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# RIPPER'S
# STEAM ENGINE THEORY AND PRACTICE

EIGHTH EDITION
Rewritten and Enlarged

BY

WILLIAM J. GOUDIE, D.Sc.
MEMBER OF THE INSTITUTION OF MECHANICAL ENGINEERS; MEMBER OF THE INSTITUTION OF ENGINEERS AND SHIPBUILDERS IN SCOTLAND; MEMBER OF THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS; ASSOCIATE MEMBER OF THE INSTITUTION OF CIVIL ENGINEERS; JAMES WATT PROFESSOR OF THE THEORY AND PRACTICE OF HEAT ENGINES IN THE UNIVERSITY OF GLASGOW

*WITH 503 ILLUSTRATIONS*
*AND NUMEROUS EXAMPLES*

LONGMANS, GREEN AND CO.
LONDON • NEW YORK • TORONTO
1932

# EDITOR'S PREFACE

THE first edition of Professor Ripper's *Steam Engine Theory and Practice* was published in 1899. It was one of the pioneers of this branch of technical literature, and ran through seven editions, the last being issued in 1914.

In 1928 the writer was asked by the Publishers to act as editor of the eighth edition, which Professor Ripper, owing to advanced age and indifferent health, felt unable to undertake.

Within recent years great advances, some of them almost revolutionary, have taken place in the theory and practice of steam engineering, and in consequence the writer found that a large part of the text was out of touch with modern conditions. He decided, therefore, to rewrite and considerably enlarge the text.

The result of this task is a new text, containing about thirty per cent. of the original illustrations, with many additions illustrating the latest modern practice.

The original scheme of this text-book, with slight rearrangement in the order of the sections, has been adhered to ; but the theoretical portion has been considerably extended and developed, to meet the modern requirements of students in the day and evening classes of Technical Schools and Colleges.

The extended text is divided into twenty-four Chapters ; and an endeavour has been made to enhance its educational value by the provision of a comprehensive series of worked examples.

In order to keep the size of the book within reasonable limits it has been necessary to compress the purely descriptive matter and illustrations into a smaller compass, and to refer the student, for fuller information on practical details, to special text-books and the publications in the Technical Journals and Transactions of Engineering Societies.

With the exception of a short note, in the section on locomotives, the subject of steam generation has been omitted. It would involve a discussion of fuels and combustion and many modern arrangements of high-pressure boiler plants and their accessories ; and an adequate treatment of these would make the size of the book prohibitive.

While primarily intended for students, it is hoped that this new edition may be found useful, for reference, by engineers concerned with the design or operation of steam engines and turbines.

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for permission to reproduce illustrations which have recently appeared in these journals; and also the various engineering firms, whose names are given in the text, for the large amount of descriptive matter, drawings, and blocks they have kindly placed at his disposal.

He has to thank Jas. Small, B.Sc., Ph.D., and George G. McDonald, B.Sc., for assistance in the preparation of a number of the illustrations, and especially E. W. Geyer, B.Sc., who undertook the tedious work of calculation in connection with the steam table compilation in 1929, from the revised Callendar equations, and drew down the temperature-entropy and heat-entropy diagrams accompanying the book.

W. J. GOUDIE.

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# STEAM ENGINE THEORY AND PRACTICE

## CHAPTER I

### THERMODYNAMIC PRINCIPLES

**1. Heat.**—According to the molecular theory, the matter of a body is an aggregation of minute particles, called molecules, in a constant state of motion.

The state of aggregation may be that of a solid or a liquid or a gas.

The heat of a body is a manifestation to the senses of the internal molecular kinetic energy of the body in the given state. The temperature or hotness indicates the rate of the molecular vibration. Heat is thus a form of molecular energy, and when a hot body gives heat to a cold body it transmits part of its internal molecular energy to the molecules of the cold body.

In such an operation, (1) the temperature of the cold body is raised due to increased rate of molecular vibration, that is, increase of internal kinetic energy; (2) wider separation of the molecules or change of aggregation is produced and internal molecular work is done against the resisting cohesive forces. The heat absorbed for this purpose is stored as internal potential energy. The sum of these two quantities, the internal kinetic energy and the internal potential energy, is called the "internal" or "intrinsic" energy; (3) an alteration of volume or change of configuration of the body takes place against external resistance and involves the performance of external work. This represents potential energy stored in the external system—that is, although supplied from the heat source it does not exist as molecular energy *in* the body. This is known as the external work and its equivalent as the external heat.

Denoting the heat supplied by Q, the kinetic heat by $k$, the internal potential heat by $\rho$, and external heat by $h_e$, the energy equation can be written as

$$Q=k+\rho+h_e$$

Denoting the internal energy by $I=(k+\rho)$ in heat units

$$Q=I+h_e \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

When the heat supplied is sufficient to change the physical state of the substance heated, *i.e.* to convert it from solid to liquid or liquid to gas, the increase of internal potential energy ($\rho$) is usually much greater than the external work ($h_e$).

The energy equation (1) holds whether there is a change of physical state or not.

The heat which is converted to internal and external potential energy ($\rho + h_e$) is said to become "latent."

The absolute amount of internal energy of a body cannot be determined, but it is always possible to estimate changes of internal energy from changes of condition of the body when heat is supplied or abstracted.

**2. Units of Heat.**—Heat is measurable from the effects it produces on substances. Rise of temperature is one of these effects, and it is the universal custom to estimate heat quantities by means of it, water being taken as the standard substance. There are various ways of defining a unit of heat, but the unit now generally adopted for engineering purposes is the mean value for a range of temperature between two fixed limits.

The British Thermal Unit (B.Th.U.) is $\frac{1}{180}$ of the quantity of heat required to raise a mass of 1 lb. of water from freezing to boiling point, 32° Fahr. to 212° Fahr., under a pressure of 1 atmosphere.

The Pound Calorie or Centrigrade Heat Unit (C.H.U.) is $\frac{1}{100}$ of the quantity of heat required to raise 1 lb. of water from 0° C. to 100° C. under a pressure of 1 atmosphere. As the ratio of the temperature divisions between the fixed limits of the thermometric scales is 180/100 or 1·8 to 1, then to convert C.H.U. into B.Th.U. the value has to be multiplied by 1·8.

The Continental engineering unit of mass is the kilogramme, and the corresponding Centrigrade heat unit is the kilocalorie. It is $\frac{1}{100}$ of the quantity of heat required to raise 1 kilogramme of water from 0° Cent. to 100° Cent. under a pressure of 1 atmosphere. As 1 kilogramme=2·203 lb.—

$$1 \text{ kilocalorie} = 2{\cdot}203 \text{ C.H.U.} = 3{\cdot}96 \text{ B.Th.U.}$$

The heat unit here defined is a *mean* value of the heat per degree rise between freezing and boiling points of water. This choice is due to the fact that the heat necessary to raise the water one degree in temperature varies slightly at different points of the scale between these limits.*

As the British Thermal Unit is most extensively used by engineers in this country, it is employed in all heat calculations throughout this text.

**3. Specific Heat.**—This is defined as the heat required to raise unit mass of the substance one degree in temperature at some

* For further discussion of heat units, see Callendar's *Properties of Steam*, p. 7.

standard temperature level. Taking the lb. and the Fahr. degree as units it is expressed as B.Th.U. per lb. per degree Fahr. or $c \times 1$ B.Th.U./lb. °Fahr. It is the ratio of two quantities of heat, and thus has no dimensions, so that the numerical value is the same both for Fahrenheit and Centrigrade scales.

An alternative definition is to state it as the ratio of the heat required to raise unit mass of the substance one degree in temperature, at some standard temperature, to the heat required to raise unit mass of water one degree at the same temperature.

For solids and liquids it is usually a constant quantity. In the case of a gas or a superheated vapour it may vary both with temperature and pressure.

Further, in the case of gas or superheated vapour its value is affected by the condition under which heat is taken in by the substance. The gas may take in heat while its volume remains constant and its pressure and temperature increase, or it may take it in while the pressure remains constant and the volume and temperature increase. In the first case the specific heat is called the specific heat at constant volume $c_v$, and in the second the specific heat at constant pressure, $c_p$.

Instead of referring the specific heat of a gas to the unit mass it is very convenient for many physical calculations to refer it to a multiple of unit mass, represented by the molecular weight of the substance. Thus in the case of oxygen, which has a molecular wt.=32, the "molecular specific heat" is the ratio of the heat to raise 32 lb. $O_2$ one degree in temperature to that to raise 1 lb. water one degree at the same temperature level.

The molecular specific heat of a gas is thus the specific heat referred to unit mass, multiplied by the numerical value of the molecular weight of the gas.

Molecular sp. ht. $C_v = mc_v$.
or, $C_p = mc_p$.

The capital letter is used to distinguish between the two forms of specific heat. Given the specific heat of a substance and its rise of temperature the amount of heat taken in by the substance is calculable.

Let a substance of weight W be raised in temperature $d$T degrees, then

Heat taken in $$dQ = cd\text{T}.\text{W} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

If the substance is a gas—

$$\left.\begin{array}{l} dQ = c_v d\text{T}.\text{W} \\ \text{or} \quad dQ = c_p d\text{T}.\text{W} \end{array}\right\} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

according to whether heat reception is at constant volume or constant pressure. $dQ$ and $d$T are very small increases in heat and temperature, and $c$, $c_v$ and $c_p$ are the true or "instantaneous"

specific heats. These statements are true whether the specific heat is constant or variable.

When, in the case of variable specific heat, a mean value of $c_v$ or $c_p$ is given, then for a rise from $T_1$ to $T_2$ the heat taken in

$$Q = {}_{(mean)}W(T_2-T_1) \quad \text{or} \quad Q=c_{p(mean)}W(T_2-T_1) \; . \quad (4)$$

If a definite law of variation of $c$ with temperature is known

$$Q=W\int_{T_1}^{T_2} c_v dT \quad \text{or} \quad Q=W\int_{T_1}^{T_2} c_p dT \quad . \; . \; . \quad (5)$$

It will be noted that the specific heat of a gas at constant volume is simply the increase of internal energy per degree rise per unit mass of gas.

*Example* 1.—The rate of air supply to a boiler is 19 lb. air/lb. coal. The air is drawn from the boiler room, and heated in a tubular heater by the flue gases from 80° Fahr. to 300° Fahr. before it enters the furnace. The temperature of the flue gases entering the heater is 600° Fahr. Neglecting radiation losses, calculate the approximate temperature of the gases at entrance to the chimney and find the percentage of the heat of the gases available to 80° Fahr. which has been saved by the use of the preheater. Take $c_p$=0·24 for air and $c_p$=0·25 for flue gases. Combustible in coal 97 per cent.

Here, neglecting radiation, heat given to air=heat given up by gases.

$$c_{pa}W_a(t_2-t_1)=c_pW_g(t_1'-t_2')$$

Air $\begin{cases} c_{pa}=0\cdot24 & t_1=80^\circ \text{ F.} \\ W_a=19 \text{ lb.} & t_2=300^\circ \text{ F.} \end{cases}$

Gas $\begin{cases} W_g=W_a+0\cdot97=19\cdot97 \text{ lb.} \\ t_1'=600^\circ \text{ F.} \quad t_2'=\; ? \end{cases}$

The weight of gas per lb. of coal is the wt. of air+combustible in coal.

Substituting the given values—

$$0\cdot24\times19 \text{ lb. } (300-80)^\circ \text{ F.}=0\cdot25\times19\cdot97 \text{ lb. } (600^\circ \text{ F.}-t'_2)$$

$$\frac{0\cdot24\times19\times\text{lb.}\times220\times{}^\circ \text{ F.}}{0\cdot25\times19\cdot97\times\text{lb.}}=600\times{}^\circ \text{ F.}-t_2'$$

$$t_2'=(600-200)=400^\circ \text{ F.}$$

Available heat of gases from 80° F.$=c_pW_g(t'-t_1)$

Heat saved $=c_pW_g(t_1'-t_2')$

$\therefore$ Heat saved/heat available

$$=\left(\frac{t_1'-t_2'}{t_1'-t_1}\right)=\frac{600-400}{600-80}=\frac{200}{520}=0\cdot384$$

Heat saved=38·4 per cent. of the heat available in the flue gases.

*Example* 2.—If the gases from the combustion chamber of a boiler enter the smoke tubes at 2000° Fahr. and leave them at 600° Fahr., calculate the heat transmitted to the water through

the tubes per lb. of gas flowing. The molecular specific heat of the gases at constant pressure is given by $C_p=6{\cdot}8+0{\cdot}00056T$ where T is the temperature in abs. Fahr. and the molecular weight is $m=29$.

Specific heat referred to the lb. mass $c_p=\frac{C_p}{m}=b+st$.

By Equation 5—

$$\text{Heat supplied } Q=W\int_{T_1}^{T_2}\frac{C_p}{m}dT$$

$$=W\int_{T_1}^{T_2}(b+st)dT.$$

On integration $Q=b(T_2-T_1)+\frac{s}{2}(T_2^2-T_1^2)$

$T_2=2000+460=2460°$ F. $T_1=600+460=1060°$ F.

$b=\frac{6{\cdot}8}{29}=0{\cdot}234$; $s=\frac{0{\cdot}00056}{29}=0{\cdot}0000193$; $W=1$

$\therefore Q=0{\cdot}234(2460-1060)+0{\cdot}0000096(2460^2-1060^2)$

$=0{\cdot}234\times1400+0{\cdot}0000096\times1400\times3520$

$=327{\cdot}6+47{\cdot}4=375{\cdot}0$ B.Th.U./lb. gas.

**4. First Law of Thermodynamics.**—According to the principle of the Conservation of Energy, energy can neither be created nor destroyed, but only transformed from one form into another. It is not possible for a machine to create work from nothing, or to deliver more energy than it receives.

The First Law of Thermodynamics, established by Dr. Joule, is a statement of this principle in relation to the conversion of heat to work and work to heat. It may be stated as follows :—

" Heat and mechanical energy are mutually convertible, and heat requires for its production and produces by its disappearance a definite number of units of work for each unit of heat converted."

Put in algebraic form, if E units of mechanical energy are converted to H units of heat

$$\left.\begin{matrix}E\propto H\\ \text{or}\quad E=JH\end{matrix}\right\} \quad \ldots\ldots\ldots \quad (6)$$

where J is a constant called Joule's equivalent.

Its dimension is dependent on the units of work and heat chosen. If the British units of ft.-lb. and B.Th.U. or ( lb.° Fahr.) are chosen then

$$J=778\ \frac{\text{ft.-lb.}}{\text{B.Th.U.}}=778\text{ ft.-lb./lb.° F.}$$

If the Centrigrade scale of temperature is taken—

$$J=1400\ \frac{\text{ft.-lb.}}{\text{C.H.U.}}=1400\text{ ft.-lb./lb.° C.}$$

The exact value is 777·8,* but for engineering purposes the round number 778 is quite sufficient.

The first law merely states the equivalence between heat and work and work and heat, and gives no information regarding the *availability* of the heat for work in a heat engine. This condition is controlled by the Second Law of Thermodynamics, which may be stated in the form given by Clausius.

**5. Second Law of Thermodynamics.**—" Heat cannot pass from a cold body to a hot one by a purely self-acting process."

Transference of heat between two bodies, without external aid, thus necessitates a drop in temperature. A heat engine acts as a transformer of heat to work by means of a carrier substance, called the working substance, interposed between a " hot " body and a " cold " body. The mechanical work is done by the substance at the expense of the heat supplied by the hot body through variation of condition of the substance as the heat passes through it.

While heat can flow through it from the zone of higher to the zone of lower temperature, it cannot of itself flow in the reverse direction from the cold to the hot zone.

The heat discharged by the substance to the cold body at the lower temperature is thus unavailable for conversion into mechanical work, between the temperature limits of the system.

If the hot body transmits $Q_1$ units of heat to the substance and this in turn delivers $Q_2$ units to the cold body, the heat transformed into work is $(Q_1 - Q_2)$. This is the available heat of the system and $Q_2$ is the unavailable or wasted heat. The availability is given by the ratio of the available heat to the total heat supplied, *i.e.* $\frac{Q_1 - Q_2}{Q_1}$. It is the fraction of the energy supplied to the system in the given initial state that can be converted into mechanical work.

It goes by the general name of the " absolute thermal efficiency " of the engine.

The conditions required to give this efficiency a maximum value are fixed by Carnot's principle, discussed later.

**6. Heat Engine Cycles.**—During the processes of reception, transformation, and rejection of heat, the working substance goes through a recurring series of changes of condition termed the " cycle " of the engine.

In all practicable engines using fluid substances a pressure difference is created in the cycle in order to produce statical and dynamical forces which do work on the mobile part of the engine.

There are three fundamental heat engine cycles, defined with reference to the mode in which the heat is received and rejected by the working substance. These are :

(1) Constant Temperature Cycle :—Heat received while the sub-

* For a discussion of values of J, see Callendar's *Properties of Steam*, p. 9.

stance remains at constant high temperature and rejected while it remains at constant low temperature. This is the cycle of the perfect heat engine of Carnot.

(2) Constant Volume Cycle :—Heat received while the volume of the substance remains constant and rejected while the volume remains constant.

(3) Constant Pressure Cycle :—Heat received while the pressure exerted by the substance remains constant and rejected while the pressure remains constant.

There are various combinations of these cycles in practical operation. As regards the steam engine and turbine, the *ideal* cycle is a combination of (1) and (2) heat being received by the substance partly at constant temperature and partly at constant volume, and rejected at constant temperature.

**7. Cycle Processes.**—Two processes are involved in the performance of any heat engine cycle : (1) transmission of heat to and from the working substance ; (2) conversion of heat into work by the substance and conversion of work done on the substance into heat, since the process of conversion is always accompanied by a change of configuration of the substance, that is, by expansion and compression.

In some cases actions (1) and (2) occur separately and in others simultaneously. In a constant volume engine, heat is received at constant volume, there is no change of configuration and no external work is done, that is, there is no conversion of heat into work. During subsequent expansion, while the substance does work on the mobile part, no heat is received, but there is a continuous conversion of heat into work.

In a constant temperature cycle (Carnot) heat is continuously taken in during isothermal expansion and converted to work, that is, there is simultaneous reception and conversion. During isothermal compression work is done continuously on the substance and heat simultaneously rejected.

**Reversible and Irreversible Processes.**—A thermodynamic process may be either reversible or irreversible. Where on reversal of the direction of the process the system passes through the direct states in reverse order, and is brought back to its initial state, while the action of external agencies remains unchanged or differs only in an infinitesimal degree, the process is said to be reversible. Any process which does not conform to these conditions is irreversible.

The conduction of heat from one body to another at a lower temperature is an irreversible process, since the heat cannot of itself flow back from the colder to the hotter body. The degradation of work to heat through friction between two rubbing surfaces is irreversible, as the frictional heat cannot be reconverted to work when the bodies in contact cool. The flow of a fluid through a valve or nozzle from a zone of high to a zone of low pressure is also irreversible, as the fluid of itself cannot flow back into the zone

of higher pressure. Restoration to its initial condition can only be accomplished through its recompression by an external agent.

Considering the case of the working substance of a heat engine, this may be regarded as completely constrained if the effort exerted by the substance on the mobile part of the engine is just balanced by the external resistance. If during an expansion in which this condition holds, that is in which the forces of action and reaction are "in finite terms equal," the balance is disturbed, a reversal of the operation is possible. The external resistance becomes an effort and the original effort a resistance. In such a reversal the energy expended by the substance or the external agency is completely restored and the substance is brought back to its initial condition at the beginning of the expansion. No energy is absorbed in giving sensible motion to the mass of the substance and no frictional heat is generated.

This, however, is only a partial statement of the case of reversibility in an engine cycle. Besides the energy conversion process there are the processes of heat reception and rejection. At any point of the cycle during reception from the source there must be no sensible difference of temperature between the hot body or source and substance, and similarly during rejection no sensible difference between the temperature of the substance and cold body or cooler. Under such conditions a slight disturbance of the temperature equilibrium will cause a reversal of the flow of heat.

The conditions then for reversibility at any point of a heat engine cycle are: (1) equality of effort and resistance, (2) equality of temperature.

It will be apparent from the foregoing considerations that since no sensible motion can be given to the substance and only an infinitesmal difference of temperature can be permitted, the time of performance of a perfectly reversible engine cycle must be indefinitely great.

A completely reversible cycle is thus not possible in practice.

If a completely reversible heat engine were possible, it would abstract as much heat from the cooler or cold body, when worked backwards as a heat pump, as it would discharge to it if worked directly; and would discharge as much heat to the source as it would abstract from it working directly.

**8. Engine of Maximum Thermal Efficiency.**—Consider a self-contained system consisting of two identical engines A and B, a source or hot body, and a cooler or cold body, and let the engines be coupled so that one can drive the other as a heat pump. Let the engine A be completely reversible.

If A works directly and drives B, it will take $Q_1$ units of heat from the source for each unit of work done on B, and discharge $Q_2$ units to the cooler. If it is driven by B, it will take $Q_2$ from the cooler and discharge $Q_1$ to the source, for each unit of work done by B.

Now suppose it is assumed that B is more efficient than the reversible engine A. It will draw a smaller quantity $(Q_1-q)$ from the source and discharge a smaller quantity $(Q_2-q)$ to the cooler. Engine A, however, returns $Q_1$ to the source, which is $q$ in excess of the quantity drawn by B, so that for every unit work done there is a gain of heat by the source. Also, since A draws $Q_2$ from the cooler while B discharges $(Q_2-q)$ into it, the cooler loses $q$ units of heat for every unit of work done. The ultimate result is a transfer of heat from the cold body to the hot body, and as by hypotheses the system is self-contained or self-acting the result is contrary to the Second Law of Thermodynamics. The engine B cannot, therefore, be more efficient than the reversible engine A. It is assumed here that the capacities of the hot and cold bodies are so great that the abstraction and addition of the heat quantities $Q_1$ and $Q_2$ do not affect the temperatures.

In general, then, this negative method of proof shows that all reversible heat engines which receive and reject heat at the same temperatures, that is, work between the same temperature limits, are equally efficient. The efficiency of the perfect heat engine is thus the efficiency of *any* reversible engine. It should be noted that in the foregoing statement nothing is said regarding the nature of the working substance or the nature of the cycle.

As will be seen presently the constant temperature cycle embodying the principle of Carnot is a completely reversible one, and its absolute thermal efficiency is given by—

$$\eta=\frac{T_1-T_2}{T_1} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

where $T_1$ and $T_2$ are the upper and lower limits of the cycle, or hot body temperature and cold body temperature. This, therefore, is the maximum possible efficiency for any engine cycle working between these limits.

It is instructive to note that if it were physically possible to reduce the lower temperature limit to absolute zero the efficiency would be unity, that is, all heat supplied would be convertible into work, or the availability would be 100 per cent.

In order to reach a proper understanding of the thermodynamic conditions of operation of the working substance of the steam engine and turbine (which is a saturated vapour) it is essential to consider first the properties of the substances known as permanent gases.

**9. Boyle's Law.**—The volume of a given mass of gas varies inversely as the pressure when the temperature is kept constant, that is, the product of pressure and volume is a constant.

If pressure is P and volume V, then

$$PV_{(t)}=C \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

The graph *abc*, Fig. 1, of this equation, on the PV co-ordinate field, is a rectangular hyperbola since $P_1V_1=P_2V_2=P_3V_3$.

**10. Charles' Law.**—Equal volumes of different gases increase equally for the same increment of temperature when the pressure is constant; and the volume changes proportionally to the absolute temperature.

The algebraic statement of this law is—

$$V_{(p)}=CT \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

If air is used as a thermometric substance and allowed to expand without change of pressure, and equal intervals of temperature are defined as those corresponding to equal expansions of the air, a thermometric scale is obtained in very close agreement with the ordinary mercurial thermometer scale, which defines equal intervals of temperature to correspond with equal expansions of the mercury.

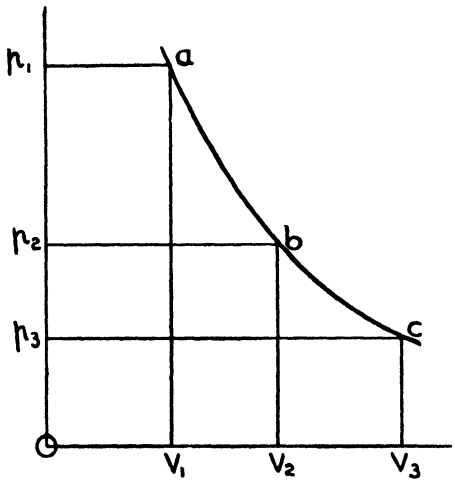


FIG. 1.

At 32° Fahr. it is found that the change of volume is $\frac{1}{492}$ per degree, that is, a volume of 492 in.$^3$ will become 493 at 33° Fahr., 491 at 31° Fahr., 482 at 10° Fahr., and so on, till at 0° Fahr. it will be (492—32) =460 in.$^3$. If it were possible to reduce the volume to zero value then the temperature would be —460° Fahr. Such a condition is impossible, as the substance would change its physical state before this low temperature could be reached.

**Absolute Temperature.**—This result means that if the temperature of a gas is reckoned from a point 460° below the Fahr. zero, the volume of the gas at constant pressure is proportional to the temperature reckoned from this zero, which is called the "absolute zero of temperature." * If the Centigrade scale is used the absolute zero is at —273° below 0° Cent.

If $t$ is the ordinary temperature and T the absolute value, then

$$T=t+460° \text{ F.}$$
$$T=t+273° \text{ C.}$$

**11. Characteristic Law of a Gas.**—Consider a unit mass of gas at volume $v$, temperature T, and pressure P.

* The exact value is 459·58, but the round number is sufficient for ordinary work.

Let it expand isothermally to $v'$ and pressure $P_1$.

Then $Pv=P_1v'$ or $v'=\frac{Pv}{P_1}$ . . . . . (*a*)

Let it further expand at constant pressure $P_1$ till the volume is $v_1$ and temperature $T_1$, then

$$\left.\begin{aligned} v'&=CT \\ v_1&=CT_1 \end{aligned}\right\} \quad \therefore\ v'=\frac{v_1T}{T_1} \quad . \ . \ . \ . \ . \quad (b)$$

substitute from (*b*) in (*a*) and

$$\frac{v_1T}{T_1}=\frac{Pv}{P_1}$$

hence $$\frac{Pv}{T}=\frac{P_1v_1}{T_1} \text{ or } \frac{Pv}{T}=\text{constant}=R \quad . \ . \ . \quad (10)$$

This important equation giving the interdependent relations of P, $v$, and T when these three "characteristic properties" of the gas vary simultaneously is known as the "Characteristic Law" of the gas.

If the mass of gas having volume V at P and T is W, its specific volume is $v=\frac{V}{W}$. Substituting in (10)

$$P\frac{V}{W}=RT$$

or $$PV=WRT \quad . \ . \ . \ . \ . \ . \ . \quad (11)$$

where R is the constant known as the "gas constant."

The dimension of this constant depends on the units of pressure, volume, temperature, and mass chosen. Taking lb. for mass, lb./ft.$^2$ for pressure, ft.$^3$ for volume, and ° Fahr. for temperature, R is dimensionsally expressed in ft.-lb./lb. ° Fahr., that is, as work per lb./° Fahr. If the mass unit is taken as the pound mol. or $m\times 1$ lb., where $m$ is the molecular wt. of the gas, the dimension is ft.-lb./lb.-mol. ° Fahr.

In this case equation $Pv=RT$ becomes

$$P(mv)=(mR)T \quad . \ . \ . \ . \ . \ . \quad (12)$$

$(mv)=\bar{v}$ is the volume of the pound mol. and $(mR)=\bar{R}$ is called the Universal Constant, since for all permanent gases it has sensibly the same value. On the Fahr. scale it is usually taken as 1544, and on the Cent. scale as 2780.

This form of the characteristic equation

$$P\bar{v}=\bar{R}T \quad . \ . \ . \ . \ . \ . \quad (13)$$

is very useful for calculation of specific volumes of gases, when the molecular weights are known.

*Example* 3.—Given that the molecular weight of air is 28·86 calculate the gas constant and the specific volume at 32° F. and 14·7 lb./in.$^2$ abs. $\bar{R}=1544$.

Here $R=\frac{\bar{R}}{m}=\frac{1544}{28{\cdot}86}=53{\cdot}49$ ft.-lb./lb.°F. (usual value taken is 53·33)

By Equation 11— $144pv=RT$. $p=14{\cdot}7$ lb./in.$^2$, $T=492°$ F., $W=1$

(Dimensionally)

$$v=\frac{53{\cdot}49 \text{ ft.-lb.}}{\text{lb.} \cdot °\text{F.}} \times \frac{492 \cdot °\text{F.}}{144\times 14{\cdot}7 \text{ lb./ft.}^2}$$

$$=\frac{53{\cdot}49\times 492}{144\times 14{\cdot}7} \times \frac{\text{ft.}\times\text{lb.}\times °\text{F.}\times\text{ft.}^2}{\text{lb.} \cdot °\text{F.}\times\text{lb.}}=12{\cdot}39 \frac{\text{ft.}^3}{\text{lb.}}$$

*Example* 4.—In the case of the boiler (*Example* 1), calculate the total volume of the flue gases discharged to the chimney at 14·7 lb./in.$^2$ abs. and 400° Fahr., when the rate of coal consumption is 1 ton per hour.

Here $144p\bar{v}=\bar{R}T$ $T=860°$ F. abs.

$v=\frac{\bar{R}T}{144p}$ $\bar{R}=1544 \frac{\text{ft.-lb.}}{\text{lb. mol.} \cdot °\text{F.}}$

$p=14{\cdot}7$ lb./in.$^2$ abs. $=144\times 14{\cdot}7$ lb./ft.$^2$ abs.

Writing dimensionally

$$\bar{v}=1544 \frac{\text{ft.-lb.}}{\text{lb.-mol.} \cdot °\text{F.}} \times \frac{1\times\text{ft.}^2\times 860° \text{F.}}{144\times 14{\cdot}7\times\text{lb.}}$$

$$\bar{v}=\frac{1544\times 860}{144\times 14{\cdot}7} \times \frac{\text{ft.}\times\text{lb.}\times\text{ft.}^2\times °\text{F.}}{\text{lb.-mol.}\times\text{lb.}\times °\text{F.}}=\frac{630 \text{ ft.}^3}{\text{lb.-mol.}}$$

1 lb.-mol.$=m$ lb.$=29$ lb.

$$\therefore v=\frac{630 \text{ ft.}^3}{29 \text{ lb.}}=21{\cdot}7 \frac{\text{ft.}^3}{\text{lb.}}$$

At. 14·7 lb./in.$^2$ abs. and 400° Fahr.
Gas discharged $=19{\cdot}97$ lb./lb. coal.
Coal burned $=2240$ lb./hr.

Total discharge of gas $=19{\cdot}97 \frac{\cancel{\text{lb. gas}}}{\cancel{\text{lb. coal}}} \times 2240 \frac{\cancel{\text{lb. coal}}}{\text{hr.}} \times \frac{21{\cdot}7 \text{ ft.}^3}{\cancel{\text{lb. gas}}}$

$=19{\cdot}97\times 2240\times 21{\cdot}7=970{,}000$ ft.$^3$/hour.

**12. Relation between the Gas Constant and Specific Heats.**—According to the energy law equation (1), for a small change of state, when heat $dQ$ is given to unit mass of a gas, and increases the temperature $dT$ and volume $dv$ under constant pressure P—

$$dQ = c_v dT + \frac{P dv}{J}$$ where $c_v dT = dI$

and $\frac{P dv}{J} = dh_e$ is the external work.

But this is also—

$$dQ = c_p dT \text{ (Art. 3)}$$

so that, $$c_p dT = c_v dT + \frac{P dv}{J}$$ But by the characteristic law—

$$P dv = R dT$$

$$\therefore \quad (c_p - c_v) dT = \frac{R}{J} dT$$

or
$$\frac{R}{J} = (c_p - c_v) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (14)$$

when referred to 1 lb. mass.

If the mass of the lb. mol. is taken

$$\frac{\bar{R}}{J} = (C_p - C_v) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (15)$$

The ratio of the specific heats is usually denoted by $\gamma$ or

$$\gamma = \frac{c_p}{c_v}$$

Since
$$c_p = \gamma c_v$$

$$\frac{R}{J} = c_v(\gamma - 1) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (16)$$

or
$$\frac{R}{J} = c_p\left(\frac{\gamma - 1}{\gamma}\right) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (17)$$

or
$$\frac{\bar{R}}{J} = C_v(\gamma - 1) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (18)$$

$$\frac{\bar{R}}{J} = C_p\left(\frac{\gamma - 1}{\gamma}\right) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (19)$$

The value of $\frac{\bar{R}}{J}$ is the same for either temperature scale. Thus taking the Fahr. scale, $\frac{\bar{R}}{J} = \frac{1544}{778} = 1{\cdot}986$. Taking the Cent. scale $\frac{\bar{R}}{J} = \frac{2780}{1400} = 1{\cdot}986$. This is a very handy figure for calculations relating to fuel combustion. If the specific heat varies with temperature and follows such a law as

$$c_v = a + sT + uT^2 \; ; \quad c_p = b + sT + uT^2.$$

then
$$\frac{R}{J} = (b - a) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (20)$$

and
$$\frac{\bar{R}}{J} = m(b - a) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (21)$$

*Example* 5.—At moderate temperatures the specific heat of air at constant pressure is 0·238. Calculate the specific heat at constant volume and the value of the ratio $\gamma$.

Taking R=53·33 and J=778, $c_p$=0·238—

$$\therefore \quad \frac{R}{J}=(c_v-c_p). \qquad \therefore c_v=\left(c_p-\frac{R}{J}\right)$$

$$\text{and} \quad \gamma=\frac{c_p}{c_v}=\frac{0\cdot238}{0\cdot169}=1\cdot41 \qquad =0\cdot238\times\frac{53\cdot33}{778}$$

$$=0\cdot238-0\cdot069=0\cdot169$$

Alternatively with $m$=28·86, $\frac{\bar{R}}{J}=1\cdot986$

$$\frac{\bar{R}}{J}=(C_p-C_v)=m(c_p-c_v)$$

$$1\cdot986=28\cdot86(0\cdot238-c_v)$$

$$c_v=0\cdot238-\frac{1\cdot986}{28\cdot86}=(0\cdot238-0\cdot069)=0\cdot169$$

$$\gamma=\frac{0\cdot238}{0\cdot169}=1\cdot41$$

*Example* 6.—For the flue gases of the boiler (*Example* 2) having variable specific heat given by $C_p$=6·8+0·00056T, compare the values of the specific heat ratio at 2000° Fahr. and 600° Fahr.

T=2000+460=2460° F. abs. T=600+460=1060° F. abs.

$C_{p2000}$=6·8+0·00056×2460 $C_{p600}$=6·8+0·00056×1060

=6·8+1·377=8·177 =6·8+0·5936=7·394

$$\frac{\bar{R}}{J}=1\cdot986=(C_p-C_v)$$

$C_{v2000}$=(8·177−1·986) $C_{v600}$=(7·394−1·986)

=6·191 =5·408

$\gamma_{2000}=\frac{8\cdot177}{6\cdot191}=1\cdot32$ $\gamma_{600}=\frac{7\cdot394}{5\cdot408}=1\cdot365$

$\gamma$ is decreased by $\frac{0\cdot045\times100}{1\cdot365}=3\cdot3$ per cent. between 600° and 2000° Fahr.

**13. Joule's Law for Intrinsic Energy of a Gas.**—When a gas expands without doing external work and without taking in or giving out heat, its temperature does not change. Joule connected two copper vessels by a pipe and stop-cock. One vessel was exhausted and the other filled with air at 22 atmospheres pressure. The combination was immersed in a vessel of water the temperature of which was accurately recorded. When the cock was opened the air rushed into the empty vessel until a balance of pressure was produced.

No external work was done, and there was no rise in the temperature of the surrounding water. Thus no heat passed out of the system and the temperature of the gas at the end of the operation was the same as at the beginning. Since it had neither gained nor lost heat and had done no external work, the gas had the same internal energy at the end as at the beginning. Although there was a change of pressure and volume there was no change of temperature.

It follows from this result that the internal energy of a given mass of gas depends only on its temperature and not upon its pressure and volume.

**14. Adiabatic Expansion and Compression.**—When a gaseous working fluid expands in a heat engine and does work against the external resistance (or in generating kinetic energy of its mass) without loss of heat to or gain of heat from the surroundings (by conduction) the process is said to be "adiabatic." The converse holds when the gas is compressed. In the first case there is a decrease of internal energy equivalent to the external work done by the gas. By the energy law for a small change of state—

$$dQ = dI + \frac{dE}{J};$$

then, since by hypothesis $dQ=0$, $-dI = \frac{dE}{J}$, or decrease of I=increase of E.

**15. Isothermal Expansion and Compression.**—If during the expansion of the gas the temperature remains constant throughout, then by Joule's Law the intrinsic energy must be constant and the change of intrinsic energy during expansion is zero. It follows then, from the energy law, that the external work can only be done by a continuous supply of heat from the surroundings and this heat is the equivalent of the work done. Thus in an isothermal expansion the heat is transformed into external work as fast as it is supplied to the substance. Again for a small change of state—

$$dQ = dI + \frac{dE}{J}, \qquad \text{then, since } dI = 0,$$

$$dQ = \frac{dE}{J}, \text{ or heat supply=increase of E} \quad . \quad . \quad (22)$$

Conversely, during isothermal compression work done on the gas is immediately converted into heat which is conducted away preventing rise of temperature, so that again the internal energy remains constant.

Any other mode of expansion or compression is classed as "polytropic."

**16. Calculation of Work done by an Expanding Gas.**—If the values of the pressure at various volumes are plotted on the PV co-ordinate field the curve obtained is AC for adiabatic and AB for isothermal

expansion (Fig. 2). Since (pressure×volume) gives a quantity of work the area ACDE below the adiabatic gives the absolute amount of external work E done by the fluid. The student must note carefully that this PV area gives no information about the heat quantities except that $\frac{E}{J}$ is the quantity of heat transformed into mechanical work.

Similarly the area ABDE is the external work done during isothermal expansion, which for the same volume limits, $V_1$ and $V_2$, is greater than that with adiabatic expansion. The isothermal falls above the adiabatic and the terminal pressure $P_3$ is greater than $P_2$.

The dotted curve AF is a polytropic, the conditions of expansion

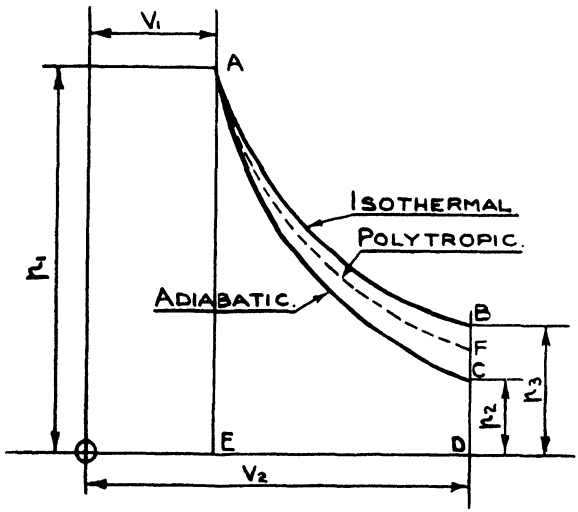



FIG. 2.

falling between those of the isothermal and adiabatic. Here again the external work is given by the area AFDE. The polytropic curve, however, need not necessarily fall between AB and AC.

When the equation to any one of these curves is known the corresponding area below the curve is calculable analytically. These curves can be fitted generally by an equation of the form $PV^n=C$, where $n$ is some *constant* index. It can be shown, in the case of a gas when the specific heat remains constant, that $n=\gamma$ for adiabatic expansion. In the isothermal, since this curve is given by Boyle's Law, $n=1$. Taking the general case where $n$ is greater than 1:

At volume V ft.$^3$ and pressure P lb./ft.$^2$ (Fig. 3), work done during a small increment $dV$, is the hatched area $dE=PdV$. The whole

work area AFDE is the summation of these elementary areas between $V_1$ and $V_2$, or

$$E=\int_{V_1}^{V_2} P dV.$$

Since $PV^n=C$, $P=\frac{C}{V^n}$, substituting

$$E=C\int_{V_1}^{V_2} dV.V^{-n}=\frac{P_1V_1^n}{1-n}[V_2^{1-n}-V_1^{1-n}]$$

Since $P_1V_1^n=P_2V_2^n$

$$\therefore E=\frac{P_2V_2-P_1V_1}{(1-n)}=\frac{P_1V_1-P_2V_2}{(n-1)} \quad . \quad . \quad . \quad (23)$$

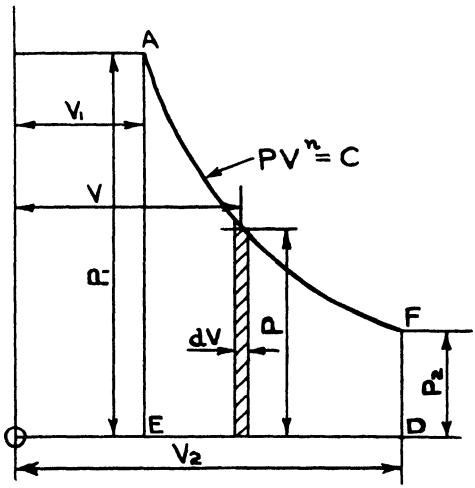



FIG. 3.

**Alternative Calculation.**—It can be seen (Fig. 4) that the work area AFDE is (KAFG+GFDO−KAEO) or $E=KAFG+P_2V_2-P_1V_1$.

Since pressure decreases $dP$ is negative; $C^{\frac{1}{n}}=VP^{\frac{1}{n}}$,

$$\therefore KAFG=-\int_{P_1}^{P_2} V dP=-\frac{C^{\frac{1}{n}}}{\frac{n-1}{n}}\left[P_2^{\frac{n-1}{n}}-P_1^{\frac{n-1}{n}}\right]$$

$$=-\frac{n}{n-1}\left[P_2^{\frac{n-1}{n}}\times P_2^{\frac{1}{n}}V_2-P_1^{\frac{n-1}{n}}\times P_1^{\frac{1}{n}}V_1\right]$$

$$=-\frac{n}{n-1}(P_2V_2-P_1V_1)=\frac{n}{n-1}(P_1V_1-P_2V_2)$$

$$\therefore E=\frac{n}{n-1}(P_1V_1-P_2V_2)+P_2V_2-P_1V_1$$

$$=P_1V_1\left(\frac{n}{n-1}-1\right)-P_2V_2\left(\frac{n}{n-1}-1\right)$$

$$=\frac{1}{n-1}(P_1V_1-P_2V_2)$$

This alternative calculation is given to direct the student's attention particularly to the close resemblance between the expressions for the areas AFDE, and KAFG. The latter is the former expression with $n$ in the numerator, and is of great importance in connection with the ideal cycle of the steam engine and turbine (see p. 72).

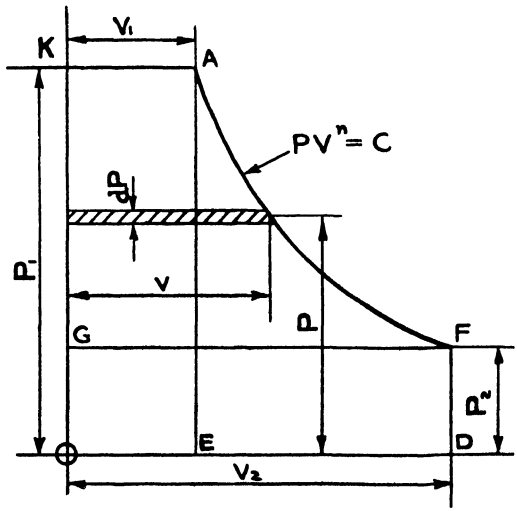



FIG. 4.

Care must be exercised not to confuse these two work expressions. The expression for E can be written in the alternative form

$$E=\frac{1}{n-1}P_1V_1\left\{1-\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}\right\} \quad . \quad . \quad . \quad . \quad (24)$$

When the expansion curve is an adiabatic then $n=\gamma$. When it is an isothermal, Equation 24 does not hold.

Here $PV=C$, so that $P=\frac{C}{V}$

$$E=\int_{V_1}^{V_2}P\,dV=C\int_{V_1}^{V_2}\frac{dV}{V}=C[\log_e V_2-\log_e V_1]$$

$$=P_1V_1\log_e\frac{V_2}{V_1} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (25)$$

Since PV=WRT each of the foregoing work expressions may be stated in terms of the weight of gas and the temperatures.

Thus $$E=\frac{WR}{n-1}(T_1-T_2)$$

(Polytropic) $$E=\frac{1}{n-1}WRT_1\left\{1-\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}\right\} \quad . \quad . \quad . \quad (26)$$

(Isothermal) $$E=WRT_1\log_e\frac{V_2}{V_1}=WRT_1\log_e\frac{P_1}{P_2} \quad . \quad . \quad (27)$$

**17. Temperature Ratio.**—The temperature ratio for polytropic or adiabatic expansion or compression of a gas can be expressed in terms of either the volume ratio or the pressure ratio.

Since $$\frac{P_1V_1}{T_1}=\frac{P_2V_2}{T_2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (a)$$

$$P_1V_1^n=P_2V_2^n \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (b)$$

Divide (*b*) by (*a*) and $T_1V_1^{n-1}=T_2V_2^{n-1}$

$$\therefore \frac{T_1}{T_2}=\left(\frac{V_2}{V_1}\right)^{n-1} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (28)$$

From (*b*) $\left(\frac{V_2}{V_1}\right)=\left(\frac{P_1}{P_2}\right)^{\frac{1}{n}}$ $\therefore \frac{T_1}{T_2}=\left(\frac{P_1}{P_2}\right)^{\frac{n-1}{n}}$ . . . . . (29)

It should be noted that the PV forms of the work equation are equally applicable to a gas and a vapour when the index $n$ is definitely constant. The other forms involving the temperatures are not valid for the vapour, unless when it is "superheated" throughout the whole range of expansion, and is therefore treated as a gas.

*Example* 7.—A mass of gas behind a piston at the beginning of the stroke occupies 0·1 ft.$^3$ at 400 lb./in.$^2$ abs. and 1000° Fahr. It expands isothermally till the pressure is 146 lb./in.$^2$ abs. The supply of heat is then cut off, and the gas expands adiabatically till the final volume is 0·45 ft.$^3$. Taking the adiabatic index as 1·38 and $c_v$=0·18, calculate: (1) the heat supplied; (2) the temperature and pressure at the end of the stroke; (3) the decrease of internal energy; (4) the total work done on the piston (in heat units).

(1) Referring to Fig. 5, since AB is an isothermal, then by the energy law—

Work done=heat supplied.

$$\frac{ABba}{J}=\frac{E_i}{J}=\frac{R}{J}WT_a\log_e\frac{p_a}{p_b}=Q$$

$$W=\frac{144p_aV_a}{RT_a}=\frac{144\times400\times0{\cdot}1}{53{\cdot}3\times1460}=0{\cdot}074 \text{ lb.}$$

$p_a=400$, $p_b=146$, $V_a=0{\cdot}10$, $\frac{p_a}{p_b}=2{\cdot}75$, $T_a=1460$, $c_v=0{\cdot}18$, $\gamma=1{\cdot}38$

$$\frac{R}{J}=c_v(\gamma-1)=0{\cdot}18\times0{\cdot}38=0{\cdot}0684$$

$$\therefore\ Q=0{\cdot}0684\times0{\cdot}074\times1460\times2{\cdot}3\log 2{\cdot}75=7{\cdot}50 \text{ B.Th.U.}$$

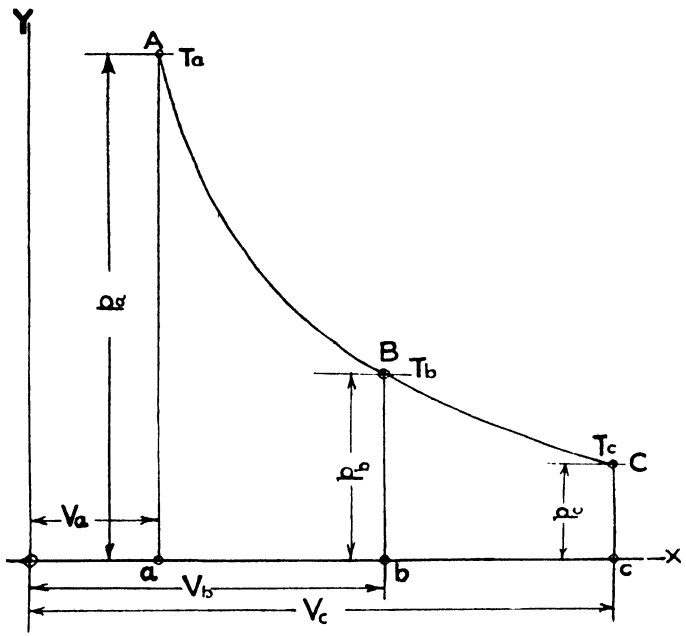



FIG. 5.

(2) $\frac{T_b}{T_c}=\left(\frac{V_c}{V_b}\right)^{\gamma-1}$, $V_b=V_a\frac{p_a}{p_b}=\frac{0{\cdot}1\times400}{146}=0{\cdot}275$, $\frac{V_c}{V_b}=\frac{0{\cdot}45}{0{\cdot}275}=1{\cdot}63$

$$T_c=\frac{1460}{(1{\cdot}63)^{0{\cdot}38}}=\frac{1460}{1{\cdot}204}=1210^\circ \text{ F. abs.}$$

$$p_c=\frac{p_b}{\left(\frac{V_c}{V_b}\right)^{1{\cdot}38}}=\frac{146}{(1{\cdot}63)^{1{\cdot}38}}=\frac{146}{1{\cdot}96}=74{\cdot}5 \text{ lb./in.}^2 \text{ abs.}$$

(3) Since BC is an adiabatic, by Joule's Law—

$$BCcb=(I_c-I_b)=c_v(T_c-T_b)W=0{\cdot}18(1210-1460)0{\cdot}074$$
$$=-0{\cdot}18\times250\times0{\cdot}074=-3{\cdot}32 \text{ B.Th.U}$$

the negative sign denotes decrease.

(4) Heat converted to work=equivalent area ABC*ca*

$$Q' = Q + (I_b - I_c) = Q + hc = (7{\cdot}50 + 3{\cdot}32) = 10{\cdot}82 \text{ B.Th.U.}$$

**18. Carnot Cycle.**—The cycle of the perfect heat engine is carried out in four stages. Referring to Fig. 6, the source S is assumed to be so large that its temperature $T_1$ is not affected by heat given to the substance. Likewise the cooler or refrigerator R is so large that its temperature $T_2$ is not affected by the heat discharged to it.

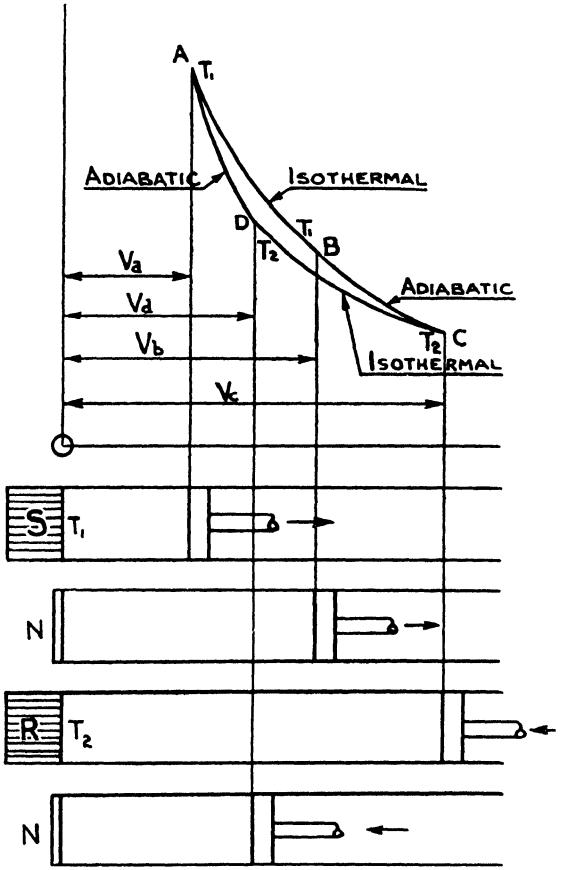



FIG. 6.

The cylinder is assumed to be perfectly non-conducting, except at the end.

(1) Source S is placed on the cylinder end, and transmits heat to the substance at $T_1$ while it expands isothermally from A to B.

(2) Source is removed and a non-conducting body N is placed on the cylinder end. The substance expands adiabatically from B to C and $T_1$ to $T_2$.

(3) The refrigerator R is placed on the cylinder end and the substance is compressed isothermally, C to D, at $T_2$, by the action of the mechanism, in which part of the positive work has been stored, and heat is rejected to the cooler.

(4) The non-conducting body N is placed on the cylinder end and the substance is compressed adiabatically, D to A, and from $T_2$ to $T_1$.

The substance in this way is passed through a reversible cycle of operations and returned at the end to the same condition as at the start.

In order to determine the expression for the thermal efficiency, suppose that the working substance is a permanent gas. The thermal efficiency is—

$$\eta = \frac{\text{Heat received} - \text{heat rejected}}{\text{Heat received}}$$

$$= \frac{Q_1 - Q_2}{Q_1} \quad \ldots\ldots\ldots\ldots \quad (30)$$

Since reception is isothermal, then by the energy law the heat supplied=work done, the internal energy change being zero. For W lb. of gas—

$$\text{Isothermal work A to B} = W\frac{R}{J}T_1 \log_e \frac{V}{V_a} = Q_1$$

$$\text{Isothermal work C to D} = W\frac{R}{J}T_2 \log_e \frac{V_c}{V} = Q_2$$

But since BC and DA are adiabatics,

(Art. 17) $$\frac{T_1}{T_2} = \left(\frac{V_c}{V_b}\right)^{\gamma-1}; \quad \frac{T_1}{T_2} = \left(\frac{V_d}{V_a}\right)^{\gamma-1}$$

$$\therefore \quad \frac{V_c}{V_b} = \frac{V_d}{V_a} \text{ or } \frac{V_b}{V_a} = \frac{V_c}{V_d} = r.$$

$$\therefore \quad \eta = \frac{Q_1 - Q_2}{Q_1} = \frac{\frac{WR}{J}(T_1 - T_2)\log_e r}{\frac{WR}{J}T_1 \log_e r} = \frac{T_1 - T_2}{T_1} \quad . \quad (31)$$

the expression (7) already quoted (Art. 8) as the maximum possible efficiency of any engine whatever, working between the temperature limits $T_1$ and $T_2$.

Alternatively this result is obtained by considering the net work done as shown by the closed PV curve ABCD$=E_c/J=(Q_1-Q_2)$.

$$\text{Positive work} = WRT_1 \log_e r + WR\left(\frac{T_1 - T_2}{\gamma - 1}\right)$$

$$\text{Negative work} = WRT_2 \log_e r + WR\left(\frac{T_1 - T_2}{\gamma - 1}\right)$$

$\therefore$ net work$=WR(T_1-T_2)\log_e r$, since the adiabatic work of expansion cancels the adiabatic work of compression.

In this cycle of Carnot all the heat supplied is taken in at the upper limit $T_1$ and all the heat rejected is rejected at the lower limit $T_2$.

The greater the range of temperature available the nearer the value of the efficiency approaches unity. The range $(T_1-T_2)$ may be increased by increasing the value of $T_1$ or decreasing the value of $T_2$. There is not much scope for reduction of $T_2$, which is practically fixed by the limiting value of the surrounding atmosphere, or about 60° Fahr. (520° Fahr. abs.), and improvement has been sought by a gradual increase of the upper limit for steam engines and turbines, which in modern practice has now reached the value of 800° Fahr., a figure controlled by the question of safe strength of boiler materials under high stress.

Taking $t_1=800°$ Fahr. or $T_1=1260°$ Fahr. abs., $T_2=520°$ Fahr. abs.

$$\text{(Carnot) } \eta=\frac{1260-520}{1260}=\frac{740}{1260}=0{\cdot}587$$

Thus for these practicable temperature limits the greatest possible efficiency is below 60 per cent.

The actual thermal efficiency does not reach half this value.

**19. Regenerative Cycle.**—The Carnot engine efficiency can also be obtained by substitution of a regenerative heat process for the adiabatic expansion and compression processes. If at the end of isothermal reception the substance is passed through some absorbent body, within the engine, which has a gradient of temperature between $T_1$ and $T_2$ such that at all points of the path there is balance of temperature between it and the substance, then the condition of reversibility of flow will be fulfilled, while the substance is progressively reduced in temperature from $T_1$ to $T_2$. Again, after isothermal rejection at $T_2$ to the cooler, if the substance is passed in the reverse direction through this internal body, the latter will restore the heat previously deposited and the substance will be progressively heated from $T_2$ to $T_1$, the original temperature at the beginning of the cycle.

This internal element, or heat sponge, which alternately stores and restores the heat is called a "regenerator," * and as the two heat quantities are the equivalents of the positive and negative adiabatic work quantities of the Carnot cycle, it follows that the efficiency of the perfectly regenerative cycle is the same as the Carnot efficiency.

Like the Carnot, the completely regenerative cycle is impracticable. It has, however, indicated one method for the improvement of the thermal efficiency of steam engines and turbines, which is now

* The regenerative cycle was first enunciated by Rev. Robert Stirling, minister of Craigie Parish, Ayrshire, in 1827.

being applied, with a fair measure of success, to large modern steam turbine plants.

This is the system of progressive heating of the feed water by means of steam extracted from the engine or turbine, so that the temperature of the feed is raised (by restoration of latent heat from the steam) from the lower condenser limit of temperature to some intermediate temperature before heat is again taken in from the external source (the boiler furnace). (See Art. 64, Chapter IV.)

If it were practically possible to employ an indefinitely large number of heating stages the feed could be raised to the upper limit by this purely "internal" process. All the heat would then

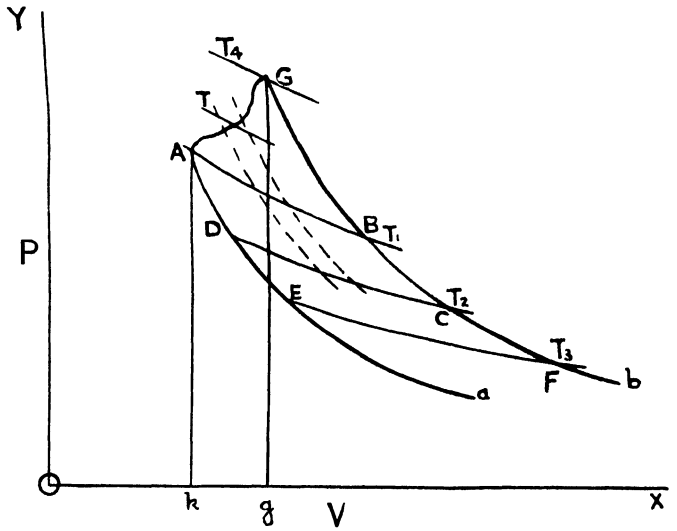



FIG. 7.

be taken in from the source at the upper and rejected to the cooler at the lower limit (assuming reversible conditions) and the maximum (Carnot) efficiency would be obtained.

Up to the present, it has not been considered advisable to employ more than six reheating stages in large turbine plants.

**20. Entropy.**—Consider the Carnot cycle as shown in Fig. 7, bounded by the adiabatics A*a* and G*b*, and the isothermals AB and CD.

Heat taken in along AB at $T_1 = Q_1$.
Heat rejected ,, CD ,, $T_2 = Q_2$

and $$\frac{Q_1 - Q_2}{Q_1} = \frac{T_1 - T_2}{T_1}, \quad \text{from which} \quad \frac{Q_1}{T_1} = \frac{Q_2}{T_2}.$$

Now suppose that the heat rejected is used in another cycle

between $T_2$ and $T_3$, bounded by adiabatics $Gb$ and $Aa$ and isothermals DC and EF, then

$$\frac{Q_2-Q_3}{Q_2}=\frac{T_2-T_3}{T_2} \quad \text{and} \quad \frac{Q_2}{T_2}=\frac{Q_3}{T_3}, \quad \text{so that} \quad \frac{Q_1}{T_1}=\frac{Q_3}{T_3},$$

and so on, for any further temperature differences. Hence at all temperature levels between the given adiabatics $Gb$ and $Aa$, the quotient of the heat taken in or rejected and the absolute temperature of reception or rejection, is a constant,

or
$$\frac{Q}{T}=\text{const.}=\phi$$

This constant ratio is called the " change of entropy " of the substance between the given adiabatics.

Now instead of adding heat at constant temperature $T_1$ along AB, suppose it is added reversibly at a series of increasing temperatures between $T_1$ and some higher temperature $T_4$, so that the locus of the state points is now given by the arbitrary curve AG, the state point G falling on the $T_4$ curve at the adiabatic $Gb$. Suppose that an element of heat $dq$ is added at temperature T between any two adjacent (dotted) adiabatics, then $\frac{dq}{T}$, as seen above, will have the same value as $\frac{dq_1}{T_1}$ at $T_1$ between the same pair of adiabatics, and the total change of entropy between $Aa$ and $Gb$ will be the summation of these elementary changes, so that—

$$\Sigma\frac{dq}{T}=\Sigma\frac{dq_1}{T_1}=\frac{Q}{T}=\phi.$$

If, instead of a series of temperature steps, there is a continuous change of temperature along AG, then, in the limit, this expression is written as—

$$\phi=\int\frac{dq}{T}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (32)$$

The significance of the foregoing deduction is that change of entropy between given adiabatics is independent of the path of the state point, along any arbitrary curve such as AG.

It is to be noted, however, that the quantity of heat required to produce the given change is not the same as that with isothermal reception.

If the substance were expanded to zero pressure and hence infinite volume the adiabatic $Gb$ would meet the axis at X, an infinite distance from O. The whole of the internal energy would be used up, and the area below $GbX$ to infinity thus represents the internal energy of the substance which has its initial state point at G and temperature $T_4$. Similarly the area below the adiabatic $AaX$ gives the internal energy for the substance, with initial state point A at temperature $T_1$.

During the pressure-volume change between A and G the work done by the substance is the area *ka*G*g*.

By the energy law—

$$\begin{aligned}\text{Heat added (along AG)} &= \text{Change of internal energy} + \text{work done}\\ &= (g\text{G}b\text{X} - k\text{A}a\text{X}) + k\text{AG}g\\ &= (g\text{G}b\text{X} + k\text{AG}g) - k\text{A}a\text{X}\\ &= \text{G}b\text{X}a\text{A}\end{aligned}$$

That is, the heat added is given by the indefinite area between the arbitrary curve and the two adiabatics.

In the same way it can be shown that the heat added along AB at $T_1$ is given by the smaller area between AB and the two adiabatics, that is, by B*b*X*a*A.

It will be evident from what has been shown regarding heat reception at varying temperature, that the same conditions will hold for heat rejection at varying temperature, provided the conditions of reversibility are fulfilled throughout the operations, and it follows that, for a completely reversible cycle, the summation of the positive and negative entropy changes will be zero, that is, for a completely reversible cycle—

$$\phi = \int \frac{dq}{\text{T}} = 0 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (33)$$

In the foregoing brief statement an attempt has been made to give the engineering student some idea of the origin of the peculiar quantity called "entropy," which he will find of great practical value for the solution of thermodynamic problems relating to the steam engine and turbine.

Entropy, as can be seen from the foregoing, is not susceptible of direct definition. It can only be indicated in a general way, as follows: "When a substance takes in or rejects heat it is said to change its entropy."

Thus if the substance takes in an element of heat $dq$ at absolute temperature T, the change of entropy is given by-

$$d\phi = \frac{dq}{\text{T}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (34)$$

Multiplying across—

$$dq = \text{T}d\phi \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (35)$$

Thus, change of entropy multiplied by temperature is a quantity of heat. Entropy is usually reckoned per unit mass of the substance, so that its dimension is heat units per degree per unit mass. It is, in fact, an abstract quantity, and entropy change is the same for any scale of temperature. It has never received a definite name, although the late Prof. John Perry suggested the name "rank" for it, as a tribute to Rankine, who first used the entropy function for thermodynamic analysis.

It will be apparent from Equation 35 that just as the area below a pressure-volume curve gives a quantity of work, the area below a temperature-entropy curve will give a quantity of heat. If entropy is taken as abscissa and temperature as ordinate, then curves of constant entropy will be straight lines parallel to the temperature axis. Change of entropy can be measured from any arbitrary adiabatic datum curve, the adiabatic being represented by a constant entropy line on the heat diagram. This is due to the fact that, by definition, since no heat is received from or rejected to external bodies during an adiabatic change, there is no change of entropy. This statement of " adiabatic " condition, however, requires qualification, as will be seen at a later stage, when it becomes necessary to differentiate between adiabatic and isentropic change in the working substance.

In Fig. 8 let OY be zero of entropy, and relative to this let $aL$ be the adiabatic or constant entropy curve corresponding to the PV adiabatic $AaX$ (Fig. 9) and let $g$K be the constant entropy line $\phi_b$ corresponding to the adiabatic $GbX$ (Fig. 9). The entropy change at any temperature level is thus $\phi=(\phi_g-\phi_a)=(\phi_b-\phi_a)=(\phi_c-\phi_a)$ (Fig. 8). The four state points $a$, $b$, $c$ and $d$ correspond to the four PV points A, B, C and D of the Carnot cycle (Fig. 9) between $T_1$ and $T_2$, and the heat area is given by $abcd=(T_1-T_2)\phi=(T_1\phi-T_2\phi)=$(heat received—heat rejected).

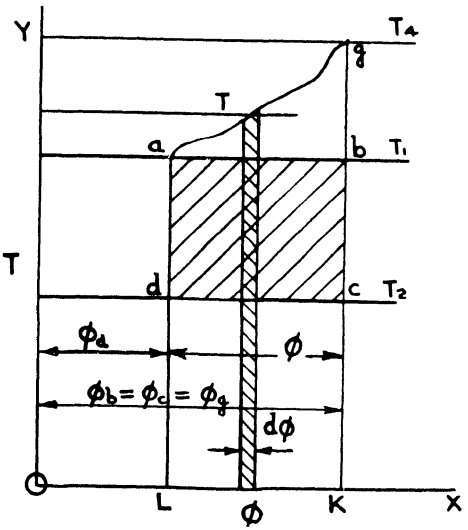



FIG. 8.

Just as $ab$ represents the isothermal AB on the PV diagram (Fig. 9), so does $ag$ correspond to the arbitrary curve AG on this diagram. At temperature T the element of heat taken in between two adjacent adiabatics or constant entropy lines is, as already stated, $dq=Td\phi$, and obviously the summation between $\phi_a$ and $\phi_g$, or the area below $ag$, gives the total amount of heat taken in during the change of temperature from $T_1$ to $T_4$. It will be seen that these two diagrams are complementary. The PV area below AG shows the external work done but gives no indication of heat changes. The $T\phi$ area below $ag$ shows the heat addition but gives no indication of the external work done during the reception.

**21. Elements of a Heat Engine.**—The essential elements of any

In order to determine the volume for any given range of superheat at a given pressure read the value of $r$ corresponding to $t_s$ and multiply

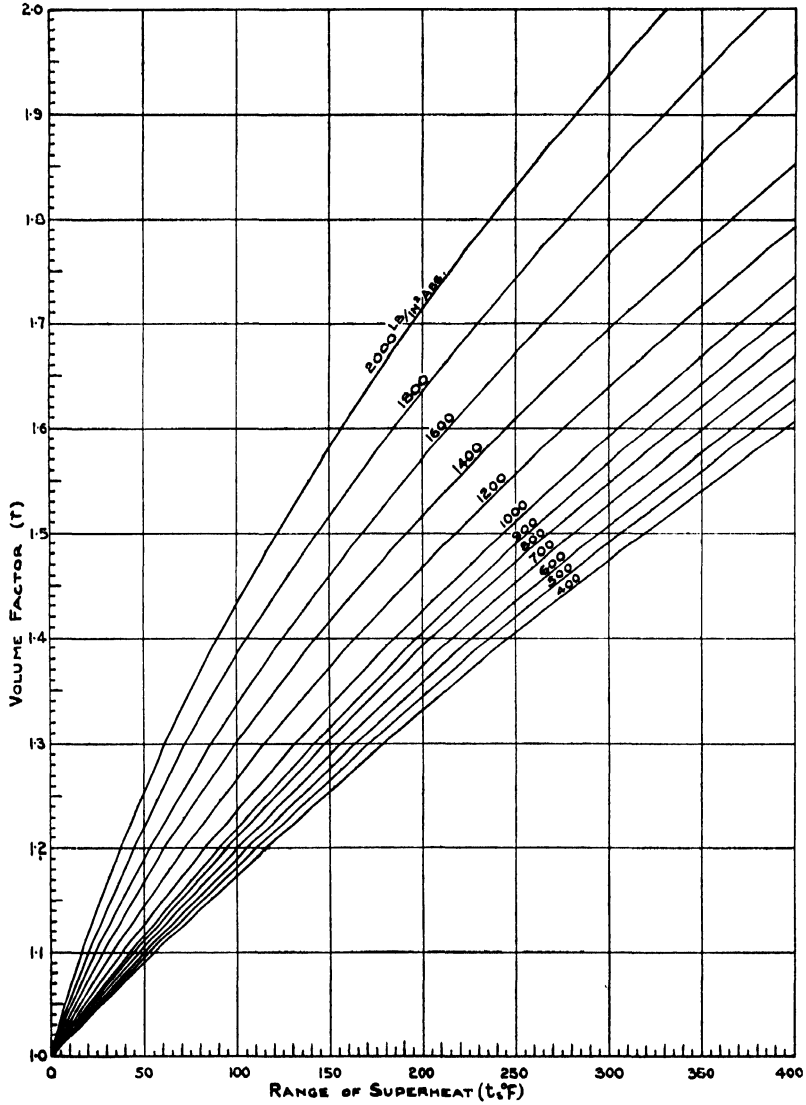



FIG. 15.

the dry steam volume $v$ by this figure. These curves are plotted from the data given in Callendar's (1929) superheated steam tables. It will be noted that between 400 and 1000 lb./in.$^2$ abs. and $t_s =$

# CHAPTER II

## THE PROPERTIES OF STEAM

**23.** THE properties of saturated steam were originally investigated experimentally by Regnault ; and the results of his classical research were embodied in the saturated steam table. The properties of superheated steam, however, remained practically untouched.

Regnault's work was repeated by many later experimenters, with the aid of more modern and improved methods of physical research, and tables based on these results were published from time to time. While for general purposes these tables were fairly satisfactory, they were not thermodynamically consistent. The necessity for a rational basis of calculation for all the properties of steam was recognised by Prof. H. L. Callendar more than thirty years ago, and he instituted an extensive thermodynamic research by means of electrical calorimetric methods, and in 1900 published the results of the investigations in the Proceedings of the Royal Society. His thermodynamic equations were left unnoticed by engineers in this country ; but Prof. Mollier, of Dresden, subsequently adopted and used them for the calculation of a new set of steam table values, which were, in consequence, thermodynamically consistent throughout. In 1915 Prof. Callendar published a table of steam properties for pressures extending up to 500 lb./in.$^2$, and in 1920 his standard work on the *Properties of Steam*. The increasing trend of practice, especially after the European war, towards higher and higher pressures, rendered further research imperative, and Callendar and the leading investigators on the Continent and in America continued to carry out investigations, along various lines, right up to the critical pressure and temperature—that is, to a pressure of 3,200 lb./in.$^2$ abs. and 704° Fahr. The American schemes of research were carried out under the auspices of the American Society of Mechanical Engineers. A good deal of controversy arose regarding the different results obtained, and with a view to reconciling these differences an international conference of physicists met in London in July 1929. The leading investigators from Great Britain, America, Germany, and Czechoslovakia were present. This conference decided to draw up, in the first place, an agreed skeleton table, with tolerances for differences of experimental results, to be made the basis for the compilation of a complete set of international steam tables. A

draft of this skeleton table has been published,* but, so far, the international conference has not issued a complete table for practical use.

Previous to this conference Prof. Mollier issued a new table † based on a modification of the original Callendar equations, and extending up to the critical limit. As the result of his later researches up to this limit, Prof. Callendar also found it necessary to modify his original equations, and an amended table extending from 400 lb./in.$^2$ abs. to 4000 lb./in.$^2$ abs. was published in the Proceedings of the Institution of Mechanical Engineers, May, 1929.‡ This research was carried out with the aid of the British Electrical and Allied Industries Research Association.

The table of properties of saturated steam given at the end of this text has been compiled from the 1929 Callendar results, pending the publication of the agreed International Table.§

A discussion of the involved thermodynamic reasoning on which the Callendar equations for steam properties are based is quite beyond the scope of this intermediate text, and only a concise statement of the latest equations is given at the end of this chapter.‖

**24. Formation of Steam at Constant Pressure.**—From the thermodynamic standpoint, the complete steam engine consists of a boiler, in which heat is added to the water, a working cylinder with a piston for energy conversion, a condenser for rejection of heat, and a feed pump to return the condensate to the boiler. The substance is assumed to circulate round the system, which constitutes a " closed cycle."

In order to obtain a clear idea of the process of steam formation, suppose that the cylinder is used successively as a boiler and working chamber, while unit mass (1 lb.) of substance goes through the cyclic changes of condition. Assume that the operation starts with 1 lb. of water at 32° Fahr., drawn from the condenser and discharged by the pump to the cylinder. Let the piston of unit area (1 ft.$^2$) and loaded with a pressure P=144$p$lb./ft.$^2$, touch the end of the cylinder, so that the initial volume is zero.

If the volume of 1 lb. mass at 32° Fahr. is $b$, the piston will move out a distance $b$=OA (Fig. 10), and an amount of pressure energy P$b$=OAGB, will be given to it by the pump. The pound of water is thus placed in the cylinder at pressure P and 32° Fahr. Heat is now supplied to the cylinder acting as a boiler and transmitted to

* *Engineering*, 6 December, 1929.

† An English translation of this comprehensive table, edited by Dr. H. Moss, has since been published, and also a table by Prof. J. H. Keenan, based on the American research.

‡ See reprint *Engineering*, 3 May, 1929.

§ Since this book went to press revised Callendar Steam Tables (1931), compiled by Dr. H. Moss, have been published by Edward Arnold.

‖ See *Properties of Steam*, by H. L. Callendar. Also "Steam Research in Europe and America," by Prof. Dr. Ing. Max Jacob. *Engineering*, 31 July, 23 October; 20, 27 November; 4, 11, 25 December, 1931.

the water, and the process of formation of steam under constant pressure proceeds as follows :—

(1) The temperature rises until the boiling-point $t$ of the water, corresponding to the pressure $p$, is reached, and the volume increases from $b$ to $u$, while the piston is moved from A to E. An amount of external work $P(u-b)$ is done at the expense of some of the heat supplied. the final volume of the water at the temperature $t$ being $u$.

(2) After the water reaches the boiling-point $t$, which is also termed the " saturation temperature " of the steam at pressure $p$, there is no further rise of temperature. Steam begins to form and the piston moves out against the external pressure.

The water gradually disappears, the weight of steam formed

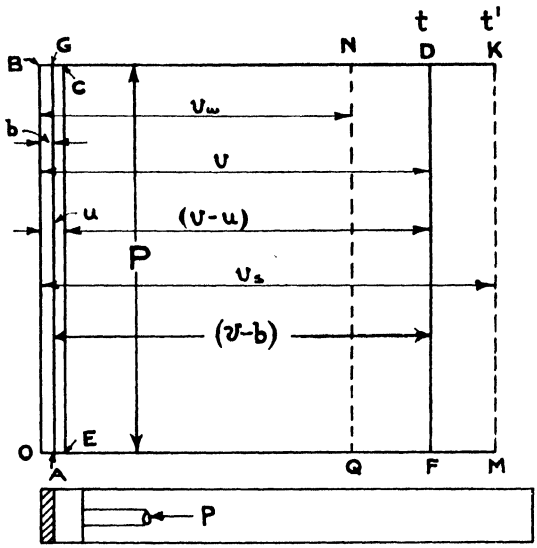



FIG. 10.

corresponding to the weight of water evaporated, till the whole pound has been converted into " dry saturated " steam, and the volume has increased to $v$. The steam during formation remains at the same temperature as the water from which it is produced. The heat added while evaporation takes place is said to become " latent," because the continued supply of heat during formation does not raise the temperature. This heat is absorbed in changing the physical state of the substance from liquid to vapour. Part of it is stored as potential molecular energy in the substance, and part as potential energy in the external system, during increase of volume from $u$ to $v$. The latter part, called the external work during evaporation, is given by the area $ECDF = P(v-u)$.

The saturated steam thus formed is at its greatest density for

the pressure $p$. It is said to be "dry," as it contains no water in suspension.

(3) When the unit mass of water is completely evaporated, that is, converted into dry saturated steam, further supply of heat produces rise of temperature with further increase of volume from $v$ to $v_s$, and performance of additional external work given by the area FDKM=P($v_s$—$v$), where $v_s$ is the volume of the unit mass when a temperature $t'$ is reached. The steam is now "superheated," or at a temperature higher than the saturation temperature at pressure $p$, and is now no longer at its greatest possible density.

**25. Pressure and Temperature of Saturated Steam.**—The temperature of saturated steam in contact with water is the same as that of the water, and there is one temperature only for steam at a given pressure. At any other pressure the temperature has some other value, but always fixed for that particular pressure.

If the temperature falls, then the pressure falls, and a portion of the steam is at the same time condensed; if the temperature increases the pressure also increases, and some of the water present is converted into steam.

In practice the water in certain types of internally fired tank boilers, when circulation is bad, is not all at the same temperature throughout. The temperature of the upper portion above the fire level is that of the steam, while the temperature of that below this level is not necessarily the same. Where this condition obtains the result is unequal expansion of the boiler, and it often gives rise to serious boiler trouble in the case of a tank boiler.

The equation connecting the pressure and temperature of saturated steam is given later. The general relationship between them is shown graphically by the curve of Fig. 11, which is plotted from the values in the Callendar (1929) table between 10 and 2000 lb./in.$^2$ abs. This curve shows that the pressure not only varies with the temperature, but that the rate of change of pressure becomes increasingly greater as the temperature increases.

The rise of pressure per degree rise of temperature between 15 and 2000 lb./in.$^2$ is as follows:—

| $p$ lb./in. abs. . . | 15 | 100 | 200 | 300 | 500 | 1000 | 1500 | 2000 |
|---|---|---|---|---|---|---|---|---|
| Press. rise per degree $dp/dt$ . . . | ·30 | 1·4 | 2·4 | 3·3 | 4·5 | 8·4 | 11·1 | 14·4 |

**26. Specific Heat of Water and Superheated Steam.**—The specific heat of water is not quite constant, but for practical purposes it is sufficient to take it as unity. The value given by Callendar as the minimum specific heat is 0·99666.

In heating 1 lb. of water from 32° Fahr. to any given temperature $t$, the approximate amount of heat supplied at low and moderate pressures may be taken as ($t$—32) B.Th.U./lb., since the external work due to increase of volume is negligibly small. At high pressures, however, this figure is slightly in error. The specific heat

of superheated steam at constant pressure, whereby the superheat for any given range can be estimated, is a variable quantity depending both on the pressure and temperature. The mean values of $c_p$, which increase with pressure and decrease with temperature, can be obtained from the specific heat chart, Figs. 12 and 13, plotted from the Callendar (1929) superheated steam table for pressures ranging from 20 to 2000 lb./in.$^2$ abs. and superheat ranges from 0 to 400° Fahr.

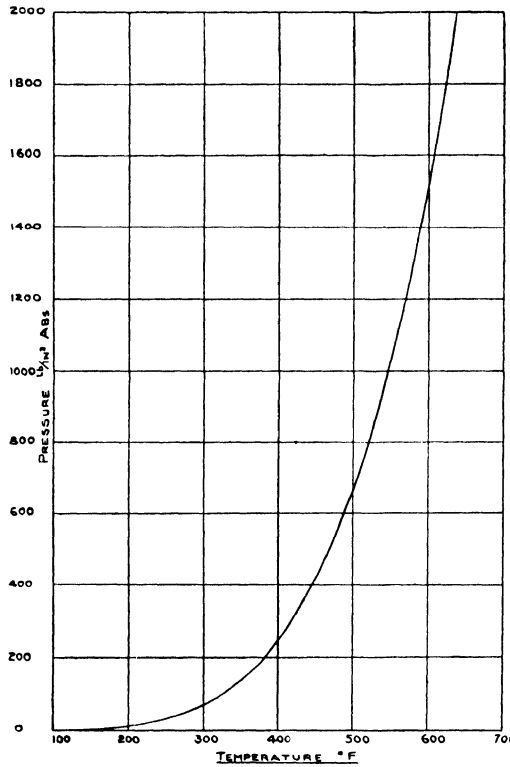



FIG. 11.

If at a given pressure $p$ the saturation temperature is $t$, and the superheated steam temperature is $t'$, the range of superheat is $t_s=(t^1-t)$, and the superheat is given by—

$$h_s=c_p(t'-t)=c_pt_s \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

**27. Total Heat of Water** ($h$).—When the pound of water in the cylinder (Art. 24) is heated from 32° Fahr. to the saturation temperature $t$, the heat thus added goes to increase the internal kinetic molecular energy by an amount $k$, and the internal potential mole-

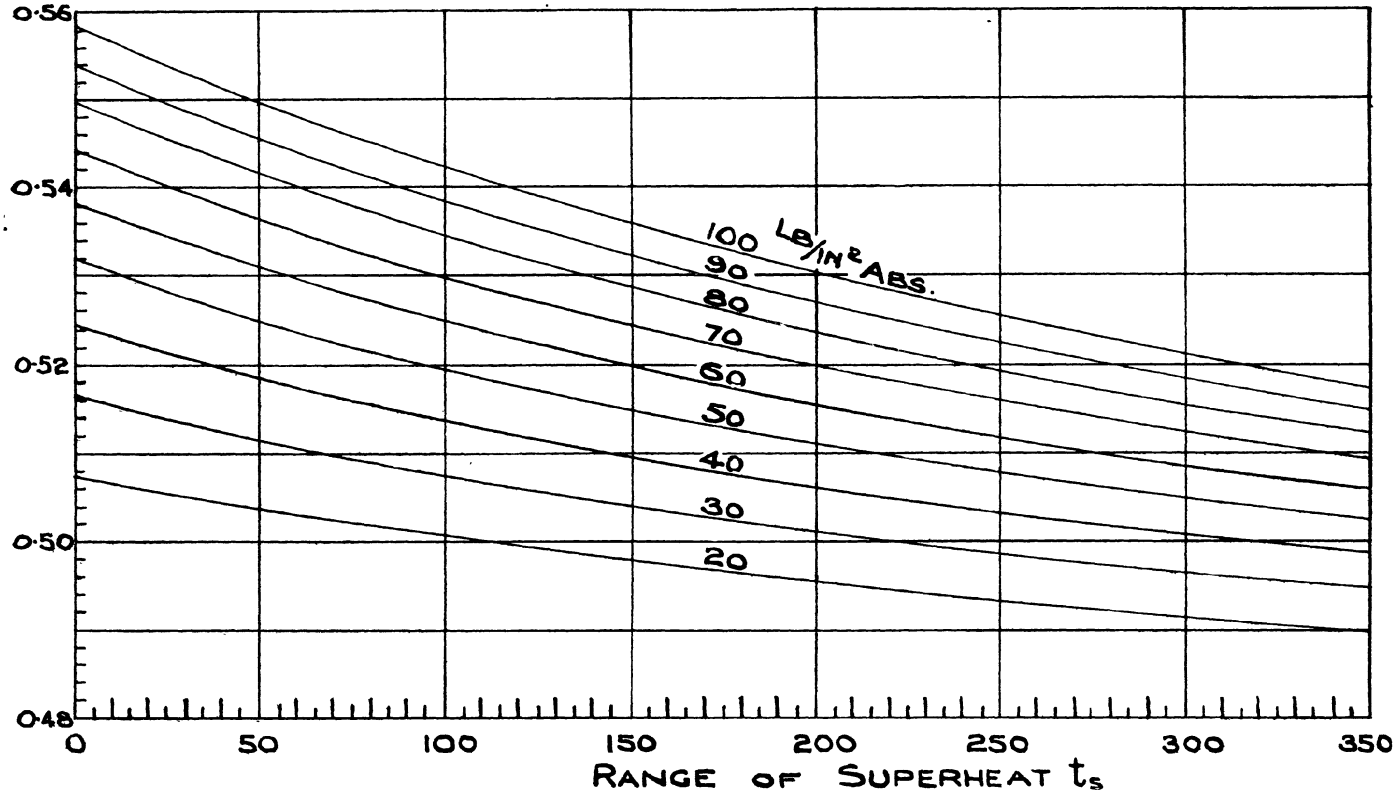

0·56
0·54
0·52
0·50
0·48
100 LB/IN² ABS.
90
80
70
60
50
40
30
20
0
50
100
150
200
250
300
350
RANGE OF SUPERHEAT $t_s$

FIG. 12.

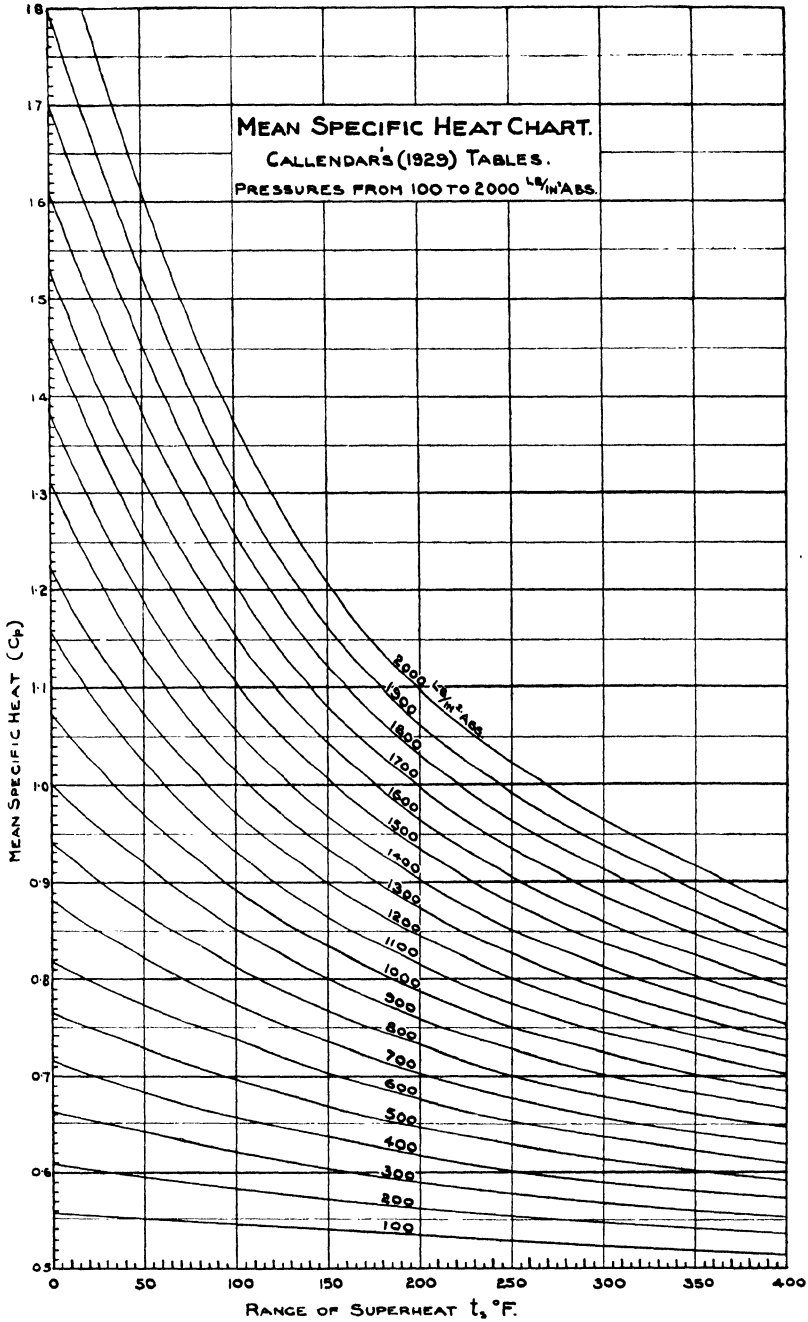

MEAN SPECIFIC HEAT CHART.
CALLENDAR'S (1929) TABLES.
PRESSURES FROM 100 TO 2000 LBS/IN² ABS.
MEAN SPECIFIC HEAT ($C_p$)
RANGE OF SUPERHEAT $t_s$ °F.
2000 LBS/IN² ABS.
1900
1800
1700
1600
1500
1400
1300
1200
1100
1000
900
800
700
600
500
400
300
200
100
1·8
1·7
1·6
1·5
1·4
1·3
1·2
1·1
1·0
0·9
0·8
0·7
0·6
0·5
0
50
100
150
200
250
300
350
400

FIG. 13.

cular energy by an amount $l$, and to do an amount of external work $\frac{P}{J}(u-b)$, in heat units. The heat added in raising the temperature at pressure P is thus—

$$q=(k+l)+\frac{P}{J}(u-b) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

The sum of the kinetic and potential molecular energy $(k+l)$ is the increase $\rho_w$ of the internal energy of the water (Art. 1), so that—

$$q=\rho_w+\frac{P}{J}(u-b) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

According to the older system of definition, this quantity has been called the "sensible" or "liquid heat." At low pressures the second term is negligibly small, and for all practical purposes the liquid heat is equal to the change of internal energy between 32° Fahr. and $t$, or $q=\rho_w=(t-32)$. As already stated in (Art. 26), this is not quite the case at high pressures.

In (Art. 24) it has been shown that in order to get the water into position in the boiler for heating from 32° Fahr. to $t$ at pressure $p$, an amount of pressure energy $Pb/J$ (in heat units) is first supplied by the pump. It has now become the practice to add this energy to the heat taken in to raise the temperature, and the increased quantity $(q+Pb/J)$ is defined as the total heat of the water,* denoting it by $h$—

$$h=q+\frac{Pb}{J}$$

$$=\rho_w+\frac{P}{J}(u-b)+\frac{P}{J}b$$

$$\text{or } h=\rho_w+\frac{Pu}{J} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

The total heat of the water is thus the sum of the internal energy increase between 32° Fahr. and $t$° Fahr., and the product of the pressure and the volume of the water at the saturation temperature $t$ (expressed in heat units).

The value of the total heat of the water calculated by the Callendar thermodynamic equation can be obtained from the steam table.

*Example* 1.—If steam is generated in a boiler at 1000 lb./in.$^2$ abs., find, with the aid of the saturated steam table, the internal energy of the water and compare it with the value by the approximate method. Also find the pressure energy of the water. Take specific volume of water as 0·0218.

* Some engineers call this the "total energy."

Here $h=\rho_w+\frac{144pu}{J}$ $\quad p=1000$, $u=0{\cdot}0218$ (table), $J=778$, $h=543{\cdot}6$

$$\therefore 543{\cdot}6=\rho_w+\frac{144\times1000\times0{\cdot}0218}{778}$$

$$\therefore \rho_w=543{\cdot}6-\frac{144\times1000\times0{\cdot}0128}{778}$$

$$=(543{\cdot}6-4{\cdot}03)=539{\cdot}57 \text{ B.Th.U./lb.}$$

$t=544{\cdot}1$ (table)

Approximate value $\rho_w=(t-32)=(544{\cdot}1-32)=512{\cdot}1$ B.Th.U./lb.
Difference 27·47 or 5·1 per cent. in defect.

Pressure energy $=\frac{144pu}{J}=\frac{144\times1000\times0{\cdot}0218}{778}=4{\cdot}03$ B.Th.U./lb.

**28. Specific Volume of Dry Steam** ($v$).—The exact value is calculable from Callendar's fundamental thermodynamic equation; but between 1 lb./in.$^2$ abs. and 200 lb./in.$^2$ abs. an approximation sufficient for practical purposes can be obtained from the simpler expression—

$$pv^{1\cdot3}=490$$

$$\text{or } v=\left(\frac{490}{p}\right)^{1\cdot3} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

It should not be used for pressures above 200 lb./in.$^2$ abs.

As a rule, however, calculation of dry steam volume is unnecessary, as it can be obtained either directly or by interpolation, from the saturated steam table.

*Example* 2.—The dry steam volume given in the table for 1 lb./in.$^2$ abs. is 333·1 ft.$^3$/lb., and for 1800 lb./in.$^2$ abs. 0·2255 ft.$^3$/lb. Find in each case the error involved by the use of the simple approximate equation (5).

At 1 lb./in.$^2$ abs. $v=\left(\frac{490}{1}\right)^{1\cdot3}=332{\cdot}7$ ft.$^3$/lb.

The difference is 0·4 or about 0·12 per cent. in defect.

At 1800 lb./in.$^2$ abs. $v=\left(\frac{490}{1800}\right)^{1\cdot3}=\frac{1}{3{\cdot}388}=0{\cdot}295$ ft.$^3$/lb.

The difference is 0·06950 or about 30 per cent. in excess.

**29. Latent Heat of Steam** (L).—This, as already seen from (Art. 24), is the amount of heat added to convert 1 lb. of water at pressure $p$ and temperature $t$ into dry saturated steam at that temperature.

During this change of physical state a part $\rho$ of this heat is stored as internal potential energy of the molecules of the substance, and is called the internal latent heat, while a part is expended in doing external work, $\frac{P}{J}(v-u)$.

This latter represents potential energy stored in the external system, or external latent heat, and it does not represent heat present in the substance. It is, however, recoverable energy when the steam is condensed at pressure $p$ and the volume is reduced from $v$ to $u$. Thus—

$$L=\rho+\frac{P}{J}(v-u) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

and the internal latent heat is given by

$$\rho=L-\frac{P}{J}(v-u) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

*Example* 3.—Calculate the external and internal latent heat of dry steam at 1600 lb./in.$^2$ abs. Take the specific volume of the water as 0·024 ft.$^3$/lb.

External latent heat $=\frac{P}{J}(v-u)$ $\quad v=0{\cdot}2635$, $u=0{\cdot}024$, $p=1600$, $L=548{\cdot}7$

$=\frac{144\times1600\times0{\cdot}2395}{778}$

$=70{\cdot}92$ B.Th.U./lb.

Internal latent heat $\rho=L-\frac{P}{J}(v-u)$

$=(548{\cdot}7-70{\cdot}92)=477{\cdot}78$ B.Th.U./lb.

**30. Total Heat of Dry Steam** (H).—According to the older definition this was the amount of heat necessary to raise the water initially at 32° Fahr. to the saturation temperature $t$ at pressure $p$, and to convert it into dry steam at this temperature. It was then given by the sum of the liquid heat and the latent heat.

By the amended definition the total heat of the steam is now the sum of the total heat of the water and the latent heat of the steam, or

$$H=h+L \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

Substituting for $h$ from (4) and for L from (6)—

$$H=\rho_w+\frac{Pu}{J}+\rho+\frac{P}{J}(v-u)$$

$$=(\rho_w+\rho)+\frac{P}{J}v \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

Now $(\rho_w+\rho)$ is the total increase of internal molecular energy of 1 lb. of substance during the whole formation process, or the total internal energy of the steam, which may be denoted by I. Hence

$$H=I+\frac{Pv}{J} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (10)$$

That is, the total heat of dry steam is the sum of the increase of internal energy between 32° Fahr. and $t$ and the product of the pressure and specific volume of the dry steam.

During inflow of heat and formation of 1 lb. of dry steam the total amount of external work done (external potential energy) is—

$$\underset{\text{Heating of water.}}{\frac{P}{J}(u-b)} + \underset{\text{Evaporation.}}{\frac{P}{J}(v-u)} = \frac{P}{J}(v-b) = \frac{E}{J} = h_e \quad . \quad . \quad (11)$$

in heat units.

The total amount of internal energy increase being $(\rho_w + l)$ the total heat can be alternatively expressed as

$$H = I + h_e + \frac{P}{J}b \quad . \quad . \quad . \quad . \quad . \quad . \quad (12)$$

Values of latent heat and total heat also specific volume ($v$) of dry steam can be obtained from the saturated steam table. Values of internal energy are calculable from Equation 10.

*Example* 4.—Calculate the internal energy of dry steam at 500 lb./in.$^2$ abs., and the total external work of formation.

Here $H = I + \frac{144pv}{J}$ $\qquad$ $H = 1212{\cdot}4$

$\therefore\ I = 1212{\cdot}4 - \frac{144 \times 500 \times 0{\cdot}938}{778}$ $\qquad$ $v = 0{\cdot}938$, $p = 500$

$= (1212{\cdot}4 - 86{\cdot}82) = 1125{\cdot}58$ B.Th.U./lb.

Total external work $E = 144p(v-b)$ $\qquad$ $b = 0{\cdot}016$

$= 144 \times 500 \times 0{\cdot}922$ $\qquad$ $(v-b) = (0{\cdot}938 - 0{\cdot}016) = 0{\cdot}922$

$= 66384$ ft.-lb.

or $h_e = \frac{E}{J} = \frac{66384}{778} = 85{\cdot}31$ B.Th.U./lb.

**31.** The heat quantities considered in the foregoing sections have been calculated for pressures of 1, 15, 100, 500, 1000, 1500, and 2000 lb./in.$^2$ abs., and are collected in Table I. The curves of total heat (H), latent heat (L) and ($\rho$), and internal energy I are plotted to a base of temperature in °Fahr. in Fig. 14.

The intercept between the L and $\rho$ curves gives the external work done during the evaporation period, or $\frac{P}{J}(v-u)$, while that between the H and I curves gives the total external work $h_e$ plus the pressure energy of the water,

or $$\left(h_e + \frac{Pb}{J}\right) = \frac{Pv}{J}$$

The temperature scale has been extended to the critical value of 704° Fahr., at which value, as will be seen later, the latent heat becomes zero.

Taking these quantities in the order given in the table.

(2) The temperature of the feed is taken at the standard value of 32° Fahr., from which all heat values in the tables are reckoned.

(3) The saturation temperature $t$ increases with the pressure, as has already been shown graphically (Fig. 11): at 15 lb./in.$^2$ abs. the rate of increase is 3° Fahr. per lb. rise of pressure, while at 2000 lb./in.$^2$ it is 0·07° Fahr. per lb. increase. These decreasing rates are the reciprocals of the rates given in Art. 25.

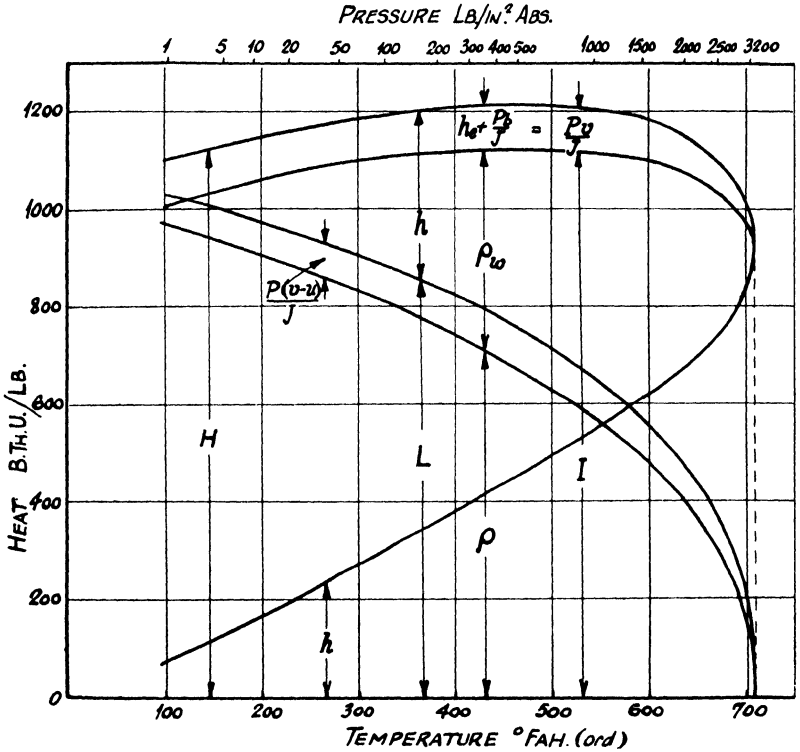



FIG. 14.

(4) The specific volume of the steam ($v$) decreases with the increase of pressure, that is the density progressively increases with pressure.

It will be noted that at 1 lb./in.$^2$ abs., corresponding to a vacuum of 27·96 in. mercury, the volume is 333 ft.$^3$/lb., while at atmosphere pressure, or 15 lb./in.$^2$ abs., it is 26·27 or only about 8 per cent. of the value at 1 lb./in.$^2$ abs., while at higher pressures, 500 lb./in.$^2$ abs. and upwards, it is only a small fraction of a per cent. of the first value. At 29·1 in. vacuum, a common figure in central power stations, the volume runs about 650 ft.$^3$/lb. This excessively large volume at very low pressure give rise to considerable difficulties

TABLE I.

| | | | | | | | |
|---|---|---|---|---|---|---|---|
| 1. Pressure $p$ lb./in.$^2$ abs. | 1 | 15 | 100 | 500 | 1000 | 1500 | 2000 |
| 2. Feed temperature $t_f$ °F. | 32 | 32 | 32 | 32 | 32 | 32 | 32 |
| 3. Saturation temperature $t$ °F. | 101·7 | 213 | 327·4 | 466·0 | 544·1 | 596·1 | 636·5 |
| 4. Specific volume steam $v$ ft.$^3$/lb. | 333·1 | 26·27 | 4·428 | 0·938 | 0·457 | 0·2873 | 0·1936 |
| 5. Specific volume water $u$ ft.$^3$/lb. | 0·0161 | 0·0167 | 0·0177 | 0·0199 | 0·0218 | 0·0236 | 0·0259 |
| 6. Total heat water $h$ B.Th.U./lb. | 69·5 | 181 | 298·3 | 451·2 | 543·6 | 614·73 | 676·8 |
| 7. Total heat steam H B.Th.U./lb. | 1102·4 | 1151·2 | 1190·6 | 1212·4 | 1203·4 | 1181·4 | 1149·8 |
| 8. Latent heat steam L B.Th.U./lb. | 1032·9 | 970·2 | 892·3 | 761·2 | 659·8 | 567·1 | 473·0 |
| 9. Internal latent heat $\rho$ B.Th.U./lb. | 971·03 | 899·32 | 810·65 | 676·2 | 579·23 | 493·87 | 410·9 |
| 10. External heat $\frac{P}{J}(v-b)=he$ B.Th.U./lb. | 61·87 | 72·88 | 81·65 | 85·0 | 80·57 | 73·23 | 62·10 |
| 11. $\frac{100he}{H}$ | 5·612 | 6·33 | 6·86 | 7·011 | 6·695 | 6·199 | 5·40 |
| 12. Pressure energy steam $144\frac{pv}{J}$ B.Th.U./lb. | 61·63 | 72·92 | 81·98 | 86·83 | 84·61 | 79·79 | 71·69 |
| 13. Internal energy $I=\left(H-\frac{pv}{J}\right)$ B.Th.U./lb. | 1040·77 | 1078·28 | 1108·62 | 1125·57 | 1118·79 | 1101·61 | 1078·11 |
| 14. Pressure energy water $144\frac{pb}{J}$ B.Th.U./lb. (at 32° F.) | 0·00296 | 0·0445 | 0·2965 | 1·481 | 2·965 | 4·447 | 5·93 |

in the design of the L.P. ends of large steam turbines. On the other hand the very small volumes at high pressures give rise to similar difficulties in the design of the small high-speed super-pressure type of turbine.

(5) The specific volume of the water ($u$) at saturation temperature increases over the range between 1 and 2000 lb./in.$^2$ abs. by 60 per cent.

(6) The total heat of the water ($h$) increases with the pressure. The rate of increase with temperature is nearly constant over the range from 1 to 2000 lb./in.$^2$ abs. At 1 lb./in.$^2$ abs. it is 1 B.Th.U. per degree increase, and at 2000 lb./in.$^2$ abs. it is 1·36 B.Th.U. per degree increase.

In the case of a tank boiler of Lancashire or Scotch marine type working at fairly high pressure, it can be seen that a considerable store of heat is carried by the water. The usual limit of pressure for this type is about 250 lb./in.$^2$ abs., for which $h$=376·3B.Th.U./lb., so that it has a store of (376·3—181)=195·3 B.Th.U./lb. in excess of what it would have at atmospheric pressure. If the pressure in such a boiler falls from $p_1$ to $p_2$, the total heat of the water changes from $h_1$ to $h_2$, and the difference of the heats is available for evaporation at constant pressure. If $L_2$ is the latent heat at $p_2$, the amount of water turned to steam at $p_2$ due to the drop is $w=(h_1 - h_2)/L_2$. This condition is taken advantage of in exhaust steam accumulators. If the steam in a shell boiler is raised to some pressure above atmospheric, but with all the valves closed, the surface of the water remains quiescent during the process. When, however, the stop valve is opened or the safety valve lifted, or the pressure in any other way is relieved, more or less violent ebullition immediately takes place, due to the fact that the heat stored in the water is in excess of that required at the reduced pressure, and this liberated heat evaporates some of the water. A similar effect is produced in the case of water in a steam engine cylinder when the pressure is reduced by expansion or during exhaust. The heat present in the water at the initial pressure and temperature exceeds that which the water can retain at the lower pressure, hence a portion of this water is re-evaporated as soon as the pressure falls.

(7) The total heat of the steam increases with the pressure, but to a small extent, between 1 and 2000 lb./in.$^2$ abs. the increase is 47·4 B.Th.U./lb. or 4·3 per cent. It can be seen from this fact that the cost or heat expenditure per pound of very high pressure steam is little more than that of low pressure steam. The practical significance of this is shown in Chapter XV. The amount of energy which can be converted into useful work during expansion in a cylinder or a nozzle for the same expenditure of heat in production of the steam is greater in the case of the high pressure steam. Thus, with ideal adiabatic expansion from 1000 lb./in.$^2$ abs. to 1 lb./in.$^2$ abs. the theoretical energy available is 420 B.Th.U./lb., while that from expansion from 100 lb./in.$^2$ to 1 lb./in.$^2$ abs. is 292·0 B.Th.U./lb.

The additional heat available for work is 128 B.Th.U./lb., while the additional heat for the formation of steam at 1000 lb./in.$^2$ abs. over that at 100 lb./in.$^2$ abs. is 12·8 B.Th.U./lb. This is the ideal, but the real advantage is much less, and, moreover, there are practical difficulties attendant on the use of very high pressure steam which do not arise with low pressures, and somewhat discount the advantage. On the whole, however, the use of high pressure steam is a paying proposition, and within the past ten years the pressure limits have risen from 200 lb./in.$^2$ abs. till from 600 to 800 lb./in.$^2$ abs. is now a usual figure in large power stations. In one or two recent cases on the Continent and in America installations working at pressures from 1200 to 3000 lb./in.$^2$ abs. are in operation. These high pressures are being used with as high a superheat temperature as can be safely permitted with regard to the strength of the boiler materials.

(8 and 9) While the total heat alters little with increase of pressure the latent L and $\rho$ decrease. L is 45 per cent. and $\rho$ 50 per cent. less at 2000 lb./in.$^2$ abs. than at 1 lb./in.$^2$ abs. They both vanish at the critical pressure of 3200 lb./in.$^2$ abs., and temperature of 704° Fahr. The external work done during evaporation $\frac{P}{J}(v-u)$ varies little between 1 and 2000 lb./in.$^2$ abs., but rapidly diminishes beyond this limit and becomes zero at 3200 lb./in.$^2$ abs., where $v=u$.

(10 and 11) The total external heat during the whole formation process or $\frac{P}{J}(v-b)$ is the figure of importance as it shows up clearly the inefficiency of the process when steam is used to do work non-expansively. As already seen from Fig. 10, this represents the work done against the external resistance by the piston under the action of the gradually forming steam. The condition is the same as if the cylinder were connected by a steam pipe to the boiler.

As a percentage of the total heat of the steam this heat value varies from 5·6 per cent. to 5·75 per cent. between 1 lb./in.$^2$ abs. and 2000 lb./in.$^2$ abs.—a very small proportion. This work, as already pointed out, is due to the continuous supply of heat to the substance isothermally in the boiler. This heat is immediately transformed into work and never exists as heat in the steam. Suppose, however, a much longer cylinder is employed and the steam supply is shut off after the pound of dry steam is admitted, then any further expansion work done against the external resistance must be done at the expense of the internal energy, and a much larger proportion of the total heat can be converted to useful work.

For efficient utilisation of the heat of the steam, it should be used with as great a range of expansion as possible. In the reciprocating engine there is a definite limit to the extent of the expansion, fixed by economic considerations; but in the case of the steam turbine,

the same conditions do not operate to the same extent and very wide ranges of expansion are employed.

(12) The pressure energy of the steam $144pv/J$ is of the same order as the external heat, but increases slightly as the pressure increases. Its value is about 18 per cent. greater at 2000 lb./in.$^2$ than at 1 lb./in.$^2$ abs.

(13) The internal energy, like the total heat, increases slightly with the pressure. It is about 9 per cent. greater at 2000 lb./in.$^2$ abs. than at 1 lb./in.$^2$ abs.

(14) It will be seen that the pressure energy of the water $Pb/J$ is negligible at 1 lb./in.$^2$ abs., hardly appreciable at moderate pressures, while at 2000 lb./in.$^2$ abs. it only represents about 0·5 per cent. of the total heat.

So far this discussion has been confined to dry saturated steam. Steam, however, can exist in four states. (1) dry; (2) wet; (3) superheated; (4) supersaturated or undercooled.

**32. Wet Steam.**—If the supply of heat is stopped when a fraction $q$ of the pound of water is converted to steam the cylinder will contain a volume $v_w$ of steam and unevaporated water (Fig. 10). The external work done during this fractional evaporation is given by the area $ECNQ=P(v_w-u)$.

The total heat of the water is the same as that for dry steam, but only the fraction $q$L of the latent heat will be absorbed. The steam is said to be "wet saturated." The fraction $q$ of the pound of water converted to steam is called the "dryness fraction" and $(1-q)$ the "wetness fraction" of the steam. Usually the dryness is spoken of as the "quality," but in this text the term quality is used in the wider sense to denote the condition either of dry, wet, superheated, or undercooled steam.

The methods of obtaining approximate steam dryness experimentally are discussed in Arts. 37, 38, 40.

In this case the total heat of wet saturated steam is—

$$\left.\begin{aligned} H_w&=h+qL \\ \text{or } H_w&=H-(1-q)L \end{aligned}\right\} \quad . \quad . \quad . \quad . \quad (13)$$

Expressed in terms of the internal energy and pressure energy, this is—

$$H_w=I_w+\frac{P}{J}v_w \quad . \quad . \quad . \quad . \quad . \quad . \quad (14)$$

where $v_w$ is the specific volume of the wet steam.

From Equation 14 the internal energy is calculable when $H_w$ and $v_w$ are known—

$$I_w=H_w-\frac{P}{J}v_w \quad . \quad . \quad . \quad . \quad . \quad . \quad (15)$$

Alternatively it can be calculated from—

$$I_w=I-(1-q)\left(L-\frac{Pv}{J}\right) \quad . \quad . \quad . \quad . \quad (16)$$

$$\text{or } I_w=qI+(1-q)\rho_w \quad . \quad . \quad . \quad . \quad . \quad . \quad (17)$$

**33. Specific Volume of Wet Steam** ($v_w$).—This is directly calculable, for a given dryness fraction, from the specific volume ($v$) of the dry steam given in the tables.

With $q$ dryness the volume of steam formed is $qv$, but in addition to this there remains in the cylinder the volume $(1-q)u$ of unevaporated water. The sum of the quantities is the volume of the 1 lb. of $H_2O$ stuff partly steam and partly water in suspension, or—

$$v_w=qv+(1-q)u \quad . \quad . \quad . \quad . \quad . \quad (18)$$

As can be seen from Table I, except at very high pressures, the volume $u$ of the water is so small that the second term of (18) becomes negligible.

Steam at high pressure is almost invariably used with more or less superheat, so that the question of dryness seldom arises. At moderate and low pressures the simpler expression—

$$v_w=qv \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (19)$$

is quite sufficient for practical purposes. Equation 18 can be used for any doubtful case at high pressure.

*Example* 5.—The steam supplied by a boiler at 500 lb./in.$^2$ abs. is 5 per cent. wet. Calculate its total heat, pressure energy, and internal energy.

Total heat $H_w=H-(1-q)L$ $\quad H=1212{\cdot}4$

$=1212{\cdot}4-0{\cdot}05\times761{\cdot}2$ $\quad L=761{\cdot}2$

$=1212{\cdot}4-38{\cdot}06$ $\quad v=0{\cdot}938$

$=1174{\cdot}34$ B.Th.U./lb. $\quad q=0{\cdot}95$

Sp. vol. $v_w=qv=0{\cdot}95\times0{\cdot}938=0{\cdot}891$.

$$\text{Pressure energy}=\frac{144pv_w}{J}=\frac{144\times500\times0{\cdot}891}{778}=82{\cdot}45 \text{ B.Th.U./lb.}$$

$$\text{Internal energy } I_w=H_w-\frac{144pv_w}{J}$$

$=(1174{\cdot}34-82{\cdot}45)=1091{\cdot}89$ B.Th.U./lb.

$$\text{Alternatively } I_w=I-(1-q)\left(L-\frac{144pv}{J}\right)$$

$$\frac{144pv}{J}=\frac{144\times0{\cdot}938\times500}{778}=86{\cdot}82$$

$I=(1212{\cdot}4-86{\cdot}82)=1125{\cdot}58$ B.Th.U./lb.

$\therefore\ I_w=1125{\cdot}58-0{\cdot}05(761{\cdot}2-86{\cdot}82)$

$=(1125{\cdot}58-33{\cdot}719)=1091{\cdot}86$ B.Th.U./lb.

The third alternative method is not quite so reliable, as any slight error in $u$ has an appreciable effect on $\rho_w$—

$I_w=qI+(1-q)\rho_w$ $\qquad \rho_w=h-\dfrac{144\times pu}{J}$

$=0{\cdot}95\times1125{\cdot}58+0{\cdot}05\times432{\cdot}8$ $\qquad =451{\cdot}2-\dfrac{144\times500\times0{\cdot}0199}{778}$

$=1069{\cdot}3+21{\cdot}64$ $\qquad =(451{\cdot}2-18{\cdot}4)=432{\cdot}8$

$=1090{\cdot}94$ B.Th.U./lb. instead of 1091·89, about 0·1 per cent. less.

*Example* 6.—Dry steam at 100 lb./in.$^2$ abs. is admitted to a cylinder having a stroke volume of 3·194 ft.$^3$, and no clearance. Cut-off takes place at 0·347 × stroke. After adiabatic expansion the terminal pressure is 30 lb./in.$^2$ abs. Find the weight of steam supplied, the quality at the end of expansion, and the decrease of internal energy per lb.

Stroke vol. V=3·194. Sp. vol. at $p_1$=100, $v_1$=4·428. Cut-off $x$=0·347.

$$\therefore \text{ wt. of supplied } w=\frac{x\mathrm{V}}{v_1}=\frac{0\cdot347\times3\cdot194}{4\cdot428}=0\cdot25 \text{ lb./stroke}$$

Let $q_2$=quality at $p_2$=30 and V=3·194, then

$$q_2v_2w=\mathrm{V} \text{ and } q_2=\frac{\mathrm{V}}{v_2w}=\frac{3\cdot194}{13\cdot74\times0\cdot25}=0\cdot93 \qquad \left.\begin{matrix}\\ \\ \end{matrix}\right\} v_2=13\cdot74$$

Decrease of internal energy—

$$(\mathrm{I}_1-\mathrm{I}_2)=(\mathrm{H}_1-\mathrm{H}_{w2})-\tfrac{1}{778}(p_1v_1-p_2q_2v_2) \qquad \left.\begin{matrix}\mathrm{H}_1=1190\cdot6\\ \mathrm{H}_{w2}=1099\cdot3\end{matrix}\right.$$

$$=(1190\cdot6-1099\cdot3)-\tfrac{1}{778}(100\times4\cdot428-30\times0\cdot93\times13\cdot74)$$

$$=91\cdot3-\tfrac{1}{778}\times59\cdot5=(91\cdot3-11)=80\cdot3 \text{ B.Th.U./lb.}$$

**34. Total Heat of Superheated Steam.**—As already stated in Art. 24, after the pound of water has been converted to dry saturated steam, further addition of heat superheats or raises the steam to some temperature $t'$ greater than the saturation value $t$.

The amount of superheat is given by

$$h_s=c_p(t'-t)=c_pt_s \quad . \quad . \quad . \quad . \quad . \quad (20)$$

where $c_p$ is the mean specific heat for the range $t_s$, as given by the specific heat chart (Figs. 12 and 13).

The total heat of the superheated steam is the total heat of the dry steam plus the superheat, or

$$\mathrm{H}_s=\mathrm{H}+h_s \quad . \quad . \quad . \quad . \quad . \quad . \quad (21)$$

If $\mathrm{I}_s$ and $v_s$ are the internal energy and specific volume of the superheated steam, then

$$\mathrm{H}_s=\mathrm{I}_s+\frac{\mathrm{P}v_s}{\mathrm{J}} \quad . \quad . \quad . \quad . \quad . \quad . \quad (22)$$

The internal energy is thus calculable from

$$\mathrm{I}_s=\left(\mathrm{H}_s-\frac{\mathrm{P}v_s}{\mathrm{J}}\right) \quad . \quad . \quad . \quad . \quad . \quad (23)$$

**35. Specific Volume of Superheated Steam.**—This is calculable exactly by the Callendar thermodynamic equation given later. Between $p$=400 and $p$=2000 a sufficiently close approximation can be obtained with the aid of the curves shown in Fig. 15. These curves give the value of the ratio $r=\frac{v_s}{v}$, where $v_s$=specific volume of the superheated and $v$=the specific volume of the dry steam.

In order to determine the volume for any given range of superheat at a given pressure read the value of $r$ corresponding to $t_s$ and multiply

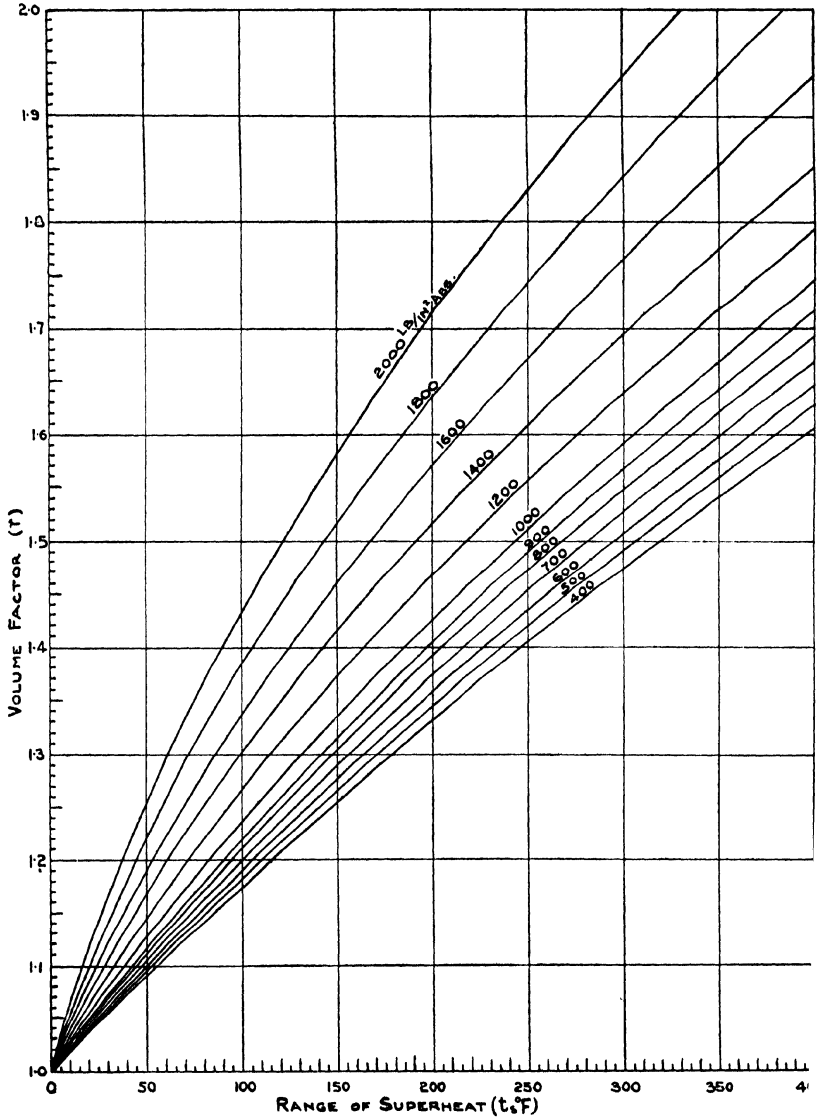



FIG. 15.

the dry steam volume $v$ by this figure. These curves are plotted from the data given in Callendar's (1929) superheated steam tables. It will be noted that between 400 and 1000 lb./in.$^2$ abs. and $t_s$=

100° Fahr. and $t_s$=400° Fahr., these curves are nearly straight lines. Between 20 lb./in.$^2$ abs. and 400 lb./in.$^2$ abs., the curves are crowded into a narrow strip and are all practically straight lines having a common slope.

Between 20 lb./in.$^2$ abs. and 400 lb./in.$^2$ abs., therefore, the specific volume of the superheated steam can be determined with quite sufficient accuracy on the assumption that the volume factor $r$ is a linear function of the range of superheat at any pressure between these limits, that is,

$$r=(1+at_s) \quad . \quad . \quad . \quad . \quad . \quad . \quad (24)$$

The average value of the coefficient $a$ is 0·0015. The error involved by this simple method of calculation does not as a rule exceed 1 per cent. The variation is from 0·5 to 1 per cent. Where extreme accuracy is required, then the tedious calculation from the thermodynamic equation is necessary.

The approximate specific volume between 20 lb./in.$^2$ abs. and 400 lb./in.$^2$ abs. can thus be determined from

$$v_s=v(1+0{\cdot}0015t_s) \quad . \quad . \quad . \quad . \quad . \quad (25)$$

*Example* 7.—The specific volume of steam at $p$=1400 lb./in.$^2$ abs. and 313° Fahr. superheat given in the Callendar table is 0·534 ft.$^3$/lb. Compare this figure with the value obtained by means of the chart, Fig. 15.

At $p$=1400, $r$=1·72. From table, $v$=0·3111

$\therefore$ $v_s=rv=1{\cdot}72\times0{\cdot}3111=0{\cdot}535$ as against 0·534.

*Example* 8.—Find the specific volume of steam at 200 lb./in.$^2$ abs. and 308·2° Fahr. superheat.

From table, $v$=2·29. Applying Equation 25—

$$v_s=v(1+0{\cdot}0015t_s)=2{\cdot}29(1+0{\cdot}0015\times308{\cdot}2)$$
$$=2{\cdot}29\times1{\cdot}4623=3{\cdot}348 \text{ ft.}^3\text{/lb.}$$

*Example* 9.—Calculate the total heat, the pressure energy, and the internal energy of steam at 1400 lb./in.$^2$ abs. and 313° Fahr. superheat.

Referring to the specific heat chart, Fig. 13, at $t_s$=313° Fahr., $c_p$=0·8. From tables, H=1186·9, then

$$H_s=H+c_pt_s$$
$$=1186{\cdot}9+0{\cdot}8\times313$$
$$=1186{\cdot}9+250{\cdot}4=1437{\cdot}3 \text{ B.Th.U./lb.}$$

The value given in the superheated steam table is 1436·9.

$$\text{Pressure energy}=\frac{144pv_s}{J}=\frac{144\times1400\times0{\cdot}535}{778} \qquad v_s=0{\cdot}535$$
$$=138{\cdot}5 \text{ B.Th.U./lb.}$$

$$\text{Internal energy } I=H_s-\frac{144pv_s}{J}$$
$$=(1437{\cdot}3-138{\cdot}5)=1298{\cdot}8 \text{ B.Th.U./lb.}$$

**36. Supersaturated Steam.**—When dry steam is allowed to expand freely in a nozzle passage from a zone of high to a zone of low pressure, it acquires a high velocity, due to conversion of part of its heat energy into kinetic energy of its mass. If this energy were converted into mechanical work on a piston in a cylinder, part of the steam would condense, due to the conversion of heat into work. In the nozzle, however, condensation does not begin until there is a definite drop of pressure. The steam remains in the gaseous state, and at any pressure, down to this limit, its temperature is less than the saturation temperature corresponding to the pressure. It in fact becomes "undercooled." The steam is in a state of thermal instability and is said to be "supersaturated." When the critical limit, known as the "Wilson limit," is reached, it is supposed that the thermal equilibrium is restored, and the steam then begins to condense, the liberated latent heat bringing the wet steam up to the saturation temperature. This phenomenon, which was not known to the early investigators, was first demonstrated by C. T. R. Wilson of Cambridge,* and by H. M. Martin,† who defined the Wilson limit. Recent investigations by C. F. Powell ‡ have shown that Martin's estimate was too low, and that the supersaturation limit of pressure corresponds to that at which the steam would be about 2 per cent. wet if it expanded adiabatically in thermal equilibrium. A detailed explanation of this involved branch of molecular physics cannot be attempted here, and the student is referred to the articles quoted in the footnote.

In a general way, however, it may be stated that expansion is so rapid in the initial stage that the molecules cannot aggregate with sufficient rapidity round the nuclei of the mass to produce condensation.

This supersaturated condition is an abnormal one which occurs to a minor extent during part of the expansion of steam through a compound turbine, where it passes from the superheated to the saturated region.

The specific volume and total heat of supersaturated steam are calculable from Callendar's general thermodynamic equations.

If the range of undercooling $t_u$ is regarded as a negative superheat, having a corresponding mean specific heat $c_u$, the volume and total heat can be determined approximately, as in the case of superheated steam, from the corresponding equation

$$v_u = rv \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (26)$$

and

$$H_u = H - c_u t_u \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (27)$$

The $c_u$ and $r$ values are to be obtained from the corresponding specific heat and volume ratio charts, plotted from Callendar's (1929) table.

* *Phil. Trans. A.*, 1897, Vol. 139, p. 305.
† *Engineering*, 1913, Vol. 95, p. 37.
‡ *Ibid.*, 1929, Vol. 127, p. 711.

This aspect of the question, however, is of minor importance. It rarely arises; and the expenditure of time and labour on the construction of these charts is not justified. They have, therefore, been omitted here.

**37. Experimental Determination of the Quality of Steam.**—The quality of superheated steam ($t_s$ degrees of superheat) is obtained by direct thermometric reading of the steam temperatures either by a thermometer or a thermocouple. If the temperature registered is $t'$ and the saturation temperature $t$, the quality of superheat is $t_s=(t'-t)$.

There is no means of ascertaining directly the exact quality (degree of undercooling) of supersaturated steam. A film of moisture always forms on the thermometer or thermocouple wire, and the temperature registered is that of the water film and not of the gas surrounding it.

The quality of wet steam (wetness or dryness) may be determined approximately by some form of steam calorimeter. An exact determination is not possible, no matter what form of instrument is used, on account of the difficulty of withdrawing a representative sample from boiler or steam-pipe.

**38. The Barrel Calorimeter.**—A rough approximation can be obtained (provided very considerable care is exercised in carrying out the test) from a barrel calorimeter. This consists of a barrel placed on a weighing-machine and partly filled with cold water. The sampling-pipe carrying the steam is immersed in the water so that the pipe outlet, which has a perforated end, is quite close to the bottom of the barrel. A stirring arrangement is used to properly mix the hot and cold water layers and obtain a correct final temperature. The steam, discharged from the pipe, is condensed by the water and raises the temperature of the original mass. The weight of steam condensed is given by the difference of the calorimeter weights before and after the test.

If W=wt. of cold water,
$w$=wt. of steam condensed,
$t_1$=initial temperature of water,
$t_2$=final temperature of water,
L=latent heat of the steam,
$q$=dryness,

then $\quad W(t_2-t_1)=[H-(1-q)L-h_2]w$ ; $h_2$=water heat at $t_2$

$$\therefore \text{ wetness } (1-q)=\frac{H-\frac{W}{w}(t_2-t_1)-h_2}{L} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (28)$$

*Example* 10.—The initial weight of water in a barrel calorimeter is 200 lb., at a temperature of 60° Fahr. Steam at 85 lb./in.$^2$ abs. is discharged into it. At the end of the test the weight has increased by 10 lb. and the temperature of the mixture is 110° Fahr. Calculate the initial quality of the steam.

Here $t_1=60$, $t_2=110$, $h_2=78{\cdot}5$, W$=200$, $w=10$.
From table H$=1187{\cdot}2$ and L$=901{\cdot}1$

$$\therefore \text{ wetness } (1-q)=\frac{\text{H}-\frac{\text{W}}{w}(t_2-t_1)-h_2}{\text{L}}$$

$$=\frac{1187{\cdot}2-\frac{200}{10}(110-60)-78{\cdot}5}{901{\cdot}1}$$

$$=\frac{1187{\cdot}2-1078{\cdot}5}{901{\cdot}1}=\frac{106{\cdot}7}{901{\cdot}1}=0{\cdot}118$$

$\therefore$ dryness $q=88{\cdot}2$ per cent.

Instead of directly mixing the steam with condensing water, a small surface condenser might be used. Types of condensing calorimeters have been tried, but have never come into favour; and this form need not be considered here.

**39. The Separating Steam Calorimeter.**—This is a form of steam drier which frees the steam from the major portion of the entrained moisture, and discharges it with a small remanent amount. One form of this instrument, designed originally by Prof. Carpenter, is shown in section in Fig. 16.

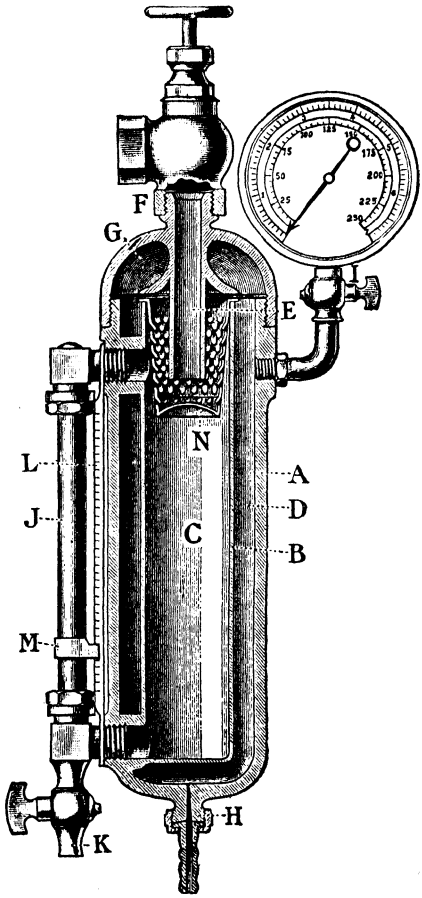


FIG. 16.

It consists of two vessels, A and B, one within the other, with a steam space between them. Wet steam enters from the steam supply valve F. The entrained water hits the convex bottom of the cup N, is thrown outward through the holes in the cup sides and falls into the chamber C. The steam passes over the edge of the cup into the annular chamber D, and is discharged through the orifice H, which is so small that the steam in the calorimeter suffers no sensible reduction in pressure. In the original instrument the gauge I had a scale

showing the rate of flow through the orifice. For an accurate determination, however, the steam passing the orifice should be condensed in a coil and weighed. The water separated in a given time is registered by a gauge glass J and scale L, a cursor M being fitted to facilitate the readings. A cock K is provided for drainage of the chamber C. The weight of water collected during the test can also be checked by weighing this drainage.

If W=wt. of steam passed in lb. (condensed)
$w$=wt. of water separated (wt. of drain)

$$\text{then the dryness is } q=\frac{W}{W+w} \quad . \quad . \quad . \quad . \quad (29)$$

$$\text{and the wetness } (1-q)=\frac{w}{W+w} \quad . \quad . \quad . \quad . \quad (30)$$

*Example* 11.—In a separating calorimeter test the steam passing out of the instrument after being condensed weighs 25 lb. The weight of separated moisture collected during the same time is 2·5 lb. Find the approximate dryness of the steam entering the instrument.

Here $W=25$, $w=2{\cdot}5$

$$\therefore\ q=\frac{W}{W+w}=\frac{25}{27{\cdot}5}=0{\cdot}91 \text{ or } 91 \text{ per cent.}$$

As this instrument cannot completely dry the steam, the value of the dryness given by it is only a rough approximation.

**40. Throttling Steam Calorimeter.**—This type of instrument, originally designed by Prof. Peabody, depends for its action on the fact that during a throttling process the total heat of the steam remains constant.

The steam is throttled with very little generation of kinetic energy, through a small orifice from the initial pressure of the boiler, or steam-pipe, or steam chest to a pressure approximately atmospheric. Any small amount of kinetic energy is reconverted to heat in eddy and shock in the chamber. If the steam is initially dry before throttling, then it will be superheated at the lower pressure (assuming that the instrument is efficiently lagged to prevent loss by radiation). If it is initially wet the excess heat evaporates the entrained moisture, and slightly superheats the steam. From the principle involved it will be apparent that there is a limit to the amount of moisture which can be evaporated before all the excess heat is used up. Thus while the previous type of instrument can be used with very wet steam, the throttling type is uitable only for steam having a high dryness fraction (0·97 to 0·99).

Obviously, in order to obtain a more accurate estimate in the case of very wet steam, a combination of the two types should be used. This is the modern practice, a separating and a throttling calorimeter being run in series.

One commercial form of the combined separating and wire-drawing calorimeter, as made by Dobbie, McInnes and Clyde, is illustrated in Fig. 17.

The steam sampling pipe A, screwed into the steam-pipe, has several saw-cuts through which the moist steam is conveyed when the valve B is opened to the separating chamber D. The major part of the entrained moisture collects in the chamber and the dried steam passes at the top into the throttling calorimeter X, which is provided with a long pocket F to take a thermometer. This is supposed to register the saturation temperature of the steam at the high-pressure side of the throttle plate H, which is clamped between the two halves of the calorimeter by the flanges G. The

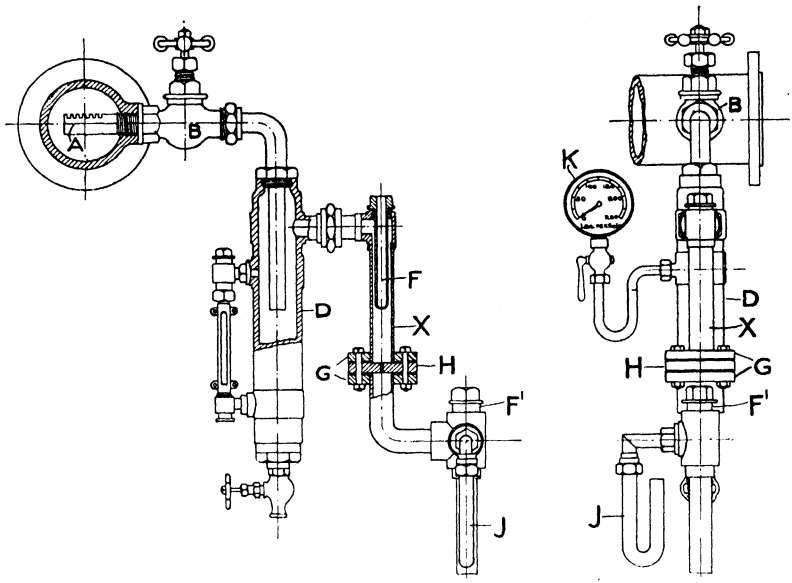



FIG. 17.

pressure in X is registered by the gauge K, and it should correspond to the saturation temperature registered by the thermometer in F. The throttle plate H is provided with a small orifice, about $\frac{1}{32}$ in. diameter. The low-pressure side is provided with a second pocket at F′ for a thermometer, which registers the final temperature at approximately atmospheric pressure. The exact value of the pressure is registered by a water manometer J. The outlet should be connected to a condensing coil open to the atmosphere, so that the steam passed through the throttling instrument can be condensed and accurately weighed. It is most important that the connections between the sampling source and the calorimeter should be as short as possible and very thoroughly lagged, otherwise the sample as it reaches the calorimeter may be appreciably

wetter than that taken from the source. Both parts of the instrument should also be thoroughly lagged with first-class non-conducting composition.

Let $q_1$=dryness of the steam at $P_1$ entering the throttling calorimeter.

The total heat per lb. of wet steam at $p_1$ is $(h_1+q_1L_1)$, where $h_1$ and $L_1$ are the water and latent heats.

Let $t_2$=saturation temperature of the steam at the lower pressure $p_2$, after throttling, and $t_2'$=temperature of the superheated steam at $p_2$, then the total heat at $p_2$ is

$$H_2+c_p(t_2'-t_2)$$

Equating the initial and final total heats

$$h_1+q_1L_1=H_2+c_p(t_2'-t_2)$$

and the initial dryness is

$$q_1=\frac{H_2+c_p(t_2'-t_2)-h_1}{L_1} \quad . \quad . \quad . \quad . \quad (31)$$

If the throttling instrument does not register a value of $t_2'>t_2$, then it is obviously useless for the estimation of $q_1$. If $t_2'=t_2$ the superheat is zero, and Equation 31 then, for a given case, gives the limiting value of $q_1'$ down to which the instrument will record. Thus—

$$\text{Limiting dryness } q_1'=\frac{H_2-h_1}{L_1} \quad . \quad . \quad . \quad (32)$$

*Example* 12.—Find the limiting value of the steam quality which can be recorded if the initial pressure is 200 lb./in.$^2$ abs. and final pressure 15 lb./in.$^2$ abs.

$$\text{Here } q_1'=\frac{1151{\cdot}2-355{\cdot}5}{847{\cdot}6}=\frac{795{\cdot}7}{847{\cdot}6}=0{\cdot}94.$$

This illustration serves to show the limitation of this instrument where steam of considerable wetness has to be handled. For the throttling calorimeter the value of the specific heat $c_p$ may be taken as 0·48.

Taking the combination of separating and throttling instruments:

Let $w$=wt. of water collected in the separator.

W=wt. of steam passed through the throttling calorimeter and condenser,

then the apparent dryness of the steam entering the combination is given by

$$q_2=\frac{W}{W+w} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (33)$$

Thus for 1 lb. of wet steam having a real quality $q_1$ entering the

combination, only $q_2$ lb. passes on to the throttling instrument. Hence the real initial quality of the steam sample is given by

$$q=q_1\times q_2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (34)$$

where $q_1$=quality as determined by the throttling calorimeter.

*Example* 13.—The quality of steam in an engine steam chest is determined by a combined separating and throttling calorimeter. The receiver pressure is 120 lb./in.$^2$ abs. and the pressure after throttling 15 lb./in.$^2$ abs. The temperature registered at the lower pressure is 253° Fahr.

The condensed steam from the throttling instrument is 22·3 lb. and the water collected in the separator is 1·4 lb. Calculate the quality of the steam drawn from the receiver.

$$W=22\cdot3, \quad w=1\cdot4 \quad \therefore\ q_2=\frac{W}{W+w}=\frac{22\cdot3}{23\cdot7}=0\cdot94$$

$$t_2'=253, \quad t_2=213, \quad H_2=1151\cdot2, \quad h_1=312\cdot3, \quad L_1=881\cdot8$$

$$\therefore\ q_1=\frac{H_2+0\cdot48(t_2'-t_2)-h_1}{L_1}$$

$$=\frac{1151\cdot2+0\cdot48\times40-312\cdot3}{881\cdot8}$$

$$=\frac{858\cdot1}{881\cdot8}=0\cdot975$$

$$\therefore\ q=q_1q_2=0\cdot975\times0\cdot94=0\cdot916.$$

**41. Entropy of Dry Steam, Φ.**—The total entropy of dry steam, which can be obtained from the saturated steam table, consists of two parts: (*a*) the entropy change in heating the water from 32° Fahr. to the saturation temperature $t$ (or T) corresponding to the pressure $p$; (*b*) the entropy of evaporation during change of physical state at constant temperature T.

The entropy of evaporation is $\phi_e=\frac{L}{T}$ . . . . . (35)

When $\phi_e$ and the total entropy Φ are known, the exact value of the entropy of the water or liquid entropy is given by their difference, thus—

$$\phi_w=(\Phi-\phi_e) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (36)$$

The value of Φ can be obtained by direct calculation from Callendar's (1929) equation (47).

An approximate value of the liquid entropy, sufficiently good for all practical purposes, can be obtained from—

$$\phi_w=\int_{492}^{T}\frac{dT}{T}=\log_e\frac{T}{492} \quad . \quad . \quad . \quad . \quad (37)$$

the assumption being made that the specific heat of the water is unity.

**Entropy of Wet Steam** ($\Phi_w$).—In this case the liquid entropy is the same and the evaporation entropy is obviously $\frac{qL}{T}=q\phi_e$, so that

$$\left.\begin{aligned}\Phi_w&=\phi_w+q\phi_e\\ \text{or } \Phi_w&=\Phi-(1-q)\phi_e\end{aligned}\right\} \quad . \quad . \quad . \quad . \quad (38)$$

**42. Entropy of Superheated Steam, $\Phi_s$.**—It can be shown that if the steam be treated as a gas the increase of entropy in superheating it from $t$ to $t'$ or T to $T_s$ absolute at constant pressure, is given by—

$$\phi_s=c_p \log_e \frac{T_s}{T} \quad . \quad . \quad . \quad . \quad . \quad (39)$$

Where $c_p$ is the mean specific heat for the superheat range $t_s=(t'-t)$ or $(T_s-T)$. This value can be obtained from the specific heat chart (Fig. 13).

The total entropy change is thus—

$$\left.\begin{aligned}\Phi_s&=\Phi+\phi_s\\ &=\Phi+c_p \log_e \frac{T_s}{T}\end{aligned}\right\} \quad . \quad . \quad . \quad . \quad . \quad (40)$$

**43. Callendar's (1929) Equations for Steam Properties. Specific Volume** ($v$).—The relationship between the pressure, volume, and temperature cannot be expressed by such a simple law as that of a permanent gas.

Callendar has shown that the specific volume is expressed in general form by the characteristic equation—

$$v=\frac{RT}{ap}+b-\frac{c}{(1-z^2p^2)} \quad . \quad . \quad . \quad . \quad (41)$$

Where $a=\frac{144}{J}=\frac{144}{778}=0{\cdot}18514$ ; $R=0{\cdot}11012$

$$\frac{R}{a}=\frac{0{\cdot}11012}{0{\cdot}18514}=0{\cdot}59479 \text{ ; } b=-0{\cdot}002805$$

$c$ = coaggregation volume $=c_1\left(\frac{T_1}{T}\right)^n$

$c_1=0{\cdot}4213$, $n=10/3$, $T_1=671{\cdot}58°$ F. abs., so that

$$c=0{\cdot}4213\left(\frac{671{\cdot}58}{T}\right)^{10/3}=\frac{1118\times10^6}{T^{10/3}}$$

$$z=5\times\tfrac{1}{3}ac/T=4{\cdot}012\frac{c}{T} \text{ ; } z^2=16{\cdot}096\left(\frac{c}{T}\right)^2$$

where T is the temperature of the steam at pressure $p$. This equation gives the volume of saturated, superheated, or supersaturated steam according as the temperature is that of saturation, superheat, or undercooling, that is, $v$, $v_s$, or $v_u$, for $T_1$, $T_s$ and $T_u$, where the temperature is in ° Fahr. absolute.

Substituting the numerical values of the constants—

$$v=0{\cdot}59479\frac{T}{p}-0{\cdot}002805-\frac{1118\times10^6}{T^{10/3}\left\{1-\left(\frac{4{\cdot}012c}{T}\right)^2p^2\right\}} \quad . \quad (42)$$

**44. Total Heat of Steam** (H).—The equation giving the total heat of dry superheated or supersaturated steam has this form when the heat is expressed as a function of the pressure and temperature—

$$H=\left[\tfrac{1}{3}^3R-\tfrac{1}{3}^3\frac{acp}{(1-z^2p^2)T}+\frac{abp}{T}\right]T+B$$

$$\text{or } H=\left[\tfrac{1}{3}^3R-\frac{pz}{5(1-z^2p^2)}+\frac{abp}{T}\right]T+B \quad . \quad . \quad . \quad (43)$$

R, $a$, $c$, $b$, and $z$ have the values already quoted in connection with the volume calculation, and the constant B has a value of 835·2. The total heat is expressed in B.Th.U./lb.

Substituting the various values of the factors—

$$H=\left[0{\cdot}4772-\frac{4{\cdot}012cp}{5\left\{1-\left(\frac{4{\cdot}012c}{T}\right)^2p^2\right\}T}-\frac{0{\cdot}001668p}{T}\right]T+835{\cdot}2$$

$$\text{or} \quad H=0{\cdot}4772T-\frac{0{\cdot}8024\times\frac{1118\times10^6p}{T}}{\left\{1-\left(\frac{4{\cdot}012c}{T}\right)^2p^2\right\}}-0{\cdot}001668p+835{\cdot}2$$

$$=0{\cdot}4772T-\frac{897\times10^6p}{\left\{1-\left(\frac{4{\cdot}012c}{T}\right)^2p^2\right\}T}-0{\cdot}001668p+835{\cdot}2 \quad . \quad (44)$$

The total heat can also be expressed as a function of the pressure and volume, thus—

$$H=\tfrac{1}{3}^3ap(v-\tfrac{1}{3}^0b)+B$$

$$\text{or } H=\frac{p}{1{\cdot}2464}(v+0{\cdot}00216)+835{\cdot}2 \quad . \quad . \quad . \quad (45)$$

**45. Total Heat of Water** ($h$).—This is given by—

$$h=s(t-32)+\frac{u}{v}[H-s(t-32)]-0{\cdot}003\left(1-\frac{u}{v}\right). \quad . \quad (46)$$

where $s=0{\cdot}9966$ and $u$ and $v$ are the specific volumes of the liquid and saturated vapour at temperature $t$.

**46. Total Entropy** ($\Phi$).—The total entropy for dry superheated or supersaturated steam is given by—

$$\Phi=1{\cdot}09876\log_{10}\frac{T}{T_1}-0{\cdot}25356\log_{10}\frac{p}{p_1}$$

$$+0{\cdot}05314\log_{10}\left(\frac{1+zp}{1-zp}\right)-\frac{pz}{5(1-z^2p^2)}+1{\cdot}76300 \quad . \quad (47)$$

where $T_1=671{\cdot}58°$ F. abs. and $p_1=14{\cdot}69$ lb./in.$^2$ abs.

**47. Relation between Pressure and Temperature.**—Finally the relation between the saturation temperature and the pressure is given by—

$$\log_{10} p = 22{\cdot}278595 - \frac{5226{\cdot}12}{T} - 4{\cdot}7174 \log_{10} T + 0{\cdot}20956 \log_{10}\left(\frac{1+zp}{1-zp}\right) + 0{\cdot}002048\frac{p}{T} \quad . \quad (48)$$

The last two terms are small and negligible except near the critical temperature (704° Fahr.).

Approximate values are assumed and the identity is established by successive approximation.

# CHAPTER III

## ENERGY DIAGRAMS FOR STEAM

**48.** When the characteristics of the working substances of heat engines are graphed to form a set of intersecting curves in the co-ordinate field the combination constitutes an energy diagram.

By means of it estimations of heat and work and energy transformation can be made with a minimum of calculation.

When the substance is a permanent gas the complete diagram consists of a set of six curves, one set for each characteristic property.

When it is a saturated vapour, such as steam, an additional set of quality curves, dryness and superheat, may be added.

It is seldom, however, that a complete energy diagram is drawn. The number of curves provided on a partial diagram depends on the particular purpose for which the diagram is to be used.

By taking the characteristics in pairs, as current co-ordinates, fourteen energy diagrams can be constructed.

Two of these are of special importance in the thermodynamic study of the steam engine and turbine, the Temperature-entropy (TΦ) and Heat-entropy (HΦ) diagrams.

The first is very serviceable for thermodynamic analysis relating to the reciprocating steam engine. The second is also serviceable for this purpose, and is an indispensible instrument for the rapid calculation of the thermodynamic data required in steam turbine design.

As already pointed out in Art. 50, a third combination, the Pressure-volume (PV) diagram, is complementary to the (TΦ) diagram. This diagram, however, will only be referred to as occasion requires in the discussion of the thermodynamic changes in the reciprocating engine.

**49. Temperature-entropy Diagram** (TΦ).—This diagram was first brought to the notice of engineers by the late J. McFarlane Gray, who demonstrated its importance as an instrument for thermodynamic analysis of steam engine performance.

It was developed further by the late Capt. H. R. Sankey, R.E., who published a monograph on the subject in 1905,* and the complete (TΦ) diagram, as drawn by him, was reproduced in one of the early editions of this text.

* *The Energy Chart. Practical Applications to Reciprocating Engines* (Albert Frost & Co., Rugby). See also *The Entropy Diagram and its Applications*, by J. Boulvin (translation by Bryan Donkin).

In the present edition it is replaced by a new diagram embodying Callendar's 1929 figures, and has been carried from the original limit of 250 lb./in.$^2$ abs. to 2000 lb./in.$^2$ abs.

As already shown in Art. 20, "entropy" is the co-ordinate of absolute temperature on a heat diagram. In other words, the area below any TΦ curve gives a quantity of heat. When the term "entropy" is used, what is meant is "change of entropy," as the absolute value, just like that of the internal energy of a substance, cannot be ascertained. Entropy change can be reckoned from any arbitrary datum level of temperature, and for $H_2O$ stuff this datum is always taken as 0° C. or 32° Fahr., as already pointed out in connection with the properties of steam.

When the entropy values for water ($\phi_w$), as given in the saturated steam table, are plotted against the corresponding temperatures, the curve ABC (Fig. 18) is obtained. When the entropy values for the dry steam (Φ) are plotted, the curve ESC results, the two curves merging at C. These are termed respectively, the "liquid limit or water curve" (AC) and the "gas limit or saturation curve" (EC). The pressure at which they merge is the "critical pressure," and has a value about 3,200 lb./in.$^2$ abs. The corresponding temperature is 704° Fahr.

The limit curves divide the co-ordinate field into three zones. The space to the left of the water curve is the liquid field, that between the curves the saturation field, and that to the right of the saturation curve the superheat field.

Any point in the liquid field gives the condition or state of the liquid; in the saturation field the condition of the wet steam; in the superheat field the condition of the superheated steam.

The diagram is always drawn for unit mass of $H_2O$ stuff. Referring again to the steam formation process in the cylinder (Art. 24), A (Fig. 18) represents the initial or starting point of the heating operation under pressure $p$, after the water is placed in the cylinder at 32° Fahr., or 491·6° Fahr. abs. During the heating the state point travels from A to D, and from 491·6 to T. OK is the corresponding entropy change, and the area OADK is the liquid heat, or heat taken in to raise the temperature between those limits. It is not the total heat of the water at $p$ and T, as defined in Art. 27, since this diagram does not take account of the pressure energy given to the water in forcing it into the cylinder, but only the heat added. At D evaporation begins at constant temperature T, with increasing volume and entropy. When the whole pound is evaporated the state point has moved to S on the saturation curve. DS is the evaporation entropy ($\phi_e$) and the area KDSM is the latent heat (L). The amount of heat supplied to form 1 lb. of dry saturated steam at $p$ and T, from 1 lb. water at 32° Fahr., or 491·6° Fahr. abs., is now given by the area below the broken curve ADS, or OADSM.

This under the older nomenclature was called the "total heat."

Under the new definition it is less than the total heat by the pressure

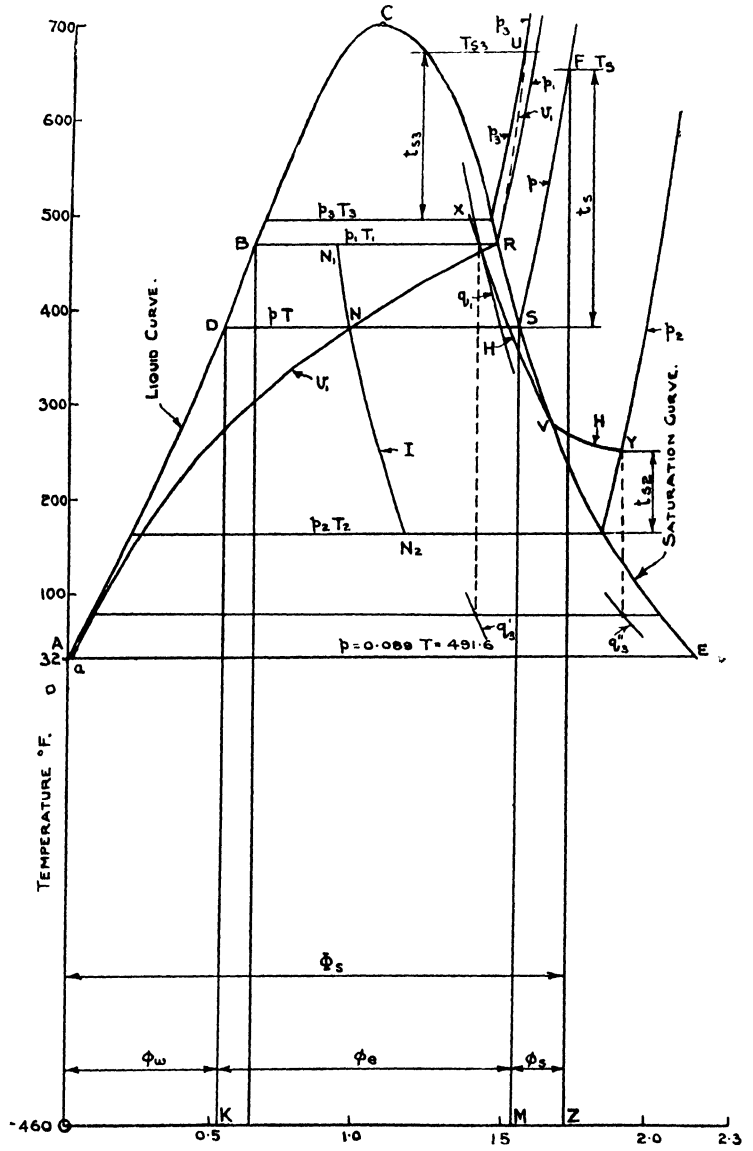



FIG. 18.

energy of the water at 32° Fahr. or $\left(\frac{144pb}{J}\right)$. Unless at very high pressure this small difference is negligible—in fact, in all practical

calculations it is disregarded. At 2000 lb./in.$^2$ abs. it amounts to about 0·475 per cent. of the total heat.

**50.** The occurrences during steam formation are shown on the PV diagram in Fig. 19. The corresponding state points are indicated by the small letters. Owing to the difficulty of showing the very small water volume and the very great steam volume at low pressure on a readable diagram, the curves have been broken, and the extreme portions at maximum pressure and volume are linked with the main parts by brackets. The water curve is shown by *adc*, the volumes being greatly exaggerated for clearness, and the saturated steam curve by *cse*. As already seen from Art. 24 the small amount of expansion work of the water is given by the area below *md*. The point *m* does not lie on the axis OY, but is inside this by the

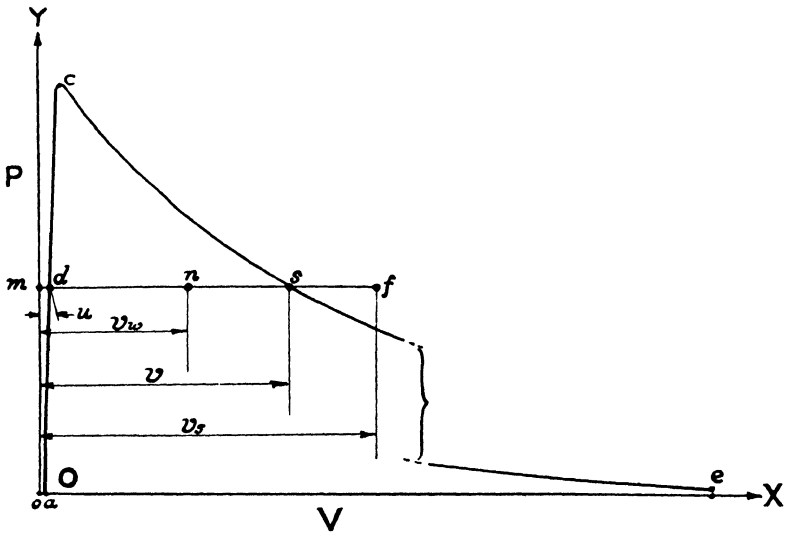



FIG. 19.

amount (*b*), the water volume at 32° Fahr., which cannot be shown on such a small scale. This diagram does not show the corresponding change of internal energy during the heating process from 32° Fahr. to T.

During evaporation at constant temperature T, the volume increases from *u* to *v*, and the state point moves from *d* to *s*. The area below *ds* gives the external work done during evaporation, but does not show the accompanying internal latent heat.

At any point N (Fig. 18) or *n* (Fig. 19) the stuff is in the state of wet saturated steam and the dryness is given by $q$=DN/DS (Fig. 18) and $q \doteqdot dn/ds = v_w/v$ (Fig. 19). The latent heat of the wet steam is given by the area below DN (Fig. 18) or $L_w = qT\phi_e$.

When the dry steam state is reached, let the supply of heat be

continued at pressure $p$. The volume, temperature, and entropy will increase as the steam becomes superheated, and the locus of the state point is the curve SF (Fig. 18). MZ is the entropy of superheat $\phi_s$, and the area below SF, or MSFZ, is the superheat $h_s$ at temperature $T_s$ and pressure $p$. The heat supplied to produce 1 lb. of superheated steam at pressure $p$ and temperature $T_s$, or superheated through a range $t_s=(T_s-T)$, is given by the area below the broken curve ADSF, that is, OADSFZ. This broken curve is called a " curve of constant pressure."

In the PV diagram (Fig. 19) the corresponding state point $f$ lies on an extension of the pressure line $ds$ where the volume is $v_s$, and the area below $sf$ gives the additional external work done during the superheating process.

Where a series of constant pressure curves is drawn, it is possible to determine by inspection, at any given state point in the T$\Phi$ field, the corresponding values of the pressure, temperature, and entropy of the stuff. The usefulness of the diagram is enhanced, for calculations in the saturation region, by the addition of constant volume curves. Pressure, temperature, entropy, and volume values can then be ascertained and various heat calculations can be facilitated.

**51.** Consider the dry steam state point R (Fig. 18), at pressure $p_1$ and temperature $T_1$. At this pressure the steam has a definite specific volume $v_1$, as given in the steam table. R may thus be defined as a point on a constant volume curve $v_1$. At the lower pressure $p$ the volume of dry steam at $s$ is $v$ and greater than $v_1$, but at some point N, corresponding to quality or dryness $q$, the volume of the wet steam at $p$ will have the same value as that of the dry steam at $p_1$, and N will thus be a point on the volume curve $v_1$. Thus $v_w=v_1=qv$, or the quality at N is given by $q=\frac{v_1}{v}$. In this way, by dividing the given volume by the dry steam volume at a series of pressures, the qualities at these pressures and corresponding state points on the curve can be located and the curve RN$a=v_1$ can be drawn in the saturation field. It can be carried, if desired, into the superheat region, although this is seldom done. At the saturation curve the volume curve RU (shown dotted) undergoes a very sharp change of curvature, and its slope at any point differs very little from that of the corresponding pressure curve $p_1$.

It is obtained by finding the superheat range ($t_s$) at a series of ascending pressures above $p_1$. Suppose that the $v_1$ curve cuts the constant pressure curve $p_3$ at temperature $T_{s3}$, then the range of superheat at this pressure is $t_{s3}=(T_{s3}-T_3)$ and $v_1=v_{s3}$, the volume of the superheated steam at $T_{s3}$.

By calculation from Equation 25 (p. 48), or by reference to the Callendar superheated steam tables, the superheat $t_{s3}$ and temperature $T_{s3}$ corresponding to $v_{s3}$, at $p_3$ can be found.

**52.** The four characteristics $p$, $v$, T, and $\Phi$ are now determinable

from the diagram, and there remain the internal energy and total heat.

The total heat can be determined, with sufficient accuracy for all practical purposes, from the TΦ diagram, by finding the area below the given constant pressure curve. At very high pressures (between 1000 and 2000 lb./in.$^2$ abs.) the pressure energy of the water at 32° Fahr., or $\frac{144pb}{J}$, should be added to the result. Since $b$=0·016 ft.$^3$, J=778 ft.-lb./B.Th.U., $\frac{144pb}{J}$=0·00296 $p$. This tedious and troublesome integration process can be avoided by making use of the alternative HΦ diagram, discussed in (Art. 53).

The internal energy of the steam in any state is most conveniently obtained, directly by calculation, from the total heat value, by the methods given in Chapter II. There is a roundabout integration method by which it can be found from the TΦ diagram: it is stated here simply as a matter of interest.

If the volume curve $v_1$ (Fig. 18) cuts the pressure line $p$=0·089, corresponding to 32° Fahr., at $a$, then it can be shown that the internal energy at any point N, where the curve cuts the pressure $p$ line, is the area below $a$N plus a small quantity 0·3095$v_1$. If the area below $a$N is denoted by $I_N$, then the internal energy of the wet steam of quality $q$ at $p$ is—

$$I_w=(I_N+0\cdot3095v_1) \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

If the steam is finally superheated $t_s$, and $I_U$ is the area below $a$NRU, the internal energy of the superheated steam is—

$$I_s=(I_U+0\cdot3095v_1) \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

In Fig. 18, $N_1NN_2$ shows an internal energy curve (I) passing through N.

A series of total heat curves can also be drawn in the TΦ field, since the quality at any pressure is given by

$$q=1-\frac{H-H_w}{L} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

where $H_w$ is the constant total heat of the wet steam and H and L the total and latent heats of the dry steam at the pressure where the quality is $q$.

The curve XV, Fig. 18, is a curve of total heat, and it will be noted that it differs very little in slope from that of the internal energy curve $N_1NN_2$. The inclusion of these two sets of curves with the pressure and dryness curves on the TΦ field simply leads to needless confusion, and they are practically never drawn. As a rule the five sets of curves, pressure, temperature, entropy, volume, and quality (dryness and superheat) are drawn.

**53. Total Heat-entropy Diagram (HΦ).**—This form of energy diagram was introduced by Prof. R. Mollier, of Dresden, principally to facilitate calculations relating to steam turbine design. It is,

however, equally serviceable for many thermodynamic calculations relating to the reciprocating steam engine. In this case the co-ordinates chosen are total heat and entropy.

When the total heats of water ($h$), given in the saturated steam table, are plotted to a base of entropy, the liquid limit, or water curve ABC (Fig. 20), is obtained.

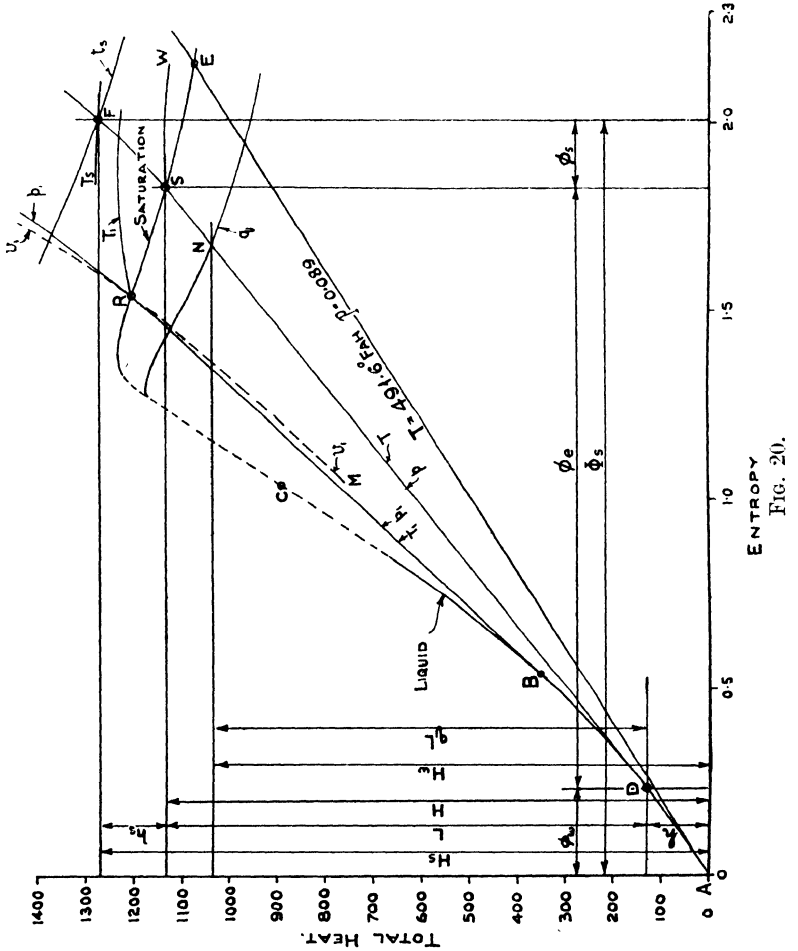



Fig. 20.

Similarly the total heats of dry steam when plotted give the gas limit, or saturation curve ESRC, which merges into the liquid limit curve at C, the critical point. As in the case of the TΦ diagram, the space to the left of ABC is the liquid field, the space between ABC and ESRC, the saturation field, and the space to the right above ESR the superheat field.

The initial point for the heating of water under any pressure $p$ is A. During this heating process between 32° Fahr. or 491·6° Fahr. abs. and T, the state point moves from A to D, and the vertical distance between these points is the exact value of the total heat of the water ($h$) at $p$ and T. At D evaporation begins, and the state point moves along the straight line DS, which is the line of constant pressure $p$ and also of constant temperature T. The vertical distance between D and S is the latent heat (L) at $p$, and the distance between A and S is the total heat (H) of the dry steam. If the supply of heat is continued the steam becomes superheated, the state point moves along the flat curve SF to F in the superheat field, which is the state point of the steam superheated from T to $T_s$ or through the range $t_s=(T_s-T)$. The vertical distance between S and F gives the superheat ($h_s$) at $t_s$, and the vertical distance between A and F the total heat ($H_s$) of the pound of superheated steam at pressure $p$ and superheat range $t_s$.

Similarly the horizontal distances between A and D, D and S, and S and F give respectively the water entropy $\phi_w$, the evaporation entropy $\phi_e$, and the entropy of superheat $\phi_s$; while the total distance between A and F gives the total entropy of the superheated steam $\Phi_s$.

Let N be the state point on the $p$ line at which the dryness is $q$, then the latent heat $q$L is given by the vertical distance between D and N, and since DS is a straight line it follows that $q$=DN/DS. The total heat of the wet steam ($H_w$) is given by the vertical distance between A and N.

When the constant pressure lines are proportionately divided to give the same fractional value ($q$), the locus of the points is the curve of constant dryness $q$.

The constant temperature line at S merges into a flat curve SW, and DSW is the curve of constant temperature T. The state point F, for the superheated steam at $p$, lies on a similar curve of $T_s$ temperature, which is $t_s$ degrees in excess of T. If the corresponding points for $t_s$ excess at several pressures are found, their locus is the curve of constant superheat $t_s$.

As a rule a series of these constant superheat curves is drawn, but in some cases a set of constant temperature curves is added. A complete set of dryness curves is always provided in the saturation field.

As in the case of the TΦ diagram, consider another pressure $p_1$ at $T_1$, for which the dry steam state point is R. The volume of dry steam corresponding to this point is $v_1$. At some lower pressure $p$, which has the dry steam volume $v$, the volume $v_1$ will correspond to some dryness fraction $q=v_1/v$, hence the point on the $p$ curve which gives dryness $q$ is also a point on the $v_1$ curve. By finding the qualities at several pressures the volume curve RM for $v_1$ can be drawn in the saturation field. This can be continued into the superheat field (as indicated by the dotted curve). In this case

there is no quick change of curvature as the curve crosses the gas limit curve. The points can be obtained in the superheat field at several pressures by noting the range of superheat at each pressure which corresponds to the required volume in the superheated steam table. The slope of the volume curve is very little greater than that of the pressure curve, and where a full set of curves is drawn in addition to the pressure and dryness curves the resulting network is very confusing. As a general rule these volume curves are omitted. The complete form of the HΦ diagram is shown in Fig. 20. For steam turbine work only the upper portion on the right is drawn to a large scale.

**54. Representation of Expansion and Throttling Processes on TΦ and HΦ Diagrams.—Isentropic or Frictionless Adiabatic Expansion.**—When steam expands adiabatically without friction, doing work on a piston against an external resistance, or in generating kinetic energy in its mass, the entropy remains constant. When the initial state point is located at the upper pressure limit, the final point determining the condition at the lower limit is therefore obtained by drawing a vertical between the two pressure curves. The general case for steam initially superheated and finally wet is shown in Figs. 21 and 22. The initial state point at $p_1$ with superheat range $t_{s1}$ and specific volume $v_{s1}$ is F. The vertical FG′ cuts the lower pressure curve $p_2$ at G′, and determines the quality $q_2'$ and specific volume $v_{w'2}$. In Fig. 22 the total heats $H_{s1}$ and $H_{w'2}$ at $p_1$ and $p_2$ are read directly. In Fig. 21 they are obtained by integrating the areas below the curves ABRF and ADG′, and adding the corresponding pressure energies of the water at $p_1$ and $p_2$. The latter, however, as already stated, are very small and are always neglected. The superiority of the HΦ diagram for this estimation is obvious.

During the expansion, the value of the total heat drops from $H_{s1}$ at $p_1$ to $H_{w'2}$ at $p_2$, and the difference $(H_{s1}-H_{w'2})$ is called the adiabatic heat drop and may be denoted by $H_\phi$. It is given by the vertical intercept FG′ on the HΦ diagram (Fig. 22) and with sufficient accuracy by the difference of the areas OABRF$g'$ and OADG′$g'$ on the TΦ diagram (Fig. 21).

If expansion takes place adiabatically—that is, without loss of heat to or gain of heat from external sources—but is accompanied by internal frictional effects, causing degradation of mechanical energy to heat (an irreversible process (Art. 7)), then the progressive addition of heat to the steam results in a progressive increase of entropy, and the expansion is non-isentropic. The resulting expansion curve is indeterminate, since the details of the reheating process cannot be traced throughout the expansion. The final state point can, however, usually be determined. This occurrence is also shown on the TΦ and HΦ diagrams (Figs. 21 and 22). The indeterminate expansion curve is indicated by the dotted curve FG, which in each case falls to the right of the adiabatic FG′. The ultimate

effect of the reheat, as shown, is to move the state point at $p_2$ from G′ to G, increasing the quality from $q'_2$ to $q_2$ and the specific volume

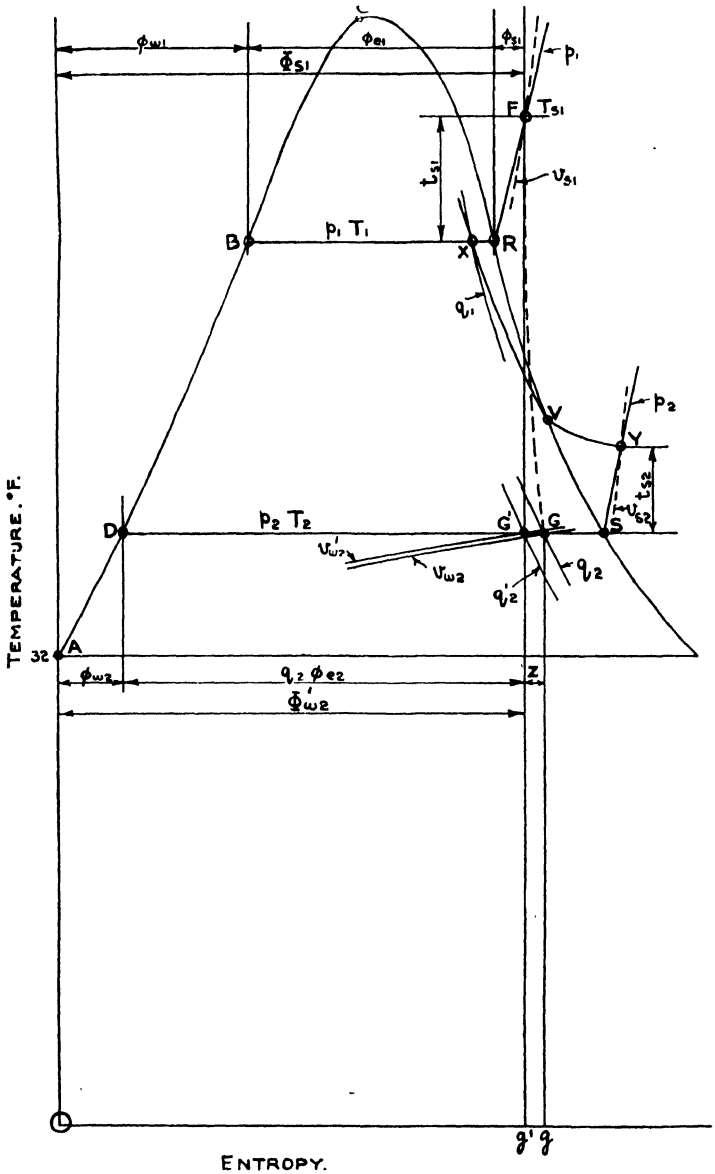



FIG. 21.

from $v_{w'2}$ to $v_{w2}$ and the final total heat from $H_{w'2}$ to $H_{w2}$. The drop of total heat with friction ($H_{s1}$ - $H_{w2}$) is less than that with

isentropic expansion, and may be termed the net heat drop $H_N$. Its equivalent on the TΦ diagram (Fig. 21) is the difference between the areas DBRFG′ and G′G*gg*′, the latter being the equivalent of the reheat, or the vertical distance between G and G′ in Fig. 22.

The methods of calculation for the determination of the total

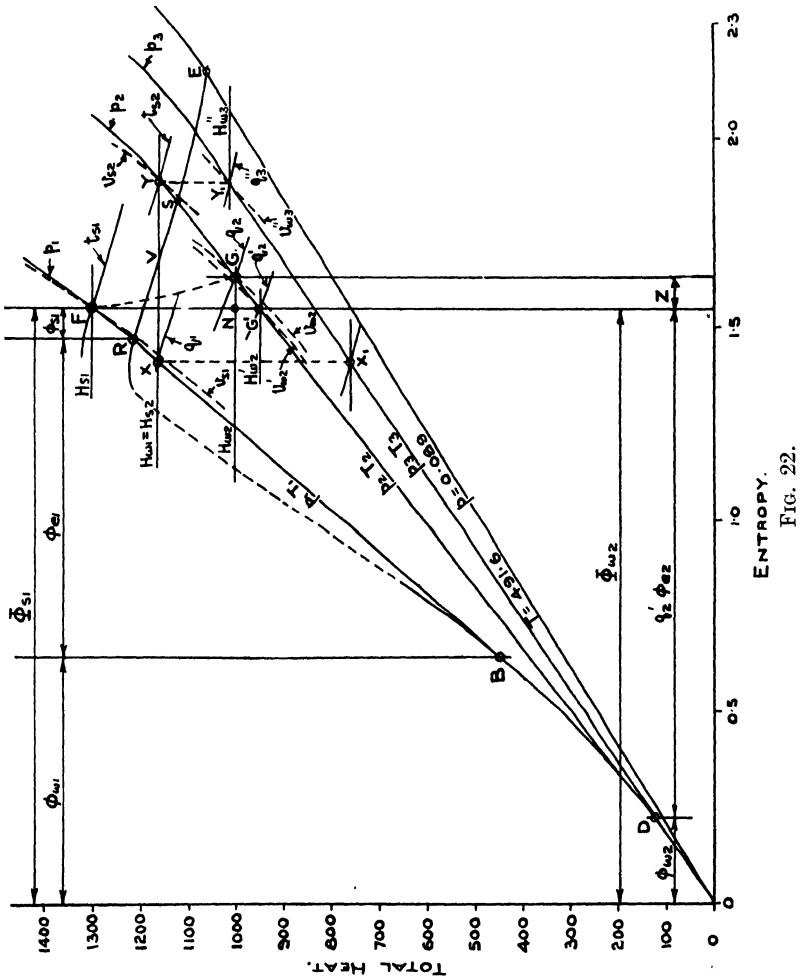



FIG. 22.

heats, qualities, volumes, etc., and the applications of these charts for the estimation of the efficiencies of various cycles, are discussed in Chapter IV.

**55. Throttling or Wiredrawing.**—In the cases considered the pressure drop is accompanied by a transformation of heat to mechanical energy. The pressure drop can, however, be accomplished without

energy transformation, by the process of throttling or wiredrawing. This action occurs at a partially opened stop-valve, at the valves and ports of a reciprocating steam engine having a slide-valve, at steam-reducing valves, and in the throttling calorimeter (Art. 40). In a case of this kind the fluid in passing the restricted opening into the lower pressure region has its velocity reduced to sensibly the same value as it has at the higher pressure region, the kinetic energy generated being reconverted into heat, so that the total heat at the lower pressure $p_2$ is the same as at the higher pressure $p_1$. Thus in a throttling or wiredrawing operation the total heat remains constant, and on the TΦ and HΦ diagrams a throttling curve is a curve of constant total heat.

The occurrence is shown in Figs. 21 and 22, for the case where the steam at $p_1$ is initially wet and at $p_2$ is finally superheated. The initial state point for the steam of quality $q_1$ is X, and the throttling curve passing through this point is XVY (Fig. 21), crossing the saturation curve at V, and the $p_2$ curve at $t_{s2}$ and $r_{s2}$. The corresponding curve in Fig. 22 is the total heat line XVY, crossing the saturation curve at V and the $p_2$ curve at $t_{s2}$ and $r_{s2}$.

If the steam were adiabatically expanded without friction from $p_1$ to some lower pressure $p_3$, the heat drop would be $XX_1$ (Fig. 22), but if it were first throttled down to $p_2$ and then expanded to $p_3$, the heat drop would be reduced to the value $YY_1$. The final quality and volume, however, would be considerably increased.

If H is the constant total heat value represented by the throttling curve, then in the saturation field the quality $q_3$ at any pressure $p_3$ is given by—

$$q_3 = 1 - \frac{H - H_3}{L_3} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

where $H_3$ and $L_3$ are the total and latent heats of the dry steam at $p_3$. When $H = H_3$ the steam is just dry at $p_3$, and the throttling curve crosses the saturation curve at this pressure. When H is greater than $H_3$, the steam is superheated at $p_3$. In this case the superheat range $t_{s3}$ should be obtained either directly by inspection or by interpolation from the Callendar superheated steam table. If a value is assumed for the probable specific heat of the superheated steam $c_{p3}$, an approximation to the range can be obtained from—

$$t_{s3} \doteq \frac{H - H_3}{c_{p3}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

This method, however, is not a satisfactory one. The simple and practical method is to use the HΦ diagram and read the quality values directly where the given constant heat line cuts the various pressure curves.

# CHAPTER IV

## ENGINE CYCLES AND EFFICIENCIES

**56.** In the discussion of an ideal cycle it is assumed that cyclic changes in the working substance take place under conditions in which there are no losses of energy due to friction, radiation, convection, wiredrawing, etc.

The most perfect heat engine (Art. 18) works on the reversible cycle of Carnot. A practicable engine on this cycle is not possible, on account of the complications involved by the fourth operation, the adiabatic compression of the $H_2O$ stuff between the lower and upper temperature limits; and as a thermodynamic standard of reference the Carnot cycle is not suitable.

The replacement of the adiabatic compression process by a feed-heating one affords a more practicable standard, and this modified cycle, originally suggested by Rankine and now termed the "straight" Rankine cycle, has until within recent years been universally used as the thermodynamic standard of reference for steam engines and turbines.

The developments of modern practice in power generation, however, have created conditions that are not covered by the "straight" Rankine definition.

Superheating, reheating, partial regenerative feed-heating are now superimposed on the conditions chosen for the original Rankine standard, and comparisons of efficiency have to be made with regard to the particular operative conditions of the given cycle.

**57. Carnot Cycle.**—In the case of the Carnot engine working with $H_2O$ stuff, the cylinder serves as both working chamber and condenser, while unit mass (1 lb.) of the stuff goes through the series of cyclic changes of condition.

The PV and TΦ diagrams for the cycle are shown in Fig. 23 and Fig. 24.

In this case the cycle starts at B, with the water at $p_1$ and $T_1$, occupying a volume $u_1$. The operations are as listed below:

(1) Isothermal reception during evaporation at constant temperature $T_1$ between B and R.

(2) Adiabatic expansion between $p_1$ and $p_2$ and $T_1$ and $T_2$, from R to G.

(3) Isothermal rejection at $T_2$ between G and L.

(4) Adiabatic compression of the wet steam between $p_2$ and $p_1$ and $T_2$ and $T_1$, with accompanying condensation till the stuff is brought back to the initial condition of water at B.

The positive work done by the substance on the piston is the $pv$ area below BRG (Fig. 23); the negative work done by the piston on the substance is the area below GLB. The difference is the net work done on the piston, or $E_c$=BRGL. This is the ideal work of

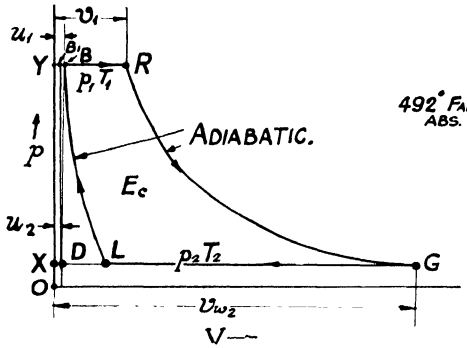



FIG. 23.

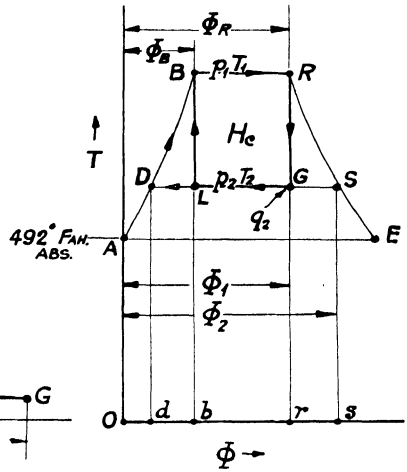



FIG. 24.

the cycle, and its heat equivalent in the TΦ diagram (Fig. 24) is the area $H_c$=BRGL.

Denoting entropy values by the subscripts of the corresponding state points—

Cycle Heat $\quad H_c=(T_1-T_2)(\Phi_R-\Phi_B)$ . . . (1)

Heat supplied $\quad H=T_1(\Phi_R-\Phi_B)$ . . . . . (2)

Hence the Carnot cycle efficiency is —

$$\eta=\frac{H_c}{H}=\frac{T_1-T_2}{T_1} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

If it were possible to go right up to the critical limit with steam, a little over 704° Fahr. at 3200 lb./in.² abs., the maximum possible value of the ideal efficiency to be obtained with a modern condensing engine or turbine (all losses neglected) with a lower condenser limit of, say, 29·25 in. vacuum or 70° Fahr., would be —

$$\eta=\frac{1164-530}{1160}=\frac{634}{1160}=0{\cdot}545.$$

Taking the lower condenser limit of 29·25 in. vacuum, the corre-

sponding Carnot efficiencies for a series of initial pressures between 3200 and 120 lb./in.$^2$ abs. are as follows:

| Press. $p$ lb./in.$^2$ abs. | 3200 | 2400 | 2000 | 1400 | 1200 | 800 | 400 | 200 | 100 |
|---|---|---|---|---|---|---|---|---|---|
| Carnot effy. $\eta$ . | 0·545 | 0·527 | 0·516 | 0·495 | 0·485 | 0·458 | 0·415 | 0·370 | 0·327 |

The curve of thermal efficiency to a base of absolute pressure is shown by AB (Fig. 25).

**58. The " Straight " Rankine Cycle.**—In this case the various operations are assumed to take place in the separate organs: boiler, cylinder or turbine casing, and condenser. The boiler pressure and temperature are assumed to be those at the engine or turbine stop-valve, and the temperature of the condensate, returned as feed to

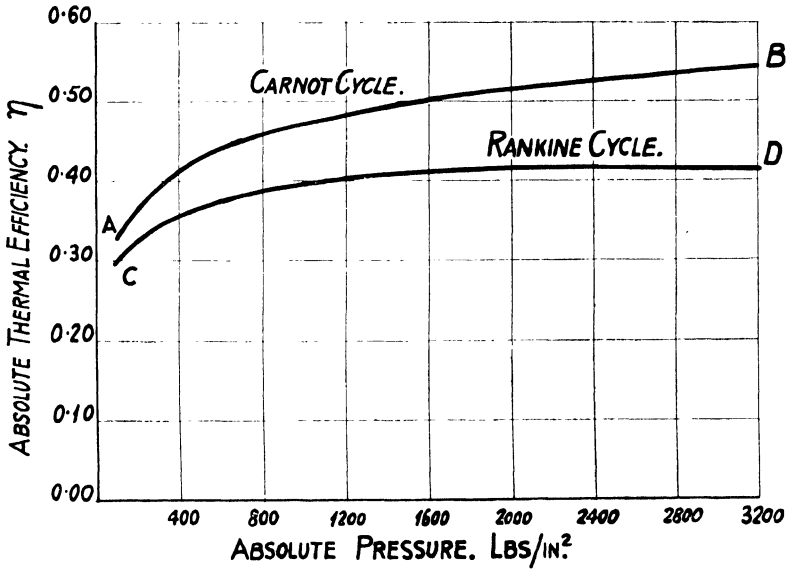



FIG. 25.

the boiler, is the same as the temperature corresponding to the vacuum pressure at the exhaust flange of the engine. No account is taken here of any thermal losses between boiler and engine, or of vacuum drop across the condenser or undercooling or subsequent heating of the condensate between condenser and boiler.

The feed at the exhaust steam temperature is supposed to be supplied with heat in the boiler to raise it to the saturation temperature corresponding to the stop-valve pressure and to convert it into dry saturated steam at this pressure.

Referring to Fig. 23, the isothermal rejection of heat at $T_2$ to the condenser is not stopped at L, as in the case of the Carnot, but carried on to D, where the $H_2O$ stuff is reduced to the liquid

condition and the volume $u_2$ at $p_2$ and $T_2$. This volume of stuff is forced into the boiler against the pressure $p_1$ by the feed pump. In the boiler the feed now occupies the volume $u_2$ at pressure $p_1$ and temperature $T_2$, and is then heated at $p_1$ from $T_2$ to $T_1$. Consequently the state point moves from $B_1$ to B before evaporation at $T_1$ commences. During this process there is a slight increase of volume and $u_1$ is not exactly the same as $u_2$; but the difference is so insignificant that it need not be considered, and for all practical purposes it may be assumed that $B_1$ and B are coincident. With this qualification, the work done by the feed pump in forcing the water into the boiler may be written as $u_2\ (p_1-p_2)\ 144=E_p$.

Compared with the cycle work this quantity is also insignificant and is always neglected. Assuming $B_1$ and B coincident the positive work done by the substance on the piston is the area below BRG, and the negative work done by the piston on the substance is the area below GD. The ideal work of the cycle is the difference, or the area $E_c$=BRGD.

It is less than the whole area YRGX by the small rectangle XYBD, representing the feed pump work (which is greatly exaggerated in the figure).

Again, the usual practice is to neglect this and to take the whole area YRGX as representing the ideal work $E_c$ of the straight Rankine cycle.

In the discussions of the various cycles which follow, this assumption is made; but the student should bear in mind that the results are not rigorously exact.

Since RG (Fig. 23) is an adiabatic, the absolute amount of work done during expansion by the steam is the difference of the internal energies at R and G or $J(I_R-I_G)=J(I_1-I_{w2})$. During expansion at constant pressure $p_1$, the work done by the steam (which up to point of cut-off R is still in communication with the boiler) is $144\ p_1v_1$; and during condensation at $p_2$ and $T_2$, when the steam is in communication with the condenser, the work done by the piston on the steam is $144\ p_2v_{w2}$. The net or cycle work is thus given by—

$$E_c=144p_1v_1+(I_1-I_{w2})-144p_2v_{w2}$$

$$\text{or } \frac{E_c}{J}=H_c=\left(144\frac{p_1v_1}{J}+I_1\right)-\left(144\frac{p_2v_{w2}}{J}+I_{w2}\right)$$

$$\text{and } H_c=(H_1-H_{w2}) \quad . \quad . \quad . \quad . \quad . \quad (4)$$

where $H_1$=total heat of dry steam at $p_1$ and $T_1$

$H_{w2}$=total heat of wet steam of quality $q_2$ at G (Fig. 24).

To be rigorously exact the feed-pump work $\frac{144}{J}(p_1-p_2)u_2$ should be subtracted from this value.

The Rankine cycle heat between the specified pressure limits may thus be taken with sufficient accuracy as the adiabatic heat

drop $H_\phi$ between those limits, and it can always be readily obtained by inspection from the HΦ diagram.

If the adiabatic expansion curve RG is fitted by a law of the form $pv^\gamma = c$, where $\gamma$ is a constant, then as already shown (page 17), the whole work area is—

$$\text{BRGD} = -\int_{p_1}^{p_2} v\,dp$$

$$\text{and } E_i = 144\,\frac{\gamma}{\gamma - 1}(p_1 v_1 - p_2 v_{w2}) \quad . \quad . \quad . \quad (5)$$

This equation is not valid when the steam is initially superheated and finally wet. It is not advisable, in any case, to use it for work calculation. In all instances the TΦ or HΦ method of estimation should be employed.

The estimation of the total heat ($H_{w2}$) of the wet steam at G can be made in two ways. The quality $q_2$ can be obtained, by equating the initial and final entropy values, from—

$$q_2 = \frac{\Phi_1 - \phi_{w2}}{\phi_{r2}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

a calculation involving both water and evaporation entropies at $p_2$. Then—

$$H_{w2} = H_2 - (1 - q_2)L_2 \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

Alternatively, since in Fig. 24 $H_{w2}$ is given by OADG$r$ = OADS$s$ − $r$GS$s$), then—

$$H_{w2} = H_2 - T_2(\Phi_2 - \Phi_1) \quad . \quad . \quad . \quad . \quad . \quad (8)$$

This is the preferable equation to use.

The absolute thermal efficiency of the straight Rankine cycle is now given by—

$$\eta = \frac{H_\phi}{H} = \frac{H_1 - H_{w2}}{H} \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

In this case the heat reception by the stuff takes place between the lower exhaust temperature limit $T_2$ and the upper saturation limit $T_1$ at $p_1$. It is given by the area $d$DBR$r$ (Fig. 24) which, to a very close approximation, is the difference between the total heat $H_1$ of the dry steam at $p_1$ and $T_1$ and the total heat of the water $h_2$ at $p_2$ and $T_2$. Hence—

$$H = (H_1 - h_2) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (10)$$

and the absolute thermal efficiency is—

$$\eta = \frac{H_\phi}{H} = \frac{H_1 - H_{w2}}{H_1 - h_2} \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$$

An inspection of the TΦ diagram (Fig. 24) will show that a larger amount of work is ideally possible per lb. of steam with the straight Rankine than with the Carnot cycle. It is, however, a smaller proportion of the heat received from the source, or in other words

the Rankine cycle is less efficient than the Carnot between the same temperature limits. This is due to the irreversible process of feed heating between D and B.

*Example* 1.—Taking a lower temperature limit of 70·2° F. for the feed, calculate the straight Rankine thermal efficiency, with initial temperatures corresponding to the saturation temperatures at pressures of 3200, 2800, 2400, 2000, 1400, 1000, 600, 400, 200, 100 lb./in.$^2$ abs., and plot the curve of thermal efficiency against absolute pressure.

At $p_2$=0·367 lb./in.$^2$ abs. or 29·25 in. vacuum (from Callendar's table) $h_2$=38·1, $L_2$=1049·6, $H_2$=1087·7, $T_2$=529·8, $\Phi_2$=2·0556.

$$\therefore\ \phi_{e2}=\frac{L_2}{T_2}=\frac{1049\cdot6}{529\cdot8}=1\cdot9811$$

$$\text{and } \phi_{w2}=(\Phi_2-\phi_{e2})=(2\cdot0556-1\cdot9811)=0\cdot0745.$$

Taking the critical pressure value of $p_1$=3200, $H_1$=1002·2, and $\Phi_1$=1·1438,

$$\text{Then}\quad q_2=\frac{\Phi_1-\phi_{w2}}{\phi_{e2}}=\frac{1\cdot1438-0\cdot0745}{1\cdot9811}=\frac{1\cdot0693}{1\cdot9811}=0\cdot540$$

$$\text{and}\quad H_{w2}=H_2-(1-q_2)L_2=1087\cdot7-0\cdot46\times1049\cdot6$$
$$=(1087\cdot7-482\cdot816)$$
$$=604\cdot884 \text{ B.Th.U./lb.}$$

Alternatively
$$H_{w2}=H_2-T_2(\Phi_2-\Phi_1)$$
$$=1087\cdot7-529\cdot8(2\cdot0556-1\cdot1438)$$
$$=(1087\cdot7-483)=604\cdot7 \text{ B.Th.U./lb.}$$

The heat drop is
$$H_\phi=(H_1-H_{w2})$$
$$=(1002\cdot2-604\cdot7)=397\cdot5 \text{ B.Th.U./lb.}$$

also the heat received is
$$H=(H_1-h_2)=(1002\cdot2-38\cdot1)$$
$$=964\cdot1 \text{ B.Th.U./lb.}$$

Hence the absolute thermal efficiency is—

$$\eta=\frac{H_\phi}{H}=\frac{397\cdot5}{964\cdot1}=0\cdot412$$

against the value of 0·545 for the Carnot cycle between the same temperature limits.

Calculating the efficiency values, as above, for the other initial pressures, the following figures are obtained:

| Pressure $p$ lb./in.$^2$ abs. | 3200 | 2800 | 2400 | 2000 | 1400 | 1000 | 600 | 400 | 200 | 100 |
|---|---|---|---|---|---|---|---|---|---|---|
| $\eta$ . . . . | 0·412 | 0·417 | 0·417 | 0·416 | 0·407 | 0·398 | 0·376 | 0·36 | 0·328 | 0·296 |

It will be noted that there is little variation in the value of $\eta$ between 3200 and 1000 lb./in.$^2$ abs.

The curve of efficiency ($\eta$) against pressure ($p$) is also drawn in Fig. 25 as CD, and shows an increasing inferiority in efficiency of the Rankine cycle to that of the Carnot as the upper pressure limit is raised.

**59. Rankine Cycle with Initially Superheated Steam.**—At a given upper pressure limit the temperature limit may be raised by superheating the steam before admission to the engine. This again introduces the condition of heat reception at varying temperatures between the saturation and superheat temperatures. It is a further departure from the condition of reversibility.

Its thermodynamic effect is to very slightly increase the absolute thermal efficiency, and to increase the cycle heat. The use of highly superheated steam, as will be shown later, is now quite general for reciprocating engines and is universal with steam turbines. The reason for its adoption, however, is not thermodynamic but mechanical. Highly superheated steam is advantageous in raising the actual thermal efficiencies of engines and turbines, especially the former, by the suppression of wasteful condensation and leakage in the engine cylinders, and the prevention

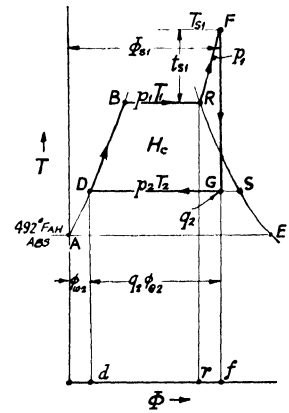



FIG. 26.

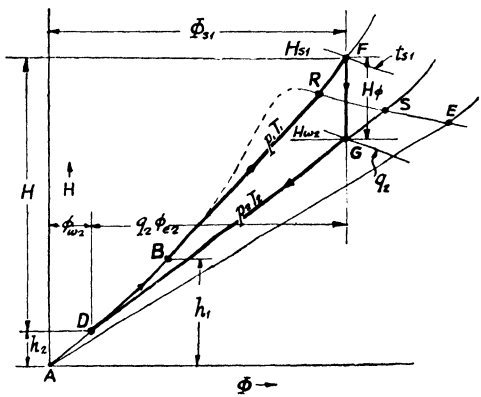



FIG. 27.

of unduly wet steam in the lower stages of turbines, which reduces the "internal" efficiencies.

The PV diagram for the Rankine engine with initially superheated steam is the same as that of Fig. 23, except that the state point at the beginning of adiabatic expansion lies at F to the right of R, since the initial volume is increased from $v_1$ to $v_{s1}$, and the final state point G lies further to the right. The diagram area is thus slightly increased.

The TΦ diagram is shown in Fig. 26, the steam being finally wet at G. If the lower pressure limit were raised the steam might be superheated at $p_2$. This case is shown in Fig. 28. The HΦ diagrams for these two cases are shown in Fig. 27 and Fig. 29.

Referring to Fig. 26, 1 lb. of stuff after conversion into dry steam at R is superheated through a range of $t_{s1}$ degrees to a final temperature $T_{s1}$ at $p_1$, so that the state point at the beginning of

adiabatic expansion is F. The superheat given is represented by the area *r*RF*f*. Reckoned again from the lower temperature limit $T_2$ the heat supply is given by the area *d*DBRF*f*, that is—

$$H=(H_{s1}-h_2) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (12)$$

The heat rejected is given by the area *d*DG*f*, and, therefore, by difference, the cycle heat is DBRFG, or—

$$H_c=H_\phi=(H_{s1}-H_{w2}) \quad . \quad . \quad . \quad . \quad . \quad (13)$$

The quality $q_2$ at G is higher than in the previous case. The absolute thermal efficiency is given by—

$$\eta=\frac{H_c}{H}=\frac{(H_{s1}-H_{w2})}{(H_{s1}-h_2)} \quad . \quad . \quad . \quad . \quad . \quad (14)$$

In the second case shown in Fig. 28, where the steam is super-

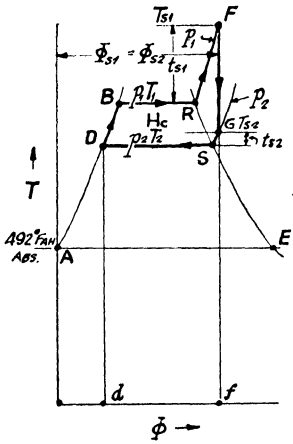



FIG. 28.

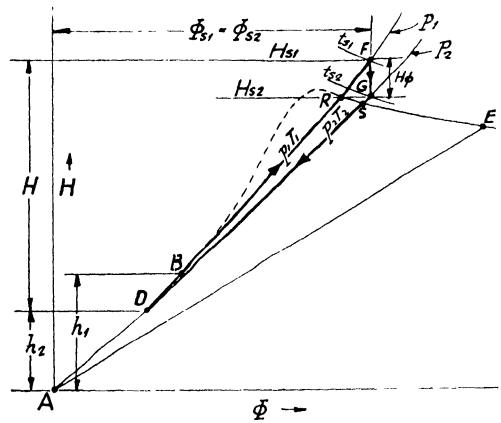



FIG. 29.

heated $t_{s2}$ at $p_2$ after adiabatic expansion, the heat received is again *d*DBRF*f*, or—

$$H=(H_{s1}-h_2) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (15)$$

At $p_2$ and $T_{s2}$ the steam has a total heat $H_{s2}$, and the heat rejected is *d*DSG*f*. The difference (*d*DBRF*f*—*d*DSG*f*)=DBRFGS is the ideal cycle heat and obviously the difference of the total heats at F and G, or—

$$H_c=H_\phi=(H_{s1}—H_{s2}) \quad . \quad . \quad . \quad . \quad . \quad (16)$$

To obtain $H_{s2}$ the final temperature $T_{s2}$ has to be calculated from the adiabatic equation—

$$T_{s2}=T_{s1}\left(\frac{p_2}{p_1}\right)^{\frac{3}{13}} \quad . \quad . \quad . \quad . \quad . \quad . \quad (17)$$

This gives $t_{s2}=(T_{s2}-T_2)$ and $c_{p2}$ can be obtained from the specific heat chart,

$$\text{then} \quad H_{s2}=H_2+c_{p2}t_{s2} \quad . \quad . \quad . \quad . \quad . \quad . \quad (18)$$

When a large scale HΦ chart is available the total heats and heat drops can be scaled directly from it as indicated in Figs. 27 and 29.

*Example* 2.—Calculate the cycle heat and absolute thermal efficiency of an engine working on the straight Rankine cycle with initially dry steam at 2000 lb./in.$^2$ abs. and temperature 636·5° Fahr., and exhausting at 29·25 in. vacuum or 0·367 lb./in.$^2$ abs., and temperature 70·2° Fahr.

In this case $H_1=1149{\cdot}8$, $\Phi_1=1{\cdot}2985$ (tables).

At $t_2=70{\cdot}2°$ F., $H_2=1087{\cdot}7$, $h_2=38{\cdot}1$, $\Phi_2=2{\cdot}0556$,

$T_2=529{\cdot}8°$ F.

$\therefore H_{w2}=H_2-T_2(\Phi_2-\Phi_1)$

$=1087{\cdot}7-529{\cdot}8\ (2{\cdot}0556-1{\cdot}2985)$

$=1087{\cdot}7-529{\cdot}8\times0{\cdot}7571$

$=(1087{\cdot}7-401{\cdot}1)=686{\cdot}6$ B.Th.U./lb.

Cycle heat $H_c=H_\phi=(H_1-H_{w2})$

$=(1149{\cdot}8-686{\cdot}6)=463{\cdot}2$ B.Th.U./lb.

Heat received $H=(H_1-h_2)=(1149{\cdot}8-38{\cdot}1)=1111{\cdot}7$ B.Th.U./lb.

Absolute thermal efficiency $\eta=\dfrac{H_\phi}{H}=\dfrac{463{\cdot}2}{1111{\cdot}7}=0{\cdot}416$

*Example* 3.—Find the cycle heat and absolute thermal efficiency of the engine (Example 2) if the same temperature limits are retained but the initial pressure is reduced from 2000 to 400 lb./in.$^2$ abs.

Here at $p_1=400$ and $t_1'=636{\cdot}5°$ Fahr., the steam is superheated $t_{s1}=(t_1'-t_1)=(636{\cdot}5-443{\cdot}7)=192{\cdot}8°$ Fahr.

To find $H_{s1}$, the value of $c_{p1}$ from the specific heat chart (Fig. 13), at $p_1=400$ and $t_{s1}=192{\cdot}8°$ Fahr., is 0·62 ; $H_1=1211{\cdot}6$.

Hence $H_{s1}=H_1+c_{p1}t_{s1}=1211{\cdot}6+0{\cdot}62\times192{\cdot}8$

$=1211{\cdot}6+119{\cdot}6=1331{\cdot}2$ B.Th.U./lb.

$T_{s1}=636{\cdot}5+459{\cdot}6=1096{\cdot}1°$ F., $T_1=(443{\cdot}7+459{\cdot}6)=903{\cdot}3°$ F.

The entropy of superheat is—

$$\phi_{s1}=c_{p1}\log_e\frac{T_{s1}}{T_1}=0{\cdot}62\times2{\cdot}3\log\frac{1096{\cdot}1}{903{\cdot}3}$$

$$=0{\cdot}62\times2{\cdot}3\times\log 1{\cdot}2134$$

$$=0{\cdot}62\times2{\cdot}3\times0{\cdot}084=0{\cdot}1198$$

$\therefore \Phi_{s1}=\Phi_1+\phi_{s1}=(1{\cdot}4933+0{\cdot}1198)=1{\cdot}6131$ which checks on the HΦ chart.

$\Phi_{s1}=1{\cdot}6131$

$\Phi_2=2{\cdot}0556$

$H_2=1087{\cdot}7$

$H_{w2}=H_2-T_2(\Phi_2-\Phi_{s1})$

$=1087{\cdot}7-529{\cdot}8(2{\cdot}0556-1{\cdot}6131)$

$=1087{\cdot}7-529{\cdot}8\times0{\cdot}4425$

$=(1087{\cdot}7-234{\cdot}43)=853{\cdot}27$ B.Th.U./lb.

which checks on the HΦ chart within the limit of accuracy possible.

Cycle heat $H_c=H_\phi=(H_{s1}-H_{w2})$

$=(1331{\cdot}2-853{\cdot}27)=477{\cdot}93$ B.Th.U./lb.

Heat received $H=(H_{s1}-h_2)=(1331{\cdot}2-38{\cdot}1)=1293{\cdot}1$ B.Th.U./lb.

Absolute thermal efficiency $\eta = \frac{H_\phi}{H} = \frac{477 \cdot 93}{1293 \cdot 1} = 0 \cdot 3696$.

In this second case with lower initial pressure and moderately superheated steam there is an increase of only 3 per cent. in the cycle heat and a decrease of 11 per cent. in thermal efficiency.

From a thermodynamic standpoint this case indicates that the higher temperature ranges should be obtained by increasing the upper pressure limit instead of retaining lower pressures and raising the temperature by superheating. This is the trend at present in power-station practice.

*Example* 4.—Compare the cycle heats and absolute thermal efficiencies for the engine (Example 3) when the steam is (*a*) initially superheated 192·8° Fahr.; (*b*) initially dry at the initial pressure of 400 lb./in.$^2$ abs.

For case (*b*) $p_1 = 400$, $H_1 = 1211 \cdot 6$, $\Phi_1 = 1 \cdot 4933$, $\Phi_2 = 2 \cdot 0556$, $H_2 = 1087 \cdot 7$, $h_2 = 38 \cdot 1$, $T_2 = 529 \cdot 8°$ Fahr.

$$\therefore H_{w2} = 1087 \cdot 7 - 529 \cdot 8\,(2 \cdot 0556 - 1 \cdot 4933)$$
$$= 1087 \cdot 7 - 529 \cdot 8 \times 0 \cdot 5623$$
$$= (1087 \cdot 7 - 297 \cdot 9) = 789 \cdot 8 \text{ B.Th.U./lb.}$$

$H_\phi = H_\phi = (H_1 - H_{w2}) = (1211 \cdot 6 - 789 \cdot 8) = 421 \cdot 8$ B.Th.U./lb.

$H = (H_1 - h_2) = (1211 \cdot 6 - 38 \cdot 1) = 1173 \cdot 5$ B.Th.U./lb.

$$\eta = \frac{421 \cdot 8}{1173 \cdot 5} = 0 \cdot 3594$$

Case (*a*) $H_\phi = 477 \cdot 93$ $\eta = 0 \cdot 3696$
(*b*) $H_\phi = 421 \cdot 8$ $\eta = 0 \cdot 3594$

While the cycle heat is increased 13 per cent., the absolute thermal efficiency is increased only 2·8 per cent. by the particular range of superheat employed at 400 lb./in.$^2$ abs.

**60. Incomplete Rankine Cycle.**—In the Rankine cycle the expansion is carried down from the initial to the exhaust pressure in the working chamber. This condition is fulfilled in practice in the steam turbine, but not by the reciprocating engine. At pressures below atmospheric the specific volume of the steam, as shown by the tables, increases very rapidly, and impracticably large cylinder volumes result if the complete range of expansion is attempted.

In the reciprocator, therefore, for constructional and economic reasons the steam is always released from the cylinder at a pressure higher than the exhaust pressure, which, in the ideal case, is the condenser pressure.

The condition of reversibility is thus further departed from during the exhaust from the cylinder to the condenser, as the steam is throttled down from the terminal to the exhaust or condenser pressure. This irreversible process results in an increase of entropy of the wet steam at the exhaust pressure.

The modified PV diagram of the Rankine engine is shown in Fig. 30 for the general case, in which the steam is initially super-

heated and finally wet. The corresponding TΦ diagram is shown in Fig. 31.

In Fig. 30 the steam is expanded adiabatically from the initial pressure $p_1$ to the release pressure $p_2$ between F and G, the cut-off and release points, the initial volume being $v_{s1}$ and final volume $v_{w2}$. In the TΦ diagram (Fig. 31) the state point G lies at $p_2$ on the constant volume (dotted) curve $v_{w2}$. In the cylinder there is a drop of pressure ($p_2$—$p_3$) at constant cylinder volume $v_{w2}$. If expansion were carried down to the exhaust pressure $p_3$, an additional amount of energy, represented by the " toe " area GG'K, would be converted into work. Owing to the early release at G this energy is carried over and appears as heat in the exhaust at $p_3$ (assuming no radiation or conduction loss). The effect of this carry-over reheat is to raise the quality of the steam at exhaust, and the state point is moved

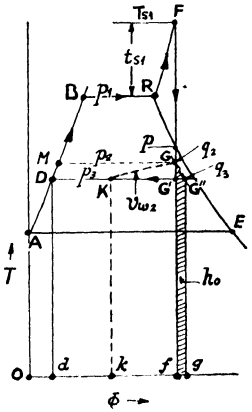



Fig. 31.

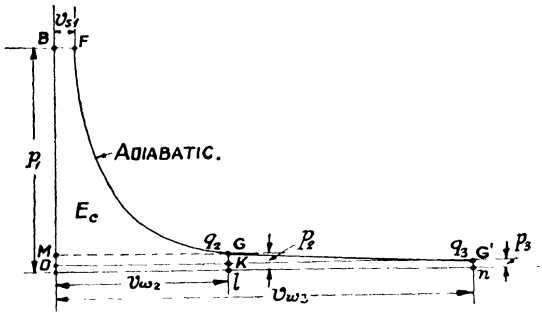



Fig. 30.

from position G at $p_2$ to G'' at $p_3$, defining the final quality $q_3$ (Fig. 31). If the expansion were complete the final state point would be G' at $p_3$. It will be evident, therefore, that by the displacement of the state point from G' to G'' an additional amount of heat, the " toe " carry-over, represented by the rectangular area G'G''$gf$ or $h_o$, is carried away in the exhaust. The initial total heat of the steam at $p_1$ and $t_{s1}$ is $H_{s1}$; the final total heat of the wet steam at $p_3$ is OADG''$g$ or $H_{w3}$, and the heat converted to work is their difference, or the cycle heat is—

$$H_c=(H_{s1}-H_{w3}) \quad . \quad . \quad . \quad . \quad . \quad . \quad (19)$$

Let the total heat at G' with complete expansion be denoted by $H_{w'3}$, then—

$$H_{w3}=H_{w'3}+h_o \quad . \quad . \quad . \quad . \quad . \quad . \quad (20)$$

The carry-over $h_o$ can be determined by the energy law. In Fig. 30, GG' is an adiabatic. The triangular area KGG', representing

the heat $h_o$ convertible into work, is the difference of the internal energies $I_{w2}$ and $I_{w'3}$ at G and G', less the heat equivalent of the rectangular area $l$KG'$n$.

Thus $$h_o=(I_{w2}-I_{w'3})-\frac{144}{J}p_3(v_{w'3}-v_{w2})$$

$$=I_{w2}+\frac{144}{J}p_2v_{w2}-I_{w'3}+\frac{144}{J}p_3v_{w'3}+\frac{144}{J}p_3v_{w2}-\frac{144}{J}p_2v_{w2}$$

$$=I_{w2}+\frac{144}{J}p_2v_{w2}-H_{w'3}=(H_{w2}-H_{w'3})-\frac{144}{J}(p_2-p_3)v_{w2}$$

Substituting in (20)—

$$H_{w3}=H_{w2}-\frac{144}{J}(p_2-p_3)v_{w2}$$

Substituting in (19)—

$$H_c=(H_{s1}-H_{w2})+\frac{144}{J}(p_2-p_3)v_{w2}$$

$$=H_{\phi 2}+\frac{144}{J}(p_2-p_3)v_{w2} \quad . \quad . \quad . \quad . \quad . \quad . \quad (21)$$

where $H_{\phi 2}$ is the adiabatic heat drop between the initial pressure $p_1$ and the release pressure $p_2$.

This result can be ascertained directly from the PV diagram, Fig. 30. The area MBFG is the equivalent of the adiabatic heat drop between $p_1$ and $p_2$ or $H_{\phi 2}$ and the rectangle DMGK is the equivalent of $\frac{144}{J}(p_2-p_3)v_{w2}$. The total work area DBFGK is the sum of the two.

The corresponding areas on the TΦ diagram (Fig. 31) are MBRFG and DMGK, so that the area DBRFGK also represents $H_c$. While this area bounded by the constant volume curve $v_{w2}$ at the lower end gives the heat value, the curve does not give any indication of the state of the stuff during the pressure drop between $p_2$ and $p_3$. The actual condition curve, which is the locus of the state points between G and G'', and shown dotted, is indeterminate. The student should bear in mind that, for the conditions specified, the curve GK has no physical significance, although the heat rejected to the condenser is given both by the area $d$DKG$f$ and the area $d$DG''$g$. The triangular area KGG' is equal to the carry-over or reheat area $f$G'G$g$.

This important item, the heat in the exhaust steam, in the case of an actual steam engine cycle is further considered in Chapter VI.

The absolute thermal efficiency of the incomplete Rankine cycle is given by—

$$\eta=\frac{H_c}{H}=\frac{H_{\phi 2}+\frac{144}{J}(p_2-p_3)v_{w2}}{H} \quad . \quad . \quad . \quad . \quad (22)$$

The heat received is $H=(H_{s1}-h_3)$ . . . . . (23)

To find the value of $v_{w2}$, the final specific volume at the release pressure $p_2$. If the steam is superheated $t_{s1}$ at $p_1$, the adiabatic FG (Fig. 31) will cross the saturation curve at some intermediate pressure $p$, having total entropy $\Phi$; and the condition $\Phi_{s1}=\Phi$ enables $p$ to be obtained from the steam table. Alternatively, $p$ can be located by inspection on the H$\Phi$ diagram and the value of the corresponding dry steam volume $v$ from the table. Let $x$=fraction of stroke at which steam is cut off, then if $v_{s1}$ is the initial specific volume at $p_1$ and $t_{s1}$, the final volume at $p_2$ is given by $v_2'=\frac{v_{s1}}{x}$, where $v_2'$ will have a value $v_{w2}=q_2v_2$ if the steam is finally wet. The value of the terminal pressure is thus given by—

$$p_2=p\left(\frac{xv}{v_{s1}}\right)^{1\cdot135} \quad . \quad . \quad . \quad . \quad . \quad . \quad (24)$$

where $\gamma=1\cdot135$ in the adiabatic exponent for the expansion of initially dry steam.

If the value of $v_2'$ obtained with the specified cut-off $x$ is less than

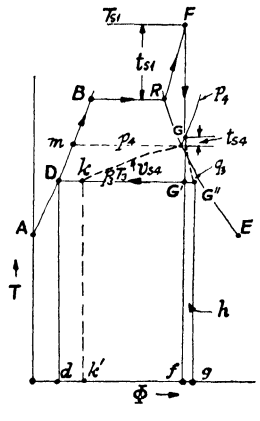



FIG. 33.

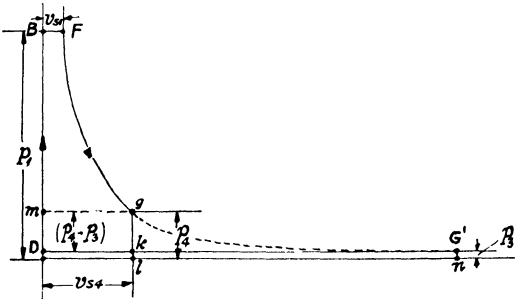



FIG. 32.

$v$, then the steam will be finally superheated at some higher terminal pressure $p_4$ given by—

$$p_4=p_1x^{1\cdot3} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (25)$$

where $\gamma=1\cdot3$ is the adiabatic exponent for superheated steam.

The latter case is shown in Figs. 32 and 33. The adiabatic F$g$ cuts the release pressure curve $p_4$ at $g$, where the range of superheat is $t_{s4}$. In this case the final specific volume obtained from the equation

$$v_{s4}=v_{s1}\left(\frac{p_1}{p_4}\right)^{\frac{1}{1\cdot3}} \quad . \quad . \quad . \quad . \quad . \quad . \quad (26)$$

should check with $v_2'$, as can be seen on the PV diagram. In Fig. 32 the cycle work is given by DBF$gk$=$m$BF$g$+D$mgk$, or—

$$H_c=H_{\phi 4}+\frac{144}{J}(p_4-p_3)v_{s4} \quad . \quad . \quad . \quad . \quad . \quad (27)$$

where $H_{\phi 4}$ is the adiabatic heat drop between $p_1$ and $p_4$. As in the previous case, the heat rejected is given in Fig. 33 by the areas $d$D$k$G$f$=$d$DG″$g$ and $h_0$=$kg$G′=G′G″$gf$. The student will find it an instructive exercise to draw down the HΦ diagrams corresponding to these two cases of the incomplete Rankine cycle.

*Example* 5.—An engine working on the incomplete Rankine cycle is supplied with steam at 500 lb./in.² abs. and temperature 700° Fahr., and exhausts at 15 lb./in.² abs. Cut-off takes place at 0·2×stroke. Calculate the cycle heat and the absolute thermal efficiency.

$p_1$=500, $t_{s1}$=(700−466)=234° F., $H_1$=1212·4.
Specific heat chart, $c_{p1}$=0·638 at $p_1$=500 and $t_{s1}$=234° Fahr.
$\therefore$ $H_{s1}=H_1+c_{p1}t_{s1}$
$=1212\cdot4+0\cdot638\times234$
$=(1212\cdot4+149\cdot29)=1361\cdot69$ B.Th.U./lb.
This checks closely on the HΦ diagram.

Total entropy $\Phi_{s1}=\Phi_1+c_{p1}\log_e\frac{T_{s1}}{T_1}$
$=1\cdot4732+0\cdot638\times2\cdot3\times\log\frac{1159\cdot6}{925\cdot6}$
$=1\cdot4732+0\cdot1432=1\cdot6164$

$T_{s1}$=1159·6° F.
$T_1$=925·6° F.
$\Phi_1$=1·4732.

From tables $\Phi$=1·6164 is approximately the total entropy at 90 lb./in.² abs. Take dry steam pressure $p$=90 and vol. $v$=4·891. This pressure also checks on the HΦ chart.

Terminal pressure $p_2=p\left(\frac{xv}{v_{s1}}\right)^{1\cdot135}$ $x$=0·2

$=90\left(\frac{0\cdot2\times4\cdot891}{v_{s1}}\right)^{1\cdot135}$

From volume ratio chart (Fig. 15), $r$=1·39 at $p_1$=500 and $t_{s1}$=234. $v_1$=0·938. $\therefore$ $v_{s1}=rv=1\cdot39\times0\cdot938=1\cdot304$ ft.³/lb.

$\therefore$ $p_2=90\left(\frac{0\cdot2\times4\cdot891}{1\cdot304}\right)^{1\cdot135}$

$=\frac{90}{1\cdot386}=65$ lb./in.² abs.

(Tables) $\Phi_2$=1·6415, $H_2$=1181·8, $T_2$=757·3.

$\therefore$ $H_{w2}=H_2-T_2(\Phi_2-\Phi_{s1})$
$=1181\cdot8-757\cdot3(1\cdot6415-1\cdot616)$
$=1181\cdot8-757\cdot3\times0\cdot0255$
$(1181\cdot8-19\cdot312)=1162\cdot5$ B.Th.U./lb.

This checks closely on the HΦ chart.

Heat drop to terminal pressure $p_2$,

$$H_{\phi 2}=(H_{s1}-H_{w2})=(1361{\cdot}69-1162{\cdot}5)=199{\cdot}19 \text{ B.Th.U./lb.},$$

which again checks closely on the chart.

$$v_{w2}=\frac{v_{s1}}{x}=\frac{1{\cdot}304}{0{\cdot}2}=6{\cdot}52 \text{ ft.}^3\text{/lb.}$$

$$\therefore \ \frac{144}{J}(p_2-p_3)v_{w2}=\frac{144}{778}(65-15)6{\cdot}52=\frac{144\times 50\times 6{\cdot}52}{778}=60{\cdot}4$$

$$\text{Cycle heat } H_c=H_{\phi 2}+\frac{144}{J}(p_2-p_3)v_{w2}$$

$$H_c=(199{\cdot}19+60{\cdot}4)=259{\cdot}59 \text{ B.Th.U./lb.}$$

$$\text{Heat supplied } H=(H_{s1}-h_3) \qquad h_3=181$$
$$=(1361{\cdot}69-181)=1180{\cdot}69.$$

Absolute thermal efficiency,

$$\eta=\frac{H_c}{H}=\frac{259{\cdot}59}{1180{\cdot}69}=0{\cdot}22.$$

With complete expansion the cycle heat would be $H_\phi=303$, and the thermal efficiency $\eta=0{\cdot}256$; but a much larger and more expensive engine would be required.

*Example* 6.—Find the cycle heat and absolute thermal efficiency of the engine working on the incomplete Rankine cycle (Example 5) when the cut-off is lengthened to 0·3×stroke, the exhaust pressure remaining the same.

With the same initial conditions, $p=90$ and $v=4{\cdot}891$.

With cut-off at 0·3, $v_3'=\frac{v_{s1}}{x}=\frac{1{\cdot}304}{0{\cdot}3}=4{\cdot}346$.

$v_3'<v$ and hence the steam is superheated at $p_4$, and $p_4=p_1x^{1{\cdot}3}=500\times 0{\cdot}3^{1{\cdot}3}=0{\cdot}209\times 500=104{\cdot}5$ lb./in.$^2$ abs.

Here $v_3'=v_{s4}=4{\cdot}346$ and

$$\frac{144}{J}(p_4-p_3)v_{s4}=\frac{144}{778}(104{\cdot}5-15)4{\cdot}346$$
$$=\frac{144\times 89{\cdot}5\times 4{\cdot}346}{778}=72.$$

Temperature at $p_4=104{\cdot}5$

$$T_{s4}=T_{s1}\left(\frac{p_4}{p_1}\right)^{\frac{3}{13}}$$
$$=1159{\cdot}6\left(\frac{104{\cdot}5}{500}\right)^{\frac{3}{13}}$$
$$=\frac{1159{\cdot}6}{1{\cdot}435}=806^\circ \text{ F. abs.}=346{\cdot}4^\circ \text{ F.}$$

$t_{s4}=(346{\cdot}4-330)=16{\cdot}4°$ F., $c_{p4}=0{\cdot}55$

$$H_{s4}=H_1+c_{p4}t_{s4}$$
$$=1191+0{\cdot}55\times16{\cdot}4$$
$$=1191+9{\cdot}02=1200 \text{ B.Th.U./lb.},$$

which checks on the HΦ chart.

Cycle heat, $H_c=H_{\phi4}+\frac{144}{J}(p_2-p_2)v_{s4}=(161{\cdot}69+72)=233{\cdot}69$ B.Th.U./lb., and efficiency, $\eta=\frac{H_c}{H}=\frac{233{\cdot}69}{1180{\cdot}69}=0{\cdot}198$.

**61. Reheat Cycle.**—In the Rankine cycle, the mass of stuff entering the high-pressure end of the engine or turbine receives all its heat from the boiler before adiabatic expansion commences in the working chamber.

Under modern conditions of practice in many cases further heat is added to the stuff after part of the expansion has taken place.

The steam may be withdrawn from the working chamber, led back to a heater at the boiler and resuperheated at the extraction pressure, then passed back at this or a slightly lower pressure, and further expanded in the working chamber to a still lower pressure, then again extracted and resuperheated, and finally expanded to the release pressure. Alternatively the steam at intermediate pressures may be passed through receivers containing heating coils, and reheated without extraction from the engine.

The ideal cycle is now no longer "straight," although after each reheat the mass of steam expands adiabatically between the extraction pressure limits.

The term "reheat cycle" will be applied to this case. It is illustrated for one reheat by the PV, TΦ, and HΦ diagrams, Figs. 34, 35, and 36, the expansion being complete, that is, carried down to the exhaust pressure. The lettering corresponds in the three diagrams.

In Fig. 34 the total work done per lb. is given by the area $BFGF_1G_1D$, and its heat equivalent in Fig. 35 is $BRFGF_1G_1D$. This is the sum of $BRFGR_1B_1$, the adiabatic heat drop $H_{\phi2}$ between $p_1$ and the reheat pressure $p_2$, and $B_1R_1F_1G_1D$, the adiabatic heat drop $H_{\phi3}$ between the reheat pressure $p_2$ and the exhaust pressure $p_3$. The heat received before expansion commences is represented by the area $dDBRFf$ or $(H_{s1}-h_3)=H'$. The reheat at $p_2$ is given by the area $GF_1f'f$, and is the difference of the total heats at G and $F_1$, that is, $h'_r=(H_{s2}-H_{s'2})$ (Fig. 36).

The gross amount of heat received is thus—

$$H=(H_{s1}-h_3)+(H_{s2}-H_{s'2}) \quad . \quad . \quad . \quad . \quad (28)$$

The heat rejected is given by the area $dDG_1f'$ or $(H_{w3}-h_3)=h$ (Fig. 35); the work done or cycle heat is $(H-h)$,

or
$$H_c=(H_{s1}-h_3)+(H_{s2}-H_{s'2})-(H_{w3}-h_3)$$
$$=(H_{s1}-H_{s'2})+(H_{s2}-H_{w3})$$

or
$$H_c=H_{\phi2}+H_{\phi3} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (29)$$

as already inferred by inspection of Fig. 34.

The absolute thermal efficiency now becomes—

$$\eta = \frac{H_c}{H} = \frac{(H_{\phi 2} + H_{\phi 3})}{(H_{s1} + h_r' - h_3)} \quad . \quad . \quad . \quad . \quad . \quad . \quad (30)$$

or

$$\eta = \frac{(H_{s1} - H_s'_2) + (H_{s2} - H_{w3})}{(H_{s1} + H_{s2} - H_s'_2 - h_3)} \quad . \quad . \quad . \quad . \quad (31)$$

*Example* 7.—Find the cycle heat and absolute thermal efficiency of an engine working with one reheat between an initial pressure of 500 lb./in.$^2$ abs. and 700° Fahr. and exhaust pressure 0·5 lb./in.$^2$

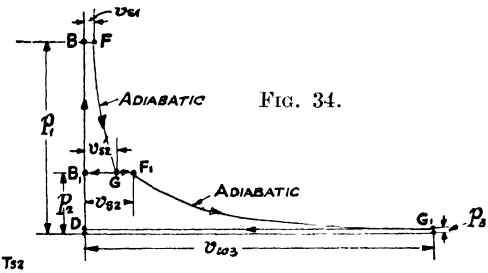



FIG. 34.

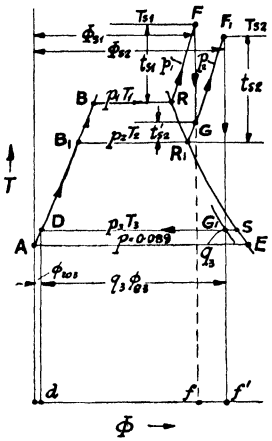


FIG. 35.

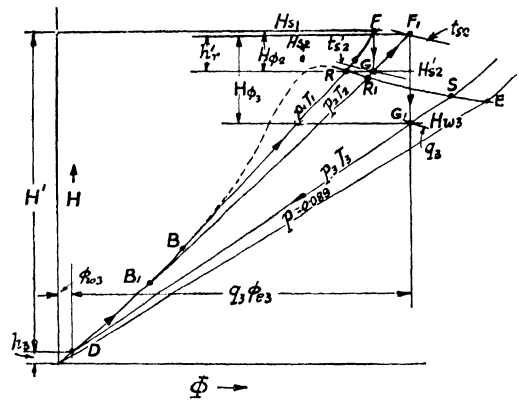


FIG. 36.

abs., the reheat pressure and temperature being 150 lb./in.$^2$ abs. and 600° Fahr. Assume complete expansion.

As the initial steam conditions, $p_1 = 500$ and $t_{s1} = 234°$ F., are the same as in Example 5, $H_{s1} = 1361{\cdot}69$ B.Th.U./lb. By the methods of calculation previously illustrated, and by check on the HΦ diagram, the total heat at G (Fig. 35), after adiabatic expansion to $p_2 = 150$, is $H_s'_2 = 1233{\cdot}8$.

Hence $H_{\phi 2} = (H_{s1} - H_s'_2) = (1361{\cdot}69 - 1233{\cdot}8) = 127{\cdot}89$ B.Th.U./lb., at $p_2 = 150$, $t_2' = 600$ ; $t_{s2} = (t_2' - t_2) = (600 - 358) = 242°$ Fahr.

Total heat, at beginning of expansion after reheat at $p_2$, is

$H_{s2}=1330$, by calculation and check on HΦ diagram, hence the reheat $h_r'=(H_{s2}-H_s'_2)=(1330-1233{\cdot}8)=96{\cdot}2$ B.Th.U./lb.

The exhaust pressure $p_3=0{\cdot}5$ and $h_3=47{\cdot}4$.

Hence the heat received $H=(H_{s1}-h_3+h_r')$
$=(1361{\cdot}69-47{\cdot}4+96{\cdot}2)=1410{\cdot}49$ B.Th.U./lb.

The adiabatic heat drop between $p_2=150$ at 600° Fahr. and $p_3=0{\cdot}5$, by calculation and check on HΦ diagram, is $H_{\phi 3}=405$ B.Th.U./lb.

Hence the cycle heat is, $H_c=(H_{\phi 2}+H_{\phi 3})=(127{\cdot}89+405)=532{\cdot}9$ B.Th.U./lb.

The absolute thermal efficiency is—

$$\eta=\frac{H_c}{H}=\frac{532{\cdot}9}{1410{\cdot}49}=0{\cdot}377.$$

The student should work out this calculation on the HΦ chart for the same initial and exhaust conditions and the same reheat temperature for a series of progressively decreasing reheat pressures, and compare the cycle heats and efficiences. The values can be plotted to a base of reheat pressure.

Only the case of complete expansion between initial and reheat pressures and reheat and exhaust pressures has been considered. Where, as in the general case, expansion is incomplete in the reciprocator in each stage, the steam conditions at the reheat pressure before reheat and those at the release can be determined by applying the calculation for the incomplete Rankine cycle in succession to the two stages of the expansion. It is not proposed to go into this in further detail.

**62. Jacketed Cycle.**—A variant of the reheat cycle has been employed for many years in the form of the jacketed steam engine cycle. Instead of additional heat being supplied at the particular pressure at an intermediate stage of expansion in the working chamber, it is continuously transmitted through the chamber wall to the expanding steam. In the ideal jacketed case the steam is initially dry at beginning of expansion and is maintained dry throughout expansion, whether complete or incomplete, by the heat transmitted through the jacket wall, which acts essentially as a "reheater" conveying heat from the external source.

There is little justification thermodynamically for jacketing, but there is mechanically, on account of its beneficial effect on the temperature of the cylinder wall, which helps to suppress wasteful condensation. Highly superheated steam admitted directly to the cylinder, which is much more effective for this purpose, is now being used instead of jacket steam. The subject of steam jacketing is considered in Chapter VI, p. 206, and the practical use of superheated steam in Chapters VII and XXIV.

The PV and TΦ diagrams of the ideal jacketed cycle are shown in Figs. 37 and 38. The PV diagram, whether for complete or incomplete expansion, has the same form as the Rankine, the difference being

that the expansion curve is the curve of dry saturated steam $pv^{1\cdot065}=c$, instead of the adiabatic $pv^{1\cdot135}=c$. The cycle work $E_c$ is given by the $pv$ area BRG′D (Fig. 37) and the cycle heat $H_c$ by BRG′D=($d$DBR$r$+$r$RG′$g'$−$d$DG′$g'$) (Fig. 38). The heat supplied by the jacket is given by the area $r$RG′$g'$. The cycle heat can be expressed in terms of the $pv$ area—

$$H_c=\frac{E_c}{J}=\frac{144}{J}\frac{n}{n-1}(p_1v_1-p_3v_3)$$

Since $n=\frac{16}{15}$, $\frac{n}{n-1}=16$, so that

$$H_c=2\cdot961(p_1v_1-p_3v_3) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (32)$$

Referring to the TΦ diagram (Fig. 38), the heat supplied is given by $d$DBRG′$g'$=DBRG′+$d$DG′$g'$, or—

$$H=H_c+(H_3-h_3) \quad . \quad . \quad . \quad . \quad . \quad . \quad (33)$$

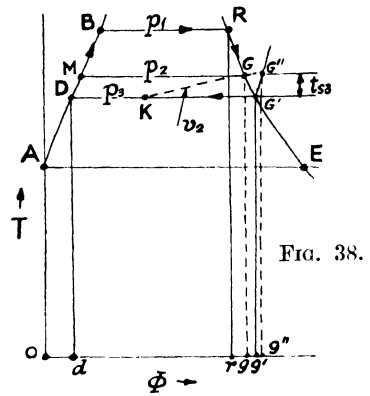



Fig. 38.

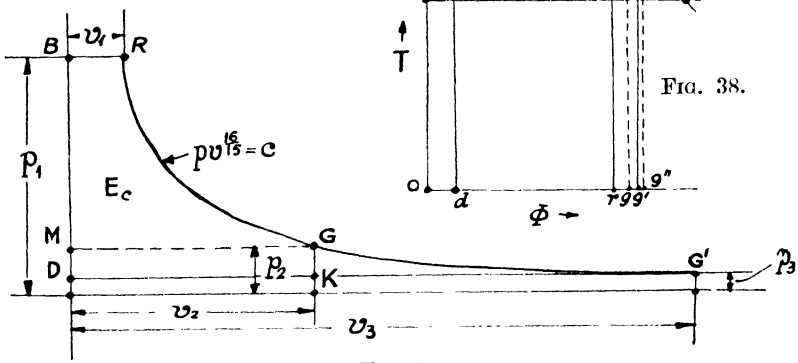



Fig. 37.

The heat from the jacket, during expansion, is—

$$h_J=(H_c+H_3)-H_1 \quad . \quad . \quad . \quad . \quad . \quad . \quad (34)$$

This result can also be deduced from the energy law thus—

$$h_J=(I_3-I_1)+\frac{144}{J}\frac{(p_1v_1-p_3v_3)}{n-1}$$

Expressing the internal energies in terms of the total heats and pressure energies and reducing, this becomes—

$$h_J=2\cdot961(p_1v_1-p_3v_3)+H_3-H_1$$
$$=(H_c+H_3)-H_1 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (35)$$

The absolute thermal efficiency is—

$$\eta=\frac{H_c}{H}=\frac{2{\cdot}961(p_1v_1-p_3v_3)}{2{\cdot}961(p_1v_1-p_3v_3)+(H_3-h_3)} \quad . \quad . \quad . \quad (36)$$

*Incomplete Expansion.*—In this case the cycle work is given by the area BRGKD=MBRG+DMGK (Fig. 37), that is—

$$H'_c=2{\cdot}961(p_1v_1-p_2v_2)+\frac{144}{J}(p_2-p_3)v_2$$

Denoting MBRG by $H_{c2}$—

$$H_c'=H_{c2}+0{\cdot}185(p_2-p_3)v_2 \quad . \quad . \quad . \quad . \quad (37)$$

The heat supplied is $d$DBRG$g$=MBRG+$d$DMG$g$ (Fig. 38),

$$\text{or } H'=2{\cdot}961(p_1v_1-p_2v_2)+(H_2-h_3)$$
$$=H_{c2}+(H_2-h_3) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (38)$$

The jacket heat is given by $h_J=H'-(H_1-h_3)$

$$\text{or } h_J'=2{\cdot}961(p_1v_1-p_2v_2)+H_2-H_1$$
$$=(H_{c2}+H_2)-H_1 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (39)$$

With a given cut-off $x$, the pressure at release is given by—

$$p_2=p_1x^{\frac{1}{1{\cdot}5}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (40)$$

As in the case of the incomplete Rankine cycle, the state point of the steam at $p_2$ at exhaust to the condenser is G″ not K, since the volume curve GK would only represent the true locus of the state points if the cooling at constant volume took place reversibly in the cylinder with constant mass of stuff.

The area KGG′ (Fig. 38), as before, is the carry over $h_o$, and is the difference between the cycle heats with complete and incomplete expansion, that is—

$$h_o=2{\cdot}961(p_2v_2-p_3v_3)+0{\cdot}185(p_2-p_3)v_2 \quad . \quad . \quad (41)$$

It is obvious that since the steam is dry at release pressure $p_2$, it will be superheated at the exhaust pressure $p_3$ by a range $t_{s3}$ (Fig. 38). The total heat here is given by $H_{s3}=(H_3+h_o)$, and the quality of superheat can be ascertained either from a superheated steam table or by inspection from the HΦ diagram. The heat discharged to the condenser is given by—

$$d\text{DG′G″}g''=d\text{DKG}g, \quad \text{or} \quad h_e=(H_{s3}-h_3)=(H_2-h_3) \quad . \quad (42)$$

*Example* 8.—A jacketed engine works between the initial and exhaust pressure limits of 200 lb./in.² abs. and 15 lb./in.² abs. Calculate the cycle heat, absolute thermal efficiency, jacket heat, heat discharged to the condenser, and the cylinder volume per lb. of steam, when expansion is complete.

$p_1=200$, $v_1=2{\cdot}29$, $p_3=15$, $v_3=26{\cdot}27$.

Cycle heat $H_c=2{\cdot}961(p_1v_1-p_3v_3)$

$=2{\cdot}961(200\times2{\cdot}29-15\times26{\cdot}27)$

$=2{\cdot}961\times64=189{\cdot}5$ B.Th.U./lb.

Heat supplied $H=H_c+(H_3-h_3)$ $H_3=1151{\cdot}2$, $h_3=181$

$=(189{\cdot}5+970{\cdot}2)$

$=1159{\cdot}7$ B.Th.U./lb.

$\therefore$ Absolute thermal efficiency

$$\eta=\frac{H_c}{H}=\frac{189{\cdot}5}{1159{\cdot}7}=0{\cdot}163$$

Jacket heat $h_J=(H_c+H_3)-H_1$ $H_1=1203{\cdot}1$

$=(189{\cdot}5+1151{\cdot}2)-1203{\cdot}1$

$=1340{\cdot}7=1203{\cdot}1=137{\cdot}6$ B.Th.U./lb.

Heat to condenser in exhaust

$$h_e=(H_3-h_3)=970{\cdot}2 \text{ B.Th.U./lb.},$$

which is simply the latent heat at $p_3$.

The cylinder volume per lb. is the specific volume of dry steam at $p_3$, or $v_3=26{\cdot}27$.

*Example* 9.—Determine the values of cycle heat, thermal efficiency, jacket heat, heat to condenser in exhaust, and cylinder volume per lb. of steam, for the jacketed engine of Example 8 when expansion is incomplete, and the release pressure is 40 lb./in.$^2$ abs.; and compare these with the values for complete expansion.

For purposes of comparison the values here are denoted by a dash.

Cycle heat

$H_c'=2{\cdot}961(p_1v_1-p_2v_2)+0{\cdot}185(p_2-p_3)v_2$ $p_1=200$, $v_1=2{\cdot}29$, $p_2=40$, $v_2=10{\cdot}5$, $p_3=15$

$=2{\cdot}961(200\times2{\cdot}29-40\times10{\cdot}5)+0{\cdot}185(40-15)10{\cdot}5$

$=2{\cdot}961\times38+0{\cdot}185\times262{\cdot}5$

$=(112{\cdot}48+48{\cdot}56)=161{\cdot}04$ B.Th.U./lb.

Heat supply $H'=2{\cdot}961(p_1v_1-p_2v_2)+(H_2-h_3)$ $H_2=1171{\cdot}7$, $h_3=181$

$=112{\cdot}48+990{\cdot}7=1103{\cdot}18$ B.Th.U./lb.

$\therefore$ absolute thermal efficiency

$$\eta'=\frac{H_c'}{H'}=\frac{161{\cdot}04}{1103{\cdot}18}=0{\cdot}146.$$

Jacket heat

$h_J'=2{\cdot}961(p_1v_1-p_2v_2)+H_2-H_1$ $H_1=1203{\cdot}1$

$=1284{\cdot}18-1203{\cdot}1=81{\cdot}08$ B.Th.U./lb.

Heat in exhaust to condenser

$$h_e'=(H_2-h_3)=(1171{\cdot}7-181)=990{\cdot}7 \text{ B.Th.U./lb.}$$

The cylinder volume per lb. of steam is the specific volume at release pressure $p_2$, or $v_2=10{\cdot}5$ ft.$^3$/lb.

Comparison of values—

$$\frac{H_c'}{H_c}=\frac{161{\cdot}04}{189{\cdot}7}=0{\cdot}85\text{; 15 per cent reduction.}$$

$$\frac{\eta'}{\eta}=\frac{0{\cdot}146}{0{\cdot}163}=0{\cdot}895\text{; 10·5 per cent. reduction.}$$

$$\frac{h_J'}{h_J}=\frac{81{\cdot}08}{127{\cdot}6}=0{\cdot}635\ ;\ 36{\cdot}5 \text{ per cent. reduction.}$$

$$\frac{h_e'}{h_e}=\frac{990{\cdot}7}{970{\cdot}2}=1{\cdot}021\ ;\ 2{\cdot}1 \text{ per cent. increase.}$$

$$\frac{v_2}{v_3}=\frac{10{\cdot}5}{26{\cdot}27}=0{\cdot}40\ ;\ 60 \text{ per cent. reduction.}$$

The student will note that in this case by the sacrifice of 15 per cent. of available energy, or 10·57 per cent. thermal efficiency, a substantial reduction of 60 per cent. can be effected in cylinder capacity. This is a very important result from the constructional and economic standpoint.

**63. Complete Regenerative Cycle.**—This cycle, by means of which an efficiency equal to the Carnot reversible cycle can be obtained, has already been outlined in Art. 19, p. 23. The regenerative effect there stated could be produced in the case of steam as working substance by providing an indefinitely large number of surface heaters, within the engine, in which the feed could be raised reversibly from the lower temperature limit to the upper limit of the saturated steam. By passing the feed contra-flow to the steam (one pound of steam in one direction giving up heat to one pound of water flowing in the opposite direction), and carrying the progressive condensation due to heat abstraction forward with the steam, the condition of equality of temperature at all points of the expansion would be obtained. One pound of feed would be heated reversibly from the lower temperature limit $T_2$ to the upper limit $T_1$ of the saturated steam. As in the Carnot cycle, the heat would thus be taken from the external source at $T_1$ and rejected to the cooler at $T_2$.

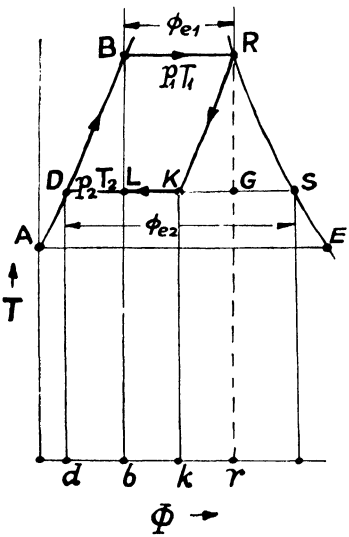


FIG. 39.

The TΦ diagram for this case is shown in Fig. 39. Under the operative conditions stated above the steam becomes progressively wetter during the expansion-regenerative process than it would during purely adiabatic expansion, and the expansion curve RK runs parallel to the water curve BD; BR=DK=LG; and $d$DB$b$=$k$KR$r$=regenerated heat.

The net heat received from the external source is given by the area $d$DBRK$k$ and the heat rejected, by the area $d$DK$k$.

Since all the heat is taken in from the source at $T_1$, $dDBRKk - bBRr$, the latent heat $L_1$. The heat rejected at $T_2$ is the latent heat at K.

Thus heat received $= H = T_1\phi_{e1}$

heat rejected $= h = T_2\phi_{e1}$

and the absolute thermal efficiency is

$$\eta = \frac{H-h}{H} = \frac{T_1-T_2}{T_1}, \text{ the Carnot efficiency.}$$

From the economic standpoint an engine of this nature is quite impracticable, and further discussion of it is unnecessary.

The nearest approach to the regenerative condition is the progressive heating of feed water by steam extracted from the engine or turbine at several stages of the expansion.

This may be termed the "partial regenerative cycle."

**64. Partial Regenerative Cycle.**—For the ideal case it is assumed that the expansion is adiabatic throughout, and that the feed

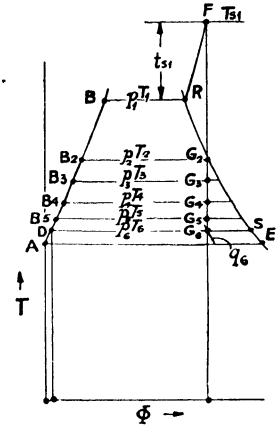



FIG. 40.

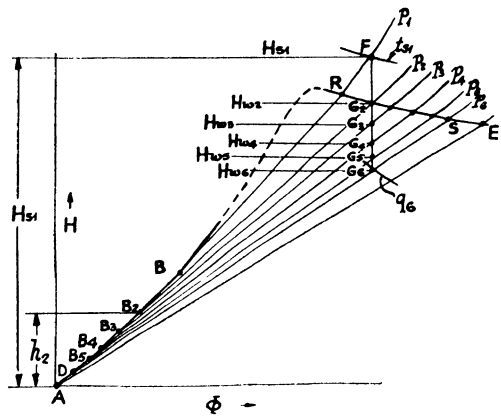



FIG. 41.

temperature at each heater is equal to the saturation temperature of the extracted steam which heats it.

The general case of the ideal cycle with initially superheated steam, for four feed heating stages, is shown on the TΦ diagram (Fig. 40) and the HΦ diagram (Fig. 41).

The first extraction is made at $p_2$, the steam for the case shown being just dry at that pressure, although this need not necessarily be the case. The extracted steam is supposed to be condensed to water at $T_2$ in the heater, and heats the feed (in contra-flow) to $T_2$, at which temperature it is then pumped to the boiler for re-evaporation and superheat at $p_1$ and $T_{s1}$. The next extraction takes place at some arbitrarily chosen lower pressure $p_3$. The extracted steam is condensed to water at $T_3$ and heats the feed to $T_3$. The process is

repeated at the successive heaters, where the extraction pressures are $p_4$ and $p_5$. The heater condensates can be disposed of in several ways. The following refer principally to steam-turbine practice.

Referring to Fig. 42, (*a*) the condensate from one heater may be cascaded to the heater below it, and together with the condensate from this heater may again be cascaded to the next lower heater, and so on until at the last heater (next the condenser) on the feed line the total heater condensate is collected. It is then cascaded to the condenser, where its water heat, reckoned from the condenser temperature, is carried away in the condensing water. With condenser cascade, as shown in Fig. 42, the corresponding extraction pressures are indicated at the heaters. The main condensate, which in this case includes the heater drains, is pumped by the condensate extraction pump to the hot well (not shown), and from thence is pumped to the boiler through the feed line by the high-pressure boiler feed pump.

(*b*) The drain from each heater may be taken by a drain pump and discharged into the feed line on the boiler side of the heater, as shown in Fig. 43. This arrangement has little appreciable effect on the efficiency if substituted for (*a*), and has a number of serious practical drawbacks.

(*c*) The drain from each heater may be passed through a cooler placed between the heater and the next one on the condenser side of it, as shown in Fig. 44. This system, more complicated than the cascade system (*a*), has only a slight effect on the thermal efficiency, as compared with (*a*).

(*d*) The drain from the heater next the condenser may be cascaded to the condenser, and the drains from the heaters on the boiler side of the feed pump may be cascaded to the heater on the condenser side of this pump, as shown in Fig. 45. The combined drain can then be drawn from the heater by a low-pressure drain pump, and discharged to the feed line on the low pressure side of the feed pump—that is, to the hot well or its modern equivalent.

This is regarded as the most desirable system for practical use, as it eliminates difficulties arising from drain pumps in operation and testing, and eliminates the disadvantages of small multi-stage high-pressure extraction pumps. In any case, however, arrangements should be made for direct cascade of the drain to the condenser if desired (case (*a*)).

In case (*a*) the water heat of all the heater condensate between $T_5$ and $T_6$ is rejected to the condenser; in case (*d*) only the water heat of the lowest pressure heater drain is rejected. By whatever method the heater drains are returned to the main feed line, the total feed entering the boiler, in a given time, is the same as the total steam supply to the H.P. end of the engine or turbine in the same time. The methods of calculation for the ideal cycle heat and absolute thermal efficiency for cases (*a*) and (*d*) only are considered here. The student can readily extend them to cover cases (*b*) and (*c*).

Referring to the TΦ and HΦ diagrams (Figs. 40 and 41), at the heater extraction pressures $p_2$, $p_3$, $p_4$, and $p_5$ the total heats per lb.

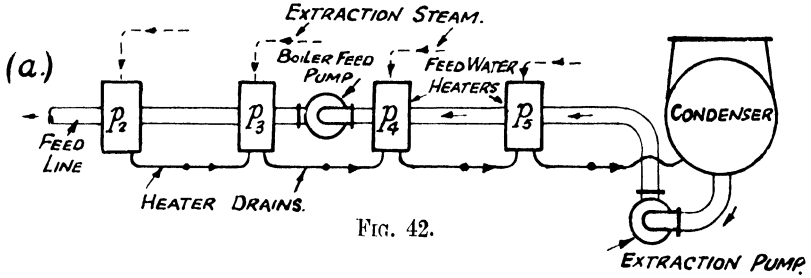



FIG. 42.

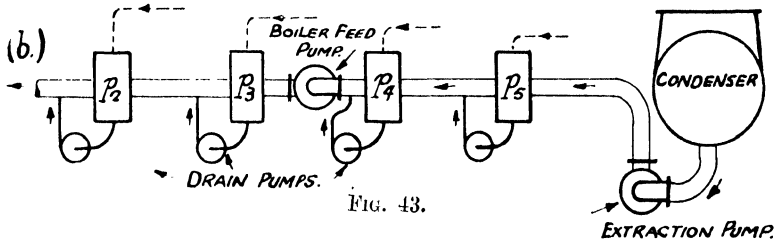



FIG. 43.

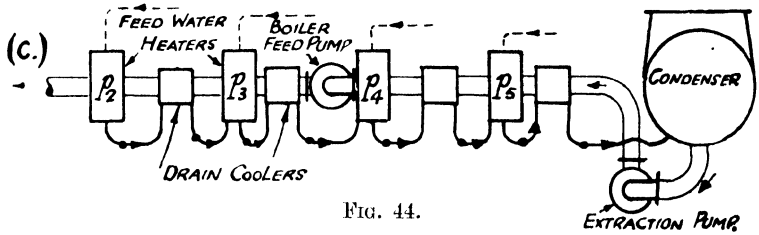



FIG. 44.

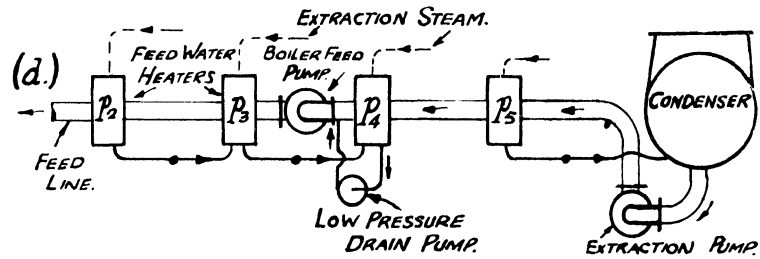



FIG. 45.

of steam, as defined by the state points $G_2$, $G_3$, $G_4$, and $G_5$ on the adiabatic line $FG_6$, are $H_{w2}$, $H_{w3}$, $H_{w4}$, and $H_{w5}$. The final total heat at the exhaust pressure $p_0 = p_6$, is $H_{w6}$, defined by point $G_6$.

Let W be the gross amount of steam entering the H.P. end and $H_{s1}$ the total heat at $p_1$ and $t_{s1}$, then the gross heat supply reckoned from the final feed temperature $T_2$ is $W(H_{s1}-h_2)$. Let $w_2$, $w_3$, $w_4$, and $w_5$ be the heater condensates at $p_2$, $p_3$, $p_4$, and $p_5$, then the net amount of exhaust steam discharged at the L.P. end is $W_o=(W-w_2-w_3-w_4-w_5)$. The latent heat of this steam at $p_6$ plus the water heat of the total drain is carried away in the circulating water Hence the gross amount of heat rejected is $W_o(H_{w6}-h_6)+(w_2+w_3+w_4+w_5)(h_5-h_6)$.

Let $z_2=\frac{w_2}{W}$, $z_3=\frac{w_3}{W}$, etc., be the fractional condensates or steam condensed per lb. of steam entering the H.P. end, then the heat per lb. of steam supply converted to work, or the cycle heat, is given by—

$$H_c=\frac{W(H_{s1}-h_2)-[W_o(H_{w6}-h_6)+(w_2+w_3+w_4+w_5)(h_5-h_6)]}{W}$$

or $$H_c=(H_{s1}-h_2)-[(1-z_2-z_3-z_4-z_5)(H_{w6}-h_6)+(z_2+z_3+z_4+z_5)(h_5-h_6)] \quad . \quad (43)$$

The absolute thermal efficiency is $\eta=\frac{H_c}{H}$, or—

$$\eta=1-\frac{[(1-z_2-z_3-z_4-z_5)(H_{w6}-h_6)+(z_2+z_3+z_4+z_5)(h_5-h_6)]}{(H_{s1}-h_2)} \quad (44)$$

The fractional condensates are obtained as follows:

Case (*a*).—At the final heater (next the H.P. end), the extraction pressure is $p_2$ and total heat $H_{w2}$. The W lb. of feed from the previous heater enters at temperature $T_3$ and is assumed to leave it at $T_2$.

Heat given by the steam to W lb. of feed is $W(h_2-h_3)$. The heater condensate is $w_2$ at $T_2$, and the heat given up by it is $(H_{w2}-h_2)w_2$, equating these—

$$(H_{w2}-h_2)w_2-W(h_2-h_3)$$

dividing by W—

$$(H_{w2}-h_2)z_2=(h_2-h_3),$$

or the ideal fractional condensate for the final heater is—

$$z_2=\frac{(h_2-h_3)}{(H_{w2}-h_2)} \quad . \quad . \quad . \quad . \quad . \quad . \quad (45)$$

In the next heater the pressure is $p_3$ and total heat of extracted steam $H_{w3}$; the final temperature of the condensate $w_3$ and the previous condensate $w_2$, cascaded to the heater, is $T_3$, while the feed is assumed to be raised from $T_4$ to $T_3$. The water heat of $w_2$ lb. entering this heater is $h_2$ and on leaving it $h_3$, so that in addition to the heat $w_3(H_{w3}-h_3)$ given up by $w_3$ lb. of steam at $p_3$, $w_2(h_2-h_3)$ is also given up to the feed, hence—

$$W(h_3-h_4)=w_3(H_{w3}-h_3)+w_2(h_2-h_3)$$

$$\text{and } z_3=\frac{(h_3-h_4)-z_2(h_2-h_3)}{(H_{w3}-h_3)} \quad . \quad . \quad . \quad (46)$$

By similar reasoning the other condensates are given by—

$$z_1=\frac{(h_4-h_5)-(z_2+z_3)(h_3-h_1)}{(H_{w4}-h_4)} \quad . \quad . \quad . \quad . \quad (47)$$

$$\text{and } z_5=\frac{(h_5-h_6)-(z_2+z_3+z_4)(h_4-h_5)}{(H_{w5}-h_5)} \quad . \quad . \quad . \quad (48)$$

Case (*d*).—The fractional condensates at $p_2$, $p_3$, and $p_4$ are the same as in (*a*), but that at $p_5$ is given by

$$z_5=\frac{h_5-h_6}{H_{w5}-h_5} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (49)$$

since the previous condensates do not enter this heater. The heat now rejected in the condenser is the latent heat of $W_o$ lb. plus the water heat of $w_5$ lb. between $T_5$ and $T_6$, returned to the condenser. Hence the heat rejected per lb. of steam supply is given by—

$$h=\frac{W_o(H_{w6}-h_6)+w_5(h_5-h_6)}{W}$$

$$\text{or } h=(1-z_2-z_3-z_4-z_5)(H_{w6}-h_6)+z_5(h_5-h_6) \quad . \quad (50)$$

The cycle heat for case (*d*) is now—

$$H_c=(H_{s1}-h_2)-[(1-z_2-z_3-z_4-z_5)(H_{w6}-h_6)+z_5(h_5-h_6)] \quad (51)$$

With a final heater temperature $T_2$ the heat supplied per lb. entering the H.P. end is, as before, $H=(H_{s1}-h_2)$, and the absolute thermal efficiency is

$$\eta=1-\frac{[(1-z_2-z_3-z_4-z_5)(H_{w6}-h_6)+z_5(h_5-h_6)]}{(H_{s1}-h_2)} \quad (52)$$

The student can readily modify the foregoing equations for a larger or smaller number of heaters than given here.

*Example* 10.—A steam turbine running on the partial regenerative cycle is supplied with steam at 600 lb./in.$^2$ abs. and 700° Fahr. (214° Fahr. superheat), and exhausts at 0·5 lb./in.$^2$ abs. There are four extraction feed heaters, and the extraction pressures are 110, 38, 10 and 2 lb./in.$^2$ abs. Calculate the ideal cycle heat, the absolute thermal efficiency, and the ideal rate of heat consumption, if the heater drains are progressively cascaded and discharged to the condenser.

The total heat values read on the adiabatic line $FG_6$ (Fig. 41) and the corresponding water heats are as given below—

| | | |
|---|---|---|
| $p_1=600$ | $H_{s1}=1355$ | $t_{s1}=214$ |
| $p_2=110$ | $H_{w2}=1187$ | $h_2=305{\cdot}6$ |
| $p_3=38$ | $H_{w3}=1107$ | $h_3=232{\cdot}8$ |
| $p_4=10$ | $H_{w4}=1016$ | $h_4=161{\cdot}1$ |
| $p_5=2$ | $H_{w5}=923$ | $h_5=93{\cdot}9$ |
| $p_6=0{\cdot}5$ | $H_{w6}=857$ | $h_6=47{\cdot}4$ |

Then $z_2=\dfrac{h_2-h_3}{H_{w2}-h_2}=\dfrac{305{\cdot}6-232{\cdot}8}{1187-305{\cdot}6}=\dfrac{72{\cdot}8}{881{\cdot}4}=0{\cdot}0826$

$$z_3=\frac{(h_3-h_4)-z_2(h_2-h_3)}{H_{w3}-h_3}=\frac{(232{\cdot}8-161{\cdot}1)-0{\cdot}0826\times72{\cdot}8}{1107-232{\cdot}8}$$

$$=\frac{65{\cdot}687}{874{\cdot}2}=0{\cdot}0751$$

$$z_4=\frac{(h_4-h_5)-(z_2+z_3)(h_3-h_4)}{H_{w4}-h_4}=\frac{(161{\cdot}1-939)-0{\cdot}1577\times71{\cdot}7}{1016-161{\cdot}1}$$

$$=\frac{55{\cdot}893}{854{\cdot}9}=0{\cdot}0654$$

$$z_5=\frac{(h_5-h_6)-(z_2+z_3+z_4)(h_4-h_5)}{H_{w5}-h_5}=\frac{(93{\cdot}9-47{\cdot}4)-0{\cdot}2231\times67{\cdot}2}{923-93{\cdot}9}$$

$$=\frac{31{\cdot}5}{829{\cdot}1}=0{\cdot}0380.$$

Hence cycle heat

$$H_c=(H_{s1}-h_2)$$
$$-[(1-z_2-z_3-z_4-z_5)(H_{w6}-h_6)+(z_2+z_3+z_4+z_5)(h_5-h_6)]$$
$$=(1355-305{\cdot}6)-[(1-0{\cdot}2611)(857-47{\cdot}4)+0{\cdot}2611(93{\cdot}9-47{\cdot}4)]$$
$$=1049{\cdot}4-[598{\cdot}2+12{\cdot}14]=(1049{\cdot}4-610{\cdot}34)=439 \text{ B.Th.U./lb.}$$

$H=(H_{s1}-h_2)=1049{\cdot}4.$

$\therefore$ absolute thermal efficiency—

$$\eta=\frac{H_c}{H}=\frac{439}{1049{\cdot}4}=0{\cdot}418.$$

Ideal rate of heat consumption (see Art. 67)—

$$Q_\phi=\frac{3412}{\eta}=\frac{3412}{0{\cdot}418}=8150 \text{ B.Th.U./kw. hr.}$$

*Example* 11.—Calculate for the turbine (Example 10) the cycle heat, absolute thermal efficiency, and rate of heat consumption when only the lowest heater drain is discharged to the condenser and the other drains are pumped back to the feed line (case (*d*)).

At the lowest pressure heater, the condensate becomes—

$$z_5=\frac{h_5-h_6}{H_{w5}-h_5}=\frac{93{\cdot}9-47{\cdot}4}{923-93{\cdot}9}=\frac{46{\cdot}5}{829{\cdot}1}=0{\cdot}056.$$

The other fractional condensates are unaltered, so that the discharge to the condenser per lb. of gross supply is—

$$(1-z_2-z_3-z_4-z_5)=(1-0{\cdot}0826-0{\cdot}0751-0{\cdot}0654-0{\cdot}056)$$
$$=(1-0{\cdot}2791)=0{\cdot}7209 \text{ lb.}$$

The cycle heat is—

$$H_c=(H_{s1}-h_2)-[(1-z_2-z_3-z_4-z_5)(H_{w6}-h_6)+z_5(h_5-h_6)]$$
$$=(1355-305{\cdot}6)-[0{\cdot}7209\times809{\cdot}6+0{\cdot}056\times45{\cdot}5]$$
$$=1049{\cdot}4-[583{\cdot}64+2{\cdot}6]=(1049{\cdot}4-586{\cdot}24)=463{\cdot}16 \text{ B.Th.T./lb.}$$

Absolute thermal efficiency—

$$\eta=\frac{463\cdot16}{1049\cdot4}=0\cdot442.$$

The theoretical improvement on (*a*) is 5·7 per cent.

$$\text{Also } Q_\phi=\frac{3412}{0\cdot442}=7720 \text{ B.Th.U./kw. hr.}$$

*Example* 12.—Compare the cycle heat and absolute thermal efficiency of the turbine (Examples 10 and 11) with the values given by a turbine working on the straight Rankine cycle between the same initial and exhaust pressure and temperature conditions.

For the straight Rankine cycle,

$$p_1=600,\ t_{s1}=214,\ H_{s1}=1355,\ p_2=0\cdot5,\ h_2=47\cdot4.$$

Heat supplied to the steam entering the turbine is

$$H=(H_{s1}-h_2)=(1355-47\cdot4)=1307\cdot6 \text{ B.Th.U./lb.}$$

The final total heat at $p_2=0\cdot5$ is $H_{w2}=857$.

Cycle heat $H_c=H_\phi=(H_{s1}-H_{w2})=(1355-857)=498$ B.Th.U./lb. against 439 in case (*a*) and 463·16 in case (*d*).

Absolute thermal efficiency

$$\eta=\frac{H_\phi}{H}=\frac{498}{1307\cdot6}=0\cdot38$$

as against 0·418 in case (*a*) and 0·442 in case (*d*), so that in (*d*) there is an ideal increase of about 16% in the absolute thermal efficiency from the use of the four heaters and drain arrangements.

The student should note, however, that these are not the values to be expected in an actual case where various losses occur and the temperature gradients of the heaters are not ideal. It will, however, be apparent that the addition of partial regenerative heating is decidedly advantageous. The increase of efficiency rises with the number of heaters employed.

A large number, however, leads to undesirable complication of plant. The number varies with the size of the plant. The following have been suggested.* For sets up to 6000 kw., one ; to 12500 kw., two ; to 25000 kw., three or four ; for larger plants four or five stages, on the assumption that the steam pressure of the large plant is not less than 300 lb./in.$^2$

**65. Reheat-regenerative Cycle.**—Finally there remains to be considered the combination cycle which is now adopted in large high-pressure power station plants. In this cycle the steam is subjected to one or more reheats in the high-pressure region, and partial regenerative feed heating is applied in the low-pressure region.

This cycle may be termed the " reheat-regenerative " cycle.

* " Tendencies in Steam Turbine Development," by H. L. Guy, *Proc. I. Mech. E.*, January 1929.

The theoretical case for one reheat, which is the usual condition of present practice, and three extraction feed heaters is considered here. The student should be able after a study of this and the previous methods to deal with the case of more than one reheat.

Obviously it will be advantageous to choose the reheat pressure as the extraction pressure for the H.P. heater, and this should be a pressure at which the steam will be approximately dry. If, as before, the total weight of steam entering the H.P. section at $p_1$ and $t_{s1}$ is W lb. and if this is withdrawn from the turbine for reheating, then by extracting the necessary steam $w_2$ for the H.P. heater at $p_2$, the steam to be passed back to the boiler is the smaller amount

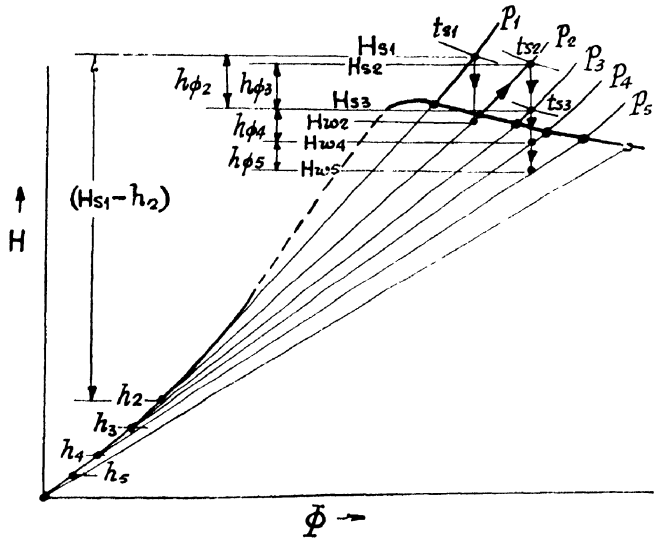



FIG. 46.

$(W-w_2)$. Its volume is smaller than that of W lb., and hence the piping and heater connections can be reduced in size.

It will be assumed, therefore, in this theoretical discussion that W lb. enters the H.P. end, $(W-w_2)$ is passed to the reheater, heated to the specified reheat temperature, and passed back to the turbine at $p_2$ and adiabatically expanded to the exhaust pressure. Actually the pressure after reheat, due to throttling and other conditions, would be less than $p_2$.

Starting the second phase of this cycle with $(W-w_2)$ lb. of steam initially at $p_2$ and $T_{s2}$, the calculations for the fractional condensations and heat received and rejected can be carried out in the manner already stated for the partial regenerative cycle, the procedure in this case, which is illustrated by the HΦ diagram (Fig. 46), is as follows.

The initial condition for W lb., at the H.P. end, is $p_1$, $t_{s1}$ and total heat $H_{s1}$.

Expansion adiabatically to $p_2$, the reheat-extraction pressure, gives, say, total heat $H_{w2}$ (if the steam were slightly superheated $H_s'_2$ would be used.)

The theoretical amount of steam now tapped off for the feed-heater at $p_2$ is—

$$w_2=\frac{(h_2-h_3)W}{(H_{w2}-h_2)} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (53)$$

where $h_3$ is the water heat at the next heater pressure $p_3$; and

$$z_2=\frac{h_2-h_3}{H_{w2}-h_2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (54)$$

$(W-w_2)$ lb. of steam is now reheated from $t_2$ to $t_2'$, the chosen reheat temperature, or from $T_2$ to $T_{s2}$ absolute temperature; the superheat is $t_{s2}$ and the total heat is $H_{s2}$.

The heat received by W lb., reckoned from the feed temperature $T_2$, is $W(H_{s1}-h_2)$, and the amount given during reheat is $(W-w_2)(H_{s2}-H_{w2})$.

Hence the gross amount of heat given by the boiler is

$$W(H_{s1}-h_2)+(W-w_2)(H_{s2}-H_{w2}),$$

and the heat received per lb. of gross feed entering the boiler is

$$H-(H_{s1}-h_2)+(1-z_2)(H_{s2}-H_{w2}) \quad . \quad . \quad . \quad (55)$$

Here again the particular method adopted for the disposal of the heater drains determines the estimation of the heat rejected in the condenser.

Assume, as in case (*a*), progressive cascading through the heaters to the condenser.

After readmission to the turbine at $p_2$ and $T_{s2}$ the steam is assumed to expand adiabatically to the next extraction pressure $p_3$, and it may be slightly superheated or wet according to the particular conditions chosen. For generality, suppose it is superheated $t_{s3}$ and the total heat is $H_{s3}$.

Then the condensate in the heater where pressure is $p_3$ is—

$$w_3=\frac{(h_3-h_4)W-w_2(h_2-h_3)}{(H_{s3}-h_3)}$$

and

$$z_3=\frac{(h_3-h_4)-z_2(h_2-h_3)}{(H_{s3}-h_3)} \quad . \quad . \quad . \quad . \quad . \quad . \quad (56)$$

Similarly

$$z_4=\frac{(h_4-h_5)-(z_2+z_3)(h_3-h_4)}{(H_{w4}-h_4)} \quad . \quad . \quad . \quad . \quad (57)$$

As in the case of the partial-regenerative cycle, the net weight of steam discharged to the condenser at $p_5$ is $W_o=(W-w_2-w_3-w_4)$; the corresponding amount of latent heat given to the circulating water is $W_o(H_{w5}-h_5)$. Again, with the final cascade of

$(w_2+w_3+w_4)$ lb. of heater drain to the condenser the additional water heat $(w_2+w_3+w_4)(h_4-h_5)$ is given up. Hence the heat rejected to the condenser is

$$(W-w_2-w_3-w_4)(H_{w5}-h_5)+(w_2+w_3+w_4)(h_4-h_5).$$

The heat rejected per lb. of gross feed to the boiler or per lb. of steam entering the H.P. section of the turbine is

$$\begin{aligned} h&=(1-z_2-z_3-z_4)(H_{w5}+h_5)+(z_2+z_3+z_4)(h_4-h_5)\\ &=(1-\Sigma z)(H_{w5}-h_5)+\Sigma z(h_4-h_5)\end{aligned}$$

The cycle heat is then given by

$$\begin{aligned} H_c&=(H-h)\\ &=[(H_{s1}-h_2)+(1-z_2)(H_s'_2-H_{w2})]\\ &\quad -[(1-z_2-z_3-z_4)(H_{w5}-h_5)+(z_2+z_3+z_4)(h_4-h_5)] \quad . \quad (58)\end{aligned}$$

And, as before, the absolute thermal efficiency is given by

$$\eta=\frac{H_c}{H}.$$

If instead of continuous cascade to the condenser the conditions of (*d*), Art. 64, are taken, then the second term in the square bracket on the right becomes $z_4(h_4-h_5)$.

If the condition as in (*e*) is taken, that is, if all the heater drains are cascaded to the L.P. heater and pumped back to the feed line, the second term in the square bracket vanishes.

*Alternative Calculation for the Cycle Heat.*—In the cases of the partial regenerative and reheat-regenerative cycles, the cycle heat has been determined from the difference of the heat supplied and the heat rejected per lb. of steam entering the H.P. end.

It can also be obtained by the summation of the effective partial heat drops between the successive extraction pressures. Denoting these partial heat drops, as given by the HΦ diagram, by $h_\phi$ and the subscripts corresponding to the extraction pressure at the end of the partial expansion, then

$$H_c=h_{\phi 2}+(1-z_2)h_{\phi 3}+(1-z_2-z_3)h_{\phi 4}+(1-z_2-z_3-z_4)h_{\phi 5} \quad . \quad (59)$$

where $h_{\phi 2}$ is the adiabatic heat drop per lb. between $p_1$ and $p_2$, $h_{\phi 3}$ the heat drop between $p_2$ and $p_3$, and so on.

*Example* 13.—Calculate the cycle heat and absolute thermal efficiency of a turbine installation supplied with steam at 600 lb./in.$^2$ and 700° F. which exhausts at 0·5 lb./in.$^2$ abs., when the steam is reheated at 100 lb./in.$^2$ abs. to 600° Fahr. and there are three regenerative feed-heaters, the extraction pressures being 100, 20, and 4 lb./in.$^2$ abs. Assume that the drains are progressively cascaded through the heaters and finally to the condenser.

In this case $p_1=600$, $t_{s1}=214°$ Fahr., and $H_{s1}=1350$.

$p_2=100$. The adiabatic vertical in the HΦ diagram (Fig. 46) cuts $p_2=100$ at $H_{w2}=1179$ ; also $h_2=298·3$.

Pressure at second heater $p_3=20$ and $h_3=196{\cdot}1$. Hence the fractional condensate at the H.P. heater is

$$z_2=\frac{h_2-h_3}{H_{w2}-h_2}=\frac{298{\cdot}3-196{\cdot}1}{1179-298{\cdot}3}=\frac{102{\cdot}2}{880{\cdot}7}=0{\cdot}1160$$

After reheat at $p_2=100$ to 600° Fahr. or 272° Fahr. superheat, $H_{s2}=1333$. Hence the reheat is

$$(H_{s2}-H_{w2})(1-z_2)=(1-0{\cdot}116)(1333-1179)=0{\cdot}884\times154=136{\cdot}14.$$

Heat supplied per lb. of gross feed is

$$\begin{aligned}H&=(H_{s1}-h_2)+(1-z_2)(H_{s2}-H_{w2})\\&=(1350-298{\cdot}3)+136{\cdot}14=1187{\cdot}84\text{ B.Th.U./lb.}\end{aligned}$$

The total heats at the extraction pressures and at exhaust, from the HΦ chart, with the corresponding water heats, are as given below.

| | | |
|---|---|---|
| $p_3=20$ | $H_{s3}=1180$ | $h_3=196{\cdot}1$ |
| $p_4=4$ | $H_{w4}=1067$ | $h_4=120{\cdot}8$ |
| $p_5=0{\cdot}5$ | $H_{w5}=949$ | $h_5=47{\cdot}4$ |

then
$$z_3=\frac{(h_3-h_4)-z_2(h_2-h_3)}{H_{s3}-h_3}$$
$$=\frac{(196{\cdot}1-120{\cdot}8)-0{\cdot}116(298{\cdot}3-196{\cdot}1)}{(1180-196{\cdot}1)}$$
$$=\frac{75{\cdot}3-11{\cdot}86}{983{\cdot}9}=\frac{63{\cdot}44}{983{\cdot}9}=0{\cdot}0645$$
$$z_4=\frac{(h_4-h_5)-(z_2+z_3)(h_3-h_4)}{H_{w4}-h_4}$$
$$=\frac{(120{\cdot}8-47{\cdot}4)-0{\cdot}1805(196{\cdot}1-120{\cdot}8)}{(1067-120{\cdot}8)}$$
$$=\frac{73{\cdot}4-13{\cdot}6}{946{\cdot}2}=\frac{57{\cdot}8}{946{\cdot}2}=0{\cdot}0632.$$

Heat rejected to the condenser—

$$\begin{aligned}h&=(1-z_2-z_3-z_4)(H_{w5}-h_5)+(z_2+z_3+z_4)(h_4-h_5)\\&=(1-0{\cdot}2437)(949-47{\cdot}4)+0{\cdot}2437(120{\cdot}8-47{\cdot}4)\\&=0{\cdot}7563\times901{\cdot}6+0{\cdot}2437\times73{\cdot}4\\&=681{\cdot}9+17{\cdot}9=699{\cdot}8\text{ B.Th.U./lb.}\end{aligned}$$

Cycle heat $H_c=(H-h)$
$$=(1187{\cdot}84-699{\cdot}8)=488\text{ B.Th.U./lb.}$$

*Alternative calculation for* $H_c$.—Adiabatic heat drop between $p_1$ at $t_{s1}$ and $p_2$ (from HΦ chart values)—

$$h_{\phi 2}=(1350-1179)=171$$

Heat drop between $p_2$ at $t_{s2}$ and $p_3$,

$$h_{\phi 3}=(1333-1180)=153\ ;\quad(1-z_2)=0{\cdot}884$$

Heat drop between $p_3$ and $p_4$,

$$h_{\phi 4}=(1180-1067)=113\,; \quad (1-z_2-z_3)=0{\cdot}819$$

Heat drop between $p_4$ and $p_5$,

$$h_{\phi 5}=(1067-949)=118\,; \quad (1-z_2-z_3-z_4)=0{\cdot}756$$

$$\therefore H_c=h_{\phi 2}+(1-z_2)h_{\phi 3}+(1-z_2-z_3)h_{\phi 4}+(1-z_2-z_3-z_4)h_{\phi 5}$$

$$=171+0{\cdot}884\times 153+0{\cdot}819\times 113+0{\cdot}756\times 118$$

$$=171+\quad 135\quad +\quad 93\quad +\quad 89\quad =488 \text{ B.Th.U.}$$

which checks with the figure by the first method.

Absolute thermal efficiency is

$$\eta=\frac{H_c}{H}=\frac{488}{1187{\cdot}84}=0{\cdot}41$$

**66. Indicated and Brake Thermal Efficiencies.**—In an actual engine arranged to work on any of the foregoing cycles the proportion of heat supplied per lb. of steam entering the working chamber, which is actually converted into work, is always less than the ideal cycle heat. Similarly the actual thermal efficiency is less than the absolute value.

In the reciprocator the gross amount of heat expended in doing useful work on the mobile part, the piston, can be determined from an autographic record of the pressure (driving force) and volume changes during the cycle, as obtained by some form of indicator. This is the indicated cycle heat per lb. and may be denoted by $H_i$.

In the case of a steam turbine this gross expenditure on the mobile part, the blading of the rotor, has to be estimated indirectly, since the energy conversion is of a dynamical nature, and no graphical record of the driving forces can be obtained. This gross value per lb. of steam, as calculated, may also be denoted by $H_i$; but the term "indicated" heat in this instance is a misnomer and should not be used.

The net work done at the driving end of either engine or turbine, or the effective amount of heat per lb., is the brake cycle heat, and may be denoted by $H_B$. It can always be determined with considerable accuracy from dynamometric or electrical measurements of the net output on test. The ratio of the net to the gross expenditure is the mechanical efficiency.

As regards the actual thermal efficiency it will be evident that this has two values, according to whether the gross or net expenditure is compared with the heat supplied. The indicated thermal efficiency may be denoted by $\eta_I$ and the net or brake thermal efficiency by $\eta_B$.

The figure of commercial importance is the brake thermal efficiency, as it covers all mechanical and thermal losses of the engine.

Dealing with the reciprocator,

Let I=total I.H.P. and B=total B.H.P.

W=total steam supply in lb./hr.

Then the indicated rate of steam consumption is—

$$w_i = \frac{W}{I} \text{ lb./I.H.P. hr.}$$

and the brake rate of consumption is

$$w_B = \frac{W}{B} \text{ lb./B.H.P. hr.}$$

Also $1 \times \text{H.P.} \times \text{hr.} = 60 \times 42{\cdot}4 = 2545$ B.Th.U.

Let H = heat supply in B.Th.U./lb. Then—

$$\text{Indicated thermal efficiency } \eta_I = \frac{2545I}{WH} = \frac{2545}{w_i H} \quad . \quad . \quad . \quad . \quad . \quad (60)$$

$$\text{Brake thermal efficiency } \eta_B = \frac{2545B}{WH} = \frac{2545}{w_B H} \quad . \quad . \quad . \quad . \quad . \quad . \quad (61)$$

As the output is frequently expressed in kilowatts and 1 kw. = 1·34 B.H.P., the brake thermal efficiency can also be expressed by—

$$\eta_B = \frac{2545 \times 1{\cdot}34K}{WH} = \frac{3412}{w_K H} \quad . \quad . \quad . \quad . \quad . \quad (62)$$

where K = kilowatt output and $w_K = \frac{W}{K}$ the brake consumption rate in lb./kw. hr.

**67. Rate of Heat Consumption.**—Before the advent of superheated steam, it was the general practice to express the performance of an engine or turbine in terms of the rate of steam consumption, $w_i$, $w_B$, or $w_K$. Such a proceeding is no longer valid, when reliable comparison has to be made between the performance of engines or turbines working under varying steam conditions. The real criterion is not the rate of steam consumption but the rate of heat consumption expressed in thermal units per horse-power hour or minute, or thermal units per kilowatt hour or minute.

This is the value of the denominator in the efficiency equations, and may be denoted by the general symbol Q, with corresponding subscripts. When the horse-power hour rate is used—

$$Q_I = \frac{2545}{\eta_I} \text{ B.Th.U./I.H.P. hr.} \quad . \quad . \quad . \quad . \quad (63)$$

$$Q_B = \frac{2545}{\eta_B} \text{ B.Th.U./B.H.P. hr.} \quad . \quad . \quad . \quad (64)$$

$$Q_K = \frac{3412}{\eta_B} \text{ B.Th.U./kw. hr.} \quad . \quad . \quad . \quad . \quad (65)$$

When the horse-power minute rate is used—

$$Q_I = \frac{42{\cdot}4}{\eta_I} \text{ B.Th.U./I.H.P. min.} \quad . \quad . \quad . \quad (66)$$

$$Q_B = \frac{42{\cdot}4}{\eta_B} \text{ B.Th.U./B.H.P. min.} \quad . \quad . \quad . \quad (67)$$

$$Q_K = \frac{56{\cdot}816}{\eta_B} \text{ B.Th.U./kw. min.} \quad . \quad . \quad . \quad (68)$$

**68. Efficiency Ratios.**—As the term indicates, this is the ratio of the actual thermal efficiency of the engine or turbine to the absolute thermal efficiency of the particular cycle on which the machine is supposed to work. Here, again, it may be expressed with reference either to the indicated or brake efficiencies. It may be denoted by $\epsilon$. Thus—

Indicated efficiency ratio $$\epsilon_I = \frac{\eta_I}{\eta} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (69)$$

Brake efficiency ratio $$\epsilon_B = \frac{\eta_B}{\eta} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (70)$$

In the case of a steam turbine the second is called the " turbine " efficiency ratio. The equivalent of the first is called the " internal " efficiency ratio $\epsilon_1$, as it excludes the external frictional losses of the machine. It is discussed in Chapter XXII.

This factor, the efficiency ratio, or, as it is also termed, the " thermodynamic efficiency " of the engine or turbine, is of first importance in facilitating the solution of various thermodynamic calculations which arise in the discussion of the probable operating conditions obtaining throughout an actual engine or turbine cycle. Where the process of compounding or stage expansion is used, it can be applied to the successive stages, if the whole engine or turbine is taken as equivalent to a number of Rankine engines working in series. In these cases the efficiency ratio for any stage is called the stage efficiency ratio, and may be denoted by $\epsilon_s$.

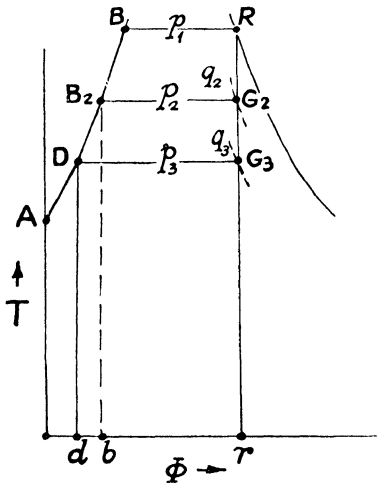



FIG. 47.

**69.** Consider the cycle of a Rankine engine working between the initial and exhaust pressure limits $p_1$ and $p_3$, as shown by $BRG_3D$ on the T$\Phi$ diagram (Fig. 47).

Suppose this area is divided into two parts at some arbitrarily chosen pressure level $p_2$. Then the top area $BRG_2B_2$ may be taken to represent the cycle heat, or heat drop $H_\phi$, of an engine in which the steam is initially dry at beginning of adiabatic expansion, working between the limits of $p_1$ and $p_2$, and obtained from an input of heat given by the area $bB_2BRr$ or $(H_1 - h_2)$. Such an engine would have an absolute thermal efficiency—

$$\eta_1 = \frac{H_{\phi 2}}{(H_1 - h_2)} \quad . \quad . \quad . \quad . \quad . \quad . \quad (71)$$

The bottom area $B_2G_2G_3D$ may similarly be taken as the cycle heat $H_{\phi 3}$ of a second Rankine engine working between $p_2$ and $p_3$ with steam of $q_2$ dryness at beginning of adiabatic expansion and obtained from a heat input $d DB_2G_2r$ or $(H_{w2}-h_3)$ and having an absolute thermal efficiency of—

$$\eta_2=\frac{H_{\phi 3}}{(H_{w2}-h_3)} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (72)$$

The complete diagram can thus be taken to represent two engines in series, the exhaust heat conditions of the first being the same as the initial heat conditions of the second.

As a single engine, working between $p_1$ and $p_3$, $BRG_3D$ is the cycle heat $H_\phi$, from a heat input $d DBRr$ and $\eta=\frac{H_\phi}{H_1-h_3}$.

The condition is

$$\eta(H_1-h_3)=\eta_1(H_1-h_2)+\eta_2(H_{w2}-h_3)$$

or

$$H_\phi=h_{\phi 2}+h_{\phi 3} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (73)$$

This procedure can obviously be extended to any number of subdivisions of the total area—that is, to any number of engines in series or stages of expansion, as in the case of a compound, triple, or quadruple engine, or a multistage steam turbine.

Treating each of a series of such stages as a single engine working between the stage pressure and temperature limits, the partial or stage efficiency ratio ($\epsilon_s$) value can be calculated when the actual cycle heat for the given stage is determined, from experimental results.

Thus if the indicated cycle heat of the first stage of a compound engine with initial and exhaust pressure limits $p_1$ and $p_2$ is $H_{i1}$, the stage thermal efficiency is—

$$\eta_{s1}=\frac{H_{i1}}{H_1-h_2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (74)$$

and the stage efficiency ratio is $\epsilon_{s1}=\frac{\eta_{s1}}{\eta_1}$, where $\eta_1$ is the absolute thermal efficiency of the stage.

Since $\eta_{s1}=\frac{h_{\phi 2}}{H_1-h_2}$, it follows that the stage efficiency ratio is—

$$\epsilon_{s1}=\frac{H_{i1}}{h_{\phi 2}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (75)$$

In other words, the stage efficiency ratio, or thermodynamic efficiency, is also the ratio of the actual or indicated stage cycle heat (heat usefully converted to work on the mobile part per lb.) and the adiabatic heat drop, between the stage pressure limits, defined by the lower pressure subscript.

Similarly for the second stage between the limits $p_2$ and $p_3$, the efficiency ratio is given by

$$\epsilon_{s2}=\frac{H_{i2}}{h_{\phi 3}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (76)$$

The method can be extended to any number of stages, whatever the initial and final conditions of the steam may be, at each stage of the expansion.

**70.** In determining the value of the indicated heat per lb. ($H_i$) for any given stage of expansion, care must be taken that the partial or stage indicated rate of consumption $w_i$ is calculated from the I.H.P. of the stage and the *actual* weight of steam passing through the particular stage.

When extraction feed-heating is adopted the amount of steam extracted before this stage is reached must be subtracted from the gross supply entering the H.P. end of the engine or turbine. This, in general notation, will be given by $W(1-\Sigma z)$, where $\Sigma z$ represents the sum of the *actual* fractional condensations up to the stage in question, either determined from test results or deduced from the other test data.

The actual $z$ values are not the same as the ideal values calculated by the methods given in Art. 64. The general principle of this calculation is the same for the actual case, but it involves temperature and heat values different from the ideal figures. The modified calculation for the actual extraction quantities is dealt with in Chapter VII, Art. 151.

When in the case of a multiple expansion engine sufficient test data are available, it is possible to show the steam conditions at each stage of the expansion on the HΦ diagram.

This important method of representation and the general principle on which it is based are discussed in detail for a single stage engine in Chapter VI.

For the multiple expansion case it can be applied to each stage in succession, when the stage efficiency ratios, determined as above, and also radiation losses, are estimated.

In general, the latter, which are relatively small, are neglected.

The general method of procedure is illustrated for a triple stage engine in Fig. 48, when radiation effect is neglected.

The initial state point $F_1$, at $p_1$ and $t_{s1}$, for the first stage expansion is located, and the adiabatic vertical is drawn between $p_1$ and $p_2$. On this vertical $F_1f_2=H_{i1}=\epsilon_{s1}h_{\phi 2}$ is set off, and a horizontal projector is drawn, to cut the $p_2$ curve at $F_2$, which is the initial state point for the next stage of the expansion. This process is repeated for the second stage, taking $F_2f_3=H_{i2}=\epsilon_{s2}h_{\phi 3}$. This gives the initial state point $F_3$ for the third stage. Again repeating the process with $F_3f_4=H_{i3}=\epsilon_{s3}h_{\phi 4}$, the final state point $G_4$, at the exhaust pressure, is obtained.

This gives the actual final quality $q_4$ at $p_4$.

If the steam were to expand adiabatically in a single engine (represented by the cylinder of the third stage), the final state point would be $G_4'$ and the adiabatic heat drop $F_1G_4'=H_\phi$. For the straight cycle—that is, where no reheat or regenerative heating occurs—the actual or net heat drop is the summation—

$$\epsilon_{s1}h_{\phi 2}+\epsilon_{s2}h_{\phi 3}+\epsilon_{s3}h_{\phi 4},$$

which is simply the vertical distance between the initial state point $F_1$ and the actual final state point $G_4$, or $\epsilon_I H_\phi$, where $\epsilon_1$ is the efficiency ratio for a single stage expansion between $p_1$ and $p_3$.

In this case the efficiency ratio is given by

$$F_1f_5/F_1G_4' \quad \text{or} \quad \frac{\Sigma H_i}{H_\phi}=\frac{\eta_I}{\eta}=\epsilon_1$$

When there is reheating or regenerative feed heating, or a combination of these, the efficiency ratio $\epsilon_I$ of the whole machine

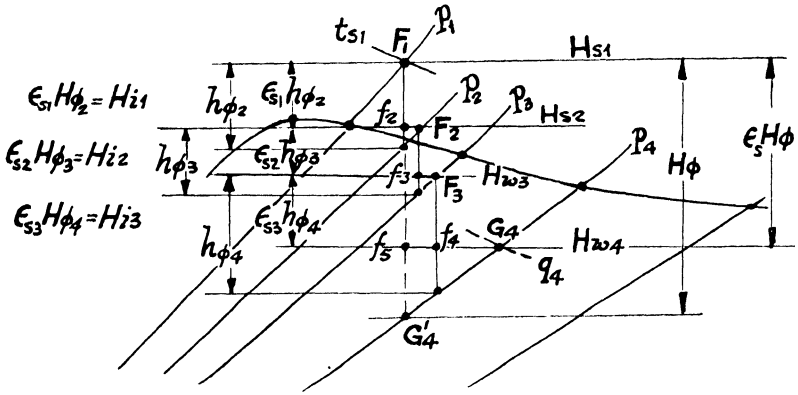



Fig. 48.

must be calculated from the ratio of the actual indicated thermal efficiency $\eta_I$ and the absolute thermal efficiency $\eta$, determined by the methods just discussed.

The heat drop ratio must not be taken.

*Example* 14.—A triple expansion marine engine is supplied with steam at 225 lb./in.$^2$ abs. and 162° Fahr. superheat, and exhausts at 2·2 lb./in.$^2$ abs.

The initial pressures at the second and third stages are 75 lb./in.$^2$ abs. and 23 lb./in.$^2$ abs. The indicated horse-powers developed are H.P. 985, I.P. 823, L.P. 797, and the steam consumption is 28,900 lb./hr. There are no reheaters or regenerative feed heaters.

Calculate the stage efficiency ratios, the efficiency ratio of the engine, referred to the straight Rankine cycle, and the rate of heat consumption.

As the same weight of steam W=28,900 lb./hr. passes through each stage, the indicated cycle heats are—

$$\text{H.P., } H_{i1}=\frac{2545I_1}{W}=\frac{2545\times985}{28900}=86{\cdot}7 \text{ B.Th.U./lb.} \qquad \begin{matrix} I_1=985 \\ W=28{,}900 \end{matrix}$$

$$\text{I.P., } H_{i2}=\frac{2545I_2}{W}=\frac{2545\times823}{28900}=72 \text{ ,,} \qquad I_2=823$$

$$\text{L.P., } H_{i3}=\frac{2545I_3}{W}=\frac{2545\times797}{28900}=70 \text{ ,,} \qquad I_3=797$$

Referring to the HΦ diagram (Fig. 48), the total heats and heat drops obtained are as follows:

$H_{s1}=1302$. $h_{\phi2}=103$. $H_{s2}=(H_{s1}-H_{i1})=(1302-86{\cdot}7)=1215{\cdot}3$

$$\epsilon_{s1}=\frac{H_{i1}}{h_{\phi2}}=\frac{86{\cdot}7}{103}=0{\cdot}84$$

$h_{\phi3}=92$. $H_{w3}=(H_{s2}-H_{i2})=(1215{\cdot}3-72)=1143{\cdot}3$

$$\epsilon_{s2}=\frac{H_{i2}}{h_{\phi3}}=\frac{72}{92}=0{\cdot}783$$

$h_{\phi4}=152$. $H_{w4}=(H_{w3}-H_{i3})=(1143{\cdot}3-70)=1073{\cdot}3$; $q_4=0{\cdot}96$

$$\epsilon_{s3}=\frac{H_{i3}}{h_{\phi4}}=\frac{70}{152}=0{\cdot}46$$

The total indicated heat drop $\Sigma H_i=228{\cdot}7$ B.Th.U./lb.

Adiabatic heat drop between $p_1=225$, $t_{s1}=162°$ Fahr., and $p_4=2{\cdot}2$ is $H_\phi=341$ B.Th.U./lb. Hence the efficiency ratio of the engine is

$$\epsilon_I=\frac{\Sigma H_i}{H_\phi}=\frac{228{\cdot}7}{341}=0{\cdot}67$$

Shorter calculation from the indicated and absolute thermal efficiencies—

$$\eta_I=\frac{2545I}{W(H_{s1}-h_4)} \qquad \begin{matrix} I=2605 \\ H_{s1}=1302 \\ h_4=97{\cdot}4 \\ W=28{,}900 \end{matrix}$$

$$=\frac{2545\times2605}{28900(1302-97{\cdot}4)}$$

$$=\frac{2545\times2605}{28900\times1204{\cdot}6}=0{\cdot}19$$

Absolute thermal efficiency—

$$\eta=\frac{H_\phi}{(H_{s1}-h_4)}=\frac{341}{1204{\cdot}6}=0{\cdot}2835$$

$$\therefore\ \epsilon_I=\frac{\eta_I}{\eta}=\frac{0{\cdot}19}{0{\cdot}2835}=0{\cdot}67$$

Rate of heat consumption—

$$Q_i = \frac{2545}{\eta_i} = \frac{2545}{0{\cdot}19} = 13{,}395 \text{ B.Th.U./I.H.P. hr.}$$

$$\text{or } Q_i = \frac{13395}{60} = 223{\cdot}2 \text{ B.Th.U./I.H.P. min.}$$

**71. The Binary Vapour Cycle.**—In the discussion of the various practicable steam engine and turbine cycles in Arts. 57 to 65 it is assumed that the energy transformations are effected by means of a single working substance, which is steam.

Until the advent of the steam turbine, the most economical use of the heat energy of the steam could not be obtained in the reciprocator on account of the large volumes at low pressures. In the highest class of modern steam turbine practice the lower limit is now about 29·1 in. vacuum or 0·442 lb./in.$^2$ abs.

In order to utilise that portion of the energy in the exhaust steam of the reciprocator which could be made available by further expansion, a system whereby the latent heat of this steam could be used to vaporise a more volatile fluid, which in turn could be used as the working substance of a second engine, was adopted. The combination was termed a " binary vapour " engine. The engine using the secondary substance was termed a " waste heat " engine.

The binary vapour combination gives a greater output than the steam engine alone. The increase may amount to 30 or 40 per cent., for the same input of heat to the primary substance, the steam, or, in other words, for the same fuel consumption. There is also the corresponding reduction in the rate of heat consumption.

The secondary substance used with a condensing steam engine must have a low boiling-point, 100° Fahr. to 140° Fahr., with corresponding pressure sufficiently low for constructional purposes.

A number of these engines using steam and $SO_2$ vapour were put into service in Germany. The advent of the exhaust steam turbine, however, rendered the use of this type unnecessary, as the turbine remedies the defect which the binary vapour combination was designed to overcome, much more effectively and with less mechanical complication. As far as the reciprocator is concerned the type is now obsolete, but the principle is still in the forefront of modern turbine practice. In this case, mercury is used as the primary and steam as the secondary substance. This modern adaptation of the binary vapour principle is discussed in Chapter XXII.

$SO_2$ has been found the most suitable substance for use in the auxiliary waste heat engine.

With a vacuum of 24 in. or about 3 lb./in.$^2$ abs., a common figure for the earlier reciprocators, the exhaust temperature is 141° Fahr. This is nearly the saturation temperature of $SO_2$ at 156 lb./in.$^2$ abs. If the $SO_2$ vapour, obtained by the condensation of the steam, is

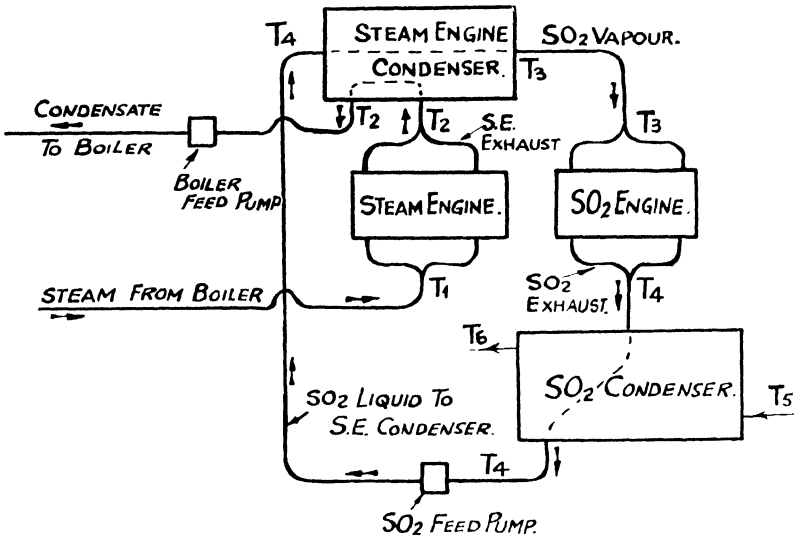

T4
STEAM ENGINE
CONDENSER.
SO2 VAPOUR.
T3
CONDENSATE
TO BOILER
T2
T2
S.E.
EXHAUST
T3
BOILER
FEED PUMP
STEAM ENGINE.
SO2 ENGINE.
STEAM FROM BOILER
T1
SO2
EXHAUST
T4
T6
SO2 CONDENSER.
T5
SO2 LIQUID TO
S.E. CONDENSER.
T4
SO2 FEED PUMP.

FIG. 49.

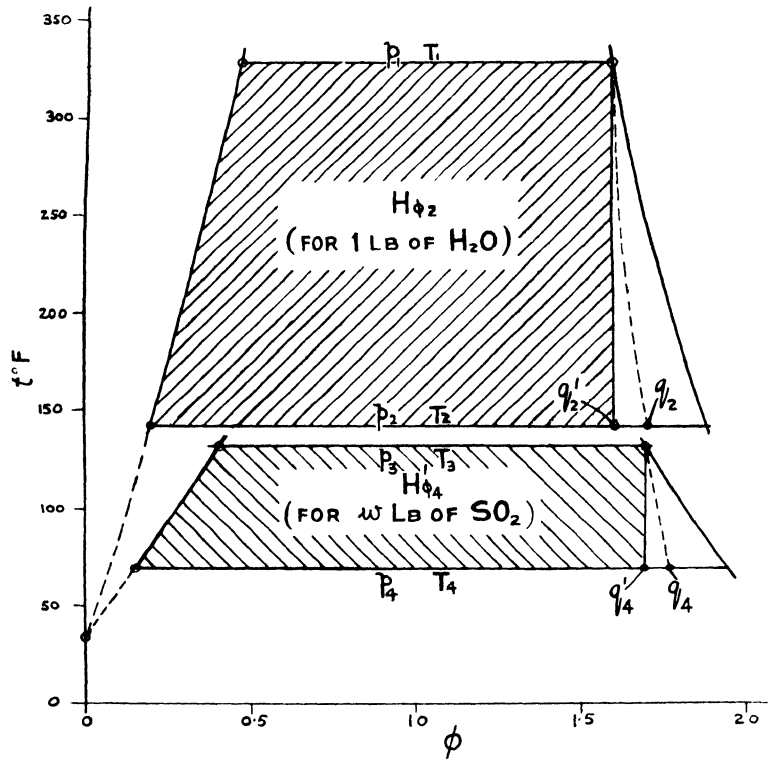

350
300
250
200
150
100
50
0
t°F
p1 T1
Hφ2
(FOR 1 LB OF H2O)
q'2
q2
p2 T2
p3 T3
Hφ4
(FOR w LB OF SO2)
p4 T4
q'4
q4
0
0·5
1·0
1·5
2·0
φ

FIG. 50.

expanded in the $SO_2$ engine cylinder and condensed at an ordinary circulating water temperature, say 70° Fahr., the vapour pressure at that temperature would be 49·6 lb./in.² abs.

The upper pressure limit of the $SO_2$ engine, on account of the temperature drop across the tubes of the steam engine condenser, is necessarily less than that corresponding to the exhaust temperature of the steam. The temperature drop may be taken as, say, 10° Fahr., so that dry saturated $SO_2$ vapour produced would have a saturation temperature 10° Fahr. lower than the steam temperature. On

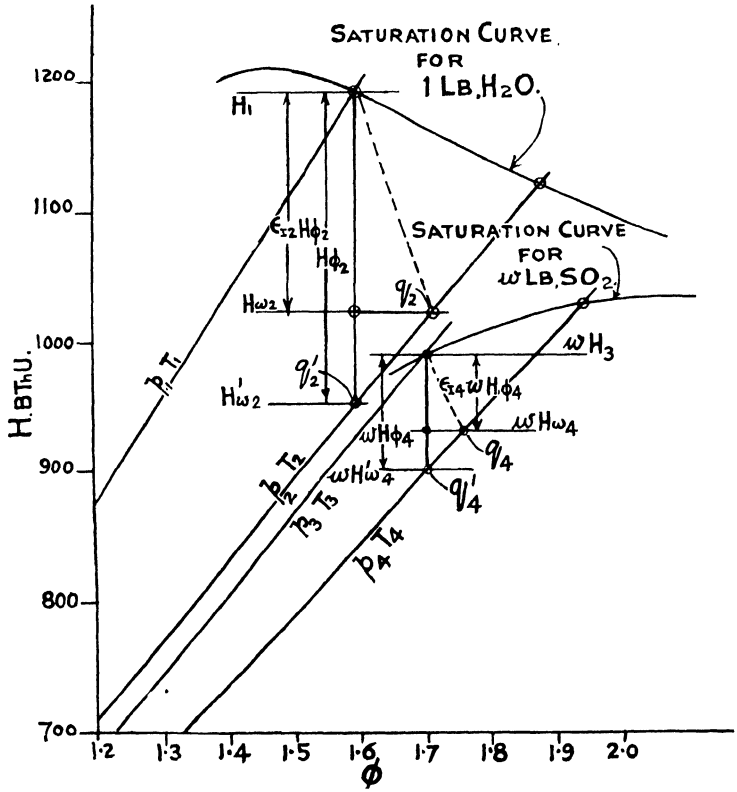



FIG. 51.

the basis of 3 lb./in.² abs. for the steam engine exhaust, the saturation temperature of the $SO_2$ would be 131° Fahr. and the corresponding pressure 137 lb./in.² abs.

**72.** A diagrammatic sketch of the arrangement of an ($H_2O$–$SO_2$) binary vapour combination is shown in Fig. 49, and the combined TΦ and HΦ diagrams in Figs. 50 and 51. Steam at pressure $p_1$ and temperature $T_1$ (or $T_{s1}$ if superheated) is supplied by the boiler to the steam engine. It is exhausted at $p_2$ and $T_2$, with some

quality $q_2$, to the steam engine condenser, which also acts as an $SO_2$ boiler. It is condensed while the liquid $SO_2$ in the condenser is evaporated at pressure $p_3$ and saturation temperature $T_3$. The $SO_2$ vapour produced is assumed to be dry. This vapour is supplied to the $SO_2$ engine, expanded and exhausted at the lower pressure and temperature limits $p_4$ and $T_4$, with some quality $q_4$. It is then condensed to liquid at $T_4$, in giving up its latent heat to the circulating water of the $SO_2$ condenser. This water is raised from $T_5$ to $T_6$. The liquid $SO_2$ is pumped to the steam condenser at $T_4$. Actually the condensate temperature may be slightly less than this value, due to undercooling, but for the theoretical case $T_4$ may be assumed.

The actual performance of each element of the combination can be deduced from the ideal when suitable values of the indicated efficiency ratios are chosen.

For every pound of steam passing through the system $w$ lb. of $SO_2$ are required for its condensation. The theoretical T$\Phi$ diagram (Fig. 50) is drawn for 1 lb. of steam, and the T$\Phi$ diagram, drawn for $w$ lb. of $SO_2$, is superposed on it. The corresponding H$\Phi$ combination is shown in Fig. 51. The liquid and saturation curves for the $SO_2$ are drawn by setting off $w$ times the liquid and total entropy values given in a table of properties of $SO_2$. It is assumed that there will necessarily be a 10° Fahr. drop in temperature, so that the initial $SO_2$ temperature line for $T_3$ is drawn 10° Fahr. below the steam exhaust temperature line.

A difficulty arises in the estimation of the $SO_2$ weight $w$, as this is not the same in the ideal and actual cases. The value is dependent on the quality of the exhaust steam, which will be $q_2'$ for the adiabatic case and $q_2$ with the actual. To be strictly accurate the value $w'$ calculated on the basis of $q_2'$ should be used for estimation of the absolute thermal efficiency $\eta$ of the combination, and $w$ based on $q_2$ for the actual or indicated thermal efficiency $\eta_1$. In Figs. 50 and 51 the curves have been drawn for 1 lb. $H_2O$ and $w$ lb. $SO_2$, that is, for the actual case.

The ideal weight of $SO_2$ is given by—

$$w' = \frac{{}^{(H_2O)}q_2'L_2}{{}_{(SO_2)}(H_3 - h_4)} = \frac{H_{w'2} - h_2}{H_3 - h_4} \quad . \quad . \quad . \quad . \quad . \quad (77)$$

and the actual weight by

$$w = \frac{q_2 L_2}{H_3 - h_4} = \frac{H_{w2} - h_2}{H_3 - h_4} \quad . \quad . \quad . \quad . \quad . \quad (78)$$

where $q_2'$ and $H_{w'2}$ are the ideal and $q_2$ and $H_{w2}$ the actual qualities and total heats of the steam engine exhaust, $H_3$ the total heat of dry saturated $SO_2$ at $p_3$, and $h_4$ the liquid heat of $SO_2$ at the exhaust pressure $p_4$.

In the ideal case the equivalent adiabatic heat drop in the $SO_2$ engine, per lb. of steam, is given by—

$$H_{\phi'4}=w'H_{\phi4}=w'(H_3-H_{w'4}) \quad . \quad . \quad . \quad . \quad (79)$$

The adiabatic heat drop in the steam engine is $H_{\phi2}=(H_1-H_{w'2})$

The gross theoretical heat drop or cycle heat of the combination is thus—

$$\left.\begin{aligned} H_c=H_\phi&=(H_{\phi2}+H_{\phi'4}) \\ \text{or } H_c=H_\phi&=(H_1-H_{w'2})+w'(H_3-H_{w'4}) \end{aligned}\right\} \quad . \quad . \quad (80)$$

The heat from the external source, the boiler, is received by the steam, as primary substance. Ideally the feed temperature is the exhaust temperature $T_2$, so that heat supplied per lb. of steam is—

$$H=(H_1-h_2) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (81)$$

The absolute thermal efficiency of the combination is thus—

$$\left.\begin{aligned} \eta&=\frac{H_c}{H}=\frac{(H_{\phi2}+H_{\phi'4})}{(H_1-h_2)} \\ \text{or } \eta&=\frac{(H_1-H_{w'2})+w'(H_3-H_{w'4})}{(H_1-h_2)} \end{aligned}\right\} \quad . \quad . \quad (82)$$

In the actual case with $w$ lb. $SO_2$ per lb. of steam (which is greater than $w'$), $H_{\phi'4}=w(H_3-H_{w'4})$, and if $\epsilon_I$ is the internal efficiency ratio of the $SO_2$ engine, the actual or indicated heat per lb. of steam (Fig. 51) is—

$$H_{i4}=\epsilon_{I4}H_{\phi'4}=\epsilon_{I4}w(H_3-H_{w'4})=\epsilon_{I4}wH_{\phi4} \quad . \quad . \quad (83)$$

The indicated heat per lb. for the steam engine, if the internal efficiency ratio is $\epsilon_{I2}$, is—

$$H_{i2}=\epsilon_{I2}H_{\phi2}=\epsilon_{I2}(H_1-H_{w'2}) \quad . \quad . \quad . \quad . \quad (84)$$

The total indicated (or net heat) of the combination is—

$$H_i=(H_{i2}+H_{i4})=[\epsilon_{I2}H_{\phi2}+\epsilon_{I4}w(H_3-H_{w'4})] \quad . \quad (85)$$

and the indicated thermal efficiency is

$$\eta_I=\frac{H_i}{H}$$

$$=\frac{\epsilon_{I2}(H_1-H_{w'2})+\epsilon_{I4}w(H_3-H_{w'4})}{(H_1-h_2)} \quad . \quad . \quad . \quad (86)$$

Then the internal efficiency ratio of the combination is given by

$$\epsilon_I=\frac{\eta_I}{\eta} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (87)$$

$$\text{or } \epsilon_I=\frac{H_i}{H_c}=\frac{\epsilon_{I2}(H_1-H_{w'2})+\epsilon_{I4}w(H_3-H_{w'4})}{(H_1-H_{w'2})+w'(H_3-H_{w'4})} \quad . \quad . \quad (88)$$

This rather lengthy method of determining the efficiencies and efficiency ratios is given to bring out clearly the interdependent relations of the $H_2O$ and $SO_2$ engines ; but for practical purposes the values can be obtained by the much easier method based on the heat supplied to the steam by the boiler and the heat rejected by the $SO_2$ to the cooler or condenser, per lb. of steam circulating in the system.

From the steam H$\Phi$ chart the initial and the ideal and actual exhaust total heats $H_1$, $H_{w'2}$ and $H_{w2}$ are obtained in the usual way. From an H$\Phi$ diagram for $SO_2$ (drawn for 1 lb. $SO_2$) the corresponding $H_{w'4}$ and $H_{w4}$ exhaust total heats are likewise obtained, when the pressure limits $p_3$ and $p_4$ and the internal efficiency ratio $\epsilon_{I4}$ are given. The value of $H_3$ is also given by this chart. These values are then multiplied by $w'$ and $w$ for the ideal and actual values per lb. of steam in the cycle. Then the heat supplied from the boiler is $H=(H_1-h_2)$ B.Th.U./lb. steam, and the heat rejected to the condenser is—

Ideal $h'=w'(H_{w'4}-h_4)$ B.Th.U./lb. steam
Actual $h=w(H_{w4}-h_4)$ ,, ,,

Hence, the absolute thermal efficiency is—

$$\eta=1-\frac{h'}{H}=1-\frac{w'(H_{w'4}-h_4)}{H_1-h_2} \quad . \quad . \quad . \quad . \quad (89)$$

and the actual or indicated thermal efficiency is—

$$\eta_I=1-\frac{h}{H}=1-\frac{w(H_{w4}-h_4)}{H_1-h_2} \quad . \quad . \quad . \quad . \quad . \quad (90)$$

The brake thermal efficiency is obtained from

$$\eta_B=\eta_m\eta_I \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (91)$$

where $\eta_m$ is the assumed mechanical efficiency of the combined machine.

The indicated and brake rates of heat consumption (Q) and the efficiency ratios are then directly calculable.

The percentage increase of output of the combination over that of the steam engine alone is given by—

$$\left.\begin{aligned} x&=\frac{\epsilon_{I4}w(H_{w'4}-h_4)}{\epsilon_{I2}(H_{w'2}-h_2)} \\ &=\frac{\epsilon_{I4}wH_{\phi4}}{\epsilon_{I2}H_{\phi2}} \end{aligned}\right\} \quad . \quad . \quad . \quad . \quad . \quad (92)$$

*Example* 15.—In a binary vapour ($H_2O$–$SO_2$) engine steam is supplied initially dry at 100 lb./in.$^2$ abs. and expanded to 3 lb./in.$^2$ abs. The dry $SO_2$ vapour produced is expanded from 137 lb./in.$^2$ abs. to 49·6 lb./in.$^2$ abs.

Assuming an indicated efficiency ratio of 0·7 for the steam engine and 0·65 for the $SO_2$ engine, estimate (*a*) the probable percentage

increase of power output due to the addition of the $SO_2$ engine, (*b*) the indicated thermal efficiency and efficiency ratio of the combination, (*c*) the indicated rate of heat consumption.

Referring to Fig. 51—

$p_1=100$ ; $T_1=787{\cdot}3°$ F. abs. ; $p_2=3$ ; $T_2=601°$ F. abs.
$H_1=1191{\cdot}3$ ; $\Phi_1=1{\cdot}608$ ; $H_2=1120{\cdot}6$ ; $h_2=109{\cdot}3$ ; $\Phi_2=1{\cdot}8833$.

Total heat after adiabatic expansion,

$$H_{w'2}=H_2-T_2(\Phi_2-\Phi_1)=1120{\cdot}6-601\times0{\cdot}275=955{\cdot}3 \text{ B.Th.U./lb.}$$

Adiabatic heat drop

$$H_{\phi2}=(H_1-H_{w'2})=(1191{\cdot}3-955{\cdot}3)=236 \text{ B.Th.U./lb.,}$$ which checks on the HΦ diagram.

From a table of the properties of $SO_2$ the following values are obtained :

$p_3=137$ ; $T_3=591$ ; $H_3=156{\cdot}1$ ; $\Phi_3=0{\cdot}269$.
$p_4=49{\cdot}6$ ; $T_4=529$ ; $H_4=162{\cdot}73$ ; $h_4=12{\cdot}58$ ; $\Phi_4=0{\cdot}3083$.

Total heat after adiabatic expansion,

$$H_{w'4}=H_4-T_4(\Phi_4-\Phi_3)=162{\cdot}73-529\times0{\cdot}0393$$
$$=141{\cdot}94 \text{ B.Th.U./lb. } SO_2$$

Adiabatic heat drop

$$H_{\phi4}=(H_3-H_{w'4})=(156{\cdot}1-141{\cdot}94)=14{\cdot}16 \text{ B.Th.U./lb. } SO_2$$

Ideal weight of $SO_2$

$$w'=\frac{H_{w'2}-h_2}{H_3-h_4}=\frac{955{\cdot}3-109{\cdot}3}{156{\cdot}1-12{\cdot}58}=\frac{846}{143{\cdot}52}=5{\cdot}9 \text{ lb. } SO_2/\text{lb. steam.}$$

With $\epsilon_{I2}=0{\cdot}7$ by calculation or direct scaling on HΦ chart for steam, $q_2=0{\cdot}902$ and $H_{w2}=1022$.

Actual weight of $SO_2$

$$w=\frac{H_{w2}-h_2}{H_3-h_4}=\frac{1022-109{\cdot}3}{143{\cdot}52}=\frac{912{\cdot}7}{143{\cdot}52}=6{\cdot}34 \text{ lb. } SO_2/\text{lb. steam}$$

and $wH_{\phi4}$ in Fig. 51 is $6{\cdot}34\times141{\cdot}6=92{\cdot}56$ B.Th.U.

(*a*) Percentage increase of output

$$x=\frac{w\epsilon_{I4}H_{\phi4}}{\epsilon_{I2}H_{\phi2}}=\frac{6{\cdot}34\times0{\cdot}65\times14{\cdot}16}{0{\cdot}7\times236}=0{\cdot}353 \qquad \begin{matrix}\epsilon_{I4}=0{\cdot}65\\ \epsilon_{I2}=0{\cdot}7\end{matrix}$$

or 35·3 per cent.

(*b*) Indicated thermal efficiency

$$\eta_I=\frac{\epsilon_{I2}(H_1-H_{w'2})+\epsilon_{I4}w(H_3-H_{w'4})}{H_1-h_2}$$

$$=\frac{0{\cdot}7\times236+0{\cdot}65\times6{\cdot}34\times14{\cdot}16}{1191{\cdot}3-109{\cdot}3}=\frac{165{\cdot}2+58{\cdot}35}{1082}=\frac{223{\cdot}55}{1082}=0{\cdot}206$$

Alternatively, $$\eta_I=1-\frac{w(H_{w4}-h_4)}{H_1-h_2}$$

On an HΦ diagram for $SO_2$, $q_4=0{\cdot}9$, and $H_{w4}=147{\cdot}715$, $(H_1-h_2)=1082$.

$$\therefore\ \eta_1=1-\frac{6{\cdot}34(147{\cdot}715-12{\cdot}58)}{1082}=(1-0{\cdot}794)=0{\cdot}206.$$

Efficiency ratio, $$\epsilon_1=\frac{\epsilon_{12}(H_1-H_{w'2})+\epsilon_{14}w(H_3-H_{w'4})}{(H_1-H_{w'2})+w'(H_3-H_{w'4})}$$

$$=\frac{0{\cdot}7\times236+0{\cdot}65\times6{\cdot}34\times14{\cdot}16}{236+5{\cdot}9\times14{\cdot}16}$$

$$=\frac{223{\cdot}55}{307{\cdot}1}=0{\cdot}727$$

(*c*) Rate of heat consumption,

$$Q_1=\frac{2545}{\eta_1}=\frac{2545}{0{\cdot}206}=12{,}354\ \text{B.Th.U./I.H.P. hr.}$$

or 206 B.Th.U./I.H.P. min.

# CHAPTER V

## INDICATED AND BRAKE HORSE-POWER

**73. The Indicator.**—The gross expenditure of energy by the working substance in the case of a reciprocating steam engine is calculated from the autographic pressure-volume record obtained by some form of indicator.

This closed PV curve is called the "indicator diagram," and its area represents the actual amount of energy expended by the steam on one side of the piston during a revolution of the engine.

It shows the pressure-drop between boiler and engine cylinder, the pressure-drop in the cylinder due to wiredrawing, and the character of the expansion, back-pressure, and compression curves.

By the comparison of indicator diagrams taken from each end of the cylinder, defective steam distribution due to faulty valve action can be detected from a consideration of the points of admission, cut-off release, and compression.

The diagram also enables the condition of the steam to be ascertained at any point of the expansion curve when the correct cylinder feed or weight of steam supplied per stroke is known.

The indicator was originally devised by James Watt. It has gone through a progressive series of developments,* and the design of the present-day instrument is very different from the original of Watt. A discussion of this development, however, would occupy too much space, and only three representative types of piston instruments now on the market and suitable for steam engine work are described here. In the domain of the internal combustion engine, where more exacting conditions of record arise, various indicators which are not of the piston type are used. The student who desires to study these instruments can refer to the information in the Proceedings of the Institution of Mechanical Engineers, quoted in the footnote.†

A steam engine indicator consists essentially of a small cylinder; a piston and a spring to regulate its displacement, in accordance with the varying steam pressure; a pencil carried by a system of light levers constituting a parallel motion, which magnifies the movement of the piston several times; a drum to which a paper or

* See Pickworth's *Indicator Handbook*.

† *Proc. I. Mech. E.*, 1921, p. 681; 1923, pp. 95, 111, 127, 137; 1925, pp. 1, 9, 69, 333.

card is fixed and which is reciprocated about its axis by means of a cord driven through a reducing gear from some reciprocating part of the engine. The closed PV indicator diagram results from a

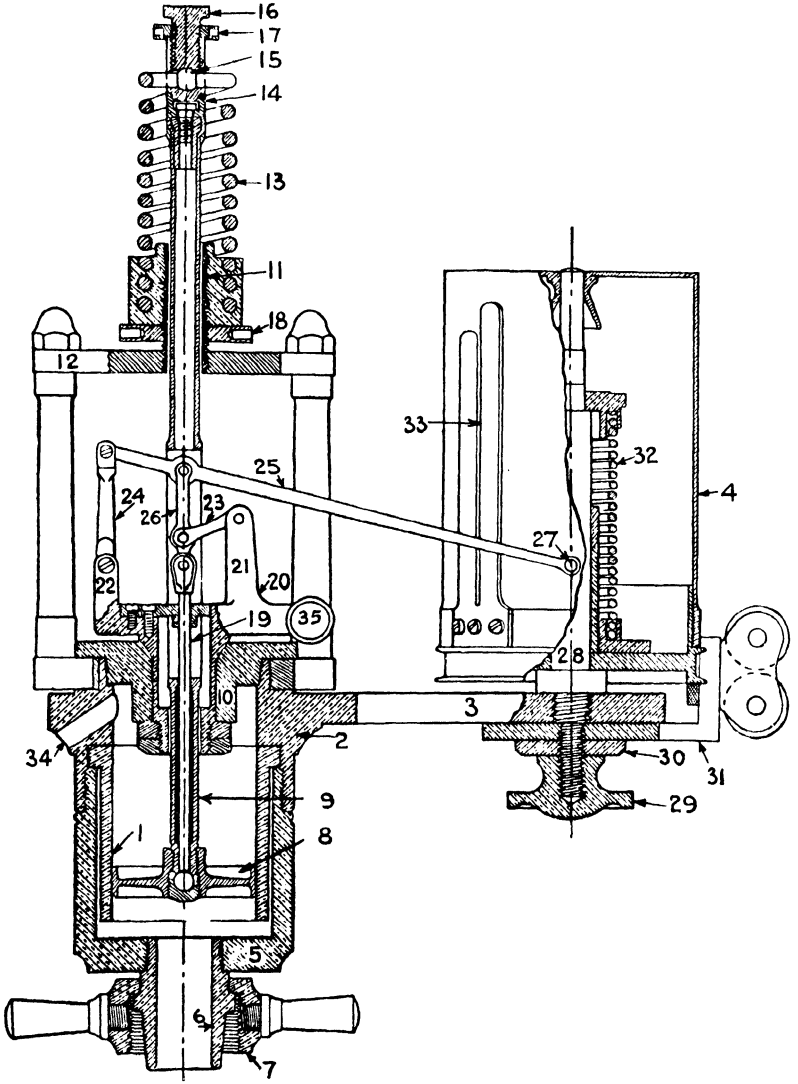



FIG. 52.

combination of the motion of the pencil in one direction and the motion of the paper in a direction at right-angles to it.

**74. Crosby Indicator.**—The latest design of this instrument is shown in section in Fig. 52, and a photograph of it, attached to the

indicator cock, in Fig. 53. The gunmetal cylinder 1 is held in the recess in the cylindrical projection 2 of the platform 3, carrying the drum 4, by the outer cylinder 5, to which is attached the connecting nipple 6 and union nut 7. The piston 8 is made of steel and has a spherical contour, to prevent sticking through any want of alignment. The piston rod 9, which is hollow, passes up through a bushing in the cap 10 and for a part of its length is slotted to accommodate the

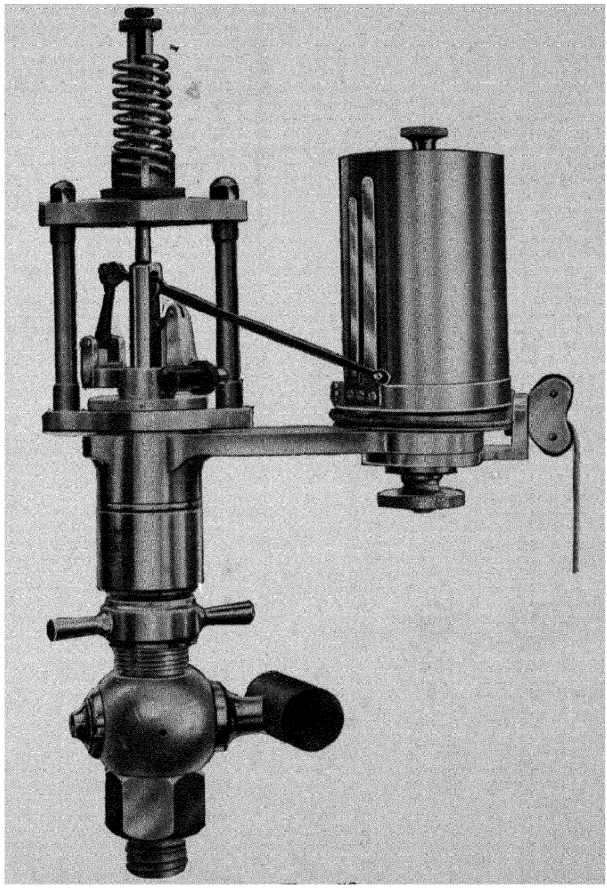


FIG. 53.

parallel motion linkage, as can be clearly seen on the photograph (Fig. 53). The hollow rod passes through a long bush 11 fixed in the crosshead 12 on to which the spring 13 can be screwed.

The top of the rod is provided with a swivel head 14 having a slot into which the ball 15, of the spring, fits, so that when the spring ball is clamped hard down on the bottom of the slot by the set-pin 16 the spring can be vertically adjusted, by turning it on the screw,

without rotation of the piston rod. The set-pin is locked by a lock-nut 17, and when the gear is properly adjusted the spring is finally locked by the nut 18. The parallel motion is not directly attached to the hollow piston rod, but to an auxiliary rod 19 having a spherical end, which passes down the rod and bears on a spherical seating. The force of the steam is thus transmitted through the rod to the spring independently of the linkage.

A swivel head 20 is mounted on the cylinder cap 10 and carries two lugs 21 and 22, to which are pivoted the links 23 and 24. The pencil lever 25 is pin-jointed to link 24 at one end and to a vertical link 26, which is pin-jointed at its lower end to the auxiliary rod 19. The link 26 is also jointed to link 23.

The links are so proportioned that the recording point 27 (which is of soft brass) at the end of the pencil lever moves in a straight line parallel to the cylinder and drum axes. The magnification of the piston motion, in this case, is 6 to 1.

The drum spindle 28 is screwed rigidly into the platform and is extended to take the clamping nut 29 and washer 30, which fix the cord pulley bracket 31 in position (see also Fig. 53). The drum, when at rest, is held against a stop by the coiled spring 32. The lower part of this spring is fixed to a flange inside the drum ; the upper part is fixed in a brass head with a square hole which fits over a corresponding squared part on the fixed spindle. The resistance of the spring can be adjusted by giving the spring several turns before it is slipped on to the spindle. The driving cord passes through a hole at the bottom of the drum, and over the guide pulleys, as clearly shown in Fig. 53. The indicator card is fixed on the drum by the two steel clips 33.

As shown in Fig. 53, the instrument is attached to the indicator cock by the union coupling. The cock has a two-way plug, and when the handle is in the position shown, the cylinder is put into communication with the atmosphere, through the small hole seen in front. When in this position the atmospheric line is drawn on the indicator paper, giving the datum line for pressure measurement on the diagram. It will be noted, from Fig. 52, that the space above the piston is open to the atmosphere through the passage 34, which allows any leakage of steam past the piston to escape.

**75.** In earlier types of indicators the springs were placed inside the cylinder and were subjected to varying steam temperatures.

Experiment has shown that indicator diagrams taken with internal springs under modern conditions of high pressure and temperature are unreliable, and it is now the practice to fit external springs which are removed from the influence of high temperature steam. These external springs can be changed without disturbance of the other parts of the indicator, which is necessary when the internal type is used. The spring is made of a single piece of spring steel wire, wound from the middle into a double coil, the ends of which are screwed into a brass head. At the opposite end of the

spring a small steel ball (already mentioned) is firmly fixed. The pressure scale, which has to be used for the estimation of the diagram pressure, is stamped on the head. The number represents the pressure in lb./in.$^2$ required to compress the spring sufficiently to move the pencil vertically 1 inch on the diagram.

When a diagram is to be taken the swivel head 20 carrying the parallel motion and pencil lever is rotated, by means of a knob 35,

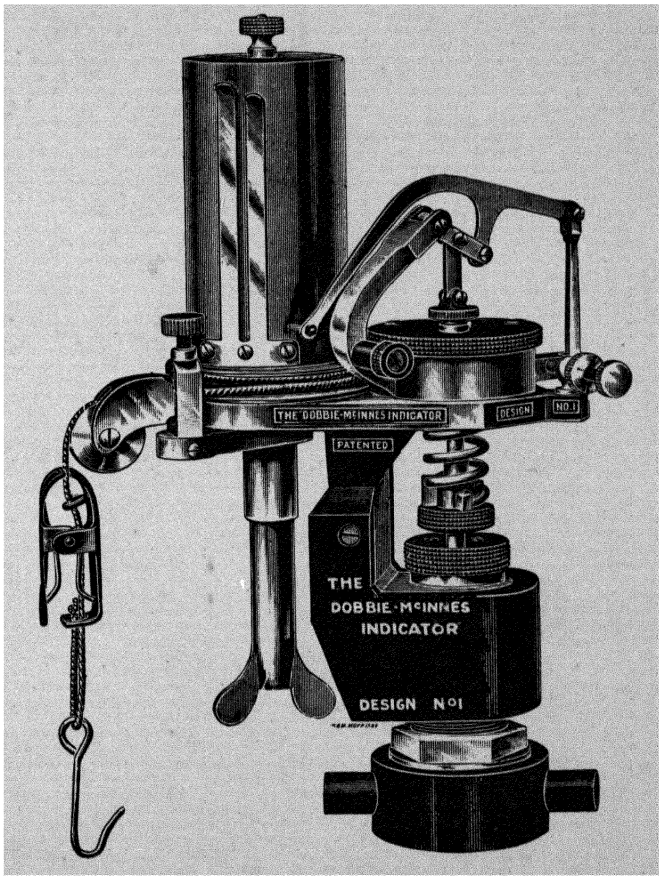



FIG. 54.

until the brass recording point just touches the paper. In order to avoid undue pressure of the point on the paper, the knob is made adjustable in the head, and arranged to come in contact with a stop-pin when the point touches the paper.

**76. Dobbie-McInnes Indicator.**—A photograph of the standard pattern of this external spring instrument is shown in Fig. 54, and a vertical section through the instrument, which is now being

fitted with a modified swivel head carrying the pencil gear, is shown in Fig. 55.

The body 1 of the indicator is made of brass, and the cylinder 2, also of brass, is screwed into the reduced portion 3 by means of a a screwdriver key which fits into the cross-slot 4. This lower portion is coned inside, to fit a corresponding conical spigot on the

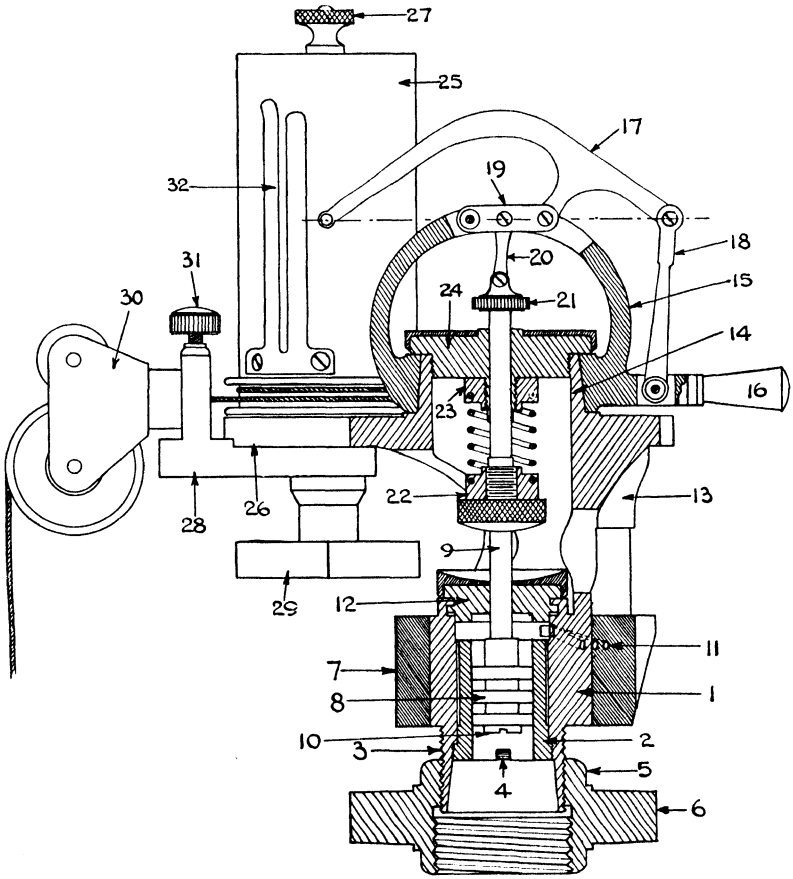



FIG. 55.

indicator cock, and is provided with a fine thread screwed to take the upper half of the coupling nut 5. The lower half of the nut is screwed with a special thread to take the screw of the cock. The nut is rotated by means of the cylindrical projections 6. As shown in the photograph (Fig. 54), it is sheathed in vulcanite. The body of the indicator is also covered with a thick sheathing of vulcanite 7, shown in section in Fig. 55.

The piston 8 is made of hardened steel and secured to the piston rod 9, which is a special steel tube, employed to obtain strength with light weight, by the screwed pin 10.

Two deep grooves are cut in the piston and act as throttling chambers, to prevent undue leakage of steam. Any steam or water which leaks past the piston is discharged downward through three holes 11, bored through the sheathing and body, to the inside of the space above the cylinder. This space is closed by a brass clamp head 12, sheathed on the top with vulcanite, which serves the purpose of a closing cap. By giving this head a half turn the clamp screw thread, shown in the section, is turned clear of the thread in the body, and the piston and head can then be withdrawn.

The frame 13 supporting the drum table and pencil mechanism is of the bayonet type, as clearly shown on the photograph (Fig. 54). It has the upper part 14 turned conical to take the swivel head 15, which is rotated by the handle 16. In the design shown in Fig. 54 this head is replaced by a cylindrical band with an overhanging arm for the suspension of the link work, to which the handle is attached as shown on the left; and a projection on the right makes contact with an adjustable stop-pin screwed into a knob on the platform. This stop arrangement does not show on the section (Fig. 55).

The curved pencil lever 17 of the parallel motion, as in the case of the previous instrument, is pivoted at the end of a vertical swing link 18, and to a horizontal link 19, which is pivoted to the swivel head. The two pin-joints of lever 17 are in the same straight line as the pencil point, so that the action of the mechanism is the same as though a straight link were used. The link 19 is pin-jointed to a vertical link 20, which in turn is pin-jointed to a detachable head 21, screwed on to the piston rod 9. This rod is screwed and enlarged near the middle to give the wide facing with knurled surface against which the lower brass end-piece 22 of the spring is screwed. The upper brass end-piece 23 is screwed to a projection on the under side of the top cap 24, which takes the thrust when the spring is compressed under the action of the steam pressure. This cap, which is covered with vulcanite, is screwed into the conical projection with a fine thread, to a considerable depth, to obtain secure fixture.

When the clamp head 12 is released and the top cap 24 is screwed out of the body, the piston, rod, spring, cap, and swivel head can be lifted out as one piece, and can then be taken adrift.

When a spring is to be changed, the rod, piston, and cap 24 are rotated by the knurled enlargement, until the rod is screwed off the detachable head 21, and the swivel head is removed. The spring and rod are then screwed off the cap 24, and finally the spring is screwed off the rod. This process is reversed when a spring is replaced. The springs are made of steel and the ends of the wire are inserted in holes drilled in the brass end-pieces, as shown in Fig. 55.

All springs intended for both vacuum and pressure are made the

requisite length, so that when screwed full home at both ends the pencil point of the indicator rests in the correct position, to afford space for the vacuum and pressure diagram below and above the

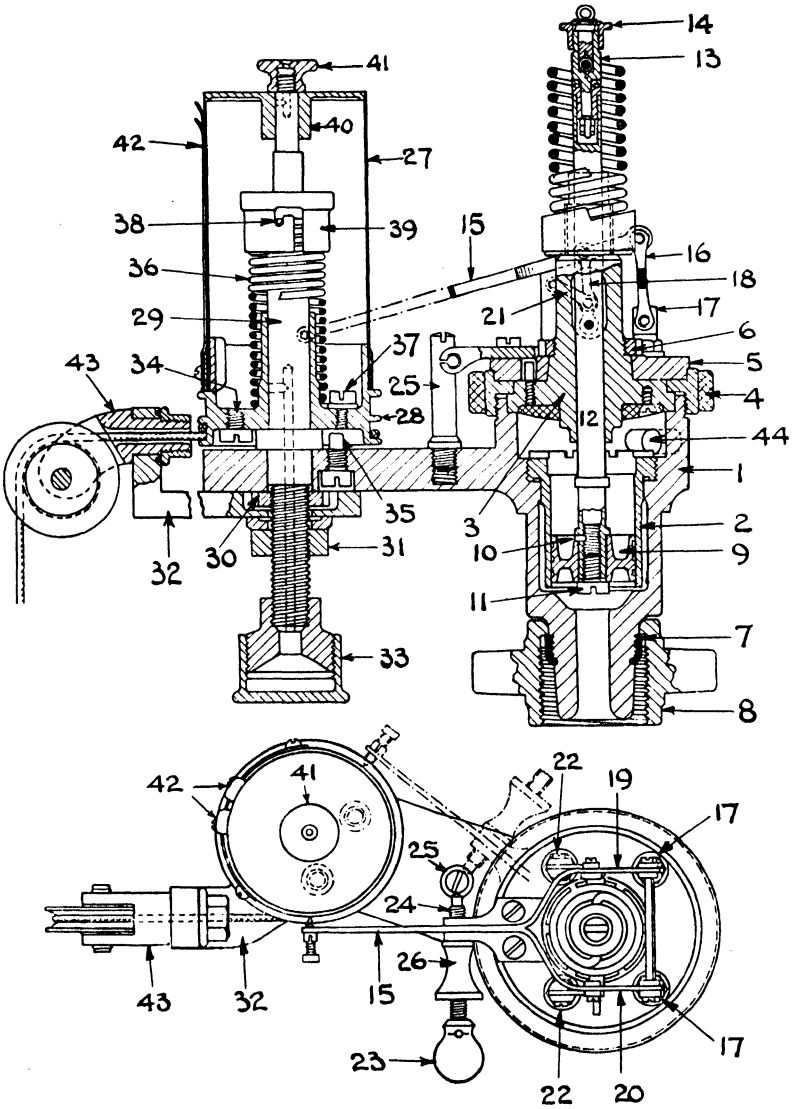



FIG. 56.

atmospheric line. This position varies with the strength of the spring. If it is desired to raise the atmospheric line, this can be done by unscrewing the piston rod and inserting a washer.

The drum 25, as in the case of the Crosby indicator, rotates on a spindle fixed on the platform 26, and is held against a stop, when at rest, by a coiled spring. It is prevented from rising by the milled nut 27 screwed on the reduced end of the drum spindle. The cord pulley bracket 28 is clamped on to the under side of the platform by the thumb nut 29. The pulley carrier 30 can be turned in the bracket to incline the pulleys in any desired direction. It is clamped in position by the set-pin 31. The small pulley prevents the cord from being thrown out of the guide pulley groove. The two steel clips 32 fixed on the drum by the screws at the bottom are used to secure the paper on the drum.

**77. Maihak-McAughtry Indicator.**—A section and plan of this indicator are shown in Fig. 56, and a photograph in Fig. 57.

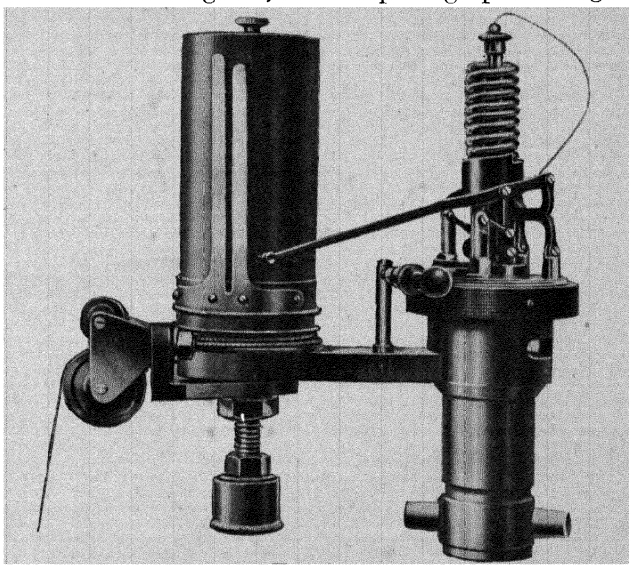


FIG. 57.

The body 1 of the instrument which, like the Crosby, is cylindrical throughout, is in a piece with the platform carrying the drum. The working cylinder 2 is screwed into this body from the top. Several sizes of cylinder can be used, and the indicator is equally serviceable for steam engine work with a large piston or internal combustion engine work with a small piston. The combined cylinder cover and spindle guide 3 is clamped down on the body by the coupling nut 4, which is sheathed with vulcanite. On the top of this is mounted the turntable 5, which carries the parallel motion linkage. This table is held in place by the lock-nut 6. A lock ring 7 is fitted to the union coupling nut 8, which is screwed on to the indicator cock body. This body is coned to take the coned end of the indicator. The piston 9 is made of steel hardened and polished. It is slipped over

the end of the piston rod, kept in place by a dowel pin 10, and secured on the rod by the screw 11. The rod 12 is turned from the solid steel bar and bored to reduce weight to a minimum. It is provided with a swivel head 13 with a slot into which the ball of the spring fits. When the spring is screwed on to the upper part of the spindle-guide 3, this head rotates. The spring is screwed down till the ball is hard on the bottom of the slot in the swivel head 13, and is locked in position by the clamping nut 14.

The parallel motion is identical with that of the Crosby indicator, but the pencil lever 15, as shown in Fig. 57, is bifurcated and carried by two swing links 16, which are pivoted in lugs 17 on the turntable. The vertical links 18, as shown clearly in Fig. 57, are pin-jointed to the two arms 19 and 20 of the pencil lever 15 and to the piston rod. The two horizontal links 21 are pin-jointed to links 18 and two pillars 22 fixed on the turntable. The turntable 5 is rotated by the hand-knob 23 on the end of the screw 24, which is adjusted to bear on the stop-pin 25, so as to give the requisite pressure of the pencil point at the drum surface. The screw is locked in position by the long nut 26.

The drum 27 is fixed to the driving pulley 28, which is provided with a sleeve that fits over the spindle 29. This is bolted to the platform by the nut 30. The lower screwed part of the spindle is extended to take the washer and nut 31, which clamp the guide pulley bracket 32 in position. The spindle, as indicated by the dotted lines, is bored at the lower end to provide a duct for lubrication of the driving pulley sleeve, and a Stauffer lubricator 33 is screwed on the end. The travel of the drum is limited by the head of screw 34 fixed in the pulley and the point of the stop-pin 35 screwed into the platform, when they come into contact. The drum spring 36 is fixed to the driving pulley by screw 37. When at rest the drum is held against the stops by this spring, the resistance of which can be varied by giving it one or two turns before it is slipped over the pin 38 on the spindle, by way of the vertical slot shown in the head 39. The spindle is reduced at the top to take the top guide sleeve 40 of the drum. The drum is held down by the milled lock-nut 41. The paper is attached to the drum by the spring clips 42, which are clearly shown on the photograph (Fig. 57). The head 43 carrying the single guide pulley (Fig. 56) and double pulleys (Fig. 57) can swivel in the bracket 32. The opening 44 in the body, shown clearly on the right side of the photograph (Fig. 57), permits any leakage steam to escape into the atmosphere.

It will be noticed that, as in the case of the Crosby indicator, the spring can be removed without disturbing the cylinder head or piston. Also, in both cases, the spring is in tension under the action of the steam pressure on the piston.

**78. Attachment of the Indicator.**—A hole which is usually not less than $\frac{3}{8}$-inch diameter is bored at each end of the cylinder, so as to be quite clear of the piston face when it is at either end of the stroke.

This hole is enlarged at the outside and tapped to take the $\frac{3}{4}$-inch Whitworth thread of the indicator cock, or a pipe coupling nipple.

It is a common practice to connect each end of the cylinder to a single indicator by means of union couplings and connecting pipes, as shown in Fig. 58.

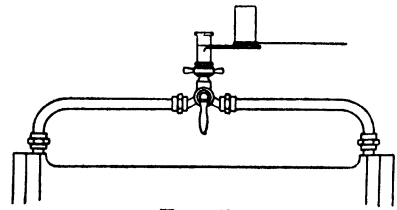


FIG. 58.

This is an unsatisfactory arrangement involving several more or less sharp bends and a considerable length of pipe, all of which may appreciably affect the accuracy of the pressure record on the diagram.

Two illustrations of this effect are shown in Figs. 59 and 60. In Fig. 59 * diagrams marked A were taken with the single indicator arrangement similar to that of Fig. 58, while diagrams marked B were taken from indicators attached directly to the cylinder ends. In Fig. 60 † the diagrams show the effect of a long pipe between indicator and cylinder.

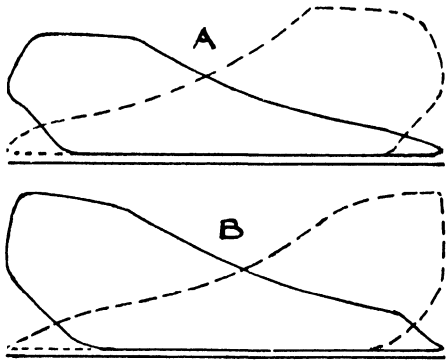



FIG. 59.

As far as indicator attachment is concerned, the rule, where accuracy is aimed at, is to avoid bends and angles and to keep the passage between the cylinder barrel and indicator as short and straight as possible.

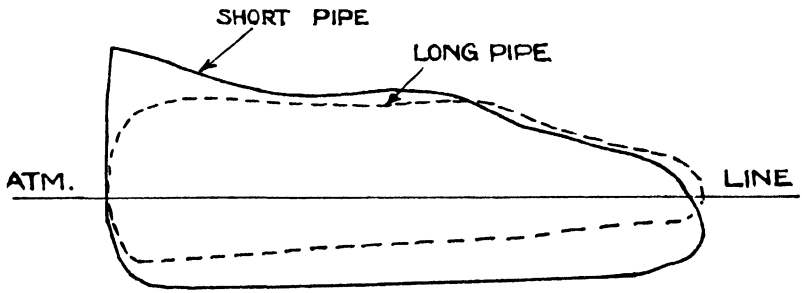



FIG. 60.

* Indicator diagrams by W. A. Tookey, Association of Engineers-in-Charge, 10 March, 1910.

† *Proc. I. Mech. E.*, May 1910.

This end can, as a rule be attained by screwing an indicator cock directly into each end of the cylinder. It is immaterial whether the indicators are placed with their axes horizontal or vertical. This arrangement requires an indicator at each end, and separate diagrams have to be taken.

**79. Reducing Gears.**—The motion or stroke of the indicator drum must exactly reproduce the motion of the engine piston, on a proportionately reduced scale. The diagram length corresponding to the drum stroke may vary from 2 inches with engines running at high rotational speed to 4 inches with engines running at low speed. The drum motion, which has to be in phase with the engine crank, is obtained by some form of reducing gear driven from a reciprocating part of the engine, usually the crosshead.

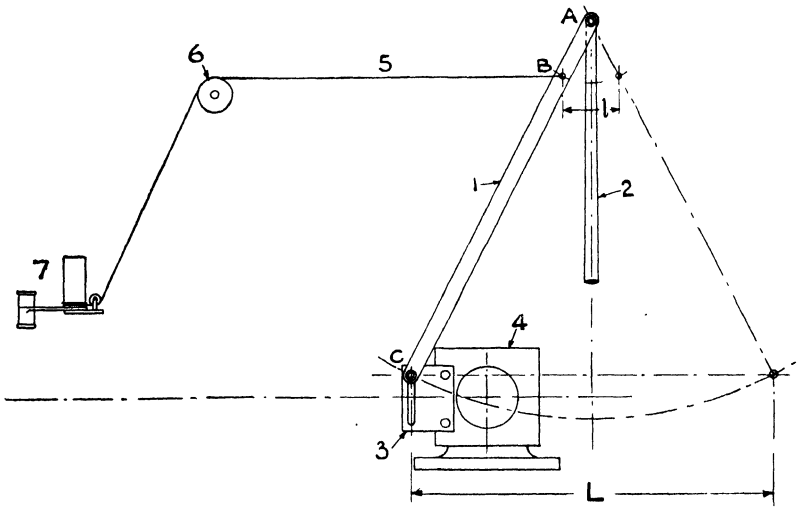



FIG. 61.

The particular form of drive for this gear depends on the type of engine. It should involve as few pin-joints as possible, where "lost motion" may occur, and the lengths of indicator cord connecting it to the drum should be kept as short as possible, to reduce errors due to stretching.

One form of lever drive for a horizontal engine is shown in Fig. 61. A swing lever 1 is pivoted at A on a standard 2, fixed to the frame. At the lower end C it has a pin which works in the vertical slot cut in a plate 3, attached to the crosshead 4. The indicator cord 5 is attached to a point B on the link 1. In order that the travel $l$ of the point B from which the cord is driven may be a correct reduction of the crosshead travel L, the ratio AB/AC must be constant for all crosshead positions. The pin and slot ensure this condition. It will be obvious that if the displacement of the indicator drum is

to be the same as that of point B at all positions of the piston stroke, the direction of the cord leaving B must be parallel to the line of stroke. This condition is ensured by the use of a guide pulley 6. As shown in Fig. 61, there is no break in the cord connection between B and the indicator 7. The cord has either to be made detachable from the driving gear or some form of clutch has to be provided at the indicator, to throw the drum into and out of gear as required. In some instances the slot in the plate is replaced by a fixed pin on the crosshead, and the end of the swing lever is slotted to take this.

An arrangement of this type fitted on a horizontal uniflow engine is shown in Fig. 62. The swing lever 1 is pivoted to the standard 2 at 3. It is provided with a slot at the end 4, which engages with a projecting stud 5 screwed into the side of the crosshead 6. Another slot 7 is provided near the top of the link, which engages with a pin

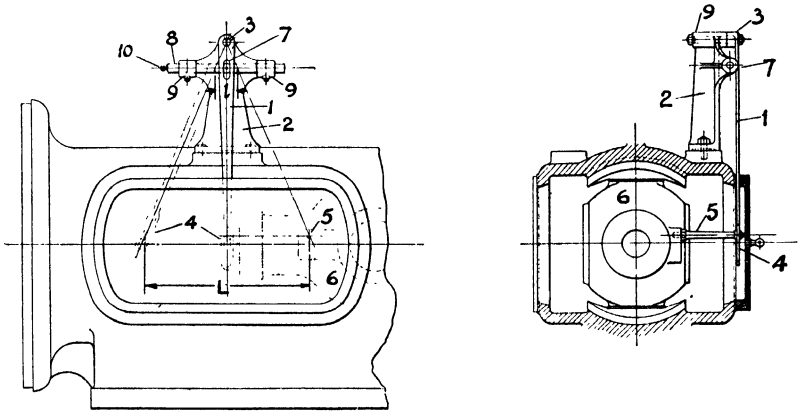



FIG. 62.

fixed in the plunger 8, supported in bearings 9 in the bracket. The indicator cord is attached to a hook 10 at the end of the plunger. The sources of slackness in these arrangements are the pins A and C and the slots. The bearing surfaces should be of ample proportions and hardened to reduce wear to a minimum.

Another arrangement fitted to a horizontal engine is shown in Fig. 63. The swing link 1 is pivoted at A on a standard and at C to the horizontal link 2, which is pivoted on the gudgeon pin D of the crosshead. The link 1 is provided with a grooved sector 4, to which a wire 5 is attached, and kept in tension, at all positions of the link, by the spring of the indicator 6.

It is not necessary in this case that the direction of the cord leaving the sector should be parallel to the line of stroke. When the cord runs on and off the sector the travel of any point on it is the same for all directions of the cord. A guide pulley, therefore, is unnecessary, when the indicator is situated above or below the level of the sector.

An arrangement of indicator gear used by Prof. Ripper on a small experimental vertical steam engine is shown in Fig. 64.

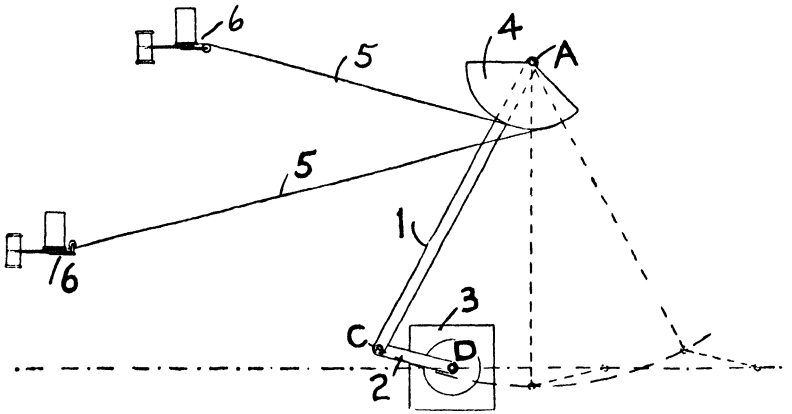



FIG. 63.

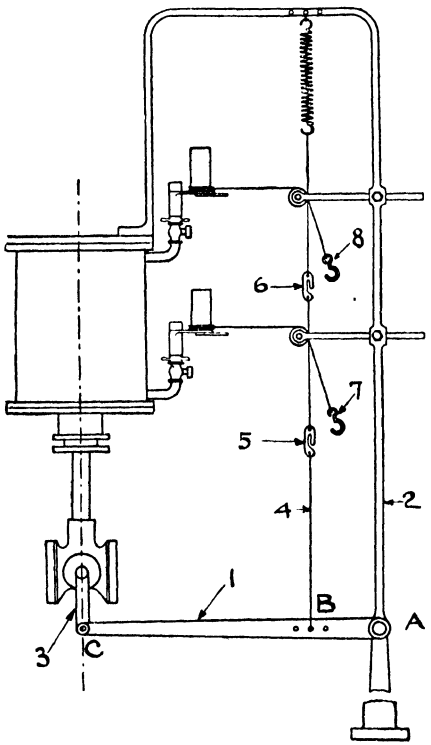



FIG. 64.

The swing link 1 is pivoted to the standard 2 at A, and at C to the vertical link 3, pivoted to the crosshead. The tension wire 4, fixed to link 1 at B, carries clips 5 and 6 to engage the hooks 7 and 8 of the indicator cords. This hook and clip arrangement with a high speed engine requires careful handling, as the operator runs the risk of damage to his fingers in the attempt to slip the hooks into the rapidly moving clips.

The lever type of reducing gear may be replaced by a pantograph arrangement, such as is shown in Fig. 65. The swing link 1 pivoted to the crosshead at A is pin-jointed to the lower link 2 at D. Link 2 is pivoted at the fixed centre C, as also link 3. The latter is jointed to link 4, which in turn is jointed to the swing link 1. At all positions of the crosshead the line joining A and C cuts link 4 at the point E, which moves parallel

to the line of stroke. The indicator cord has therefore to be attached at E to obtain the correct reduction $x/L$, and a guide pulley 5 has to be employed to lead the cord from E parallel to the line of stroke.

80. Instead of a lever arrangement the reduced drum travel can be obtained by reducing cord-pulleys. A cord connected to the cross-head is passed round a large drum which is rotated against the action of a spring in a similar way to the indicator drum. This pulley, either directly or indirectly, drives a smaller pulley over which the cord from the indicator drum is passed. These fittings are made by the indicator manufacturers and can be bolted to the

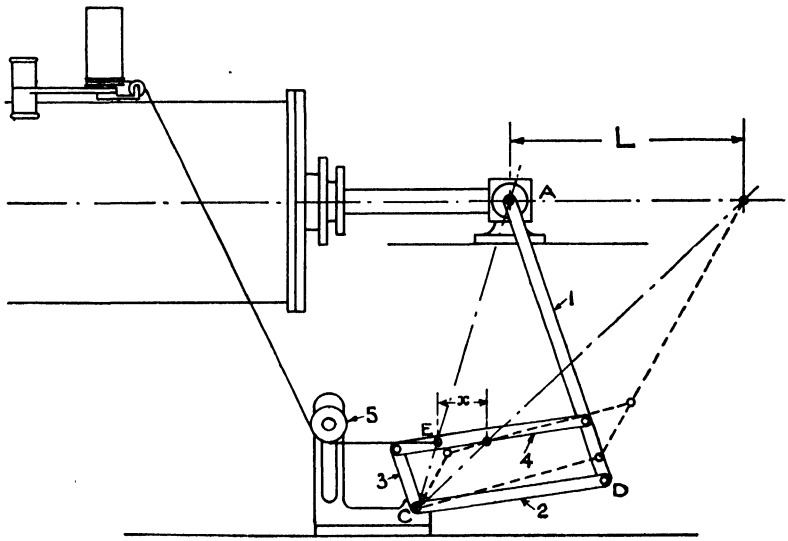



FIG. 65.

under side of the indicator drum platform in place of the ordinary guide pulley bracket.

The type of pulley reducing gear for use with the Maihak indicator (Fig. 56) is shown in elevation and plan in Fig. 66. A bush 1 is screwed into the bracket 2 bolted to the indicator platform 3, and is locked by the set-pin 4. This bush is screwed to take the fine pitch square thread on the spindle 5, to which the drum 6 is rigidly attached by the coned end and nut. The sleeve of this drum is connected to the coiled spring 7, carried in the casing 8, attached to the bottom of the bush 1. The guide pulley carrier 9 is clamped round the bush by the bolt 10, and thus can be adjusted to any desired position to suit the cord lead.

The screwed spindle at the upper end carries the reduced motion pulley 11 on which the cord from the indicator pulley 12 is wound.

When the driving cord 13 is pulled by the crosshead the drum is rotated against the resistance of the spring, and with it the screwed spindle and the reducing pulley. The screw causes both pulley and drum to rise and fall, and so maintains the correct pitch of the cords on these surfaces. This prevents any overlapping of cord, and consequent error in the indicator drum displacement.

**81. Method of Taking an Indicator Diagram.**—Couple up the indicator to the cock (as shown in Fig. 53). With the engine in motion hook the cord to the reducing gear mechanism and adjust its length, until the drum does not knock at either end of its stroke.

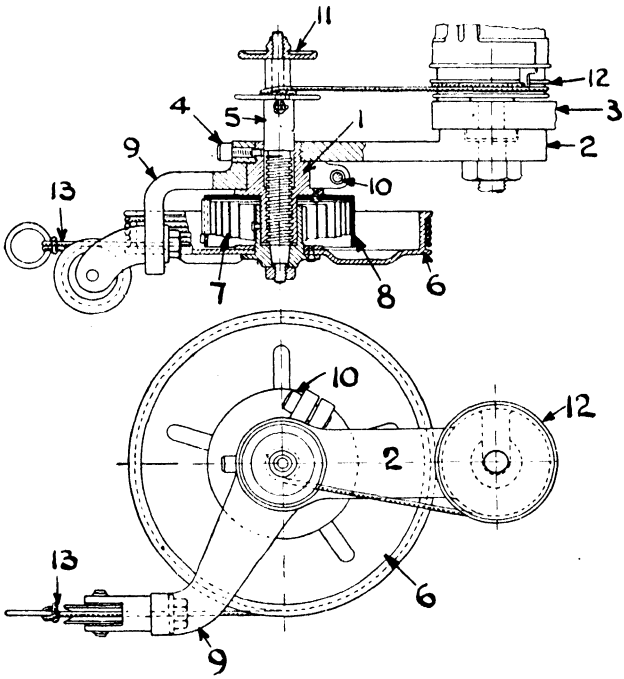



FIG. 66.

Withdraw the piston, clean the cylinder, oil the piston and replace it. See that the indicator spring is properly secured. Clip a card on the drum, turn the indicator cock handle until the air port is open to the cylinder, press the pencil point against the paper, and pull the drum cord *by hand* through the full drum stroke.

The line drawn on the paper is the atmospheric pressure reference line.

Again hook the cord to the reducing gear mechanism and set the drum in motion ; turn the indicator cock handle to the open position to admit steam to the cylinder and allow the piston to reciprocate for a few cycles. Then press the pencil point *lightly* on the paper

during several cycles. Shut the indicator cock, detach the cord, and slip the card from the drum.

Mark on the card, the end of cylinder (crank or head, or top or bottom), the time of record, and the scale of the indicator spring. Various other data can be entered later, but these are the three essentials which *must* be recorded immediately the " card " is taken.

**82. Theoretical Indicator Diagrams.**—In the discussion of the engine cycle from the thermodynamic standpoint in Chapter IV, it has been assumed that the expansion curve of the ideal diagram is an adiabatic. Such a condition never obtains in practice. Unless the steam is highly superheated on admission, some of it condenses before cut-off takes place, and the steam is always more or less wet at the beginning of expansion. Condensation proceeds during part of the expansion, and then a certain amount of re-evaporation occurs towards the end of expansion.

The complex actions taking place in the cylinder produce a curve which in the majority of cases is approximated by a law of the form $PV^n=c$, where $n$ may vary from 0·7 with wet steam to 1·30 with highly superheated steam.

It is the general practice, in order to simplify the design calculations, to adopt a modified theoretical diagram of reference having a rectangular hyperbola as expansion curve, for which the simple law is $PV=c$, or $n=1$.

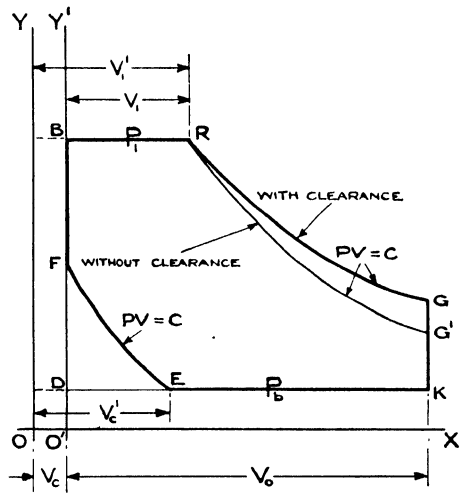



FIG. 67.

The simplest theoretical standard diagram is one for an engine which has no clearance. This is shown in Fig. 67 by DBRG'K, the co-ordinate axes showing lines of zero pressure and zero volume, being O'X and O'Y'.

Its area gives the theoretical work done on one side of the piston by the steam in the cycle between the initial and back-pressure limits $p_1$ and $p_b$, with incomplete expansion.

In the actual case, however, clearance always exists, owing to the passages behind the piston and the space between the cylinder cover and piston face, required to prevent mechanical knock.

Average values of clearance expressed as percentages of the piston displacement or stroke volume are given in Table II below for various types of engine. It depends on the size of the engine,

ratio of stroke to diameter, and the type and disposition of the valve gear employed.

TABLE II

CYLINDER CLEARANCE

| Type of Engine. | Per cent. Clearance. |
|---|---|
| Corliss valve | 2 to 4 |
| Long slide valve | 1·75 to 2·75 |
| Double beat valve | 5 to 7 |
| Ordinary slide valve | 6 to 12 |
| Locomotives | 7 to 12 |
| Small horizontal and vertical | 9 to 20 |
| Marine slide valve | 10 to 18 |
| Marine piston valve | 12 to 25 |

The effect of the clearance ($V_c$) on the diagram is to raise the expansion curve from RG′ to RG ; and the theoretical diagram for engine with clearance is given by DBRGK, OX and OY being the co-ordinate axes. O′ is the origin for the curve RG′, and O for RG. For the same stroke volume $V_o$ the terminal pressure is increased.

In addition, however, to clearance it is necessary for dynamical reasons to have more or less compression. The discharge of the steam is stopped before the end (D) of the exhaust stroke, and the remaining steam in the cylinder is compressed above the back-pressure value $p_b$. If the point of compression (E) is made early enough it is possible to compress the steam to the admission pressure or above it. Normally the maximum compression pressure (at F) is less than the admission pressure $p_1$, that is, the compression is incomplete. A third theoretical diagram of reference allowing for compression, BRGKEF, may, therefore, be chosen for the purpose of comparison with the actual indicator diagram obtained from the engine.

**83. Nominal and Actual Ratios of Expansion.**—In the case of the engine without clearance the total volume of the steam at the cut-off point R (Fig. 68) is $V_1$, and the final volume at the end of the stroke is $V_o$. The ratio $r=\frac{V_o}{V_1}$, is called the "nominal" ratio of expansion, or nominal number of expansions.

In the engine with clearance, the total volume at cut-off is $V_1'=(V_1+V_c)$, while the final volume is $V_2=(V_o+V_c)$. The ratio $V_2/V_1=\frac{V_o+V_c}{V_1+V_c}=r'$ is called the actual ratio of expansion.

Let the clearance volume be denoted as a fraction of the stroke volume, $V_c=cV_o$, then since $V_o=rV_1$

$$r'=\frac{(1+c)V_o}{V_1+cV_o}=\frac{1+c}{\left(\frac{1}{r}+c\right)} \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

In each case the arbitrarily chosen law of expansion is $PV=C$, so that the corresponding terminal pressures without and with clearance are $p_2=rp_1$ and $p_2'=r'p_1$.

**84. Theoretical Indicated Work per Cycle.** *First Case. No clearance and no compression.*—The work per cycle is given by the area of DBRG'KD.

Expressing pressures in lb./ft.$^2$ (P).

This is, $E=P_1V_1+P_1V_1 \log_e r-P_bV_o$ | $P=144p$, where $p$ is in lb./in.$^2$

$$=P_1V_1(1+\log_e r)-P_bV_o$$

$$=P_1\times\frac{1}{r}V_o(1+\log_e r-P_bV_o)$$

$$=V_o\left[P_1\frac{(1+\log_e r)}{r}-P_b\right] \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

The quantity in the square brackets is obviously the mean ordinate or mean effective pressure ($P_m$) of the diagram in lb./ft.$^2$

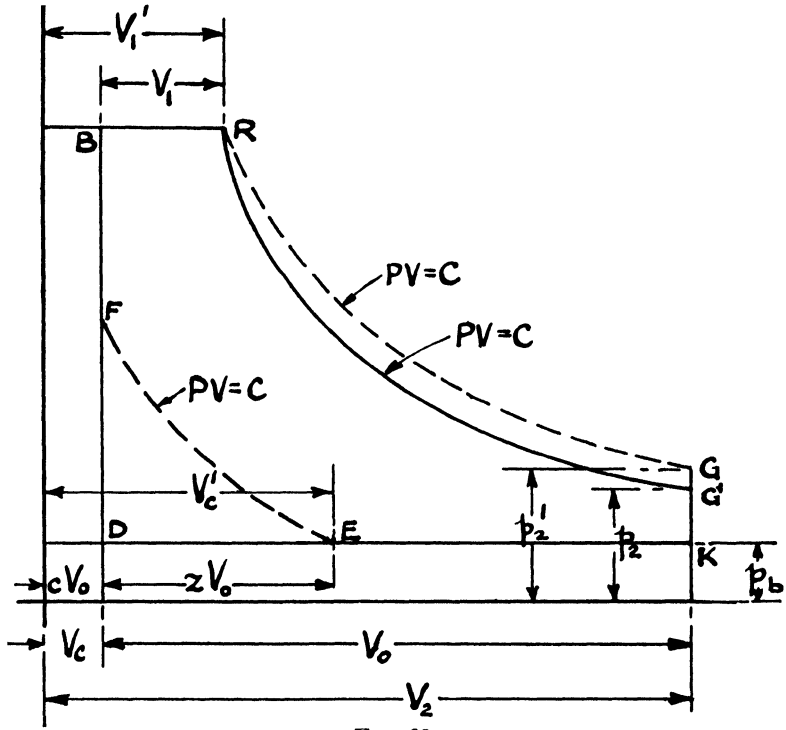



FIG. 68.

*Second Case. Clearance and No Compression.*—The work per cycle is given by the area of BRGKD,

or $$E'=[P_1V_1+P_1V_1' \log_e r'-P_bV_o]$$

$$=P_1\frac{V_o}{r}+P_1\frac{V_2}{r'}\log_e r'-P_bV_o$$

Substituting for $V_2$ and $r'$ in terms of $V_o$ and $V_c$—

$$E'=V_o\left[P_1\left\{\frac{1}{r}+\left(\frac{1}{r}+c\right)\log_e\frac{1+c}{\left(\frac{1}{r}+c\right)}\right\}-P_b\right] \quad . \quad . \quad (3)$$

The term in the square brackets is the theoretical mean effective pressure ($P_m'$) in lb./ft.$^2$

*Third Case. Clearance and Compression.*—As in the previous case the first term in the square brackets is the same, but the second, $P_b$ has to be multiplied by a reducing factor.

Let the compression ratio be $r_c=V_c'/V_c$, and let $DE=zV_o$, then

$$V_c'=V_o(z+c) \quad \text{and} \quad r_c=\left(\frac{z+c}{c}\right)$$

The area DEF is thus given by—

$$P_bV_c'\log_e r_c-zP_bV_o=P_bV_o\left[(z+c)\log_e\frac{z+c}{c}-z\right]$$

Then the work per cycle represented by the area BRGKEF is

$$E''=[P_1V_1+P_1V_1'\log_e r'-P_bV_o]-DEF$$

$$=V_o\left[P_1\left\{\frac{1}{r}+\left(\frac{1}{r}+c\right)\log_e\frac{1+c}{\left(\frac{1}{r}+c\right)}\right\}-P_b\left\{1-z+(z+c)\log_e\frac{z+c}{c}\right\}\right] \quad (4)$$

The quantity in the square brackets is the theoretical mean effective pressure ($P_m''$) in lb./ft.$^2$

It will be seen from (4) that when $z=0$, there is no compression and the expression reduces to that of (3), and when also $c=0$, there is no clearance and it reduces to that of (2).

The relation of these arbitrary standards of comparison to the actual work diagram given by the indicator is discussed in Art. 92.

*Example* 1.—Compare the values of the theoretical mean effective pressure of a single cylinder non-condensing engine, having a cut-off at 0·6 stroke, when: (a) clearance and compression are neglected; (b) when 12 per cent. clearance is allowed; (c) when 12 per cent. clearance and compression at 80 per cent. stroke are allowed for.

Initial pressure 100 lb./in.$^2$ abs.; back pressure 17 lb./in.$^2$ abs.

(a) $p_m=p_1\left(\frac{1+\log_e r}{r}\right)-p_b$ $\qquad p_1=100$, $p_b=17$

$=100\left(\frac{1+\log_e 1\cdot666}{r}\right)-17$ $\qquad r=\frac{1}{0\cdot6}=1\cdot666$

$=(90\cdot68-17)=73\cdot68$ lb./in.$^2$

(b) $p_m' = p_1\left\{\frac{1}{r}+\left(\frac{1}{r}+c\right)\log_e\frac{1+c}{\frac{1}{r}+c}\right\}-p_b$ $\quad c=0\cdot12$, $(1+c)=1\cdot12$, $\frac{1}{r}=0\cdot6$

$=100\left\{0\cdot6+(0\cdot6+0\cdot12)\log_e\frac{1\cdot12}{0\cdot72}\right\}-17$

$=100(0\cdot6+0\cdot72\log_e 1\cdot555)-17$

$=100(0\cdot6+0\cdot72\times0\cdot4414)-17$

$=(91\cdot78-17)=74\cdot78$ lb./in.$^2$

(c) $p_m'' = p_1\left\{\frac{1}{r}+\left(\frac{1}{r}+c\right)\log_e\frac{1+c}{\frac{1}{r}+c}\right\}$ $\quad z=0\cdot2$, $z+c=0\cdot32$

$-p_b\left\{1-z+(z+c)\log_e\frac{z+c}{c}\right\}$

$=100\left\{0\cdot6+(0\cdot6+0\cdot12)\log_e\frac{1\cdot12}{0\cdot12}\right\}-17\left\{0\cdot8+0\cdot32\log_e\frac{0\cdot32}{0\cdot12}\right\}$

$=91\cdot78-17(0\cdot8+0\cdot32\log_e 2\cdot66)$

$=(91\cdot78-17\times1\cdot113)$

$=(91\cdot78-18\cdot92)=72\cdot86$ lb./in.$^2$

**85. The Actual Indicator Diagram.**—A representative diagram of a single cylinder non-condensing engine running under satisfactory

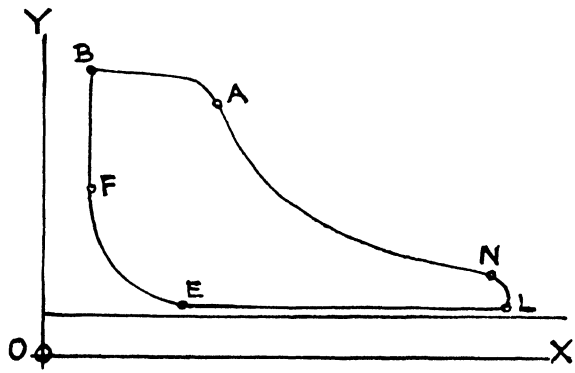



FIG. 69.

conditions is shown in Fig. 69, the expansion and compression curves AN and EF being approximately fitted by the law $PV^n=C$.

With correct valve setting the admission line FB should be practically vertical as shown. Excess of lead (see Art. 165) causing too early admission gives an admission line *m* (Fig. 70) sloping to the left; defect of lead gives a line *n* sloping to the right (Fig. 71).

During flow of steam into the cylinder between B and cut-off, there is always more or less drop of pressure as shown by the dotted line $p$ (Fig. 70) and by BA (Fig. 69). This is due to increased demand for steam as the piston moves out and causes a sudden and large displacement.

This gradual fall of pressure is defined as "wiredrawing." The effect on the steam line of regulating the engine by throttle valve, and thus varying the opening for the supply of steam, is shown in Fig. 72, which was obtained by successively removing portions of

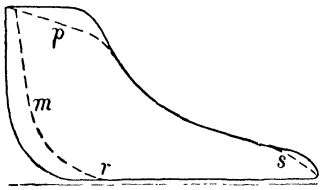



FIG. 70.

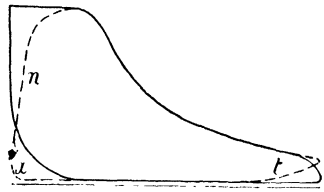



FIG. 71.

the load on the engine, and maintaining the speed constant by partially closing the steam valve.

The admission pressure line A for a heavy load fell to B for a medium and to C for a light load, the points of cut-off, release, and compression remaining constant.

The cut-off point A (Fig. 69) is more or less sharp with trip-valve gears, which cut off the steam suddenly by the action of a strong spring. The cut-off with a slide-valve, however, is more gradual: the corner is rounder and the exact point of cut-off is more difficult to locate. In such a case the point of cut-off may be

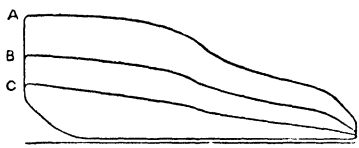



FIG. 72.

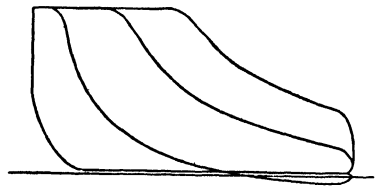


FIG. 73.

taken as the point where the concave curve of the expansion line meets the convex curve of the cut-off corner. A close determination can be obtained if the diagram curve is plotted to a logarithmic scale (see Art. 127, p. 218).

The effect on the diagram of varying the point of cut-off is shown in Fig. 73 for non-condensing engines and in Fig. 74 for condensing engines, with a trip-valve gear, the cut-off being fairly sharp. The effect of regulating the cut-off with a slide-valve high-speed engine is shown by Fig. 75. Here the cut-off point is much less definite than with a trip gear (Fig. 74). In the non-condensing

engine diagrams (Fig. 73) with an early cut-off the expansion line falls below the atmospheric line and forms a loop at the end of the diagram. This is due to the pressure of steam during expansion falling below atmospheric pressure, and hence, when the exhaust port opens, the pressure will rise instead of falling to the back pressure line. This is a wasteful form of diagram.

In condensing engines with a good vacuum a loop is not formed, even with a very early cut-off (Fig. 74).

The release point N (Fig. 69) occurs just before the end of the stroke. The higher the rate of revolution of the engine the earlier the exhaust; and the trouble with high-speed engines is not how to get the steam into but out of the cylinders.

The exhaust line NL (Fig. 69) represents the fall of pressure which takes place when the exhaust port is opened. Early opening to exhaust is shown at *s* (Fig. 70) and late opening to exhaust at *t* (Fig. 71).

The back pressure line LE (Fig. 69) shows the pressure on the

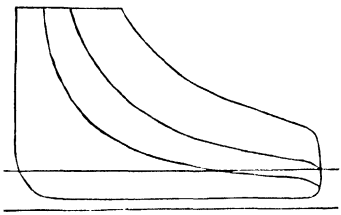


FIG. 74.

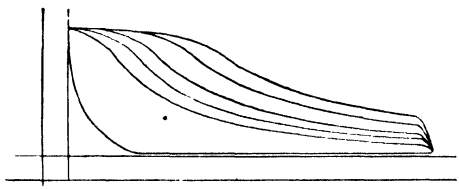


FIG. 75.

piston during its return stroke, its value being reckoned from the back pressure line down to the line of zero pressure OX.

In non-condensing engines with ample exhaust passages the back pressure approximates closely to that of the atmosphere. In good designs the back pressure is about 1 lb./in.$^2$ above the atmosphere. In condensing engines the back pressure runs from 3 to 4 lb./in.$^2$ abs.

The compression curve EF (Fig. 69) commences at the point of exhaust closure E. This point depends on the amount of inside lap of the valve and the angular advance of the eccentric (see Art. 165), and the nature of the curve formed will depend on the pressure of the steam trapped, as well as upon the volume of the clearance space. A valve with inside lap suitable for a condensing engine, when the pressure of the steam at the beginning of compression is, say, 3 lb./in.$^2$ abs., will probably show an excessive amount of compression when the engine is used non-condensing and compresses the steam at 16 lb./in.$^2$ abs. (see also uniflow engines, Chapter XV).

The shape of the compression corner of the diagram is, however, due to several causes, including the compression of the clearance

steam, and any air trapped with the steam, especially in non-condensing engines, leakage of steam into and out of the cylinder during compression (see Art. 127), and the early admission of steam before the piston has reached the end of the return stroke, especially with an expansion gear linked up towards mid-position.

**86. Losses due to Incomplete Compression, Expansion, Wiredrawing, etc.**—The various losses of effective work per cycle are indicated by the cross-hatched areas in Fig. 76.

The line TS is drawn to represent the stop-valve pressure $p_1$. If there were no drop of pressure in the connecting steam pipe between it and the boiler, $p_1$ would also be the boiler pressure. As a rule there is more or less drop; but since comparisons of

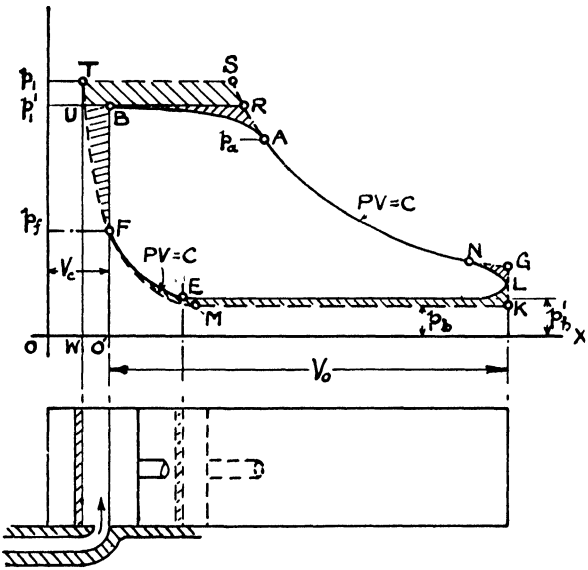



FIG. 76.

engine performance are usually based on the pressure at the boiler side of the stop-valve the stop-valve pressure has been taken here as an upper limit.

If the steam, therefore, were admitted without any drop of pressure at $p_1$ the expansion curve passing through A would start at S; and if the compression were carried out on the clearance steam from the back pressure $p_b'$ to this initial pressure $p_1$, it would have T as its final point. If it were admitted at the lower pressure $p_1'$ the corresponding points on the expansion and compression curves would be R and U. The area TSRU outside the actual diagram thus represents a loss of effective energy due to accelerating the steam in the pipe and in overcoming frictional resistance to flow into the cylinder casing. This is really a loss external to the cylinder, and

may be termed the pipe loss. Coming to the losses occurring in the cylinder, the first to be considered is that due to incomplete compression.

If the steam were further compressed from $p_f$ to $p_1'$, or from F to U, the clearance steam would occupy a final volume OW. Compression by the piston, however, is only carried to $p_f$, and between F and U the work has to be done by the additional boiler steam (at $p_1'$) admitted to the clearance space. Imagine a light diaphragm which can move along with the piston placed behind it, as illustrated by the dotted sketch below. When the piston reaches the " in " position it stops, and the pressure of the clearance steam behind it and the diaphragm is $p_f$. The admission valve then opens and the fresh steam is imagined to flow in between the diaphragm and the stationary piston, forcing the diaphragm outward and doing the work of compression on the steam to raise the pressure from $p_f$ to $p_1'$. The volume displacement of the imaginary diaphragm will be O'W, which will show the volume occupied by the new boiler steam, while OW will show the volume occupied by the cushion steam. The work done by the boiler steam in compressing the cushion steam from $p_f$ to $p_1'$ is the area below FU, or UFO'W, and when the expansion begins behind the piston after cut-off this energy will be expended again on the piston as the cushion steam expands between U and F. If, however, the boiler steam had been admitted to the clearance space with the cushion steam at pressure $p_1'$, the constant pressure work it would do on the piston, would be represented by the rectangle below UB, or UBO'W. Its actual effective work is less than this amount by the triangular area UBF, which represents a waste of energy due to kinetic effect and shock, as the boiler steam, which is throttled by the admission valve, rushes into the lower pressure steam in the clearance. This clearance loss may be considerable when the clearance volume is proportionally large, as in short-stroke high-speed engines.

The compression of the steam towards the end of the exhaust stroke is of more importance from the point of view of smooth running and prevention of shock than of steam economy ; and the higher the speed the greater the necessity for a good compression corner on the diagram.

As shown in Art. 179, the Stephenson link motion affords a very efficient means for the fulfilment of this condition, by providing an increasingly early closure to exhaust when " linking up " takes place at high speed.

The result of this action is the large compression corner shown in Fig. 75, which improves the smoothness of running of the engine.

Owing to the throttling action at the admission valve the pressure drops from $p_1$ at B to $p_a$ at cut-off A, and the triangular area BRA thus represents the loss of available energy due to wire-drawing on admission.

The total loss of available energy between stop-valve and cut-off, by the working steam admitted per cycle, represented by the area FTSAB, might be regarded as stored in the steam in the form of friction heat at A, provided no heat passed into the cylinder wall between F and A.

Practically it is stored in the walls, in addition to heat arising from initial condensation, unless in the case of highly superheated steam.

A discussion of these heat interchanges between the steam and cylinder wall is given in Chapter VI.

The next loss occurs at release N. As already pointed out in Chapter IV, even if expansion were carried down to G, the end of the stroke, there is a loss of available energy due to cutting off the toe of the diagram. Based, however, on this diagram with incomplete expansion, the area NGL represents the loss of effective energy due to the necessarily early opening of the exhaust valve at N.

Finally, due to the throttling effect and the resistance of valve and exhaust passages, the real back pressure $p_b'$ is always more or less in excess of the exhaust pressure $p_b$, or the pressure in the zone, atmosphere, condenser, or receiver into which the steam is exhausted, and the exhaust line is KM, where M is the intersection of KM with the compression curve UF produced. The area ELKM thus represents the loss of available energy due to back pressure resistance.

**87. Limit of Useful Expansion.**—Cylinder condensation and excessive variation of piston load place a restriction on the expansion ratio in the cylinder of the contra-flow type. In addition to these two influences the work done in overcoming back pressure and frictional resistance has also to be considered in the cases of both simple and compound engines of all classes.

The proportional loss due to work done against the back pressure increases directly as the expansions increase, while the gain due to increased expansion is a gradually decreasing one. A limit is thus reached beyond which further expansion would involve a loss.

This is well illustrated by Fig. 77, which is a modification of a diagram by Willans.

A scale of volume swept by the piston, or of expansion ratio ($r$), is laid off on the axis OX, and a scale of work (E) done per ft.$^3$ of piston displacement on the axis OY. It is assumed here that the engine has no clearance and the expansion is hyperbolic. With a given initial pressure $P_1$ and volume $V_1$ (to cut-off), the value of the absolute work done by the steam on one side of the piston during the forward stroke is $E_e = P_1V_1(1+\log_e r)$. Values of $E_e$ calculated for $r=1$, 2, 3, etc., are set up at the corresponding values from OX, and the positive work curve K$mf$ is obtained.

The net work done on the piston face up to any point of the stroke is, however, the difference between the absolute positive work done by the steam on one side and the absolute negative work done by the constant back pressure on the other side, plus the negative work of the frictional resistance.

If $P_b$ is the constant back pressure and $P_f$ a constant additional back pressure equivalent to the frictional resistance (in lb./ft.$^2$ of piston area), then the negative work done during a piston displacement of V ft.$^3$ is $E_b'=(P_b+P_f)V=P_b'V$, where $P_b'$ is a virtual back pressure.

Values of $E_b'$, calculated for displacements corresponding to $r=1, 2, 3$, etc., when set up from OX, give the straight line O*e* passing through the origin O in the case of a non-condensing engine.

If there were no friction then $P_f=0$, and the line would be O*d*.

It will be noted that as the piston displacement and hence the

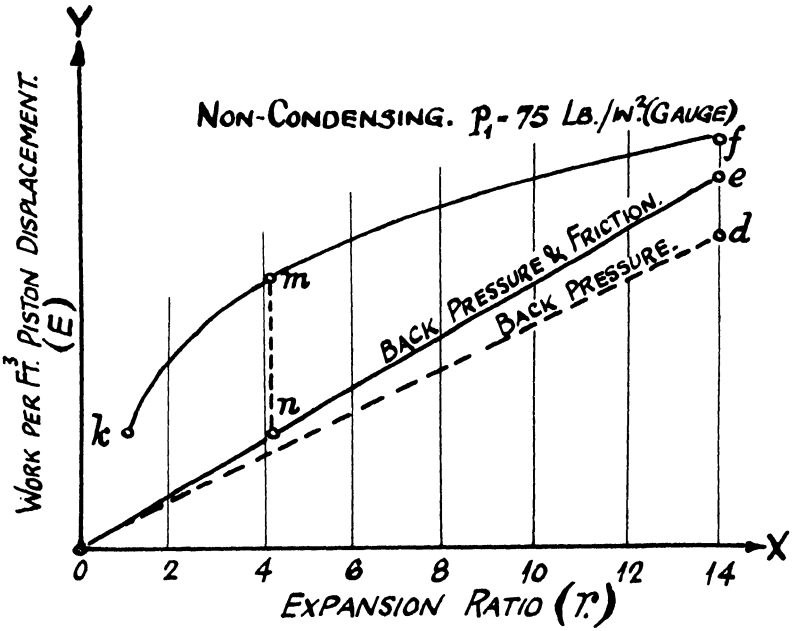



FIG. 77.

expansion ratio increases, with a given cylinder, the vertical intercept between the positive and negative work curves, which represents the net energy expended on the piston, increases to a limiting value, and then decreases.

For the particular initial pressure chosen here in the case of the non-condensing engine, this limiting value is *mn* where the expansion ratio is slightly in excess of 4.

The limiting ratios of expansion given by Willans were as follows:

$$\text{Non-condensing engines } r=\frac{p_1}{25}$$

$$\text{Condensing engines } r=\frac{p_1}{14}$$

where $p_1$=initial pressure in lb./in.$^2$ abs.

**88. Calculation of Indicated Horse-power.**—The area of the indicator card BANLEF (Figs. 69 and 76) represents the work done on *one* side of the piston during a revolution. It is equal to the product of the mean ordinate and the length of the diagram. Since the ordinates represent pressure per in.$^2$ of piston area, the mean ordinate is the mean effective pressure during the stroke. The diagram length represents the volume swept out by the piston during the stroke, or stroke volume.

Let $p_{e1}$=mean effective pressure in lb./in.$^2$
$V_o$=stroke volume in ft.$^3$

then the work done by the steam on the piston in one stroke is—

$$E_1=144p_{e1}V_o \text{ ft.-lb.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

Let $A_1$=piston area in in.$^2$
L=stroke in ft.

$$\text{Then } V_o=\frac{A_1L}{144} \text{ ft.}^3 \quad \text{Substitute in (5)—}$$

$$\text{and } E_1=p_{e1}A_1L \text{ ft.-lb./stroke} \quad . \quad . \quad . \quad . \quad . \quad (6)$$

Let N=rotational speed in revs./min., then work done on one side of the piston is given by—

$$E_1N=p_{e1}LA_1N \text{ ft.-lb./min.} \quad . \quad . \quad . \quad . \quad . \quad (7)$$

By definition 1 horse-power=33,000 ft.-lb./min., hence, dividing by this rate, the indicated horse-power developed on one side of the piston is—

$$IHP_1=\frac{NE_1}{33000}=\frac{144p_{e1}V_oN}{33000}=\frac{p_{e1}LA_1N}{33000} \quad . \quad . \quad . \quad (8)$$

Equation (8) gives the indicated horse-power for a single acting engine.

Since $C_1=LA_1/33{,}000$ is constant for a given engine the horse-power can also be expressed by—

$$IHP_1=C_1p_{e1}N \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

where $C_1$ is the "cylinder constant" for *one* side of the piston. When the steam acts on the other side of the piston the engine is double acting. As a rule the mean effective pressure $p_{e2}$ for this side is not the same as $p_{e1}$ for the other side; and unless there is a tail rod the piston area $A_2$ is less than $A_1$. Here the cylinder constant is—

$$C_2=LA_2/33{,}000$$

and the horse-power developed on this side of the piston is—

$$IHP_2=C_2p_{e2}N \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (10)$$

For the double acting engine, therefore, the total amount of indicated horse-power is—

$$IHP=(IHP_1+IHP_2)=(C_1p_{e1}+C_2p_{e2})N \quad . \quad . \quad (11)$$

This is the accurate method of calculation; but, as a rule, a simpler process is adopted, which is sufficient for most practical purposes. In this case the mean of the two mean effective pressures is taken, and the mean area of the two sides of the piston.

Thus, $p_e=\frac{p_{e1}+p_{e2}}{2}$, and if $a$=area of the piston rod in in.$^2$ the mean area is—

$$A=\frac{A_1+A_2}{2}=\frac{(A_1+A_1-a)}{2}=\left(A_1-\frac{a}{2}\right). \quad . \quad . \quad (12)$$

Since these quantities are now assumed to be the same for each side of the piston the total indicated horse-power of the double-acting engine is given to a first approximation by—

$$IHP=\frac{2p_eLAN}{33000}=Cp_eN \quad . \quad . \quad . \quad . \quad . \quad (13)$$

where the cylinder constant is $C=\frac{2LA}{33000}$. Alternatively they can be written—

$$IHP=p_eSA \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (14)$$

where $S=2LN$=mean piston speed.

**89. Measurement of Mean Effective Pressure from the Diagram.**—Two methods may be adopted to find the mean ordinate or mean width of the diagram: (1) by ordinates; (2) by integration of the diagram area by planimeter.

(1) As shown in Fig. 78, two verticals *ca* and *db* are drawn at right angles to the atmospheric line touching the diagram at the ends. The base *cd* is then divided into 10 or more equal parts and the (dotted) ordinate lines are drawn across the diagram, as shown, at the mid-points of these divisions. The pressure values represented by these intercepts can be measured to the scale of the diagram spring, and tabulated as shown below the diagram.

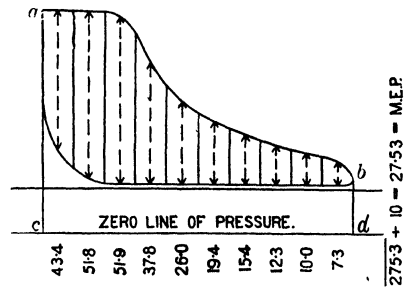



FIG. 78.

The sum of these pressures divided by the number gives the mean effective pressure, which in the case illustrated is 27·53 lb./in.$^2$

This process is troublesome and there is the "personal" error in reading each pressure value. The mean ordinate can be more easily and accurately determined by summing the ordinates on the edge of a strip of paper, measuring the total length thus obtained, and dividing it by the number of ordinates. This mean length multiplied by the scale of the spring gives the mean effective pressure.

(2) A still more accurate result is obtained by finding the diagram area and dividing it by the length. This area is easily and rapidly ascertained by means of a planimeter, which is usually of the Amsler type. The pointer of this instrument is traversed over the diagram outline and the area of the diagram is read on the scale of a roller, the movement of which depends on the path of the pointer.

Let $a$=diagram area in in.$^2$; $l$=diagram in length in ins.; $s$=scale of spring in $\frac{\text{lb/in.}^2}{\text{inch}}$

$$\text{Then } p_e=\frac{a}{l}s \text{ lb./in.}^2 \quad . \quad . \quad . \quad . \quad . \quad (15)$$

**90. Rotational Speed** (N).—In the case of a steam engine the number of revolutions it makes in a given time represents the number of working cycles in this time, if single acting, and half the number if double acting.

This number divided by the time interval gives the mean rate of rotational speed N, when the speed is not constant during the interval. It is always expressed in revolutions per minute. The number of revolutions can be measured by some form of revolution counter, which may be either of the rotary or reciprocating type. This should be positively driven from the engine shaft or a reciprocating part, and not by a belt. A tachometer reading should not be used for calculation of horse-power.

When during the period of an engine trial the rotational speed varies appreciably, indicator diagrams should be taken at frequent intervals, and the mean rotational speeds, by counter, used for each interval. The indicator diagram should be taken at the middle of an interval, a convenient time for which is ten minutes.

For an accurate determination of the horse-power during the trial the power developed during each interval should be calculated for each side of the piston by Equations 9 and 10, and the mean of the series of powers should be taken. This work can be most expeditiously done in tabular form as shown in the illustration given below of the heading of an Indicated Horse-Power log sheet.

| 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 |
|---|---|---|---|---|---|---|---|---|
| | | | | Mean Effective Pressure | | Indicated Horse-power | | |
| Time | Counter Reading | Difference $d$ | Revs./min. $N=\frac{d}{t}$ | Head $p_{e_1}$ | Crank $p_{e_2}$ | Head $I_1$ | Crank $I_2$ | Total $(I_1+I_2)=I$ |
| | | | | | | | | |

$t$=time interval in minutes; Cylinder constant $C_1=$ , $C_2=$ .

The correct value of the mean indicated horse-power is obtained by taking the mean value of I in Column 9. It should not be calculated from the mean values of columns 4, 5, and 6. If the power, as is often the case, is calculated to the first approximation by equation (13), only one column is required for $p_e$ and one for I.

The maximum rotational speed permissible with a given engine is conditioned by the type of valve gear employed. Engines governed by trip gears, Corliss or drop-valves cannot be run much above 100 rev./min. Small slide-valve engines may be run up to 400 rev./min. The usual rate for medium-sized engines is from 150 to 200 rev./min. The vertical inverted closed type may run up to 500 rev./min.

**91. Piston Speed** (S).—The mean piston speed, given by S=2LN, varies from 400 ft./min. to 1000 ft./min. according to the type of engine.

For ordinary slide and piston-valve engines it may vary from 300 to 500 ft./min. For high-class Corliss or drop-valve engines, with relatively long strokes, from 500 to 700 ft./min. For marine engines it varies from 600 to 800 ft./min., and for locomotives from 900 to 1200 ft./min. For high-speed electric engines from 500 to 1000 ft./min.

The piston speed of pumping engines is low, owing to the nature of the service. It varies from 150 to 300 ft./min.

**92. Diagram Factor.**—As already indicated in Art. 84, the theoretical mean effective pressure or the mean positive pressure acting on the piston during a cycle, in the three cases stated, is given by the quantity in the square brackets of Equations 2, 3, and 4. These are, in lb./in.$^2$, for—

No clearance
$$p_m=\left[p_1\left(\frac{1+\log_e r}{r}\right)-p_b\right] \quad . \quad . \quad . \quad . \quad . \quad (16)$$

Clearance and no compression
$$p_m'=\left[p_1\left\{\frac{1}{r}+\left(\frac{1}{r}+c\right)\log_e\frac{1+c}{\frac{1}{r}+c}\right\}-p_b\right] \quad . \quad (17)$$

Clearance and compression
$$p_m''=\left[p_1\left\{\frac{1}{r}+\left(\frac{1}{r}+c\right)\log_e\frac{1+c}{\frac{1}{r}+c}\right\}-p_b\left\{1-z+(z+c)\log_e\frac{z+c}{c}\right\}\right] \quad (18)$$

The actual mean effective pressure as calculated from the indicator diagram is always less than the mean effective pressure of the particular theoretical diagram taken as a standard of reference.

If this is denoted by the general symbol $p_e$, then, according to the three values of theoretical mean effective pressure given by Equations 16, 17, and 18, three values of the ratio of actual and

theoretical mean pressure, which was originally termed the "diagram factor" by Prof. Unwin, are obtained, thus—

$$f = \frac{p_e}{p_m} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (19)$$

$$f' = \frac{p_e}{p_{m'}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (20)$$

$$f'' = \frac{p_e}{p_{m''}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (21)$$

The calculation of the probable mean effective pressure for any proposed type of engine is, at the best, a very tentative process. It has to be based on a factor determined from experience with the type, and calls for the exercise of a considerable amount of practical judgment on the part of the designer. Under these circumstances these refinements of $f'$ and $f''$ are hardly justified, when clearance and compression effects can be allowed for in the choice of $f$, the figure which is commonly used.

This factor, however, is looked on with disfavour by some engineers, who consider that the correction should apply only to the mean absolute pressure $p_1\left(\frac{1+\log_e r}{r}\right)$; and they deduce the probable mean effective pressure from—

$$p_e = \mathrm{F}\left\{p_1\left(\frac{1+\log_e r}{r}\right)\right\} - p_b \quad . \quad . \quad . \quad . \quad (22)$$

In this case, F is slightly greater than $f$, for the same value of $p_e$, the relation between $f$ and F being given by—

$$\mathrm{F} = f + \frac{p_b(1-f)}{p_1\left(\frac{1+\log_e r}{r}\right)} \quad . \quad . \quad . \quad . \quad . \quad (23)$$

It is a matter of opinion whether the one method or the other is preferable. Equally satisfactory results should be got, provided the $f$ factors derived from typical engines in the ordinary way are not used instead of the F factors in Equations 22 and 23.

The value of the diagram factor is affected by various influences. It depends on the type, whether simple or compound, unjacketed or jacketed; on the initial pressure and number of expansions; on the compression; and also on the back pressure.

With regard to the back pressure, C. H. Wingfield, in the discussion of the Willans Engine Trials* in 1893, pointed out that for any type of engine there was some "virtual back pressure" $p_b'$, which if used to calculate F from actual mean pressure values (Equation 22), gave a diagram factor which was practically independent of the speed and not greatly affected by variation in the number of the expansions.

* *Proc. Inst. C. E.*, vol. cxiv., 1892–1893.

This virtual back pressure can be obtained, for any engine, by plotting the actual mean effective pressure values obtained from a series of trials at varying loads against the mean absolute pressure values $\left(p_1\frac{(1+\log_e r)}{r}\right)$. The locus of the points is found to be a straight line, which crosses the horizontal axis to the right of the origin and intersects the vertical axis at the value of the virtual back pressure $p_b'$ (measured on the M.E.P. scale).*

Considering the number of variable factors which influence the relation between the actual and the theoretical mean effective pressures, it is not possible to do more than indicate the probable limits of $f$ (or F), for the various types of reciprocators. Compound or multistage engines are considered in Chapter VII; but for the simple unjacketed type of slide or piston-valve engine and simple Corliss, which may run at initial pressures from 80 to 120 lb./in.$^2$ abs., at moderate piston speeds, the diagram factors may be taken between the values given below—

| | $f$ | $f'$ |
|---|---|---|
| Simple slide or piston-valve engine . | 0·8–0·9 | 0·7–0·8 |
| „ „ Corliss-valve engine . | 0·85–0·95 | 0·8–0·9 |

Average values of the probable M.E.P. for the simple types are as follows:

| | | | |
|---|---|---|---|
| Boiler pressure . . . . | 80 | 100 | 120 lb./in.$^2$ gauge |
| M.E.P. . . . . | 35 | 40 | 45 non-condensing. |
| M.E.P. . . . . | 30 | 34 | 40 condensing. |

These refer to conditions of normal load; a 25 per cent. overload can be obtained, either by increasing the initial admission pressure or lengthening the cut-off, according to the type of valve gear used. The M.E.P. values are thus proportionately increased. The mean admission pressure is always less than the boiler pressure in an expansion governed engine.

*Example* 2.—A simple non-jacketed slide-valve engine is required to develop a maximum of 15 I.H.P. at 200 rev./min. It may be assumed, allowing for wiredrawing on admission, that the mean admission pressure is 100 lb./in.$^2$ abs. If the valve is set to give a constant cut-off at 0·4 stroke, and a diagram factor $f=0{\cdot}8$ is assumed, find a suitable cylinder diameter when the stroke$=1{\cdot}50\times$diameter. Take the back pressure at 18 lb./in.$^2$ abs.

$$p_m=p_1\left(\frac{1+\log_e r}{r}\right)-p_b \qquad \begin{matrix} p_1=100 \\ p_b=18 \\ r=2{\cdot}5 \end{matrix}$$

$$=100\left(\frac{1+2{\cdot}3\times 0{\cdot}9151}{2{\cdot}5}\right)-18=(76{\cdot}6-18)$$

$$=58{\cdot}6 \text{ lb./in.}^2$$

* For a later discussion of this point, by Wingfield, see *Engineering*, 20 October, 1893.

$p_e = f p_m = 0{\cdot}8 \times 58{\cdot}6 = 46{\cdot}88$ lb./in.$^2$ I.H.P.$=15$, $L=1{\cdot}5D$, $N=200$

$$\text{I.H.P.} = \frac{2p_e LAN}{33000}$$

$$A = \frac{15 \times 33000}{2 \times 46{\cdot}88 \times 1{\cdot}5D \times 200} = \frac{\pi}{4}D^2$$

$$\therefore\ D^3 = \frac{4 \times 15 \times 33000}{2\pi \times 46{\cdot}88 \times 1{\cdot}5 \times 200} = 224 \text{ in.}^3$$

$$\therefore\ D = \sqrt[3]{224} \doteqdot 6 \text{ ins.}$$

$\therefore$ size of cylinder, 6 ins. diam. $\times$ 9 ins. stroke.

The probable economical load of this engine would run from 11 to 12 I.H.P., or allowing a mechanical efficiency (Art. 93), of say, 0·92, the net or effective horse-power would run from 10 to 11 at the rated speed.

*Example* 3.—If the simple engine (Example 2) is to have a clearance, 12 per cent. of the stroke volume, find the value of the modified diagram factor $f'$ to give the actual mean effective pressure of 46·88 lb./in.$^2$

Here $r=2{\cdot}5$, $c=0{\cdot}12$, $\frac{1}{r}=0{\cdot}4$, $\left(\frac{1}{r}+c\right)=0{\cdot}52$, $p_1=100$, $p_b=18$.

$$\text{Then } p_m' = p_1\left\{\frac{1}{r}+\left(\frac{1}{r}+c\right)\log_e\frac{1+c}{\frac{1}{r}+c}\right\}-p_b$$

$$= 100\left(0{\cdot}4+0{\cdot}52\times 2{\cdot}3\log\frac{1{\cdot}12}{0{\cdot}52}\right)-18$$

$$= 100(0{\cdot}4+0{\cdot}52\times 0{\cdot}769)-18$$

$$= (80-18) = 62 \text{ lb./in.}^2 \text{ (against 58·6 lb./in.}^2 \text{ with no clearance)}$$

$$\therefore\ f' = \frac{p_e}{p_m'} = \frac{46{\cdot}88}{62} = 0{\cdot}76.$$

*Example* 4.—If the engine (Examples 2 and 3) has its compression point at 80 per cent. of the stroke, find the modified diagram factor $f''$ corresponding to the actual M.E.P. of 46·88, when both clearance and compression are allowed for.

Here $z=0{\cdot}2$, $c=0{\cdot}12$, $p_1=100$, and $p_b=18$,

$$\text{and } p_m'' = \left[p_1\left\{\frac{1}{r}+\left(\frac{1}{r}+c\right)\log_e\frac{1+c}{\frac{1}{r}+c}\right\}-p_b\left\{1-z+(z+c)\log_e\frac{z+c}{c}\right\}\right]$$

$$= \left[80-18\left(1-0{\cdot}2+0{\cdot}32\times 2{\cdot}3\log\frac{0{\cdot}32}{0{\cdot}12}\right)\right]$$

$$= 80-18\times 1{\cdot}1139 = (80-20) = 60 \text{ lb./in.}^2, \text{ against 62 lb./in.}^2 \text{ with clearance only.}$$

$$\therefore\ f'' = \frac{p_e}{p_m''} = \frac{46{\cdot}88}{60} = 0{\cdot}781.$$

**93. Brake Horse-power.**—The brake horse-power of an engine is the time rate of expenditure of energy against the external resistance, or load on the engine. The difference between it and the gross or indicated horse-power gives the loss of energy due to mechanical friction of the moving parts of the engine. The ratio of the brake to the indicated horse-power is consequently termed the mechanical efficiency of the engine, and is expressed by—

$$\eta_m = \frac{B}{I} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (24)$$

where B=brake horse-power
I=indicated horse-power.

According to Lucke,* $\eta_m$ is independent of the speed, and may be expressed as a function of the mean effective pressure $p_e$ lb./in.$^2$

Thus $$\eta_m = 1 - K_1 - K_2/p_e \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (25)$$

where $K_1$ is a constant varying from 0·02 to 0·05.
$K_2$ ,, ,, ,, ,, 1·3 to 2·0.

With high-class engines $\eta_m$ runs from 0·9 to 0·96 ; with less efficient engines it may fall to 0·85.

Brake horse-power is measured by some form of dynamometer. This may be either an absorption or a transmission machine. In land work the absorption type is almost exclusively used. In marine work it is impracticable, and a transmission dynamometer, in the form of either a torsion meter or a thrust meter, has to be employed to calculate the shaft horse-power. In the case of a locomotive a special dynamometer car is used to automatically register the " draw-bar pull " from which the net or " draw-bar horse-power," can be calculated.

**94. Absorption Dynamometers.**—These may be roughtly classified under Prony and rope brakes ; electrical brakes ; fan brakes ; and hydraulic brakes.

The Prony brake (Fig. 79), sometimes used on engines of small power, consists of two blocks of hard wood 1 and 2 clamped on the brake wheel 3. The lower block has an arm 4 carrying an adjustable weight 5 at the end and balanced by a counterweight 6. The load is adjusted by means of the wheel nut and spring 7 until the required speed is obtained and the end of the arm bears on the stop 8. The end weight 5 is then adjusted till the arm " floats " between the stops 8 and 9. It is advisable, as an additional precaution, to fit a safety catch at the weight carrier, in case the blocks may seize and carry round the arm and weights. A mixture of tallow and graphite may be used to lubricate the wheel rim. The constant friction torque at the wheel is given by T=WL, where W=weight in lb. and L=distance in ft. from the weight to the shaft centre. Instead of the upper block a steel band fitted with a series of hard-wood

* *Engineering Thermodynamics.*

friction blocks may be employed. A brake of the Prony type tends to be jerky, requires lubrication, is difficult to regulate, and its capacity is very limited. It is, however, cheap and easily constructed and applied. There are several variants of the type with the strap and blocks, on which further information can be obtained in the series of articles quoted in the footnote.*

When the wooden blocks and lever arm are replaced by a rope, embracing all or part of the wheel circumference, a much more serviceable brake is obtained.

An illustration of an " all-round " rope brake, as fitted by the writer to a small experimental steam engine, is shown in Fig. 80. The disc brake wheel 1, keyed to the engine shaft, has a channel shaped rim 2 to which water is supplied through the service pipe 3,

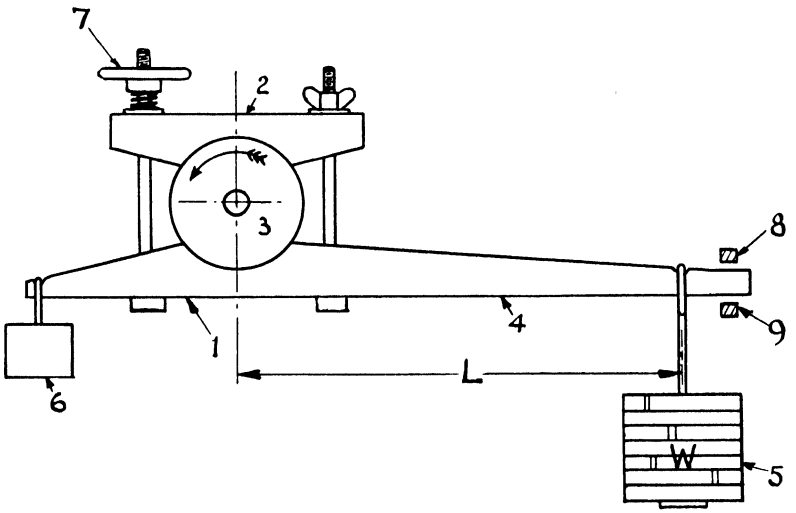



FIG. 79.

and distributed to each side of the disc through the holes 4. After being carried round in the rim it is scooped out by the pipe 5. In this way the frictional heat is carried away by the water and the wheel rim kept cool. The rope 6 is formed in two lengths joined at the ends, and is passed completely round the circumference of the wheel. The parallel lengths are kept apart and on the rim by several ⌐ shaped wooden cleats 7. One end of the rope is attached to a weight-carrier 8, on to which a series of weights is threaded, giving a total load of W lb. The other end of the rope is attached to the hook of a spring balance 9.

The difference between the brake load W and the effective spring

* " Commercial Dynamometers," by P. F. Foster, *Mech. World*, 26 April, 1918 *et seq.* See also *Proc. I. Mech. E.*, April 1908. Appendix to Paper, " Gas Engine Governing," by J. Atkinson.

balance reading $w$ is the frictional force $F=(W-w)$ at the circumference of the brake wheel. The line of action of this force is along the centre line of the rope, so that the effective radius is $R_1=R+d/2$ ft., where R=brake wheel radius in feet and $d$=diameter of rope in feet. The friction torque, in this case, is $T=FR_1$ ft.-lb. The size of rope may run from $\frac{5}{8}$ to 1 in., depending on the brake load. The

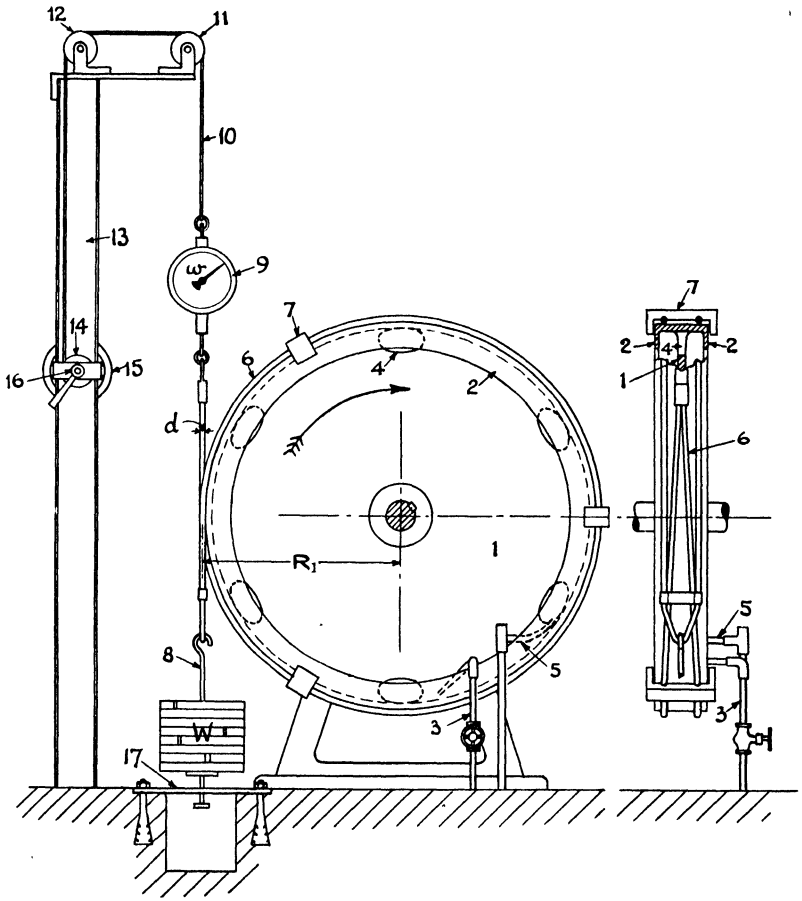



FIG. 80.

rope always stretches more or less, especially when new, but this is immaterial as long as the load W is kept well clear of the floor. The height adjustment can be made by raising the end of the rope at the spring balance.

In the case shown the balance is not attached to a fixed point, but to a cord 10, passing over two small guide pulleys 11 and 12, and down a channel-shaped standard 13, fixed to the floor. It is

wound round a small pulley 14 by means of a hand-wheel 15, which in turn is locked by the handle-nut 16. By this arrangement not only can the load be adjusted to any desired height, but it can be instantly lowered to the floor should it become necessary to suddenly throw the load off the engine.

A brake of this type should always be provided with a safety stop, so that should the rope "seize," the weights cannot be hurled round the wheel. In this case a hole is cut in the floor to take the lower half of the weight-carrier 8, which is passed through a plate 17 bolted to the floor by substantial rag bolts. A nut and washer at the end of this rod engage with the keeper plate, if the weights are pulled up too far. If the plate is properly bolted down, the weights can be effectively prevented from rising under a heavy frictional pull by the rope.

Some engineers attach a small link chain to the weight-carrier and anchor the other end to the floor. This is a bad arrangement. The chain, according to its size and the length suspended from the carrier, adds an indeterminate amount to the brake load W, which may appreciably affect the accuracy of the calculated B.H.P.

The arrangement shown here is very suitable for small experimental engines of from 10 to 30 B.H.P., such as are used in laboratories. In the case of a commercial test, however, it is seldom that a special brake wheel can be applied, and the engine fly wheel has to do duty as the brake wheel and a temporary rig has to be provided to carry the balance.

In such a case the duration of a test is controlled by the safe limits of temperature of the wheel rim, since the major part of the frictional heat is stored in the metal, and only a small proportion carried away by convection and radiation to the air. When the engine is at rest the spring balance pointer does not read the zero value, unless specially set back to this. It registers the weight of the suspension hook and the vertical part of the rope between the balance and wheel circumference. This reading has therefore to be deducted from the dial reading when the engine is run on load in order to obtain the correct balance reading $w$. Similarly the weight of the carrier rod must be added to the sum of the brake weight in order to obtain the correct value of the brake load W.

Instead of a dead load and spring balance, a platform weighing machine may be used to register the friction force F at the wheel rim. One arrangement of this type is shown in Fig. 81. The weighing machine 1 has an upright 2 bolted to the platform 3. This upright carries an adjusting screw 4 to which is attached a hook 5. The screw is raised and lowered by means of the hand-nut 6. About mid-height of the standard a small pulley 7, operated by a hand-wheel 8, which can be locked by a handle-nut 9, is mounted. The rope is passed round the whole circumference of the wheel. One end is attached to the hook 5. A light cord attached to the other end is passed round a small guide pulley 10 and carried

up to the straining pulley 7. The downward pull W of the rope on 5 is transmitted to the platform. The pull $w$ on the other end of the rope is transmitted upward through the cord wound on pulley 7, so that the net load registered on the steel-yard of the weighing machine, when the jockey weight is adjusted to keep the beam floating between the stops, is the difference of these forces, or the friction force at the wheel rim, $F=(W-w)$. The friction torque is $FR=T$.

This arrangement obviates the handling of heavy weights and gives a direct reading of the net force. On the other hand, it is inferior to the directly loaded rope brake, which automatically allows for rope stretch and maintains a constant load on the wheel.

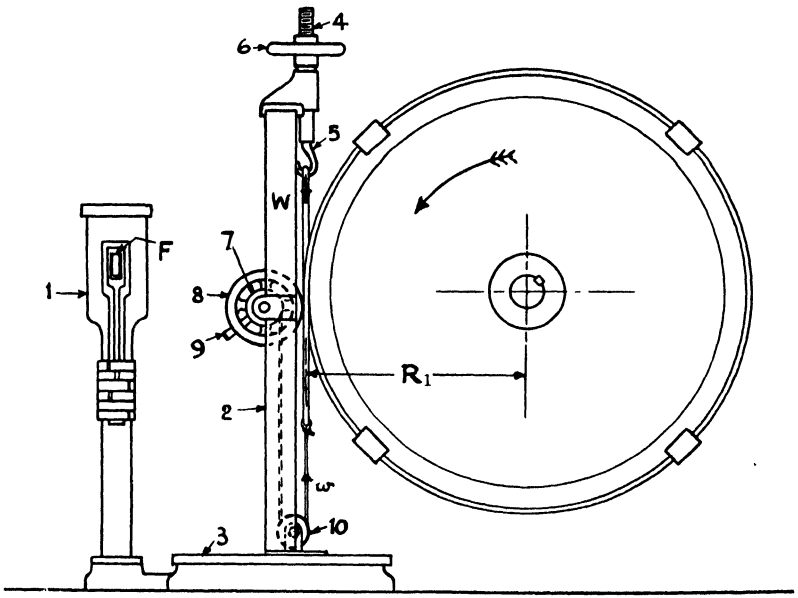



FIG. 81.

It requires the constant attention of an observer throughout a trial to adjust the straining screw 4 and take up cord stretch, so that the beam of the machine may be kept floating between the stops. The outfit is also more expensive than that of the ordinary rope brake.

Either cotton or hemp rope may be used, but the latter is preferable, as it gives less trouble by stretching.

In the case of an electrical brake the resistance of the magnetic flux to the rotation of the armature of a dynamo replaces the frictional force F at the circumference of the brake wheel.

An arrangement of electrical brake fitted to an enclosed high-speed steam engine running normally at 500 r.p.m. is shown in Fig. 82.

The frame 1 of a direct current dynamo carrying the field magnets 2 is mounted on ball bearings, so that it can rotate about the shaft axis. It is provided with a long arm 3, which is balanced by an adjustable balance weight 4. The brake load W is hung at the end of the arm. This end can be pulled upward by a cord through the medium of a spring balance 5. This can be adjusted, when the engine runs under load so that the end of the arm is kept floating horizontally. A safety stop is fitted at 6. When an electrical resistance is put in the circuit the armature 8 of the dynamo tends to rotate the frame, under a torque $T=(W-w)L=FL$, where W and $w$ are the brake load and spring balance reading in lb. and L the lever arm in ft. to the centre of the shaft.

This is a heavy and expensive outfit for any but small powers, such as can be quite well handled by the ordinary rope brake, unless the

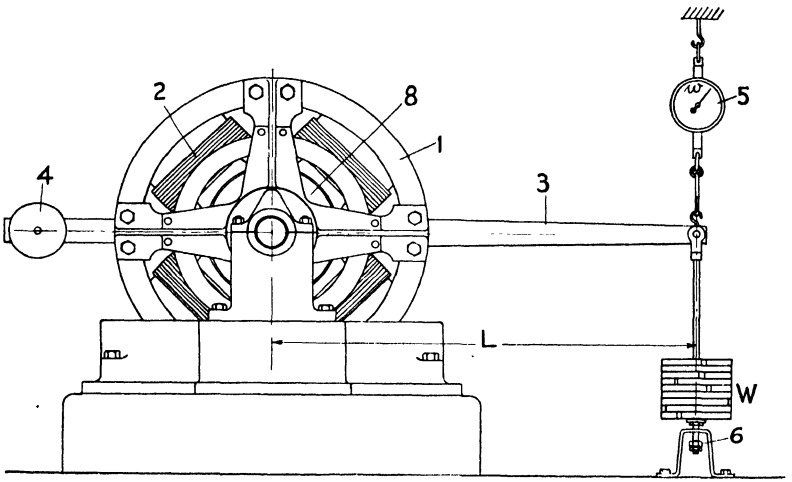



FIG. 82.

rotational speed is very high, a condition which in general does not arise in the case of small steam engines, although it is quite common with steam turbines.

In the case of a large output, where a rope brake is impracticable and the electrical method of absorption is employed, the electrical output of the dynamo has to be measured and correction made for the dynamo losses. The accuracy of the engine output depends on the correctness of the electrical instruments employed and dynamo efficiency. The temperature of the windings affects the electrical losses, and hence the efficiency.

The fan brake dynamometer consists of two equal arms, which are clamped to the engine shaft and carry two rectangular steel plates. The air resistance to the motion of these plates, when the engine is run at speed, is equivalent to the friction force F at the

brake wheel. This type was originally developed for the purpose of getting a load on high-speed engines over a long endurance test. It can, however, be used to give a rough approximation to the B.H.P. output, although it is not as reliable as the other absorption types discussed. It is really a two-bladed fan, and has to be calibrated by first driving it at a series of speeds by an electric motor, and finding the inputs after correction for motor efficiency. The power absorbed by a brake of this type is proportional to the cube of the rotational speed and the density of the air in which it rotates. The accuracy of the B.H.P. obtained is in the first place dependent on the correctness of the calibration curves of the makers. It may be seriously affected by variation of temperature of the air, by draughts, by proximity of buildings which affect air currents, etc. While in the other cases the load can be changed when the engine is running, it cannot in this one until the engine is stopped and the plates are changed. There is also a considerable element of danger, from the risk of fracture of the fixing bolts under the centrifugal force on the plates. This brake also produces a deafening noise.

95. For small, moderate, and large outputs, running up to several thousand brake horse-power, the hydraulic dynamometer or water brake invented by W. Froude is now generally used. It is made in several designs by Heenan and Froude. The makers claim the following advantages for this type of brake.

The starting resistance is practically negligible and rises rapidly with the speed. It has a great range of output: the maximum power capacity may be one hundred times greater than the minimum at the same speed. The inertia of the rotor is slight. The brake operates silently. The load can be altered when the brake is running, and taken off quickly without exertion. It is suitable for very long non-stop running at full load. It exerts a pure torque resistance and imposes no load on the bearings.

A diagram of the arrangement of the latest design of this dynamometer for medium and large powers is shown in Fig. 83. In earlier designs the torque was measured by heavy loose weights adjustable on a long lever arm. When an attempt was made to economise space by shortening the lever these weights often became impracticably heavy. The new design, which embodies a system of compound levers actuating the steel-yard of a weighing machine, gets over this difficulty.

A casing 1 which can swing on roller bearings about the shaft axis is connected at 2 and 3, by rods 4 and 5 having articulated joints, to the end of levers 6 and 7, which are fitted with roller bearings. Lever 6 is pivoted at 8 on a short link attached to the base plate, and lever 7 is pivoted at 9, on a block also fixed to the base plate. The outer end of the lever 6 is jointed at 10 to a forked lever 11, which in turn is jointed to the outer end of lever 7 at 12. The other end 13 of the forked lever 11 is connected by the vertical

rod 14 to the steel-yard 15 of a weighing machine. The impeller, which is mounted on the shaft, is provided with cup vanes. Similar vanes are provided at the side of the casing, so that when the casing is filled with water and the impeller is rotated, the action of these cups on the water produces a torque which tends to turn the casing about the axis. With counter-clockwise rotation, as indicated by the directional arrow, rod 5 pulls upward on lever 7, while rod 4 pushes down on lever 6. The result is that, as indicated by the

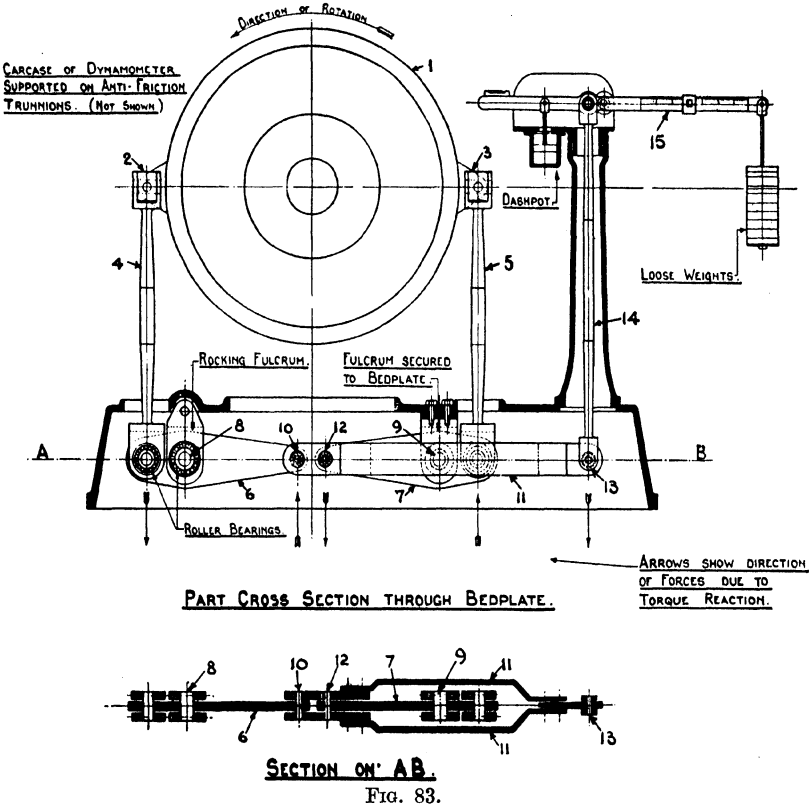



FIG. 83.

arrow lines, lever 6 applies an upward force to the forked lever 11 at 10, while lever 7 applies a downward force on it at 12. These forces constitute a couple or torque which is balanced by the upward pull of the steel-yard at 13, acting through the length of lever between 13 and 12 as an arm. This system of levers is the equivalent of a lever arm on the casing having an effective length 600 times as great as the radius from the shaft axis to the point 2 or 3.

The result is that a very large torque can be measured by very small weights on the steel-yard, instead of by enormously heavy ones on a direct arm.

These dynamometers can be constructed to absorb up to 30,000 B.H.P. at speeds from 80 to 150 rev./min. and upwards.

A longitudinal section through this dynamometer is shown in Fig. 84. The casing 1 is provided with elliptical-shaped pockets 2,

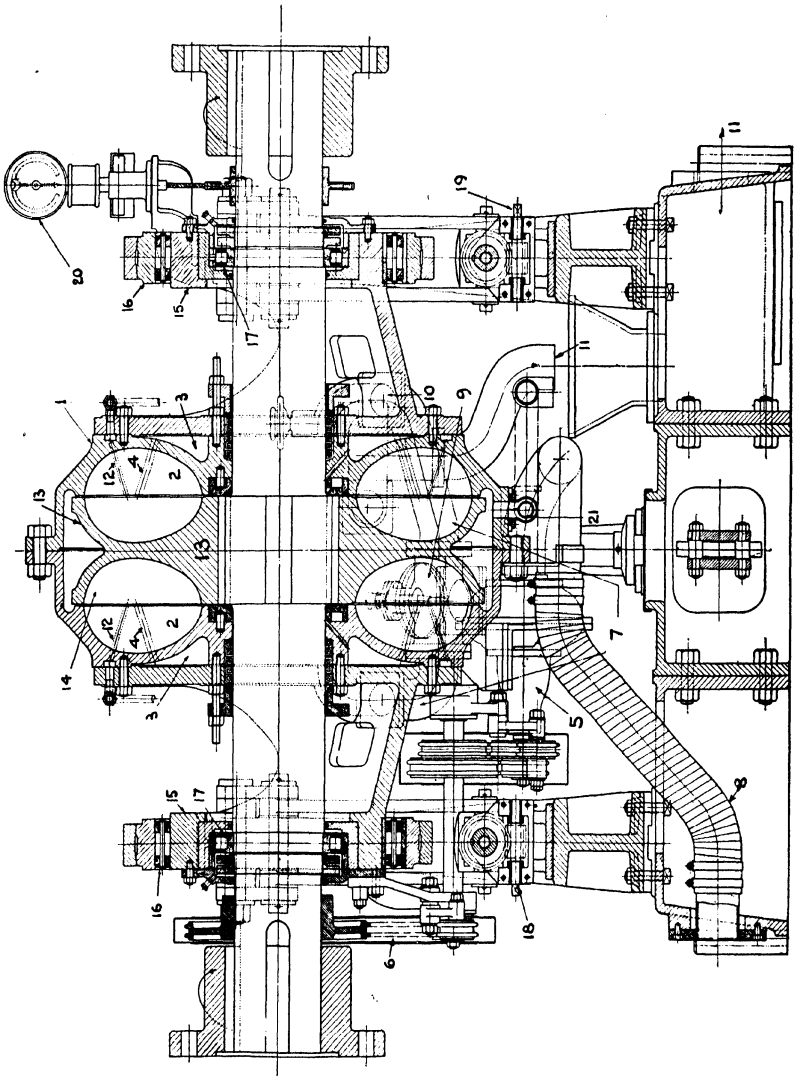



FIG. 84.

supplied with water from the inlet water chamber 3, through water holes 4. The chamber is supplied with water from a centrifugal pump 5, mounted on the side of the casing, and driven by a chain gear 6 from the rotor shaft. The pump discharges the water to

the casing through the distributing pipes 7. The water is supplied to the pump through the flexible pipe 8. An inlet control valve is provided at 9, on the pump discharge, and an outlet control valve at 10. The water flows away through the open-ended pipe 11. Air holes 12 are provided in the casing pockets. The rotor 13 is provided with corresponding pockets 14. When run at speed the water from the rotor pockets is discharged with a high velocity at the periphery into the casing pockets, by which it is returned at reduced velocity at the inner circumference of the rotor pockets next the shaft. The pockets in the rotor and casing form elliptical chambers round which the water courses at high speed, creating vortices which absorb the power of the engine as quickly as it is transmitted. The energy is finally carried away as heat in the circulating water. The air holes 12 in the casing pockets are provided to maintain atmospheric pressure at the centre of the vortices. Although the water pressure at the periphery of each pocket, under running conditions, is high, it is always atmospheric at the centre of the vortex. The water can thus flow freely into the pocket from the casing chamber. The vortices may "cavitate" when there is an insufficient supply of water, and this may course round the peripheries of the pockets in thin films. When such a condition is set up the resistance to rotation is small. With a steady supply of water by the pump, if the outlet valve is partly closed down, the gain in water content and film thickness increases the resistance to rotation. The load can thus be regulated by hand adjustment of the outlet water valve. Closure of the valve increases the load and opening decreases it.

The introduction of the centrifugal pump, the speed of which varies with the speed of the rotor shaft, is a recent improvement. Any change of speed in the dynamometer simultaneously changes that of the pump, which thus automatically delivers water into the power absorbing pockets at a rate dependent on the speed of the prime mover.

If the speed begins to increase and the engine tends to race, the pockets are automatically packed with water, and resistance to motion is thereby increased until the racing tendency is checked. Conversely, with a decrease of speed the resistance is diminished. The introduction of the pump thus ensures great stability of speed under fluctuating load. The casing is carried on the bearing brackets 15 by trunnion roller bearings 16. The shaft is also carried in roller bearings 17. Vertical displacement of the heavy bracket and the whole dynamometer, for shaft adjustment, is effected by a worm and screw gear 18 and 19. The speed is recorded by a tachometer 20. A drain valve is fitted on the casing at 21. A photograph of one of this type of dynamometer, similar to that shown in section although not the same in all details, is shown in Fig. 85.

It has a capacity of 15,000 B.H.P. at 105/225 revs./mins. The vertical pipe on the left is the supply pipe to the pump suction;

the discharge and outlet pipes are on the other side. The horizontal wheel valve, at the circumference of the casing at the shaft level, is the outlet control valve.

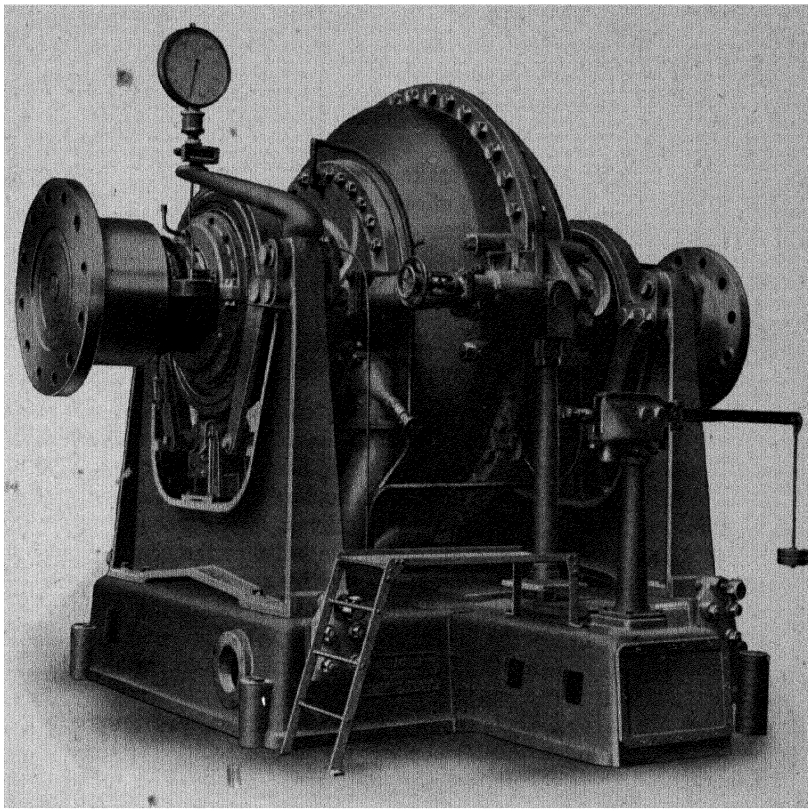


FIG. 85.

**96. Calculation of Brake Horse-power.**—In the foregoing cases of absorption dynamometers when—

T=torque in ft.-lb.
$\omega$=angular velocity of the brake wheel in radians/sec.
B=brake horse-power—

$$B=\frac{T\omega}{550} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (26)$$

$$\omega=\frac{2\pi N}{60} \quad \text{where} \quad N=\text{rev./min.}$$

$$\therefore B=T\times\frac{2\pi N}{33000}$$

Substituting for T in the several cases—

$$\left.\begin{matrix}\text{Prony}\\ \text{Electrical}\end{matrix}\right\} \quad B=\frac{WL\times 2\pi N}{33000}=C_bWN \quad . \quad . \quad . \quad . \quad . \quad (27)$$

$$\left.\begin{matrix}\text{Rope}\\ \text{Platform}\end{matrix}\right\} \quad B=(W-w)\frac{R_1\times 2\pi N}{33000}=C_b(W-w)N \quad . \quad (28)$$

where $C_b=\frac{2\pi L}{33000}$ or $\frac{2\pi R_1}{33000}$ is the brake constant, since L and $R_1$ are constant for the particular brake considered. In the case of the Froude hydraulic brake the effective radius $R_1$ is suitably chosen and the value of the brake constant is given by—

$$C_b=\frac{1}{K} \quad \text{and} \quad \text{B.H.P.}=\frac{WN}{K} \quad . \quad . \quad . \quad . \quad (29)$$

The value of K is given for each machine by the makers, to suit the particular value of $R_1$ chosen in the design.

**Torsion Meters.**—The power transmitted to the propeller by a marine reciprocating engine or turbine cannot be directly measured, and some form of torsion meter has to be used. It is not proposed to deal with these here in detail. Particulars of various makes can be obtained from the publications mentioned in the footnote.*

According to the laws of elasticity the twist of the shaft is proportional to the torque producing it, hence by measuring the angular displacement of two sections of the shaft at a given distance apart, the torque and hence the power transmitted at a given speed can be estimated. The function of the torsion meter is to register this angular displacement.

There are several types of meter on the market, and they may be classified under three heads: (1) mechanical; (2) electro-mechanical; (3) optical.

The mechanical type is designed to give either a continuous diagram of mean torsion or variation of torsion during a revolution. The electro-mechanical gives mechanical multiplication of torsion with electrical transmission to a distant indicator. The optical registers the torsion by the deflection of a beam of light from fixed and moveable mirrors.

Assuming that the angle of twist is ascertained with a reasonable degree of accuracy by one or other of these forms of meter, the shaft horse-power, S.H.P., is calculable as follows. Referring to Fig. 86, A and B are two discs, each of radius R inches, fixed on the shaft L inches apart. The end points of the radii OE and OF when the shaft rotates idly remain in the same plane.

* Articles on "Commercial Dynamometers," by P. F. Foster, *Mechanical World*, 27 June and 25 July, 1919. Also Sothern's *Marine Steam Turbine*, and "Development and Present Status of Steam Turbines," by E. M. Speakman, *Trans. Inst. Engineers and Shipbuilders in Scotland*, 1907. A "Universal Torsion Meter," *Engineer*, 26 July, 1912.

When the shaft transmits power let the deflection of OE relatively to OF be $\theta$ radians, the radius OE taking the position $OE_1$. The corresponding arc at the radius R is $x$ inches, and hence $\theta=\frac{x}{R}$.

If $f_s$ is the skin shear stress produced in the shaft of diameter

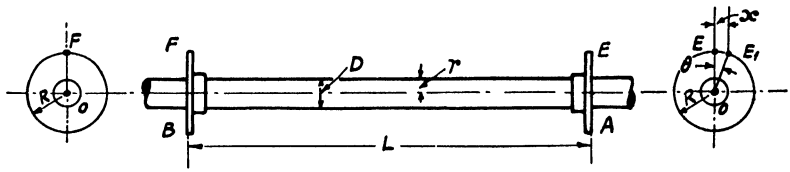



FIG. 86.

D inches by a torque T, and J is the polar second moment of the shaft section, then—

$$\frac{T}{J}=\frac{f_s}{\frac{D}{2}}.$$

Also the rigidity modulus is—

$$G=\frac{f_s}{\frac{r\theta}{L}}=\frac{2f_sL}{\theta D} \quad \text{where} \quad r=\text{shaft radius}=\frac{D}{2}$$

Hence $$G=\frac{TL}{J\theta}$$

For a solid shaft $J=\frac{\pi}{32}D^4$

For a hollow shaft $J=\frac{\pi}{32}(D^4-d^4)$, where $d$ is the internal diameter.

$$\therefore G=\frac{32TL}{\pi D^4\theta} \quad \text{or} \quad \frac{\frac{32}{\pi}TL}{(D^4-d^4)\theta}, \text{ and the torque in ft.-lb. is given by}$$

Solid shaft.

$$T=\frac{\pi}{32}\times\frac{Gd^4\theta}{12L} \text{ ft.-lb.} \quad . \quad (30)$$

Hollow shaft.

$$T=\frac{\pi}{32}\times\frac{G(D^4-d^4)\theta}{12L} \text{ ft.-lb.} \quad . \quad (31)$$

The modulus of rigidity varies slightly for different shafts, and it is desirable that the actual torque for any given angle of twist should be experimentally determined before a shaft is put in the vessel. A torque angle diagram can thus be plotted, and the torque read directly from it, for any recorded value of the angle obtained under service conditions.

Failing any such preliminary adjustment the value of G for the

calculation of the torque may be taken as $11{\cdot}76\times10^6$ lb./in.$^2$ If the shafting is run at N rev./min. the power transmitted is—

$$\text{S.H.P.}=\frac{2\pi NT}{33000} \quad . \quad . \quad . \quad . \quad . \quad . \quad (32)$$

The angle of twist is usually measured in degrees.

If D=external diameter in inches
$d$ =internal diameter in inches
L=length in ft.
$\theta$ =angle of twist in degrees
G=$11{\cdot}76\times10^6$ lb./in.$^2$

the value of the torque in ft.-lb. reduces to

Solid shaft. Hollow shaft.

$$T=\frac{140D^4\theta}{L} \quad \text{or} \quad T=\frac{140(D^4-d^4)\theta}{L}$$

If the reading obtained from the particular meter used gives the value of the arc of twist $x$ at radius R, then $\theta$ degrees$=\frac{57{\cdot}3x}{R}$. Substituting this value in these equations for T, the horse-power, from (32), is given by—

Solid shaft. Hollow shaft.

$$\text{S.H.P.}=\frac{1{\cdot}53D^4Nx}{RL} \quad . \quad (33) \quad \text{or} \quad \text{S.H.P.}=\frac{1{\cdot}53(D^4-d^4)Nx}{RL} \quad . \quad (34)$$

**97. Draw-bar Horse-power of a Locomotive.**—The draw-bar pull which the engine exerts on the train is equal to the effective tractive force at the rail, these forces constituting a couple which has half the diameter of the driving wheel as its arm. If there were no frictional losses of energy in the engine mechanism, the tractive force would have a higher value and consequently the pull on the train would be greater. This larger value is termed the indicated tractive effort, and may be denoted by $T_i$. If the actual tractive effort or draw-bar pull is denoted by T the mechanical efficiency is given by their ratio, or—

$$\eta_m=\frac{T}{T_i} \quad . \quad . \quad . \quad . \quad . \quad . \quad (35)$$

The indicated tractive effort is readily deduced from the indicated horse-power.

Let I =indicated horse-power
$v$ =velocity of the engine in ft./sec.
N =revolution/min. of the engine
$n$ =number of simple cylinders
$d$ =diameter of cylinder in inches
D =diameter of driving wheel in ft.
L =stroke in ft.
$p_e$ =mean effective pressure in lb./in.$^2$

then $$T_iv=550\times I \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (36)$$

The total I.H.P. developed by $n$ simple cylinders is—

$$I=\frac{n\times 2p_e L\frac{\pi}{4}d^2N}{33000} \quad . \quad . \quad . \quad . \quad . \quad . \quad (37)$$

and the velocity is $v=\frac{\pi DN}{60}$. Substituting these values in (36) the indicated tractive effort is given by—

$$T_i=\frac{550\times I}{v}=\frac{n}{2}\times\frac{p_e L d^2}{D} \text{ lb.} \quad . \quad . \quad . \quad . \quad (38)$$

It is the usual practice to take the mean effective pressure $p_e$ as proportional to the *gauge* pressure $p$ of the boiler, thus $p_e=Cp$ the factor C being termed the " card factor " and varying from 0·8 to 0·9 under the ordinary cut-off conditions for the development of the maximum horse-power of the engine. The indicated tractive effort can thus be written

$$T_i=\frac{n}{2}\frac{CpLd^2}{D} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (39)$$

for the simple type of engine with several cylinders of the same diameter. A modification, however, is necessary in the case of a compound engine in which the cylinder ratio has to be taken into account. The following values of the indicated tractive effort for simple and compound engines are given in the *Trans of A.S.M.E.* in the paper quoted in the footnote.*

SIMPLE LOCOMOTIVES

Two cylinder $T_i=C\times\frac{pd^2L}{D}$

Three cylinder $T_i=C\times\frac{1{\cdot}5pd^2L}{D}$

Simple Mallet $T_i=C\times\frac{2pd^2L}{D}$

COMPOUND LOCOMOTIVES

Cross compound $T_i=C\times\frac{pd_L{}^2L}{D(R+1)}$

Three cylinder compound $T_i=C\times\frac{3pd_L{}^2L}{D(R+2)}$

Mallet compound $T_i=C\times\frac{2pd_L{}^2L}{D(R+1)}$

where R=cylinder ratio and $d_L$=L.P. cylinder diameter. The difference between the indicated tractive effort and the draw-bar pull or effective tractive effort represents the loss of pull due to engine friction, thus—

$$(T_i-T)=T_f$$

* " Preliminary Power Rating and Tractive Force of Modern Locomotives," by R. Eksergian, American Soc. of Mechanical Engineers, 22 May, 1926.

Professor Dalby has deduced the following expression for the friction draw-bar loss *—

$$T_f = \frac{120W_c}{D} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (40)$$

where $W_c$=total adhesion load on the coupled driving wheels
D=driving wheel diameter in ft.

The approximate value of the draw-bar pull may thus be estimated

$$T = \left(T_i - \frac{120W_c}{D}\right) \text{ lbs.} \quad . \quad . \quad . \quad . \quad . \quad (41)$$

The leading railway companies use special dynamometer cars for locomotive testing, from which accurate records of the variation of the draw-bar pull under the actual running conditions of the road are obtained and from which the corresponding draw-bar horse-powers can be calculated; and from these the mechanical efficiencies

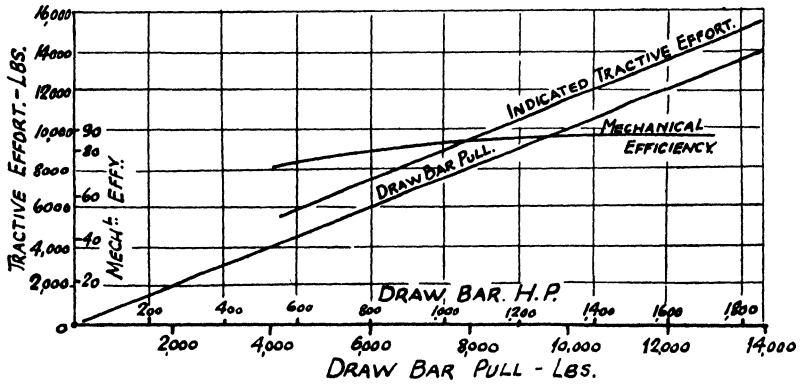



FIG. 87.

can be obtained. There are also test plants on which the stationary locomotive is run on friction wheels at steady speeds and loads. A typical diagram giving curves of indicated tractive effort, experimentally observed draw-bar pull, and the derived mechanical efficiency, for the Schmidt high-pressure compound engine described in Chapter XVIII is shown in Fig. 87.†

When the draw-bar pull is ascertained the draw-bar horse-power is calculable from

$$\text{D.H.P.} = \frac{Tv}{550} \quad . \quad . \quad . \quad . \quad . \quad . \quad (42)$$

where $v$=velocity in ft.-sec.

or

$$\text{D.H.P.} = \frac{TV}{375} \quad . \quad . \quad . \quad . \quad . \quad . \quad (43)$$

where V=velocity in miles/hour.

* *Steam Power*, by W. E. Dalby.

† "The Schmidt High-Pressure Locomotive of the German State Railway Company," by R. P. Wagner, *Trans. A.S.M.E.*, December 1928 (Vol. 51).

# CHAPTER VI

## CONDITION OF THE STEAM IN THE CYLINDER

**98.** In the ordinary type of engine, using saturated steam, the apparent amount of steam in the cylinder at the point of cut-off is less than that actually admitted to the cylinder per stroke, the remainder being present as water, or having passed away to exhaust by leakage at the valve or piston.

The amount of the loss from these two causes, termed the "missing quantity," varies from 20 to 50 per cent. of the total weight of steam supplied per stroke. The causes of the presence of water in the cylinder may be stated in detail as follows:

(1) Wetness of the steam originally supplied by the boiler.

(2) Wetness due to condensation in long ranges of steam-pipes and in the valve chests, especially when these parts are not well covered with non-conducting material.

(3) "Initial condensation" of the steam on entering the working barrel of the cylinder.

(4) Condensation due to work done by the steam during expansion in the cylinder after cut-off.

(5) Condensation due to external radiation and conduction from the cylinder walls.

**99. Wetness of Steam supplied by the Boiler.**—When steam carries over with it from the boiler to the engine, water which has not been evaporated, but which passes away mixed with the steam, the phenomenon is known as "priming."

The conditions which determine, to a greater or less extent, the quality of the steam supplied from a boiler as to dryness are: (*a*) the rate at which the steam is generated—whether by natural draught or accelerated draught: the greater the rate the greater the tendency to wetness. (*b*) The area of the water surface at the water-level of the boiler per pound of steam generated per minute: the smaller the surface area for a given weight of steam delivered, the greater the disturbance of the surface, and the more probability of the steam being wet—hence the steam is usually dryer from a Lancashire boiler than from a vertical-type boiler. (*c*) The volume of the steam space: within certain limits, the smaller the volume the greater the tendency to wetness; hence one means of reducing the amount of priming in boilers is to work with the water-level low in the gauge-glass. (*d*) The size of the boiler compared with the weight of steam required per stroke by the engine: thus a large slow-running engine, if

supplied with steam from a relatively small boiler, causes a fluctuation of pressure in the boiler at each stroke of the engine, which induces surface agitation of the water as the pressure varies, and tends to increase the wetness of the steam. This effect may be remedied by throttling down the steam-supply at the stop-valve, so as to reduce the extent of the fluctuation of the pressure in the boiler.

**100. Wetness due to Condensation in Steam-pipes and Valve Chest.**—The loss of heat from uncovered steam-pipes is considerable, and varies directly as the difference of temperature of the steam and the external air, and inversely as the thickness of the pipe.

The loss from iron steam-pipes uncovered, per degree difference of temperature between steam and external air, is approximately 2·4 thermal units per hour per square foot of external surface of pipe. By covering the pipe with woollen felt $\frac{1}{2}$ in. thick, this loss is reduced to 0·7 thermal unit per hour per square foot of external surface of metal pipe; with 1-in. covering the loss is 0·4 thermal unit, and with a 2-in. covering 0·24 thermal unit.*

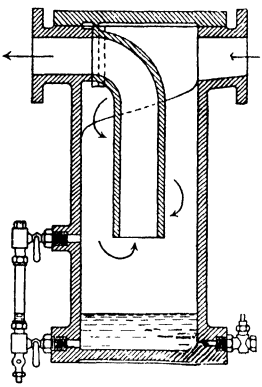


Fig. 88.

All water present in the steam should, as far as possible, be separated from it, so that the steam may enter the cylinder dry, and for this purpose it is usual to fix a separator as near as possible to the engine. The action of one form of separator, of which there are various designs, will be understood by reference to Fig. 88. The wet steam enters the chamber at the top, and passes through a spiral passage downwards towards the bottom of the separator. The whirling motion of the steam thus set up causes the particles of water present in the steam to strike the sides of the chamber, and to flow to the bottom of the vessel. The steam passes forward in a more or less dry condition, in an upward direction through the exit pipe, the bottom of which is some distance from the bottom of the separator. Connected with the separator is a "steam trap" into which the water is collected, and from which it is passed into the feed-tank. A gauge-glass is fitted to show the height of water present in the bottom of the separator.

**101. Initial Condensation.**—Next to the loss of heat at the exhaust, that due to initial condensation of steam in the cylinder is the most serious of the losses connected with the use of steam as a working fluid; and the endeavour to prevent the loss from this cause has accounted for most of the improvements in the steam engine since the time when James Watt invented the separate condenser. Before

* For a recent discussion of "Transmission of Heat through Insulation," by R. H. Heilman, see *Mech. Engineering*, July 1930, p. 693.

this time the cylinder was used alternately as a steam cylinder and a condenser.

When steam from the boiler is admitted to the cylinder with the piston at the beginning of the stroke, it comes in contact with the metallic surfaces of the cylinder cover, the face of the piston, the walls of the steam ports, and more or less area of the circumferential surface of the cylinder barrel.

If all these surfaces were as hot as the steam which enters the cylinder, no transfer of heat would take place between the steam and the metal, and therefore there would be no initial condensation of the steam.

But in practice the temperature of the walls is always lower than that of the entering steam, the walls being cooled during expansion and during exhaust by having been in contact with the comparatively cool steam of reduced pressure during these periods. Consequently, during admission of steam at the beginning of the stroke, condensation takes place, till the walls are heated up to a temperature approaching that of the initial steam. Hence the weight of steam admitted to the cylinder per stroke, up to point of cut-off, is greater than that present in the cylinder as steam, by the amount condensed during admission in the process of warming up the cylinder walls.

Condensation, then, up to point of cut-off is due to the heat lost in warming up the metallic walls with which the steam comes in contact in the cylinder.

In addition to this there is the condensation which takes place after cut-off due to the work done during expansion at the expense of the internal energy of the steam.

Considering the amount of steam condensed in the cylinder, it would seem, at first sight, that the cylinder must gradually become choked with water. Such is more or less the case when the engine is started and before the cylinders have been properly heated up, and to get rid of this water, relief-cocks are fitted at each end of the cylinder, which are always opened when the engine is started, so as to blow through and relieve the cylinder of the water deposited.

As the temperature of the cylinder walls gradually increases, less water is deposited. If the relief-cocks are now shut, more or less condensation will still continue, but the water deposited is usually removed from the cylinder by re-evaporation.

**102. Effect of Clearance and Barrel Surface on Condensation.**— The portion of the double stroke, during which heat passes from the steam to the metal, begins at admission of the steam to the cylinder, and terminates usually almost immediately after cut-off, where the expanding steam has fallen in temperature to that of the mean temperature of the walls. The greater the area of the internal surface per pound of steam admitted, the greater the condensation. Hence the importance of reducing this surface as much as possible, especially the clearance portion of it, where, by care in designing,

considerable reductions might often be made, with a corresponding improvement in the economy of the engine.

In respect of clearance surface and cylinder surface generally, it may be well to compare the effect of difference of ratio between cylinder diameter and length of stroke.

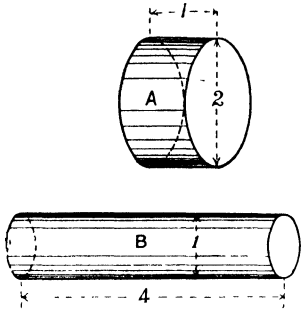


FIG. 89.

Taking cylinders (Fig. 89) of equal capacity A and B, A having 4 sq. ft. of piston area and 1 ft. stroke, and B having 1 sq. ft. of piston area and 4 ft. stroke—A representing the short-stroke, high-speed type of engine of relatively large piston area, and B representing the long-stroke type with relatively small piston area: then, neglecting clearance volume and steam passages, the relative area of clearance and cylinder surface exposed to equal weights of steam in the two cases will be seen from the following table:

| | *A* | *B* |
|---|---|---|
| Piston area . . . . . sq. ft. | 4·0 | 1·0 |
| Stroke . . . . . ft. | 1·0 | 4·0 |
| Piston displacement . . . . . cub. ft. | 4·0 | 4·0 |
| Surface of barrel . . . . . sq. ft. | 7·09 | 14·18 |
| Clearance surface . . . . . ,, | 8·0 | 2·0 |
| ,, ,, + barrel to ¼ stroke . . . ,, | 9·78 | 5·56 |
| ,, ,, + ,, ½ ,, . . . ,, | 11·56 | 9·13 |
| ,, ,, + ,, ¾ ,, . . . ,, | 13·35 | 12·69 |
| ,, ,, + ,, full ,, . . . . ,, | 15·13 | 16·26 |
| Surface exposed per unit weight of steam admitted with cut-off at ¼ stroke . sq. ft. | 9·78 | 5·56 |
| Ditto ditto ½ ,, . ,, | 5·78 | 4·56 |
| Ditto ditto ¾ ,, . ,, | 4·45 | 4·23 |
| Ditto ditto full ,, . ,, | 3·78 | 4·06 |

Comparing the two cases, on admission of steam to the respective cylinders the clearance surface of the short-stroke engine is four times as great as with the long-stroke. If cut-off takes place at ¼ stroke, the actual wall surface of the cylinder enclosing the steam up to cut-off is 5·56 sq. ft. in the long-stroke engine, and 75 per cent. more than this in the short-stroke engine. That is, if condensation be directly proportional to surface, the condensation in the short-stroke engine will be 75 per cent. greater than in the long-stroke engine when cut-off takes place at ¼ stroke in each engine.

It will, however, be observed that as the cut-off is made later, the respective areas of cylinder surface in the two cases become more and more nearly alike, and if steam were admitted to the end of the stroke, the surface is 7 per cent. greater in the long-stroke engine; in other words, the loss by condensation in the two cylinders would be more nearly the same as the cut-off is later. The earlier the cut-off the more the advantage lies with the long-stroke engine.

It has been so far assumed that the engines are run at the same rotational speed; but if they are run at the same piston speed, as would be more probably the case, then the short-stroke engine would make four times as many revolutions as the long-stroke.

But if the assumption is correct that cylinder condensation is inversely proportional to the square root of the number of rotations, then the loss by condensation in the long-stroke engine as compared with that in the short-stroke is as $1 : \frac{1}{\sqrt{4}} = 1 : \frac{1}{2}$.

Short-stroke engines are therefore most economical when run at high speeds and with a late cut-off.

This statement takes no account of clearance volume. In short-stroke engines the clearance volume is proportionally large, and in such engines great care must be taken to make the best use of the compression steam, so as to fill the clearance space with steam at a pressure as near as possible to the admission pressure, otherwise the difference must be made up by boiler steam, and the amount required for this purpose may be a large proportion of the total steam used.

The advantage of turned and polished surfaces for the clearance spaces in reducing cylinder condensation has been proved in numerous instances; and the most economical results yet recorded with saturated steam have been obtained in engines having the piston face and inner surface of the cylinder cover turned and polished. The surfaces appear to be thereby rendered more nearly non-conducting.

**103. Re-evaporation.**—During the stroke of the piston, as soon as cut-off takes place, the pressure of the steam gradually falls, and the water present, owing to the removal of the pressure upon it, begins to re-evaporate as soon as the pressure of the steam falls below that corresponding to the temperature of the water in the cylinder. This point generally occurs soon after cut-off, and the re-evaporation continues as the expansion continues, the weight of steam present, as steam, gradually increasing towards the end of the stroke. When the exhaust port opens, the pressure is, more or less suddenly, still further reduced and the rate of re-evaporation accelerated, and during the exhaust stroke the water of initial condensation more or less completely disappears as dry steam.

The heat required for re-evaporation is obtained partly from the heat in the water itself, but chiefly at the expense of the heat in the cylinder walls; hence the greater the re-evaporation the more heat flows from the walls, and the more heat must be given up to the walls on the succeeding stroke by the steam during admission.

The heat given up by the steam to the walls of a single cylinder engine is practically lost (except for the small amount of work done by re-evaporated steam), because this heat, which is again returned by the walls, is given up during exhaust, and thus increases the already large exhaust waste.

**104. Mean Temperature of Cylinder Walls.**—From the experiments

of Messrs. Bryan Donkin, Callendar, and Nicolson, it has been shown that the cylinder walls may be divided into two parts, namely, the outer portion, where the temperature is constant; and the inner or "periodic" portion, where the temperature fluctuates with the temperature of the steam in contact with it (see Art. 130).

The depth of the periodic portion is usually very small, and the less so as the time of interaction is less between the steam and the cylinder walls.

In all cases economy results from raising the mean temperature of the walls nearer to that of the initial steam in the cylinder. The mean temperature of the walls is raised as the weight of steam passing through the engine per minute is increased, and the condensation is thus reduced per pound of steam supplied.

Conversely, all causes tending to reduce the mean temperature of the cylinder walls tend also to increased cylinder condensation, and therefore to increased consumption of steam per I.H.P. per hour.

**105. Range of Temperature.**—The range of temperature of the steam in the cylinder is the difference $(t_1 - t_2)$, where $t_1$ is the temperature during admission and $t_2$ the temperature of exhaust. The range of temperature is thus independent of the point of cut-off.

But cylinder condensation depends not directly on the range of temperature of the steam but on the *mean* temperature of the internal portion of the cylinder walls, and the following relations should be noted between range of temperature of the steam and mean temperature of the walls :—

(1) For a given constant range of temperature of the steam in a cylinder, the mean temperature of the walls increases as the point of cut-off is later; hence the mean temperature, and also the amount of cylinder condensation, may vary considerably with the same range of temperature.

(2) The mean temperature of the walls may remain constant for any number of different ranges of temperature above and below the mean; hence the amount of cylinder condensation may vary considerably with the same mean temperature, being greater as the difference between the initial temperature of the steam and the mean temperature of the walls is greater, and *vice versâ*.

**106. Effect of Initially Wet Steam.**—Experiment has shown that steam admitted to the cylinder initially wet tends still further to increase initial condensation up to a certain limit, and conversely the drier the steam the smaller the heat interaction between the steam and the cylinder walls. Hence the importance of draining steam pipes and valve chest.

Water entering the cylinder with the steam tends to partially evaporate in passing through the cylinder, because the heat contained in the water on entering the cylinder is greater than will be retained by it on leaving, hence a portion of the original water is evaporated by the heat liberated at the lower pressure.

**107. Condensation Due to Work done during Expansion.**—In addition to initial condensation due to interchange of heat between the steam and the cylinder walls, there is during expansion in the cylinder a liquefaction due to work performed at the expense of the internal energy of the steam ; therefore the greater the expansion the wetter the steam becomes.

**108. Conduction and Radiation Loss.**—A small proportion of the heat of the steam is lost by conduction through the cylinder metal to surrounding parts and by radiation from the external surfaces. This is conditioned by the effectiveness of the insulation provided, and the state of such unlagged surfaces as cylinder and valve covers. Radiation loss is reduced by giving these surfaces a high polish. This heat loss may run from $\frac{1}{3}$ per cent. of the heat supply in the case of a large engine to 2 per cent. in a small engine. A good average for practical calculation is 1 per cent.

**109. Indicated Steam and Missing Quantity.**—To find the weight of the stuff in the cylinder existing partly as steam at any point of the stroke, the volume occupied by it at the chosen point has to be ascertained from the indicator diagrams. This being known, the corresponding weight is got by dividing it by the specific volume of the steam (from the tables) corresponding to the pressure at the point.

The diagram (Fig. 90) has first to be calibrated by drawing the

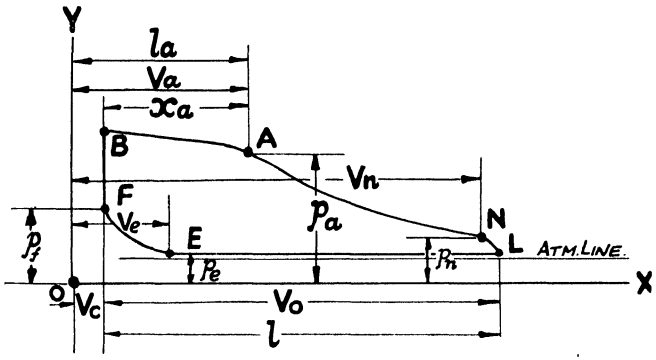



FIG. 90.

lines of zero volume and pressure OY and OX. On this diagram the points of admission (F), cut-off (A), release (N), and compression (E) are shown. The corresponding volumes measured from the axis OY are $V_f = V_c$, $V_a$, $V_n$, and $V_e$ ; the stroke volume is $V_o$.

The corresponding absolute pressures are $p_f$, $p_a$, $p_n$, and $p_e$, and the specific volumes $v_f$, $v_a$, $v_n$, and $v_e$.

The volume at any point, for example at A, can be obtained by scaling the length $l_a$ from OY and finding the value from $V_a = \frac{l_a}{l} V_o$,

where $l$ is the length of the diagram. Alternatively by scaling the length $x_a$ from the end of the diagram and calculating—

$$V_a=\left(V_c+\frac{x_a}{l}V_o\right) \quad \text{or} \quad V_a=V_o\left(\frac{x_a}{l}+c\right) \quad . \quad . \quad . \quad (1)$$

where $c$ is the fractional clearance or $\frac{V_c}{V_o}$.

The weight of steam in the cylinder at cut-off A is given by—

$$w_a=\frac{V_a}{v_a} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

and at release by—

$$w_n=\frac{V_n}{v_n} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

At the point of compression E it is usually assumed that the steam is in the dry condition and the weight in the clearance is obtained from—

$$w_e=\frac{V_e}{v_e} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

This is called the " cushion " or " play " steam.

This weight of steam is enclosed in the clearance space, when admission commences at F. When cut-off takes place at A there is therefore an apparent amount $(w_e+w_a)$ lb. of steam in the cylinder and clearance. The apparent supply or " cylinder-feed " during admission is thus $w_{f_a}'=(w_a-w_e)$; but this is *not* the actual amount which has been supplied if the steam is slightly wet, dry, or slightly superheated on admission, on account of condensation and leakage losses.

The actual cylinder feed, or steam admitted per stroke from the steam pipe, can only be estimated from the measured weight of condensate, after leakage steam which has by-passed the cylinder during the admission period at valve and piston, has been deducted. This leakage quantity is always indeterminate (see Art. 126).

Assuming, however, that the leakage in a given case has been approximated and deducted from the measured condensate, and the actual cylinder feed entering the cylinder per stroke is $w_f$, then the total weight of stuff in the cylinder and clearance during expansion (assuming no further leakage between A and N) will be—

$$w=(w_f+w_e) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

At cut-off only the portion $w_a$ of this amount exists as steam; the rest is in the form of water, which is assumed to be deposited as a film on the cylinder surfaces. The difference $w_{m_a}=(w-w_a)$ lb./str. is the " missing quantity " at cut-off A, and $w_{m_n}=(w-w_n)$ the missing quantity at release N. These values are also given by the differences of the actual and apparent or " indicated " cylinder feeds $(w_f-w_{f_a}')$ and $(w_f-w_{f_n}')$.

If W is the total steam consumption in lb./hr., less the estimated

leakage at valve and piston during admission, the mean value of the cylinder feed, when the engine is double acting, is—

$$w_f = \frac{W}{60 \times 2N} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

and the mean value of the indicated feed at cut-off is—

$$w_{f_a}' = \frac{V_o}{l}\left(\frac{l_a}{v_a} - \frac{l_e}{v_e}\right) \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

where $l_a$ and $l_e$ are the mean values of the lengths measured from OY for the two ends of the cylinder, or the lengths measured from the mean card for the two ends. It is the usual practice to base these calculations on the mean card.

If the engine is supplied with highly superheated steam, initial condensation can be suppressed, and the steam may either be dry at cut-off or more or less superheated. In such a case there is no missing quantity at cut-off, and the specific volume of the steam at A will be $v_{sa}$, a higher value than $v_a$, depending on the amount of superheat. Here the weight of stuff at A is—

$$w_{sa} = \frac{V_a}{v_{sa}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

and $w_f = (w_{sa} - w_e)$.

The steam is superheated at A when $(w_f + w_e)v_a < V_a$; it is dry when $(w_f + w_e)v_a = V_a$, and wet when $(w_f + w_e)v_a > V_a$.

The indicated steam can also be expressed by a rate of consumption, that is in lb./H.P. hr.

Let I = total I.H.P.

N = speed in rev./min.

$p_{me}$ = mean effective pressure in lb./in.$^2$

Then $I = \frac{2p_{me}V_oN \times 144}{33000}$ for a double-acting engine.

The total indicated feed is $2 \times 60N w_{f_a}'$ and the indicated rate is given by—

$$w_{i_a}' = \frac{2 \times 60 \times N w_{f_a}' \times 33000}{2 \times 144 \times V_o N p_{me}}$$

Substitute for $w_{f_a}'$ from (7) and

$$w_{i_a}' = \frac{13750}{lp_{me}}\left(\frac{l_a}{v_a} - \frac{l_e}{v_e}\right) \text{lb./I.H.P. hr.} \quad . \quad . \quad . \quad . \quad (9)$$

at cut-off.

This is also applicable to the single-acting engine. If the rate at release (N) is taken, then

$$w_{i_n}' = \frac{13750}{lp_{me}}\left(\frac{l_n}{v_n} - \frac{l_e}{v_e}\right) \text{ lb./I.H.P. hr.} \quad . \quad . \quad . \quad (10)$$

at release.

In the case of a multiple-expansion engine $p_{me}$ is the mean

effective pressure referred to the cylinder under consideration (see Art. 143).

Since the actual rate of feed is $w_i=\frac{W}{I}$ the missing quantity is

$$w_{mi}=(w_i-w_{ia}') \text{ lb./I.H.P. hr.} \quad . \quad . \quad . \quad . \quad (11)$$

at cut-off.

and $$w_{mi}=(w_i-w_{in}') \text{ lb./I.H.P. hr.} \quad . \quad . \quad . \quad . \quad (12)$$

at release.

When the steam is wet $w_{mi}$ is less at release than at cut-off, due to re-evaporation.

*Example* 1.—The mean indicator diagram * for crank and head ends of an unjacketed double-acting engine, using saturated steam, is shown in Fig. 91.

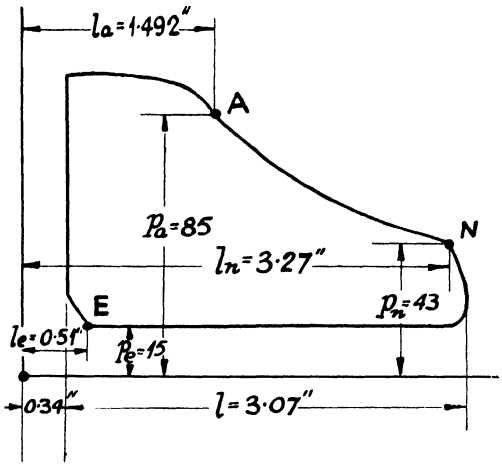



FIG. 91.

The mean stroke volume is 0·261 ft.³ and the percentage clearance is 11·1. The steam consumption, corrected for leakage, is estimated as 590·4 lb./hr., when the engine speed is 150 rev./min.

Calculate, from the figures given on the diagram, the indicated steam and missing quantity at cut-off and release, and the percentage re-evaporation.

$$V_a=\frac{l_a V_o}{l}=\frac{1\cdot492\times0\cdot261}{3\cdot07}=1\cdot492\times0\cdot085$$

$=0\cdot1268$ ft.³

$p_a=85$; $v_a=5\cdot184$, hence

$V_o=0\cdot261$
$l_a=1\cdot492$ in.
$l=3\cdot07$ in.

* From First Report of Steam Engine Research Committee, I.Mech.E., March 1905.

$$w_a=\frac{V_a}{v_a}=\frac{0\cdot1268}{5\cdot184}=0\cdot02446 \text{ lb.}$$

$$V_n=\frac{l_n V_o}{l}=3\cdot27\times0\cdot085$$

$$=0\cdot278 \text{ ft.}^3$$

$l_n=3\cdot27$ in.
$p_n=43$
$v_n=9\cdot8$

$$w_n=\frac{V_n}{v_n}=\frac{0\cdot278}{9\cdot8}=0\cdot02836 \text{ lb.}$$

$$V_e=\frac{l_e}{l}V_o=0\cdot51\times0\cdot085$$

$$=0\cdot04335 \text{ ft.}^3$$

$l_e=0\cdot51$
$p_e=15$
$v_e=26\cdot27$

$$w_e=\frac{V_e}{v_e}=\frac{0\cdot04335}{26\cdot27}=0\cdot00165 \text{ lb.}$$

Indicated feed at cut-off $w_{f_a}'=(w_a-w_e)=(0\cdot02446-0\cdot00165)$
$=0\cdot02281$ lb./str.

,, ,, ,, release $w_{f_n}'=(w_n-w_e)=(0\cdot02836-0\cdot00165)$
$=0\cdot02671$ lb./str.

Mean actual cylinder feed $w_f=\frac{W}{2\times60N}=\frac{590\cdot4}{120\times150}$ | W=590·4, N=150

$=0\cdot0328$ lb./str.

Missing quantity at cut-off $w_{ma}=(w_f-w_{f_a}')=(0\cdot0328-0\cdot02246)$
$=0\cdot01034$ lb.

,, ,, ,, release $w_{mn}=(w_f-w_{f_n}')=(0\cdot0328-0\cdot02671)$
$=0\cdot00609$ lb.

Re-evaporation $=(w_a-w_n)=(w_{f_a}'-w_{f_n}')=(0\cdot01034-0\cdot00609)$
$=0\cdot00425$ lb.

Percentage re-evaporation $=\frac{(w_{f_a}'-w_{f_n}')}{w_m}=\frac{0\cdot00425}{0\cdot01034}=0\cdot410$

or $=41$ per cent.

*Example* 2.—The area of the indicator diagram (Fig. 91) is 4·14 in.$^2$ and the scale of the spring 1 in.=43 lb./in.$^2$ Calculate the indicated rate of cylinder feed and missing quantity at cut-off and release.

Mean effective pressure $p_{me}=\frac{as}{l}=\frac{4\cdot14\times43}{3\cdot07}$

$=57\cdot95$ lb./in.$^2$

$a=4\cdot14$
$s=43$
$l=3\cdot07$
$V_o=0\cdot261$
$N=150$

$$\text{I.H.P.}=\frac{2p_{me}V_oN\times144}{33000}=\frac{2\times57\cdot95\times0\cdot261\times150\times144}{33000}$$

or I=19·8

Actual rate of cylinder feed $w_i=\frac{W}{I}=\frac{590\cdot4}{19\cdot8}=29\cdot8$ lb./I.H.P. hr.

Indicated rate at cut-off $w_{ia}=\frac{13750}{lp_{me}}\left(\frac{l_a}{v_a}-\frac{l_e}{v_e}\right)$

$=\frac{13750}{3\cdot07\times57\cdot42}\left(\frac{1\cdot492}{5\cdot184}-\frac{0\cdot51}{26\cdot27}\right)$

$=78(0\cdot288-0\cdot0194)$

$=78\times0\cdot2686=20\cdot95$ lb./I.H.P.hr.

$l=3\cdot07$, $l_a=1\cdot492$, $l_e=0\cdot51$, $v_a=5\cdot184$, $v_e=26\cdot27$, $p_{me}=57\cdot42$

Indicated rate at release $w_{in}=\frac{13750}{lp_{me}}\left(\frac{l_n}{v_n}-\frac{l_e}{v_e}\right)$

$=\frac{13750}{3\cdot07\times57\cdot42}\left(\frac{3\cdot27}{9\cdot8}-\frac{0\cdot51}{26\cdot27}\right)$

$=78(0\cdot334-0\cdot0194)$

$=78\times0\cdot3146=24\cdot53$ lb./I.H.P. hr.

$l_n=3\cdot27$, $v_n=9\cdot8$

Missing quantity at cut-off $w_{mi}=(29\cdot8-20\cdot95)=8\cdot85$

„ „ „ release $w_{ni}=(29\cdot8-24\cdot53)=5\cdot27$

Percentage re-evaporation $=\frac{(8\cdot85-5\cdot27)100}{8\cdot85}=\frac{3\cdot58}{8\cdot85}=0\cdot405$

against 0·410 by the other method.

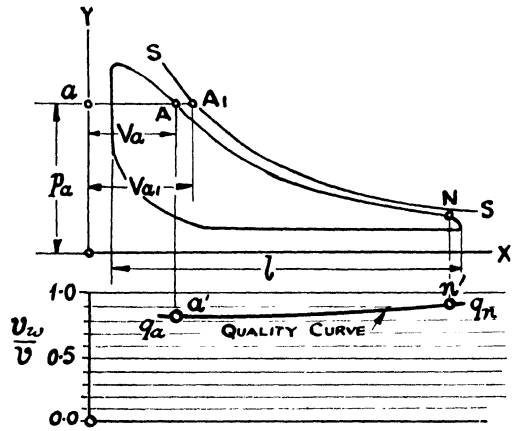



Fig. 92.

**110. Dryness Fraction and Quality Curve.**—At any point on the expansion curve the ratio of the volume of steam in cylinder and clearance, as shown by the diagram, to the volume of dry steam corresponding to the total weight of $H_2O$ stuff in the cylinder, is called the " dryness fraction."

Alternatively it may be defined as the ratio of the weight of steam shown by the diagram to the weight of $H_2O$ stuff in the cylinder.

If the curve of dry steam volume SS for the total weight of $H_2O$ stuff ($w$) is drawn on the diagram (Fig. 92), then when the steam is

wet the quality of dryness is given by the ratio of the intercepts between the axis OY and the expansion and saturation curves.

Thus at A, $q_a = aA/aA_1 = V_a/V_{a1} = \dfrac{w_a}{(w_f+w_e)}$

On the volume scale of the diagram at any point where the pressure is $p$ and specific volume $v$, the volume to be set off from OY is $V=(w_f+w_e)v$.

Thus at A, $V_{a1}=(w_f+w_e)v_a$, and the corresponding length, to the same scale as $l$, is $aA_1=(w_f+w_e)v_a l/V_o$.

When the dryness quality ($q$) is calculated for several points on the expansion curve and plotted on a stroke base, the resulting curve $a'n'$ (Fig. 92) is called the " quality curve."

Since $V_a/w=v_{wa}$, the specific volume of the wet steam of quality $q_a$, and $V_{a1}/w=v_a$, the specific volume of the dry steam, then $q_a=\dfrac{v_{wa}}{v_a}$, or in general $q=\dfrac{v_w}{v}$.

In modern practice superheated steam is used with considerable initial superheat and the steam at cut-off may be either dry or appreciably superheated. It may pass from the superheated to the dry or wet condition before release or may remain superheated throughout the expansion, according to the type of engine, the initital conditions of superheat, ratio of expansion, etc.

In a case of this kind, the saturation curve SS may fall wholly inside the expansion curve of the diagram, or partly inside and partly outside this curve, the point where it crosses the indicator curve denoting the pressure at which the dry steam ($q=1$) condition is reached.

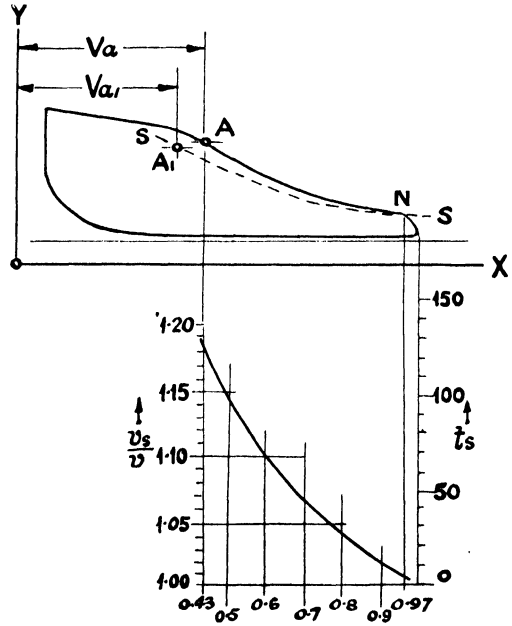



FIG. 93.

One of the diagrams from Prof. Ripper's paper on " Superheated Steam Engine Trials," * is shown in Fig. 93.

The steam had 338° Fahr. degrees of superheat on admission. It was still considerably superheated at cut-off A, and slightly super-

* *Proc. Inst. C. E.*, vol. cxxviii, 1897.

heated at release N, as indicated by the saturation curve SS. Here $V_a/w=v_{sa}$ and $V_{a1}/w=v_a$, so that the quality value is greater than unity, indicating superheat. The quality values given by the curve may be expressed either as volume ratios or degrees of superheat, and both scales are shown on the figure.

An alternative method of determining the qualities of dryness or superheat, without reference to the PV saturation curve, is given in connection with the transference from PV to TΦ field (Art. 111).

*Example* 3.—Find the dryness fraction at cut-off and release for the engine of Example 1.

In this case $w=(w_f+w_e)=(0{\cdot}0328+0{\cdot}00165)=0{\cdot}03445$ lb.

At cut-off $w_a=0{\cdot}02445$. Hence—

$$q_a=\frac{w_a}{w}=\frac{0{\cdot}02445}{0{\cdot}03445}=0{\cdot}71 \text{ or } 71 \text{ per cent.}$$

At release $w_n=0{\cdot}02838$. Hence—

$$q_n=\frac{w_n}{w}=\frac{0{\cdot}02838}{0{\cdot}03445}=0{\cdot}824 \text{ or } 82{\cdot}4 \text{ per cent.}$$

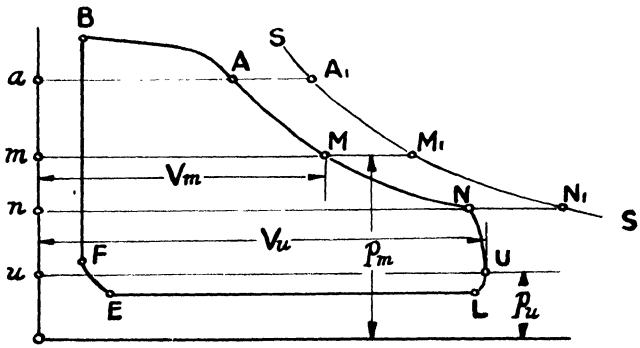



FIG. 94.

**111. Application of the Indicator Diagram to the TΦ Chart.**—The closed PV curve, obtained from the cylinder by the indicator, can be represented by a corresponding curve on the TΦ chart.

The TΦ indicator diagram area gives the heat units converted to work for an equivalent cylinder feed, while the PV diagram gives the work value in mechanical units for $w_f$ lb. of feed.

The TΦ diagram is serviceable in showing the nature of the heat losses incurred during a complete cycle.

Referring to Fig. 94, which shows the mean diagram for the two ends of the double-acting engine considered in Example 1, at any point M on the expansion curve the volume of steam in the cylinder and clearance is $V_m$ at pressure $p_m$, and the quality as defined above is $q_m=m\mathrm{M}/m\mathrm{M}_1$, where $m\mathrm{M}_1=wv_m$, the volume of $w$ lb. of dry steam at $p_m$.

The corresponding state point in the TΦ field (Fig. 95) is M′ at the intersection of some constant volume curve $V_m'$ and the pressure line $p_m$.

The volume $V_m'$ represents a specific volume or volume per lb. of stuff for which the chart is drawn, and since $V_m$ is the volume of $w$ lb. of the wet steam,

$$V_m' = \frac{V_m}{w} = CV_m \quad . \quad . \quad . \quad . \quad . \quad . \quad (13)$$

This may be termed the TΦ volume, and written $V_{(T\Phi)m}$. The reducing factor C is called the "volume factor." * It is the reciprocal of the total weight of stuff in the cylinder, between cut-off and release.

The expansion curve AN, Fig. 94, can be transferred to the TΦ field by scaling the volumes, multiplying these by the volume factor,

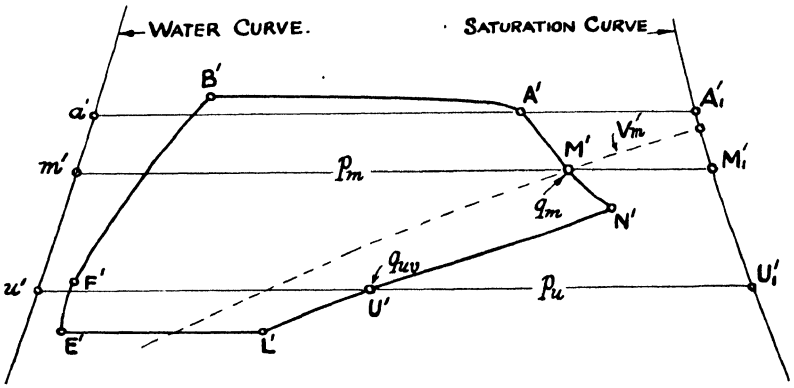



FIG. 95.

and then locating the intersection of the $V_{T\Phi}$ curves and the corresponding pressure lines; when a sufficiently complete set of constant volume curves is provided. Alternatively the points can be located without reference to constant volume curves.

At any point to be transferred let the fractional displacement of the piston be denoted by $x$, then the total volume of cylinder and clearance is given by $(xV_o + V_c)$, where $V_o$ is the stroke volume and $V_c = cV_o$ is the clearance volume. Then the TΦ volume is given by

$$V_{T\Phi} = V_o \frac{(x+c)}{w} \quad . \quad . \quad . \quad . \quad . \quad . \quad (14)$$

The ratio of the intercepts between the water curve and the expansion curve A′N′ and that between the water and saturation curves, Fig. 95, give the quality, or ratio of weight of steam to total weight of stuff present in the cylinder, and clearance at the point considered. Thus for M′, $q_m = m'M'/m'M_1'$. All points on

* This term was introduced by Captain Sankey, *Proc. Inst. Mech. E.*, February 1894.

this transferred expansion curve are *real* quality points, as none of the stuff supplied during admission escapes from the cylinder to exhaust.

Between N and A, however, during exhaust, compression, and admission, the quality of stuff, steam and water, in the cylinder varies continuously. Hence if this method of transference is applied all round the cycle, the points located for the open portion give fictitious values of quality, that is, they give no clue to the relative proportions of steam and water present. Now instead of carrying out the heat and work conversions in the separate organs of the steam engine, they could equally well be performed in a substituted closed vessel, and an identical indicator diagram would be produced, without variation of the quantity of stuff. On this assumption, that the real engine may be replaced by this hypothetical one in which the whole of the $w$ lb. of stuff is retained during the cycle and is alternately heated and cooled reversibly by external means, the foregoing process of transference may be applied at all points round the closed indicator curve.

Thus, at a point U, Fig. 94, after release, at pressure $p_u$ the volume will be $V_{T\Phi u}=CV_u$, giving the point U′ in the TΦ field, Fig. 95.

It must be clearly borne in mind, however, that the ratio $u'U'/u'U_1'$ is *not* a real quality value for the stuff in the engine cylinder. It is the quality which would obtain under the theoretical conditions in the substituted closed vessel.

As, however, this ratio is of importance for the quick and easy location of points, without reference to volume curves in the TΦ field, it will be termed, in what follows, the "virtual quality" and denoted by a subscript ($v$). Thus at U the TΦ virtual quality is $q_{uv}$.

Instead of carrying out the process of direct calculation of $V_{T\Phi}$ by Equation 14 for each point of transference, a scale of TΦ volume can be used in the following manner. As shown in Fig. 96, calibrate the diagram by drawing the lines of zero volume OY and zero pressure OX, and draw a vertical $Xy$ touching the toe of the diagram, thus defining the total (cylinder+clearance) volume $V_1$. Calculate the corresponding TΦ volume $V_{(T\Phi)1}=CV_1$.

Choose any suitable TΦ volume scale, place it with the zero value at the origin O and the value of $CV_1$ on the vertical $Xy$, and mark off the scale divisions on $Oy$, as shown. The TΦ volume at any point of the stroke is then obtained by dropping a vertical to the inclined scale. This arrangement is somewhat more convenient than drawing the scale directly on OX, as the scale of fractional divisions of stroke can, as shown, be placed on this line and the transferred points can be numbered in the TΦ field to correspond to fractions of the stroke. This method, however, when the diagram is small, is less exact than that of direct calculation.

In working out a given case the process of transference is simplified

if the pressures are marked round the diagram at the various points, as shown in Fig. 96. The determination of the real and virtual quality values can now be carried out by a tabular method for out-stroke and in-stroke. The tabular arrangement is shown below. In the case of the compression curve the real quality values are given by $q=q_v/Cw_e$ (see Art. 112).

When the $V_{T\Phi}$ value at any point on the expansion curve in the case of an engine using initially superheated steam is greater than the corresponding specific volume $v$, the steam is superheated at that point. Other points in the open part of the cycle may appear to show superheat, but it must be borne in mind that for the actual cylinder these are fictitious values and apply only to ideal conditions in a substituted closed vessel.

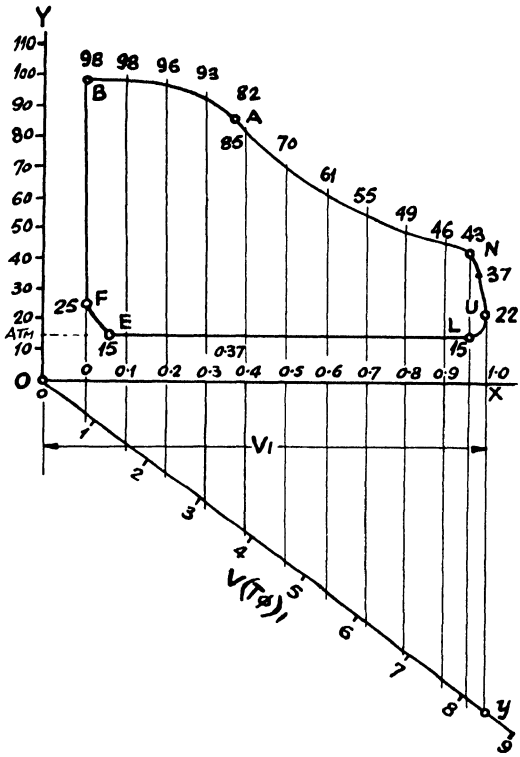



FIG. 96.

The next example shows the practical application of this method to an engine supplied with saturated steam.

*Example* 4. — The indicator diagram, Fig. 96, is the mean diagram for the double-acting engine (Example 1), supplied with saturated steam assumed to be initially dry. The stroke volume is 0·261 ft.$^3$, percentage clearance 11·1, cylinder feed 0·0328 lb., weight of clearance steam $w_e$=0·00165 lb. Transfer the PV diagram to the TΦ field.

Total weight of stuff in the cylinder during expansion $w=(w_f+w_e)$ $=(0·0328+0·00165)=0·03445$ lb.

Volume factor $C=\frac{1}{w}=\frac{1}{0·03445}=29.$

Total cylinder and clearance volume $V_1=1·11V_o=1·11\times0·261$ $=0·28997$ ft.$^3$.

Hence $V_{T\Phi}=CV_1=29\times0·28997=8·41.$

Cut-off in this case takes place at 0·37 stroke and release about 0·96, so that these fractions are included in the data table, as also the point for 0·98 stroke.

OUT-STROKE

| Fraction of Stroke $x$ | 0 | 0·1 | 0·2 | 0·3 | 0·375 | 0·4 | 0·5 | 0·6 | 0·7 | 0·8 | 0·9 | 0·96 | 0·98 | 1·0 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| essure $p$ . . . | 98 | 98 | 96 | 93 | 85 | 82 | 70 | 61 | 55 | 49 | 45 | 43 | 37 | 22 |
| ecific volume $v$ . | 4·535 | 4·535 | 4·624 | 4·765 | 5·184 | 5·362 | 6·218 | 7·075 | 7·90 | 8·68 | 9·40 | 9·80 | 18·29 | 18·3 |
| p volume $V_{T\Phi}$ . | 0·84 | 1·60 | 2·37 | 3·12 | 3·68 | 3·89 | 4·67 | 5·43 | 6·18 | 6·92 | 7·68 | 8·10 | 8·30 | 8·41 |
| uality $\frac{V_{T\Phi}}{v} = q_v$ . | 0·185 | 0·353 | 0·511 | 0·655 | — | — | — | — | — | — | — | — | 0·785 | 0·45 |
| . . . . . | — | — | — | — | 0·71 | 0·725 | 0·75 | 0·767 | 0·782 | 0·798 | 0·817 | 0·824 | — | — |
| | | | | | ← EXPANSION → | | | | | | | | | |

On the in-stroke since the exhaust pressure is constant at 15 lb./in.$^2$ abs. between 0·96 and 0·055 × stroke, it is not necessary to determine all the intermediate points, and only the point for 0·5 is given.

IN-STROKE

| $x$ | 0 | 0·055 | 0·5 | 0·96 | 1·0 |
|---|---|---|---|---|---|
| $p$ . . . . . | 23 | 15 | 15 | 15 | 22 |
| $v$ . . . . . | 17·62 | 26·27 | 26·27 | 26·27 | 18·37 |
| $V_{T\Phi}$ . . . . . | 0·84 | 1·26 | 4·67 | 8·16 | 8·41 |
| $\frac{V_{T\Phi}}{v} = q_v$ . . . . | 0·0472 | 0·048 | 0·177 | 0·31 | 0·457 |
| $q$ . . . . . | 0·983 | 1·0 | — | — | — |
| | ← COMPRESSION → | | | | |

The true and virtual quality points are shown in Fig. 97, by the numbers corresponding to the fractions ($x$) of the stroke chosen on Fig. 96.

Alternatively to the method of diagram transference discussed above, a purely graphical process, due originally to Prof. J. Boulvin of Ghent * can be employed.

**112. Estimation of Heat Exchanges from the TΦ Diagram.**—In Fig. 98 the diagram of Fig. 97 is replotted to a smaller scale on the TΦ field, which is extended to the absolute zero of temperature.

The full line curves WW and SS show the water and saturation curves for 1 lb. of $H_2O$ stuff.

* See footnote, Art. 49. Also see "The Graphical Conversion of the PV into the TΦ Diagram," by W. J. Goudie, *Engineering Review*, August 1906. This is a modification of the Boulvin method.

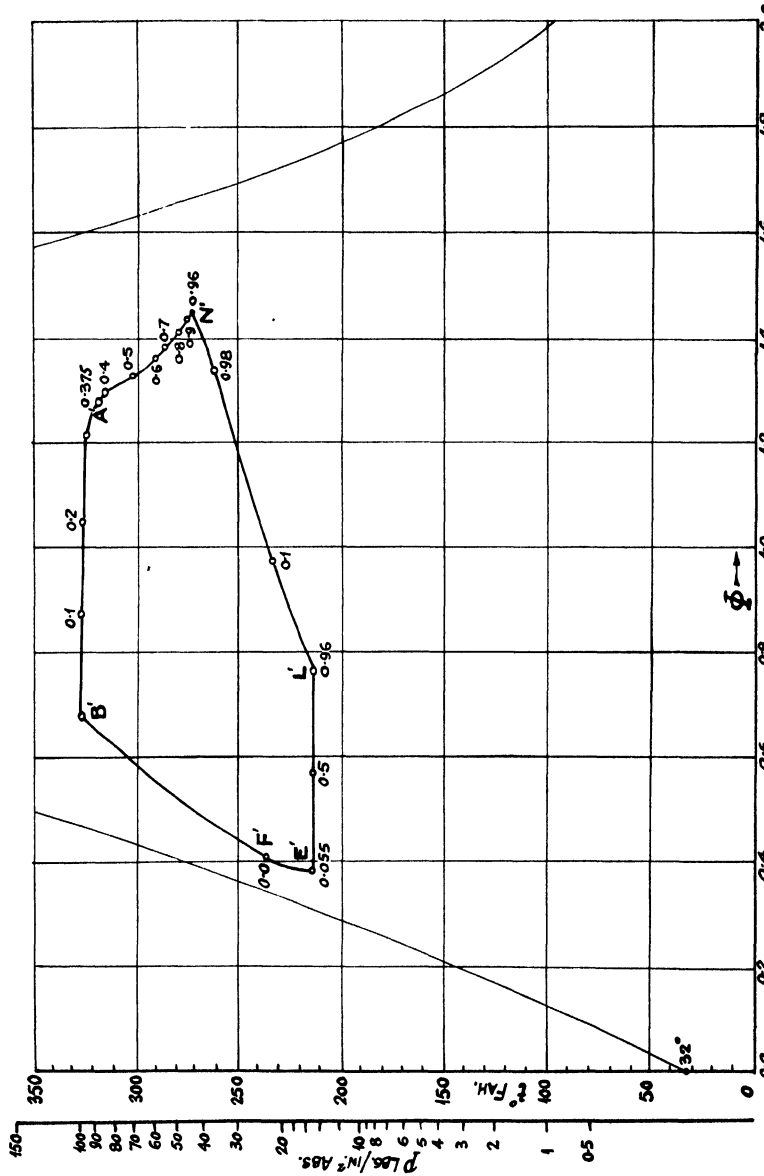

P Lbs./in² Abs.
t° Fah.
Φ
A
B'
N'
L'
E'
F'
0·375
0·4
0·5
0·6
0·7
0·8
0·9
0·96
0·98
0·2
0·1
0·055
32°

Fig. 97.

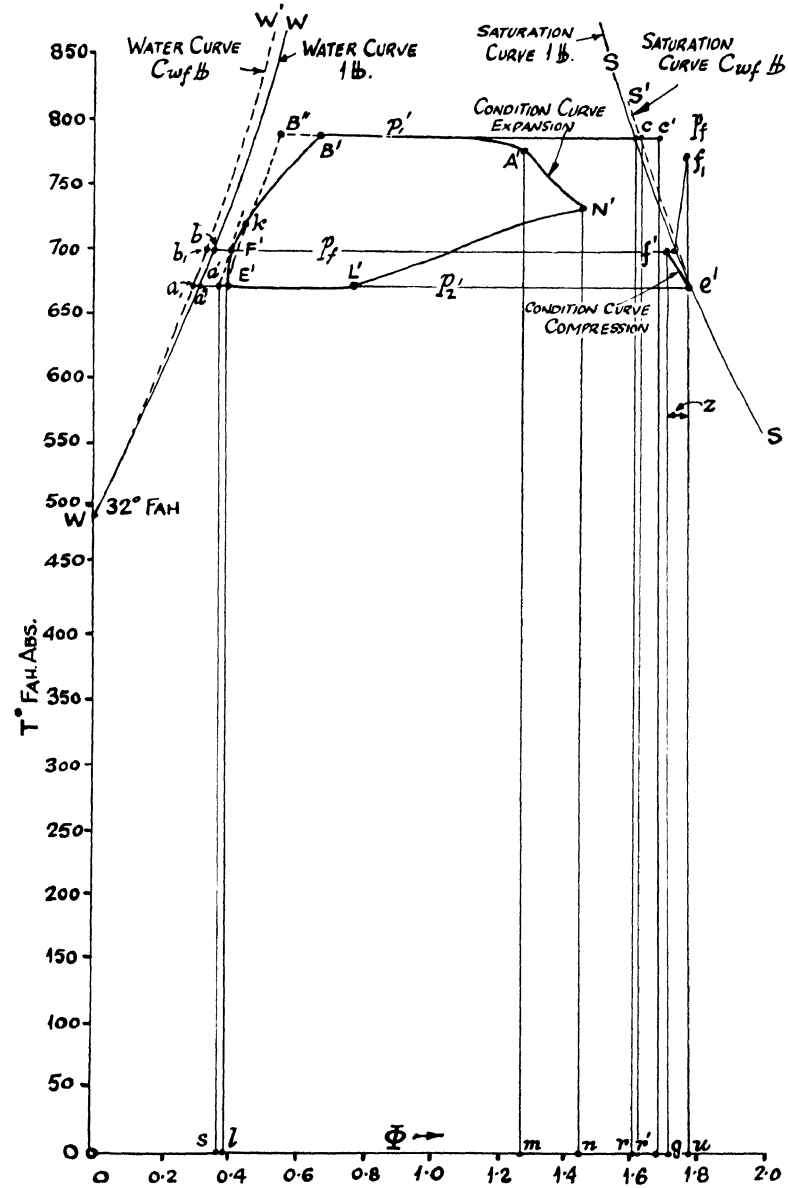

W' W
WATER CURVE C wf lb
WATER CURVE 1 lb.
SATURATION CURVE 1 lb.
SATURATION CURVE C wf lb
S
S'
CONDITION CURVE EXPANSION
CONDITION CURVE COMPRESSION
B''
B'
p1'
A'
N'
c c'
pf
f1
f'
e'
b
b1
k
F'
pf
a
a1
E'
L'
p2'
z
S
W
32° FAH
T° FAH. ABS.
850
800
750
700
650
600
550
500
450
400
350
300
250
200
150
100
50
0
Φ
s
l
m
n
r
r'
q
u
0
0·2
0·4
0·6
0·8
1·0
1·2
1·4
1·6
1·8
2·0

FIG. 98.

By definition the volume factor used in the transference of the diagram is—

$$C=\frac{1}{w}=\frac{1}{(w_e+w_f)}$$

so that

$$1=Cw_e+Cw_f \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (15)$$

At the point of compression E, where the pressure is $p_e$ let $\phi_{we}$ be the water entropy per lb., then by multiplying (15) throughout by $\phi_{we}$

$$\phi_{we}=Cw_e\phi_{we}+Cw_f\phi_{we}$$

and

$$Cw_e\phi_{we}=\phi_{we}-Cw_f\phi_{we} \quad . \quad . \quad . \quad . \quad . \quad . \quad (16)$$

If then the water curve WW′ is drawn for the TΦ cylinder feed of $Cw_f$ lb., the intercept $a_1a$ between it and the water curve WW for 1 lb. will give the value of $Cw_e\phi_{we}$, the water entropy for $Cw_e$ lb. As the steam is assumed to be dry at E′, $a$E′ will be the evaporation entropy for $Cw_e$ lb. and $a_1$E′ the total entropy. Since the weight of clearance steam between E′ and F′ remains constant, the intercept between WW and E′F′ at any pressure level gives the corresponding evaporation entropy. Thus at F′, the end of compression, the evaporation entropy of $Cw_e$ lb. is given by $b$F′. The evaporation entropy of this quantity of steam, if dry, would be $Cw_e\phi_{ef}$, where $\phi_{ef}$ is the evaporation entropy per lb. at $p_f$. Hence the real quality of the steam in the clearance at F′ is given by—

$$q_f=\frac{b\text{F}'}{Cw_e\phi_{ef}}=\frac{q_{fv}}{Cw_e} \quad . \quad . \quad . \quad . \quad . \quad . \quad (17)$$

where $q_{fv}$ is the virtual quality at F′.

If the *real* quality points at one or two pressures during compression are calculated from Equation 17 and located on the TΦ field, the condition curve $e'f'$ for compression is obtained, as shown in Fig. 98.

The area $f'e'ug$, below this curve down to the absolute zero of temperature shows the heat $h_{c1}$ given to the wall per lb. during compression, and hence the actual heat absorbed by the wall during compression per $Cw_e$ lb. is

$$h_c'=Cw_eh_{c1} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (18)$$

This heat can be determined in another way. The water curve and virtual condition curve during compression, Fig. 98, are shown to a larger scale in Fig. 99. If the water curve $agb$ and saturation curve E′$f$ are drawn for $Cw_e$ lb. of steam, as shown in Fig. 100, then the correspondingly lettered lengths on the two diagrams are the same; $agb$ is the water curve, $a'g'$F′ the adiabatic through F′. In Fig. 99 F′$a'$ is drawn through the final point F′ parallel to the water curve WW′, since obviously this gives the total entropy a constant value between E′ and F′. The area below E′F′$a'$, Fig. 99, down to absolute zero of temperature is the same as the area below

the curve E′F′, Fig. 100, and is therefore the value of the heat $h_c'$ given to the wall by $Cw_e$ lb. during compression.

In order to avoid appreciable errors $h_{c1}$ should be calculated from the area below the actual condition curve $e'f'$. It is given to a first approximation by—

$$h_{c1}=w_e z\,\frac{(T_f-T_2')}{2} \quad . \quad . \quad . \quad . \quad . \quad . \quad (19)$$

where $T_f$ and $T_2'$ are the absolute temperatures at $p_f$ and $p_2'$, and $z$ the difference of the total entropies $\Phi_2$ and $\Phi_{wf}$ of dry and wet steam at $p_2'$ and $p_f$.

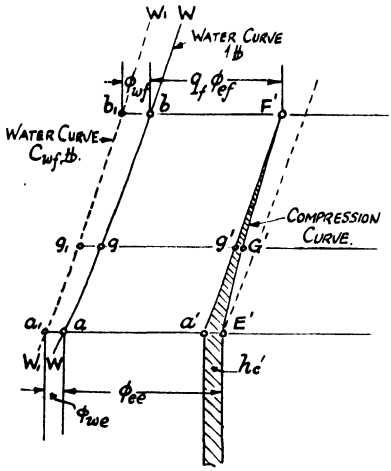



FIG. 99.

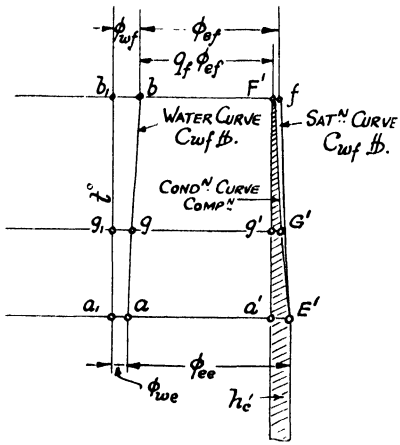



FIG. 100.

The clearance steam, however, need not necessarily be wet at the end of compression. As already pointed out, if the compression curve E′F′, Fig. 98, is parallel to WW, the water curve for 1 lb., the compression is adiabatic, and with dry steam at E′ it follows that the steam is superheated at all pressures between $p_2'$ and $p_f$, as the compression curve referred to 1 lb. of clearance steam falls in the superheat field.

In this case $e'f_1$, Fig. 98, is the compression curve and the compression heat area is reduced to zero, or $h_{c1}=0$.

When $f'$ falls to the right of $e'f_1$ the compression curve will slope to the right and the area below it will represent heat given to the clearance steam by the wall, so that $h_{c1}$ has to be reckoned as a negative quantity in the heat balance. If $f'$ falls to the left of $e'f_1$ the compression curve is inclined to the left, and as in the case of wet steam the area below it represents heat given by the steam to the walls, and $h_{c1}$ is reckoned as a positive quantity.

The position of $f_1$ in the superheat field is located when the total entropy of the superheated steam is determined.

In this case $q_{fv}/Cw_e > 1$, when the steam is superheated. Let this ratio be denoted by $k$, then the total entropy value will be—

$$\Phi_{sf} = \phi_{wf} + k\phi_{ef} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (20)$$

where $\phi_{wf}$ and $\phi_{ef}$ are the water and evaporation entropies at $p_f$. The intersection of the entropy vertical with the $p_f$ curve in the superheat field gives the point $f_1$.

Intermediate points between $e'$ and $f_1$ can be obtained by the same process.

The TΦ cylinder feed which has heat added and abstracted by the external agency is $Cw_f$ lb., and if it is supposed, in the first instance, to start as $Cw_f$ lb. of water at E′, Fig. 98, where the pressure and temperature are $p_2'$ and $t_2'$, then E′ may be taken as the initial state point for the water curve E′B″, which will run parallel to the curve for WW′.

The evaporation entropy $Cw_f\phi_{we}$, when set off from E′, will give the point $e'$, which will fall on the saturation curves SS for 1 lb. The evaporation entropy at $p_1'$ will be $Cw_f\phi_{ef}$, and when set off from B″ will give the point C on the curve S′$e'$, which is the saturation curve for $Cw_f$ lb. of TΦ feed.

Actually if the steam at stop-valve pressure $p_1$ is greater than the admission pressure $p_1'$ the steam may have a slight superheat at $p_1'$, and the state point C will fall above the position shown on the $p_1'$ curve in the superheat field. To avoid confusion this condition is not shown in Fig. 98. In either case the heat supplied per $Cw_f$ lb. is given by the area $l$E′B″C$r$=H′. It is not necessary, however, to find this area, as the heat value can be calculated from the known steam conditions at the stop-valve.

Thus $$H' = Cw_f(H_1 - h_2) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (21)$$

where $H_1$ is the total heat of dry steam at stop-valve pressure $p_1$ and $h_2$ the water heat at back pressure $p_2'$. If the steam is superheated at $p_1$, then $H_{s1}$ is to be used instead of $H_1$. The actual heat supplied per cycle is H=H′/C or

$$\left.\begin{aligned} H &= w_f(H_1 - h_2) \\ \text{or} \quad H &= w_f(H_{s1} - h_2) \end{aligned}\right\} \quad . \quad . \quad . \quad . \quad . \quad . \quad (22)$$

The heat converted into useful work, or the indicated heat per $Cw_f$ lb., is given by the area E′B′A′N′L′=$h_i'$, which can also be obtained by reduction from the PV diagram area. The indicated heat per cycle is given by—

$$h_i = h_i'/C \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (23)$$

During admission between F′ and A′ the walls absorb an amount of heat $h_a$ per cycle. During expansion between A′ and N′ the walls return per $Cw_f$ lb. an amount of heat given by the area below A′N′ or A′N′$nm$=$he''$, and the actual amount restored per cycle is

$$h_e' = h_e''/C \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (24)$$

At release the heat in $Cw_f$ lb. is given by the area below E'L'N' or $l$E'L'N'$n$=$h_{e1}$; the actual amount per cycle is—

$$h_e = h_{e1}/C \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (25)$$

The amount of heat radiated per cycle can be most conveniently expressed as a fraction of the heat supply H, and denoted by $h_r$.

The heat supplied per cycle less the indicated heat and radiated heat must obviously equal the net heat to the walls plus the heat in the exhaust at release—

$$\text{or } (H - h_i - h_r) = (h_c + h_a - h_e') + h_e$$

Hence the heat given to the wall per cycle during admission is

$$h_a = H - h_i - h_e - h_c + h_e' - h_r \quad . \quad . \quad . \quad . \quad (26)$$

During exhaust between N' and E' the heat which is refunded by the walls is given by—

$$h' = (h_c + h_a - h_r - h_c') \quad . \quad . \quad . \quad . \quad . \quad (27)$$

This refund, however, can be more conveniently determined directly by difference when $h_i$, $h_e$, and $h_r$ are evaluated.

Thus
$$h' = (H - h_i - h_r - h_e) \quad . \quad . \quad . \quad . \quad . \quad (28)$$

When these items are found from the chart or by calculation, a heat balance showing the distribution of the heat supply during a cycle can be drawn up, as shown below.

When the engine has a steam jacket (see Art. 116), and the condensate in the jacket is $w_J$ lb./cycle and the latent heat at the jacket pressure is L, the heat supply per cycle becomes—

$$(H + w_J L) = {}_1H \quad . \quad . \quad . \quad . \quad . \quad . \quad (29)$$

Alternatively the value of H, in each case, can be obtained directly from the tables, since

$$H = (H_1' - h_2')w_f \quad . \quad . \quad . \quad . \quad . \quad . \quad (30)$$

where $H_1'$ is the total heat of the steam at $p_1'$ and $h_2'$ the total heat of the water at $p_2'$.

In order to take account of the jacket heat on the TΦ diagram, Fig. 98, the additional entropy value CC' is set off at $p_1'$ from C for $Cw_J L$ heat units, that is $cc' = Cw_J L'/T_1'$, where $T_1'$ is the absolute temperature at $p_1'$. The larger area $l$E'B''C'$r'$, thus obtained, is then used for the graphical evaluation of the foregoing heat balance quantities.

*Example* 5.—The reduced diagram shown in Fig. 98 is that for the engine using initially dry saturated steam and considered in Examples 1 to 4.

This has a TΦ cylinder feed $Cw_f$=0·951 lb. The actual cylinder feed is $w_f$=0·0128 lb. and the clearance steam $w_e$=0·00165 lb., while the volume factor is C=29. Find the distribution of heat during the cycle and draw up a tabular heat balance.

The heat scale of this smaller TΦ diagram (original size) is 1 in.$^2$ =40 B.Th.U. The area values obtained by planimeter are:

$lE'B''Cr$=24·1 in.$^2$; $E'B'A'N'L'$=2·030 in.$^2$; $f'e'ug$=1·028 in.$^2$; $lE'N'n$=18·6 in.$^2$; $A'N'nm$=3·68 in.$^2$ Assuming a radiation loss, say, 1 per cent. of the heat supply per cycle, draw up a heat balance showing the distribution of the heat in the cylinder per cycle.

Heat supplied $H=\frac{24\cdot1\times40}{29}=33\cdot17$ B.Th.U./cycle

Indicated heat $h_i=\frac{2\cdot03\times40}{29}=2\cdot80$ ,,

Refund by wall during expansion—

$$h_e'=\frac{3\cdot68\times40}{29}=5\cdot076 \text{ B.Th.U./cycle}$$

Heat in exhaust at release $h_e=\frac{18\cdot6\times40}{29}=25\cdot655$ ,,

Heat radiated (assumed)—

$$h_r=0\cdot01H=0\cdot01\times33\cdot17=0\cdot3317 \quad ,,$$

Heat absorbed by wall during compression—

$$h_c=1\cdot028\times40\times w_c=1\cdot028\times40\times0\cdot00165=0\cdot0680 \quad ,,$$

Heat absorbed by wall during admission—

$$h_a=H-h_i-h_r-h_e-h_c+h_e'$$
$$=33\cdot17-2\cdot8-0\cdot3317-25\cdot655-0\cdot068+5\cdot076$$
$$=(38\cdot246-28\cdot861)=9\cdot385 \text{ B.Th.U./cycle.}$$

Heat refunded by the wall during exhaust—

$$h'=(H-h_i-h_r-h_e)$$
$$=(33\cdot17-2\cdot8-0\cdot331-25\cdot655)=4\cdot384 \text{ B.Th.U./cycle .}$$

HEAT BALANCE (Example 5)

| | B.Th.U. | Per cent. | | B.Th.U. | Per cent. |
|---|---|---|---|---|---|
| Heat supplied per cycle, H or $H_1$ . . | 33·170 | 100 | Indicated heat $h_i$ . | 2·800 | 8·455 |
| | | | Radiation $h_r$ . . . | 0·3317 | 1·000 |
| | | | Heat in exhaust, $h_e$ . | 25·655 | 77·344 |
| | | | Heat refunded by wall during exhaust, $h'$ . . . | 4·384 | 13·215 |
| Total . . . . | 33·170 | 100 | Total . . | 33·170 | 100 |

**113. Thermal Efficiency and Efficiency Ratio.**—In order to evaluate the heat exchanges in the cylinder alone, the limiting pressures taken are the admission and back pressures. As already stated in Chapter IV in connection with thermal efficiency and efficiency ratio on the straight Rankine cycle, the pressure limits usually taken are those at the stop-valve and the exhaust flange. On this

basis the heat supply as shown in Fig. 101 is given for $Cw_f$ lb. of TΦ feed by the area below $E_1B_1C_1$.

The Rankine cycle area for this quantity is given by $B_1C_1L_1E_1$

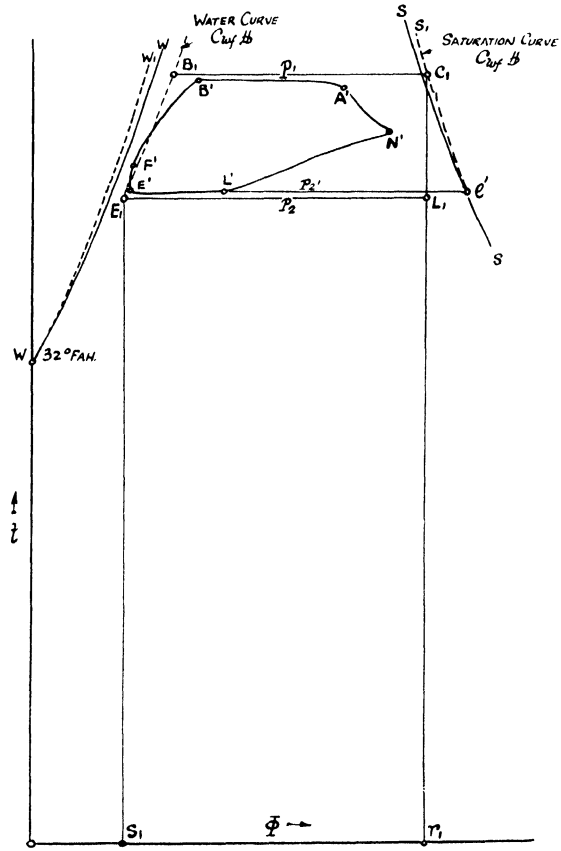



FIG. 101.

and the actual indicated area by E′B′A′N′L′. The efficiencies are, therefore—

$$\text{Actual thermal efficiency (indicated) } \eta_1 = \frac{E'B'A'N'L'}{S_1E_1B_1C_1r_1}$$

$$\text{Rankine } \eta_r = \frac{B_1C_1L_1E_1}{S_1E_1B_1C_1r_1}$$

$$\text{and the efficiency ratio is } \epsilon = \frac{E'B'A'N'L'}{B_1C_1L_1E_1}$$

The accuracy of the transference from PV to TΦ field can be checked by a comparison of the PV and TΦ indicator card areas.

If the areas are—

TΦ, A in.², and scales TΦ, 1 in.²=S B.Th.U./lb.
PV, $a$ in.², „ „ PV, 1 in.²=$s$ „
and C=volume factor,

then $\frac{AS}{C}=as$ and the TΦ area should be—

$$A=\frac{Cas}{S} \text{ in.}^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (31)$$

The TΦ areas are, of course, not necessary for the calculation of either the efficiency or efficiency ratio values. These can be more expeditiously determined by direct calculation, using the methods given in Chapter IV.

Thus if heat supply per lb. feed, $H=(H_1-h_2)$
indicated consumption rate=$w_i$ lb./I.H.P. hr.

then $$\eta_I=\frac{2545}{w_iH}=\frac{2545}{w_i(H_1-h_2)} \quad . \quad . \quad . \quad . \quad . \quad (32)$$

If $H_\phi$=adiabatic heat drop between $p_1$ and $p_2$

$$\eta_r=\frac{H_\phi}{(H_1-h_2)} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (33)$$

and $$\epsilon=\frac{\eta_I}{\eta_r}=\frac{2545}{w_iH_\phi} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (34)$$

*Example* 6.—Check the accuracy of the TΦ indicator card transference for the diagram of Example 5, and find the thermal efficiency and efficiency ratio from the diagram areas and by direct calculation.

The stop-valve pressure is 105 lb./in.² abs.; exhaust pressure 15 lb./in.² abs.; total consumption (corrected for leakage) 590·4 lb./hr. I.H.P.=19·8.

The diagram scales of the originals are—
PV, 1 in.=43 lb./in.²; 1 in.=0·85 ft.³
TΦ, 1 in.=50° F.; 1 in.=0·2 entropy unit.

$$\therefore \text{ PV heat scale is, 1 in.}^2=\frac{43\times0{\cdot}85\times144}{778}=0{\cdot}6764 \text{ B.Th.U.}=s$$

TΦ „ „ „ 1 in.²=10 B.Th.U./lb.=S

The areas of the diagrams when planimetered are PV, $a$=4·14 in.²; TΦ, A=8·10 in.² Volume factor C=29

$$\therefore \text{ the TΦ area should be } A=\frac{Cas}{S}=\frac{29\times4{\cdot}14\times0{\cdot}6764}{10}=8{\cdot}12 \text{ in.}^2$$

which is a sufficiently close check and shows that the transference may be taken as satisfactory. In work of this nature the student should always bear in mind the liability of "personal" error in the planimeter work.

The Rankine cycle area $E_1B_1C_1L_1$, Fig. 101, is 13·6 in.², which checks with the heat drop from the HΦ diagram, since $\frac{143\times0{\cdot}951}{10}=13{\cdot}6$. The area below $E_1L_1$ works out at 82·57 in.²,

and the heat input for $Cw_f$=0·951 lb. is (13·6+82·57)=96·17 in.$^2$ Hence the thermal efficiency is—

$$\eta_1=\frac{8\cdot12}{96\cdot17}=0\cdot0844,$$

and the efficiency ratio is—

$$\epsilon=\frac{8\cdot12}{13\cdot6}=0\cdot597$$

By direct calculation, since $H_1$=1192·3 ; $h_2$=181 ; $(H_1-h_2)$ =1011·3 ; $H_\phi$=143 ; and $\frac{W}{I}=\frac{590\cdot4}{19\cdot8}=29\cdot8=w_i$

$$\eta_1=\frac{2545}{w_i(H_1-h_2)}=\frac{2545}{29\cdot8\times1011\cdot3}=0\cdot0844$$

and

$$\epsilon=\frac{2545}{w_iH_\phi}=\frac{2545}{29\cdot8\times143}=0\cdot597$$

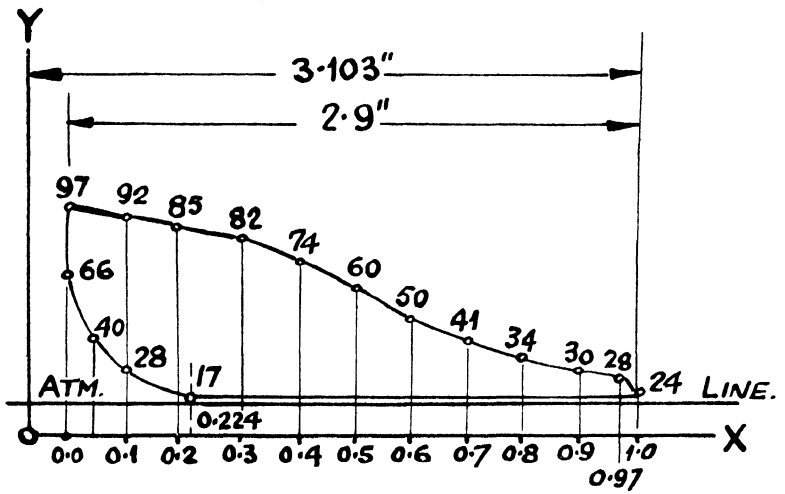



FIG. 102.

The foregoing methods of diagram transference and calculation for heat exchanges given in Art. 112, are also applicable in the case of the unjacketed engine supplied with superheated steam. The next example is worked out for the diagram from Prof. Ripper's paper, which, with the condition curve, is shown in Fig. 93.

*Example* 7.—The indicator diagram, Fig. 102, is the mean of the diagrams taken from a two-cylinder Schmidt single-acting non-condensing engine, supplied with steam at 102 lb./in.$^2$ abs., superheated 338° Fahr. Stroke volume, 0·27 ft.$^3$ ; clearance, 7 per cent. ; total consumption (two cylinders) at speed of 173·5 rev./min., 323·08 lb./hr. ; cut-off at 0·43×stroke ; release at 0·97×stroke ; compression at 0·224×stroke. Transfer the diagram to the TΦ field

and determine the heat exchanges between the steam and cylinder wall, on the assumption that there is a radiation loss of 2 per cent. of the heat supply.

Cylinder feed, $w_f = \frac{W}{60 \times 2N} = \frac{323{\cdot}08}{120 \times 173{\cdot}5}$ $\quad W = 323{\cdot}08$ lb./hr. $\quad N = 173{\cdot}5$

$$= 0{\cdot}0155 \frac{\text{lb.}}{\text{str.}}$$

Stroke volume $V_o = 0{\cdot}27$, $l = 2{\cdot}9$ in., and 1 in. $= \frac{0{\cdot}27}{2{\cdot}9} = 0{\cdot}0931$ ft.$^3$

$$V_e = 0{\cdot}850 \times 0{\cdot}0931 = 0{\cdot}07915 \text{ ft.}^3$$

$$P_e = 17, \; v_e = 23{\cdot}37 \quad \therefore \; w_e = \frac{V_e}{v_e} = \frac{0{\cdot}07915}{23{\cdot}37} = 0{\cdot}00338 \text{ lb.}$$

$$\therefore \quad w = (w_f + w_e) = (0{\cdot}0155 + 0{\cdot}00338) = 0{\cdot}01888 \text{ lb.}$$

$$\text{Volume factor } C = \frac{1}{0{\cdot}01888} = 53$$

The PV volume values scaled from the diagram and the calculated $V_{T\Phi}$ volumes are given in the tables below, with the derived virtual and actual qualities.

OUT-STROKE.

| $x$ | 0 | 0·1 | 0·2 | 0·3 | 0·35 | 0·4 | 0·43 | 0·5 | 0·6 | 0·7 | 0·8 | 0·9 | 0·97 | 0·98 | 1·0 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| $p$ | 97 | 92 | 85 | 82 | 78 | 74 | 72 | 60 | 50 | 42 | 35 | 31 | 28 | 24 | 20 |
| $v$ | 4·58 | 4·813 | 5·183 | 5·362 | 5·618 | 5·902 | 6·056 | 7·184 | 8·52 | 10·03 | 11·9 | 13·4 | 14·66 | 16·93 | 20·08 |
| $V_{T\Phi}$ | 1·00 | 2·432 | 3·864 | 5·30 | 6·01 | 6·725 | 7·155 | 8·156 | 9·587 | 11·018 | 12·45 | 13·88 | 14·88 | 15·13 | 15·31 |
| $\frac{V_{T\Phi}}{v}$ | 0·218 | 0·504 | 0·7455 | 0·99 | 1·069 | 1·14 | 1·18 | 1·134 | 1·125 | 1·068 | 1·030 | 1·013 | 1·005 | 0·89 | 0·765 |
| $t_{sv}$ or $t_s$ | — | — | — | — | 48 | 96 | 125 | 92 | 77 | 44 | 20 | 10 | 4 | — | — |
| | | | | | | | ← EXPANSION → | | | | | | | | |

IN-STROKE.

| $x$ | 0 | 0·05 | 0·1 | 0·2 | 0·224 | 0·5 | 0·97 | 1·0 |
|---|---|---|---|---|---|---|---|---|
| $p$ | 66 | 40 | 29 | 18 | 17 | 17 | 17 | 20 |
| $v$ | 6·27 | 10·5 | 14·18 | 22·16 | 23·37 | 23·37 | 23·37 | 20·08 |
| $V_{T\Phi}$ | 1·0 | 1·716 | 2·43 | 3·87 | 4·2 | 8·2 | 14·7 | 15·31 |
| $\frac{V_{T\Phi}}{v} = q_v$ | 0·160 | 0·164 | 0·171 | 0·175 | 0·1797 | 0·35 | 0·63 | 0·765 |
| $q$ | 0·90 | 0·92 | 0·954 | 0·98 | 1·0 | — | — | — |
| | ← COMPRESSION → | | | | | | | |

The real qualities during compression are obtained from $q=q_v/Cw_e$ $=q_v/0{\cdot}17914$.

The transferred diagram E′B′A′N′L′, obtained by locating the real and virtual quality points in the TΦ field, is shown in Fig. 103.

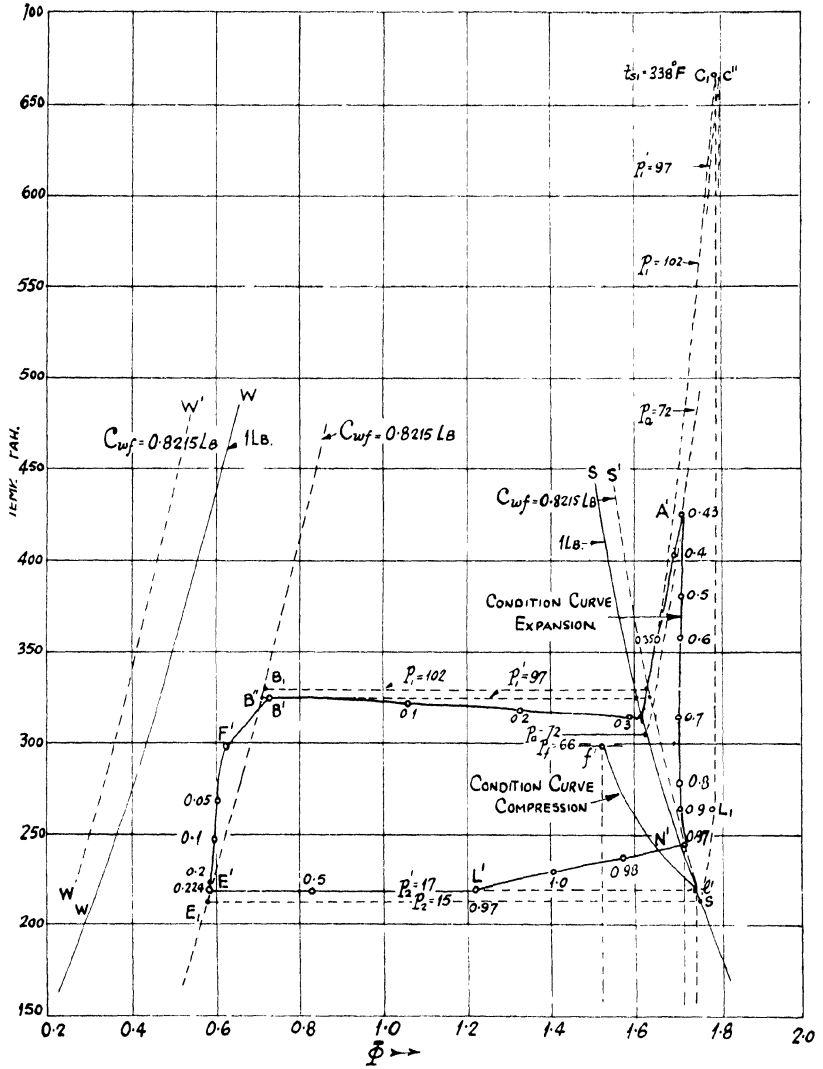



FIG. 103.

These points are marked with the corresponding fractional values ($x$) of the stroke.

In this case the value of $Cw_f$ is $53\times0{\cdot}0155=0{\cdot}8215$ lb.

The Rankine cycle diagram $E_1B_1C_1L_1S$ is drawn for this quantity

between the stop-valve and exhaust pressures 102 lb./in.$^2$ abs. and 15 lb./in.$^2$ abs. By planimeter the area of the transferred indicator diagram is 10·8 in.$^2$, and the Rankine cycle area is 15·65 in.$^2$

In order to avoid confusion the curve of $p_1'=97$ has not been drawn in completely in the superheat field, but merely indicated at the upper end. C″ being the state point at the maximum temperature obtained by throttling the steam at the stop-valve from 102 to 97 lb./in.$^2$ abs. The area below $E_1B''C''$ down to absolute zero is 96·87 in.$^2$, so that $H'=968·7$ and $H=H'/C=968·7/53=18·28$ B.Th.U./cycle.

The indicated heat is $h_i=108/53=2·037$ B.Th.U./cycle.

The expansion curve is practically an adiabatic to 0·8 and the small area below the curve between 0·8 and 0·97 is approximately 1·08 in.$^2$, giving $h_e'=10·8$ and $h_e'=\frac{10·8}{53}=0·0203$ B.Th.U./cycle or a negligible refund during expansion. The radiated heat (assumed) is $h_r=0·02\times H=0·02\times 18·28=0·03656$ B.Th.U./cycle. The heat absorbed by the wall during compression per lb. of steam is the area below $e'f'$. This is 15·9 in.$^2$ and $h_c=15·9\times 0·00338=0·5374$ B.Th.U./cycle. The area below the exhaust curve E′L′N′ is 77·64 in.$^2$, so that $h_e=776·4/53=14·65$ B.Th.U./cycle.

The heat absorbed by the wall during admission (F′ to A′) is—

$$h_a=(H-h_i)-h_e+h_e'-h_c+h_r$$
$$=(18·28-2·037-14·65+0·0203-0·5374+0·3656)=1·44$$

or 7·9 per cent. of the heat supply.

The heat refunded by the wall during exhaust is—

$$h'=H-h_i-h_r-h_e$$
$$=(18·28-2·037-0·3656-14·65)=1·2274.$$

HEAT BALANCE

| | B.Th.U. | Per cent. | | B.Th.U. | Per cent. |
|---|---|---|---|---|---|
| Heat supplied per cycle, H . . . | 18·28 | 100 | Indicated heat $h_i$ . | 2·0370 | 11·1 |
| | | | Radiation, $h_r$ . . | 0·3656 | 2·0 |
| | | | Heat in exhaust $h_e$ . | 14·650 | 80·2 |
| | | | Heat refunded by wall during exhaust, $h'$ . . . | 1·2274 | 6·7 |
| Total . . . . | 18·28 | 100 | Total . . . | 18·28 | 100 |

*Example* 8.—Check the accuracy of the diagram transference in the case of Example 7, and calculate the thermal efficiency and efficiency ratio from the diagram areas and by direct calculation.

By planimeter the area of the indicator diagram $a=1{\cdot}473$ in.$^2$, $l=2{\cdot}9$ in., and the scale is 1 in.$=80$ lb./in.$^2$

Mean effective pressure $p_e=\dfrac{1{\cdot}473\times80}{2{\cdot}9}=40{\cdot}64$ lb./in.$^2$; $N=173{\cdot}5$, $V_o=0{\cdot}27$.

$$\therefore \text{ I.H.P.}=\frac{40{\cdot}64\times144\times0{\cdot}27\times173{\cdot}5}{33000}=8{\cdot}3$$

W$=161{\cdot}54$ lb./hr. for 1 cylinder.

$$\therefore\ w_i=\frac{161{\cdot}54}{8{\cdot}3}=19{\cdot}4 \text{ lb./I.H.P. hr.}$$

Volume scale of diagram 1 in. $=\dfrac{0{\cdot}27}{2{\cdot}9}=0{\cdot}0931$ ft.$^3$

Pressure scale of diagram 1 in. $=80\times144$ lb./ft.$^2$

Heat scale 1 in.$^2=\dfrac{0{\cdot}0931\times80\times144}{778}=1{\cdot}378$ B.Th.U.

Area of TΦ diagram should be $A=\dfrac{Cas}{S}$ S, 1 in.$^2=10$ B.Th.U.

$$=\frac{53\times1{\cdot}378\times1{\cdot}473}{10}=10{\cdot}76 \text{ in.}^2$$

The planimetered area is 10·8, and shows the transference is satisfactory.

From the areas $\eta=\dfrac{10{\cdot}8}{96{\cdot}87}=0{\cdot}1114$ or 11·14 per cent.; $\epsilon=\dfrac{10{\cdot}8}{15{\cdot}65}=0{\cdot}69$.

By calculation $\eta=\dfrac{2545}{w_i(H_{s1}-h_2)}=\dfrac{2545}{19{\cdot}4\times1185{\cdot}2}=0{\cdot}1107$ or 11·07 per cent. against 11·14.

The heat drop between 102 lb./in. abs. and 15 lb./in.$^2$ abs. is $H_\phi=191$ B.Th.U./lb. Hence the efficiency ratio is—

$\epsilon=\dfrac{2545}{w_iH_\phi}=\dfrac{2545}{19{\cdot}4\times191}=0{\cdot}688$ against 0·69 from the area calculation.

**114. Actual Condition of Steam after Exhaust from the Cylinder.** — The condition of the steam after it issues from the exhaust branch of the cylinder, either to the atmosphere or to a condenser or the receiver of a multiple-expansion engine, can be determined from the law of conservation of energy.

Under conditions of steady flow through the engine the total energy of the steam as it passes the stop-valve must be equal to its total energy as it passes the exhaust branch plus the heat converted into mechanical work on the piston and heat lost by radiation. The total energy per lb. of steam is the sum of the pressure energy $144pv/J$, the internal energy I, and the kinetic energy K. Denoting

the conditions at the stop-valve and exhaust flange by subscripts 1 and 2—

$$144p_1v_1/J+I_1+K_1=144p_2v_2/J+I_2+K_2+H_i+H_r$$

where $H_i$ and $H_r$ are indicated and radiated heat per lb. This can be written as—

$$H_1+K_1=H_2'+K_2+H_i+H_r$$

so that the total heat of the steam after passing the exhaust branch is

$$H_2'=H_1-(H_i+H_r)+(K_1-K_2)$$

When the steam is initially superheated $H_1$ is replaced by $H_{s1}$, and when wet by $H_{w1}$.

In general, the kinetic term $(K_1-K_2)$ is a negligibly small quantity and for all practical purposes the total heat may be taken as—

$$H_2'=H_1-(H_i+H_r)$$

If $H_2'$ is greater than the total heat of dry steam at $p_2$, the exhaust pressure, the steam is superheated ; if less, it is wet. Thus—

$$H_2+c_{p2}t_{s2}=H_2'$$

$$\text{and} \quad t_{s2}=\frac{H_2'-H_2}{c_{p2}} \quad . \quad . \quad . \quad . \quad . \quad . \quad (35)$$

$$\text{Again} \quad H_2-(1-q_2)L_2=H_2'$$

$$\text{and} \quad (1-q_2)=\frac{H_2'-H_2}{L_2} \quad . \quad . \quad . \quad . \quad . \quad (36)$$

It is a common practice in applying this calculation to the successive stages of a multiple-expansion engine to neglect radiation effect and take the rougher approximation—

$$H_2'=(H_1-H_i) \quad . \quad . \quad . \quad . \quad . \quad . \quad (37)$$

The value of the total heat at exhaust is thus approximated by the difference of areas $(S_1E_1B_1C_1r_1-E'F'B'A'N'L')$, Fig. 101, divided by $Cw_f$.

The estimation of $H_2'$ can be most expeditiously carried out on the HΦ chart. The case for initially dry and finally wet steam is shown in Fig. 104. Using the HΦ chart notation of Chapter III, F is the initial state point at the stop-valve pressure $p_1$ and $H_1$ the total heat per lb. of dry steam. The indicated heat per lb. is given by—

$$H_i=\frac{2545\times I}{W} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (38)$$

where I is the total horse-power developed with a consumption of W lb./hr. If $x$ is the fraction of the heat supply per lb. lost by radiation, then—

$$H_r=x(H_1-h_2) \quad . \quad . \quad . \quad . \quad . \quad . \quad (39)$$

If in the first instance $H_r$ is neglected, the difference $H_{w2}''$

$=(H_1-H_i)$ gives the state point G'' and quality $q_2''$. With complete adiabatic expansion between the initial and exhaust pressures $p_1$ and $p_2$ the heat drop would be $FG'=H_\phi$, and the difference N'G' would represent an equivalent reheat $h_f$ due to wall action and incomplete expansion, any small kinetic effect being neglected.

The ratio FN''/FG' gives the efficiency ratio on the Rankine cycle, reckoned on the stop-valve and exhaust flange conditions. When the radiation is allowed for, the total heat of the exhaust is decreased by $H_r$. This on subtraction from $H_{w2}''$ gives the final total heat $H_{w2}$ at $p_2$, the state point being moved to G, which defines the final quality $q_2$. The equivalent reheat is thus reduced to $G'N=h_f'$.

Thus to estimate the probable quality after exhaust, from the

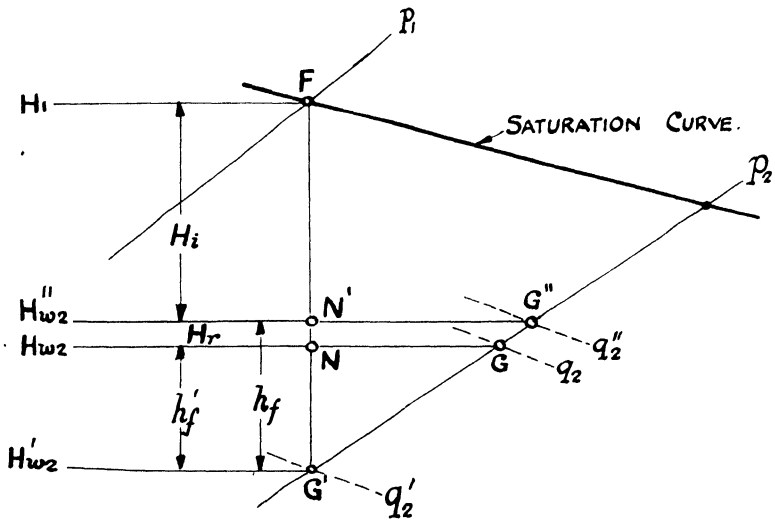



FIG. 104.

$H_\phi$ chart, for an estimated radiation loss $H_r$ and indicated heat $H_i$, set off $FN=(H_i+H_r)$ vertically from the initial state point F on $p_1$, and draw a horizontal NG to cut the $p_2$ curve at G, and read the corresponding quality $q_2$.

The same process applies in the case of initially superheated steam. This method is very serviceable for ascertaining the probable condition of the exhaust steam at the successive stages of a multiple-expansion engine.

It is of interest to note that this method acts as a check on the accuracy of the heat balance items deduced from the TΦ diagram analysis.

From Equation 28

$$h'=H-(h_i+h_r)-h_e$$

so that

$$(h'+h_e)=H-(h_i+h_r),$$

for $w_f$ lb. cylinder feed ; hence per lb. of feed—

$$\frac{h'+h_e}{w_f}=\frac{H-(h_i+h_r)}{w_f}=(H_1-h_2)-(H_i+H_r)$$

or

$$h_2+\frac{h'+h_e}{w_f}=H_1-(H_i+H_r)=H_2' \quad . \quad . \quad . \quad (40)$$

In the case of a jacketed engine $(H_i+H_r-\frac{w_J}{w_f}L_1)$ has to be used, where $w_J$ is jacket condensate per cycle and $w_f$ cylinder feed per cycle and $L_1$ latent heat of jacket steam.

*Example* 9.—Find the probable condition of the exhaust in the case of an engine using dry steam at 105 lb./in.$^2$ abs. at the stop-valve, and exhausting at 15 lb./in.$^2$ abs., when the I.H.P. is 19·8, the total consumption 590·4 lb./hr. and a radiation loss of 10·1 B.Th.U./lb. is assumed.

This is the case considered in Examples 1 to 5.

Here $$H_i=\frac{25451}{W}=\frac{2545\times 19\cdot 8}{590\cdot 4}=85\cdot 35 \text{ B.Th.U./lb.}$$

From the TΦ data of Example 6—

$$H_i=h_i/Cw_f=\frac{81\cdot 2}{0\cdot 951}=85\cdot 3 \qquad Cw_f=0\cdot 951$$

which is a good check.

$H_r=10\cdot 1$ $\therefore$ $(H_i+H_r)=(85\cdot 35+10\cdot 1)=95\cdot 45$ B.Th.U./lb.

at $p_1=105$, $H_1=1192\cdot 3$.

At $p_2=15$, $H_2=1151\cdot 2$, $L_2=970\cdot 2$

$H_2'=H_1-(H_i+H_r)=(1192\cdot 3-95\cdot 45)=1096\cdot 85$ B.Th.U./lb.

$$\therefore \quad (1-q_2)=\frac{H_2-H_2'}{L_2}=\frac{1151\cdot 2-1096\cdot 85}{970\cdot 2}=\frac{54\cdot 35}{970\cdot 2}=0\cdot 056$$

and $q_2=(1-0\cdot 056)=0\cdot 944$. This checks with the value given by the HΦ chart.

If the radiation were neglected, $H_2'=1106\cdot 95$ and $q_2$ would be increased to 0·954.

*Alternative Calculation from Heat Balance Items* (*Example* 5).—$h_2'=h_2=181$, $h_e=25\cdot 655$, $h'=4\cdot 384$, $w_f=0\cdot 0328$.

$$\therefore \ H_2'=h_2'+\frac{h_e+h'}{w_f}$$

$$=181+\frac{25\cdot 655+4\cdot 384}{0\cdot 0328}=181+\frac{30\cdot 039}{0\cdot 0328}$$

$$=181+915\cdot 67=1096\cdot 67 \text{ B.Th.U./lb.}$$

which practically checks with the previous figure.

*Example* 10.—Find the probable quality of the exhaust steam in the case of the engine, Example 7, supplied with steam at 102 lb./in.$^2$ abs. and superheated 338° F. and exhausting at 15 lb./in.$^2$ abs. ; I.H.P.=8·3 ; W=161·54 ; $H_r$=2 per cent. of heat supplied.

Check against the value obtained from the heat balance items $h_e$=14·65, $w_f$=0·0155, $h'$=1·2274.

At 102 lb./in.$^2$ and 338° F., $H_{s1}$=1366·2. At $p_2$=15, $H_2$=1151·2.

$$H_i=\frac{2545\times I}{W}=\frac{2545\times 8\cdot 3}{161\cdot 54}=130\cdot 75 \text{ B.Th.U./lb.}$$

$$H_r=(H_{s1}-h_2')x=0\cdot 02(1366\cdot 2-187\cdot 5)=0\cdot 02\times 1178\cdot 7=23\cdot 6:$$

$$\therefore H_2'=H_{s1}-(H_i+H_r)=(1366\cdot 2-130\cdot 75-23\cdot 6)$$
$$=1211\cdot 85 \text{ B.Th.U./lb.}$$

The steam, as might be expected, has a considerable superheat. From the HΦ diagrams $t_{s2}$=120° F.

Alternatively $H_2'=h_2'+\dfrac{h_e+h'}{w}$

$h_2'$=187·5
$h_e$=14·65
$h'$=1·2274
$w_f$=0·0155

$$=187\cdot 5+\frac{14\cdot 65+1\cdot 2274}{0\cdot 0155}$$

$$=187\cdot 5+\frac{15\cdot 8774}{0\cdot 0155}$$

$$=187\cdot 5+1024\cdot 34$$

$$=1211\cdot 84 \text{ B.Th.U./lb.}$$

which checks with the previous figure.

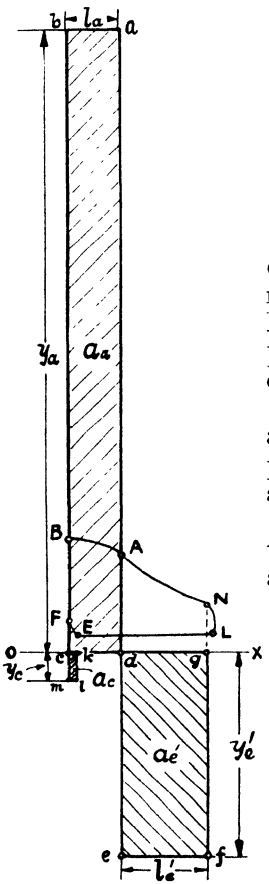



FIG. 105.

**115. Representation of Heat Exchanges on the PV Diagram.**—The graphical TΦ method of analysis, originally developed by Prof. Boulvin, has been substituted for the purely numerical method of Hirn, which was outlined in the previous editions of this text.

Hirn's method necessitates the progressive application of the energy law to successive phases of the cycle, and involves an intricate and tedious series of calculations.

In this system the heat quantities have to be exhibited by equivalent rectangular areas, which are superposed on the PV indicator diagram. An illustration is shown in Fig. 105.

Areas representing heat given up by the steam are sectioned thus ////, and those representing heat returned to the steam thus \\\\.

During compression E to F heat given to the wall is $h_c$, and is represented by the rectangle *cklm* of area $a_c$ drawn on *ck* as base.

$$a_c=\frac{h_c}{s} \quad \text{and} \quad y_c=\frac{a_c}{l_c}$$

OX is the line of zero pressure; $s$ is the heat scale of the diagram.

During admission between F and A heat given to the wall is $h_a$ and represented by the rectangle *cbad* of area $a_a$ drawn on *cd* as base.

$$a_a = \frac{h_a}{s} \quad \text{and} \quad y_a = \frac{a_a}{l_a}$$

During expansion between A and N the wall refunds heat $h_e'$ to the steam. This is represented by the rectangle *defg* of area $a_c'$ drawn on *dg* as base.

Since the total area *mbadkl* or $(a_c+a_a)$ is the heat given to the wall by the steam, and *defg* or $a_e'$ the refund up to release at N, the difference must be equal to the refund $h'$ by the wall during the exhaust period N to E, plus the heat $h_r$ lost during the cycle by radiation. Denoting the areas by the corresponding subscripts—

$$(a_c+a_a-a_e')=a'+a_r=a$$

The area $a$ can be most conveniently represented by a rectangle *nqrs* drawn on the stroke length $l$ as a base, as shown in Fig. 106.

$$a=\frac{h'+h_r}{s} \quad \text{and} \quad y=\frac{a}{l}$$

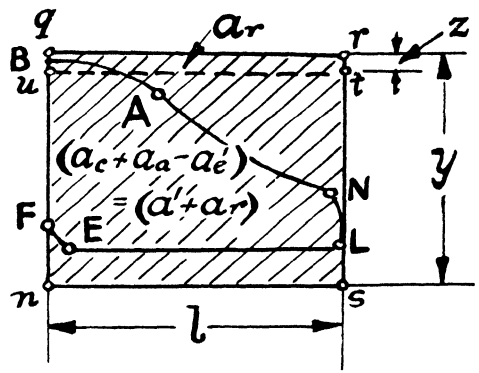



FIG. 106.

If the dotted line *tu* is drawn so that $z=\frac{h_r}{sl}$ and $qrtu=a_r$, then the refund during exhaust is given by $a'=nuts$.

*Example* 11.—For the case considered in Example 5, draw the heat exchange rectangles on the PV diagram. The heat values are $h_a=9{\cdot}385$, $h_c=0{\cdot}068$, $h_e'=5{\cdot}076$, $h_r=0{\cdot}331$, and lengths scaled from the diagram are $l_a=1{\cdot}152$ in., $l_c=0{\cdot}17$ in., $l_e'=1{\cdot}1778$ in., $l=3{\cdot}07$ in. Heat scale of diagram, 1 in.$^2$=0·6764 B.Th.U.

$$a_a=\frac{9{\cdot}385}{0{\cdot}6764}=13{\cdot}9 \text{ in.}^2, \qquad y_a=\frac{14{\cdot}8}{1{\cdot}152}=12{\cdot}06 \text{ in.}$$

$$a_c=\frac{0{\cdot}068}{0{\cdot}6764}=0{\cdot}10 \text{ in.}^2, \qquad y_c=\frac{0{\cdot}1}{0{\cdot}17}=0{\cdot}59 \text{ in.}$$

$$a_e'=\frac{5{\cdot}076}{0{\cdot}6764}=7{\cdot}5 \text{ in.}^2, \qquad y_e'=\frac{7{\cdot}5}{1{\cdot}1778}=4{\cdot}22 \text{ in.}$$

$$(a_a+a_c-a_e')=(a'+a_r)=(14-7{\cdot}5)=6{\cdot}5 \text{ in.}^2=a$$

$$y=\frac{6{\cdot}5}{3{\cdot}07}=2{\cdot}12 \text{ in.} \quad \text{and} \quad z=\frac{0{\cdot}331}{0{\cdot}6764\times 3{\cdot}07}=0{\cdot}159 \text{ in.}$$

The rectangles shown in Figs. 105 and 106 are shown to these dimensions on the original diagram.

**116. Steam Jacketing.**—As stated in the discussion of the ideal cycle of an engine in which the steam is maintained in the dry saturated condition during expansion, the additional heat is transmitted through the cylinder wall from a steam jacket.

The actual conditions in an unjacketed cylinder are, however, far removed from the ideal, where the steam is assumed to be dry at beginning of expansion, and the primary object in applying the jacket is the maintenance of the temperature of the cylinder walls as nearly equal as possible to that of the steam entering the cylinder, so as to reduce the loss due to initial condensation.

As already stated, whatever tends to increase the mean temperature of the walls tends also to reduce initial condensation, and experiment has shown that the heat expended in the jacket for this purpose is more than compensated for by the increased efficiency of the working steam in the cylinder. The extent of the gain following the use of the jacket varies greatly according to the conditions of working, the construction of the jacket, including the arrangements for drainage and for supply of steam to the jacket, the temperature of the steam supplied to the jacket, the speed of the engine, the quality of the steam entering the cylinder, the ratio of internal cylinder surface to weight of steam used, the range of temperature of the steam in the cylinder, etc.

**117. Action of the Jacket.**—An unjacketed cast-iron cylinder is practically a heat sponge absorbing from the steam, each stroke during admission, heat which should have been employed in doing useful work, and rejecting the same amount of heat, but at a lower temperature, during expansion and exhaust, by re-evaporation of the water deposited at the beginning of the stroke. This heat is almost entirely wasted, as the useful work done by it by re-evaporation of water during expansion is extremely small, and the remainder goes, during exhaust, to increase the already large exhaust waste.

The greater the proportion of water deposited in the cylinder, up to a certain limiting point, the greater the demand upon the store of heat in the cylinder walls for the purpose of re-evaporation; the deeper, also, the ebb and flow of the heat-wave in the metal walls each stroke, and the larger the proportion of heat taking part in this most wasteful process.

The jacket therefore reduces the extent of the heat-interchange between the steam and the walls, and, as a consequence, reduces also the weight of water to be re-evaporated at the expense of heat from the cylinder walls. Thus with a jacket a smaller proportion of the heat of the steam is wasted in merely passing into and out of the walls, and a larger proportion is employed in the performance of useful work than when no jacket is used.

All heat transmitted through the cylinder walls from the jacket is accompanied by a corresponding loss due to condensation in the jacket, but the net result is in favour of jacketing. For it should be noted that each lb. of steam condensed in the jacket leaves it

as *water*, containing only the liquid heat at the jacket temperature, while each pound of steam condensed in the cylinder passes away as *steam* and carries with it the latent heat, which has been given to and then taken from the cylinder walls. There is no re-evaporation of the water in the jacket; but it may be usefully employed as hot feed to the boiler.

**118. Effect of Rotational Speed on Jacket Action.**—Since the amount of heat transmitted through the cylinder walls varies with the time, then, as the rate of rotation increases, the extent of the heat interchanged per stroke between the steam and the walls will become less, until, when the speed is indefinitely great, the heat-interchange is zero, and the jacket is of no effect. From this it follows that the efficiency of the jacket increases as the rotational speed decreases.

**119. Effect of Expansion Ratio on Jacket Action.**—With given initial and back pressures—that is, with a fixed range of temperature in the cylinder, but with a variable cut-off—the efficiency of the jacket will vary with the point of cut-off. For since the flow of heat through the cylinder walls varies with the temperature on the two sides of the walls, it is evident that the jacket heat will pass more readily the earlier the cut-off, since the mean temperature of the internal portion of the cylinder walls is lower as the cut-off is earlier.

Hence the jacket is more effective in cylinders having a large ratio of expansion. During expansion the re-c ration is greater without a jacket than with one, and the dryness fraction, especially at release, is always greater with a jacket than without one.

The following diagram (Fig. 107) shows the varying efficiency of a steam-jacket at different ratios of expansion. It was prepared by Mr. Bryan Donkin from trials made by him with an engine having a cylinder 6 in. diameter, stroke 8 in.; speed in all experiments about 220 revolutions per minute; steam pressure, about 50 lbs. above atmosphere.

Water in the cylinder, from any cause whatever, is a source of loss of heat from the cylinder walls. Owing to the property which liquids possess of absorbing heat from surrounding bodies during the process of evaporation which follows a fall of pressure, it is easy to account for the rapid flow of heat from the walls to the water in the cylinder during expansion, but especially when the exhaust port opens.

However dry the condition of the steam on entering the cylinder, there is always *some* initial condensation (except when a high degree of superheat is used), and even if the steam were dry at cut-off, there is still the water formed during expansion by conversion of heat into work.

Assuming that all possible care has been taken to obtain dry steam in the cylinder by lagging steam-pipes, cylinders, and valve-chests, by separators, and by drain-cocks on valve-chests, by reduction as far as possible of the proportion of clearance surface,

and by polished internal surfaces of cylinder-cover and pistons, then, unless the speed of rotation is high, the steam-jacket or superheating will still be necessary to secure dry steam at release.

**120. Construction of the Jacket.**—The success of jacketing depends very largely upon the care exercised both in the design and use of the jacket. If a jacket is so constructed as to leave pockets which will inevitably remain filled with water, or flat horizontal surfaces upon which will continually lie a layer of water through which heat has to pass to the cylinder, the result will be disappointing. Great

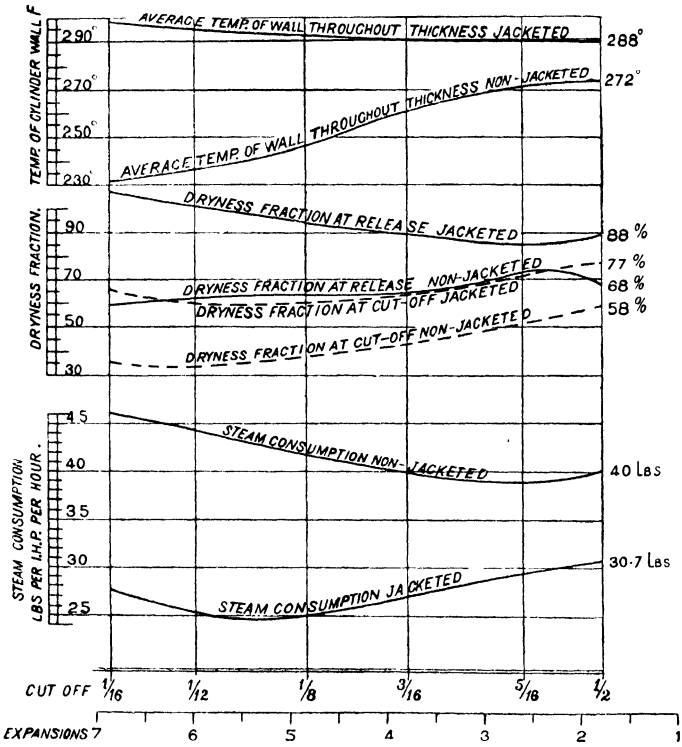



FIG. 107.

care should be taken to arrange for the proper flow of the water deposited in the jacket to the jacket drain by sloping surfaces and properly placed drain-pipes.

The steam-supply pipe to the jacket should be made of ample diameter; also care should be taken to provide for efficient circulation of the steam in the jacket.

Cylinders have sometimes been made with a surrounding jacket through which the steam from the boiler first passes before entering the cylinder. The object of this is, no doubt, to accelerate the action of the jacket by a rapid circulation of the jacket steam. Such

jackets should be well drained. The jacket is never in direct connection with the exhaust side of the piston, hence its mean temperature is higher than that of the internal cylinder walls.

The value of the jacket is greater, other things being equal, as the dimensions of the cylinder are smaller; because in small cylinders the cylinder surface per unit weight of steam passing through the engine is greater than in large cylinders.

One effect of jackets is to increase the tendency to loss of heat by radiation, as the area of the external surface and the temperature of the surface are both increased by the addition of jackets.

It appears, from various experiments, that there is no advantage in making the pressure of the steam in the jacket of any cylinder more than a few pounds greater than that of the initial steam pressure in the cylinder. Thus it is usual to reduce the pressure in the jackets of the second and succeeding cylinders of triple and quadruple expansion engines.

**121.** When, however, the steam is thus throttled down from the initial H.P. pressure to that of an intermediate receiver pressure, it enters the jacket of the intermediate cylinder with more or less superheat. The jacket heat transmitted per lb. will therefore be $(L+h_s)$, where L is the latent heat of the steam at the intermediate jacket pressure and $h_s$ the heat in excess of the saturation total heat at this pressure. For example, in the case of a triple-expansion engine with initial H.P. supply at 210 lb./in.$^2$ or 225 lb./in.$^2$ abs., and L.P. receiver pressure and also jacket pressure at 25 lb./in.$^2$ abs. or 40 lb./in.$^2$ abs., the total heat at 40 lb./in.$^2$ abs. of the throttled steam will be $H_{s2}=H_1=1202{\cdot}7$. Total heat of dry steam $H_2=1171{\cdot}7$, so that the superheat is $h_{s2}=(1202{\cdot}7-1171{\cdot}7)=31$ B.Th.U./lb. The latent heat is $L_2=935{\cdot}8$, so that the jacket heat is $(h_{s2}+L_2)=(31+935{\cdot}8)=966{\cdot}8$ B.Th.U./lb. or $(H_{s2}-h_2)=1202{\cdot}7-235{\cdot}9=966{\cdot}8$ B.Th.U./lb., and this value has to be used in the heat exchange calculations.

The effect of the jacket on the condition of the steam in the cylinders of a triple-expansion engine jacketed as in Fig. 108 * is shown in Fig. 109. H, I, and L denote respectively high, intermediate, and low pressure cylinders, and J and N jacketed and non-jacketed. The heights of the darkened portions show the percentages of water present at cut-off. In the H.P. the dryness is increased 4·7 per cent.; in the I.P., 15·2 per cent.; and in the L.P., 41·5 per cent. by the use of the jacket. The advantage of the jacket in suppressing initial condensation, especially in the L.P. cylinder, is very considerable in this case.

In general, however, the improvement in economy or decrease in rate of heat consumption Q B.Th.U./I.H.P. min. is very variable, ranging from 3 per cent. to 25 per cent. as the conditions and extent of the jacketing vary. Prof. Denton found, for a compound pumping engine completely jacketed with steam at boiler pressure,

* Inst. Mech. E. Reports on "Value of Steam Jacket," *Proc.*, 1889, 1892, and 1894.

that the gain amounted to 3 per cent. Prof. Peabody * found for a triple-expansion engine when jacketed on the cylinder ends with boiler steam, a gain of 4·85 per cent., and when completely jacketed 7·2 per cent.; while Prof. Osborne Reynolds found for the triple-expansion experimental engine at Owen's College, Manchester, fully jacketed with boiler steam, a gain of 25 per cent.

Superheating of the steam supplied to the cylinder is a much more effective and simple means of obtaining increased efficiency and economy, and this method, with the further provision of reheating between stages in the case of multiple-expansion engines, has in modern practice practically superseded the steam jacket.

**122. Superheated Steam.**—In the discussion of the theoretical cycles in Chapter IV of engines using saturated and superheated

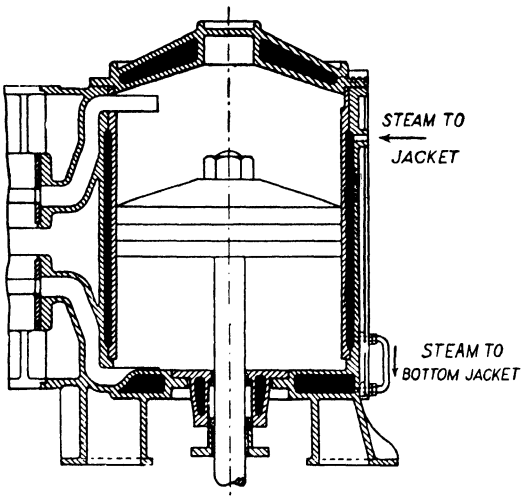



FIG. 108.

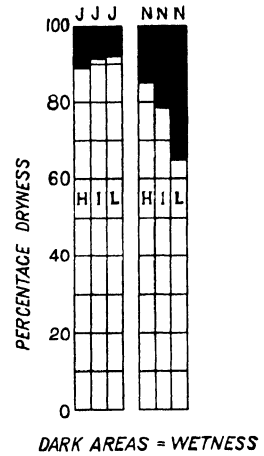



FIG. 109.

steam, it was shown that the gain of thermal efficiency due to superheating was a relatively unimportant figure. Although thermodynamically the use of superheated instead of saturated steam is not a paying proposition, it is so practically, since the superheat effects more thoroughly in a direct manner what the steam jacket does indirectly, that is, the suppression of condensation on the cylinder walls.

When the working fluid is saturated steam, no transfer of heat, however small in amount, can take place from the steam to the metal without accompanying deposition of water, which during the exhaust stroke is evaporated at the expense of the heat in the cylinder walls. The result is that the mean temperature of the walls with saturated steam is much below that of the steam on entering the cylinder. On the other hand, when the steam is given a

* *Thermodynamics of the Steam Engine*, 1907.

sufficiently high degree of superheat it is in a far more stable condition than when dry saturated. It can give up the whole of its superheat to the walls without condensation, and a greater amount of work is done per lb. of steam passing through the cylinder.

The drier the steam at release the less the demand upon the cylinder walls during exhaust for heat of re-evaporation, and the higher the mean temperature of the walls. A dry cylinder at release parts with little heat to the comparatively non-conducting medium passing away during exhaust.

Hence when the steam is given a sufficiently high superheat to ensure complete dryness at release, the principal source of the loss of heat by the walls, that is, water in the cylinder at release, is removed, and the heat exchange between the steam and the walls is reduced to a minimum.

**123. Range of Superheat to maintain the Steam Dry up to Cut-off.** —If an engine supplied with dry saturated steam has a quality $q_a$ at cut-off, and it is desired to supply it instead with initially superheated steam, so as to ensure completely dry steam at cut-off ($q_a=1$), the range of superheat can be approximated by the following rule. This was deduced by Mr. Michael Longridge from Prof. Ripper's experiments on a small experimental Schmidt engine.*

It is strictly valid only for the particular engine used and subject to modification for other types and for variation of operative conditions obtaining in practice. The range of superheat in degrees Fahr. is given by—

$$t_s=750(1-q_a) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (41)$$

where $q_a$ is the dryness at cut-off with saturated steam.

For example, if the dryness at cut-off in an unjacketed engine using saturated steam is 75 per cent., or $q_a=0{\cdot}75$, the steam will require to be superheated $t_s=750(1-0{\cdot}75)=750\times0{\cdot}25=187{\cdot}5°$ Fahr., to ensure dry steam at cut-off.

If with any range of superheat the quality of the steam at cut-off is increased from $q_a$ to some higher value $q_a'$, then the general form of the rule becomes—

$$t_s=750(q_a'-q_a) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (42)$$

For dry steam at cut-off $q_a'=1$, and Equation 41 results.

From Equation 42 the resulting quality for varying ranges of superheat is calculable for a given value of $q_a$—

$$q_a'=\left(\frac{t_s}{750}+q_a\right) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (43)$$

Thus in the previous case, if steam superheated 150° Fahr. instead of 187·5° Fahr. is supplied, the probable quality at cut-off will be raised to—

$$q_a'=\left(\frac{150}{750}+0{\cdot}75\right)=0{\cdot}95.$$

* See footnote, p. 181.

**124. Effect of Superheat on Heat Absorption by the Wall during Admission.**—The extent to which the heat absorption by the wall during admission can be reduced by progressive superheat of the steam before admission is illustrated in Fig. 105. These diagrams show the heat areas ($h_a$) constructed, as explained in Art. 115, from the trial results obtained by Prof. Ripper on the small Schmidt engine already referred to.

**125. Effect of Superheat on the Indicated Rate of Heat Consumption.**—In order to avoid complication of the discussion

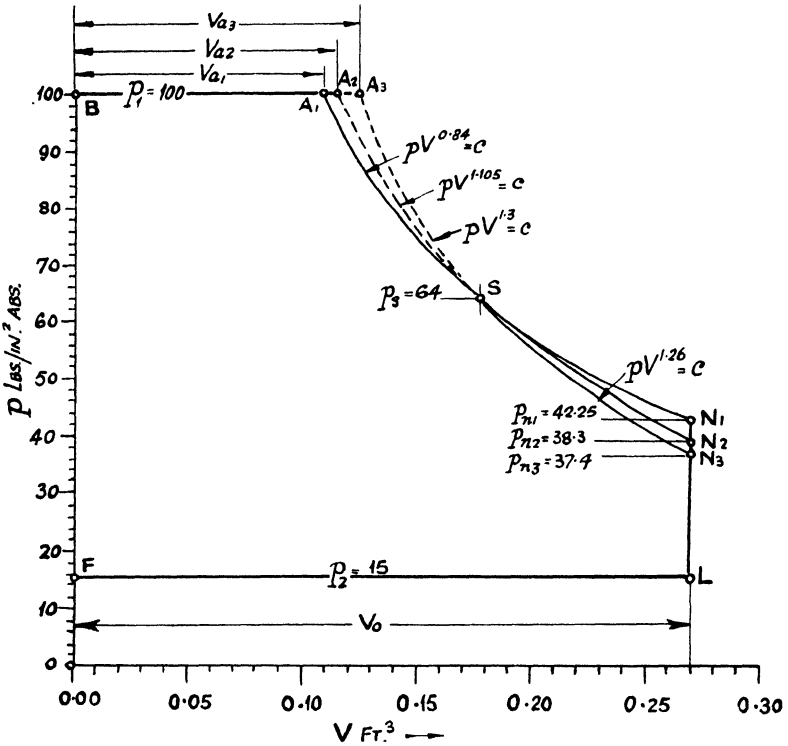



FIG. 110.

consider a single cylinder unjacketed engine, running at constant speed on constant load, without clearance or compression and with no wiredrawing losses.

The indicator diagram (Fig. 110), in this case shown for initially saturated steam, is $FBA_1N_1L$. The initial and back pressure limits are $p_1$ and $p_2$, and the expansion law is taken as $PV^n=c$. The ratio of expansion is $r_1=V_o/V_{a1}$.

When superheated steam is substituted for saturated steam, experiment has shown that the index of the expansion law increases

with the increase of superheat due to the increase in quality of the steam at cut-off. With moderately superheated steam the curve of expansion becomes steeper than that with saturated steam. With high superheat it becomes more steep. For maintenance of constant work per stroke the cut-off has thus to be extended from $A_1$ to $A_2$ for the moderate superheat, and to $A_3$ for the high superheat.

The law of expansion is $PV^m=c$ and $PV^k=c$, where $m>n$ and $k>m$. The terminal pressure $p_n$ is lowered in each case, but the diagram areas are equal: that is, for the condition of constant cycle work, $FBA_1N_1L=FBA_2N_2L=FBA_3N_3L$. If in the case of high initial superheat this is sufficient to give a considerable range of superheat at cut-off, the steam may remain superheated during expansion. It may, however, pass from the superheated to the wet condition before the end of expansion. In the latter event the law will change when the saturated condition is reached. The real expansion curve will thus consist of two curves. The first part in the superheat region will have a steeper slope than the last part. This general case is shown in Fig. 110, by the curve $A_3SN_3$, the dry steam condition being reached at S. The corresponding pressure can be readily determined on the TΦ or HΦ diagram.

In general, if $h_i$ is the constant value of the indicated heat per stroke in B.Th.U. represented by the diagram area, and N the constant speed in strokes/min., the indicated horse-power is—

$$\mathrm{I}=\frac{h_i\mathrm{N}}{42{\cdot}4} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (44)$$

If $v_{w_a}$ or $v_{s_a}$ is the specific volume of the wet or superheated steam at cut-off A in ft.$^3$, the cylinder feed is $w_f=\frac{\mathrm{V}_a}{v_w}$ or $w_f=\frac{\mathrm{V}_a}{v_{s_a}}$, and the consumption per minute is $w_f\mathrm{N}$.

The indicated rate of consumption is—

$$w_i=\frac{42{\cdot}4\times 60\times w_f}{h_i}=\frac{2545w_f}{h_i} \text{ lb./I.H.P. hr.} \quad . \quad . \quad (45)$$

If H is the heat input to the steam per lb. from the exhaust temperature $t_2$, then the indicated thermal efficiency is—

$$\eta_{\mathrm{I}}=\frac{2545}{w_i\mathrm{H}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (46)$$

and the rate of heat consumption—

$$\mathrm{Q}_i=\frac{42{\cdot}4}{\eta_{\mathrm{I}}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (47)$$

For the simpler case considered here $w_i$ and $\mathrm{Q}_i$ will be less than the actual values, when clearance, compression, and wiredrawing are included. The probable value can be approximated by means of a correction factor.

By calculating the $w_i$ and $Q_i$ values for the conditions of initially saturated and superheated steam, an approximation to the gain in economy, due to the suppression of condensation by superheating the steam, can be obtained.

These calculations have been carried out for a small unjacketed engine with stroke volume 0·27 ft.$^3$; and the PV and TΦ diagrams are shown in Figs. 110 and 111. The conditions assumed are: (*a*) saturated steam, cut-off at 0·4 stroke, $p_1$=100, $p_2$=15; dryness at cut-off taken as 0·6; and expansion law, $PV^{0·94}=c$: (*b*) steam initially superheated 150° Fahr., and expansion law, $PV^{1·105}=c$; cut-off at 0·42; dryness at cut-off, 0·8; (*c*) steam initially superheated 350° Fahr.; expansion law, $PV^{1·30}=c$ in the superheat region and $PV^{1·26}=c$ in the saturation region; cut-off at 0·465; steam 50° Fahr. superheated at cut-off and dry at 0·655 stroke, with pressure 64 lb./in.$^2$ abs.

Each of these diagrams when planimetered gives the same area, the common heat value being $h_i$=3·14. The value of $w_i$, $Q_i$, and $\eta$ obtained for the three cases are as follows, and expressed in terms of the saturated steam value as a unit—

| | (*a*) *Saturated* | (*b*) *Moderate Superheat* | (*c*) *High Superheat* |
|---|---|---|---|
| $W_i$ . . | 1 | 0·785 | 0·642 |
| $Q_i$ . . | 1 | 0·847 | 0·767 |
| $\eta$ . . | 1 | 1·175 | 1·315 |

Under the conditions arbitrarily assumed there are theoretical reductions of 21·5 per cent. and 35·8 per cent. in steam consumption; 15·3 per cent. and 23·3 per cent. in heat consumption; and increases of 17·5 per cent. and 31·5 per cent. in thermal efficiency.

In an actual case with clearance, compression, and wire-drawing, the values would be about 85 per cent. of the theoretical.

Average figures for saving in heat economy with steam of total temperature of about 600° Fahr. are 20 per cent. in simple, 16 per cent. in compound, and 8 per cent. in triple-expansion engines.

The corresponding conventional TΦ diagrams are given in Fig. 111. The area in each case gives the indicated heat per lb. per cycle, that is, the value of $h_i/w_f$. The condition or expansion lines $A_1N_1$, $A_2N_2$, $A_3N_3$ are located on the TΦ field and the diagrams are completed by drawing the constant volume curves which pass through the terminal points. The volume per lb. of steam in the cylinder at release is given, for any case, by—

$$v_w = \frac{V_o}{w_f} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (48)$$

and the corresponding real quality at pressure $p_n$ is obtained from $q_n=\frac{v_{w_n}}{v_n}$, where $v_n$ is the specific volume of dry steam at $p_n$. The

virtual quality at any other chosen pressure $p$ on the constant volume curve is obtained from the quality equation—

$$q=\frac{v_{w_n}}{v} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (49)$$

where $v$ is the specific volume at the chosen pressure $p$.

The heat values represented by these TΦ diagram areas, on integration by planimeter, check with the calculated values. They are in the ratio 1 : 1·27 : 1·54, so that per lb. of steam supplied per cycle the indicated work is increased 27 per cent. with moderately. and 54 per cent. with highly superheated steam. The values in an actual case would be proportionately less.

It will be noted that while the heat returned by re-evaporation in the case of the saturated steam is considerable, that with moderate

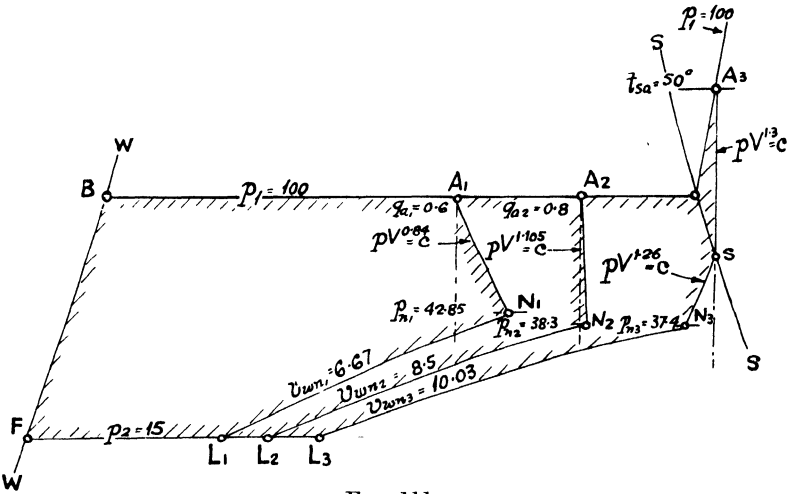



FIG. 111.

superheat is small and the expansion curve $A_2N_2$ approaches closely to the adiabatic vertical at $q_{a_2}=0·8$. At higher qualities it will reach the adiabatic and then change in slope, so that from about $q_a=0·85$ to 1, heat will be taken by the walls from the steam during the whole of the expansion. This condition was experimentally ascertained in the study of the logarithmic form of indicator diagram by J. P. Clayton * for a large number of unjacketed engines.

In this case it has been assumed that as long as the steam remains superheated during expansion, the heat exchange is practically negligible, or the expansion is sensibly adiabatic; but after the saturated condition is reached the wall takes heat from the steam during the remainder of the expansion, with consequent conden-

* Bulletin No. 58. "A New Analysis of the Cylinder Performance of Reciprocating Engines." Engineering Experiment Station, University of Illinois, Urbana, U.S.A.

sation, as indicated by the slope to the left of the adiabatic, of the expansion curve, between S and $N_3$.

In addition to initial superheating of the steam, further reheating between stages, in the case of a multiple-expansion engine, leads to increased economy. This particular aspect of the superheat question and the actual gain due to superheat are discussed in Chapter VII.

**126. Leakage.**—In the foregoing discussions it has been assumed that the true cylinder feed $w_f$ is known. This quantity is seldom quite the same as the weight of measured condensate per stroke, as there is always more or less leakage past the valves from the steam chest to the exhaust, and sometimes past the piston.

When leakage does occur the missing quantity recorded by the indicator diagram is thus due both to condensation and leakage. The difficulty that arises in a given case is to disentangle the two quantities. The relation between condensation and leakage loss has been the subject of much controversy for many years, one school of engineers maintaining that the major part of the missing quantity is due to condensation and the other school that it is due to leakage.

In a classical paper on the Law of Condensation of Steam,* Profs. Callendar and Nicholson, in 1897, stated as a result of their investigation on an experimental engine fitted with a " balanced " slide-valve, that the leakage varied directly as the difference of pressure on the two sides of the valve, that is, the difference between the steam chest and exhaust pressures. If $w_l$=leakage in lb./hr., $\delta p$=the difference of pressure then—

$$w_l = k\delta p$$

where $k$ is a constant termed the " rate of leak," and gives the leak per hour per lb. difference of pressure. This, they stated, should be proportional to the perimeter $x$ of the port divided by the mean overlap $l$ of the valve, that is, $k = C\frac{x}{l}$, C being a coefficient depending on the nature of the lubricating oil film. Thus—

$$w_l = \frac{Cx\delta p}{l} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (50)$$

From the experiments on this engine they found that the value of C was 0·02. Similar tests made on the unbalanced H.P. and L.P. valves of a quadruple expansion engine gave C=0·019.

Their conclusion regarding the nature and effect of valve leakage was stated as follows : " The foregoing experiments would make it appear probable that a moving valve, however well fitted, is subject to a regular leakage of a peculiar type which has not been previously suspected. The leakage appears to take place in the following manner. So long as the valve is stationary the oil film may suffice

* " On the Law of Condensation of Steam deduced from Measurements of Temperature-Cycles of Walls and Steam, in the Cylinder of a Steam Engine," *Proc. Inst. C. E.*, vol. 131, 1897–98.

to make a perfectly tight joint, but as soon as it begins to move the oil film becomes broken up and partly dissipated. Water is being continuously condensed on the colder parts of the surface exposed by the motion of the valve. This water works its way through and breaks up the oil film under the combined influence of the pressure and the motion. The continual re-evaporation taking place in the exhaust tends to keep the valve and bearing surfaces of the seat cool, and to maintain the leaking fluid in the state of water. The exhaust steam from the cylinder has the same tendency. The coefficients of viscosity of steam and water, at the temperatures which occur in a steam engine, are not accurately known. Whereas that of steam increases with rise of temperature, that of water diminishes very rapidly. It is not improbable that the quantity of water which can leak through a given crack, under a given difference of pressure, may be from twenty to fifty times greater than the quantity of steam that can leak under similar conditions. This agrees with well-known facts in regard to leakage, and explains how it is that the leakage in the form of water is so great."

They further stated that the analysis of a large number of observations, in addition to the few made in their experiments, " leads to the conclusion that all valves leak more or less when in motion, and that in many cases the greater part of the missing quantity is to be attributed to leakage of this description. Whatever the precise manner in which the leak takes place, it appears to be nearly proportional to the difference of pressure, and to be in most cases independent of the speed. In any case, it appears probable that the leakage is connected in some way with the condensation taking place on the valve surfaces, and if so, it may evidently be greatly reduced, if not entirely cured, by jacketing or otherwise heating the valve seat to minimise the condensation."

Subsequent experiments by Prof. Capper on a slide valve engine described in the First Report of the Steam Engine Research Committee,* appear to confirm Callendar and Nicholson's conclusions regarding the nature of the leakage, and that this was reduced by jacketing the engine, and also when the steam was wiredrawn between stop-valve and steam chest so that it was slightly superheated at the cylinder. He found, however, that the leakage was not directly proportional to pressure at *all* speeds, and that it was doubtful if it was exactly inversely proportional to the overlap. Taking the latter assumption, however, as correct, he obtained a *mean* value for the constant C (Equation 50) of 0·02, the same as that given by Callendar and Nicholson. Prof. Mellanby has also submitted evidence that valve leakage probably accounts for the major proportion of the missing quantity.†

* *Proc. Inst. Mech. E.*, 1905.

† " Effect of Jacketing of a Compound Engine," *Proc. Inst. Mech. E.*, 1905, p. 519. Also " Surface Condensation in Steam Engine Cylinders," *Proc. Inst. of Engineers and Shipbuilders in Scotland*, 1911.

Slide-valve engines may be expected to show more leakage than those fitted with piston-valves or Corliss or drop-valves, as a slide-valve is much more liable to warp and distort when heated. In the discussion on the Steam Engine Research Report, Captain Sankey gave the results of experiments on leakage past piston-valves. He found that the constant C in the Callendar and Nicholson equation was only 0·003 for the standard piston-valve with rings, as used, or about $\frac{1}{7}$ of the value obtained for the slide-valve.

When the piston-valve was replaced by a solid plug valve, and the clearance between the plug and cylinder progressively increased from 1/1000 to 5/1000 in. the leakage at all pressures increased proportionately with the clearance, dry steam being used. When the plug was well lubricated with oil, at 4/1000 in. clearance, the leakage was reduced about 40 per cent. When wet steam was supplied the leakage greatly increased. As regards Corliss and drop-valves definite experimental data are wanting; but it might reasonably be inferred that the Corliss in good condition would be less subject to leakage than the slide-valve, and the drop-valve the least subject of all.

In dealing with the heat analysis of a given case it will be obvious that every precaution must be taken to have the engine in first-class running condition with valves carefully overhauled beforehand to ensure as tight a condition during test as possible.

Captain Sankey suggested the following allowances as suitable for the different classes of valves *: engines with slide-valves a correction of 5 per cent. of the cylinder feed if it is calculated from the total consumption (W); engines with piston-valves fitted with rings and springs, zero correction, but if the piston valves are not thus fitted, an allowance of anything between 5 and 20 per cent., may be required; engines with Corliss or drop-valves an allowance from 1 per cent. to 3 per cent.

Dean and Wood † found that for a series of flat-valve and piston-valve engines, after from 1 to 5 years of 24 hour-day service the rate of consumption was increased from 1 per cent. to 45 per cent. (average 16 per cent.) over the values obtained when the engines were new. In these cases the valves had only been adjusted, not refitted.

**127. The Logarithmic Indicator Diagram.**—It is a well-known fact that, in the majority of cases, the expansion and compression curves of steam engines can be fitted closely by a polytropic law $PV^n=C$, where $n$ is a constant. In many cases the value of $n$ is not far removed from unity, and this has led to the adoption of an arbitrary standard $PV=C$ giving equilateral hyperbolas for the expansion and compression curves of the diagrams of reference used for design purposes. The student, however, should bear clearly in mind that this is purely an arbitrary choice made to simplify provisional calculation and has no thermodynamic significance.

* *The Energy Chart*, p. 120.
† *Trans. A.S.M.E.*, vol. 30, p. 6.

When the absolute values of pressures and volumes obtained from an indicator card are plotted logarithmically (that is, from the logarithmic scale of a slide-rule or on logarithmic squared paper) the expansion and compression curves, when they conform closely to the law $PV^n=C$, are transformed into straight lines, the negative slope of each line giving the value of the index $n$.

The law $PV^n=C$ applies only in cases where the weight of steam is a *constant* quantity during expansion or compression.

If sensible leakage takes place either into or out of the cylinder the law does not hold and the straight logarithmic expansion or compression line is replaced by a curved line.

This method of plotting thus reveals the existence of any material leakage past the admission valve, the piston, or the exhaust valve during the expansion and compression phases of the cycle. This

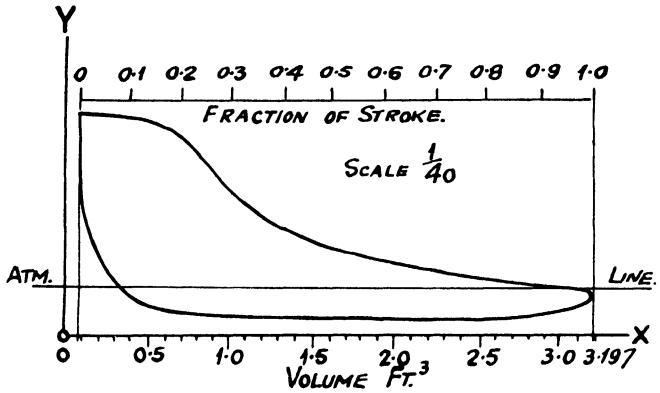



FIG. 112.

leakage is usually due to wear of working parts, which affects most of the possible sources of leakage in equal proportion.

In a steam engine three determining pressures condition the expansion and compression leakage.

These are the steam chest pressure, the pressure in the cylinder at the particular point at which the leakage condition is being considered, and the pressure in the exhaust passage. Clayton, in the investigation already cited (p. 215), states that leakage into or from a steam cylinder has been found in most cases to occur only when the pressure difference is over 20 lb./in.$^2$

The expansion and compression lines may be roughly divided into three equal parts, on the logarithmic diagram, where they extend from the initial to nearly the back pressure: (1) the upper third, influenced by leakage from the cylinder; (2) the middle third, practically uninfluenced by leakage; (3) the lower third, influenced by leakage into the cylinder. A typical case of this double form of leakage, from Clayton's paper, is shown in Figs. 112 and 113.

The indicator diagram (Fig. 112) was taken from a Corliss engine

with cylinder 14 in. diameter × 35 in. stroke, which was known to have a leaky piston and valves. Examination showed that one of the piston rings was broken. The leakage into the cylinder through the valve is shown in Fig. 113 by the rise of the expansion curve above the dotted line, which is a continuation of the straight central portion. It commences at an absolute pressure of 25 lb./in.$^2$, which is 35 lb./in.$^2$ lower than the admission pressure. Leakage out of the cylinder in the first part of the expansion is shown by the rise of the curve above the dotted line between 57 lb./in.$^2$ abs. and 45 lb./in.$^2$ abs. Leakage out of the cylinder during the latter part of the compression stroke is revealed by the fall of the compression curve below the straight dotted line of

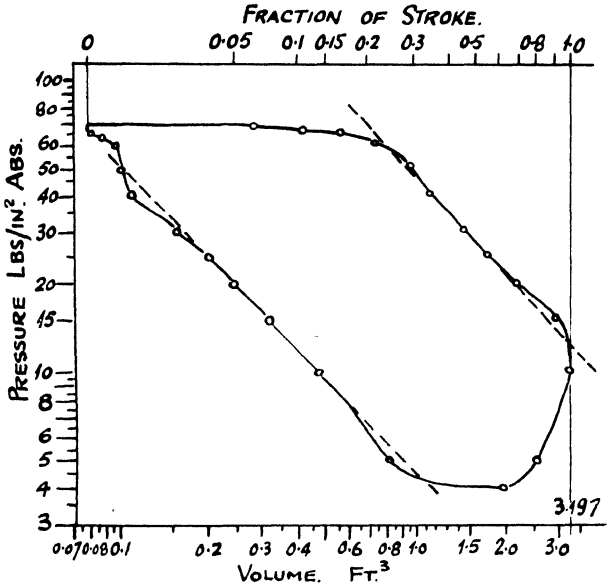



FIG. 113.

constant steam weight. This begins at a pressure of 25 lb./in.$^2$ Leakage into the cylinder from the admission valve during the first part of the compression is shown by the fall of the curve below the straight line between 4·25 lb./in.$^2$ abs. and 7·25 lb./in.$^2$ abs.

There is always a slight leakage out of the cylinder on closure of the valve until it has obtained a sufficient cover, and this should be allowed for in estimating the clearance steam by prolonging the back pressure line to cut the straight compression line produced, and taking the volume value at the intersection.

It can be seen from the illustration that the logarithmic form of diagram is useful for the detection of leakage and as an indicator to show if an engine needs an overhaul of valves and piston. A

deduction was drawn from the logarithmic diagrams taken from the cylinders of forty-seven engines of various types that the majority of engines in good condition are "practically tight" as regards leakage into or out of the cylinder.

128. In addition to the information it gives regarding the law of expansion or compression, the logarithmic diagram can also be used to approximate the clearance volume of an engine and to define the characteristic points of cut-off, release, compression, and admission, which are often indistinguishable on the PV diagram.

In order to determine the clearance, the information required for the calculation is the scale of the indicator spring and the atmospheric line on the diagram. The method is one of trial and error. For the type of engine considered, a clearance value expressed as a fraction of the stroke volume is assumed, and with this value the log graph of the compression curve of the diagram is drawn. If it is a flat curve bending to the left the clearance assumed is too small; if the curve bends to the right the assumed clearance is too large.

If two values are tried giving defect and excess, the true clearance is given by a straight line which will lie between the other two and is readily obtained by a further trial. Clayton states that "the closeness of location of the straight line region, lying between the two families of diverging curves, will be found to be within 5 per cent. to 10 per cent. of the clearance volume between 20 per cent. and 2 per cent. respectively of the piston displacement."

The following example serves to illustrate this method.

*Example* 12.—Find the approximate value of the clearance volume of the Schmidt superheated steam engine from the indicator diagram shown in Fig. 102.

In this case the stroke volume is $V_o=0{\cdot}27$ ft.$^3$ and the pressures from the compression curve are as follows:

| | | | | | |
|---|---|---|---|---|---|
| $x$ . . . . . | 0·224 | 0·2 | 0·1 | 0·05 | 0 |
| $p$ . . . . . | 17 | 18 | 29·0 | 40 | 66 |

Choose a low volume of clearance for a first trial, 4 per cent. or $V_c=0{\cdot}04\times V_o=0{\cdot}0108$ ft.$^3$

Using the lower scale of a slide rule lay off a scale of volume between 0·01 and 0·1, and a scale of pressures between 10 and 100 lb./in.$^2$ abs., as shown in Fig. 114.

| Assumed Clearance Vc | $x$ | 0·224 | 0·20 | 0·1 | 0·05 | 0 |
|---|---|---|---|---|---|---|
| | $p$ | 17 | 18 | 290 | 40 | 66 |
| (1) 0·0108 . . . | $V_1$ | 0·07128 | 0·0648 | 0·0378 | 0·0243 | 0·0108 |
| (2) 0·0270 . . . | $V_2$ | 0·0875 | 0·0710 | 0·0540 | 0·0405 | 0·0270 |
| (3) 0·0189 . . . | $V_3$ | 0·0794 | 0·0739 | 0·0459 | 0·0324 | 0·0189 |

Tabulate, as shown, the stroke fractions ($x$) and corresponding total volumes ($V_1$) when $V_c=0{\cdot}0108$ is used, and then plot the $p$ and $V_1$ values to the log scales. The result is a flat curve $E_1F_1$ (Fig. 114) bending to the left, showing that the assumed value of $V_c$ is too low. For a second trial, take a higher limit of, say, 10 per cent. or $V_c=0{\cdot}027$. Again tabulate and plot the $p$ and $V_2$ values, and the result is a flat curve $E_2F_2$ bending to the right, showing that this value is too high. For a third trial take, say, the average of 7 per cent., or $V_c=0{\cdot}07\times0{\cdot}27=0{\cdot}0189$ ft.$^3$; tabulate and plot

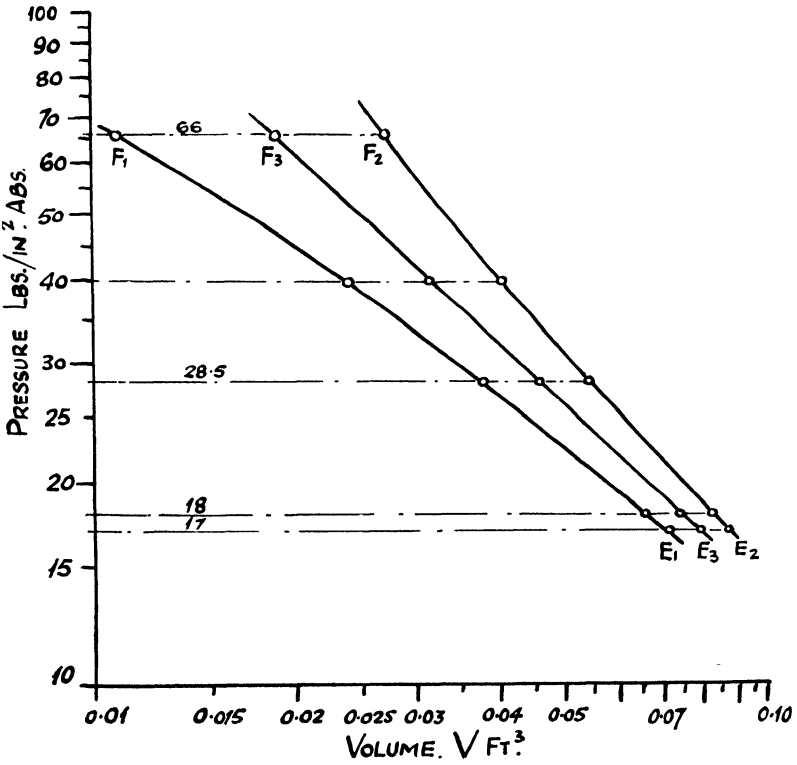



FIG. 114.

the $p$ and $V_3$ values. The result is a straight line $E_3F_3$, showing that the correct value of the clearance is 7 per cent. The slope of this line is $n=0{\cdot}905$.

The next example illustrates the usefulness of the logarithmic diagram in the estimation of the characteristic points of the expansion and compression curves. The diagrams chosen were taken from the H.P. cylinder of a triple-expansion marine engine, using superheated steam and working in series with an exhaust steam turbine.*

* From a paper on " Recent Developments of the Exhaust Steam Turbine," by Dr. Gustav Bauer. *Trans. Inst. Eng. and Shipbuilders*, January 1928.

*Example* 13.—Plot the logarithmic diagrams from the PV diagrams shown in Fig. 115, and find approximately the points of cut-off, release, and compression, and the laws of expansion and

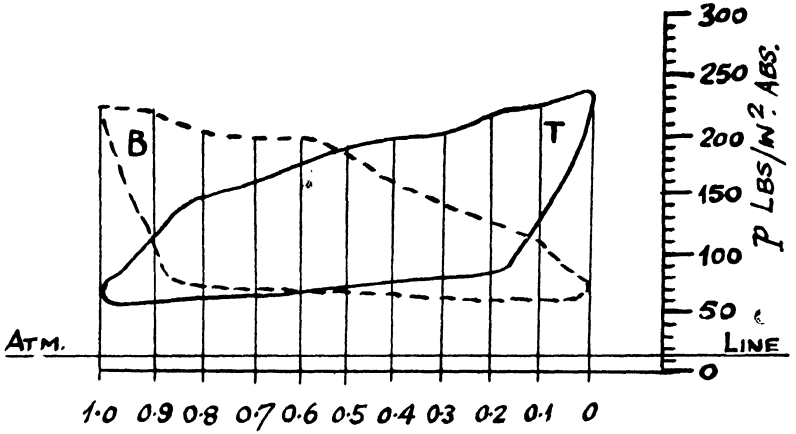



FIG. 115.

compression. The stroke volume is 16·7 ft.$^3$ and clearance is 15 per cent. or 2·505 ft.$^3$

When the total volumes at different fractions ($x$) of the stroke are

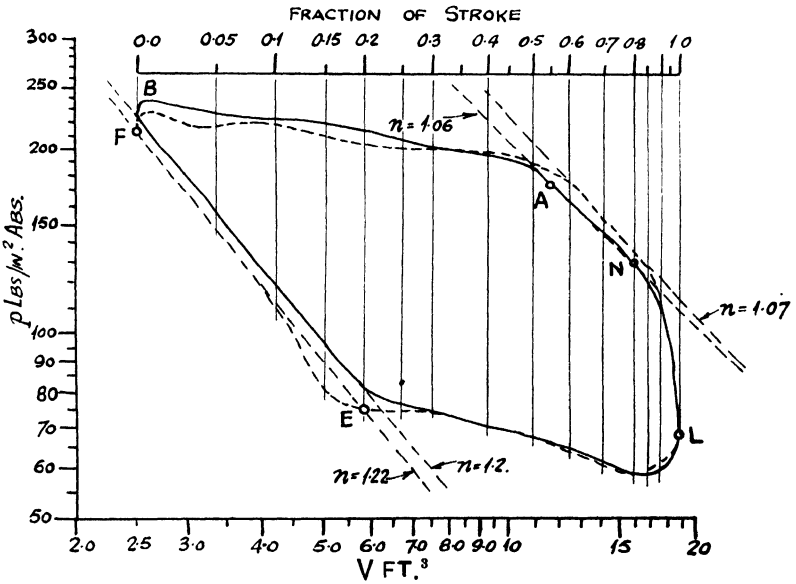



FIG. 116.

calculated from $V=V_c(x+0{\cdot}15)$ and the corresponding pressures are scaled from the PV diagrams and logarithmically plotted, the diagrams shown in Fig. 116 are obtained.

The full line diagram is for the top and the dotted diagram for the bottom of the cylinder.

On the top diagram the points between 0·55 and 0·80 lie on a straight line, indicating cut-off at 0·55 and release at 0·80. From 0·2 to practically the end of the instroke the points lie on a straight line, defining the compression point at 0·2.

On the bottom diagram the points for expansion again lie on a straight line, between 0·6 and 0·85, indicating later cut-off at 0·6. The mean cut-off is thus apparently 0·575. The value stated on the original diagrams is 0·55.

With the exception of those at 0·12 and 0·15 stroke the points for the compression lie on a straight line from 0·2 to the end of the stroke. The bulge on the diagram between 0·2 and 0·1 indicates a leakage into the cylinder through the admission valve, at the beginning of the compression. The slope of the expansion line for the top diagram is $n=1·07$, and for the bottom diagram $n=1·06$. The corresponding slopes of the compression lines are $n=1·2$ top and $n=1·22$ bottom.

**129.** Clayton states that "the values of $n$ for most engines of ordinary size using saturated steam at normal cut-off is between 0·95 and 1·05, while for superheated steam the range is usually from 1·0 to 1·3.

"For unjacketed cylinders in good condition which exhaust at or near atmospheric pressure it appears there is a known relationship between the expansion index $n$ and the quality $q_a$ of the steam at cut-off."

This deduction was made first from a systematic series of tests on a 12 in. diam.$\times$24 in. stroke Corliss experimental engine, and its applicability to various engines for which diagrams and reliable test data were available was subsequently tested. Four classes of American engines were examined. The results from cylinders of simple Corliss, two valve and four valve types, the high pressure cylinders of compound engines, the intermediate pressure cylinders of triple-expansion engines, cylinders of high speed and simple locomotive engines were compared, the restricting conditions imposed being (1) that the tests results should come from reliable sources; (2) that the data supplied should be complete enough to enable the quantities required for comparison to be computed; (3) the cylinders should be unjacketed; (4) the diagrams should be representative of average conditions; (5) the back pressure in the cylinder should be *practically atmospheric;* (6) that no large leaks should exist.

This relation of $q_a$ and $n$ was found to be practically independent of the size of cylinder and cut-off, and was not seriously affected by variation of pressure at cut-off between 75 lb./in.$^2$ abs. and 150 lb./in.$^2$ abs., and of speed between 70 rev./min. and 150 rev./min.

At the average pressure between these limits of 120 lb./in.$^2$ abs.,

back pressure about atmospheric, and average speed of 120 rev./min. the law suggested is—

$$q_a = 1{\cdot}245n - 0{\cdot}576 \quad . \quad . \quad . \quad . \quad . \quad (51)$$

An examination of the large table of results given in this Bulletin shows that the value of $q_a$ ranged from 0·508 to 0·825. It is claimed that within the limits stated above the expression will enable the actual consumption of an engine to be determined from the indicator diagram to within an average of 4 per cent. of the measured consumption on test.

In the case of the well authenticated test figures given in Example 1 the quality at cut-off is 0·71 and according to this rule, which should be applicable to it, the index of expansion should be $n = 1{\cdot}030$. When, however, the log graph of the pressures and volumes is drawn it is found to have a slope of $n = 0{\cdot}94$, corresponding to a quality $q_a = 0{\cdot}594$ by the rule. This gives a cylinder feed of 0·0394 instead of the experimental value of 0·0328 lb., showing an excess of 20 per cent. In this particular case, at least, the rule gives an erroneous result.

Taking the other case of Prof. Ripper's test on the Schmidt engine (Example 7) with highly superheated steam, when the log graph of the expansion curve is drawn it is found to have a slope $n = 1{\cdot}35$. The corresponding volume ratio at cut-off, where $p_a = 73$, is 1·18. According to this rule, which should also apply here, the ratio should be $q_a = (1{\cdot}245 \times 1{\cdot}35 - 0{\cdot}576) = (1{\cdot}681 - 0{\cdot}576) = 1{\cdot}105$. This corresponds to a superheat range about 78° Fahr. as against the TΦ value of 125° Fahr. at cut-off, or an increased weight in the cylinder of $0{\cdot}01888 \times \frac{1{\cdot}18}{1{\cdot}105} = 0{\cdot}02$ lb. Deducting the clearance steam, $w_f' = (0{\cdot}02 - 0{\cdot}00338) = 0{\cdot}01662$, instead of 0·0155, an excess of 7 per cent., which is more reasonable.

It would appear from these two cases that results given by this equation, even within the narrow limitations specified, should be accepted with considerable reserve.

**130. Condensation.**—The action which occurs between the cylinder walls and the steam was originally demonstrated by Callendar and Nicholson in the investigation already cited, by means of time-temperature diagrams showing variation of wall and steam temperatures during a cycle. The steam temperatures were obtained from a platinum resistance thermometer carried by the piston and the wall temperatures by means of thermocouples bedded in the walls.

By means of Fourier's method of analysis the temperatures at the wall surface were calculated from these couple readings. For the particular area of surface examined the steam and wall temperatures were plotted to a base of cycle time, this whole time being divided into sixty elements, so that each division of the base line represented $\frac{1}{60}$ rev. One of these time-temperature diagrams is shown in

Fig. 117. It shows the cyclic variation of steam and cover wall temperatures, with the thermocouple at a depth of 0·01 in. in the cover, when the engine ran at 100 rev./min.

The steam and total temperature scales are not the same. Unit length on the steam temperature scale represents five times the temperature difference for the same length on the metal temperature scale. While the range of steam temperature in one cycle is from 215° Fahr. to 320° Fahr., or 105° Fahr., the range of the metal wall of the cover (where about 90 per cent. of the whole condensation takes place) is from 295° to 298°, or 3° Fahr. The mean metal temperature shown by the heavy line is 295° Fahr.

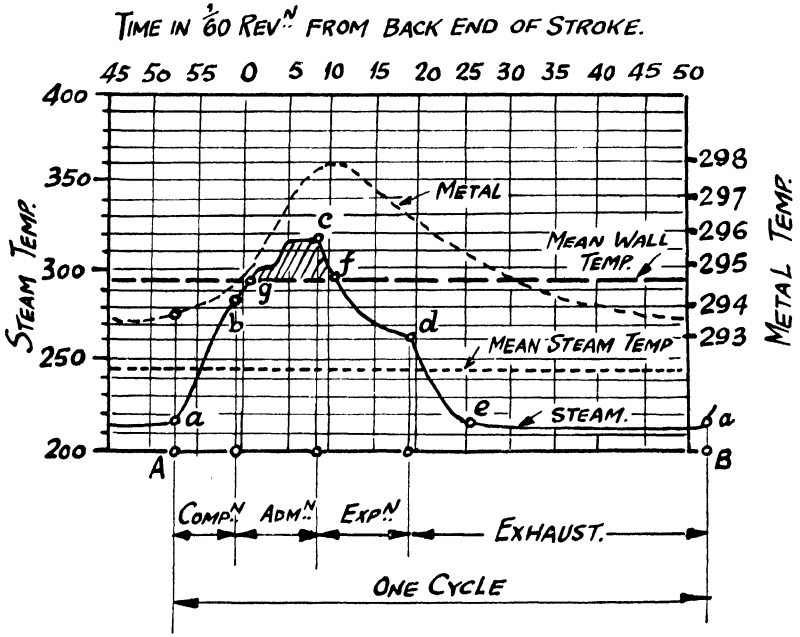



Fig. 117.

(to the steam temperature scale). If the metal temperature were plotted to the same scale as that of the steam the flat wavy curve would hardly be distinguishable from the mean straight line.

Compression starts at *a* with the steam at 215° Fahr., and the lower surface (and clearance surface) at 293·5° Fahr. During compression *a* to *b* as the pressure and temperature of the steam rises to 300° Fahr., the cover temperature rises to 295° Fahr., due to heat absorption from the compressed steam. During admission *b* to *c* the metal temperature rises rapidly and continues to rise to a maximum of 298° Fahr. during part of the expansion stroke. The steam meantime has reached the maximum of 320° Fahr. at end of admission, and fallen to about 290° Fahr. before the wall reaches

its maximum of 298° Fahr. The steam temperature falls between cut-off (*c*) and release (*d*) from 320° Fahr. to 260° Fahr., while the wall temperature, due to refund for evaporation, falls from 298° Fahr. to 296·5° Fahr.

On release, *d* to *e*, the steam temperature falls from 260° Fahr. to 215° Fahr. The wall refund goes on during exhaust *e* to *a* at constant steam temperature of 215° Fahr., the final fall of temperature of the wall being from 295·5° Fahr. to 293·5° Fahr.

During the period between *g* and *f* the steam temperature is in excess of the wall temperature, and heat passes from the steam to the metal. Assuming that the rate of transfer is proportional to the difference of temperature, the product of the mean temperature difference in degrees Fahrenheit and the time of absorption in seconds, or the area of the shaded figure bounded by the steam temperature curve *gcf* and the metal temperature curve *gf*, will be a measure of the amount of heat given by the steam to the metal. Callendar and Nicholson called this the "condensation area." The amount of condensation taking place on any part of the walls per cycle is assumed to be proportional to the average excess of temperature of the steam multiplied by the time during which the temperature is above that of the walls. The heat exchanged is measured by the product of the mean excess temperature in degrees Fahr. and the number of sixtieths of a revolution. This product gives the result in degree-seconds per minute. If it is divided by the number of revolutions per minute, the area is thus expressed in degree-seconds per cycle.

Let N=speed in rev./min.

$n$=number of $\frac{1}{60}$ of revolution in *gf* (base of area).

$t_m$=mean of temperature difference in ° Fahr. (mean ordinate of area).

Then the condensation area is expressed as

$$\left.\begin{aligned} A_1 &= nt_m \ \frac{\text{degree-seconds}}{\text{min.}} \\ A &= \frac{nt_m}{N} \ \frac{\text{degree-seconds}}{\text{cycle}} \end{aligned}\right\} \quad . \quad . \quad . \quad . \quad (52)$$

For example, in the case shown in Fig. 117, $n$ =8 sixtieths, N=100, $t_m$=18·8° Fahr.,

$$A_1 = 8 \times 18{\cdot}8 = 150{\cdot}4 \text{ deg.-sec./min.}$$

or

$$A = \frac{150{\cdot}4}{100} = 1{\cdot}504 \text{ deg.-sec./cycle.}$$

When the values of the heat quantities (Q) given to the metal over the surface area considered were calculated and expressed in B.Th.U./ft.$^2$ deg.-sec., and then compared with the corresponding condensation areas, it was found that the ratio F=Q/A was fairly constant. It varied between the limits of 0·58 and 0·76, and

Callendar and Nicholson suggested a mean value of 0·61 B.Th.U./deg.-sec. ft.$^2$, so that the heat transmitted to the surface can be expressed as—

$$h=0{\cdot}61A \text{ B.Th.U./ft.}^2 \text{ cycle}=0{\cdot}61AN \text{ B.Th.U./ft.}^2 \text{ min.} \quad . \quad (53)$$

and for the surface of area $s$ ft.$^2$ the heat transmitted is—

$$h_s=0{\cdot}61ANs \text{ B.Th.U./min.} \quad . \quad . \quad . \quad . \quad (54)$$

For example, in one of the trials the surface area of the cylinder cover was $s=0{\cdot}6$ ft.$^2$; condensation area $A=1{\cdot}9$ $\frac{\text{degree-sec.}}{\text{cycle}}$; speed N=97 rev./min., hence the heat absorbed by the cover surface was—

$$h_s=0{\cdot}61\times1{\cdot}9\times97\times0{\cdot}6=67{\cdot}46 \text{ B.Th.U./min.}$$

Various elements of the cylinder surface can be dealt with in this way and the heat absorption for the whole condensation surface can thus be approximated.

There is a limit of cyclical condensation for any given cycle. If the conditions of external and internal heat loss are such that the mean temperature of the clearance surface (on which the major part of the condensation takes place) is reduced to the mean temperature of the steam, the condensation and evaporation areas on the cycle diagram will be equal. If the temperature of the clearance surface falls below this point evaporation will be incomplete. Water will thus accumulate in the cylinder until a balance is produced by mechanical removal of the excess. It follows then that whatever the quantity of condensed steam rejected in the form of water, it represents the communication of a quantity of heat to the walls equivalent to the total heat of this condensed steam reckoned from the temperature at which the water is thrown off from the walls.

The mean temperature of the steam cycle is thus a natural minimum of temperature for the wall surface corresponding to a maximum limit of condensation for any given cycle.

In order to deduce this limiting value of the condensation per square foot per hour, it is simply necessary to draw on the cycle diagram the steam temperature curve and the mean steam temperature line (dotted), as shown in Fig. 117.

The area above this line is the maximum condensation area corresponding to this particular cycle.

This value increases with speed, because the temperature range of the metal surface is reduced and the effective condensation area is increased.

In the majority of practical cases the maximum point of the wall surface temperature coincides with the steam temperature at cut-off, while the point at which the wall temperature crosses the steam temperature curve is usually near the release point.

For the estimation of the limiting condensation at cut-off and

release Callendar and Nicholson modified the constant F from 0·61 to 0·74 and introduced the following correction factors for speed. Taking N as the speed in rev./min., the correction factor at cut-off is $a=\frac{(1+\sqrt{N})}{(3+\sqrt{N})}$. The correction factor at release is $a=\frac{\sqrt{N}}{(3+\sqrt{N})}$.

In either case the heat absorbed under the limiting conditions is given by—

$$H=0{\cdot}74aA(C+b) \text{ B.Th.U./min.}$$

where C is the total clearance surface, which includes the surface of the cover, the surface of the piston face, the counter bore surface, and $b$ the surface of the barrel up to the point of the stroke considered, all in ft.$^2$

To obtain the corresponding condensation value the heat absorbed has to be divided by the mean value of the latent heat. In general, between pressure limits of 100 to 150 lb./in.$^2$, the latent heat varies from 890 to 870 B.Th.U./lb. Taking an average of 885, then to an approximation the limiting condensation may be estimated in general from—

$$\left.\begin{aligned} W_c&=\frac{H}{885}=\frac{0{\cdot}74}{885}aA(C+b) \text{ lb./min.} \\ \text{or } W_c&=\frac{H\times 60}{885}=0{\cdot}05aA(C+b) \text{ lb./hour} \end{aligned}\right\} \quad . \quad (55)$$

In this way the maximum possible condensation may be estimated for a given case, but whether this limit actually is reached or not is a question which can only be settled by the determination experimentally of the mean temperature of the cylinder cover wall, a figure which is difficult to estimate with accuracy. Provided, in a special case, thermometers are bedded in the wall at various points and the mean temperature obtained, then by drawing the mean wall temperature line on the cycle diagram, it can be seen how near the condition of limiting condensation is approached. If the mean metal and steam temperatures have the same value, then the limiting condition has been reached. This method of Callender and Nicholson, by which the heat absorption and the condensation is derived by calculation for the wall temperatures, as a rule, however, gives values which fall much below the apparent condensation rates, and their validity is open to question. It seems hardly reasonable to attribute the large discrepancies usually found as due to leakage alone, and consequently many engineers do not accept the results obtained by this method.

**131.** Several empirical formulæ have since been evolved for the calculation of the approximate cylinder condensation. One of these, which appears to give reasonable results, is due to Prof. Heck,* who states that it " harmonises the variant data about as well as can be expected, and has a wide range of application.

* *Steam Engine and Turbine* (1911), p. 175.

" The formula is intended to give the missing quantity at cut-off in an engine which is tight enough to show but an inappreciable amount of leakage at valves and piston, tested when standing still in the usual fashion. The engine is supposed to receive ordinary ' commercially dry ' steam, which generally shows from 0·5 per cent. to 1·5 per cent. of moisture when tested with the throttling calorimeter. The formula is not applicable to the lower cylinders of multiple-expansion engines, except in the few cases where the water in the exhaust from the high cylinder is allowed to separate out in the comparative quietude of a large receiver and is drained away, leaving dry steam to enter the lower cylinder. It does not cover the use of superheated steam, nor does it apply when steam jackets are in action."

The expression is as follows:

$$m=\frac{C}{\sqrt[3]{N}}\sqrt{\frac{s\delta t}{pe}} \quad . \quad . \quad . \quad . \quad . \quad . \quad (56)$$

where $m$=ratio of missing quantity to total stuff at cut-off=$w_m/w$.

C=coefficient varying with type of engine; range 0·25 to 0·30; average 0·27.

N=speed in rev./min.

$s$ =factor for condensing surface=(cylinder surface)/(stroke volume).

$\delta t$=a virtual temperature range in cylinder between initial and back pressures, $p_1$ and $p_2$.

$p$ =pressure at cut-off or point immediately after it, for which the steam consumption is found.

$e$ =ratio of total volume at cut-off to stroke volume=$V_a/V_o$.

With regard to the factor $s$, which is the normal surface per cubic foot of piston displacement, this is obtained by dividing the sum of the cover surface, the piston face surface, and the total barrel surface, by the stroke volume,

$$\text{or} \qquad s=\frac{1}{144}\left(2\frac{\pi}{4}D^2+\pi DL\right)\Big/\left(\frac{\pi}{4}D^2L\right)\frac{1}{1728}=24\left(\frac{1}{L}+\frac{2}{D}\right) \quad . \quad (57)$$

where D is the diameter in inches and L the stroke in inches. With regard to the temperature range in the cylinder Heck states, " it was found that the very large increase in range, caused by dropping to condenser temperatures, exerted too great an influence in the formula; while in the first cylinder of a multiple-expansion engine the range was too small to account for the condensation. To use the range of pressure would give an error in just the opposite direction, making the ' moisture ' figure too small with condensing engines and too large with compounds."

To get over this difficulty, in a simple way, he substitutes what may be termed a virtual temperature curve in place of the real saturation temperature curve, the temperatures being progressively

increased between 80 lb./in.$^2$ abs. to 300 lb./in.$^2$ abs. and decreased between 80 and 10 lb./in.$^2$ abs.

This curve AC, which is reproduced in Fig. 118, is purely empirical, and laid out by trial for various cases. The curve AB of saturation temperatures is shown dotted for comparison.

If $p_1$ and $p_2$ are the highest and lowest pressures in the cylinder,

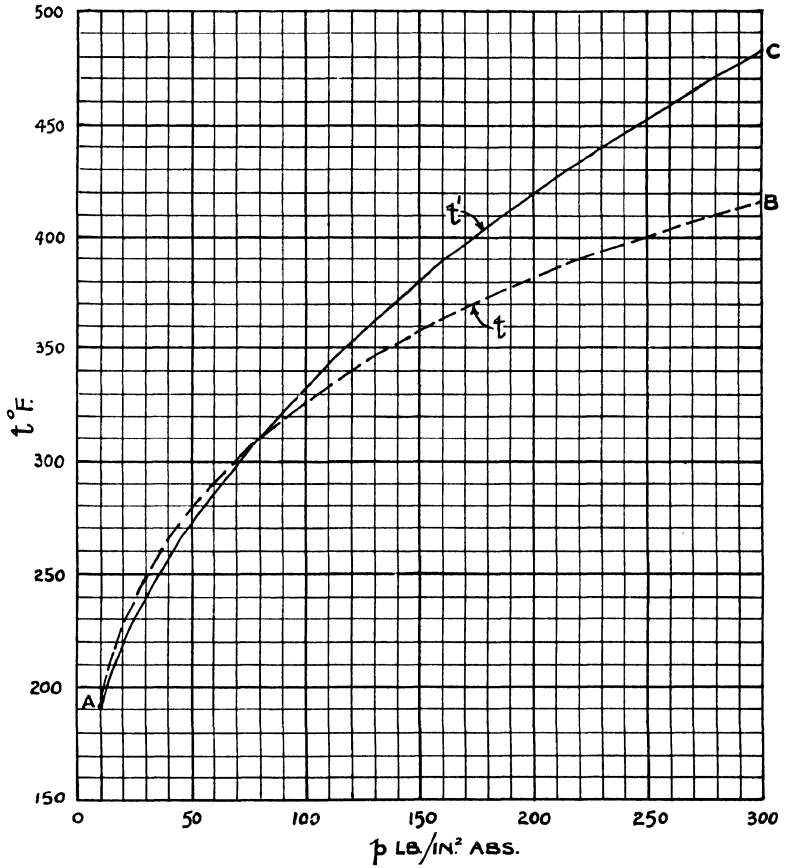



FIG. 118.

the value of $t_1'$ is read at $p_1$ and $t_2'$ at $p_2$ and $\delta t=(t_1'-t_2')$. The temperatures are in Fahrenheit degrees.

*Example* 14.—Apply Heck's empirical equation to calculate the missing quantity for the case of the engine of Example 1, having cylinder 6·5 in. diam. and 14 in. stroke, running at 150 rev./min. The maximum initial pressure is 98 lb./in.$^2$ abs., the minimum 15 lb./in.$^2$ abs., and cut-off pressure, 88 lb./in. abs. The stroke

volume is 0·261 ft.$^3$ and total volume to cut-off is 0·1268 ft.$^3$ Total weight of stuff in cylinder during expansion, 0·03445 lb.

Here $N=150$ and $\sqrt[3]{N}=\sqrt[3]{150}=5{\cdot}3$.

$D=6{\cdot}5$, $L=14$ $\therefore s=24\left(\frac{1}{L}+\frac{2}{D}\right)=24\left(\frac{1}{14}+\frac{2}{6{\cdot}5}\right)$

$=24\times(0{\cdot}0175+308)=24\times0{\cdot}3795=9{\cdot}108$

$p_1=98$, $t_1'=333$
$p_2=15$, $t_2'=207$ } from temperature curve Fig. 118.

$\therefore \delta t=(333-207)=126$

$V_a=0{\cdot}1268$, $V_o=0{\cdot}261$. $\therefore e=V_a/V_o=\frac{0{\cdot}1268}{0{\cdot}261}=0{\cdot}485$; $p_a=85$

$pe=85\times0{\cdot}485=41{\cdot}2$ $C=0{\cdot}27$ (average).

Then $m=\frac{0{\cdot}27}{5{\cdot}3}\sqrt{\frac{9{\cdot}108\times126}{41{\cdot}2}}=\frac{0{\cdot}27}{5{\cdot}3}\sqrt{27{\cdot}85}=\frac{0{\cdot}27\times5{\cdot}28}{5{\cdot}3}=0{\cdot}27$

Missing quantity $w_m=mw=0{\cdot}27\times0{\cdot}03445=0{\cdot}0093$. Compared with the value of 0·01 in Example 1, a defect of 7 per cent.

# CHAPTER VII

## MULTIPLE-EXPANSION ENGINES

**132.** In the previous chapter the reduction of thermal loss and consequent improvement in economy and efficiency by jacketing and superheated steam have been considered relatively to the use of moderate initial pressure with a low ratio of expansion in a single-cylinder engine

Where steam of high pressure is used with a large range of expansion in the customary type of engine, where "contra-flow" conditions exist during admission and exhaust, expansion in a single cylinder is attended with serious disadvantages; and to avoid these, compounding or the division of the pressure drop into several stages is resorted to, that is, instead of a single expansion a multiple-expansion engine is employed.

The objection to working the steam at high pressure and large expansion in one cylinder, which become more serious as the pressure and number of expansions are increased, are as follows:

1. The cylinder must have a volume sufficiently large for the required amount of expansion, and it must be sufficiently strong to safely withstand the maximum pressure. The working parts of the engine have to be made large enough to transmit the maximum load; and since the maximum pressure is greatly in excess of the mean, when the number of expansions or ratio of expansion is large, the single-cylinder engine is excessively heavy and costly in proportion to the power developed.

2. The initial condensation loss increases rapidly, as the number of expansions in a single cylinder increases.

3. The turning effort on the crank-pin is very variable.

By the use of multiple expansion, the range of stress on the working parts, initial condensation loss, and variability of turning effort are much reduced. Hence, as regards smoothness of working and economy, with wide range of pressure and expansion, the multiple-expansion engine is a decided advance on the simple or single-cylinder engine in which the "contra-flow" principle for admission and exhaust is adhered to.

In modern practice, however, the single-cylinder "contra-flow" and the compound (two-stage) contra-flow type are often replaced by a single-cylinder "uniflow" engine, when a moderate output is required. The particular features of this type are discussed in Chapter XV.

The reduction of the initial condensation loss in the multiple-expansion engine may be accounted for as follows. As shown in the previous chapter, the heat transfer between steam and metal is dependent on the difference between the initial steam temperature and that of the metal this steam comes in contact with. In the multi-stage engine the only cylinder metal in contact with steam at condenser pressure is that of the last or low-pressure cylinder. The further any given cylinder is removed from the low-pressure cylinder the higher the mean temperature of its walls, so that as the steam flows through the successive stage cylinders the temperature range in each cylinder is greatly reduced and also the corresponding difference between the steam and metal temperatures. The natural result is a reduction of the total condensation which would result from the use of a single-stage engine.

Further, the initial condensation loss in the successive cylinders is not cumulative, but is approximately that due to the L.P. cylinder. The water of condensation in any cylinder before the low-pressure one is usually re-evaporated during the exhaust stroke and passes on again as steam to the next cylinder, and part of its energy is utilised in this cylinder.

**133. Methods of Compounding.**—The essential feature of compounding is to exhaust the steam from one cylinder into a second cylinder of larger volume, where it may do further work by continued expansion. This process may be repeated through three, four, or more successive cylinders.

The multiple-expansion engine thus formed is equivalent to a set of simple engines working " in series."

The compounding may be accomplished by exhausting from a higher pressure into a lower-pressure cylinder in any of the following ways:

1. A cylinder of larger diameter, the stroke for the two cylinders being the same. This is the most common arrangement.
2. A cylinder of the same diameter having a longer stroke.
3. A cylinder having the same dimensions as those of the higher-pressure cylinder, with its piston making a larger number of reciprocations per minute and the engines working on independent cranks.
4. Any combination of these methods.

In all cases the work done in a cylinder, or between the pistons of a compound engine, or in any combination whatever of cylinders and pistons, is—

$$E = P_m(V_2 - V_1)$$

where $P_m$ is the mean effective pressure, $V_1$ the initial and $V_2$ the final volume while the steam is enclosed; it is independent of the manner in which the increase of volume is obtained.

Two-stage or double-expansion engines may have a tandem arrangement of H.P. and L.P. cylinders, as shown in Fig. 119, or may have the cylinders placed side by side, as shown in Fig. 120.

In the latter case there are two cranks at 90°, ensuring a better turning moment. In the case of the tandem (Fig. 119), which is known as the Woolf type, the steam is exhausted directly from the top of the H.P. piston to the bottom of the L.P. piston, and vice versa, during the whole stroke, as indicated by the directional arrow lines.

Alternatively, this type may have side-by-side cylinders with cranks at 180°, so that the two pistons commence their respective strokes simultaneously, as in the case of the tandem.

In the usual type there is an intermediate chamber between the H.P. and L.P. cylinders called a "receiver," into which the H.P. exhaust is discharged, so that it may be retained until the admission valve of the L.P. cylinder opens. In the case of a triple

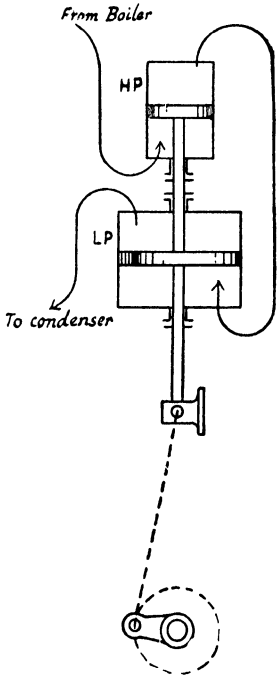



Fig. 119.

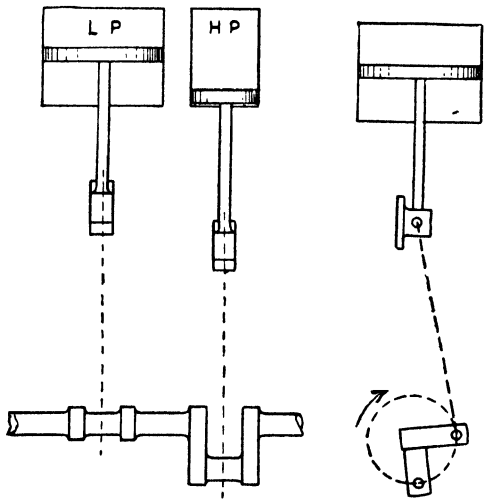



Fig. 120.

there is another receiver between the I.P. and L.P. cylinders, and so on for any further stages.

**134. Number of Cylinders.**—When the L.P. terminal pressure at the end of expansion (usually taken as hyperbolic) and the number of expansions or cut-off in each cylinder are fixed, the number of cylinders will depend on the total range of pressure, and will increase as the initial pressure increases.

For condensing engines the L.P. terminal pressure varies from 10 to 12 lb./in.$^2$ abs., and for non-condensing engines, from 20 to 25 lb./in.$^2$ abs.

Single-cylinder or simple engines are usually run non-condensing

with initial pressures from 90 to 100 lb./in.$^2$ abs. Compounds condensing with pressures from 100 to 150 lb./in.$^2$ abs., triples with pressures 150 to 200 lb./in.$^2$ abs., and quadruples with pressures from 200 to 300 lb./in.$^2$ abs.*

In order to show the general action of the steam in its passage through the various stages of a multi-cylinder engine, the theoretical conditions and corresponding indicator diagrams only are considered in the following section—that is, all effects due to clearance, compression, wiredrawing, condensation, etc., are neglected, in order to simplify the explanations. Hyperbolic expansion is assumed throughout. The real diagrams for any given case may differ materially from the theoretical.

**135.** The tandem type of compound engine with no receiver

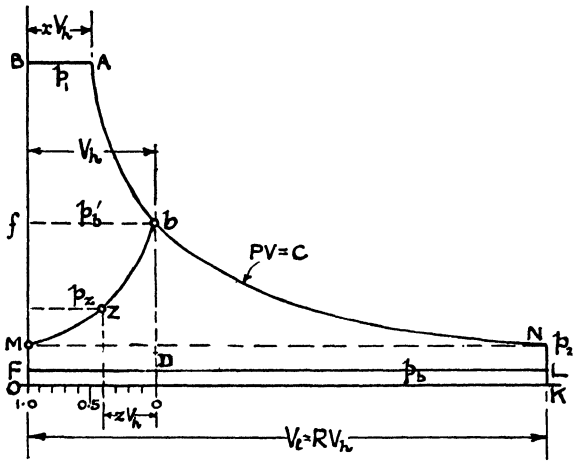



FIG. 121.

and working on one crank, or its equivalent, the side-by-side with two cranks at 180°, is considered first.

In Fig. 121—

let $fb$=FD=$V_h$, the stroke volume of the H.P. cylinder.
„ FL=$V_l$ „ „ „ L.P. cylinder.

The volume ratio is $R=\frac{V_l}{V_h}$, and is usually written as $\left(\frac{\text{L.P.}}{\text{H.P.}}\right)$.

* Within the past five years proposals for the use of very high pressures in reciprocators have been made. A vertical triple-expansion set

$$\frac{14\frac{1}{2}\text{ in.}+21\frac{1}{2}\text{ in.}+25\text{ in.}}{25\text{ in. str.}}$$

running at 225 r.p.m., with superheated steam at 1840 lb./in.$^2$ and 820° Fahr., and developing 6000 I.H.P. was put in operation at the works of the Philip Carey Co., U.S.A., on 3 May, 1931. See *Power*, 19 May, 1931.

Let steam at initial pressure $p_1$ be admitted above the H.P. piston and cut-off be at $x$ fraction of the (common) stroke. This steam expands until the end of the stroke, between A and $b$, when it has a pressure $p_b'$, and communication is opened to the under side of the L.P. piston. It continues to expand during the up-stroke below the L.P., while the H.P. piston is moved up against the gradually decreasing resistance. At the end of the up-stroke the H.P. cylinder is empty and the L.P. cylinder full of steam at the L.P. terminal pressure $p_2$. The original volume $x\mathrm{V}_h$ at $p_1$ has been expanded to the final volume, $\mathrm{V}_l=\mathrm{R}\mathrm{V}_h$ at $p_2$, and the absolute amount of work done by it is given by the area below BAN down to the line of zero pressure, or OBANK. On the down-stroke this steam is exhausted to the condenser at constant back pressure $p_b$, and the negative work done on it is the area below FL, or OFLK. The net ideal indicated work per cycle is thus given by BANLF.

The distribution of this work between the cylinders is shown by the compound diagrams BA$b$M for H.P. and M$b$NLF for L.P. The curve $b$M shows the gradually diminishing back pressure on the top of the H.P. piston during the up-stroke. The points on this curve can be obtained as follows. Suppose the pressure is required when the H.P. piston has moved through a fraction of the upward stroke. The volume above the H.P. piston will be $\mathrm{V}_h(1-z)$ and the volume below the L.P. piston $z\mathrm{V}_l=z\mathrm{R}\mathrm{V}_h$, so that the total volume occupied by the steam at pressure $p_z$ is $\mathrm{V}_h[1+z(\mathrm{R}-1)]$.

Assuming the usual convention that the expansion curve is sufficiently approximated by a rectangular hyperbola, with saturated steam, then—

$$p_b'V_h=p_z\mathrm{V}_h[1+z(\mathrm{R}-1)]$$

or

$$p_z=\frac{p_b'}{1+z(\mathrm{R}-1)} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

The intersection of the vertical at fraction $z$ and the horizontal at $p_z$ give the required point Z on the H.P. back pressure or exhaust curve $b$M.

At M the pressure $p_m$ should be the same as $p_2$, since $z=1$.

$$p_m=p_z=\frac{p_b'}{\mathrm{R}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

*Example* 1.—A double-acting pumping engine of Woolf type has cylinders 21 in. and 42 in. and stroke 36 in. Initial H.P. pressure, 100 lb./in.$^2$ abs.; L.P. back pressure, 3 lb./in.$^2$ abs.; H.P. cut-off, 0·5 stroke; speed, 20 rev./min. Draw the theoretical combined indicator diagram, assuming hyperbolic expansion; find the ratio of the powers developed in the L.P. and H.P. cylinders, and the total horse-power.

Referring to Fig. 121, $p_1=100$; $x=0{\cdot}5$—

$$p_b'=\frac{p_1x\mathrm{V}_h}{\mathrm{V}_h}=p_1x=100\times0{\cdot}5=50 \text{ lb./in.}^2$$

This fixes point $b$ and stroke volume $V_h$.

$$\text{Cylinder ratio } R=\frac{V_l}{V_h}=\left(\frac{D_l}{D_h}\right)^2=\left(\frac{42}{21}\right)^2=4 \qquad \begin{matrix} D_l=42 \text{ in.} \\ D_h=21 \text{ in.} \end{matrix}$$

Set off OK=4FD, and draw KN. This gives the total volume $V_l=RV_h$. The L.P. terminal pressure is—

$$p_2=\frac{p_1 x V_h}{RV_h}=p_1\times\frac{x}{R}=\frac{100\times 0{\cdot}5}{4}=12{\cdot}5 \text{ lb./in.}^2 \text{ abs.}$$

Other points on AN can be obtained by taking intermediate expansion ratios. Also $p_b=3$ lb./in.$^2$ abs. To draw in $b$M, consider, for example, the point corresponding to $z=0{\cdot}4$ on the H.P. upstroke, then by Equation 1—

$$p_z=\frac{p_b'}{1+z(R-1)}=\frac{50}{1+0{\cdot}4\times 3}=\frac{50}{2{\cdot}2}=22{\cdot}73 \text{ lb./in.}^2 \text{ abs.}$$

On the original diagram (Fig. 121) OK=4 in., so that the volume scale is—

$$1 \text{ in.}=\frac{V_l}{4}=\frac{28{\cdot}863}{4} \qquad V_l=\frac{\pi}{4}\times 3{\cdot}5^2\times 3$$
$$=7{\cdot}215 \text{ ft.}^3 \qquad =28{\cdot}863 \text{ ft.}^3$$

The pressure scale is 1 in.=40 lb./in.$^2$

Hence the heat scale of the diagram is—

$$1 \text{ in.}^2=\frac{144\times 40\times 7{\cdot}215}{778}=53{\cdot}416 \text{ B.Th.U.}$$

By planimeter, the H.P. area BA$b$M $=1{\cdot}488$ in.$^2$
,, ,, L.P. ,, M$b$NLF$=1{\cdot}937$ in.$^2$
Total area $=3{\cdot}425$ in.$^2=$A

$$\therefore \quad \frac{\text{Indicated work L.P.}}{\text{Indicated work H.P.}}=\frac{1{\cdot}937}{1{\cdot}488}=\frac{1{\cdot}3}{1}$$

The work done in the L.P. is thus 30 per cent. greater than that in the H.P. cylinder.

Total work per cycle (one side of piston) in heat units—

$$h_i=sA=53{\cdot}416\times 3{\cdot}425=182{\cdot}95 \text{ B.Th.U./cycle.}$$

Total horse-power $I=\frac{2h_iN}{42{\cdot}4}=\frac{2\times 182{\cdot}95\times 20}{42{\cdot}4}=172{\cdot}5.$

The horse-power actually developed was 141, or $\frac{141\times 100}{172{\cdot}5}$ $=81{\cdot}7$ per cent. of the theoretical value.

**136. Compound Engine with Receiver between H.P. and L.P. Cylinders.**—In this case the steam is exhausted from the H.P. cylinder into the receiver, and during part of the stroke it flows directly through the receiver into the L.P. cylinder. The L.P. admission valve, however, cuts off the steam before the whole

exhaust stroke is completed in the H.P. cylinder, with the result that the H.P. piston compresses the steam in the receiver. The condition is shown in Fig. 122 for the tandem engine with receiver. The H.P. back pressure falls between $b$ and Z, the point of the H.P. stroke at which the L.P. admission valve closes, and then rises between Z and M. The receiver pressure at M should be the same as the release pressure $p_b'$ from the H.P. cylinder, otherwise a "receiver drop" occurs. In the L.P. cylinder expansion takes place between M and Q, while the H.P. is exhausting, and the back pressure falls from $p_b'$ to $p_z$. The rest of the expansion is carried out wholly in the L.P. cylinder between $p_q=p_z$ and the terminal pressure $p_2$. The H.P. diagram is BA$bz$M, and the L.P., which overlaps it, is MQNLF.

The exhaust and compression curves $b$Z and ZM of the H.P.

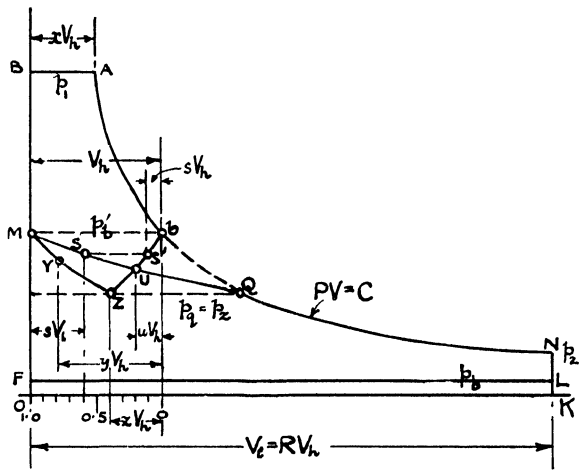



FIG. 122.

cylinder and the expansion curve of the L.P. cylinder are determined as follows. Let the volume of the intermediate receiver be expressed as a multiple of the H.P. stroke volume, or $V_r=kV_h$. Let Z be the point of the up-stroke in the H.P. cylinder at fraction $z$ when the L.P. admission valve closes and compression begins in the H.P. and receiver.

When the H.P. exhaust valve opens, the receiver volume $kV_h$ and the H.P. stroke volume $V_h$ are filled with steam at $p_b'$ and the total initial volume is $V_h(1+k)$.

When the H.P. piston has travelled $z$ of the up-stroke the volume filled with steam at $p_z$ in H.P., receiver, and L.P. is—

$$(1-z)V_h+kV_h+zRV_h=V_h(1-z+k+Rz)$$

Hence
$$p_b'(1+k)=p_z[1-z+k+Rz] \quad . \quad . \quad . \quad . \quad (3)$$

When compression starts in the H.P., the volume of steam at $p_z$

in H.P. and receiver is $(1-z+k)V_h$, and at the end of the compression, it is the receiver volume $kV_h$, and the pressure is $p_b'$.

Hence $$p_b'k=p_z(1-z+k)$$

or $$p_z=\frac{p_b'k}{(1-z+k)} \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

Substituting from (4) in (3) and reducing, the final result is—

$$z=\frac{(1+k)}{(1+Rk)} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

$p_z$ is calculable from (4) and the point Z is then defined.

The pressure at any point U on the H.P. exhaust curve, when the piston has travelled a fraction $u$ of the up-stroke, is given by—

$$p_u=\frac{p_b'(1+k)}{(1+k)+u(R-1)} \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

and that at any point Y on the compression curve after the piston has travelled a fraction $y$ of the stroke is given by—

$$p_y=p_z\frac{(1-z+k)}{(1-y+k)} \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

When a very large receiver is used, that is, when $k$ becomes indefinitely great, the fractional displacement of the H.P. piston reaches a limiting value of $z=\frac{1}{R}$, and the point Z falls on the constant pressure exhaust line $b$M.

Any point S on the expansion curve of the L.P. cylinder between admission M and cut-off Q can be obtained by projection from the corresponding point on the H.P. back pressure curve $b$Z. Draw a vertical through the fractional displacement value $sV_h$ on the H.P. base scale to cut $b$Z in S′, and draw the pressure horizontal S′S. Through the $sV_l$ value on the L.P. base scale draw a vertical to cut S′S in S.

By taking several S′ points between $b$ and Z, the corresponding S points on MQ can be located.

*Example* 2.—In the case of the tandem pumping engine of Example 1, draw the theoretical H.P. and L.P. diagrams when a receiver of the same capacity as the H.P. cylinder is used, and find the work ratio and the total horse-power.

As before, $p_b'=p_m=50$. For this condition

$$z=\frac{1+k}{1+Rk}=\frac{2}{1+4}=\frac{2}{5}=0{\cdot}4 \qquad \begin{matrix} k=1 \\ R=4 \end{matrix}$$

and $$p_z=\frac{p_b'k}{1-z+k}=\frac{50\times1}{1-0{\cdot}4+1}=\frac{50}{1{\cdot}6}=31{\cdot}25 \text{ lb./in.}^2 \text{ abs.}$$

This fixes points $b$ and Z. To find any other point, say U, for $u=0{\cdot}2$—

$$p_u=\frac{p_b'(1+k)}{(1+k)+u(R-1)}=\frac{50\times2}{2+0{\cdot}2\times3}=\frac{100}{2{\cdot}6}=38{\cdot}5 \text{ lb./in.}^2 \text{ abs.}$$

To find any point, say Y, on the compression curve ZM, corresponding to displacement $y$ of H.P. piston of 0·8,

$$p_y=p_z\frac{(1-z+k)}{(1-y+k)}=31{\cdot}25\frac{(1-0{\cdot}4+1)}{(1-0{\cdot}8+1)}=\frac{31{\cdot}25\times1{\cdot}6}{1{\cdot}2}=41{\cdot}6 \text{ lb./in.}^2 \text{ abs.}$$

To find a point, say S, at $\frac{1}{10}$ of the L.P. stroke, draw the vertical through $s=0{\cdot}1$ on the H.P. base scale to cut $b$Z in S′, and draw S′S; project from $s=0{\cdot}1$ on the L.P. base scale to cut S′S in S. The cut-off point Q is obtained by drawing ZQ to cut the vertical through $s=0{\cdot}4$ on the L.P. base scale.

The terminal pressure is—

$$p_2=p_a\times s=31{\cdot}25\times0{\cdot}4=12{\cdot}50$$

The points A, $b$, Q, and N lie on the rectangular hyperbola drawn through A.

In this case the area of the H.P. diagram BA$b$ZM is 1·116 in.², and that of the L.P. diagram MQNLF is 2·2475 in.²

Hence $$\frac{E_L}{E_H}=\frac{2{\cdot}2475}{1{\cdot}116}=\frac{2{\cdot}01}{1}$$

The work done on the L.P. is thus 100 per cent. greater than that in the H.P. cylinder.

The total area is A=3·3635 in.², and the indicated horse-power is

$$I=\frac{53{\cdot}416\times3{\cdot}3635\times2\times20}{42{\cdot}4}=170$$

**137.** If the volume of the receiver or chamber between the cylinders of compound engines (including the exhaust chamber and exhaust pipe of the first cylinder, and the valve chest of the second cylinder) were indefinitely large, then the back pressure line of the small cylinder and the forward pressure line of the large cylinder would each be a horizontal straight line, as shown in the approximate diagrams (Figs. 123 and 124).

In practice the receiver volume is from one and a half to several times the volume of the high-pressure cylinder; and the effect of the restricted volume of the receiver is to make the back pressure line of the high and the admission line of the low-pressure diagram somewhat irregular.

The theoretical form of these portions of the diagrams is seen in Fig. 137, which represents the pressures in the receiver, assuming PV=C.

The receiver volume is usually made as small as possible to avoid loss of heat by radiation, but the necessities of the design determine the volume. Other things being equal, the effect on the power-distribution, of a small receiver, is to increase somewhat the back pressure against the high-pressure piston, and increase the initial pressure on the low. Increasing the volume of the receiver, therefore, increases to a small extent the area of the high-pressure diagram, and decreases the area of the low.

An increase of pressure sometimes occurs in the low-pressure cylinder towards the point of cut-off, shown in practice as a more or less sudden increase of pressure during admission on the low-

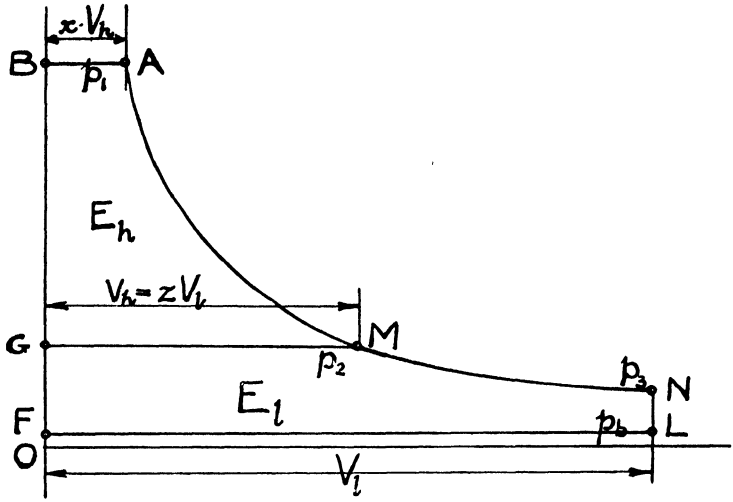



FIG. 123.

pressure diagram, especially when the engine is running slowly. This is due to the high-pressure cylinder exhaust passing into the receiver before cut-off has taken place in the low.

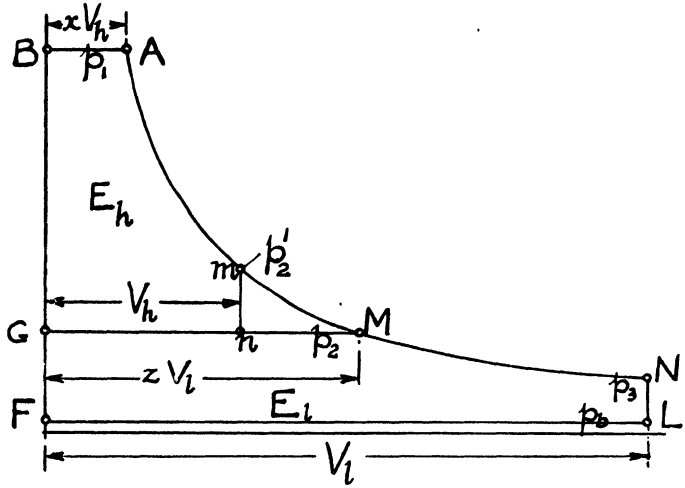



FIG. 124.

**138. Distribution of Work and Piston Loads.**—Consider the case of the double-expansion engine with receiver of sufficiently large capacity, so that the pressure during the H.P. exhaust stroke is

maintained constant and equal to the constant admission pressure of the L.P. cylinder to cut-off, as shown in Fig. 123.

With initial pressure $p_1$, terminal pressure $p_3$, back pressure $p_b$, and hyperbolic expansion, the total ideal work done in the two stages is FBANL=GBAM+FGMNL, or $E=(E_h+E_l)$. The expansion here, as is always the case, is incomplete in the L.P., but complete in the H.P. cylinder. The H.P. stroke volume is $V_h=zV_l$, the volume of the L.P. cylinder to cut-off.

The cylinder ratio is $R=\frac{FL}{GM}=\frac{V_l}{V_h}$. For the condition of complete expansion in the H.P. cylinder, its value will depend on the value fixed for the receiver pressure $p_2$, that is, on the point of cut-off M of the L.P. cylinder.

It will be apparent that for the development of the total power, widely different values of the cylinder ratio may be adopted, with corresponding variation of the indicated work areas GBAM and FGMNL.

Two conditions have to be considered : the distribution of piston load and the distribution of power. The ratio of volume between any two cylinders should be so fixed that the work areas and initial loads on the respective pistons may be made as nearly as possible equal.

The value of R in the case shown in Fig. 123 gives equality of work area, that is, FGMNL=GBAM or $E_l=E_h$. It causes, however, an unequal distribution of piston load. Each load is proportional to the product of the difference of initial and back pressure, and the cylinder volume. Denoting these by $W_l$ and $W_1$,

L.P. piston load $W_l \propto (p_2-p_b)V_l$
H.P. ,, ,, $W_1 \propto (p_1-p_2)V_h$

and the ratio is—

$$\frac{W_l}{W_1}=\frac{(p_2-p_b)}{(p_1-p_2)}R \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

which is less than unity.

The H.P. piston is more heavily loaded than the L.P.

This case is illustrated numerically by the next example.

*Example* 3.—The initial pressure of a double-expansion engine is 100 lb./in.$^2$ abs., the back pressure 4 lb./in.$^2$ abs., the total nominal expansion ratio 8, the cylinder ratio is chosen as 2, and the corresponding receiver pressure is 25 lb./in.$^2$ abs.

Find the ratio of the piston loads and the indicated work values.

$\frac{W_l}{W_1}=\frac{(p_2-p_b)}{(p_1-p_2)}R$ $\qquad p_1=100,\ p_b=4,\ p_2=25,\ R=2.$

$=\frac{(25-4)2}{(100-25)}=\frac{21\times 2}{75}=\frac{42}{75}=\frac{1}{1\cdot 8}$

The load on the H.P. piston is 80 per cent. greater than on the low.

$$E_l = p_{ml} V_l \text{ ; and } E_1 = p_{mh} V_h$$

or $$\frac{E_l}{E_1} = \frac{p_{ml} R}{p_{mh}}$$

$$p_{ml} = p_2 \frac{(1+\log_e r_l)}{r_l} - p_b$$

$$= 25\left(\frac{1+\log_e 2}{2}\right) - 4$$

$$= (21{\cdot}15 - 4) = 17{\cdot}15$$

$$p_{mh} = p_1 \frac{(1+\log_e r_h)}{r_h} - p_2$$

$$= 100\frac{(1+\log_e 4)}{4} - 25$$

$$= (59{\cdot}65 - 25) = 34{\cdot}65$$

$$\therefore \frac{E_l}{E_1} = \frac{17{\cdot}15 \times 2}{34{\cdot}65} = \frac{34{\cdot}30}{34{\cdot}65} = 1{\cdot}0$$

$$p_3 = \frac{p_1}{r} = \frac{100}{8} = 12{\cdot}5$$

$$p_2 = 25$$

$$r_l = \frac{p_2}{p_3} = \frac{25}{12{\cdot}5} = 2$$

$$p_b = 4$$

$$r_h = \frac{p_1}{p_2} = \frac{100}{25} = 4$$

In this case the condition of equal power distribution is accompanied by a very unequal distribution of piston load. The latter inequality can be reduced by an increase in the value of R, that is, by a reduction of H.P. volume, the receiver pressure being kept the same, that is, without alteration to the L.P. cut-off point M.

The effect of this reduction is to produce incomplete expansion in the H.P. cylinder with resulting "receiver drop" from some H.P. release pressure $p_2'$ to $p_2$ (Fig. 124). The work done in the H.P. cylinder is reduced to the area GBA*mn*, and the total work done between the initial and back pressure limits is reduced by the triangular area M*mn*. A compromise between these two conflicting conditions has to be made to approximately equalise the work done and the initial piston loads. The same reasoning holds for any successive pairs of cylinders of a multi-stage engine.

In the general case (Fig. 125) of a multi-stage engine having $m$ stages with incomplete expansion in the last or L.P. stage and complete expansion in all the other stages, the common receiver pressure ratio for equal power distribution is $\rho' = \frac{p_1}{p_2} = \frac{p_2}{p_3}$ etc., and is given by the expression—

$$\log_e \rho' = \frac{1}{m}\left(1 + \log_e r - \frac{r p_b}{p_1}\right) \quad . \quad . \quad . \quad . \quad . \quad (9)$$

where $r$ = total nominal ratio of expansion ; $p_1$ = H.P. initial pressure and $p_b$ = L.P. back pressure.

*Example* 4.—In the case of the compound engine of Example 3

with complete expansion in the H.P. cylinder, find the receiver pressure to give equal power in each cylinder.

Here $m=2$, $p_1=100$, $r=8$, $p_b=4$.

$$\therefore \log_e \rho' = \tfrac{1}{2}\left(1+\log_e 8-\frac{8\times 4}{100}\right)$$

$$= \tfrac{1}{2}(3{\cdot}0795-0{\cdot}32) = \tfrac{1}{2}\times 2{\cdot}7595 = 1{\cdot}3797$$

$$\log \rho' = \frac{1{\cdot}3797}{2{\cdot}3026} = 0{\cdot}5999 \quad \text{and} \quad \rho' = 3{\cdot}981$$

$$\therefore p_2 = \frac{p_1}{\rho_1} = \frac{100}{3{\cdot}981} = 25 \text{ lb./in.}^2 \text{ abs.}$$

the value given in Example 3.

*Example* 5.—In the case of the compound engine (Example 4) find the effect on the piston load and power distribution if the cylinder ratio is increased from 2 to 3 and the L.P. cut-off remains the same.

Here $p_2=25$, but $R=3$.

$$\therefore \frac{W_l}{W_1} = \frac{(25-4)3}{(100-25)} = \frac{21\times 3}{75} = \frac{63}{75} = \frac{1}{1{\cdot}2}$$

The H.P. piston load is now 20% greater than the L.P. one. The reduced expansion ratio in H.P. cyclinder is—

$$r_h = \frac{r}{R} = \frac{8}{3} = 2{\cdot}666$$

then

$$p_{mh} = 100\left(\frac{1+\log_e 2{\cdot}666}{2{\cdot}666}\right) - 25$$

$$= (70-25) = 45 \text{ lb./in.}^2 \text{ abs.}$$

and

$$\frac{E_l}{E_1} = \frac{17{\cdot}15\times 3}{45} = \frac{51{\cdot}45}{45} = \frac{1{\cdot}14}{1}$$

The power developed in the L.P. cylinder is now 14 per cent. greater than that in the H.P. The terminal pressure in the H.P. cylinder is now $p_2 = \frac{100}{2{\cdot}666} = 37{\cdot}5$ lb./in.$^2$ abs. and the receiver drop is $(37{\cdot}5-25) = 12{\cdot}5$ lb./in.$^2$

In addition to an alteration of the cylinder ratio the L.P. cut-off can also be reduced, arbitrarily.

*Example* 6.—In the case of the engine (Example 4), find the effect on the piston load and power distribution of an increase of cylinder ratio from 2 to 2·8 and a decrease in L.P. cut-off from 0·5 to 0·48.

Here $R=2{\cdot}8$, $r_l = \frac{1}{0{\cdot}48} = 2{\cdot}08$, and $r_h = \frac{r}{R} = \frac{8}{2{\cdot}8} = 2{\cdot}86$.

Receiver pressure $p_2 = r_l p_n = 2{\cdot}08 \times 12{\cdot}5 = 26$ lb./in.$^2$

and
$$\frac{W_l}{W_1} = \frac{(26-4)2{\cdot}8}{(100-26)} = \frac{22 \times 2{\cdot}8}{74} = \frac{61{\cdot}6}{74} = \frac{1}{1{\cdot}2}$$

$$p_{ml} = 26\left(\frac{1+\log_e 2{\cdot}08}{2{\cdot}08}\right) - 4$$

$$= 26 \times 0{\cdot}833 - 4$$

$$= (21{\cdot}66 - 4) = 17{\cdot}66 \text{ lb./in.}^2 \text{ abs.}$$

$$p_{mh} = 100\left(\frac{1+\log_e 2{\cdot}86}{2{\cdot}86}\right) - 26$$

$$= 100 \times 0{\cdot}717 - 26$$

$$= (71{\cdot}7 - 26) = 45{\cdot}7 \text{ lb./in.}^2 \text{ abs.}$$

then
$$\frac{E_l}{E_1} = \frac{17{\cdot}66 \times 2{\cdot}8}{45{\cdot}7} = \frac{49{\cdot}5}{45{\cdot}7} = \frac{1{\cdot}08}{1}$$

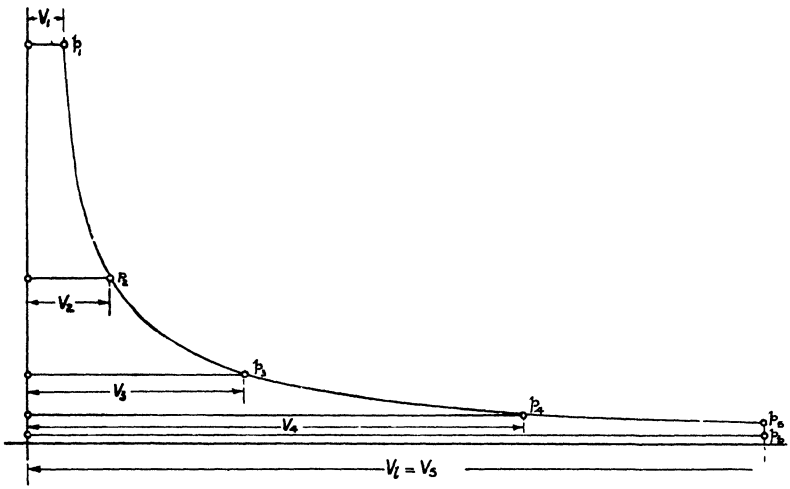



FIG. 125.

The H.P. piston load is still 20 per cent. greater than the L.P. one, but the power in the L.P. is now 8 per cent. instead of 14 per cent. greater than that in the H.P.

The H.P. terminal pressure is $\frac{100}{2{\cdot}86} = 35$, and the receiver drop $(35-26) = 9$ lb./in.$^2$, against 12·5 lb./in.$^2$ in the previous case.

**139.** In the case of a multi-stage engine where an arbitrary choice of cylinder ratios and receiver pressures is made, the corresponding expansion ratio in the successive cylinders can be obtained, for the calculation of the mean effective pressures, from the following expressions, which are given below for the general case of four-stage expansion.

Referring to Fig. 125, let the receiver pressure ratios be denoted by—

$$\rho_1=\frac{p_1}{p_2}\;;\quad \rho_2=\frac{p_2}{p_3}\;;\quad \rho_3=\frac{p_3}{p_4}$$

and the cylinder ratios by—

$$R_1=\frac{V_3}{V_2}\;;\quad R_2=\frac{V_4}{V_3}\;;\quad R_3=\frac{V_l}{V_4}$$

The ratios of expansion, which are the reciprocals of the cut-off fractions, are given by—

$$\text{H.P.}\qquad r_1=\frac{r}{R_1R_2R_3}=\frac{r}{R}\quad\ldots\ldots\quad(10)$$

$$\text{1st I.P.}\qquad r_2=\frac{r}{\rho_1R_2R_3}\quad\ldots\ldots\quad(11)$$

$$\text{2nd I.P.}\qquad r_3=\frac{r}{\rho_1\rho_2R_3}\quad\ldots\ldots\quad(12)$$

$$\text{L.P.}\qquad r_l=\frac{r}{\rho_1\rho_2\rho_3}\quad\ldots\ldots\quad(13)$$

For the case of a triple expansion $R_3=1$ in (12)
,, ,, two-stage ,, $R_2R_3=1$ in (11)
,, ,, simple engine $R=1$ in (10)

*Example* 7.—A triple-expansion marine engine has cylinder ratios $\frac{\text{L.P.}}{\text{H.P.}}=7{\cdot}1$ ; $\frac{\text{I.P.}}{\text{H.P.}}=2{\cdot}78$ ; $\frac{\text{L.P.}}{\text{I.P.}}=2{\cdot}56$.

The initial pressure is 195 lb./in.$^2$ abs. and the L.P. back pressure is 3 lb./in.$^2$ abs., while the total nominal ratio of expansion is 13. Find the values of the H.P., I.P. and L.P. cut-off fractions to give (ideal) receiver pressures of 64 lb./in.$^2$ abs. and 21 lb./in.$^2$ abs., and determine the relative values of the theoretical piston loads and indicated powers, referred to the L.P. cylinder.

The diagram for this case is shown in Fig. 126.

To compare the piston loads—

$$p_2=64,\ p_1=195\quad\therefore\ \rho_1=\frac{195}{64}=3{\cdot}04$$

$$p_3=21\quad\therefore\ \rho_2=\frac{p_2}{p_3}=\frac{64}{21}=3{\cdot}04\ ;\ p_4=3,\ R=7{\cdot}1$$

$$\therefore\ \frac{W_l}{W_1}=\frac{(p_3-p_b)R}{(p_1-p_2)}=\frac{(21-3)7{\cdot}1}{(195-64)}=\frac{18\times7{\cdot}1}{131}=\frac{127{\cdot}8}{131}=\frac{1}{1{\cdot}025}$$

$R_2=2{\cdot}56$.

$$\therefore\ \frac{W_l}{W_2}=\frac{(p_3-p_b)R_2}{(p_2-p_3)}=\frac{(21-3)2{\cdot}56}{(64-21)}=\frac{18\times2{\cdot}56}{43}=\frac{1}{0{\cdot}93}$$

The H.P. piston load is thus 2·5 per cent. greater and the I.P. load 7 per cent. less than the L.P. piston load.

To compare the powers—

H.P. expansion ratio $r_1=\frac{r}{R}=\frac{13}{7\cdot1}=1\cdot83$

Cut-off $x=\frac{1}{1\cdot83}=0\cdot546$

I.P. expansion ratio $r_2=\frac{r}{\rho_1 R_2}=\frac{13}{3\cdot04\times2\cdot56}$
$=1\cdot67$

Cut-off $y=\frac{1}{1\cdot67}=0\cdot60$

$R_2=2\cdot56$
$\rho_1=3\cdot04$
$r=13$

L.P. expansion ratio $r_3=\frac{p_3}{p_n}=\frac{21}{15}=1\cdot4$ $\qquad p_n=p_4=\frac{p_1}{r}=\frac{195}{13}=15$

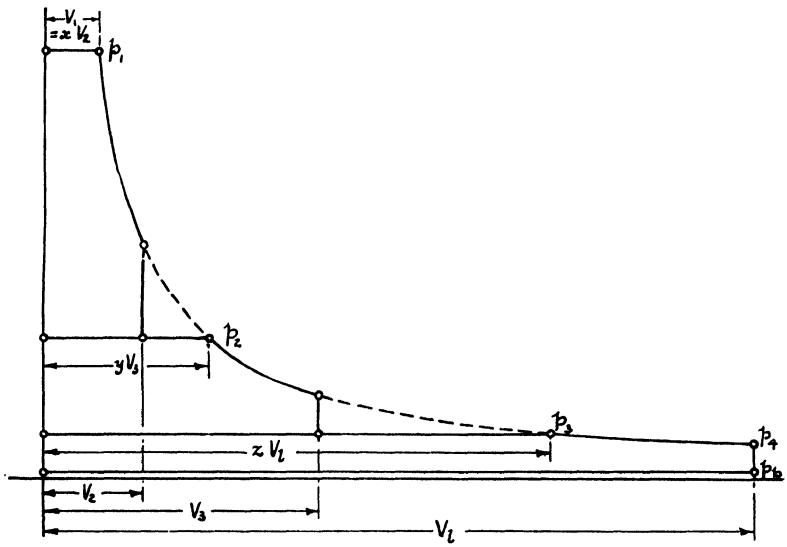



FIG. 126.

Cut-off $z=\frac{1}{1\cdot4}=0\cdot714$

M.E.P. H.P., $p_{mh}=p_1\frac{(1+\log_e r_1)}{r_1}-p_2$

$=195\frac{(1+\log_e 1\cdot83)}{1\cdot83}-64$

$=195\times0\cdot875-64$

$=(170\cdot62-64)=106\cdot62$ lb./in.$^2$

M.E.P. I.P., $p_{m2}=p_2\dfrac{(1+\log_e r_2)}{r_2}-p_3$

$=64\dfrac{(1+\log_e 1{\cdot}67)}{1{\cdot}67}-21$

$=64\times 0{\cdot}906-21$

$=(58{\cdot}00-21)=37{\cdot}00$ lb./in.$^2$

M.E.P. L.P., $p_{ml}=p_3\dfrac{(1+\log_e r_l)}{r_l}-p_b$

$=21\dfrac{(1+\log_e 1{\cdot}4)}{1{\cdot}4}-3$

$=21\times 0{\cdot}9545-3$

$=(19{\cdot}05-3)=16{\cdot}05$ lb./in.$^2$

Hence—

$$\frac{E_l}{E_1}=\frac{p_{ml}R}{p_{mh}}=\frac{16{\cdot}05\times 7{\cdot}1}{106{\cdot}62}=\frac{114}{106{\cdot}62}=\frac{1}{0{\cdot}935}$$

$$\frac{E_l}{E_2}=\frac{p_{ml}V_l}{p_{m2}V_3}=\frac{p_{ml}R_2}{p_{m2}}=\frac{16{\cdot}05\times 2{\cdot}56}{37{\cdot}0}=\frac{1}{0{\cdot}902}$$

The power developed in the H.P. is 6·5 per cent., and in the I.P. 9·8 per cent. less than that developed in the L.P. cylinder.

*Example* 8.—In the case of the engine of Example 7, find the theoretical loss of indicated work due to incomplete expansion in the H.P. and I.P. cylinders.

For the whole engine with complete expansion in H.P. and I.P. cylinders the total work done (Fig. 125) would be—

$E=p_mV_l$ $\qquad p_1=195$, $r=13$, $p_b=3$

$p_m=p_1\dfrac{(1+\log_e r)}{r}-p_b$

$=195\dfrac{(1+\log_e 13)}{13}-3$

$=195\times 0{\cdot}2785-3$

$=(54{\cdot}3-3)=51{\cdot}3$ lb./in.$^2$

Theoretical work done with incomplete expansion is—

$E'=p_{m1}V_2+p_{m2}V_3+p_{ml}V_l$ $\qquad p_{mh}=106{\cdot}62$, $R=7{\cdot}1$, $p_{m2}=37{\cdot}0$, $R_2=2{\cdot}56$, $p_{ml}=16{\cdot}05$

$=V_l\left(\dfrac{p_{mh}}{R}+\dfrac{p_{m2}}{R_2}+p_{ml}\right)$

$\therefore\quad \dfrac{E'}{E}=\left(\dfrac{p_{mh}}{R}+\dfrac{p_{m2}}{R_2}+p_{ml}\right)\Big/p_m$

$=\left(\dfrac{106{\cdot}62}{7{\cdot}1}+\dfrac{37}{2{\cdot}56}+16{\cdot}05\right)\Big/51{\cdot}3$

$$=(15{\cdot}0+14{\cdot}44+16{\cdot}05)/51{\cdot}3$$

$$=\frac{45{\cdot}49}{51{\cdot}3}=0{\cdot}885$$

$\therefore$ E′=0·885E, or 11·5 per cent. of the total possible theoretical work is lost due to incomplete expansion in H.P. and I.P. cylinders.

A reduction of the external load on an engine necessitates a corresponding reduction of the indicated work. This reduced power can be obtained in two ways : (*a*) by cut-off governing ; (*b*) by throttle governing.

In case (*a*) the initial pressure in the H.P. cylinder is maintained constant, and the supply of energy is regulated by a decrease of

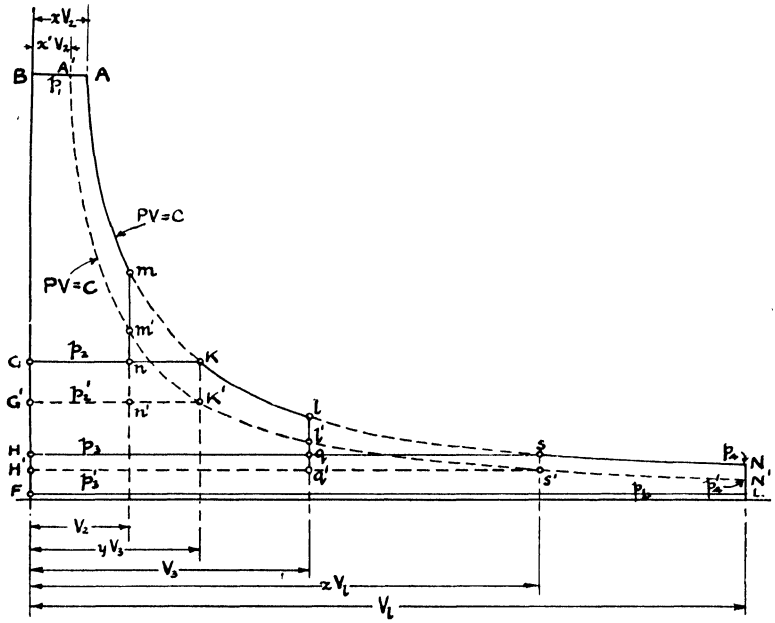



FIG. 127.

the cut-off fraction of the H.P. cylinder, the cut-off values in the I.P. and L.P. cylinders remaining the same.

In case (*b*) the cut-off values of all the cylinders are kept constant, and the initial pressure at the H.P. cylinder is reduced by throttling the steam.

**140. Cut-Off or Expansion Governing.**—The general effect of variation of the H.P. cut-off is shown on the theoretical indicator diagram (Fig. 127) for a triple expansion engine, having incomplete expansion in each cylinder. The H.P., I.P., and L.P. cylinder volumes are $V_2$, $V_3$ and $V_l$ respectively.

The full load diagram is B$\Lambda$*mn*K*lqs*NLF, when the H.P. cut-off fraction is $x$, or the initial H.P. volume is $xV_2$.

For a reduced load the cut-off is $x'$ or the initial steam volume is reduced to $x'V_2$.

The reduced load diagram shown by the dotted lines is BA′$m'n'$K′$l'q's'$N′LF.

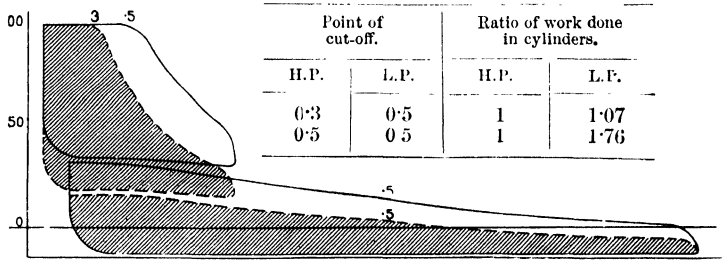
The cut-off volumes of the I.P. and L.P. cylinders are, respectively, $yV_3$ and $zV_4$.

The H.P. diagram is reduced from BA$mn$G on full load to BA′$m'n'$G′ on light load; the I.P. diagram is reduced from GK$lq$H to G′K′$l'q'$H′; the L.P. diagram is reduced from HSNLF to H′$s'$N′LF.

The relative proportions of the total power developed in the two cases are as given below—

| | *H.P.* | *I.P.* | *L.P.* |
|---|---|---|---|
| Full load . . . | 0·327 | 0·323 | 0·350 |
| Reduced load . . | 0·40 | 0·31 | 0·29 |

This case shows that the general effect of varying the H.P. cut-off is to increase the proportion of the total work of the H.P. on reduced




VARIABLE CUT-OFF IN H.P. CYLINDER.
CONSTANT „ L.P. „

| Point of cut-off. | | Ratio of work done in cylinders. | |
|---|---|---|---|
| H.P. | L.P. | H.P. | L.P. |
| 0·3 | 0·5 | 1 | 1·07 |
| 0·5 | 0 5 | 1 | 1·76 |

FIG. 128.

load, and to decrease the proportion done in the I.P. and L.P. cylinders. On a low load in the case of a non-condensing engine the L.P. diagram may be so reduced that the expansion curve may cross the atmospheric line and produce a negative work loop, a very undesirable result.

The foregoing conditions are reversed when the H.P. cut-off fraction is increased to increase the indicated power on overload. The proportion of total power developed in the L.P. is increased and the proportion in the H.P. is decreased.

The actual diagrams obtained from a two-stage compound on full and reduced load are shown in Fig. 128. For full load the H.P. cut-off is 0·5 and for reduced load 0·3. In each case the L.P. cut-off is constant at 0·5. The relative proportions of the total power developed are—

| | *H.P.* | *L.P.* |
|---|---|---|
| Full load . . . . . | 0·360 | 0·640 |
| Reduced load . . . . | 0·485 | 0·515 |

These figures serve to confirm the general conclusion drawn from the theoretical diagrams.

**141. Throttle Governing.**—The effect of maintaining a constant cut-off in the H.P., I.P. and L.P. cylinders and reducing the initial H.P. pressure by throttling is shown, for the triple-expansion engine, in Fig. 129.

The initial point of expansion is moved from A at $p$ to A″ at $p_1'$, the reduced initial pressure. The initial steam volume at cut-off remains $xV_2$.

The same reduced load curve is taken, for convenience of explanation, but any lower curve corresponding to a lower initial

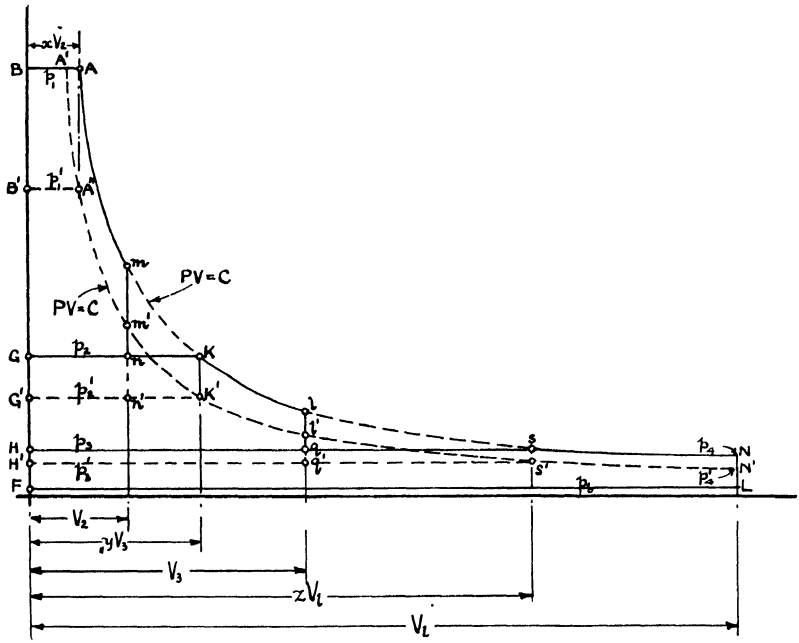



FIG. 129.

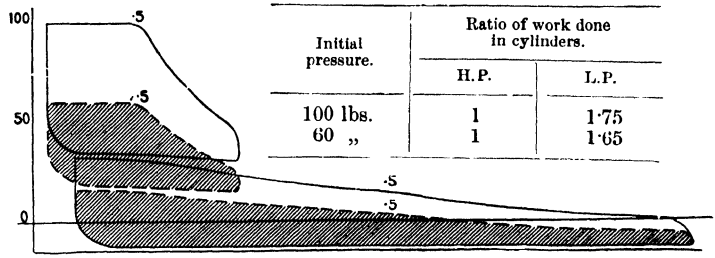
pressure than $P_1'$ might be used. As in the previous case, the full load diagram is BA*mn*K*lqs*NLF, while the reduced load diagram is B′A″*m*′*n*′K′*l*′*q*′*s*′N′LF. The H.P. diagram is reduced from BA*mn*G, on the full load, to B′A″*m*′*n*′G′, on the light load. The I.P. and L.P. full and light load diagrams are the same as for the previous case. The relative proportions of the total power in the two cases are as given below—

| | *H.P.* | *I.P.* | *L.P.* |
|---|---|---|---|
| Full load . . . . . | 0·327 | 0·323 | 0·350 |
| Reduced load . . . . | 0·365 | 0·321 | 0·314 |

The general effect again is to increase the proportion of the total power on reduced load in the H.P., and decrease it in the I.P. and

A point of some importance relating to economy arises in this case.

In each of the cases considered the terminal pressure $p_4'$ and volume $V_l$ on the reduced load are the same, so that the same weight of steam is exhausted from the engine on light load. The total power developed under throttling conditions is less than that with cut-off governing, so that the work done per lb. of steam is less.



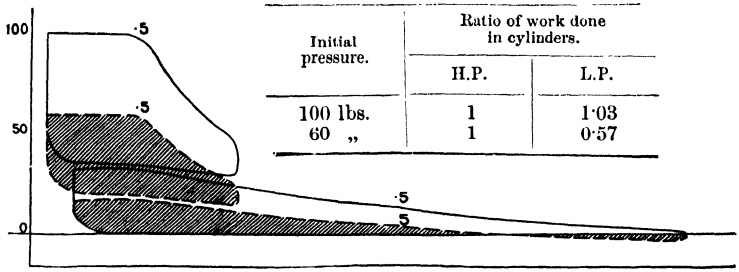

THROTTLING GOVERNING IN H.P. CYLINDER.
CONDENSING.

| Initial pressure. | Ratio of work done in cylinders. | |
|---|---|---|
| | H.P. | L.P. |
| 100 lbs. | 1 | 1·75 |
| 60 „ | 1 | 1·65 |

FIG. 130.

The general inference is that throttle governing is less economical than cut-off governing. The theoretical advantage of the cut-off system is not, however, obtained in the actual case, owing to the greater condensation with an early cut-off.

Actual diagrams from a two-stage condensing compound engine, running on full and reduced load under throttling conditions, are shown in Fig. 130, the initial pressure on reduced load being about half that for full load.




NON-CONDENSING. THROTTLING.

| Initial pressure. | Ratio of work done in cylinders. | |
|---|---|---|
| | H.P. | L.P. |
| 100 lbs. | 1 | 1·03 |
| 60 „ | 1 | 0·57 |

FIG. 131.

The relative proportions of the total power developed are—

| | *H.P.* | *L.P.* |
|---|---|---|
| Full load . . . . . | 0·364 | 0·636 |
| Reduced load . . . . . | 0·378 | 0·622 |

Similar diagrams are shown for a non-condensing two-stage compound engine in Fig. 131.

The power of the L.P. is so much reduced that, for part of the stroke, the piston is being dragged instead of exerting a driving force.

The relative proportions of the total power developed are—

| | *H.P.* | *L.P.* |
|---|---|---|
| Full load . . . . . | 0·49 | 0·51 |
| Reduced load . . . . . | 0·636 | 0·364 |

These two sets of figures confirm the general deduction from the theoretical diagrams.

Another set of diagrams taken from a four-crank triple-expansion engine is shown in Fig. 132.

This engine was one of a twin set fitted in a British cruiser, H.M.S. *Powerful*, before the days when the steam turbine replaced the reciprocating engine in the capital ships.

This four-crank triple had two L.P. cylinders, designated "forward" and "after" low-pressure, or F.L.P. and A.L.P.

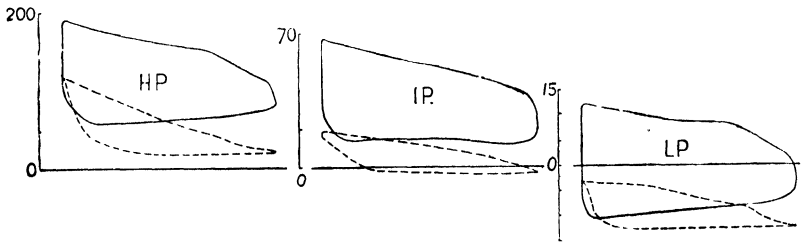



FIG. 132.

By the use of two L.P. cylinders instead of a single cylinder, the necessary volume was obtained without excessively large cylinder diameters.

The power records of a full and a reduced load trial (less than one-quarter full power) run under throttling conditions are as given below *—

| | | *Full Power* | *Reduced Power* |
|---|---|---|---|
| Boiler pressure lb./in.² gauge . . | | 205 | 190 |
| Vacuum (inches) . . . . . | | 26·25 | 26·75 |
| Revolutions per min. . . . . | | 116·95 | 69·18 |
| I.H.P. | H.P. . . . . | 4288 | 982 |
| | I.P. . . . . | 4233 | 729 |
| | F.L.P. . . . . | 2263 | 426 |
| | A.L.P. . . . . | 2238 | 426 |
| Total . . . . . | | 13,022 | 2563 |

The relative proportions of the total power, in each case, developed in the H.P., I.P., and L.P. cylinders are—

| | *H.P.* | *I.P.* | (2)*L.P.* |
|---|---|---|---|
| Full power . . . . . | 0·330 | 0·324 | 0·346 |
| Reduced power . . . . | 0·384 | 0·284 | 0·332 |

* See Paper by Sir A. J. Durstan, *I.N.A.*, April 1897.

The same effect of the throttle governing on the power distribution at reduced load is again apparent.

**142. Variation of Cut-off in I.P. and L.P. Cylinders.**—It will be obvious from the foregoing discussion that in the case of a multiple-expansion engine, if there is a fairly uniform distribution of the power among the cylinders at full load, at a reduced load the distribution, with either form of governing, will be more or less unequal.

In each of the cases considered it has been assumed that the I.P. and L.P. cut-off values remain constant.

The distribution resulting from this condition can, however, be modified by an adjustment either of I.P. or L.P. cut-off, or of both.

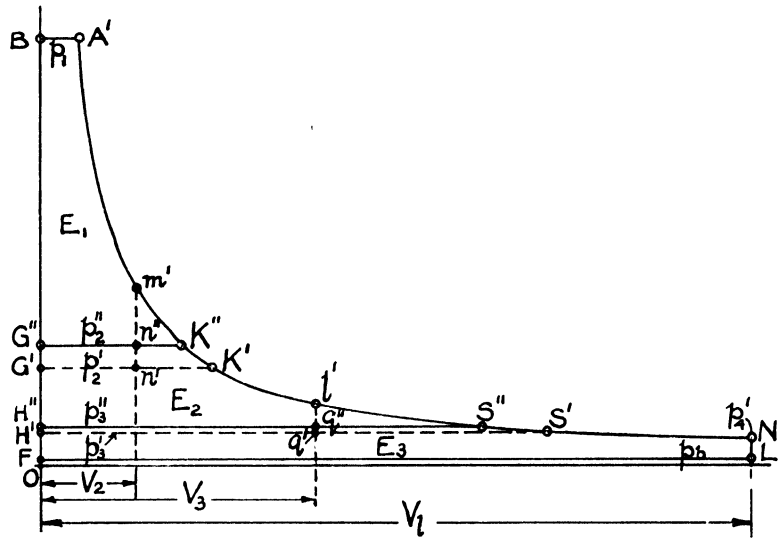



FIG. 133.

The total reduced power remains sensibly the same, since this is controlled by the initial conditions at the H.P. cylinder.

An inspection of any of the foregoing (theoretical) multi-stage engine diagrams will show that if the cut-off fraction in any cylinder is reduced, the initial pressure of that cylinder is raised, and with it the back pressure of the previous cylinder. The result is an increase of power in the cylinder having the earlier cut-off and a decrease in the cylinder before it.

The various methods of independently varying the cut-off in several cylinders of a multi-stage engine are discussed in Chapter IX on valves and valve gears.

The effect on the theoretical diagrams of varying the cut-off in the I.P. and L.P. cylinders of the triple-expansion engine is shown in Fig. 133. The diagram with the dotted receiver pressure lines is

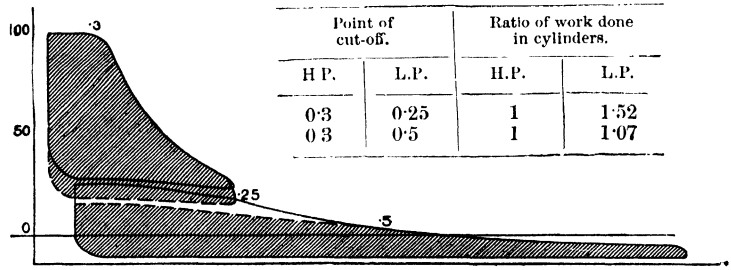
the reduced load diagram of Fig. 129. When the I.P. receiver pressure is raised from $p_2'$ to $p_2''$ the cut-off point is moved from K′ to K″. When the L.P. receiver pressure is raised from $p_3'$ to $p_3''$ the cut-off point is moved from S′ to S″.

The power areas are now H.P., $E_1=BA'm'n''G''$; I.P., $E_2=G''K''l'q''H''$; L.P., $E_3=H''S''NLF$. These areas are sensibly equal, or $E_1=E_2=E_3$, so that one-third of the theoretical power on the reduced load is developed in each cylinder.

The equalising effect of increase of L.P. cut-off in a compound engine is shown in Fig. 134, the power distribution being for—

| *L.P. Cut-off* | *H.P.* | *L.P.* |
|---|---|---|
| 0·25 | 0·398 | 0·602 |
| 0·50 | 0·482 | 0·518 |

**143. Referred Mean Pressure of a Multiple-Expansion Engine.**—If the total I.H.P. of a multiple-expansion engine were developed in




CONSTANT CUT-OFF IN H.P. CYLINDER.
VARIABLE „ L.P. „

| Point of cut-off. | | Ratio of work done in cylinders. | |
|---|---|---|---|
| H.P. | L.P. | H.P. | L.P. |
| 0·3 | 0·25 | 1 | 1·52 |
| 0·3 | 0·5 | 1 | 1·07 |

FIG. 134.

the last or L.P. cylinder, the mean effective pressure required is termed the mean pressure referred to the L.P. cylinder, or, more concisely, the referred mean pressure.

The actual referred mean pressure may be calculated either from the stage mean pressures and the cylinder ratios, or directly from the I.H.P. equation, when the total horse-power of the engine is used.

Let $p_E$=actual referred mean pressure.
$p_{eh}$=mean pressure in H.P.
$p_{e2}$= „ „ „ I.P.
$p_{el}$= „ „ „ L.P.

$$R=\frac{\text{L.P.}}{\text{H.P.}}=\text{total cylinder ratio.}$$

$$R_2=\frac{\text{L.P.}}{\text{I.P.}}=\text{I.P.} \quad \text{„} \quad \text{„}$$

$$\text{Thus} \quad p_E=\frac{p_{eh}}{R}+\frac{p_{e2}}{R_2}+p_e \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (14)$$

The more direct method of calculation is that from the total I.H.P.

Let $I$ = total horse-power.
$S$ = mean piston speed in ft./min.
$A_l$ = L.P. piston area in in.$^2$
$p_E$ = referred mean pressure in lb./in.$^2$

Then $$p_E = \frac{33000I}{SA_l} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (15)$$

Corresponding to the actual there is the theoretical referred mean pressure $p_M$ of the single-cylinder engine represented by the L.P. engine of the multiple set.

In order to simplify calculation it is usual to estimate the theoretical referred mean pressure for the L.P. cylinder without reference to clearance.

If $p_1$ = H.P. initial pressure
$p_b$ = L.P. back pressure
$r$ = total nominal ratio of expansion
$$= \frac{V_l}{xV_h}$$

where $V_l$ = L.P. cylinder volume
$V_h$ = H.P. ,, ,,
$x$ = H.P. cut-off fraction

then $$r = \frac{R}{x} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (16)$$

and the theoretical referred mean pressure is given by—

$$p_M = p_1\left(\frac{1+\log_e r}{r}\right) - p_b \quad . \quad . \quad . \quad . \quad (17)$$

An alternative expression for the theoretical mean effective pressure which takes account of the H.P. and L.P. clearance values can be obtained as follows. Intermediate cylinders do not enter into this calculation. As shown on the theoretical diagram of a two-stage compound (Fig. 135), let—

$c_h$ = fractional clearance of H.P. cylinder
$c_l$ = ,, ,, ,, L.P. ,,

Then initial volume $V_1 = xV_h$
final volume $V_2 = (V_l + c_l V_l) - c_h V_h$
$= (1+c_l)V_l - c_h V_h$
= volume of the substituted single-cylinder engine.

Hence $$r' = \frac{V_2}{V_1} = \frac{(1+c_l)V_l - c_h V_h}{xV_h}$$

or $$r' = \frac{(1+c_l)R - c_h}{x} \quad . \quad . \quad . \quad . \quad . \quad (18)$$

The fractional clearance of the substituted cylinder of volume $V_2$ is—

$$c'=\frac{KD}{KL}=\frac{c_hV_h}{(1+c_l)V_l-c_hV_h}=\frac{c_h}{(1+c)R-c_h} \quad . \quad . \quad (19)$$

The corresponding theoretical referred mean pressure is given by—

$$p_M'=p_1\left\{\frac{1}{r'}+\left(\frac{1}{r'}+c'\right)\log_e\frac{1+c'}{\frac{1}{r'}+c'}\right\}-p_b \quad . \quad . \quad (20)$$

*Example* 9.—A marine quadruple-expansion engine has cylinder diameters 26½ in., 37½ in., 59 in., 77 in., and stroke 57 in. The H.P. initial pressure is 230 lb./in.² abs. and the L.P. back pressure 3 lb./in.² abs. The H.P. cut-off is at 0·57 of the stroke.

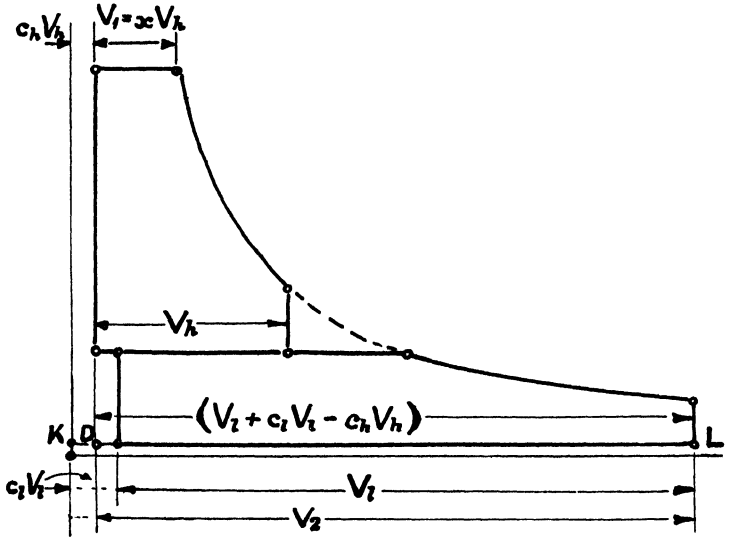



FIG. 135.

Calculate the ideal referred mean pressure: (*a*) when clearance is neglected; (*b*) when clearance of 15 per cent. is assumed for the H.P. and 8 per cent. for the L.P. cylinder.

(*a*) $R=\left(\frac{77}{26\cdot5}\right)^2=8\cdot44$. Hence $r=\frac{R}{x}=8\cdot44=14\cdot8$ $\quad x=0\cdot57$

Thus $p_M=p_1\left(\frac{1+\log_e r}{r}\right)-p_b$ $\quad p_1=230$, $p_b=3$

$$=230\left(\frac{1+\log_e 14\cdot8}{14\cdot8}\right)-3$$

$$=\frac{230\times3\cdot695}{14\cdot8}-3$$

$$=(57\cdot4-3)=\underline{54\cdot4}\text{ lb./in.}^2$$

(*b*) $c_h=0{\cdot}15$, $c_l=0{\cdot}08$ $\quad\therefore\ c'=\dfrac{c_h}{(1+c_l)\mathrm{R}-c_h}=\dfrac{0{\cdot}15}{1{\cdot}08\times 8{\cdot}44-0{\cdot}15}$

$$=\frac{0{\cdot}15}{8{\cdot}965}=\underline{0{\cdot}0168}$$

and $$r'=\frac{(1+c_l)\mathrm{R}-c_h}{x}=\frac{8{\cdot}965}{0{\cdot}57}=\underline{15{\cdot}7}$$

$$p_{\mathrm{M}}'=p_1\left\{\frac{1}{r'}+\left(\frac{1}{r'}+c'\right)\log_e\frac{1+c'}{\frac{1}{r'}+c'}\right\}-p_b \qquad \frac{1}{r'}=0{\cdot}0635$$

$$\frac{1}{r'}+c'=0{\cdot}0803$$

$$=230(0{\cdot}0635+0{\cdot}0803\ \log_e 12{\cdot}66)-3$$

$$=230(0{\cdot}0635+0{\cdot}2038)-3$$

$$=230\times 0{\cdot}2673-3=(61{\cdot}48-3)=58{\cdot}48\ \text{lb./in.}^2$$

**144. Diagram Factors of Multiple-Expansion Engines.**—As in the case of the simple type the actual referred mean effective pressure can be estimated from the ideal value by means of a diagram factor $f$ or F. Thus—

$$f=\frac{p_{\mathrm{E}}}{p_1\left(\frac{1+\log_e r}{r}\right)-p_b} \qquad (21)$$

or

$$\mathrm{F}=\frac{p_{\mathrm{E}}-p_b}{p_1\left(\frac{1+\log_e r}{r}\right)} \qquad (22)$$

where $r$=total expansion ratio and $p_b$=L.P. back pressure. As already indicated, the calculation of $p_{\mathrm{E}}$ is necessarily very tentative. Average values of the ordinary diagram factor ($f$) for condensing engines given in a paper * by F. Foster are reproduced below in Table III.

TABLE III

DIAGRAM FACTORS ($f$) FOR CONDENSING ENGINES

| *Type of Engine.* | *Conditions.* | *Diagram Factor ($f$).* |
|---|---|---|
| Compound . . | Piston and slide-valves ; saturated steam | 0·65–0·75 |
| ,, . . | Corliss or drop-valves ; saturated steam | 0·7–0·80 |
| ,, . . | Corliss or drop-valves ; superheated steam | 0·7–0·75 |
| ,, . . | Corliss, low-speed ; superheated steam | 0·8–0·90 |
| Triple land . . | Corliss ; superheated steam | 0·7–0·75 |
| ,, . . | Corliss low-speed ; superheated steam | 0·75–0·9 |
| Triple Marine . | Piston and slide-valves ; saturated steam | 0·63–0·7 |
| ,, . | Piston, high-speed ; saturated steam | 0·6–0·65 |

The following average values are also quoted for the actual mean effective pressures adopted by British makers. They are based on the normal full load, but can be increased to permit of considerable

* " Essential Principles of Engine Design," by F. Foster, *Proc. Manchester Association of Engineers*, 1915.

overload on the engine. The overload capacity is dependent principally on the value of the maximum cut-off at normal full load. Thus the valve gear of the typical compound condensing Corliss or drop-valve engine, working with 160 lb./in.$^2$ boiler pressure can be adjusted to give a maximum of 45 lb./in.$^2$ mean effective pressure, which is equivalent to an overload of about 50 per cent. At lower boiler pressures the percentage is reduced.

| | Mean Effective Pressures in lb./in.$^2$ | | | | | | |
|---|---|---|---|---|---|---|---|
| Boiler pressure . . | 80 | 100 | 120 | 140 | 160 | 180 | 200 |
| Compound condensing . | 25 | 27 | 28 | 29 | 30 | 31 | 32 |
| „ non-condensing | 26 | 30 | 34 | 37 | 40 | — | — |
| Triple condensing . . . | — | — | — | 27 | 29 | 30 | 30·5 |
| Uniflow condensing . . | 30 | 33 | 35 | 37 | 38 | 39 | 40 |
| „ non-condensing (about) . . | 26 | 30 | 34 | 37 | 40 | — | — |

If the alternative method of calculation for the mean effective referred pressure, which allows for clearance, is used, the equation for the multiple-expansion case becomes—

$$p_E = f' p_M' = f'\left[p_1\left\{\frac{1}{r'}+\left(\frac{1}{r'}+c'\right)\log_e\frac{1+c'}{\frac{1}{r'}+c'}\right\}-p_b\right] \quad . \quad (23)$$

$r'$ and $c'$ being calculated for the assumed case from Equations 18 and 19.

*Example* 10.—In the case of the quadruple-expansion engine (Example 9) the total I.H.P. developed at a mean piston speed of 760 ft./min. was 4159. The mean effective pressures from the indicator cards were: H.P., 75·88; 1st I.P., 41·97; 2nd I.P., 16·82; and L.P., 9·94 lb./in.$^2$ Calculate the actual referred mean pressure and deduce the diagram factors $f$ and $f'$.

Referred mean pressure, from cylinder ratios—

$$\frac{\text{L.P.}}{\text{H.P.}}=\text{R}=8{\cdot}44\,; \quad \frac{\text{L.P.}}{\text{1st I.P.}}=\text{R}'=\left(\frac{77}{37{\cdot}5}\right)^2=4{\cdot}215\,;$$

$$\frac{\text{L.P.}}{\text{2nd I.P.}}=\text{R}''=\left(\frac{77}{59}\right)^2=1{\cdot}7$$

$$p_{eh}=75{\cdot}88, \quad p_{e2}=41{\cdot}97, \quad p_{e3}=16{\cdot}82, \quad p_{el}=9{\cdot}94 \text{ lb./in.}^2$$

$$\therefore\ p_E=\frac{p_{eh}}{\text{R}}+\frac{p_{e2}}{\text{R}'}+\frac{p_{e3}}{\text{R}''}+p_{el}$$

$$=\frac{75{\cdot}88}{8{\cdot}44}+\frac{41{\cdot}97}{4{\cdot}215}+\frac{16{\cdot}82}{1{\cdot}7}+9{\cdot}94$$

$$=8{\cdot}99+9{\cdot}95+9{\cdot}88+9{\cdot}94=38{\cdot}76 \text{ lb./in.}^2$$

Shorter method, from I.H.P.— $I=4159$, $S=760$ ft./min., $A_l=4660$ in.$^2$

$$p_E=\frac{33000I}{SA_l}$$

$$=\frac{33000\times4159}{760\times4660}=38{\cdot}76 \text{ lb./in.}^2$$

Diagram factor (Case $a$), $f=\frac{p_E}{p_M}=\frac{38{\cdot}76}{54{\cdot}4}=0{\cdot}71$ $\quad p_M=54{\cdot}4$

(Case $b$), $f'=\frac{p_E}{p_M'}=\frac{38{\cdot}76}{58{\cdot}48}=0{\cdot}66$ $\quad p_M'=58{\cdot}48$

**145. Calculation of Cylinder Dimensions of Multiple-Expansion Engines.**—In the first place, the diameter of the L.P. cylinder, for the development of the total power I at the stated piston speed S, has to be obtained by determining the probable referred mean pressure $p_E$ (or $p_E'$), then the L.P. cylinder area $A_l$ from the I.H.P. equation is given by—

$$A_l=\frac{33000I}{p_ES} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (24)$$

and $$D_l=\sqrt{\frac{4A_l}{\pi}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (25)$$

or the value can be read directly from a table of areas of circles.

When the total cylinder ratio R is fixed the diameter $D_1$ of the H.P. cylinder is obtained from—

$$R=\left(\frac{D_l}{D_1}\right)^2$$

or $$D_1=\frac{D_l}{\sqrt{R}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (26)$$

Similarly, when the intermediate pressure cylinder ratios have been decided, the corresponding diameters are obtained by dividing the L.P. diameter by the root of the corresponding cylinder ratio. Thus, for a triple, if $\frac{\text{L.P.}}{\text{I.P.}}=\left(\frac{D_l}{D_2}\right)^2=R_2$, the I.P. diameter is—

$$D_2=\frac{D_l}{\sqrt{R_2}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (27)$$

Values of cylinder ratios adopted for various British types of multiple-expansion engines are given in Tables IV and V. As is customary in practice the boiler pressures are stated in lb./in.$^2$ gauge. The figures for land (horizontal) engines are reproduced from F. Foster's paper already cited. They refer to engines with cut-off governing. The probable absolute initial pressures can be obtained by adding 10 to the gauge pressures.

## TABLE IV

### Compound Non-Condensing

| Boiler Pressure | H.P Volume | L.P. Volume |
|---|---|---|
| 90–100 lb./in.² g. . . | 1 | 2·25–2·5 |
| 120 ,, . . | 1 | 2·50–2·7 |
| 140 ,, . . | 1 | 2·8–3·0 |
| 160 ,, . . | 1 | 3·0–3·2 |

### Compound Condensing

| Boiler Pressure | H.P. Volume | L.P. Volume |
|---|---|---|
| 90–100 lb./in.² g. . . | 1 | 3·0–3·3 |
| 120 ,, . . | 1 | 3·3–3·5 |
| 140 ,, . . | 1 | 3·7–3·8 |
| 160 ,, . . | 1 | 3·8–4·0 |

### Horizontal Triple Condensing

| Boiler Pressure | H.P. Volume | I.P. Volume | L.P. Volume |
|---|---|---|---|
| 140 lb. in.² g. . . | 1 | 2·3–2·36 | 5·2–5·6 |
| 160 ,, . . | 1 | 2·36–2·45 | 5·6–6·0 |
| 180 ,, . . | 1 | 2·43–2·50 | 5·9–6·3 |
| 200 ,, . . | 1 | 2·45–2·67 | 6·0–6·6 |

## TABLE V

### Marine Two-Stage Compound Condensing (Mercantile Vessels)

| Boiler Pressure | H.P. Volume | L.P. Volume |
|---|---|---|
| 100 to 140 lb./in.² g. . | 1 | 4–5 |

### Marine Triples Condensing (Mercantile Vessels)

| Boiler Pressure | H.P. Volume | I.P. Volume | L.P. Volume |
|---|---|---|---|
| 160 lb./in.² g. . . | 1 | 2·3 | 6·5 |
| 180 ,, . . | 1 | 2·7 | 7·5 |

TABLE V.—*continued.*

MARINE TRIPLES CONDENSING (CROSS-CHANNEL STEAMERS)

| Boiler Pressure | H.P. Volume | I.P. Volume | L.P. Volume |
|---|---|---|---|
| 180–200 lb./in.² g. . | 1 | 2·55 | 6·75 |

MARINE QUADRUPLE CONDENSING

| Boiler Pressure | H.P. Volume | 1st I.P. Volume | 2nd I.P. Volume | L.P. Volume |
|---|---|---|---|---|
| 200 lb./in.² g. . . | 1 | 1·7 | 3·8 | 8 |
| 220 ,, . . | 1 | 2·05 | 4·3 | 9 |

The ratios given above for marine engines are deduced from data in *Kempe's Engineer's Yearbook.*

The student should refer to Seaton's standard work on marine engineering, in which tables of data relating to the dimensions and performances of many marine engines of various types are given.

As regards the locomotive, there has been for many years much difference of opinion on the subject of compounding, and there is a considerable variation in the cylinder ratios adopted. The ratio of L.P. and H.P. may vary from 1 to 1·69 to 1 to 3 for engines working at 200 lb./in.² In recent high-pressure design, with pressures about 850 to 900 lb./in.², a ratio of about 1 to 6 has been used, with steam at 700° Fahr.

*Example* 11.—Determine suitable cylinder diameters and stroke for a compound horizontal slide-valve engine, with automatic expansion gear, to develop a maximum of 100 I.H.P., at a piston speed of 400 ft./min., the boiler pressure being 100 lb./in.² g. and the vacuum 26 in. Assume a cylinder ratio of 3·0, a H.P. cut-off at 0·5, and a diagram factor $f=0{\cdot}7$. Stroke $=1{\cdot}2\times$ L.P. diameter.

For the cylinder ratio and cut-off chosen, the nominal number of expansions is—

$$r=\frac{R}{x}=\frac{3{\cdot}0}{0{\cdot}5}=6 \qquad \begin{matrix} R=3{\cdot}0 \\ x=0{\cdot}5 \end{matrix}$$

The absolute admission pressure may be taken as boiler pressure $+10$ or 110 lb./in.² abs. The vacuum pressure is $p_v=2$ lb./in.² abs., but the back pressure will be greater than this, say 3 lb./in.² abs.

$$p_m=p_1\left(\frac{1+\log_e r}{r}\right)-p_b \qquad \begin{matrix} p_1=110 \\ p_b=3 \end{matrix}$$

$$=110\left(\frac{1+2{\cdot}3\log 6}{6}\right)-3$$

$$=\frac{110\times 2{\cdot}8}{6}-3=(51{\cdot}3-3)=48{\cdot}3 \text{ lb./in.}^2$$

$$p_e = f\,p_m = 0{\cdot}7 \times 48{\cdot}3 = 33{\cdot}81 \qquad f = 0{\cdot}7$$

$$\text{L.P. area, } A_l = \frac{33000\,I}{p_E S} \qquad S = 400,\ I = 100$$

$$= \frac{33000 \times 100}{33{\cdot}81 \times 400} = 245 \text{ in.}^2$$

and L.P. diameter, $D_l = 17\frac{3}{4}$ in.

$$\text{H.P. diameter } D_h = \frac{D_l}{\sqrt{R}} = \frac{17{\cdot}75}{\sqrt{3}} = 10 \text{ in.}$$

$$\text{Stroke, } L = 1{\cdot}2 \times D_l = 1{\cdot}2 \times 17{\cdot}75 = 21 \text{ in.}$$

$$\text{Provisional size of engine, } \frac{10 \text{ in.} + 17\frac{3}{4} \text{ in.}}{21 \text{ in. str.}}$$

$$\text{Rotational speed, } N = \frac{S}{2L} = \frac{400 \times 12}{2 \times 21} = 115 \text{ r.p.m.}$$

The standard size of this type of engine, listed in an engine maker's catalogue for the specified maximum load of 100 I.H.P., is $\frac{10 \text{ in.} + 17\frac{1}{2} \text{ in.}}{20 \text{ in.}}$; boiler pressure 100 lb./in.$^2$ g. and speed 120 r.p.m.

The normal or economic load at which the engine would run would be about three-quarters of the maximum, or 75 I.H.P., or taking a mechanical efficiency of, say, 0·92, the economical brake horse-power would be about 69.

*Example* 12.—A triple-expansion marine engine is to develop 2000 I.H.P. at a piston speed of 700 ft./min. with initial pressure 195 lb./in.$^2$; back pressure 3 lb./in.$^2$ abs. Full load cut-off in H.P. cylinder 0·546, and cylinder ratios 7·1 and 2·56; stroke 0·75×L.P. diameter.

Determine the necessary cylinder dimensions assuming an average diagram factor $f = 0{\cdot}66$.

This is the case for which the relative values of the theoretical piston loads and indicated powers have been worked out in Example 7.

$$\text{Total expansion ratio } r = \frac{R}{x} \qquad R = 7{\cdot}1,\ x = 0{\cdot}546,\ p_1 = 195,\ p_b = 3,\ f = 0{\cdot}66$$

$$= \frac{7{\cdot}1}{0{\cdot}546} = 13$$

$$p_M = p_1\left(\frac{1 + \log_e r}{r}\right) - p_b$$

$$= 195\left(\frac{1 + 2{\cdot}3 \log 13}{13}\right) - 3$$

$$= (54{\cdot}3 - 3) = 51{\cdot}3 \text{ lb./in.}^2$$

$$p_E = f p_M = 0{\cdot}66 \times 51{\cdot}3 = 33{\cdot}86 \text{ lb./in.}^2$$

L.P. diameter, $A_l = \dfrac{33000I}{Sp_E}$ $\quad I=2000$, $S=700$

$$= \frac{33000 \times 2000}{700 \times 33{\cdot}86} = 2780 \text{ in.}^2$$

$D_l = 59\frac{1}{2}$ in., or, say, the even figure of 60 in.

H.P. diameter $D_h = \dfrac{D_l}{\sqrt{R}} = \dfrac{60}{\sqrt{7{\cdot}1}} = \dfrac{60}{2{\cdot}67} = 22{\cdot}6$, say $22\frac{1}{2}$ in. $\quad R=7{\cdot}1$

I.P. diameter $D_i = \dfrac{D_l}{\sqrt{R_2}} = \dfrac{60}{\sqrt{2{\cdot}56}} = \dfrac{60}{1{\cdot}6} = 37{\cdot}5$ in. $\quad R_2=2{\cdot}56$

Stroke $L = 0{\cdot}75\ D_l = 0{\cdot}75 \times 60 = 45$ in. $= 3$ ft. 9 in.

The provisional sizes to the nearest half inch may therefore be taken as—

$$\frac{22\frac{1}{2} \text{ in.} + 37\frac{1}{2} \text{ in.} + 60 \text{ in.}}{45 \text{ in. str.}}$$

The rotational speed here is $N = \dfrac{S}{2L} = \dfrac{700}{7{\cdot}5} = 93{\cdot}5$ r.p.m., which is a common value for this class of engine.

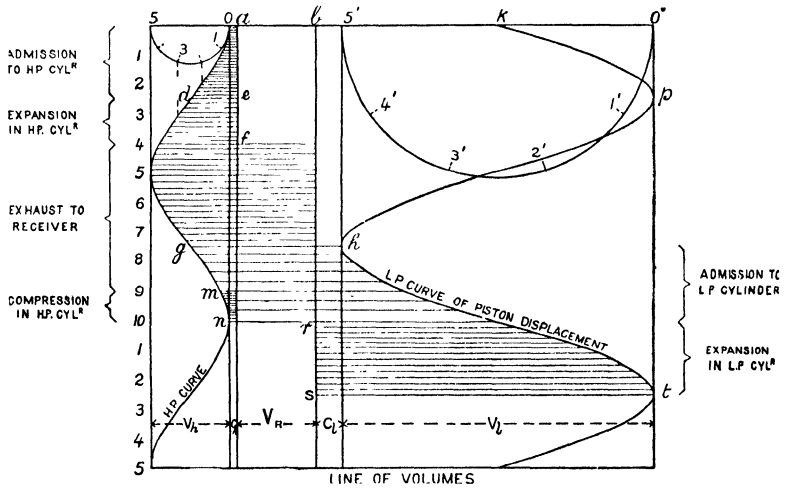



FIG. 136.

**146. Diagram of Relative Piston Displacement in Compound Engines.**—Having given the ratios of cylinders and clearance and receiver volumes for a given compound engine, it is possible to follow the steam through the engine, and to construct diagrams representing the nature of the changes of volume and pressure between the points of entrance to and exit from the cylinder.

In Fig. 136 horizontal lines are lines of volume, and vertical lines are subdivided into portions of a revolution. Thus, starting at *a* on

the top line, let $a$O=volume of high-pressure clearance ($c_h$); O5 =volume of high-pressure piston displacement ($V_h$); $ab$=volume of receiver ($V_R$); $b5'$=volume of low-pressure clearance ($c_l$); and 5′O′=volume of low-pressure piston displacement ($V_l$).

On the lines O5 and O′5′ draw semicircles representing a half-revolution of the crank-pin, and divide it into any number of equal parts—say five, as shown. On the vertical line to the left of the figure set off ten equal spaces representing parts of a revolution. The diagram is completed for one and a half revolutions. The cranks are at right angles, the H.P. piston is at the beginning of its stroke O, and the L.P. piston is at half-stroke K. A curve is now drawn for each cylinder, called the "curve of piston displacement," through the points of intersection of the horizontals from the divisions on the line of revolution, and of the verticals from the corresponding numbers on the crank-pin circles. This curve gives, by horizontal measurement to it, as shown by the shade lines, the volume of steam in the H.P. cylinder, including clearance, for any position of the piston, before exhaust. After exhaust, it gives the volume of the steam on the exhaust side of the piston, and including the receiver volume; and finally, when both cylinders are in communication, the horizontal distance between the lines gives the volume of the steam for any relative position of the pistons, the displacement curves having been drawn so that the cranks have the required relative position with one another.

The varying volume of the steam in its passage through the engine may now be followed.

At the beginning of the stroke the H.P. clearance volume $a$O is full of steam at the initial pressure, and admission continues at this pressure to cut-off, where the volume in the cylinder and clearance is $de$. Expansion then takes place until the exhaust port opens near the end of the stroke at $f$, and the steam passes from the H.P. cylinder to the L.P. receiver.

The exhaust side of the H.P. piston is in communication with this receiver, as indicated by the horizontal lines, between $f$ and $h$, where the L.P. admission valve opens. The H.P. piston is in communication with the receiver and L.P. cylinder till the H.P. exhaust port closes at $m$, the total volume of steam between the two pistons being shown by the horizontal intercept between the two displacement curves. Compression continues in the H.P. cylinder from $m$ to $n$, the end of the stroke; admission takes place to the L.P. cylinder between $h$ and $r$, the L.P. cut-off, and then the steam expands to the final volume of the L.P. cylinder and clearance $st$.

The corresponding theoretical H.P. and L.P. indicator diagrams are shown below the respective displacement curves in Fig. 137. The law of expansion and compression is taken as PV=C. The combined diagram with continuous expansion curve is obtained by superposing the H.P. diagram on the L.P. diagram on the right side of Fig. 137.

In the following equations the subscripts of volume and pressure refer to the corresponding numbers of the points on the theoretical indicator diagrams.

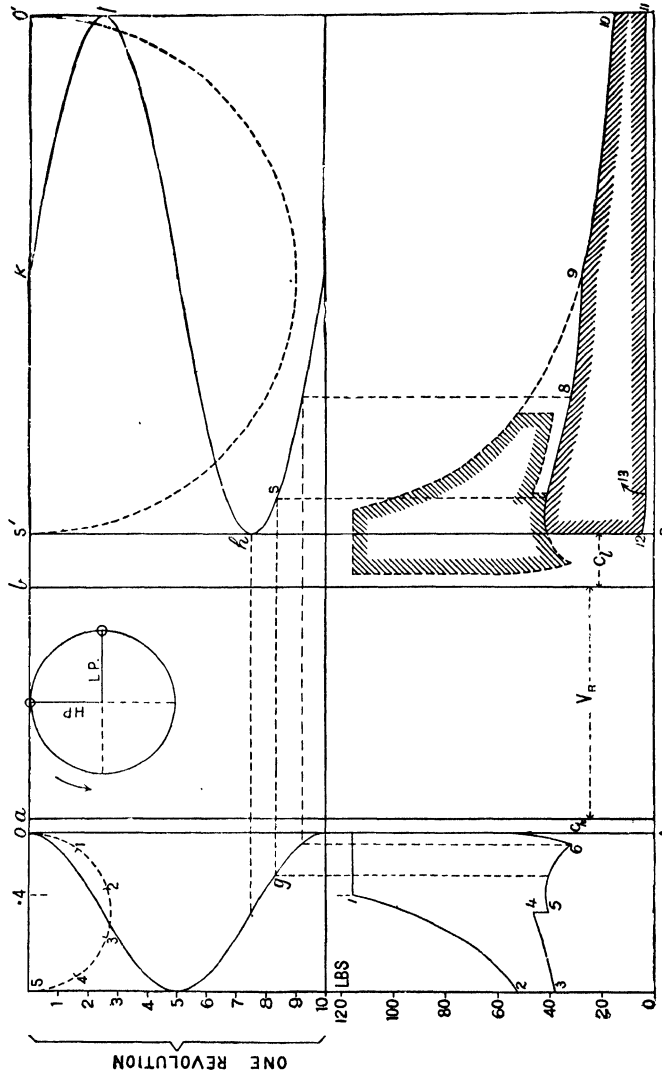



FIG. 137.

For example, $V_3$=volume of steam at the point 3, measured from the beginning of the stroke, that is from the vertical line AO, to the left, on the H.P. diagram, and $V_{10}$=volume at point 10 measured from the vertical line through B, to the right, on the L.P. diagram.

With continuous hyperbolic expansion throughout the two cylinders, Fig. 137—

$$p_1(V_1+c_h)=p_{10}(V_{10}+c_l) \quad . \quad . \quad . \quad . \quad . \quad (28)$$

and the L.P. terminal pressure $p_{10}$ is calculable.

The point of cut-off (9) in the L.P. cylinder is known, so that

$$p_9(V_9+c_l)=p_{10}(V_{10}+c_l) \quad . \quad . \quad . \quad . \quad . \quad (29)$$

This gives the L.P. cut-off pressure $p_9$, which is also the receiver pressure at this point.

The pressures at all other points can now be obtained. In the case of the H.P. cylinder the release pressure $p_2$ is given by—

$$p_1(V_1+c_h)=p_2(V_2+c_h) \quad . \quad . \quad . \quad . \quad . \quad (30)$$

At 2 the steam exhausts to the receiver of volume $V_R$ filled with steam at $p_9$. During this exhaust, until the L.P. admission valve opens at 4, the pressure rises from $p_3$ to $p_4$. The initial compression pressure $p_3$ in receiver is given by—

$$(p_2V_2+c_h)+p_9V_R=p_3(V_3+c_h+V_R) \quad . \quad . \quad . \quad (31)$$

and the pressure $p_4$ by—

$$p_3(V_3+c_h)+V_R)=p_4(V_4+c_h+V_R) \quad . \quad . \quad . \quad (32)$$

When the L.P. admission valve opens at 4 there is a drop of pressure at constant volume $V_4$ from $p_4$ to $p_5$ at 5. The L.P. clearance is filled with exhaust steam at $P_{12}$, the pressure at point 12, so that the volume occupied by the steam at 5 is $(V_5+c_h+V_R+c_l)$; also, since the L.P. back pressure $p_{11}=p_{13}$ is fixed, $p_{12}$ is given by—

$$p_{12}c_l=p_{11}(V_{13}+c_l) \quad . \quad . \quad . \quad . \quad . \quad . \quad (33)$$

and $p_5$ by—

$$p_4(V_4+c_h+V_R)+p_{12}c_l=p_5(V_5+c_h+V_R+c_l) \quad . \quad . \quad (34)$$

As the L.P. piston moves out, the pressure in H.P. cylinder, the receiver, and L.P. cylinder gradually falls until, at the point of compression 6 in the H.P. cylinder, it has a value $p_6$, which is given by—

$$p_5(V_5+c_h+V_R+c_l)=p_6(V_6+c_h+V_R+c_l+V_8) \quad . \quad (35)$$

where $V_8$ is the volume of L.P. piston displacement, when the H.P. exhaust closes. Pressures $p_9$ and $p_{10}$ at cut-off and release in the L.P. cylinder are already known.

The H.P. and L.P. compression curves are obtained by drawing rectangular hyperbolas through the compression points 6 and 13, and the continuous expansion curve by drawing a rectangular hyperbola through the H.P. cut-off point. Intermediate points are obtained on curve 3–4 by substituting one or more intermediate volumes for $V_4$ in Equation 32.

Similarly, points on the curve 5–6 are obtained by substituting one or more intermediate volumes for $V_6$ in Equation 35.

It should be noted that the H.P. and L.P. theoretical diagrams are drawn to common scales of pressure and volume.

The foregoing method can be applied to triple and quadruple-expansion engines by taking successive pairs of cylinders.

The rectangular hyperbolas for any case can readily be plotted from the values of pressure given by $PV=C$. They can, however, if desired, be obtained by the purely graphical method given below.

**147. Method of Drawing a Hyperbolic Curve on an Indicator Diagram.**—Let E (Fig. 138) be the compression point of the diagram through which a hyperbolic compression curve is to be drawn. Draw the lines of zero volume OY and zero pressure OX and QR the terminal line of the stroke, so that $OQ=V_c$ the clearance volume.

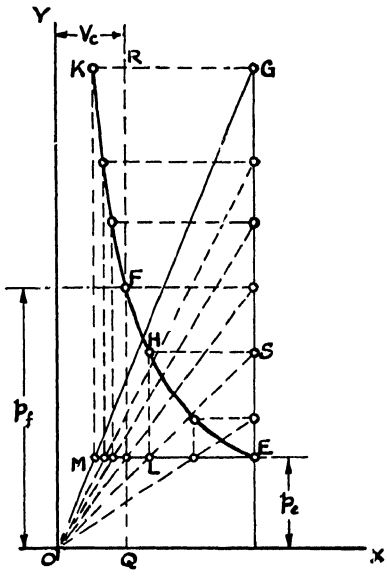


FIG. 138.

Through E draw a vertical EG of any convenient length and join O and G. Draw EM cutting OG in M. To find any point H on the compression curve EF, take a point S on EG and draw the dotted diagonal line SO to cut EM in L. Draw the horizontal SH through S and the vertical LH through L and the point of intersection H is the corresponding point on the curve. The other points shown are obtained in a similar manner, giving the curve KE, which intersects the terminal line of the stroke QR at the point of final compression F, which defines the final pressure $p_f$.

Let A (Fig. 139) be the cut-off point through which a hyperbolic expansion curve is to be drawn.

Draw OY and OX; set off $V_c$ and $V_o$, and draw the terminal line of the stroke CD. Take any convenient point C on CD and join O and C. Through the cut-off point A draw the vertical AZ to cut OC in Z and draw the horizontal ZN cutting CD in N. Take any point T on CD and draw TO to cut ZN in W. Draw TU horizontally through T and WU vertically through W and the point of intersection U is a point on the expansion curve. Other points shown are obtained in the same way. The intersection ZN with CD gives the release point N, which defines the final pressure $p_n$.

**148. Combined Indicator Diagram of a Multiple-Expansion Engine.**—When the indicator diagrams, obtained from the stage cylinders of a multiple-expansion engine, are replotted to common scales of

pressure and volume, a combined diagram is obtained. This is useful in showing the actual relative work areas to a uniform scale, and indicating the value of the energy losses peculiar to the type of engine considered.

In Fig. 140 the three stage diagrams, which are the means of top and bottom indicator cards of a marine triple-expansion engine, are shown at (*a*), (*b*), and (*c*), and the combined indicator diagram is shown at (*d*).

In each case the lines of zero volume and zero pressure are drawn, and the diagram base line is divided into 10 or more equal parts and perpendiculars are erected, as shown on the L.P. diagram (*c*).

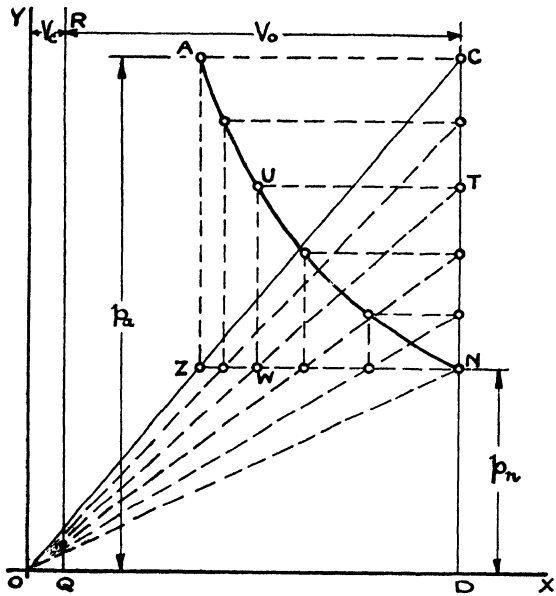



FIG. 139.

To obtain the combined diagram (*d*) a length QM is set off to a convenient scale to represent the volume of the L.P. cylinder $V_l$, and this is divided into the same number of parts as the base of (*c*). To the same scale the clearance volume $V_{cl}=c_l V_l$ is set off as QO, and the line of zero volume OY is drawn, and on it is set off a scale of pressures to any convenient unit. The pressures at 0, 1, 2, etc. in (*c*) are measured and set off at the corresponding divisions on QM in (*d*), and the enlarged L.P. diagram is drawn in. Next, a line $O_1Q_1M_1$ is drawn above the L.P. diagram in (*d*) with $O_1Q_1=c_iV_i=V_{ci}$, the I.P. clearance volume, and $Q_1M_1=V_i$, the I.P. stroke volume. The base of (*b*) and $Q_1M_1$ are divided into ten parts, the pressures scaled and transferred from (*b*) to (*d*), and the I.P. enlarged diagram is drawn in.

Similarly a line $O_2Q_2M_2$ is drawn above the I.P. diagram in (*d*) with $O_2Q_2=c_hV_h=V_{ch}$, the H.P. clearance volume, and $Q_2M_2=V_h$,

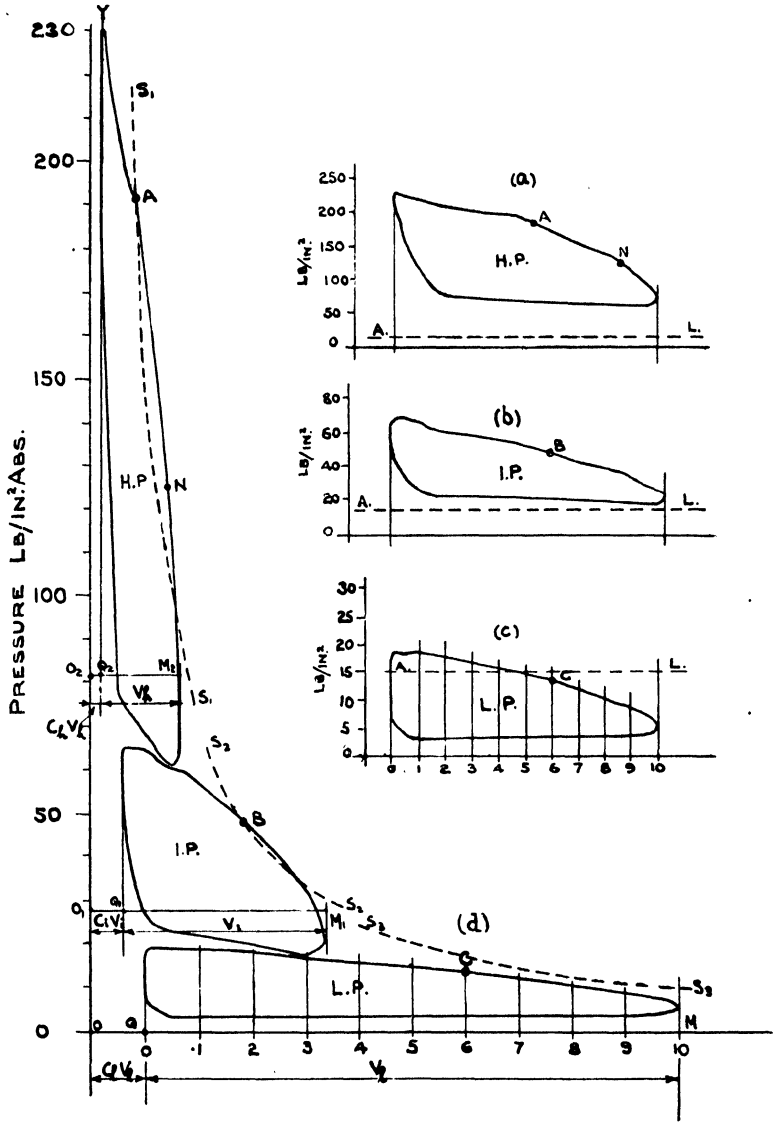



FIG. 140.

the H.P. stroke volume. The base of (*a*) and $Q_2M_2$ are then divided into ten equal parts, the pressures measured and transferred, and the enlarged H.P. diagram is drawn in.

The saturation curves $S_1S_1$, $S_2S_2$, $S_3S_3$ for the H.P., I.P., and L.P. cards can then be drawn in accordance with the methods explained in Chapter VI.

In general, a continuous saturation curve will not be obtained on the combined diagram (*d*), as the weight of cushion steam varies in the different cylinders.

The logarithmic diagram of the H.P. cylinder of this set has already been drawn (Fig. 116), and indicates a release about 0·857 per cent. at N. The mean cut-off is at A about 0·55 per cent. The saturation curve $S_1S_1$ in (*d*) falls inside the expansion curve AN and shows that the steam remains superheated during expansion, and is still superheated at release. It appears to have a slight superheat in the I.P. cylinder, as shown by the close approach of $S_2S_2$ to the expansion curve, but it is definitely wet when it enters the L.P. cylinder, as shown by the curve $S_3S_3$.

These saturation curves show the expansion of the cylinder feed plus clearance steam in each case. It is more satisfactory to eliminate the cushion steam and obtain a single saturation curve for the whole engine, as shown at (*d*), Fig. 141. The corrected diagrams are shown dotted at (*a*), (*b*), and (*c*). The method of eliminating the clearance steam is illustrated, for the L.P. cylinder, on diagram (*c*).

The compression point E is located and the compression curve EG is drawn (without sensible error) as a rectangular hyperbola, up to the admission pressure limit.

Let *b* be any point on the expansion curve at a given pressure level, *a* the corresponding point of the admission line, and *c* the point of the continued compression curve.

Shift the line *cab*, to the left, till *c* coincides with the co-ordinate axis OY, and mark the positions *b'* and *a'* of the points *a* and *b*. Repeat the process all round the diagram, and obtain the dotted and corrected diagram as shown. Divide the base line of the diagram into 10 parts, as in the previous case.

In Fig. 141 (*d*) draw the co-ordinate axes OM and OY and set off $OM=V_l$, the L.P. stroke volume. Similarly set off $V_i$ and $V_h$, the H.P. and I.P. volume values, from OY, and draw the boundary verticals for the I.P. and H.P. cylinders. Then proceed to transfer the diagrams from the one base to the other, as in the previous cases. Finally, using only the cylinder feed ($w_f$), draw in the single saturation curve SS.

**149. The Use of Superheated Steam in Multiple-Expansion Engines.**—The effect of initially superheated steam on the economy and efficiency of the single-cylinder contra-flow type of engine has already been discussed in Chapter VI. Highly superheated steam was tried in multiple-expansion engines many years ago, and was discarded on account of the difficulties attending the internal lubrication and the destructive effect of the high temperature on the gland packings.

The introduction of mineral oils as lubricants and the adoption of metallic packings remedied these drawbacks, and for some

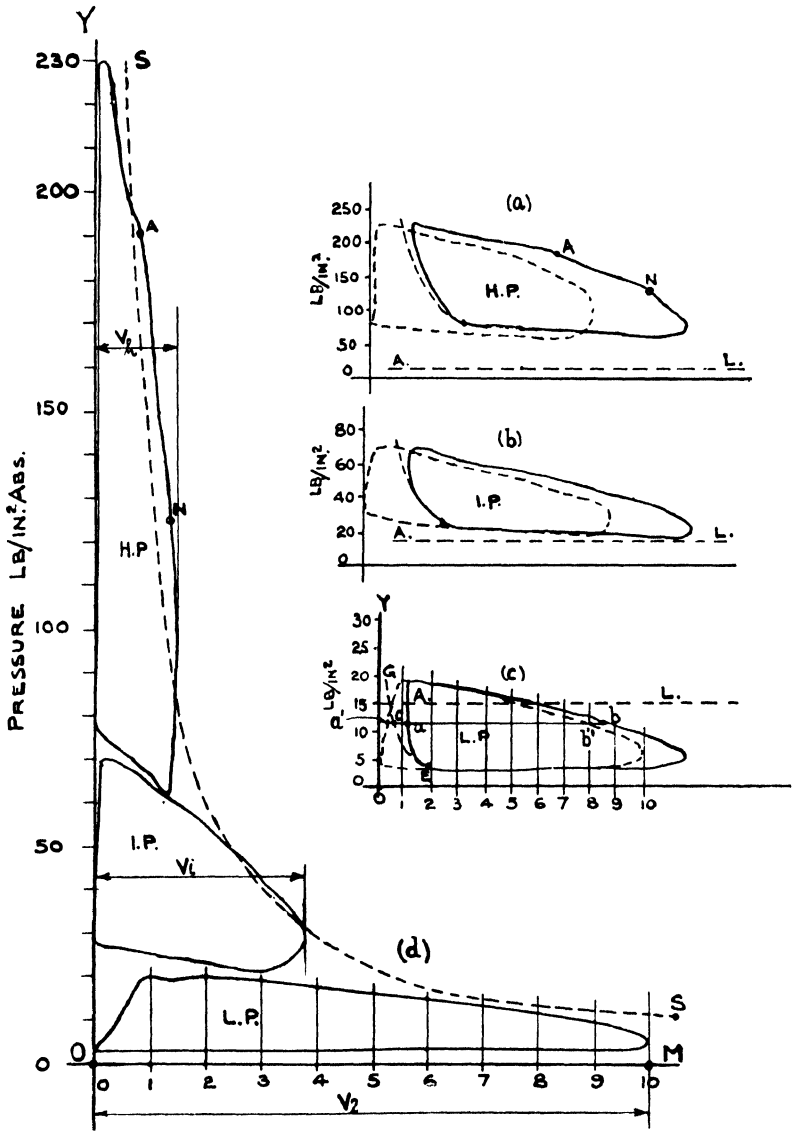



FIG. 141.

twenty years superheated steam with a gradually increasing range of superheat has been used in multiple-expansion engines. In steam turbine practice its use is universal.

In some up-to-date designs of multiple-expansion engines special H.P. castings are provided to stand temperatures from 700° Fahr. to 750° Fahr.; but, in general, at present, the initial temperature is limited to a maximum of 600° Fahr.

By the use of this high temperature the steam can usually be maintained in a superheated condition in the H.P. cylinder and may be slightly superheated on admission to the I.P. cylinder. The

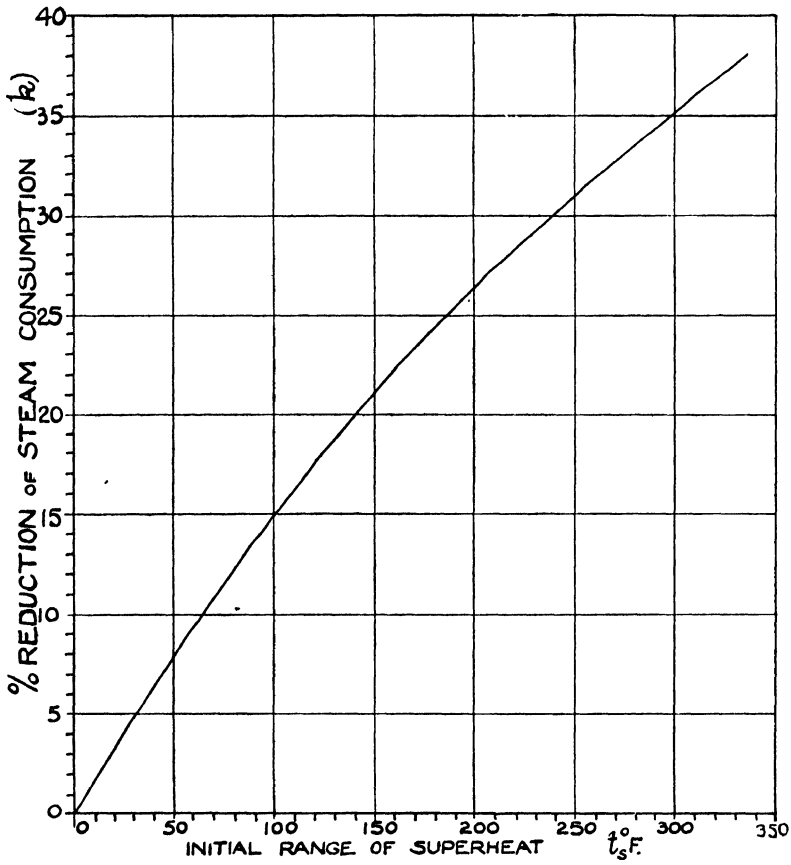



FIG. 142.

steam entering the L.P. cylinder is in consequence much drier than it would be if the engine were supplied with saturated steam. The cumulative effect of the suppression of the major part of the cylinder condensation and leakage of wet steam is an appreciable improvement in efficiency and economy.

The percentage reduction of the rate of steam consumption resulting from every 10° Fahr. increase of superheat depends on the type and size of the engine and the initial superheat. In general, the higher

the superheat the smaller will be the influence of the size of the engine on the steam consumption; and if a very high range of superheat is used all sizes of engine of a given type may be expected to have approximately the same rate of steam consumption (see Art. 444).

From data published by Messrs. Belliss and Morcum relating to tests on their high-speed type of triple-expansion engines, it appears that between 0° and 100° Fahr. a reduction of 1·5 per cent. in steam consumption rate per 10° Fahr. rise of superheat may be expected; between 100° Fahr. and 200° Fahr. a reduction of 1·2, and between 200° Fahr. and 300° Fahr. a reduction of 0·8.

For a given case these figures may be used tentatively to estimate the probable improvement in economy and efficiency of this type on full load when the stop-valve and exhaust conditions are specified between 170 and 200 lb./in.$^2$ abs. and 25 in. to 27 in. vacuum.

The curve shown in Fig. 142 has been derived from the above figures, and enables the reduction on the " dry steam " consumption to be read directly for the specified range of superheat.

If $w$ is consumption of dry steam in lb./H.P. hr., then the reduced consumption at superheat $t_s$, for which the factor $k$ is ascertained from the curve, is—

$$w'=(1-k)w \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (36)$$

For the specified initial pressure $p_1$ the dry steam total heat $H_1$ is known, and also the total heat $H_{s1}$ of the superheated steam at $p_1$ and $t_{s1}$. The water heat $h_2$ corresponding to the given vacuum pressure is known, and—

(Dry) $$\eta=\frac{2545}{w(H_1-h_2)} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (37)$$

(Superheated) $$\eta'=\frac{2545}{w'(H_{s1}-h_2)} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (38)$$

The corresponding economy values or rates of heat consumption are—

$$Q=\frac{42{\cdot}4}{\eta} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (39)$$

$$Q'=\frac{42{\cdot}4}{\eta'} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (40)$$

The consumptions may be given either in lb./I.H.P. hr. or lb./B.H.P. hr. and corresponding subscripts I and B are to be used for efficiency and economy.

Increase of thermal efficiency is—

$$x=\left(\frac{\eta'}{\eta}-1\right) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (41)$$

Increase of economy or decrease of heat consumption—

$$y=\left(1-\frac{Q'}{Q}\right)=\left(1-\frac{\eta}{\eta'}\right) \quad . \quad . \quad . \quad . \quad (42)$$

*Example* 13.—A high-speed triple-expansion engine when run on full load with initially dry steam at 170 lb./in.$^2$ abs. shows a consumption rate of 16·7 lb./B.H.P. hr., the vacuum being 26 in. (30″ Bar). Find the probable consumption rate, the increase in thermal efficiency, and economy, if superheated steam at 568·4° Fahr. is supplied, the vacuum remaining the same.

Saturated steam: $p_1=170$ lb., $H_1=1200{\cdot}4$, $p_2=2$, $h_2=93{\cdot}9$ B.Th.U./lb.

Heat supplied per lb. $H=(H_1-h_2)=(1200{\cdot}4-93{\cdot}9)=1106{\cdot}5$

Consumption rate $w_B=16{\cdot}7$ lb./B.H.P. hr.

Brake thermal efficiency $\eta_B=\dfrac{2545}{w_B H}=\dfrac{2545}{16{\cdot}7\times1106{\cdot}5}=0{\cdot}138$

Economy $Q_B=\dfrac{42{\cdot}4}{\eta_B}=\dfrac{42{\cdot}4}{0{\cdot}138}=308$ B.Th.U./B.H.P. min.

Superheated steam—

$p_1=170$, $t_1=368{\cdot}4$, $t'=568{\cdot}4$, $t_{s1}=200°$ Fahr.
$H_{s1}=1313$, $h_2=93{\cdot}9$, $(H_{s1}-h_2)=1219{\cdot}1$.

Reduction in consumption rate (for dry steam) from curve (Fig. 142) at $t_{s1}=200°$ F., $k=26{\cdot}5$ per cent.

Reduced consumption rate—

$$w_B'=(1-k)w_B=(1-0{\cdot}265)16{\cdot}7$$
$$=0{\cdot}735\times16{\cdot}7=12{\cdot}275 \text{ lb./B.H.P. hr.}$$

$$\eta_B'=\frac{2545}{w_B'(H_{s1}-h_2)}=\frac{2545}{12{\cdot}275\times1219{\cdot}1}=0{\cdot}170$$

$$Q_B'=\frac{42{\cdot}4}{0{\cdot}170}=250 \text{ B.Th.U./B.H.P. min.}$$

Increase in thermal efficiency—

$$x=\left(\frac{\eta_B'}{\eta_B}-1\right)=\left(\frac{0{\cdot}170}{0{\cdot}138}-1\right)=(1{\cdot}23-1{\cdot}0)=0{\cdot}23$$

or 23 per cent.

Decrease of heat consumption—

$$y=\left(1-\frac{\eta_B}{\eta_B'}\right)=\left(1-\frac{0{\cdot}138}{0{\cdot}170}\right)=(1-0{\cdot}81)=0{\cdot}19$$

or 19 per cent.

The engines from which these figures are derived run at speeds from 250 rev./min. to 450 rev./min.

Slower speed engines may be expected to show less improvement than indicated by this typical example for the high-speed type.

Average values much less favourable are quoted by Ennis in the *Mechanical Engineers Handbook* * for American and Continential engines. For an initial temperature of 600° Fahr. he concludes from several tables of experimental results quoted that the improvement of economy is about 20 per cent. for simple engines,

* Second edition edited by Prof. L. S. Marks.

15 per cent. for compounds, and only 8 per cent. for triples. The speed is not stated for any of these cases. The initial pressures for the triples range from 180 to 225 lb./in.$^2$ abs. See also Chapter XXIV.

**150. Reheating between Stages.**—In addition to initial superheat of the steam, it may be resuperheated in the receivers between the stages. The system, however, is not much used in present practice, where highly superheated steam is the rule at the H.P. cylinder. The cylinder which benefits most is the L.P. Live steam is passed from the main through a heater coil placed in the receiver, and may be either desuperheated and condensed in heating the exhaust steam in the receiver or merely desuperheated and reduced in pressure before it passes on to the H.P. stop-valve. In the latter case the steam has to be supplied at much higher pressure and temperature by the boiler than are required at the engine.

The efficiency to be expected from combined superheat and stage reheat is indeterminate. Where, however, the indicated powers with reheat and the receiver temperature and pressures are ascertained from a test, it can be calculated as follows. Taking the general case of a triple with I.P. and L.P. reheating coils in which a certain amount of steam is desuperheated and condensed and the condensate is returned to the hot well, let—

$W$ = steam consumption in lb./hour.
$I_1$ = Indicated horse-power in H.P. cylinder.
$I_2$ = ,, ,, ,, I.P. ,,
$I_3$ = ,, ,, ,, L.P. ,,

The indicated heats in B.Th.U./lb. are—

$$\text{H.P.} \qquad H_{i1}=\frac{2545I_1}{W} \quad . \quad . \quad . \quad . \quad . \quad . \quad (43)$$

$$\text{I.P.} \qquad H_{i2}=\frac{2545I_2}{W} \quad . \quad . \quad . \quad . \quad . \quad . \quad (44)$$

$$\text{L.P.} \qquad H_{i3}=\frac{2545I_3}{W} \quad . \quad . \quad . \quad . \quad . \quad . \quad (45)$$

Let the receiver pressures be $p_1$, $p_2$, $p_3$, and the superheat ranges, determined from the test, be $t_{s1}$, $t_{s2}$, and $t_{s3}$, giving the corresponding total heat values $H_{s1}$, $H_{s2}$, $H_{s3}$, then the state points at admission to each cylinder will be A, B, C (Fig. 143). Scaling the indicated heat values $H_{i1}$, $H_{i2}$, $H_{i3}$ vertically from A, B, and C, the approximate condition in each receiver and at L.P. exhaust is given by $a$, $b$, and $c$, and the total heats are $H_s'_2$, $H_{w3}$, $H_{w4}$. Owing to vacuum drop across the condenser the condensate is cooled below $t_4$, and the feed temperature will be $t'_4$; but if the heater drains are led to the hot well, or if, as is sometimes the case, the exhaust steam is also passed through an exhaust feed heater, the temperature may be

raised to some higher value, which in general may be denoted by $t_f$, giving a water heat $h_f$ for the feed entering the boiler.

The heat supplied to the steam is—

$$(H_{s1}-h_f)+h_{r1}+h_{r2}=H \quad . \quad . \quad . \quad . \quad . \quad (46)$$

where $$h_{r1}=(H_{s2}-H_{s'2}) \quad . \quad . \quad . \quad . \quad . \quad . \quad (47)$$

the I.P. reheat, and $$h_{r2}=(H_{s3}-H_{w3}) \quad . \quad . \quad . \quad . \quad . \quad . \quad (48)$$

the L.P. reheat.

The heat rejected to the circulating water (on the assumption that the heater drains go back directly to the feed) is—

$$h=(H_{w4}-h_4) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (49)$$

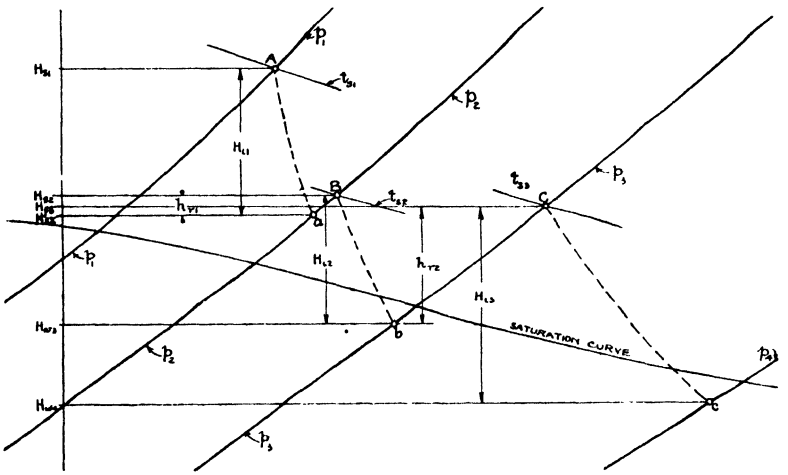



FIG. 143.

The thermal efficiency is then given by—

$$\left.\begin{aligned}\eta_I'&=1-\frac{h}{H}\\&=1-\frac{(H_{w4}-h_4)}{(H_{s1}+h_{r1}+h_{r2}-h_f)}\end{aligned}\right\} \quad . \quad . \quad (50)$$

or $$\eta_I'=1-\frac{(H_{s1}+h_{r1}+h_{r2}-H_{w4})}{(H_{s1}+h_{r1}+h_{r2}-h_f)}$$

The economy is obtained as before.

The following example illustrates the application of the foregoing method of calculation.

*Example* 14.—An unjacketed compound mill engine with an L.P. stage reheater coil has cylinders 16·07 in. diam., 28·03 in. diam., and stroke 42 in. It is run with initial pressure 156 lb./in.$^2$ abs., superheat 273·4° Fahr., and exhaust pressure 2·4 lb./in.$^2$ abs.; the receiver pressure is 38 lb./in.$^2$ abs. and the temperature 408° Fahr.

The indicated horse-powers at 103·28 rev./min. are: H.P., 225, L.P., 249·5; the consumption is 4633 lb./hr. and the feed temperature 133° Fahr. Calculate the thermal efficiency and economy.

$p_1=156$; $t_{s1}=273{\cdot}4$, and $H_{s1}=1347$ $\quad I_1=225$, $W=4633$

$$H_{i1}=\frac{42{\cdot}4\times60\times I_1}{W}=\frac{42{\cdot}4\times225\times60}{4633}$$

$=124$ B.Th.U./lb.

Total heat in exhaust, $H_s'_2=(H_{s1}-H_{i1})=(1347-124)$
$=1223$ B.Th.U./lb.

Superheat at $p_2=38$, $t_{s2}=(t_2'-t_2)=(408-264)$ $\quad t_2'=408$, $t_2=264$
$=144°$ Fahr.

$H_{s2}=1243$. Reheat $h_{r1}=(H_{s2}-H_s'_2)=(1243-1223)$
$=20$ B.Th.U./lb.

$$H_{i2}=\frac{2545\times I_2}{W} \qquad I_2=249{\cdot}5,\ W=4633$$

$$=\frac{2545\times249{\cdot}5}{4633}=137 \text{ B.Th.U./lb.}$$

Exhaust $p_3=2{\cdot}4$, $t_3=133=t_f$, $h_3=h_f=100{\cdot}7$ B.Th.U./lb.

$H_{w3}=(H_{s2}-H_{i2})=(1243-137)=1106$ B.Th.U./lb.

$$\therefore\ \eta_I'=1-\frac{H_{w3}-h_3}{H_{s1}+h_{r1}-h_f}=1-\frac{1106-100{\cdot}7}{1347+20-100{\cdot}7}$$

$$=1-\frac{1005{\cdot}3}{1266{\cdot}3}=(1-0{\cdot}794)=\underline{0{\cdot}206}$$

and economy $Q'=\frac{42{\cdot}4}{0{\cdot}206}=206$ B.Th.U./I.H.P. min.

*Example* 15.—The engine of Example 14, when run with saturated steam and the heater out of action, gives the following results.

Initial pressure 160 lb./in.$^2$ abs., exhaust pressure 2·7 lb./in.$^2$ abs. Total I.H.P. at 102·2 rev./min. 406·7; consumption 5630 lb./hr.; feed temperature 136° Fahr.

Compare the efficiency and economy with those obtained with superheat and reheat.

$p_1=160$ lb./in.$^2$ abs., $H_1=1201$ B.Th.U./lb., $I=406{\cdot}7$, $W=5630$, $t_f=136$, $h_f=103$.

$$\therefore\ w_i=\frac{W}{I}=\frac{5630}{406{\cdot}7}=13{\cdot}84 \text{ lb./I.H.P. hr.}$$

$$\eta_I'=\frac{2545}{w_i(H_1-h_f)}=\frac{2545}{13{\cdot}84(1201-103)}$$

$$=\frac{2545}{13{\cdot}8\times1098}=\underline{0{\cdot}168}$$

Or alternatively—

$$H_{w3}=(H_1-H_i)=(1201-183{\cdot}5)$$
$$=1017{\cdot}5 \text{ B.Th.U./lb.}$$

$$\eta_1=1-\frac{H_{w3}-h_3}{H_i-h_f}$$
$$=1-\frac{1017{\cdot}5-104}{(1201-103)}=1-\frac{913{\cdot}5}{1098}$$
$$=(1-0{\cdot}832)=\underline{0{\cdot}168}$$

$$H_i=\frac{42{\cdot}4\times 60\times I}{W}$$
$$=\frac{42\times 60\times 406{\cdot}7}{5630}$$
$$=183{\cdot}5$$
$$h_3=104$$

$$Q_I=\frac{42{\cdot}4}{0{\cdot}168}=253 \text{ B.Th.U./I.H.P. min.}$$

$\therefore$ Increase of thermal efficiency due to superheat and reheat

$$x=\left(\frac{\eta_I'}{\eta_I}-1\right)=\left(\frac{206}{168}-1\right)=(1{\cdot}22-1)=0{\cdot}22 \text{ or } 22 \text{ per cent.}$$

Decrease of heat consumption—

$$y=\left(1-\frac{Q_I'}{Q_I}\right)=\left(1-\frac{206}{253}\right)=(1-0{\cdot}815)=0{\cdot}185 \text{ or } 18{\cdot}5 \text{ per cent.}$$

**151. Multiple-Expansion Engine with Regenerative Feed-Heating.**—The thermal efficiency of the multiple-expansion engine can be further improved by the partial application of the principle of regeneration, already outlined in Chapter IV. In this case the steam is extracted from the receivers and used to heat the feed in contra-flow surface heaters, on its way from the condenser to the boiler. A diagrammatic sketch of such a heater arrangement, applied to a triple-expansion engine, is shown in Fig. 144. The I.P. heater condensate is cascaded to the L.P. heater, and mixes with the L.P. heater condensate, giving up part of its liquid heat. The combined condensate may then be cascaded to the condenser, as indicated by the full lined pipe, or discharged into the hot well, as indicated by the dotted pipe. In the case of the condenser cascade, the water heat of the heater condensate, reckoned from the condenser temperature, is thrown away in the circulating water. When the condensate is returned to the hot well, this heat is saved, and goes back to the boiler in the feed. The initial temperature of the feed, as drawn from the condenser by the extraction pump, is, as already mentioned, always several degrees below the vacuum temperature at the engine exhaust. This temperature, in the case of drain discharge to the hot well, is raised again by the water heat of the condensate, and the feed is pumped to the first or L.P. heater, next the condenser, at this increased temperature.

In the actual case the pressure in the heater coils is less than the receiver pressure by from 5 per cent. to 7 per cent., due to throttling action. Also the temperature of the feed leaving any heater may be taken as 10° Fahr. less than the saturation temperature of the steam in the coils.

In the L.P. heater coils the pressure will be $p_{h3}$, about 93 per cent. to 95 per cent. of $p_3$, the L.P. receiver pressure, and similarly in the I.P. heater coils it will be $p_{h2}$. If the saturation temperatures for $p_{h3}$ and $p_{h2}$ are $t_3$ and $t_2$, the corresponding feed temperatures are to be taken as $t_3' = (t_3 - 10)°$ Fahr. and $t_2' = (t_2 - 10)°$ Fahr., and the water heats, at these temperatures (from the steam table), as $h_3'$ and $h_2'$. The feed is raised from the hot well temperature $t_f$ to $t_3'$ in the L.P., and from $t_3'$ to $t_2'$ in the I.P. heater.

Let the L.P. and I.P. heater condensates be denoted by $w_3$ and $w_2$ lb. and the total steam supply to the engine by W lb.

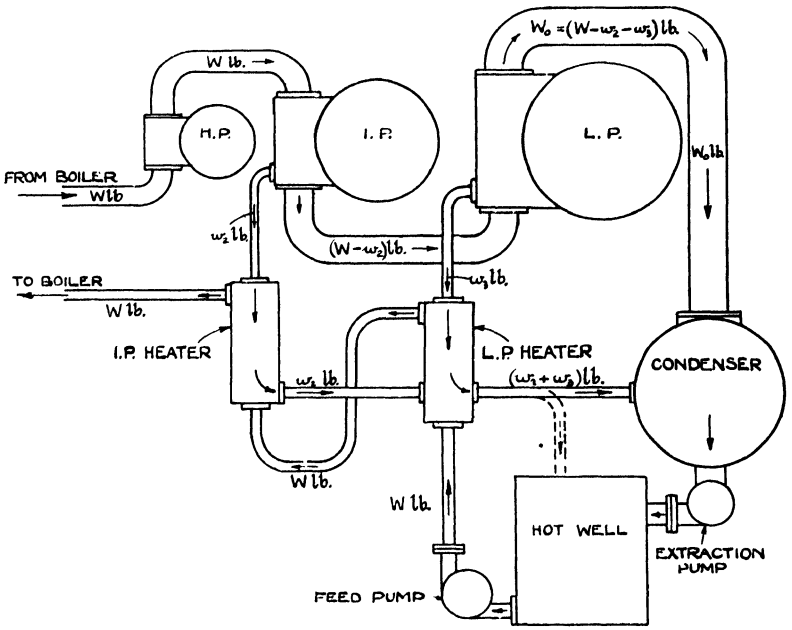



FIG. 144.

Consider the general case of a triple with initially superheated steam, for which the HΦ diagram is shown in Fig. 145. Neglecting radiation loss, then to a first approximation, the state points B, C, and D, giving the qualities and the total heats of the steam at the receiver pressures $p_2$, $p_3$ and the exhaust pressure $p_4$, are found by scaling vertically, in succession, the indicated heats per lb., $H_{i1}$, $H_{i2}$, $H_{i3}$. Alternatively, without the aid of a diagram, the final total heats are given by: $H_{s2} = (H_{s1} - H_{i1})$; $H_{w3} = (H_{s1} - H_{i1} - H_{i2})$; $H_{w4} = (H_{s1} - H_{i1} - H_{i2} - H_{i3})$. The water heats at $p_{h2}$ and $p_{h3}$ or $h_2$ and $h_3$ are obtained from the tables, as also the feed-water heats $h_2'$ and $h_3'$, and the final condensate heat $h_4$ at $t_4$.

The condition for heat transmission to the feed in the I.P. heater is

$$w_2(H_{s2}-h_2)=W(h_2'-h_3')$$

or

$$z_2=\frac{w_2}{W}=\frac{h_2'-h_3'}{(H_{s2}-h_2)} \quad . \quad . \quad . \quad . \quad (51)$$

If the steam should be wet at $p_2$, then $H_{w2}$ is to be used instead of $H_{s2}$. The estimation of the fractional condensate $z_3$ at the L.P. heater is dependent on the method of disposal of the combined heater drain.

If this is cascaded to the condenser, then the temperature of the

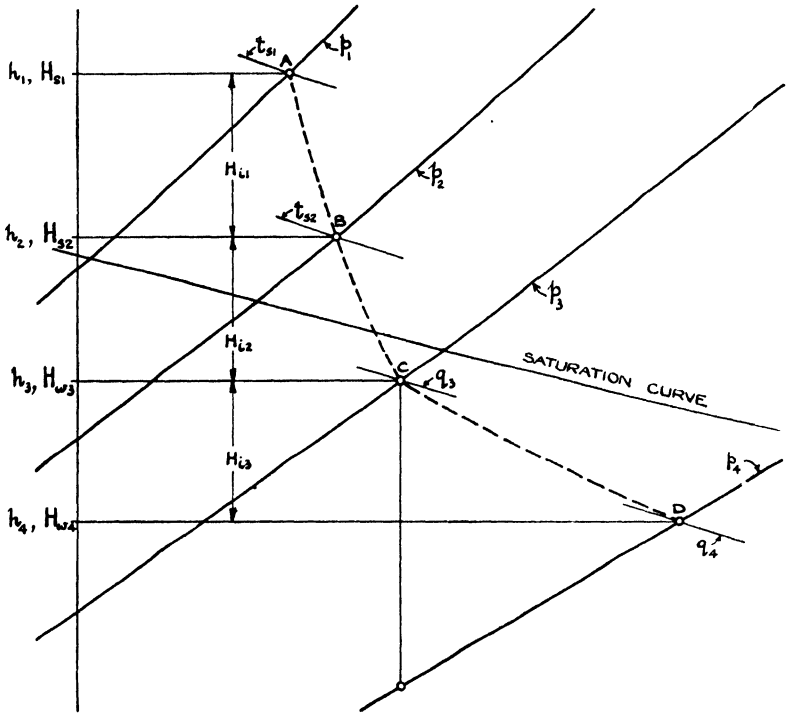



FIG. 145.

feed in the hot well will be that of the condensate drawn by the extraction pump from the condenser. This will usually be $t_5$, some value below the vacuum temperature $t_4$ (see Chapter XXIII).

For this case, the condition of heat transference in the L.P. heater is given by—

$$w_3(H_{w3}-h_3)=W(h_3'-h_5)-w_2(h_2-h_3)$$

or

$$z_3=\frac{w_3}{W}=\frac{(h_3'-h_5)-z_2(h_2-h_3)}{H_{w3}-h_3} \quad . \quad . \quad (52)$$

If, however, the heater drain is taken directly to the hot well

and raises the feed temperature to some value $t_f$, the following relations hold—

$$z_3(H_{w3}-h_3)=(h_3'-h_f)-z_2(h_2-h_3) \quad . \quad . \quad . \quad . \quad (a)$$

$$h_f=(1-z_2-z_3)h_5+(z_2+z_3)h_3 \quad . \quad . \quad . \quad (b)$$

When $h_f$ is eliminated between equations $(a)$ and $(b)$ the result is

$$z_3=\frac{(h_3'-h_5)-z_2(h_2-h_5)}{(H_{w3}-h_5)} \quad . \quad . \quad . \quad . \quad . \quad (53)$$

The thermal efficiency is calculable from the heat supplied and the heat actually rejected in the circulating water.

The feed at entrance to the boiler is raised to the temperature $t_2'$ and has a total heat per lb. of $h_2'$.

At $p_1$ and $t_{s1}$, the steam has a total heat $H_{s1}$. Hence the heat supplied to W lb., the amount of feed circulating, is—

$$W(H_{s1}-h_2')=H \quad . \quad . \quad . \quad . \quad . \quad . \quad (54)$$

The heat rejected depends on the disposal of the heater drains. Taking the case of cascade to the condenser, where $(w_2+w_3)$ lb. is cooled from $t_3$ to $t_5$, the water heat from this rejected in the circulating water is $(w_2+w_3)(h_3-h_5)$. The steam discharged to the condenser at $p_4$ and $t_4$ is $W_o=(W-w_2-w_3)$ and the heat given up by it down to $t_5$ is $(W-w_2-w_3)(H_{w4}-h_5)$. Hence the amount of heat rejected per cycle is—

$$h=(W-w_2-w_3)(H_{w4}-h_5)+(w_2+w_3)(h_3-h_5) \quad . \quad (55)$$

The indicated thermal efficiency is given by—

$$\eta_I'=1-\frac{h}{H}$$

$$=1-\frac{(1-z_3--z_3)(H_{w4}-h_5)+(z_2+z_3)(h_3-h_5)}{(H_{s1}-h_2')} \quad . \quad (56)$$

In the case where the heater drain is taken directly to the hot well, the heat rejected in the circulating water is reduced to $(W-w_2-w_3)(H_{w4}-h_5)$, and the efficiency is—

$$\eta_I'=1-\frac{(1-z_2-z_3)(H_{w4}-h_5)}{(H_{s1}-h_2')} \quad . \quad . \quad . \quad (57)$$

When the heaters are thrown out of action and the engine works as an ordinary triple, the fractional condensations become zero, the feed temperature is $t_5$, and the efficiency is given by—

$$\eta_I=1-\frac{H_{w4}-h_5}{H_{s1}-h_5} \quad . \quad . \quad . \quad . \quad . \quad . \quad (58)$$

The foregoing method can only be applied tentatively to estimate the probable improvement of efficiency or reduction of heat consumption to be expected when regenerative heaters are fitted to an ordinary compound or triple-expansion engine for which test data are available. If the assumption is made that the stage efficiency ratios $(\epsilon_s)$ of the three cylinders will be sensibly the same with as without heaters, or in other words that the indicated heat

per lb. of steam flowing through any stage remains unaltered, and the receiver pressures are kept the same by adjustment of the cut-off fractions, an approximate value of the improvement due to the use of heaters may be obtained.

In dealing with the purely theoretical case of reheating the receiver and heater pressures are to be taken as equal, and the feed temperatures the same as the receiver steam temperatures.

*Example* 16.—A triple-expansion marine engine is supplied with steam at 225 lb./in.$^2$ abs. and 554° Fahr. and exhausts to the condenser at 2·2 lb. /in.$^2$ abs. Find the absolute thermal efficiency and the theoretical steam consumption, when (*a*) it is run on the straight Rankine cycle ; (*b*) on the partial regenerative cycle with I.P. and L.P. heaters, and the heater drain is discharged to the hot well. The receiver pressures are 75 lb./in.$^2$ abs. and 24 lb./in.$^2$ abs.

From the HΦ chart the total heats shown by the adiabatic vertical are—

| | | |
|---|---|---|
| $p_1=225$ | $H_{s1}=1300$ | $h_1=366{\cdot}3$ |
| $p_2=75$ | $H_{s2}=1199$ | $h_2=277{\cdot}3$ |
| $p_3=23$ | $H_{w3}=1108$ | $h_3=203{\cdot}7$ |
| $p_4=2{\cdot}2$ | $H_{w4}=962$ | $h_4=97{\cdot}4$ |

(*a*) Straight Rankine cycle. Absolute thermal efficiency—

$$\eta=1-\frac{H_{w4}-h_4}{H_{s1}-h_4}$$

$$=1-\frac{962-97{\cdot}4}{1300-97{\cdot}4}$$

$$=(1-0{\cdot}72)=0{\cdot}28$$

Theoretical steam consumption—

$$w=\frac{2545}{\eta(H_{s1}-h_4)}=\frac{2545}{0{\cdot}28\times1202{\cdot}6}=7{\cdot}55 \text{ lb./I.H.P. hr.}$$

(*b*) Partial regenerative cycle. Ideal fractional condensation

$$z_2=\frac{h_2-h_3}{H_{s2}-h_2}=\frac{(277{\cdot}3-203{\cdot}7)}{(1199-277{\cdot}3)}=\frac{73{\cdot}6}{921{\cdot}7}=0{\cdot}08 \text{ lb.}$$

$$z_3=\frac{(h_3-h_4)-z_2(h_2-h_4)}{H_{w3}-h_4}=\frac{(203{\cdot}7-97{\cdot}4)-0{\cdot}08(277{\cdot}3-97{\cdot}4)}{(1108-97{\cdot}4)}$$

$$=\frac{106{\cdot}3-14{\cdot}4}{1010{\cdot}6}=\frac{91{\cdot}9}{1010{\cdot}6}=0{\cdot}091$$

In the ideal case $t_5=t_4$ ; $h_2'=h_2$.

Absolute thermal efficiency—

$$\eta'=1-\frac{(1-z_2-z_3)(H_{w4}-h_4)}{H_{s1}-h_2}$$

$$=1-\frac{(1-0{\cdot}08-0{\cdot}091)(962-97{\cdot}4)}{(1300-277{\cdot}3)}$$

$$=1-\frac{0{\cdot}829\times864{\cdot}6}{1022{\cdot}7}=(1-0{\cdot}7)=0{\cdot}3$$

The theoretical increase of thermal efficiency due to the feed-heating is 7·15 per cent. The theoretical steam consumption is—

$$w'=\frac{2545}{\eta'(H_{s1}-h_2)}=\frac{2545}{0{\cdot}3\times 1022{\cdot}7}=8{\cdot}3 \text{ lb./I.H.P. hr.}$$

The theoretical increase of steam consumption is 9·9 per cent.

*Example* 17.—The triple-expansion engine (Example 16) when run without heaters gave the following figures on test: $p_1=225$, $t_{s1}=162°$ Fahr., $p_2=75$, $p_3=24$, $p_4=2{\cdot}2$ lb./in.$^2$ abs.; indicated horse-power, H.P. 985; I.P., 823; L.P., 797. Total feed, 28,900 lb./hr.

Find the indicated thermal efficiency and the steam consumption. Neglect under-cooling of condensate.

Assuming the same receiver pressures and indicated work per lb. in the several stages, calculate the corresponding approximate thermal efficiency and consumption, when I.P. and L.P. regenerative heaters are used.

H.P. $H_{i1}=\epsilon_{s1}h_{\phi 1}=\frac{2545I_1}{W}=\frac{2545\times 985}{28900}=86{\cdot}7$ B.Th.U./lb.

I.P. $H_{i2}=\epsilon_{s2}h_{\phi 2}=\frac{2545I_2}{W}=\frac{2545\times 823}{28900}=72$ ,,

L.P. $H_{i3}=\epsilon_{s3}h_{\phi 3}=\frac{2545I_3}{2545}=\frac{2545\times 797}{28900}=70$ ,,

$I_1=985$
$W=28900$
$I_2=823$
$I_3=797$

Setting off the $H_i$ values (indicated heat per lb.) as shown in Fig. 145, the total heats obtained are—

$H_{s1}=1302$
$H_{s2}=1206$, $h_2=272{\cdot}5$, $h_2'=262$
$H_{w3}=1134$, $h_3=201{\cdot}3$, $h_3'=191$
$H_{w4}=1065$, $h_4=97{\cdot}4$

$p_{h2}=0{\cdot}93\times p_2'=.70$ lb./in.$^2$ abs.
$p_{h3}=0{\cdot}93\times p_3=22$ ,, ,,
$t_2=303$, $t_2'=293°$ Fahr.
$t_3=233$, $t_3'=223°$ ,,
$t_4=129{\cdot}6°$ Fahr.

Indicated thermal efficiency without heaters—

$$\eta_I=1-\frac{H_{w4}-h_4}{H_{s1}-h_4}=1-\frac{1065-97{\cdot}4}{1037-97{\cdot}4}$$

$$=1-\frac{967{\cdot}6}{1204{\cdot}6}=(1-0{\cdot}803)=0{\cdot}197$$

Steam consumption without heaters—

$$w_i=\frac{2545}{\eta_I(H_{s1}-h_4)}=\frac{2545}{0{\cdot}197\times 1204{\cdot}6}=10{\cdot}72 \text{ lb./I.H.P. hr.}$$

Fractional condensates with heaters—

$$z_2=\frac{h_2'-h_3'}{H_{s2}-h_2}=\frac{262-191}{1206-272{\cdot}5}=\frac{71}{933{\cdot}5}=0{\cdot}076 \text{ lb.}$$

$$z_3=\frac{(h_3'-h_4)-z_2(h_2-h_4)}{(H_{w3}-h_4)}=\frac{(191-97{\cdot}4)-0{\cdot}076(272{\cdot}5-97{\cdot}4)}{(1134-97{\cdot}4)}$$

$$=\frac{93{\cdot}6-0{\cdot}076\times 175{\cdot}1}{1036{\cdot}6}=\frac{93{\cdot}6-13{\cdot}3}{1036{\cdot}6}=\frac{80{\cdot}3}{1036{\cdot}6}=0{\cdot}0774 \text{ lb.}$$

Indicated thermal efficiency with heaters—

$$\eta_I' = 1 - \frac{(1-z_2-z_3)(H_{w4}-h_4)}{(H_{s1}-h_2')} = 1 - \frac{(1-0{\cdot}1534)(1065-97{\cdot}4)}{1302-262}$$

$$= 1 - \frac{819{\cdot}17}{1040} = (1-0{\cdot}787) = 0{\cdot}213$$

Steam consumption with heaters—

$$w_i' = \frac{2545}{\eta_I'(H_{s1}-h_2')} = \frac{2545}{0{\cdot}213 \times 1040} = 11{\cdot}5 \text{ lb./I.H.P. hr.}$$

Actual increase in thermal efficiency due to use of heaters is 8·1 per cent. ; and the increase of steam consumption is 7·27 per cent.

The rate of heat consumption without heaters is—

$$Q_i = \frac{2545}{\eta_I} = \frac{2545}{0{\cdot}197} = 12{,}918 \text{ B.Th.U./I.H.P. hr.}$$

Rate of heat consumption with heaters—

$$Q_i' = \frac{2545}{\eta_I'} = \frac{2545}{0{\cdot}213} = 11{,}948 \text{ B.Th.U./I.H.P. hr.}$$

Reduction of heat consumption due to heaters is 7·5 per cent.

# CHAPTER VIII

## ENGINE DETAILS

**152. Cylinders.**—The general proportions of a cylinder depend on the relation between the length of stroke and the diameter. The ratio varies from 1·5 to 2 for slow-speed horizontal mill engines, from 1·25 to 0·6 for vertical high-speed engines, for locomotives from 1·3 to 1·5, and for marine engines from 0·6 to 0·7, referred to the L.P. diameter.

Where slide-valves are used the ports are made from 0·6 to 0·8 of the cylinder diameter in length. The longer the port the smaller the port opening for admission and exhaust of steam at given velocities, and the shorter the valve travel. The full port area is conditioned by the exhaust of the steam. Large ports, however, may involve large clearance volumes and surfaces; and care should be taken to keep the latter as small as possible.

When the inner surface of the cover and the piston face can be turned, the linear cylinder clearance may be reduced to the lowest practicable limit for safe operation. This linear clearance between piston and cover may vary from $\frac{1}{8}$ in. to $\frac{1}{2}$ in. according to the size of the engine. With a small clearance considerable care is necessary in the adjustment of crosshead and connecting rod brasses.

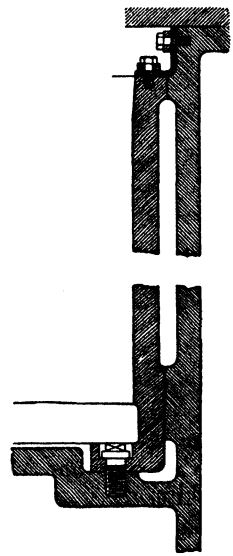


FIG. 146.

**153. Cylinder Barrels and Liners.**—In the case of a multiple-expansion engine a liner is often fitted in the H.P. cylinder. This is made of a hard, close-grained quality of cast-iron, to provide a suitably hard bearing surface for the piston rings. In small cylinders it may simply be a pressed fit in the ends of the cylinder barrel (see Fig. 368, p. 560). For larger cylinders it is usually secured top and bottom to the cylinder barrel and the space between it and the cylinder is used as a steam jacket. One arrangement of liner, as fitted on the vertical cylinder of a marine engine, is shown in Fig. 146. Generally, in present practice, in the intermediate and low-pressure

cylinders of marine engines, liners are dispensed with on the score of expense. The student can refer to the various cylinder designs shown on the types of engines illustrated in Chapters XV to XVIII. A section through the cylinder and piston-valve casing of a small vertical engine is shown in Fig. 147.

Where different stock sizes of a given type of stationary engine are made it is very important to keep the number of cylinder and other patterns down to a minimum. A large stock represents capital which to a considerable extent is unremunerative. For the manufacture of a series of a given type to run at standard piston speed, say 600 to 650 ft./min., with standard steam conditions, the increased or decreased power can be obtained by varying the stroke, so that one head pattern can be used with but slight alteration, and the barrel can be lengthened or shortened to obtain the required stroke. Engine frames or bed-plates can be similarly stocked and slightly altered to take several sizes of cylinders.

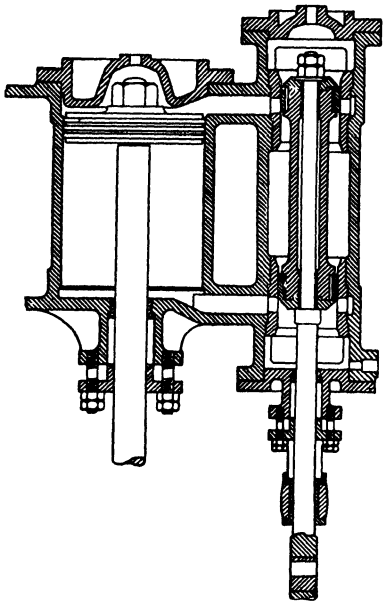


FIG. 147.

**154. Glands.**—A great variety of piston rod and valve rod glands and packings is in use. For low-pressure saturated steam the ordinary stuffing-box type shown in Fig. 147 is used, with one or other of the many "soft" packings or semi-metallic packings. These soft packings are inserted in rings in the box and squeezed up by the gland. They are unsuited for high-pressure and high-temperature steam. In this case one of the various forms of floating metallic ring packing is used. One of the most extensively used packings of this class is the United States Metallic Packing, and it is chosen here to illustrate this important type. The design used for high-pressure steam is shown in longitudinal and transverse section in Fig. 148. The combination consists of a block packing in a case 1, which is fixed to the stuffing box A by stud bolts 2, and a set of cone packing rings in the stuffing box.

The block packing rests on a specially seated neck bush 3, which presses on a sliding ring 4, which in turn is held in the projection of the horn ring 6. The packing block sections 5 are held up against the piston rod B by the radial springs 7. The guide blocks 5A are held up by a second sliding plate 4A, which bears on the block

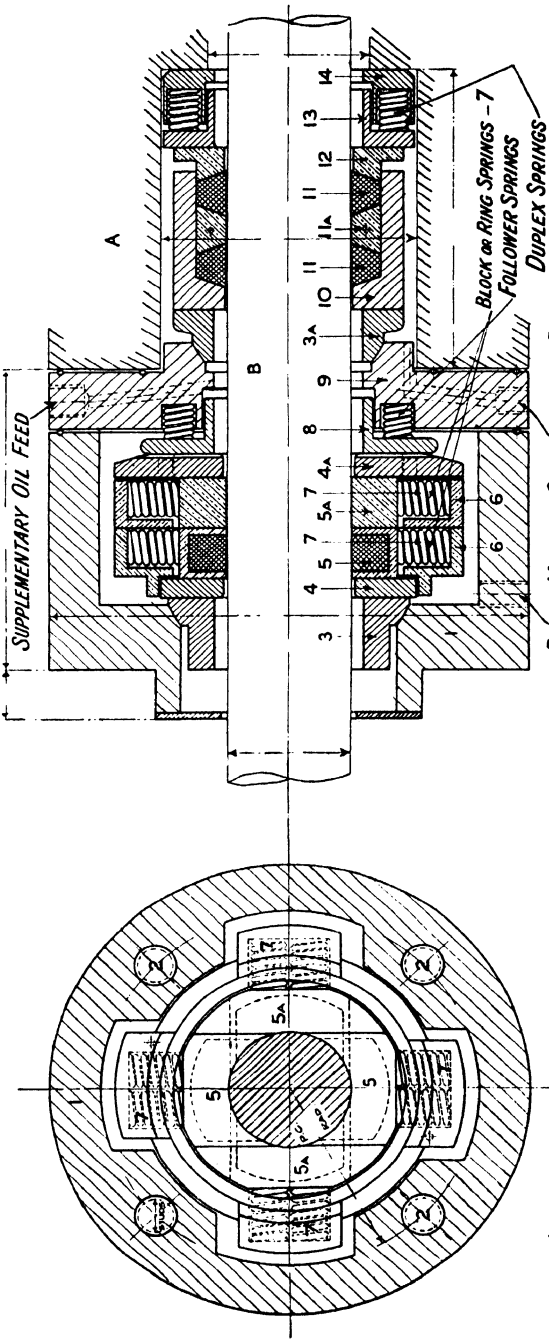

SUPPLEMENTARY OIL FEED
A
B
BLOCK OR RING SPRINGS - 7
FOLLOWER SPRINGS
DUPLEX SPRINGS
DRAIN VALVE
SUPPLEMENTARY DRAIN

FIG. 148.

spring cover 8, pressed axially by the follower springs 7. These springs are fitted in holes in the block plate 9 of the casing, which bears on the end of the stuffing box. Another spherical-seated bush 3A fits on the face of the plate 9, and takes the pressure of the duplex vibrating cup or casing 10 carrying the set of cone packing rings 11, separated by the duplex ring 11A. These are held in

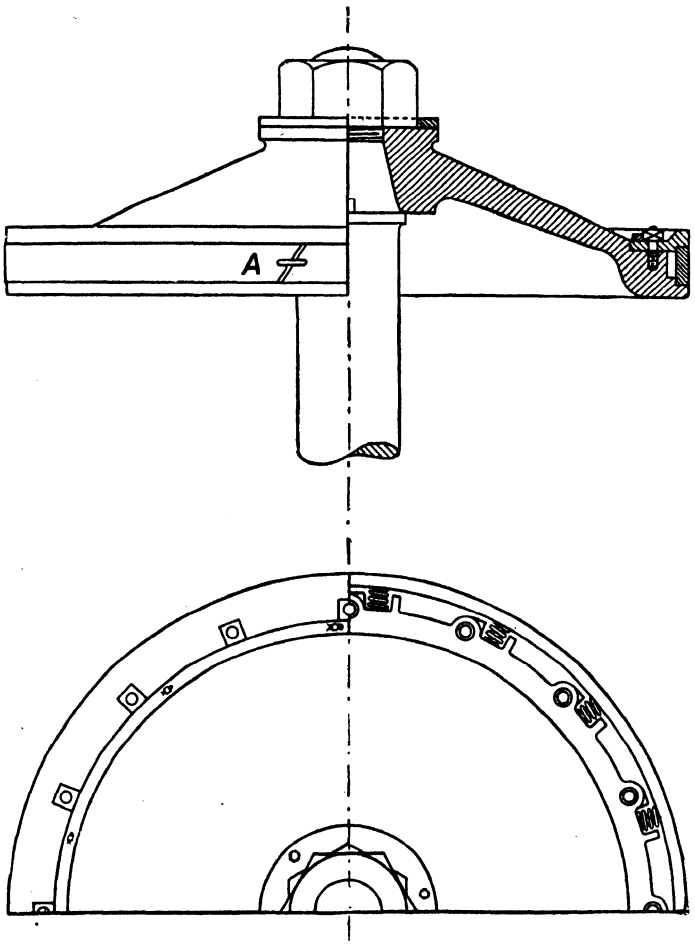



FIG. 149.

by the duplex gland bush or follower 12. The combination is held up by the spring cover 13 and the axial springs in the duplex spring holder 14, which bears on the bottom of the stuffing box.

The packing blocks are put together in sections, four blocks per section. Each section consists of two working blocks 5 lined with Babbit metal, and two guide blocks 5A. The ball joints at 3

and 3A give the packing free play, and keep it steam-tight without regard to the vibration of the rod. With this combination the cone packing next the cylinder checks the steam pressure, and the outer block packing completely prevents the escape of any leakage steam.

**155. Pistons.**—A perfect piston should be frictionless and ensure perfect steam tightness. It is impossible to attain both of these objectives. Some form of packing is required between the piston and cylinder wall, and there is always more or less friction. There are many types of piston packings.

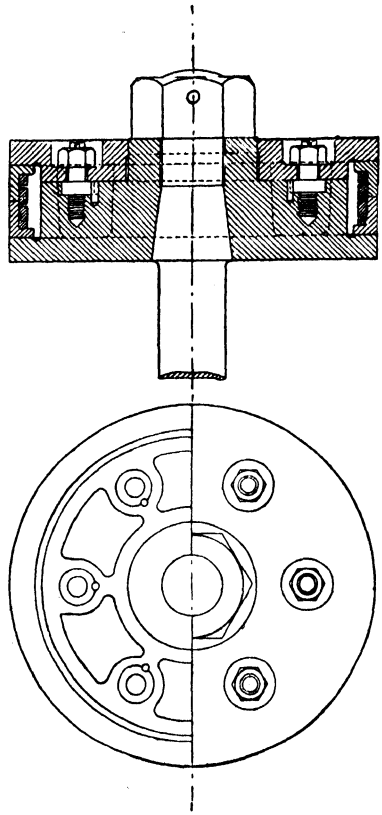


FIG. 150.

The simplest type of packing is a cast-iron ring cut obliquely and sprung over the end of the piston block into a groove cut on the circumference. Usually two rings, either in one groove or separate grooves, are fitted, and placed so as to "break joint." They are known as Ramsbottom rings. Before being cut from the cast-iron solid ring they are usually turned to a diameter slightly greater than the cylinder diameter, and when cut across are compressed into the cylinder. Either an oblique or a tongued joint is made. By breaking joint any steam leaking at one joint is stopped by the adjacent ring.

In very large pistons an old form of packing consisted of a single ring, Fig. 149, held on the piston body by a bottom flange and by a junk ring fixed by studs, as shown in the elevation. This ring had an oblique joint at A into which a tongue piece was fitted to prevent leakage. The necessary pressure radially was obtained by the use of small spiral springs fitting into recesses in the body, as shown in the plan.

These springs gave a pressure about 2 lb. per in.$^2$ bearing surface of the packing ring. The illustration shows a coned cast steel piston, which is both light and strong, and is still generally used for the I.P. and L.P. cylinders of marine engines (see Fig. 374, p. 566). The tapered piston rod end and nut connection is typical of all classes of pistons.

A type of piston used on stationary engines is shown in

Fig. 150. This has a solid block with junk ring holding in two L-section rings which are pressed out by a coiled spring. Another form of conical cast steel marine engine piston is shown in Fig. 151. In this case two spring rings are carried between a spacing ring and the combination is kept in place by a junk ring. The majority of pistons used in present day practice are fitted with some form of combined spring packing rings. One of this type in very general use is the Lockwood and Carlisle packing.

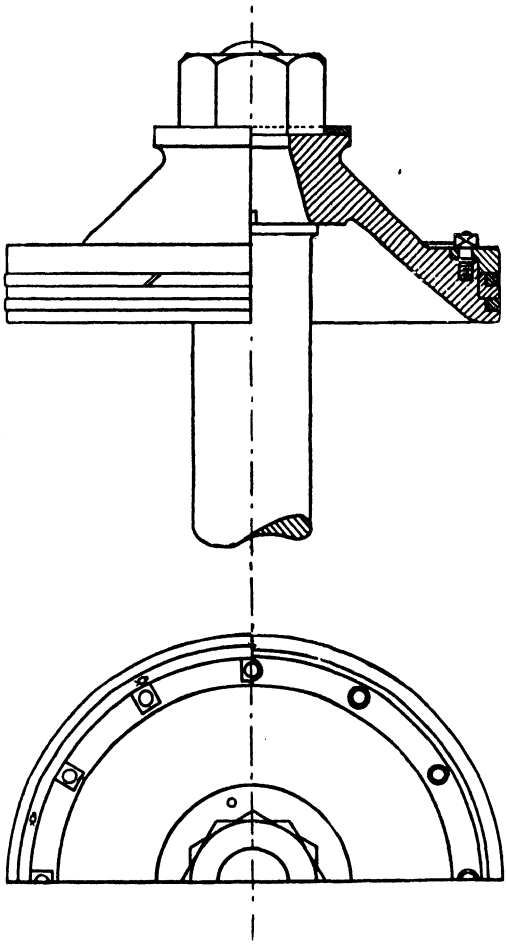


Fig. 151.

This is illustrated in Fig. 152. The piston ring is in two parts, showing the L-shaped sections. Inside these a compound spring is placed. One part of this is helical and the other flat coiled, as clearly shown on the half elevation and part plan, and also indicated on the cross-section. The helical portions press the packing rings out, and the flat coil parts press them against the piston flange and junk ring face, and leakage of steam is prevented at these faces. This type of packing is also much used for large piston valves. An illustration of a piston valve fitted with it is shown in Fig. 153.

The piston rod, generally made of mild steel, has to be considered as a strut, as well as a tension and compression member, and designed to resist the maximum piston load. When connected to the crosshead by a cottered end, ample margin of strength has to be allowed at the cotter section. The safe stress should be kept within 8000 lb./in.$^2$ for mild steel. Similarly at the piston end a safe margin of stress has to be allowed for the screwed end in tension.

**156. Crossheads.**—A crosshead provides a support at the outer end of the piston rod, which would, if unsupported, bend under the oblique thrust or pull of the connecting rod, which reaches a maximum value about mid stroke. This effect is illustrated in Fig. 154.

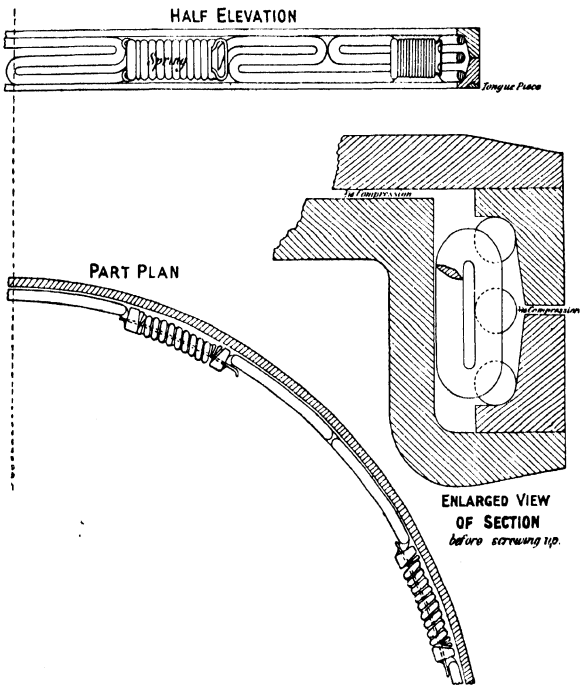



FIG. 152.

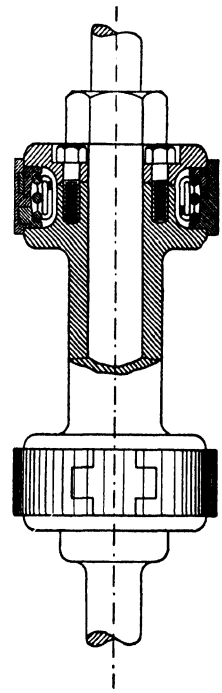


FIG. 153.

The resistance to motion of the crank from C causes a thrust Q along the rod. This has a horizontal component $Q \cos \theta$ which balances the piston load P, and a vertical component $T = Q \sin \theta$ which tends to bend the piston-rod.

$$\text{Since } Q = \frac{P}{\cos \theta}, \quad \text{then} \quad T = P \frac{\sin \theta}{\cos \theta} = P \tan \theta.$$

The condition for the out-stroke is shown here, the crank pin being pushed. On the in-stroke it is pulled, and the vertical thrust T on the crosshead is again downward. Conversely, if the engine runs in the reverse direction the vertical force T will be upward for each stroke. Also, if the engine continues to rotate after the steam is shut off, the crank will drive the piston and reverse the thrust on

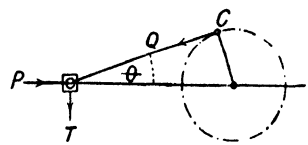



FIG. 154.

the guides. Hence, under all circumstances, it is necessary to have top and bottom guide surfaces. The maximum value of T occurs when the crank and connecting rod are at right-angles, provided cut-off does not take place before half stroke, and the tendency is thus for the guide surfaces to wear most in the centre.

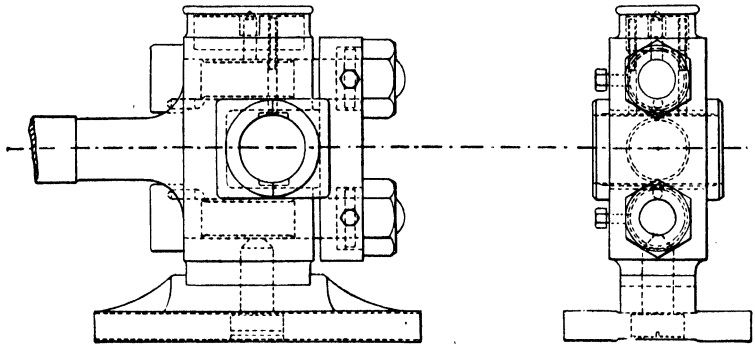


FIG. 155.

It is important, in the case of a horizontal engine, that it should rotate in the direction which brings the thrust T on to the bottom guide, as this surface can be more effectively lubricated.

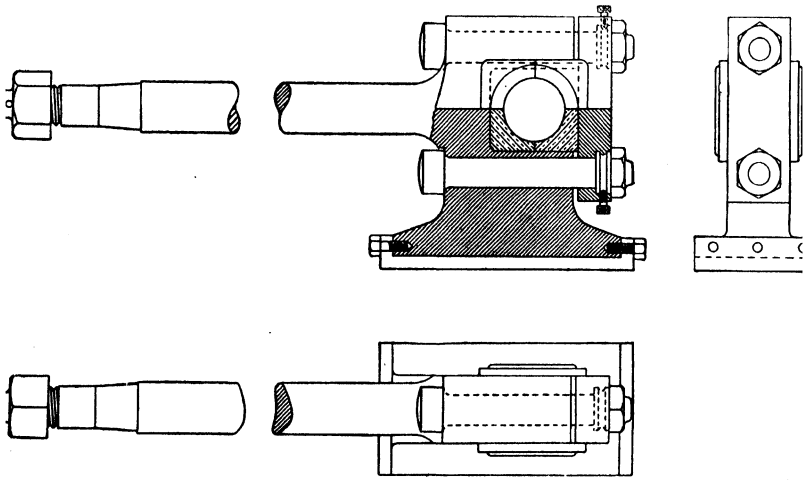


FIG. 156.

When an engine has to rotate equally in either direction, the crosshead surface in contact with the guide is made the same for top and bottom. When, however, only occasional reversal of the engine is necessary, as in the case of a marine engine, the surface which normally transmits this thrust is made sufficiently large, but the opposite surface may be reduced. A crosshead for this condition

is shown in Fig. 155, which is provided with a slipper or shoe guide. The bottom of the shoe takes the normal thrust and the smaller upper surface of the projecting sides, as shown in the end elevation, takes any occasional reversed thrust. A somewhat similar design is shown in Fig. 156, the slipper arrangement being slightly different. In each case the block is forged solid with the piston rod and slotted out to take the brasses, which are held in by cap and bolts. A forked-end connecting rod has to be used here. The standard form of forked crosshead used on inside cylinder locomotives with four guide bars is shown in Fig. 157. Here the gudgeon or crosshead pin is fixed between the jaws and projects at each side, to take the rectangular shell guide blocks of cast-iron. A strap end-connecting rod has to be used, since the gudgeon pin is a fixture.

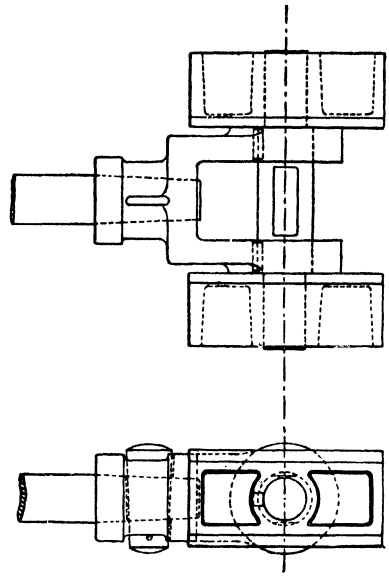


FIG. 157.

A type of crosshead used between channel slide bars and on trunk frames is shown in Fig. 158. Adjustment for wear is made by taking out liners and drawing up the slippers on the inclined faces. This crosshead block is usually made of cast steel and the the slippers of cast-iron.

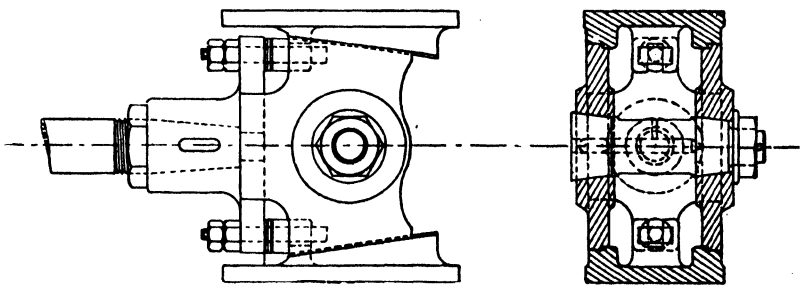


FIG. 158.

An example of a crosshead used on high-speed engines is shown in Fig. 159. The wearing surfaces are turned to fit the bored-out frame guides. The gudgeon pin is fitted with an oil box, which is drop fed for vertical and wiper fed for horizontal engines. It is

cut away at each side to form a relief corresponding to the recess in the brasses and give the wearing surfaces an overrun, and prevent the formation of a shoulder when the pin wears.

The pin is tapered at the head end, and next the nut it is parallel, and fits into a split cone bush inserted in the crosshead. This bush enables the pin to be held truly, and allows slack fit to be taken up. The jaws of the crosshead are kept from closing in, under the pull of the gudgeon pin and nuts, by projections on the

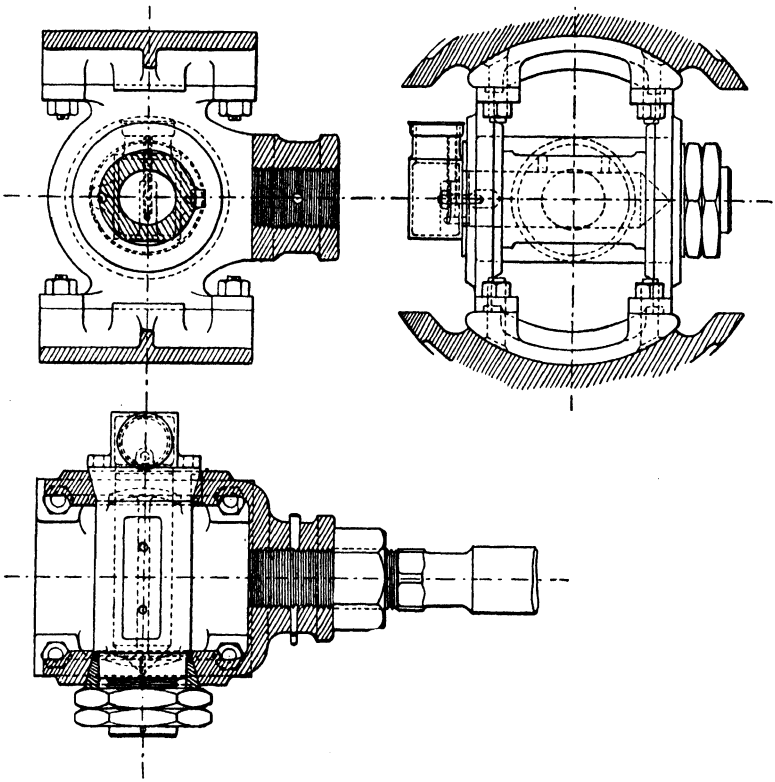


FIG. 159.

slippers, which are fixed on the crosshead by studs. The slippers can be adjusted by the insertion of brass liners.

Crossheads of good design have a low bearing pressure on the slipper faces ; and if this is kept below 40 lb./in.$^2$, adjustments are practically unnecessary. The pressure on the guide surface should not exceed a maximum of 100 lb./in.$^2$, and on the gudgeon pin a maximum of 1200 lb./in.$^2$ of projected area.

**157. Connecting Rods.**—What is usually termed the marine type of connecting rod is shown in Figs. 160 to 163. It is largely used on

horizontal and vertical land engines, as well as on marine engines. Its weight is less than that of other types suited for the same size of crank pin. It is very suitable for high-speed engines on account of the smaller crank balance required. The interspace between the cap and body of the large end is usually fitted with brass horse-shoe fillers, as shown at A, Fig. 162. These are kept in position by small studs, and they can be withdrawn when adjustment of the rod end is necessary. The main nuts are slacked back and the fillers can be removed with the studs in position.

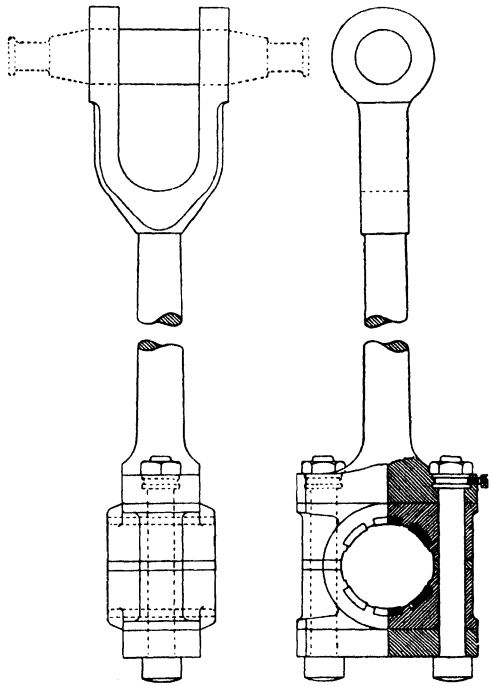


FIG. 160.

The tendency of half brasses, when heated, is to close in on the crank pin; and in order to reduce the risk of seizure, recesses may be cut at the edges, as shown at C, Fig. 161. Also, side screws, shown on the view, may be used to hold the brasses out. In this case, and on the design shown in Fig. 163, one of the cylindrical brasses fits into the bored end of the connecting rod. In the type shown in Fig. 160 the rod has a palm end forged on it, and square brasses are bolted on by means of the bolts and a rectangular cap. This is the form usually fitted to marine engines. These large end brasses are usually provided with strips of anti-friction metal, as

indicated by the black sections. They usually project about $\frac{1}{32}$ in. above the surface. Oil gutters are also cut across the surface at the top.

It will be noted that—with the exception of Fig. 163—collar nuts and locking set pins are used to fix the bolts; and the clumsy and heavy arrangement of jamb nuts is avoided. In the designs of Figs. 161 and 163 the bolts pass through part of the circular brasses and keep these from turning.

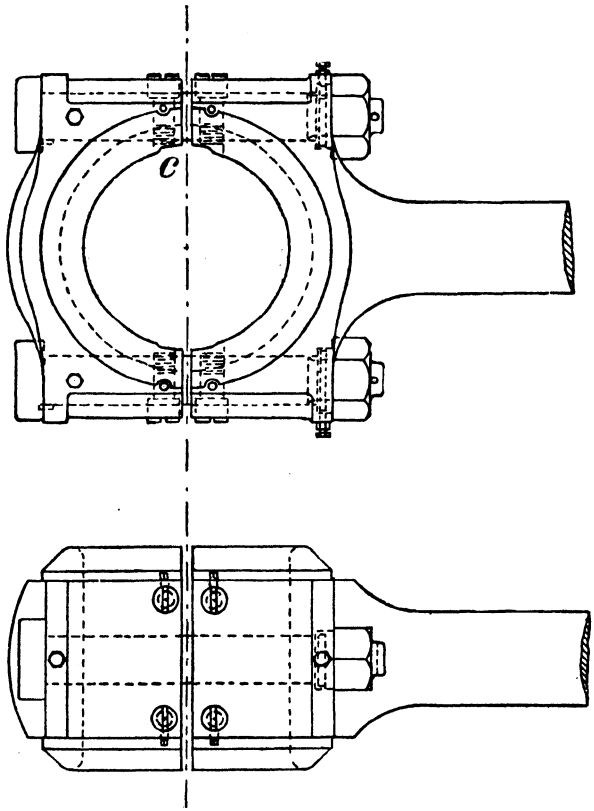



FIG. 161.

With regard to the crosshead ends of connecting rods, a type of forked end with brasses and caps is shown in Fig. 164. It used to be fitted to mill engines, but has given place to types more suited to the modern conditions of running. It is, however, the usual form of rod end used on marine engines, which have solid crossheads with projecting gudgeon pins. A type of forked end suitable for use with the large end design (Fig. 163) is shown in Fig. 165. Care has

to be taken in the manufacture of this fork type to ensure that no flaws occur in the metal of the fork.

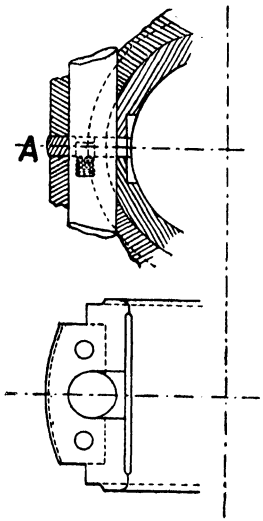



FIG. 162.

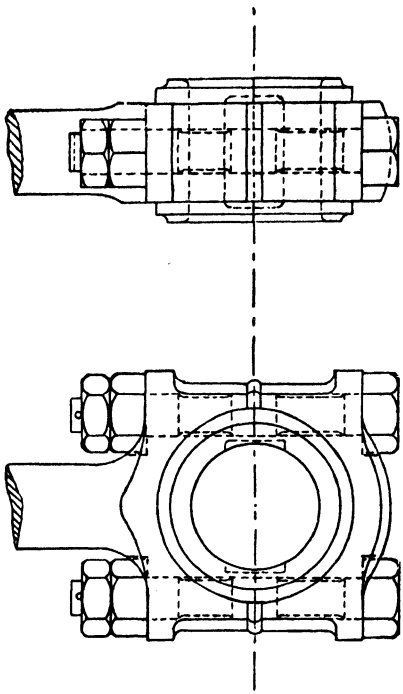


FIG. 163.

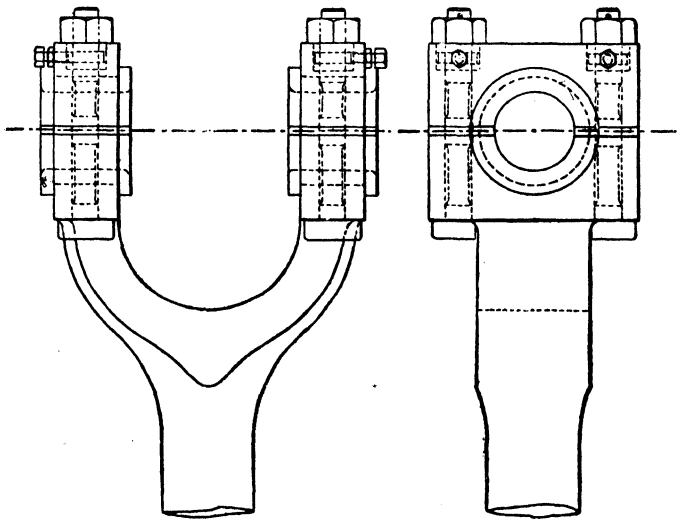


FIG. 164.

One form of a gib and cotter type of connecting rod is shown in Fig. 166. Adjustment of the brasses in the strap is made by the cotter, and the gib prevents the strap from opening. Sometimes two gibs have been used, one on each side of the cotter, the taper of the cotter being halved and the gibs made with the same taper.

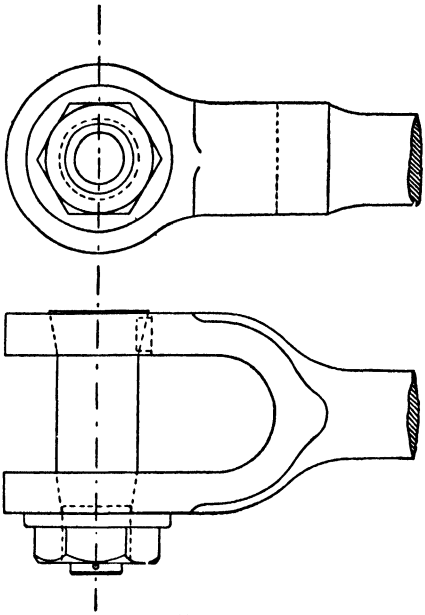


FIG. 165.

For high speed, the strap end is not as good as the marine end, as the strap has a tendency to open out and produce a slack fit of the brasses. This form of gib and cotter end is seldom used in present practice. The crosshead end, as seen on the right, is of the "solid" form. It is slotted out to take the rectangular half brasses. A steel pad piece moved by a fine-thread screw is used for adjustment. While this design is suitable for small engines, the fine screw adjustment makes it unsuitable for large ones. A much more serviceable type of solid end is

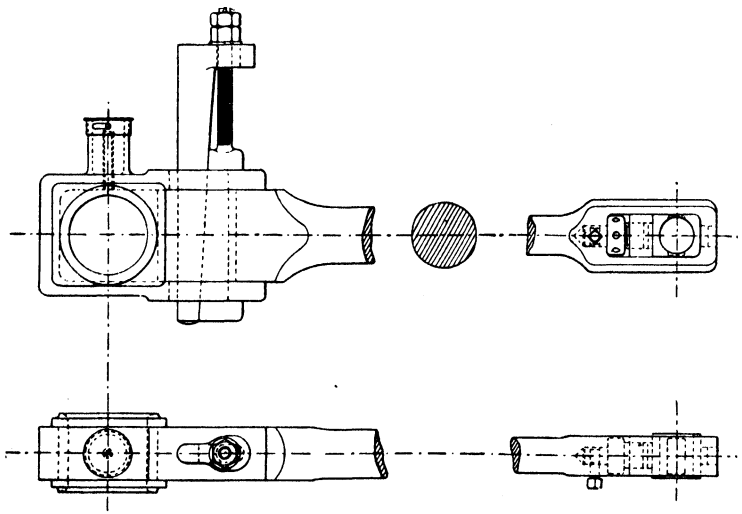


FIG. 166.

shown in Fig. 167. Here the adjustment is made by a steel wedge block behind the end brass, adjusted by a heavy bolt, which can be locked in position by jamb nuts.

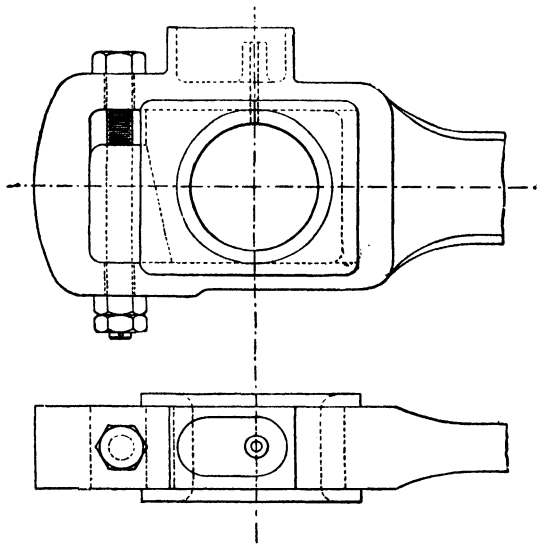


FIG. 167.

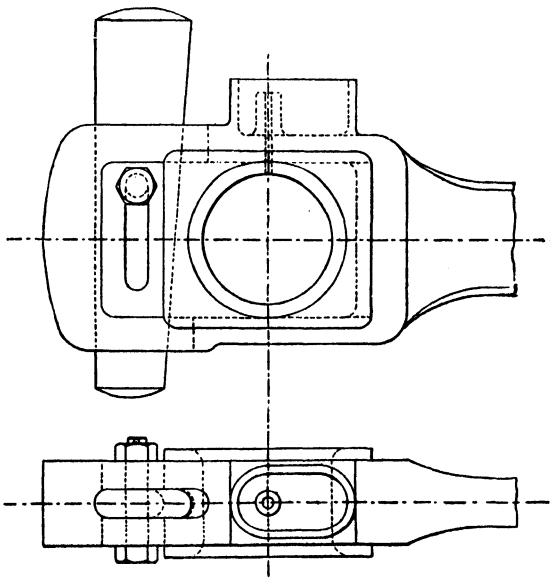


FIG. 168.

Two other forms of this end are shown in Figs. 168 and 169, the adjustment being made by cotters. The design of Fig. 168, with the

simple bolt to lock the cotter in position, is the preferable one. Usually the cotter is locked simply by a set pin screwed into the rod to bear hard on the side of the cotter. The standard design of big or crank end of the inside cylinder locomotive connecting rod is shown in Fig. 170. The rod is forged with a butt end on to which the strap is slipped and securely fastened by two fitted bolts. Adjustment is made by a cotter, which bears on a steel pad placed behind the inside brass. The cotter is locked by set-screws, one being screwed in each arm of the strap. As shown in all these cases, a wick lubricator is provided on the top of the solid end or on the strap.

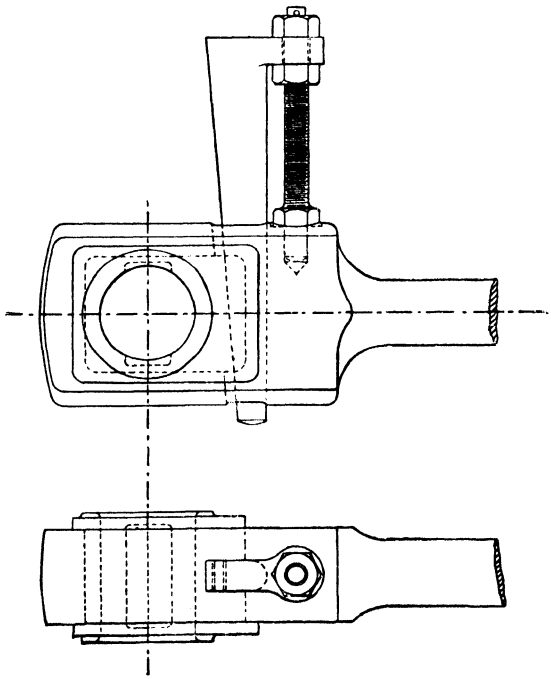


FIG. 169.

These lubricator cups at the crank ends should have screw-down caps to prevent the oil from being thrown off at speed. The cups on the crosshead end, which have little rotary motion, can be left open, as shown in Figs. 167 and 168.

An alternative design of crank end for a locomotive is shown in Fig. 171. The end of the rod is forked, and after the brasses are inserted it is closed by a steel block and the head converted into an equivalent closed end by the bolt shown.

**158. Bearings.**—The main bearings carrying the crank shaft transmit the piston loads and inertia forces to the engine framing. They must be set so as to maintain the crank shaft in proper alignment

and be provided with suitable means of adjustment for wear. The construction should also be such that the brasses may be readily removed while the shaft is in place. The bearing surface should be

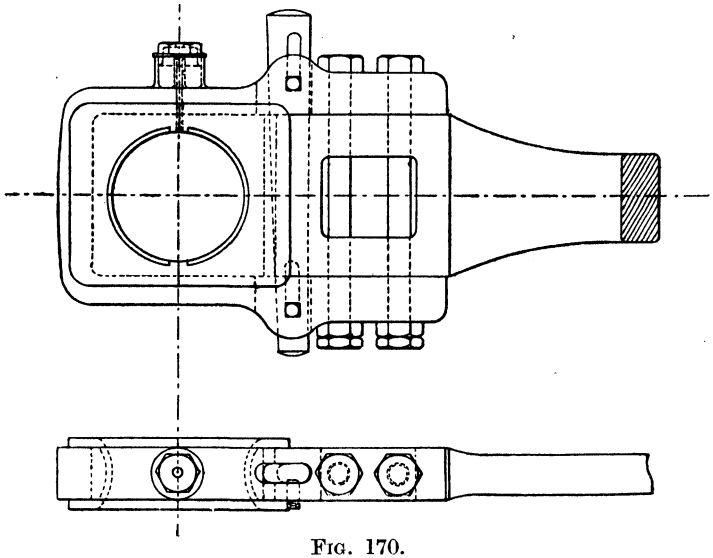


FIG. 170.

sufficient to limit the pressure per square inch of projected surface to 600 lb. for slow rotation and 400 lb. for high rotation engines (Unwin). The brasses on which the shaft journals run should be made of a metal which can readily stand a greater pressure than is normally likely to come on the

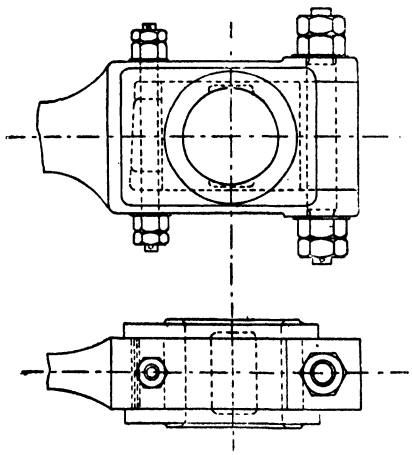


FIG. 171.

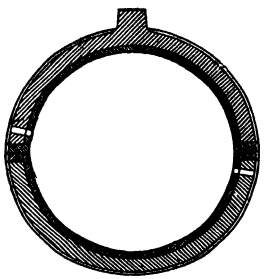


FIG. 172.

bearing; it has also to have anti-friction properties to reduce the possibility of seizure, due to heating.

In a new engine the bearing surfaces must be carefully bedded

down on the seatings, and sufficient running clearances must be provided, especially with vertical engines. These clearances range from 0·001 in. to 0·002 in. per inch diameter of shaft.

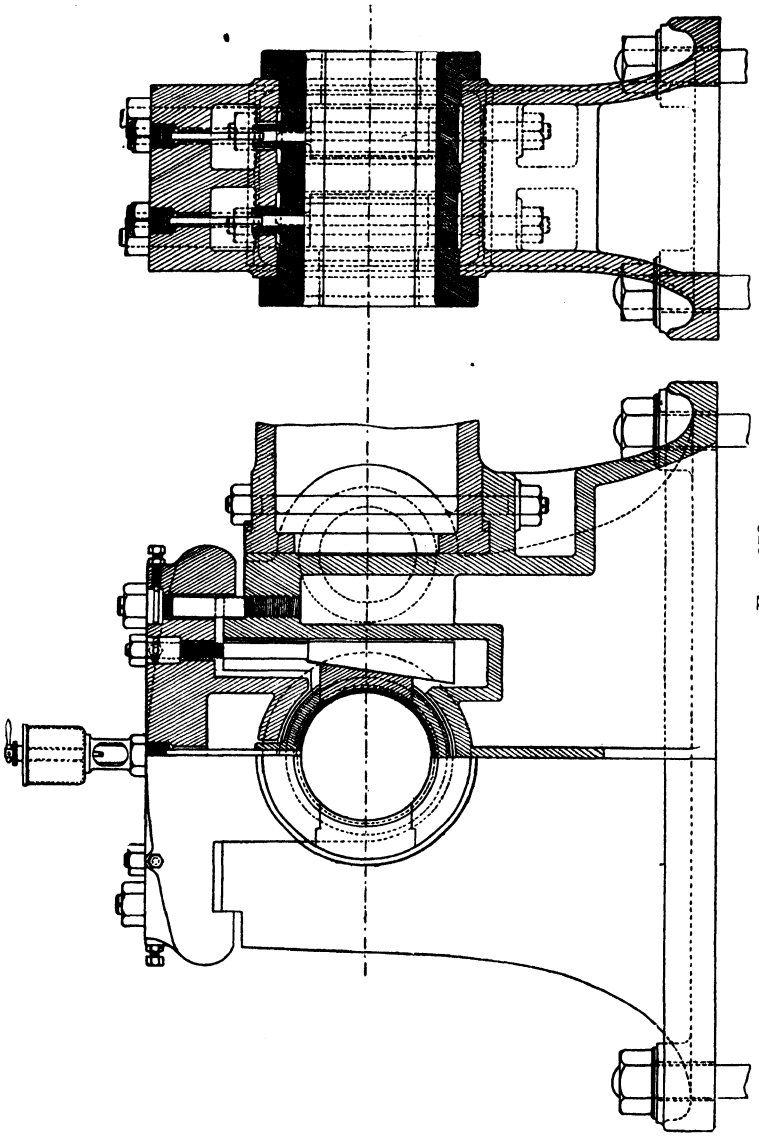


FIG. 173.

According to Dewrance* "the oil for lubrication should be introduced to the bearing surfaces at a point which takes the least

* *Proc. Inst. C. E.*, vol. cxxv., p. 359.

load, and an escape should not be provided for it at the part that has to bear the greatest load." The proper point to introduce the oil is just above the joint of the bearing at the side, as indicated in Fig. 172. There the oil is distributed over the shaft and carried

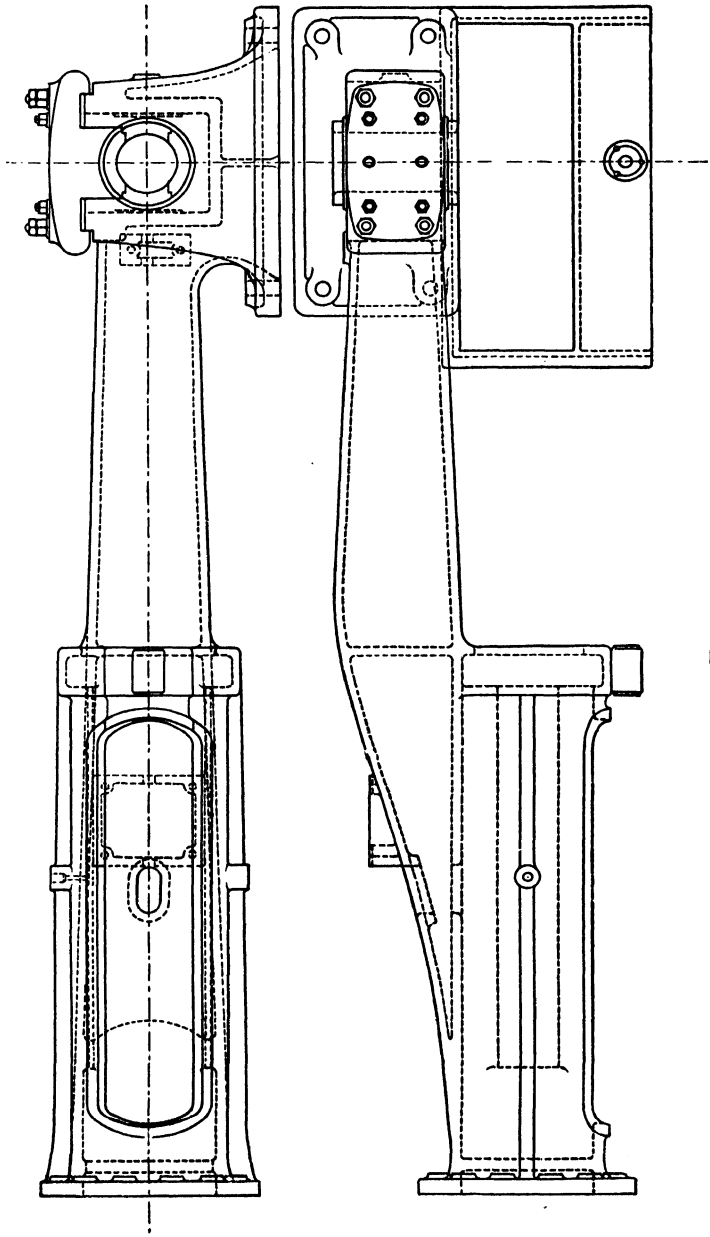


FIG. 174.

to the point of greatest pressure. It is desirable that each half bearing should have its own supply of oil. An example of the type of bearing fitted on large horizontal mill and other engines is shown in Fig. 173. The bearing casing can either be cast with the engine frame or bolted to it. The bearing brasses are in four parts, the horizontal sections being adjustable for wear by means of wedge bolts. The cap is clipped over the upstanding jaws of the pedestal casting, so that when bolted down it provides a practically solid eye, in which the bearing brasses are gripped.

A sight-feed lubricator is fitted on the cap. In modern high-speed engine practice the shafts run in the bearings under forced lubrica-

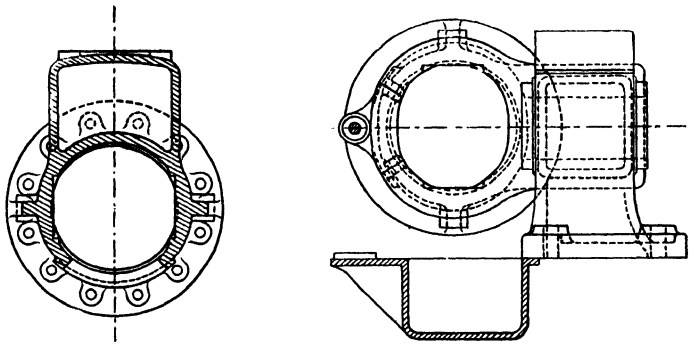


FIG. 175.

tion. Illustrations are given in Chapters XVI and XX on High-Speed Engines and Steam Turbines.

**159. Engine Frames.**—The engine frame has to be designed to give sufficient rigidity to allow for as even a distribution of the loading over its section as possible.

A side elevation and plan of the cast-iron bayonet type of frame for a horizontal engine is shown in Fig. 174 and an end view and cross-section in Fig. 175. More modern designs are illustrated in connection with the various classes of engine discussed in Chapters XIV. XV, XVI, and XVII.

# CHAPTER IX

## VALVES AND VALVE GEARS

**160.** THE distribution of steam to the cylinders of a reciprocating engine may be effected in the following ways.

(*a*) By a single valve, either flat or cylindrical, which covers and uncovers suitable ports.

(*b*) By means of a valve of type (*a*), with another flat or cylindrical valve working on or within it, and covering and uncovering a supplementary set of ports in this valve.

This superposed valve is called an expansion valve ; the other is called the main valve.

(*c*) By a series of four rotating valves each of which covers and uncovers a port at one end of the cylinder. Two of these valves serve for admission and two for exhaust of the steam. Such valves are called Corliss-valves, after the American engineer G. H. Corliss who introduced them.

(*d*) By a series of four conical drop-valves, either single or double beat, which perform the same functions as the four Corliss-valves (*c*).

In one type of engine, the Uniflow, the piston also acts as an exhaust valve, and there are only two admission valves.

In each case the requisite motion for the correct periodic operation of the valve or valves—to admit, cut-off, and exhaust the steam from the cylinder—is obtained from some moving part, usually the driving shaft of the engine ; and the particular linkage employed in any of the systems is given the general name " valve gear."

In certain cases the motion of the gear is quite positive, that is, there is a completely closed chain between the driving shaft and the valve or valves.

In others the chain can be automatically broken by the action of the governor. The latter case occurs partly with classes (*c*) and (*d*), the admission valves being " tripped." The exhaust valves, however, are, in general, positively operated.

Under (*a*) in some cases although the drive of the valve may always be positive, the motion may be varied automatically, by a change of the configuration of the linkage or gear employed.

Such a variation of valve motion is also obtained by the action of a governor. These arrangements are called " expansion gears."

**161.** (*a*) **Slide and Piston-Valves.**—In the majority of small and medium-sized land engines, locomotive and marine engines, the steam distribution to the cylinder is effected by the sliding type of valve.

The valve may work either on a flat face or on a cylindrical one. In the first instance it is termed simply a " slide-valve," in the second a " piston-valve."

As far as the principle of the action is concerned there is no distinction between the two types.

A longitudinal and end section of a common or " D " slide-valve suitable for a small engine are shown in Fig. 176. This valve is

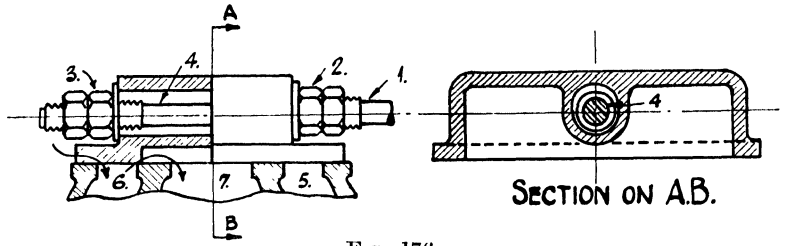



FIG. 176.

driven by the spindle or rod 1, to which it is loosely attached by lock-nuts 2 and 3, which are adjusted so that it has freedom to move down on the face to take up wear, and yet remain in correct adjustment axially. The cored hole 4 through the valve is made oval, to permit of this automatic adjustment.

The steam is admitted over the outer edges to the steam ports

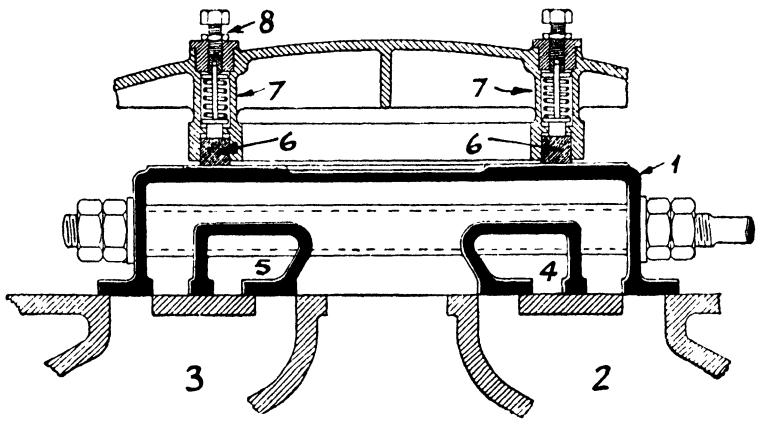



FIG. 177.

5 and 6 when the valve uncovers these, and is exhausted past the inside edges to the exhaust port 7.

In the case of a large cylinder, for instance the L.P. cylinder of a marine engine, the necessary travel of the valve to give a sufficiently wide port opening for the steam would be inconveniently great.

In order to limit the work done in moving the valve and to reduce the dimensions of parts of the driving gear, the slide-valve is made " double ported," as shown in Fig. 177. It will be seen that the

outer darkened section corresponds to the section in Fig. 176. The steam ports 2 and 3 are divided at the cylinder face, and what is virtually an inner slide-valve (cast integrally with the outer one), and represented by the inner darkened section, controls the steam distribution at the inner pair of ports. Steam is admitted to the spaces 4 and 5 for distribution, through passages from the side of the outer shell 1.

The portions of the valve face covering the two ports at one end are identical, so that to obtain the same area of opening for admission of steam at the divided ports the travel of the valve is halved.

In the case of a large valve like this, there is a very considerable unbalanced area subjected to the steam pressure in the casing. The downward load on the valve face may, in consequence, be so great and cause so much friction between the valve and face that a serious load may be imposed on the working parts of the valve gear. This may happen, even with a comparatively small valve, if the pressure in the casing is high. In such cases the valve is

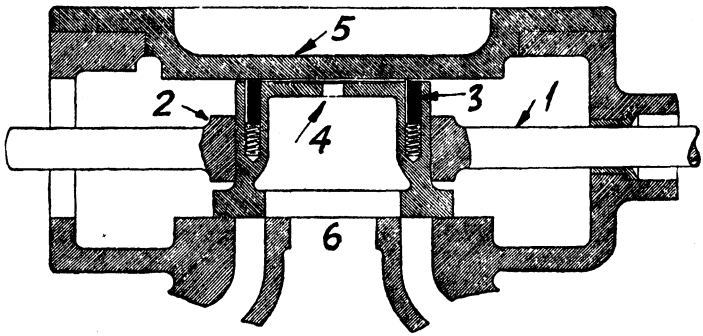



FIG. 178.

balanced by the provision of some form of equilibrium ring fitted either to the casing cover or to the back of the valve.

An old method of balance is shown in Fig. 177, where the equilibrium ring 6, acting like an annular piston in the circular channel, turned in the cylindrical projection 7 on the cover, bears on the machined face of the back of the valve. The pressure of this ring is adjusted by several compression springs and adjusting screws 8. The area of the back, equal to the inner area of the ring circle, thus remains unaffected by the pressure in the casing. Usually this central space is connected to the condenser.

This method of valve balancing is still used in locomotive practice. A longitudinal section through one form of locomotive balanced valve is shown in Fig. 178.

In this case the valve spindle 1 does not pass through the body. It is formed with a strap 2, either square or circular, which fits over the valve, giving it the necessary freedom for wear adjustment. Here the relief ring 3 works in the groove in the back of the valve,

and is held up, by a set of spiral springs, against the face of the cover 5. A hole is provided at 4 in the back of the valve to ensure that the pressure within the equilibrium circle will not rise above the pressure in the engine exhaust port 6.

In addition to the trouble with unbalanced steam pressure the flat type of slide-valve, as has already been pointed out in Chapter VII, is very liable to warp and distort with variation of temperature, and more or less leakage of steam results.

A modern type, extensively used on marine engines, is the Andrews balanced slide-valve. Two longitudinal sections are shown at (*a*) and (*b*), Fig. 179. In (*a*) the valve is at the mid position; in (*b*) at the end position, so as to indicate the flow of the live and exhaust steam through the ports. The valve proper 1 has two faces exactly

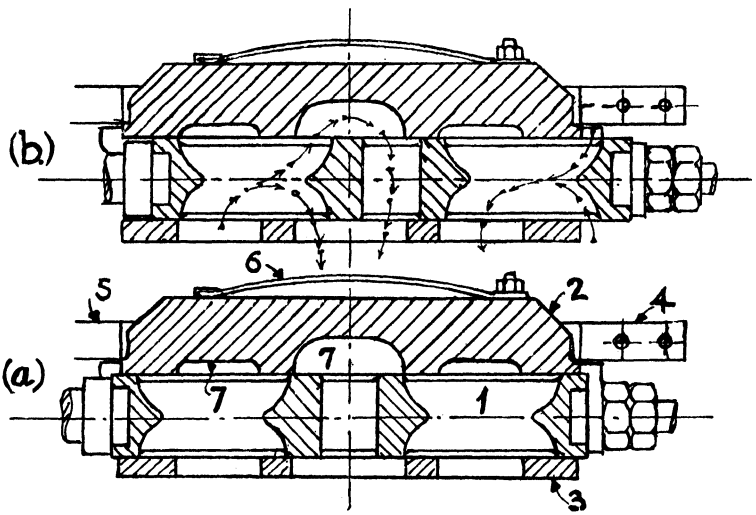



FIG. 179.

alike. It is made a close fit between a strong saddle 2 and the cylinder port face 3. The saddle rests on the cylinder port face by means of side supports, and is prevented from moving longitudinally by stops 4 and 5.

The valve and saddle are free to lift off the face and relieve any excessive pressure in the cylinder, being returned and held down by the coach spring 6 bearing on the casing cover.

The saddle is provided with a port face exactly the same as the cylinder face, but it has cavities 7 instead of through ports.

The steam is admitted between the two inside edges on the end bars of the valve and the extreme edges of the saddle, while the exhaust is controlled by the outside edges on the middle bars of the valve.

The steam distribution is the reverse of that of the common D

slide-valve and the same as that of a piston-valve, taking steam at the inside edges.

The saddle takes the whole steam load, and the valve is thus in perfect balance at all positions of its travel.

The steam and exhaust openings are very direct, and the resistance to flow is much less than in the case of a double-ported slide-valve. The double openings reduce the size of the valve to 50 per cent. of that of the ordinary D slide-valve, and it is 35 per cent. smaller than the double-ported valve.

This valve has met with much success in marine practice on account of the elimination of wear of the valve faces; reduction of load on the valve gear, with consequent saving in wear and tear on the gear, and improved mechanical efficiency; improvement in steam conditions causing reduced condensation and leakage losses.

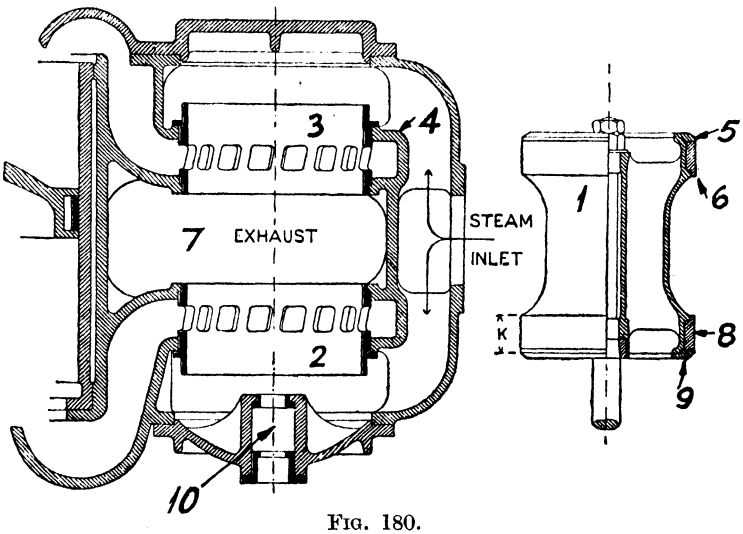



FIG. 180.

162. While special provision has to be made to balance the flat type of slide-valve, the piston type is self-balanced.

A section through the casing of a cylinder of a vertical marine engine fitted with the piston type of sliding valve is shown in Fig. 180. The valve 1, which is shown half in section, and half in elevation on the right, works in liners 2 and 3 fitted in the casing 4. These liners are provided with diagonal bars across the ports, so that scoring of the piston rings may be avoided and that they may be able to move up and down across the port edges without catching on these.

It will be seen that the section on the right is similar to that of the simple slide-valve (Fig. 176), the depth K being the equivalent of the flat part of the slide-valve covering the steam port. Steam here, as can be seen from the directional arrows, is admitted at the

outer edges 5 and exhausted at the inner edges 6 to the exhaust belt 7. The working face consists of a spring ring 8 held in place by a junk ring 9. Another piston-valve with more modern packing is shown in Fig. 153, p. 293. In some instances the admission of steam takes place at the inner edges and exhaust at the outer ones, and through the interior of the valve. This arrangement prevents steam at high temperature and pressure, especially when it is superheated, from getting at the packings of the valve spindle gland.

Although it has the advantage of steam tightness the piston-valve involves the use of large steam passages with accompanying clearances. It is too clumsy and inefficient for use on L.P. marine cylinders, and is usually confined to the high-pressure cylinder. It is much used, however, on some types of small high-speed engine (see Figs. 364, 366, and 369).

**163.** (*b*) **Expansion Valves.**—Either a single- or a double-plate valve may be used on the back of the main valve. In the first case the point of cut-off is a fixed quantity; in the second, by adjustment

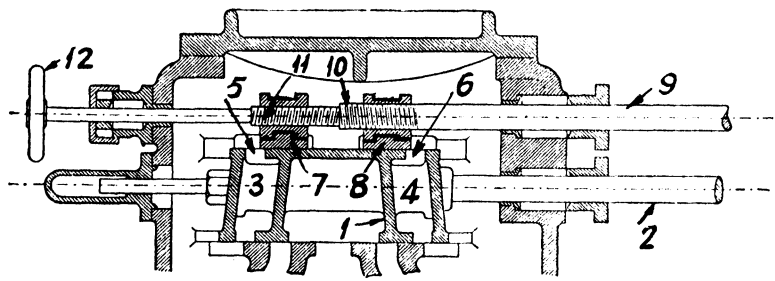



FIG. 181.

of the two plates, the cut-off may be varied, usually by hand, when the engine is running. This is the type commonly used. There are, however, variants of it in the piston-valve form, the expansion valve working inside the main valve, which is provided with inclined ports, the cut-off being varied by a slight rotation of the expansion valve about its axis. The form of the flat and divided type of expansion valve generally used is Meyer's. A longitudinal section through the valve casing of an engine fitted with this type is shown in Fig. 181.

The main valve 1, driven from an eccentric on the shaft by the valve spindle 2, is provided with passages 3 and 4 leading from ports 5 and 6 in the back of the valve. The admission of the steam to the outer edges of the main valve is controlled by the expansion plates 7 and 8, which form a divided expansion valve, and are driven, by a second eccentric on the shaft, through the valve spindle 9. This spindle is provided with a right-handed screw 10 which engages a nut carried loosely by the expansion plate 8 and a left-handed screw 11 which engages another nut carried by the plate 7. The spindle is

so arranged that it can be rotated by a hand-wheel 12 while the engine is running. In this way the plates can be brought together or separated to cut off the supply of steam to the main valve at an earlier or later point of the piston stroke, to suit the variation of load on the engine.

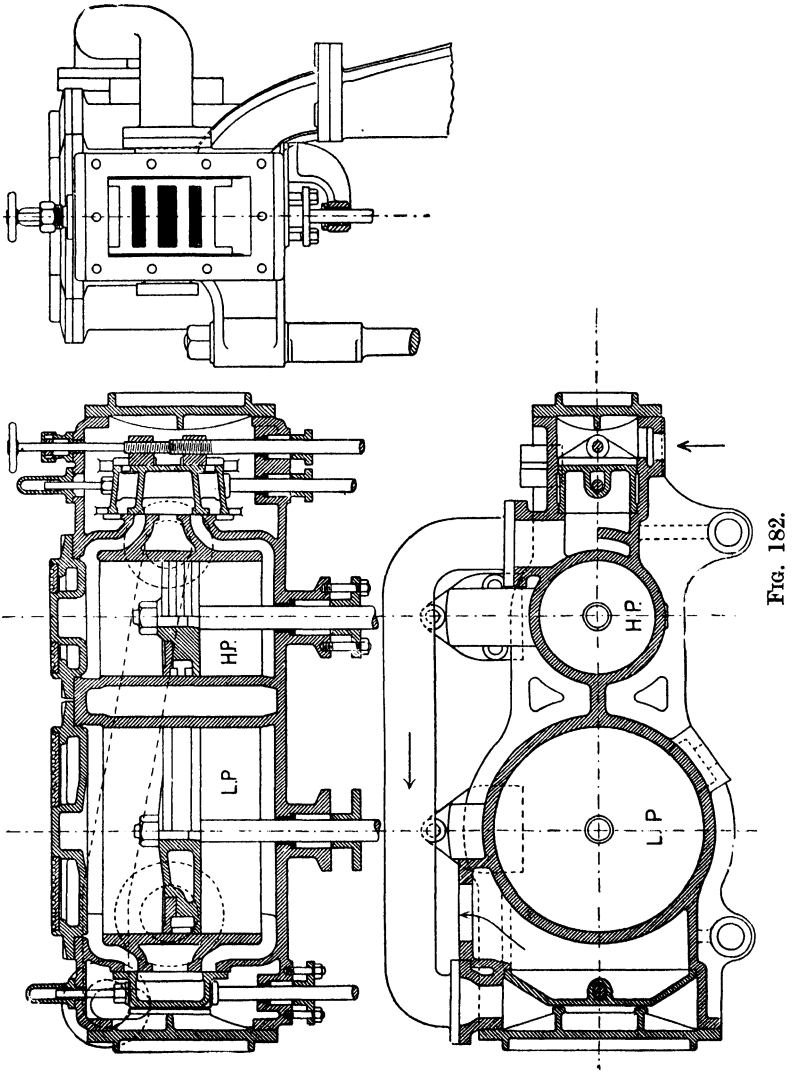



FIG. 182.

A sectional elevation, plan, and end elevation of the cylinders of the compound vertical engine on which the Meyer's gear (Fig. 181) is fitted to the H.P. cylinder is shown in Fig. 182. An ordinary D slide-valve is used for the L.P. cylinder.

**164. The Eccentric.**—In the majority of cases, a slide-valve is driven from the engine shaft by means of an eccentric and a rod connecting this to the valve spindle. This combination constitutes the simplest and most direct type of valve gear.

In some cases the motion of the eccentric is transmitted through a linkage which modifies the action of the valve. Also, when an engine has to run in either direction, that is, has to be capable of reversal, provision has to be made for the accompanying reversal of the valve motion. Usually this is accomplished by means of two eccentrics, one for forward and the other for backward running, in conjunction with a linkage which serves the double purpose of a reversing gear and an expansion gear.

In some types of radial reversing gears there is only one eccentric, in others none. The motion of the valve in the latter case can, however, be regarded as that which would be produced by an

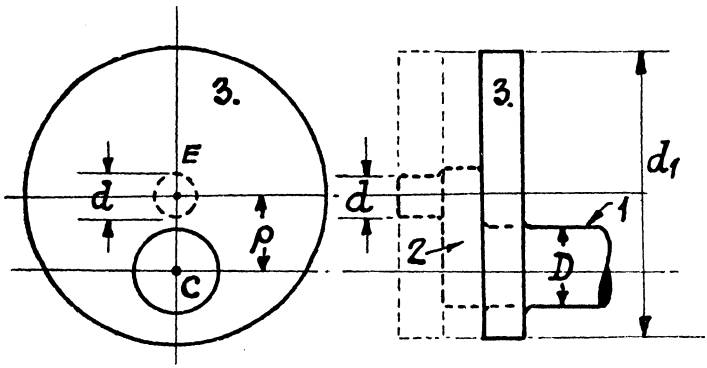



FIG. 183.

equivalent eccentric keyed to the shaft, and the valve design can be based on this equivalent.

An eccentric is simply a small crank with a very large crank pin which embraces the shaft. This will be apparent from Fig. 183. A shaft 1 having a diameter D is provided with a small crank arm 2, having a length $\rho$, and pin, of diameter $d$. If a connecting rod is attached to the pin and the shaft is rotated, the rod will have a total travel of $2\rho$.

Obviously if the diameter $d$ is increased to some value $d_1$ and the rod end enlarged to suit, the travel will not be affected, since $\rho$ remains the same. A disc 3 mounted eccentrically on the shaft can be substituted for this enlarged crank pin. The distance $\rho$ between the shaft centre C and disc centre E is now called the "eccentricity" of the eccentric; and when the disc is connected by a valve rod to the valve spindle the travel of the valve is given by twice the eccentricity, or $T=2\rho$.

In practice, this disc is called the eccentric sheave, and the end,

corresponding to the connecting rod end in the case of a crank, is called the strap. An illustration of one type of design of an eccentric sheave and strap is shown in side elevation and section in Fig. 184.

The sheave is in halves 1 and 2 which are fastened together by studs and cotters 3 and 4. In order to get in this fitting the lower half has to be made smaller than the upper one. The sheave is keyed in position on the shaft by a sunk key 5. As indicated in the section at 8, these halves are checked into one another, to prevent their axial displacement. The split sheave is necessary when the eccentric cannot be slipped over the end of the shaft.

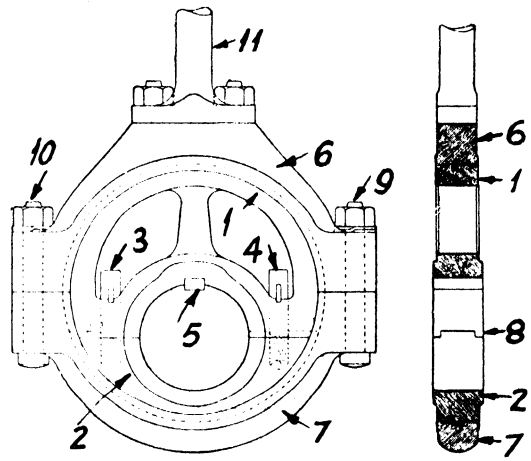



FIG. 184.

The strap has necessarily to be in halves 6 and 7. It is divided on a line through the sheave centre, and the halves are fastened together by bolts 9 and 10, having thimble points and split pins. Packing strips are usually inserted at the faces, for adjustment as the strap wears. The eccentric rod 11 is bolted to the strap, and pin-jointed at the other end to the valve spindle, when the drive is " direct."

The sheave may be of cast-iron or mild steel, and the strap of cast-iron, although it is usually made of steel and lined with an anti-friction metal. In some cases the strap is made of gunmetal. Although not indicated in Fig. 184, a lubricating oil cup and syphon has to be provided.

**165. Definitions relating to the Slide-Valve.**—Refer to Fig. 185, which shows sections of the cylinder steam and exhaust ports and the slide-valve in positions corresponding to the four positions of the engine crank $oC$, shown on the right—$oE$ is the eccentric crank, which is set at an angle $\phi$ considerably in advance of the engine crank, for a reason which will be explained later. It is assumed that the rod connecting the eccentric and the valve is so long relatively to the eccentricity $oE$ of the eccentric that the effect of its obliquity may be neglected, and the motion of the valve is the same as would be obtained if the eccentric pin E worked in the block of a slotted crosshead connected to the valve.

*Outside and Inside Laps.*—In position (*a*) the eccentric is at right

angles to the line of stroke, and it follows that the valve must be in the mid-stroke position on the face. It covers the outer or steam edge of the port by an amount O, and the inner or exhaust edge by an amount $i$, and the piston, which is still moving backward, is compressing the steam. This outer cover is called the outside lap

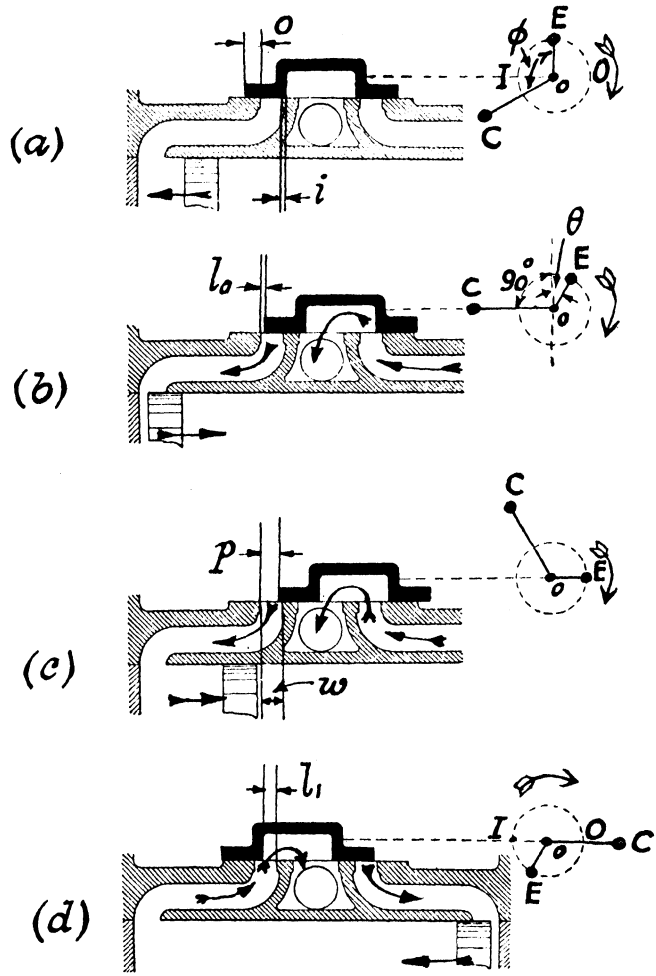



FIG. 185.

and the inner cover the inside lap. Sometimes, in the case of L.P. cylinders of marine engines, this inside lap is made negative.

*Lead.*—In position (*b*) the crank pin has reached the inner dead centre I, and the valve is displaced the amount of the outside lap plus a small amount $l_o$, which represents port opening to steam at

the beginning of the out-stroke of the piston. This is called the lead of the valve.

The lead of the valve depends principally on the piston speed. It runs from $\frac{1}{16}\rho$ in slow-speed to $\frac{1}{8}\rho$ in high-speed engines. It is generally made the same at each end in the case of a horizontal engine, but it is always greater at the bottom end of a vertical engine than at the top. The larger lead assists the cushioning, and reduces the inertia and gravity load on the piston. Also, it allows for wear on the gear links, which tends to reduce the lead value. In the case of a new vertical engine, sometimes the bottom lead is made twice the top value.

In position (*d*) the crank pin has reached the outer dead centre O and the valve has moved back and opened the crank end port by the amount of the corresponding lead. It has also, for this position of the piston, opened the head end port on the inside to exhaust by an amount $l_i$, which is called the inside or exhaust lead.

*Port Opening.*—In position (*c*) the eccentric crank *o*E is in the line of stroke again, and has thus displaced the valve to its extreme position on the right. The valve is therefore at the end of its travel and it has opened the port to steam by an amount $p$, which is less than the total port width $w$. This is called the port opening.

For a given length of port, the port opening depends on the mean velocity of steam chosen. In high-pressure cylinders this may run from 5000 ft./min. to 6500 ft./min., but in low-pressure cylinders it may run over 7000 ft./min. For marine practice Seaton gives H.P. 7500, I.P. 9000, L.P. 12000 ft./min.

In position (*d*) the head end port is already well open to exhaust, and before the eccentric crank *o*E reaches the inner dead centre position, it is obvious that the inside edge will be well over the outer edge of the port, so that the maximum opening to the exhaust from the cylinder port is the full port width $w$.

Again for a given port length this width is fixed by the mean exhaust steam velocity allowed. This may run from 5000 ft./min. with H.P. to 6500 ft./min. with L.P. cylinders. Seaton, again, gives for marine engines, H.P. 5800, I.P. 7200, L.P. 8600 ft./min.

In arranging the ports in the cylinder face care must be taken to make the central exhaust port sufficiently wide, so that when the valve is in the extreme " in " position, the exhaust opening between it and the exhaust port edge is sufficient to give an exhaust velocity, not less than is allowed in the port, based on the mean piston speed.

The method of calculating port areas on the basis of mean velocities of steam and piston is purely empirical. The values indicated above are those which experience shows give satisfactory results in practice.

*Valve Travel.*—While the eccentric crank *o*E rotates from its position at right angles to the line of stroke, in (*a*), to the line of stroke in (*c*), the valve travels a distance equal to the outside lap O plus the port opening $p$, equal to half the diameter of the eccentric crank

circle, or $\rho$. The half travel of the valve equal to the eccentricity of the eccentric is thus given by—

$$\rho=(O+p)$$

and the valve travel by—

$$T=2\rho=2(O+p) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

*Angle of Advance of the Eccentric.*—Suppose that in position (*b*) the engine crank and shaft were kept fixed at the inner dead centre position and the eccentric sheave were rotated back on the shaft to the position in (*a*) at right angles to the line of stroke. Then the valve would be returned to the central position it occupies in (*a*). The valve would thus be displaced backward by an amount equal to the lap O plus the lead $l_o$ for a rotation of the eccentric crank through an angle $\theta$.

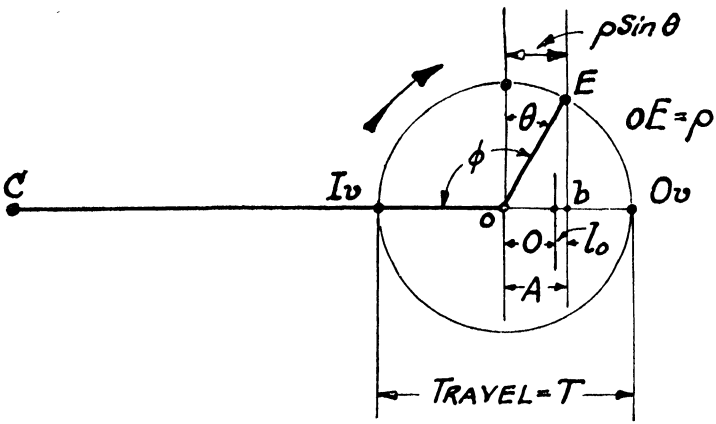



FIG. 186.

The total phase angle between the main crank and the eccentric crank, or $\phi$, is thus the sum of two angles, or—

$$\phi=(90+\theta)^\circ$$

This angle $\theta$ in excess of 90°, by which the eccentric crank leads or is in advance of the main crank, is defined as the "angle of advance" of the eccentric. It is obviously due to the lap and the lead of the valve. If there were neither lap nor lead, then the angle of advance would be zero. The valve face would just equal the port width, and no expansive working of the steam would be possible.

*Linear Advance of the Eccentric.*—As can be seen in Fig. 186, the horizontal projection of OE on the line of stroke is $Ob=\rho \sin \theta$, where $Ob=(O+l_o)=A$; and this is called the linear advance of the eccentric.

The student should note that for a fixed valve travel, that is a fixed eccentricity, the linear advance is the same for both ends of the valve.

**166. Piston and Valve Displacement.**—There are four positions of the piston which in each revolution have to be considered in the design of a valve gear.

On the indicator diagram they correspond, for the out-stroke, to the points of cut-off and release, and for the in-stroke to those of compression and admission, since in the latter case the valve, on account of the lead, opens the port to steam before the piston completes the in-stroke.

In Fig. 187, IO, on the left, is the stroke length L, I being the inner and O the outer dead centre. IO=L=2$r$, where $r$ is the length of the crank arm $o$C.

The angular positions of the crank at cut-off and release are $oC_2$ and $oC_3$. The corresponding positions of the piston (on the left) are obtained by striking arcs from $C_2$ and $C_3$ as centres, with the length $l$ of the connecting rod as radius, to cut the line of stroke in $c_2$ and $c_3$. I$c_2$ is the displacement of the piston from the inner

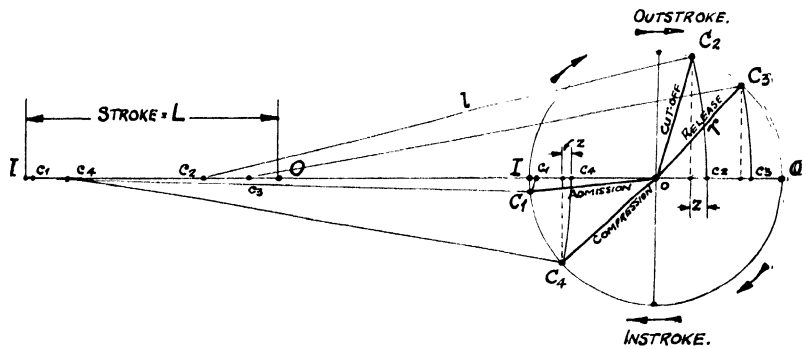



FIG. 187.

dead centre at cut-off, and I$c_3$ the displacement at release. The points of compression and admission $c_4$ and $c_1$ on the in-stroke are obtained in the same way.

For practical purposes the left-hand part of this diagram is not required. All the necessary points can be located on the diameter of the crank pin circle by drawing the arcs $C_2c_2$, $C_3c_3$, $C_4c_4$, and $C_1c_1$ through the respective crank pin centres with a radius equal to the connecting rod length, and centres on OI produced to the left. Actually these arcs need not be drawn. The connecting rod length $l$ can be marked off on the edge of a piece of paper and laid with the ends on the crank pin centre and the line of stroke. It can then be swung down with outer end on the stroke line as a centre until it lies along IO and the position of the other end can be marked on the line of stroke.

If the connecting rod were infinitely long these arcs would reduce to straight lines at right angles to the line of stroke, as indicated by the dotted verticals.

On the out-stroke the piston travels a greater and on the in-stroke a less distance than it would travel from the dead centre if the rod were infinite. This difference of travel $z$ is zero at the dead centres, and reaches a maximum near the mid-stroke. It increases as the ratio of connecting rod to crank, or $n=\frac{l}{r}$, decreases. This effect of the obliquity of the connecting rod must always be taken into account in getting out the design of a slide-valve.

In the case of the valve displacement, if instead of an infinitely long eccentric rod, a rod of finite length $l_1$, giving a small value of the ratio $l_1/\rho$ is used, then the same conditions hold as in the case of the piston displacement. When a drawing on a sufficiently large

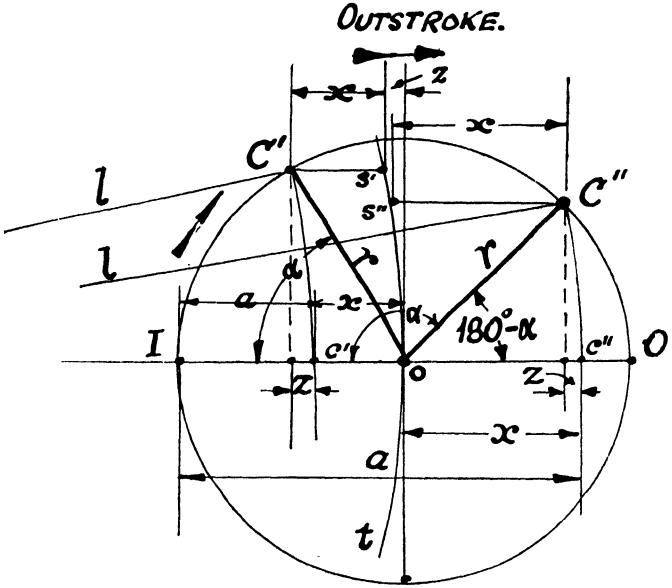



FIG. 188.

scale is made, it is possible to determine the piston displacements on the out- and in-strokes, with sufficient accuracy for all practical purposes, by drawing the arcs with the connecting rod length $l$ as radius, as indicated in Fig. 188 for the out-stroke and by Fig. 189 for the in-stroke, and scaling the corresponding displacement $a$.

Alternatively the displacement can be derived by the use of a single arc $s't$, drawn through the centre position $o$.

The horizontal intercept, such as C'$s'$ or C''$s''$, between the crank pin and the curve, is equal to the corresponding displacement $x$ of the piston from the mid-stroke position $o$.

The intercept between the vertical through $o$ and the curve also gives the corresponding value of the obliquity displacement $z$. The displacement of the piston is given by $a=(r\pm x)$, the sign being

determined by the position of the crank. It is negative if the crank arm, Figs. 188 and 189, falls in the second or fourth quadrant, and positive if the arm falls in the first or third.

**167. Analytical Calculation of Piston and Valve Displacement.**—The displacement of the piston, which is usually expressed as a fraction of the stroke, can be determined by the following expressions. It is not proposed here to deduce these, but to leave the work as a trigonometrical exercise for the student.

Referring to Fig. 188, it can be shown that on the out-stroke, for any angular displacement $\alpha$ of the crank arm $oC'$, from the inner

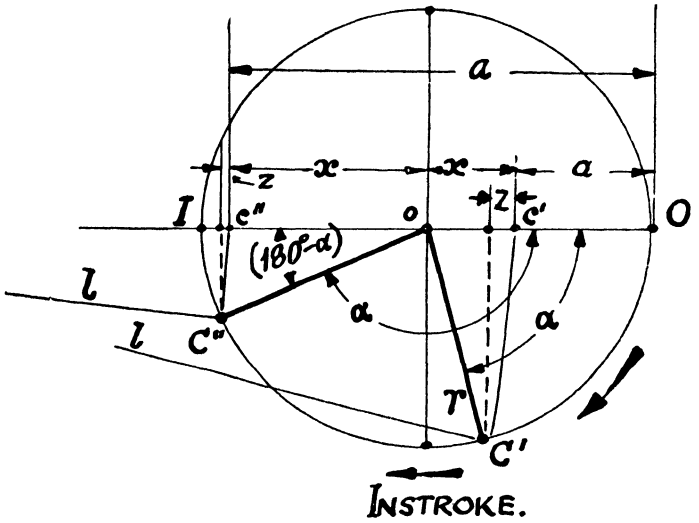



FIG. 189.

dead centre position $oI$, the corresponding piston displacement is given by—

$$a=r\left[1-\cos\alpha+\frac{\sin^2\alpha}{2n}\right] \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

where $\alpha<90°$;

and

$$a=r\left[1+\cos(180-\alpha)+\frac{\sin^2(180-\alpha)}{2n}\right] \quad . \quad . \quad . \quad (3)$$

when $\alpha>90°$.

Similarly, referring to Fig. 189, on the in-stroke for any angular displacement $\alpha$ from the outer dead centre position $oO$—

$$a=r\left[1-\cos\alpha-\frac{\sin^2\alpha}{2n}\right] \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

where $\alpha<90°$;

and

$$a=r\left[1+\cos(180-\alpha)-\frac{\sin^2(180-\alpha)}{2n}\right] \quad , \quad . \quad (5)$$

where $\alpha>90°$.

In each of these expressions the third term in the square bracket is the obliquity displacement $z$; the algebraical sum of the second and third terms is the displacement $x$ of the piston from the mid-stroke position.

Since the stroke $L=2r$, the fractional displacement of the piston is obtained by dividing the quantity in the square bracket by 2.

As regards the valve displacement from the centre, for any main crank angle $\alpha$ from the inner dead centre position, the corresponding eccentric crank displacement from its inner dead centre position $oI_v$, Fig. 190, will be $\delta=(\alpha+\phi)$. If $\delta<180°$, then the displacement of the valve from the mid position is—

$$x=\rho \cos (180-\delta) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

If $\delta>180°$, then—

$$x=\rho \cos (\delta-180) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

assuming an infinitely long valve rod. When $\alpha=0°$, then $\delta=\phi$, and $x=\rho \cos (90-\theta)=\rho \sin \theta$, and when $\alpha=180$, $x=\rho \sin \theta$.

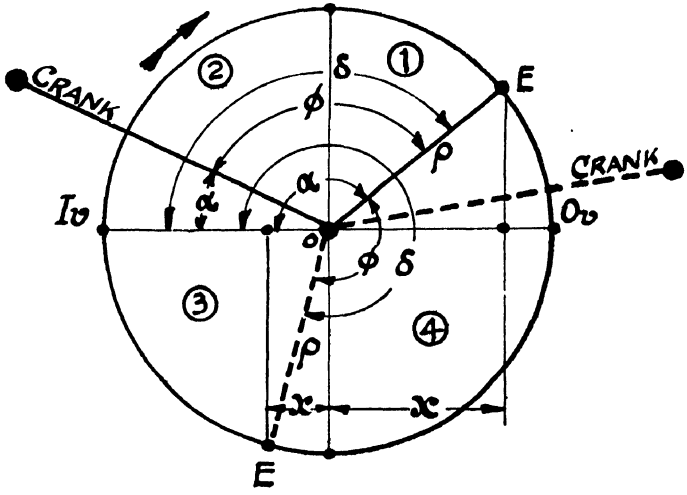



FIG. 190.

These equations also hold for crank positions during the in-stroke. The direction of the valve displacement will thus be positive, or to the right of the centre $o$, if the eccentric crank falls in the first (1) or fourth (4) quadrant, and negative, or to the left of the centre, if it falls in the third (3) or second (2) quadrant.

The displacement is expressed as a fraction of the valve travel by dividing the $x$ value by $2\rho$.

In the majority of cases the eccentric rod is long relatively to the eccentric crank length, and this method of determining the displacement on the basis of an infinitely long eccentric rod is sufficiently accurate. If, however, in any special case an exceptionally short

rod has to be used, the correction for the valve displacement can be made in the same way as for the piston displacement.

*Example* 1.—A horizontal engine has a connecting rod length of 4 × crank length. Cut-off takes place on the out-stroke when the crank angle is 109°, and on the in-stroke when it is 111°. The angle of advance of the eccentric is 37°. Calculate the fractional displacements of the piston from the inner and outer dead centres, and the fractional displacements of the valve from its mid position. Check the results by graphical construction.

Piston displacement on the out-stroke. Here $\alpha=109°$ and

$$\frac{a}{L}=\tfrac{1}{2}\left[1+\cos(180-\alpha)+\frac{\sin^2(180-\alpha)}{2n}\right]$$

$(180-\alpha)=71°$, $\cos 71=0{\cdot}3256$, $\sin 71=0{\cdot}9455$, $n=4$.

$$\therefore \quad \frac{a}{L}=\tfrac{1}{2}(1+0{\cdot}3256+0{\cdot}111)=\tfrac{1}{2}\times 1{\cdot}4366=0{\cdot}718 \text{ stroke.}$$

Piston displacement on the in-stroke.

$\alpha=111°$, $(180-\alpha)=69°$, $\cos 69=0{\cdot}3584$, $\sin 69=0{\cdot}9336$, $n=4$.

$$\frac{a}{L}=\tfrac{1}{2}\left[1+\cos(180-\alpha)-\frac{\sin^2(180-\alpha)}{2n}\right]$$

$$=\tfrac{1}{2}(1+0{\cdot}3584-0{\cdot}1089)=\tfrac{1}{2}\times 1{\cdot}2495=0{\cdot}6247 \text{ stroke.}$$

The mean value of the cut-off is thus 0·6713 or 67·13 per cent.

Valve displacement from the centre.

Out-stroke: $\theta=37°$, $\phi=(90+\theta)=127°$, $\alpha=109$.

$\delta=(\alpha+\phi)=236°$, $\delta>180$ $\therefore x=\rho\cos(\delta-180)$

$(\delta-180)=56°$, and $\frac{x}{T}=\frac{1}{2}\cos 56°=\frac{1}{2}\times 0{\cdot}5592$

$=0{\cdot}2796\times$ travel right of centre.

In-stroke: $\alpha=111°$ $\therefore \delta=(\alpha+\phi)=(111+127)=238°$

$\delta>180°$ $\therefore x=\rho\cos(\delta-180)$; $(\delta-180)=58°$

$\therefore \frac{x}{T}=\frac{1}{2}\cos 58=\frac{1}{2}\times 0{\cdot}5299$

$=0{\cdot}2649\times$ travel left of centre.

**168.** In Fig. 191 the main crank pin is shown on the inner dead centre I, where $\alpha=0$, and in this position the valve, with an infinite rod, would be displaced from the inner dead centre position $I_v$ of the eccentric crank $o$E a distance from its mid position equal to the linear advance A. With the rod of finite length $l_1$, owing to the obliquity, its displacement is increased by the amount $z$ to $a_1$ from the inner dead centre $I_v$, or $(A+z)=x$ from the mid position. When the eccentric crank pin E is at the inner dead centre $I_v$ or outer

centre $O_v$, the value of $z$ is zero. On the return stroke, between the outer and inner dead centres $O_v$ and $I_v$, the displacement is decreased by the amount $z$.

The displacement $x$ from the central position at any position of the main crank, which is the figure used in dealing with the valve design, can be obtained by setting the eccentric crank the total phase angle $\phi$ in advance of the main crank and striking the arc with $l_1$ as radius. In the usual case, with indefinitely long rod, the displacement is obtained by drawing the perpendicular from E to the line of stroke.

**169. Slide-Valve Diagrams.**—As a general rule for the design of a slide-valve the following provisional figures are available. The cut-off, which is fixed arbitrarily to suit the particular operative conditions of the engine; the lead, also arbitrarily fixed; and the port opening

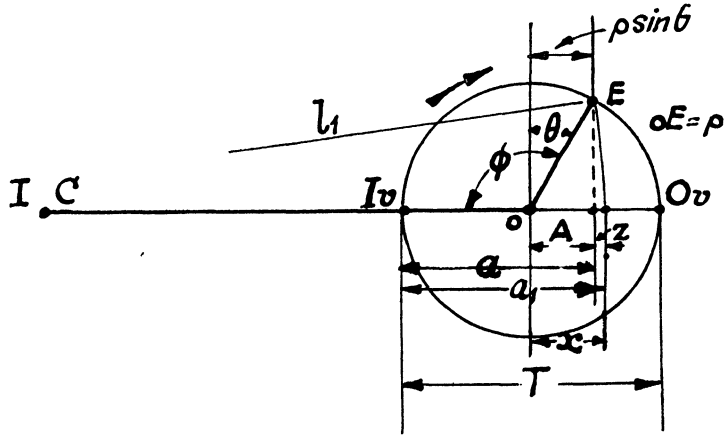



FIG. 191.

to steam, which is determined with reference to a suitable steam velocity. The unknown quantities which have to be found are the valve travel and the outside lap for the end of the cylinder for which the above quantities are given.

Once the travel is fixed, the inside lap for specified release and compression conditions can be determined, and vice versa.

The valve travel and steam lap can be found by a geometrical construction due to Hugo Bilgram, known as the Bilgram diagram, which is shown in Fig. 192.

**170. The Bilgram Diagram.**—The dotted crank pin circle $IC_{2I}O$ is drawn to any convenient scale, and the piston displacement $a_I$ from the inner dead centre I to give the required cut-off point $c_{2I}$ is set off. The arc $c_{2I}C_{2I}$ is struck with connecting rod length as radius, and the crank radial $oC_{2I}$ at cut-off angle $\alpha$ is drawn. A line $ab$ is drawn parallel to and above the line of stroke IO, at a distance equal to the chosen lead $l$, and an arc DHB is struck

from $o$ as centre with radius equal to the given port opening $p$. By trial, the centre E of a circle KHG which touches $oC_{2I}$, $ab$ and the arc DHB is found, and $o$E is drawn.

$o$E is the eccentricity of the eccentric, and the circle $EO_vI_v$ is the valve travel circle; $I_vO_v$=T=valve travel. The angle E$o$F=$\theta$, the angle of advance; EH=EG=outside lap O, and EF=A=linear advance of the eccentric.

The construction of Fig. 192 gives the valve travel, and the lead, lap, port opening and cut-off for the head end of the cylinder only. It is necessary to find the corresponding values for the crank end.

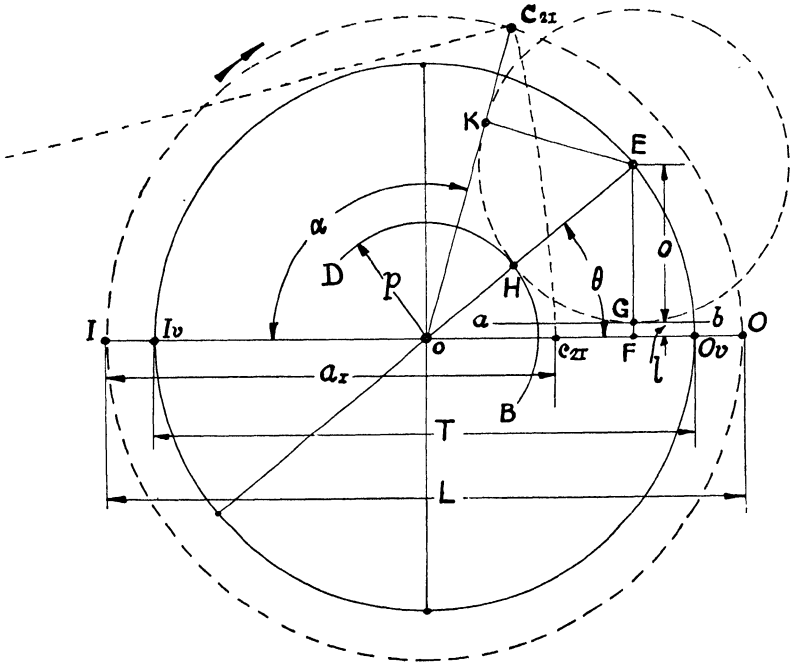



FIG. 192.

As already pointed out, the linear advance A is a constant quantity. Denoting the head and crank end values by subscripts 1 and 2 (Fig. 193) A=$E_1F_1$=($O_1$+$l_1$)=$E_2F_2$=($O_2$+$l_2$). The crank end lead $l_2$ being arbitrarily fixed, if $F_2G_2$ is set off below IO equal to $l_2$, then the corresponding outside lap is $O_2$=$E_2G_2$. Draw the lap circle $G_2H_2K_2$, cutting $E_1E_2$ in $H_2$, then $oH_2$ is the corresponding port opening $p_2$ for the crank end. The radial $oC_{2o}$ drawn tangential to the bottom circle at $K_2$, gives the position of the crank at cut-off on the in-stroke, and the arc $C_{2o}c_{2o}$, struck with the connecting rod length as radius, determines the cut-off point $c_{2o}$ and the piston displacement $a_o$ from the outer dead centre O.

If the inside lap is made zero, then the radial $oE_1$, Fig. 194, gives the crank position at release, on the out-stroke, and $oE_2$, the position at compression on the in-stroke. The corresponding release and compression points $c_3'$ and $c_4'$ are obtained by striking the arc from $C_3'$ and $C_4'$ with the connecting rod length as radius.

If positive inside lap $i$ is given at the head end, strike a circle from $E_1$ as centre with radius $i$, and draw the radial $oC_3$ through the intersection with the valve travel circle below $oC_3'$ ; $oC_3$ is the crank position when release takes place on the out-stroke, and $c_3$ the corresponding position of the piston.

The release is later with positive inside lap. If negative lap is given, then the radial is drawn to the lap circle at the top, and the

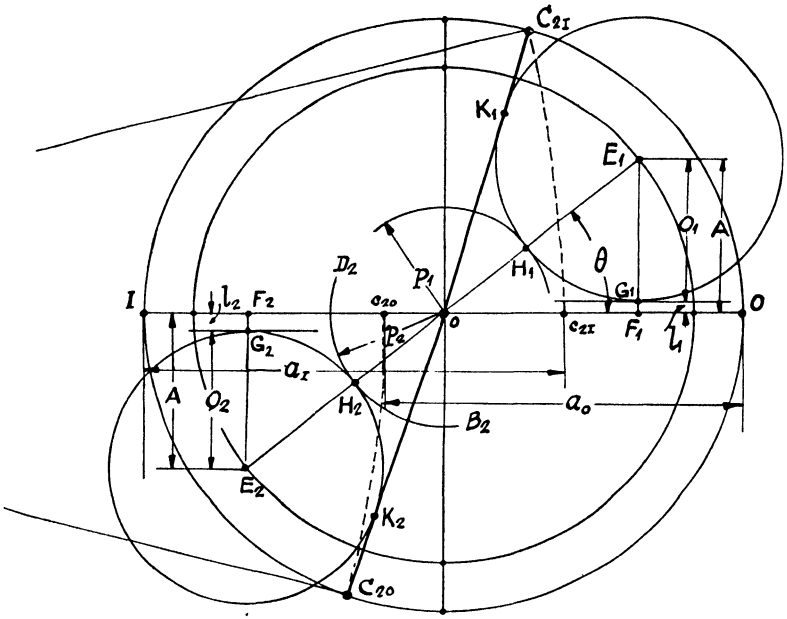



FIG. 193.

crank radial, as indicated by the dotted line, lies above $oE_1$. The corresponding release point on IO lies to the left of $C_3'$. The release is thus earlier with the negative inside lap.

Again draw the circle with radius equal to $i$ from $E_2$ as centre, and the radial $oC_4$ through the intersection below $oC_4'$. Then $oC_4$ is the crank position at compression, and $c_4$ the corresponding position of the piston on the in-stroke. If the lap is made negative the crank radial is drawn through the intersection on the valve travel circle above $oC_4'$, as shown by the dotted line. It will be evident here that with positive inside lap the compression is earlier and with negative inside lap it is later than compression with zero inside lap.

Finally there is the position of the crank and piston near the end of the in-stroke, when admission begins. At the inner dead centre of the eccentric crank, describe a circle with radius equal to the lead. Draw a line *gh* parallel to $E_2E_1$, touching this circle on the under side, and cutting the valve travel circle in *g*. Join *o* and *g*. This gives the crank position $oC_1$, at which admission commences.

The values of cut-off, release, compression and admission, obtained as above, are those for the head end of the cylinder only. The procedure has to be repeated, using the lap and lead for the crank end.

*Example* 2.—The port opening at the head end of a horizontal engine cylinder is to be 1 in., the cut-off fraction on the out-stroke is to be 0·71, and the lead $\frac{1}{8}$ in. Find a suitable valve travel, the

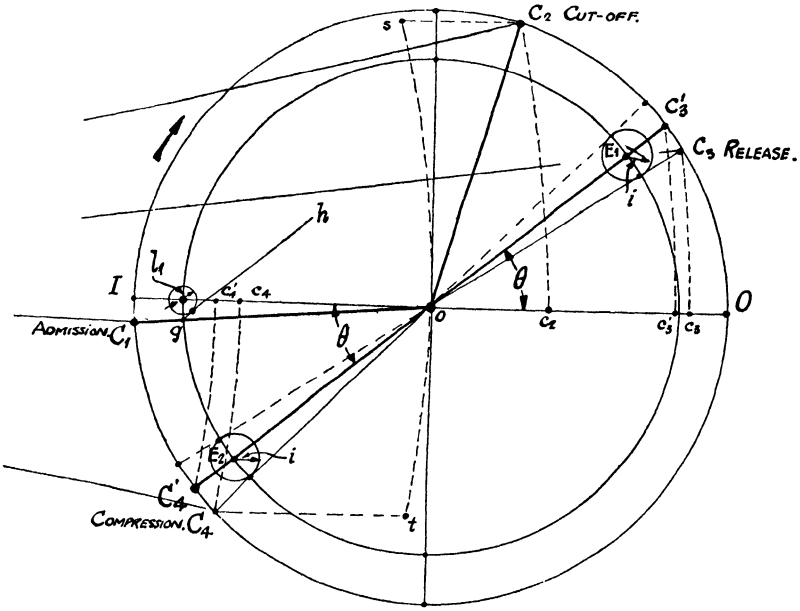



FIG. 194.

head end lap, and the points of release and compression if a positive inside lap of $\frac{1}{4}$ in. is given. The ratio of connecting rod length to crank is 4.

Referring to Fig. 192, draw the crank pin circle on a diameter of $5\frac{1}{2}$ in. Set off $FG=l=\frac{1}{8}$ in. and $a_1=0{\cdot}71\times5{\cdot}5=3{\cdot}9$ in. Lay off $l=4\times2{\cdot}75=11$ in. on the edge of a strip of paper and strike the arc $c_{2I}C_{2I}$ and draw the crank radial $oC_{2I}$.

With *o* as centre and $p=1$ in. as radius draw the arc DHB. Then find by trial the centre E of the circle KHG, and $OE=\rho=2{\cdot}3$ in., giving a valve travel of 4·6 in. The angle of advance is $\theta=39°$. The lap is $EG=(\rho-p)=(2{\cdot}3-1)=1{\cdot}3$ in.

At $E_1$ and $E_2$ with radius equal to the inside lap $i=\frac{1}{4}$ in. describe

the circles, Fig. 194, draw the crank radials $oC_3$ and $oC_4$ tangential to them, and strike the arcs, thus at release on the out-stroke $Ic_3=5{\cdot}2$ in. and at compression on the in-stroke $Oc_4=4{\cdot}57$.

Hence release takes place at $\frac{5{\cdot}20}{5{\cdot}5}=0{\cdot}945$ stroke, and compression at $\frac{4{\cdot}57}{5{\cdot}5}=0{\cdot}835$ stroke.

*Example* 3.—In the previous case find the points of cut-off, release, and compression for the crank end, if a lead of $\frac{3}{16}$ in. is given and the inside lap is kept at $\frac{1}{4}$ in.

The linear advance is $A=(1{\cdot}3+0{\cdot}125)=1{\cdot}425$ in.; $l_2=0{\cdot}1875$. Crank end lap $O_o=(A-l_2)=(1{\cdot}425-0{\cdot}1875)=1{\cdot}2375$ in.

Drawing the lap circle (Fig. 193) with $E_2G_2=O_o=1{\cdot}2375$ as radius and $E_2$ as centre, and the radial $oC_{2o}$ tangential to it, and striking the arc, the cut-off point on the in-stroke is $c_{2o}$ at $a_o=3{\cdot}45$ in. from the outer dead centre. The cut-off fraction is thus $\frac{3{\cdot}45}{5{\cdot}5}=0{\cdot}627$.

The mean cut-off for the two strokes is thus $\frac{0{\cdot}71+0{\cdot}627}{2}=0{\cdot}668$ stroke. To find the release point on the in-stroke draw the dotted radial in Fig. 194 tangential to the top of the inside lap circle at $E_2$, strike the arc to the line of stroke; and the displacement from the outer dead centre at release scales $a_o=5{\cdot}05$ in. This is $\frac{5{\cdot}05}{5{\cdot}5}=0{\cdot}918$ stroke. Drawing the dotted radial tangent to the top of the lap circle at $E_1$, and striking the arc, the piston displacement from the inner dead centre when compressions begins is $a_I=4{\cdot}9$ in. or $\frac{4{\cdot}9}{5{\cdot}5}=0{\cdot}89$ stroke. Since the lead at the crank end is increased from $\frac{1}{8}$ in. to $\frac{3}{16}$ in., the port opening at this end is reduced by $\frac{1}{16}$ in., or $p_2=(1-\frac{1}{16})=1\frac{5}{16}$ in.

The Bilgram construction has been considered first since it is the diagram which is commonly used when the valve travel has to be found.

It is, however, only one of several diagrams applicable to the valve design problem. Where the valve travel is one of the known quantities, either the Reuleaux or the Zeuner diagram is generally used in preference.

**171. The Reuleaux Diagram.**—In this instance the valve travel circle $C_2C_3C_4$ (Fig. 195) also serves for the crank pin circle. As in the case of the Bilgram, the line $E_2E_1$ is drawn at the angle of advance $\theta$, to the line of stroke IO. This is a datum line from which the valve displacements from the central position are reckoned. It is termed the "valve displacement line." At any position of the main crank such as $oC$ where the crank displacement from the line of stroke is $\alpha$, the perpendicular distance CN from C to the line $E_2E_1$ gives the displacement of the valve from its central position on the face. The

line $C_1C_2$ is drawn parallel to $E_2E_1$ at a distance equal to the outside lap $O_1$ for the head end, and the line $C_4C_3$ is drawn parallel to $E_2E_1$ at a distance equal to the positive inside lap $i_1$. If the lap were negative the line would be drawn above the centre $o$. Since the (lap+port opening)=half the valve travel, when the crank reaches the dotted position at right angles to $E_2E_1$ the distance $p_1$ will give the maximum opening to steam. This, in general, will be less than the port width W, as indicated by the line outside the valve circle parallel to $E_2E_1$. Where the lap line cuts the travel circle at $C_1$, $oC_1$ is the crank position at admission and the vertical distance from $C_1$ to $E_2E_1$ is simply the outside lap $O_1$.

As the crank rotates the port begins to open and at position OC

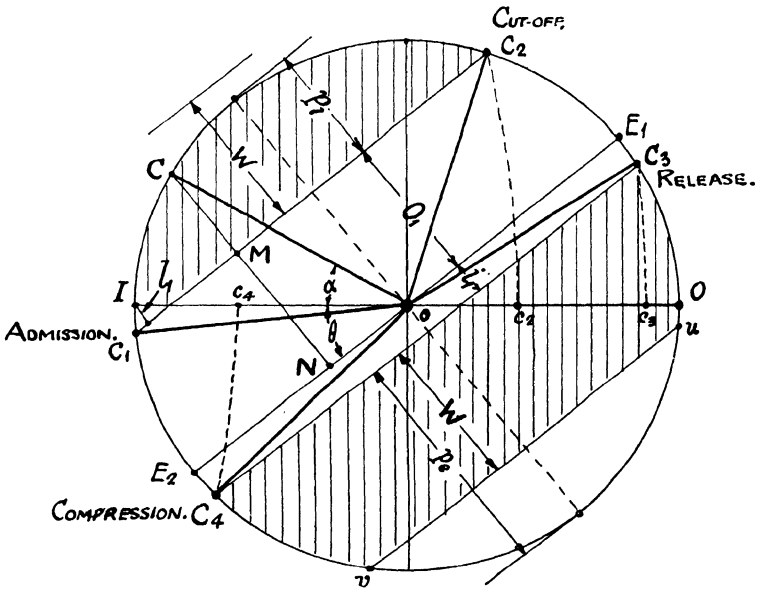



FIG. 195.

the port opening is CM. This increases to the maximum value $p_1$ and then decreases until at $C_2$, where the lap line again cuts the travel circle, it is zero ; $OC_2$ is the crank position at cut-off.

The point $C_3$, where the positive inside lap line cuts the valve travel circle, gives the crank position $oC_3$ at release, and the point $C_4$ gives the crank position $oC_4$ at compression. When the crank has reached the position shown by the dotted line at right angles to $E_2E_1$ on the in-stroke, the maximum displacement of the valve from the central position is $p_e$ and, as shown here, it is greater than the port width W. The full port width W is therefore open to exhaust between the crank positions at $u$ and $v$ on the in-stroke. If a perpendicular be drawn from the inner dead centre I to the lap

line $C_1C_2$ the length of this gives the lead at the head end, that is, a circle drawn with I as centre and $l_1$ as radius will touch $C_1C_2$. This is the method already employed to fix $C_1$ in Fig. 194.

As in the previous case (Fig. 194) the piston positions $c_1$, $c_2$, $c_3$, and $c_4$ have to be determined either by striking the arcs or by measuring the intercepts between the crank pin positions and the mid arc *st* and then setting them off from the centres. Thus at $oC_2$, $sC_2=oc_2$, and at $oC_4$, $tC_4=oc_4$.

The diagram, Fig. 195, as drawn applies only to the head end. For the complete set of design figures the process has to be repeated for the crank end, using the chosen lead and inside lap values. In this case the outside lap line $C_1C_2$ is drawn below the centre *o* and the inside lap line $C_3C_4$ above it.

The student should draw down the complete Bilgram and

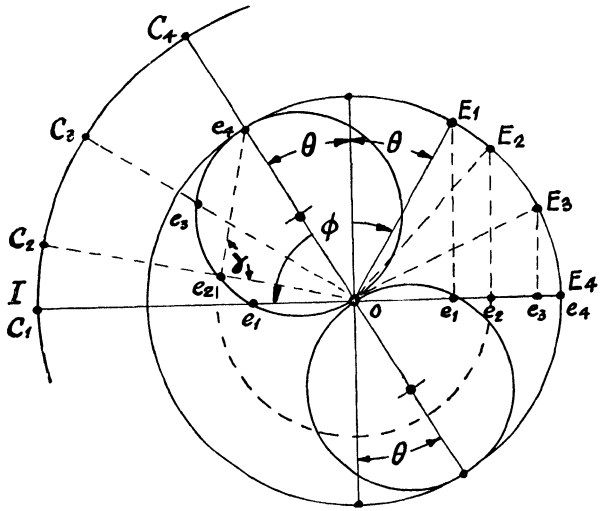



FIG. 196.

Reuleaux diagrams for the horizontal engine given in Examples 2 and 3, and compare the results.

**172. The Zeuner Diagram.**—This diagram, originated by Prof. Zeuner, is probably the most popular and widely used construction for valve design. It is of the polar type, the valve displacements from the central position being set off on the corresponding crank radials. The principle of this diagram is illustrated in Fig. 196.

$oC_1$ gives the position of the engine crank on the inner dead centre, and $oE_1$ the position of the eccentric crank.

As the shaft rotates and the engine crank takes up the positions $oC_2$, $oC_3$, $oC_4$, etc., the eccentric crank takes up the corresponding positions $oE_2$, $oE_3$, $oE_4$, etc. In position $oE_4$ the eccentric crank has reached its outer dead centre position, and hence the angular displacement of the engine crank from the vertical position at $oC_4$

will be $\theta$, the angle of advance of the eccentric. On the assumption that the obliquity of the eccentric rod can be neglected, the displacement of the valve from mid position corresponding to the crank positions are $oe_1$, $oe_2$, $oe_3$, $oe_4$. If these lengths are laid off from centre $o$, on the corresponding crank radials, they will be found to lie on a circle having the radial length $oe_4=\rho$ the eccentricity of the eccentric, as diameter.

This condition is easily proved. Consider the crank position $oC_2$. Join $e_4$ and $e_2$ and let the angle $e_4e_2o$ be denoted by $\gamma$. In the triangles $e_4oe_2$ and $E_2oe_2$, $oe_4=oE_2$, $\angle e_2oe_4=\angle e_2OE_2$, and the triangles are similar, hence $\gamma=90°$.

The angle at the circumference of a circle subtended by a diameter is always a right angle,* hence the point $e_2$ lies on the circle $e_1e_2e_4$ which has $oe_4$ as a diameter.

If, then, the eccentric crank arm is drawn with $\theta$ negative, that is, with $\theta°$ lag instead of advance on 90° from the dead centre, and a circle is described on it as diameter, the intercept for any crank position between the inner and outer dead centre positions I and O, between the centre $o$ and this circle, gives the displacement of the valve from its mid position. The displacements for crank positions between O and I, on the in-stroke, are obtained similarly, from the intercepts with the circle drawn on the lower half diameter of the valve travel circle.

In a case in which the ratio of the valve rod length to the eccentricity is small, the obliquity effect changes the displacement curves from circles into ovals. The curves have to be drawn by finding the displacements, as affected by the obliquity, and transferring the lengths to the crank radials.

It is very seldom, however, that this construction is necessary.

The complete Zeuner diagram for the head end is shown in Fig. 197. The crank radials with the subscripts 1, 2, 3, and 4 give, as before, the positions at admission, cut-off, release, and compression. The eccentric centre line $E_1E_2$ is drawn at $\theta°$ to the vertical (negatively). The line $gh$ drawn at right angles to $E_1E_2$ is tangential to the lap circle struck with radius $O_1$ from the centre $o$.

When the valve displacement circle is drawn on $oE_1$ as diameter it cuts the lap circle at $m$ on the crank radial $oC_2$ at cut-off, and at $n$ on the crank radial $oC_1$ at admission. At $n$ the valve is opening the port and at $m$ it is closing it. The intercept in the shaded field between the arc $nm$ and the valve displacement circle shows, at any crank position, the corresponding port opening to steam. It reaches a maximum when the crank arm coincides with $oE_1$.

As in the previous case, the line $gh$ is tangential to the lead circle struck from the inner dead centre of the valve travel.

The release and compression positions are controlled by the inside or exhaust lap, which may be either positive or negative. With positive lap, when the lap circle is struck from $o$ as centre with

* *Euclid*, Bk. III, proposition 31,

radius $i$, it cuts the lower valve displacement circle in $q$ on the crank radial $oC_3$ at release, and in $r$ on the crank radial $oC_4$ at compression. As the crank rotates from $oC_3$ to $oC_4$, the port opening to exhaust at any given position is the intercept in the shaded field between the inside lap circle $qvr$ and the irregular curve $qsutr$.

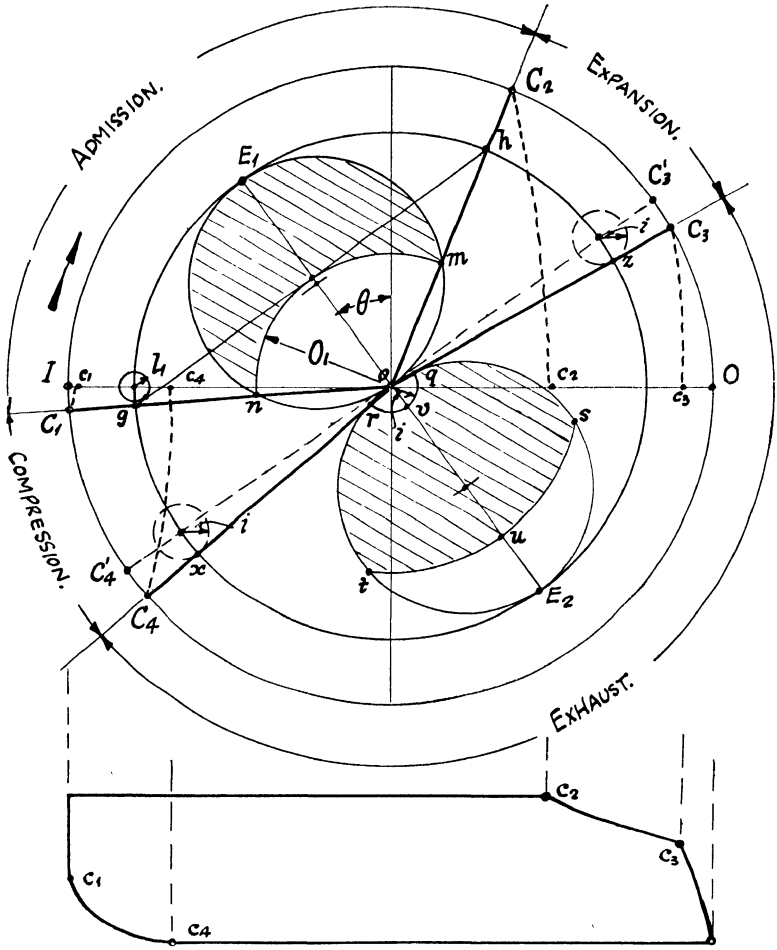



FIG. 197.

The arc $st$ is struck from $o$ with radius $(W+i)$, where W is the port width and equal to $vu$. The exhaust edge of the valve, at the extreme position, overruns the port by the amount $uE_2$.

When the inside lap, as is often the case with a vertical engine, is negative the crank radials $oC_3$ and $oC_4$ have to be drawn through the intersection of the lap circle and the upper valve displacement circle.

With a small lap it is usually difficult to obtain the correct positions $oC_3$ and $oC_4$ of the crank at release and compression. It is much more satisfactory to replace the original Zeuner method, given here, by that used on the Reuleaux diagram to ascertain these crank positions.

If a parallel to the displacement line $C_4'C_3'$ is drawn tangential to the inside lap circle it will cut the valve travel circle in $z$ and $x$ on the crank radials. These two points can obviously be obtained by drawing the line $C_4'C_3'$ through $o$ perpendicular to the eccentric line $E_1E_2$ and describing circles at the valve travel circle with radius equal to the inside lap $i$. The crank radials can then be drawn through $z$ and $x$. As shown here, with positive inside lap the radials fall below the displacement line $C_4'C_3'$. With negative lap they would fall above the displacement line.

The approximate form of the theoretical indicator diagram which this " setting " for the head end would give is shown below the diagram.

A similar valve diagram has to be drawn for the crank end. In practice this double procedure is not, as a rule, followed. The transference of the inside lap circles to the circumference of the valve travel circle enables the lower valve displacement circle to be used as the circle for the steam side at the crank end, and the complete set of particulars can be obtained from the single diagram.

This practical drawing-office method (really a combination of Zeuner and Reuleaux methods) is shown in Fig. 198, which is drawn for the general case of a vertical marine engine with unequal leads, positive inside lap at bottom (corresponding to crank end), and negative lap at top (corresponding to head end of a horizontal engine).

The valve travel circle is drawn full size, but for convenience in construction and to bring out the various points clearly, the crank pin circle is of less diameter than the valve travel circle, and the obliquity arc is drawn for each crank position. The fractional displacements of the piston are marked against the corresponding points on the line of stroke T.B.

This diagram gives the solution of the following example.

*Example* 4.—The low-pressure cylinder of a triple-expansion marine engine is 70 in. diam. and the stroke is 48 in.; the connecting rod length is 96 in.; and the engine is to run at 75 rev./min. It is to be fitted with a double-ported slide-valve (see Fig. 177), designed to give a mean cut-off about 0·7 stroke, a top release of 0·9 stroke, and a bottom compression 0·87 stroke. The arbitrarily chosen leads suitable for a large valve like this may be taken as $\frac{7}{16}$ in. top and $\frac{9}{16}$ in. bottom.

Assuming a port length of 54 in., and a mean exhaust velocity of 8600 ft./min., calculate a suitable port width; and allowing a mean steam velocity of 11,000 ft./min., the necessary top port opening.

Find a suitable valve travel by the Bilgram construction, using

this opening, $\frac{7}{16}$ in. top lead, and a provisional top cut-off at 0·75 stroke.

Then draw down the modified Zeuner valve diagram, find the mean cut-off, the top and bottom inside laps, the top compression and bottom release; and complete the design data listed in the table below.

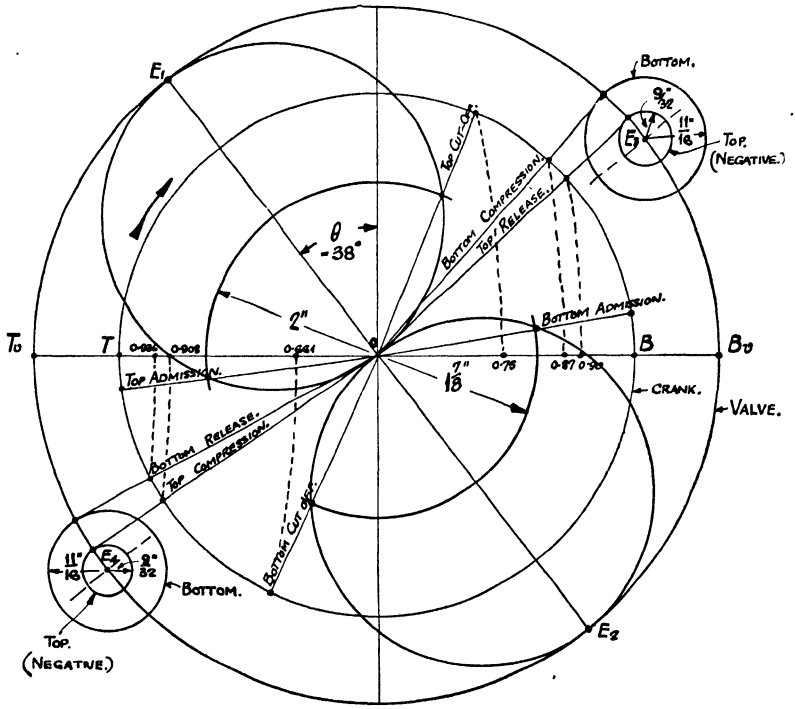



FIG. 198.

Port width.

Let A = cylinder area in in.$^2$
$a$ = divided port area in in.$^2$
$b$ = port length in ins.
$w$ = port width in ins.
$V_e$ = exhaust steam velocity in ft./min.
S = mean piston speed in ft./min.

Then since the total port area = $2a$

$AS = 2aV_e$; $a = wb$

$$\therefore w = \frac{AS}{2bV_e}$$

$$= \frac{3848{\cdot}5 \times 600}{2 \times 54 \times 8600} = .2\tfrac{1}{2} \text{ in.}$$

D = 70; A = 3848·5 in.$^2$
L = 4 ft., N = 75/min.
S = 2LN = 4 × 75 = 600 ft./min.
$V_e$ = 8600 ft./min.
$b$ = 54 in.

Port opening (top) with $V_s$=11,000 ft./min.—

$$p_1 = w\frac{V_e}{V_s} = \frac{2{\cdot}5 \times 8600}{11000} = .2 \text{ in.}$$

Drawing the Bilgram diagram (Fig. 192) with $p$=2 in. $Ic_{2I}$=0·75 stroke, lead FG=$\frac{7}{16}$ in., the eccentricity $o$E=4 in. or travel =T=8 in., the outside lap EG=2 in., and the angle of advance $\theta$=38°.

Using this travel value in the modified Zeuner construction (Fig. 198), draw $E_1E_2$ at $\theta$=38° and describe the valve displacement circles on $oE_1$ and $oE_2$. Also draw through $o$ the dotted line $E_4E_3$ at right angles to $E_1E_2$, cutting the valve travel circle in $E_4$ and $E_3$.

The linear advance is A=$\frac{7}{16}$+2=2$\frac{7}{16}$ in. Since the bottom lead is made $\frac{9}{16}$ in., the bottom lap is (A−$\frac{9}{16}$)=1$\frac{7}{8}$ in.

In Fig. 198 the valve travel circle is drawn full size, that is, with diameter 8 in., but for convenience in striking the obliquity arcs a smaller circle of 6 in. diameter has been used for the crank pin. The valve travel circle can be used for the crank pin circle if desired.

The lap arcs are struck from centre $o$ with radii 2 in. and 1$\frac{7}{8}$ in., the crank radials, for top and bottom admission and cut-off, are drawn, and the obliquity arcs are struck to the line of stroke T.B.

The top cut-off displacement of the piston is 0·75 stroke and the bottom cut-off 0·66 stroke. Hence the mean cut-off is 0·705 stroke, the required value. If this mean value were not satisfactory, then an alteration could be made in the top cut-off value, keeping the same travel and angle of advance. If this value is too high, then a reduction in the cut-off, as can be seen by inspection of the diagram, will increase the lap and decrease the lead on the top, and vice versa.

To determine the top inside lap, set off the displacement to release of 0·9 stroke, strike the obliquity arc to the crank pin circle, and draw the crank radial for top release. As shown, this falls above the level $E_4E_3$, so that the inside top lap is negative. Its value is $\frac{9}{32}$ in. Set this off at $E_4$, draw the crank radial for top compression, and strike the obliquity arc; and the displacement from the bottom centre, when compression commences on the top, is 0·908 stroke.

To determine the bottom inside lap, set off from the top dead centre T the compression displacement 0·87 stroke, strike the obliquity arc, and draw the crank radial. This also falls above $E_4E_3$, so that the lap is positive for the bottom end. Its value is $\frac{11}{16}$ in. Set this off from $E_4$, draw the crank radial for bottom release and the obliquity arc. The corresponding displacement from the bottom centre, at bottom release, is 0·936 stroke.

Since there is a decrease of $\frac{1}{8}$ in. in the bottom lap there is a

corresponding increase of port opening to $2\frac{1}{8}$ in. and a decrease of steam velocity to $\frac{11000\times 2}{2\cdot 125}=10{,}320$ ft./min.

DESIGN DATA. DOUBLE-PORTED MARINE SLIDE-VALVE

| | Top | Bottom |
|---|---|---|
| Lead | $\frac{7}{16}$ in. | $\frac{9}{16}$ in. |
| Outside lap | 2 in. | $1\frac{7}{8}$ in. |
| Inside lap | $-\frac{9}{32}$ in. | $+\frac{11}{16}$ in. |
| Cut-off, per cent. | 0·75 | 0·66 |
| Release | 0·90 | 0·936 |
| Compression | 0·908 | 0·87 |
| Port opening, ins. | 2 in. | $2\frac{1}{8}$ in. |
| Steam velocity, ft./min. | 11,000 | 10,320 |
| Exhaust „ „ | 8600 | 8600 |
| Angle of advance | 38° | |
| Linear advance | $2\frac{7}{16}$ in. | |
| Valve travel | 8 in. | |
| Cylinder diameter | 70 in. | |
| Stroke | 48 in. | |
| Connecting rod length | 96 in. | |
| Revs. per minute | 75 | |

This design data, which can be used in getting out the drawing of the valve, is based on a direct eccentric drive from the engine shaft. The drive is actually through a link motion, and the figures refer only to the case of "full gear" when the eccentric rod and spindle are sensibly in line. The effects of "linking up" such a gear on the operation of the valve are discussed in the section on link motions.

The three forms of valve diagram which have been discussed are sufficient for most design requirements. There are, however, two other diagrams which may be used. These are the harmonic or rectangular diagram and the oval diagram. The latter is serviceable only for the examination of the action of a valve gear already constructed.

**173. The Harmonic or Rectangular Diagram.**—This diagram, shown in Fig. 199, consists of a curve of piston displacement and a curve of valve displacement plotted to a base of crank angle. A line XX as a datum for piston and valve displacement from mid position is drawn to the right of the crank and eccentric circles, and the angles from 0 to 360 are set off to any convenient scale. The crank and eccentric circles are divided up by 30° sectors, giving twelve divisions on each circle, and marked consecutively, as shown, from the inner dead centre position for both crank and eccentric.

The obliquity arcs are drawn for the crank and the verticals for the eccentric on the assumption that the eccentric rod is indefinitely long.

The horizontal displacements of the piston from the centre are transferred to the corresponding verticals drawn through the angle

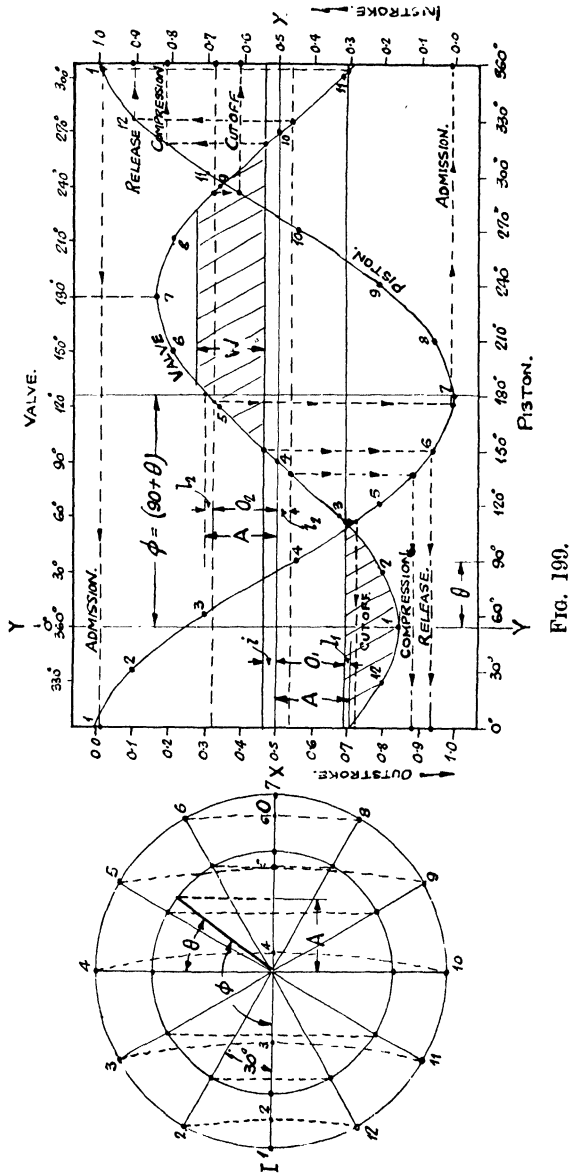



Fig. 199.

values on the lower angle scale, the measurements being set off from XX. The points on the piston displacement curve correspond to the numbered crank positions.

The curve of valve displacement is set off in the same way, using the valve scale of angles at the top of the diagram. The vertical YY passing through the zero value, or the eccentric angle on its inner dead centre, is obtained by setting off the value of the angle of advance $\theta$ to the left of 90° on the crank angle scale, or at $\phi=(90+\theta)$ from the 180° vertical of this scale.

The valve displacements for the successive angular positions 0°, 30°, 60°, etc., are then set off from XX, and the valve displacement curve is obtained. Since with an infinite rod the displacement varies as the cosine of the angle, the valve displacement curve is a curve of cosines. If the connecting rod were infinitely long, the piston displacement curve would also be a curve of cosines.

When these displacement curves are drawn in their correct relative positions the valve travel curve cuts the vertical through the angle 0°, the head dead centre value, at a distance below XX equal to the linear advance A. It also cuts the vertical through the 180° angle, or the crank end dead centre value, at the distance A above XX.

A line (full) drawn below XX at a distance equal to the head outside lap $O_1$, cuts the valve displacement circle at the cut-off crank angle, and, on the extreme right, at the admission angle for the out-stroke.

A line (full) drawn above XX at a distance equal to the head inside lap $i$ cuts the valve travel circle on the vertical through the crank angle at release on the out-stroke, and on the extreme right it cuts the valve displacement curve on the vertical through the crank angle at compression on the in-stroke. The vertical intercepts in the lower shaded field give the port opening to steam, and those in the upper shaded field the openings to exhaust. The port width is W.

When the outside lap line (dotted) for the crank end is drawn above XX, and the inside lap line (dotted) is drawn below XX, the admission, cut-off, release, and compression values for this end are determined, in the same way, as indicated by the dotted directional lines.

The corresponding fractions of the stroke are obtained by drawing horizontals from the piston displacement curve to the out-stroke and in-stroke verticals.

When the valve displacement curve is drawn on a piece of tracing paper, and moved on the diagram to vary the angle of advance $\theta$, the effect of this variation on the valve events can be studied.

This diagram, although serviceable for the analysis of some types of valve gear motion, involves a tedious amount of work, and for practical purposes of design is not to be compared with the simple polar diagram of Zeuner, which consists of straight lines and circles.

**174. The Valve Ellipse or Oval Diagram.**—This diagram, which has been much used by locomotive engineers, can be applied

to study the effects of various adjustments of a gear already constructed.

In this case the valve displacement is plotted to a base of piston displacement. The result is an oval form of closed diagram, which would be an ellipse if the connecting rod were infinitely long. Hence the term "valve ellipse" given to the diagram.

A graphical method for its construction is shown in Fig. 200. The horizontal line XX represents the line of piston motion and the vertical line YY the line of valve motion. The crank pin circle is drawn on the stroke length IO as diameter. The valve travel circle is drawn on the valve travel length $E_1E_2$ as a diameter, $E_1E_2$ being set off at the total phase angle $\phi=(90+\theta)$ from the vertical crank position $o$Y. The crank pin circle is divided into twelve equal parts, starting at the inner dead centre with point 1. The

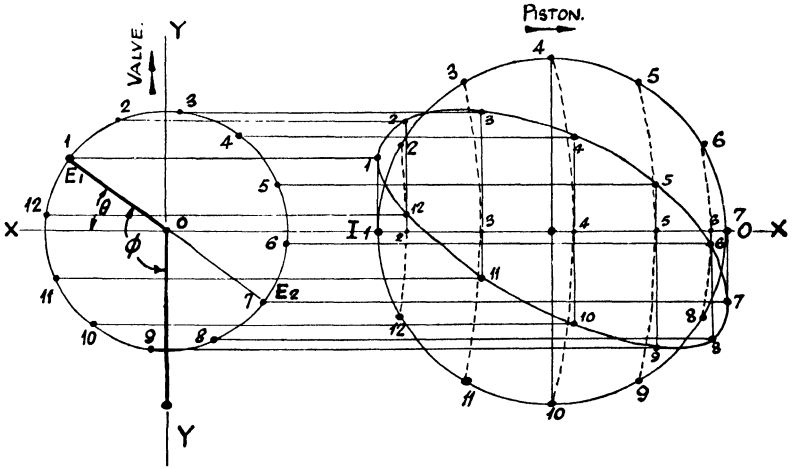



FIG. 200.

valve travel circle is likewise divided into twelve equal parts, starting at $E_1$ with point 1.

The obliquity arcs are struck from the crank pin positions and verticals drawn through the corresponding points 1, 2, 3, etc., on the line of stroke. Horizontal projectors are drawn from points 1, 2, 3, etc., on the valve travel circle to intersect the corresponding verticals from the line of stroke. The oval curve shown is the locus of these points of intersection.

This crank pin circle and the oval diagram are reproduced in Fig. 201, free from the complication of the construction lines. A line *ab* is drawn above IO at a distance equal to the outside lap $O_1$ at the head end, and cuts the curve at *a* and *b*. Projectors from these points to IO give the head end admission and cut-off positions. The vertical distance between *ab* and the curve in the shaded field

at any piston position gives the corresponding port opening to steam. The maximum opening is $p_1$.

A line *cdg* is drawn below IO at a distance equal to the positive inside lap $i_1$ for the head end, and cuts the curve at *c* and *d*. Projectors from these points to the line of stroke IO give the head end release and compression positions.

A line *ef* is drawn below *cd* at a distance equal to the port width W. Between release and the end of the out-stroke, that is, for any piston position between *d* and *g*, the vertical intercept between *dg* and the curve gives the amount the port is open to exhaust. On the in-stroke between *f* and *e* the full port width W is open to exhaust. The opening gradually reduces till at *c* the port is closed. For the in-stroke, therefore, the port opening for any piston position is given by the vertical intercept between *cg* and the irregular curve *cef*. The distance between the point *a* and the vertical through I gives the head end lead $l_1$. From the nature of the curve at this end it is difficult to get an accurate value of $l_1$. This is one of the drawbacks of this method. These "settings" refer only to the out-stroke, or for the head end of the cylinder.

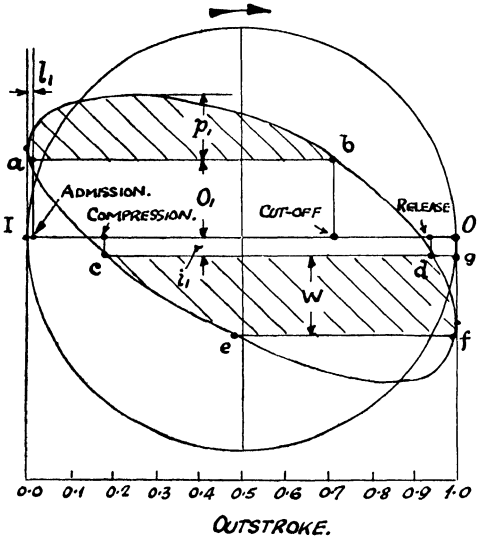



FIG. 201.

The crank end outside lap $O_2$ has to be set off below, and the inside lap $i_2$ set off above IO, and the process has to be repeated for the crank-end values. In Fig. 201 these lines have been omitted to prevent confusion.

**175. Expansion Valve Gears.**—Referring to Fig. 197, which shows the Zeuner diagram for the head end of the cylinder of a horizontal engine, suppose it is desired to cut-off earlier than the fraction of the stroke shown, the same lead and inside lap being retained. The outside lap will obviously be increased, and hence with it the linear advance A and the angle of advance $\theta$. Since $C_4'C_3'$ is perpendicular to $E_1E_2$, then the angle $IoC_4'$ will be correspondingly increased, and with the same value of the inside lap *i*, compression will occur earlier on the in-stroke. This is one of the drawbacks in the use of a single slide-valve for steam distribution with an early cut-off. Another is the reduction of port opening and the excessive

wire-drawing which results when the angle of advance $\theta$ is kept constant and the earlier cut-off is obtained by increasing the outside lap.

In practice it is found desirable to limit the cut-off to about half stroke, to avoid excessive compression and restricted port opening.

When an earlier cut-off is required in the case of a stationary engine running in one direction, some form of expansion gear is employed. This is usually in the form of a separate expansion valve, driven independently from the shaft, either directly or through some form of linkage.

Alternatively, as is often the case with slow-speed mill engines, the Corliss or drop-valve arrangement, with separate steam and exhaust valves, may be used.

In the case of an engine—marine, locomotive or winding—which must be capable of running in either direction, that is, which has to be mechanically reversible, some form of link motion has to be used, and this is also arranged to act as an expansion gear.

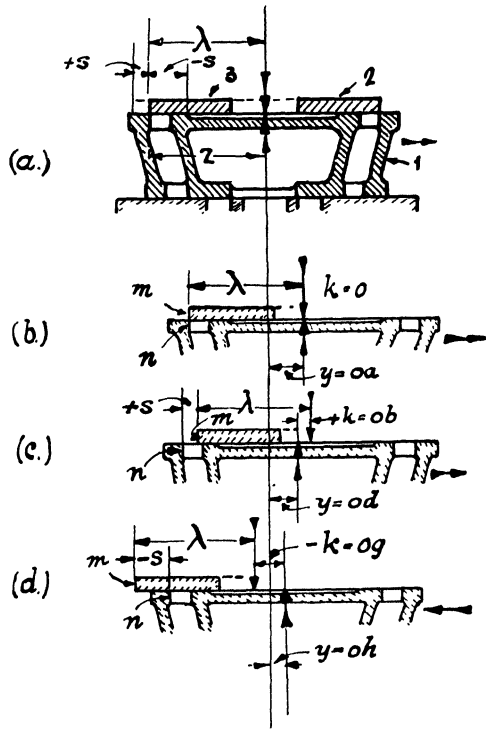



FIG. 202.

**176. The Meyer Valve Gear.**—The most common form of expansion gear used on stationary engines with slide-valves is the Meyer, already described in Art. 163. The principle of this gear is best illustrated by the Zeuner polar construction, although the Reuleaux and rectangular constructions can also be applied to it. The main valve 1 (Fig. 202 (*a*)) is driven from an eccentric on the shaft as an ordinary slide-valve, and the expansion valve, shown diagrammatically in the form of two adjustable plates 2 and 3, is driven by a second eccentric on the shaft which is set in advance of the " main " eccentric, driving the valve 1.

In Fig. 203, *o*C is the inner dead centre position of the crank; *o*E is the main eccentric, with an eccentricity $\rho$ and an angle of advance $\theta$; OE′ is the expansion eccentric, with a somewhat

greater eccentricity $\rho'$, usually about $1{\cdot}2\rho$, and $\theta'$ is its angle of advance.

The latter is often made 90°. This arrangement is necessary when the engine has to reverse. The distance EE′ is a fixed quantity for this eccentric arrangement.

When the crank is on the dead centre $o$C, the main valve is displaced a distance $oe$ from the mid position and the expansion valve is displaced a distance $oe'$. The difference $z=ee'$ is the displacement of the expansion valve relatively to the main valve. When the crank rotates to position $o\mathrm{C}_2$, the line EE′ is vertical, and hence

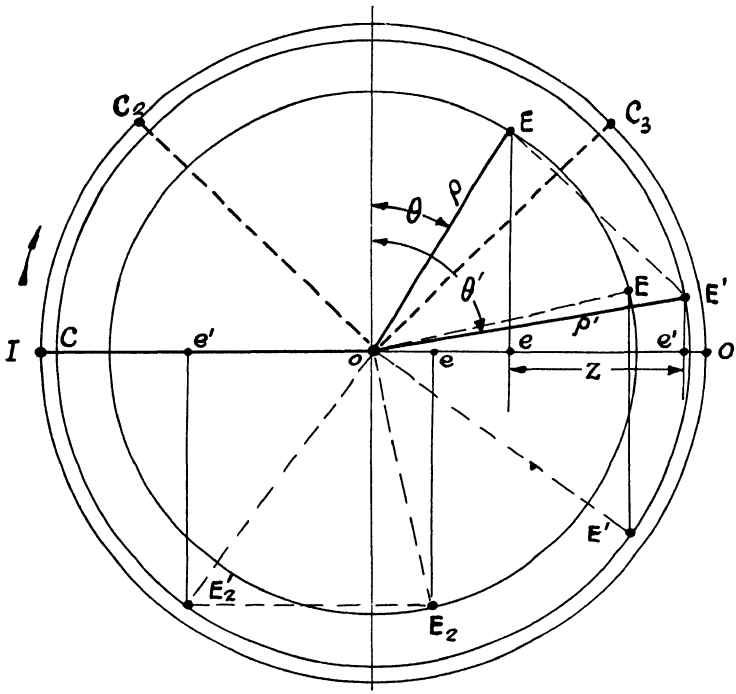



Fig. 203.

$z=0$, that is, the centres of the two valves coincide. When the crank reaches the position $o\mathrm{C}_3$ the line $\mathrm{E}_2'\mathrm{E}_2$ joining E and E′ is horizontal and $z=\mathrm{E}_2\mathrm{E}_2'$. This is obviously the maximum value of the relative displacement $z$. In moving from $o\mathrm{C}_2$ to $o\mathrm{C}_3$, therefore, the relative displacement of the valves varies from $o$ to $\mathrm{E}_2\mathrm{E}_2'$, and $\mathrm{E}_2\mathrm{E}_2'$ is thus the half travel of the expansion valve relatively to the main valve which could be produced by a virtual or equivalent eccentric with eccentricity $\mathrm{E}_2\mathrm{E}_2'=\rho''$ and angle of advance $\theta''$.

As already pointed out, with the Zeuner method of construction the angle of advance has to be set off negatively. This arrange-

ment is shown in Fig. 204. At the inner dead centre the relative valve displacement is $ee'$. If this is set off from $o$ on the crank radial the point $f$ is obtained. If this process is repeated for a number of crank positions in the third quadrant, the locus of the points will be a circle $of$E″, having a diameter $o$E″=EE′, with its centre on the line E″E‴, drawn through $o$ parallel to the direction of EE′. The line OE″ is the virtual eccentric for relative displacement, and $\theta''$ is the corresponding angle of advance. OE″ is readily obtained by completing the parallelogram $o$EE′E″. When the crank reaches the position in which it is at right angles to E″E‴, the radial will be tangential to the relative displacement circle, and the valve centres will be coincident. If the displacements are again

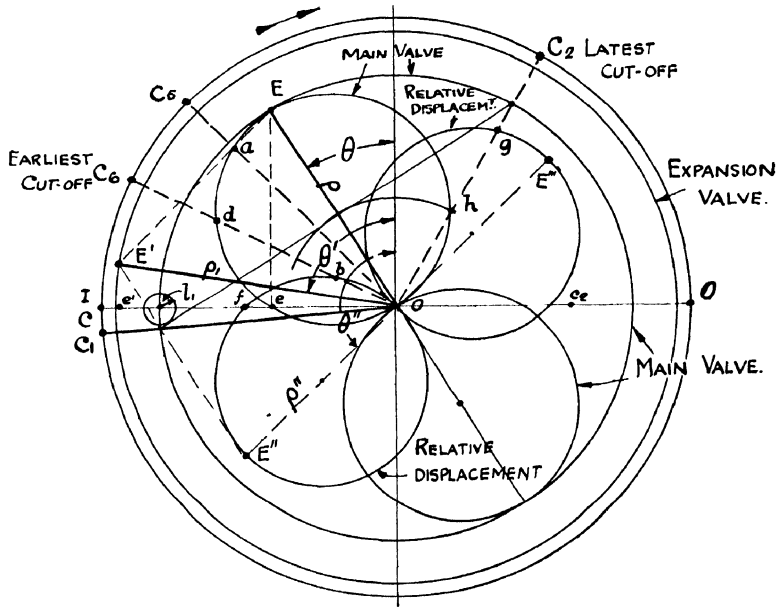



FIG. 204.

set out as the crank sweeps through the first quadrant, another relative displacement circle $Og$E‴ on diameter $o$E‴ is obtained. The intercepts on the lower circle give displacements of the expansion valve to the right of the centre of the main valve, and those on the upper circle, the displacements to left of the centre.

Let $oC_5$ be the position of the crank radial which is a common tangent to the two relative displacement circles, or a line perpendicular to EE′. In this position the main valve (Fig. 202 (*b*)) is displaced a distance $y=oa$ from the central position, but the relative displacement of the two valves is zero. Hence the centre line of the expansion valve coincides with that of the main valve, as indicated by the arrow lines. If the steam is to be cut off when the crank is in

position $oC_5$, it is obvious that the outer edge $m$ of the expansion valve must just coincide with the outside edge $n$ of the steam port in the main valve. The mid position of the two valves relatively to the cylinder face is shown in Fig. 202 (*a*), and the expansion valve has therefore neither positive nor negative lap. The length from the expansion valve centre to the edge $m$ is denoted by $\lambda$.

If now the expansion valve is to cut off steam from the main valve at some position $oC_6$ of the crank in the second quadrant, then the displacement of the main valve, as shown by the main displacement circle (Fig. 204), will be $y=od$, while the positive displacement of the expansion valve centre from the main valve centre, as shown by the relative displacement circle, will be $+k=ob$. When the expansion valve centre (Fig. 202 (*c*)) is displaced by this amount to the right of the main valve centre, the edge $m$ falls short of the steam port edge $n$ by an amount +S. Hence if the valve is to close the port at crank position $oC_6$, an amount of positive lap $S=+k$ has to be given to the expansion valve, as indicated in Fig. 202 (*a*). If now the latest cut-off by the expansion valve is to be arranged to coincide with the cut-off by the main valve, the crank radial position will be $oC_2$ (Fig. 204).

For this position the main valve centre has still a positive displacement $oh$, but the expansion valve has a negative displacement relative to the main valve centre of $-k=og$. When the expansion valve is displaced by this amount, the edge $m$ overruns the edge $n$ of the steam port by an amount −S (Fig. 202 (*d*)), so that in order to cut off at crank position $oC_2$ the distance from the centre of the expansion valve to the edge $m$, or $\lambda$, has to be shortened by −S, or the expansion valve has to be given a negative lap of the amount $-k=og$, as indicated in Fig. 202 (*a*).

In order to bring out the possibility of positive laps the crank radial $oC_6$ has been taken in the second quadrant for a very early cut-off, in the case shown about 10 per cent. of the stroke. It is seldom that the earliest cut-off is made less than 20 per cent., and for this condition, in this instance, the crank radial $oC_6$ would cut the upper displacement circle and the lap would have to be negative. This is the usual condition ; but the student should bear in mind that the lap of the expansion valve is not necessarily always negative.

The settings of the main valve should first be determined by the Zeuner construction and then the particulars of the expansion valve should be determined. These refer only to the head end, and the procedure has to be partly repeated to determine the values for the crank end.

Whatever may be the sign of the lap, if $\lambda_1$ is the distance from the expansion valve centre to the edge $m$ for the earliest cut-off and $\lambda_2$ the distance for the latest cut-off, the total displacement of the expansion plates which has to be produced by rotation of the screwed spindle will be $(\lambda_1-\lambda_2)$. It is thus a simple matter to produce a cut-off scale on which the cut-off for any setting of the spindle

can be read by means of a pointer on the sleeve nut operated by the hand-wheel. This index scale and pointer are not indicated in Figs. 181 and 182.

When the Reuleaux diagram is used in place of the Zeuner, the procedure is as follows.

Draw the main valve travel circle and proceed with the main valve design to obtain the outside laps for the required maximum cut-off and the desired release and compression conditions.

In Fig. 205 only the displacement and outside lap lines $E_2E_1$ and $C_1C_2$ for the head end are shown. The expansion valve travel circle and the eccentric line are drawn and the parallelogram

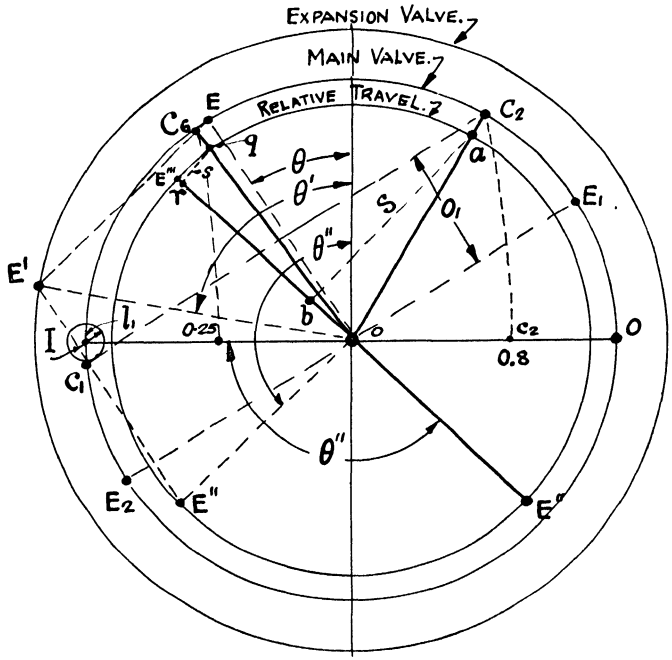



FIG. 205.

$oEE'E''$ is completed to obtain $oE''$ and the virtual angle of advance $\theta''$. The relative travel circle is drawn with $oE''$ as radius, the radial $oE''$ is set off from the horizontal IO at $\theta''$, and the diameter $E''E'''$ is drawn.

At the latest cut-off, $oC_2$ cuts the relative travel circle at $a$, and the perpendicular $ab$ between $a$ and $E''E'''$ gives the negative lap required for this cut-off. It is equal to $og$ in Fig. 204. For any other cut-off position the perpendicular to $E''E'''$ gives the corresponding lap. If the crank pin position is above $E''E'''$ this lap is negative; if it is below, the lap is positive. If $oC_6$ is the earliest cut-off, then $qr$ is the lap S, still negative, which is required. This

can be more accurately determined on the Reuleaux diagram than on the Zeuner, as in this position it is very difficult to locate the exact point of intersection of the crank radial with the relative displacement circle. An inspection of both diagrams will show that if the crank radial falls below E″E‴ the intercept will take place in the lower displacement circle. In fact, E″E‴ (Fig. 205) corresponds to the common tangent of the two displacement circles in Fig. 204.

Between the positions $oC_1$ and $oC_2$ of the crank the port in the cylinder face is uncovered by the main valve, and for earlier cut-off than at $oC_2$ by the expansion valve the width of plate must be such that it does not overrun the inside edge of the port in the main valve and re-admit steam.

If a cover of value $c$ is given at the inside edge and the port width is W, then the width of the plate should be—

$$w=(W+c+\rho''\pm S)$$

where S is the positive or negative lap for the earliest cut-off and $\rho''=EE'$.

The distance from the centre of the main valve to the outside edge of the steam port (Fig. 202 (*a*)), is given by—

$$z=(S+w+d)$$

where S is the negative lap for latest cut-off, and $d$ is half the distance allowed between the inner edges of the plates when these are adjusted to the full " in " position.

*Example* 5.—The main valve of a Meyer's expansion gear is to have a travel of 4 in. ; the head end lead is to be $\frac{1}{8}$ in., and the latest cut-off 0·8 stroke.

The expansion valve travel is to be 20 per cent. greater than the main valve travel, and the plates are to be arranged to vary the cut-off from 0·25 stroke to 0·8 stroke ; the angle of advance of the expansion eccentric is to be 80°. The connecting rod length is 4 × crank length.

Find the angle of advance of the main eccentric and the outside lap to give the latest cut-off, and the lap values of the expansion valve required to give the limiting values of the cut-off. Allowing an inside cover of $\frac{1}{4}$ in. and a port width of $1\frac{1}{4}$ in. determine the minimum width of the expansion plate for the head end, and the distance from the centre of the main valve to the outside edge of the steam port. Allow a central clearance between the plates of $\frac{1}{4}$ in.

Referring to Fig. 205, draw the lead circle with radius $l_1=\frac{1}{8}$ in., $Ic_2=0{\cdot}8\times4=3{\cdot}2$ in. The line $oE$ drawn perpendicular to $C_1C_2$ gives $\theta=31\frac{1}{2}°$, and the outside lap $O_1=0{\cdot}92$ in. Draw the expansion valve travel circle with radius $\rho'=1{\cdot}2\rho=1{\cdot}2\times2=2{\cdot}4$ in. and $oE'$ at $\theta'=80°$ to the vertical, and $EE'=\rho''=1{\cdot}81$ in. Complete the parallelogram $oEE'E''$, and $\theta''=134°$. Draw E″E‴ at $\theta''=134°$ to the horizontal and describe the relative travel circle with

radius $\rho''=1{\cdot}81$ in. Locate the crank radial $oC_6$ for cut-off at 0·25 stroke. This cuts the relative travel circle at $q$ above E″E‴, showing that the plate lap is negative. Draw $qr$ perpendicular to E″E‴, and S=0·35 in. The crank radial $oC_2$ at latest cut-off cuts the relative travel circle at $a$. Draw the perpendicular $ab$ to E″E‴, and the maximum negative plate lap is S=1·75 in.

Minimum width of plate—

$w=(W+c+\rho''-S)$
$=(1{\cdot}25+0{\cdot}25+1{\cdot}81-0{\cdot}35)=(3{\cdot}31-0{\cdot}35)$
$=2{\cdot}96$, say 3 in.

W=1·25
$c$ =0·25
$\rho''=1{\cdot}81$ in.
S =0·35

Minimum distance from centre of main valve to port edge—

$z=(S+w+d)$
$=(1{\cdot}75+3+0{\cdot}125)=4{\cdot}875$ in.
An overall value of 5 in. may be used.

S =1·75
$w$ =3
$d$ =0·125

**177. The Link Motion.**— There are several types of link motion — Stephenson, Allan, Gooch; but the Stephenson gear, first applied to the locomotive, has until recently been almost universally used for locomotive, winding, and marine engines. One arrangement of this gear is shown in Fig. 206 for a marine engine, and another for a locomotive in Fig. 207.

The curved slot link L (Fig. 206), from which the gear takes its distinctive title, is introduced primarily for the reversal of motion of the engine, but it serves the further purpose of an expansion link. Before this motion can be discussed, it is necessary to outline the general principle of the action of a reversing gear.

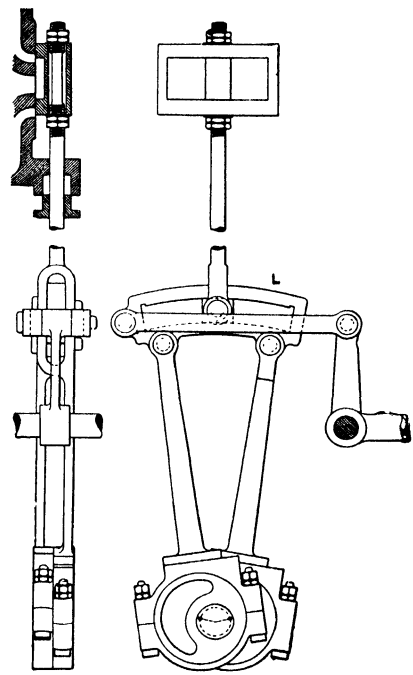



FIG. 206.

In Fig. 208 the crank $oC$ is shown on the inner dead centre. If it is to rotate in the clockwise direction then the eccentric $oE$ must be set at the angle of advance $\theta$ in the first quadrant, so that the valve may open the head end port to steam. If it is to rotate in the counter-clockwise direction the eccentric must obviously be set at the angle of advance $\theta$ in the fourth quadrant, if the valve is to

open the port. According to the direction of rotation, therefore, the valve spindle must be connected either to E or E′, and the problem of the gear is this shift of centres. One obvious and simple method

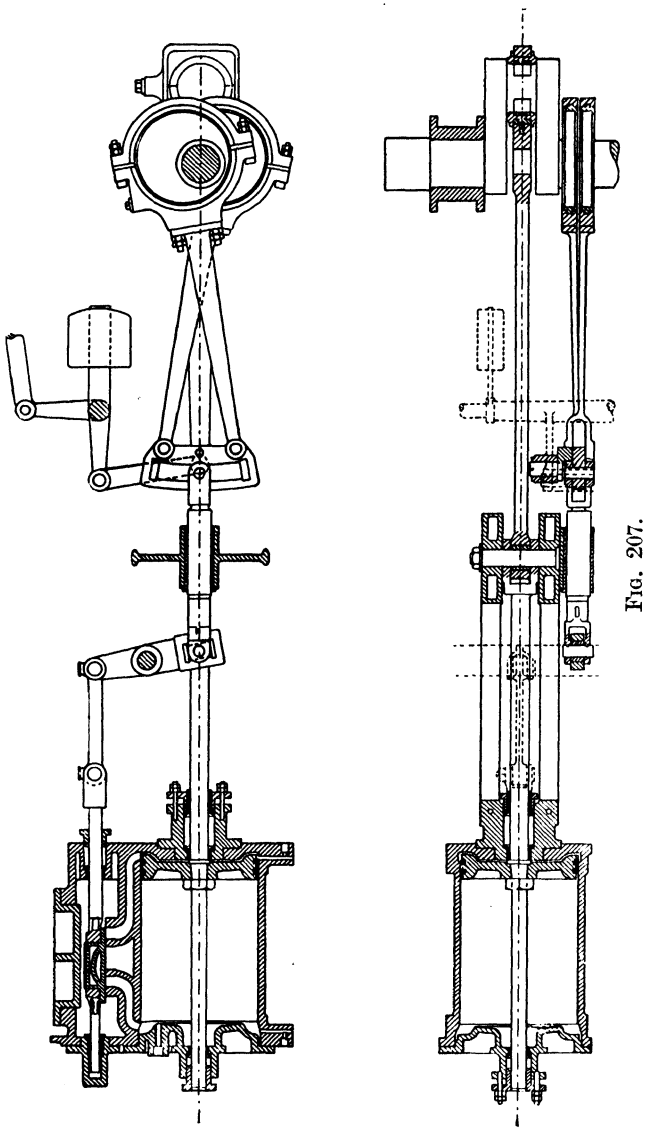


Fig. 207.

is to use one eccentric which can be displaced transversely, so that the centre may be placed at E or E′, as desired, or fixed at any intermediate point on EE′. One arrangement of this kind is shown

in Fig. 209. A disc 1 is fixed on the shaft, and the eccentric sheave 2 is provided with a wide slot 3 to enable its centre to be displaced from E to E′. It is locked in any desired position by the stud and nut 4, which clamp it to the fixed disc 1. It will be noted that not

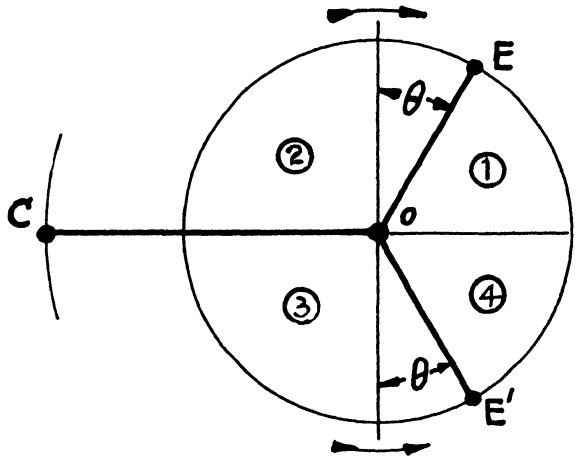



FIG. 208.

only does this arrangement permit of reversal of the engine, but at any position of the eccentric centre between E and E′ the angle of advance and eccentricity vary. Hence a variable cut-off is obtained, as well as means of reversal.

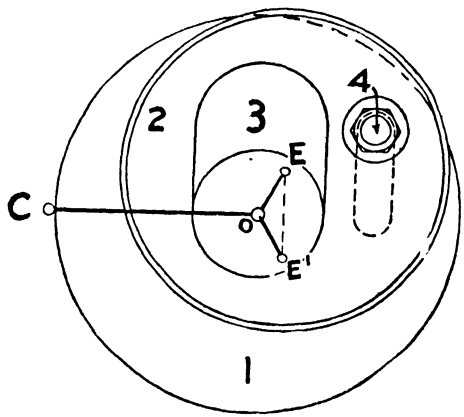



FIG. 209.

It will be apparent, however, that such variable expansion cannot be made adjustable while the engine is running, as in the case of the Meyer gear. This form of shifting eccentric is now practically obsolete, but its modern successor exists in the form

of the swinging eccentric used with various forms of shaft governors (see Chapters X and XVI).

The effect of varying the position of the eccentric centre is clearly shown by the Zeuner construction in Fig. 210. For a given valve the outside lap is constant.

In " full " gear the eccentric centre is at E, the eccentricity is $o$E, and the angle of advance E$o$Y. When the eccentric is shifted to some point F, it has the radius $o$F and the angle of advance is increased to F$o$Y. With the same lap the cut-off now takes place at position $oC_3$. When the eccentric centre is displaced to the mid position G, the eccentricity is further reduced to $o$G and the angle

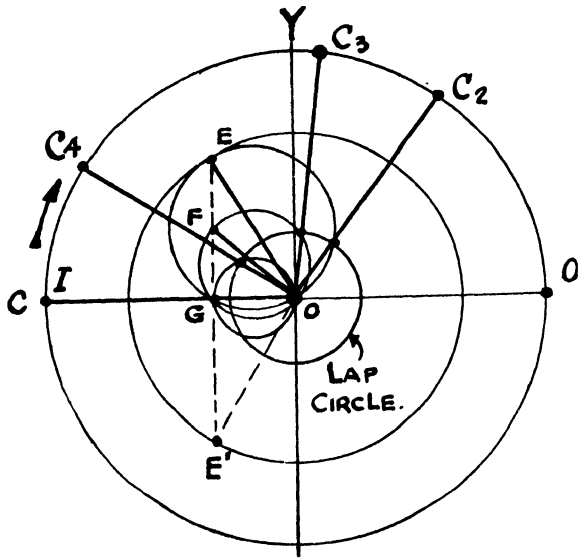



FIG. 210.

of advance increased to G$o$Y, or 90°. The cut-off now takes place at crank position $oC_4$. Similar conditions will hold for reverse running. The locus of the " virtual " eccentric centres E, F, G, etc., or EE′, is termed the " characteristic " curve of the gear. In this case it is a straight line. In other cases it is a curved line.

**178. Stephenson Link Motion.**—In the case of the link motion the single movable eccentric is replaced by two fixed eccentrics, one for forward and the other for backward running. The eccentric rods, Fig. 206, are connected to the ends of the slotted link L, and by means of the bell-crank lever and the horizontal rod or " drag link." This can be moved so that either rod is brought in line with the valve spindle. This spindle is pin-jointed to a block on which the link slides. In the position shown the arrangement is in " mid gear," the block being in the middle of the link. The valve is thus equally

influenced by the eccentrics, and the engine in consequence cannot run in either direction. When partly "linked up," the nearer the position of the block is to that of mid gear, the shorter the travel of the valve and the earlier the cut-off.

A skeleton drawing of the link and eccentric rods for a vertical engine is shown in Figs. 211 and 212. The arrangement in Fig. 211 is that also shown in Fig. 206, and is one of "open rods," since each rod is attached to the end of the link on the same side of the vertical

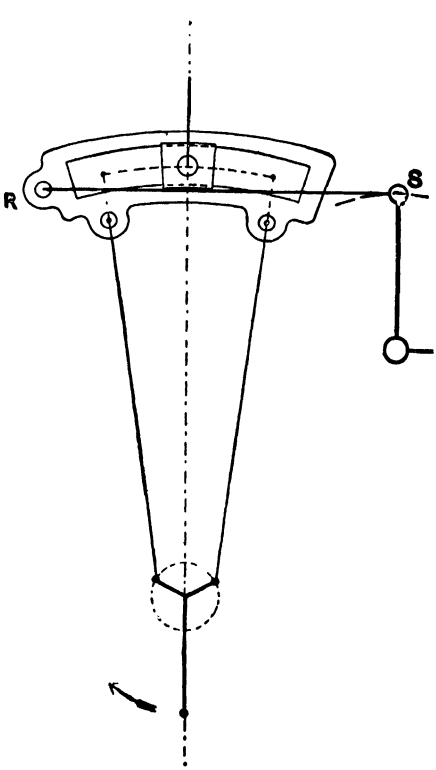



FIG. 211.

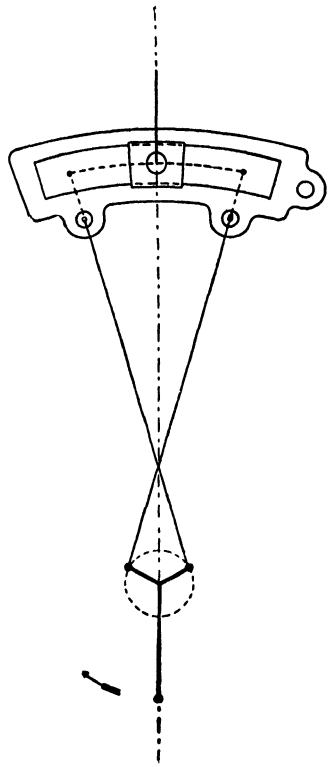


FIG. 212.

as the eccentric centre. In Fig. 212 each rod is attached to the link end on the other side of the vertical. The rods cross each other and the arrangement is one of "crossed rods." With open rods, as the gear is linked up the lead increases; with crossed rods it decreases.

The motion of the slot link is somewhat complex, being a compound of the motions imparted by the two eccentric rods and the radius or drag link SR (Fig. 211), the end R of which moves in the arc of a circle with S, the pin of the reversing lever arm, as centre.

This motion involves more or less " slotting " or rubbing of the link on the block.

The nearer the block to the suspension point R on the drag link, the less the slotting action. The link is thus, as shown in Fig. 211, usually attached to the end of the drag link at the end next the forward eccentric rod, so as to reduce slotting and wear to a minimum at the normal working position of the link. When this is done, however, it will be evident that for backward running the slotting will be considerable. This disadvantage is conceded to obtain better action under forward running conditions. If an equality is

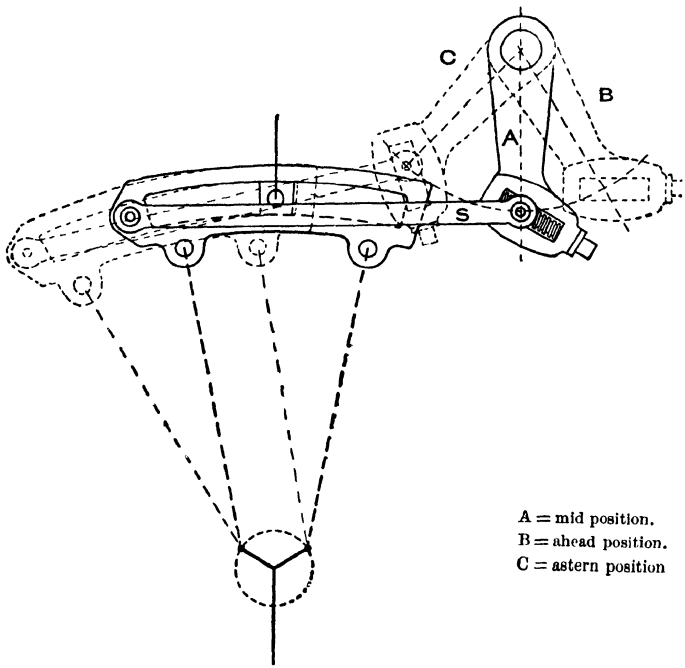



FIG. 213.

desired, then the link is suspended or attached to the drag link end at the centre.

The location of the fixed centre S of the drag link SR for full gear is of some importance.

The path of the pin R of the slotted link is first traced with the drag link removed, by drawing the gear in several positions throughout a revolution, for the full forward setting.

The process is then repeated for the full backward setting. With the length of the proposed drag link as radius an arc is drawn from some centre so as nearly as possible to bisect the irregular curved figure obtained with forward gear. Again, with the drag link length as radius, a centre is found for the arc bisecting the irregular

curve for backward gear. The two centres thus obtained fix approximately the path of the end of the reversing lever pin S to which the drag link is attached. The lever length and its centre of rotation can then be determined. Usually some compromise is made in favour of full forward gear.

*Expansion Regulator Arm.*—In compound and triple-expansion engines, with link motions all connected to one reversing shaft, it is desirable, as already shown in Chapter VII, for the power distribution between the cylinders, to adjust the positions of the slot link for one cylinder relatively to that of another, so as to obtain an earlier or later cut-off.

The method usually adopted is shown in Fig. 213. The outer end of the suspension or drag link S is connected to a block which is adjustable by means of a screw in the slot of the reversing lever arm A. The slot in A is so inclined that, if sudden reversal of the engine is necessary, the link may be thrown into full backward

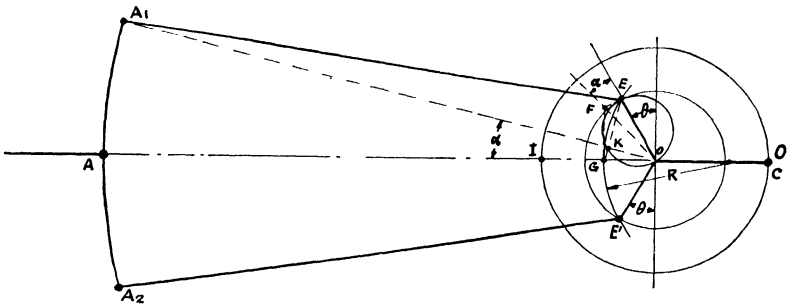



FIG. 214.

gear without recourse to the adjustment of this supplementary fitting.

In whatever position in the slot the end of the drag link may be, there can be no appreciable effect on the action of the link in full backward gear, as can be seen for positions C, in full backward or astern gear, the centre line of the slot is nearly at right angles to that of the drag link. In full forward or ahead gear, shown by the position B, the drag link is practically in line with the slot.

**179. The Characteristic Curve of the Link Motion.**—Consider the Stephenson gear shown in skeleton for open rods in Fig. 214 and for crossed rods in Fig. 215. $o$C is the crank position at the outer dead centre, $o$E is the eccentric for forward and $o$E′ the eccentric for backward running, each set at the angle of advance $\theta$. The slot link $A_1AA_2$ has a radius equal to the length of the eccentric rod $A_1E=A_2E'$. In Fig. 214 join the shaft centre $o$ and the link end $A_1$ by the dotted line $oA_1$. On the eccentric radius $o$E draw the valve displacement circle $o$EK for full forward gear, that is, when the link is swung down till $A_1$ coincides with A. Through E and the point K where the displacement circle cuts $oA_1$ draw the

line EKG cutting the line of stroke in G. Then E, G, and E′ are three points on the characteristic curve, or locus of the virtual eccentric centres between full forward and full backward gear. This curve EGE′ is concave towards the shaft. The point G can be obtained by drawing EF at 90° to the line $o$F at $\alpha$ to $o$E and projecting vertically to the line of stroke at G.

Again, in Fig. 215 join $o$ and $A_1$, draw the valve displacement circle $o$EF cutting $oA_1$ in K, then draw EK cutting the line of stroke in G. Alternatively, find point F as before and draw FG perpendicular to IO. In this case the characteristic curve EGE′ is convex to the shaft.

Many years ago Mr. McFarlane Gray gave the following rule for the determination of the approximate value of the radius of the

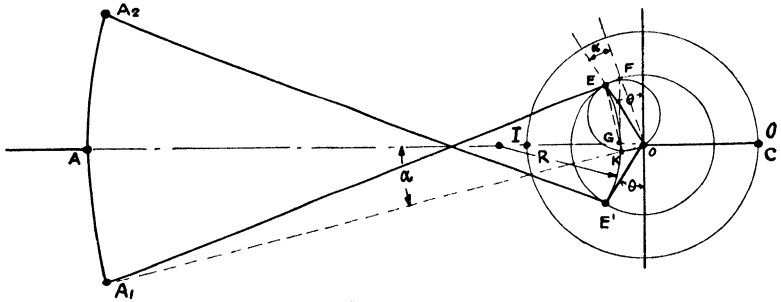



FIG. 215.

characteristic curve. In terms of the lettering on Fig. 214, this is given by—

$$R=\frac{A_1E\times\frac{1}{2}EE'}{A_1A_2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

By means of this the construction of a skeleton linkage on a large scale is avoided.

In the case of a link gear with long eccentric rods, a properly suspended link, and a relatively short valve travel, the actual valve action for intermediate settings of the link, from full to mid gear, should approximate fairly close to that shown by the diagram on which the characteristic curve is used. In general, however, exact conditions cannot be expected, owing to the various disturbing influences of the linkage. The only way in which an exact record can be obtained is to draw down the particular arrangement of gear decided upon, to a large scale, for a number of positions throughout a revolution of the engine, and then measure the actual valve displacements and other data from it.

On full forward or ahead and on full backward or astern gear, however, provided the link block centre is brought into coincidence with the end of the eccentric rod, the Reuleaux or Zeuner construction will give sufficiently correct results.

The following examples, relating to the gear of a small marine engine, bring out the salient points of the link motion, and serve as a guide to the student for the examination of types of link gear other than the Stephenson.

*Example* 6.—The following particulars are taken from the drawings of the link motion for the L.P. cylinder of a triple-expansion marine engine having an "open rod" gear and a double-ported L.P. valve.

Eccentricity $2\frac{1}{4}$ in.; angle of advance 36°; distance on the link ($A_1A_2$) between the eccentric rod ends, at mid gear position, $11\frac{5}{8}$ in.;

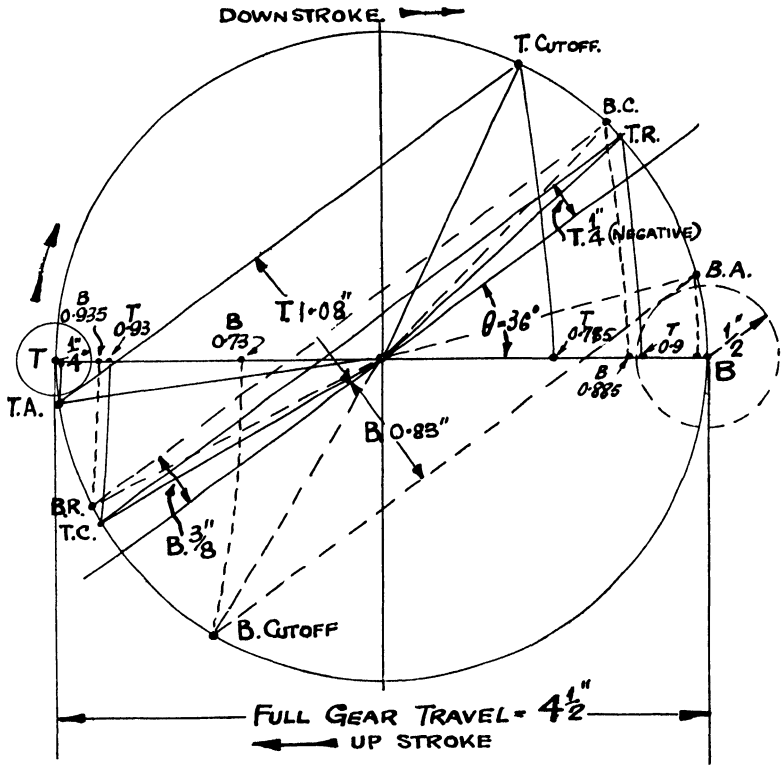



FIG. 216.

length of eccentric rod 3 ft. $9\frac{5}{8}$ in.; length of connecting rod 3 ft. 6 in.; stroke 21 in.

Allow for the top end $\frac{1}{4}$ in. lead and $\frac{1}{4}$ in. negative inside lap, and for the bottom end $\frac{1}{2}$ in. lead and $\frac{3}{8}$ in. positive inside lap. Draw the complete Reuleaux diagram for full ahead gear, and find the values of the outside laps, and the cut-off, release, and compression fractions for the down- and up-strokes.

The diagram, drawn according to the method given in Art. 171, is shown in Fig. 216. The full lines refer to the top (T) and the

dotted lines to the bottom (B) end of the cylinder. Since $\theta=36°$ and the linear advance $A=\rho \sin\theta$, then $A=2{\cdot}25\times \sin 36°=2{\cdot}25 \times 0{\cdot}588=1{\cdot}33$ in.

Top lead $l_1=\frac{1}{4}$ in., so that top lap $O_1=(A-l_1)=(1{\cdot}33-0{\cdot}25)=1{\cdot}08$ in.; bottom lead $l_2=\frac{1}{2}$ in. and bottom lap $O_2=(A-l_2)=(1{\cdot}33-0{\cdot}5)=0{\cdot}83$ in. These figures check on the drawing.

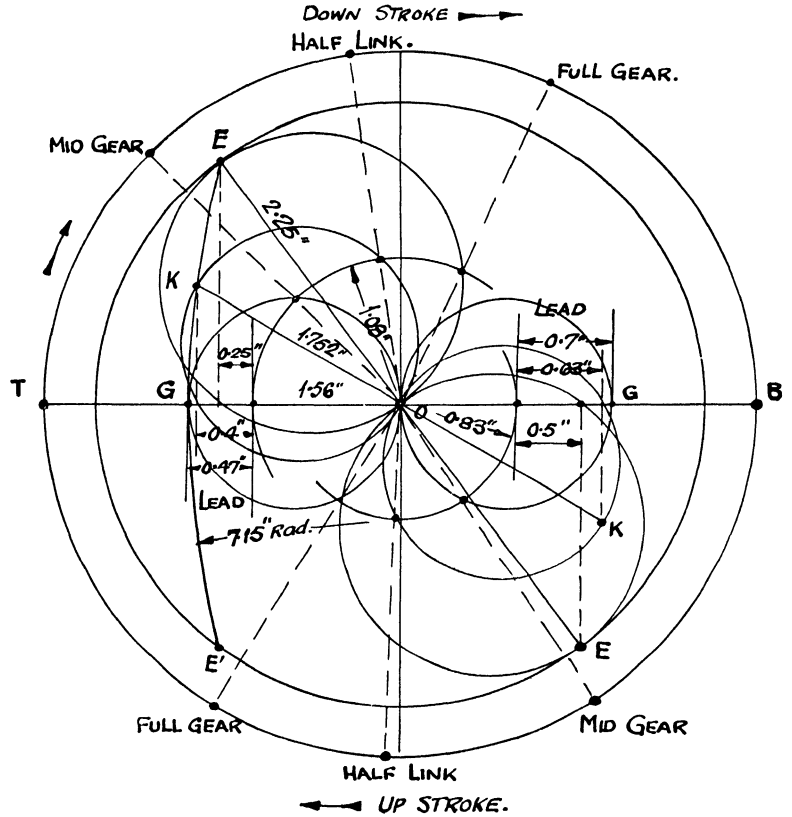



FIG. 217.

The obliquity arcs are drawn with radius $4\rho=4\times 2{\cdot}25=9$ in. and the corresponding stroke fractions are as follows:

| | *Top* | *Bottom* |
|---|---|---|
| Cut-off . . . . . | 0·765 | 0·73 |
| Release . . . . . | 0·90 | 0·935 |
| Compression . . . . . | 0·93 | 0·885 |

*Example* 7.—Draw the characteristic curve of the marine engine link gear of Example 6, and by means of the Zeuner construction find the reduced travel when the gear is set at half link, that is, half-

way between full ahead and mid gear, and then when set at mid gear. Find the corresponding leads and cut-off fractions for the top and bottom ends.

The diagram is shown in Fig. 217. The radius of the characteristic curve EGE′ is—

$$R=\frac{A_1E\times\frac{1}{2}EE'}{A_1A_2}$$

$$=\frac{45\cdot625\times1\cdot82}{11\cdot625}=7\cdot15 \text{ in.}$$

$A_1E=45\cdot625$ in.
$EE'=3\cdot64$ in.
$A_1A_2=11\cdot625$ in.

The point K for half link position is at the centre of the arc EG. This gives the virtual eccentricity at half link $oK=1\cdot752$ in. At mid gear the virtual eccentricity is $oG=1\cdot56$. K*o* and G*o* are produced into the fourth quadrant and the valve displacement circles are drawn on them as diameters. Taking the top end, $O_1=1\cdot08$ in. The upper lap arc is described with this as radius, and through the intersections with the full gear, half link, and mid gear displacement circles the dotted crank radials are drawn, giving the cut-off positions for the down-stroke. Taking the bottom end, the lower lap circle is described with the bottom lap, $O_2=0\cdot83$ in., as radius, and the crank radials are drawn through the intersections with the three displacement circles, giving the cut-off positions for the up-stroke. The cut-off fractions are then obtained, either by drawing the obliquity arcs or by using a central obliquity arc drawn through the centre *o* and setting off the displacements shown by it from the centre. The full gear cut-off values should coincide with those obtained from the Reuleaux diagram.

To find the leads, draw the dotted perpendiculars from E and K to the line of stroke T.B. The intercept between any perpendicular and the outside lap circle gives the corresponding lead. This diagram shows up clearly the fact that as the open rod gear is linked up, the lead increases. If a crossed rod gear were tried, it would be found to decrease. If the characteristic curve were straight, as in the case of the shifting eccentric, the lead would be constant.

The results given by this diagram are—

| | *Top* | | | *Bottom* | | | *Travel* |
|---|---|---|---|---|---|---|---|
| | F. | M.L. | M. | F. | M.L. | M. | F., 4·5 in. |
| Cut-off . . | 0·765 | 0·51 | 0·195 | 0·73 | 0·445 | 0·177 | M.L., 3·54 in. |
| Lead . . | 0·25 | 0·4 | 0·47 in. | 0·5 | 0·63 | 0·7 in. | M., 3·12 in. |

*Example* 8.—For the case of the open rod link motion (Examples 6 and 7) determine the release and compression fractions for the three settings of the gear.

The diagram is shown in Fig. 218. Lines are drawn through *o* at right angles to EE, KK, and GG, and at their intersections with the corresponding valve travel circles the negative top inside lap circle (full) with radius $\frac{1}{4}$ in. and the positive bottom lap circle

(dotted) with radius $\frac{3}{8}$ in. are described. The crank radials are then drawn through the points of intersection with the travel circles, in accordance with the method given in Art. 172. The full-line radials refer to the top and the dotted radials to the bottom end. The letters R and C refer to release and compression. The release and

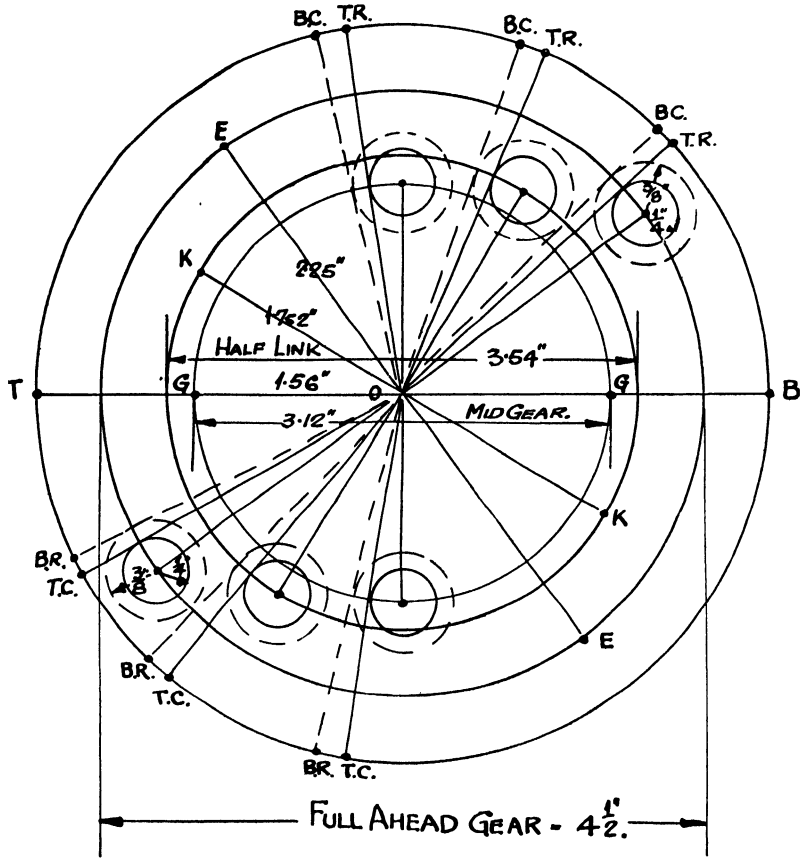



FIG. 218.

compression values obtained, when the correction for obliquity has been made, are as follows:

| | *Top* | | | *Bottom* | | |
|---|---|---|---|---|---|---|
| | F. | M.L. | M. | F. | M.L. | M. |
| Release . . | 0·90 | 0·75 | 0·49 | 0·935 | 0·822 | 0·57 |
| Compression . | 0·93 | 0·795 | 0·522 | 0·885 | 0·733 | 0·445 |

**180. Oblique Eccentric Drive.**—When an eccentric drives a valve directly, that is, when the eccentric rod is in line or parallel to the direction of the valve spindle, the travel or displacement of the valve is twice the eccentricity of the eccentric. When, however, the

eccentric rod is inclined to the axis of the valve spindle, this condition does not hold. The motion of the valve is that which would be imparted by an equivalent or virtual eccentric, having an eccentricity $\rho'$ greater than the eccentricity $\rho$ of the actual eccentric. and an angle of advance $\theta'$ either greater or less than the actual angle $\theta$, according to whether the inclined eccentric rod crosses the

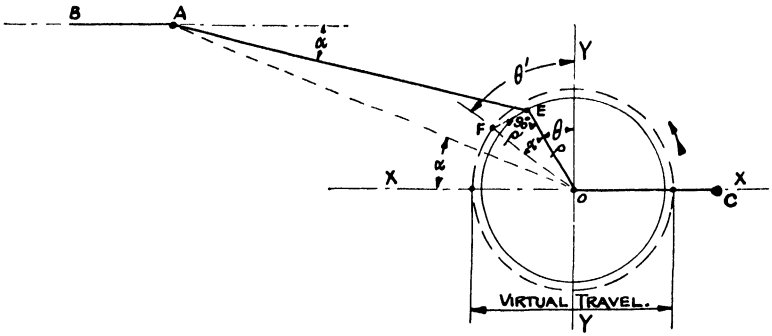



FIG. 219.

line of stroke or not. This condition arises in certain link motions, with both crossed or open rods.

Referring to Fig. 219, let AB represent the valve spindle and AE an inclined eccentric rod connecting it with the eccentric crank $o$E, set at the angle of advance $\theta$ to the line $o$Y, at right angles to the line of stroke XX, when the crank is at the outer dead centre

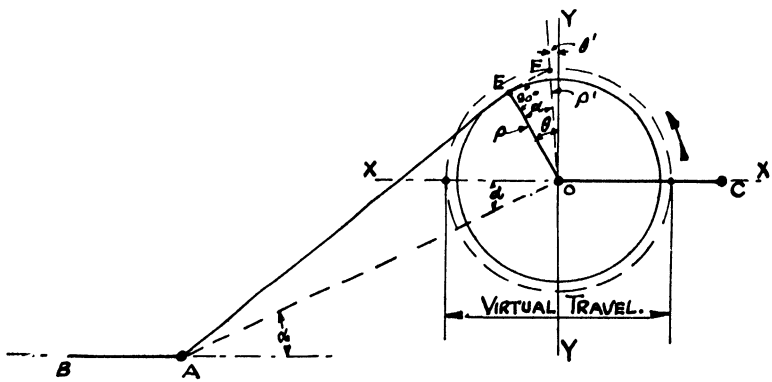



FIG. 220.

position $o$C. The rod AE in this case does not cross the line of stroke XX. Join $o$ and A, making the angle $\alpha$ with XX and AB. Draw the line $o$F at the angle $\alpha$ to $o$E, and through E draw EF at right angles to $o$E, to cut $o$F in F. Then $o$F is the virtual eccentric with eccentricity $\rho'$, and the virtual angle of advance is $\theta' = (\alpha + \theta)$.

Referring to Fig. 220, in which the eccentric rod AE crosses the

line of stroke XX, again join $o$ and A, and set off the angle $EoF=\alpha$, to the right of $oE$; draw EF perpendicular to $oE$ to cut $oF$ in F, and again $oF=\rho'$, the eccentricity of the virtual eccentric. The virtual angle of advance is $\theta'=(\theta-\alpha)$.

The first case arises in certain forms of link motion with open rods; the second with crossed rods. This is the alternative construction used to find the point G on the characteristic curve in Figs. 214 and 215.

**181. The Gooch Link Motion.**—In the Stephenson motion the link is moved on the block attached to the valve rod actuating the valve spindle. This rod has motion in the same direction as the spindle, but none in the direction at right angles to it.

In the Gooch motion the link is suspended from the end of a lever arm, which enables it to move in the line of the valve spindle only, and to pivot about the end. The link has no motion in the direction at right angles to the line of the valve spindle. On the other hand, the valve rod connecting the link block with the spindle swings about the spindle end as a centre, so that the block can be adjusted in the link to any desired position between full and mid gear. This rod is now termed the "radius rod."

A skeleton drawing of the Gooch motion for a locomotive with open rods is shown in Fig. 221. The expansion link $A_1AA_2$ in this case is convex towards the shaft, and has a radius equal to the length of the radius rod BA. The link is suspended at A from the arm JA, which is pivoted on a fixed centre at J. When the radius rod BA is swung up by the link FD, which is operated by the balanced reversing lever GF on the reversing shaft H, to the position $BA_1$, the arrangement is in full forward gear. When the radius rod is swung to the position $BA_2$ it is in full backward gear. In any setting between mid and full gear, the radius rod, during the valve travel, is inclined to the valve spindle axis BN. The rod $EA_1$ drives this inclined rod obliquely to the direction of the valve spindle axis, and hence, under this condition, the motion imparted to $A_1$ and B and therefore to the valve spindle will be that due to a virtual eccentric having a larger eccentricity and angle of advance than the actual eccentric keyed on the shaft.

On the outer dead centre, as shown, the valve will have a certain lead in full forward gear. If the radius rod is swung either up or down till the block $A_1$ occupies some intermediate position between full and mid gear, the end B will remain stationary, that is, the valve will not be moved due to the "linking-up," and hence the lead will remain the same at all positions of the block in the link.

The Gooch motion, therefore, gives a constant lead, and in this respect is an improvement on the Stephenson motion. It is, however, not quite so simple an arrangement. As already pointed out in connection with the shifting eccentric, a gear giving constant lead has a straight line as the characteristic curve. In this case it is easily found by determining the centres of the virtual eccentrics

for full forward and full backward gear and joining these by a straight line. In Fig. 221 the dotted line $oA_1$ is drawn and the angle $\alpha$ is determined. In Fig. 222, showing the crank and the forward and backward eccentrics in their correct relation, the dotted line $o$F is drawn at the angle $\alpha$ to the forward eccentric $o$E and $o$F′ at $\alpha$ to the backward eccentric $o$E′, giving the virtual angle of advance $\theta'$.

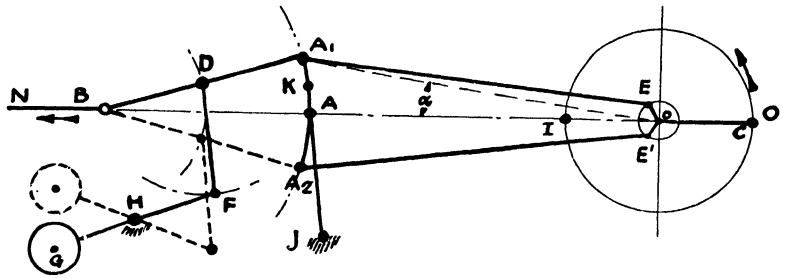



FIG. 221.

The centres F and F′ of the virtual eccentrics are obtained by the construction of Fig. 219, that is, by drawing EF perpendicular to $o$E and E′F′ perpendicular to $o$E′. The line FF′ is the characteristic curve of this gear, and $o$G represents the virtual eccentric radius for mid gear. In applying the Zeuner construction to determine

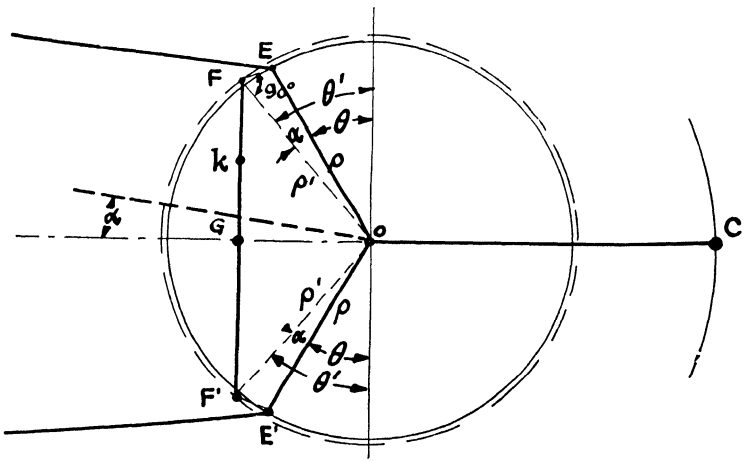



FIG. 222.

the valve motion at various settings, the angle of advance $\theta'$ has to be set off negatively; and the line FGF′ lies on the same side of the centre as the crank. In order to determine the virtual eccentric at any setting between full and mid gear the length FG or F′G is divided in the same proportion as the division of the link length between $A_1$ and A by the block of the radius rod.

For instance, if the rod is adjusted till the block is at K, midway between $A_1$ and A (Fig. 221), then the corresponding point $k$ on the characteristic curve (Fig. 222) is located midway between F and G. When once the characteristic line is drawn, the procedure is the same as in the case of the Stephenson motion already considered in detail. When a crossed rod arrangement is employed the radials $o$F and $o$F′ are drawn on the upper and lower sides of $o$E and $o$E′ and $\theta' = (\theta - \alpha)$, as in the case of the Stephenson gear.

**182. The Allan Straight Link Motion.**—In the Stephenson gear the valve rod and block are fixed in direction and the link moves transversely to the direction of the valve spindle axis. In the Gooch gear the rod and block move transversely to the spindle axis, while the link has no tranverse displacement.

In another type of gear both the link and the valve rod block have transverse motions in opposite directions. This arrangement, which gives a " straight " link, is known as the Allan straight link motion. A skeleton drawing of one arrangement of this gear is shown in Fig. 223, for open rods. The expansion link $A_1A_2$ is straight. It

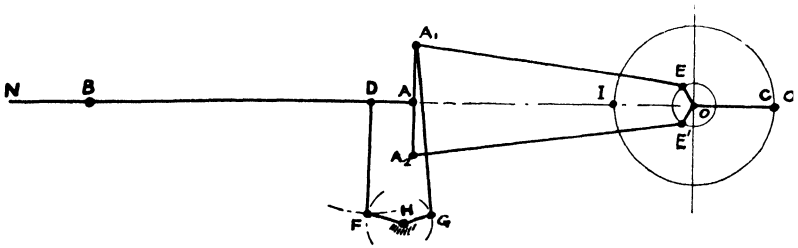



FIG. 223.

is suspended at $A_1$ from a link $GA_1$ attached to the arm HG of a bell-crank lever GHF, which is keyed on the reversing shaft H. The radius rod BA carrying the link block at the end A is connected at D to a link FD, pin-jointed to the other arm HF of the bell-crank lever. In the position shown the arrangement is in mid gear, and the link block A is acted on equally by the eccentric rods $EA_1$ and $E'A_2$. When the reversing shaft is turned counter-clockwise the link FD is lowered and with it the block A, while the link $GA_1$ is raised and with it the end $A_2$ of the eccentric rod $E'A_2$. The full backward position is reached when $A_2$ and A coincide. The action is the same for forward running, when the reversing shaft and bell-crank are turned clockwise. The expansion link is lowered and the link block is raised until A and $A_1$ coincide.

Owing to the double shift of the link and block, it is obvious that the displacement of the expansion link between the full forward and full backward gear is less than that of the Stephenson. This is an advantage where head-room is limited. There is, however, still a variation of lead—increase with open and decrease with crossed rods, as the gear is linked up—although it is less than with the

Stephenson gear. The Allan gear, however, takes up more length and has normally shorter eccentric rods, so that the reduction of lead variation is really small. There are also two extra pin-joints, an undesirable feature. The link, being straight, is more easily constructed. On the whole this gear does not seem to possess much advantage over the simpler Stephenson, and it is not now fitted on locomotives.

The examination of the valve action is more troublesome than that of the previous cases, on account of the simultaneous displacement of the link and the block in the linking up process. It is not proposed to deal with this here in detail, but the general procedure is as follows.

The gear has to be drawn in the full forward position and the virtual eccentric centre F determined by the method of Art. 180, the dotted line determining the angle $\alpha$ being drawn between the shaft centre and the eccentric rod end at the link. The virtual eccentric centre F′ for full backward gear is obtained in the same manner. In this way the two extreme points on the characteristic curve are found. A third point G giving the virtual eccentric centre on the line of stroke for mid gear setting is obtained by drawing a perpendicular through the eccentric centre to the line joining the shaft centre and the eccentric rod end at the link in the mid gear position. A circular arc can then be drawn through F, G, and F′ to give the characteristic curve.

**183. Radial Valve Gears.**—In order to remedy the defects of the ordinary type of link motion reversing gears, many forms of link work, designated " radial gears," have from time to time been tried.

These, however, have the mechanical disadvantage attending the use of a number of links with pin-joints, which are objectionable on account of the wear and slackness produced under the exacting conditions of the drive.

A few of these have survived to the present time, the two principal gears being the Joy and the Walschaert. The latter is very extensively used in locomotive practice in this country, the Continent, and America. The Joy gear is also used on locomotives, but to a less extent. Radial gears are now seldom used in marine practice.

In any form of radial valve gear the valve rod is actuated from a point A on a link, which may be termed the " characteristic link " of the gear. One end $A_1$ of this link is constrained to move in a curved path, which may be circular or oval, and the other end $A_2$ is constrained to move in a path either slightly curved or straight, the direction of which can be varied. The " linking up " of a gear of this type is accomplished by the change of direction of this path of the second point on the link. When the mean inclination of this path is 90° to the line of the valve spindle, the arrangement is in mid gear. When the path is inclined to one side of the mid position the engine runs in one direction, when it is inclined to the other

side the engine runs in the opposite direction. Just as in the case of the link motion, the nearer the inclination of the path approaches the 90° value the shorter the travel of the valve. In mid gear the travel is due only to the motion of the point $A_1$ and is equal to twice the lap plus the lead, that is, twice the linear advance of the virtual eccentric for mid gear.

It is possible to determine approximately for any given setting of the gear the eccentricity and angle of advance of the virtual eccentric, which, if coupled directly to the valve rod, would give the same motion to the valve as the radial gear in this position.*

The general case is illustrated in Figs. 224 and 226. Referring to Fig. 224, $A_1A_2$ is the characteristic link connected by a pin joint

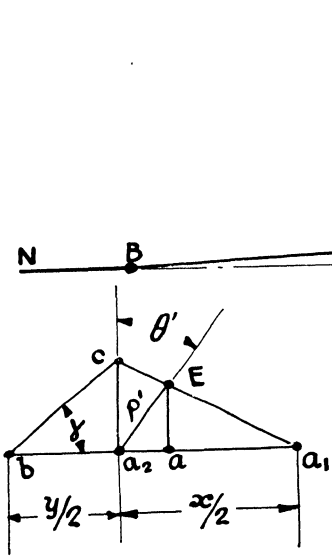



FIG. 225.

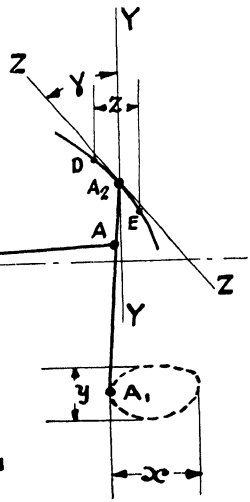



FIG. 224.

at a point A, between $A_1$ and $A_2$, to the valve rod BA. BN is the axis of the valve spindle. The line YY drawn through $A_2$ is perpendicular to this axis.

The end $A_1$ of the characteristic link, connected to some moving part of the engine mechanism, in this case moves round the closed oval curve, shown dotted. It has a total displacement $x$ parallel to the axis of the valve spindle and a total displacement $y$ in the direction at right angles to the spindle. The path of $A_1$ is shown as an oval for generality, but it may be circular, elliptical, oval, or flat loop, according to the particular gear considered. The end $A_2$

* For an analytical treatment of this subject see paper on "Radial Valve Gears: Analysis of the Motion of the Valve," by Joseph Harrison, *Proc. Inst. C. E.*, 1892–93, vol. cxiii, p. 170.

is guided in a path DE, which may be slightly curved or straight, and which has a mean inclination $\gamma$ to the line YY, at right angles to the valve spindle axis, for the particular setting of the gear shown.

The average direction of the characteristic link to the valve rod during a revolution is 90°.

The point $A_1$ moves either synchronously with or at a half period ahead of the piston, in other words it reaches the end of its stroke as the piston reaches the end of its stroke. The point $A_2$ has a motion which is either a quarter of a period behind $A_1$ or in advance of $A_1$, that is, it is always at the middle of its stroke when $A_1$ is at the end of its stroke. Expressed in another way, the motion of A may be regarded as the resultant motion of two virtual eccentrics, one placed either at 180° or at 0° to the crank, that is with $\theta = \pm 90°$, and the other at 90° to the crank, or with $\theta = 0°$.

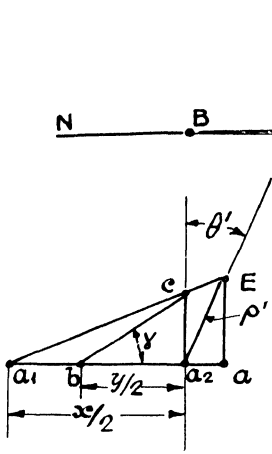



FIG. 227.

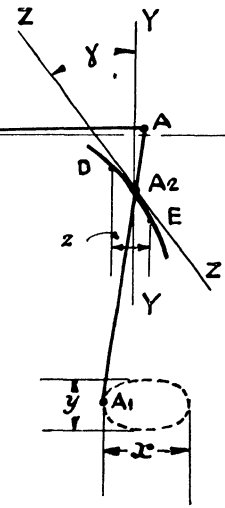



FIG. 226.

184. There are two methods by which the virtual eccentric and angle of advance for any setting of the gear may be determined. The first and more general method is as follows. When the point of attachment A of the valve rod with the characteristic link is situated between $A_1$ and $A_2$, as in Fig. 224, draw $a_2c$ and $ba_1$ at right angles (Fig. 225). From $a_2$ set off to a convenient scale $a_2b$ equal to half the displacement of $A_1$ in the direction at right angles to the valve spindle, that is $\frac{y}{2}$; and in the other direction set off $a_2a_1$ equal to half the displacement of $A_1$ in the direction parallel to the valve spindle, that is $\frac{x}{2}$. Find the average angle of inclination $\gamma$, defined by the line $zz$, which the path of $A_2$ makes with the direction perpendicular to the valve spindle (Fig. 224), and from $b$ draw $bc$ at

the angle $\gamma$ to $ba_1$, cutting $a_2c$ in $c$. Join $a_1$ and $c$. Divide $a_1a_2$ at $a$ in the same proportion as the characteristic link $A_1A_2$ (Fig. 224) is divided at A, that is, make $a_2a = a_1a_2 \times \frac{A_2A}{A_1A_2}$. Draw the line $a$E perpendicular to $a_2a_1$ to cut $ca_1$ in E, and join $a_2$ and E. Then $a_2$E is the virtual eccentric radius $\rho'$, and $ca_2$E is the virtual angle of advance $\theta'$, for the particular setting of the gear.

Referring to Fig. 226, in this case the point of attachment A to the valve rod is on an extension of the link $A_1A_2$. In Fig. 227 draw $a_1a_2$ and $a_2c$ at right angles. From $a_2$ set off, as before, $a_2b$ equal to $\frac{y}{2}$, and on the same side of $a_2c$ set off $a_2a_1$ equal to $\frac{x}{2}$.

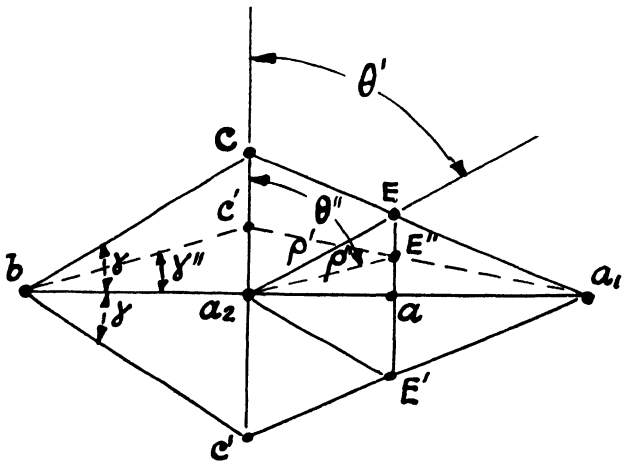



FIG. 228.

Draw the line $bc$ through $b$ at the angle $\gamma$ to $ba$, to cut $a_2c$ in $c$; then join $a_1$ and $c$, and produce the line to the right. Find the point $a$ on $a_1a_2$ produced so that $a_2a = a_1a_2 \times \frac{A_2A}{A_1A_2}$. Draw $a$E perpendicular to $a_1a$ to cut $a_1c$ produced in E, then $a_2$E is the virtual eccentric radius $\rho'$ and $ca_2$E the virtual angle of advance $\theta'$. Since for any setting of the gear, that is, for any value of the angle $\gamma$, $a_2b$, $a_2a_1$, and $a_2a$ are constant, if the gear is linked up so that this angle is reduced to $\gamma''$ (Fig. 228), then, repeating the construction, the reduced virtual eccentric radius $a_2E'' = \rho''$ and the increased virtual angle of advance $\theta''$ are obtained. In mid gear, when $\gamma$ is reduced to zero, that is, when the inclination of the path of $A_2$ is 90° to the valve spindle, then $\rho'' = a_2a$ and $\theta'' = 90$.

Any further shift of the gear will change the direction of running. The lower triangle in Fig. 228 shows the condition when the gear is linked to the same angle $\gamma$ for reverse running; $a_2E' = a_2E$ is

the virtual eccentric, the virtual angle of advance being $c'a_2E'=\theta'$. The locus of the virtual eccentric centres between these forward and backward settings is the straight line $Ea E'$; and hence for any radial gear the lead remains constant at all settings. In Fig. 228 $a_2b$ may be taken to represent the crank position at the inner dead centre, and $a_2E$ the equivalent eccentric for forward running, while $a_2E'$ is the equivalent eccentric for backward running.

**185.** In the foregoing method the virtual eccentric radius and angle of advance for a given setting of the gear are determined without reference to the displacement of the point $A_2$ in the inclined path, only the two displacements of the operative point $A_1$ being taken into account.

Alternatively the virtual eccentric radius can be found when the displacements of $A_1$ and $A_2$, in the direction of the valve spindle, for one revolution are determined. The displacement of $A_1$ can be treated as that obtained from a virtual eccentric set either at 0° or 180° to the crank, and that of $A_2$ from another virtual eccentric set at 90° to the first one. In Fig. 224, for the case of the characteristic link, with the point A between $A_1$ and $A_2$ the displacement of A parallel to the direction of the valve spindle is $x$. During a revolution the point $A_2$ in moving between its extreme positions E and D, has a displacement $z$ parallel to the axis of the valve spindle. When these two virtual radii $\frac{x}{2}$ and $\frac{z}{2}$ are found, the virtual eccentric and angle of advance can be determined as follows. In Fig. 229 draw $oa_1$ and $oa_2$ at right angles to one another. Set off $oa_1$ to the right of $oa_2$ equal to $\frac{x}{2}$, and $oa_2$, vertically, equal to $\frac{z}{2}$. Join $a_1$ and $a_2$ and divide $a_1a_2$ at $a$ in the same ratio as $A_1A_2$ is divided at A (Fig. 224). Join $o$ and $a$, and the line $oa=\rho'$, the eccentricity of the virtual eccentric, and $a_2oa=\theta'$, the virtual angle of advance, for the given setting. The triangle $a_2oa_1$ is the triangle $ca_2a_1$ (Fig. 225) obtained by the first construction.

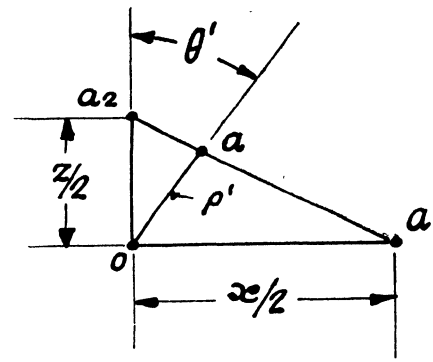


FIG. 229.

For the case where A lies outside $A_2$ on a continuation of the link, draw, as before, $oa_1$ and $oa_2$ at right angles (Fig. 230), but set off $oa_1$ to the left of $oa_2$. Join $a_1$ and $a_2$ and produce the line $a_1a_2$ to the right. Locate the point $a$ on this line so that $a_2a=a_1a_2\times\frac{A_2A}{A_1A_2}$ and join $o$ and $a$. Then $oa=\rho'$, the virtual eccentric radius, and

$a_2oa=\theta'$, the virtual angle of advance. Here again the triangle $a_1oa_2$ is the triangle $a_1a_2c$ in Fig. 227.

The methods given are perfectly general, and can be applied to any of the following radial gears, which are usually designated by the names of the engineers who first evolved them.

It should be borne in mind, however, that the results obtained are, as in the case of link motions, only close approximations. The exact valve motions can only be obtained by drawing down the particular gear in the given setting for a number of crank positions during a revolution and measuring the actual displacements, a process which is extremely tedious. When the characteristic line is determined as shown in Fig. 228, the approximate action of the gear for any setting between full forward and full backward positions can be examined by the polar method of Zeuner, in the same manner as that of the link motion.

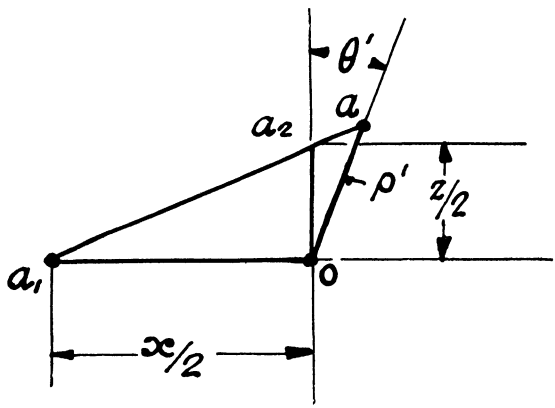



FIG. 230.

**186. Hackworth Gear.**—This was the forerunner of the family of radial valve gears, and was patented by Hackworth in 1859. A skeleton drawing of this gear is shown in Fig. 231, the arrangement of linkage being shown for full forward running. In this case the end $A_2$ of the characteristic link $A_1A_2$ is guided by a block working in a straight reversing guide DE which is fixed to the reversing shaft F and rotated about F as centre. The end $A_1$ of the link is driven by an eccentric $oA_1$ on the shaft set at 180° to the crank $oC$. The centre F is fixed so that when the crank is on either dead centre the end $A_2$ coincides with it, that is, the sliding block is at the centre of reversal. The path of $A_1$ is thus a circle and $y=x$. With the given setting the point A driving the valve spindle describes the elliptical curve shown, the major axis of which has the same direction as that of the guide. When the guide is changed round for backward running to the dotted position, this ellipse changes round with it. The distance between the tangents to this curve, drawn per-

pendicular to the direction of the valve spindle BN, is the travel T of the valve for the particular setting, and half of this is the radius $\rho'$ of the virtual eccentric of the gear.

*Example* 9.—The stroke of a locomotive engine fitted with a Hackworth gear is 15 in., and the eccentric keyed opposite the crank has an eccentricity of 3 in. The characteristic link $A_1A_2$ is 21 in. long and $A_2A$ is 10 in. The valve rod BA is $30\frac{3}{8}$ in. long, and the distance between the line of stroke IO and the valve spindle axis BN is 11 in. When the gear is set at the full forward position the guide inclination is $\gamma=45°$.

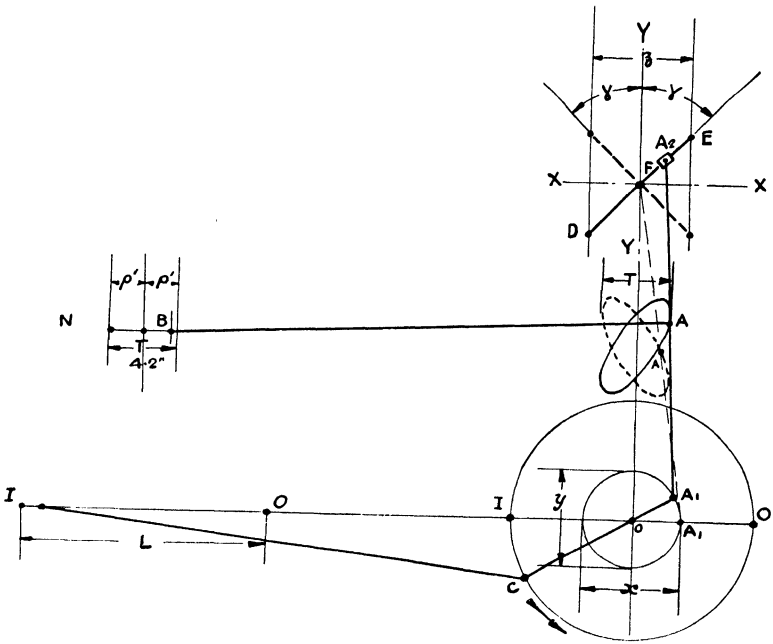



FIG. 231.

Draw the path of the valve rod end A; find the virtual eccentric radius and angle of advance for full and mid gear, and check the full gear results by the measurement from the elliptical figure described by A.

Draw the crank pin circle ICO and the line of stroke IO, and $o$YY at right angles to it (Fig. 231). Set off $oA_1$ opposite the crank on the inner dead centre and draw the eccentric circle. With $A_1$ as centre and $A_1A_2$=21 in. describe an arc cutting $o$YY in F. This fixes the reversing shaft centre. As the engine is to run forward it will be evident that as $A_1$ will move $A_2$ upwards the guide DE must slope in the direction shown by the full line, at $\gamma=45°$. Locate the point A on $A_1A_2$, 20 in. from $A_2$; place this link with $A_1$ on the eccentric circle at a number of points, keeping $A_2$ on the line DE,

and mark the positions of A. The locus of these points is the elliptical curve shown. The characteristic link and valve rod are shown for the position of the crank at 30° past the inner dead centre. Draw the vertical tangents to this curve and measure the valve travel T=4·2 in.

To find this travel without reference to the skeleton diagram, since the path of $A_1$ is a circle of 6 in. diameter, $x$=6 in. and $y$= 6 in. Apply the general method of Art. 183 as shown in Fig. 232. Set off $a_2b$=3 in. and $a_2a_1$=3 in. Draw $bc$ at 45° to $ba_1$ ; join $c$ and $a_1$. To locate $a$ on $a_2a_1$, $a_2a=a_2a_1\times\frac{A_2A}{A_1A_2}=3\times\frac{10}{21}=1{\cdot}428$ in. Draw $a$E perpendicular to $ba_1$ to cut $ca_1$ in E, and join $a_2$ and E. The virtual eccentric radius is $a_2$E=2·1 in. and the angle of advance $\theta'=42\frac{1}{2}°$. The two travel values check. Alternatively, again by measurement

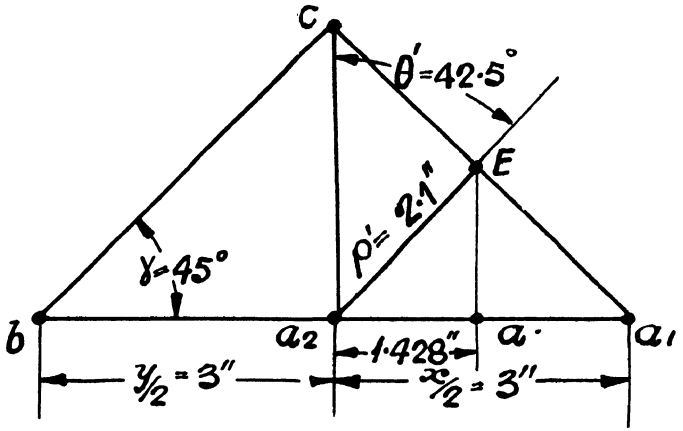



FIG. 232.

from the skeleton gear (Fig. 231) find the extreme points D and E of the block travel in the guide and draw perpendiculars to XX. The horizontal displacement $z$ of the block or of $A_2$ is 6 in., so that in the construction of Fig. 229 $oa_2=oa_1$=3 in. ; $a_2a_1$ scales 4·25 in. and $a_2a=4{\cdot}25\times\frac{10}{21}=2{\cdot}02$ in. ; $oa=\rho'$=2·1 in., as in the previous case, and $\theta'=42\frac{1}{2}°$.

In each case the virtual eccentric radius, for mid gear, is $a_2a$ (Fig. 228)$=a_1a_2\times\frac{A_2A}{A_1A_2}=1{\cdot}428$ in., and $\theta'=90°$.

**187.** In the arrangement of Hackworth gear illustrated in Fig. 231 the eccentric is fixed opposite or 180° in advance of the crank, and the valve operating point A falls between $A_1$ and $A_2$ on the characteristic link. Instead of fixing the eccentric at 180°, it may be fixed at 0° to the crank, that is, $oA_1$ is measured from $o$ towards the

inner dead centre and the point $A_1$ lies on the other side of the vertical. In this case the point A must be located on the extension of $A_1A_2$, as in the case shown in Fig. 226, and the method of procedure shown in Fig. 227 must be applied to determine the virtual eccentric radius and angle of advance.

There are two variants of the Hackworth gear, the Marshall and

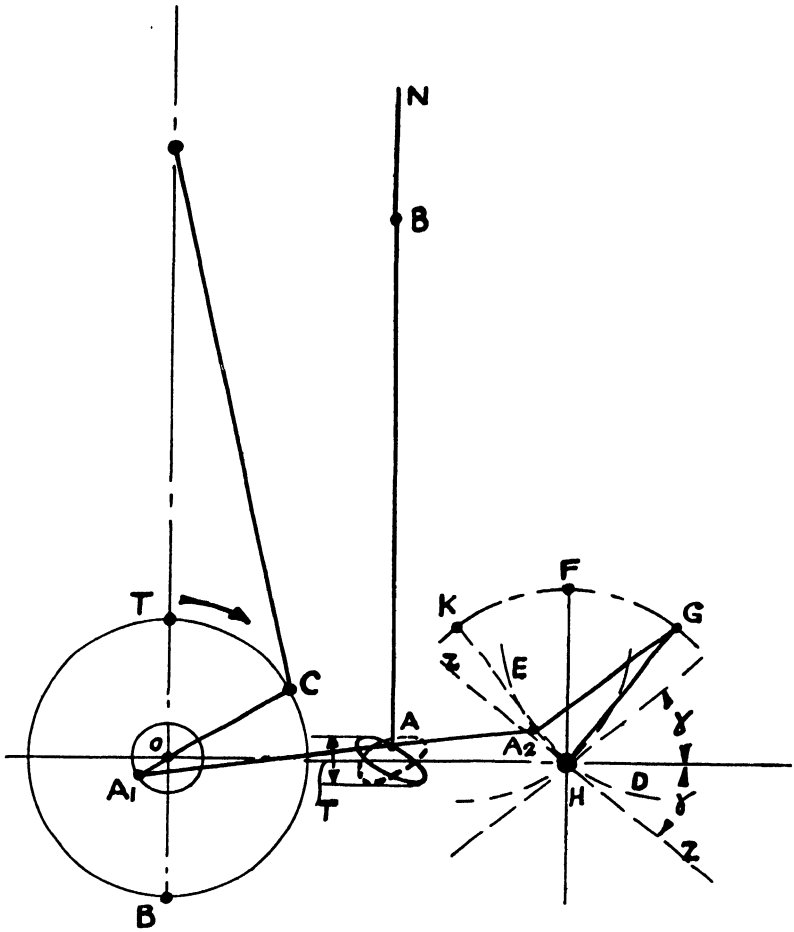



FIG. 233.

the Bremme. In each case the straight guide path for the link point $A_2$ is replaced by a circular arc. The point $A_2$ is connected to the end of a swing link, the centre of suspension of which can be altered so as to alter the mean slope of the arc to the line at right angles to the valve spindle. In the Marshall gear the eccentric is fixed at 180° in advance of the crank and the point A lies between

$A_1$ and $A_2$ on the characteristic link, as in the particular case of the Hackworth just described. In the Bremme gear, the eccentric is fixed at 0° to the crank and the point A is situated on the extension of $A_1A_2$, beyond $A_2$.

The skeleton outline of a Marshall gear, for a marine engine, is shown in Fig. 233. The reversing shaft H has an arm HG which can be fixed in any position between full forward setting HG and full backward setting HK. A swing link $GA_2$ is pivoted at the

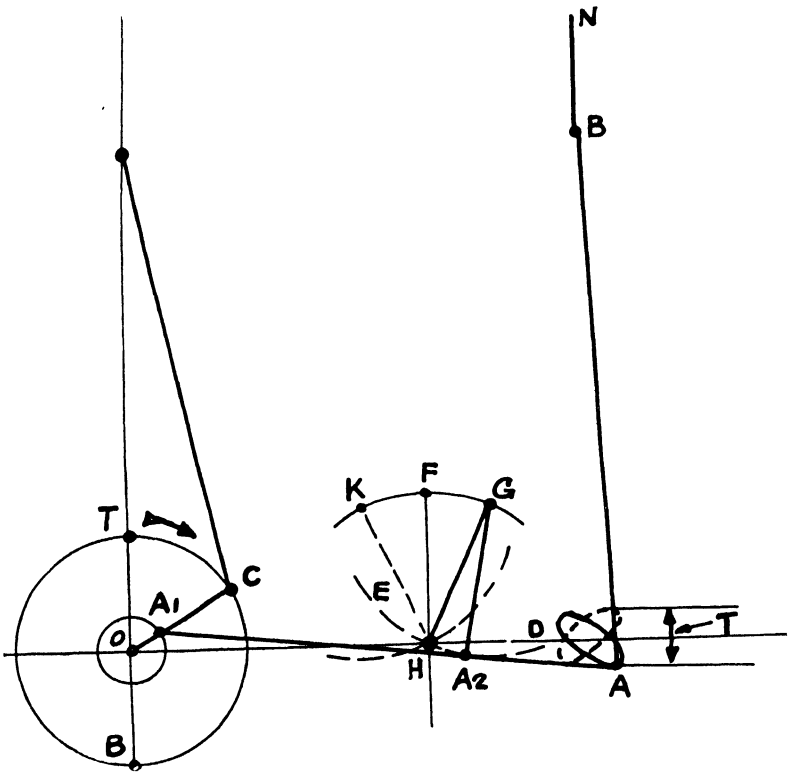



Fig. 234.

end of the arm HG and describes the arc DE about G as centre. This swing link is pin-jointed at $A_2$ to the characteristic link $A_1A_2$. The line ZZ shows the mean inclination of the path of $A_2$ during a revolution. Again the locus of A is an oval curve, the total width parallel to the valve spindle BN giving the valve travel T.

The procedure to determine the virtual eccentric radius and angle of advance is exactly the same as for the Hackworth.

The outline of a Bremme gear is shown in Fig. 234. In this case the eccentric $oA_1$ is fixed at 0° to the crank $oC$, that is, $A_1$ is between

*o* and C and A falls outside of $A_2$ on the extension $A_2A$. The arrangement of reversing lever and swing link is the same as in the Marshall gear.

**188. Joy's Valve Gear.**—This gear, designed by David Joy in 1879, was taken up by what, before the railway amalgamation, were the London and North Western Railway, the Great Eastern Railway, the North Eastern Railway, and the Lancashire and Yorkshire Railway.

The original arrangement, as then fitted on the express passenger engines of the latter railway, is shown in Fig. 235. The single eccentric which is an essential feature of the previous gears is dispensed with and the motion of the driving end of the characteristic link is derived from the connecting rod.

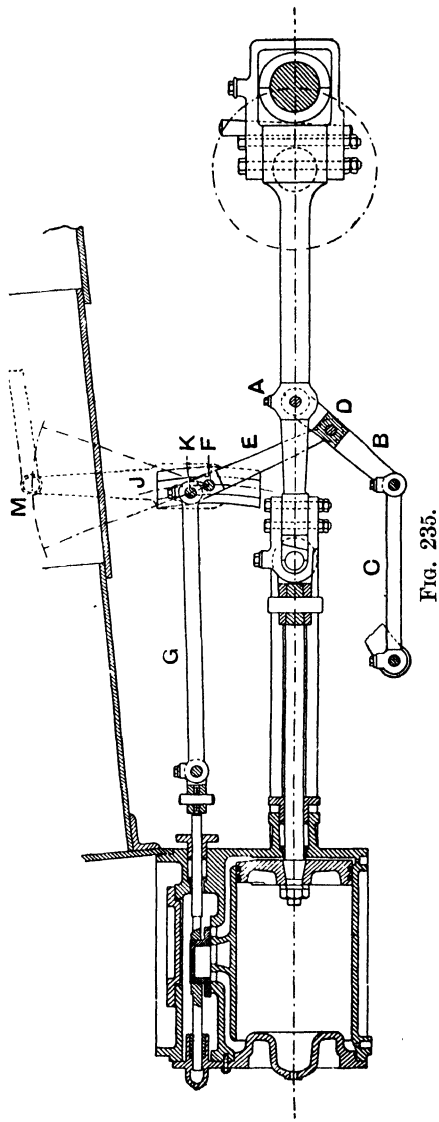



FIG. 235.

A radius rod C is pivoted on a bracket fixed to the engine frame. A swinging link B is pin-jointed to this rod at the lower end, and at the upper end to the connecting rod at A. The characteristic link is DFK. It is pin-jointed to link B at D and to the sliding block in the curved guide J, the centre of rotation of which, in the dead centre position shown, is coincident with F. The radius of this guide is equal to the length of the valve rod G. The guide is swung from the full forward to full backward gear by means of the reversing lever and rod M. The lever occupies the dotted position on the left when the gear is set for full forward running.

A skeleton drawing of this gear is shown in Fig. 236, the notation used being the same as in the case of the Hackworth gear. The points $A_1$, $A_2$, and A correspond to D, F, and K in Fig. 235. During a revolution the sliding block in the guide, to which $A_2$ is pin-jointed, travels from E to D and back again. The linkage is here shown for the engine on the inner dead centre and the block is in the centre of the curved guide, so that $A_2$ coincides with the centre of rotation F. The tangent to the curved path of the guide for full forward gear is inclined at $\gamma$ to the vertical. The radius rod *cb* swings about the pin *c*, through an angle $cbb_1$ during a revolution of the engine crank $oC_1$, and the end *b* of the vibrating link *ab* moves in the path $bb_1$. The other end *a* of this link, pin-jointed to the connecting rod, describes the elliptical curve shown, while the point on this link at which the end $A_1$ of the characteristic link is jointed describes an

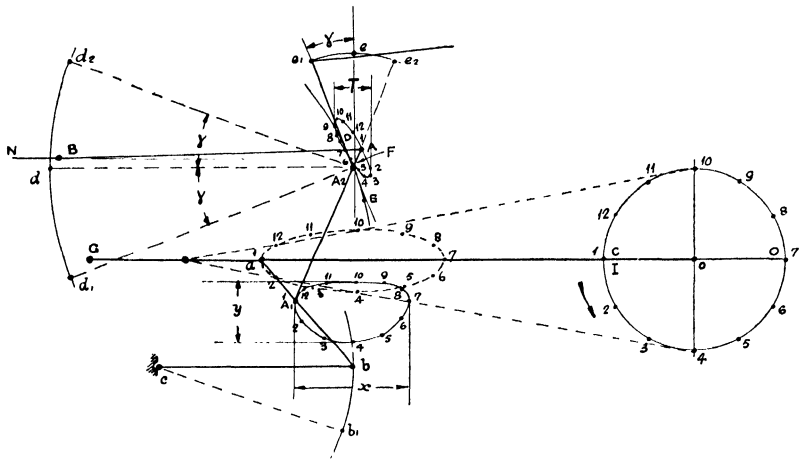


FIG. 236.

oval flattened at the top. The numbers on these curves correspond to the positions of the points *a* and $A_1$ at the correspondingly numbered crank positions during a revolution, when the engine is running in full forward gear.

As in the previous cases the operative end A of the characteristic link, jointed to the valve rod BA, describes an elliptical curve, which is also numbered to correspond to the crank pin positions. The horizontal distance between the vertical tangents to this curve gives the full gear valve travel T. The points $d$, $d_1$, and $d_2$ indicate the centre of curvature of the guide DE when it is in the mid, full forward, and full backward positions. The corresponding positions of the end of the reversing arm are $e$, $e_1$, and $e_2$.

The virtual eccentric radius $\rho'$ and angle of advance $\theta'$ for this gear can be obtained, to a first approximation, by the general methods given in Arts. 184 and 185.

In the case of a narrow-gauge locomotive with small wheels which give a limited ground clearance, there is not sufficient headroom to accommodate the radius rod C and the full swing link B (Fig. 236). To meet this difficulty Joy modified the original gear. He retained the upper part AD of the swing link and connected it to the end of an eccentric rod, driven by an eccentric set at 0° to the engine crank. As in the case of the Walschaert gear, actually what is known as a "fly crank" is used. That is, a small crank arm is fixed on the end of the main crank pin with its driving pin at the distance $\rho$, the eccentricity of the equivalent eccentric, from the shaft centre. This modification is shown in skeleton outline in Fig. 237. C is the crank pin, shown on the inner dead centre, and CK the fly crank; $A_1K$ is the eccentric rod pin-jointed at $A_1$ to the swing link $aA_1$, which is jointed to the connecting rod GC at $a$. The rest of the gear is the same as shown in Fig. 236.

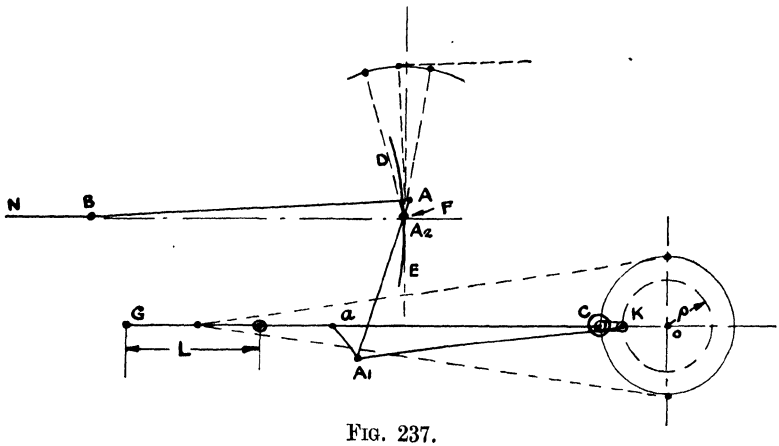



FIG. 237.

*Example* 10.—Draw down the skeleton linkage of the Joy valve gear (Fig. 236) to a scale of $1\frac{1}{2}$ in.=1 ft., using the following dimensions. Divide the crank pin circle into twelve equal parts, and then trace the paths of the points $a$, $A_1$, and A throughout a revolution.

Apply the general method of Art. 184 to find the approximate values of the virtual eccentric radius $\rho'$, the angle of advance $\theta'$, and the characteristic line of the gear.

Check the value obtained for $\rho'$ by the travel scaled from the elliptical path of A.

| | |
|---|---|
| Length of crank *o*C | 1 ft. 1 in. |
| „ connecting rod GC | 6 ft. 2 in. |
| „ „ *Ca* | 4 ft. 1 in. |
| „ „ *cb* | 2 ft. 4 in. |
| Horizontal distance of F from shaft centre | 4 ft. 1 in. |
| Vertical „ „ „ | 1 ft. $1\frac{5}{8}$ in. |
| Horizontal distance of end *c* of link *bc* from shaft centre | 6 ft. 5 in. |
| Vertical „ „ „ | 1 ft. $3\frac{5}{16}$ in. |

| | |
|---|---|
| Vertical distance of valve spindle BN from shaft centre | 1 ft. $2\frac{3}{4}$ in. |
| Length of valve rod BA = $dF$ | 3 ft. $7\frac{1}{2}$ in. |
| Angle $\gamma$ for full gear | $21\frac{1}{2}°$ |
| Length of $ab$ | 1 ft. 8 in. |
| „ $A_1A_2$ | 1 ft. 9 in. |
| „ $AA_2$ | 3 in. |

When the curves (Fig. 236) are drawn it will be found that $x=16{\cdot}624$ in., $y=8{\cdot}75$ in., and $T=4{\cdot}8$ in.

Applying the general construction (Art. 184), as shown in Fig. 238, $a_2a_1=\frac{x}{2}=8{\cdot}312$ in., $a_2b=\frac{y}{2}=4{\cdot}375$ in., angle $\gamma=cba_2=21\frac{1}{2}°$. The distance $a_2a$ is given by—

$$a_2a=a_1a_2\times\frac{AA_2}{A_1A_2}=8{\cdot}312\times\frac{3}{21}=1{\cdot}187 \text{ in.}$$

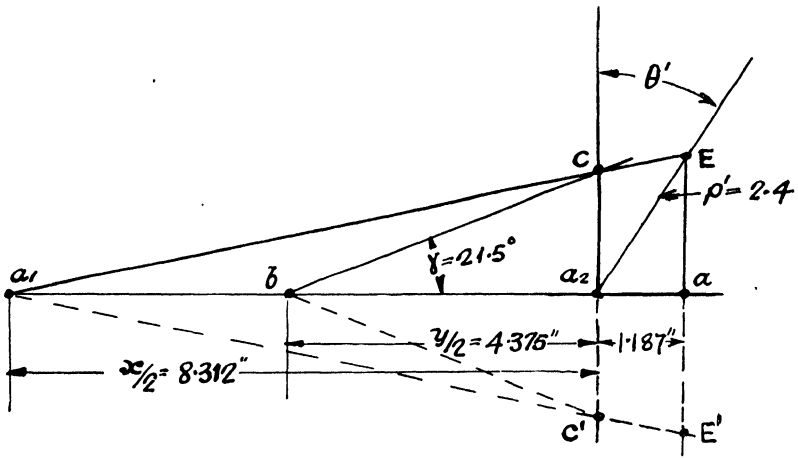



FIG. 238.

Setting this off and projecting to cut $a_1cE$ in E, the virtual eccentric radius is found to be $\rho'=2{\cdot}4$ in., and $\theta'=32°$. The check on the travel value in full forward gear is satisfactory. The lower dotted part of Fig. 238 refers to the full backward gear and $EaE'$ is the approximate characteristic line for the gear, which can be used in conjunction with the Zeuner construction to obtain approximate values of the cut-off, release, compression, etc., under various conditions of linkage of the gear, as in the case of the Stephenson link motion.

This diagram should be drawn full-size.

**189. Walschaert Gear.**—This form of gear, which was originally patented by Walschaert in 1844 and first used on the engines of the Belgian State Railways, is still in extensive use on locomotives. It is a modification of the Gooch link motion in which one of the eccentrics is dispensed with.

A side elevation of one arrangement of this gear, as fitted to an outside cylinder, narrow-gauge, tank locomotive, is shown in Fig. 239. A casting 1, which carries the outer end of the slide bar 2, is bolted outside the frame below the gangway plate. This carries a forked bracket 3 which supports the trunnions of the Gooch slot link 4. The tail 5 of this link is connected to the rod 6, which in turn is connected to the pin 7 of a fly crank 8. This crank is keyed and clamped to a projection 9 on the end of the crank pin on the wheel. The centre of the pin 7 is the centre of an equivalent eccentric which has an eccentricity $\rho$. Hence the end of the link tail 5 has a total displacement $2\rho$. The slipper crosshead 10 is provided with a vertical tail 11, which is coupled by a link 12 to the characteristic link 13 of the gear. This has the operative point A, driving the valve spindle 14, outside the end point $A_2$, as in the case of the Joy gear. The drawing shows the radius rod 15 in the mid position, so that if the engine could turn the Gooch link would simply oscillate

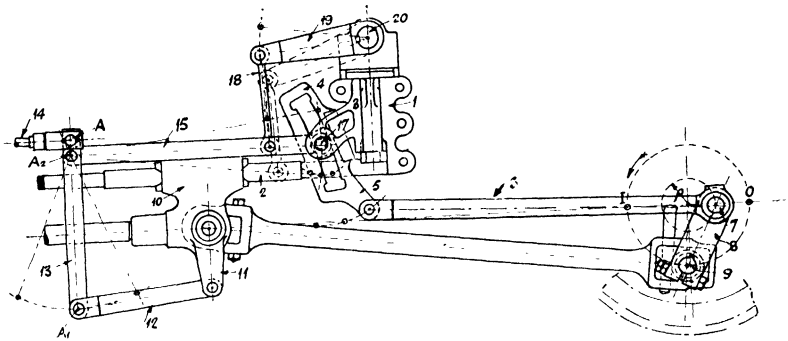



FIG. 239.

about its centre 17, and the rod would remain stationary. The characteristic link would thus pivot about $A_2$ under the action of the crosshead link, the travel of the valve would reduce to twice the lap plus lead, and the port would only open by the amount of lead. The additional travel for full gear is obtained by swinging the radius rod down or up to the dotted positions shown. The three positions of the tail pin and the shifting block pin for the bottom position are shown by the circles for the mid and end positions of the equivalent eccentric at the shaft. When the extreme positions are thus located the horizontal distance between the points gives very closely the travel of $A_2$ due to this eccentric. The horizontal displacement of $A_1$ is the piston travel. These values being known, the virtual eccentric radius and angle of advance are easily obtained by the alternative construction of Art. 185. The same process serves for linked-up positions. The radius rod is raised and lowered by the links 18 operated by the lever 19 on the reversing shaft 20.

**190. Corliss Valve Gears.**—In this system of steam distribution

the single valve is replaced by two steam admission and two exhaust valves having cylindrical instead of flat faces. It was evolved by the American engineer G. A. Corliss in 1850, and was for many years extensively used in America and this country, principally on mill engines. The advantages claimed for this system over the single valve distribution were—

(1) A reduction of clearance volume and surface, due to the shortness of the admission and exhaust passages, which results from placing the valves close to the ends of the cylinders. The clearance runs from 3 per cent. to 5 per cent. of the piston displacement.

(2) The steam admission valves being placed at the top and the exhaust valves at the bottom of the cylinder in the case of the horizontal engine, then during the flow of steam from the cylinder, the surfaces in contact with the exhaust are swept clear of water. The cylinder is thus automatically drained.

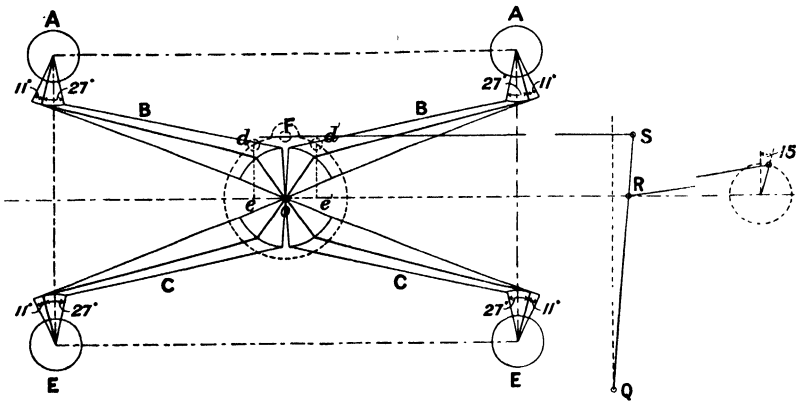



FIG. 240.

(3) The Corliss type of valve ensures a wide opening of the steam port during admission and a sharp cut-off. This prevents the wire-drawing during admission, which is one of the objectionable features of the slide-valve.

(4) The use of four valves ensures the independent adjustment of the admission, cut-off, release, and compression periods.

(5) An engine of large power can be easily and effectively governed, by regulation of the cut-off, through the action of a governor, which operates the comparatively light working parts of the valve tripping gear.

It has been frequently claimed that the use of separate steam and exhaust valves reduces initial condensation, because the live steam entering through a separate passage and not through that by which the exhaust steam is discharged, does not come in contact with the cooled surfaces of the latter. This claim is valid only if the area of the clearance surface is reduced by the arrangement of separate

valves, because in any case all the surface up to the exhaust valve must be heated up at each stroke, whether the steam is admitted through the same or through a separate port.

One of the disadvantages of the Corliss gear is the limitation it places on the rotational speed of the engine. This is due to the difficulty of getting the automatic trip gear mechanism to act effectively at high speed. A rate of about 150 rev./min. is the limit with an automatic trip gear. For higher rotational speeds the trip gear is dispensed with: the admission valves are driven directly from the wrist-plate and the cut-off is regulated by the action of the governor, which controls the valve rod motion through the medium of a link.

**191.** The skeleton arrangement of the linkage of a single eccentric Corliss gear for a horizontal engine is shown in Fig. 240. On the right an eccentric on the shaft is connected by the eccentric rod to a rocking lever QRS. This in turn is connected by a rod SF at F to a circular plate, mounted on the side of the cylinder, with its centre of rotation at O, called the "wrist-plate." Two rods B, B are attached to the wristplate at suitable points, and operate the steam admission valves at A, A through the valve lever shown. Similarly two rods C, C are attached to the wristplate, and operate the exhaust valves at E, E. The latter rods are directly connected to the exhaust valve levers. The admission valve rods B, B, however, do not operate the valve levers directly, but through some form of disengaging mechanism or "trip" controlled by the governor, as explained later.

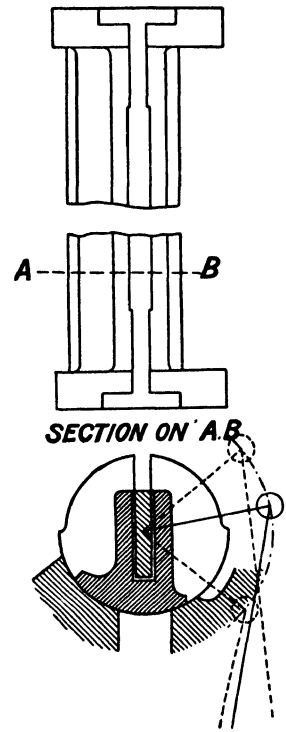



FIG. 241.

A longitudinal plan and a cross section of a single-ported Corliss admission valve is shown in Fig. 241. It is formed with a longitudinal slot into which the rectangular body of the valve spindle fits loosely. The valve has thus freedom to set itself on the cylindrical valve face of the cylinder. The ends of the spindle are cylindrical and the driving end passes through a stuffing box, while the other end works in a bush. In some cases no stuffing box is used, the spindle simply passes through a long bush. Frictional effects are thus avoided, and there is practically no leakage.

When it is necessary to restrict the valve travel, a double-ported valve may be used. A section of such an admission valve is shown

in Fig. 242. The port opening at the face is increased to the width $W_1$, so that $W_1=(W+b)$, in order to obtain the clearance when the valve has opened the port fully.

**192.** With regard to the tripping arrangements, the original form of trip gear used on Corliss engines is shown in Fig. 243. It was known as the crab-claw gear. Here B is the valve rod, driven from the wristplate, as shown in Fig. 240. A bell-crank lever AA is keyed to the valve spindle C. A square-headed bolt D with a hole through the head is attached to the lower arm of the bell-crank AA. The valve rod B passes freely through the hole in the head of D. A fork EE is pin-jointed to B at F. The lower limb, which is straight, carries a hardened steel piece or catch H, which can engage with the lower edge of the head of D. This limb is held up by means of a spiral spring G and adjusting screw. The upper limb of EE is curved and provided with a hardened steel piece

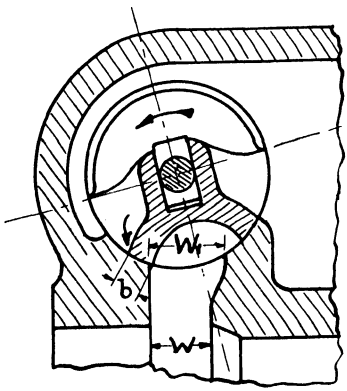



FIG. 242.

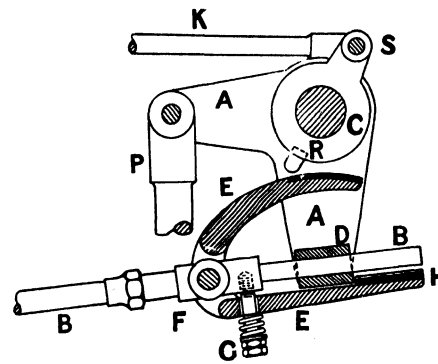



FIG. 243.

(shown in black). A ring C is loosely mounted on the valve spindle. It has on top an arm S to which the rod K, operated by the governor, is attached, and at the bottom a projecting pin R, which in the position of the gear illustrated is clear of the upper limb of the fork. As the valve rod B moves to the left it pulls on the bell-crank AA, rotates the valve spindle, and opens the valve. When the upper limb comes in contact with the pin R the lower limb is forced down and H is disengaged from D. The horizontal arm of the bell-crank AA is connected through the rod P to a spring-loaded piston in a dash pot. When the catch is released the valve is instantly closed by P under the action of the dash pot spring. The position of the pin R is regulated by the governor. If the load falls off and the speed increases, the governor pulls K to the left and moves R to the right, and the trip acts earlier. The operation is reversed for an increase of load. In this way the engine can be expansively governed.

**193.** A more modern form of trip gear is shown in Fig. 244. It consists of four parts. A lever A carrying a hard-steel block B is keyed to the valve spindle C. This lever is attached to the dash pot rod M. A double-armed lever EE is loosely mounted on a wrought iron tube on the valve bracket. The lower arm is connected to the valve rod J driven from the wristplate, the upper arm has a two-armed lever DD pivoted on a stud. The lower arm of this swinging lever has a hardened steel plate which acts as a catch and engages the steel block on A. The upper arm acts as a pawl. There is a supplementary adjustment of the trip catch, in this case by a spring H, to assist the lever DD as it falls into the trip position through the action of gravity. The disengagement of the trip is accomplished by a cam F, mounted loosely on the boss of the driving lever EE. This is operated through the arm on the right, which is connected to the governor rod at K. As the valve rod J pulls the driving lever EE to the left, with the trip catch in gear, it rotates the valve lever A and opens the valve. When the pawl arm comes in contact with the cam F it is raised, the trip catch is released from the block B, and the rod M, under the action of the dash pot spring, instantly closes the valve. The duration of the contact of the cam and pawl is very short, and a moderately powerful governor is not affected by it. The governor is free to move during admission of steam, and can be made sufficiently sensitive to keep the speed variation about the order of 1 per cent.

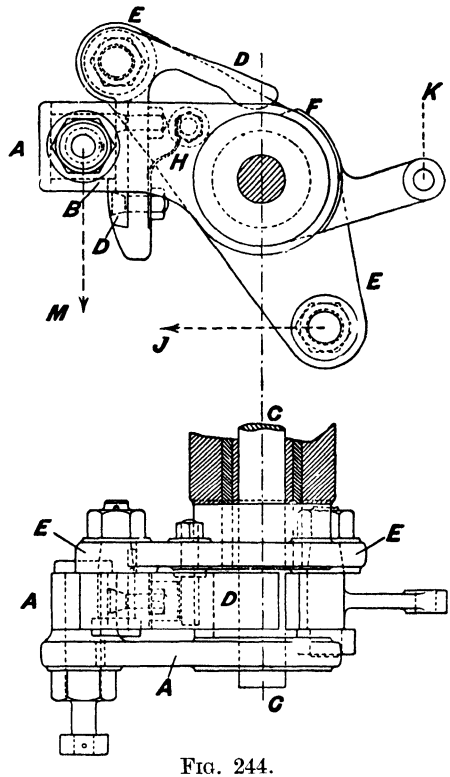


FIG. 244.

**194.** The function of the dash pot is to return the valve to the closed position quickly and noiselessly when the governor releases the trip catch. Details of one type of dash pot arrangement are shown in Figs. 245 and 246. In some instances the dash pot piston is operated by atmospheric pressure due to the formation of a vacuum below it as it is raised by the rod M (Fig. 244) during steam admission.

Preferably, however, the dash pot can be operated by a spring, as shown in Fig. 245. An air escape plug is fitted at the bottom of the dash pot cylinder and is shown in section in Fig. 246. As the piston rises the air is drawn in past the ball valve; as it descends the air is forced through the adjustable escape plug controlled by the milled head A. The speed of closing can thus be accelerated or retarded by the adjustment of the plug.

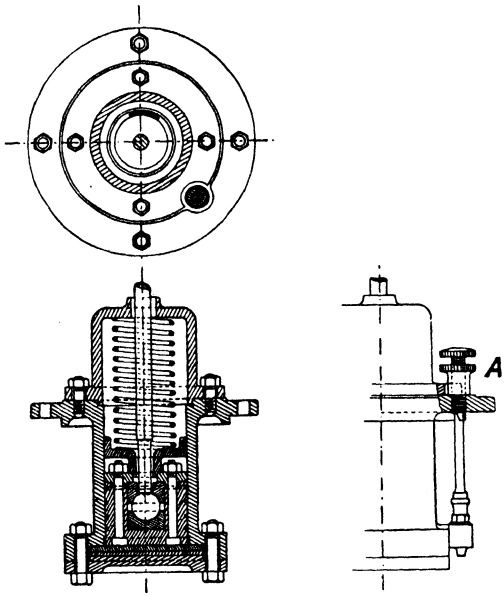



FIG. 245.

The lower cover of the dash pot (Fig. 245) holds in place two pieces of leather, which form a pad or buffer that finally brings the dash pot piston to rest.

**195.** Another skeleton arrangement of a Corliss gear for a vertical engine, with sections of the steam and exhaust valves, is shown in Fig. 247.* S, S are the steam admission and E, E the exhaust valves. Steam is admitted through the port *a* and discharged through the port *b* when the exhaust valve uncovers the exhaust port *c*. In this illustration the position of each valve corresponds to the mid position of the eccentric, that is, the exhaust valves are just about to open or close the respective ports, and the steam admission valves overlap their ports by the amount of "steam lap" corresponding to the outside lap of the ordinary flat slide-valve. The positions of the

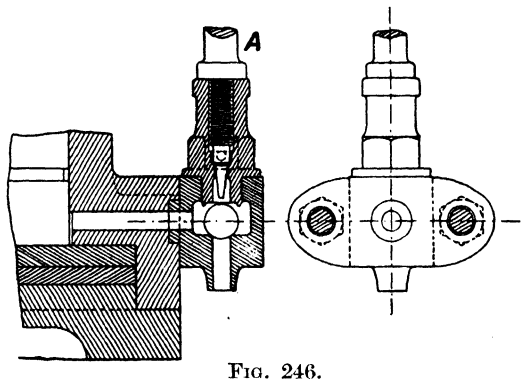



FIG. 246.

* For details of a more modern arrangement, see "The Nordberg Winding Engine," *Engineering*, 18 Novembe[illegible]21.

wristplate pins, the admission and exhaust valve rods, and the valve levers, for this "mid" position of the valves, are indicated by full lines. The steam lap is the amount by which the edge of the valve covers the port at 2 on the right side of the port. It will be noted that on the other side of the port the overlap is much greater. When the valve is liberated by the trip gear it rotates further to the right, the valve lever ultimately taking up the extreme lower or upper (dotted) position, and the cover is much greater than the steam lap. The left side of the valve must be wide enough to prevent re-opening of the port at the extreme position of the valve. The angle which the valve lever describes between the extreme position and the mid position is known as the "dwelling angle," and should be kept as small as possible.

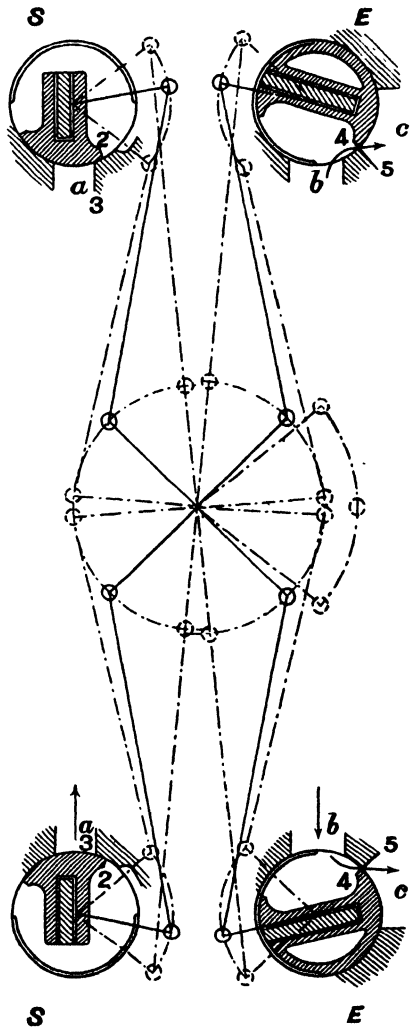



FIG. 247.

**196.** In order to set the valves of a Corliss engine marks are placed on the ends of the valves and on the valve box, to show the relative positions of the working edges of the valves and ports.

In Fig. 247 the working edge of the steam valve is marked 2, and the edge of the steam port is marked 3. The working edge of the exhaust valve is marked 4 and the edge of the exhaust port is marked 5.

A centre line is drawn on the boss of the wristplate and three lines are drawn on the periphery of the wristplate support corresponding to the middle and two end positions of the wristplate centre line.

The wristplate is first set in its mid position with its centre line ' for a horizontal engine and horizontal for a vertical engine. valves are then set so that each has the desired amount own at 2.

The lap given depends on the size of the engine. It may vary from $\frac{1}{8}$ in. to $\frac{1}{4}$ in. in small engines to $\frac{5}{8}$ in. or more in larger engines. It can be varied by shortening or lengthening the valve rod by means of adjusting nuts (see Fig. 337, p. 518).

Next, with the wristplate still in the mid position, the edge 4 of each exhaust valve (Fig. 247) is adjusted to the exhaust port edge 5 for whatever exhaust lap may be desired. Frequently no exhaust lap is given and the edge of each exhaust valve then coincides with the edge of the exhaust port when the wristplate is in mid position.

The rocking arm driven by the eccentric rod, which in turn drives the wristplate, is vertical in the case of a horizontal engine (see QRS, Fig. 240) and horizontal for a vertical engine when the wristplate is in mid position.

**197. Wristplate Diagram.**—The wristplate diagram for the left-

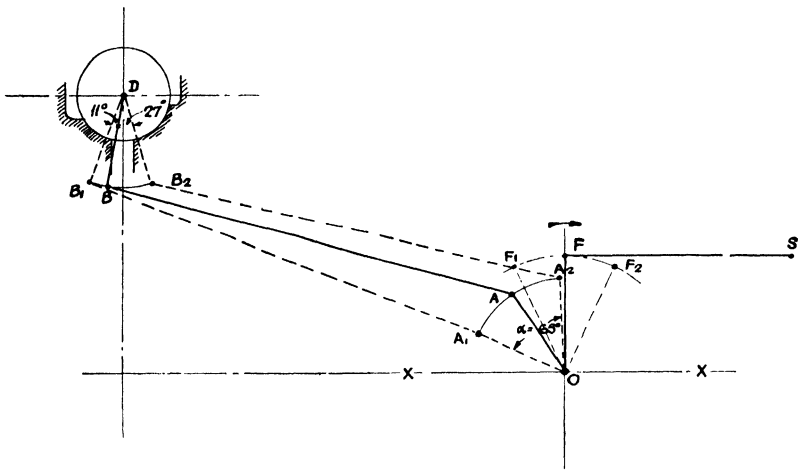



FIG. 248.

hand steam valve of the horizontal engine, Fig. 240, is shown in Fig. 248. The wristplate is centred at O and is driven by the rod FS from the rocking lever QRS (Fig. 240), which oscillates it through an angle $F_1OF_2$, in this instance 65°.

The wristplate is shown in mid position with OF vertical. The wristplate pin is located for mid position at A, and it describes the arc $A_1AA_2$ with radius OA. The corresponding central position of the valve lever, driven from the wristplate by the valve rod AB, is DB, and as already stated the valve is in the mid position, just as in the case of the ordinary slide-valve, and overlaps the port edge by the amount of the steam lap.

When the wristplate begins to move the valve the radius lies in the position $OA_1$. While this describes the angle $A_1OA$, case $32\frac{1}{2}°$, the valve lever is displaced from $DB_1$ to DB through the much smaller angle $B_1DB$, which is

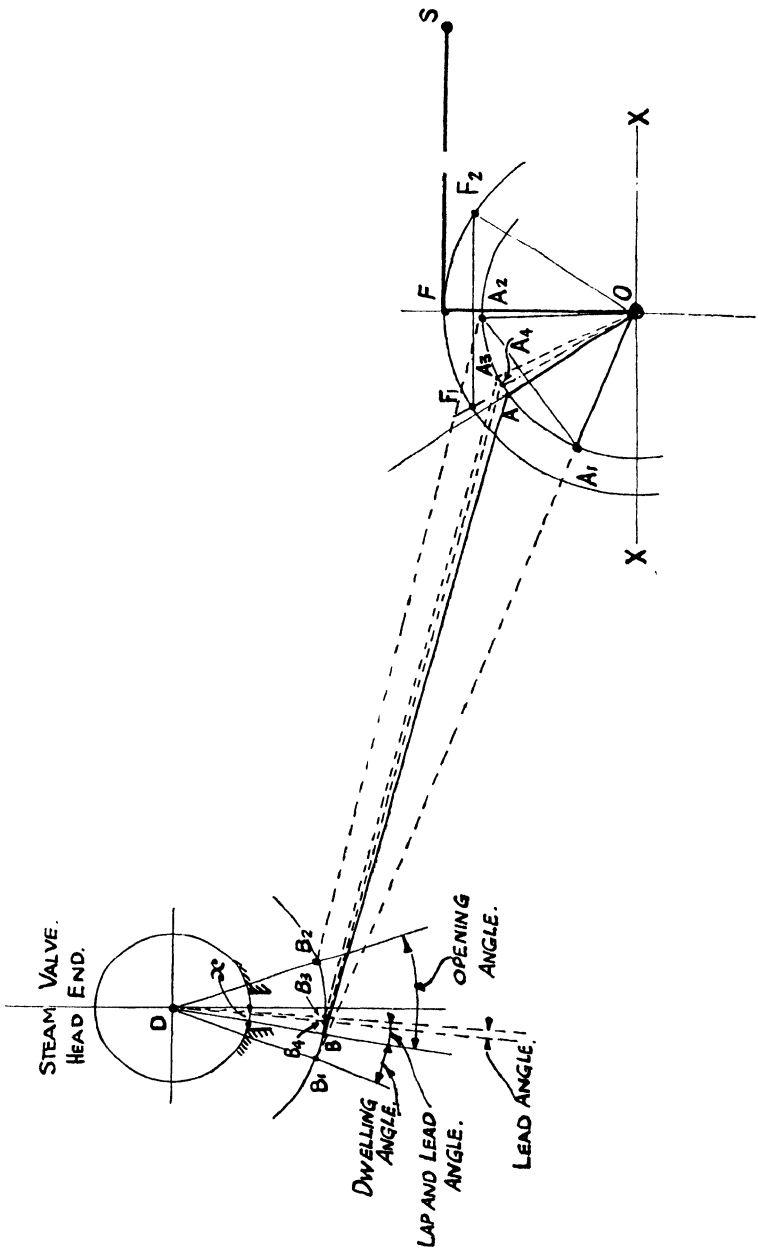

S
F
F2
F1
A2
A3
A4
A
A1
O
X
X
STEAM VALVE.
HEAD END.
D
B2
B3
B4
B
B1
OPENING ANGLE.
DWELLING ANGLE.
LAP AND LEAD ANGLE.
LEAD ANGLE.

FIG. 249.

the "dwelling angle," and should be kept as small as possible. As the wristplate radial OA sweeps out the angle $AOA_2$ to the full open position, the valve lever is moved from DB to $DB_2$, through the much larger angle $BDB_2$ of 27°. As A moves uniformly, it is obvious that the smaller the dwelling angle and the larger the angle $BDB_2$ the more rapid the opening of the port by the valve and the better the admission line of the indicator diagram.

It will also be evident that the dwelling angle can be further reduced and the opening angle increased by locating A on the arc $A_1A_2$, nearer the horizontal line XX. There is, however, a limit to this shifting of the wristplate pin, and which, in fact, is reached in this particular case. Let a line be drawn from the wristplate centre O tangential to the arc $B_1BB_2$ described by the end of the valve lever, then if $A_1$ falls above this line, the stress on the rods and pins is of the same sign for any position between $OA_1$ and $OA_2$.

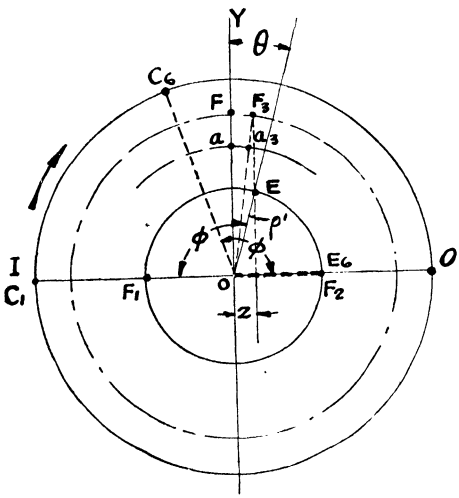



FIG. 250.

If, however, at the extreme of the travel, $OA_1$ falls below this tangent line, then there is a reversal of stress every time $OA_1$ crosses the tangent line. This is an objectionable condition that should be avoided. In Fig. 248, $OA_1$ lies almost on the tangent line, so that for the proportions given the dwelling angle has been reduced to the practicable minimum.

This diagram is repeated in Fig. 249. The rod FS driven from the rocking lever oscillates the wristplate driving pin F through the arc $F_1FF_2$, displacing F a distance $F_1F_2$ parallel to the line of stroke. This displacement may be regarded as that of the crank pin of an equivalent eccentric on a shaft at S running synchronously with the engine shaft. The virtual eccentric travel may thus be taken as $F_1F_2$.

In Fig. 250 let $o$ be the centre of the equivalent shaft, $F_1EF_2$ the virtual eccentric travel circle, and $C_1C_6O$ the crank pin circle to any convenient scale. Let the eccentric be given an angular advance $\theta$, so that $oE$ is the eccentric radius $\rho'$ and $\phi$ the total phase angle. When the crank is on the inner dead centre position $oC_1$, at I, the displacement of the eccentric centre from the mid position $oY$ is $z$ in the direction of $F_2$. With $oF$ as radius and $o$ as centre describe the dotted circle, and set off $FF_3=z$. Since $FF_3$ is a relatively

small dimension it is sufficiently accurate simply to project vertically from E to cut $FF_3$ in $F_3$. Draw the radial $oF_3$, and the angle $FoF_3$ is the (lap+lead) displacement angle of the wristplate. With radius $oa$=OA (Fig. 249) draw the arc $aa_3$ cutting $oF_3$ in $a_3$. Then set off the length $AA_3$ (Fig. 249) equal to $aa_3$ and draw the wristplate radial $OA_3$. With $A_3$ as centre and the valve rod length AB as radius cut the valve lever arc $B_1BB_2$ in $B_3$ and draw $DB_3$. The angle $BDB_3$ is the (lap+lead) angle, and the intercept $x$ at the valve face between DB and $DB_3$ is the (lap+lead)=$(O+l)$. The lead, as in the case of the slide-valve, is an arbitrary quantity varying from $\frac{1}{32}$ in. to $\frac{3}{32}$ in., according to the size of engine. Setting off the chosen lead value from $DB_3$ at the valve face, the valve lever position $DB_4$ is obtained. The angle between this line and DB is the lap angle. With $B_4$ as centre and the valve rod length AB as radius, cut the wristplate arc $A_1A_2$ at $A_4$ and measure $AA_4$.

In Fig. 251 again draw an arc $aa_4$ with OA (Fig. 249) as radius and set off $aa_4$=$AA_4$ (Fig. 249) on this towards $F_2$. Draw the radial $oa_4F_4$ cutting the dotted circle through F in $F_4$. Project from $F_4$ to cut the eccentric circle in $E_4$ and draw the radial $oE_4$.

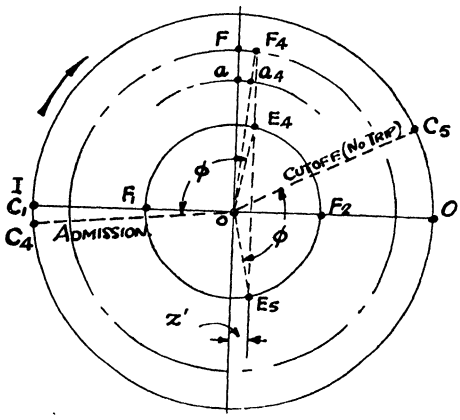



FIG. 251.

This is the position of the virtual eccentric just as the valve begins to uncover the left-hand edge of the steam port.

Draw the crank radial $oC_4$ at the phase angle $\phi$ to $oE_4$, and $oC_4$ is the position of the crank at admission. Now suppose, in the first instance, that the trip gear is locked so that the valve remains under the control of the valve rod during the complete revolution, then when the valve again reaches the left edge of the port, that is, when cut-off takes place, the virtual eccentric centre will again be displaced the distance $z'$ to the right of $o$ and the radial will have the position $oE_5$. Set off the total phase angle $\phi$, as shown, and the crank radial at cut-off (with no trip) is $oC_5$. In this case the cut-off is very late.

In the second instance suppose the trip gear is arranged to come into action under control of the governor, in the manner already described in Arts. 192 and 193.

When the crank is on the dead centre $oC_1$ (Fig. 250) the eccentric is at $oE$. When the eccentric reaches its outer dead centre position

$oE_6$ the motion of the wristplate is reversed and with it the motion of the valve lever connected to the trip through the trip catch. As already seen, the trip can only act when the rod pulls on the valve lever. Also, it has got to be operated before the end of the valve lever displacement. Hence, if the trip is in proper working order, it must release the valve before the virtual eccentric reaches the position $oE_6$. For this extreme position of the eccentric the corresponding position of the crank is $oC_6$, so that the cut-off will take place slightly before the crank reaches this position, that is, about $\frac{3}{8}$ stroke. It will be seen presently that between positions

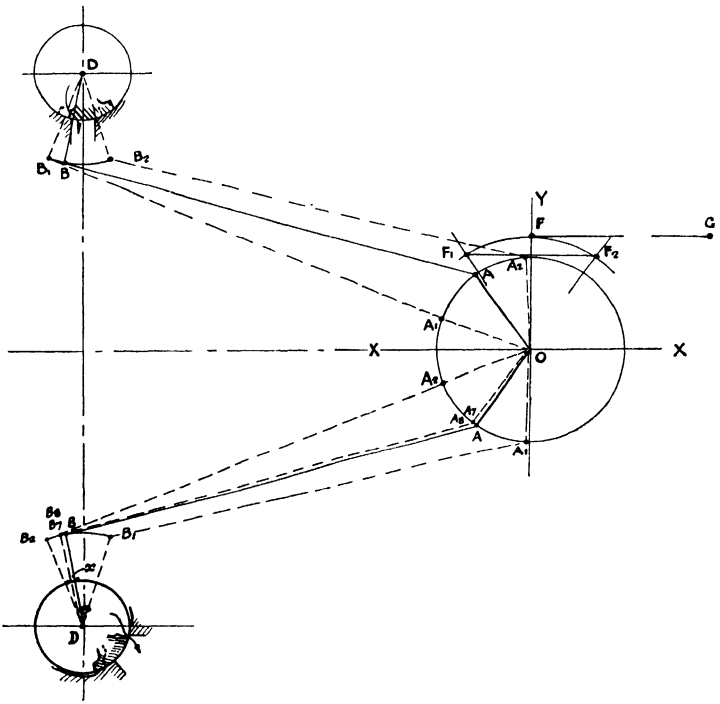



FIG. 252.

$oC_6$ and $oC_5$, where a single eccentric is used to operate the steam and exhaust valves, the trip gear is of no use. It should be possible to control the cut-off by the trip gear up to at least 60 per cent. of the stroke, to allow for variation of load on the engine.

**198.** An inspection of Fig. 250 will show that as the angle of advance or the total phase angle $\phi$ is decreased the cut-off takes place later. Hence by giving the eccentric a negative angle of advance it is possible to obtain a sufficiently late cut-off. Such an adjustment of the single eccentric, however, makes the operation of the exhaust valve impracticable. The combination of wristplate

valve rods and levers for the head end of the engine driven by a single eccentric is shown in Fig. 252. The valves are shown in the mid position corresponding to the full line linkage. When the driving rod GF pulls over the wristplate clockwise the wristplate radials for the steam valve (top) and exhaust valve (bottom) are at $OA_1$, and the valve levers at $DB_1$. When the wrist pin F reaches position $F_2$ the wristplate radial is at $OA_2$ and the valve lever at $DB_2$. The steam valve has opened the port, the trip has acted and instantly closed it, and the exhaust valve edge has moved up and covered the port edge to its fullest extent. As F moves back, the wristplate pulls over the exhaust valve lever from position $DB_2$ to $DB_1$. When it reaches some position $DB_7$ the wristplate pin is at $A_7$ and the edge of the valve is just uncovering the edge of the exhaust port, that is, $OA_7$ is the position of the wristplate radial at release. In this case the radial has to move backward in the direction from $A_2$ to $A_1$ through the angle $AOA_7$ before the centre position OA is reached, and the valve lever has to move through the angle $B_7DB$. At the mid position DB, therefore, the port will be open by the amount $x$ intercepted at the valve face between $DB_7$ and DB. Here $x=-i$, the negative exhaust lap. If $B_7$ were to fall on the other side of B the exhaust lap would be positive. In many cases the exhaust lap is zero, that is, the valve edge and steam port edge are coincident when the wristplate is in the mid position.

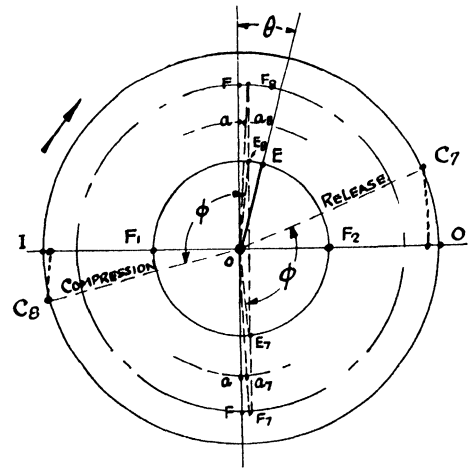



FIG. 253.

Referring now to Fig. 253, in which the crank pin circle, the eccentric travel circle, and the dotted circle of the wristplate driving pin F are the same as in Figs. 250 and 251, let the position of the crank radial, at the selected point of release on the out-stroke, be $oC_7$, then the position of the virtual eccentric radius, which is $\phi°$ in advance, is $oE_7$. The vertical projector from $E_7$ cuts the dotted circle at $F_7$, and the radial $oF_7$ defines the inside lap angle $FoF_7$ which is equal to $AOA_7$ (Fig. 252). As before, with $oa=OA$ (Fig. 252) draw the arc $aa_7$ cutting $oF_7$ in $a_7$. Measure $aa_7$ and set it off as $AA_7$ towards $A_2$ on the arc $A_1A_2$ (Fig. 252). With the valve rod length AB as radius and $A_7$ as centre cut the valve lever arc at $B_7$ and draw the radial $DB_7$, which, as just stated, then

determines the inside lap $i$ required, and whether this is positive or negative.

The difficulty which arises with the use of the single eccentric for both valves is the provision of satisfactory compression, and is clearly shown by this particular case. It will be obvious that when the valve edge again reaches the edge of the exhaust port and cuts off the exhaust the eccentric crank centre will be displaced again the same amount as at $E_7$ and will occupy the position $oE_8$ (Fig. 253). Repeating the construction to find the lap angle $FoF_8$, the arc $aa_8$ will be the same as $aa_7$, a condition which follows from the fact that $A_7$ and $A_8$ must be coincident for the compression position of the valve edge.

To determine, therefore, the position of the crank at compression the process is reversed. Draw the arc $aa_8$ with $aa_8 = AA_7$. Draw $oa_8$ and produce it to cut the dotted circle at $F_8$. Project vertically from $F_8$ to cut the virtual travel circle in $E_8$ and draw the eccentric radial $oE_8$. Then draw the crank radial $oC_8$ at angle $C_8oE_8 = \phi$ to $oE_8$. This gives the position of the crank when compression commences. In this case it is at about 97 per cent. of the return stroke.

An inspection of Fig. 253 will show that if the angle of advance $\theta$ is increased, then for a given release position $oC_7$ the centre $E_7$ will fall closer to the vertical $oF$, and if the increase is considerable it will fall on the other side of it. It follows, therefore, that $E_8$ will likewise fall to the left of $oF$, and with this new position and the increased phase angle $\phi$ a much earlier compression can be obtained.

The remedy for limited cut-off under trip action is obviously a reduction of the angle of advance from a positive to some negative value; and for late compression the increase of the angle of advance. These two conditions cannot be fulfilled by a single eccentric, and the usual practice is to employ two eccentrics, one for the steam and the other for the exhaust valves, the steam valve eccentric having a negative angle of advance.

An arrangement of valve gear with two eccentrics and wrist-plates is shown in Figs. 337, 338, and 339.

**199.** In some designs the wristplate is replaced by a rocking lever, driven directly at one end by the eccentric valve rod. The upper end of this lever is provided with "catch plates" which engage with corresponding plates on the horizontal valve rods. These rods are pin-jointed to the valve levers keyed on the valve spindles. Each valve lever is double-ended, the end of the upper arm being connected to the rod of a horizontal spring dash pot on the top of the cylinder. The valve rods and catches are lifted and thrown out of gear or tripped by coming in contact with rollers, the heights of which are adjusted by the governor. From the nature of the arrangement the valve rods push on the valve levers of the steam valves, and this type is called a "push" gear. The exhaust valves are operated directly by a second eccentric, the valve levers being

coupled directly to the eccentric rod, and they vibrate through the same angle on each side of the vertical.

When a separate eccentric is used to operate the steam admission valves, the dwelling period, necessary with the combined drive from a single eccentric, disappears. The travel of the valve is due only to the clearance necessary for the engagement of the trip catches, the lap, and the port opening. A smaller angular movement of the steam valve lever therefore results.

**200.** Referring to the trip gear, Fig. 244, as shown, the steel block of the valve lever A is in gear with the steel catch plate of the vertical fork arm of the pawl lever DD, and the valve is just about to move, as the wrist pin valve rod J pulls the swing lever EE, carrying DD, to the left. The valve covers the steam edge of the port by the amount of lap. Before this initial geared position of the plates is reached, however, the vertical arm of DD, which swings to the left after the upper arm has fallen clear of the cam ring F on the return stroke, has to bring the steel catch plate below the steel block B, and obviously a small amount of vertical clearance is necessary to enable the catch plate to slip under the block and ensure engagement. The lever EE, therefore, in the position shown, has already moved through the small angle necessary to bring the steel catch plate from the lowest position at the end of the return stroke to the contact position, that is, through the catch plate clearance. This angle may be regarded as the equivalent of the dwelling angle with the single eccentric gear. When the trip clearance angle is defined, the wristplate diagram corresponding to the negative angle of advance required for the latest cut-off can be drawn by the methods of Art. 197, and the proportions of the valve can be determined.

**201. Drop-Valve Gears.**—The valves used on these gears are usually of the double-beat type, but piston-valves are also employed. The gearing is so arranged that the valve is lifted quickly from its seat, or, in the case of the piston type, quickly drawn across the ports in the valve casing liner.

It is held in the full open position until the piston reaches the point of cut-off, and then either instantly dropped on to its seating, through the action of a trip gear, or quickly returned by the action of a cam. There are many varieties of valve design and gearing, but limitation of space precludes a detailed discussion of these.

Three examples of modern design only are shown here. With the advent of higher pressures and highly superheated steam, the double-beat type of valve has been replacing the Corliss on mill and other stationary types of engine. It is also being used on locomotives, principally on the Continent; and in one or two cases it has recently been applied to the marine engine. One of the advantages of the double-beat valve is that of balance. Unlike the slide-valve and its modification, the Corliss, it is nearly in equilibrium under the action of the steam pressure, when closed, and only a small lifting force is necessary to open it. A light

operative valve gear can thus be employed. Like the slide and Corliss types, however, it is liable to leak, especially where high-pressure superheated steam is used, on account of differential expansion. Attempts have been made to mitigate this defect by making one of the valve rims flexible, but failure has been frequent owing to fatigue fracture. The usual method is to provide the valve seatings with a flexible facing, so that under all conditions of temperature the valve is pressed hard down on both seats. This type is illustrated in Fig. 257.

**202. Marshall Drop-Valve Trip Gear.**—The standard drop-valve gear now fitted by Marshall, Sons and Co. to their horizontal types of mill engines is shown in Fig. 254. A cage 1, on which the upper and lower valve seats are formed, is fitted at the bottom of the valve chest 2. The double-beat valve 3 (shown black in section) is connected to a spindle, with throttling grooves cut on it to prevent leakage, which passes through a long sleeve attached to the cover 4. An adjustable steel block 5 is screwed on this spindle and locked in position by a jamb nut. The upper end of the spindle is fixed to the piston of an air cushion cylinder 6. This piston is loaded by an adjustable spiral spring 7. When the trip gear releases the valve it is forced down by this spring, and progressively cushioned by the air piston as it nears the seating. Pounding of the seating is thus prevented. The speed of closure is regulated by an adjustable air valve 8 on the bottom of the cylinder. The trip gear is operated by an eccentric 9 on the lay shaft 10, which is driven from the engine shaft, at the same speed, through a machine-cut mitre gear. The eccentric rod 11 oscillates a two-armed lever 12 pivoted on the valve bonnet 13. A loose bell-crank lever 14 is pivoted between these arms. The horizontal arm of this lever carries a steel roller and the vertical arm a steel contact plate (shown in black). A shaft 15, connected to the centrifugal governor, supports the projecting arm 16, which carries a trip plate (shown in black) with which the roller engages. This arm is fixed to an eccentric 17 on shaft 15, by means of which its inclination can be adjusted. The valve is raised by the lever 18 pivoted on the valve bonnet 13. The slotted end engages the spindle block 5, and the outer end has a steel trip plate which engages the corresponding plate on the end of the bell-crank 14.

In the position shown the valve is closed, the roller on 14 is clear of the plate on arm 16, and the trip plates on 14 and 18 are out of gear. As the eccentric rod moves up the lower end of lever 14 clears the end of lever 18, the roller comes down and moves along to the inner end of the arm 16. On the down-stroke of the rod the lever arm 14 pulls down the arm 18, and the valve is raised from its seat. During the downward stroke the roller on 14 moves up the inclined plate on 16, causing the bell-crank 14 to rotate, and the lower trip plate slides out of contact with the plate on 18. At the instant the two plates clear, the valve is closed by the spring. The

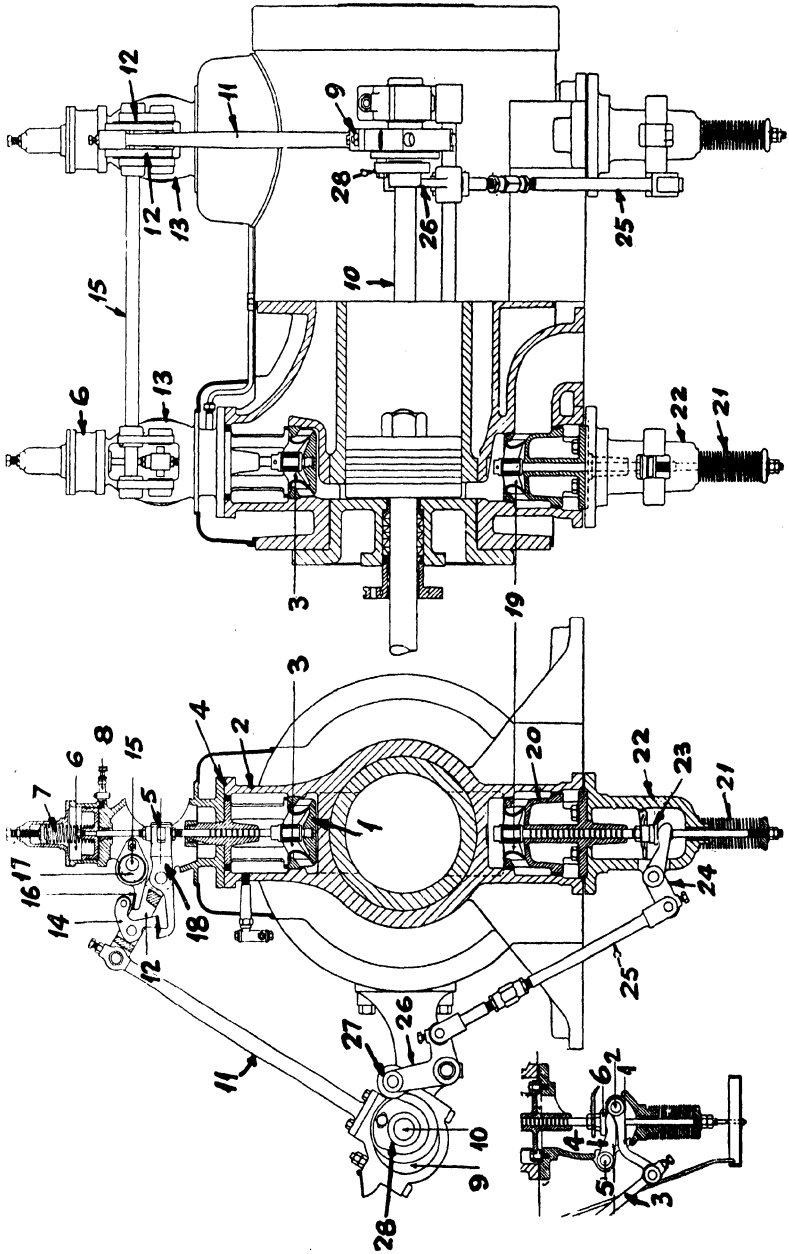



FIG. 254.

FIG. 255.

point of release is determined by the governor, which controls the shaft 15 and varies the inclination of the arm 16. In this way the speed is regulated to suit the load, by keeping the initial pressure constant and varying the cut-off.

At the exhaust end, the valve 19 is seated on a valve cage 20, which is provided with a long sleeve for the labyrinth grooved spindle. This spindle is pulled down by a strong spiral spring 21 fitted below the valve bonnet 22. An adjustable bearing block 23 is screwed on the spindle, to engage one arm of the lever 24, pivoted on the valve bonnet. This lever is operated by the adjustable rod 25 connected to another bell-crank 26, pivoted on the frame.

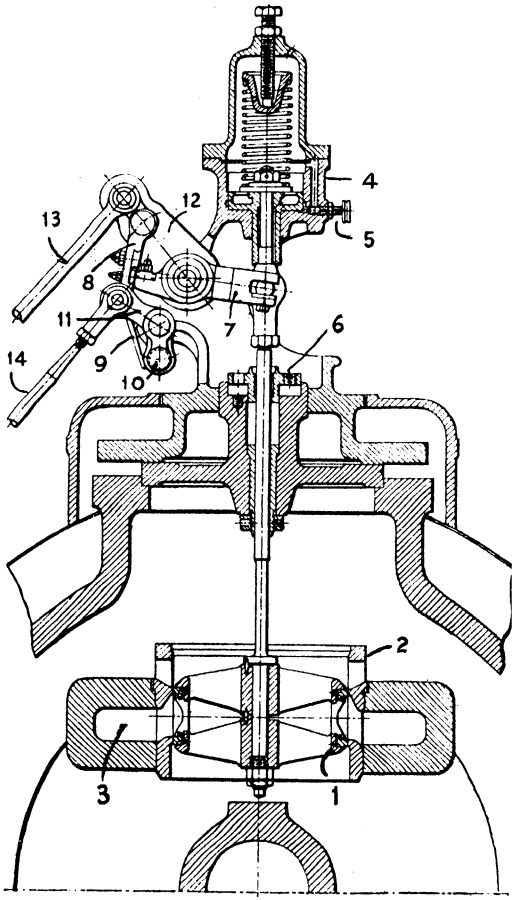



FIG. 256.

The vertical arm of this lever carries a roller 27, which makes contact with the driving cam 28. It is held up to the cam face by the pull of the spring 21 on the linkage. The cam is so formed that the valve is quickly opened and closed. Both steam and exhaust valves have flat seats. On larger-sized engines the arrangement of exhaust valve levers shown in Fig. 255 is fitted.

The lever 24 (Fig. 254) is replaced by a compound arrangement. A curved lever 1, pivoted on the valve bonnet at 2, is positively operated by the rod 3 of an eccentric on the lay shaft. A second lever 4, pivoted on the bonnet at 5, engages the valve spindle block 6 on the top side and is formed to fit the curved end of the lever 1 on the bottom. This curving accelerates the rate of valve opening and ensures a free exhaust.

**203. Piston Drop-Valve Gear.**—An example of another recent design of trip gear with drop piston valves, by Hick, Hargreaves and Co., is shown in Fig. 256. The valve body 1 carries spring rings held in place by two junk rings attached by ribs to each boss. The whole combination is fastened by the collar and nut on the valve spindle. The valve is double-ported. When raised from the closed position shown, steam is admitted at top and bottom through the ports in the valve liner 2, to the steam passage 3. In the closed position the valve has a lap of from $\frac{3}{8}$ in. to $\frac{1}{2}$ in. It requires this lift before steam can be admitted, and for the earliest cut-off this motion is ample for the full engagement of the trip gear, which can operate satisfactorily at a speed of 155 rev./min. As in the previous case, the valve is closed by the spring and cushioned by the piston of the air cylinder 4, which is fitted with an air regulating valve 5. In this case the spindle is plain, passes through a long bush, and is packed by an ordinary stuffing box and gland 6. The valve is lifted by one arm of lever 7 ; the other arm carries a steel trip plate which engages a similar plate on a pawl 8. This pawl has an extension 9 which engages with a roller 10 on the bell-crank lever 11, pivoted on the valve bonnet. The lever 12, from which the pawl 8 is hung, is driven from an eccentric on the lay shaft by the rod 13 ; and the bell-crank 11 is controlled by the governor through the rod 14. The valve is raised when the pawl is in gear and the lever 12 is pulled down by rod 13. As the roller 10 runs on to the extension 9, it throws the pawl outward, the gear trips and the valve closes under the action of the spring.

**204. Sulzer Flexible-seated Drop-Valve.**—This valve, shown in section in Fig. 257, is one of the admission valves of the uniflow engine cylinder, Fig. 353, p. 535. The valve body 1, having the lower face of smaller diameter than the upper, is connected to the central boss 2 by coned top 3 and several ribs. The boss is joined to a circular guide 4 by a connecting cone 5. The upper valve face rests on the corresponding face of the seating ring 6, which is held down in place by the junk ring 7 and stud bolts. The lower face rests on the outer edge of an adjustable piston 8, which is pressed upward by several springs 9. It will be obvious that whatever may be the expansion of the valve under temperature, these springs will maintain contact between the lower faces, the upper faces being kept in contact by the valve spindle through the action of the closing spring in the valve bonnet (see Fig. 353). The inner cylinder 10, which takes the lower part 4 of the valve, is bolted to the cylinder 11 carrying the piston 8, which is packed by two spring rings, and in turn is bolted to the valve chest 12 by the central stud 13. Projections 14 and 15 are turned on the inner circumference of 8 and the outer circumference of 10, so that when the valve is raised off its seatings these projections come into contact, and limit the rise of the piston 8. Holes are tapped in 8 and 10 for lifting screws.

By means of this elastic lower seating, a steam-tight valve is ensured under all conditions of steam temperature, if the faces are properly ground.

Unlike the previous cases, this valve is not operated by a trip gear, but by a cam action, as can be seen by a reference to Fig. 353, p. 535. The eccentric rod 1 drives the cam lever 2, pivoted on the valve bonnet 3. The upper arm of this lever carries a cam roller 4 which runs on the cam 5, at the end of the valve spindle 6. The surface of the cam is so formed that when the rod pulls the lever downward the valve is quickly opened, and on the return stroke it is quickly shut.

**205. Caprotti Valve Gear.**—This gear is a recent development due

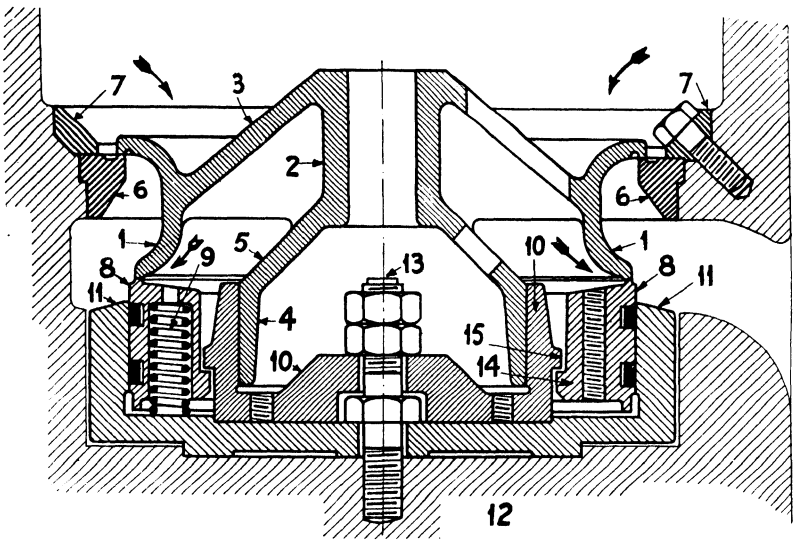



FIG. 257.

to Signor Arturi Caprotti of Milan. It is designed to overcome the drawbacks attending the use of the ordinary link motion, such as variable lead and excessive compression under linked-up conditions, and further to ensure a light and easily controlled reversible mechanism. It is in use on the engines of the Italian State Railways,* has been tried out on some engines of the L.M.S. Railway,† and is now being applied to marine engines ‡ in the form of the "Beardmore-Caprotti" gear. Full illustrations of its applications, with detailed drawings, can be referred to by the student in the articles quoted in the footnote, and therefore only a brief description

* See *Engineering*, 27 April, 1923, p. 522.

† See *Engineering*, 2 September, 1927, p. 292, and 31 January, 1930.

‡ See *Engineering*, 2 November, 1928, p. 554, and *Marine Engineer*, August 1930.

and illustration of the gear, as applied to the L.P. cylinder of a marine engine by Messrs. Wm. Beardmore and Co., is given here.

This design is a radical departure from the orthodox form of link

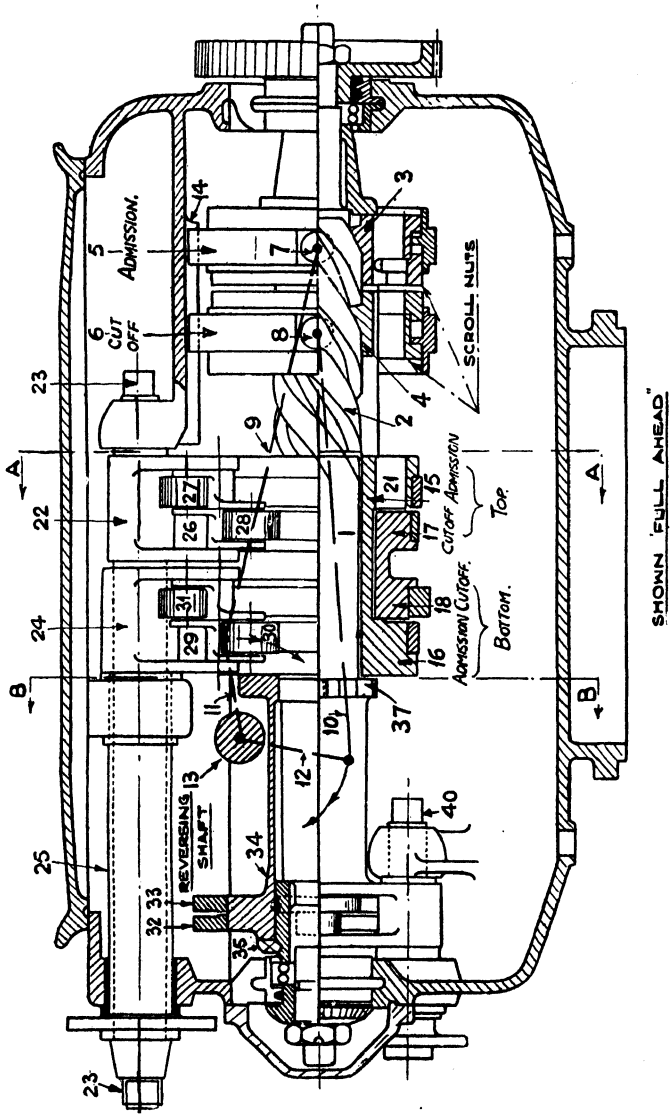



FIG. 258.

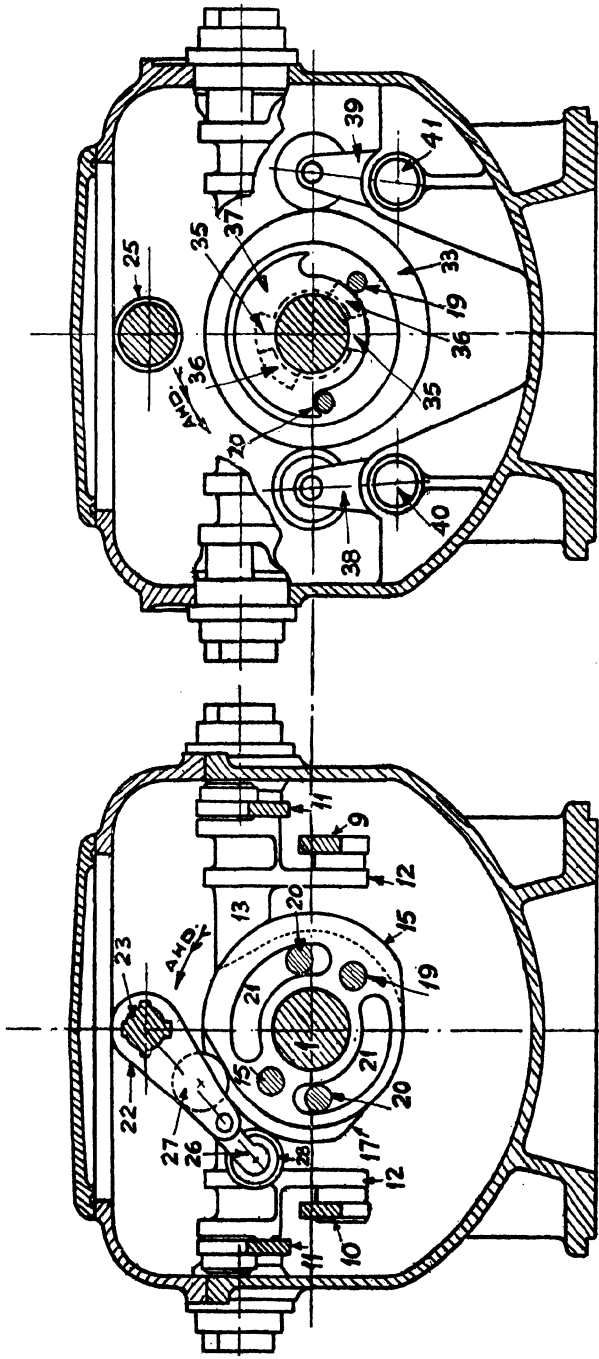
reversing and expansion gear. Four double-beat drop valves are used in place of the slide or piston valve, two for steam and two for exhaust. These valves are operated by cams against the resistance

of springs, which normally keep the valves shut on their seats. Each exhaust valve is controlled by a cam, giving a constant condition of release and compression. Each admission or steam valve is controlled by two cams, one determining the admission and the other the cut-off point. All eccentrics, rods, links, etc., are dispensed with, and the whole gear, which is driven by a jack shaft through bevels from the driving shaft, is contained in a small air and dust-tight gear box, partly filled with lubricating oil, fixed on the cylinder.

The arrangement of the Beardmore-Caprotti gear for the L.P. cylinder of a marine engine mentioned above is shown in Figs. 258, 259, and 260. The configuration of the parts as shown corresponds to the " full ahead " condition.

The steel cam shaft 1 (Fig. 258) is driven by spur gearing from the crank shaft. A portion of this, to the right, is cut with a quick pitch screw 2. Two cylindrical scroll nuts 3 and 4 are screwed on the shaft to the extreme position shown. These would rotate with the shaft, just like ordinary nuts, if held between lateral guides preventing their displacement axially. These nuts are embraced by straps 5 and 6 having trunnion projections 7 and 8 which fit into the ends of connecting links 9 and 10. They are jointed respectively to the horizontal and vertical arms 11 and 12 of a reversing shaft 13, mounted in bearings on the gear box. These straps are kept from turning by the guide ridge 14 on the casing, but can be moved axially along it. It will be obvious that if shaft 13 is turned sufficiently in the clockwise direction the links 9 and 10 will draw the straps and nuts to the left, and this motion will cause the nuts to rotate relatively to the shaft. The amount of rotation will depend on the axial displacement in each case. In the positon shown, lever 11 is nearly horizontal while lever 12 is nearly vertical. For a considerable angular displacement of 13 the displacement of the nut 3 by 11 will be very slight, while that of 4 by the arm 12 will be considerable. It is by this particular configuration of lever arms and links that the admission is kept sensibly constant while the cut-off can be varied over a wide range.

The cams which operate the valves are not driven directly by the cam shaft, but by these scroll nuts, by means of horizontal rods which pass through the nuts and cams. On the middle portion of the cam shaft, turned plain, two cams 15 and 16 are threaded, and between these on their connecting sleeve are threaded other two cams 17 and 18. Cams 15 and 16 control the admission of steam to top and bottom respectively, and cams 17 and 18 the cut-off. The admission cams 15 and 16 are driven from the scroll nut 3 by means of two rods 19, not visible in the elevation but shown in section in Fig. 259, which represents a section at AA, Fig. 258, as seen looking from right to left. These rods pass through a slot in the scroll nut 4. The cut-off cams 17 and 18 are driven by two rods 20 (Fig. 259) which are good fits in the nut 4 and pass through




Section at BB.
Fig. 260.

Section at AA.
Fig. 259.

the slots 21 in cam 15. As the cam shaft rotates the scroll nuts rotate with it and carry the cams round with them.

A forked lever 22 is provided on a shaft 23 which by means of outside levers operates the top steam valve. Another forked lever 24 is provided on a sleeve 25 mounted on 23, through which the bottom steam valve is operated. A swing lever 26 is pivoted on the end of the lever 22 and carries two rollers 27 and 28 (Fig. 258). A similar swing lever 29 is pivoted on the end of lever 24, and carries two rollers 30 and 31 (not seen in Fig. 259).

Roller 27 runs on the top admission cam 15, and roller 28 runs on the top of cut-off cam 17 (see Fig. 259). Similarly, roller 30 runs on the bottom admission cam 16 and roller 31 on the bottom cut-off cam 18 (Fig. 258).

As the reversing shaft 13 is turned clockwise the scroll nut 4 is first displaced at a faster rate along the cam shaft than 3, rotating the cut-off cam and gradually reducing the cut-off. As, however, lever 12 approaches nearer the horizontal position and 11 the vertical, the nut 4 slows up and the motion of 3 increases. At the intermediate position of the reversing shaft (corresponding to mid gear) the scroll nuts and cams are so situated that the admission cams lift the valves and the cut-off cams immediately drop them on their seats again. In other words, admission and cut-off take place almost simultaneously. As the reversing shaft is turned further, the order of operation is reversed, 17 and 18 become the admission and 15 and 16 the cut-off cams for " astern " running. When the reversing shaft has been rotated till lever 12 is horizontal and 11 is vertical, the gear is in the " full astern " position.

With regard to the exhaust valves, these are operated by two cams 32 and 33 shown at the left end of the cam shaft (Fig. 258). They are fitted on a sleeve 34 which is loose on the cam shaft. This shaft drives the sleeve by means of a dog clutch 35. The shaft dogs are shown at 35 on Fig. 260, and the sleeve dogs at 36. A catch disc 37 is fitted at the right end of the sleeve 34, having the horse-shoe form shown in Fig. 260. The projecting ends of the scroll nut rods 19 and 20 can engage with the catches, as shown for the case of rod 20, for " full ahead " gear. As the reversing shaft is turned from the full ahead towards the mid gear position, rods 19 and 20 move in the direction of turning of the cam shaft and the dog clutch continues to drive the exhaust cams, so that the release and compression points remain the same at all cut-off positions. When mid gear position is reached rod 20 has moved down clear of the left catch and rod 19 has moved up and engaged the right catch. When the reversing shaft pulls the gear through this mid position, rod 19 turns the catch disc and carries the dogs 36 clear of the shaft dogs 35, and when the shaft reaches the " full astern " position rod 19 has turned the catch disc and cam sleeve so as to bring the faces of dogs 35 round into contact with the opposite faces of the cam shaft dogs 36. The clutch is then in the position

for astern drive of the exhaust cam sleeve. The cam 32 operates the top exhaust valve lever 38, and cam 33 the bottom exhaust valve lever 39. These are keyed respectively to the operative shafts 40 and 41, which are connected by suitable levers and rods to the top and bottom exhaust valves.

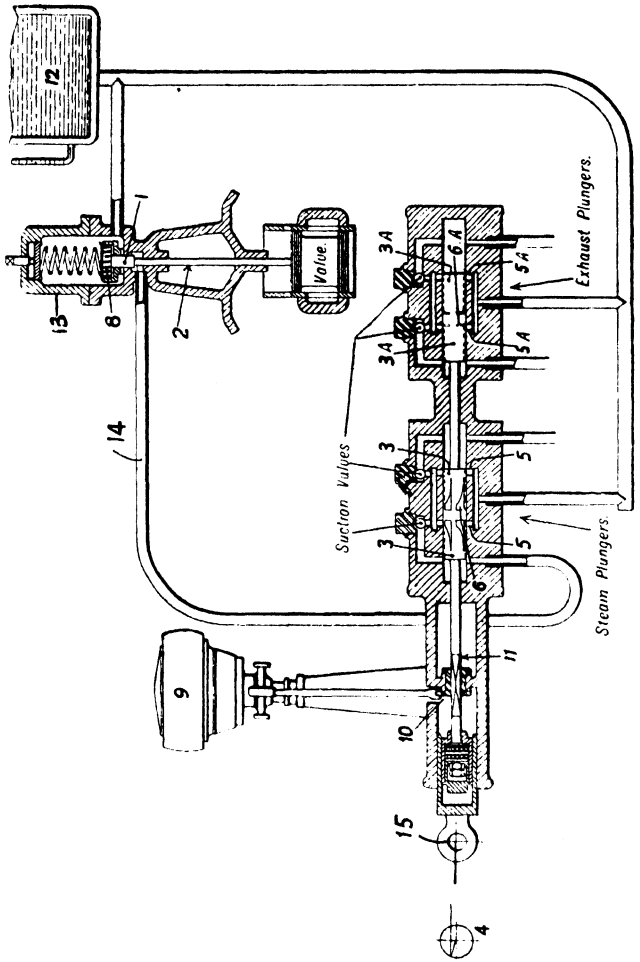



FIG. 261.

**206. Meier-Mattern Hydraulic Valve Gear.**—In this gear due to I. Meier-Mattern of Werkspoor, Amsterdam, four double-beat valves, two steam and two exhaust, are operated under oil pressure. There is no mechanical linkage. The principle of this gear is illustrated diagrammatically in Fig. 261.* The spindle of the steam admission valve shown is fitted with a piston I working in a

* Reproduced by permission from *The Engineer*, 3 February, 1928.

cylinder at the bottom of an oil vessel 13 and provided with a dash pot 8. A strong spiral spring holds the valve down on the seating. The shaft eccentric 4 drives a plunger crosshead 15, which in turn drives the spindle 11, which can be partially rotated by the lever 10, actuated by the rod of the governor 9. Two hollow plungers 3 and 3A are attached to the spindle 11. The plunger 3 regulates the steam admission and cut-off, and 3A the release and compression. The end of the cylinder is connected with the underside of piston 1 by the pipe 14, and exhaust ports are provided at 5 leading to the oil reservoir 12 and the oil chamber 13. Plunger 3 is provided with ports 6 which can overrun the exhaust ports 5, and 3A with ports 6A which can overrun the exhaust ports 5A. As 3 moves forward the top edge closes port 5 and the oil is forced through 14 and raises piston 1 until its lower edge uncovers a small port leading to chamber 13. As long as oil is forced through 14, the valve floats in the open position. As soon as the slot 6 uncovers port 5, then release of oil takes place, the flow in 14 ceases, and the valve shuts down under the action of the spring.

The timing of closure and opening of ports 5 by the slot port 6 is regulated by turning the spindle 11, so that the spiral-shaped slots close or open ports 5 earlier or later. A similar action takes place in the case of the exhaust valve at the other end of the cylinder. Suction ball valves are provided in order that the pipe connections may always run full of oil. A small light governor can be used to operate the gear, as the power required to rotate the spindle 11 is negligible. The gear acts very promptly, and it is stated that a reduction of 17 per cent. to 22 per cent. in steam consumption is obtained as compared with that given by an engine with ordinary valve gear.

# CHAPTER X

## GOVERNORS

**207.** VARIATION in speed of an engine may be due either to variation of the external load or of the mean steam pressure in the cylinder. When it is of the nature of a gradual increase or decrease of speed, which extends over a number of revolutions, it is controlled by a governor; when the speed varies in a single revolution, due to either unequal crank effort or to a sudden change of load, it is regulated by a fly-wheel.

The primary object of a governor is the maintenance, as nearly as possible, of a uniform speed of rotation independently of a change of load and steam pressure.

A governor driven by the engine cannot fulfil this function perfectly, as it cannot come into action until a change in the speed of the engine causes the regulating mechanism to act. A good governor, however, acts very quickly, and as a rule the percentage variation of the engine speed above and below the mean value is of a small order.

The engine speed may be regulated in two ways. (*a*) By throttle governing: the governor operates a throttle valve on the steam admission branch, and varies the admission pressure, while the cut-off remains constant. (*b*) By expansion governing: the governor varies the point of cut-off and alters the volume of steam admitted to the cylinder, while the initial steam pressure remains constant (see Chapter VII, Arts. 140, 141).

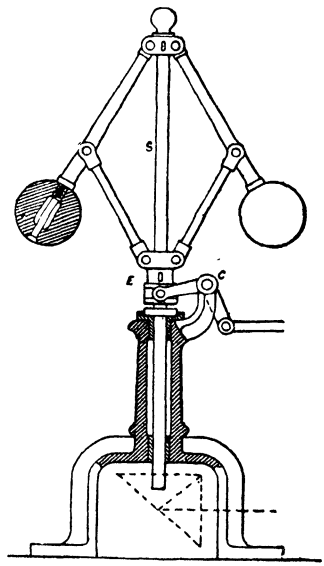


FIG. 262.

**208. Watt's Pendulum Governor.**—The original type of governor used by James Watt on the early steam engines (which ran at very low speeds of rotation) is shown in Fig. 262. When running at speed the balls are displaced outward under the centrifugal force. The linkage pulls the sleeve E upwards on the spindle S, and operates the bell-crank lever C connected to a butterfly valve in the steam pipe

which partly closes and throttles the steam when the engine begins to overspeed ; the reverse action takes place when the speed falls off.

This slow-speed centrifugal governor is unsuited for modern steam engine practice ; and its place has been taken by various forms of centre-weighted or spring-loaded high-speed and crank shaft governors. The reasons for this change are given in the following discussion of the principles on which the design of an efficient centrifugal governor must be based.

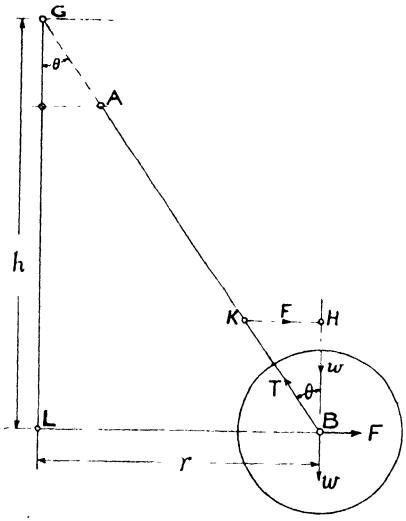



FIG. 263.

**209. Relation between Speed and Height of the Simple Pendulum.**—Referring to Fig. 263, let the pendulum arm AB, carrying the ball of mass $w$ lb., be pivoted either on a cross-head at A or at the spindle axis at G where the line BA cuts the axis. With either method of suspension, at a constant rotational speed of N rev./min. the ball centre B rotates in a circle of $r$ ft., and the arm describes a cone of height $h$ ft. At any point on the circle the ball is in equilibrium under the weight of the ball mass $w$, the pull of the arm T, and the (dynamical) centrifugal force F, acting radially outward. Draw the triangle of forces HBK for B. It is similar to the triangle LGB, and hence—

$$\frac{LB}{LG}=\frac{HK}{HB}=\tan\theta$$

$$\tan\theta=\frac{r}{h}=\frac{F}{w}$$

or

$$F=\frac{wr}{h} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

By the laws of dynamics the centrifugal force is given by—

$$F=\frac{w}{g}\omega^2 r \text{ lb.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

where $\omega$ is angular velocity in radians/sec., $r$ is in ft. and $w$ in lb.

$$\therefore \quad \frac{w}{g}\omega^2 r=\frac{wr}{h}, \text{ where } h \text{ is in ft.}$$

or

$$\omega=\sqrt{\frac{g}{h}}$$

If N=speed in rev./min., $\omega=\frac{2\pi N}{60}$

and $$\frac{2\pi N}{60}=\sqrt{\frac{g}{h}}$$

hence $$N=\frac{60}{2\pi}\sqrt{\frac{g}{h}}=\frac{54{\cdot}187}{\sqrt{h}} \quad . \quad . \quad . \quad . \quad . \quad (3)$$

or $$h=\frac{2936{\cdot}3}{N^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

In the case of the simple pendulum, therefore, the height of cone varies inversely as the square of the speed, and is independent of the radius and the weight of the ball.

**210.** In any form of pendulum governor, however, the pendulum arm and ball form only a part of a linkage which is necessary for the operation of the steam regulating gear. It is connected by suspension links to the operating sleeve on the governor spindle. The weight of the ball arm (especially in a slow-speed governor), the weight of the suspension links, the weight of the sleeve, the length of the arm and link, and the particular points of suspension of the links, together with any central load and frictional resistances, all affect the speed at which the balls will rotate, for a given height.

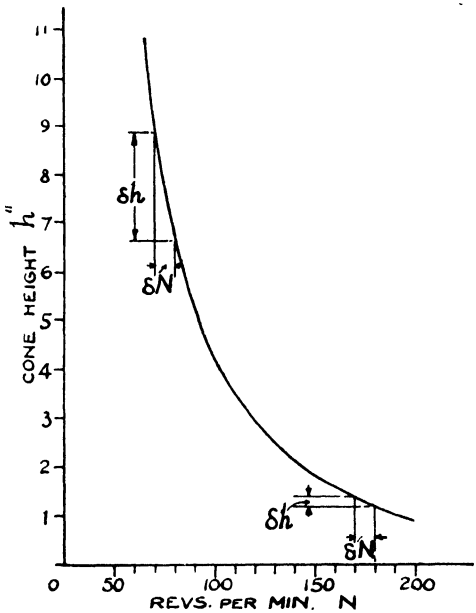



FIG. 264.

Whatever the nature of the linkage, suspension, or loading, the centrifugal force on each ball, at any speed N, is given by the fundamental equation (2), which can be written in the form—

$$F=\frac{wN^2r}{2936{\cdot}3} \quad \text{or} \quad F=0{\cdot}00034wN^2r \quad . \quad . \quad . \quad (5)$$

and the speed is—

$$N=54{\cdot}187\sqrt{\frac{F}{wr}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

The condition that the height of the cone of revolution of the pendulum varies as the square of the speed restricts the case of the

simple or Watt type to low speeds. While the change of height for a given change of speed may be sufficient for the proper travel of the governor sleeve at a low rate of rotation, it may be quite inadequate for the same change at a much higher rate.

This is clearly shown by the curve of height plotted to a base of speed for a Watt governor, with effect of sleeve, etc., included (Fig. 264). The rate of change of height with speed, or $\delta h/\delta N$, between 70 and 80 rev./min. is large as compared with the rate between 170 and 180 rev./min. The higher the speed the worse the condition becomes.

In the development of the steam engine, as speeds increased, this

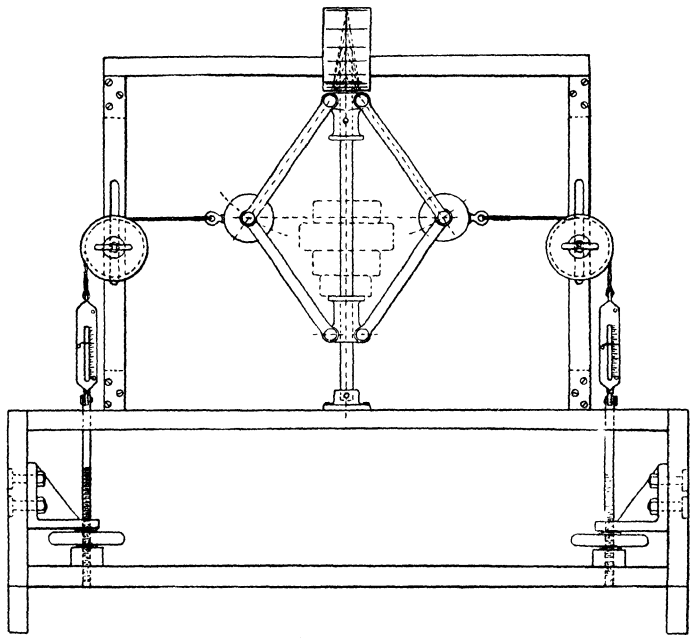


FIG. 265.

limitation of the pendulum governor, together with the demand for increased "powerfulness," led to the introduction of various forms of loaded governor. In these governors the "controlling force," that is, the inward radial pull on the ball opposing the centrifugal force at any given speed, is greatly increased by the addition to the sleeve of a central load, either dead weight or spring.

**211. Controlling Force.**—Where a governor is run from the condition of rest up to the rated speed N, at some radius $r$, the outward motion of the ball is stopped by the inward pull of the controlling force C which just balances the outward (dynamical) centrifugal force F on the mass of the ball at this speed.

The controlling force for any governor linkage, load, and configura-

tion of the governor, when friction is neglected, can be determined approximately when the governor is at rest by pulling on the balls with spring balances, as indicated on the experimental model for a loaded governor shown in Fig. 265, and tapping the spindle with a mallet to eliminate friction at the sleeve and joints. In general, it can be obtained by the ordinary methods of graphical statics, or by calculation for the configuration of the linkage corresponding to various positions of the balls. The spring balance readings, when plotted to a base of ball circle radius ($r$), give a curve of controlling force (C) per ball.

Now for a constant speed N, Equation 5, for the centrifugal force on a ball, is that of a straight line passing through the origin. If a series of such lines, usually termed "speed lines," is drawn to a base of radius, then since the condition for constant speed is C=F, the intersection of a given speed line with the curve of controlling force gives the corresponding ball circle radius; or conversely, at a given radius the intersection with the controlling force curve defines the speed. This system of representation was introduced by Wilson Hartnell, and is valuable for the study of the characteristic properties of any centrifugal governor.*

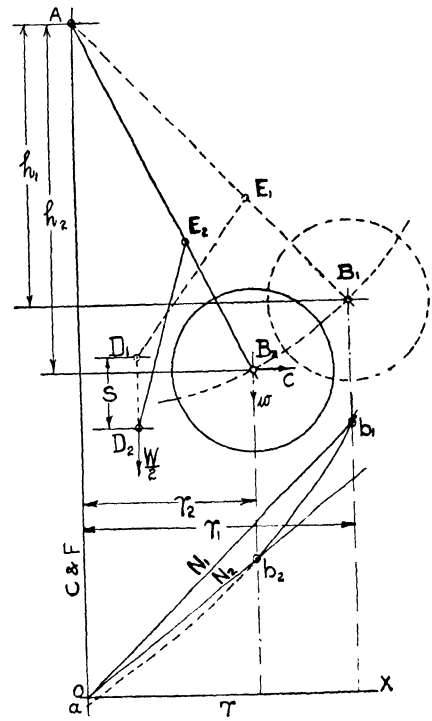



FIG. 266.

**212. Controlling Force Curves and Speed Lines on Characteristic Diagram.**—The configuration of the linkage of a Watt governor is shown in Fig. 266, for the upper and lower positions of the sleeve, which has a total travel S. The subscripts 1 and 2 denote the "full-out" and "full-in" positions of the ball centres and link joints. In this case the ball arm is pivoted at A on the spindle axis, and as is usually the practice, the lifting link DE is pivoted on the sleeve to the right of the axis. Its upper end is pivoted at E slightly below the mid point of the arm AB. The governor is really a loaded one, as the controlling force (friction neglected) has to overcome the

* See Paper on "Automatic Expansion Gear," by Wilson Hartnell, *Proc. Inst. Mech. E.*, August 1882.

downward load of the sleeve and links in addition to the ball weight. The downward load on the right is half of the central load, or W/2.

By one of the methods given later the controlling forces at various radii between $r_2$ and $r_1$ have been determined, and the controlling force curve $b_2b_1$ has been plotted. If the linkage could permit the ball arm to move into the vertical position, the curve would be continued to the left as shown by the dotted portion. The straight lines drawn from the origin through $b_2$

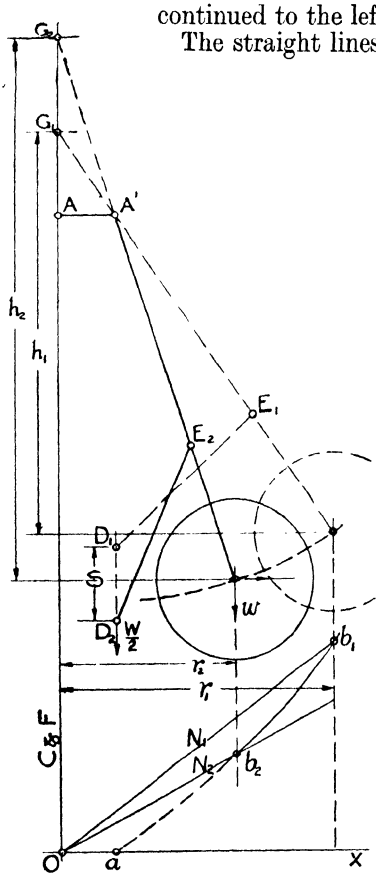



Fig. 267.

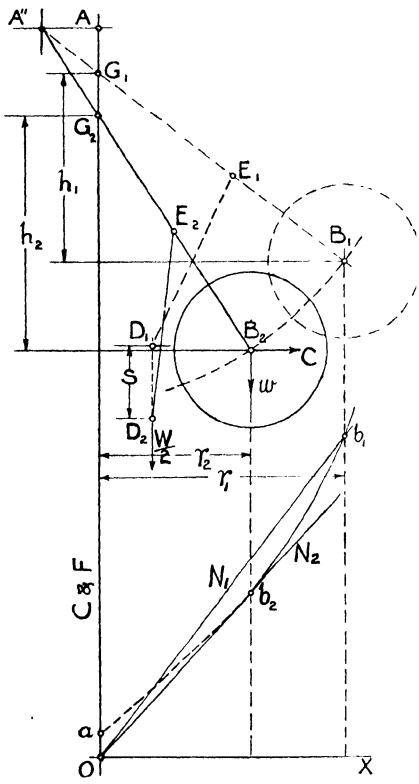



Fig. 268.

and $b_1$ give the speeds of rotation $N_2$ and $N_1$, at the respective radii $r_2$ and $r_1$. These are obtained by substituting the corresponding values of C in Equation 6.

This combination of controlling force and speed lines may be termed the characteristic diagram of the governor. Using the same linkage and sleeve the configuration of the governor with "open arms," pivoted off the axis at A', is shown in Fig. 267, and with "crossed arms, pivoted at A", in Fig. 268. The corresponding controlling force curves and speed lines are shown below the governor,

as in Fig. 266. It will be noted that in Figs. 266 and 267 at any radius the curve to the left of the point $b$ falls below the speed line drawn through the point, throughout the whole range of the curve.

In Fig. 268, however, at smaller radii than $r_2$, the curve falls above the speed line. In fact, for a short distance below $b_2$, the $N_2$ line and the dotted curve are nearly coincident.

**213. Stability of a Governor.**—A governor is said to be stable or static when it maintains a definite position of equilibrium at a given speed. It is unstable or astatic when at a given speed it assumes indifferently any position throughout its range of movement. In a stable governor (apart from friction effect), a small finite change in speed from the value which corresponds to a given configuration, causes the balls to move out or in by a finite amount, and to attain a new position of equilibrium at the new speed.

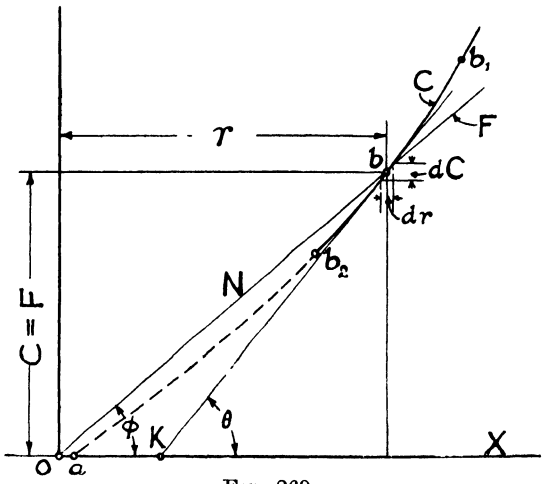



FIG. 269.

The condition for stability is that the ratio of the change of controlling force to change of radius must be greater than the ratio of the controlling force to the radius. The graphical interpretation is shown in Fig. 269, in which the controlling force curve $b_1b_2$ for the Watt governor is reproduced. If the governor ball is rotating at radius $r$ and change of speed causes an increase $dr$, the corresponding change of C is $dC$. The ratio $\frac{dC}{dr}$ is the tangent of the angle $\theta$, the line K$b$ being drawn tangential to the curve $b_2b_1$ at $b$. The speed line corresponding to F=C is O$b$, and the ratio $\frac{C}{r}=\frac{F}{r}$ is the tangent of the angle $\phi$ which this line makes with the axis OX. For stability—

$$\frac{dC}{dr}>\frac{C}{r}>\frac{F}{r}$$

In other words, the angle $\theta$ must be greater than $\phi$. It will be obvious that for any configuration of the governor, if the curve of controlling force falls below the speed line, to the left of the point of intersection $b$, the governor will be stable. Obviously the greater the excess of angle $\theta$ over angle $\phi$ the more stable the governor. When, however, the curve of controlling force reduces to a straight line through the origin, this line also becomes the speed line at all radii ;

$\theta=\phi$, and $\frac{dC}{dr}=\frac{C}{r}$.

In a case of this kind the governor is said to be " isochronous." The balls may take up any position within the range of their motion. The very slightest alteration of speed is sufficient to throw them to the one extreme or the other of the range. This kind of governor is quite impracticable. It continually oscillates or " hunts," and causes the engine speed to oscillate with it.

Where, as in the case illustrated in Fig. 268, at a certain speed the speed line is tangent to the controlling force curve, and the curve thus falls above the line to the left of the point, a state of instability is reached, and $\frac{dC}{dr}<\frac{C}{r}<\frac{F}{r}$, and the governor will fail to act below this speed.

**214. Sensitiveness of a Governor.**—Provided the condition of stability is satisfied, any change of speed of a governor would, if it were free from the influence of frictional resistance, cause a change of the position of the balls, that is, a change of radius. Hence only one position of the stable governor would be possible for any one speed. It will be evident, from this consideration, that a stable governor cannot maintain a perfectly constant speed of the engine it controls, because when either the steam pressure or the external load varies a certain displacement of the balls must occur before the governor can regulate the steam supply to suit the changing conditions. The maximum change of speed of the engine under governor control is that which corresponds to the extreme range of the governor balls, that is, from the position (2) for full steam supply to position (1) for no steam supply (Fig. 266). When a small variation of speed is sufficient to produce this result or to cause the full displacement of the governor sleeve, the governor is said to be " sensitive." The sensitiveness of a governor is defined in various ways ; but the usual meaning attached is, the ratio of the difference between its maximum and minimum speed to the mean speed, or

$$\text{Sensitiveness} \quad v=\frac{N_1-N_2}{N}=\frac{N_1-N_2}{\frac{1}{2}(N_1+N_2)} \quad . \quad . \quad . \quad . \quad . \quad (7)$$

where $N_2$ and $N_1$ are lower and upper limits of speed and N the mean value.

A more suitable term for this ratio is " static fluctuation,"

and a governor with a small static fluctuation may be termed a "sensitive" governor. An isochronous governor is an infinitely sensitive one, since $N_2=N_1$ and the static fluctuation is reduced to zero.

*Example* 1.—In the case of the Watt governor, Fig. 266, the controlling force at the full-in position (2) is 10·5 lb., the radius being 5·25 in. At the full-out position (1) the values are 20·5 lb. and 8·25 in. Calculate the static fluctuation for the full range, and the free variation (that is, without friction) from the mean speed. The weight of each ball is 17 lb.

Applying equation (6)—

$$N=54{\cdot}187\sqrt{\frac{C}{wr}}$$

$$N_2=54{\cdot}187\sqrt{\frac{10{\cdot}5\times12}{5{\cdot}25\times17}}=64{\cdot}5 \text{ rev./min.}$$

and $$N_1=54{\cdot}187\sqrt{\frac{20{\cdot}5\times12}{8{\cdot}25\times17}}=71{\cdot}5 \quad ,,$$

Static fluctuation $v=\frac{N_1-N_2}{\frac{1}{2}(N_1+N_2)}=\frac{7}{68}=0{\cdot}103$, or 10·3 per cent.

The variation from the mean speed is 5·15 per cent.

The effect of crossing the pendulum arms as shown in Fig. 268 is to make the governor more sensitive, that is, to reduce the static fluctuation. When the offset AA″ and lengths of the links and arm are so arranged that the path of the ball centre is approximately a parabola, the control force curve would reduce to a straight line through the origin, the condition as already pointed out for isochronism. By off-setting the arm to A′, as in Fig. 267, the governor is made more stable and the static fluctuation is increased.

**215. The Effect of Frictional Resistances.**—When the governor is rotating and governing the engine through the medium of the steam control mechanism, the central load W, or resistance of the frictionless governor at the sleeve at any given speed, is increased by some amount R, due to internal frictional resistance of the governor itself, and the resistance of the external mechanism it has to set in motion before a change of height of the sleeve can take place.

The central load per ball becomes $\frac{W+R}{2}$, and the controlling force is in consequence increased at a given radius. Hence before the ball can move out, the centrifugal force, with no frictional resistance, must be increased until balance with the new control force is reached. The result of the frictional detention, transferred to the sleeve, is thus an increase or decrease of speed at constant

radius. Thus at a given radius while the sleeve remains motionless the speed may vary from a higher limit $N_h$ to a lower limit $N_l$. The speed variation, reckoned on the mean value, is—

$$q = \frac{N_h - N_l}{\frac{1}{2}(N_h + N_l)} = \frac{N_h - N_l}{N} = \frac{2\left(\sqrt{\frac{C_h}{C_l}} - 1\right)}{\left(\sqrt{\frac{C_h}{C_l}} + 1\right)} \quad . \quad . \quad (8)$$

This ratio is given various names: "detention by friction" appears to be a suitable term. This lagging effect is conditioned by the particular design of governor, the efficiency of the lubrication, the effect of vibration in neutralising friction, and the character of the operative gear the governor moves. The value of $q$ may vary from 0·5 to 4 per cent.

When the value of $\frac{W+R}{2}$ is provisionally fixed, the corresponding values of the controlling force can be calculated and plotted to a base of radius. This curve, for rising speed, will lie above the control force curve for the frictionless governor.

When a decrease in speed occurs the transferred friction force acts negatively, that is, resists the downward motion of the sleeve, so that the downward central force on the sleeve becomes $\frac{W-R}{2}$, and the controlling force at the given radius will fall below the value for the frictionless governor before the sleeve moves downward. The corresponding controlling force curve then lies below the curve for the frictionless governor.

The three controlling force curves for a centre-loaded governor are shown in Fig. 270. The controlling force curve for the frictionless governor is $b_2b_1$. The curves with frictional detention are $d_2d_1$ for decreasing and $e_2e_1$ for increasing speed, the conditions accompanying increase and decrease of load on the engine. When the balls are rotating at the in position $r_2$, if a decrease of load occurs the engine will speed up until the governor can move the sleeve up and operate the steam supply mechanism. Before the sleeve can rise, the centrifugal force must increase from $hb_2$ to $he_2$ and hence the speed increases from $N_2$ to $N_{h2}$ at the constant radius $r_2$. Conversely at the full-out position $r_1$, where the engine is running on a light load, a decrease of load will cause an increase of engine speed, and the speed of the governor will rise from $N_1$ to $N_{h1}$ before the sleeve will rise. If the external load increases, the centrifugal force has to fall to the value $gd_1$ before the frictional resistance at the sleeve is balanced, and the speed will fall from $N_1$ to $Nl_1$, given by the line drawn through $d_1$, before the sleeve moves down. The detention by friction ($q$), in any case, at the lower and upper radii or any intermediate value, is calculable from Equation 8 when the corresponding speeds or

control forces are determined. The "retarded sensitiveness" is given by—

$$v_f = \frac{N_{h1} - N_{l2}}{\frac{1}{2}(N_{h1} + N_{l2})} \quad \ldots \ldots \quad (9)$$

*Example* 2.—Find from the controlling force curves of the loaded governor, Fig. 270, the static fluctuation or the "sensitiveness" between the limiting radii of 6 and 8 in. and the retarded sensitiveness. The weight of each ball is 10 lb.

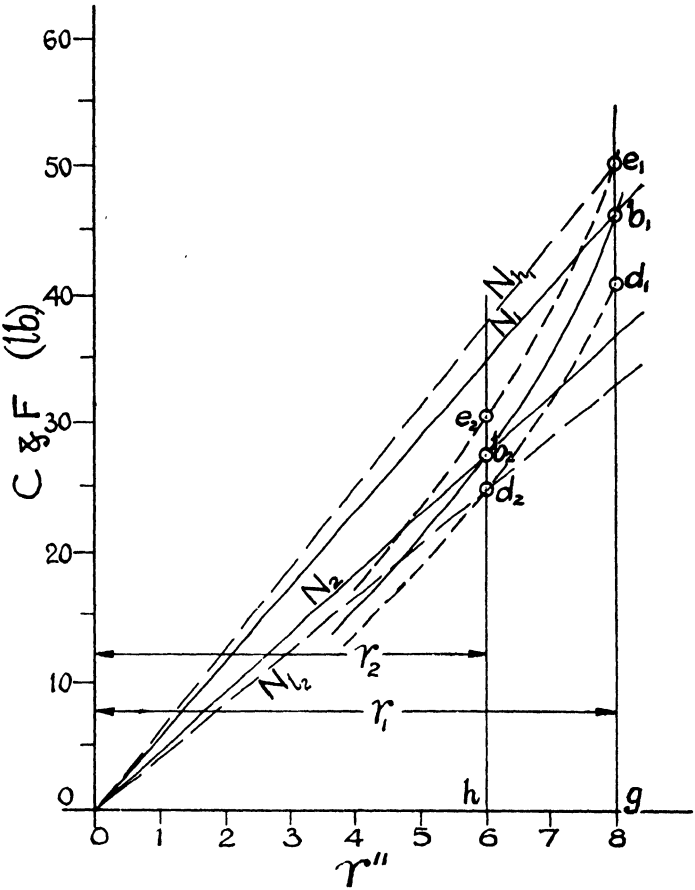



FIG. 270.

From the free control curve $b_2b_1$ the controlling force values are $C_1=46$ at $r_1=8$ in., and $C_2=27{\cdot}8$ at $r_2=6$ in. The corresponding speeds from equation (6) are—

$$N_1 = 54{\cdot}187\sqrt{\frac{46\times12}{8\times10}} = 142 \text{ rev./min.}$$

$$N_2=54{\cdot}187\sqrt{\frac{27{\cdot}8\times12}{6\times10}}=128 \text{ rev./min.}$$

$$\therefore \text{ Sensitiveness } v=\frac{N_1-N_2}{\frac{1}{2}(N_1+N_2)}=\frac{14}{135}=0{\cdot}104$$

The speed thus varies about 5·2 per cent. above and below the mean, on the extreme range with no friction.

Taking account of the friction, from curve $e_2e_1$ the control force is $C_{h1}=50$ at $r=8$ in., and from curve $d_2d_1$ $C_{l2}=25$ at $r=6$ in.

$$\therefore N_{h1}=54{\cdot}187\sqrt{\frac{50\times12}{8\times10}}=148{\cdot}5 \text{ rev./min.}$$

$$N_{l2}=54{\cdot}187\sqrt{\frac{25\times12}{6\times10}}=122$$

The retarded sensitiveness is—

$$v_f=\frac{N_{h1}-N_{l2}}{\frac{1}{2}(N_{h1}+N_{l2})}=\frac{26{\cdot}5}{135{\cdot}2}=0{\cdot}195$$

The variation is here increased to nearly 10 per cent. above and below the mean. This condition, however, is for the extreme case of complete unloading or loading of the engine.

*Example* 3.—In the case of the governor (Example 2) find the detention by friction at the inner and outer limits $r_2$ and $r_1$.

From the curves $e_2e_1$ and $d_2d_1$, $C_{h1}=50$, $C_{l1}=40{\cdot}8$ at $r_1=8$ in.
,, ,, ,, $C_{h2}=30{\cdot}5$, $C_{l2}=25$ at $r_2=6$ in.

Then $$q_8=\frac{2\left(\sqrt{\frac{C_{h1}}{C_{l1}}}-1\right)}{\left(\sqrt{\frac{C_{h1}}{C_{l1}}}+1\right)}=\frac{2(1{\cdot}1-1)}{2{\cdot}1}=\frac{0{\cdot}2}{2{\cdot}1}=0{\cdot}095$$

$$\frac{C_{h1}}{C_{l1}}=\frac{50}{40{\cdot}8}=1{\cdot}22$$

$$\frac{C_{h2}}{C_{l2}}=\frac{30{\cdot}5}{25}=1{\cdot}22$$

Similarly $$q_6=\frac{2\left(\sqrt{\frac{C_{h2}}{C_{l2}}}-1\right)}{\left(\sqrt{\frac{C_{h2}}{C_{l2}}}+1\right)}=\frac{0{\cdot}2}{2{\cdot}1}=0{\cdot}095.$$

Hence a variation of fully 4½ per cent. will take place up or down at either end of the range before the governor sleeve is moved from a given position. This will also be the approximate variation at any intermediate position of the balls.

**216. Hunting.**—An oversensitive governor may cause speed irregularity of the engine known as "hunting." The sensitiveness permissible in the governor is conditioned to some extent by the variation permitted by the fly-wheel. If the cyclic variation is greater than the sensitiveness the governor will oscillate. The fly-wheel and governor have to be designed so that the sensitiveness of the governor at normal load may be greater than the cyclic variation, so that the engine may run steadily. It may happen,

however, that at light loads this condition is not fulfilled, and the governor will hunt, if very sensitive. Except where very close speed regulation of the engine is necessary, as for instance in an engine driving an electric generator, the sensitiveness may be kept fairly large.

This condition more or less counteracts the time lag between any change of load on the engine and the change of supply of energy by the steam. If the governor operates a throttle and acts quickly on a decrease of load, it shuts the throttle partly, but the speeding up continues, due to the reserve steam flowing from the valve chest, with the result that the governor overcontrols and further closes the throttle, and there is insufficient steam supply even for reduced load. When the speed goes down again a reverse action occurs. In addition to this condition of over and under supply of steam, friction of the governor and its gear may cause too great a time lag in its action, with the result that when it does act, at the increased speed it overruns the limit for sufficient steam supply, and vice versa.

**217. Governor Power.**—Referring again to Fig. 270, when the balls are pulled out, in the case of the frictionless governor, from $r_2$ to $r_1$, the controlling force increases from the value $hb_2$ to $gb_1$, or from $C_2$ to $C_1$. The area below the force-distance curve $b_2b_1$ thus gives the energy expended per ball in changing the configuration of the governor and raising the sleeve through its full travel. The work thus done and stored in the governor at the full-out position, and which is restored when the balls and sleeve return to the full-in one, is called the "governor power." This was the term originally employed by Hartnell in his valuable paper, and is now in common use, although "energy" would be a more suitable expression, since power means a time rate of performance of work. The governor power per ball is thus given by the product of the mean value of the controlling force in lb. between $r_2$ and $r_1$, and the difference of radii in ft.

For the two balls—

$$\text{Total governor power } E = 2C_{mean}(r_1 - r_2) \text{ ft.-lb.} \quad . \quad . \quad (10)$$

or if the planimetered area $hb_2b_1g$ is A in.$^2$, $y$ the scale of force in lb., and $x$ the scale of radius in ft.

$$E = 2xyA \text{ ft.-lb.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$$

The work done in actual service against friction of the governor and its attached mechanism is given by the area $b_2b_1e_1e_2$ for the outward displacement of the ball, and by $b_1b_2d_2d_1$ for the inward return. These areas, as a rule, are not equal unless, in the case of throttle governing, the throttle valve is well balanced.

Hartnell stated that "in order to be ample the governor power should be, say, 20 times the friction to be overcome; it may be 40 times with advantage." For link expansion gears he also stated

that from 5 to 8 ft.-lb. per I.H.P. would be sufficient. In a later publication he gave the empirical rule—

$$E = Ad^2 \text{ ft.-lb.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (12)$$

where $d$=cylinder diameter in inches, and A=constant, varying from 0·5 to 0·7 for link expansion gear, and 0·2 for throttle valves. These figures refer to the spring-loaded types of governor, in which far higher values of power can be obtained than in the case of the centre-weight governor. In the latter type the power is limited by the size of central weight which can be got in between the arms, and which in turn fixes the size of ball for a given speed. The governor power is, in consequence, always a low figure.

*Example* 4.—Find from the controlling force curves of the centre-weighted governor (Fig. 270) the ratio between the friction work and the governor power. On the original drawing the scale of radius is half full size, and the force scale is 1 in.=10 lb. When planimetered the power area $hb_2b_1g$=3·55 in.$^2$, and the friction area $d_2e_2e_1d_1$ =0·712 in.$^2$

Length scale 1 in.=$\frac{2}{12}$ ft.=$x$
Force scale 1 in.=10 lb.=$y$
Work scale 1 in.$^2$=$\frac{20}{12}$=1·666 ft.-lb.=$e$.

The total power is—

$E = 2eA = 2 \times 1{\cdot}666 \times 3{\cdot}55 = 2 \times 5{\cdot}9 = 11{\cdot}8$ ft.-lb. $\quad A = 3{\cdot}55$

$a = 2ea' = 2 \times 1{\cdot}666 \times 0{\cdot}712 = 1{\cdot}185 \times 2 = 2{\cdot}370$ ft.-lb. $\quad a' = 0{\cdot}712$

$$\frac{E}{a} = \frac{3{\cdot}55}{0{\cdot}712} = \frac{4{\cdot}85}{1}$$

The power is less than five times the friction work, which is a small value. The figure could be improved by the use of a larger centre weight if room were available.

**218. Calculation of Controlling Force of Centre-Weight Governor.**—Three representative types of the centre-weight centrifugal governor are shown in Figs. 271, 273, and 275. The loaded type was first introduced by the American engineer Porter, to obtain a high-speed governor with small balls, more powerful than the older Watt type. The Porter governor, as shown in Fig. 271, usually has the ball arm and suspension link equal in length, and jointed on the ball axis. The ball arm and link may either be pivoted on the axis, or, as shown here, may be equally off-set from it. In some instances the arms are crossed to obtain greater sensitiveness. The type shown in Fig. 273 is simply a centre-loaded Watt governor. In some designs of it a spiral spring is fitted inside the weight and bears on a collar at the top of the spindle. A very considerable increase in the central load is thus made possible.

The third type, shown in Fig. 275, is known as a Proell governor. The ball arm is cranked and pivoted at the sleeve, and the suspension link is pivoted on the upper crosspiece. Usually the link and arm

lengths are equal. In none of these cases need the arms necessarily be equal. The ball arm arrangement of the Proell governor enables the sensitiveness of the governor to be reduced until an almost isochronous condition is reached.

**219.** The controlling force of any of these types can be obtained either graphically or by calculation. Both methods are given here.

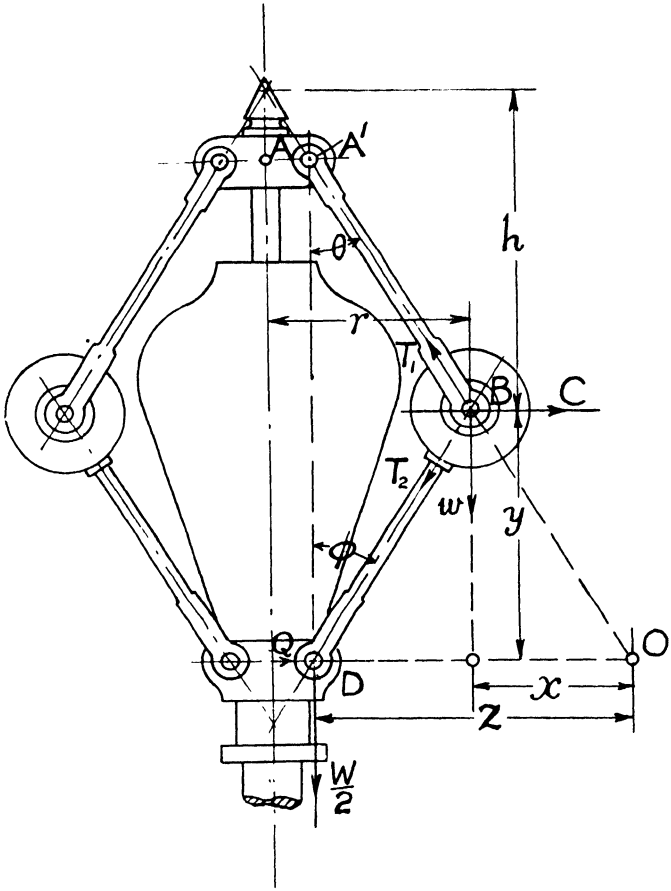



FIG. 271.

Take first the case of the Porter governor. For generality let the ball arm AB and suspension link BD (Fig. 271) be unequal.

Draw the configuration of the linkage for the given position of the ball. Consider the static equilibrium of the suspension link BD under the action of the external system of forces: the load $\frac{W}{2}$ and the horizontal reaction Q at D; the pull $T_1$ of the arm A'B; the

weight of the ball mass $w$ and the controlling force C at B. The algebraic sum of the moments of this system about any point is zero, for equilibrium. Choose the point O, at the intersection of

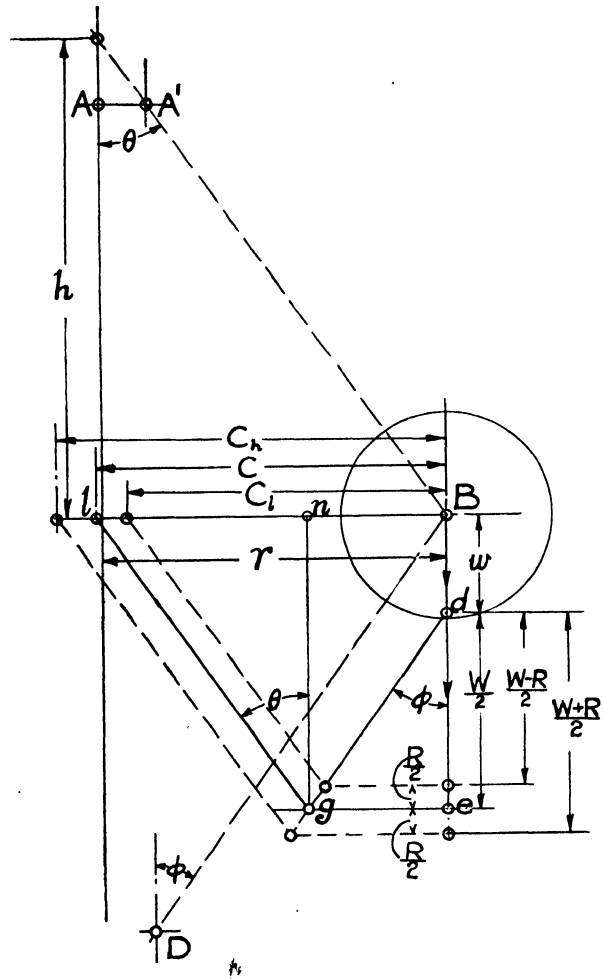



FIG. 272.

the horizontal through D and the arm AB produced. Then the moment of Q and of $T_1$ about O is zero,

and
$$Cy=\frac{W}{2}z+wx$$

$$\therefore \quad C=\frac{\frac{W}{2}z+wx}{y} \qquad \ldots\ldots\ldots \quad (13)$$

Instead of using this momental method, the conditions of equilibrium may be obtained by drawing the polygon of forces for the ball. This is under the action of the system: $w$, $T_1$, C, and the downward pull $T_2$ of the suspension link BD. The pull $T_2$ is obtained by first drawing the triangle of forces for the point D.

In Fig. 272, scale vertically, $Bd=w$, and $de=\frac{W}{2}$; draw *eg* hori-

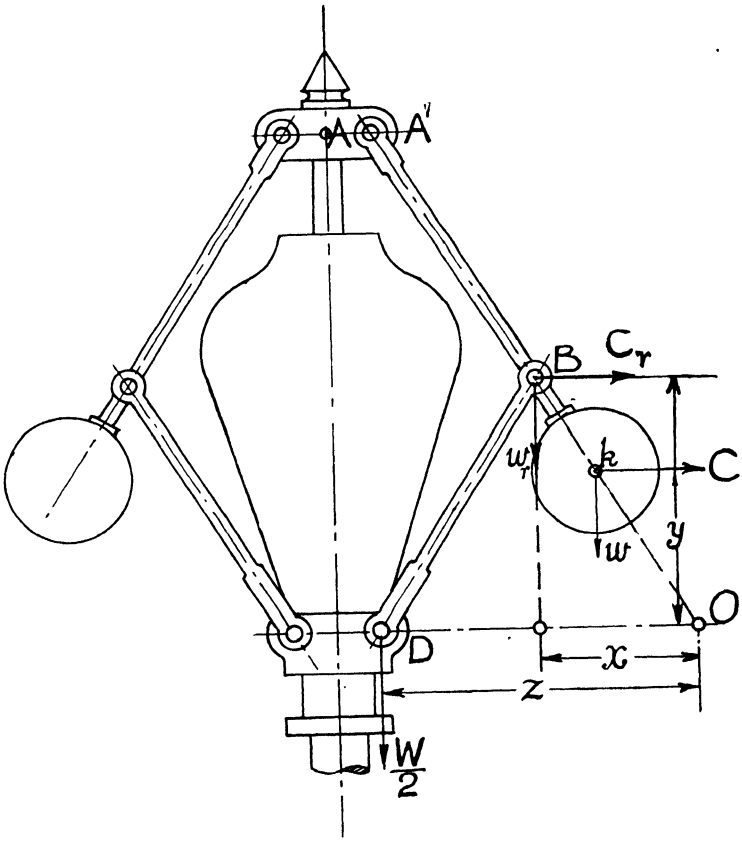



FIG. 273.

zontally, and *dg* parallel to BD. Then *deg* is the force triangle for D, and $dg=T_2$. The directions of the other two forces $T_1$ and C are known. Draw *gl* parallel to the ball arm BA′ and B*l* horizontal to meet it in *l*. Then *l*B=C, the controlling force. It will be found most convenient to make the centre of the ball B, as shown, the starting-point of the polygon. The parallels to the links can then be more easily drawn.

The contruction has to be repeated for several values of the radius,

in order to obtain a sufficient number of points for the controlling force curve. The value of $w$ should include half the weight of the ball arm and the suspension arm, and W/2 should include half the weight of the suspension arm.

**220.** The value of C can be analytically calculated as follows. Let $\theta$=angle the ball arm makes with the vertical, and $\phi$ the angle the suspension arm makes with it. Then in Fig. 272 the angle $lgn=\theta$, and angle $gde=\phi$, and $ln=\left(w+\frac{W}{2}\right)\tan\theta$, $nB=ge=\frac{W}{2}\tan\phi$.

$$\therefore \qquad C=(ln+nB)=\left(w+\frac{W}{2}\right)\tan\theta+\frac{W}{2}\tan\phi$$

or
$$C=\frac{W}{2}(\tan\theta+\tan\phi)+w\tan\theta \quad . \quad . \quad . \quad . \quad . \quad (14)$$

By equating this against the corresponding centrifugal force, a general expression for the speed of rotation in terms of the head $h$, the angle $\phi$, and radius $r$ can be obtained.

Thus
$$C=F=\frac{wN^2r}{2936\cdot3}=\left(\frac{W}{2}+w\right)\tan\theta+\frac{W}{2}\tan\phi.$$

$$\text{Tan }\theta=\frac{r}{h} \quad \therefore \; N^2=\frac{2936\cdot3}{w}\left\{\left(\frac{W}{2}+w\right)\frac{1}{h}+\frac{W}{2}\frac{\tan\phi}{r}\right\} \quad . \quad (15)$$

In the special case where the links are equal $\theta=\phi$, and $\tan\theta=\tan\phi$. Substituting for $\tan\phi$ in (15), then—

$$N^2=2936\cdot3\left(\frac{W+w}{w}\right)\frac{1}{h} \quad . \quad . \quad . \quad . \quad . \quad (16)$$

That is, for the Porter governor with *equal* arms, the speed is inversely proportional to the square root of the head, or the head is inversely proportional to the square of the speed, the condition already established for the simple pendulum (Equation 4).

The general equation for the controlling force can be put in another way. From the principle of virtual velocities, since O (Fig. 271) is the instantaneous centre of the link BD, it follows that for a rise $\delta h$ of the ball the rise of sleeve or point D is $z/x\delta h$ or $k\delta h$. The work done in raising the ball and weight is—

$$w\delta h+\frac{W}{2}k\delta h=\left(w+k\frac{W}{2}\right)\delta h=w'\delta h$$

The same work would thus be done on a simple pendulum represented by the ball arm if a mass $w'=\left(w+k\frac{W}{2}\right)$ were (by moments) substituted for $w$. Hence by Equation 1—

$$C=F=\left(w+k\frac{W}{2}\right)\frac{r}{h} \quad . \quad . \quad . \quad . \quad . \quad . \quad (17)$$

or
$$N^2=\frac{2936\cdot3}{w}\left(w+k\frac{W}{2}\right)\frac{1}{h} \quad . \quad . \quad . \quad . \quad . \quad (18)$$

It should be noted that $k$ is *not* a constant, with unequal arms. It

varies with each position of the linkage. It is constant, however, for equal arms, and has the value $\frac{z}{x}=2$.

Hence for the *equal* armed Porter governor, Equation 18 becomes the same as 16.

The foregoing constructions and equations give the controlling force for the free or frictionless governor. They apply equally to the governor with frictional detention when the central force W/2 is replaced by the virtual load $\left(\frac{W\pm R}{2}\right)$, the positive sign being used for increasing and the negative sign for decreasing speed. The modified polygon of forces for the ball is shown in Fig. 272, the construction lines being dotted. The general equation (14) becomes—

$$C=\left(\frac{W\pm R}{2}\right)(\tan\theta+\tan\phi)+w\tan\theta \quad . \quad . \quad . \quad (19)$$

In the particular case where the links are of equal length it becomes—

$$\left.\begin{aligned} C&=(W\pm R+w)\tan\theta \\ \text{or}\quad C&=(W\pm R+w)\frac{r}{h} \end{aligned}\right\} \quad . \quad . \quad . \quad . \quad . \quad (20)$$

*Example* 5.—The Porter governor (Figs. 271 and 272) has equal links $10\frac{3}{8}$ in. long, pivoted at 1 in. from the spindle axis. The centre weight is 40 lb., and the weight of each ball is 10 lb. Find the controlling force and speed when the ball circle radius is 7 in.

In this case $\theta=\phi$, and Equation 17 becomes—

$$C=(w+W)\tan\theta=(w+W)\frac{r}{h}$$

Since A'B=10·375 in. and AA'=1 in., $r$=7 in.—

$$\text{Sin}\,\theta=\left(\frac{r-AA'}{A'B}\right)=\frac{7-1}{10\cdot375}=\frac{6}{10\cdot375}=0\cdot58 \text{ and } \theta=35^\circ\ 30'$$

$$\text{Tan}\,\theta=0\cdot7133 \quad \therefore\ C=(10+40)0\cdot7133 \qquad w=10$$
$$=0\cdot7133\times50=35\cdot666 \text{ lb.} \qquad W=40$$

Drawing the linkage (to half size scale) and taking force scale as 1 in.=10 lb., draw the force polygon B*degl*, and B*l*=C=35·6 lb. The student can check the value by the momental method. The height is

$$h=\frac{r}{\tan\theta}=\frac{y}{0\cdot7133}=9\cdot81 \text{ in.},$$

which checks on the drawing.

By Equation 6 the speed is—

$$N=54\cdot187\sqrt{\frac{C}{wr}}$$

$$=54\cdot187\sqrt{\frac{35\cdot67\times12}{10\times7}}=134 \text{ rev./min.}$$

Alternatively from Equation 16—

$$N=54{\cdot}187\sqrt{\left(\frac{W+w}{w}\right)\frac{1}{h}}=54{\cdot}187\sqrt{\frac{50}{10}\times\frac{12}{9{\cdot}81}}=134 \text{ rev./min.}$$

221. For the case of the loaded Watt governor (Fig. 273), the conditions of equilibrium can be considered by taking moments of the external forces on the ball arm about A′. It is easier, however, to replace the ball weight $w$ and controlling force C at $k$ by equivalent values $w_r$ and $C_r$ at the link joint B. Let the ratio $A'B/A'k=\rho$, then $w_r=\frac{w}{\rho}$ and $C_r=\frac{C}{\rho}$. Taking moments as before about the virtual centre O of the suspension link—

$$C_r=\frac{\frac{W}{2}z+w_r x}{y} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (21)$$

Then

$$C=\rho C r \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (22)$$

As in the case of the previous governor, the force polygon is drawn for the joint B, giving $C_r$, and this is graphically reduced to $C=\rho C_r$. From B (Fig. 274) scale off vertically $Bn=w$. Join A′ and $n$ and produce the line to meet the vertical through the ball centre at $q$, then $kq=w_r$. Scale off $kd=w_r$ and $de=\frac{W}{2}$, and complete the force polygon *kegl*. Then $kl=C_r$, the referred controlling force at B. Join A′ and $l$, and draw the horizontal BM through B to cut A′$l$ in M. Then BM=C, the controlling force on the ball at $k$.

As in the previous case, the referred controlling force at B is given by—

$$C_r=\frac{W}{2}(\tan\theta+\tan\phi)+w_r\tan\theta \quad . \quad . \quad . \quad (23)$$

so that

$$C=\rho C_r=\rho\frac{W}{2}(\tan\theta+\tan\phi)+w\tan\theta \quad . \quad (24)$$

and the speed is

$$N=54{\cdot}187\sqrt{\frac{C}{wr}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (25)$$

For the case of equal links $\tan\theta=\tan\phi$, and the equations reduce to—

$$C=(\rho W+w)\tan\theta=(\rho W+w)\frac{r}{h} \quad . \quad . \quad . \quad . \quad (26)$$

and

$$N=54{\cdot}187\sqrt{\left(\frac{\rho W+w}{w}\right)\frac{1}{h}} \quad . \quad . \quad . \quad . \quad . \quad (27)$$

also

$$h=\frac{2936{\cdot}3}{N^2}\left(\frac{\rho W+w}{w}\right) \quad . \quad . \quad . \quad . \quad . \quad . \quad (28)$$

The controlling force with frictional detention for increasing and decreasing speed is obtained by the same method as already

described, the values of $\frac{(W \pm R)}{2}$ being used in the construction of the force polygon. The construction is again clearly shown by the dotted lines in Fig. 274. In the equations (W+R) or (W—R) is to be used in place of W.

*Example* 6.—The loaded Watt governor, Figs. 273 and 274, has a

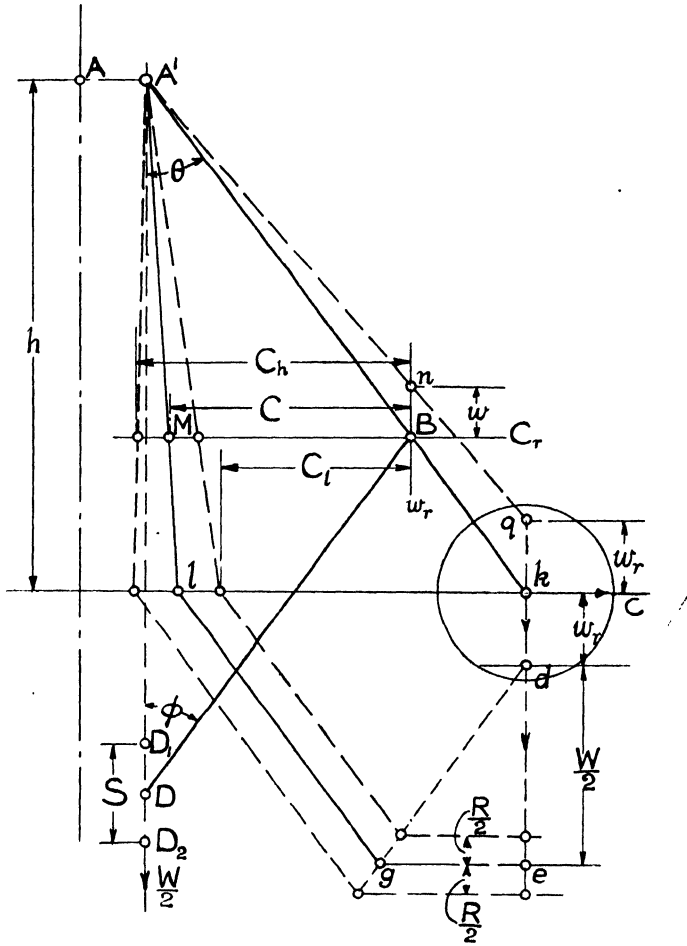



FIG. 274.

ball weight of 5 lb. and a centre weight of 40 lb. A'B=BD=9 in. AA'=$1\frac{1}{4}$ in., A'$k$=13 in., when the sleeve is in the mid position the radius of the ball circle is 9 in. If the internal and external friction is equivalent to an additional load of 6 lb. at the sleeve, find the speed of the governor at which the sleeve will rise and the percentage increase when the load on the engine decreases.

Here $\rho=\frac{9}{13}$, W=40, R=6, $r$=9 in.,

$$\text{Sin }\theta=\frac{r-AA'}{A'k}=\frac{9-1\cdot25}{13}=\frac{7\cdot75}{13}=0\cdot595$$

By Equation 24—

$$C_h=\rho\left(\frac{W+R}{2}\right)2\tan\theta+w\tan\theta$$

$$=\tfrac{9}{13}\times\tfrac{46}{2}\times1\cdot48+5\times0\cdot74$$

$$=23\cdot6+3\cdot7=27\cdot3\text{ lb.}$$

$\theta=36°\ 30'$
$\tan\theta=0\cdot74$
$w=5$
$(\tan\theta+\tan\phi)=2\tan\theta$

and $$C=\rho\frac{W}{2}\times2\tan\theta+w\tan\theta$$

$$=\tfrac{9}{13}\times20\times1\cdot48+3\cdot7$$

$$=20\cdot5+3\cdot7=24\cdot2\text{ lb.}$$

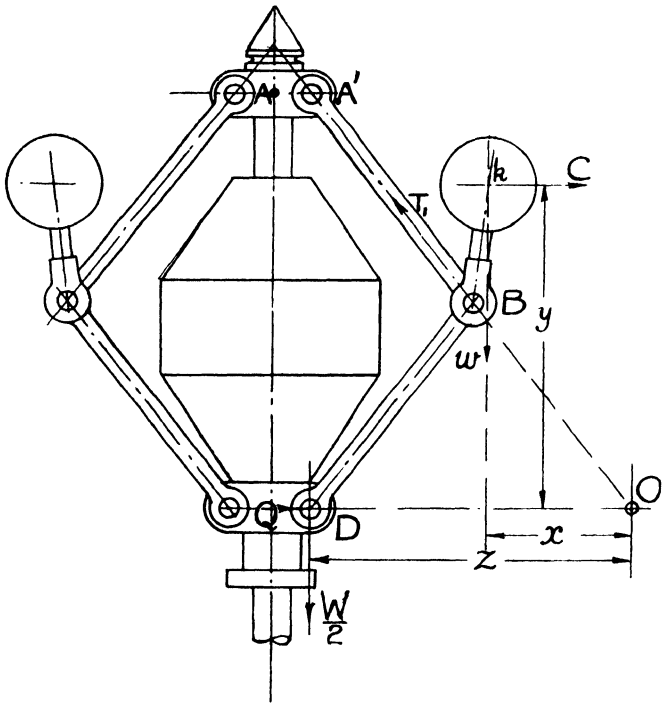



FIG. 275.

By Equation 6—

$$N_h=54\cdot187\sqrt{\frac{C_h}{wr}}$$

$$=54\cdot187\sqrt{\frac{27\cdot3\times12}{5\times9}}=146\text{ rev./min.}$$

$$\frac{N_h}{N}=\sqrt{\frac{C_h}{C}}=\sqrt{\frac{27\cdot3}{24\cdot2}}=\sqrt{1\cdot13}=1\cdot06$$

∴ Increase of speed is 6 per cent., before the sleeve moves.

222. In the case of the Proell governor (Fig. 275) the cranked ball arm, pivoted at D on the sleeve, is in equilibrium under the ball weight $w$, the controlling force C, the pull of the suspension arm A'B, the load $\frac{W}{2}$, and reaction Q at D. The condition of equilibrium can be obtained by taking moments either about the centre O or about D. Taking O, as before, the moments of $T_1$ and of Q become zero,

and $$Cy=\frac{W}{2}z+wx$$

or $$C=\frac{\frac{W}{2}z+wx}{y} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (29)$$

The values of $z$, $x$, and $y$ are scaled from the drawing.

The result can be obtained by the following entirely graphical method. Draw the diagonal OG (Fig. 276) and complete the rectangle DOEG. At G set up $Gg=w$. Join E and $g$. The line cuts the ball vertical at $l$. Draw $kn$ parallel to OG, set up $lm=\frac{W}{2}$ and draw $mn$ horizontally to cut $kn$ in $n$. Finally, draw the vertical $n$H to cut EG in H. Then $Hk=C$.

As in the previous cases, the values of controlling force with frictional detention, $C_h$ and $C_l$, are obtained by scaling up $\frac{W+R}{2}$ or $\frac{W-R}{2}$ from $l$ and completing the projection as shown by the dotted lines.

In order to obtain an analytical expression, consider the equilibrium of the lower arm, with respect to the point D at the sleeve. Here

$$Cy+wa=T_1v$$

and $$C=\frac{T_1v-wa}{y} \quad . \quad . \quad . \quad . \quad . \quad . \quad (30)$$

Join D and $k$, cutting A'BO in B'. The sum of the vertical components on the link A'B is $\left(\frac{W}{2}+w\right)$. Set up $fe=\left(\frac{W}{2}+w\right)$ from DO, then $oe=T_1$, $Dd=v$ is drawn at right angles to OB. From the geometry of the figure it will be seen that $\frac{v}{z}=\cos\theta$ or $v=z\cos\theta$ and $\left(\frac{W}{2}+w\right)/T_1=\cos\theta$.

$$\therefore T_1v=z\left(\frac{W}{2}+w\right);\quad a=y\tan\phi,\ z=(q+s)=B'b(\tan\phi+\tan\theta)$$

By proportion $B'b/y=DB'/Dk=\rho$, hence $B'b=\rho y$

$$\therefore \qquad C=\frac{\left(\frac{W}{2}+w\right)z-wy\tan\phi}{y}$$

or
$$C=\rho\left(\frac{W}{2}+w\right)(\tan\phi+\tan\theta)-w\tan\phi \quad . \quad . \quad . \quad (31)$$

The corresponding speed is obtained from Equation 6—

$$N=54{\cdot}187\sqrt{\frac{C}{wr}} \quad . \quad . \quad . \quad . \quad . \quad . \quad (32)$$

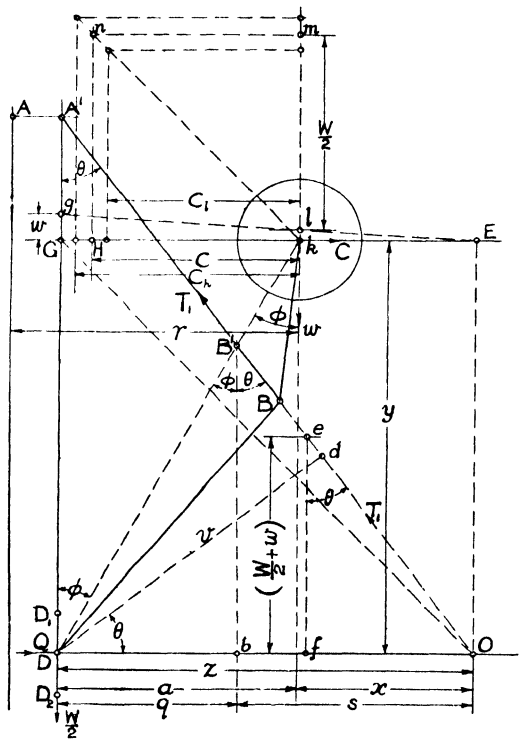



FIG. 276.

*Example* 7.—The Proell governor (Figs. 275 and 276) is to run at a normal speed of 185 rev./min. when the ball circle at mid lift is $r=9$ in. The proportions of the linkage are $A'B=11$ in., $DB=10\frac{1}{4}$ in., $Bk=5$ in., $AA'=1\frac{5}{8}$ in., diameter of ball $3\frac{3}{4}$ in. If the estimated external and governor resistance is 10 lb. at the sleeve and the detention by friction is not to exceed 5 per cent., find the necessary centre weight W.

The weight of a $3\frac{3}{4}$ in. cast-iron ball is 7·2 lb.$=w$.

For the frictionless governor, the controlling force at $r=9$ in. is, by Equation 5—

$$C=F=\frac{wN^2r}{2936\cdot3} \qquad \begin{matrix} w=7\cdot2 \\ N=185 \\ r=9 \end{matrix}$$

$$=\frac{7\cdot2\times180^2\times9}{2936\cdot3}=63 \text{ lb.}$$

By Equation 8, detention by friction—

$$q=\frac{2\left(\sqrt{\frac{C_h}{C_l}}-1\right)}{\left(\sqrt{\frac{C_h}{C_l}}+1\right)}$$

Let $$\sqrt{\frac{C_h}{C_l}}=k$$

then $$q=\frac{2(k-1)}{(k+1)}=0\cdot050$$

and $$k=\frac{2\cdot050}{1\cdot95}=1\cdot052$$

$$\frac{C_h}{C_l}=k^2=1\cdot052^2=1\cdot106\ ;\ \frac{C_h+C_l}{2}=C=63$$

$$\therefore \quad C_h=\frac{126}{1+\frac{1}{1\cdot106}}=\frac{126\times1\cdot106}{2\cdot106}=66\cdot16.$$

Drawing the linkage and scaling, $\rho=DB'/Dk=0\cdot74$ ; $\frac{R}{2}=5$, $w=7\cdot2$, $\theta=37°$, tan $\theta=0\cdot7536$, $\phi=30°$, tan $\phi=0\cdot5774$.

By Equation 31—

$$C_h=\rho\left(\frac{W+R}{2}+w\right)(\tan\phi+\tan\theta)-w\tan\phi$$

$$66\cdot16=0\cdot74\left(\frac{W}{2}+5+7\cdot2\right)1\cdot331-7\cdot2\times0\cdot5744$$

$$\frac{W}{2}=\frac{66\cdot16+4\cdot14}{1\cdot331\times0\cdot74}-12\cdot2$$

$$=(71\cdot4-12\cdot2)=59\cdot2$$

and W=118·4 lb.

Graphically, scale $C_h=66\cdot16$ from the ball axis, project to $kn$ produced, and $\frac{W+R}{2}$ scales 64 lb.

$$\therefore \frac{W}{2}=(64-5)=59 \text{ lb., and W}=118,$$

which is a close enough agreement with the calculated value. If $f$ is the additional controlling force at $r_1$ required to overcome frictional resistance for increase or decrease of speed—

$$f=(C_h-C)=(66{\cdot}16-63)=3{\cdot}16 \text{ lb.}$$

or $\frac{3{\cdot}16\times100}{63}=5$ per cent. of the free governor controlling force.

**223. Effect of Ball Arm and Link on the Centrifugal Force.**—As already pointed out, for the calculation of the static controlling force, half the weight of the ball arm and link should be added to the weight of the ball, and the sum taken as $w$. This procedure is not quite correct for the calculation of the centrifugal force. It can easily be shown that the centrifugal force at speed on the arm or link is the force due to the whole mass of the arm or link concentrated at a point one-third the length from the ball end. Hence, for a very exact result, half the mass may be added for the static calculation of controlling force, and one-third for the centrifugal force. To avoid undue complication on the characteristic diagram it is generally sufficient to neglect this refinement.

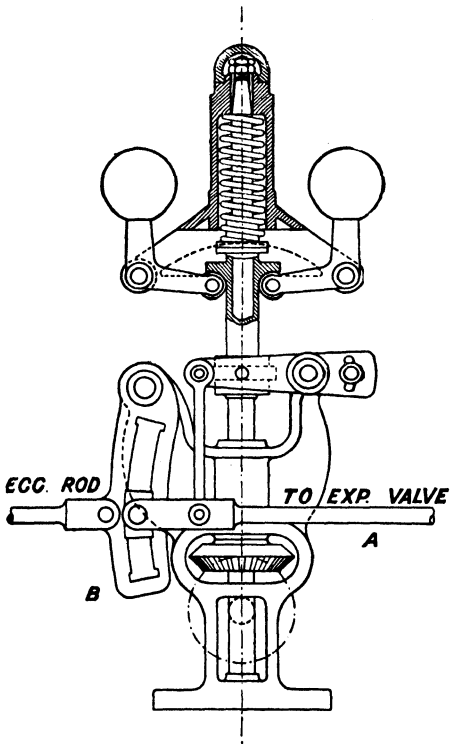



FIG. 277.

**224. Spring-loaded Vertical Governors.**—As already mentioned, the "powerfulness" of the centre-loaded governor is restricted by constructional conditions. By the substitution of a spring for the dead load, a large range of power may be obtained from a comparatively small governor running at high speed. Four types of vertical spring-loaded governors are shown in Figs. 277, 278, 279. The original form developed by Hartnell is shown in Fig. 277. A bonnet to which the cranked ball arms are pivoted is fixed on the spindle, and holds the loading spring which bears on the sleeve. The horizontal arm of each bell-crank has a roller which presses on the under side of the sleeve collar, and raises it against the spring

resistance as the balls move outward. The sleeve operates a lever on the standard, which raises the expansion valve rod A provided with a block which slides in the slot link B, driven by the eccentric rod. As the block rises, the valve travel decreases, and vice versa. Originally the expansion valve on the main valve, unlike the Meyer plates, was a single piece "gridiron" valve, that is, it had a large number of narrow ports, so that the full steam area could be obtained by a comparatively short travel. This arrangement was extensively used on small engines for expansion governing; but for this class of work one or other of the types of spring-loaded shaft governor is now commonly employed. The Hartnell governor, however, is still used, principally for throttle governing.

A spring-loaded type of Proell governor is shown in Fig. 278. The cylindrical head fixed on the spindle has arms to which the cranked levers L are pivoted at C. The ball arms are pivoted not on their centre lines but at points A outside them, so that the balls as they move out travel in practically a horizontal line. The lever rollers press down the compression plate above the controlling spring S. By giving sufficient initial compression to S the governor may be made as nearly isochronous as desired. A fine speed adjustment is provided, in the shape of an auxiliary spring E, fitted into a trunion sleeve F on the bracket at K. The plunger is attached to a sliding block in the inclined slot of the lever QPR. The block is adjusted by the hand-wheel W and screw. The push of this auxiliary spring, transferred to the sleeve, increases the central load. This gear enables the speed to be adjusted slightly when the engine is running.

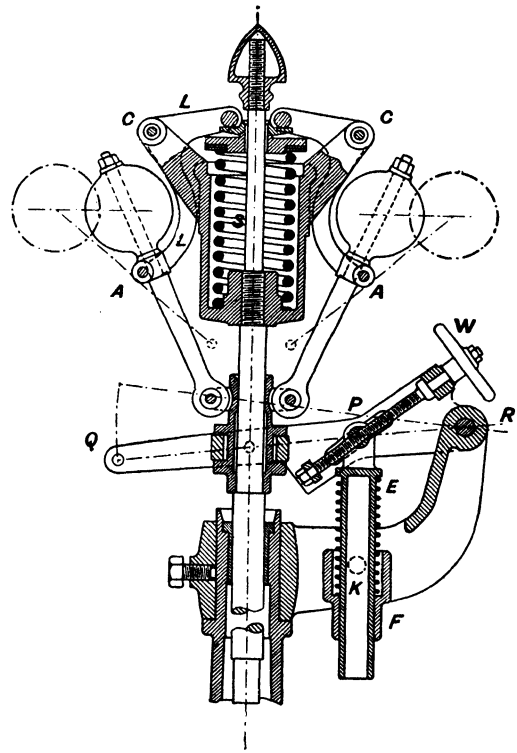


FIG. 278.

A different type, known as the Hartung governor, is shown in

Fig. 279. It is in general use in modern practice, and is employed in steam turbine work, not for operation of the steam control valves but the relay systems which do the governing.

The head 1 fixed to the spindle 2 carries two horizontal springs 3 contained in sliding cylindrical weights 4, which are operated by the bell-crank levers 5. The lower arm of each lever is attached to the sleeve by links 6.

This governor is very compact. The two cylinders replace the ball weights of the vertical type and the centrifugal force at each mass at speed is directly balanced by the controlling force of the spring. In this design the friction is reduced to a minimum. The

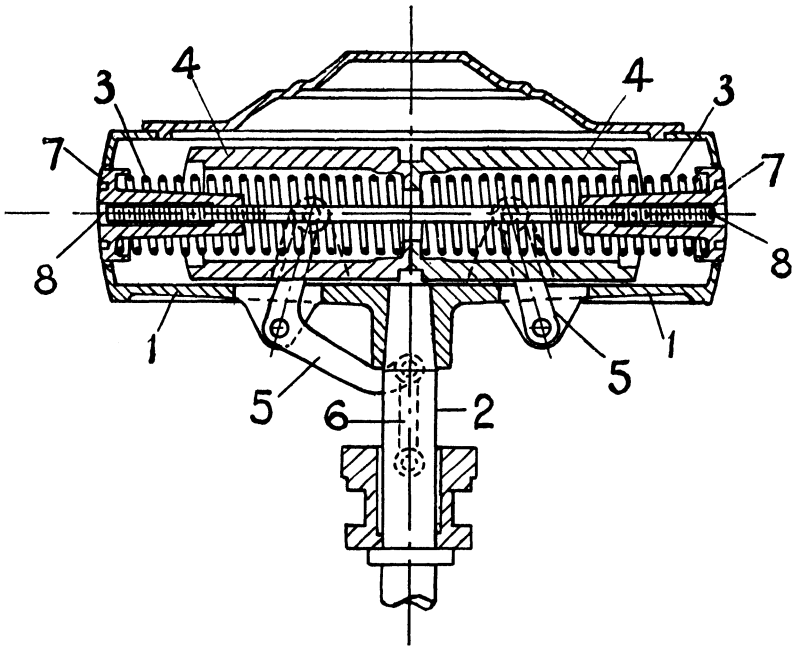



FIG. 279.

initial compression of each spring can be varied by the screwed sleeve 7 and central rod 8.

The type most commonly used for small engines is the Pickering governor, shown in Fig. 280. It is usually combined with a double-beat throttle valve V and a stop valve A. In this case the spiral spring is replaced by three coach springs fixed to the rotating head and the sleeve. Each ball is mounted on the middle of a spring. As the spring bends outwards under the centrifugal force on the ball mass it provides the direct controlling force, and the upper and lower halves act as ball arm and lifting lever, since the deflection at the centre causes a shortening of the length, between head and

sleeve. The outward displacement of the ball is limited by a hoop stop, fitted into the ball and passing round the spindle. The resistance of the plate springs can be increased by the auxiliary spring S, adjustable by the worm and screw. This is for fine regulation of speed.

**225. Characteristic Diagram of the Spring-loaded Governor.**—In the centre-weight or gravity type the axial sleeve load W of the frictionless governor remains constant as the balls move out. In the spring-loaded type it increases proportionally with the rise of the sleeve, that is, with the compression of the spring. At the lower

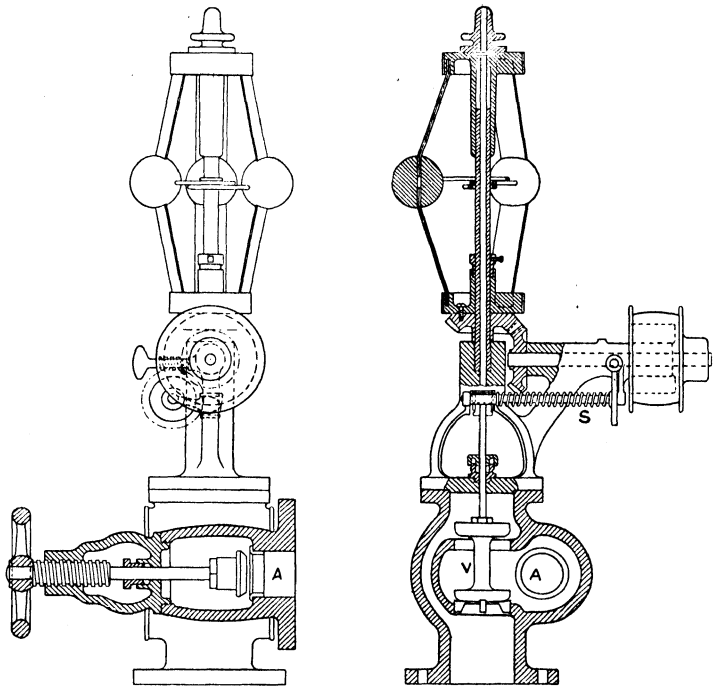



FIG. 280.

sleeve limit the spring is always given an initial compression. The substitution of a spring for a dead load reduces the control curve to an approximate straight-line form, as shown in the characteristic diagram for a Hartnell governor in Fig. 281. This drawing shows the ball arm in the middle position. At the full-in position $A_2$ with the sleeve at the bottom position the ball weight produces a negative turning moment about the central pin O. At the out position $A_1$ the moment is positive.

The sleeve load $\frac{W}{2}$ now consists of three parts: the weight of the sleeve and attached parts and the proportion of the weight of the

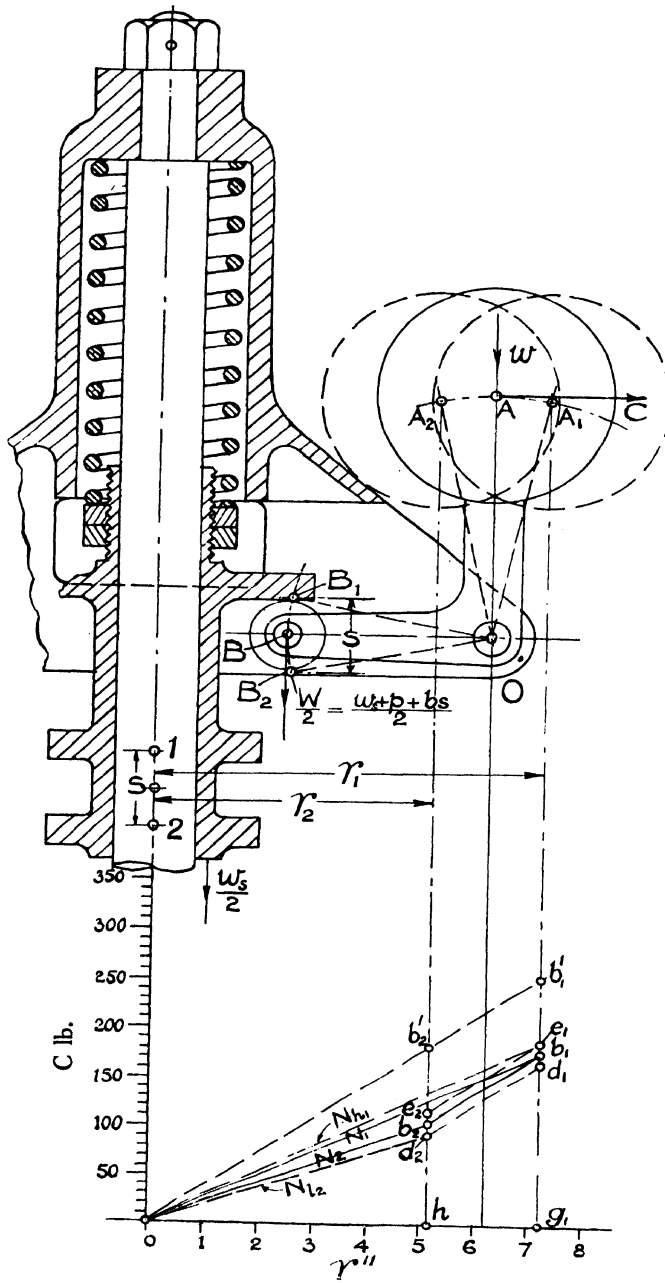

w
A2
A
A1
C
B1
B
S
B2
W/2 = w1+p+bs/2
O
1
S
2
r1
r2
ws/2
350
300
250
200
150
100
50
C lb.
b'1
b'2
e1
b1
d1
e2
b2
d2
NH1
N1
N2
N12
h
g1
0
1
2
3
4
5
6
7
8
r''

FIG. 281.

spring to be raised, which can be taken as one-third; the initial compression force $p$ lb.; and the additional compression force due to the displacement $s$ of the sleeve from the lowest position $B_2$, as shown in Fig. 282. If the strength of the spring, that is, the force required to compress it one inch, is $b$ lb., then the additional load due to a displacement of $s$ inches is $bs$. Hence—

$$\frac{W}{2}=\frac{(w_s+p+bs)}{2} \quad \ldots\ldots\ldots \quad (33)$$

In order to determine the controlling force for any position of the

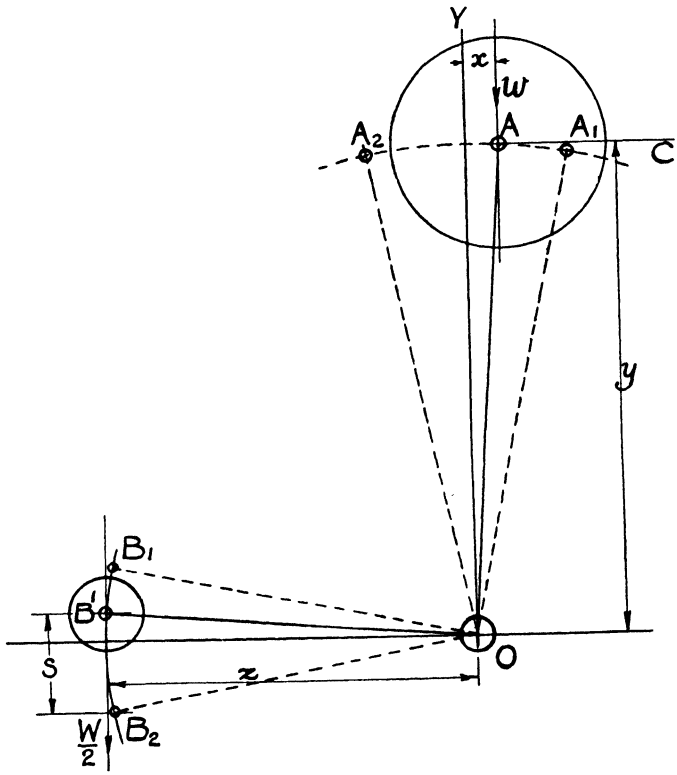



FIG. 282.

balls between $A_2$ and $A_1$, the simplest process is to consider the static equilibrium of the bell-crank lever with reference to the pivoting pin O.

The lever is in equilibrium under the four external forces: the ball weight $w$ and centrifugal force at A, the sleeve load $\frac{W}{2}$ and the reaction of the pin at O. By taking moments about O, the latter is eliminated.

From the configuration shown in Fig. 282 it will be obvious that—

$$Cy+wx-\frac{W}{2}z=0 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (34)$$

If A were situated to the left of the vertical the moment of $w$ about O would be $-wx$.

Hence, in general—

$$C=\frac{\frac{W}{2}z\pm wx}{y} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (35)$$

The values of $z$, $x$, and $y$ are scaled from the drawing. When the C values are thus calculated for several positions of the ball and plotted, the curve $b_2b_1$, Fig. 281, is obtained. As before, to take account of frictional detention, the sleeve load is increased by the equivalent friction force R and $\frac{(W\pm R)}{2}$ is substituted in (35). The dotted curves $d_2d_1$ and $e_2e_1$ are then obtained. The effect of increasing the initial load $p$ on the spring is simply to raise each curve by this amount. When this initial load is such that the control force curve $b_2'b_1'$, when produced to the left, passes through zero radius, the governor is isochronous. It will, in fact, become isochronous before this limit is reached on account of the higher dotted friction curve for increasing speed, and for decreasing speed it will still have a slight stability.

The momental method is the quick one to use, but it is of interest to indicate an approximate graphical one. In Fig. 283, which shows the bell-crank lever B'OA' in the same position as in Fig. 282, set-off $Oa=w$, the ball weight, above and below the base OB, and draw parallels. From the out and in positions $A_1$ and $A_2$ draw verticals $A_1b_1$ and $A_2b_2$ to cut the parallels through $a$ in $b_1$ and $b_2$. At $b_2$ set up the calculated value $b_2d=\left(\frac{W}{2}\right)_2$ at full-in, and at $b_1$ the value $b_1e=\left(\frac{W}{2}\right)_1$ at full-out position, and join $d$ and $e$. To find C for the position A', draw the perpendicular A'E to cut $de$ in E, and through the corresponding position B' draw the vertical B'F. Draw the horizontal EF cutting B'F on F; join O and F, and draw $a$G parallel to OF, cutting EF in G; draw the horizontal A'K through A' and project vertically from G to cut A'K in K. Finally, join K and O. The line KO cuts EG in L and the controlling force is given by C=LM. When A' is to the left of $OA_1$ draw $a$G from the lower point $a$.

To find the controlling force with friction effect, draw the dotted spring lines as shown, using the corresponding values $\left(\frac{W\pm R}{2}\right)_2$ and $\left(\frac{W\pm R}{2}\right)_1$, and proceed as above.

As in the previous cases, the speed for any configuration is given by—

$$N=54{\cdot}187\sqrt{\frac{C}{wr}} \quad . \quad . \quad . \quad . \quad . \quad . \quad (36)$$

**226. Spring Dimensions.**—In order to ascertain the initial spring load $p$ Equations 33 and 34 can be combined; the values for the position (2), where $s=0$, are used. The resulting equation is—

$$p=2\left(\frac{wr_2N_2{}^2y}{2936{\cdot}3\times 12z}-\frac{wx}{z}\right)-w_s \quad . \quad . \quad . \quad . \quad (37)$$

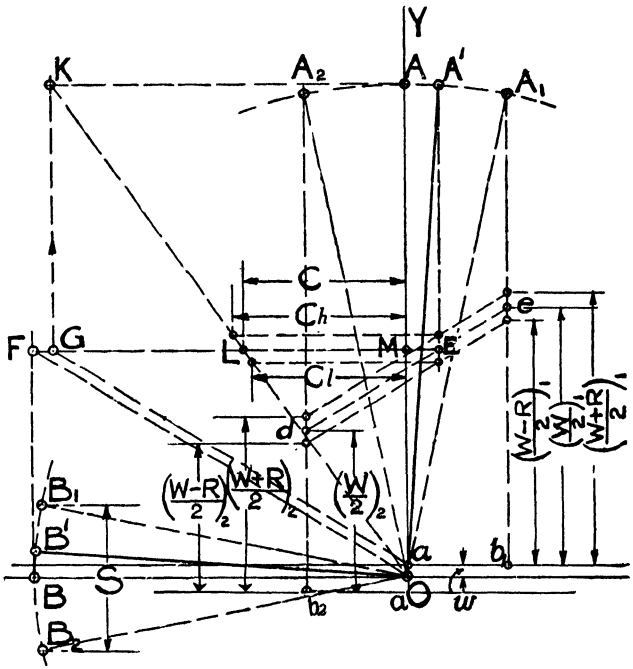



FIG. 283.

When the minimum speed $N_2$, at which the governor balls are to lift the sleeve from the full-in position, is provisionally fixed $p$ can be found.

The selection of the spring suitable for a given case belongs to the province of machine design, but a note on it is desirable here. There are four variables involved: the diameter of coil D; the size of wire, diameter $d$ or side of square; the number of coils; and the free length L (Fig. 284). The initial load $p$ compresses the spring a distance $(a)$, giving the initial compressed length $l_2$, and the rise

of sleeve S further compresses it to the final length $l_1$. The total displacement S of the sleeve is fixed by consideration of the steam distributing mechanism, and is known.

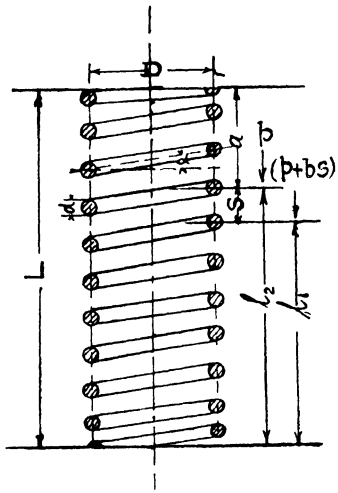



FIG. 284.

An arbitrary choice has to be made of the wire diameter $d$, number of coils $n$, and diameter of spring coil D. The latter can be provisionally fixed to suit the proposed dimensions of the head, and a choice has to be made between $n$ and $d$. Care must be taken to see that when the spring is compressed to length $l_1$ the coils are not completely closed. At least $\frac{1}{16}$ in. should be allowed between each coil.

There are $(n-1)$ spaces with $n$ free coils, so that the minimum value of the compressed length is given by—

$$l_1 = nd + \tfrac{1}{16}(n-1) \text{ in.} \quad . \quad . \quad (38)$$

where $d$=wire diameter in inches.

If an axial load W lb. is required to produce a deflection of $\delta$ in., this is given by—

$$\delta = \frac{64WD^3n}{d^4G} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (39)$$

where D and $d$ are in inches and G is the rigidity modulus for the steel, varying from $12\times10^6$ to $11{\cdot}0\times10^6$ lb./in.$^2$ Taking $G=11{\cdot}5\times106$ and expressing the wire diameter $d$ in sixteenths inch, this reduces to the simple form—

$$\delta = \frac{WD^3n}{22d^4} \text{ in.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (40)$$

When $d$ is the side of a square spring the constant to be used is 30.

The strength of the spring of cylindrical wire is—

$$b = \frac{W}{\delta} = \frac{22\times d^4}{D^3n} \text{ lb./in. deflection} \quad . \quad . \quad . \quad . \quad (41)$$

The " stiffness " is the reciprocal of this figure.

The deflection given by (40) is the extension of a close-coiled spring; but unless the angle ($\alpha$) is large, it can be applied without sensible error to the open-coiled spring used for compression.*

*Example* 8.—In the proposed design of a Hartnell governor, shown in Figs. 281 and 282, the weight of ball is $w=11{\cdot}5$ lb.; weight of sleeve and attachments $w_s=14$ lb.; the "full-in" radius $r_2=5{\cdot}2$ in.,

* See Morley's *Strength of Materials*, p. 279.

and the leverages about the fulcrum pin O are, $x=1$ in., $y=4{\cdot}9$ in., and $z=3{\cdot}7$ in.

If the sleeve is to "float" at a lower speed $N_2=250$ rev./min., calculate the approximate value of the initial spring load $p$ required.

By Equation 37—

$$p=2\left(\frac{wr_2N_2^2y}{2936{\cdot}3\times12z}-\frac{wx}{z}\right)-w_s$$

$$=2\left(\frac{11{\cdot}5\times5{\cdot}2\times250^2\times4{\cdot}9}{2936{\cdot}3\times12\times3{\cdot}7}-\frac{11{\cdot}5\times1}{3{\cdot}7}\right)-14$$

$$=2(140{\cdot}2-3{\cdot}1)-14$$

$$=(2\times137-14)$$

$$=(274-14)=\underline{260 \text{ lb.}}$$

$w=11{\cdot}5$
$w_s=14$
$r_2=5{\cdot}2$ in.
$x=1$ in.
$y=4{\cdot}9$ in.
$z=3{\cdot}7$ in.
$N_2=250$

If the sensitiveness or static fluctuation for the whole range is decided, then for the full-out configuration the upper speed limit $N_1$ can be calculated by Equation 7, and the final spring load is given by—

$$p'=2\left(\frac{wr_1N_1^2y}{2936\times12z}+\frac{wx}{z}\right)-w_s \quad . \quad . \quad . \quad . \quad (42)$$

The increase of load for a total (sleeve) deflection S is thus $(p'-p)$, and the strength of the required spring is—

$$b=\frac{p'-p}{S}=\frac{22d^4}{D^3n} \quad . \quad . \quad . \quad . \quad . \quad . \quad (43)$$

For the given coil diameter and assumed number of coils the size of spring wire can be calculated from (43). In general, $n$ may be taken from 8 to 12.

*Example* 9.—In the case of the Hartnell governor of Example 8, the sensitiveness, or static fluctuation, for the whole range is not to exceed 10 per cent. (giving 5 per cent. above and below the mean). Calculate a suitable size of spring wire, assuming provisionally $D=2{\cdot}5$ in. and $n=10$. The ball circle radius at the full-out position is $r_1=7{\cdot}2$ in., the sleeve travel is $S=1{\cdot}5$ in., and the leverages are $x=1$ in., $y=4{\cdot}9$ in., $z=3{\cdot}7$ in.

Static fluctuation $v=\dfrac{N_1-N_2}{\frac{1}{2}(N_1+N_2)}$

$$2(N_1-250)=(N_1+250)\times0{\cdot}10$$

$$\therefore\ N_1=\frac{525}{1{\cdot}9}=276 \text{ rev./min.}$$

$v=0{\cdot}10$
$N_2=250$

$$p'=2\left(\frac{wr_1N_1^2y}{2936{\cdot}3\times12z}+\frac{wx}{z}\right)-w_s$$

$$=2\left(\frac{11{\cdot}5\times7{\cdot}2\times276^2\times4{\cdot}9}{2936{\cdot}3\times12\times3{\cdot}7}+\frac{11{\cdot}5\times1}{3{\cdot}7}\right)-14$$

$$=2(237+3{\cdot}1)-14=466$$

$w=11{\cdot}5$
$r_1=7{\cdot}2$ in.
$N_1=276$
$y=4{\cdot}9$ in.
$x=1$ in.
$z=3{\cdot}7$ in.
$w_s=14$

$$b=\frac{p'-p}{S}=\frac{22d^4}{D^3n}$$

$$=\frac{466-260}{1\cdot5}=\frac{22d^4}{2\cdot5^3\times10}$$

$$\therefore\ d^4=\frac{206\times2\cdot5^3\times10}{1\cdot5\times22}=975\cdot2$$

$\therefore\ d=5\cdot59$ sixteenths inch.

or $d=\frac{5\cdot59}{16}=0\cdot35$ in.

D $=2\cdot5$
$n$ $=10$
$p$ $=260$
S $=1\cdot5$

The minimum length of the compressed spring should be—

$$l_1=nd+\tfrac{1}{16}(n-1)$$

$$=10\times0\cdot35+\tfrac{9}{16}=4\cdot06 \text{ in.}$$

The length of the initially loaded spring, between the adjusting nuts on the spindle and the head, is 7 in. The actual length of the freely compressed spring is $l_1=(7-1\cdot5)=5\cdot5$ in., so there is ample room between the coils. The initial compression will be $a=\frac{p}{b}=\frac{260}{137}=1\cdot9$ in., so that the spring may be specified: $2\frac{1}{2}$ in. dia. coil, 0·35 in. dia. wire, 10 coils and free length, say, 10 in.

**227.** In this case, since the controlling force curve is approximately straight the "governor power" (per ball) may be taken as the product of the mean controlling force between full-in and full-out positions, and the difference of the radii.

Thus $E=\frac{1}{2}(C_1+C_2)(r_1-r_2)$ ft.-lb. . . . . (44)

The frictional work (per ball) is given (Fig. 281) for—

Increasing speed by the area $(he_2e_1g-hb_1b_2g)$
Decreasing speed by the area $(hb_2b_1g-hd_2d_1g)$, or—

Increasing speed $E_{fh}=\frac{1}{2}(r_1-r_2)[(C_{h1}-C_1)+(C_{h2}-C_2)]$ ft.-lb. (45)

Decreasing speed $E_{fl}=\frac{1}{2}(r_1-r_2)[(C_1-C_{l1})+(C_2-C_{l2})]$ ft.-lb. (46)

The forces are in lb. and radii in feet.

*Example* 10.—In the case of the Hartnell governor of Examples 8 and 9, the controlling force values for the free or frictionless governor are $C_1=178$ lb., $C_2=105$ lb. With frictional detention, for increasing speed $C_{h1}=188$, $C_{h2}=115$; for decreasing speed $C_{l1}=168$, $C_{l2}=95$. Find the total governor power and the ratio of power to friction work on the full range, where $r_1=7\cdot2$ in., and $r_2=5\cdot2$ in.

$$E=\tfrac{1}{2}(C_1+C_2)(r_1-r_2)$$

$$=\tfrac{1}{2}(178+105)(7\cdot2-5\cdot2)\tfrac{1}{12}$$

$$=\frac{283\times2}{24}=23\cdot6 \text{ ft.-lb./ball.}$$

$r_1=\frac{7\cdot2}{12}$

$r_2=\frac{5\cdot2}{12}$

Total power $=2\times23\cdot6=47\cdot2$ ft.-lb.

Friction work increasing and decreasing will be the same here, so—

(Increasing) $E_{fh}=\frac{1}{2}(r_1-r_2)[(C_{h1}-C_1)+(C_{h2}-C_2)]$

$$=\frac{1}{2}\left(\frac{7\cdot2-5\cdot2}{12}\right)[(188-178)+(115-105)]$$

$$=\frac{20}{12}=1\cdot66 \text{ ft.-lb.}$$

Frictional work per ball $E_{fh}+E_{fl}=2\times1\cdot66=3\cdot32$ ft.-lb.

Total friction work (two balls)$=6\cdot64$ ft.-lb.

Ratio $\dfrac{\text{Governor power}}{\text{Friction work}}=\dfrac{47\cdot2}{6\cdot64}=7\cdot1$

**228.** In the case of the Proell type, Fig. 278, for which the equivalent linkage is shown in Fig. 285, the ball arm DB$k$ is held up by the arm BE of the rigid bell-crank GEB. This exerts a pull $T_1$ on the pin at B. The transferred spring force at B due to the spring load $\frac{W_s}{2}$ at G will act at right angles to the arm BE, or at some angle $\alpha$ to the horizontal. This force is—

$$F=\frac{W_s}{2}\cdot\frac{u}{v} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (47)$$

By drawing the parallelogram of forces the resultant of $T_1$ and F on B or F′ is determined. It is easy to show that the pull $T_1$ is given by—

$$T_1=\frac{F\sin\alpha+w+\frac{w_s}{2}}{\cos\alpha} \quad . \quad . \quad . \quad . \quad . \quad (48)$$

The loading of the ball arm DB$k$, under which it is in equilibrium for the given position, is F′ at B, $w$ and C at the ball centre $k$, and Q and $\frac{w_s}{2}$ at the lower pin D. The link can be replaced by a rigid link D$bk$ joining D and $k$ and cutting the line of the force F′ at $b$, which is the point of application to the substituted link. The same momental effect at D will be produced if force F′ is removed and a parallel force F″ is applied at the ball centre $k$. The magnitude of this force is given by—

$$F''=F'\frac{Db}{Dk}=\rho F' \quad . \quad . \quad . \quad . \quad . \quad . \quad (49)$$

The equivalent system is now $w$, C and F″ acting at $k$, and Q and $\frac{w_s}{2}$ at D. The latter have a resultant $T_2$ along the line D$bk$, so that if moments are taken about D, its moment is zero.

Hence $Cy+wz=F''x$

and $$C=\frac{F''x-wz}{y} \quad . \quad . \quad . \quad . \quad . \quad . \quad (50)$$

the values of $x$, $y$, and $z$ being scaled for the particular configuration of the linkage.

When frictional detention is considered, then $\frac{w_s \pm R}{2}$ is to be substituted in (48) for $\frac{w_s}{2}$ for the calculation of $T_1$. The corre-

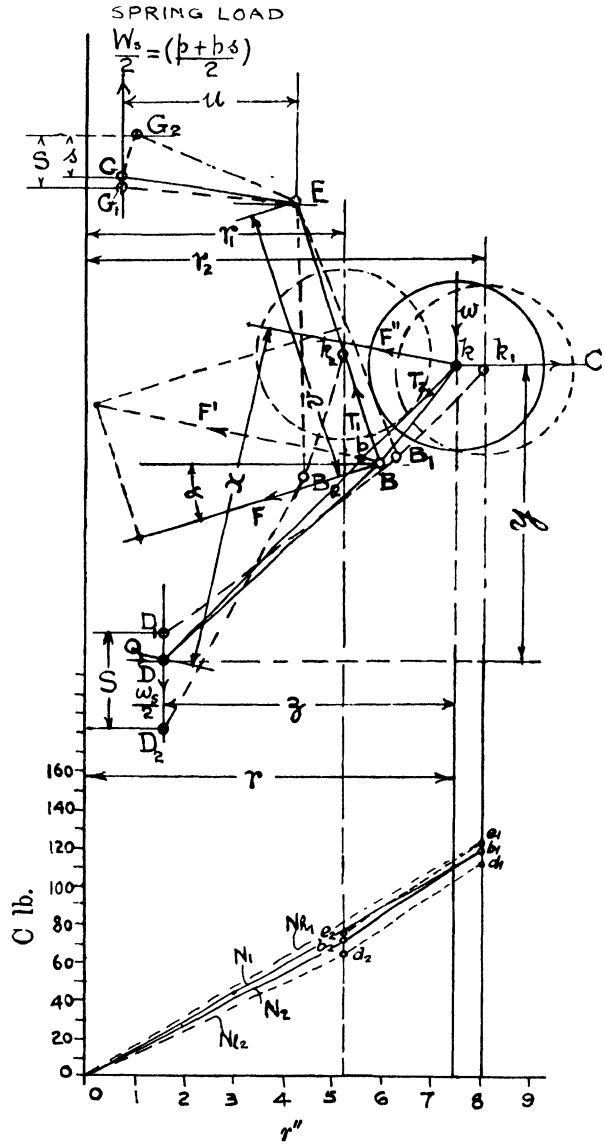



FIG. 285.

sponding $C_h$ and $C_l$ values for the controlling force can then be determined in the same manner as C.

**229.** The controlling force values can also be obtained by the following purely graphical method, shown in Fig. 286.

Draw vertically $ge=\frac{w_s}{2}$, and $ed=w$, and the horizontals $gk$ and $em$ through $g$ and $e$. From $d$ draw $df$ equal and parallel to the transferred spring force F, then through $f$ draw $fk$ parallel to the bell-crank arm BE (Fig. 285), to cut $gk$ in $k$. Join $d$ and $k$ and make $db=\rho dk$ $\left(\text{or } dk\frac{Db}{Dk}\right)$. Through $b$ draw a parallel to the line $Dbk$ (Fig. 285) to cut $em$ in $l$, then $el=C$=the controlling force.

With frictional detention, as in the previous cases set off $\frac{R}{2}$ above and below $gk$, and draw the dotted horizontals, find the

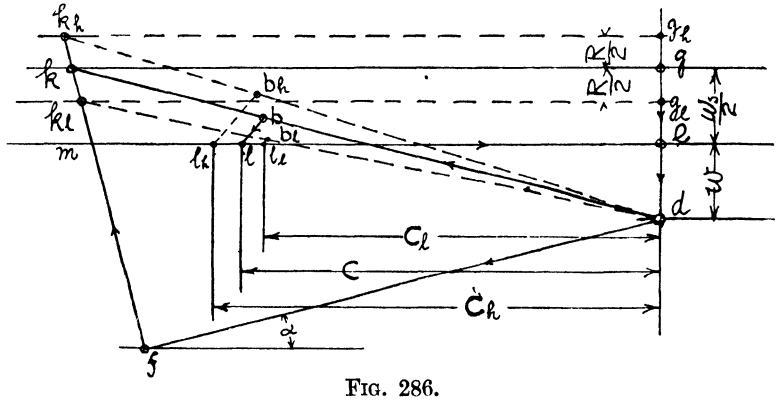



FIG. 286.

points $k_h$ and $k_l$ on $fk$. Draw the dotted diagonals $dk_h$ and $dk_l$, divide them in the proportion $\rho$ at $b_h$ and $b_l$, and draw the parallels to $bl$ to cut $em$ at $l_h$ and $l_l$, then $l_he=C_h$, and $l_le=C_l$.

*Example* 11.—For the configuration of the Proell governor shown in Fig. 285, the following values are obtained from the drawing and other design data: $w=7$ lb.; $w_s=12$ lb.; initial compression load on the spring $p=350$ lb.; strength of spring $b=166{\cdot}66$ lb./in. The leverages are $u=3{\cdot}5$ in.; $v=5{\cdot}7$ in.; $y=6{\cdot}2$ in.; $z=5{\cdot}9$ in. The spring compression, from the full-in position (2) of the balls, is $s=0{\cdot}9$ in.

Calculate the controlling force required to pull the ball to the given radius $r=7{\cdot}5$ in., and find the speed at which the governor will run in this position.

$$\frac{W_s}{2}=\frac{p+bs}{2}=\frac{350+166{\cdot}66\times 0{\cdot}9}{2}=250 \text{ lb.}$$

$$F=\frac{W_s}{2}\cdot\frac{u}{v}=\frac{250\times 3{\cdot}5}{5{\cdot}7}=153{\cdot}5 \text{ lb.}$$

The inclination of F to the horizontal is $\alpha=15°$; sin 15=0·2588, and cos $\alpha$=0·966.

$$\therefore T_1=\frac{F \sin \alpha+w+\frac{w_s}{2}}{\cos \alpha} \qquad \begin{matrix} w=7 \\ \frac{w_s}{2}=6 \end{matrix}$$

$$=\frac{153\cdot5\times0\cdot2588+13}{0\cdot966}$$

$$=\frac{39\cdot73+13}{0\cdot960}=\frac{52\cdot73}{0\cdot966}=54\cdot5$$

Drawing the parallelogram of forces for F=153·5 and $T_1$=54·5 F′=163 lb. The value of $Db/Dk=\rho=0\cdot67$, and $F''=0\cdot67\times153\cdot5=109\cdot2$ lb. Drawing F″ through $k$ and scaling, $x$=6·6 in., then by Equation 50 the controlling force—

$$C=\frac{F''x-wz}{y}$$

$$=\frac{109\cdot2\times6\cdot6-7\times5\cdot9}{6\cdot2}$$

$$=\frac{720\cdot72-41\cdot3}{6\cdot2}=\frac{679\cdot42}{6\cdot2}=109\cdot6 \text{ lb.}$$

The speed is $N=54\cdot187\sqrt{\frac{C}{wr}}$ $\qquad r=7\cdot5$

$$=54\cdot187\sqrt{\frac{109\cdot6\times12}{7\times7\cdot5}}$$

$$=54\cdot187\sqrt{25\cdot06}$$

$$=5\times54\cdot187=271 \text{ rev./min.}$$

A similar calculation for the full-in position shows that the controlling force is $C_2$=72·6 lb., and the speed $N_2$=264. The control force line $b_2b_1$ in Fig. 285 is drawn through the values $C_2$=72·6 and C=109·6 lb.

The values calculated above check with the graphical construction of Fig. 286.

*Example* 12.—If a heavy valve gear resistance increases the sleeve load ($w_s$) by R=30 lb., find the detention by friction at $r$=7·5 in. radius, and the speed variation above or below the mean value of 271 rev./min.

In Fig. 286, by setting off $eg_h=\frac{w_s+R}{2}=\frac{12+30}{2}=21$ from $e$, and $eg_l=\frac{w_s-R}{2}=\frac{12-30}{2}=-9$, drawing the diagonals $dk_h$ and $dk_l$ and parallels $b_hl_h$ and $b_ll_l$, the corresponding values of the control force are $C_h$=118 lb., and $C_l$=101 lb.

The dotted lines $e_2e_1$ and $d_2d_1$ are drawn through these values parallel to the control line $b_2b_1$ (Fig. 285).

Here $$q_{7\cdot5}=\frac{2\left(\sqrt{\frac{C_h}{C_l}}-1\right)}{\left(\sqrt{\frac{C_h}{C_l}}+1\right)} \qquad \sqrt{\frac{118}{101}}=1\cdot078$$

$$=\frac{2(1\cdot078-1)}{1\cdot078+1}=\frac{2\times0\cdot078}{2\cdot078}=\frac{0\cdot156}{2\cdot078}=0\cdot075.$$

The variation will be 3·75 per cent. above or below the mean of 271 rev./min. before the sleeve will move.

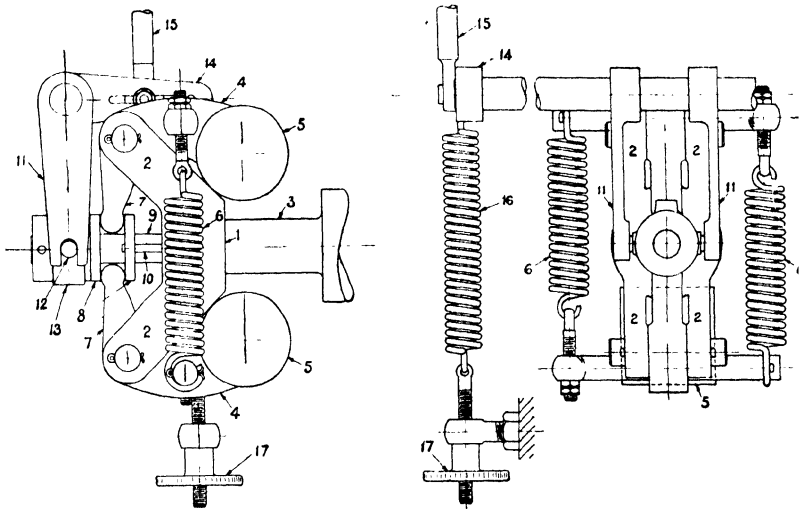



FIG. 287.

When the characteristic diagram is drawn as shown in Fig. 285 the various calculations, already illustrated, can be worked out.

**230. Shaft Governors for Throttle Governing.**—As the name indicates, in this type the centrifugal masses are mounted on and rotate with the shaft, and the centrifugal force at speed is balanced by spring resistances. In the throttle governing type each ball or suitably formed mass is mounted on one arm of a bell-crank lying parallel to the shaft axis, and the other arm, perpendicular to it, engages a groove on the governor sleeve, mounted on an extension of the shaft. The bell-crank lever connected with the valve spindle rod engages another groove in the sleeve, so that as the governor bell-crank lever moves the sleeve, the throttle valve lever is also moved. The centrifugal masses are connected by two spiral springs. A supplementary spring is attached to the throttle valve bell-crank and can be adjusted to regulate the speed while the engine is running.

The details of the design vary with the maker.

One arrangement fitted on a high-speed enclosed vertical engine is shown in Fig. 287.

The boss 1 with arms 2 is keyed on the shaft extension 3. The arms carry the bell-crank levers 4. The centrifugal masses 5 are held together by the two springs 6, which resist the outward motion under the centrifugal force on the masses. The vertical arms 7 of the bell-crank levers fit the recess between the collars 8 on the spindle 9, which is a sliding fit in an axial hole in the shaft 3. The guide pins 10 prevent the spindle from rotating relatively to the shaft. As shown, when the masses move out the sleeve is pushed towards the right. The forks of the vertical arms 11, of the throttle bell-crank, fit over the pins 12 attached to the sleeve 13 in which the axial spindle rotates. This bell-crank is pivoted on a bracket fixed to the casing, and the horizontal arm 14 is connected to the throttle valve rod 15. The supplementary spring 16 is fixed to the casing and to the outer end of the horizontal crank arm 14. The tension is adjustable by the screw and hand-wheel 17. By means of this spring the speed can be brought back to the normal value when the load is changed, so that the engine can be kept at normal speed at any load from zero to maximum. A similar type of governor is shown on the high-speed engine (Fig. 364, p. 555).

Two other arrangements of this type are shown in skeleton in Figs. 288 and 289. In Fig. 288 the governor springs are attached directly to the centre of the centrifugal masses, but the supplementary spring is attached to an auxiliary lever HLG, pin-jointed to the valve rod at L so that the force at G is increased in the ratio of $m$ to $n$ at A.

In Fig. 289 the governor springs are attached to the bell-crank arms at N, the leverage being reduced from $l$ to $q$, and the pull of the spring at N, to balance against pull on the mass, is increased in the ratio of $l$ to $q$ above the value at the mass centre K. In this case the supplementary spring is attached to the bell-crank arm AB at a point G intermediate between A and B. The force exerted by the spring at G is thus reduced at D on the sleeve in the ratio of $u$ to $d$.

**231. Calculation of Controlling Force of Shaft Governor.**—Consider, as a general case, the arrangement of Fig. 289. The governor is to be arranged to run at some normal speed N with an overall static stability or sensitiveness defined by the speed $N_2$ at full-in radius $r_2$, and $N_1$ at full-out radius $r_1$.

Suppose from other design data the value of $r_2$ is fixed. This will be the radius of the mass centre circle (K) when the masses are closed in and the governor is at rest. The force at each mass necessary to move the masses out at this radius (controlling force) will be the force to overcome the initial pull of *two* springs and the transferred resistance of the supplementary spring acting on the sleeve through the leverage system. Given the static fluctuation

($v$) and mean speed N, the minimum and maximum speeds $N_2$ and $N_1$ are known, and—

$$N_2 = N\left(1 - \frac{v}{2}\right) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (51)$$

Let $w$=weight of one governor mass

$C_2$=controlling force at radius $r_2$ on this mass.

$$\text{Then} \quad C_2 = \left(\frac{N}{54 \cdot 187}\right)^2 \times \frac{wr_2}{12} \quad . \quad . \quad . \quad . \quad (52)$$

If, as in Fig. 288, the main springs (two to each mass) are attached at the mass centre K, this force is the initial tension of the main springs plus the transferred force from the auxiliary spring.

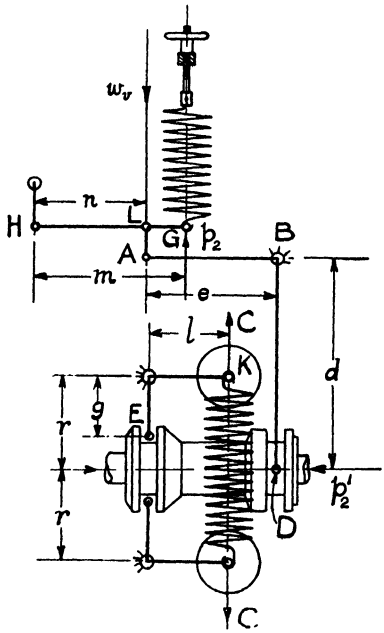



FIG. 288.

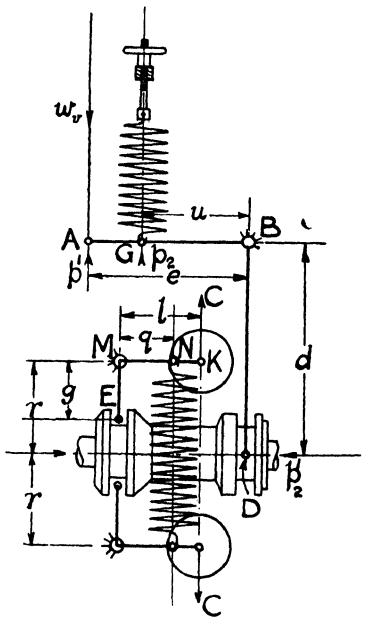



FIG. 289.

If, however, the main springs, as in Fig. 289, are attached to the arms at N, the force acting in the line of the main springs is increased to—

$$C'_2 = C_2 \times \frac{l}{q} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (53)$$

when $q=l$, $C'_2=C_2$.

This force can be divided in any arbitrary manner between the main and supplementary spring loads. A common proportion is 20 per cent. for the supplementary load and 80 per cent. for the main spring load.

Let the force at N be $f_2'=xC_2'$, where $x$ is the fraction of the load taken by the main springs.

The proportion $(1-x)C_2'$ due to the supplementary spring may be denoted by $f_{p2}$.

Let the strength of each main spring be $b$ and of the supplementary spring $b'$.

Then if $a$ is the initial stretch of each main spring—

$$f_2'=2ba \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (54)$$

(for the two springs).

Let $a'$=initial stretch of the supplementary spring to produce the force $f_{p2}$ on each governor arm at N. The pull at G is $b'a'=p_2$; and this is reduced by the compound leverage $\frac{u}{d}\times\frac{g}{q}$, to $2f_{p2}=p_2'$, since the force transmitted to the sleeve is taken by *two* bell-cranks, or—

$$2f_{p2}=p_2\frac{u}{d}\times\frac{g}{q}$$

$$\therefore\ f_{p2}=\frac{p_2}{2V} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (55)$$

since $\frac{d}{u}\times\frac{q}{g}=V$=velocity ratio $\left(\frac{\delta_r}{\delta_s}\right)$ of the point N to the point G ($\delta_r$ is the small change of radius producing a small displacement $\delta_s$ of G.)

The transferred central force at N (in line of main spring) is thus given by—

$$C_2'=f_2'+f_{p2}$$

$$=\left(2ba+\frac{b'a'}{2V}\right) \quad . \quad . \quad . \quad . \quad . \quad (56)$$

Given $C_2'$, suitable springs can be determined for arbitrarily chosen initial extensions, or vice versa.

Now let the masses be pulled out to some radius $r$ between $r_2$ and $r_1$. The radial increase is $(r-r_2)$ and the additional stretch of the main springs would be $2(r-r_2)$ if these were connected at K. The stretch in the line of N is, however, less, or $2(r-r_2)\frac{q}{l}$, on each of the two springs, and the main spring controlling force at N becomes—

$$f_s'=f_2'+2\times 2b(r-r_2)\frac{q}{l}$$

$$=2b\left(a+2(r-r_2)\frac{q}{l}\right) \quad . \quad . \quad . \quad . \quad . \quad (57)$$

When the point N moves the distance $(r-r_2)\frac{q}{l}=\delta r$, the end G of the supplementary spring moves $\delta s=\frac{\delta r}{V}$. The additional tension

in the supplementary spring, due to the radial increase, is thus $b'\delta s = b'\left(\frac{r-r_2}{V}\right)\frac{q}{l}$, and the total tension is—

$$\begin{aligned} p &= p_2 + b'\delta s \\ &= b'a' + b'\left(\frac{r-r_2}{V}\right)\frac{q}{l} \\ &= b'\left\{a' + \left(\frac{r-r_2}{V}\right)\frac{q}{l}\right\} \end{aligned}$$

The transferred force at N in the line of the main springs is $f_p = \frac{p}{2V}$, so that—

$$f_p = \frac{b'}{2}\left(\frac{a'}{V} + \frac{(r-r_2)q}{V^2 l}\right) \quad . \quad . \quad . \quad . \quad . \quad (58)$$

and the referred controlling force at N becomes—

$$C' = (f_s' + f_p) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (59)$$

$$C' = 2b\left(a + 2(r-r_2)\frac{q}{l}\right) + \frac{b'}{2}\left(\frac{a'}{V} + \frac{(r-r_2)q}{V^2 l}\right)$$

$$\text{or} \quad C' = C_2' + \frac{q}{l}\left(4b + \frac{b'}{2V^2}\right)(r-r_2) \quad . \quad . \quad . \quad . \quad . \quad . \quad (60)$$

The actual controlling force referred to the mass centre K will be less, or—

$$C = C'\frac{q}{l} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (61)$$

Any effect of the weight of valve rod and throttle valve has been neglected here. If appreciable, this will reduce the transferred supplementary spring load at N. This referred value will be $f_v = \frac{w_v}{2V_v}$, where $w_v$ is the weight of the valve and rod, and, if desired, the value of $(f_p - f_v)$ can be substituted for $f_p$ in Equation 59. The velocity ratio here is that of N to A, or—

$$V_v = \frac{d}{e} \times \frac{q}{g} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (62)$$

The frictional resistance of the valve and spindle will have the same effect as a positive or negative spindle load ($\pm w_f$), increasing the controlling force at N for increase of speed and decreasing for decrease of speed. The value is given by $\pm\frac{w_f q}{2V_v l} = f_f$.

Then

$$C_h = C + f_f \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (63)$$

$$C_l = C - f_f \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (64)$$

The value of $w_f$ may be taken at from 2 to 3 lb. per in. diameter of throttle valve.

As in the previous cases, the characteristic diagram can be plotted and power and other calculations can be made.

It is not proposed to consider this work again in detail. The following example will give the student an idea of how the foregoing equations may be applied in a preliminary design.

*Example* 13.—A high-speed engine running at a normal speed of 400 rev./min. is to be fitted with a shaft governor of the design shown in Fig. 289. The radius of the governor masses at the full-in position (2) is fixed at $r_2=4{\cdot}25$ in. It is proposed to use main springs with strength of 21 lb./in. and a supplementary spring with strength of 88 lb./in. The proposed values of the leverages are: for governor bell-cranks, $\frac{q}{g}=1$, $\frac{l}{q}=1{\cdot}33$; throttle rod bell-crank, $\frac{d}{u}=1{\cdot}85$, $\frac{d}{e}=1{\cdot}3$. An overall statical stability of 10 per cent. is to be allowed. The weight of each governor mass is 8 lb. Find the initial extensions of the main and supplementary springs and fix suitable spring dimensions. Also find the speed at $r=5$ in. and $r=5{\cdot}75$ in.

Here $N=400$, $v=10$ per cent.

$$N_2=N\left(1-\frac{v}{2}\right)=400(1-0{\cdot}05)=0{\cdot}95\times400=380 \text{ rev./min.}$$

$$C_2=\left(\frac{N}{54{\cdot}187}\right)^2\frac{wr_2}{12} \qquad w=8,\ r=4{\cdot}25$$

$$=\left(\frac{380}{54{\cdot}187}\right)^2\frac{8\times4{\cdot}25}{12}=139{\cdot}3 \text{ lb.}$$

Referring this to the line of the main spring—

$$C_2'=C_2\frac{l}{q}=139{\cdot}3\times1{\cdot}33=185{\cdot}3 \text{ lb.}$$

Taking $x=0{\cdot}8$, $f_2'=xC_2'=185{\cdot}3\times0{\cdot}8=148{\cdot}24$

$$f_2'=2ba \qquad b=21 \text{ lb./in.}$$

$$\therefore\ a=\frac{f_2'}{2b}=\frac{148{\cdot}24}{2\times21}=3{\cdot}53 \text{ in.,}$$

the initial extension of each main spring. A spring of 10 free coils, 2·5 in. coil diameter, and $\frac{7}{32}$ in. wire, gives a strength of 21 lb./in. (Equation 41). Stretched length is $2r_2=8{\cdot}5$ in., assuming N at same radius as K, hence the free length of the spring will be $l_o=(2r_2-a)=(8{\cdot}5-3{\cdot}53)=4{\cdot}97$ in.

The proportion of the referred controlling force at N taken by the supplementary spring is $(1-x)C_2'=0{\cdot}2\times18{\cdot}53=37$ lb.$=f_{p2}$. The velocity ratio of N to G is $V=\frac{d}{u}\times\frac{q}{g}=1{\cdot}85\times1=1{\cdot}85$

$$\therefore\ f_{p2}=\frac{p_2}{2V}=\frac{b'a'}{2V}=\frac{88\times a'}{2\times1{\cdot}85} \qquad b'=88$$

$$\therefore\ a'=\frac{2\times1{\cdot}85\times37}{88}=1{\cdot}56 \text{ in.,}$$

the initial stretch of the supplementary spring for the full-in position.

A spring of 8 free coils, 2 in. coil diameter, and $\frac{1}{4}$ in. wire gives a strength of 88 lb./in. A free length of, say, $4\frac{1}{2}$ in. to 5 in. may be specified.

The controlling force in the line of the main springs for any radius $r$ now becomes—

$$C' = C_2' + \frac{q}{l}\left(4b + \frac{b'}{2V^2}\right)(r - r_2)$$

$$= 185{\cdot}3 + \frac{1}{1{\cdot}33}\left(4 \times 21 + \frac{88}{2 \times 1{\cdot}85^2}\right)(r - r_2)$$

$$= 185{\cdot}3 + 72{\cdot}8(r - r_2)$$

$C_2' = 185{\cdot}3$
$\frac{q}{l} = \frac{1}{1{\cdot}33}$
$b = 21$
$b' = 88$
$V = 1{\cdot}85$
$r_2 = 4{\cdot}25$
$w = 8$

For $r = 5$, $C' = 185{\cdot}3 + 72{\cdot}8 \times 0{\cdot}75$

$= 185{\cdot}3 + 54{\cdot}6 = 239{\cdot}9$ lb.

$$C = C'\frac{q}{l} = 239{\cdot}9 \times \frac{1}{1{\cdot}33} = 180 \text{ lb.}$$

$$N_5 = 54{\cdot}187\sqrt{\frac{180 \times 12}{8 \times 5}} = 397 \text{ rev./min.}$$

For $r = 5{\cdot}75$ $C' = 185{\cdot}3 + 72{\cdot}8 \times 1{\cdot}5 = 295{\cdot}3$

$$C = \frac{295{\cdot}3}{1{\cdot}33} = 222$$

$$N_{5{\cdot}75} = 54{\cdot}187\sqrt{\frac{222 \times 12}{8 \times 5{\cdot}75}} = 412 \text{ rev./min.}$$

The student can vary this exercise by taking various adjustments of the supplementary spring.

*Example* 14.—Find the frictional detention for the above governor at radius $r = 5$ in. if a $3\frac{1}{2}$ in. throttle valve is used and 3 lb. per inch throttle diameter is allowed for frictional resistance. In this case the velocity ratio $V_v = \frac{d}{e} \times \frac{q}{g} = 1{\cdot}3$.

Here $w_f = 3 \times 3{\cdot}5 = 10{\cdot}5$ lb.

$$f_f = \frac{w_f}{2V} \cdot \frac{q}{l} = \frac{10{\cdot}5}{2 \times 1{\cdot}33} \times \frac{1}{1{\cdot}3} = 3{\cdot}32 \text{ lb.}$$

$$C_h = C + f_f = 180 + 3{\cdot}32 = 183{\cdot}32$$

$$\therefore\ N_{h5} = 54{\cdot}187\sqrt{\frac{183{\cdot}32 \times 12}{8 \times 5}} = 400 \text{ rev./min.}$$

$$\text{Detention } q = \frac{100(400 - 397)}{397} = \frac{3 \times 100}{397} = 0{\cdot}75 \text{ per cent.}$$

and the probable total detention is within 1·5 per cent.

**232. Shaft Governors for Expansion Governing.**—Instead of regulating the travel of an expansion valve in the manner shown in

Fig. 277, the travel and the angle of advance of the main slide-valve may be varied by means of a more powerful form of shaft governor, which operates a floating or shifting eccentric sheave or its equivalent.

There are many varieties of this class which cannot be adequately dealt with in this text. The theory of the type is also much more difficult than that of the previous forms, and only a brief outline of the conditions which have to be taken into account in a design of this kind can be given. The type is illustrated here by one of

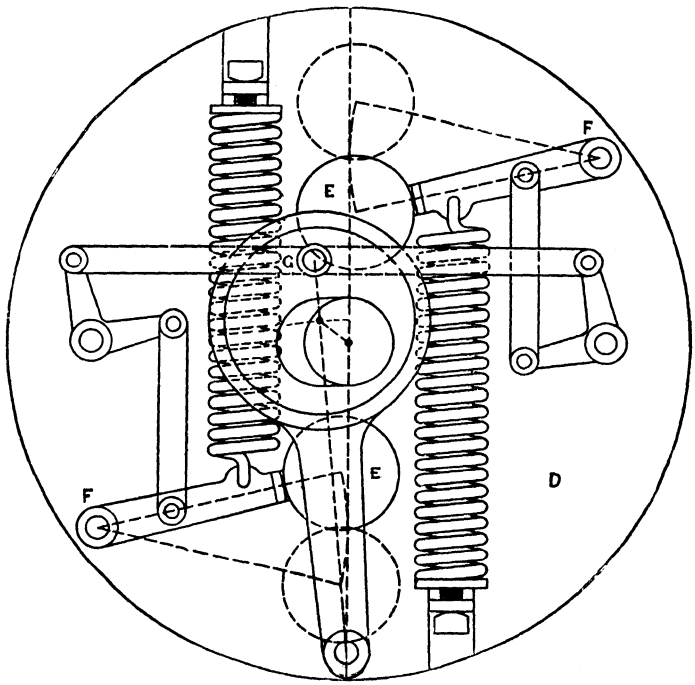



FIG. 290.

the older and by one of the most recent and compact modern designs fitted to a high-speed engine.

The illustration of the earlier form is shown in Fig. 290. A disc D, keyed to the engine shaft, carries the combination of levers and shifting eccentric. The eccentric sheave, which has an oblong slot to enable it to be moved transversely to the shaft axis, is bolted to the face of a swing arm pivoted as shown near the bottom edge of the disc D. Two levers carrying centrifugal masses or balls E are pivoted at F. Each lever has a control spring anchored at one end to the disc D. The levers are connected up by two bell-cranks and links with the horizontal rod, which is pin-jointed to the eccentric sheave at G.

It will be obvious that, as the masses E are forced out against the resistance of the springs, the horizontal rod will swing the eccentric about the link pin, reduce the eccentricity, and increase the angle of advance, when the load falls off and the speed increases, and vice versa. The eccentric centre will thus describe the " characteristic curve " of the gear, already fully discussed in Chapter IX. By making the radius of the swing link or its equivalent large, this curve can be made to approximate to a straight line.

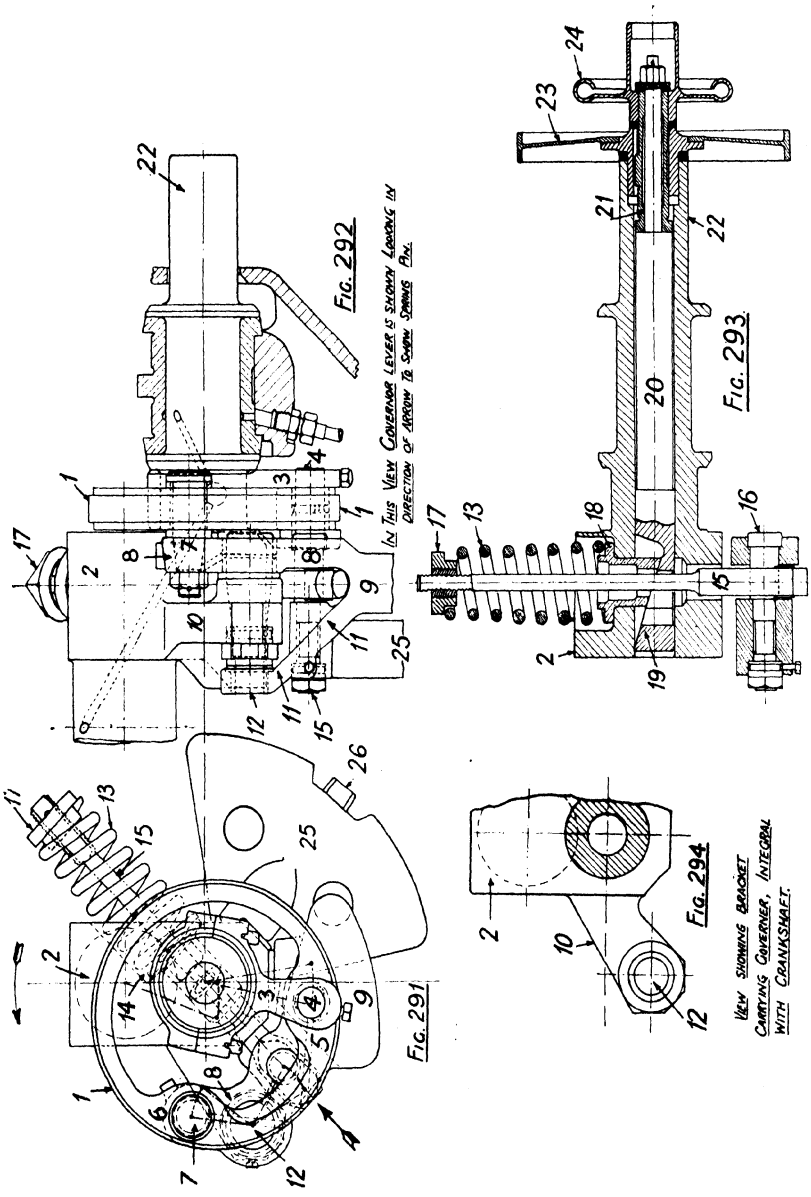
The design has too many joints and links, and two long springs, all of which set up disturbances due to centrifugal forces on them, independently of the operative centrifugal force on the masses E. Joint friction and wear is considerable, and the springs are liable to serious distortion due to the centrifugal force of their mass. This form is seldom made now. Generally, a single weight and lever and one control spring is used.

An extremely compact and effective single-weight governor for expansion governing is that fitted on the Sisson high-speed engine described in Chapter XVI. Four views of this up-to-date design are shown in Figs. 291, 292, 293, and 294. The eccentric sheave (Fig. 291) is replaced by what is simply a thick ring 1, suspended from the shaft outside the right-hand crank web 2 by a swing link 3. The link pin 4 fits into a boss 5 formed on the side of the eccentric ring. Another boss is formed at 6, into which a pivot pin 7 is fitted. This pin is fixed by coned end and nut on the inner fork 8 of the vertical arm of the governor lever, the lower arm 9 of which acts as a governor weight. As shown in Figs. 292 and 294, a lug 10 is formed on the side of the crank web, and the outer fork 11 of the governor lever is pivoted to this by the pin 12.

The eccentric ring is thus kept in a floating condition by the suspension link pin 4 and lever pin 7. If the governor lever is pulled down pin 7 moves in and forces the ring to the right, while the link 3 swings about the shaft centre. The eccentric centre describes the flat dotted curve (Fig. 291), which is the " characteristic curve " of this expansion gear. The control spring 13 is of the conical spiral type and fits at the bottom into a recess 14 cut in the side of the crank web. The shaft and web are bored out to take a fine adjustment gear shown in Fig. 293, which shows a section of the shaft in the plane of the control spring. A hole is bored through the crank web and shaft, through which the spring rod 15 is passed, and connected to the governor lever by the pin 16. By this arrangement radial distortion of the spring at the high engine speed is prevented.

Referring to Fig. 293, the spring is held on the rod by the screwed collar 17 at the top. The bottom bears on a pressure piece 18, which in turn bears on a wedge face 19 formed at the end of a draw rod 20 fitted in the shaft bore. This draw rod is operated by the adjusting sleeve 21, which is threaded on the reduced end of the rod 20 and held in place by a nut and washers.

This sleeve, which is provided with a feather, is threaded into the boss of a floating wheel 23. The outer portion is formed with a fine thread and screwed into the boss of the adjusting hand-wheel 24. When the floating wheel is held, the bush, and with it the draw rod, can be moved out or in by turning the hand-wheel 24.



The crank balance weight 25 (Fig. 291) is bolted to the crank web by the bolt 26.

**233. Conditions of Equilibrium of Expansion Shaft Governors.**—In this type the external force system acting on the governor linkage for a given configuration varies from point to point during a revolution.

Superposed on the steady centrifugal load of the governor weight there are centrifugal forces on out-of-centre masses, the principal being that due to the mass of the eccentric; periodic inertia force of the reciprocating valves, rods, and other attachments, which passes twice in a revolution from a positive maximum to a negative maximum value; frictional forces due to centrifugal loading at various joints; kinetic friction force due to resistance of valves, rods, etc., which is constant but changes from a positive to a negative value each revolution. It will be apparent that this equilibrium problem if dealt with completely is a very complicated one. A discussion of it is out of the question in this text, and only an indication of its general treatment can be given. The student can refer to special treatises on governing for detailed discussion.*

The general statement of the condition of equilibrium at any instant, is that the controlling moment of the spring about the governor lever pivot must balance the sum of the centrifugal inertia and friction moments about that centre. The inertia force moment can be expressed as the centrifugal moment of an equivalent mass at the eccentric centre and thus be included under the centrifugal moment sum.

In general terms—

If $M_s$=moment of spring force about the lever pin,

$\Sigma M_c$=sum of the centrifugal force moments about the lever pin,

$\Sigma M_f$=sum of the frictional force moments about the lever pin,

Then $$M_s=\Sigma M_c+\Sigma M_f \quad . \quad . \quad . \quad . \quad . \quad . \quad (65)$$

If L, in general, denotes the leverage of any centrifugal force, *i.e.* the perpendicular distance from the lever pin to the line of action of the force, or the line drawn from the mass considered to the shaft centre; $\omega$ is the angular velocity; and $w$, in general, is the weight of any eccentric mass, then $\Sigma M_c=\Sigma\frac{w}{g}\omega^2 r\times L$. If $F_c$ is the pressure produced at any surface due to the centrifugal force of a given mass and the inertia force superposed on it (for instance, the surface of the eccentric strap), it can be expressed as the force due to an equivalent mass $w'$ at radius $r'$, and if, in general, the leverage for this force is denoted by $l'$, then the total moment about the pivot of these equivalent friction forces will be $\Sigma\frac{w}{g}\omega^2 r' f\times l'$, where $f$ is the coefficient of friction for the surfaces in contact.

* See *Governors and Governing of Prime Movers*, by W. Trinks, M.E., Chapter VI.

Similarly, if F is the constant kinetic friction force, and in general $l$ is the leverage, $M_f = \Sigma Fl$.

The general statement then becomes—

$$M_s = \omega^2\left(\Sigma\frac{w}{g}rL + \Sigma f\frac{w'}{g}r'l'\right) + \Sigma Fl \quad . \quad . \quad . \quad . \quad (66)$$

and the angular velocity is given by—

$$\omega = \sqrt{\frac{M_s - \Sigma Fl}{\Sigma\frac{w}{g}rL + \Sigma f\frac{w'}{g}r'l'}} \quad . \quad . \quad . \quad . \quad . \quad (67)$$

The frictional conditions are very uncertain owing to the possible variation of the friction coefficient $f$ with the conditions of lubrication and the mechanical tightening of the glands, etc. $\Sigma Fl$ may be small or very considerable. The possibility of the large negative friction moment under the root sign indicates the need to make the spring moment $M_s$ and its balancing centrifugal moment $\Sigma\frac{w}{g}rL$

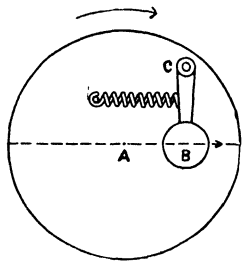



FIG. 295.

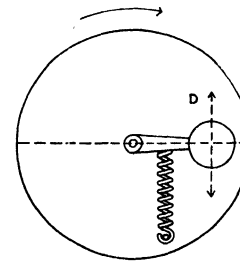



FIG. 296.

very large to prevent unsteady running, from changing frictional conditions.

This type of governor can be steadied somewhat against these impressed forces by arranging the governor weight so that its " tangential inertia " is brought into action to counteract them.

**234. Tangential Acceleration.**—In Figs. 295 and 296 the governor arm, ball, and controlling spring are shown diagrammatically. As the governor disc rotates about the shaft centre A the centrifugal force on B (Fig. 295) throws it outward and stretches the spring. In the case of Fig. 296, where the pivot $c$ coincides with the shaft centre A, the centrifugal force merely causes a pull on the shaft and there is no effect on the spring when the disc is running at a constant speed. Suppose, however, that a sudden change, say increase of speed, takes place, then the inertia of the mass of B in each case causes the ball to lag behind the disc, the inertia force produced by the acceleration of B being at right angles to the line joining the centre of B with the shaft centre, or in other words a tangential acceleration force.

In the first case (Fig. 295) this force acting along BC simply pro-

duces a pressure on C and has no effect on the spring. In the second (Fig. 296) it forces the ball up in the direction of D and produces a pull on the spring, and a controlling moment about the ball lever pivot. These cases represent the extremes giving zero and maximum moments about the lever pin. Figs. 297 and 298 show two intermediate arrangements.

In the case of Fig. 297, if the tangential inertia force is P and CD is the perpendicular from the pivot pin C to its line of action, it produces for the clockwise rotation of the disc a moment $P \times CD$ about the pivot C of the same sign as the centrifugal moment of the ball mass. In other words, the tangential acceleration is employed here to increase the centrifugal force during the momentary increase of speed. The result of this is to make the governor more responsive or sensitive. On the other hand, it will be apparent that with the arrangement shown in Fig. 298 the tangential inertia force produces a moment about C in the opposite sense to the centrifugal moment of B. The tangential acceleration force thus retards the centri-

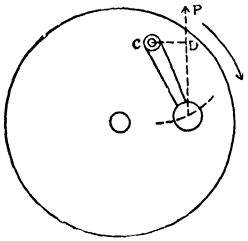



FIG. 297.

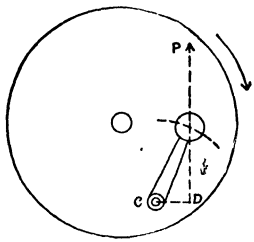



FIG. 298.

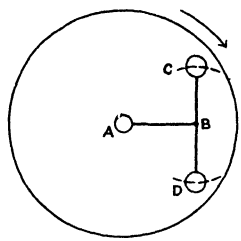



FIG. 299.

fugal force, making the governor less sensitive and causing it to act sluggishly.

**235. Angular Acceleration.**—Another form of accelerating force which has been utilised in some designs of shaft governor is that due to angular acceleration. The principle is illustrated in Fig. 299. An arm AB is rigidly attached to the shaft. The mass of the governor weight is divided into two parts, and placed at the ends *c* and *d* of a rod CD pivoted at its centre on the end of the arm AB. If the shaft rotates at a uniform velocity the rod CD will be held in a fixed position, but if the speed of the shaft is suddenly arrested each ball tends to move on along the tangent to the circle, and the arm CD will rotate clockwise about B. Conversely, if the speed of the shaft increases the balls will lag in the circle and the arm CD will be turned counter-clockwise. This angular inertia of a loaded bar pivoted at its centre is utilised in the Begtrup design of shaft governor shown in Fig. 300.

The eccentric, as shown in plan and in the sectional elevation, is bolted to the cross bar carrying the weights, and the bar is pivoted on a pin passing through the boss of the governor wheel. The right-hand weight is made slightly heavier than the left one and acts as

governor ball, under control of the spring attached to the wheel rim and the arm. Stops are provided on the rim to limit the swing of the arm. As the speed increases the right-hand weight tends to fly out and turn the bar, and thus to swing the eccentric to the right, and alter the eccentricity. At the same time the inertia of both weights assists the centrifugal force in effecting the change of position of the eccentric.

In this chapter the action of various forms of centrifugal governor has been considered with regard to the direct control of steam engines.

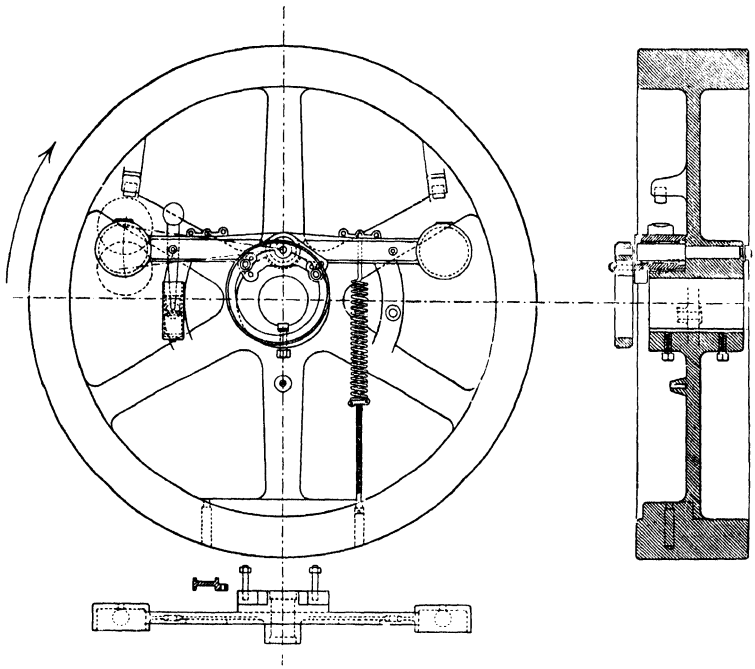


FIG. 300.

In steam turbine practice they are not used, except on small machines, for direct control but for the primary regulation of some form of relay gear, usually hydraulic, which provides the necessary "power" for the operation of the steam distributing valve system. An excellent example of one of these relay systems is illustrated and described in the Chapter XX, dealing with steam turbines.

For other systems of relay governing as applied to large steam turbines the student is referred to texts on the steam turbine.*

* See *Steam and Gas Turbines*, by Stodola, sixth edition, vol. i. Translated by L. C. Loewenstein. Also *Steam Turbines*, by W. J. Goudie, Chapter XVIII.

# CHAPTER XI

## CRANK EFFORT AND TWISTING MOMENT

**236.** At any position of the crank the force acting on the crank pin at right angles to the crank arm, and due to the effective steam load on the piston, is termed the crank effort or tangential force. This produces a torque or twisting moment about the axis of the shaft. At the inner and outer dead centres this tangential force is zero, the whole effective piston load acting along the crank arm and simply pushing or pulling on the shaft and bearings. The twisting moment varies from zero to a maximum and then to zero again in a stroke, or twice in a revolution, the maximum being reached about half stroke.

The character of this variation is affected by the varying steam pressure on the piston, by the use of a finite connecting rod, and by the inertia of the reciprocating parts, which include the piston and rod, crosshead, and part of the connecting rod.

**237.** It is important, for steady running, that the cyclical variations of crank effort and torque should be modified as much as possible.

The means employed for this purpose are—

(1) A combination of cylinders driving on separate cranks on the same shaft, the cranks being arranged so that the torque variations partly correct one another, when in combination. Two, three, or four cranks may be used. In rare cases five cranks have been tried with steam engines (see Fig. 373, p. 565).

(2) Adjustment of the cut-off and compression points to suit the speed.

(3) By a flywheel. This is the most important factor for a single-cylinder engine. The wheel acts as a reservoir of energy. It absorbs surplus energy while the torque due to the steam pressure exceeds the mean torque due to the external resistance on the engine, and vice versa. This function of the flywheel is considered in Chapter XII.

**238. Effective Pressure on the Piston.**—At any piston position on the out-stroke the force of the steam is the total steam pressure on the back minus the total steam pressure on the front of the piston. This is obtained by superposing the indicator diagrams for front and back of the piston, as shown in Fig. 301, and scaling the intercept between the steam or expansion line for the back and the exhaust line for the front side. Thus at the piston positions

on the out-stroke corresponding to the points D, E, and F on the diagram from the back of the piston, the intercepts are respectively $y_1$, $y_2$, and $y_3$. When these and other values are plotted on a stroke base $X_1X_2$, the curve of effective steam pressure DEF on the back of the piston is obtained. By a similar process the dotted curve for the front can be plotted. It will be noted that at G the pressures on each side of the piston are equal, or the effective forward pressure is zero ; and from G to the end of the stroke at $X_2$ the pressures are negative. The piston instead of pushing the crosshead and connecting rod is pulled by them. The same condition occurs at the end of the in-stroke.

While the ordinates of the curve DEF represent the force exerted by the steam on the piston face, they do not represent the net push or pull of the piston on the rod and connections when the engine runs at speed.

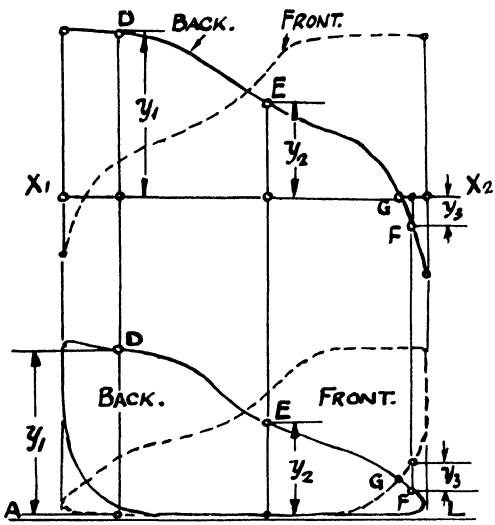



FIG. 301.

The reciprocating parts, piston and rod, crosshead, and a proportion of the connecting rod, have to be started from rest or accelerated positively at the beginning of each out-stroke, and brought to rest or accelerated negatively at the end. The same action is necessary on the in-stroke. At the point of the stroke at which the piston velocity reaches a maximum the acceleration changes from the positive to the negative sign, that is, passes through the zero value.

The actual force transmitted from the piston to the crosshead and connecting rod in the line of stroke, at any point of the stroke, is the net steam load less the inertia force required to give the piston the corresponding acceleration at that point. The total inertia force due to the reciprocating parts is reduced to an equivalent pressure ($p_f$), by dividing it by the piston area. Then the resultant or effective piston pressure is given by—

$$p_R = (p_E \pm p_f) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

the sign being positive as the piston is moving towards the centre of stroke and negative as it is moving away from it.

**239. Calculation of Inertia Force.**—According to the law of dynamics, if W is the weight of the reciprocating parts and $f$ the linear acceleration, the corresponding inertia force is—

$$F_f=\frac{W}{g}f \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

The acceleration $f$ in ft./sec.$^2$ can be obtained either analytically or graphically. The analytical solution is as follows. Referring to Fig. 302, strike the obliquity arc CD with the connecting rod length BC$=l$ as radius, and draw CE perpendicular to BO. Then ID$=z$, the displacement of the piston from the inner dead centre I

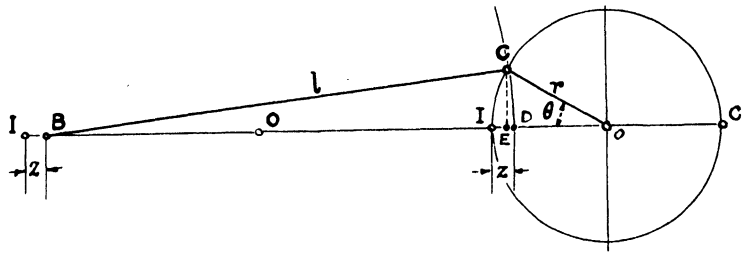



FIG. 302.

when the crank angle is $\theta$, or $z$=(IE+ED). The acceleration of the piston at position D is required.

Here $CE^2+BE^2=BC^2=BD^2$

$$\therefore\ CE^2=BD^2-BE^2$$

$$=(BD+BE)(BD-BE)=(BD+BE)ED$$

$$\therefore\ ED=\frac{CE^2}{(BD+BE)} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

Also $CE=r\sin\theta$; BD=BC$=l$. As ED is a small quantity compared with BD, as an approximation assume BD=BE and (BD+BE)$=2BD=2l$.

$$\therefore\ ED=\frac{r^2\sin^2\theta}{2l}$$

The piston displacement from the end of the stroke is $z$=IE+ED,

$$\text{or } z=(r-OD)+\frac{r^2\sin^2\theta}{2l} \qquad OD=r\cos\theta$$

$$=r(1-\cos\theta)+\frac{r^2\sin^2\theta}{2l}, \text{ approximately.}$$

The linear velocity of B is $v=\frac{dz}{dt}=\frac{d\theta}{dt}\cdot\frac{dz}{d\theta}$

By definition $\frac{d\theta}{dt}=\omega$=angular velocity of the crank.

$$\therefore\ v=\frac{dz}{dt}=\omega\frac{d}{d\theta}\left\{r(1-\cos\theta)+\frac{r^2\sin^2\theta}{2l}\right\} \qquad \frac{l}{r}=n$$

$$=\omega r\left(\sin\theta+\frac{\sin 2\theta}{2n}\right) \qquad 2\sin\theta\,.\cos\theta=\sin 2\theta$$

The acceleration is $f=\frac{dv}{dt}=\frac{d^2z}{dt^2}$

$$\therefore\ f=\omega^2 r\left(\cos\theta+\frac{1}{n}\cos 2\theta\right) \quad . \quad . \quad . \quad . \quad . \quad (4)$$

This expression is sufficiently exact for all practical purposes. The exact expression has the form—

$$f=\omega^2 r\left(\cos\theta+\frac{n^2\cos 2\theta+\sin^4\theta}{(n^2-\sin^2\theta)^{\frac{3}{2}}}\right) \quad . \quad . \quad . \quad . \quad (5)$$

**240. Klein's Construction for Inertia Pressure Curve.**—Graphically the acceleration can be easily obtained by what is known as Klein's construction.*

When the crank rotates at a constant angular velocity $\omega$, the

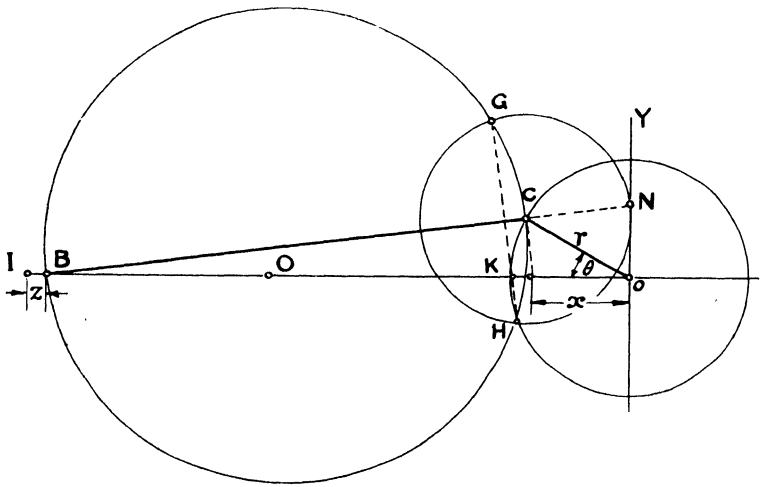



FIG. 303.

constant radial acceleration is given by $\omega^2 r$. In Fig. 303, produce the line BC of the connecting rod to cut OY in N. Describe a circle BGH on BC as diameter, and with C as centre and CN as radius the circle GNH cutting BGH in G and H. Join G and H cutting the line of stroke BO in K. Then $o$K is the acceleration $f$ of the piston and reciprocating parts in the line of stroke, to the same scale as the crank radius represents the constant radial acceleration of the

* *Journal of the Franklin Institute*, vol. 132, September 1891.

crank pin. The inertia force, for the position of crank and connecting rod shown, is $\frac{W}{g}\omega^2 r \times \frac{oK}{oC} = F_f$. If A is the effective area of the piston, the corresponding inertia pressure is—

$$p_f = \frac{F_f}{A} = \frac{W}{g} \cdot \frac{\omega^2 r}{A} \times \frac{oK}{oC} \quad . \quad . \quad . \quad . \quad . \quad (6)$$

*Example* 1.—If the effective piston area is 50·265 in.$^2$ and the weight of the reciprocating parts 100 lb., find the inertia pressure when the crank has turned through 30° from the inner dead centre, the speed being 340 rev./min., the ratio of connecting rod to crank length $n=4$, and the crank length 5 in.

By calculation, $f = \omega^2 r\left(\cos\theta + \frac{1}{n}\cos 2\theta\right)$

$$\omega = \frac{2\pi N}{60} = \frac{6{\cdot}28 \times 340}{60} = 35{\cdot}58 \ \frac{\text{rad.}}{\text{sec.}}; \quad \omega^2 = 1265{\cdot}93$$

$$f = 1265{\cdot}93 \times \frac{5}{12}\left(0{\cdot}866 + \frac{0{\cdot}5}{4}\right)$$

$$= 1265{\cdot}93 \times \frac{5}{12} \times 0{\cdot}985 = 519{\cdot}5 \ \frac{\text{ft.}}{\text{sec.}^2}$$

$$p_f = \frac{W}{g} \cdot \frac{f}{A} = \frac{100 \times 519{\cdot}5}{32{\cdot}2 \times 50{\cdot}265} = 32{\cdot}1 \text{ lb./in.}^2$$

$N = 340$
$\theta = 30°$, $\cos 30° = 0{\cdot}866$
$2\theta = 60°$, $\cos 60° = 0{\cdot}50$
$r = 5$ in. $= \frac{5}{12}$ ft.
$n = 4$
$W = 100$ lb.
$A = 50{\cdot}265$ lb./in.$^2$

Graphically, drawing the constructional circles as shown in Fig. 303 and scaling, $\frac{oK}{oC} = 0{\cdot}985$

and
$$p_f = \frac{W}{g} \cdot \frac{\omega^2 r}{A} \times \frac{oK}{oC}$$

$$= \frac{100 \times 1265{\cdot}93 \times 5 \times 0{\cdot}985}{12 \times 32{\cdot}2 \times 50{\cdot}265} = 32{\cdot}1 \text{ lb./in.}^2$$

When the crank is on the inner dead centre $\theta = 0$ and $\cos\theta = +1$, and when on the outer centre, $\theta = 180$ and $\cos\theta = -1$.

Equation 4 thus reduces to—

(Inner) $$f_I = \omega^2 r\left(1 + \frac{1}{n}\right) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

(Outer) $$f_O = \omega^2 r\left(-\frac{1}{n}\right) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

and $$p_{f_I} = \frac{W}{g} \cdot \frac{f_I}{A} = \frac{W}{g} \cdot \frac{\omega^2 r}{A}\left(1 + \frac{1}{n}\right) \quad . \quad . \quad . \quad . \quad (9)$$

$$p_{f_O} = \frac{W}{g} \cdot \frac{f_O}{A} = \frac{W}{g} \cdot \frac{\omega^2 r}{A}\left(1 - \frac{1}{n}\right) \quad . \quad . \quad . \quad (10)$$

Again, if the connecting rod were infinitely long, that is, if $n = \infty$,

the second term in the bracket becomes zero, and the acceleration force becomes—

$$F_{f\infty}=\frac{W}{g}\omega^2 r \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$$

which is equivalent to the centrifugal force of the reciprocating mass if concentrated at and revolving at uniform speed with the crank pin. At any other crank position, except a dead centre, this becomes—

$$F_{f\infty}=\frac{W}{g}\omega^2 r\cos\theta=\frac{W}{g}\omega^2 x \quad . \quad . \quad . \quad . \quad . \quad (12)$$

when $x$ is the displacement of the piston from the centre of the stroke or the centre of the crank pin circle. Thus—

$$F_{f\infty}=\text{constant}\times x$$

$$\text{or} \quad p_{f\infty}=\text{constant}\times x \quad . \quad . \quad . \quad . \quad . \quad (13)$$

This is the equation to a straight line passing through the centre of the crank pin circle as origin.

**241. Alternative Construction for the Inertia Pressure Curve.**—If the line of inertia force or equivalent inertia pressures on the piston

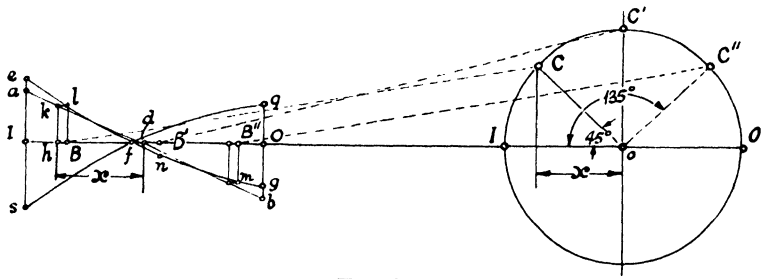



FIG. 304.

is drawn on a stroke base for the infinite connecting rod, the curve for the finite rod can be graphically derived from it by the following method.*

Find the value of—

$$p_{f\infty}=\frac{W}{g}\cdot\frac{\omega^2 r}{A}$$

In Fig. 304 scale up $Ia=p_{f\infty}$ at the inner dead centre and $Ob=p_{f\infty}$ down at the outer dead centre and draw the inertia pressure line *adb*. Set up $ae=\frac{1}{n}Ia$ and $bg=\frac{1}{n}Ob$, then $e$ and $g$ are the terminal points of the inertia pressure curve. Draw the crank radials $o$C, $o$C′, and $o$C″ at 45°, 90°, and 135° from the inner dead centre, and find

* This is due to J. W. Kershaw.

the corresponding piston positions B, B′, and B″. For position *o*C, measure *x* and set off $dh=x$ from the mid point of the stroke *d*. Draw verticals through *h* and B. Through the point of intersection *k* of *hk* with *adb*, draw the horizontal to cut B*l* in *l*, and *l* is another point of the inertia pressure curve. For position *o*C′ scale down B′*n*=*ae* and *n* is another point. For position *o*C″ repeat the construction for position *o*C, and find the other point *m*. A fair curve *efg* can then be drawn through these six points. At *f*, where it crosses the line of stroke, the crank and connecting rod should be at right angles. When the process is repeated for the in-stroke, the curve *qfs* is obtained. It is the curve *efg* reverted about IO.

When the curves of inertia pressure are superposed on the net pressure curves for back and front of the piston the resultant pressures ($p_R$) transmitted to the piston rod and connecting rod are given by the intercepts between the two curves. These intercepts, when scaled and replotted to a stroke base, give the resultant pressure diagram (Fig. 305), from which the twisting moment or torque diagram can be constructed, either as a rectangular or a polar diagram. The rectangular diagram (on an angle base) is most commonly employed.

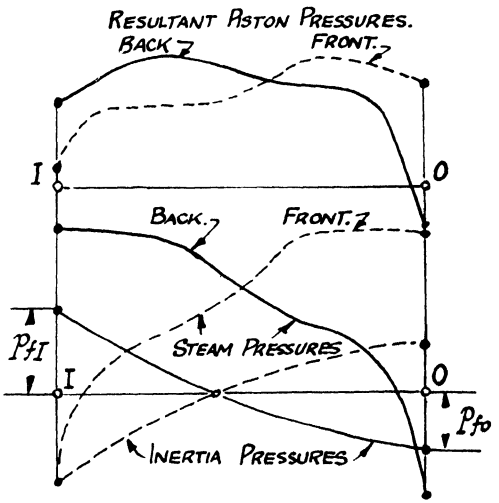



FIG. 305.

**242. Tangential or Crank Effort.**—As shown in Chapter VIII (Art. 156), the resultant piston load, which is given by the product of the resultant piston pressure and the effective area, or $P=p_RA$, is resolved into a thrust along the connecting rod and a downward thrust on the crosshead guide. The latter produces no turning effect on the shaft. The effective force for this purpose is the component of the thrust of the connecting rod at right angles to the crank arm. This is the "crank effort." It can be obtained either by direct calculation or by graphical construction. The product of the crank effort and the crank arm length gives the twisting moment or torque on the shaft.

The analytical method of determining the crank effort is as follows. In Fig. 306 draw CM parallel to Bo and CE parallel to *o*N, then *o*N=*o*M+NM.

The triangles NMC and CEB are similar,

$$\therefore \frac{NM}{CM}=\frac{CE}{BE} \text{ or } NM=CM\times\frac{CE}{BE} \qquad \begin{matrix} CM=r\cos\theta \\ CE=r\sin\theta \end{matrix}$$

$$\therefore NM=\frac{r^2\sin\theta\,.\cos\theta}{BE}$$

But $CE^2+BE^2=BC^2$ or $r^2\sin^2\theta+BE^2=l^2$

$$\therefore NM=\frac{r^2\sin\theta\,.\cos\theta}{\sqrt{l^2-r^2\sin^2\theta}}; \text{ also } OM=r\sin\theta$$

Hence
$$oN=r\sin\theta+\frac{r^2\sin\theta\,.\cos\theta}{\sqrt{l^2-r^2\sin^2\theta}} \quad . \quad . \quad . \quad . \quad . \quad (14)$$

For the position of the crank $o$C and connecting rod BC, defined by the crank angle $\theta$, the intersection O′ of the perpendicular BO′, from B and the line of the crank arm $o$C produced, gives the in-

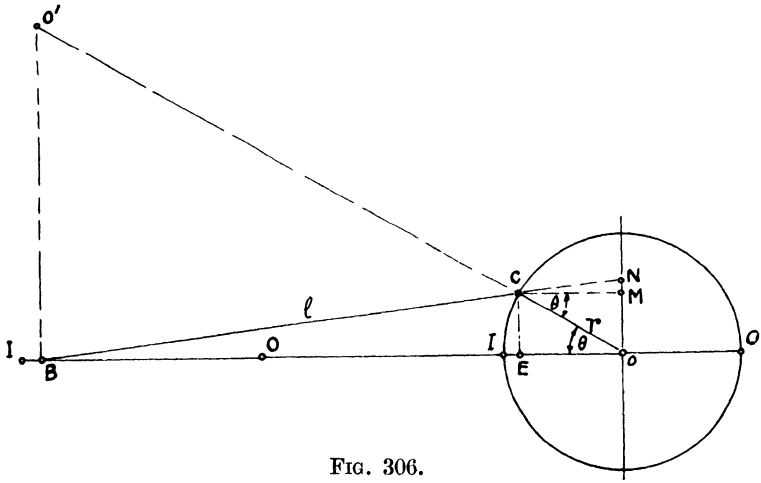



FIG. 306.

stantaneous centre of the connecting rod, that is, the centre about which it is momentarily turning at this position. The velocities of B along BO and of C tangential to $o$C are proportional to the radii O′B and O′C or

$$V_B/V_C=\frac{O'B}{O'C} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (15)$$

Let $P_T$ be the resolved component of the connecting rod thrust tangential to the crank arm, or the crank effort. According to the principle of work, the work done along BO for a small displacement of B by P is equal to the work done by $P_T$ on C along the tangent during the corresponding displacement of C, hence—

$$PV_B=P_TV_C$$

or
$$P_T=P\times\frac{V_B}{V_C}$$

Substituting from (15)—

$$P_T = P \times \frac{O'B}{O'C}$$

But since the triangles BO'C and NoC are similar, O'B/O'C $= oN/oC = \frac{oN}{r}$, hence the crank effort is given by—

$$P_T = P \times \frac{oN}{r} = p_R A \frac{oN}{r} \quad . \quad . \quad . \quad . \quad . \quad . \quad (16)$$

Substituting the value of $oN$ from (14) the crank effort is given by

$$P_T = P\left[\sin\theta + \frac{r \sin\theta \,.\, \cos\theta}{\sqrt{l^2 - r^2 \sin^2\theta}}\right] \quad . \quad . \quad . \quad (17)$$

Equation 17 enables the crank effort to be accurately obtained for any position $\theta$ of the crank, without construction; but it is simpler to draw down the crank and connecting rod to scale, measure the $oN$ values directly, and apply Equation 16.

**243. Twisting Moment or Torque Diagram.**—The twisting moment on the shaft is given by—

$$T = P_T r$$

So that from (16)—

$$T = p_R A \times oN \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (18)$$

also from (14)

$$T = p_R A r\left(\sin\theta + \frac{r \sin\theta \,.\, \cos\theta}{\sqrt{l^2 - r^2 \sin^2\theta}}\right) . \quad . \quad . \quad . \quad (19)$$

In order to construct the twisting moment diagram on a crank angle base draw in the crank pin circle, as shown in Fig. 307, and

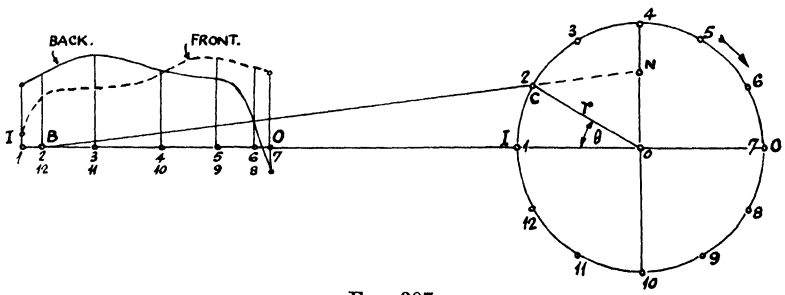



FIG. 307.

divide it into, say, twelve parts, numbering these from 1 at the inner dead centre.

Place the resultant piston pressure diagram, on the stroke base IO, in its correct position, as indicated, and with each point on the circumference as centre and the connecting rod length BC as radius mark the consecutive points 1 to 7 and 7 to 12 on IO, and erect

perpendiculars, to obtain the resultant piston pressures. At each crank position find the corresponding value of ON, to the same linear scale as the radius $r$.

Then tabulate $p_R A$, $o$N, and $T = p_R A \times o$N against the corresponding angle $\theta$, as indicated below.

| Crank Angle $\theta°$ | Piston Load $P = p_R A$ lb. | $o$N ft. | Twisting Moment $T = P \times o$N ft.-lb. |
|---|---|---|---|
| | | | |

If desired the value of the torque can be accurately calculated from Equation 17 ; but for all practical purposes the foregoing procedure is sufficient.

**244. Rectangular Diagram of Twisting Moment.**—On a base

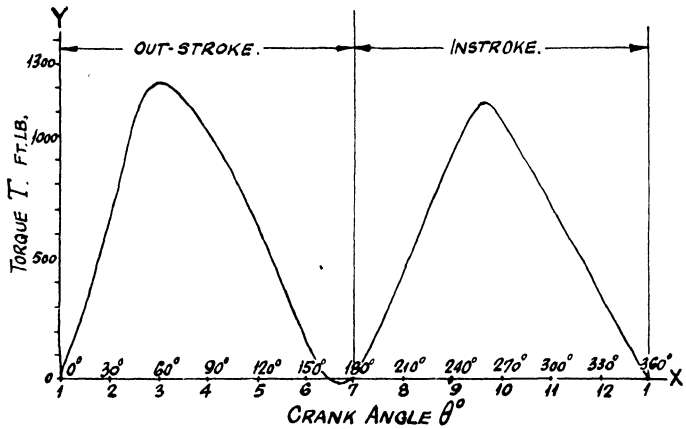



FIG. 308.

line OX (Fig. 308), lay off any convenient scale of angles in degrees between 0° and 360°, and mark the crank pin positions from 1 to 12 as shown. Set off any convenient scale of twisting moment in ft.-lb. on OY, then plot the twisting moment values at the successive angular positions of the crank. In the case given it will be seen that between $\theta = 165$ and $\theta = 180$ the torque has a slight negative value on the out-stroke.

**245. Polar Diagram of Twisting Moment.**—This is obtained, as shown in Fig. 309, by setting off the torque (T) values from the crank pin circle along the line of the crank radius produced.*

* For further examples of twisting moment diagrams, see paper on " Design and Construction of Flywheels," by A. Marshall Downie, B.Sc., *Trans. Inst. Eng. and Shipbuilders in Scotland*, 1901–02.

246. With regard to the weight W of the reciprocating parts to be used in the inertia force calculations, it is the usual practice to regard a portion of the connecting rod as a reciprocating mass and to add it to the mass of the piston, rod, and crosshead.

The part of the rod between the crosshead pin and the centre of gravity is usually taken. This method gives results which are sufficiently good for most practical purposes. When very accurate results are required, as in the case of a very high-speed engine, an

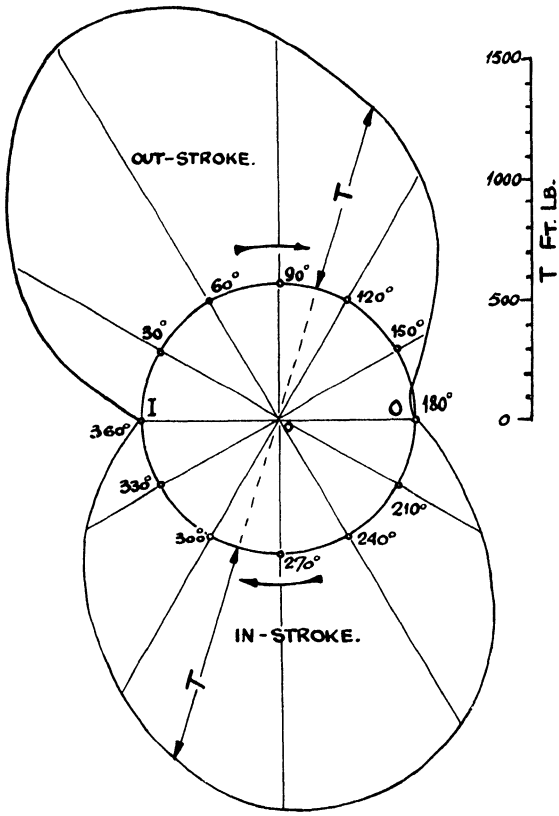



FIG. 309.

involved treatment taking account of the effect of the inertia of the connecting rod is necessary. It is not proposed to deal with it analytically in this text.* A graphical solution is given in Art. 249.

247. **Gravity Effect of the Reciprocating Masses.**—In this discussion of crank effort and twisting moment, the case of a horizontal engine has been considered. When the engine is vertical the

* For full analytical treatment see Dalby's *Balancing of Engines*, Chapter VIII.

gravitational effect on the reciprocating mass has to be taken into account. This is easily done. On the down-stroke, corresponding to the " out-stroke " of the horizontal engine, the resultant piston load P will be increased by W, and on the up-stroke it will be decreased. Hence for the vertical engine (P+W) is to be used for the down and (P—W) for the up-stroke, instead of P, when calculating the torque T.

The necessary allowance can, however, be made graphically by shifting the base line of the inertia pressure diagram before combining it with the effective steam pressure diagram to obtain the curves of resultant piston pressure (Fig. 305). Thus, as shown in Fig. 310, draw $Z_1Z_2$ above IO as the new base line of the inertia diagram for the down-stroke, and $Z_3Z_4$ below IO as the new base line for the up-stroke, and proceed to obtain the resultant piston pressure curves as before. The value of the equivalent pressure due to gravity is $\left(\frac{W}{A}\right)$.

When the modified resultant pressure diagram is used the $p_R$ values obtained are to be used directly in the calculation of the torque T.

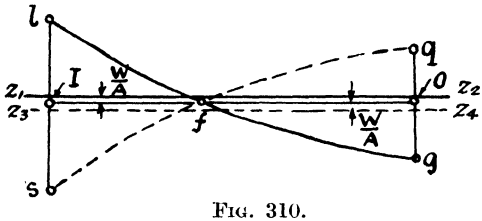



Fig. 310.

248. Since the curve of twisting moment is a curve of force plotted against angle, the area below it, between any given ordinates, represents mechanical work, since—

Torque × angle in radians = work, or

$$E = T\frac{\theta}{57{\cdot}296} \text{ ft. lb.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (20)$$

If, then, the torque scale is 1 in. $= x$ ft.-lb.

and the angle scale is 1 in. $= y$ degrees $= \frac{y}{57{\cdot}296}$ radian,

the work scale of the twisting moment diagram is—

$$1 \text{ in.}^2 = \frac{xy}{57{\cdot}296} \text{ ft.-lb. (work).}$$

The total area below the complete twisting moment curve (out and in-strokes) gives the work expended on the crank shaft in one revolution. This must obviously be the equivalent of the work done on the piston, or the work represented by the sum of the areas of the back and front indicator diagrams, as friction effect has been neglected in the calculations of the twisting moments.* When the

* For a short discussion on frictional effects, see Ewing's *Steam Engine and Other Heat Engines*, Chapter X.

friction at crosshead pin and crank pin is considered, the torque is slightly reduced and the twisting moment curve, with friction, lies slightly below the curve obtained as above when friction is neglected. The difference of areas obviously gives the energy wasted in frictional resistance during a revolution.

**249. Effect of the Inertia of the Connecting Rod on the Twisting Moment.**—At any given position during a revolution, the connecting rod has a motion of translation parallel to the line of stroke and a motion of rotation about the crosshead pin. It has an upward swing during the first half of the out-stroke, and a downward swing during the second half. It has thus a varying angular acceleration about the crosshead pin at each position.

The inertia force of the rod referred to the crank pin can be obtained graphically by the following construction. Referring to

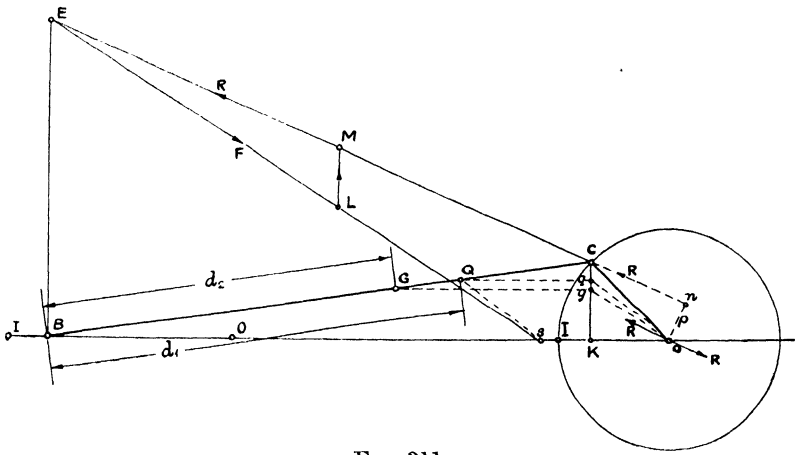



FIG. 311.

Fig. 311, let G be the centre of mass of the rod and Q the centre of percussion. The distance $d_1$ from the centre of rotation B to the centre of percussion, or BQ, is equal to the length of the simple pendulum having the same period as would be given by the connecting rod, if it were hung on a knife edge at B and set in vibration. If the period of oscillation thus obtained is $t$ secs., then—

$$t=2\pi\sqrt{\frac{l}{g}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (21)$$

and the pendulum length $l$ gives the value of $BQ=d_1$, and this fixes the position of Q.

Alternatively the value of $d_1$ can be obtained when the spin radius or radius of gyration of the rod ($k_B$) about the centre B is known.*

* For methods of calculating the moment of inertia and radius of gyration of a rotating body, see any standard work on dynamics.

If $d_2$ is the distance from the centre B to the centre of mass G, then—

$$d_1=\frac{k_B^2}{d_2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (22)$$

When G and Q are located as shown in Fig. 311, apply Klein's construction to find $o$K, the linear acceleration of B, and thus locate K. Join C and K. Through the centre of gravity G draw G$g$ horizontally to cut CK in $g$, and join $g$ and $o$. The length $go$ represents the acceleration of G in magnitude and direction to the same scale as $o$C represents the constant radial acceleration of the crank pin C.

The force producing this acceleration is—

$$F=\frac{W'}{g}\omega^2 go \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (23)$$

where W′ is the weight of the connecting rod. This line of action, however, does not pass through G, since the rod has an angular acceleration about B. To determine its direction, draw Q$q$ through the centre of percussion Q horizontally to cut CK in $q$ and join $q$ and $o$; the line $qo$ gives the acceleration of the centre of percussion Q. Through Q draw QS parallel to $qo$, to cut BO in S, and $s$ is a point on the line of action of the inertia force F. As its clinure and magnitude are given by $go$, the line of action is obtained by drawing $s$E parallel to $og$.

This force, for the position shown, is balanced by the reaction of the crosshead guide at B and the component acting on the crank pin at C. The latter must thus pass through the intersection E of $s$E with the vertical BE through B. The line joining E and C is thus the line of action of the force on the crank pin due to the rod inertia alone. Its magnitude and sense are obtained by drawing the triangle of forces. Set off EL=F=$\frac{W'}{g}\omega^2 go$ to any convenient scale, draw LM vertically to cut EC in M, and ME=R, the force acting on the crank pin which has a sense negative to that of F.

Draw $on$ perpendicular to EC produced, and the negative twisting moment due to the inertia force component R is R$\times on$=R$\rho$=T$_i$.

This value has to be subtracted from the twisting moment T obtained from the resultant pressure diagram (Art. 243) when only the inertia of the reciprocating parts is considered, in order to obtain the net twisting moment for the position of the rod shown. The inertia force effect of the rod is equivalent to a force R at the shaft centre and a couple constituted by an equal and opposite force at the shaft and the parallel force R at the crank pin, the couple arm being $\rho$. The force R thus constitutes a disturbing force on the engine frame.

*Example* 2.—A small high-speed vertical engine has a stroke of

7 in. and runs at 400 rev./min. The connecting rod weighs 60 lb., the length is 17¼ in. between centres; centre of mass is 11 in., and the centre of percussion 13 in. from the crosshead end. Determine the negative twisting moment due to the inertia force on the rod when the crank is at 45° from the inner dead centre.

Here $d_1=13$ in. and $d_2=11$ in. Draw in the crank and connecting rod, as shown in Fig. 311, to scale of half size, and locate G and Q, then find K by Klein's construction, and proceed with the rest of the construction as given above.

Scaling the drawing, $og=2{\cdot}95$ in., also W′=60 and the angular velocity is—

$$\omega\frac{2\pi N}{60}=\frac{2\times3{\cdot}1416\times400}{60}=41{\cdot}86 \text{ rad. sec.} \qquad N=400$$

Then $$F=\frac{W'}{g}\omega^2 og=\frac{60\times41{\cdot}86^2\times2{\cdot}95}{32{\cdot}2\times12}=805 \text{ lb.}$$

The couple arm $on=\rho=1{\cdot}25$ in. Setting off EL=805 and drawing LM, the force at the crank pin is EM=R=730 lb.

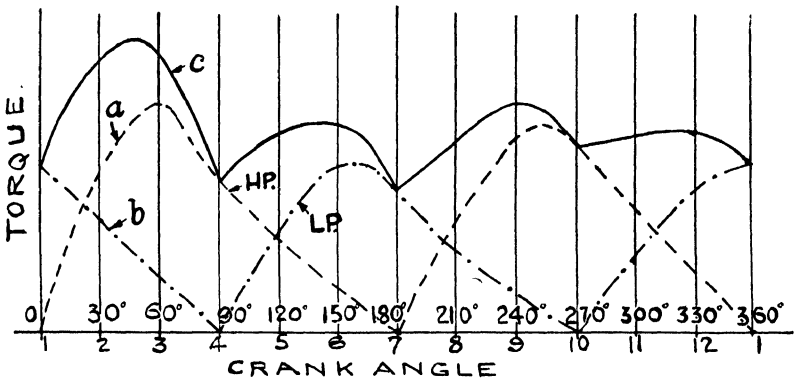



FIG. 312.

The negative twisting moment is, therefore—

$$T_i=-\frac{730\times1{\cdot}25}{12}=-76 \text{ ft.-lb.}$$

**250. Combined Twisting Moment Diagram of a Multi-cylinder Engine.**—When two or more cylinders drive on the crank shaft through separate cranks, the resulting twisting moment or torque is obtained by combining the separate diagrams for the cranks. The general result is a considerable reduction of the variation of twisting moment throughout a revolution, that is, a much more uniform torque is obtained.

This result is clearly shown in Fig. 312, in which the dotted curves are the twisting moment or torque curves for a two-cylinder compound engine with cranks at right angles. The chain dotted curve *a*

is the curve for the H.P. crank, and the dot and dash curve *b*, set 90° in advance of *a*, is the curve for the L.P. crank at right angles to it. With the engine working on the H.P. crank only, the torque (and the crank effort, which is simply proportional to it) would be zero at each dead centre. The L.P. crank, at right angles to the H.P., to a large extent remedies the deficiency of the twisting moment, ensuring a positive torque of considerable magnitude when the H.P. crank is on the dead centres. The H.P., on the other hand, supplies the torque when the L.P. is on its dead centres. The resultant torque on the shaft at any point in a revolution is given by the full line curve *c*, which is obtained by adding the ordinates of the two curves. In this case the minimum torque does not fall below 50 per cent. of the maximum.

In the case of a triple cylinder engine, with crank at 120°, the three twisting moment curves, when drawn at 120° intervals, give a much more uniform resultant torque curve.

It will be obvious that by increasing the number of the cranks, the variation of twisting moment can be reduced to a relatively small figure. The effects of this reduction is considered in the next chapter, dealing with the function of the flywheel.

# CHAPTER XII

## FLYWHEELS

**251. Function of the Flywheel.**—As already stated in Chapter XI, variations of speed from stroke to stroke are checked by the governor, but there are variations in each stroke which are beyond its control, and the main function of the flywheel is to keep the latter within reasonable limits. During the portion of a revolution in which the work done by the piston and connecting rod on the shaft is in excess of the work done against the external resistance, the wheel absorbs the surplus energy, and during the portion in which the work done is in defect of the external work, it restores energy. It accomplishes this object in virtue of its motion. It also assists in the prevention of "hunting" when a change of the external load takes place, and may, in certain circumstances, be used to obtain a slight increase of output for a short period under considerable change of load, such as occurs in the case of a rolling mill engine.

With a given flywheel the increase of its kinetic energy due to absorption of surplus energy is necessarily accompanied by increase of speed, and vice versa.

**252. Coefficients of Energy and Speed Fluctuation.**—In the majority of cases the external load is sensibly constant, so that the external-load torque or twisting moment is constant throughout a revolution.

This is represented by the mean ordinate of the torque diagram. It is readily found by integrating the area below the torque curve and dividing by the length of the diagram. It can also be calculated from the indicated horse-power, obtained from the indicator diagrams used to obtain the torque curve (by the methods of Chapter XI).

If $T_m$ = mean torque during a revolution, in ft.-lb.
$I$ = indicated horse-power
$N$ = speed in rev./min.

$$I = \frac{T_m \cdot 2\pi N}{33000}$$

and

$$T_m = \frac{33000 I}{2\pi N} \text{ ft.-lb.} \quad \ldots \quad (1)$$

The combined twisting moment diagram for a compound engine with cranks at right angles, considered in Art. 250, is reproduced

in Fig. 313, and the line $X_1X_2$, representing the constant external load torque on the shaft, has been drawn across it. The shaded areas below this line represent defects of energy which have to be supplied by the wheel; the areas above it the excesses which have to be absorbed and temporarily stored by the wheel. Between $X_1$ and $a$ since the wheel is supplying energy its speed decreases; between $a$ and $b$ it has to absorb the energy represented by the area of $abc$, and its speed increases. Similarly, between $c$ and $d$, $e$ and $f$, $g$ and $h$, $h$ and $k$ the speed decreases; and between $d$ and $e$, $f$ and $g$ it increases. There is thus a continuous fluctuation of speed during the revolution. Obviously the largest of these cyclic changes of speed is that due to the largest excess or defect, in this case the excess given by the area $abc$ on the out-stroke. The maximum fluctuation of speed between $a$ and $c$ will depend on the relative values of the excess energy and the store of energy carried by the

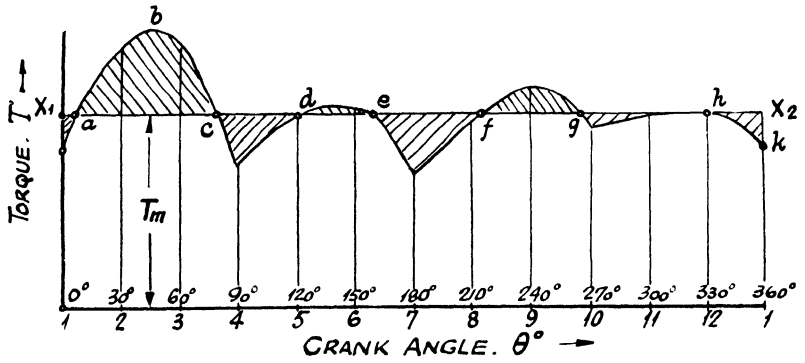



FIG. 313.

wheel running at the constant mean speed. By proportioning the wheel so that its kinetic energy is large, relatively to the maximum excess or defect shown by the torque diagram, the variation of speed can be kept within narrow limits.

Let $e$ =the maximum excess or defect of energy, or the "fluctuation of energy."

$E'$=the energy given by the torque diagram or work per revolution, then the ratio—

$$k=\frac{e}{E'} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

is termed the coefficient of fluctuation of energy.

In steam engine practice it has the average values * given in Table VI for double-acting engines.

* From a paper on "Essential Principles of Engine Design," by Frank Foster, M.Sc., Manchester Association of Engineers, 1915.

TABLE VI

COEFFICIENT OF ENERGY FLUCTUATION ($k$)

| Type | No. of Cranks | Cycles per Revolution | Crank Angles | Coefficient $k$ |
|---|---|---|---|---|
| Simple . . . . | 1 | 2 | — | 0·12–0·16 |
| Tandem . . . | 1 | 2 | — | 0·11–0·14 |
| Cross compound . | 2 | 4 | 90 | 0·06–0·09 |
| Triple . . . . | 3 | 6 | 120 | 0·025–0·03 |
| Uniflow . . . . | 1 | 2 | — | 0·11–0·13 |

In works on dynamics it is shown that the kinetic energy stored in a rotating body is given by—

$$E=\tfrac{1}{2}I\omega^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

where $\omega$=angular velocity in radians/sec.

I=second mass moment or moment of inertia of the body about the axis of rotation.

Let the subscripts 1, 0, and 2 represent the conditions at maximum, mean, and minimum speed of the wheel, then the energy stored in the wheel at mean speed is—

$$E_0=\tfrac{1}{2}I\omega_0^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

and the fluctuation of energy is—

$$e=\tfrac{1}{2}I(\omega_1^2-\omega_2^2)=\tfrac{1}{2}I(\omega_1+\omega_2)(\omega_1-\omega_2) \quad . \quad . \quad . \quad (5)$$

The mean angular velocity of the wheel for the usual ranges in practice is sensibly equal to $\omega_0=\frac{\omega_1+\omega_2}{2}$. Substituting in (5)—

$$e=I(\omega_1-\omega_2)\omega_0=2E_0\frac{(\omega_1-\omega_2)}{\omega_0} \quad . \quad . \quad . \quad . \quad (6)$$

The ratio of the fluctuation of speed $(\omega_1-\omega_2)$ to the mean speed $\omega_0$ is defined as the coefficient of fluctuation of speed, thus—

$$q=\frac{\omega_1-\omega_2}{\omega_0} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

Substituting in (6), the store of energy in the flywheel at mean speed, to give a required speed fluctuation with given fluctuation of energy, is—

$$E_0=\frac{e}{2q} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

Also, substituting for $E_0$ from (4)—

$$e=I\omega_0^2 q \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

The ratio of the energy stored in the wheel at mean speed to the work done per revolution is sometimes defined as the "flywheel coefficient." Denoting this by $c$, then—

$$c=\frac{E_0}{E'} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (10)$$

The relationship between the three coefficients is then given by—

$$q=\frac{k}{2c} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$$

Typical values of the speed fluctuation and flywheel energy coefficients are given in Table VII * for various services and drives.

TABLE VII

VALUES OF SPEED FLUCTUATION ($q$) AND FLYWHEEL ($c$) COEFFICIENTS

| Drive | Method of Drive | Type of Engine | No. of Cranks | $q$ | $c$ |
|---|---|---|---|---|---|
| Alternators in parallel . | Direct | High-speed | — | $\frac{1}{250}$ | — |
| ,, ,, . | ,, | ,, | — | $\frac{1}{250}-\frac{1}{300}$ | — |
| Direct current dynamos | ,, | ,, | — | $\frac{1}{100}-\frac{1}{125}$ | — |
| Best factory . . . . | Ropes | Steam | 1 | $\frac{1}{185}$ | 12 |
| ,, . . . . | ,, | ,, | 2 | $\frac{1}{200}-\frac{1}{340}$ | 7–12 |
| General factory . . . | ,, | ,, | 1 | $\frac{1}{125}-\frac{1}{155}$ | 8–10 |
| ,, . . . | ,, | ,, | 2 | $\frac{1}{170}-\frac{1}{200}$ | 6–9 |
| Pumping reciprocating . | Direct | ,, | 3 | — | 0·15–0·5 |
| ,, usual . . . | ,, | Triple Vertical | 3 | — | 0·45 |

If the wheel weight is W lb. and K is the radius of gyration, the moment of inertia is given by—

$$I=\frac{W}{g}K^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (12)$$

Again every point in the wheel at the radius K has a linear velocity $V=\omega K$, or $\omega^2=\frac{V^2}{K^2}$. Substituting these values in Equation 9—

$$e=\frac{W}{g}V^2q \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (13)$$

**253.** The flywheel consists of a heavy rim attached to the hub by a number of arms, and the gross weight of the wheel is the weight of these three parts. The rim contains the major portion of the flywheel mass. The radius of gyration K is less than the mean rim radius R. It is a common practice to substitute R for K and to calculate the moment of inertia with the mass of the rim for W. This, as a rule, gives a sufficiently close approximation and avoids the troublesome calculations for K. Some engineers take the velocity V as that of the rim circumference and W as 60 per cent. of the whole weight of the wheel, which is assumed to be concentrated at the rim surface.

* From Kempe's *Engineer's Year-book.*

In either case the linear velocity is given by—

$$V=\frac{2\pi RN}{60} \text{ ft./sec.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (14)$$

where N is the speed in rev./min., and R is in ft.

In any case of electrical generation, where the drive is direct and the generator armature adds to the flywheel effect, the armature weight can be allowed for by an equivalent mass added to the flywheel mass.

*Example* 1.—A compound steam engine driving a traction generator has cylinders 24 in. and 46 in. diameter and stroke 4 ft. The respective mean effective pressures are 60 lb./in.$^2$ and 17 lb./in.$^2$, and the normal speed is 90 rev./min. The flywheel has a mean rim diameter of 20 ft. and the weight is 45 tons. The armature may be taken as an addition of 5 tons to the weight of the wheel.

The fluctuation of energy is found to be 10 per cent. of the work per revolution on a constant resistance. Calculate (*a*) the work done per revolution E′; (*b*) the flywheel coefficient *c*; (*c*) the coefficient of fluctuation of speed *q*.

(*a*) $E'=\frac{\pi}{4}d_h^2P_{eh}\times 2L+\frac{\pi}{4}d_l^2P_{el}\times 2L$

$=\frac{\pi}{4}\times(24)^2\times 60\times 8+\frac{\pi}{4}\times(46)^2\times 17\times 8$

$=218000+226000=442{,}000$ ft.-lb./rev.

$d_h=24$
$d_e=46$
$P_{eh}=60$
$P_{el}=17$
$L=4$
$K=10$
$W=50\times 2240$
$V=\frac{2\pi KN}{60}$
$=\frac{2\pi\times 10\times 90}{60}$
$=94{\cdot}2$ ft./sec.

(*b*) $E_0=\frac{1}{2}I\omega^2=\frac{1}{2}\frac{W}{g}V^2$

$=\frac{50\times 2240\times(94{\cdot}2)^2}{2\times 32{\cdot}2}=15{\cdot}5\times 10^6$ ft.-lb.

$c=\frac{E_0}{E'}=\frac{15{\cdot}5\times 10^6}{442\times 10^3}=35$

(*c*) $q=\frac{e}{2E_0}=\frac{44200}{2\times 15{\cdot}5\times 10^6}=\frac{1}{702}$

$k=0{\cdot}10$
$e=0{\cdot}1\times E'=44{,}200$

In this case the cyclical variation is only $\frac{1}{7}$ of 1 per cent.

*Example* 2.—A uniflow engine, having cylinder diameter of 35 in. and stroke 40 in., develops 1100 I.H.P. at a speed of 120 rev./min. The mean rim velocity of the flywheel is not to exceed 85 ft./sec. Estimate the approximate weight of the flywheel rim to keep the cyclic variation of speed within 1 per cent., on the assumption that the coefficient of fluctuation of energy may be taken as 0·13. Also fix provisional values of the rim diameter and breadth and depth. Weight of cast-iron 0·26 lb./in.$^3$

Work per rev. $E'=\frac{33000I}{N}=\frac{33000\times 1100}{120}=302{,}500$ ft.-lb. | $I=1100$ $N=120$

Coefficient of fluctuation of energy $k=0{\cdot}13$.

$\therefore \quad e=kE'=0{\cdot}13\times 302{,}500$

$=39{,}325$ ft.-lb.

Also $q=0{\cdot}01=\frac{1}{100}$

(Equation 13) $\quad e=\frac{W}{g}V^2q$ $\qquad V=85$, $q=\frac{1}{100}$

$$\therefore \ W=\frac{32{\cdot}2\times 39325\times 100}{85\times 85\times 2240}=7{\cdot}83 \text{ tons}$$

Mean rim diameter, $D=\frac{60V}{\pi N}=\frac{60\times 85}{\pi\times 120}=13{\cdot}5$ ft.

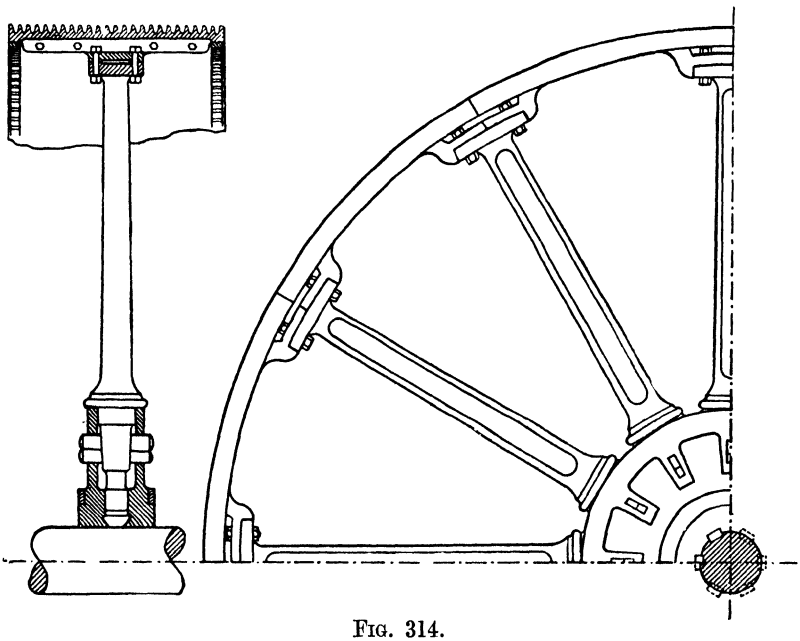


FIG. 314.

Make depth $d=12$ in., say, and the outside diameter of the wheel is—

$$D_o=14 \text{ ft.}$$

The breadth of the wheel is—

$$b=\frac{2240W}{w\pi Dd}$$

$$=\frac{2240\times 7{\cdot}83}{0{\cdot}26\times\pi\times 12\times 13{\cdot}5\times 12}=11 \text{ in.}$$

$D=13{\cdot}5\times 12$ in.
$d=12$ in.
$W=7{\cdot}83$ ton
$w=0{\cdot}26$

**254. Flywheel Construction.**—Flywheels fitted to the smaller classes of high-speed engines, mostly vertical enclosed, are usually

of the disc type, that is, the rim is joined to the boss by a disc or thin web. Sections of this type are shown on the engines illustrated in Figs. 364 to 369, Chapter XVI.

The wheels for horizontal slow-speed engines of moderate power are often made in two segments. The rim arms and boss may be cast all in one piece or built in sections. Illustrations of this type are shown on the engines of Figs. 350 and 357, Chapter XV.

Large flywheels may be made in several segments, with built-up arms. For factory rope drive the flywheel rim is usually made to serve as the rope pulley. A wheel of this type is shown in Fig. 314. It will be seen that the rim consists of a series of segments having lugs cast at the ends, by means of which they

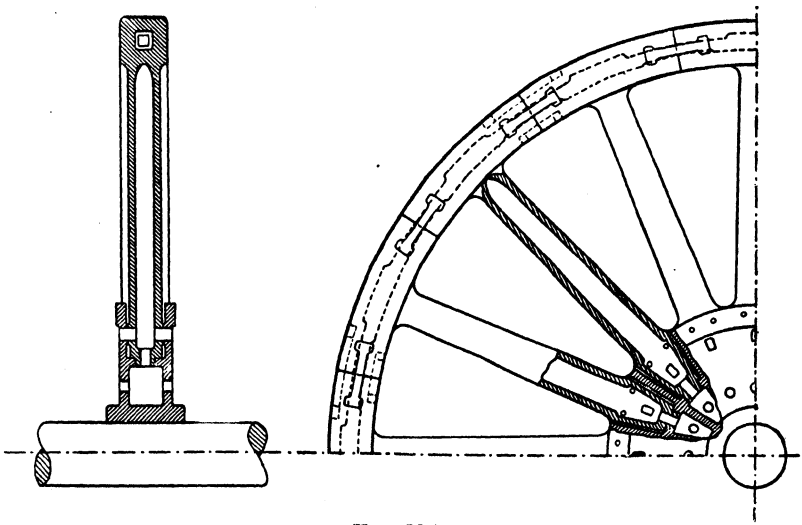


FIG. 315.

are bolted to the flanged arms of cruciform section. The arms are connected to the hub by double cotters.

An American design of flywheel, for the exacting service required on a rolling mill engine, is shown in Fig. 315.* This design was made in sizes from 20 to 30 ft. diameter. The rim is cast in segments, one arm being integral with each segment. The segment and arm are hollow.

The segments are connected by recessed steel T-pieces, proportioned so as to give the same strength at the joint as at the solid rim. The arms are cottered to the boss.

A German design of wheel † is shown in Fig. 316. It has a cast-iron rim in segments, a cast-iron boss, and steel plate arms secured in

* *Cassier's Magazine*, vol. 16, No. 2.
† From *Stahl und Eisen*, May 1899.

pairs to the boss and rim. There is a cast-iron distance piece between each pair of arms.

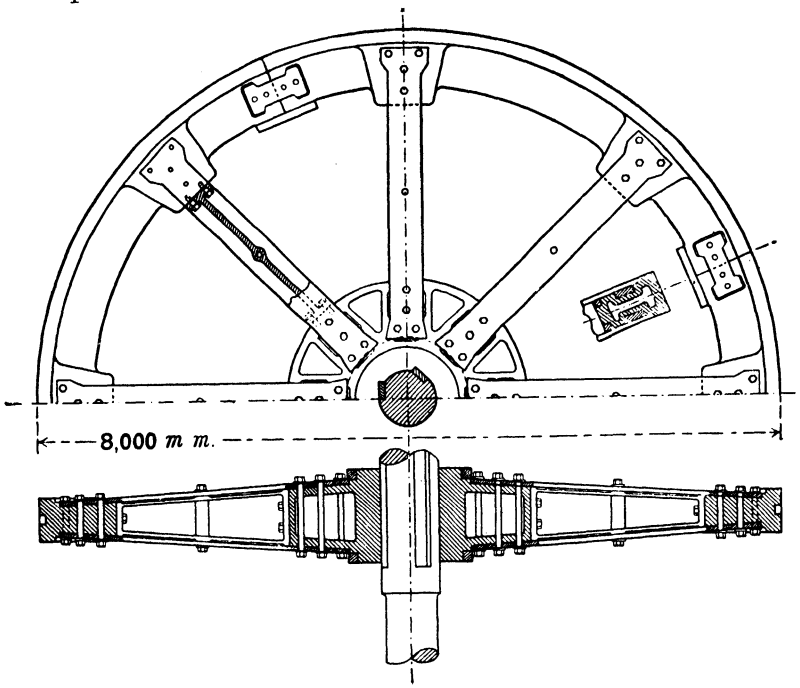



FIG. 316.

An American design of built-up wheel by the E. P. Allis Co., composed of flat steel plates, is shown in section in Fig. 317 and in elevation in Fig. 318.* It was fitted to a cross compound engine having cylinders 32 and 62 in. and stroke 5 ft. The rotational speed was 75 rev./min., giving a rim velocity of 90 ft./sec. The boss, Fig. 317, is made of cast-iron and its diameter is 8 ft. A steel plate A, 1 in. thick, is bolted on each side of the boss, and to each plate are fixed sixteen web plates, as shown in Fig. 318. These plates are stiffened by truss pieces 1 in.$\times$8 in., set diagonally as shown in Fig. 317, bolted at boss and rim ends by $\frac{1}{2}$-in. bolts, and by $1\frac{3}{4}$-in. bolts and distance pieces B near the middle. These truss pieces are fitted at the joints between the web plates. Cover plates D, 1 in. thick, are bolted outside the web plates at the hub.

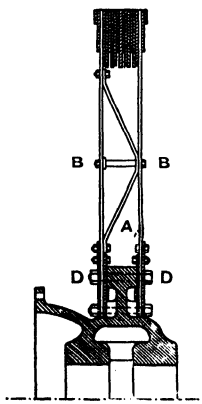



FIG. 317.

* From the *Railroad Gazette*.

The rim section consists of thirteen 1 in. plates. Cover plates 1 in.$\times$12 in. are riveted outside the web plates all round the rim. At the top there is an additional ring strip 1 in.$\times$5 in. All these plates are connected by $1\frac{3}{4}$ in. countersunk rivets.

**255.** The rim of a flywheel is subjected to a tensile or "hoop" stress due to the centrifugal force. This is normal to the radial cross-section. Due to the bending moment produced by the centrifugal force on the parts between the arms, there is, in addition, a compressive stress at the inner and a tensile stress at the outer surface. In general, the net result is a tensile stress across the section.

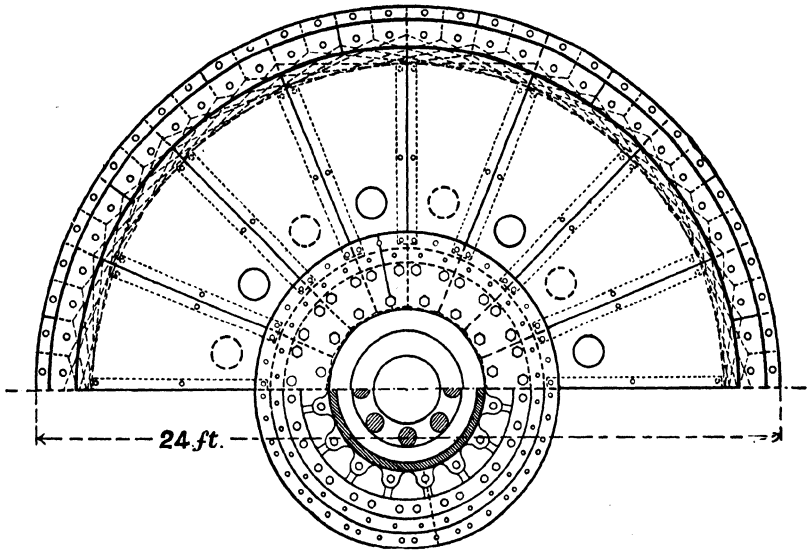



FIG. 318.

The hoop tension is independent of the section of the rim, as can easily be shown.

Let $\rho$=specific density of the material

$a$=cross-sectional area.

The centrifugal force on a unit length of the rim may be treated as a centrifugal pressure and the rim as a section of a boiler shell subject to this pressure. Then if this is $p_c$, the force tending to burst the rim across a diametral plane is, $F'=p_cD=p_c\times 2R$, where R is the mean rim radius,

$$p_c=\frac{\rho a V^2}{gR}, \quad \text{and} \quad F'=\frac{2\rho a V^2}{g}$$

But the centrifugal stress is $f_c=\frac{F'}{2a}$

$$\therefore f_c=\frac{\rho V^2}{g} \quad \ldots\ldots\ldots \quad (15)$$

For the general case of cast-iron, taking $\rho$=450 lb./ft.$^3$, this reduces to—

$$f_c=\frac{450V^2}{144\times 32\cdot 2}=\frac{V^2}{10\cdot 3} \text{ lb./in.}^2 \quad . \quad . \quad . \quad . \quad (16)$$

Some idea of the bending stresses may be obtained by treating the rim between any two arms as an encastre beam under a uniformly distributed load, $F=\frac{wV^2}{gR}$ lb./ft., where $w$=weight per unit length of the rim=$\rho a$.

The encastre conditions at the junctions of the rim and arms are uncertain, since each arm acts more or less as a flexible column and there is only connection between it and the rim at the underside of the latter. For the encastre beam the maximum bending moment is $M_{max}=\frac{Fl^2}{12}$, for rigid fixture; and at $\frac{1}{4}l$ from each end there is a point of contrary flexure and zero bending moment. In the actual case the conditions will be something between those for the encastre and simply supported beam and the constant in the bending moment equation will be between 8 and 12.

The foregoing applies to the wheel with solid rim. When it is built in segments the conditions are modified by the necessary joints of the rim. These joints are usually placed midway between the arms, and the question of the rigidity has to be considered. In the most extreme case if the joint has little rigidity compared with that of the solid rim, then the part between the two arms may be regarded as two butting cantilevers, each rigidly fixed to an arm. Here the maximum bending moment will be at the junction with each arm and its value $M'_{max}=\frac{Fl^2}{8}$, where $l$ is the space between the arms. If the joints are made at the junctions of the arms and rim, then the rim between the arms may be regarded as a singly supported beam for which the maximum bending moment is also $M'_{max}=\frac{Fl^2}{8}$. For whatever condition is chosen the maximum tensile stress due to bending can be calculated by the usual expression—

$$f_t=\frac{My_t}{I} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (17)$$

where I is the second moment of the section area about the neutral axis and $y_t$ the distance from the neutral axis to the external fibres. This can be added to the hoop stress.

**256.** With regard to the safe stress permissible in the material of the rim, it is necessary to provide a large safety margin or large factor of safety, not less than 12, as the bursting of a large flywheel may be as disastrous as the bursting of a steam boiler.

Assuming the best quality of material is used and the casting is carefully done, the safe stress of cast-iron may be taken about

1000 lb./in.$^2$ With cast-iron of ultimate strength, say 7 ton/in.$^2$ or 15,680 lb./in.$^2$, this gives a factor of safety of nearly 16. Inserting this in the stress equation (16) the corresponding maximum rim velocity is—

$$V=100 \text{ ft./sec.}$$

With good cast-iron, therefore, and allowing for bending stress the restriction of the rim speed to 100 ft./sec. ensures a factor of at least 15 for the solid rim wheel.

In fixing the diameter of wheel for a given engine to run at a stated speed, this value can thus be taken. In the case of the wheel with jointed rim a necessarily lower factor obtains, owing to the efficiency of the jointing being a little uncertain. It is undesirable to run the wheel with a lower factor of safety than 8. The weakest type of joint is the flanged one, and the maximum speed should be reduced to about 80 ft./sec. In the case of a built-up steel wheel the rim speed may be taken at 130 ft./sec.

**257.** With regard to the arms, these are subjected to direct tensile stress due to the centrifugal force, and a certain amount of the pull of the rim, and to bending stresses under centrifugal force, and when sudden fluctuations of speed occur, to additional inertia stresses.

If the rim were not rigid and depended for its support entirely on the arms, the entire load on the half rim would be taken by the connecting arms.

If the length supported by each arm is $s$, then the centrifugal pull of the rim on the arm is—

$$F_c=\frac{\rho a s V^2}{gR}$$

The total pull on $n$ arms is $F_a=\Sigma\frac{\rho a s V^2}{gR}=\frac{\rho a V^2 \Sigma s}{gR}$

$$=\frac{\rho a V^2 2\pi R}{gR}=\frac{2\rho a V^2}{g}\times\pi=F'\pi$$

where $F'$ is the force tending to burst the wheel on a diametral plane (Art. 255). Hence the load per arm is—

$$F_a=\frac{\pi F'}{n} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (18)$$

If $a'$=area of arm, the stress $f_t=f_c=\frac{F_a}{a'}=\frac{\pi F'}{na'}$

If $a$ =area of rim, the stress $f_c=\frac{F'}{2a}$

For equality of stress, then—

$$\frac{a'}{a}=\frac{2\pi}{n} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (19)$$

For instance, with eight arms the arm cross-section would thus be 78 per cent. of the rim section. Such a condition as assumed above should not arise in a well designed wheel. The rim should be sufficiently stiff to carry its centrifugal load and relieve the arms. In general, with large built-up wheels the total cross-sectional area of the arms is made from two to three times the cross-sectional area of the rim.

Each arm may be regarded as a beam fixed at the hub end and subjected to a force at the rim, due to the acceleration of the segment it connects, when a fluctuation of speed, denoted by the maximum value on the twisting moment diagram, occurs. The centrifugal force on its mass also produces a maximum stress effect at the hub end. It is not proposed to go into the question of design of arms or joints, which is outside the scope of this text. The approximate stress in the arms should be such that a factor of safety of at least 8 is ensured. When the arms are secured to the hub by bolts or cotters the strength of these, in double shear, should be equal to the strength of the arm.

**258.** The following general conclusions regarding the design of flywheels are given in the paper by Marshall Downie, already cited.* The student should refer to it for further details.

" Cast-iron flywheels should, where practicable, have solid rims ; but when jointed the best form is the ' link and lug ' type, where such can be adopted without inconvenience, and the next best is the dowel-plate and cotter. Flanged and bolted joints should be avoided ; and the best place for a joint is near one arm.

" Solid rimmed cast-iron wheels may be run at a peripheral velocity of 100 ft./sec., with the certain knowledge that the factor of safety is not under 12 ; and link and lug jointed wheels may also be run at that speed and have a certain factor of safety of 8. A lower factor should not be used ; and flange-jointed wheels should not be run above 70 to 75 ft./sec. Built steel wheels may be run up to 130 ft./sec.†

" Arms should be joined to the rims with large fillets and their fixing to the hub should be carefully fitted. The best material of its kind should be used in construction, and homogeneity should be ensured as far as practicable by having test bars cast and proved from each segment."

* See footnote, p. 466.

† Some modern wrought-steel flywheels are stated to run at 19,000 ft./min. or 317 ft./sec. See *Mechanical World Year Book*, 1931.

# CHAPTER XIII

## BALANCING

**259.** A MACHINE is perfectly balanced when the inertia forces due to the relative motions of its working parts neutralise each other in all phases of the motion, and there is therefore no resultant force or couple on the frame.

There would, in such a case, be no tendency for the machine, as a whole, to vibrate, and it could be run without any foundation fixture. It would be perfectly self-contained with regard to the forces between the various members. Such an ideal condition is never obtained in practice, although in some classes of machines, in which the motion is purely rotary, the vibration effects can be reduced to a very small order.

The steam engine, in which reciprocating is transformed into rotary motion, is subject to considerable disturbing inertia forces,

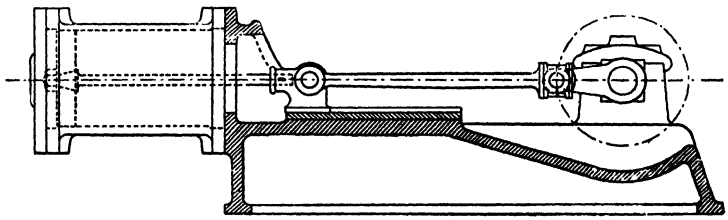


FIG. 319.

which, when the speed is high, produce serious vibration ; and it is essential that these should be balanced as completely as the conditions will admit. Balance in this case has to be considered for both reciprocating and rotating masses.

**260. Effect of Inertia of the Reciprocating Masses.**—An outline of a simple engine cylinder, frame, piston, connecting rod, and crank is shown, with the crank at the inner dead centre, in Fig. 319. The engine is supposed to be running at speed. Just as it turns the centre the motion of the reciprocating masses changes from a negative to a positive value, or, in other words, the parts momentarily come to rest at the centre. As already shown in Chapter XI, in order to start them on the out-stroke again an accelerating force has to be applied, and this can only be obtained from the

pressure of the steam in the cylinder, or, if the steam were shut off, by a pull from the crank pin.

Let $p$ be the pressure of the steam when the crank is just turning the dead centre. Then with area A the steam exerts a total force $P=pA$ on the piston face (momentarily at rest) and a pressure $P=pA$ on the cylinder cover. The piston pressure P is transmitted through the rod and connecting rod and crank to the main bearing and thence to the frame, and tends to move the frame to the right. The cover load P through the cylinder connections tends to move the frame to the left. Both forces are equal, so that, as far as frame motion on the foundation is concerned, the effects balance. There is, however, a tensile stress induced on all sections of the frame and cylinder, due to the load P. As soon as the crank turns the centre and the piston begins to move outward, a part of the steam load on the piston is required to accelerate the reciprocating mass. This must equal the inertia pressure $p_f$, so that the effective pressure on the piston face is $(p-p_f)$ and the net piston load, transmitted through the piston rod and connecting rod, is $P'=(p-p_f)A$.

This is the force tending to move the frame to the right. The cover load, however, remains as $P=pA$, and is the force tending to move the frame to the left. The result of the inertia effect is thus an unbalanced force $-F_f=-p_fA$ on the frame, tending to move it to the left. As already seen, the inertia pressure diminishes as the piston moves forward and reaches a zero value somewhere about the mid stroke, the position of the piston depending on the ratio of connecting rod to crank. At this position there is again equality of steam load on the frame to right and left, whatever the pressure of the steam may be. The inertia pressure then begins to operate in the reverse sense and reaches its maximum again at the outer dead centre position. The effective load transmitted to the frame through the shaft is now $P''=(p+p_f)A$, while the cover load is $P=pA$. There is then a positive unbalanced force $+F_f=+p_fA$ tending to move the frame to the right.

During one half revolution, therefore, the frame receives a variable push first in one direction and then in the other in the line of stroke, due to the acceleration and retardation of the reciprocating mass, of the piston, rod, and crosshead. As already seen in Chapter XI, when the connecting rod is short compared with the crank there is an oblique inertia force due to its mass, which has both linear and angular acceleration. The rod then introduces a secondary disturbing effect, which gives rise to what is termed " secondary " balancing.

As a rule, unless the ratio $n$ is small, the secondary effects are neglected, and to obtain a first approximation the mass of the rod is subdivided, one part being taken as an equivalent mass rotating with the crank pin and the other part as a mass reciprocating with the crosshead ; and the balancing calculation is carried out for a

rod of infinite length, for which condition the virtual reciprocating mass has a simple harmonic motion. For this condition the forces and couples produced are known as " primary," and the balancing effect is termed " primary balance." As already shown in Art. 240, with infinite connecting rod the inertia force is given by—

$$F_{f\infty} = \frac{W}{g}\omega^2 r \cos\theta \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

where $\frac{W}{g}$ is here the virtual reciprocating mass, $r$ the crank radius, and $\theta$ the angle of crank displacement from the inner dead centre ; $\omega$ the constant angular velocity of the crank. Since $r \cos\theta = x$, the displacement of the piston from the centre of the stroke—

$$F_{f\infty} = \frac{W}{g}\omega^2 x$$

and the magnitude of the inertia force at any crank position is given by $x$, to the same scale as $r$ represents the centrifugal force of a mass $\frac{W}{g}$ concentrated at and rotating with the crank pin. The sign is determined by the point of projection on the line of stroke. If it falls to the left of 0, Fig. 303, the force is negative, if to the right it is positive.

If now some balancing mass $\frac{W'}{g}$ is placed opposite and in the plane of rotation of the crank, so that it produces a centrifugal force equal to that of $\frac{W}{g}$, the inertia force will be balanced at all points in the line of stroke by the component of this force in that direction. Its component at right angles to this, however, will be unbalanced. It is obvious, then, that a reciprocating mass cannot be completely balanced by a rotating mass. Complete balance can only be obtained by the use of a reciprocating mass in the same plane as the out-of-balance mass and moving in the opposite direction, or two masses in parallel planes, so that the equal and opposed inertia couples and forces may balance.

**261. Effect of Centrifugal Force of Rotating Masses.**—With regard to an unbalanced rotating mass, such as that of the crank, and equivalent of the mass of the crank webs and pin concentrated at the crank pin centre, it could obviously be balanced by a mass placed opposite the crank in its plane of rotation, at such a radius that its centrifugal force would balance the force on the crank mass. This, of course, is not practicable, and the balance mass has to be fixed in some plane parallel to the plane of the crank. The centrifugal forces will balance, but as each acts in a different plane they constitute a couple tending to turn the shaft in a plane perpendicular to the plane of rotation of the masses. The two general conditions of equilibrium of a body under the action of a

system of external forces are: (*a*) there must be no resultant force—this is force balance; (*b*) there must be no resultant couple—this is couple balance.

In the case of the unbalanced rotating mass, the second condition can be fulfilled by dividing the balance mass into two portions and placing these in two planes parallel to the plane of the unbalanced mass, and so situated that the positive and negative couples produced balance each other.

The foregoing remarks apply to the simple or single cylinder and crank engine. For the general case, the multi-cylinder engine with a number of cranks has to be considered. This case involves the balancing of a number of reciprocating and rotating masses in different planes. In what follows, the condition of primary balance only is considered, as the balance of inertia couples and forces of the second and higher order, involves an advanced mathematical treatment beyond the scope of this intermediate text.

**262. Balancing of Rotating Masses in Different Planes.**—In Fig. 320 let A represent a plane section of any body which is constrained by an axis at O perpendicular to the plane, and let F be any force acting in the plane A at a perpendicular distance *a* from the axis at O. Since O is fixed in position it is obvious that the force F cannot produce a motion of translation of A, but it can rotate it about the axis. The translation is prevented by the pressure of the body on the axis equal to the force +F. The axis in turn reacts on the body with an equal and opposite force —F. The forces +F and —F acting at the arm length *a* thus form the couple producing rotation about the axis. Hence it may be stated that a force F, acting on a body fixed at one point O distant *a* from its line of action, may be resolved into—

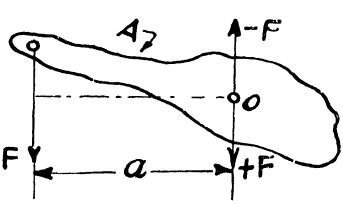


FIG. 320.

(1) An equal and parallel force F acting at the point;

(2) A couple of moment F*a*, tending to rotate the body about an axis through the point perpendicular to its plane.

Consider next the case of a shaft, as shown in Fig. 321, loaded with a radial force F, which is the centrifugal force on a mass weighing W lb. rotating in the plane A. The plane C (shaded) passing through the shaft axis is the plane of the force. Let O be any arbitrarily chosen point on the shaft axis distant *a* from the plane A, and OD an axis through O perpendicular to the plane C of the force F, and lying in the plane B. Then, according to the foregoing deduction, the loading is equivalent to the force +F in the plane B, equal and parallel to F at A, and the couple constituted by the force —F and +F in the plane C, acting at arm *a*, and tending to rotate the shaft and plane C about the axis OD. The

arbitrarily chosen plane B, containing the couple axis and the referred dynamical forces, is called the " plane of reference."

**263.** Any number of masses attached to and rotating with a shaft may be treated in this way, the forces being referred to the arbitrarily chosen plane and vectorially summed to obtain the resultant dynamical force which has to be balanced.

A couple is a directed quantity, and hence can be represented by the length of the axis in the reference plane, such as OD, which is drawn in this plane perpendicular to the plane of the couple. The couple vectors can thus be vectorially summed and the resulting couple can be found; and this has to be balanced to obtain equilibrium. The resultant couple is given by—

$$C_R = \frac{\omega^2}{g} \Sigma W r a \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

and the resultant force by—

$$F_R = \frac{\omega^2}{g} \Sigma W r \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

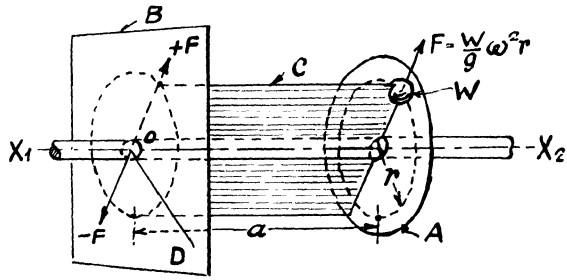



FIG. 321.

The value of $\frac{\omega^2}{g}$ is constant, so that for equilibrium $\Sigma W r a = 0$ and $\Sigma W r = 0$ and the couple and force polygons drawn for the values of $Wra$ and $Wr$, must both close.

If both the polygons do not close then two additional balance masses have to be attached to the shaft in arbitrarily chosen planes of rotation, so that when their couple and force vectors are also drawn, the polygons will close.

For convenience of reference the couple and force polygons may be termed the $Wra$ polygon and the $Wr$ polygon.

In order to illustrate the general vectorial method of determining the resultant force and couple of an unbalanced system of rotating masses, the perspective sketch of four masses weighing $W_1$, $W_2$, $W_3$, and $W_4$ lb., rotating at radii $r_1$, $r_2$, $r_3$, and $r_4$ ft. in planes 1, 2, 3, and 4 with angular velocity $\omega$ radians/sec. is shown in Fig. 322 (*a*). Choose a reference plane 5 at any point O on the shaft giving the couple arms $a_1$, $a_2$, $a_3$, and $a_4$ for the forces $F_1$, $F_2$, $F_3$, and $F_4$.

On the reference plane (turned into the plane of the paper) draw

the force vectors 1, 2, 3, and 4, parallel to the radii in the respective planes, and having the corresponding $Wr$ magnitudes, to any convenient scale, as shown at (*b*). All the sense arrows read from O outwards.

The corresponding couple axes, as already shown, are drawn at right angles to the force vectors. To avoid confusion the reference plane is again shown at (*c*), and the couple vectors 1, 2, 3, and 4, having the magnitudes $Wra$, are drawn on it, each perpendicular to the direction of the corresponding force vector.

The reference plane in this case is chosen to the left of the rotating masses, so that all the couples have the same sense and the sense arrows therefore point from O outward. If the reference plane were

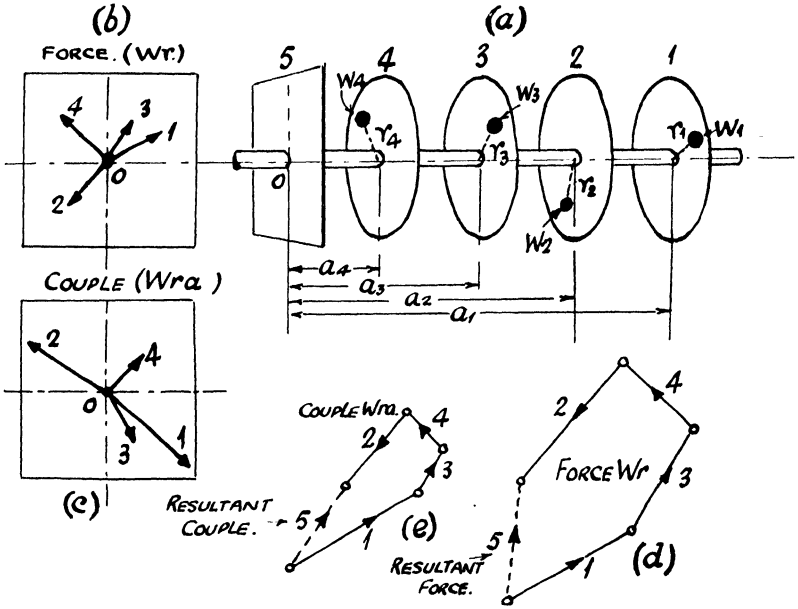



FIG. 322.

chosen with some of the masses on one side and some on the other, then the sense arrows of the couples for the masses on the left would have to be negative to those on the right, and would point to the centre O, since the couple arms are measured negatively. The space diagrams for the proportional forces and couples are now given by (*b*) and (*c*), Fig. 322, and the summation is then carried out, according to the general rule for vectorial addition of directed quantities.

The invariable rule in vector summation is, that the sense arrows must point round the diagram in one direction. If the diagram or polygon is unclosed the closing line is the resultant and has a sense in the opposite direction. The force polygon, obtained by drawing

the vectors in order, equal in magnitude, and parallel in direction to the vectors in (*b*), is shown in (*d*). It is unclosed, and the resultant unbalanced force is shown in magnitude and direction by the dotted vector 5.

Similarly the couple polygon (*e*) can be drawn with the sides parallel to the vectors in (*c*). It is also unclosed, and the resultant couple is given by the closing side 5. It will be evident that if the plane (*c*) is rotated through 90°, so that the couple vectors are parallel to the force vectors in (*d*), the same result will be obtained from the couple polygon. For practical purposes, then, the sides of the couple polygon can be drawn, as shown in (*e*), parallel to the corresponding sides of the force polygon, the condition as to sense of the couples being attended to when all the masses do not lie on one side of the reference plane. In order to facilitate the graphical work the values of W, $r$, $a$, W$r$, and W$ra$, should be set out in tabular form as shown below.

| No. of plane of revolution | Weight of Balance mass W lb. | Radius $r$ ft. | Couple Arm $a$ ft. | Products | |
|---|---|---|---|---|---|
| | | | | W$ra$ | W$r$ |
| | | | | | |

If the foregoing construction shows the system to be unbalanced, the next step is to find the two masses in arbitrarily chosen planes which will produce balance, that is, give closed polygons for the modified system. Take the reference plane as one of these arbitrarily chosen planes, so that the couple arm for the balance mass in it is zero and hence the couple is zero. Fix on any convenient position for the second plane, which in most cases may be taken as a plane containing one of the unbalanced masses. Let the couple arm for this force be $a_R$, then the proportional force value is—

$$\frac{\Sigma \mathrm{W}ra}{a_R} = \mathrm{W}'r' \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

When the arbitrary value of the radius $r'$ in the chosen plane is fixed, the necessary balance mass W′ is known from (4). In other words, to obtain the balance mass in the chosen plane, scale the closing side of the couple polygon, Fig. 322 (*e*), and divide the value by the arbitrarily chosen distance between this second balance plane and the reference plane. Again divide this value (which is a " mass moment ") by the radius $r'$ at which the balance mass is to be placed and obtain the weight W′ of one balance mass.

If the force and couple polygons, as is the usual practice, are drawn with corresponding sides parallel, the direction of this balance force due to W′ will be parallel to the closing side of the couple polygon. Add this force to the system in Fig. 322 (*d*),

and redraw and close the force polygon. The closing line will then give the proportional force or $Wr$ value for the mass in the reference plane. If the weight of this mass is W″ and the arbitrarily chosen radius is $r''$, then $W''=\frac{\Sigma Wr}{r''}$, or the magnitude of the closing side of the force polygon divided by the radius $r''$.

The foregoing may be condensed into the following practical rule to determine the balancing masses for an unbalanced system of rotating masses.

(1) Choose two planes at any arbitrary distance apart and the corresponding radii at which the balance masses are to be fixed in them.

(2) Let one of these planes be the " plane of reference," and draw the $Wra$ couple polygon referred to it. Close this polygon. Using the chosen couple arm for the second plane, determine, from the closing couple vector, the value of the mass in this plane.

(3) Draw the force polygon for the original system of masses and this added mass, and the closing line gives the $Wr$ value for the mass in the reference plane. The W value is obtained by dividing this " mass moment " by the chosen radius in the reference plane.

The accuracy of the work can be checked by taking the balanced system of masses and drawing down the force and couple polygons for a new position of the reference plane. If the work has been done correctly both polygons will close.

*Example* 1.—A three-cylinder vertical engine has cranks at 120° ; the crank radius is 10 in. and the equivalent rotating mass at each crank pin weighs 150 lb. The pitch of the cylinders is 30 in. Two rotating balance masses are to be placed at 12 in. radius in planes 75 in. apart, and disposed symmetrically to the cylinders. Find the weights of the balance masses, so that there may be no dynamic load on the bearings.

The shaft and crank arrangement is shown in Fig. 323 (*a*) and (*b*), the planes of the crank masses being numbered 1, 2, and 3, and the balancing mass planes 4 and 5. The reference plane chosen is 5. Tabulating according to the method stated on p. 491—

| No. of Plane | W lb. | $r$ ft. | $a$ ft. | $Wra$ | $Wr$ |
|---|---|---|---|---|---|
| 1 | 150 | 0·833 | 0·625 | 78 | 124·95 |
| 2 | 150 | 0·833 | 3·12 | 390 | 124·95 |
| 3 | 150 | 0·833 | 5·62 | 702 | 124·95 |
| 4 | (86·5) | 1·00 | 6·25 | (closure) 540 | (86·5) |
| 5 | (86·5) | 1·00 | 0 | 0 | (closure) 86·5 |

The couple polygon ABCD is drawn as shown in (*c*), with the sides parallel to the force vectors in the space diagram (*b*) of the cranks.

The closing side DA=540. The couple arm $a_4$=75 in.=6·25 ft., so that the proportional force value is $Wr=\frac{540}{6\cdot25}=86\cdot5$. These two values are entered in brackets in the table. The force diagram *abcde*, Fig. 323 (*d*), is then drawn, the vector *de*=86·5 being drawn parallel to the closing side DA of the couple polygon. The closing side of the force polygon is *ea*, equal to *de* reverted, *a* and *d* being coincident and *Wr* for plane 4 is the same as for plane 5. As the radius in each case is 1 ft., $W_4=W_5=86\cdot5$ lb.

The force vectors are drawn in line in Fig. 323 (*b*) and are inclined at 30° to the plane of the crank 1. It is instructive to note, in this

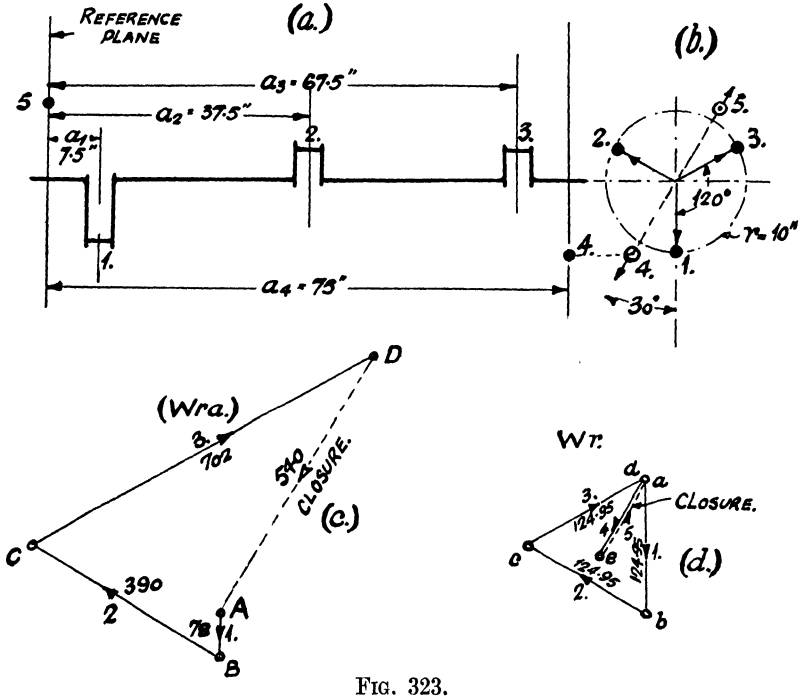



FIG. 323.

particular case with symmetrical arrangement of cranks and equal crank masses, that the centrifugal forces are in balance, but there is an unbalanced couple, which requires the addition of the balance masses.

**264. Primary Balancing of Reciprocating Masses.**—As already pointed out in Art. 260, a reciprocating mass cannot be completely balanced by a rotating mass. If, however, each connecting rod of a multi-cylinder engine is taken as infinitely long, so that the motion of the whole of the reciprocating masses may be considered as "simple harmonic," it is possible to balance the masses among

themselves by a suitable choice of their magnitudes and disposition of cranks, provided the number is greater than three.

The assumption of simple harmonic motion is not correct, but in general the approximation to complete balance obtained is sufficiently satisfactory. As already indicated, the effect of each connecting rod mass is taken into account by apportioning it between the crosshead and the crank pin.

If $l$ is the length of rod between centres, $x$ the distance between the mass centre of the rod and the crank pin end, and $W_R$ the weight of the rod, then the proportion which is to be added to the crosshead is given by $w_r=\frac{W_R x}{l}$ and the proportion (rotating) to be added at the crank pin is $w_c=(W_R-w_r)=W_R\left(1-\frac{x}{l}\right)$.

**265.** Consider a system of $n$ masses reciprocated in the same plane by cranks, through the medium of infinitely long connecting rods (or their mechanical equivalents, slotted crossheads), as for instance the reciprocating masses of a vertical engine.

With this assumed condition of simple harmonic motion, as already shown, for any mass the inertia force at any point of the stroke is proportional to the displacement of the mass from the centre position, and hence is given by the component in the line of stroke of the centrifugal force of an equal mass fixed on and rotating with the crank pin. The system of reciprocating masses can thus be treated as an equivalent system of $n$ rotating masses.

The conditions for complete self-balance, as before, are (1) closure of the W$ra$ vector polygon, (2) closure of the W$r$ vector polygon.

The problem is to find the magnitudes of the masses and the relative dispositions of the cranks so that these conditions may be fulfilled.

**266.** The complete specification of any mass involves three items: (1) the W$r$ value (or the W value referred to a unit radius); (2) the direction of the radius or the crank angle $\theta$, referred to some line of reference in the reference plane; (3) the couple arm $a$, or distance between the plane of revolution and the reference plane.

There are thus $3n$ quantities for $n$ masses. They are not all, however, independent variables.

The closures of the sides of the W$r$ vector polygon are unaltered by change of scale, or, in other words, the form of the diagram is unaltered if one of the magnitudes is taken as unit and the others expressed in terms of it.

On this hypothesis the number of independent W$r$ variables is one less than the number of masses or $(n-1)$. The same condition holds for the W$ra$ polygon, where by expressing the sides in terms of one of these as unit, the number of W$ra$ independent variables becomes $(n-1)$. For $n$ cranks there are $n$ angles, but by taking the line of one crank as reference line the number of independent crank angle variables reduces to $(n-1)$. Summing these the

number of independent variables corresponding to $n$ reciprocating or quasi-rotating masses, is—

$$\overset{Wr}{(n-1)} + \overset{Wra}{(n-1)} + \overset{\theta}{(n-1)} = \overset{\text{Total}}{3(n-1)}$$

In the closure of each polygon for balance the number of independent variables which must be left is four. Hence for the $n$ masses the maximum number of independent variables which can be arbitrarily fixed is—

$$[3(n-1)-4]=(3n-7)$$

In accordance with this line of reasoning it should be noted that the choice of two masses determines only one independent $Wr$ variable, of three masses two independent $Wr$ variables, and so on.

Again, if the reference plane is not chosen coincident with the plane of one of the cranks, the fixing of $n$ distances corresponds to the fixing of $(n-1)$ independent couple arm variables. In general the reference plane is made to coincide with the plane of one of the cranks and then one of the $(n-1)$ variables is zero.

Instead of expressing the $Wr$ values in terms of the actual weights of the masses at varying radii, these can be reduced to equivalent masses at some common radius $r$. For instance, $W_1r_1=W_1'r$ or $W_1'=\frac{W_1r_1}{r}$, and so on.

In general, however, the strokes and hence $r_1$, $r_2$, $r_3$, etc., are equal to $r$, so that the actual masses can be taken in such cases. As only the proportional and not absolute values of $Wr$ and $Wra$ need be used for the vector polygons, the common radius $r$ may be taken as unity, and for calculation the data can be tabulated as below, the unit value being used in the $r$ column.

| Crank Number | W | $r$ | $a$ | $Wr$ | $Wra$ |
|---|---|---|---|---|---|
| | | | | | |

267. The procedure for the self-balance of a series of *reciprocating* masses may be taken as follows:

I. Add the proportion $w_r$ of each connecting rod mass to the mass of the crosshead, piston, etc.

II. Reduce each reciprocating mass to an equivalent mass rotating at a common radius of crank $r$.

III. Tabulate its W, $a$, $Wr$, and $Wra$ values as above, and treat them as data for a revolving system, first drawing the $Wra$ polygon for closure, then the $Wr$ polygon, the plane of reference being taken as the plane of one of the cranks.

If the work has been correctly carried out the vector polygons will close when another reference plane is chosen.

To illustrate the general method of selection of the arbitrarily chosen data for self-balance of such a system, consider a five-crank engine. In this case there are $n=5$ reciprocating masses, so that by the rule of Art. 266 the maximum number of independent variables that can be given is $(3n-7)=(15-7)=8$.

These may be distributed over the $Wr$, $a$, and $\theta$ values as follows. One choice might be three masses defining two $Wr$ independent variables, and leaving six variables still to be fixed; next four of

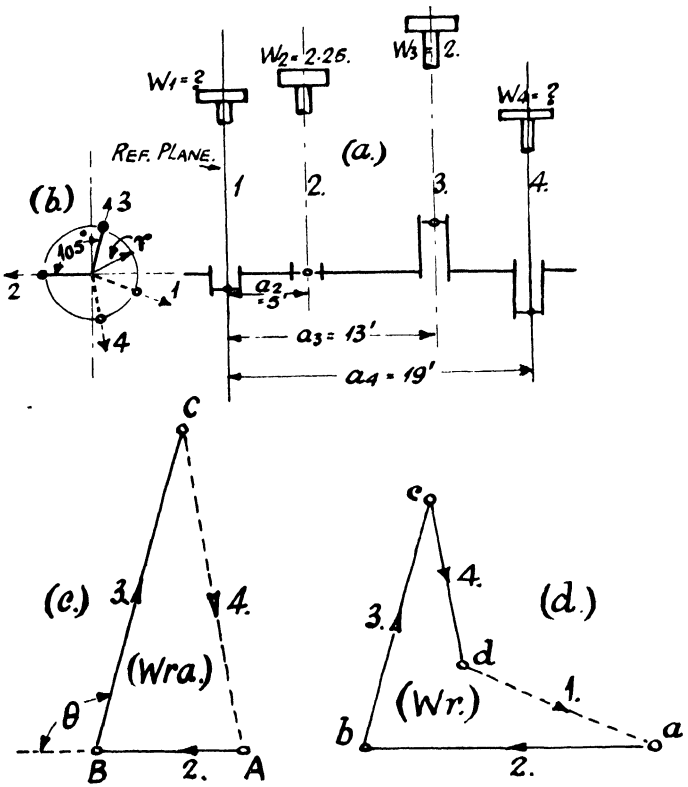



FIG. 324.

the five couple arms might be taken, leaving two angles to be fixed to make up the total.

Another choice might be four masses defining three $Wr$ variables, three couple arms, and two angles, and so on.

*Example* 2.—The cylinders 1, 2, 3, and 4, of a four-crank marine engine are arranged as shown in Fig. 324 (*a*). The weights of the reciprocating masses in tons, referred to a common crank radius and the cylinder pitches in feet, are as shown.

The angle between the cranks 2 and 3 is 105°, in (*b*). Determine

the crank angles for 1 and 4, and the weights of the corresponding reciprocating masses, so that the engine may be self-balanced for the reciprocating parts.

Here $n=4$ and the number of independent variables that can be given is—

$$(3n-7)=(12-7)=5$$

Two masses defining one $Wr$ variable are specified by $W_2$ and $W_3$. Three couple arms, by $a_2$, $a_3$, and $a_4$, and one crank angle $\theta$ are given, making the necessary total of 5. Let the plane of crank 1 be chosen as reference plane, and take the line of crank 2 as the reference line for the angles. Tabulate the data as shown below, referring all masses to a unit radius.

| No. of Crank | W ton | $r$ ft. | $a$ ft. | $Wra$ | $Wr$ | $\theta°$ |
|---|---|---|---|---|---|---|
| | | | | | (closure) | |
| 1 | (1·6) | 1 | 0 | 0 | 1·6 | 203 |
| 2 | 2·25 | 1 | 5 | 11·25 | 2·25 | 0 |
| 3 | 2·0 | 1 | 13 | 26·0 | 2·0 | 105 |
| | | | | (closure) | | |
| 4 | (1·34) | 1 | 19 | 25·5 | (1·34) | 260 |

From the given data the proportional $Wra$ values for 2 and 3 and the angle $\theta$ between 2 and 3 are known. In (*c*) draw AB=11·25 for crank 2, parallel to crank 2, then draw BC=26, at 105° to AB. The closing line CA gives the $Wra$ value for the mass at crank 4. This, on the same scale as AB and BC, measures 25·5 and is inserted under "closure" in the $Wra$ column. Since $a_4=19$, the proportional $Wr$ value for crank 4 is $\frac{25{\cdot}5}{19}=1{\cdot}34$, and this is inserted in brackets in the $Wr$ column.

Next draw the proportional $Wr$ polygon with $ab=2{\cdot}25$ parallel to AB, $bc=2$ parallel to BC, $cd=1{\cdot}34$ parallel to CA. The closing line $da=1{\cdot}6$ gives the corresponding $Wr$ value for crank 1, and this is inserted under "closure" in the $Wr$ column. The crank radials are then obtained in (*b*) by drawing the dotted crank line 4 parallel to CA in the couple polygon (*c*) and the dotted crank line 1 parallel to *da* in the force polygon (*d*). The corresponding angles, reading clockwise from 2, are 203 and 260, and these figures are entered in the angle column. Since all the masses are referred to unit radius, the weight of the mass at crank 4 is 1·34 ton and at crank 1, 1·6 ton. These are put in brackets to indicate derived values. The reciprocating masses should thus be $W_1=1{\cdot}6$, $W_2=2{\cdot}25$, $W_3=2$, $W_4=1{\cdot}34$ ton.

**268.** In order to obtain an approximately complete state of balance the rotating masses at the shaft have also to be balanced. As the crank webs are always symmetrically disposed relatively to the line of stroke, each arm can be extended in the form of an added

balance mass, as indicated in Fig. 325, so that the total centrifugal force will balance the centrifugal force of the rotating mass of webs, crank pin, and referred connecting rod mass, assumed to be concentrated at the crank pin. Alternatively the rotary balance can be effected by the use of only two rotating balance masses in separate planes of revolution, in the manner already discussed in Art. 262. The planes may be conveniently taken as those of two of the cranks.

*Example* 3.—In the case of the four-crank engine of Example 2, find the weights of two balance masses and their angular positions in the planes of the cranks 1 and 4 so that the shaft may be in dynamic balance, given that the rotating masses (including the

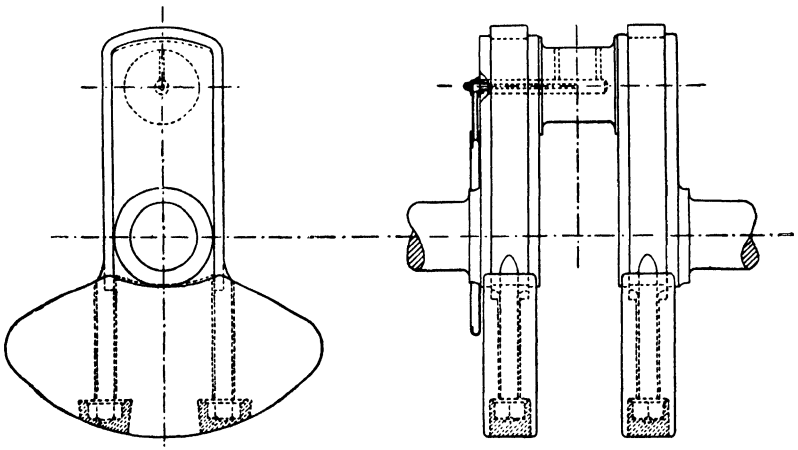


FIG. 325.

connecting rod proportions) are, 1, $1\frac{1}{2}$, $1\frac{1}{4}$, and 1 ton respectively for cranks 1, 2, 3, and 4.

Referring to Fig. 326, as before take the plane of crank 1 as reference plane and tabulate the given data, taking unit radius of crank.

| No. of Plane | W ton | *r* ft. | *a* ft. | W*ra* | W*r* |
|---|---|---|---|---|---|
| 1 | 1 | 1 | 0 | 0 | 1 |
| 2 | 1·5 | 1 | 5 | 7·5 | 1·5 |
| 3 | 1·25 | 1 | 13 | 16·25 | 1·25 |
| 4 | 1 | 1 | 19 | 19 (closure) | 1 |
| 5 | (0·168) | (1) | 19 | 3·2 | (0·168) |
| 6 | (0·125) | (1) | 0 | 0 | (0·125) |

Drawing the W*ra* polygon ABCD (*c*) with sides parallel to the

crank radials in (*b*), the closing side DA gives W*ra* for plane 5 (which is coincident with plane 4), as 3·2, and this is entered against 5, under " closure " in the W*ra* column. As the couple arm is 19 the proportional W*r* value is $\frac{3{\cdot}2}{19}=0{\cdot}168$, and this is entered in brackets in the W*r* column. Drawing the W*r* (*d*) polygon *abcdefa* with sides parallel to the respective cranks, the closing side *fa*=0·125. The radials for balance masses, 5 and 6, are drawn in the space diagram (*b*) of the crank, parallel respectively to DA and *fa* in (*c*) and (*d*).

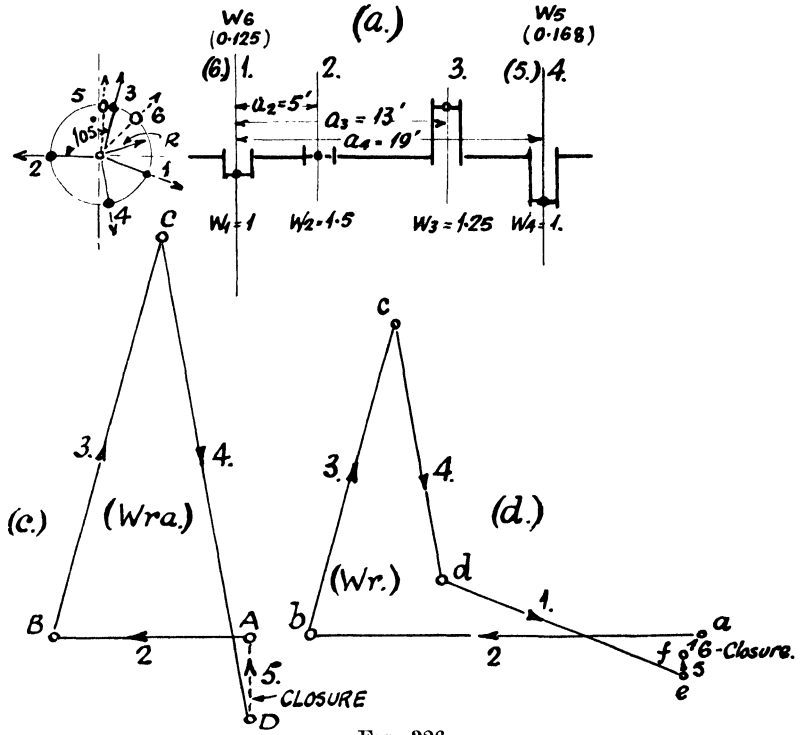



FIG. 326.

At the unit radius $W_5$=0·168 ton, $W_6$=0·125 ton. These values are entered in the W column, in brackets. The weights to be attached to the crank webs should thus be, at crank 1, 0·125 ton, and crank 4, 0·168 ton, at 1 ft. radius.

**269. Locomotive Primary Balancing.**—In the case of the locomotive the revolving masses, consisting of crank webs, crank pins, outside cranks, pins and coupling rods, can all be completely balanced, and no trouble need arise from them. As in the case of the stationary or marine engine, balance extensions may be fitted at the cranks ; but the general custom is to balance them by two

balance masses placed in the planes of the driving wheels. The inertia forces due to the reciprocating masses are the principal sources of unbalance. They produce an unbalanced force in the horizontal plane containing the cylinder axes, and a couple called the "swaying" couple, in this horizontal plane, which tends to swing the engine about a vertical axis. Also, if the line of draw-bar pull does not lie in the plane through the driving wheel axes, the unbalanced force produces a "pitching" couple, tending to oscillate the engine about a horizontal axis parallel to the driving wheel axis.

In a two-cylinder engine the cranks are always set at 90°, and the use of reciprocating balance weights is impracticable. It is usual to make a compromise and obtain partial balance of the reciprocating masses by means of rotating balance masses placed in the planes of the driving wheels.

As already pointed out, the reciprocating masses may be balanced in the line of stroke by rotating masses, but the use of these balance masses introduces an unbalanced force and couple in the plane at right angles, that is, in the vertical plane. The unbalanced force, which is of an alternating nature, gives rise to what is termed the "hammer blow" on the rail, which may appreciably affect the tractive effort, and a couple called the "rocking couple" tending to rock the engine about an axis parallel to the axis of the train. This rocking condition may become dangerous at a high speed.

**270.** If the reciprocating masses were fully balanced by rotating masses there would be no unbalanced force and no unbalanced swaying couple. The introduction of the full balance rotating masses, however, produces an excessive variation of the wheel pressure on the rail. This, in an extreme case, may be sufficient to double the rail pressure at one instant and reduce it to zero in another instant. Such conditions of running are obviously injurious to tyres, permanent way, and to the bridges, which are subjected to what is termed a "hammer blow." These conditions are also made worse by the accompanying rocking couple.

On the other hand, if the reciprocating parts are not balanced, there is the unbalanced force in the horizontal plane and the accompanying swaying couple, which may also become dangerous at high speed. A compromise has obviously to be made between these two opposing conditions of unbalance. All the revolving masses are completely balanced, and a proportion ($q$) of the reciprocating masses, which is usually two-thirds, is balanced by rotating masses placed in the planes of the driving wheels.

This system gives a state of unbalance in both horizontal and vertical planes, but the unbalanced forces and couples are reduced to more reasonable dimensions.

The problem can be treated analytically for conditions of primary balance as follows, for the usual form of two-crank engine with cranks at 90°.

**271. Unbalanced Forces and Couples due to Reciprocating Parts of a Locomotive.**—The end elevation, side elevation, and plan of the crank shaft and driving wheels of an inside-cylinder engine with cranks at 90° are shown in Fig. 327, L denoting the left and R the right-hand crank. W is the weight of the reciprocating mass at each cylinder, $d$ the pitch of the cylinders and $a$ half the pitch. The radius of each crank is $r$.

Let a reference plane be chosen midway between the cylinders, as indicated in ($a$) and ($c$). The proportional force on each crank pin, if W reciprocating mass is transferred to the crank and rotated with it, is W$r$. The reference plane turned into the plane of the paper is shown at ($d$), with the force vector for the left crank drawn horizontal and parallel to the crank, and the equal vector for the right crank

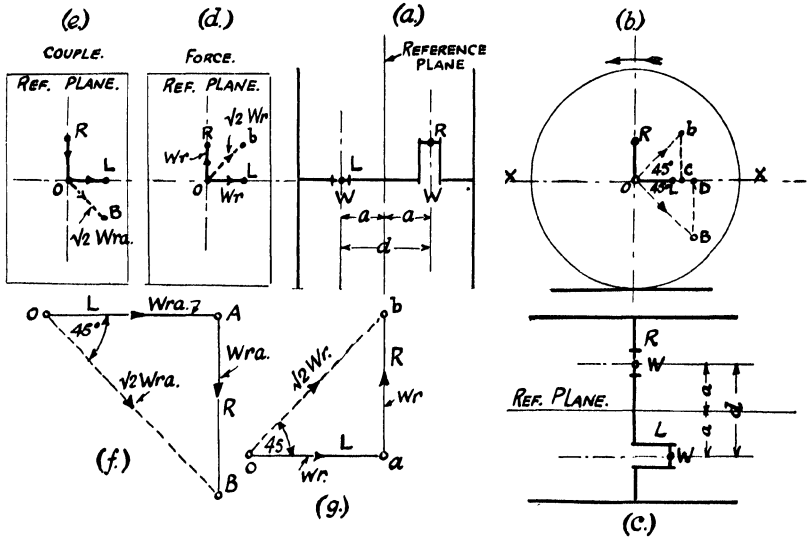



FIG. 327.

drawn vertically. The corresponding proportional vector polygon (W$r$) is shown at ($g$) and the side, $ob=\sqrt{2}$W$r$, is the resultant force referred to the reference plane. This is inclined at 45° to each crank. In ($e$) the couple vectors for W$ra$ at each crank are drawn according to the usual convention, with sides parallel to the sides of the force polygon, but since crank R is on the opposite side of the reference plane in ($a$) and ($c$), the arrow points to the centre of the shaft $o$.

The corresponding couple or W$ra$ polygon is drawn at ($f$) and the resultant couple is OB$=\sqrt{2}$W$ra$, the vector being inclined at 45° to each crank arm.

These resultant W$r$ and W$ra$ force and couple vectors have been drawn in the end elevation ($b$). For the position of the cranks shown, with the left-hand crank on the outer dead centre and the

right-hand crank leading, the unbalanced inertia force in the plane containing the cylinder axes is the horizontal projection $oC$ of $ob$, and the unbalanced swaying couple by the projection $oD$ of $oB$, on the line of stroke XX.

As these vectors rotate with the shaft and since each is inclined at 45° to the cranks, it will be obvious from Fig. 328 that when the crank makes an angle $\theta$ with the line of stroke XX the resultant unbalanced force in this direction is $oc=\sqrt{2}Wr\,.\cos(\theta+45)\frac{\omega^2}{g}$. If the line of the draw-bar pull is $t$ distance below the line of wheel centre XX for the position shown, the pitching couple will be $oc\times t$ or $C_p=\sqrt{2}Wr\frac{\omega^2}{g}\,.\cos(\theta+45)t$. The swaying couple will be $oD$ or $C_s=\sqrt{2}Wr\frac{\omega^2}{g}\,.\cos(\theta-45)a$. It will be obvious that the maximum value of the unbalanced force occurs when the force vector $ob$ coincides with XX, or when $\theta=-45$ and $+135$, while the maximum value of the swaying couple occurs when OB is coincident with XX, or when $\theta=+45$ or 225.

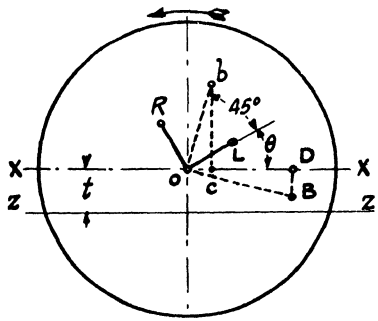


FIG. 328.

*Example* 4.—An inside-cylinder engine has driving wheels 6 ft. diameter and travels at 50 miles/hr. The stroke is 24 in., the pitch of the cylinders is 30 in. and the cranks are at 90°. The reciprocating mass per cylinder is 300 lb. The line of draw-bar pull is 1 ft. below the wheel axle. Find the maximum values of the alternating force, and the swaying and pitching couples, when none of the reciprocating mass is balanced.

Here the unbalanced force is $F=\frac{\sqrt{2}W}{g}\omega^2 r\,.\cos(\theta+45)$

$$N=\frac{5280\times S}{60\times\pi D}$$

$$=\frac{5280\times 50}{60\times\pi\times 6}=234 \text{ rev./min.}$$

$$\omega=\frac{2\pi N}{60}$$

$$=\frac{2\pi\times 234}{60}=24{\cdot}5 \text{ rad./sec.}$$

$W=300$
$r=1$
$(\theta+45)=0°$
$S=50$ mile/hr.
$D=6$ ft.

$$\therefore\ F_{max.}=\frac{300\times 24{\cdot}5^2\times\sqrt{2}}{32{\cdot}2}=7860 \text{ lb.}=3{\cdot}5 \text{ ton}$$

Swaying couple $C_{s\,max.}=\sqrt{2}\frac{W}{g}r\omega^2 . \cos 0\times a$ $a=1{\cdot}25$ ft.

$=7860\times 1{\cdot}25=9825$ ft.-lb.$=4{\cdot}38$ ft.-ton

Pitching couple $C_{p\,max.}=Ft=7860\times 1=7860$ ft.-lb. $t=1$ ft.

$=3{\cdot}5$ ft.-ton

**272. Partial Balancing of Reciprocating Parts of Locomotive.**—In order to carry out the calculation for the complete balance of rotating masses and partial balance of reciprocating masses it is not necessary to divide the process. The proportion of the reciprocating mass which is to be balanced in the line of stroke is added to the rotating masses to be balanced, and the revolving balance masses are found for the combination and located in the planes of the driving wheels.

If the weight of revolving mass to be balanced per crank, referred to the crank radius, is $W_c$ and the reciprocating mass is $W_r$, and if the proportion $q$ of the latter is to be balanced, then the equivalent rotating mass at the crank radius is $W=(W_c+qW_r)$. This, treated as a revolving mass at each crank, gives, as before, two equal force vectors at right angles. Referring to Fig. 329, which again shows the end elevation of the crank axle, cranks, and wheel disc, let the plane of reference coincide with the plane of the right-hand wheel, so that the couple arm $a_4$ of the right-hand balance mass is zero and the arms for the crank masses and left-hand balance mass are $a_3$, $a_2$, and $a_1$. Tabulate the W, $r$, $a$, W$r$ and W$ra$ values as before, draw and close the couple polygon, and determine the left-hand balance mass and its angular position in the space diagram of the cranks (elevation).

Then, as before, draw and close the force polygon, and determine the right-hand balance mass and its angular position. The process is exactly the same as that given and illustrated in Art. 263.

The balance masses in the planes of the wheels are obtained for the crank radius, but they can be referred to a much larger radius, and smaller masses can thus be fitted. Each mass is generally formed as a crescent-shaped block at the junction of the spokes with the wheel rim. The next example should make the procedure clear.

*Example* 5.—Determine the balance masses to be fitted to the driving wheels of a simple inside-cylinder locomotive at radius of 33 in., given the following data: weight of reciprocating mass per cylinder 660 lb.; weight of revolving mass per cylinder 720 lb.; distance between the planes of balance masses in wheels 58 in.; pitch of cylinders 24 in.; stroke 26 in. All the revolving masses and two-thirds of the reciprocating masses are to be balanced.

Referring to the skeleton outline of the crank shaft and wheel discs in Fig. 329 (*a*) and (*b*), choose the plane of the right-hand wheel as reference plane, then the couple arms are $a_3=1{\cdot}415$, $a_2=3{\cdot}42$, $a_1=4{\cdot}84$ ft.

$W_c$=720 lb., $W_r$=660 lb., so that the equivalent revolving mass at each crank is $(W_c+\frac{2}{3}W_r)$=(720+440)=1160 lb. Tabulate the data as below, referring the masses to a unit radius, and draw the

| No. of Plane | W lb. | $r$ | $a$ ft. | W$ra$ | W$r$ |
|---|---|---|---|---|---|
| | | | | (closure) | |
| 1. L. balance mass . . . | (888) | 1 | 4·84 | 4297 | (888) |
| 2. L.H. crank . . . . | 1160 | 1 | 3·42 | 3967·2 | 1160 |
| 3. R.H. crank . . . . | 1160 | 1 | 1·415 | 1641·4 | 1160 |
| | | | | | (closure) |
| 4. R. balance mass . . . | (888) | 1 | 0 | 0 | (888) |

couple polygon as shown in (*c*). The closing side gives W$ra$=4297, inclined at 22° 36′ to the line of the left-hand crank.

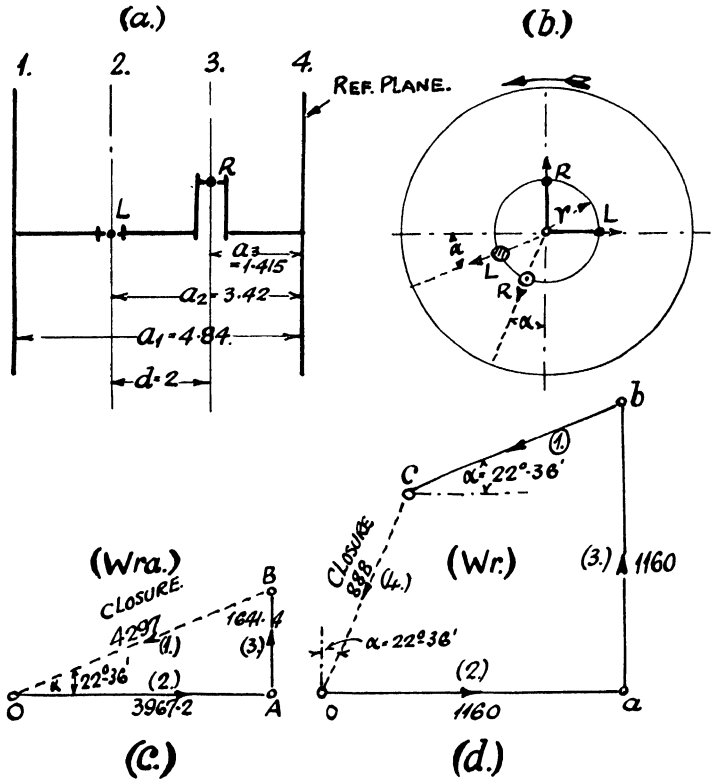



FIG. 329.

Dividing by $a_1$=4·84, the W$r$ value for plane 1 is 888, referred to $r$=1, so that the weight of the balance mass at the left-hand wheel is 888 lb. at 13 in. radius, hence the weight of the mass required at

33 in. radius is $888 \times 1\frac{3}{3} = 350$ lb. It will be obvious, from the symmetrical distribution of cranks and masses, that the balance mass in the right wheel should have the same value. It is hardly necessary in this case to draw the force polygon except to check the accuracy of the work. This is shown at (*d*) and the closing line gives the W*r* value of 888, thus checking the drawing. The angle between it and the line of the right-hand crank is again 22° 36′.

Hence if a balance mass weighing 350 lb. is fitted opposite each crank in the corresponding driving wheel at 33 in. radius and on the radial making an angle of 22° 36′ with the crank, the whole of the revolving masses and two-thirds of the reciprocating masses will be balanced.

**273. Calculation of Unbalanced Forces and Couples with Partial Balancing of Reciprocating Parts of a Locomotive.**—In the case of partial balance of reciprocating masses, and the complete balance of the rotating masses of the locomotive, the unbalanced inertia force in the horizontal plane is given by—

$$F_h = \sqrt{2}(1-q)\frac{W_r}{g}\omega^2 r \cdot \cos(\theta+45) \quad . \quad . \quad . \quad (5)$$

where $q$ is the proportion of the reciprocating mass $W_r$ which is balanced. The maximum value occurs when $\theta = -45°$ and $\cos(\theta+45) = 1$ (Fig. 328), so that the maximum unbalanced horizontal force is given by—

$$F_{h\,\max.} = \sqrt{2}(1-q)\frac{W_r}{g}\omega^2 r \quad . \quad . \quad . \quad . \quad . \quad (6)$$

The swaying couple is—

$$C_s = \sqrt{2}(1-q)\frac{W_r}{g}\omega^2 r \cdot \cos(\theta-45)a \quad . \quad . \quad . \quad (7)$$

This is a maximum for $\theta = 45$ and $\cos(\theta-45) = 1$, and

$$C_{s\,\max.} = \sqrt{2}(1-q)\frac{W_r}{g}\omega^2 r a \cdot \quad . \quad . \quad . \quad . \quad . \quad (8)$$

where $a$ is half the pitch of the cylinders, or—

$$C_{s\,\max.} = \frac{(1-q)}{\sqrt{2}}\frac{W_r}{g}\omega^2 r d \quad . \quad . \quad . \quad . \quad . \quad (9)$$

where $d$ is the pitch of the cylinders.

The force introduced in the vertical plane is that due to the mass $(qW_r) = w$. The balancing mass is obtained by drawing the couple polygon, Fig. 329 (*c*), with the vectors $qW_r r a_2$ and $qW_r r a_3$, and since the angle OAB is 90°, it follows that the balance mass for the reciprocating mass alone, in each wheel, is—

$$w' = \frac{qW_r}{a_1}(\sqrt{a_2^2 + a_3^2}) \quad . \quad . \quad . \quad . \quad . \quad (10)$$

And the angle of inclination $\alpha$ of the balance mass radius to the line of each crank at ($b$) is given by—

$$\tan \alpha = \frac{a_3}{a_2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$$

The force in the vertical plane due to mass $w'$, rotating with the wheel, is the vertical component of its centrifugal force, or—

$$F_v = \frac{w'}{g} \omega^2 r \sin \beta \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (12)$$

where $\beta$ is the instantaneous angle (read counter-clockwise from the line of stroke) between the radius line of the balance mass and the line of stroke. Obviously as the mass rotates through the third and fourth quadrants the vertical force acts downward, and in the first and second quadrants it acts upward. In the first case it causes an increased pressure, and in the second a decreased pressure on the rail. The maximum + or − value is reached when the radius line of the mass is vertical, that is when $\beta = \pm 90°$, for which $\sin \beta = \pm 1$, and—

$$F_{v\,\max.} = \pm \frac{w'}{g} \omega^2 r \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (13)$$

If S = speed of the engine in miles/hr.
D = diameter of driving wheel in ft.
N = speed in rev./min.

the angular velocity of the crank shaft is given by—

$$\omega = \frac{2\pi N}{60} = \frac{2\pi}{60} \times \frac{5280S}{60\pi D} = \frac{2 \cdot 93S}{D} \text{ radians/sec.} \quad . \quad . \quad (14)$$

Substituting in (12)—

$$F_v = \pm \frac{w'}{g} \left( \frac{2 \cdot 93S}{D} \right)^2 r \sin \beta \quad . \quad . \quad . \quad . \quad . \quad (15)$$

and $$F_{v\,\max.} = \pm \frac{w'}{g} \left( \frac{2 \cdot 93S}{D} \right)^2 r \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (16)$$

when the balance mass is in the line of stroke $\beta = 0$ or 180, and $\sin \beta = 0$.

If $W_w$ is the constant load on the wheel transmitted to the rail, the effective load or pressure between rail and wheel is given by—

$$(W_w \pm F_v) = W_e \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (17)$$

for any angular position of the balance mass for which $F_v$ is known. The variation of rail pressure per revolution can be shown by a curve of $W_e$ plotted to a base of angle $\beta$ between 0 and 360.

The rocking couple due to the balanced mass ($qW_r$) is

$$C_r = \sqrt{2} q W_r \frac{\omega^2}{g} r \sin (\theta - 45) a \quad . \quad . \quad . \quad . \quad (18)$$

where $a$ is half the pitch of the cylinders and $\theta$ is read counter-clockwise from the line of centres or the left-hand crank, Fig. 328.

When the rotating couple vector $\sqrt{2}qW_r\frac{\omega^2}{g}ra$ reaches the vertical position the angle $(\theta-45)=90$ and $\sin 90=1$, so that the maximum value of the rocking couple is given by—

$$C_{r\ max.}=\sqrt{2}qW_r\frac{\omega^2}{g}ra=\frac{qW_r}{\sqrt{2}}.\frac{\omega^2}{g}rd \quad . \quad . \quad . \quad (19)$$

*Example* 6.—In the case of the inside-cylinder locomotive (Example 5) with two-thirds of the reciprocating masses balanced, find the maximum unbalanced horizontal force and maximum swaying couple when the speed is 50 miles/hour.

$W_r=660$, $q=\frac{2}{3}$, $\omega=\left(\frac{2\cdot93S}{D}\right)$ $\quad S=50$, $D=6$, $r=1\frac{3}{12}$

$$=\frac{2\cdot93\times50}{6}=24\cdot4 \text{ rad./sec.}$$

$$F_{h\ max.}=\sqrt{2}(1-q)\frac{W_r}{g}\omega^2r$$

$$=\frac{1\cdot414\times660\times13\times24\cdot4^2}{3\times12\times32\cdot2}=6260 \text{ lb.}=2\cdot8 \text{ ton}$$

Swaying couple $C_{s\ max.}=(1-q)\sqrt{2}\frac{W_r}{g}\omega^2ra$ $\quad a=1$

$$=6260 \text{ ft.-lb.}=2\cdot8 \text{ ft.-ton}$$

*Example* 7.—In the case of the engine (Example 6) find the maximum value of the unbalanced vertical force on the rail, and the maximum value of the rocking couple. Also find the speed at which the wheel will leave the rail if the dead load on the wheel is 8 tons and the wheel diameter is 6 ft.

$a_1=4\cdot84$, $a_2=3\cdot42$, $a_3=1\cdot415$, $W_r=660$, $r=1\frac{3}{12}$, $q=\frac{2}{3}$

$$w'=\frac{qW_r}{a_1}\sqrt{{a_2}^2+{a_3}^2}$$

$$=\frac{2\times660}{3\times4\cdot84}\sqrt{3\cdot42^2+1\cdot415^2}$$

$$=90\cdot9\times3\cdot7=336\cdot3 \text{ lb.}$$

$$F_{v\ max.}=\pm\frac{w'}{g}\left(\frac{2\cdot93S}{D}\right)^2r$$

$$=\frac{336\cdot3\times24\cdot4^2\times13}{32\cdot2\times12}$$

$$=6730 \text{ lb.}=3 \text{ tons}$$

Rocking couple $C_{r\ max.}=\sqrt{2}qW_r\frac{\omega^2}{g}ra$ $\quad a=1$

$$=\frac{1\cdot414\times2\times660\times24\cdot4^2\times13\times1}{3\times32\cdot2\times12}$$

$$=12500 \text{ ft.-lb.}=5\cdot6 \text{ ft.-tons}$$

This value can also be derived directly from the swaying couple since—

$$\frac{C_{r\ max.}}{C_{s\ max.}}=\frac{q}{(1-q)}=\frac{\frac{2}{3}}{\frac{1}{3}}=2 \quad \text{or} \quad C_{r\ max.}=2C_{s\ max.}$$

$$=2\times 2{\cdot}8=5{\cdot}6 \text{ ft.-tons}$$

To find the speed at which the pressure on the rail vanishes, $W_e=(W_w-F_{v\ max.})=0$, so that $F_{v\ max.}=W_w$

and $W_w=\frac{w'}{g}\left(\frac{2{\cdot}93S}{D}\right)^2 r$ $\qquad W_w=8$ tons, $D=6$ft., $w'=336{\cdot}3$

$$S^2=\frac{W_w g D^2}{(2{\cdot}93)^2 w' r}=\frac{8\times 2240\times 32{\cdot}2\times 6^2\times 12}{336{\cdot}3\times 13\times(2{\cdot}93)^2}$$

$$=6620$$

and $S=\sqrt{6620}=81{\cdot}5$ miles/hour.

**274. Variation of Rail Pressure.**—The variation of pressure on one rail is given by Equation 12. For the simultaneous variation

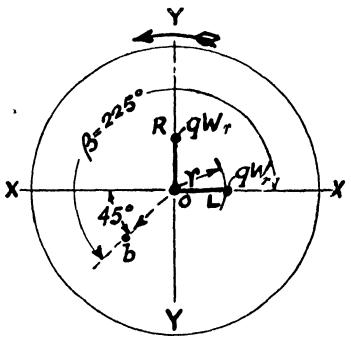



FIG. 330.

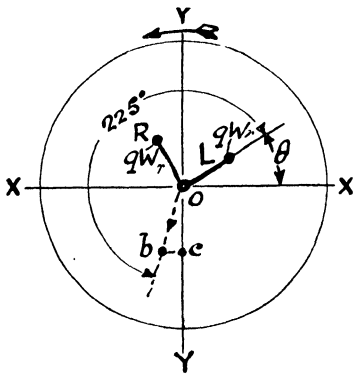



FIG. 331.

of pressure on the two rails, the resultant inertia force of the balanced mass referred to a plane midway between the cranks can be taken. The force due to the balanced mass $qW_r$ at each crank pin is given by $qW_r\frac{\omega^2}{g}r$, and the resultant is $\sqrt{2}qW_r\frac{\omega^2}{g}r$, acting at 45° to the line of stroke. The vector *ob* of this resultant is drawn in Fig. 330, and bisects the angle between the cranks. With the left crank on the dead centre, the angle measured counter-clockwise from this crank is 225°. In Fig. 331 the crank makes an angle $\theta$ with the line of centres XX, and the inclination of the rotating force vector *ob* to XX is now $\beta=(\theta+225)$. The vertical resultant force on the two rails is thus *oc*, the projection of *ob* on the axis YY, at right angles to the line of stroke, or $oc=ob\sin(\theta+225)$.

*oc* is obviously a maximum when the vector lies on YY, that is,

when $\theta=45°$ and $\beta=(\theta+225)=270°$, so that $\sin\beta=-1$. The maximum resultant pressure on the two rails is thus—

$$F_{PR}=\sqrt{2}qW_r\frac{\omega^2}{g}r \quad . \quad . \quad . \quad . \quad . \quad . \quad (20)$$

In the previous example the maximum load coming on the two rails has a total value of $1{\cdot}414\times3=4{\cdot}242$ tons.

**275. Balancing of Outside-cylinder Locomotive.**—In the case of an outside-cylinder locomotive the crank arms are formed between the spokes and project outside the plane of the wheel. The crank pin also projects outside the crank face, and thus rotates in a plane outside that of the wheel, together with the proportion of the mass of the connecting rod.

The general procedure for balance is the same as given in Art. 262 and need not be repeated. The reference plane is taken as the plane of one of the wheels, say the right-hand wheel. With this choice the crank mass and the equivalent crank pin rotating mass, at the wheel, lie on the outside of the reference plane, and the proportional $Wra$ vectors have the sense negative to those of the other masses lying to the other side of this plane (see Art. 263). The cranks are always set at 90°. The equivalent rotating mass at each crank pin is equal to the sum of the crank pin mass, the proportion of the connecting rod mass, and the proportion of the reciprocating mass which is to be balanced.

**276. Primary Balancing of Coupled Locomotives.**—In the foregoing discussion, the primary balancing of the reciprocating parts and the rotating parts of the locomotive has been considered.

The driving wheels are always connected by outside rods, either four or six wheels being coupled. Each point of a coupling rod moves in a circle with radius equal to the outside crank radius.

To take account of the effect of the coupling rod, therefore, the proportion of the weight of the rod taken by each crank pin is treated as a revolving mass at that crank pin, and simply included with the rotating masses to be completely balanced. As a rule the radius of the outside crank taking the coupling rod is less than that of the inside cranks and the rotating mass at this radius has to be reduced to the equivalent mass at the larger inner crank radius, and used in the balancing calculation.

A final example of the primary balancing of a four-coupled inside-cylinder locomotive should make the whole precedure clear.

*Example* 8.—The following data refer to a four-coupled inside-cylinder locomotive with cranks at 90°:

Stroke 24 in.; right-hand crank leading; outside cranks 11 in. radius and placed opposite to the corresponding inside cranks; cylinder pitch 2 ft.; distance between planes of wheels 4·94 ft.; distance between wheel cranks 5·166 ft.; distance between coupling rod centres (outside crank pins) 6·27 ft.

Reciprocating mass per cylinder, 642 lb. at 12 in.

Inside revolving mass per cylinder, 723 lb. at 12 in.

Mass of each coupling rod, 224·5 lb. at 11 in., or 206 lb. at 12 in.

Mass of each wheel crank, 117·6 lb. at 11 in., or 108 lb. at 12 in.

Mass of each outside crank pin (with washer), 38·2 lb. at 11 in., or 35 lb. at 12 in., and hence mass at each outside crank is 138 lb.

All revolving and two-thirds of the reciprocating masses to be balanced. The mass of each coupling rod is to be divided equally between the wheels.

The skeleton drawing of the crank shaft and wheel is shown in

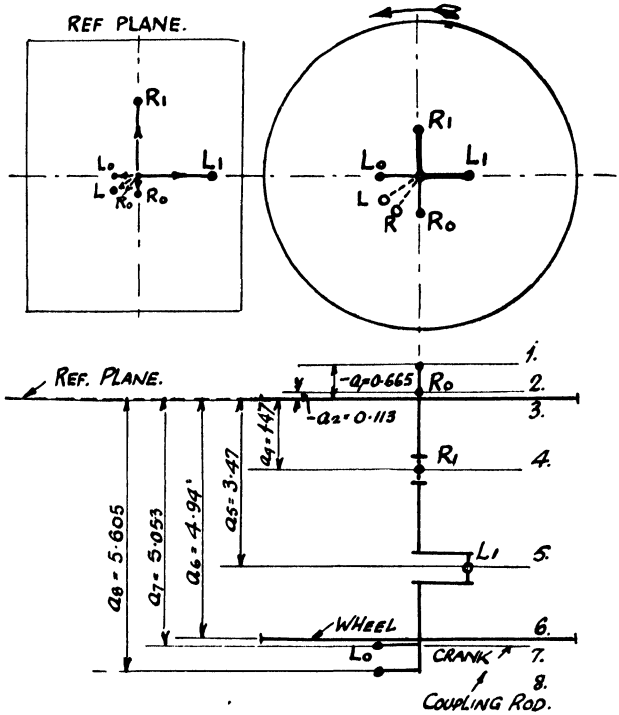



FIG. 332.

elevation and plan in Fig. 332. The plane of the right-hand wheel is taken as the reference plane. The planes are numbered 1 to 8 from the plane of the right-hand coupling rod and outer crank pin; and the couple arms are correspondingly numbered. The arm for plane 3, the reference plane, is $a_3=0$.

The referred mass at each inside crank, including two-thirds of the reciprocating mass, is $(723+\frac{2}{3}\times 642)=723+428=1151$ lb.

The data for graphical calculation are set out in the table below, the masses being referred to unit crank radius.

| No. of Plane | W lb. | $r$ | $a$ ft. | W$ra$ | W$r$ |
|---|---|---|---|---|---|
| 1. R. coupling rod . . . | 138 | 1 | −0·665 | 91·8 | −138 |
| 2. $R_0$, crank . . . . . | 108 | 1 | −0·113 | 12·2 | −108 |
| 3. R, wheel . . . . . | (650) | 1 | 0 | 0 | (closure) (650) |
| 4. $R_1$, crank . . . . . | 1151 | 1 | 1·47 | 1692 | 1151 |
| 5. $L_1$, crank . . . . . | 1151 | 1 | 3·47 | 3994 (closure) | 1151 |
| 6. L, wheel . . . . . | (650) | 1 | 4·94 | 3100 | (650) |
| 7. $L_0$, crank . . . . . | 108 | 1 | 5·053 | −546 | −108 |
| 8. L. coupling rod . . . | 138 | 1 | 5·605 | −774 | −138 |

As the couple arms for 1 and 2 are measured negatively to the others, the W$ra$ values are negative and the sign convention is

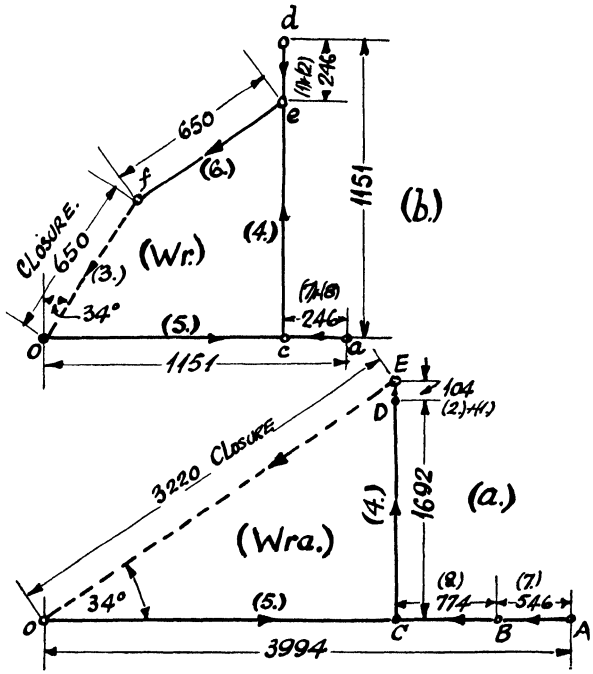



FIG. 333.

observed in drawing the W$ra$ polygon, shown in Fig. 333 ($a$). Here OA=3994 drawn parallel to the direction of the centrifugal force for the left-hand inside crank ($L_1$), as the centrifugal force on the outer ($L_0$) crank mass and the left-hand crank pin mass has a reversed sense, as shown in the reference plane, Fig. 332, AB=546 and BC=774 are drawn negatively; CD=1692 is drawn vertically

upward parallel to the direction of the centrifugal force on the inside right-hand crank ($R_1$). The directions of the forces on the right-hand outer crank ($R_0$) and the outer crank pin are vertically downward, so that, by the sign convention, the $Wra$ value DE=(91·8 +12·2)=104 is drawn vertically upward. The closure EO=3220 is inclined at 34° to the direction of the left-hand crank.

The left wheel balance mass at unit crank radius is thus $EO/a_6 = \frac{3220}{4 \cdot 94} = 650$ lb. From the symmetrical disposition of cranks and masses this ought also to be the balance mass for the right wheel.

This is checked by the force polygon, Fig. 333 (*b*): *oa*=1151; *ac*=(108+138)=246 in negative direction; *cd*=1151, *de*=246 in negative direction; *ef*=650 drawn parallel to EO of Fig. 333 (*a*). The closure *fo*=650, at 34° to the vertical, checks the accuracy of the graphical work.

These balance masses can now be referred to any convenient radius greater than the crank radius of 12 in.

The trailing wheel which is coupled to this leading driver is dealt with in the same manner for the balance of the rotating masses which, referred to 12 in. crank radius, are 138 lb. at each crank pin, and 180 lb. at the wheel crank. This estimation of the balance masses in the planes of the trailing wheels is left as a further exercise for the student.

**277.** As already shown in Art. 266, it is possible to self-balance the reciprocating masses of a four-cylinder engine (when simple harmonic motion is assumed) by a suitable choice of piston masses and disposition of cranks. The four-cylinder type of locomotive (usually compound) could be balanced in this way, with the result that both swaying and rocking couples and hammer blow would be eliminated. Self-balance, however, would involve variation of crank angles and necessitate a separate valve gear for each cylinder. This method is not employed. The arrangement is two sets of cranks at 90°, the cranks of each pair being set at 180° to each other. With equal reciprocating masses the horizontal unbalanced inertia force is zero, but there still remains the swaying couple. If this is left unbalanced no rotating balance masses have to be added and there is no resulting rocking couple and no hammer blow on the rail, and the full dead weight or static load on each driving wheel is available for adhesion.

Whether no balancing or a certain amount of partial balancing of reciprocating masses should be adopted will depend on the particular type of engine and the length and total weight, which condition the moment of inertia of the engine about a vertical axis. All the rotating masses should, however, be balanced.

**278. Secondary Balancing.**—Throughout the foregoing discussion it is assumed that the connecting rod is of such length relatively to the crank length that the condition of simple harmonic motion of the reciprocating parts is practically ensured. When the ratio $n$ of

connecting rod to crank is small this condition is not fulfilled, with the result that in addition to the primary inertia forces, or forces of the first order, there are superposed forces of the second and higher orders, arising from the irregular motion of the reciprocating masses.

In general, however, it is not necessary to consider any higher order than the second. This part of the subject of balancing is too involved to admit of discussion in this text, and the student is referred to the special texts dealing with balancing * for complete treatment and applications.

In this section only a brief note on the nature of the simpler case of secondary balancing is given.

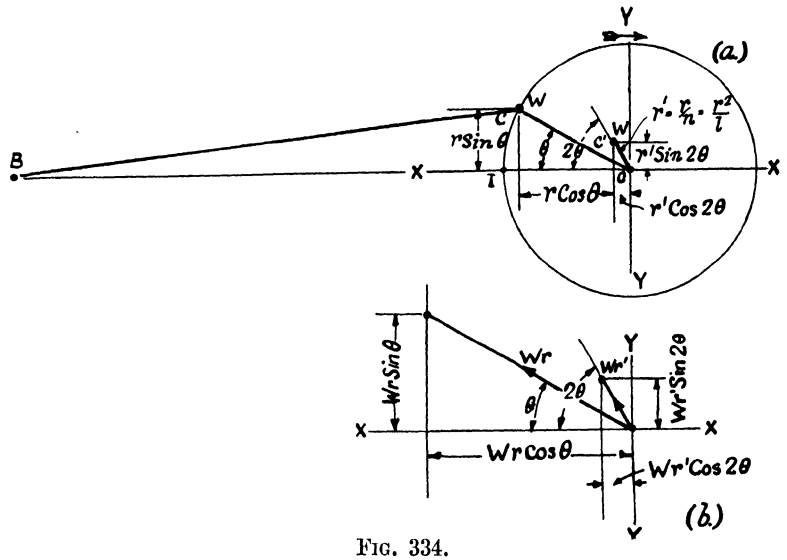



FIG. 334.

In Art. 238 it has been shown that, to a first approximation, the acceleration of the reciprocating mass attached to the connecting rod is given by—

$$\frac{dv}{dt}=f=\omega^2 r\left(\cos\theta+\frac{1}{n}\cos 2\theta\right) \quad . \quad . \quad . \quad (21)$$

With a reciprocating mass weighing W lb. the inertia force is—

$$F=\frac{W}{g}f=\frac{W}{g}\omega^2 r\left(\cos\theta+\frac{1}{n}\cos 2\theta\right) \quad . \quad . \quad (22)$$

Referring to Fig. 334 (*a*), which shows the position of the crank

* See Dalby's *Balancing of Engines;* Sharpe's *Balancing of Engines; A Treatise on Engine Balance using Exponentials,* by P. Cormac; *Engineering,* 9 December, 1921.

$oC$ after it has described an angle $\theta$ from the inner dead centre I, since $\omega$ is taken as a constant, if $t$ is the time taken by the crank to describe the angle $\theta$, then by definition $\omega=\frac{\theta}{t}$ and $\theta=\omega t$.

Hence Equation 22 can be written—

$$F=\frac{W}{g}\omega^2 r\left[\cos\omega t+\frac{1}{n}\cos(2\omega)t\right] \quad . \quad . \quad . \quad . \quad . \quad (23)$$

but $\frac{1}{n}=\frac{r}{l}$, so that

$$F=\frac{W}{g}\omega^2 r\cos\omega t+\frac{W}{g}\omega^2\frac{r^2}{l}\cos(2\omega)t$$

If the second term is multiplied and divided by 4 this becomes—

$$F=\frac{W}{g}\omega^2 r\cos\omega t+\frac{W}{g}(2\omega)^2\frac{r^2}{4l}\cos(2\omega)t \quad . \quad . \quad (24)$$

It will be seen here that the first term of (24) is the primary force dealt with throughout the previous discussion, or the component in the line of stroke of an equivalent mass weighing W lb. attached to and rotating with the crank pin C.

The second term is obviously the force which would be produced by the mass if attached to another crank having a radius $\left(\frac{r^2}{4l}\right)$ and rotating with an angular velocity $(2\omega)$, or at twice the speed of the engine crank. This imaginary crank $oC'$ has twice the angle of displacement $\theta$ of the real crank $oC$.

Equation 22 may be written in the form—

$$F=\frac{W}{g}\omega^2 r\cos\theta+\frac{W}{g}\omega^2 r'\cos 2\theta \quad . \quad . \quad . \quad . \quad (25)$$

where $r'=\frac{r}{n}=\frac{r^2}{l}$. As before, $\frac{\omega^2}{g}$ is constant, and the mass moment values $Wr$ and $Wr'$ only need be used in the calculation.

The component of the primary inertia force in the line of stroke XX is $Wr\cos\theta$, and of the secondary $Wr'\cos 2\theta$. The component, at right angles, for the primary, in direction YY, is $Wr\sin\theta$ and of the secondary $Wr'\sin 2\theta$.

For complete primary force and couple balance the sum of the $Wr$ and $Wra$ components in direction XX, in the chosen reference plane, must be zero; and the sum of the components in the direction YY must be zero.

Similarly, for complete secondary balance, the sum of the $Wr'$ and $Wr'a$ components in the direction XX must be zero; and the sum in the direction YY must be zero.

Thus,

| Primary | | Secondary | |
|---|---|---|---|
| $(a)$ | $\begin{cases}\Sigma Wr\cos\theta=0\\ \Sigma Wr\sin\theta=0\end{cases}$ | $(c)$ | $\begin{cases}\Sigma Wr'\cos 2\theta=0\\ \Sigma Wr'\sin 2\theta=0\end{cases}$ |
| $(b)$ | $\begin{cases}\Sigma Wr\cos\theta\,.\,a=0\\ \Sigma Wr\sin\theta\,.\,a=0\end{cases}$ | $(d)$ | $\begin{cases}\Sigma Wr'\cos 2\theta\,.\,a=0\\ \Sigma Wr'\sin 2\theta\,.\,a=0\end{cases}$ |

There are thus eight equations in four groups of two each which have to be satisfied for complete primary and secondary balance.

As already stated in Art. 265, where there are $n$ reciprocating or quasi-rotating masses, the number of independent variables is given by $3(n-1)$. In the case where $n=1$, no balance can be obtained. When $n=2$, then $3(n-1)=3$, and the first group $(a)$ can be satisfied and the primary forces balanced. When $n=3$, $3(n-1)=6$ three groups $a$, $b$, and $c$ can be satisfied, and the primary forces and couples and secondary forces can be balanced. When $n=4$ the complete set $a$, $b$, $c$, and $d$ can be satisfied, and complete primary and secondary force and couple balance can be theoretically obtained. Practically, this is not possible, since two of the cranks fall in the same plane.

# CHAPTER XIV

## CORLISS AND DROP-VALVE ENGINES

**279.** MANY considerations arise in the selection of an engine suitable for a particular factory or mill.

Where only the question of power output is concerned the engine may be slow or high speed, horizontal or vertical, simple or com-

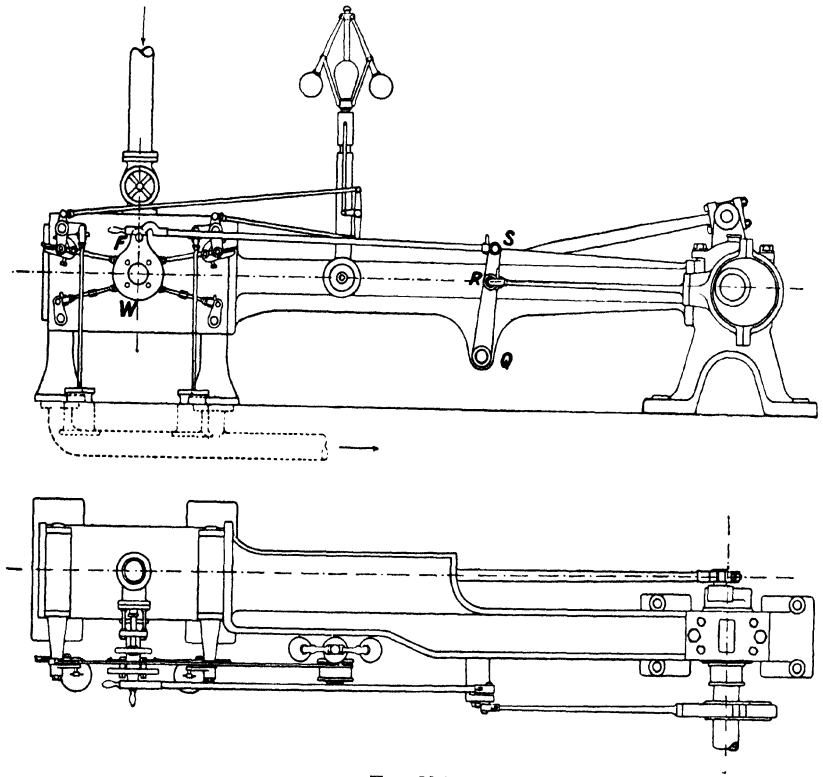



FIG. 335.

pound, non-condensing or condensing. Limitation of floor space may necessitate a vertical high-speed engine; but if there is ample floor space the slow-speed horizontal type, either Corliss or drop-

valve, may be found more suitable. The question of simple or compound will be influenced by such conditions as the power required, the character of the load fluctuation, the existing boiler pressure, the supply of condensing water available, etc. Where the load is subject to wide fluctuations a simple engine may be more flexible than a compound, and although it is less economical than the latter, its first cost is less. It is desirable in most cases to run the engine condensing. Both jet and surface condensers are in general use on small and moderate powered engines (Chapter XXIII). If a compound is to be installed to take a steady load throughout

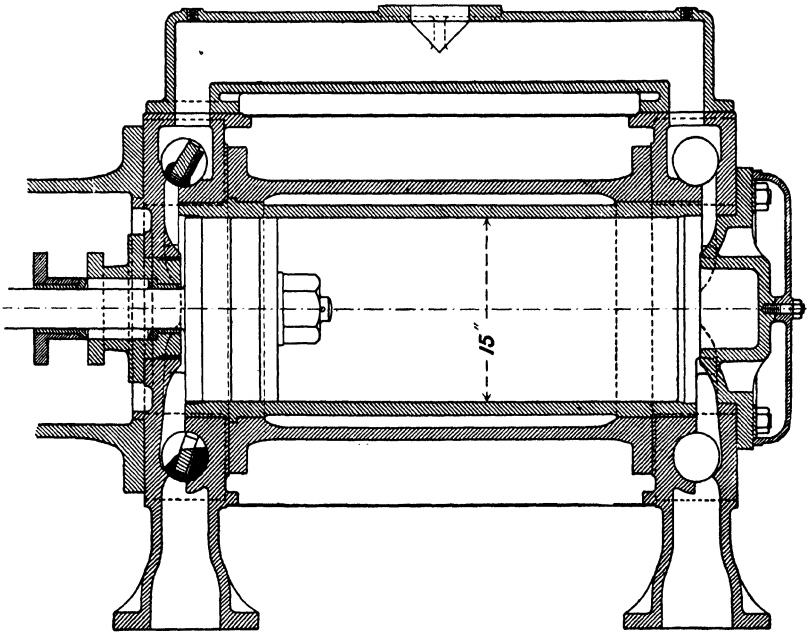



FIG. 336.

the working period it should be run condensing with high pressure superheated steam in order to obtain the most economical results.

In many industries, however, the provision of power is only one aspect of the general economic problem.

In addition to power requirements, which may be of a very variable nature, large quantities of steam are necessary for heating and manufacturing processes. In such a case the wider question of whether a back pressure or a heat extraction engine should be installed has to be considered. These types are discussed in Chapter XV. The vertical high-speed engine is briefly considered in Chapter XVI.

In this chapter only brief descriptions of a Corliss and two drop-valve engines are given, as restriction of space does not admit of fuller discussion.

The Corliss engine is that which appeared in the previous editions, and is repeated here as it is representative in its essentials of numerous mill engines of the wristplate type still in use. The first drop-valve engine also appeared in the previous editions and does not differ to any extent from the modern engine of this type. The second drop-valve engine is one of the most recent departures, installed in a woollen mill. It is a tandem compound jet-condensing

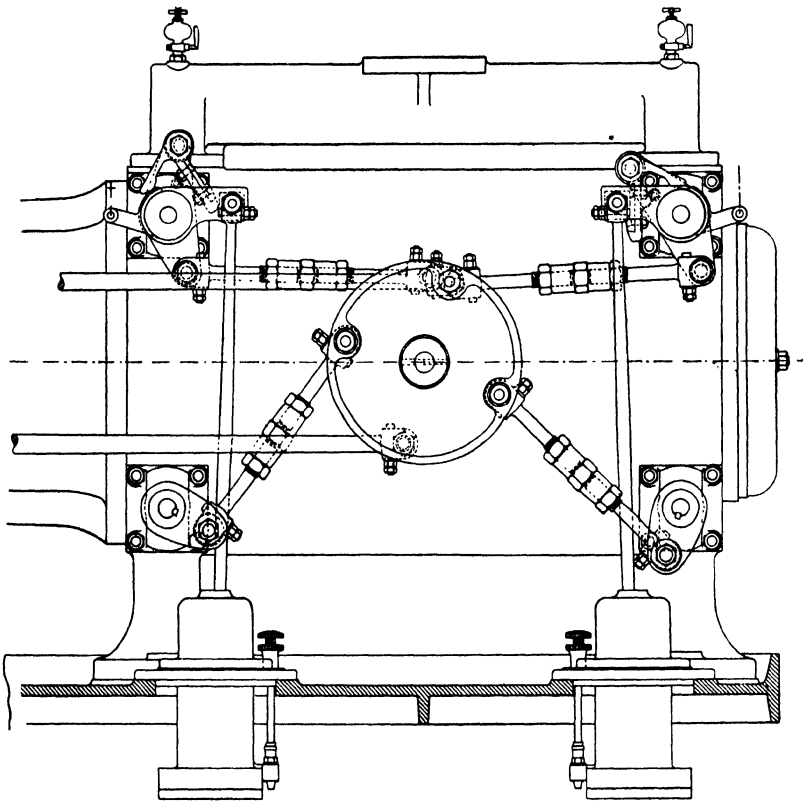


FIG. 337.

heat extraction engine, with low pressure uniflow, and drives the mill by ropes from the flywheel.

**280. Corliss Engine.**—The principal advantages claimed for this type have already been stated in Chapter IX on Valve Gears, and nothing further need be said here. A side elevation and plan of the single eccentric engine, already referred to in the discussion on the single eccentric gear, are shown in Fig. 335. The lettering corresponds to that on the skeleton arrangement of the linkage discussed in detail in Art. 191. The trip gear employed has also been illustrated and described in detail in Art. 192. The cylinder is

bolted to the end of a bayonet frame of strong construction, and supported in the standards, which also serve as exhaust passages.

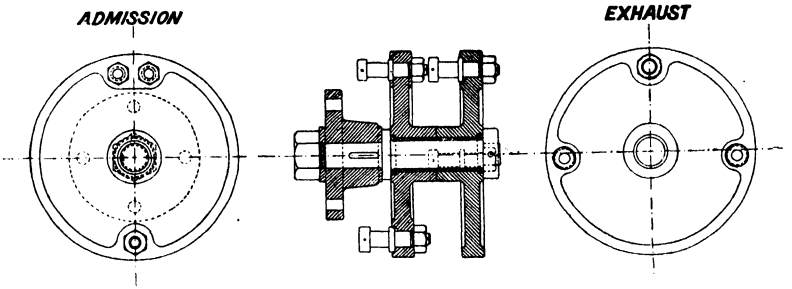



FIG. 338.

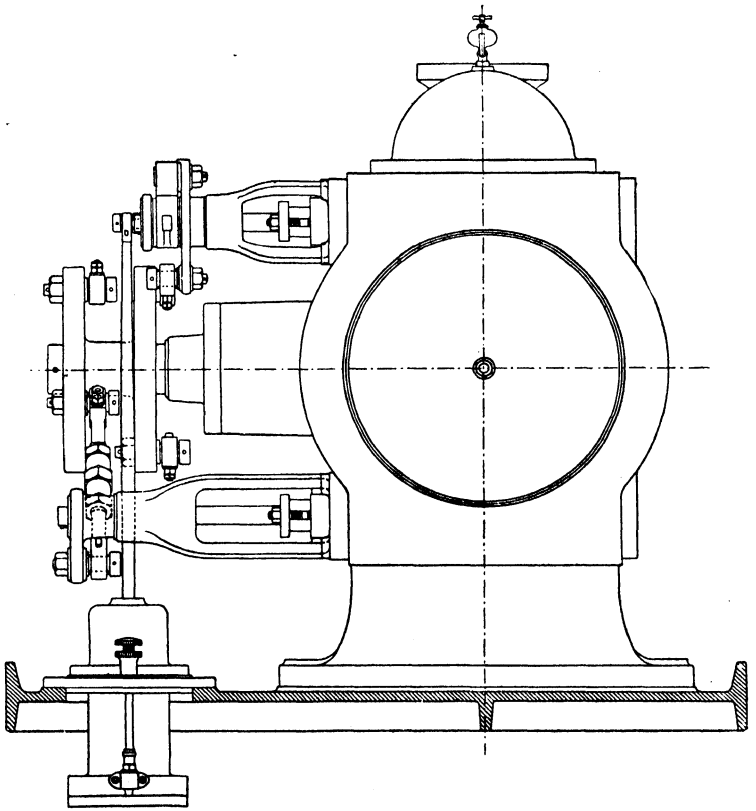


FIG. 339.

The frame has bored out guides for the crosshead, and the connecting rod works on an overhung crank on the left of the main bearing. The eccentric, rocking lever, valve gear and loaded

governor are on the right, and there is also an out-board bearing on this side and a flywheel on the shaft between it and the eccentric.

A longitudinal section through the H.P. cylinder of a cross compound double wristplate Corliss engine is shown in Fig. 336. The outside elevation is shown in Fig. 337. The exhaust valve wristplate is in front, with the valve rods coupled directly to the levers, and the horizontal driving rod is connected at the top. In this case the steam-valve rods are horizontal and connected to the inside wristplate, at the top, and the driving rod at the bottom, as indicated by the dotted lines. A section of the two wristplates mounted on the spindle and an end elevation of each plate are shown in Fig. 338. The trip gear is that illustrated and discussed in Art. 193. An end elevation of the cylinder, valve gear, and dash pot is shown in Fig. 339, and a cross-section through the valve chests and cylinder ports in Fig. 340.

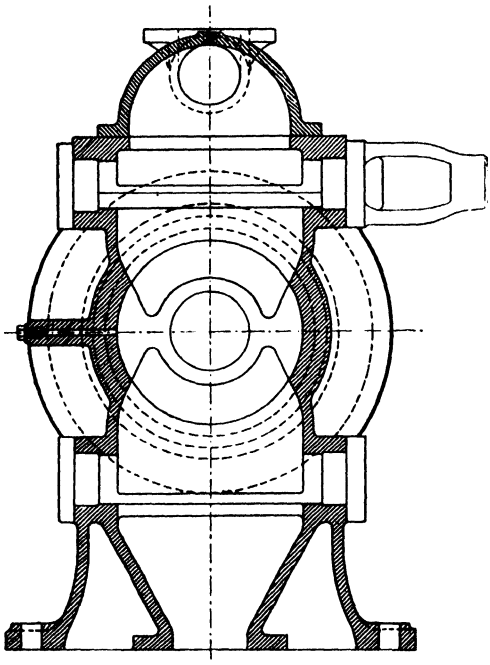


FIG. 340.

A detail of the valve spindle and bracket is shown in Fig. 341. The stuffing box arrangement is not as good as the long sleeve

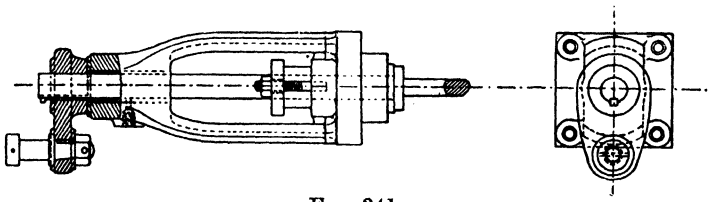


FIG. 341.

and valve spindle with labyrinth grooves, which effectively throttle the steam and prevent leakage. This arrangement reduces frictional resistance at the spindle. The dash pot details have been illustrated and described in Art. 194. A detail of the

adjustable head of the valve rod is shown in Fig. 342. This engine has cylinders 15 in.×28 in.×36 in. stroke, and develops a normal load of 340 I.H.P. at 105 rev./min. at a boiler pressure of 160 lb./in.$^2$ g. with saturated steam, and a vacuum of 26 in. It can take an overload up to 425 I.H.P.

**281. Drop-Valve Engine with Double-beat Valves.**—A side elevation half in section and a transverse section through the cylinder and valve chests of a simple horizontal trip gear engine with double-beat drop-valves are shown in Figs. 343 and 344. It is designed to run on saturated steam at 100 lb./in.$^2$ g. at 100 rev./min. The cylinder diameter is 15 in. and the stroke 36 in. As in the case of the Marshall gear described in Art. 202, the exhaust valve is operated by a cam on the lay shaft C (Fig. 344), and the trip gear by an eccentric and rod, but the details of the trip arrangement are somewhat different.

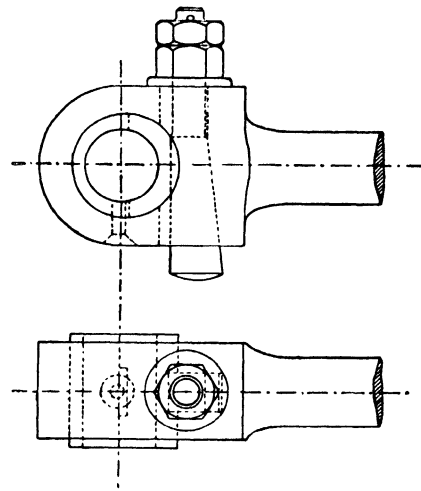


FIG. 342.

The cylinder is a cored casting of tough cast-iron, and has suitable recesses into which the steam and exhaust valve seats are forced. The steam is admitted through the stop valve (Fig. 344) on the side, at the middle of the cylinder, and passes round a belt to a passage at the top, shown in section in Fig. 343. From this passage it is supplied to the valve chests at A. When it is exhausted through the valves B it passes first into the cored chamber D and thence to the exhaust pipe. A liner E is forced into the cylinder casting. The spigot portion is $\frac{1}{16}$ in. larger at one end than at the other, to facilitate the insertion and fixture of the liner. The valves and seatings are made of a specially strong and hard mixture of cold-blast and hematite cast-iron. A section of one of the valves and its seating is shown in Fig. 344. The upper and lower beats of the valve faces are tangent to two spheres which have a common centre at the point of suspension. The effect of this arrangement is to maintain free contact between the valves and seats and minimise leakage. All castings are more or less in a state of internal stress when received from the foundry, and when subjected to heat they may twist or warp. Initial stresses may be relieved in the case of small castings by annealing them, that is, bringing them to a red heat and then letting them cool slowly in the closed

furnace. When castings cannot be treated in this way, it is necessary to examine and "let down" wearing parts which are to be a tight fit under steam pressure. With complicated valves this fitting has to be done several times before a reasonable amount of steam tightness is obtained. The final grinding, if possible, should be done when the valves and seatings are hot.

The space between the liner and cylinder barrel forms a steam

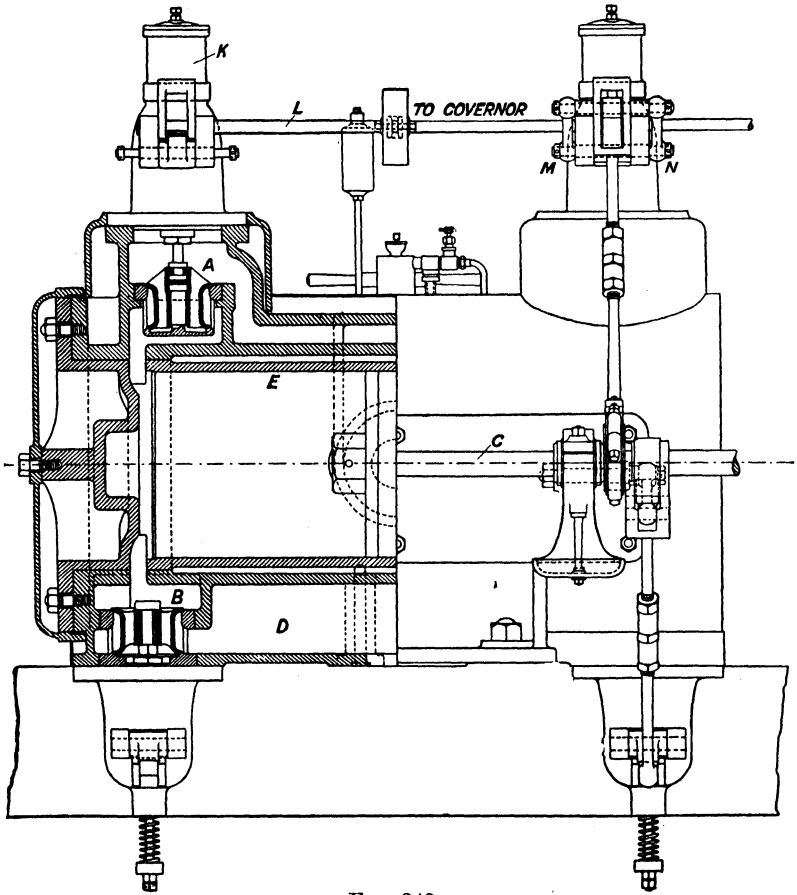



FIG. 343.

jacket, which is supplied with high-pressure steam from a small steam valve placed alongside the stop valve. The jacket condensate is drained through the passage indicated by the dotted lines, at the bottom in Fig. 343. The cylinder is automatically drained through the exhaust valves at each stroke, but to provide against the contingency of water being trapped between the piston and cover, when these are closed, spring-loaded relief valves are fitted, as shown

on the right of Fig. 344. Lubrication is effected by a sight feed lubricator F, which is connected to the steam supply passage.

The cylinder is covered with a fibrous non-conductor, such as asbestos or slag wool, and cleaded with sheet steel, which is secured by screws.

The cover is a strong ribbed casting, with a recess for the piston

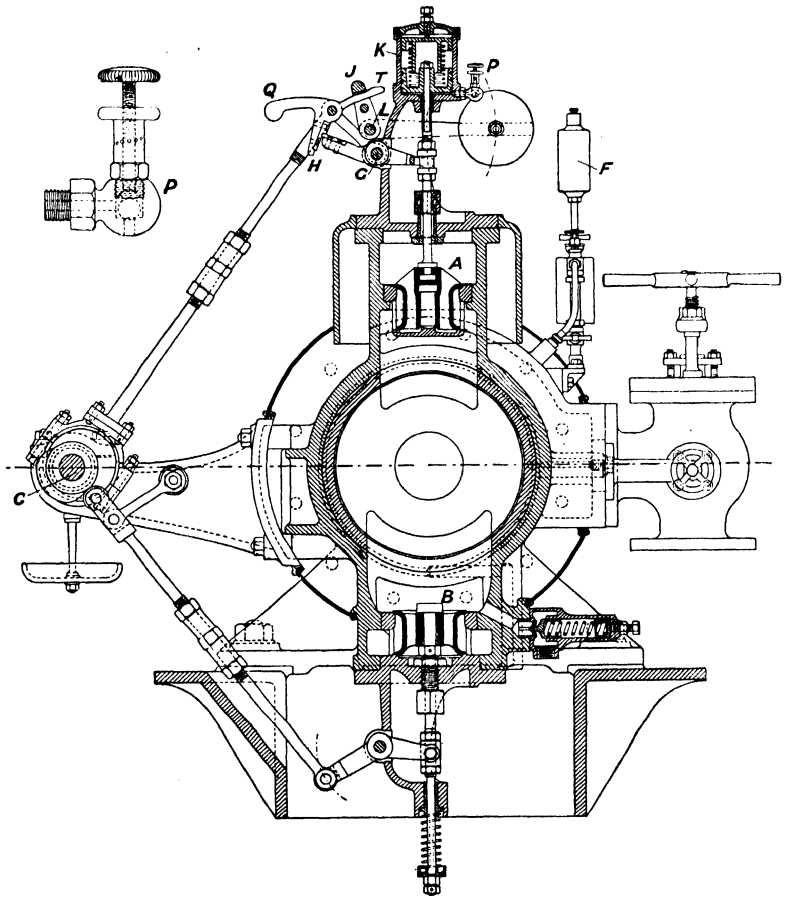



FIG. 344.

nut, which allows $\frac{3}{16}$-in. clearance. A polished cover is fitted as shown in Fig. 343 and the space between it and the cover forms an excellent air jacket.

As regards the trip gear, the lifting lever of the steam valve pivoted on the bonnet at G is fitted with a hardened steel plate at the outer end, which is engaged by the trigger arm H, carrying a similar plate. The amount of overlap or contact between these

plates, when in full gear, is $\frac{3}{16}$-in. The trigger arm H, a back balance arm Q, and a tail arm T are all in one piece. The amount of trip contact is controlled by T, and the balance Q swings H to the left and ensures effective engagement of the trip plates. An eccentric on the lay shaft C, driven by bevel gear from the crank shaft, is connected through its rod to two side links M and N, Fig. 343, between which the trigger arm is suspended. When the trip plates are in gear, these links, the lifting lever, and the trigger arm form a locked triangle. The tail arm T moves in contact with

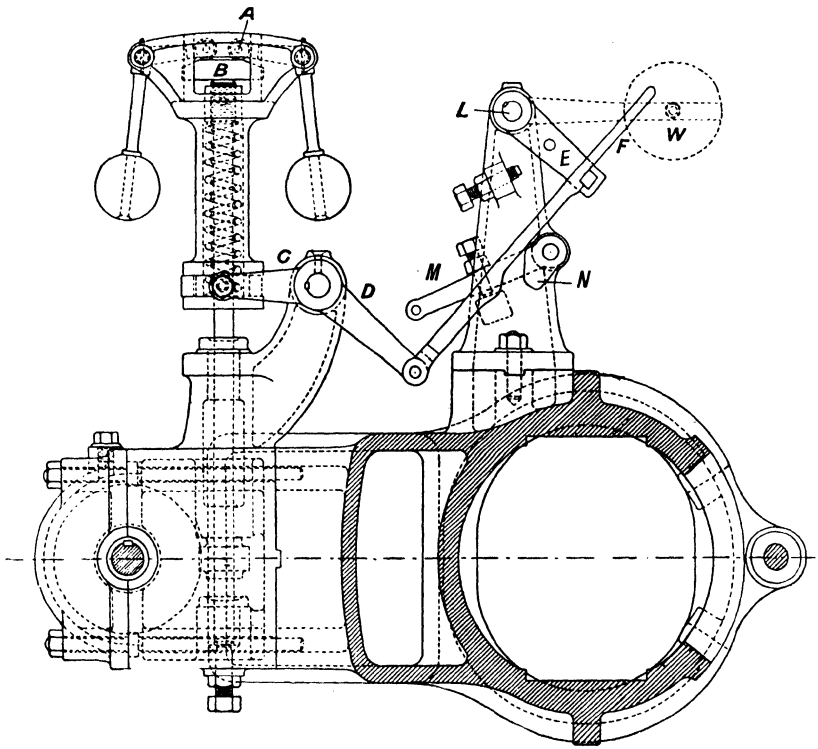



FIG. 345.

a pad on the end of the lever J keyed to the shaft L, which is controlled by the governor. It has an upper curvature towards the end, and as the rod pulls down H and the lifting lever, this curved surface, in contact with the pad, causes the tail T to move downward and throw the trigger arm out of gear. A lever with a counterweight, shown dotted in Fig. 344, is also keyed on the shaft L, and is prevented from falling by a safety catch. If this catch is released the weighted lever throws the tail T to the right and the trigger arm is permanently thrown out of action. The result is the immediate stoppage of the engine.

The auxiliary gear for the operation of this emergency cut-off lever is shown in Fig. 345. The inverted Hartnell governor controls the shaft L (also shown in Figs. 343 and 344) through the bell-crank lever D and the lever E keyed on L. The connecting rod F between D and E is pin-jointed to D, but merely engages a stop pin on the end of lever E. The weighted lever shown dotted and situated between the two steam valves, as shown in Fig. 343, swings the lever E down till the stop pin engages F, and holds it there in position under normal conditions of running.

A lever M and a cam piece N, Fig. 345, are keyed on a shaft

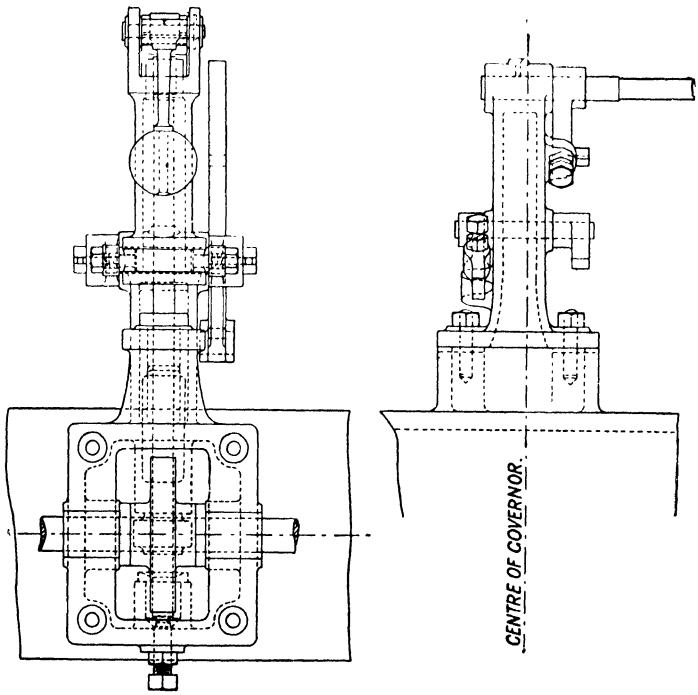



FIG. 346.

pivoted on the bracket. The position of this lever and cam can be adjusted by the screwed pin and the stop lug on the bracket.

The rod F is bulged on the under side to form a cam surface, so that when it is moved upward and engages with the surface of N it is thrown back sufficiently to detach it from the stop catch on E. When this occurs the weight W pulls round the shaft L, and the trip arm or trigger H, Fig. 344, is thrown out of gear. The movement of E is limited by the stop pin screwed into the lug on the pillar.

If the governor becomes inoperative the balls and the sleeve fall sufficiently to move F into contact with N, and release takes place.

Independently of the governor, if any emergency requiring immediate stoppage occurs in the mill, a cord attached to the end of lever M can be pulled to rotate the cam N and throw the rod F out of gear. When the trip gear liberates the valve it is quickly brought to rest under the action of the spring by the piston of the air cushion cylinder K, Fig. 344. This piston has about $\frac{1}{32}$ in. clearance when the valve is on its seat. The air regulator P with a non-return ball-valve is shown on a larger scale on the left.

As can be seen in Fig. 344, the valve operating rods are provided with right- and left-hand couplings by which the lengths can be varied for adjustment of lead and cut-off and for proper seating of the exhaust valves.

The bell-cranks of the inverted Hartnell spring governor (Figs. 345 and 346) are provided with rollers A at the ends of the short horizontal arms. These push down on the T-headed spindle B. The controlling force which the centrifugal force of the balls has to over-

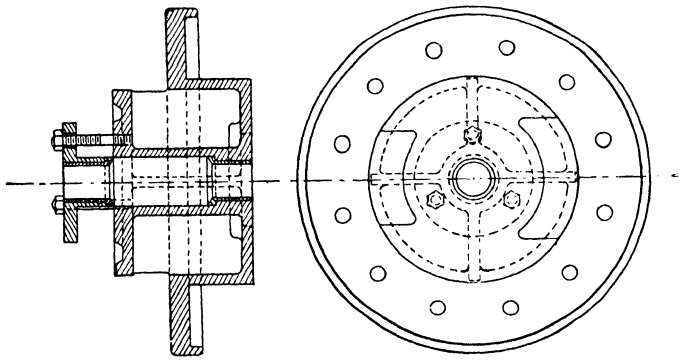


FIG. 347.

come is the resistance of the spring plus the dead weight of the complete head, which includes the box surrounding the spring and the double-ended bracket on which the ball arms are pivoted.

This arrangement ensures a heavier governor than the simple spring Hartnell, which is liable to hunt when working on a fine variation. The spindle is of steel, hardened at the bottom, and runs on a hardened adjustable toe piece. It is driven from the lay shaft by a screw gear running in an oil box, as shown by the dotted lines in Figs. 345 and 346.

The stuffing box for the piston rod is shown in Fig. 347. The front of the flange projecting to the left fits in a bored portion of the engine frame, and the flange face is bolted against the face of the frame. The space to the bottom of the stuffing box thus forms an efficient air jacket. The box and the gland are provided with long brass bushes. The projecting rim of the flange fits over the cylinder flange and holds the steel cleading in place.

The cast-iron disc crank is shown in section and elevation in Fig. 348. It is a force fit on the shaft. The pin is also a force fit and riveted over. It is lubricated by a visible drip lubricator by means of a centrifugal oiler. The oil drips into a ball on the rotating arm, in the line of the shaft axis, and it is driven by centrifugal force along the arm to the surface of the crank pin. The little rib in the ball prevents the oil from running out when the crank stops on the upper position. The drip lubricator is shown separately above Fig. 348. The girder frame of this engine has already been shown in Figs. 174 and

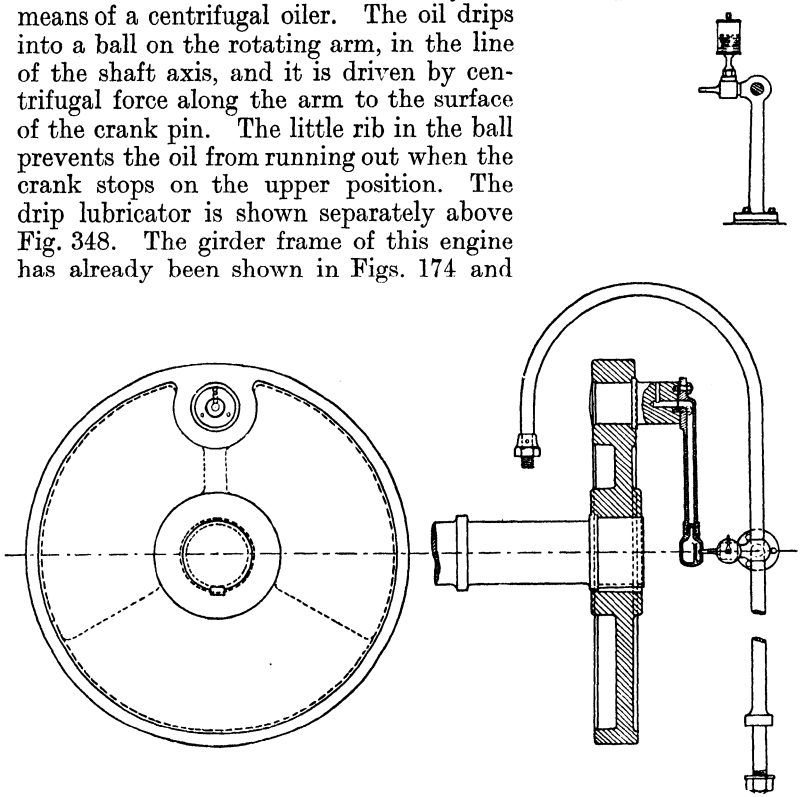


FIG. 348.

175. The arrangement of bevel drive for the lay shaft is shown in Fig. 349.

282. **Piston Drop-Valve Engine.**—A longitudinal section through a combined heat extraction and uniflow engine, with trip gear and piston drop-valves, designed on modern economic lines by Hick, Hargreaves and Co., is shown in Fig. 350.* A diagrammatic representation of the extraction system is shown in Fig. 351. The engine is a tandem compound with H.P. cylinder 28 in. diameter, L.P., 41 in. diameter, and stroke 36 in. The steam pressure is 160 lb./in.$^2$ gauge, and temperature 550° Fahr. (180° Fahr. superheat). The engine is rated at 1235 I.H.P. at a speed of 155 rev./min. on full load, with an extraction quantity of 15,000 lb. of steam per hour at 60 lb./in.$^2$ gauge for the process purposes in a woollen mill.

* Reproduced by permission from *Engineering*, 26 March, 1926. The student should refer to the complete drawings given there.

This is a case in which both the demands for power and process steam vary considerably, and provision is made for the by-pass of live steam through a reducing valve to the process steam main when the engine is running light and the exhaust is insufficient to meet the demand.

Referring to the sketch, Fig. 351, steam is supplied from the main steam pipe 1 to the H.P. cylinder 2 through the stop valve 3 to the branches 4 and 5, leading to the admission valve chests. It is exhausted through the branches 6 and 7 to the intermediate receiver 8. Part of the exhaust passes through the non-return valve 9 into the process or heating-steam main 10. The rest passes to the L.P. cylinder, through the branches 11 and 12. The uniflow system is used for the L.P., and the steam is exhausted at the centre of the cylinder into an under-hung jet condenser 13, from which the water is drawn by a horizontal air pump 14 through the branch pipe 15. The pump is driven from the crank shaft by a vertical rod and bell-crank lever. A small bore pipe 16 is led from the receiver to a cylinder 17 containing oil under pressure. Another small pipe 18 conveys the oil under pressure from the oil chamber to a relay cylinder 19. This relay is arranged to control the trip gear through the rod 20, so as to vary the cut-off in the L.P. cylinder and ensure the maintenance of a steady pressure in the receiver and heating main when the load fluctuates. At a very low load, when this control is insufficient to keep the receiver at the 60 lb./in.$^2$ limit, the relay further operates a by-pass valve 21, through the rod 22. Live steam is passed from the main 1 through a stop valve 23, and the auxiliary supply pipe 24. It is then supplied to the process main 10 at 60 lb./in.$^2$, through a reducing valve 25. As the load

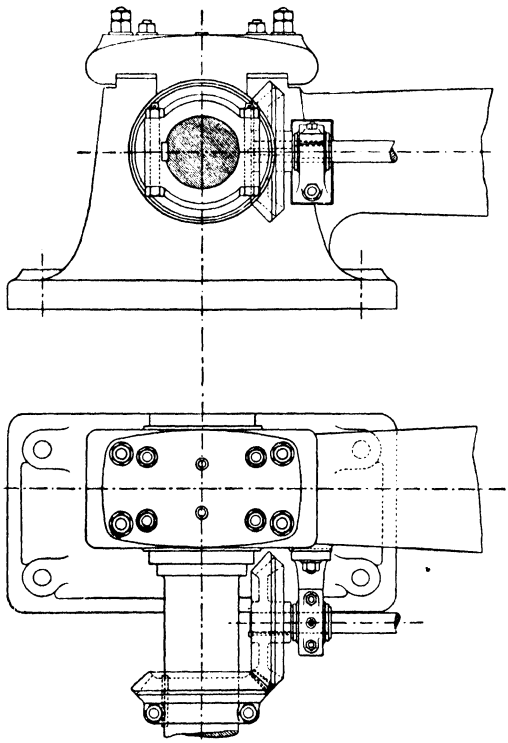


Fig. 349.

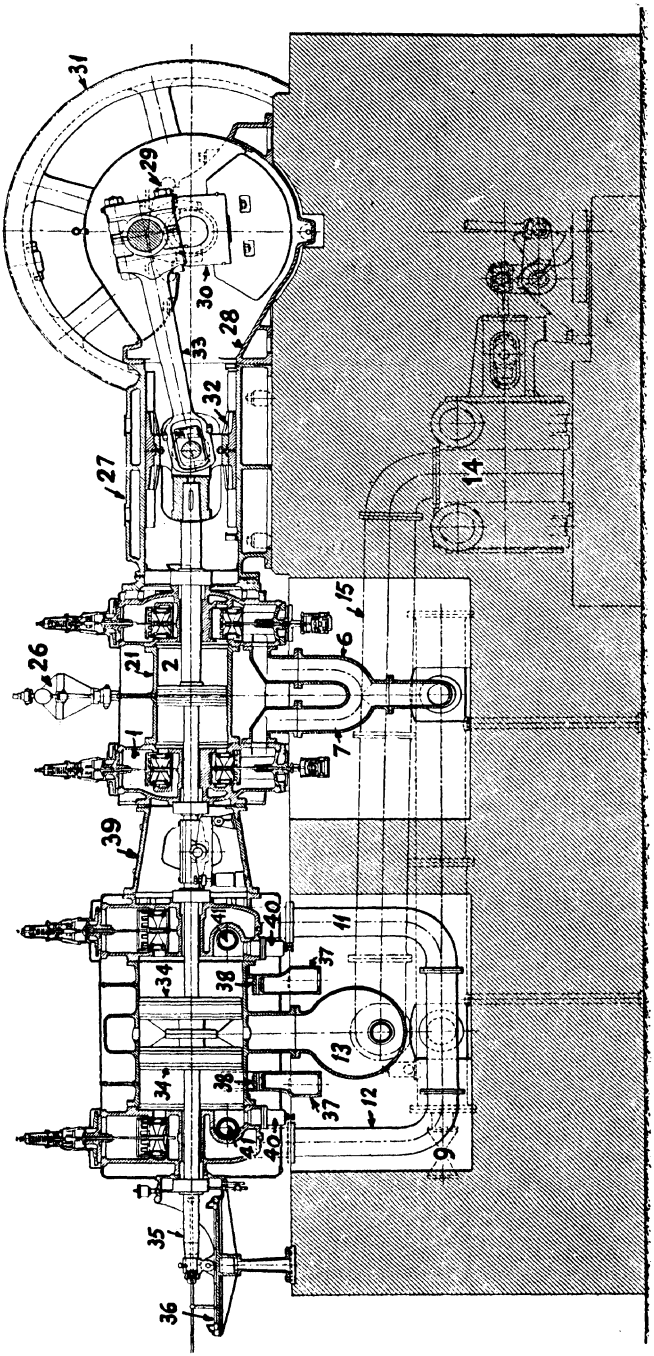



FIG. 350.

increases the relay governor closes down the by-pass, and at normal load the whole of the process steam is taken from the H.P. exhaust. Before the steam is used for process work, it is passed through an oil separator on the process main at the left.

The reference numbers of the piping, Fig. 351, are repeated on the section, Fig. 350. A detailed illustration and description of the valves and trip gear fitted on this engine has already been given in Art. 203, in the chapter on valve gears, and this section gives a clear illustration of their arrangement on the engine.

The loaded governor 26 controls the trip gear lever of the H.P. cylinder 2, as explained in Art. 203. The engine has a massive frame 27 symmetrical about the axis of the cylinder. This is bolted to the main bearing casting 28 integral with the two main bearings 29, which support the shaft on each side of the double web crank 30, fitted with balance weights. This type of construction is now replacing the older form of bayonet frame and single bearing with overhung crank. The shaft is supported by an out-board bear-

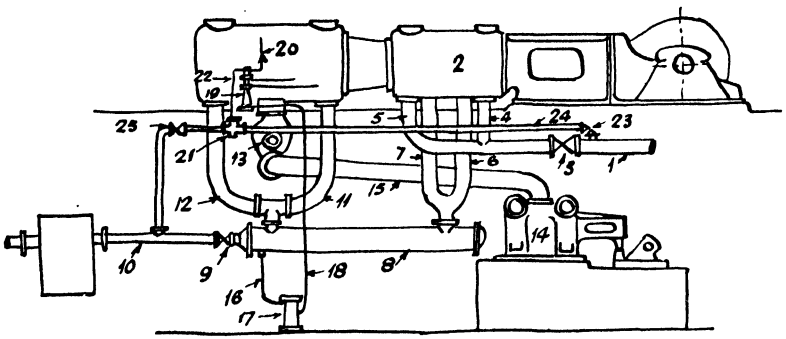



FIG. 351.

ing, and close up to this carries a flywheel 31, 12 ft. 3 in. diameter by 5 ft. 6 in. face, grooved for 26 $1\frac{3}{4}$-in. ropes. At the engine side of the rim there is a toothed ring, to gear with the pinion of a steam driven barring engine (not shown). The crosshead 32 is of the forked slipper type, to take the solid-ended connecting rod 33, which has a T head at the crank pin with double brasses and cap. The L.P. piston 34 is lightened by dividing it into two parts. It is provided with a tail rod 35 to reduce the wear on the bottom of the cylinder. This rod is supported on the slipper guide 36. The L.P. cylinder is attached to cross girders 37 by two feet 38, on each side. The H.P. cylinder is supported at the head end on the concrete foundation by a wide spreading foot, which is not shown on the drawing. The cylinders are aligned by the tapered distance piece 39, and cleaded with planished steel. Since the L.P. cylinder works on the uniflow principle it is necessary to provide relief valves 40 to guard against excessive compression, and also auxiliary clearance valves 41. The latter are discussed in Chapter XV.

# CHAPTER XV

## UNIFLOW, BACK PRESSURE, AND EXTRACTION ENGINES

**283. The Uniflow Engine.**—In the ordinary type of steam engine the steam enters one end of the cylinder, at high pressure and temperature, follows up the piston with decreasing pressure and temperature, then flows back during the exhaust stroke and out the same end at the lower exhaust pressure and temperature.

It is this contra-flow action that mainly gives rise to the injurious heating and cooling of the cylinder metal, especially that of the cylinder head, with the resulting condensation when fresh steam is admitted.

This evil is most accentuated in the slide-valve type of engine, where a long steam passage serves for both admission and exhaust of steam.

Although it is not so pronounced in the case of a Corliss or a drop-valve type of engine, in which a comparatively short passage is used for admission and another short passage for exhaust, it is still a serious cause of loss of efficiency.

Of the various expedients adopted to improve efficiency, by reduction or elimination of condensational and leakage loss, those of jacketing, initial superheating of the steam, compounding, and reheating have been briefly considered in the previous chapters.

**284.** There is a further method, which has been adopted of recent years and received considerable attention on the Continent and in America, and to some extent in this country, where it is said to have been originated by T. J. Todd, as far back as 1895.*

It received little attention at that time; but was taken up later in 1908 by Prof. J. Stumpf of Berlin, who developed and made it a commercial success.

In this system the contra-flow principle of the older type of steam engine is discarded, and a uniflow principle is adopted. The steam at the high pressure and temperature enters the cylinder at one end, as in the older type, follows up the piston with falling pressure and temperature, and is exhausted at the end of the stroke, through ports in the cylinder barrel which are uncovered by the piston. The low-pressure steam remaining in the cylinder when the piston

* See *The Engineer*, 24 and 31 May, 1895, for drawings and test results of Todd's "Quadruple Expansion Terminal Pressure Exhaust Engine."

commences the return stroke is not discharged through the cylinder head, but is compressed by the piston with increasing temperature to nearly the initial pressure of admission, and at the end of compression is in a superheated condition.

This system possesses an obvious thermodynamic advantage and certain mechanical advantages. The latter, however, are offset by disadvantages stated later. It obviates the cooling by exhaust steam of the clearance surfaces, where, as already pointed out in Art. 130, the major part of the initial condensation occurs in the contra-flow type. This condition, together with the superheating of the steam during compression, reduces the initial condensation so effectively that a single-cylinder single-stage expansion engine may be used instead of a compound or triple, without sensible reduction in economy.

The piston and exhaust ports in the cylinder barrel replace

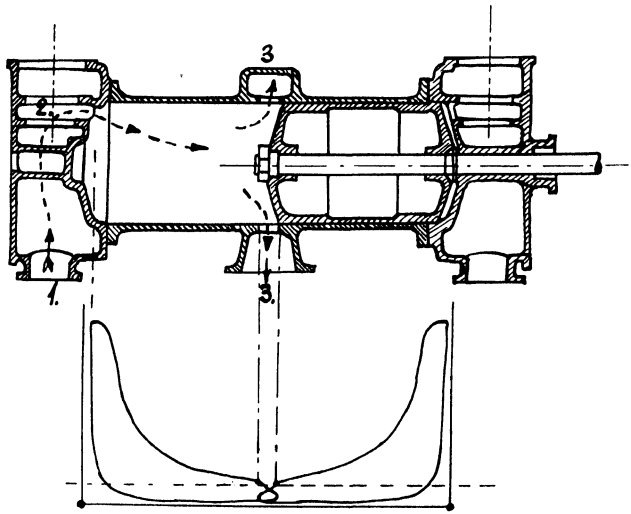



FIG. 352.

separate exhaust valves and ports, with their accompanying clearance surfaces and valve gears, and this is a decided mechanical advantage. Further, the exhaust area of the uniflow engine is about three times that possible with a slide or drop-valve contra-flow one. The resistance to exhaust flow is much reduced, and the back pressure is more nearly the same as the condenser pressure or the atmosphere than in the case of the contra-flow type.

**285.** A diagrammatic sketch of the section of a Stumpf double-acting engine cylinder is shown in Fig. 352. Steam either saturated or superheated, usually the latter, enters the valve chest at 1, passes through the seatings of a special resilient type of double-beat valve at 2, follows up the piston, as indicated by the directional

arrows, and then passes into the exhaust belt 3 as the piston uncovers a ring of ports in the wall. The width of this set of ports is about 10 per cent. of the piston stroke. When the piston overruns the ports on the return stroke, compression commences at 90 per cent. from the end of the stroke, and hence a high final compression pressure is obtained.

The form of indicator diagram produced is shown below the section, for each end of the cylinder. It is stated that such indicator cards show adiabatic curves of expansion and compression. This cylinder, as can be seen, consists of two single-acting cylinders having a common exhaust end, and on account of the long piston it is longer than the corresponding cylinder of a contra-flow engine.

There is practically a steady temperature gradient along the cylinder length, between the head and the exhaust belt. The inlet ends are maintained hot and the common exhaust cold, and the temperature changes gradually between hot inlet and cold exhaust.

**286.** Experience shows that with steam of high initial superheat, jacketing is ineffective, unless when the mean effective pressure is low. For steam of moderate initial temperature, about 450° Fahr. to 500° Fahr., a short jacket extending about a quarter of the barrel length may be provided.

When saturated or only slightly superheated steam is used, the barrel may be jacketed for part of the length. If continued right up to the exhaust belt, jacket heat would be thrown away in the exhaust. In every case the heads of the cylinders are jacketed.

These head jackets do not affect the operation of the piston. Their heating action is very effective, as the working steam is in continual contact with the surfaces which are maintained at a steady temperature ; and since no exhaust steam passes over them, as in the contra-flow case, there is no loss of this portion of the jacket heat to the exhaust.

In the case of partial jacketing of the barrel, however, there is some loss of jacket heat.

Owing to large temperature difference and difference of specific density of the live and the compression steam, the head jacketing produces an effective heating action during the expansion, exhaust, and compression on the portion of the steam close to the cylinder head.

The steam immediately following up the piston, however, is subject to drop in temperature and increase of wetness, as expansion proceeds. The major part of the moisture due to adiabatic expansion is thus situated close to the piston face, while there is a progressive dryness and increase of temperature from the piston face towards the cylinder head.

The wet steam escapes through the exhaust ports, while the portion which has received heat from the head jacket is trapped by the piston on the return stroke and compressed, first as saturated then as superheated steam, since during the first part of the com-

pression stroke heat is being transmitted to the steam from the head jacket.

The absence of exhaust valves and casings enables the clearance in this type to be reduced much below the value necessary in the design of the ordinary contra-flow engine. It runs from 1 per cent. to 2 per cent. of the piston displacement, and the clearance loss is consequently less.

The compression ratio is necessarily high. As compression starts at 10 per cent. of the return stroke, the initial and final volumes are 90 per cent. and 2 per cent. of the stroke volume, and the compression ratio is $r_c=45$.

Obviously a fairly high vacuum must be maintained in the condenser in order to avoid an abnormally high terminal compression pressure. There is always the risk due to failure of vacuum and other causes, and some form of relief valve arrangement has to be provided.

**287.** A uniflow engine designed to run normally condensing must also be able to run non-condensing under atmospheric or high back pressure, on account of liability to failure of vacuum, deficiency of circulating water, or to the demand for exhaust steam for industrial heating or process work.

Provision for these contingencies is usually made by auxiliary clearance spaces in the cylinder heads, which are normally blanked off by shut-down valves. The amount of the additional clearance depends on the initial and the required back pressure.

Professor Stumpf gives the following figures as suitable for the total clearance,* when the terminal pressure of expansion is taken as one atmosphere gauge pressure, about 30 lb./in.$^2$ abs., and the terminal pressure of compression as one atmosphere below the initial pressure:

| Initial pressure lb./in.$^2$ abs. | 120 | 135 | 150 | 165 | 180 | 195 | 210 | 225 | 240 |
|---|---|---|---|---|---|---|---|---|---|
| Total clearance per cent. | 27·9 | 23·8 | 21·3 | 19·2 | 17·6 | 16·25 | 15·15 | 14·2 | 13·4 |

The large increase of the clearance above the normal condensing value of 1 per cent. to 2 per cent. results in a considerable increase of steam consumption. With superheated steam this increase varies from 0·16 lb./I.H.P. hr. to 0·19 lb./I.H.P. hr., according to the range of superheat, for every 1 per cent. increase in clearance volume. From the point of view of economy, therefore, the additional clearance for non-condensing operation is not desirable.

The shut-down valves which blank off the additional clearance space under normal condensing conditions, have to be so designed and located in the cylinder heads that the normal clearance volume may be maintained when they are closed down on their seats. These valves may also be designed as spring-loaded relief valves.

* *The Uniflow Engine*, p. 48.

A section through the cylinder head of a Sulzer uniflow engine fitted with this combined shut-down and relief type of valve is shown in Fig. 353. The valve 7 is held down on the seating by the pressure of an adjustable spiral spring in the casing 8, and can be raised off the seating when required by means of the hand-wheel 9.

288. As an alternative to additional clearance, auxiliary exhaust valves may be used. These valves, by suitable gear, can be thrown into action when a change over from condensing to non-condensing operation takes place. They are, however, liable to leak and also cause an undesirable increase in the clearance space and condensation

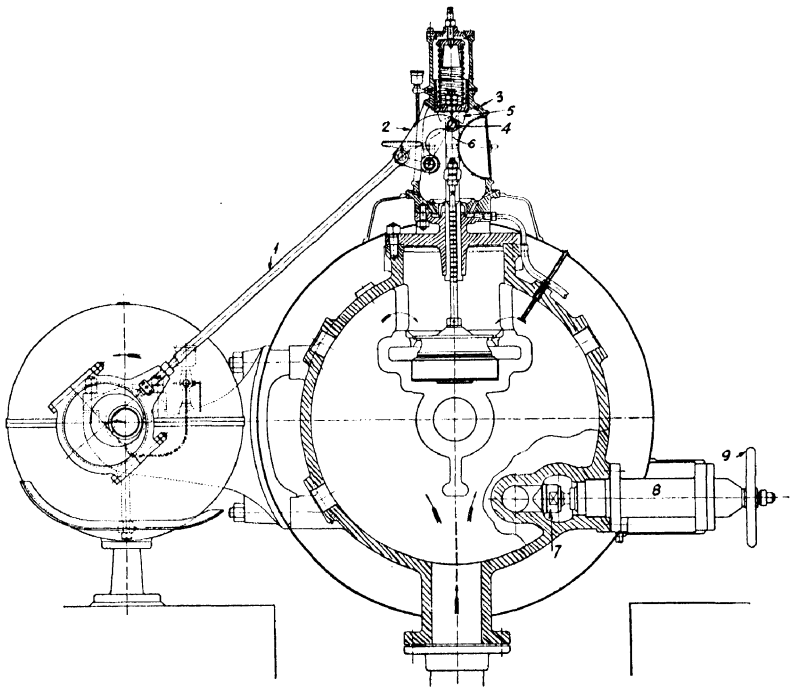



FIG. 353.

surfaces, and a partial reversion to the less economical contra-flow conditions, as the exhaust action becomes part uniflow and part contra-flow. There is a liability to over-expansion, with the formation of a negative loop on the indicator diagram, a condition that cannot arise when the engine is provided with additional clearance and shut-down valves.

Auxiliary exhaust valves may increase the consumption rate from the value already stated with non-condensing operation 0·19 lb./I.H.P. hr. per 1 per cent. clearance increase to 0·265 lb./I.H.P. hr. The permanent increase of clearance also increases

the consumption rate, under condensing operation, when these valves are cut out of action.

In some instances these auxiliary exhaust valves are made to serve as drain, relief and exhaust valves. An arrangement of this type fitted by Cole, Marchment, and Morley * is shown in Fig. 354.

The large hollow valve 1 is held on its seating by a heavy spiral spring 2, and can open against the resistance of the spring when the maximum limit of compression pressure is reached. This is the relief valve.

The exhaust valve 3 is seated on the top of this valve, and its spindle is connected to one arm 4 of a bell-crank lever pivoted at 5. The other arm 6 of this lever is pin-jointed to a rod 7, which carries a cam roller 8 at the upper end. This is jointed to a rod 9 which in turn is jointed to one arm 10 of a hand-operated lever 11, pivoted at 12. By adjustment of the hand lever, the cam roller can be brought hard up against the surface of a cam 13 on the second

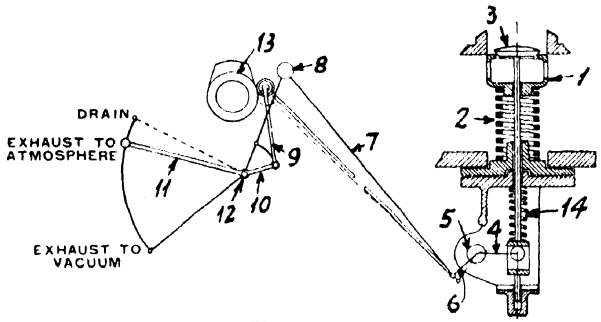



FIG. 354.

motion or " lay " shaft, or thrown entirely clear of it. To operate valve 3 as a drain valve in starting, the hand lever is set so that the roller remains in contact with the cam surface during the complete revolution. When the cylinder is cleared of water the lever is adjusted so that the roller engages the cam only on the return stroke The valve is then lifted by the arm 4 against the resistance of the smaller spiral spring 14, which assists in its closure, and then acts as an ordinary exhaust valve, for non-condensing operation. When the engine is run condensing the hand lever is adjusted to throw the roller completely clear of the cam, and the valve 3 goes out of action. If through any mishap the compression pressure should rise above the normal, the large valve 1 can open and relieve the pressure.

Instead of having hand-operated relief valves to provide against excessive compression pressure, an automatic gear can be employed.

The type fitted by Robey and Co., and shown in Fig. 355, consists

* From discussion on paper " The Uniflow Engine," by F. B. Perry, *Proc. Inst. Mech. E.*, July 1920.

of an eccentric 1 on the lay shaft 2, which can be connected to or disconnnected from an auxiliary rocker shaft 3 by means of a clutch 4. This auxiliary shaft carries the release valve cams 5, which are held in contact with the cam roller 6 on the valve spindles. The clutch is operated by a relay consisting of a piston in a cylinder 7, which has one side connected to the condenser and is provided with a spiral spring. Under normal running conditions the vacuum holds the piston down against the resistance of the spring, the clutch remains out of gear, and the rocker shaft is at rest. If the vacuum falls off appreciably or the engine runs at atmospheric pressure, the spring operates the piston which puts the clutch into

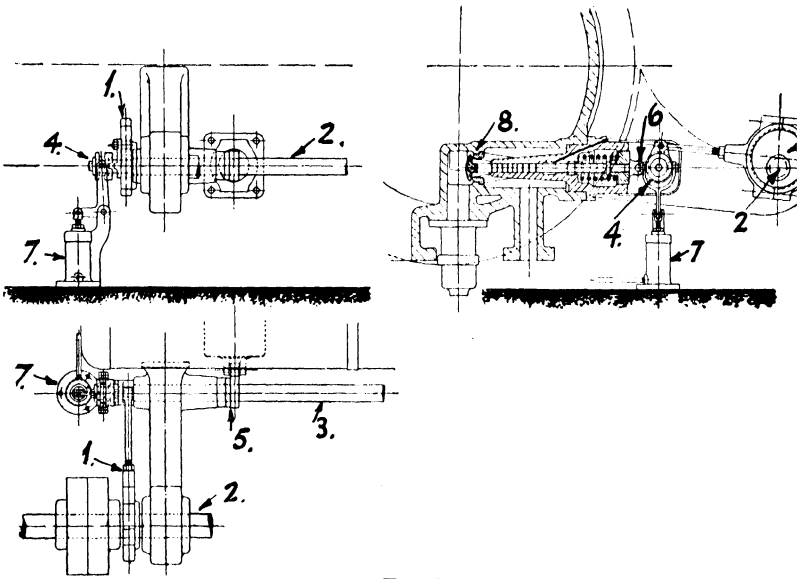



FIG. 355.

gear. The cams on the rocker shaft now rotated by the eccentric drive come into action and operate the relief valves 8.

289. Since the expansion in the uniflow engine takes place in a single stage the cut-off is necessarily much earlier than that of a multi-stage engine of the same power. It usually runs from 10 per cent. to 25 per cent. In some special cases, however, as in rolling mill engines, it may run from 40 per cent. to 50 per cent.

The following note and Table VIII of mean effective pressures have been given by F. Foster.* "The cut-off pressures are usually about 15 lb./in.$^2$ below the steam pipe pressure with a cut-off of 10 per cent. At earlier cut-off this drop is rather less, but unless the valves are specially arranged it increases a good deal at late cut-off. It has

* *Essential Principles of Engine Design*, Manchester Association of Engineers, 1915.

been assumed that the compression starts at 90 per cent. stroke and an absolute pressure of 2·5 lb./in.$^2$ abs., the clearance being 2·5 per cent. These conditions will cause the tabular M.E.P. to be low at early cut-off and too high at late cut-off. Allowances for rounded corners have been made, and hyperbolic expansion has been assumed. For superheated steam the M.E.P. must be reduced or the cut-off increased about 1·5 per cent. per 100° Fahr. superheat."

TABLE VIII

CALCULATED M.E.P. FOR CONDENSING UNIFLOW ENGINES, IN LB/IN.$^2$

| Pressure at cut-off, lb./in. abs. | 90 | 100 | 120 | 140 | 160 | 180 |
|---|---|---|---|---|---|---|
| 5 per cent cut-off . | 12·6 | 14·9 | 19·5 | 24·1 | 28·7 | 33·4 |
| 7·5 ,, ,, . | 19·9 | 23·0 | 29·3 | 34·5 | 41·6 | 47·9 |
| 10 ,, ,, . | 25·1 | 28·8 | 36·2 | 43·5 | 50·8 | 58·1 |
| 12·5 ,, ,, . | 29·9 | 34·1 | 42·5 | 50·9 | 59·1 | 67·4 |
| 15·0 ,, ,, . | 34·3 | 38·9 | 48·2 | 57·5 | 66·7 | 75·9 |
| 25·0 ,, ,, . | 49·3 | 55·5 | 67·9 | 80·3 | 92·7 | 104·8 |

There is usually difficulty in getting a trip gear to work satisfactorily enough to ensure early cut-off, and in most cases a " positive " gear is employed under the control of a powerful and sensitive governor. Alternatively, hydraulically operated valves may be used or some form of relay, controlled by a comparatively light governor.*

Early cut-off also involves a high initial load with accompanying high stresses, and thus necessitates a massive frame construction and heavy working parts. The single crank gives a very uneven torque, and a massive flywheel has to be used for steady running.

These are the mechanical drawbacks of the uniflow type already referred to. There is also a peculiar constructional feature which is absent in the case of the multiple-expansion engine cylinder. Owing to the gradient of temperature between the admission and exhaust ends of the stroke, the diameter at the admission end increases more than that at the exhaust when the cylinder is heated from the cold condition. In order to ensure a " parallel " cylinder under working conditions, it is necessary, in the case of a double-acting engine, to bore it so that it is barrel-shaped when cold.

**290.** A longitudinal section through the cylinder and valve chests of the type of horizontal uniflow engine, for the drive of electric generators, made by Robey and Co., Lincoln, is shown in Fig. 356, and outside views in elevation and plan are shown in Fig. 357. The automatic relief valves previously mentioned are shown at

* See *Engineering*, 16 March, 1928, for complete drawings of a compound uniflow engine by Galloways, Manchester. Also *Engineering*, 22 May, 1925, for a uniflow pumping engine by Worthington-Simpson Co.

1, Fig. 356. It will be noted from Fig. 357 that the engine has a massive form of frame known as the " mammoth " double bearing type. The trunk guide is cast solid with the bed and there is no chance of defective alignment. The crank casing is enclosed and the working parts are under forced lubrication. The bearing has four-part brasses, and the side wedges are formed with circular backs to enable them to automatically adjust themselves and thus ensure proper alignment. The drop-valves are driven by eccentrics on the lay shaft, through eccentric rods and cam gear, similar to that in Fig. 353. The eccentrics are of the movable type controlled by a powerful spring shaft governor in the casing 1, and the speed can be adjusted by means of the hand-wheel gear 2. A heavy flywheel with barring rack is provided. The engine exhausts into

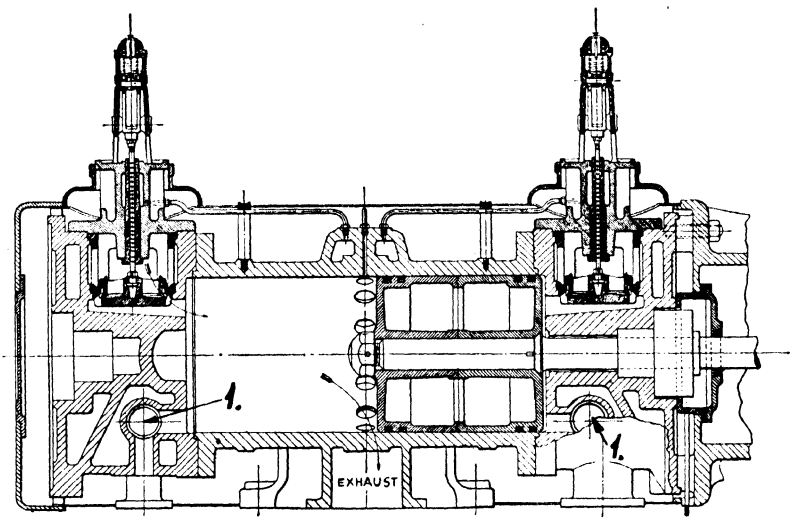
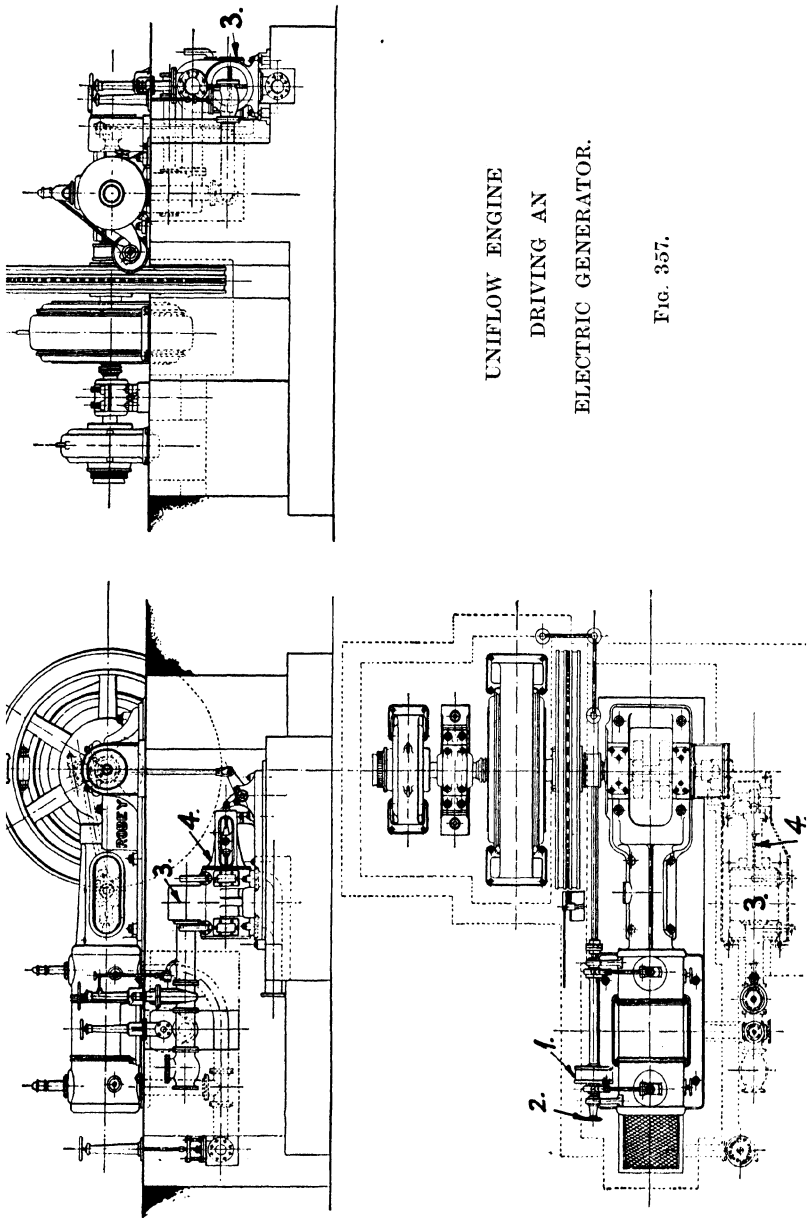



FIG. 356.

a jet condenser 3, placed below it on the right, and the double-acting air pump 4 is driven through a bell-crank and rod from a crank on the end of the main shaft.

**291. Back Pressure and Extraction Engines.**—In the discussions of thermal efficiency in Chapter IV, only the prime mover or motive element of the complete combination, which includes the steam generator, was considered. The thermal efficiency in this case is given by the ratio of the net output of the motive element to the heat supplied to the steam between the given exhaust or feed temperature limits and the initial pressure and temperature limits. It is the overall thermal efficiency of the plant, however, which determines fuel costs, and this of recent years has become an item of first importance. The engine or turbine may be very efficient, but the boiler may be far from efficient. Great advances, however,

have recently been made in boiler construction, and especially in operation, and in moderate and large-sized plants, boiler efficiencies of 80 per cent. to 84 per cent. are now being obtained. In some very




UNIFLOW ENGINE DRIVING AN ELECTRIC GENERATOR.

FIG. 357.

large central stations over 90 per cent. boiler efficiency has been reached. The thermal efficiency of the steam plant may be denoted by—

$$\eta_p = \frac{H_{Bf}}{H_f} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

where $H_{Bf}$=B.H.P. output per lb. of fuel burned.

$H_f$ =calorific value of the fuel.

Up to the present the highest value of turbine plant efficiency has been obtained in a large American power station. This is said to be 28·75 per cent.* The highest in this country is about 25 per cent.; but in the majority of central stations it runs from 12 per cent. to 16 per cent., and in industrial plants the figure is often below the 12 per cent. limit.

Taking the best figure, at least over 70 per cent. of the heat of each pound of fuel burned is lost through radiation, heat in the flue gases, and as heat carried away in the circulating water. The latter is the serious item, and it may be said that fully 60 per cent. of the heat of the fuel is thrown away in the exhaust steam of the condensing engine or turbine. The utilisation of this proportion of low-temperature heat is a matter of great economic importance.

Some progress in this direction has been made in recent years by power supply or " utility " companies in America in the distribution of waste heat in exhaust steam and hot water for industrial and domestic purposes in the immediate neighbourhood of the power stations; but little has been done so far in this country.

The application of the combined power and heating system has, however, been considerably extended in the cases of public institutions such as hospitals, asylums, hotels, residential flats, where considerable quantities of steam and hot water are required for heating and other purposes. The most profitable field for application of the system is the industrial one. Many industries require not only considerable power supply but also more or less steam for " process " work.

In many cases it is found more economical to instal separate power units with high-pressure boilers to give superheated steam, and to utilise the low-pressure exhaust steam for the manufacturing process, than to generate the low-pressure steam directly and purchase the power from the outside supply company.

A discussion of the economics of this question is outside the scope of this text, but the examples given below will indicate its importance to the student.

**292.** In such establishments as chemical, paper, soap, rubber, bleaching and dye works, laundries, textile factories, tanneries, wood mills, sugar refineries, etc., steam at pressures from 5 lb./in.$^2$ to 60 lb./in.$^2$ gauge, and even more, may be required for the process work.

* See *Engineering*, 19 December, 1930.

This steam may be obtained from batteries of low-pressure boilers, or by taking the exhaust from an engine which uses high-pressure steam, and supplies the necessary power for the establishment.

In the latter case the cost of power is so low that the power may be regarded as practically a by-product of the process steam or of the fuel used for steam generation. The underlying reason is the small variation of the total heat of the steam over a wide range of pressure.

To produce a pound of dry steam at 5 lb./in.$^2$ the total heat (from 32° Fahr.) is 1157 B.Th.U. ; for a pound of dry steam at 60 lb./in.$^2$ it is 1185. The difference here is 28 units on 1157, or an increase of 2·4 per cent. At 180 lb./in.$^2$, a common pressure for a back pressure engine, the increase is 4·15 per cent.

With superheated steam the additional heat usually runs from 10 per cent. to 12 per cent. increase, and this represents the heat from the fuel which is actually required for the production of the power. The cost of the amount of fuel to produce this additional quantity of heat is the cost of the power produced.

The nature of the combined power and process steam installation is determined by the following conditions :

(*a*) The process steam required may be just equal to that to give the required amount of power.

(*b*) The process steam required may be more than that required for power.

(*c*) The process steam required may be less than that for power.

The conditions (*a*) and (*b*) are best met by a back pressure engine which exhausts at the minimum pressure required for the process work.

In (*b*) any defect on the amount of exhaust steam can be made up by steam from the boiler which is passed through a reducing valve.

In case (*c*) there are several alternatives, but usually what is termed an " extraction " engine is used. These alternatives are—

($c_1$) A back pressure engine developing a proportion of the power that gives the required amount of process steam, run in conjunction with a separate condensing engine supplied from the same boiler. This engine supplies the balance of the power.

($c_2$) A back pressure engine only, the balance of electrical power being purchased from a supply company.

($c_3$) A back pressure engine in series with a heat accumulator. The latter is a heat reservoir which stores up the heat of the exhaust when this is in excess of the process demand, and thereafter returns the heat when the exhaust is in defect.*

($c_4$) A condensing engine from which the requisite steam is extracted at the required process pressure. This is the " extraction " type. In the case of the reciprocator, this is usually a compound engine and the process steam is drawn from the receiver.

* See *Steam Turbines*, by W. J. Goudie, 2nd edition, p. 638, for discussion of heat accumulators.

Case ($c_1$) is illustrated diagrammatically in Fig. 358. A battery of three boilers discharges into the high-pressure steam main, from which the steam can be supplied to the back pressure Belliss engine

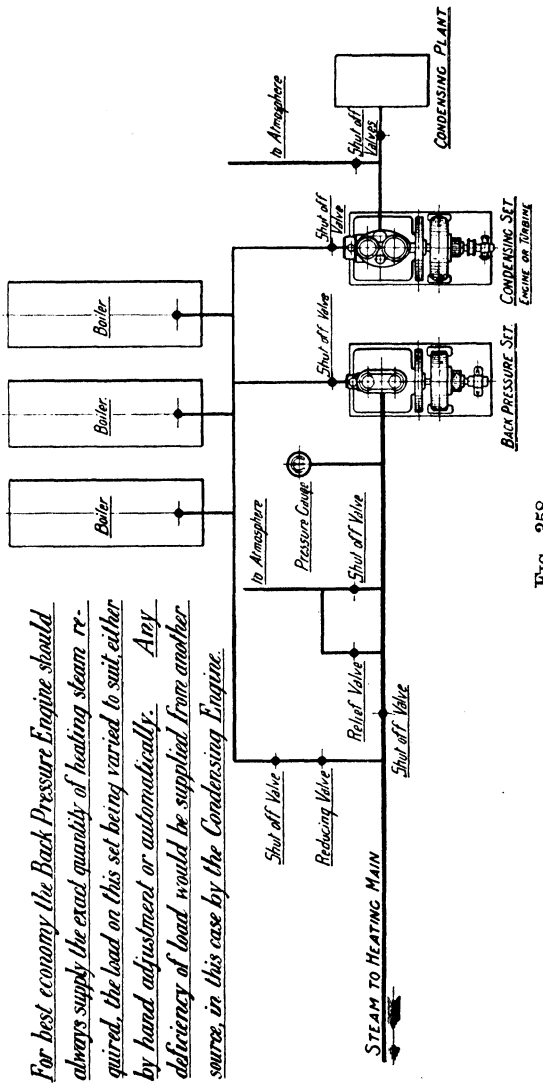



Fig. 358.

alone, or to this engine and the auxiliary condensing engine, or, in emergency, if the engines are shut down, through the reducing valve on the left directly to the process main.

Case ($c_3$) is illustrated in Fig. 359, which shows a diagrammatic arrangement of a Sulzer back pressure engine with a heat accumulator. High-pressure steam is supplied through the main 1 to the back pressure engine 2 and is exhausted to the heating main 3, provided with a safety valve 4 connected to the atmospheric exhaust pipe 5. The exhaust steam is first passed through an oil separator 6, then through a non-return valve 7, and then along the outgoing portion 8 of the heating main. When the supply of exhaust steam is in excess of the process requirements the surplus flows through the branch 9 of the non-return valve 10 to the heat accumulator 11. When the exhaust supply is in defect, the deficit is made up by re-evaporation in the accumulator, the steam

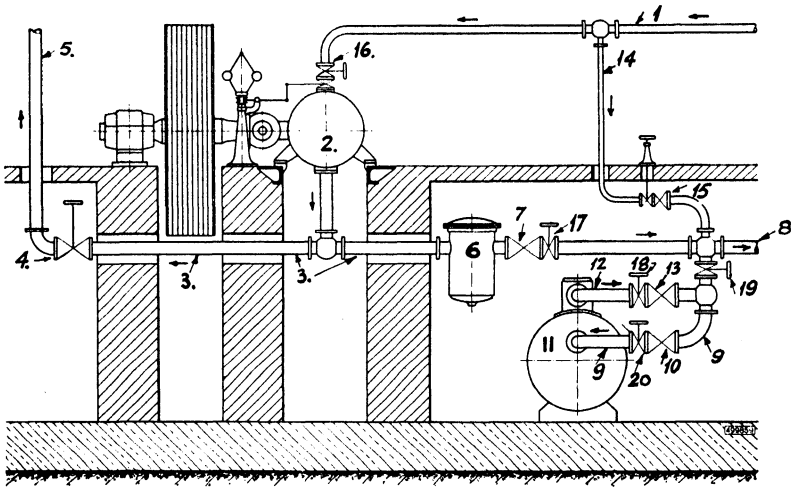



FIG. 359.

passing to the main 8 through the branch 12 and the non-return valve 13.

In event of stoppage of the engine, the necessary process steam can be obtained by passing high-pressure steam from branch 14 through a reducing valve 15 to the main 8. Shut-off valves are provided at 16, 17, 18, 19, and 20.

**293.** In the case of the back pressure engine the power obtainable for a given output of process steam is calculable when a suitable efficiency ratio is chosen for the particular initial and back pressure limits used.

Let $\epsilon_I$ = indicated efficiency ratio
$\eta_m$ = mechanical efficiency
$H_\phi$ = adiabatic heat drop between initial and back pressure
$W_p$ = weight of process steam in lb./hr.

Then the B.H.P. output of the back pressure engine is given by—

$$\left.\begin{aligned} B&=\frac{W_p\eta_m\epsilon_1 H_\phi}{2545}\\ \text{or}\quad B&=\frac{\eta_m W_p H_i}{2545}\end{aligned}\right\} \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

where $H_i$=indicated heat per lb. of steam.

When the engine drives an electric generator, the kilowatt output is—

$$K=\frac{\eta_g B}{1{\cdot}34} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

where $\eta_g$=generator efficiency.

The heat per lb. of steam given by the fuel to the water in the boiler at $p_1$ is $H=(H_{s1}-h_f)$, when the steam is superheated and $H=(H_1-h_f)$ when it is dry; $h_f$=the water heat at feed temperature $t_f$. If $\eta_b$ is the boiler efficiency, then the gross heat from the fuel for $W_p$ lb./hr. process steam is—

$$h_p=\frac{W_p H}{\eta_b} \text{ B.Th.U./hr.} \quad . \quad . \quad . \quad . \quad . \quad (4)$$

If the calorific value of the fuel is $H_f$, the weight of coal required for process steam and power is—

$$w_f=\frac{W_p H}{\eta_b H_f} \text{ lb./hr.} \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

Let N=number of operative hours per annum of the plant, then the net amount of fuel is given by—

$$W_f=Nw_f=\frac{W_p HN}{2240\eta_b H_f} \text{ ton/annum} \quad . \quad . \quad . \quad . \quad (6)$$

Also, if $k$=cost per ton of fuel in shillings, the net annual cost of fuel for process steam and power is—

$$C=kW_f=\frac{kW_p HN}{44800\eta_b H_f} \text{ £/annum} \quad . \quad . \quad . \quad . \quad (7)$$

In the case of an establishment in which the process steam is obtained from low-pressure boilers and the power supply from an outside company, if K is the supply in kilowatts, $y$ the rate in pence per unit, then for N operative hours the power cost is given by—

$$C'=\frac{yKN}{240} \text{ £/annum} \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

The following examples will serve to indicate how the foregoing expressions can be applied to estimate the fuel cost for given operative conditions and the probable saving from the adoption of a back pressure process plant in place of a low-pressure supply and purchased power.

*Example* 1.—In a dye work 5800 lb. of steam per hour is required at 50 lb./in.$^2$ gauge pressure. It is to be passed through a back

pressure engine driving a generator. Assuming the following data calculate the probable kw. output and the net yearly fuel cost for power and heating requirements.

Boiler pressure, 150 lb./in.$^2$ gauge; superheat, 100° Fahr.; back pressure, 50 lb./in.$^2$ gauge. Indicated efficiency ratio of the engine, 0·68; mechanical efficiency, 0·92; generator efficiency, 0·9; feed temperature, 150° Fahr.; boiler efficiency, 0·68; calorific value of coal, 12,000 B.Th.U./lb.; cost of coal, 15*s.* per ton. Approximate number of operative hours per annum, 3000.

$p_1$ =165; $ts_1$=100° Fahr.; from H$\phi$ chart, $Hs_1$=1260
$p_2$ =65 and $H_\phi$=80, $\epsilon_1$=0·68 $\therefore$ $H_i$=0·68×80=55.
$W_p$=5800; $\eta_m$=0·92; $\eta_g$=0·9.

$$\text{Brake horse-power output, } B=\frac{\eta_m W_p H_i}{2545}=\frac{0\cdot92\times5800\times55}{2545}=115$$

$$\text{Kw. output, } K=\frac{\eta_g B}{1\cdot34}=\frac{0\cdot9\times115}{1\cdot34}=77$$

Yearly fuel cost

$$C=\frac{kW_pHN}{44800\eta_b H_f}$$

$$C=\frac{15\times5800\times1142\times3000}{44800\times0\cdot68\times12000}$$

=£830 per annum for process steam and power

$t_f$ =150, $h_f$=118
H =($H_{s1}-h_f$)
=(1260−118)=1142
N =3000
$\eta_b$=0·68
$H_f$=12,000
$k$ =15*s.*

*Example* 2.—In the previous case find the approximate cost for power and heating requirements, if low-pressure process steam is supplied from boilers at 50 lb./in.$^2$ gauge and the electrical supply is taken from a power company at the rate of 1¼*d.* per unit.

Assume the same boiler efficiency, and estimate the probable percentage saving in cost on the coal bill when the back pressure engine is used.

With low-pressure steam $W_p$=5800, $p_1$=65, and taking dry steam, $H_1$=1182, $h_f$=118, $\eta_b$=0·68, $H_f$=12,000, N=3000, $k$=15*s.*, H=($H_1-h_f$)=(1182−118)=1064.

Cost of fuel—

$$C_1=\frac{kW_pHN}{44800\eta_b H_f}=\frac{15\times5800\times1064\times3000}{44800\times0\cdot68\times12000}=£675 \text{ per annum.}$$

In the back pressure case the actual fuel cost for power is thus £155/annum.

Cost of electrical power—

$$C'=\frac{yKN}{240}$$

$$=\frac{1\cdot25\times77\times3000}{240}=£1200 \text{ per annum}$$

$y$ =1·25
K=0·77
N=3000

Total cost process steam and electrical power—

$C=(C_1+C')$

$=(675+1200)=£1875$ per annum.

The saving by substituting the back pressure engine and high-pressure boilers is—

$$(1875-830)=£1045 \text{ per annum}$$

$$\text{Decrease in running cost}=\frac{1045\times 100}{1875}=55 \text{ per cent.}$$

294. Case ($c_4$) is illustrated diagrammatically in Fig. 360, which shows the sketch of a Sulzer horizontal compound extraction engine. A photograph of the engine cylinders and gears is shown in Fig. 361.

Referring to Fig. 360, live steam from the high-pressure main is

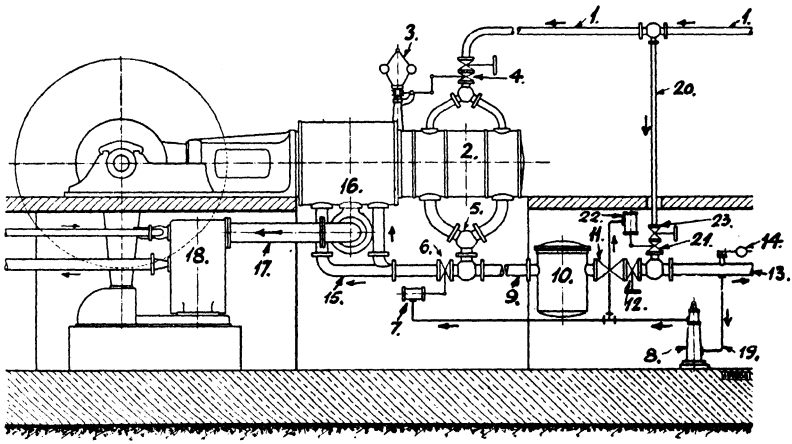



FIG. 360.

supplied through the pipe line 1 to the H.P. cylinder 2. The amount of steam supply is regulated by the engine governor 3, which actuates a special cut-off valve gear which is merely indicated by a valve 4 on this sketch. This governor and adjusting gear are contained in the casing G, Fig. 361, and shorten or lengthen the H.P. cut-off by adjustment of the admission valves E of the H.P. cylinder H.

Exhaust steam from the H.P. cylinder is discharged to a receiving vessel 5, Fig. 360, which is connected with a regulating valve 6, controlled by a servo-motor 7 worked from an oil operated pressure regulator 8, and also with branch 9 leading to an oil separator 10. The process steam is passed through this on its way to the heating main. The servo-motor is shown at S and the L.P. cylinder at N in Fig. 361. A non-return valve 11 is fitted on the outlet from the separator, Fig. 360. A shut-off valve 12 is fitted next the non-

return one. The process steam, after passing these valves, enters the heating main 13, which is provided with a safety valve 14.

Part of the H.P. exhaust passes into the heating main and the remainder through the valve 6 to the admission branch 15 of the L.P. cylinder 16, from which it is exhausted through the branch 17 to the condenser 18.

Suppose the load is steady and an increased demand for process steam arises, then a fall in pressure will ensue in the heating main 13. The reduced pressure in the connection 19 to the pressure regulator 8 will cause this to operate the servo motor 7, which will partly close the regulating valve 6. More steam will then flow into the heating main. The defect of steam to the L.P. will cause a reduction

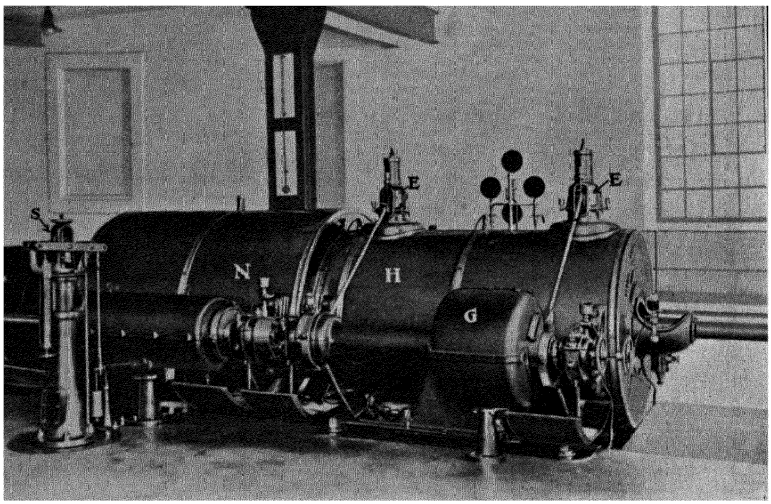



FIG. 361.

of power output and corresponding reduction of engine speed, and the main or speed governor 3 will come into action, to lengthen the H.P. admission, and increase the H.P. steam supply to the receiver 5 and the heating main 13. The engine will immediately speed up to the normal rate, and the combined output of the L.P. and H.P. cylinder will become the same as before the commencement of the regulation. This process will be reversed if the demand for heating steam diminishes.

An auxiliary supply of process steam can be obtained from the branch pipe 20, through a regulating valve 21 operated by a servo-motor 22. This can be blanked off by a stop valve 23.

**295.** If it were possible to estimate the indicated efficiency ratios of the H.P. or back pressure section and the L.P. section of a heat

extraction engine for a specified normal power and extraction output, the gross steam supply could be readily calculated. The indicated efficiency ratios, however, are indeterminate, and the probable values for any given conditions of operation can only be guessed.

If these values are arbitrarily chosen the probable total steam supply is calculable as follows:

Let W =gross steam supply in lb./hr.
$W_e$ =steam extracted in lb./hr.
$\eta_m$ =mechanical efficiency (0·92 to 0·95)
B =brake horse-power
I =indicated horse-power$=\frac{B}{\eta_m}$
$\epsilon_{I_1}$ =indicated efficiency ratio of H.P. section
$\epsilon_{I_2}$ = ,, ,, ,, L.P. ,,
$H_{\phi_1}$=adiabatic heat drop between $p_1$ and $p_e$
$H_{\phi_2}$= ,, ,, ,, ,, $p_e$ and $p_b$

where $p_e$ and $p_b$ are the extraction and condenser pressures respectively.

$$W\epsilon_{I_1}H_{\phi_1}+(W-W_e)\epsilon_{I_2}H_{\phi_2}=2545I,$$

$$\text{and} \quad W=\frac{2545I+W_e\epsilon_{I_2}H_{\phi_2}}{\epsilon_{I_1}H_{\phi_1}+\epsilon_{I_2}H_{\phi_2}} \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

The minimum brake horse-power, which gives the normal extraction quantity $W_e$, that is, the power for the engine when the H.P. section runs as a back pressure engine and the power developed in the L.P. section is zero, is given by—

$$B=\frac{\eta_m W_e \epsilon_{I_1} H_{\phi_1}}{2545} \quad . \quad . \quad . \quad . \quad . \quad . \quad (10)$$

The curve of gross steam consumption at normal and any reduced output of steam, on a base of B.H.P., is sensibly a straight line, which can be drawn when the values of W and B are obtained from Equations 9 and 10.

The following example illustrates this tentative method of estimating the gross consumption with varying power and constant extraction weight of process steam.

*Example* 3.—A compound heat extraction engine which has a normal load of 600 B.H.P. is supplied with steam at 185 lb./in.$^2$ abs. and 200° Fahr. superheat, and the pressure at the L.P. exhaust is 2 lb./in.$^2$ abs. The normal quantity of extracted steam is 8800 lb./hour, and the extraction pressure is 43 lb./in.$^2$ abs. Estimate the probable gross steam consumption and the minimum B.H.P. based on the following assumptions. Mechanical efficiency, 0·95; indicated efficiency ratios for H.P. and L.P. sections, 0·75 and 0·65.

Here $I=\frac{B}{\eta_m}=\frac{600}{0\cdot95}=631$ $\quad$ B =600
$\eta_m$ =0·95

In H.P. section adiabatic heat drop, on HΦ chart, between $p_1=185$ at $t_{s1}=200°$ Fahr. and $p_e=43$, is—

$$H_{\phi_1}=140, \qquad \epsilon_{I_1}=0{\cdot}75$$

and $\epsilon_{I_1}H_{\phi_1}=0{\cdot}75\times140=105$ B.Th.U./lb. $\qquad \epsilon_{I_2}=0{\cdot}65$

Initial condition at L.P., $p_e=43$ and $t_{s2}=70°$ Fahr.

Adiabatic heat drop to 2 lb./in.$^2$ abs., $H_{\phi_2}=209$, and—

$\epsilon_{I_2}H_{\phi_2}=0{\cdot}65\times209\doteqdot136.$

$$W=\frac{2545I+\epsilon_{I_2}H_{\phi_2}W_e}{\epsilon_{I_1}H_{\phi_1}+\epsilon_{I_2}H_{\phi_2}} \qquad W_e=8800$$

$$=\frac{2545\times631+136\times8800}{105+136}$$

$$=\frac{1606095+1196800}{241}$$

$$=\frac{2802895}{241}=11{,}630 \text{ lb./hr.}$$

Minimum brake horse-power is—

$$B=\frac{\eta_m W_e \epsilon_{I_1} H_{\phi_1}}{2545}$$

With the L.P. running idle on the low power, the mechanical efficiency will be slightly lower, say $\eta_m=0{\cdot}92$. Assuming the same efficiency ratios—

$$B=\frac{0{\cdot}92\times8800\times0{\cdot}75\times140}{2545}=334.$$

The resulting total steam consumption line AB for the normal extraction output of 8800 lb./hr. is shown in Fig. 362. It should hold up to about 720 B.H.P.

The corresponding curve of gross rate of steam consumption based on the input to the H.P. section, is shown by CD. The gross steam consumption curve determined by the foregoing calculation practically coincides with the experimental curve for 8800 lb./hr. obtained from a compound Sulzer extraction engine working under the conditions of pressure, powers, and extraction taken for the example.

The complete consumption diagram is shown in Fig. 363, and from this the probable consumption at any reduced or increased load (up to about 25 per cent. overload) can be read for any extraction quantity, at 43 lb./in.$^2$ abs., greater or less than the normal amount.

The minimum load limit curve is the locus of the points where the extraction quantities cut the corresponding gross consumption lines.

For $W_e=0$, when there is no external load, there is still a gross

supply of 600 lb./hr., representing the steam consumption for the compound engine running light.

The student as an exercise can draw down the gross consumption rate curve for each of the extraction quantities, similar to the curve CD, Fig. 362.

296. The reciprocator has been considered in the foregoing sections. The steam turbine can also be used, either as a back pressure or an extraction unit. It is not possible to state any rule regarding the choice between reciprocator and turbine, as so much depends on the particular conditions of any given case.

In general, however, for outputs below 1500 kw. the reciprocator

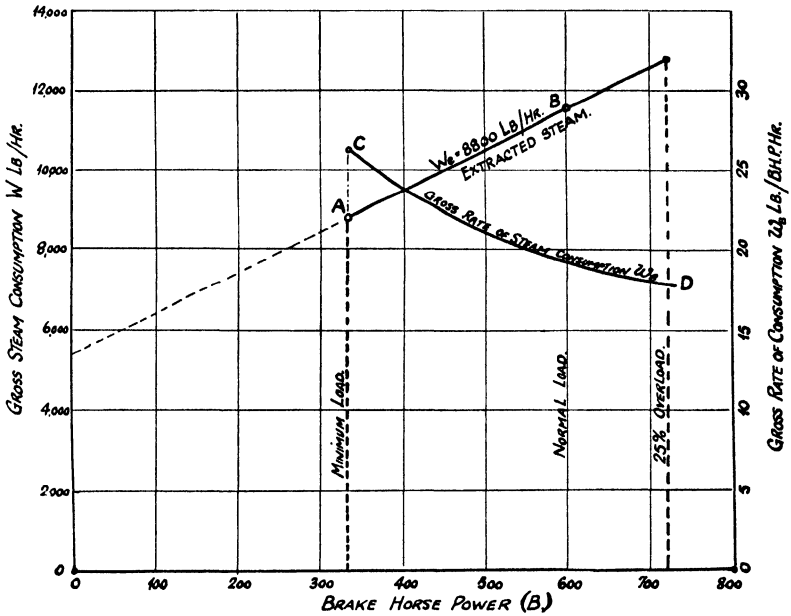



Fig. 362.

is the superior of the two. For higher powers the turbine has some advantage as regards space occupied compared with the corresponding reciprocator back pressure or extraction outfit. Steam can be extracted from it at several pressures, and this steam is free from lubricating oil, a condition which does not obtain with the reciprocator. On the other hand, the H.P. section of the turbine is essentially of lower efficiency than the L.P. section, on account of frictional and leakage losses. With high pressure and superheat it may be possible to obtain an indicated heat per lb. from the reciprocator considerably greater than the corresponding heat value from the H.P. section of the turbine. A back pressure engine would thus be more suitable than an extraction turbine

for the production of the required power for the factory. Also, with an extraction turbine the most efficient portion, the L.P. section, is lightly loaded when there is a heavy extraction, and the bulk of the power has to be developed in the less efficient H.P. section. In general, the steam supply to the turbine is regulated

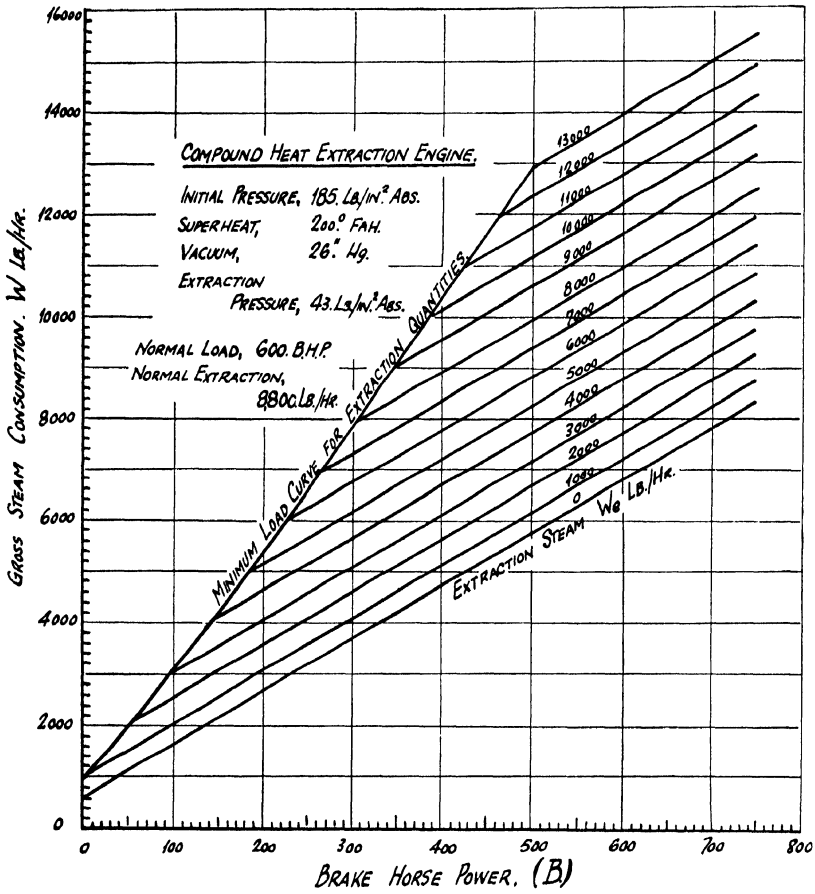



FIG. 363.

by throttle governing, which is a less efficient method than the cut-off system used as a rule on the reciprocator extraction unit.

These conditions have militated against the extraction turbine in recent years and brought the reciprocator more to the front as an extraction unit.

An illustration of a modern design of Parsons extraction turbine is shown in Fig. 410, p. 627.

# CHAPTER XVI

## HIGH-SPEED ENGINES

**297.** BEFORE the advent of the steam turbine the drive of electrical generators of small and moderate output was effected by belt or other form of intermediate gearing from what may be termed a slow-speed or low-revolution steam engine, that is, an engine running at from 100 to 150 rev./min.

As the dynamo developed, a demand arose for a higher speed engine for its direct drive; and in response to this an engine, running from 200 to 500 rev./min., was evolved. This type, however, apart from its application to electrical drive, possesses certain advantages, due to the higher speed of rotation. It is smaller for a given power output than the slow-speed engine; it has a better turning moment: and also a superior economy—due, as already indicated in Chapter VI, to shortening of the time of contact of the steam with the cylinder walls, per cycle. It is now not only extensively used for the direct drive of small and moderate power electric lighting sets, but to drive pumps, fans, and other auxiliary machinery, and also for drive of mills and factories. It is made in the simple, compound, or triple form, for outputs running from a few horse-power in the case of the simple non-condensing engine working with saturated steam of low pressure, to well over two-thousand horse-power, in the case of a triple condensing engine, working with superheated steam at pressures from 170 to 200 lb./in.$^2$

One of the early quick revolution engines was the famous Willans single-acting sleeve-valve engine, on which the late P. W. Willans carried out his classical series of systematic engine trials, and laid the foundation of the scientific system of engine testing now in universal use. This engine has been obsolete for many years and therefore is not reproduced here. The student can, however, refer to the original papers by Willans in the Proc. Inst. C. E., vols. cxiii and cxiv.

**298.** One of the troubles encountered in the early development of the high-speed type of engine was that of inefficient lubrication of the working parts, which led to seizure and breakdown. An important advance in the progress of the design was made by the introduction in 1890, by Messrs. Belliss and Morcom, of the system of forced lubrication of the working parts, which has now a world-wide application to all kinds of engines and machines. It is of interest to note that the first forced-lubrication engine, of about

20 H.P., built by the firm, and used for the direct drive of a dynamo at 625 rev./min., was in constant service from 1890 to 1919, in which year its use was discontinued. At the end of 29 years of useful service it had all the original parts, and the journal bearings showed little sign of wear, although it is estimated that in this time the shaft made more than 4000 million revolutions.

**299. Belliss High-Speed Engine.**—Three types of this vertical enclosed engine are made, two-crank simple or non-compound, two-crank compound, and three-crank triple expansion.

A longitudinal section of one form of the two-crank compound is shown in Fig. 364, and a transverse section in Fig. 365. The outstanding feature of the design is the superposed piston valves, driven from a single eccentric which serves both for the H.P. and L.P. cylinders. The H.P. cylinder is situated on the right hand and the L.P. on the left, with the valve casing between them. The H.P. piston valve is fixed at the lower part of the valve rod and the L.P. valve at the upper part. The cranks are set at 180°. As shown by the directional arrows, the H.P. valve admits steam to the cylinder over the inside edges and exhausts it over the outside edges, through the interior of the L.P. valve, which works as an ordinary piston valve with outside admission. Instead of placing the H.P. valve below the L.P. one, it may be placed above it and arranged for a variable cut-off through the medium of an expansion governor (described later).

By this means the cut-off may be adjusted to permit of 25 per cent. overload, or to suit condensing or non-condensing operation.

The cranks being set at 180° and steam simultaneously admitted to the top of one cylinder and the bottom of the other, a satisfactory balance of the reciprocating parts is obtained, vibration and load on the bearings being reduced to a minimum. The governing, in this case, is ensured by the horizontal centrifugal governor, fitted on the shaft extension at the right-hand bearing (Fig. 364). The governor sleeve operates the vertical arm of the bell-crank lever, and the horizontal arm, through the vertical rod, raises or lowers the equilibrium throttle valve at the steam inlet to the H.P. receiver passage. An additional adjustment is provided, in the form of a spiral spring attached to the vertical arm of the bell-crank lever, so that a fine regulation of the speed may be obtained while the engine is running. This governor limits the variation of speed to 3 per cent., between full and no load.

The steam connection is made at the vertical branch, shown behind the throttle valve. The directional arrows indicate clearly the path of the steam from the supply pipe through the valve to the H.P. cylinder.

The H.P. piston is provided with two rings held in a keeper block. The combination is kept in place below by the bottom piston flange and at the top by a junk ring. The piston block and this ring are secured to the rod by a coned end and nut. The

L.P. piston is of the light conical type, all in one piece, with a deep groove for two rings, which are sprung into place. The H.P. piston

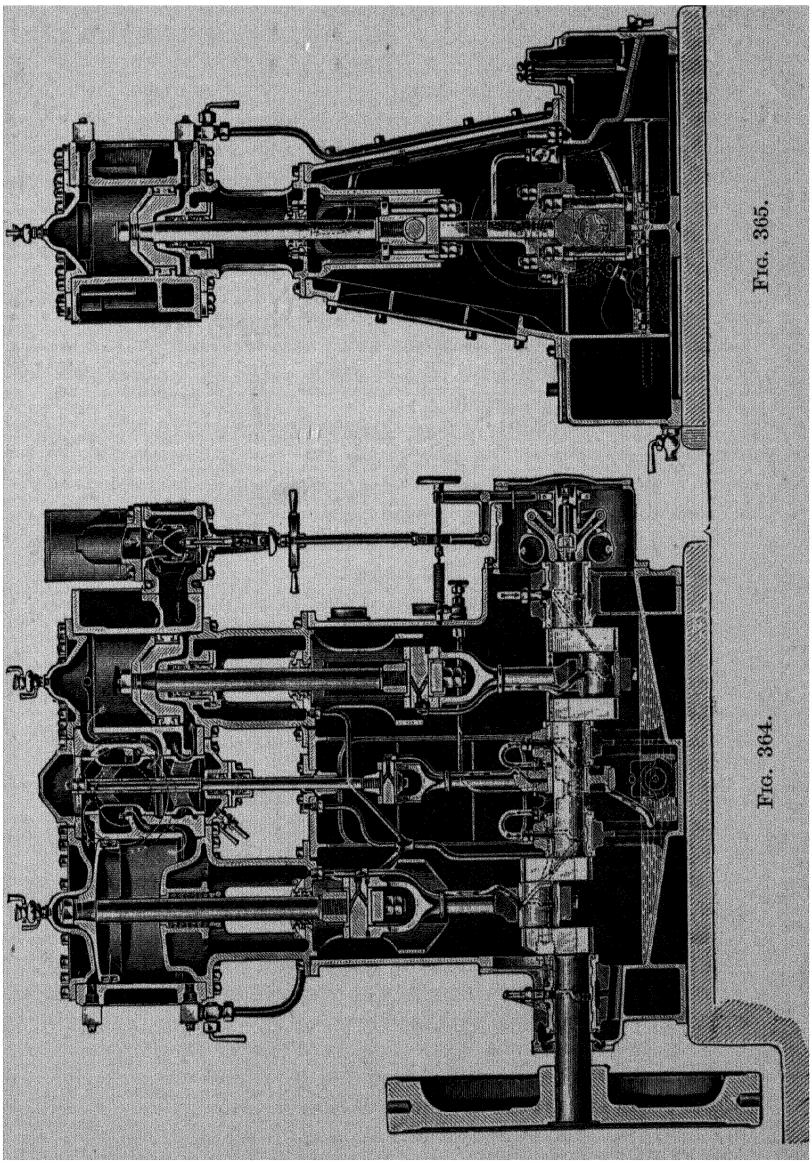


FIG. 365.

FIG. 364.

rod and valve rod are packed with metallic packing and the L.P. rod with a soft packing.

The cylinders are supported on the top of the casing by two

columns. These are made of sufficient length to ensure that the distance between the stuffing glands and the top of the casing is greater than the travel of the piston rods. These rods pass through stuffing glands at the top of the casing.

An extension of each column, into the casing, carries the guides for the crosshead, into which the piston rod is screwed. The casing is provided with large doors, at back and front, for access to the working parts. It is bolted to a wide bedplate, which is provided on the top with doors for access to the crank chamber, the bottom of which is made cup-shaped in the longitudinal direction to act as an oil well.

A side pocket is provided on the right of the bedplate chamber, Fig. 365. Oil carried up from the casing by the piston rods and water which drips from the cylinder glands are carried down, by the system of drain pipes shown into this side pocket, where the oil is automatically separated from the water and returned to the crank chamber. The water is rejected to waste. The oil consumption is thus reduced to a minimum.

The forced lubrication system consists of a plunger pump, shown dotted on the left of the crank case, Fig. 365, and driven from a lug on the bottom of the eccentric strap. In the early stages of the development an ordinary pump, with stuffing boxes and loose valves, was tried and found to be unsatisfactory. It was replaced by an oscillating pump. The plunger was made a close fit in the barrel, without a stuffing box, and a port in the barrel which oscillated on trunnions was brought alternatively opposite the suction and discharge ports in the body. This is the type now used and shown in the figure. It works immersed in the oil in the crank chamber, and runs continuously without any attention. A gauze strainer is fitted on the suction side of the pump, as shown on the right in Fig. 365.

The pump discharges the oil, through the pipes shown, to the top of each shaft bearing and to the governor joints, and it is distributed through ducts (shown dotted) from the bearing shells to the big end brasses of the connecting rod, and thence, by the vertical pipes, is led up the rod to the gudgeon pin, in which ducts are bored for the distribution of the oil to the crosshead brasses and guides. The eccentric strap and eccentric rod pin are similarly lubricated. The oil which leaks past journals and pins falls back into the crank chamber, to be strained and used over again. The oil is supplied under a pressure of from 10 to 30 lb./in.$^2$ Each connecting rod is of the marine pattern, with forked end to take the gudgeon pin, which works in the crosshead brasses held in place by the cap and bolts shown. The other end is T-shaped to take the two big end brasses and their bolts and cap. Wide outer bearings are provided, especially that on the left, which supports the end carrying the overhung flywheel. It will be noted that while the bearing at the right next the governor is inside the crank chamber, that on the left is

outside it. The cylinders are provided with spring-loaded relief valves, and at the bottoms with drain cocks ; a drain is also provided at the bottom of the valve casing. In this type grease cup lubricators are fitted on the tops of the cylinder covers. In the large triple-expansion engines a sight feed lubricator is fitted at the H.P. inlet.

**300.** Another form of the two-crank compound, which is capable of dealing with considerable overload, is the V type inclined valve engine. The H.P. and L.P. valves in this case work in separate chambers. They are again placed between the cylinders, one on each side of the engine, and are set at an angle on either side of the centre line, and operated radially from the shaft as a centre by separate eccentrics. The H.P. valve cuts off on the inside edges, as before, and exhausts directly to the L.P., and the use of outside receiver pipes is thus avoided.

This type is fitted with automatic expansion gear. It is governed on the throttle on light loads and by the expansion governor on heavy loads. This system enables overloads of 25 per cent. to be taken, and a uniform rate of steam consumption is obtained over a wide range of load. The various details are the same as those of the previous engine, with the exception of the H.P. valve.

**301.** Longitudinal and transverse sections of the vertical triple-expansion type are shown in Figs. 366 and 367.

This engine has three cranks set at 120° ; and the shaft is supported by six bearings. The high-pressure piston valve is of special design. As can be seen on the end elevation (Fig. 367), which shows the outside of the valve, the piston faces top and bottom are serrated at the admission edges, and by rotating the valve on its axis the position of these edges relative to the port in the liner can be altered and the cut-off lengthened or shortened. The valve is suspended from the top end of the spindle, on which it can rotate, and connected to a Stanhope lever arrangement, operated by the vertical rod which leads from the bell-crank lever of the centrifugal governor.

The action is as follows. As the load on the engine increases the speed falls off, the governor balls close in, and the throttle opens out, and this process continues until the valve no longer throttles the steam, which passes to the piston valve at full initial pressure. Up to this point the expansion governing mechanism has been at rest, at its earliest cut-off position, about 0·5 stroke, which may correspond to about $\frac{3}{4}$ load. As the load is further increased and the governor balls move in farther, the bell-crank operates the expansion gear, rotates the valve, and lengthens the cut-off, until the latest cut-off value is attained. This may be about 0·8 stroke, depending on the normal load or overload, as may have been determined. The governor valve or throttle during this period still continues to open, but this action has no effect on the engine, as the valve is already beyond the throttling position.

The piston valve is operated through a spring, and if the load should be suddenly thrown off, the throttle valve can close instantly,

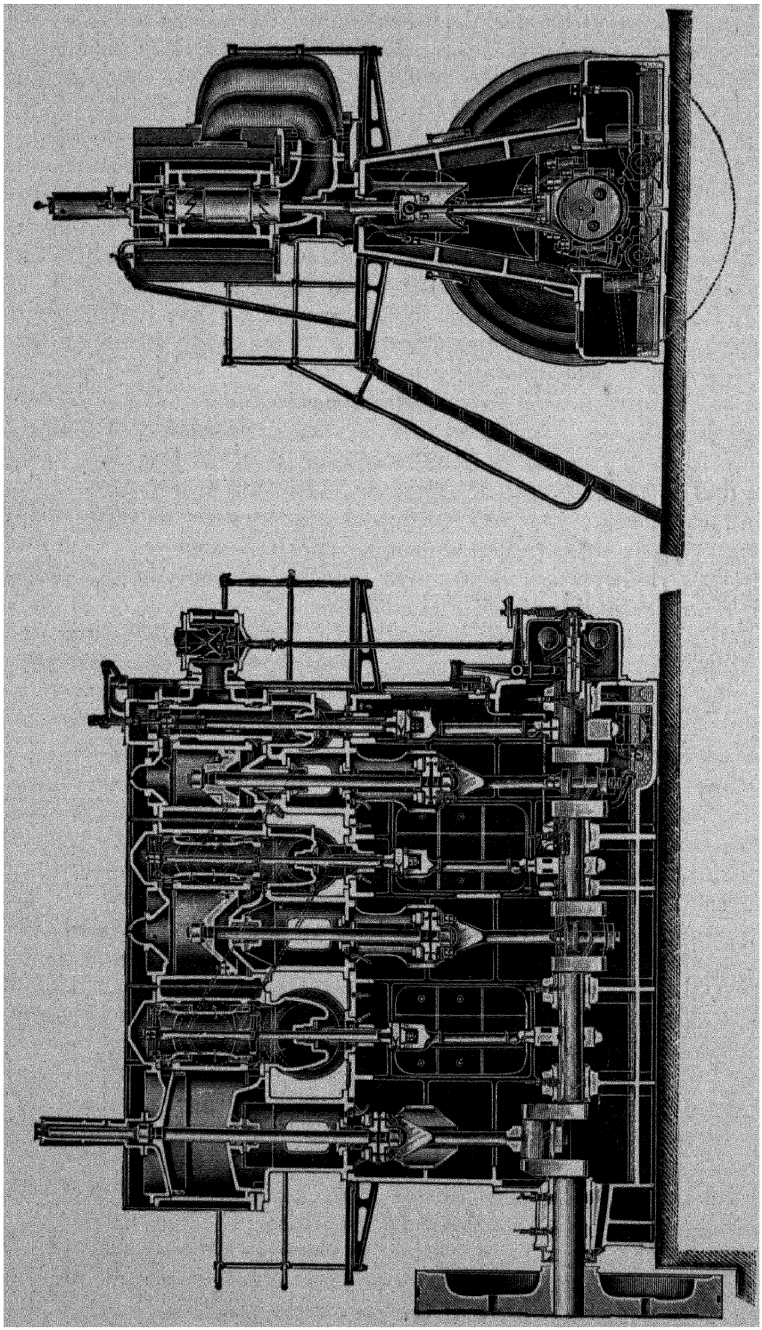


FIG. 367.

FIG. 366.

leaving the piston valve to be gradually rotated under the combined action of the spring and the movement of the engine. If desired, the expansion gear can be disconnected from the throttle and fixed in the latest cut-off position, and the engine can be run on the throttle alone.

The I.P. and L.P. piston valves, placed between the cylinders, have also inside admission. In each case the exhaust takes place through the valve to the single exhaust outlet at the bottom of the casing, and is led by the inclined exhaust branches, shown dotted in Fig. 366 and full in Fig. 367, to the centre of the next valve casing. The single L.P. exhaust is connected to the condenser. The various details are similar to those of the first engine described. It will be noted, however, that in this case two oil pumps are required to ensure efficient lubrication of the larger number of surfaces. This type is made in sizes to develop from 250 to 2500 B.H.P. at speeds from 500 to 200 rev./min. with superheated steam at pressures of 170 to 200 lb./in.$^2$

As already indicated in Chapter XV, this type of engine is also suitable for belt or rope drive of mills and factories, where it is found advantageous to retain this older system in preference to an electric drive.

In large factories with several storeys, a rope drum may be provided on the engine shaft, grooved for a large number of ropes, several of which lead to each floor. When the spans are short, however, belts are to be preferred to ropes, as they have a longer life. The limit of power transmission, however, is about 300 H.P. for belts. Above this they become too cumbersome and costly.

**302. Sisson High-Speed Engine.**—Another high-speed engine of quite different design made by Wm. Sisson and Co. is shown in longitudinal and transverse section in Figs. 368 and 369. It is a compound. The H.P. cylinder is placed above and partly overlaps the L.P. By this arrangement the cranks, which are set at 180°, are kept close together and the dynamic rocking couple on the frame is materially reduced. As shown in Fig. 369, the H.P. and L.P. piston valve casings are in line, forming, in fact, a continuous chamber, and the piston valves are mounted in tandem on the single valve spindle, which is worked from a shifting type of eccentric controlled by a shaft governor. The governing of this engine is thus entirely of the cut-off or expansion type; there is no throttle valve. The H.P. cylinder 1, Fig. 368, is bolted to the extension flange 2 of the L.P. cylinder 3. This cylinder is bolted to the top of the engine casing 4 by the corresponding flange 5. The neck 6 of the H.P. cylinder is check-jointed into the top of the L.P. extension, which has a cored passage 7. A long gun-metal sleeve or neck bush 8 with three bearing strips is fitted into this, and extends right up to the bottom of the H.P. cylinder. A stuffing box and gland 9 is provided at the bottom of 7. With this arrangement the packing does not come in contact with the steam heated portion of the H.P.

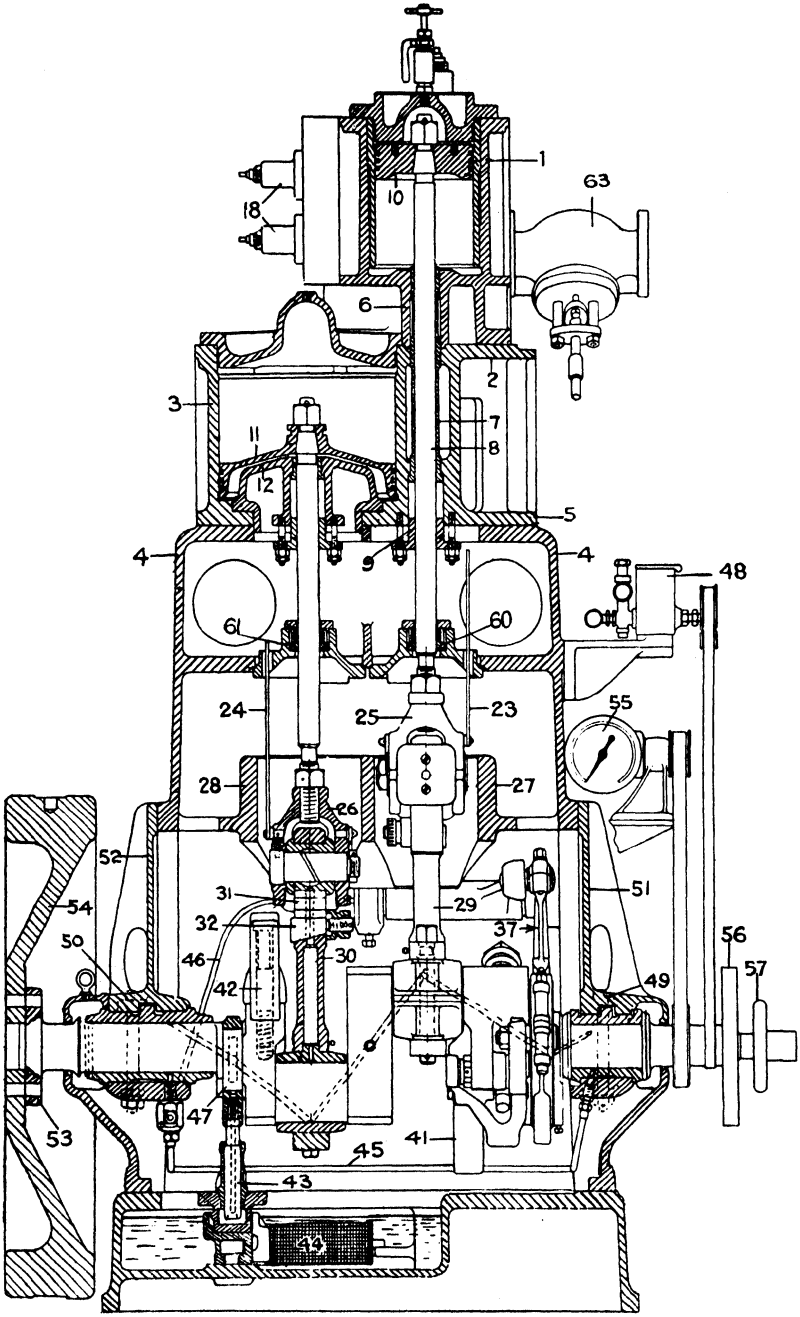

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FIG. 368.

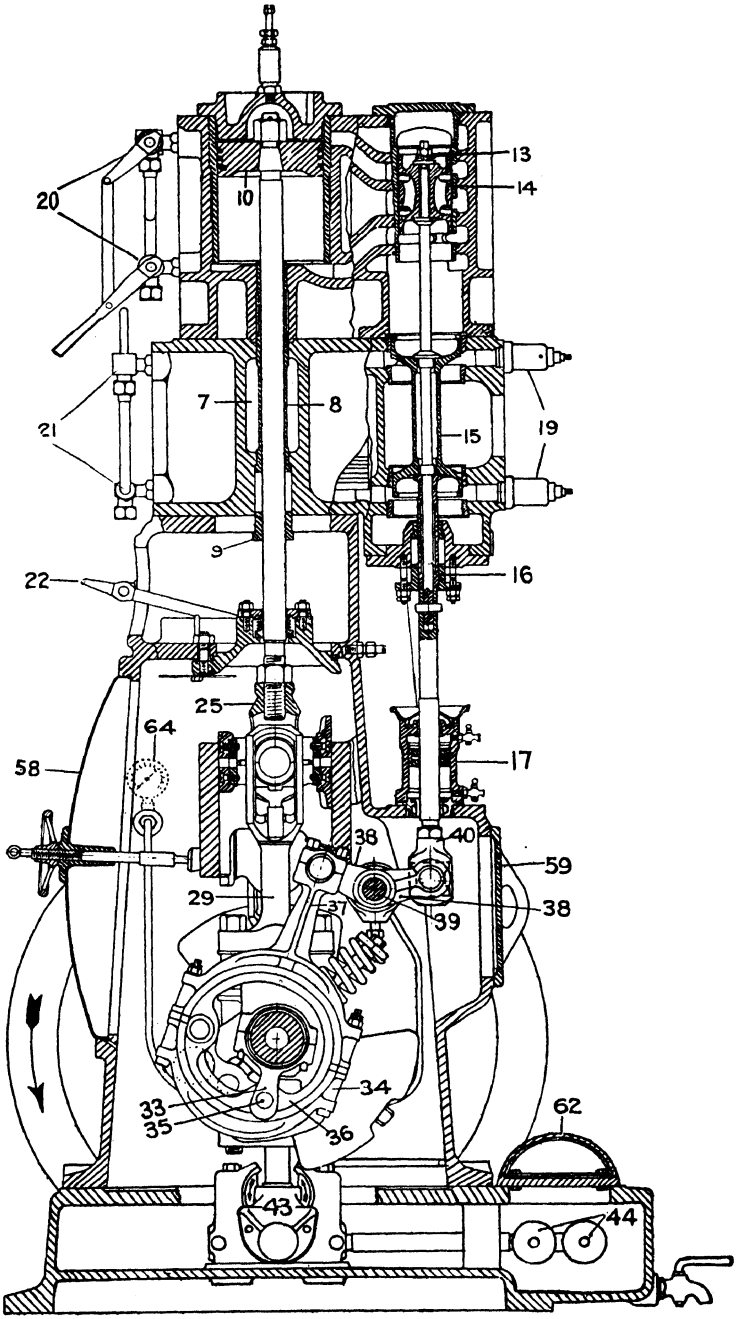

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FIG. 369.

piston rod. The cylinders are made of a special cast-iron mixture, and the H.P. is also fitted with a hard cast-iron liner. The H.P. piston 10 is of cast-iron, and fitted with three spring rings of the same quality as the liner. The coned L.P. piston 11 is of cast steel and has three similar cast-iron rings. The lower cover 12 carrying the deep stuffing box and gland is jointed from the inside of the cylinder. The top covers of the H.P. and L.P. cylinders are of the usual type, recessed to take the piston rod nuts. The H.P. piston valve 13, Fig. 369, admits steam at the inside edges. It is of cast-iron, and works in a hard cast-iron liner 14.

The L.P. valve 15 admits steam at the outer edges. It is held on the spindle by a long cast-iron sleeve, which works in the stuffing gland 16. An air cushion piston and cylinder 17, provided with test cocks, balances the inertia forces on the valve gear at the ends of the travel.

Spring-loaded escape valves 18 and 19 and lever-operated drain valves 20 and 21 are fitted top and bottom to the H.P. and L.P. cylinders. A square is formed on each top valve to take an indicator. The H.P. indicator cord is attached to the lever 22, Fig. 369, driven by a link 23, Fig. 368, from the crosshead. A similar link 24 from the crosshead operates the L.P. indicator gear. The cast-steel crossheads, 25 and 26, Fig. 368, are fitted with cast-iron slippers, which work in the bored-out guides 27 and 28. The piston rods screwed into the heads can be adjusted to give the necessary piston clearances.

The steel connecting rods 29 and 30 are bored to lighten them. Each rod end is of the solid type, and the brasses are adjusted on the gudgeon pin by a steel pad 31 and a wedge bolt 32, whose nut is locked by a taper pin. The big end is of the ordinary marine pattern, with bolts and caps, Fig. 369.

The swing link 33 of the shifting eccentric 34, Fig. 369, is pivoted on the pin 35 on the eccentric ring 36. The arm 37 of the eccentric strap is connected to the valve rocker lever 38, which rotates on the gudgeon pin 39. The pin of the outer arm of the rocker fits into a sliding block in the valve spindle crosshead 40. The governor arrangement has been described in detail in Chapter X. The crank shaft is machined from a special steel, and the cranks are fitted with balance weights 41 and 42, Fig. 368. As indicated by the dotted lines, diagonal oil channels are bored through the journals, crank webs, and pins.

Oil is drawn from the well in the base of the casing by the oscillating pump 43, through strainers 44, and discharged under pressure through the distributing pipes 45 and 46 to the bearings, journals, crank pins, crosshead gudgeon pins, guides, and eccentrics. The oil pump is driven by the eccentric 47. The valves and cylinders are lubricated by a mechanical lubricator 48, belt driven from the shaft.

The caps of the bearings 49 and 50, Fig. 368, are cast in a piece

with the casing doors 51 and 52. When these are removed and the working parts and flywheel are disconnected from the shaft, it can be withdrawn through the openings.

A flange 53 is turned on the left end of the shaft and the overhung flywheel 54 is bolted to it. The speed is recorded by a tachometer 55. The wheels 56 and 57 are part of the adjusting gear of the crank shaft governor already described. The front of the casing is closed by a large circular door 58, Fig. 369, of light section, which is held down on a machined facing by the central bolt and hand-wheel nut. A smaller hand door 59 is provided at the back, for access to the governor gear. Oil-tight packings 60 and 61, consisting of spherical neck bushes held down by springs, are fitted at the casing top. Access to the oil well is obtained by removing the cover 62.

Steam is admitted to the H.P. valve casing through the stop valve 63, Fig. 368, which is made of cast steel or cast iron to suit the operative steam conditions. The oil pressure of the forced lubrication system is registered on the gauge 64 fitted outside the casing, Fig. 369.

# CHAPTER XVII

## MARINE ENGINES

**303.** In recent years many developments have taken place in land practice, while the design of the marine reciprocating engine has scarcely altered. There has, however, been a considerable development of the engine-room equipment, in the form of auxiliary machinery, which it is out of the question to consider in this text, as the limitation of space restricts the discussion to the main propelling engine.

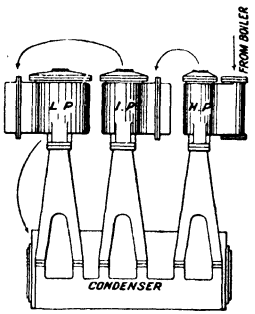



FIG. 370.

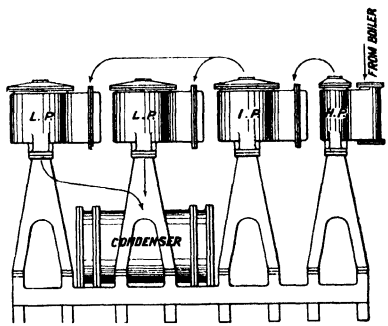



FIG. 371.

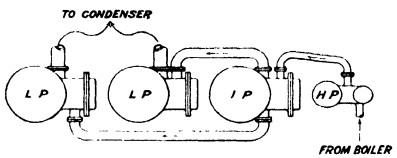



FIG. 372.

The student can refer to special texts dealing with marine engineering.* The present practice in merchant vessels is to fit the three-crank triple type, with the arrangement of cylinders as shown in Fig. 370. With large-powered engines where a very large L.P. cylinder was required, the single cylinder used to be replaced by two L.P. cylinders, and a four-crank engine was produced. This older arrangement is shown in elevation and plan in Figs. 371 and 372.

* See Seaton's *Manual of Marine Engineering.*

The direction of steam flow from the H.P. inlet to the condenser is clearly shown by the directional arrows. Besides the reduction of the L.P. cylinder diameter, this arrangement of four cranks gives a better balanced engine. Quadruple expansion engines are not as often fitted as in former years. One of the early forms of quadruple is shown in the photograph, Fig. 373, and does not sensibily differ, in its essential details, from the type being fitted at present. This engine, unlike the quadruples of its day, had five instead of four cranks, obtained by dividing the L.P. cylinder. These cranks were set at equal angles. It ran with saturated steam at the then high pressure, of 255 lb./in.$^2$ gauge. It was designed to have light reciprocating parts, to give equal load on each crank pin, reduce the bearing pressures, and obtain a more even turning moment. This was one of the early cases in which exhaust feed heaters and extraction feed heaters were employed to heat the feed, which was discharged to the boilers at 400° Fahr. This five-crank quadruple type was not generally adopted.

Within the past five years an important advance has been made toward the more efficient utilisation of the heat of the steam, at the low-pressure end of the scale, by the addition of an exhaust geared turbine to the marine reciprocator. This Bauer-Wach system is being applied to many existing marine engines and fitted in a large number of new vessels in this country and abroad. Still another development has recently been tried in the adoption of high-pressure high superheat steam. The ordinary marine engine runs with steam from 180 lb./in.$^2$ gauge to 200 lb./in.$^2$ gauge, and in the majority of cases the steam is saturated, although in some instances it may be given a moderate superheat, of the order of about 100° Fahr. One or two vessels are now running with steam at 500 lb./in.$^2$ gauge and 700° Fahr. temperature, special cast-steel H.P. cylinders being used with hydraulically operated poppet valve gear.* In these cases the usual Scotch marine boilers cannot be used, and water-tube boilers are fitted.

To economise space, the text of this chapter is confined to a description of one representative triple-expansion marine engine, and a brief discussion of the reciprocator-turbo combination of Bauer and Wach.

**304. Triple-Expansion Marine Engine for a Cargo Vessel.**—This engine, built by Rankine and Blackmore, is shown in side and end elevation in Figs. 374 and 375. It has cylinders H.P. 26 in. diameter, I.P. 43 in. diameter, L.P. 74 in. diameter, and stroke 60 in. The steam pressure is 180 lb./in.$^2$ gauge, vacuum 26 in., and the I.H.P. developed at 70 r.p.m., with initially dry saturated steam, is 2800.

The H.P. cylinder 1, Fig. 374, has a hard cast-iron liner, but the barrels of the I.P. and L.P. cylinders 2 and 3 are plain. Steam is

* See Paper "The Application of High Pressures to the Reciprocating Marine Steam Engine," by S. G. Visker, *Proc. N.E. Coast Engineers and Shipbuilders*, 17 November, 1927.

supplied through a stop valve 4, Fig. 375, opened and shut by a lever and screw gear operated by the hand-wheel 5. It is admitted to the H.P. cylinder at the inside edges of the piston valve 6, Fig. 374, and exhausted over the outer edges to a belt leading to the I.P. valve casing. It is then admitted to the I.P. cylinder by an Andrew's balanced slide-valve 7, the frame of which is held up

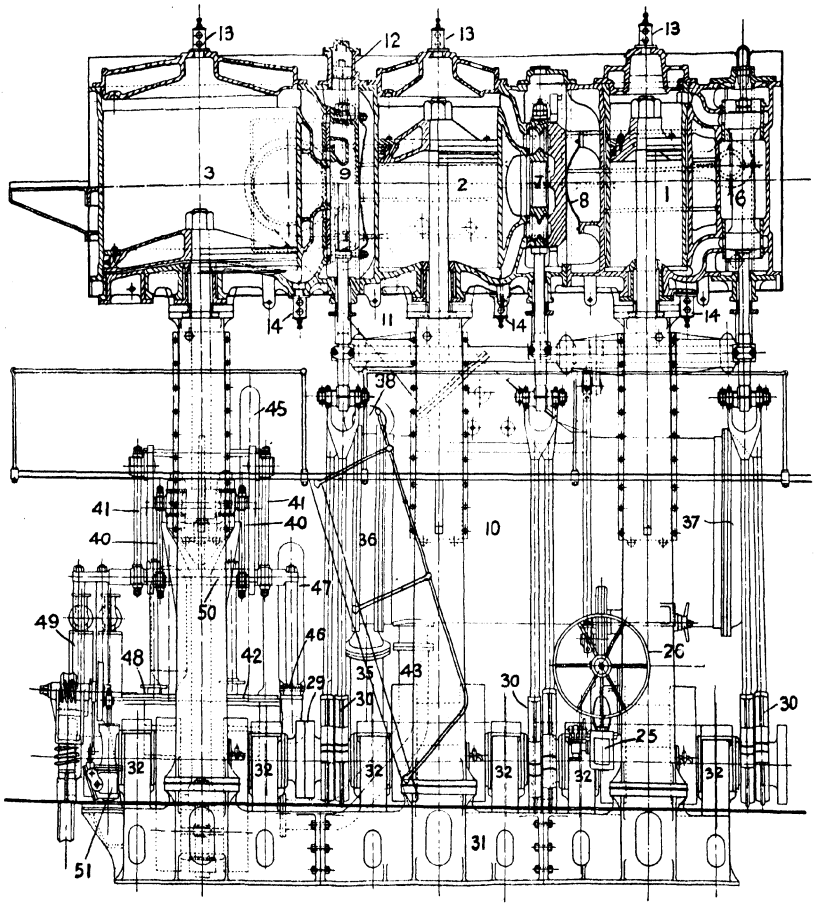
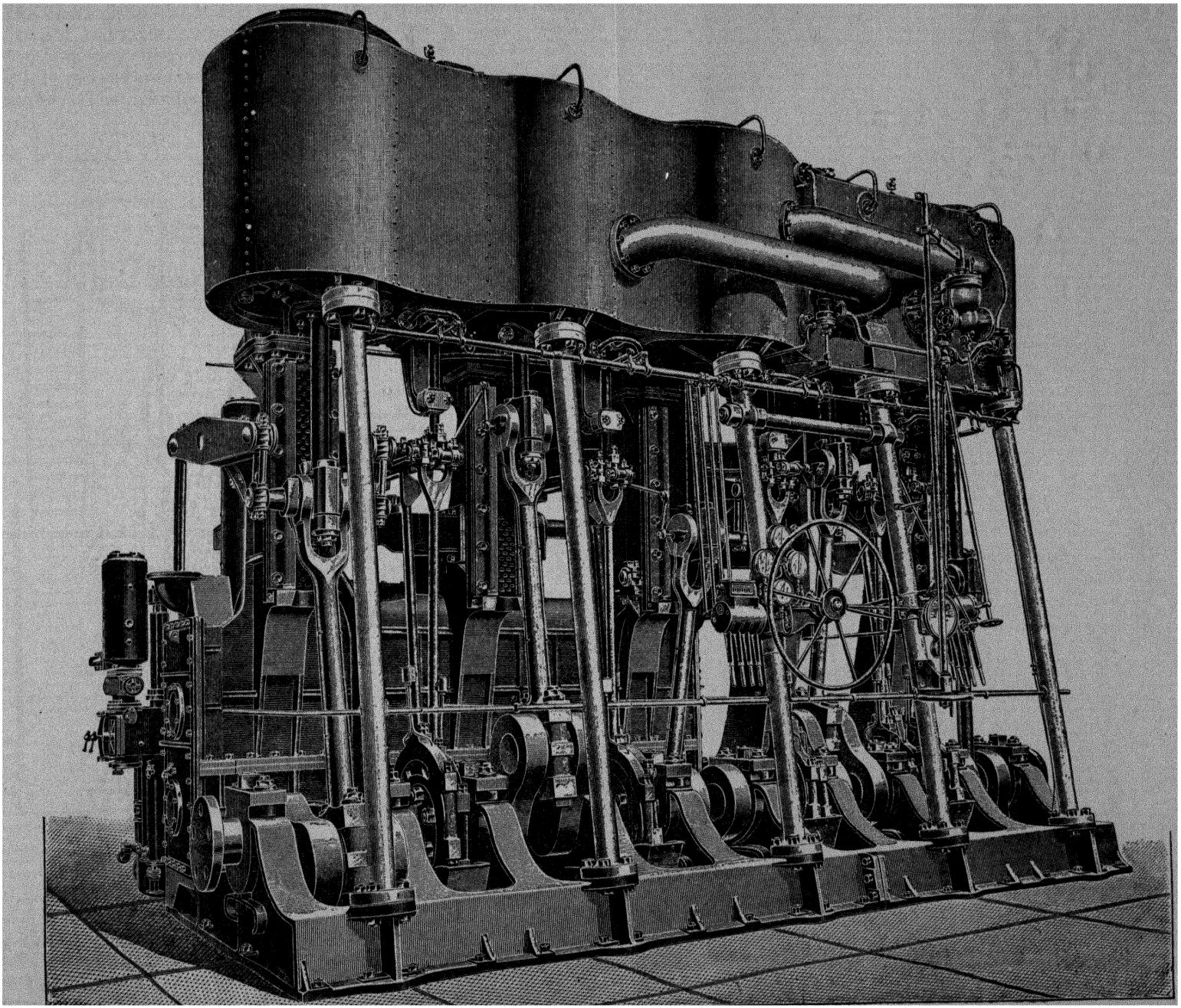



FIG. 374.

to the cylinder face by the coach spring 8. It is exhausted to a belt leading to the L.P. valve casing and admitted and exhausted at the L.P. cylinder by an unbalanced double-ported slide-valve 9. It is finally discharged to the condenser 10, through the exhaust pipe 11. The dead weight of the L.P. slide-valve and gear is counteracted by the steam pressure acting on the piston in a cylinder 12 at the top of the valve casing, Fig. 374.



[*To face page* 566.

*From "The Engineer"*]

FIG. 373.—FIVE-CRANK QUADRUPLE-EXPANSION ENGINES: S.S. "INCHMONA."

The H.P. piston-valve and each of the pistons is fitted with Lockwood and Carlisle's spring packing rings (see p. 293). Each piston is made conical in form. The H.P. piston is solid, the I.P. and L.P. pistons are made cellular to ensure combined strength and lightness. The cylinder covers are domed to suit the piston con-

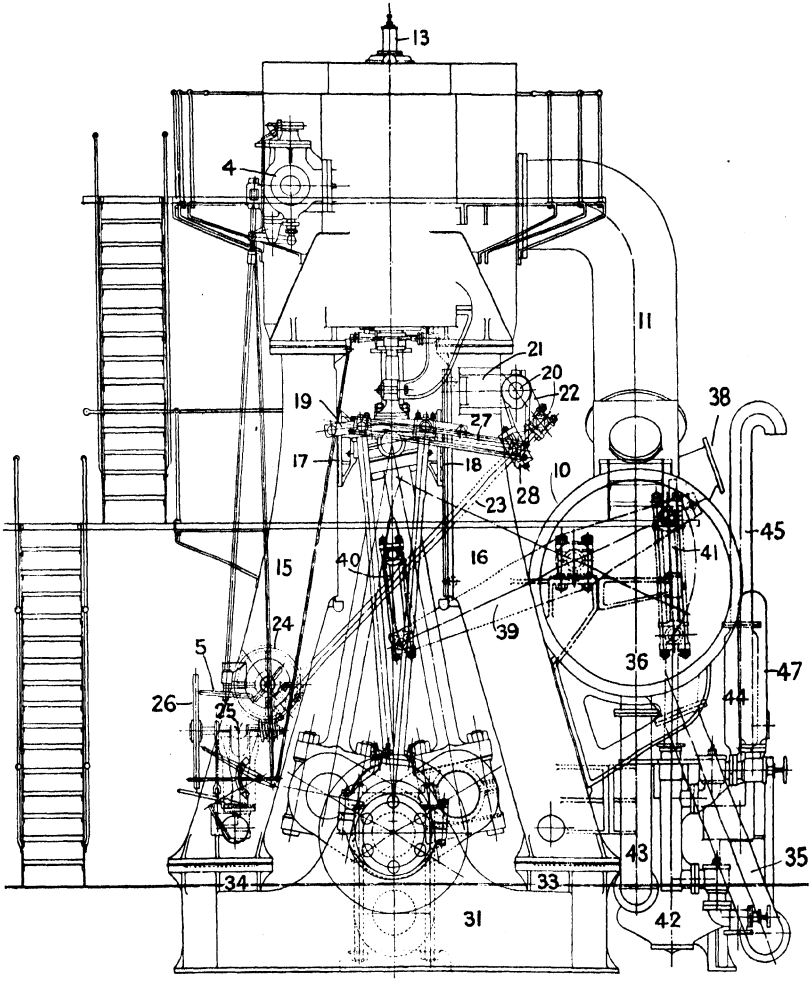



FIG. 375.

tours and to accommodate the piston rod nuts. A spring-loaded relief valve 13 is fitted on each cover, and another valve 14, Fig. 374, on the bottom of each cylinder. The piston and valve rods are fitted with United States metallic packing (see Fig. 148). Each cylinder is mounted on front and back columns 15 and 16, Fig. 375,

having plain guide faces inside to take the crosshead slippers 17 and 18. The crosshead is a solid block with gudgeon pins which take the double brasses of the forked end of the connecting rod. At the crank the rod has a palm end with double bushes lined with white metal strip, bolts, and cap. Each valve is controlled by a Stephenson link motion, having a double-bar type of expansion link 19. The reversing shaft 20 is carried along the back of the engine on three bracket bearings 21 bolted to the sides of the columns 16, Fig. 375. It is operated by the lever 22 and the reversing rod 23, which is connected to the wheel 24 of a worm gear, driven by a steam reversing engine 25. It can also be operated by the hand-wheel 26. Each drag link 27 consists of two rods carrying double brasses at the ends. At the reversing shaft end, the brasses fit on the pins of a sliding block, adjustable in the slotted end 28 of the lever, to enable the cut-off to be independently varied in the respective cylinders (see Art. 178).

The crank shaft is in three parts, which are connected by flange couplings 29 having fitted bolts. It is of the built-up type, the crank webs being forced on to the shaft and crank pins. The cranks are set at 120°, so that a fairly even turning moment is obtained. The three sets of eccentrics 30 are keyed on the shaft, close up to the bearings, the span of the bearings being kept as small as possible. The box bedplate 31 is cast in three sections which are bolted together, and is fixed to the framing of the ship. Each of these sections has two bearing housings 32 cast on it, to take the bearing brasses and cap. The end thrust of the propeller is taken by a thrust block of the Michell type, bolted to the ships framing, aft of the engine and connected to the crank shaft by a flange coupling. The back and front columns are bolted to entablatures 33 and 34 cast on the bedplate, Fig. 375. The condenser 10, supported on brackets at the back, is a cylindrical steel shell with brass tube plates and tubes. It is of the two-pass type. Circulating water, drawn through a sea cock on the ship's side by the circulating pump, is discharged through the pipe 35, Fig. 375, to the cast-iron water box 36. It flows through the lower section of tubes to the water box 37, Fig. 374, and then back through the upper set, and is discharged overboard through a discharge pipe connected to the outlet branch 38. There are five pumps driven from the L.P. crosshead by the double plate rocking lever 39 and links 40 and 41, Fig. 375. The Edwards valveless bucket air pump, 42, Fig. 374, to which the condensate flows by gravity from the bottom of the condenser by the pipe 43, lifts the water and associated air and vapour and discharges them to the hot well 44, at the top, and the air and vapour escape by the vent 45. The condensate is then drawn from the hot well by a pump 46, Fig. 374, and discharged to the feed tank, from which it is pumped to the boiler by a Weir feed pump. An air vessel 47 is fitted on the pump discharge. The other three pumps are a circulating pump 48 already mentioned,

and two bilge pumps 49 which keep the bilges clear of draining water.

All the pump plungers are directly connected to the cross-bar 50 of the rocking beam 39. A steering-gear engine is fitted at 51, Fig. 374.

**305. The Bauer-Wach Reciprocator-Turbine Combination.**—One of the advantages possessed by the steam turbine over the reciprocator is its ability to utilise the energy of the steam at the low-pressure end of the scale, that is, down to 29·3 in. vacuum (30 in. bar.).

From the thermal point of view it has no advantages over the reciprocator—at high pressures, in fact, the high-pressure end of the steam turbine is inherently less efficient than the low-pressure end; but the turbine can be used for outputs which would be out of the question to attempt with a reciprocator. It is essentially a large-power type of prime mover. Even for moderate outputs the reciprocating condensing engine is able to utilise only comparatively poor vacuum, on account principally of the restriction of the L.P. cylinder dimensions.

In the early development of the marine steam turbine, before the application of reduction gears, combinations of reciprocators and exhaust steam turbines were tried in a number of high-speed vessels, but the system was discontinued when the small high-speed geared marine turbine was evolved.

In these cases the turbines drove separate shafts, were directly coupled, and of large dimensions on account of the low rotational speed of the propeller shafts, necessary for efficient propulsion. Such arrangements were economically impracticable for the slow-speed single-screw type of cargo vessel in which, till within the last five years, only a triple reciprocator or a high-speed single or double reduction geared turbine has been employed.

A reversion to the older system has recently been made. A geared high-speed exhaust turbine is run in series with the reciprocating engine, but instead of driving on a separate shaft it is coupled to the shaft driven by the reciprocator, through the medium of an elastic hydraulic clutch, designed originally by Dr. Fottinger as a speed reducing gear for steam turbines. This is the Bauer-Wach combination already referred to. One of the special features of this arrangement is the ease with which it can be applied to existing reciprocating jobs.

It is of interest first to consider a representative case and ascertain the theoretical increase in output due to the addition of the exhaust turbine to the reciprocator.

*Example.*—A triple-expansion marine engine is supplied with dry steam at 215 lb./in.$^2$ abs. The L.P. terminal pressure is 12 lb./in.$^2$ abs., the back pressure 1·75 lb./in.$^2$ abs., and the condenser vacuum 28 in. If a steam turbine, coupled to the engine shaft, is arranged to take the reciprocator exhaust, and the same condenser

vacuum is maintained when the two work in series, find the theoretical increase in the output, over that of the reciprocator alone.

Using the method of calculation given in Art. 60, p. 81,

$p_1=215$, $H_1=1204{\cdot}2$, $\Phi_1=1{\cdot}5462$, $v_1=2{\cdot}137$
$p_2=12$, $H_2=1146{\cdot}6$, $\Phi_2=1{\cdot}7731$, $T_2=662$, $v_2=32{\cdot}37$
$L_2=976{\cdot}7$

$$\therefore\ H_{w2}=H_2-T_2(\Phi_2-\Phi_1)$$
$$=1146{\cdot}6-662\times 0{\cdot}2269$$
$$=1146{\cdot}6-150{\cdot}2=896{\cdot}4$$

Heat drop between $p_1$ and $p_2$, $H_{\phi 2}=(H_1-H_{w2})=(1204{\cdot}2-996{\cdot}4)=207{\cdot}8$ B.Th.U./lb.

Quality at $p_2$, $q_2=1-\dfrac{H_2-H_{w2}}{L_2}=1-\dfrac{150{\cdot}2}{976{\cdot}7}=(1-0{\cdot}154)=0{\cdot}846$

$v_{w2}=q_2v_2=0{\cdot}846\times 32{\cdot}37=27{\cdot}385$

Nominal ratio of expansion, $r=\dfrac{v_{w2}}{v_1}=\dfrac{27{\cdot}385}{2{\cdot}137}=12{\cdot}8$, which is a reasonable value for this class of engine.

The cycle heat of the reciprocator is—

$$H_c=H_{\phi 2}+\frac{144}{J}(p_2-p_3)v_{w2}$$
$$=207{\cdot}8+\frac{144\times 1025\times 27{\cdot}385}{778}$$
$$=207{\cdot}8+52=259{\cdot}8 \text{ B.Th.U./lb.}$$

$p_2=12$
$p_3=1{\cdot}75$
$p_4=1{\cdot}0$

The cycle heat for the combination is the heat drop between $p_1=215$ and $p_4=1{\cdot}0$. This by calculation or HΦ chart is $H_\phi=342$ B.Th.U./lb.

Increase in output$=(H_\phi-H_c)=(342-259{\cdot}8)=82{\cdot}2$ B.Th.U./lb. or 31·6 per cent. increase on the reciprocator output.

**306.** One arrangement of exhaust turbine and double reduction gear, as fitted to the triple-expansion engine of a cargo steamer, to increase the output from 4400 I.H.P. to 5500 I.H.P. or by 23 per cent., is shown in side elevation, end elevation, and plan in Fig. 376.* The forward cylinder 1 is the H.P., the middle cylinder 3 the L.P., and the after cylinder 2 the I.P.

The L.P. exhaust first passes through an oil separator 4 into a change-over valve 5, by means of which it can be discharged either to the condenser 6 or to the exhaust turbine 7. An intermediate shaft 8 connects the turbine shaft and the shaft of the first reduction pinion of the gear. The gear box 9 contains the Vulcan hydraulic clutch and the pinions and gear wheels of the double reduction gear. The speed is reduced from 3065 rev./min. at the turbine to 75 rev./min. at the propeller shaft, a reduction of 42·5. The

* Reproduced from paper "Recent Developments of the Exhaust Steam Turbine," by Dr. Gustav Bauer, *Trans. Inst. Engineers and Shipbuilders in Scotland*, December 1927. See also "Exhaust Turbine Equipment S.S. *Britannia*," *Engineering*, 28 September, 1928.

propeller shaft 10 is coupled to the shaft carrying the large gear wheel.

A sectional elevation and plan of the turbine and gear equipment is shown in Fig. 377. The turbine 7 is of the straight Parsons reaction type. The first motion pinion 11, connected to the turbine

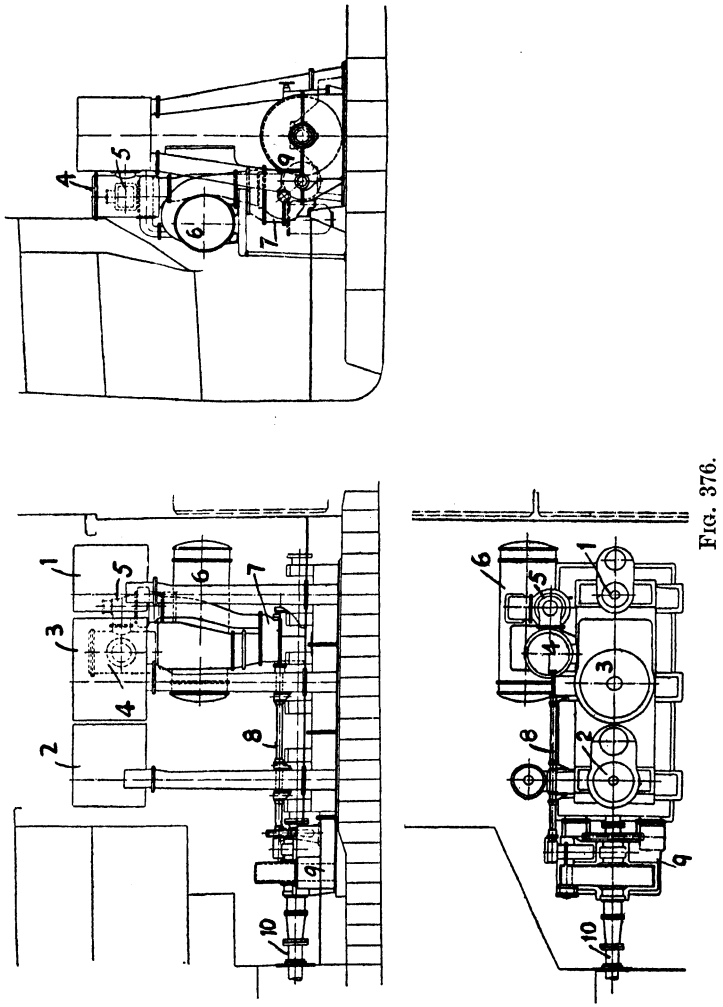



FIG. 376.

shaft by the intermediate shaft 8, gears with the first reduction wheel 12, which forms the outer rim of one half 13 of the hydraulic clutch. This combination is keyed on the inner shaft 14, which is supported on the outer bearings 15 and 16. The other half 17 of the clutch is keyed on a hollow shaft or sleeve 18, which carries the

second reduction pinion 19. This shaft is supported in the frame bearings 20 and 21. The support of the coupling halves with their

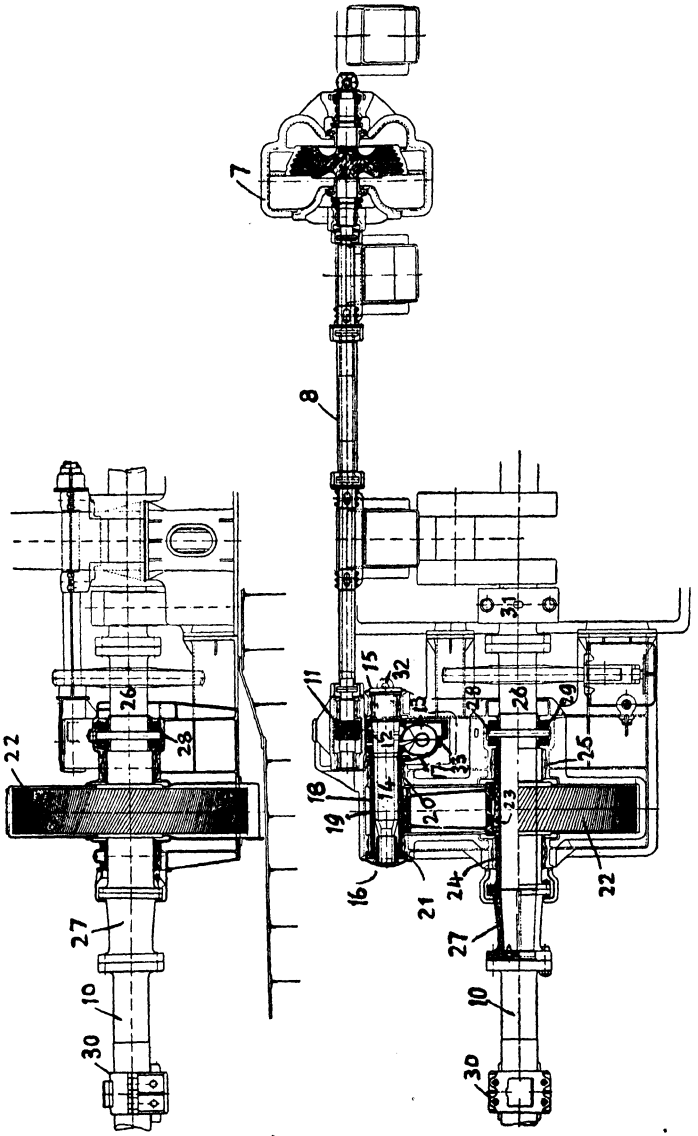



FIG. 377.

gear wheels and pinions is therefore very rigid. The second reduction pinion, gears with the large wheel 22. This wheel in turn is keyed to the hollow shaft or sleeve 23, supported on bearings 24

and 25. The crank shaft of the reciprocator is connected to an intermediate length of shaft 26, coned at the after end and having a coned sleeve 27 keyed on to it. This cone sleeve is bolted to the flange of the hollow shaft 23, and to the flange of the propeller shaft 10. If it is desired to permanently disconnect the exhaust turbine and gears, the bolts can be removed at the coupling between the cone sleeve 27 and the hollow shaft 23.

The end thrust of the propeller and gear is taken by a thrust block consisting of a single collar 28 on the intermediate shaft 26 and a bearing 29. The forward bearing of the propeller shaft is situated at 30. The span between it and the other bearing, supporting the forward end of shaft 26, is sufficiently great to permit the shaft a small amount of bending play if the main shaft bearing 31 wears down slightly. In this event the shaft 26 can move vertically, relatively to the hollow shaft 23, without detrimental effect on the bearings 24 and 25 of the main gear wheel.

This arrangement works very satisfactorily in service. The exhaust turbine is fitted to increase the output when the vessel is continuously "steaming ahead." In narrow water and harbours, where manœuvring is frequent, the piston engine is most suitable for the service, and the turbine has to be temporarily cut out. This operation is easily and quickly performed by means of the hydraulic clutch. The halves 13 and 17 of the clutch are fitted with a number of radial and axial blades. These blades divide each half into several cells which under normal running conditions are kept full of lubricating oil. The faces of the outer rims of the halves fit closely, but the blade edges are axially clear from $\frac{3}{8}$ in. to $\frac{5}{8}$ in. When filled up with oil the two halves turn with little shear effect, the speed of the driven half 17 being only from 2 per cent. to 3 per cent. less than the driving half 13. The coupling is supplied with oil by a pump through the pipe 32. An adjustable ring 33 fits closely round the outer circumference of the half 17 and covers a set of holes in it. When the ring is displaced to the left it uncovers these holes, the oil is then thrown out and the clutch is emptied. The connection between the exhaust turbine and the engine shaft is at once broken, and the reciprocator alone becomes operative. The turbine is thrown into gear again by the replacement of the ring 33 and admission of oil from pipe 32. It will be obvious that the exhaust must be deflected from or by-passed the turbine directly to the condenser, before the connection is broken, otherwise the turbine will "run away." This preliminary change-over can be effected in two ways, either by a change-over valve or by an arrangement of flap or butterfly valve in the exhaust passages. In the case under consideration the type of change-over valve shown in Fig. 378 is used.

The valve 1, when the turbine is shut down, is held on its seating by the spring acting on the upper side of the piston in the oil cylinder 2, as shown. The exhaust steam entering by branch 3

flows round the annular space 4 and is discharged directly to the condenser through the branch 5. The operative ring on the clutch half 17, Fig. 376, is hydraulically operated, and its servo-motor is connected up, in such a manner, with the cylinder of the change-over valve that the ring is returned to the closed position and the clutch filled with oil before the pressure oil is admitted at 6 below the piston in cylinder 2. In other words, the clutch is first thrown into gear before the exhaust steam is turned on to the turbine. The oil raises the piston and the valve 1, which shuts off the annular space, and the steam flows through the valve and the branch 7 to

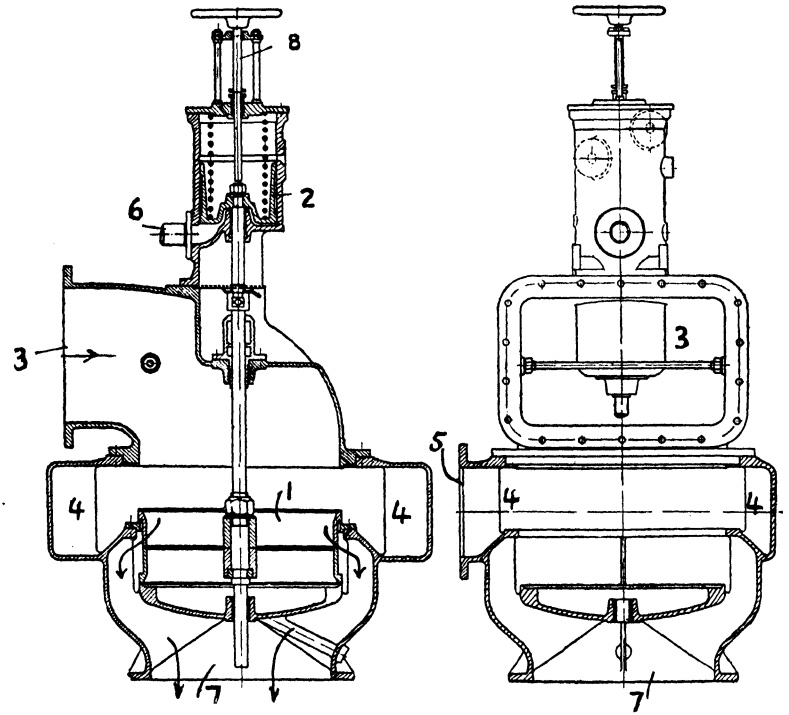



FIG. 378.

the turbine. If through any accidental cause the turbine should race, its emergency governor releases the oil from below the valve piston, and the valve is at once closed by the spring. It can be held down on the seating or limited in lift by the adjustable spindle 8.

**307.** It is of interest to note that in spite of the increased output transmitted by the shafting, no alteration of existing propeller shafting has been found necessary. The addition of the turbine very materially decreases the unevenness of the turning moment, which is imposed by the reciprocator alone. Further, the flywheel effect of the main gear wheel, the first reduction gear wheel, and the

turbine rotor, which runs at high speed, contributes materially to this improvement. It is, in fact, found that the maximum torque developed by the reciprocator-turbine combination is less than the maximum produced by the reciprocator at the same speed and output before the addition of the turbine.

The torque diagrams for one revolution, shown in Fig. 379, are reproduced from Dr. Bauer's paper, for a case where output and speed were kept the same with and without the turbine combination, so that the mean torque remained the same, that is, a value of 173,500 ft.-lb. The reciprocator alone gave the full line curve and the combination the dotted line curve. This shows a maximum torque of 274,000 ft.-lb. for the reciprocator and 246,000 ft.-lb. for the combination. The variations above and below the mean were

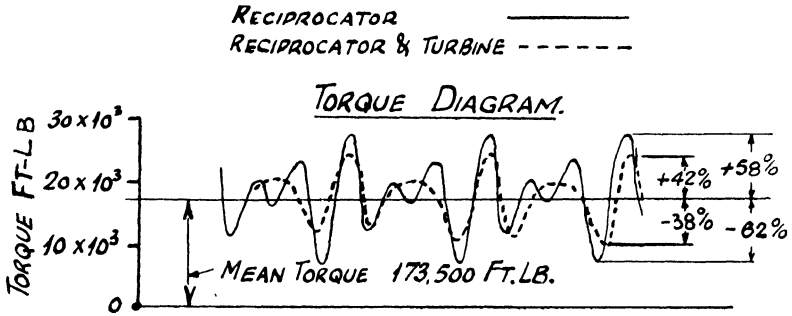



FIG. 379.

+58 per cent. and —62 per cent. for the reciprocator and +42 per cent. and —38 per cent. for the combination.

When the combination was run to give the 33 per cent. increase of output, the speed was increased from 82·6 to 89·1 rev./min., the mean torque increased to 210,000 ft.-lb., and the maximum to 260,000 ft.-lb., while the variations above and below the mean were reduced to +25 per cent. and —25 per cent.

The Parsons Marine Steam Turbine Co. are also fitting combination sets similar in arrangement to the foregoing, but in this case a special type of flexible spring coupling is used in place of the Vulcan hydraulic clutch, in order to obtain the required elasticity of drive between the steady-torque shaft of the turbine and the varying-torque shaft of the reciprocator.*

* For description and drawings of Parsons exhaust steam turbine outfit for S.S. *Kingswood*, see *Engineering*, 6 February, 1931, p. 171.

# CHAPTER XVIII

## LOCOMOTIVES

**308.** DURING the past twenty years the continually increasing demand for greater hauling capacity in main line work, especially on the Continent and in America, has taxed the resources of the locomotive engineer. Unlike the designer of marine or land power plants, he is faced with the problem of compressing a self-contained and self-moving power plant into a very restricted space.

In order to cope with the demand for increased power the steam pressure has been increased, within the past few years, from 200 lb./in.$^2$ to 300 lb./in.$^2$, then to 450 lb./in.$^2$, then to 850 lb./in.$^2$, and the latest application (in Germany) is a pressure of 1700 lb./in.$^2$

Although there has always been some difference of opinion among locomotive engineers as to the ultimate economic advantage of the compound over the simple non-condensing type of engine,* compounding is now being more extensively adopted in this country and especially on the Continent.†

The use of superheated steam has now become general for main line engines.

For the moderate steam pressures the original type of tubular locomotive boiler with rectangular firebox, but with the maximum diameter permissible by the gauge restriction and of greatly increased length, is retained. With the very high pressures now proposed this type cannot be used, and various forms of water-tube or semi-watertube boiler are being tried.

The Stephenson gear has in a large measure been superseded by the Walschaert, which besides giving a constant lead at all settings is very much lighter than the Stephenson.

Piston valves are generally used instead of slide valves, which are unsuitable for superheated steam. Other gears such as the Lentz and the Caprotti, which are light and compact and enable four double-beat valves to be operated, with resulting improvement on the cut-off and compression conditions which obtain with a single slide or piston valve in " linked-in " conditions, are now being employed to some extent.‡

* See discussion on paper " Compounding and Superheating in Horwich Locomotives," by George Hughes, *Proc. Inst. Mech. E.*, March–May, 1910.

† See " Recent Railway Locomotive Work in France," by Lord Monkswell, *Engineering*, 29 Jan. and 12 Feb., 1932; also 19 and 26 April, 1929.

‡ See footnote, p. 396.

In this chapter descriptions and illustrations of five of the latest types of main line locomotives, British and Continental, are given.

For more fully detailed drawings the student should refer to the articles quoted in the footnotes on these engines, as restriction of space does not admit of fuller discussion. A great number of designs and drawings with discussions on all branches of American locomotive construction are given in the *Locomotive Cyclopedia of American Practice*, issued in 1927 by the Railway Master Mechanics Association.

**309. L.M.S. Three-cylinder Engine "Royal Scot" Locomotive.** —A longitudinal section of a simple expansion three-cylinder express passenger engine designed by Sir Henry Fowler for the L.M.S. Railway is shown in Fig. 380, and two end elevations, partly in section, are shown in Figs. 381 and 382.* A photograph of the engine with tender is shown in Fig. 383. It is of the 4–6–0 type. The principal feature is the large boiler with Belpaire type of firebox, which, as shown in Fig. 380, is reduced in width from 5 ft. 9 in. at the top to 4 ft. $\frac{1}{2}$ in. between the frames. The length of the box between the frames is 10 ft. 3 in. This long box renders cleaning of the fire a difficult matter, and what is termed a "drop grate" is provided. As can be seen, the fire grate bars are given a considerable slope downwards, and one of the bars can be tilted by means of a rod worked from the cab, so as to provide an opening through which the ashes can be discharged. The boiler barrel is 5 ft. 9 in. in diameter and 14 ft. $\frac{3}{8}$ in. long, and the length between tube plates is 14 ft. 6 in. There are 27 large tubes $5\frac{1}{8}$ in. external diameter, which take a series of superheater elements. These tubes are reduced in diameter at the firebox tube plate

There are in the lower part of the boiler 180 small tubes 2 in. external diameter. The ends of these tubes are clearly shown on the right of Fig. 381. Each superheater element consists of a length of seamless steel tube, which passes into the large tube at the bottom, is carried along the tube, then bent up and carried back to the outside of the smoke box tube plate, bent horizontally in a wide loop, then passed along the tube again, bent upward, as indicated by the dotted curves, and returned to the smoke box end. There are thus four lengths of superheater tube subjected to the action of the hot gases flowing through the large tube. The inlet end of the superheater tubing is connected with a cross distributing box, shown in section in Fig. 380, which is supplied with steam from the saturated-steam pipe leading from the double-beat regulator valve placed in the dome. This regulator, as can be seen, is operated through the long rod carrying a lever and link. This rod is rotated by the lever in the cab, shown inclined on the left side of Fig. 382. Saturated steam, which normally carries a certain amount of moisture, passes from the main steam pipe when the regulator is

* Reproduced by permission from *The Engineer*, 7 October, 1927.

opened, and is first dried and then superheated and passed back through the return branches of the superheater tubing to the lower

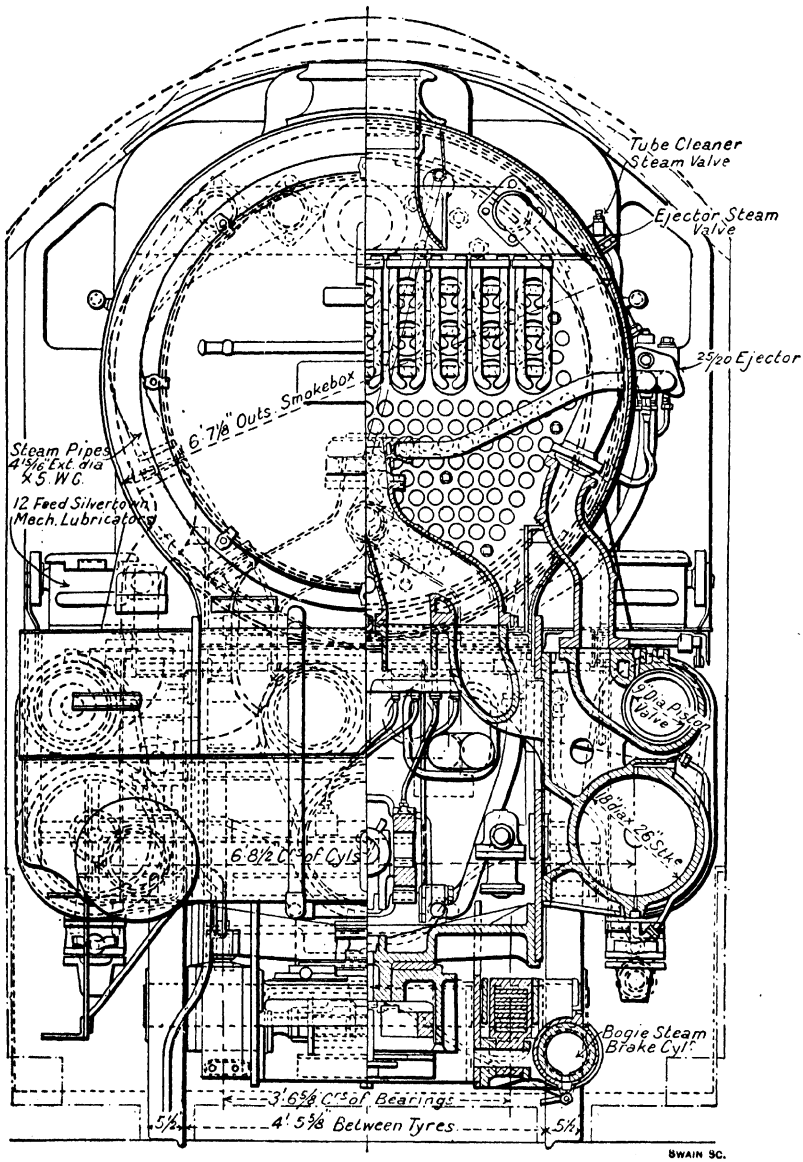



FIG. 381.

portion of the cross distributing box, shown in section in Fig. 380. From this it is supplied to the cylinders by the distributing steam

pipes, clearly shown in Fig. 380 and on the end elevation, Fig. 381. On exhaust from the cylinders the steam is passed up the inclined

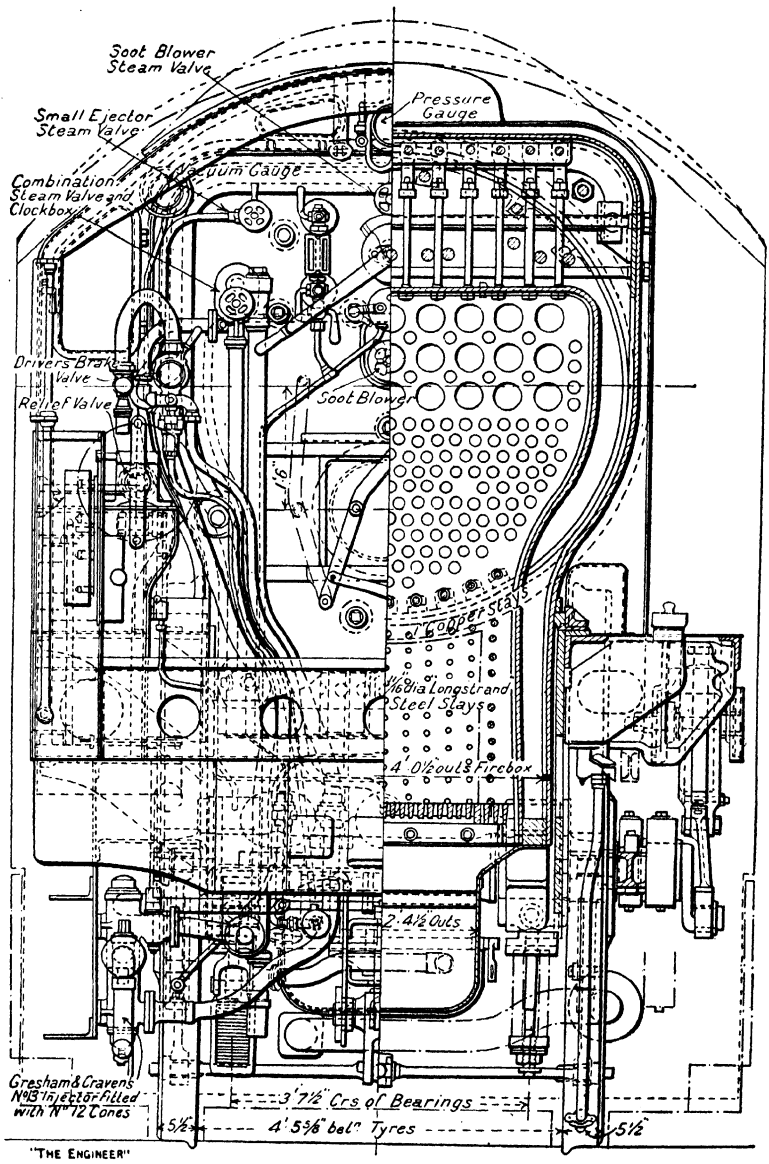



FIG. 382.

exhaust branches to the central blast pipe, shown in the end view, Fig. 381, and also in Fig. 380.

The grate area of the boiler is 31·2 ft.$^2$ ; and the heating surfaces are—

| | |
|---|---|
| Tubes | 1800 ft.$^2$ |
| Firebox | 189 „ |
| Superheater | 445 „ |
| Total | 2524 ft.$^2$ |

The working pressure is 250 lb./in.$^2$ gauge.

There are three cylinders, each 18 in. diameter by 26 in. stroke. The middle cylinder, placed between the frames and shown in section in Fig. 380, is right at the front above the leading axle of the bogie, and drives on the first axle. The other two cylinders are placed outside the frames, as shown in Fig. 381, above the bogie, and drive on the wheels of the second axle. Each cylinder is fitted with a piston valve 9 in. diameter driven by a Walschaert valve gear. The valve spindles are provided with tail rods and guides. The Walschaert gear of the middle cylinder, shown by the full-lined links in Fig. 380, is driven from an eccentric fitted on the axle of the front driving wheel, to the right of the double web balanced crank (looking forward), the piston valve being similarly placed, as shown by the dotted outline in Fig. 381. The gears for the outside cylinders are clearly shown on the photograph, Fig. 383 ; and as the type has been described in detail in Art. 189, p. 376, they do not require further comment.

They are reversed by a common reversing shaft having a counterbalance lever and weight, as shown in Fig. 380, between the first and second driving axles. The shaft is operated from the cab by the long, flat reversing rod shown dotted and passing along the side of the firebox. Each horizontal lever arm of the reversing lever engages with a block in the slot of a projecting arm of the vibrating link of the outside cylinder gear, by which the link is rotated about its centre for reversal. In the case of the middle cylinder, an auxiliary shaft and lever, operated by an intermediate rod from the reversing shaft, has to be used. The auxiliary bell-crank is placed in front on the first driving axle. The arrangement is shown in full lines. Double bar guides and slipper crossheads are used, and the connecting rods are of standard locomotive pattern, with bushed crosshead end and strap end at the cranks. The usual articulated coupling rod is used to couple the three driving wheels, each of which is 6 ft. 9 in. on the tread. The journals run in solid brass axle boxes with antifriction metal lining. Each box fits between cast-steel horn blocks bolted to the frame and closed at the bottom by a clamping strap.

The load is transmitted to each of the first two axles by two carrying springs, threaded on suspension rods from the shell cover at the bottom of the box and bearing on washers at the ends of these. The axle of the trailing driving wheel is loaded by the underhung laminated plate spring, suspended by a link from the bottom of the

axle box and by end links to the frame. Each bogie wheel is 3 ft. 3½ in. diameter, and the load on the axle box is transmitted by the end of the equalising beam, from which the inverted laminated plate spring is suspended by links. The stirrup of the spring takes the load from the truck frame, transmits it through the spring to the bent equalising beam, shown dotted in Fig. 380, which distributes it equally between the two boxes. The bogie can swivel on the central pivot shown, and the truck is also given a side play of 4¾ in.

A steam brake cylinder is fitted below the cab foot plate. It is 10½ in. diameter and is supplied with boiler steam at 250 lb./in.$^2$ The load is transmitted through a bell-crank lever, with long arm of 18 in. and short arm of 8 in. The total pull on the two longitudinal brake rods, one of which is shown in Fig. 380, is thus 2¼ times the piston load. Each swing link carrying the brake block at the centre of its length is 30 in. long, so that the total pressure exerted on the six wheel tyres when the brake is full on is 4½ times the piston load. This amounts to a total of 43·3 tons, or fully 7 tons per brake block. A steam brake is fitted to each pair of bogie wheels. The cylinder and blocks are clearly shown in the photograph, Fig. 383. A vacuum pump, driven off the engine crosshead, is fitted below the cylinder on the left-hand side of the engine. This is clearly shown on the photograph, and is indicated in Fig. 380. This pump exhausts the air from the brake piping when the engine is running, and the air ejector is used only when the engine is standing.

There are two injectors. A Metcalf exhaust steam injector, supplied with exhaust steam from the middle cylinder, through the pipe shown dotted along the length of the engine, Fig. 380, and in full lines below the ash pan in the end view, Fig. 382, is fitted below the gangway plate at the cab on the right. This utilises the exhaust steam for boiler feeding, when the engine is running, and also acts as a feed water heater.* A Gresham and Craven high-pressure injector is fitted on the left side of the engine, as shown on the end view, Fig. 382, and is used to feed the boiler when the engine is standing, or to supplement the exhaust injector. Each injector is supplied with water by the usual hose connection from the tender, and discharges the feed to the boiler through the non-return or clack valve shown on the left in front of the firebox, Fig. 382. The air ejector for the exhaustion of the vacuum brake piping is fitted on the outside of the smoke box on the left side, and exhausts to the blast pipe through the pipe shown in the end view, Fig. 381. The driver's brake control valve in the cab is shown on the left in Fig. 382. A mechanical lubricator driven from the crosshead is fitted on the top of each gangway plate, Fig. 381, above the valve gear, and supplies oil to the cylinders and valves. Each cylinder is fitted with spring loaded relief-valves and drain cocks worked from the cab. Sanding pipes are fitted at the front

* See paper on "Feed Water Heaters for Locomotives," by Prof. E. Sauvage, *Engineering*, 23 June, 1922.

and middle driving wheels, Fig. 380. Owing to the restriction in height a specially low set spring-loaded safety valve shown above the centre of the firebox had to be fitted. Strong glass plate protectors are fitted to the gauge glasses, as shown in Fig. 382.

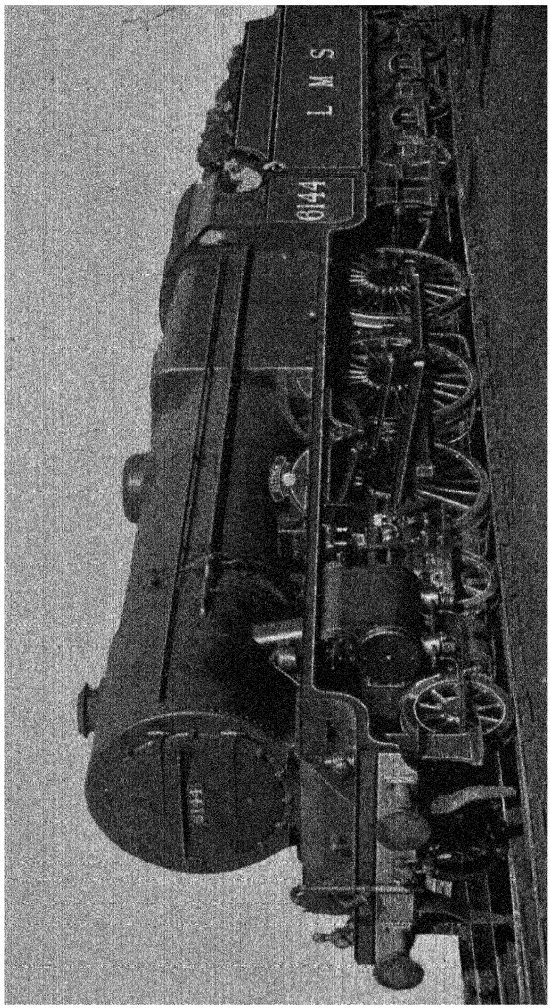



Fig. 383.

The tender, as shown on the photograph, Fig. 383, is of normal design. It is carried by six wheels, 4 ft. 3 in. diameter, by means of axle boxes with over-hung springs. A steam brake cylinder is fitted at the front end of the tender, as shown in the elevation,

Fig. 380, and operates the brake blocks on the six wheels. An emergency hand-brake gear is also connected to the brake rods. The column and hand wheel of this brake gear is shown in the elevation, at the back of the foot plate.

The tank capacity of the tender is 3550 gallons, and the coal capacity 5½ tons.

The over-all length between buffers is 63 ft. 2¾ in.; the total wheel base is 52 ft. 9¼ in.; the weight on the driving wheels 62 tons 10 cwts., and on the tender wheels 42 tons 14 cwts. The total weight of engine and tender in working order is 127 tons 12 cwts., and the tractive effort is 15 tons.

**310. L. & N.E. Railway High-pressure Compound Locomotive.**—One of the most recent developments in this country, made with a view to improved efficiency, is the four-cylinder compound designed by Mr. Gresley of the L. & N.E. Railway. A skeleton drawing of this engine is shown in Figs. 384 and 385,* and a photograph in Fig. 386.† The engine, which is intended for express passenger service, is of the 4–6–2–2 type, that is, there are four bogie wheels in front, six coupled driving wheels, and two pairs of trailing wheels. The tender has four pairs of wheels. The radical departure from standard practice is the boiler, which is a modified Yarrow watertube, the outline of which is shown in dotted lines. It consists of a steam drum 1, carried on a cast-steel cradle, two water drums 2, which are situated outside the two trailing wheels (see Fig. 386), extending the length of the firebox, and two longer water drums 3, which are placed inside the frames in the way of the two after driving wheels. Each water drum is connected with the steam drum by bent tubes. These four small drums are not supported but hang from the water tubes. The space between the water drums 2 is provided with the usual firebox grate and ash pan, and there is a brick arch at the forward end of the grate.

The gases from the firebox pass over this arch and flow into the central passage between the tubes. In this passage they are so baffled that they flow from it across the tubes, in their forward passage, into the outer flue formed between the tubes and the outer wall. An air space is formed between this wall and the outer casing, through which the air, taken in through openings at the smoke box end, is passed to the ash pan, and thence through the furnace bars. By this arrangement the air for combustion is preheated and the outer casing kept cool. The side flues merge into a single flue at the forward end of the water drums 2, and a superheater is fitted in this flue and connected to the steam drum. The following are the leading dimensions of the boiler:

Steam drum 3 ft. diameter by 27 ft. 11⅝ in. long; each firebox

* Reproduced by permission from *Engineering*, 27 December, 1929. See also *Engineering*, 30 January and 13 March, 1931.

† See paper "High Pressure Locomotives," by H. N. Gresley, C.B.E., *Proc. Inst. Mech. E.*, January 1931, for more complete details and drawings.

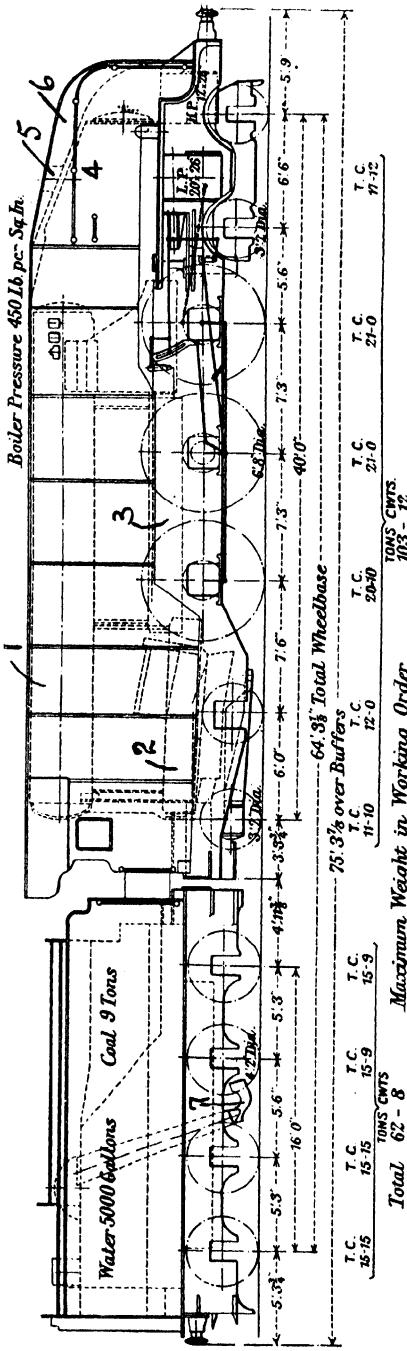



FIG. 384.

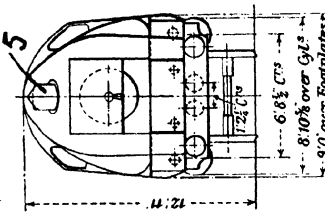



FIG. 385.

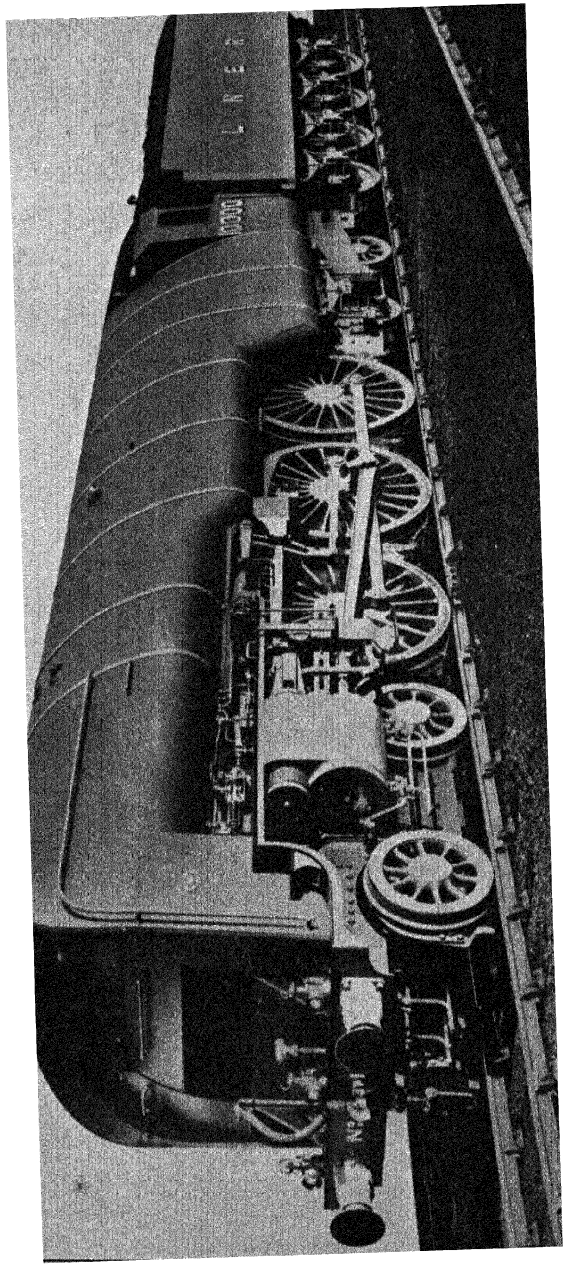



Fig. 386.

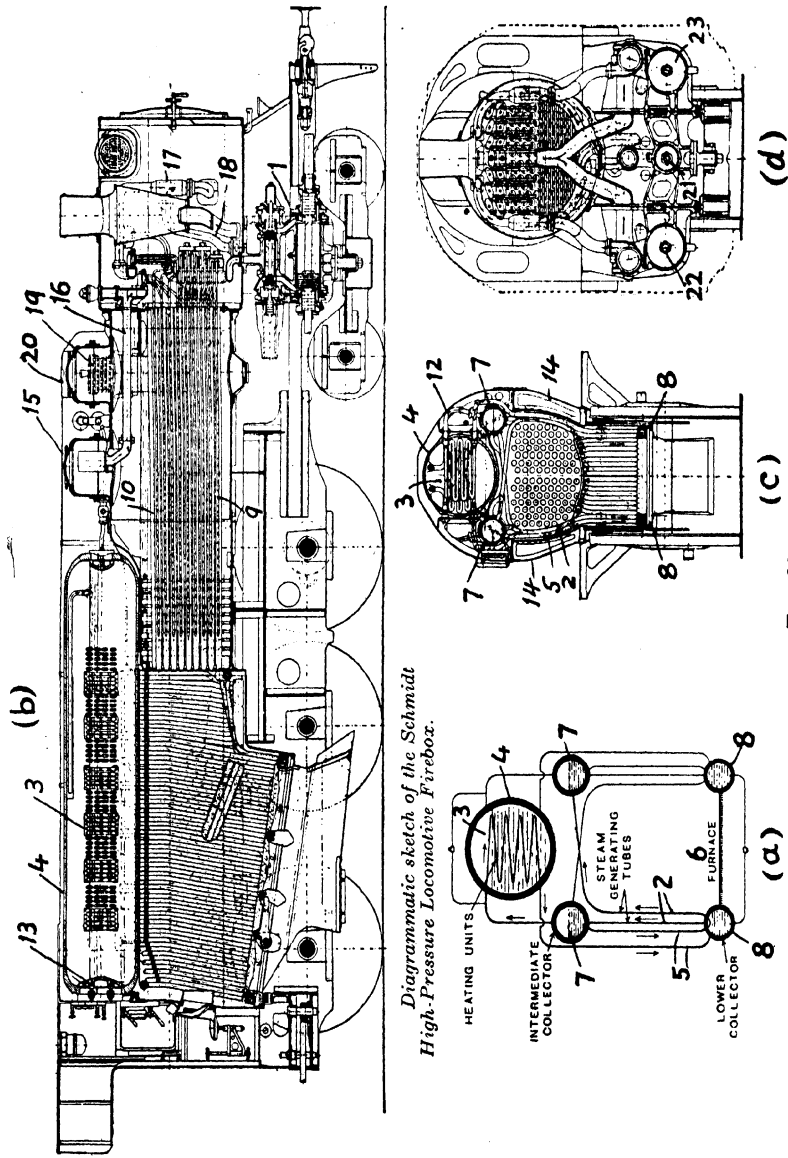
water drum 18 in. diameter by 11 ft. $\frac{5}{8}$ in. long ; each forward water drum 19 in. diameter by 13 ft. $5\frac{3}{4}$ in. long.

The firebox water drums are connected to the steam drum by 444, 2-in. and 74, $2\frac{1}{2}$-in. tubes, and the forward drums by 238 $2\frac{1}{2}$-in. tubes. The headers and the steam drum, at the front of the firebox, are connected by 12 $2\frac{1}{2}$-in. diameter tubes. The total evaporative surface is 1986 ft.$^2$ All the drums are of forged steel and are machined all over. The working pressure is 450 lb./in.$^2$ and the boiler is capable of evaporating 20,000 lb./hr. A high-pressure steam injector is used for the supply of the feed. The feed is supplied to the forward part of the steam drum, which, as can be seen in Fig. 384, projects well in front of the forward banks of tubes, and is made to pass over a weir before it can reach these evaporative surfaces. The object here is to heat the water sufficiently to cause a deposit of lime or other scale-forming matter in this part of the drum, from which it can be readily " blown down." For such high pressure the safety valves, relief valves, and regulator valve are of special design to withstand the abrading action of high-temperature steam. The main regulator, situated on the cylinder side of the superheater, supplies the high-pressure cylinder steam chests, but there is also an auxiliary valve by which reduced steam can be supplied directly to the L.P. cylinders, for starting. These cylinders are provided with relief valves which blow off at 200 lb./in.$^2$ All the auxiliaries are supplied with reduced steam at 200 lb./in.$^2$

The two high-pressure cylinders, 10 in. diameter by 26 in. stroke, as shown in the end view, Fig. 385, are placed above the leading bogie wheel between the frames, and drive on the front coupled wheel axle. The L.P. cylinders, 20 in. diameter by 26 in. stroke, are placed outside the frame, above the bogie, and drive on the mid coupled wheel axle. The H.P. cylinders are placed close together, the distance being $14\frac{1}{4}$ in. centre to centre. This arrangement permits the use of a double crank with a central web, which leaves space for large bearing axle boxes. The cylinder chests and receiver are all in one piece and cast in steel, and the barrels are fitted with cast-iron liners. Piston valves are used throughout, and are driven by Walschaert gears. Each gear (Fig. 386) is fitted, in the usual way, outside the frame to drive the L.P. valve, and its motion is transmitted to a rocking shaft the inner arm of which is connected to the valve of the corresponding H.P. cylinder. This arm is provided with a slot and a sliding block to which the valve spindle is jointed. A steam reversing gear is used for the Walschaert gear and an auxiliary steam adjusting gear is used to operate the sliding blocks of the rocking shaft lever. In this way the H.P. cut-off can be controlled independently of the L.P. cut-off.

The H.P. cylinders have piston valves 6 in. diameter, and the the L.P. have valves 8 in. diameter, the travel being $8\frac{9}{16}$ in. for the H.P. and $6\frac{11}{16}$ in. for the L.P. The H.P. valves have $1\frac{3}{8}$ in. steam lap, and the L.P. valves $1\frac{5}{8}$ in. There is no inside lap. The full gear

cut-off is 90 per cent. for the H.P. and 75 per cent. for the L.P. cylinders. The driving wheels are 6 ft. 8 in. diameter, and the front bogie and trailing wheels 3 ft. 2 in. diameter. The tender is of the




*Diagrammatic sketch of the Schmidt High-Pressure Locomotive Firebox.*

FIG. 387.

usual type with 8 wheels 4 ft. 2 in. diameter. It has a tank capacity for 5000 gallons of water, and during long non-stop express runs it is replenished at intervals from the water troughs on the line by

means of the scoop 7. The coal capacity is 9 tons. The total load taken by the driving wheels is 62 tons; the load on the tender wheels is 62 tons 8 cwt., and the maximum load of the whole engine in working order is 103 tons 12 cwts. The boiler casing has been carried up to the maximum limit permitted by the loading gauge height, and all the usual outstanding parts—funnel steam domes, safety valves—are invisible. Further, the smoke box is made of a tapering form as can be seen at 4, Fig. 384. The rudimentary funnel 5, which is left, is surrounded by the side screens 6, so that the rush of air past these may lift the steam and smoke well above the level of the cab and give the driver an unobstructed view ahead.

**311. Schmidt High-pressure Locomotive.**—Another experimental engine, working at the much higher pressure of 852 lb./in.$^2$ gauge, which has recently been tried out on the German State Railways, is illustrated in Fig. 387.* It is an ordinary three-cylinder simple engine converted into a compound by fitting a suitable H.P. cylinder 21 between the frames and the low-pressure cylinders 22 and 23, Fig. 387 (*d*), outside them. The cylinders have piston valves. The H.P. piston, which is subjected to the high-pressure steam, is a pressed steel block carrying six narrow cast-iron rings, and the deep packing gland shown on the section (*b*) has also six cast-iron packing rings. The other details of the engine mechanism and fittings follow the usual locomotive practice. As in the previous case, the special feature is the high-pressure boiler, which is of the Schmidt indirect heating type. For such high pressures it is absolutely essential that pure water should be used, and further it is most desirable that the drum containing the high-pressure steam should not be subjected to the radiant heat of the furnace, as over-heating of it might be disastrous. The Schmidt principle of indirect heating is diagrammatically illustrated in Fig. 388. A set of coils 1 filled with distilled water is connected with a set of condensing coils 3, in the drum 4, containing water, by the header 2 and the down-comer 5. The coils 1 are subjected to the radiant heat of the boiler furnace 6. Steam is generated in the lower coils, passes up the header to the coils in the drum, where it is condensed, and the

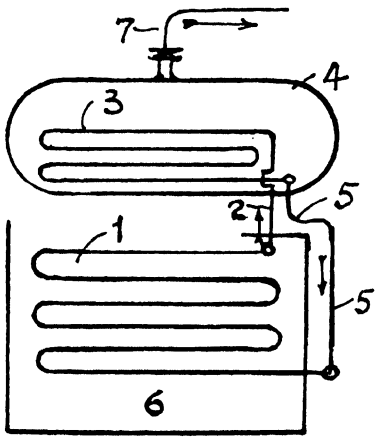


FIG. 388.

* Reproduced from *Proc. Inst. Mech. E.*, December 1927. Discussion on "Some Experimental Results from a Three-Cylinder Compound Locomotive," by L. H. Fry.

condensate passes back through the downcomer, which is not exposed to heat. The latent heat of the steam evaporates the water in the drum, and the steam produced is taken away by the outlet valve 7. The system is self-contained, and the quantity of water does not alter except when leakage occurs. As the heating system continuously receives heat from the furnace and as it is entirely enclosed, the pressure in it rises higher than the pressure maintained in the boiler drum. The difference between the pressures provides a sufficient drop of temperature for effective heat transmission between the water and coil.

A pressure of about 1360 lb./in.$^2$ gauge in the condensing coil 3 corresponds with a pressure of about 850 lb./in.$^2$ gauge in the drum. The closed system, in fact, constitutes a combination of super pressure boiler and high temperature condenser. An outline of the arrangement fitted on this engine is shown in Fig. 387 (*a*). There are two lower water drums or collectors 8, two headers 2, one leading to an intermediate collector 7 directly and the other diagonally to the opposite collector. There are two downcomers 5, one from the condensing coil 3 in the boiler drum and the other from the intermediate collector, to the lower collectors 8. The diagonal headers form a tubular furnace crown. In order to minimise the risk from leakage or explosion, the coils 3 in the boiler drum were originally arranged in six sets, as shown in Fig. 387 (*b*), the collecting chambers and tubes being similarly divided. The bottom collectors 8, Fig. 387 (*c*), are arranged round the grate so as to form a rigid frame. Each chamber consists of a steel forging bored out. The heating tubes extend upward from these and form the walls of the combustion chamber. The tubes are rolled into the chambers, not welded.

The two upper collecting chambers 7, which take the side wall tubes, are 11 in. diameter. The drum 4 is 3 ft. in diameter and 16 ft. 11 in. long. It is made of 3 per cent. nickel steel, forged over a steel mandrel. The thickness is $1\frac{3}{16}$ in.

This watertube furnace and the boiler drum replace the usual firebox of the locomotive boiler, but its barrel 9, Fig. 387 (*b*), is retained separately with the usual set of smoke tubes 10. This cylindrical portion acts as an ordinary locomotive low-pressure boiler, steam being generated in it at 200 lb./in.$^2$ gauge. The heat exchanger coils are inserted through the manhole 13 at the front of the drum, Fig. 387 (*b*). This element is supported on the angle frame 14, Fig. 387 (*c*). High-pressure steam at 850 lb./in.$^2$ gauge is passed from the drum to a set of thirty superheater elements fitted in the lower smoke tubes. These elements lead into a collecting chamber from which the high-pressure cylinder is supplied at this pressure and a temperature of 750° Fahr. The low-pressure steam, drawn from the dome 15 through the steam pipe 16, Fig. 387 (*b*), is passed through a series of fifty-six superheater units in the upper smoke tubes, and through the inlet and outlet compart-

ments of the distributing cross chamber. From this chamber it is led by two tubes, inside the front end, at a temperature of 800° Fahr., to a chamber 17, into which the exhaust from the H.P. cylinder is discharged through the pipe 18. Here mixture takes place, the H.P. exhaust is resuperheated, and the mixture is then passed on to the L.P. cylinder, at about 700° Fahr. The cold feed is pumped to the low-pressure boiler through a feed water heater and sprayed over V-shaped trays 19 in the dome 20. The water is thus quickly heated to a temperature above which scale is precipitated. This collects in the trays and can be removed. Water from this boiler, in a relatively pure condition, is pumped by a high-pressure pump to the high-pressure drum, which is thus kept clear of scale and sediment. After four months' running only fine loose dust was found in the drum.

It was also found in the road tests that varying pressures were developed in the H.P. heater elements, and these were replaced by one large element, extending over the whole length of the firebox. The system of intermediate superheating of the H.P. exhaust avoids the discharge of oil into the H.P. superheater elements, and assists in the lubrication of the L.P. cylinders. The superheated steam is finally expanded and discharged from the L.P. cylinders to the blast pipe, as indicated by the directional arrows in Fig. 387 (*d*). Recent particulars of the dimensions and road performance of this high-pressure development are given in a paper by R. P. Wagner, Chief Designer of the German State Railway, in the official journal of the American Society of Mechanical Engineers.* It has now been found from further trials that the gain in economy over that of the ordinary compound is too small to justify this design, and it is being modified to obtain a larger amount of H.P. steam, the H.P. boiler being much enlarged and the L.P. reduced to a preheater, while a resuperheater is being fitted.

**312.** More recent attempts to apply high-pressure steam to the locomotive have been made by the Swiss Locomotive and Machine Co., Winterthur, and by L. Schwartzkoff and Co., Berlin.

The Swiss locomotive has a watertube boiler working at 850 lb./in.$^2$ gauge. This boiler consists of a solid drawn-steel steam drum at the top, and two small water drums at the bottom. These are connected together by water space walls, similar to those of the ordinary locomotive boiler firebox. The space between these walls constitutes the furnace, and the walls are pierced by short tubes which allow the furnace gases to pass through. Each water tube element consists of two vertical tubes fixed at the bottom to the water drums and at the top to a cross tube which is connected to the steam drum at the centre line by a T piece. A superheater is placed in front of the water space, and in front of this is placed the feed heater. These fittings fill up the forward space between the watertube walls. The feed is preheated by exhaust steam to

* "The Schmidt High-pressure Locomotive of the German State Railway," *Mechanical Engineering*, December 1928, p. 921.

180° Fahr., and pumped through the feed heater, in which it is raised to 450° Fahr. The major part of the solid matter and scale is thus deposited in the heater, and very little gets through to the tubular elements. The boiler has a grate area of 14·4 ft.², a heating surface of 215 ft.², and a capacity of 594 gallons.

The engine, which is placed above the frame in front of the boiler, has three cylinders with cranks at 120°. It is of the high-speed uniflow type, each cylinder being $8\frac{1}{2}$ in. diameter by $13\frac{3}{4}$ in. stroke. It is geared to a jack shaft by a $2\frac{1}{2}$ to 1 flexible reduction gear. This shaft has outside cranks from which the drive is transmitted, by connecting rods, to the sets of six coupled wheels. Each driving wheel is 60 in. diameter.

This locomotive has a water tank capacity of 1320 gall., a coal bunker capacity of 2·7 ton. The weight in working order is 90·8 ton, and it develops 800 D.H.P. at 50 miles/hour.

313. The Schwartzkoff locomotive is more complicated still than the Schmidt and the Swiss engines. In this case a Loeffler boiler working at 1700 lb./in.² is used. In the Schmidt system distilled water is used in a closed circuit in conjunction with a heat exchanger coil in the secondary pressure drum. In the Loeffler system highly superheated steam is pumped through the water in the steam drum and its superheat is used to evaporate the water into saturated steam at 1700 lb./in.² gauge. This saturated steam is drawn off by the steam circulating pump. About 25 per cent. of it passes to the H.P. cylinders ; the rest is forced back through the water in the drum for the purpose of evaporation. The saturated steam is raised to the high superheat temperature in a radiant superheater coil placed in the furnace. This is briefly the principle of action of the Loeffler system of steam generation. The high-pressure and high temperature steam which is passed to the H.P. cylinders, of which there are two, is exhausted to what may be termed the L.P. boiler, which plays the same part as the heat exchanger in the Schmidt boiler, only the steam generated in this drum has a low pressure of 225 lb./in.² gauge.

On its way to the L.P. boiler coils this H.P. exhaust is passed through on oil separator and then through a feed heater. The condensate from the coils is then passed through a high-pressure feed heater, and any leakage of steam in the H.P. system is made up by water drawn from the L.P. boiler. The low-pressure steam, at 225 lb./in.² gauge, is passed through a superheater and raised to about 640° Fahr. It is supplied to the single low-pressure cylinder between the frames, expanded, and discharged to the atmosphere, in the ordinary way, through the blast pipe.

In addition to supplying the L.P. cylinder, the low-pressure boiler supplies steam for the operation of all the pumps, which are in duplicate, as any failure of the circulating system would prove disastrous. The pumps used are (1) the circulating steam pump ; (2) the feed water pump for the evaporative drum supplying the

H.P. condensate at 1700 lb./in.$^2$ gauge; (3) the pump supplying the evaporative drum with make-up feed from the low-pressure boiler. Each pump is capable of delivering 75 per cent. of the maximum requirements of the boiler and takes 4 per cent. of the power developed.

The air is preheated by the escaping gases to 300° F. before it is supplied to the furnace.

The H.P. superheater has a surface of 970 ft.$^2$, the L.P. one 344 ft.$^2$ The H.P. preheater has 764 ft.$^2$ and the L.P. boiler 882 ft.$^2$ surface. Each H.P. cylinder is $8\frac{5}{8}$ in. diameter by 26 in. stroke, and the L.P. cylinder is $23\frac{5}{8}$ in. diameter by 26 in. stroke. Each of the six coupled driving wheels is $78\frac{3}{4}$ in. diameter. The weight of the engine in working order is 113 tons.

It is very questionable if these elaborate and complicated high-pressure outfits, of high first cost and involving considerably higher maintenance costs, will survive the economic test, in spite of the higher efficiency that may be obtained over that of the moderate pressure and simpler type of compound, such as the L.N.E.R. locomotive.

**314. The Turbo-Locomotive.**—The credit for the introduction of the turbo-locomotive as a working proposition is due to the brothers Ljungström, the Swedish engineers who designed the well-known reaction turbine bearing their name. In 1922 they produced a locomotive, radically different from the conventional reciprocator design, which was put to work on the Swedish State Railways. This pioneer was fully illustrated and described in *Engineering*.* The reciprocating engine was replaced by a geared impulse turbine driving, in the ratio of 22 to 1, on to the axle of the leading wheel of what in the reciprocator is the tender but in this case was the turbine and condenser carriage, with six coupled driving wheels and a trailing wheel. The front part of the locomotive, the "engine" in the case of the reciprocator, carried the usual type of locomotive boiler, with saddle type coal bunkers at the sides. The turbine exhausted to an air-cooled condenser. The air for combustion was preheated by the flue gas and the feed was heated in regenerative heaters supplied with exhaust steam from the various auxiliaries. This engine was subjected to a prolonged test on a special power testing plant before it was put into service on the road. Its reliability was thus established, and in addition valuable information regarding performance was obtained and used in the design of subsequent locomotives, one for the Swedish State Railway, one for the Argentine State Railway, and another for the L.M.S. Railway, built by Beyer, Peacock and Co.

This later and improved design has also been very fully illustrated and described in *Engineering*. An outline sketch of the complete locomotive is shown in Fig. 389† and a photograph in Fig. 390,

* *Engineering*, 21 July; 4, 11, 18 August, 1922.

† Reproduced by permission from *Engineering*, 16 and 23 December, 1927.

and these serve sufficiently for the following general description; but the student should refer to the detail drawings in *Engineering* for further information.

**315. Ljungström Turbo-Locomotive (L.M.S. Railway).**—As shown on the photo (Fig. 390) the steam generating plant is carried by the usual frames supported on a four-wheeled bogie in front and three pairs of trailing wheels, with overhung laminated springs and equalising levers. The fan-shaped structure at the bottom of the smoke box extension is a set of louvres, through which the air for combustion passes by the duct shown dotted in Fig. 389 to the closed ash pit, and from thence through the fire bars.

Above the louvres a turbine-driven induced draught fan is fitted. This draws the gases from the smoke box through an air preheater and discharges them up the chimney in the manner explained below. The small circular casting shown on the side of the smoke box extension near the bottom, carries one of two rollers on which the circular preheater rolls inside the casing. The rollers are driven by chain gear from the suction fan spindle, as shown in Fig. 389. On each side of the boiler there is a reserve water tank, and at the rear of the cab a coal bunker. Both of these are marked on Fig. 389 and clearly shown by the photograph. The pipe running along the frame below the gangway is the main steam

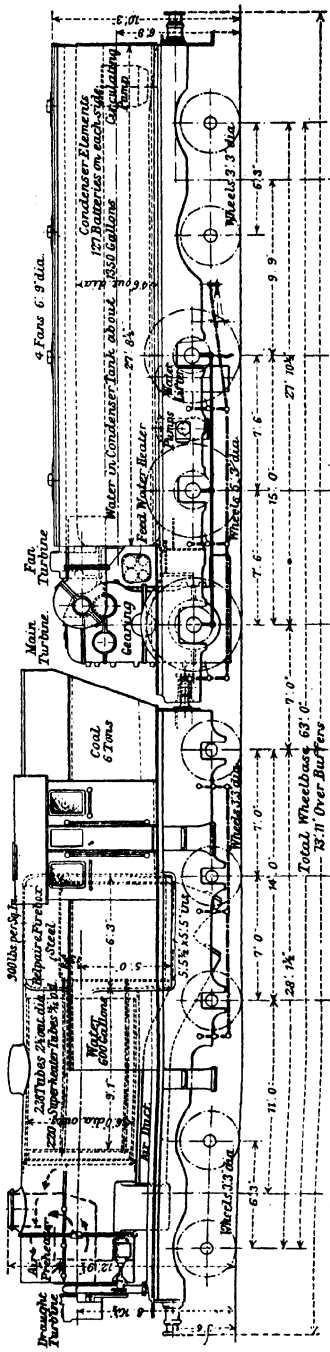



FIG. 389.

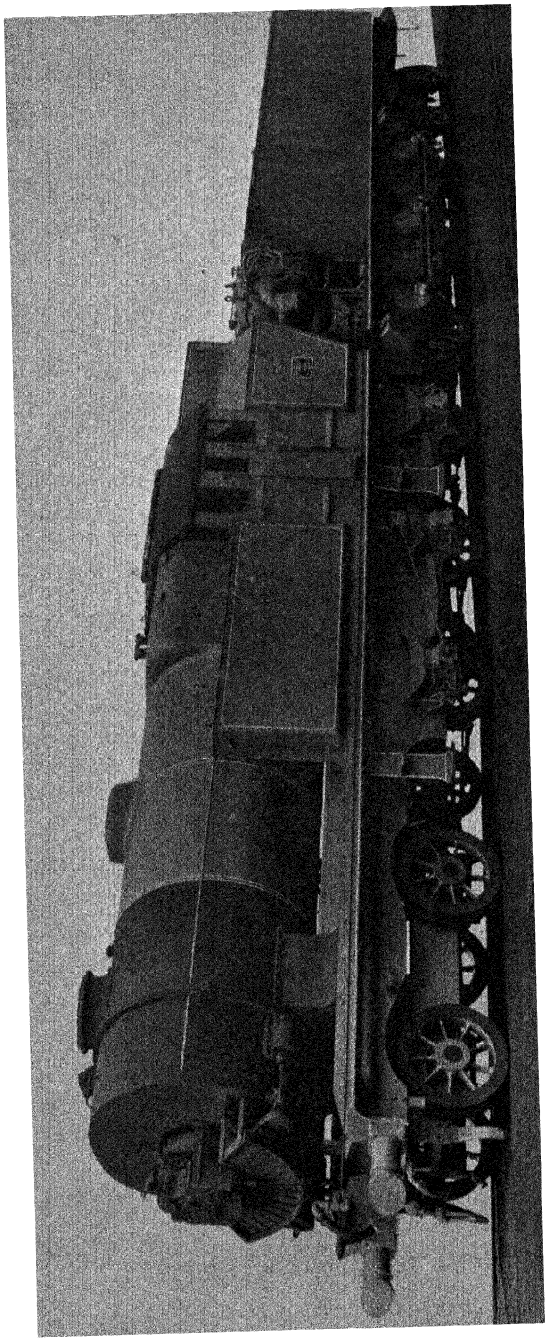


FIG. 390.

pipe leading from the superheater distribution box, in the smoke box, to the main turbine on the rear carriage. It is connected to the turbine by a ball joint. The exhaust pipe from the fan turbine runs along the other side of the engine to the feed heater, placed behind the turbine.

As shown in Fig. 389, the boiler is of ordinary locomotive type with Belpaire fire-box, and is fitted with a set of superheater elements similar to those of the piston locomotive (Fig. 380). There is a double-beat regulator valve in the dome by which the steam is admitted to the superheater system, but it is not required for the control of the engine, as the turbine has its own starting valve.

Where the smoke box door would normally be fitted, a cylinder of depth about half that of the projecting casing is placed. It is free to rotate slowly on the two rollers already mentioned. The chimney casing is carried down into the smoke box, and then out horizontally through this cylinder, as indicated by the dotted lines. The lower half of the projecting front communicates through the heater cylinder with the air duct, so that when the louvres are open there is a continual flow of air into the duct. The heater cylinder is closely packed by radial plates of thin corrugated metal. The gases are drawn by the fan from the upper part of the smoke box through these plates, which reduce the temperature to about 150° Fahr. It then discharges them up the chimney. As the cylinder moves round, the heated section passes between the louvres and the duct, and the heat deposited in the metal plates is given up to the rapidly moving cold air. The cooled plates are again heated by the gases, and the process is repeated. It is, in fact, a continuous one. This type of rotating regenerative air heater has been successfully applied to many land installations, and the possible increase of boiler efficiency is about 10 per cent. It, however, increases the first cost of the boilers considerably. In order that this preheating system may work the ash pit has to be " closed."

The louvres are opened and closed simultaneously with the fire door to prevent the flame from being blown out when the door is opened. The ash pit, as shown in Fig. 389, has two shoots with cleaning doors.

The trailing carriage with the turbine and condensing plant is carried on the axles of six coupled wheels and a four-wheeled trailing bogie. The leading wheel is the driver, operated through a treble reduction gear from the turbine. The carriage is carried by laminated springs and equalising levers, as shown on the photograph.

In order that the full thermal benefit of a turbine drive may be obtained it is necessary to run the plant condensing, at the highest vacuum obtainable. The difficulty in the case of a locomotive of providing a surface condenser suitable for from 1500 to 2500 horse-power is very great. Either an air or a water-cooled or an evapora-

tive condenser may be used. In this instance the designers chose the air-cooled type. This condenser is in two parts. A cylindrical tank 4 ft. 6 in. diameter by 27 ft. 8½ in. long, supported on the frame by flexible plates to admit of expansion and contraction, occupies the major part of the carriage length. The turbine exhausts directly into this cylinder, which is kept partly filled with the water of condensation. Two headers, one at the top and the other at the bottom, run the length of the carriage frame on each side. These are connected by about 2000 flattened copper tubes narrowly spaced, 6 in. wide and $\frac{1}{16}$ in. cross-section, which form the vertical sides of the carriage shown on the photograph. Four fans each 6 ft. 9 in. diameter, indicated in Fig. 389, at the top of the carriage, draw the air from outside through the tubular walls. This high velocity air condenses the steam in the flat tubes. The fans are driven, through a shaft running along the top, from a separate steam turbine of 300 B.H.P. It also drives a small centrifugal pump which raises water inside the tank, and discharges it on to a tray, through which it is sprayed on the steam entering from the turbine, causing partial condensation. The remainder of the steam passes through six branch pipes from the cylinder to the upper headers, and thence to the tubular sides.

The condensate flows from these tubes into a sump tank, from which it is raised into the cylinder by a jet pump. The air and vapour pass on from the sump and are exhausted by a steam ejector pump, which delivers the air and condensed vapour to the feed tank. The feed is pumped back to the boiler by a rotary high-pressure turbo-pump. On its way to the feed tank it is passed through a two-stage feed water heater, Fig. 389, which takes the exhaust steam from the auxiliary turbines, the vacuum brake ejector, etc. The average vacuum obtained is 25 in. Values as high as 27·5 in. and as low as 15 in. have been recorded.

The main turbine is of the disc and drum type. It has a single row of blading on the impulse wheel and eighteen rows of reaction blading on the drum. It runs at 10,500 rev./min. and develops 2000 H.P., the corresponding speed of the locomotive being 75 miles/hr. A pitch line diagram of the reduction gear is shown in Fig. 391. The pinion 1 on the turbine shaft 2 gears with the wheel 3 of the second motion shaft 4. Its pinion 5 gears with the wheel 6 of the third motion shaft 7, and its pinion 8 gears with the wheel 9 on the driving axle 10. The dotted circle 11 is that of the tread of the wheel. The turbine runs in one direction, and in order to reverse the motion of the engine an auxiliary shaft 12, with a wheel 13 and pinion 14, which are duplicates of 3 and 5, is provided. To reverse the engine the second motion shaft 4 is swung slightly to the right with 3 and 1 still in gear until 5 falls out of gear with 6. The "idler wheel" and its pinion are swung to the right so that 13 gears with 3 and 14 with 6. Wheel 6 and with it the driver wheel 9 then rotate in the reverse direction, and the engine moves backward.

The teeth of the gears are specially undercut to give them a slight amount of elasticity to resist shock.

The final gear wheel 9 is elastically mounted on the driving axle 10. The rim is connected by slightly flexible spokes to the hub, which is part of a sleeve extending right across the axle width. This is provided with three arms inclined at 120° at each end and forked. Through this again is passed another sleeve with three similar arms. This inner sleeve, through which the wheel axle passes, is fixed at the ends to the wheels. The drive from the pinion sleeve to the fixed sleeve is transmitted by push rods extending between the pairs of forks on the arms. A swivel block is pivoted in each fork and the push rods fit loosely into sockets in these. This loose arrangement provides against oscillation of the carriage on the springs and reduces shock on the gear. For the details of the arrangements for reversal and the prevention of fouling of the gears the student will have to refer to the *Engineering* article. The scheme is ingenious and appears to work satisfactorily in service. The simple plan, however, which has been standard practice on marine turbines for many years would get rid of the gearing complication involved here. It would result in a considerably decreased power for running backward. A locomotive, especially a main line express, is intended to run forward in regular service and backward only for shunting purposes, and this objection is thus not serious. In addition to the special features just described the boiler and locomotive is provided with the usual standard fittings. Steam brakes are provided on the front portion and vacuum brakes on the condenser carriage.

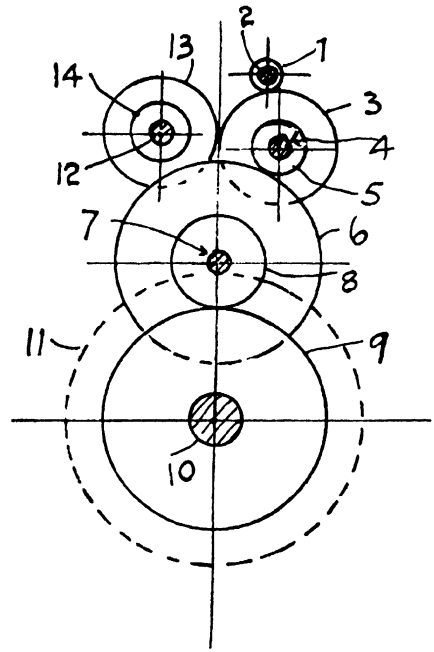


FIG. 391.

**316. Krupp-Zoelly Turbo-Locomotive.**—A different design of locomotive is being developed conjointly by Krupp of Essen and Zoelly of Escher Wyss and Co., Zurich. A photograph of one of these locomotives is shown in Fig. 392. In this case the reciprocator cylinders are replaced by a double turbine combination in one housing fitted below the smoke box, consisting of a forward

running impulse turbine of six stages and a backward turbine of three stages, one at each side with shafts in line and the gearing between them. The rotor of the backward turbine is driven round in vacuo when the engine is running forward, so that the windage loss is not excessive.

Each turbine is geared through double reduction gearing to a jack shaft, carrying an outside crank at each end. A connecting rod from each of these cranks is connected to the crank pin of the first of the three coupled driving wheels, as shown on the photograph. A surface condenser is used, divided into two portions. Each portion is a cylindrical shell carrying the necessary tubes, and is placed transversely below the boiler immediately behind the turbines. The end of these shells are shown above the leading driver, with the flexible connecting pipe. Each turbine exhaust branch is connected to the front shell at the top as shown. Compared with the Ljungström air-cooled condenser this one is small. This is due to the fact that with water as the cooling medium the average rate of heat transmission is about 500 B.Th.U./ft.$^2$ hr. ° Fahr., while with air it is about 25 B.Th.U./ft.$^2$ hr.° Fahr., with an air velocity of 4000 ft./min. This advantage is, however, offset to a considerable extent by the additional apparatus required for recooling the circulating water directly by air currents. The tender is mounted on two four-wheeled bogies. It carries the water tank and coal bunker as usual, and in addition the cooling system for the circulating water. A number of cooling elements are so arranged above one another, that the hot circulating water returned from the condenser is discharged at the top, falls through these, and is broken up into small drops. A current of air is passed through this rain in an opposite direction to the water flow. The sprayer elements are provided with fins which obstruct the air and give it a turbulent motion, in order to ensure an intimate contact with the water particles and obtain the maximum cooling effect. The cooled water collects at the bottom and is pumped back to the condensers. This system is simply an adaptation of the cooling tower used on land installations where the supply of circulating water is limited. It involves loss of circulating water, which is carried away as vapour with the cooling air, and hence reserve water has to be carried. The turbine circuit, however, being closed, only " make-up " feed water has to be supplied.

This Krupp-Zoelly locomotive works with steam at 200 lb./in.$^2$ gauge and 600° Fahr. and develops 2000 D.H.P., with a maximum speed of 62 miles/hr. The L.M.S. Ljungström works with 285 lb./in.$^2$ gauge and 600° Fahr. and also develops 2000 D.H.P., and appears from test results to be slightly superior in efficiency to the Krupp-Zoelly.

317. A turbo-locomotive which is similar to the Krupp-Zoelly in arrangement but differs considerably from it in detail is being tried out by J. A. Maffei of Munich. The turbine is placed above a bogie truck, in front of the boiler, and a special spring drive is

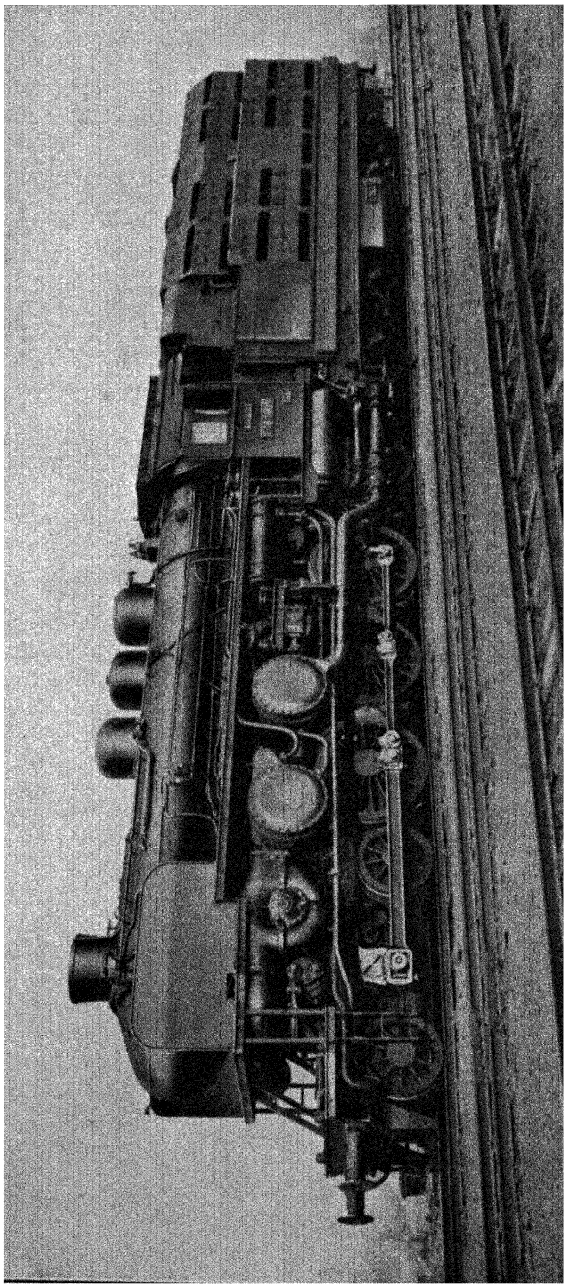


FIG. 392.

provided for the reduction gear shafts. The two condensers are arranged longitudinally at each side, and the cooling system on the tender has perforated plates instead of tubes, over which the water flows in thin streams, while air is drawn through them by a fan. This is said to be the most powerful engine of this type so far constructed. It is stated to have attained a maximum speed of 78 miles/hr. with a H.P. of 2500.

The boiler is of normal design working at 300 lb./in.$^2$ gauge, but there is neither preheating of the air nor feed heating.

The performances of the various classes of locomotives are discussed in Chapter XXIV.

# CHAPTER XIX

## TYPES OF STEAM TURBINES

In the four following chapters a brief discussion is given of the theory and action of the turbine prime mover, with illustrations and descriptions of several modern examples of the different types.

The student can refer to the special texts on steam turbines for a more complete treatment of this very extensive subject than can be attempted in a general text.*

**318. Types of Steam Turbines.**—It is necessary, at the outset, to distinguish between the operative action of the steam in the reciprocator and in the turbine.

The driving effort exerted on the mobile part of the reciprocating engine, which is the piston, is the statical pressure of the steam on the piston face. Relatively to this, the steam has no sensible velocity, and hence there is no dynamical effect. On the other hand, the effort on the mobile part of the turbine, the blading of the rotor, is a dynamical pressure, due to change of momentum of a jet of steam moving with a high velocity. This force may be due to impulse or reaction or to both.

The effort of the piston of the reciprocator is measurable directly from the indicator record. That on the turbine blading is not measurable, and can only be approximated by a dynamical calculation. There is thus no such quantity as the "indicated" horse-power of a turbine; but its counterpart, the gross energy expended by the steam on the blading, is calculable on the basis of certain assumptions, which are stated later.

**319.** As in the case of the reciprocator, a turbine may be either simple or compound, that is, the total pressure drop may be carried out in one element or stage, or it may be sub-divided into a number of stages or drops.

In a turbine stage the essential feature is a drop of pressure accompanied first by generation of kinetic energy in the mass of the steam, and secondly the conversion of this energy into mechanical work.

In an impulse stage the pressure drop takes place in a set of stationary passages, from which the steam issues with a higher velocity than at entrance. Part of this energy is then absorbed in

* See sixth edition of Prof. Stodola's *Steam and Gas Turbines*, English translation by Dr. Loewenstein; see *Steam Turbines*, 2nd edition, by W. J. Goudie.

a ring of moving blade passages, through which it is deflected. Referred to the turbine elements, the "stage" thus consists of the set of stationary nozzle passages and the succeeding set of moving ring passages. The important point to be noted here is that there is no drop of pressure and generation of kinetic energy in the moving blade ring.

In a pure reaction stage the drop of pressure, with expansion and generation of kinetic energy, takes place in a moving nozzle passage or passages, on an arm or a drum. Here the effort is purely reactive.

In the practicable type of turbine, however, a stage, as in the impulse, is taken to consist of a set of fixed nozzle passages and the succeeding moving ring, but in this case there is a further drop of pressure in the moving passages. The stage pressure drop, with this definition, really consists of two drops, one in the fixed and the other in the moving passages, and the more suitable term is a "double stage." The driving effort on the ring of moving blades is now a combination of impulse and reaction. A turbine having this operative stage condition is always called a "reaction," but the more correct term should be "impulse-reaction" turbine.

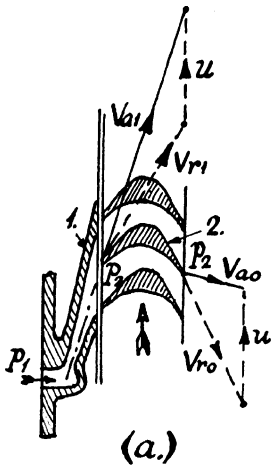


FIG. 393.

**320. Classification of Impulse Turbines.**—The impulse type can be divided into four classes:

(*a*) Simple.
(*b*) Velocity compounded.
(*c*) Pressure-velocity compounded.
(*d*) Pressure compounded.

In any of these cases the general direction of the flow of the steam between admission and exhaust may be either axial or radial. The majority of impulse turbines have axial flow.

**321. (*a*) Simple Impulse. Axial Flow.**—This is diagrammatically shown in Fig. 393. A nozzle 1, shown convergent-divergent, expands the steam from initial pressure $p_1$ to exhaust pressure $p_2$ and discharges it with an absolute velocity $V_{a1}$ on to the ring of blades 2, moving with velocity $u$, as shown in plan. After passing through the blade channels the steam is discharged with an absolute velocity $V_{a0}$. In changing the velocity from $V_{a1}$ to $V_{a0}$ the steam expends energy on the ring. The methods of construction of the velocity diagrams shown, and the calculation of the work done on the blade ring, are discussed in Chapter XXI.

**322. (*b*) Velocity Compounded Impulse. Axial Flow.**—This is diagrammatically shown in Fig. 394. As before, the steam drops

in pressure from $p_1$ to $p_2$ in the nozzle 1, and enters the first ring of moving blades 2 with the same value of $V_{a1}$. It is discharged into a ring of fixed guides 3, in which its direction is changed, and from these into a second ring of moving blades 4. It issues from these with an absolute velocity $V_{a0}$. The pressure throughout the rings is $p_2$. Since the energy is absorbed in two rings of moving blades, this pressure stage is said to be "two-velocity compounded." In this case the velocity $u$ of the blade is half the velocity in case ($a$). The reason for this modification is given later.

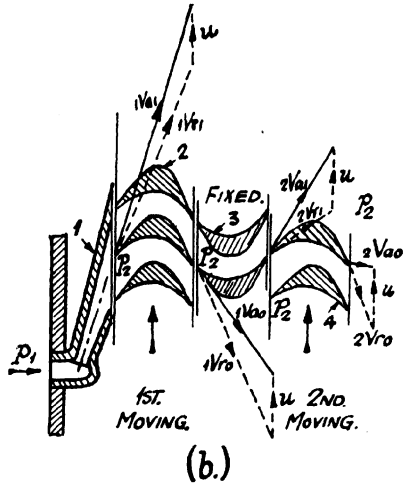



FIG. 394.

The kinetic energy of the steam at entrance and exit is proportional to the square of the velocity. In ($a$) the energy represented by $V_{a0}$ is about 6 per cent. of the jet energy at entrance to the blades, represented by $V_{a1}$, while in ($b$) it is reduced to about 1 per cent. by the velocity compounding. This energy is called the "carry over" of the stage.

**323. ($c$) Pressure Velocity Compounded Impulse. Axial Flow.**—

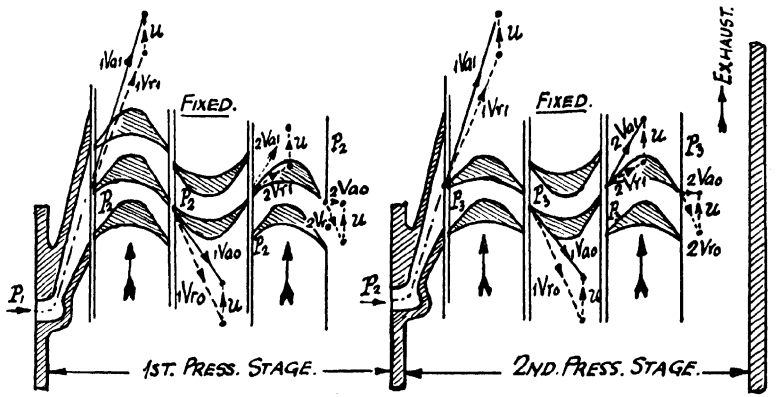



FIG. 395.

In this case, shown in Fig. 395, the total drop between the initial and exhaust pressures $p_1$ and $p_3$ is divided between two stages, each two-velocity compounded.

The machine consists of a velocity compounded turbine working between the pressure limits $p_1$ and $p_2$ in series with another velocity compound working between the limits $p_2$ and $p_3$. It is assumed that the total energy available for work between $p_1$ and $p_3$ is equally divided between the stages, and the two sets of diagrams are identical. The initial velocity ${}_1V_{a1}$ is less than in the previous case, on account of the smaller pressure drop. The first stage carry over is about $1\frac{1}{4}$ per cent. of the initial energy. If the steam could flow directly into the second stage nozzles a certain amount of this energy could be utilised. As a rule it does not, and the carry over is converted by eddy and stock into friction "reheat," a small fraction of which may be available in the next stage. The carry over at the second stage is, however, a complete but not a serious loss. In the first-stage blade passages the pressure throughout is $p_2$ and in the second-stage passages it is $p_3$.

This pressure-velocity compounding can obviously be carried to any number of stages. In the early development of this type, introduced by the American engineer Curtis, three, four, or more velocity compound stages were tried per pressure stage. A little experience showed that it did not pay to go beyond three. Even this limit is now seldom worked to, except in the cases of small machines, two velocity stages being the rule in modern turbine practice.

It will be noted that in all these cases the nozzles are shown convergent-divergent, that is, the cup-shaped entrance is narrowed down to a "throat" and after this the walls of the nozzle diverge. This is because at each stage the ratio of the exhaust to the initial pressure is below a certain limiting value, known as the "critical" limit, which is discussed in Art. 358. When the pressure ratio is above this limit the nozzle passage is made convergent.

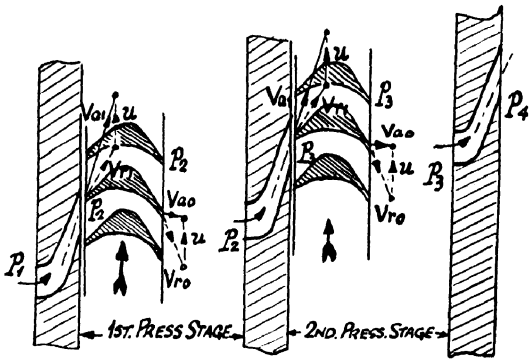
**324.** (*d*) **Pressure Compounded Impulse. Axial Flow.**—When the number of velocity stages is reduced to one per pressure stage, then type (*c*) reduces to a number of simple impulse turbines working in series. The first two stages of a machine of this class are shown in Fig. 396. It is called a "pressure compounded impulse" and is a type very generally adopted. The total drop between initial and exhaust pressures is so sub-divided that the stage pressure ratio, as a rule, is above the critical value, and the convergent type of nozzle passage, indicated in the figure, is used.

**325.** Although, for simplicity, only one nozzle has been shown in these illustrations, in actual machines the nozzles are arranged in groups or sectors. Frequently these sectors are advanced in each successive stage, as indicated in the figure, so that the steam flowing from one sector of nozzles, after passing through the wheel blades, may be discharged directly into the nozzle sector of the next stage. As a rule not much benefit is derived with a velocity compounded machine; but with this type, in which the carry over velocity may be appreciable (300 to 500 ft./sec.), a slight benefit is obtained by

this arrangement. The principal carry-over loss in this case occurs at the last stage, and as a rule is a very small fraction of the total energy available between the initial and exhaust pressures.

The total available energy between these pressures may be equally divided among the stages ; but frequently a much larger proportion is taken by the first than by the succeeding stages. The pressure drop in this stage is much greater than in the others, an important consideration where high pressure and high superheat are used, as only the small nozzle chamber is subjected to the extreme conditions and the first stage chamber and cylinder end to the more moderate pressure and temperature. Also the number of stages is reduced, as well as frictional resistance.

For this larger pressure drop a two-velocity compounded wheel is




(d.)

FIG. 396.

usually fitted. This machine is really a combination of types (*c*) and (*d*).

326. Small turbines of class (*b*) velocity compounded instead of having two or more moving rings may have a single ring of blades through which the steam is passed two or more times. In such a case the flow is usually radial. Steam is discharged radially from the nozzle into the blading, at the outer circumference of the ring (which projects from the side of the wheel), and at exit into a curved guide passage, which changes its direction and passes it radially again from the inner circumference through the blade ring in the opposite direction. It may be again discharged into an outer guide passage and passed a third time through the ring, and so on.

Alternatively it may be discharged from the nozzles into " buckets " cut on the face of the wheel and passed through these buckets again by means of reversing channels fixed above the rim. The flow here is partly radial and partly axial.

These special types are only suitable for small powers, and are

mostly used for auxiliary purposes, where the question of economy of steam does not arise and a simple and robust machine needing little attention is required. Constructionally they are impracticable for large powers.

**327. Reaction Turbines.**—These may be divided into two classes : (*e*) pure reaction ; (*f*) reaction with impulse.

Class (*e*) is of little practical importance. The turbine consists of a rotating arm, as shown in Fig. 397, or a cylinder with nozzle openings at the ends or circumference. Steam admitted at $p_1$ at the centre is expanded in the nozzles to the external pressure $p_2$ and issues with a velocity $V_{r0}$, relatively to the nozzles. It exerts a reactive force which rotates the arm or cylinder at an absolute velocity $u$. It leaves the nozzles with an absolute velocity of exit

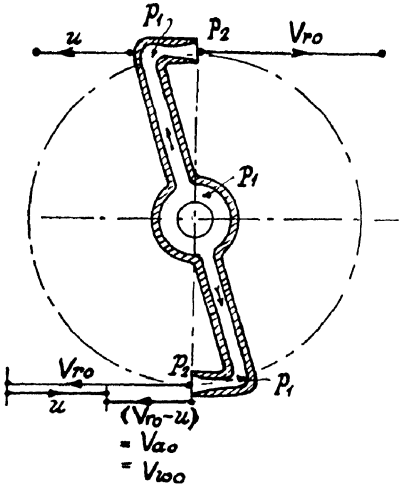



FIG. 397.

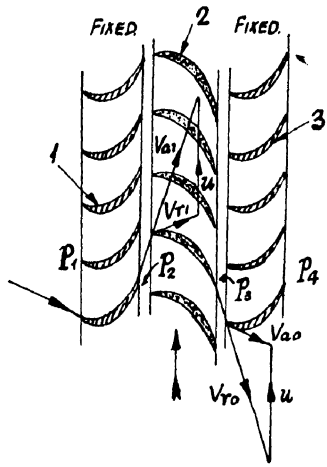



FIG. 398.

$V_{a0}=(V_{r0}-u)$, tangential to the nozzle circle. Small machines of this type were originally used by Dr. De Laval to drive cream separators, and they have had some use in America. Sir Charles Parsons experimented with the type and abandoned it.* It has no commercial representative.

**328.** Class (*f*) is the practicable type, and may be either axial or radial flow. The majority of machines are axial flow. This is the type originally developed by Sir Charles Parsons and invariably called a Parsons Turbine. As already stated in Art. 319, the driving effort on the blading is partly impulse and partly reaction. A double stage of this type is shown in Fig. 398. The steam enters a ring of blades 1, fixed on the cylinder or casing, with an absolute inlet velocity $V_{a0}$ and pressure $p_1$ and expands to pressure $p_2$ at

* See Richardson's *Evolution of the Parsons Steam Turbine*, p. 42.

exit. It enters a ring of moving blades 2, mounted on the circumference of a drum or a wheel, with absolute velocity $V_{a1}$, and expands further to $p_3$ at outlet. It then enters the next ring of fixed blades 3 on the casing with velocity $V_{a0}$ and is again expanded to $p_4$, issuing with absolute velocity $V_{a1}$, and is discharged into the next ring of moving blades, and so on.

329. At present there is only one commercial representative of the radial reaction turbine. A double stage of the outward radial-flow type is shown in Fig. 399. The steam, as before, enters the fixed ring 1 with some absolute velocity $V_{a1}'$ and expands to $p_2$ at exit and is discharged to the moving ring 2 with an absolute velocity $Va_1$. It again expands from $p_2$ to $p_3$ and is discharged into the next ring of fixed blades 3 with absolute velocity $V_{a0}$, and so on. In the previous case $V_{a1}$ and $V_{a0}$ and $u$ are constant for a series of rings of the same diameter, but in this one all three have increasing values from stage to stage, as indicated on the velocity triangles. This is an example of a stage of a "single-motion" radial turbine. If the fixed rings were transferred to a disc on one shaft, and the moving rings kept on the original shaft in line with this second shaft, then the impulse-reaction effect of the steam would drive the blade rings and the two shafts in opposite directions and the machine would become a "double-motion" turbine. The successful commercial machine of the radial type, the Ljungström, is a double-motion turbine.

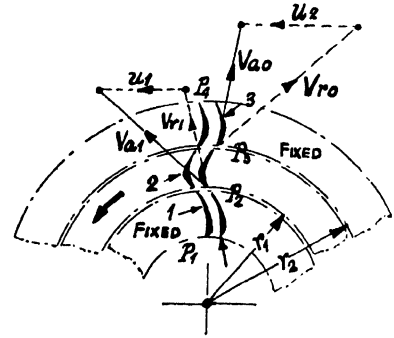


FIG. 399.

330. **Combination Turbines.**—There are certain mechanical disadvantages common to both the axial flow reaction and the impulse machines. One of these is the necessity for running clearances between the fixed and stationary parts, in order to prevent seizure. Through such clearance areas more or less leakage of steam takes place. In the Parsons type the weak spot is at the blade tips in the high-pressure section. In the impulse machine it is at the stage diaphragm glands, through which the shaft passes. Owing to the continuous nature of the expansion in the reaction blading, full peripheral admission is necessary from the first stage onward. For the passage of a given weight of steam, the blade lengths at the H.P. end are much shorter than they would be if partial admission were possible. In the majority of cases the blade tip clearance which is necessary for safe running has in consequence to be made an appreciable fraction of the radial distance between the drum and casing, and the leakage loss becomes appreciable. On the other

hand, in the impulse machine, where there are considerable pressure differences in the successive stages at the H.P. end, the reduction of the leakage area between the shaft and diaphragm glands is equally important if the leakage loss is not to become appreciable. In order to mitigate the leakage defect and also to obtain additional mechanical advantages, a considerable proportion of the total energy available between admission and exhaust pressure is developed in an impulse stage, usually two-velocity compounded, which is substituted for a number of the reaction stages. The turbine in this instance is a combination of impulse and reaction, and goes by the name of a " disc and drum " turbine. This modification materially shortens the rotor and increases its stiffness, and at the same time shortens the length of the turbine. The steam is reduced at entrance to the reaction section to a comparatively moderate pressure and temperature, and this condition simplifies the external packing glands. Further, highly superheated steam can be used, since this comes in contact only with the small externally-fixed nozzle box, and does not affect the main casing body. These advantages are purely mechanical. From the thermodynamic point of view the section of reaction blading is inherently more efficient than the impulse wheel if the radial clearance defect does not exist. The Parsons company are now remedying the H.P. reaction blade leakage by a system of what is termed " end tightening." In this construction radial flow of the steam is prevented by axial fins. This form of blading is illustrated in Chapter XX.

# CHAPTER XX

## IMPULSE AND REACTION STEAM TURBINES

**331. De Laval Simple Impulse Turbine.**—This turbine is a high-speed geared machine which has been in use for many years. It was the forerunner of the geared type now extensively used, both in land and marine practice. It is made by Greenwood and Batley in sizes from 5 to 500 B.H.P. The reduction in the small size is 10 to 1 and in the large 13 to 1. A longitudinal section through one of the small machines is shown in Fig. 400. Steam from a stop valve (not shown) enters at 1 and passes to the throttle 2, through a gauze strainer 3. The throttle valve, which is of the double-beat type, is controlled by a centrifugal governor on the second motion or driving shaft. The steam passes from the throttle to a belt 4 surrounding the wheel casing, which is drained by a cock 5. Several straight convergent-divergent nozzles are fixed in the casing wall. Each nozzle has a shut-off valve, which is simply a screwed spindle coned at the end to fit into the throat. In the larger size of machine two or three nozzle passages are provided in a nozzle plate, forming the front of a nozzle box, to which steam is admitted through a shut-off valve at the back. Several of these groups are provided to vary the steam supply in accordance with the load, and maintain a constant initial pressure.

After passing through the nozzles and blading of the wheel 7 into the exhaust belt 8, the steam is discharged at 9 into the exhaust branch, leading to the condenser.

In the small-size machine the shaft 10 is made flexible. It is turned down to a smaller diameter to the right of the wheel and supported on a spherically seated bearing 12 and a fixed bearing 14. These bearings are spaced widely apart. On account of its flexibility the shaft bends when running at speed, so that the centre of mass coincides with the axis of rotation. In the largest machine the speed is 10,000 rev./min.; in the smallest it is 30,000 rev./min. At such speeds an inflexible shaft would cause serious vibration trouble. The first application of the flexible shaft to avoid vibration was made by Dr. de Laval. In the illustration the shaft passes through a hole in the wheel boss, and bears, for a short length only, on the bushing to which it is pinned. In the larger machine there is no hole in the boss. The shaft is in two parts, each of which is bolted to the boss. The reason for this arrangement is to keep

down the very high stress, near the centre, which is set up in the material of the larger wheels at the high rotational speeds. The wheel section tapers from a large boss with very wide radius to a thin disc near the rim. In the larger machines the disc bulges

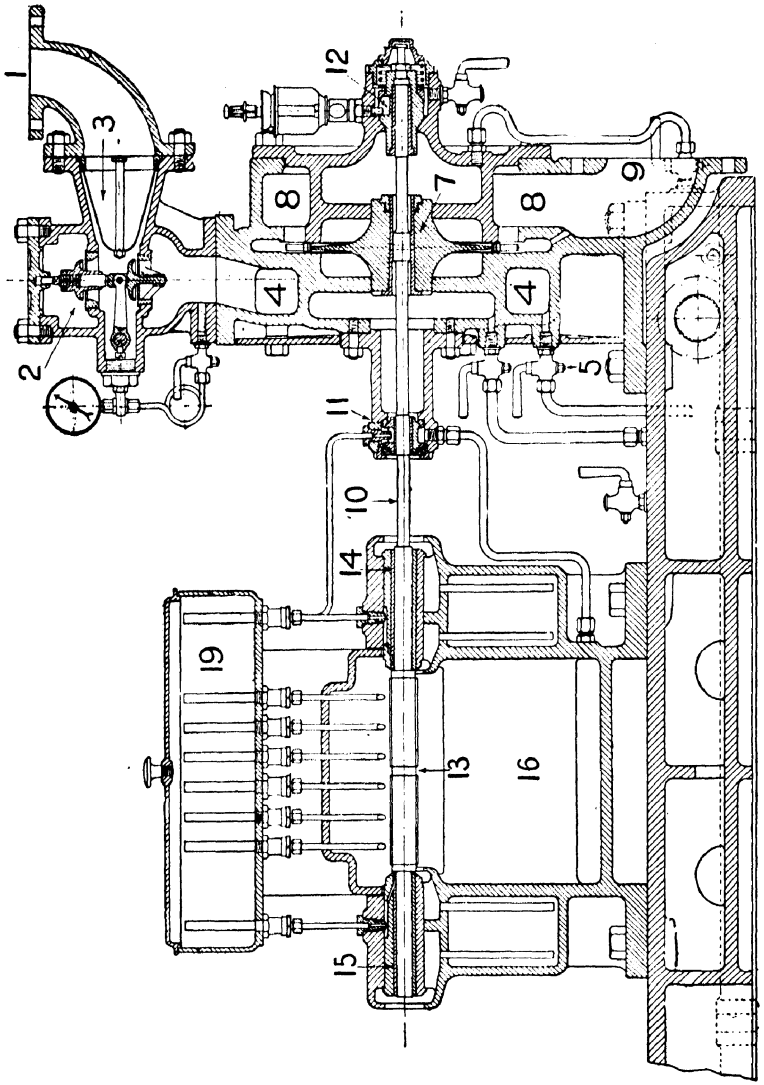



FIG. 400.

out from about the mid-radius, and merges into the boss with a smaller radius. It is known as a " disc of uniform strength." The reduction gear is double helical. The driving pinion 13 is solid with the shaft and is situated between the bearings 14 and 15, at

the ends of the gear case 16. The wheel and second motion or driving shaft, on the far side, are not shown.

In machines of 20 H.P. and over, the pinion and gear wheels are divided, and an intermediate bearing is provided between 14 and 15. The bearings in this case are syphon lubricated from the oil box 19. The driving shaft bearings are ring lubricated. A sight feed lubricator is fitted on the spherical bearing 12. Machines of over 20 H.P. are provided with forced lubrication.

A gear wheel pump driven by a worm and wheel from the driving shaft draws oil from a well in the base, in which a cooling pipe coil is placed, and discharges it under pressure to all the bearings. Leakage of air into the casing under vacuum is prevented by tightening bushes 11.

The small centrifugal governor, which is mounted on the end of the driving shaft next the casing, is very effective in controlling the throttle valve.

In case of serious over-speeding, however, an emergency governor is also fitted and is brought into action by the overrun of the centrifugal governor. This operates an air valve which admits air below a piston, held down in its cylinder by a spring, in equilibrium under vacuum on each side. This piston is connected by a link with a flap valve in the exhaust pipe. When the vacuum is broken by the admission of the air, the piston is drawn up under the unbalanced pressure and throws the throttle flap across the pipe, blocking the exhaust passage. The pressure in the casing immediately backs up and the speed is reduced, due to increased frictional resistance and decreased pressure drop in the nozzles.

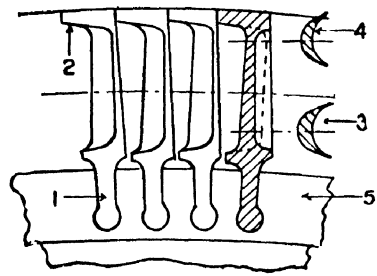


FIG. 401.

The form of the blading and method of fixture in the wheel rim is shown in Fig. 401. The blade, which is stamped from nickel steel, is formed with a tailpiece 1 enlarged and rounded at the bottom. It is pressed into the corresponding slot in the wheel rim. It is symmetrical, that is, the inlet and outlet angles are equal, and tapers from root to tip, for which sections 3 and 4 are shown. The tip projections 2 butt on one another, acting as spacers and also forming an elastic shroud, which allows for expansion under varying temperature.

**332. Rateau Velocity Compounded Impulse Turbine.**—The standard design of this type of two-velocity stage machine made by Société Rateau, Paris, for the drive of such auxiliaries as fans, pumps, etc., up to 150 H.P., is shown in Figs. 402 and 403. The wheel casing in the longitudinal section, Fig. 402, is in two parts.

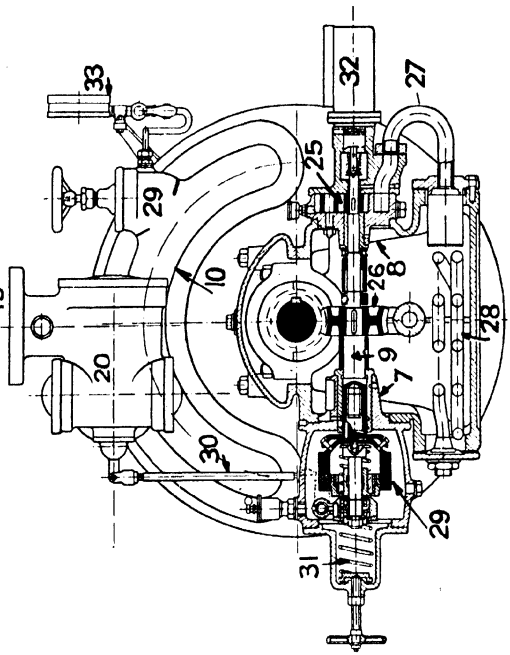



FIG. 403.

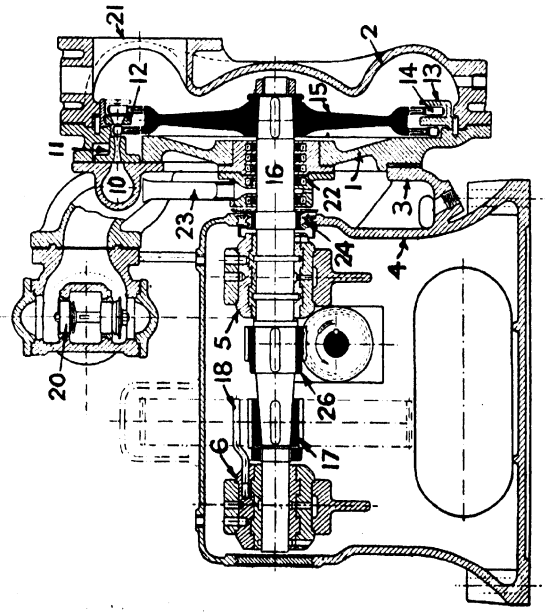



FIG. 402.

Part 1 is bolted to a semicircular flange 3 on the frame 4, which carries the shaft bearings 5 and 6, and the bearings 7 and 8 of the governor shaft 9, Fig. 403. A semicircular nozzle plate 10, shown in section in Fig. 402, is bolted on the upper half of part 1, and groups of nozzles 11 are bolted on its face. The outer part 2 of the chamber is bolted to the face of part 1, which may be of cast steel or cast iron according to the steam conditions. Part 2 is made of cast iron. The guide blades in the way of the nozzle arc are of drawn steel and dovetailed into a grooved ring 13, which is of channel section beyond the nozzle arc. The wheel blade edges fit closely to the channel and casing sides, and the loss due to pumping action on the " dead " steam is thereby minimised. The nozzles 11 are of steel and machined from the solid. The blades 14 are also machined out of solid steel and forked to straddle over the rim of the wheel 15, to which they are riveted. The wheel, which is of steel turned all over and balanced, is keyed on and fixed to the coned end of the shaft 16 by a locking nut. It will be noted that this overhung arrangement of the wheel renders access easy, and there is no horizontal joint in the wheel casing. The shaft bearings 5 and 6 are brass bushed with white metal lining, and are arranged for forced lubrication, the ducts being indicated in the covers and brasses. In this instance the turbine is arranged to transmit the power to the driven machine shaft by a straight wheel and pinion gear, the pinion 17 being keyed on the coned portion and locked by an end nut. It is lubricated from the short pipe 18 led from the left-hand bearing brass. Steam is admitted from a stop valve (not shown) at 19, Fig. 403, to the double-beat throttle valve 20, Fig. 402, from which it is supplied to one sector of nozzles, passed through the wheel and guide blading, and discharged to the condenser through the exhaust opening 21. For an overload, additional steam is supplied to the other sector by the valve 29. Another advantage of the overhung arrangement is the absence of a shaft gland at the L.P. side. When the turbine is run as a back pressure machine to give heating or process steam the single gland has to be sufficient to prevent leakage outward from the chamber. For initial pressures about 220 lb./in.$^2$ and superheated steam about 575° F., the gland 22 is composed, as shown on the section, Fig. 402, of five carbon rings in segments held up to the shaft by spiral band springs. When higher steam conditions obtain, labyrinth packing is used. A vent passage and pipe 23 are provided, between the fourth and fifth carbon rings, to carry away any leakage steam. The shaft, between the right-hand bearing and the gland, is fitted with a channel section liner 24 into which a double V ring fits. This ring is held in the casing 4. This arrangement is for the purpose of preventing the spread of oil along the shaft into the gland and the exhaust steam. The oil is pumped into the bearings by the gear wheel pump 25 on the governor spindle 9, which is driven by the worm and wheel 26. It is drawn through a strainer and pipe 27

from the bottom of the casing, where it is cooled by the circulating water in the pipe coils 28. The centrifugal shaft governor 29 controls the throttle valve, through the medium of the rod 30 operated from the bell-crank in the governor box. The speed can be regulated by the spring control and hand-wheel 31. The speed is recorded on the tachometer 32, and the pressure in the H.P. chest by the gauge 33.

**333. M.V. Self-contained Impulse Turbine Set.**—The Metropolitan-Vickers Electrical Co. has recently made an important advance in turbine practice for small and moderate power machines by the introduction of a geared type of impulse turbine in which the bottom of the turbine casing is made integral with the condenser shell, and the circulating and condensate extraction pumps are worked directly from the driving shaft, while a steam ejector pump is used to draw the air from the condenser.

This compact set can be arranged on one floor, thus avoiding the costly basement arrangement of condenser and independently driven auxiliaries, and reducing foundation costs to a fraction of the former values incurred with the older arrangements.

These sets are made for high pressure, or for extraction service, and are also very suitable for the drive of electric generators. The outputs range from 200 kw. to 6250 kw., the turbine speed varying from 5000 rev./min. for the larger to 6000 rev./min. for the smaller sizes.

In the case of electrical generation of power, the combination of a high-speed turbine with a lower-speed generator enables each machine to be designed to give its most efficient performance, with the result that high overall thermal efficiency and low rate of heat consumption are obtained.

**334.** A side view of one of these turbines is shown in Fig. 404, the turbine and condenser being in section and the reduction gear box in outside elevation.

The casing 1 is divided on the horizontal centre line, so that the top half can be lifted for the removal and insertion of the rotor. The end 2 of this half is made of cast steel, as it is subjected to the action of the high temperature steam admitted through the branch 3 to the steam belt 4, from which the nozzles are supplied. It is bolted to the casing 1, which is made of cast-iron. The whole of the lower half 5 is made of cast-iron and formed, at the L.P. end, into the top part 6 of the condenser 7. This is of the two-pass type, the water entering one end, flowing through the bottom pass of tubes 8, then back from the other end through the top pass 9, to be discharged at this end. The water is supplied by a centrifugal pump driven from the reduced motion shaft. The condensate is drawn from the bottom of the condenser by an extraction pump, also geared from the shaft. The section of tubes 10 below the baffle plate acts as an integral cooler for the air and vapour mixture which collects below the plate, and is drawn away at the

top by an air ejector, which replaces the mechanically-operated air pump.

This machine has one two-velocity stage followed by eight simple

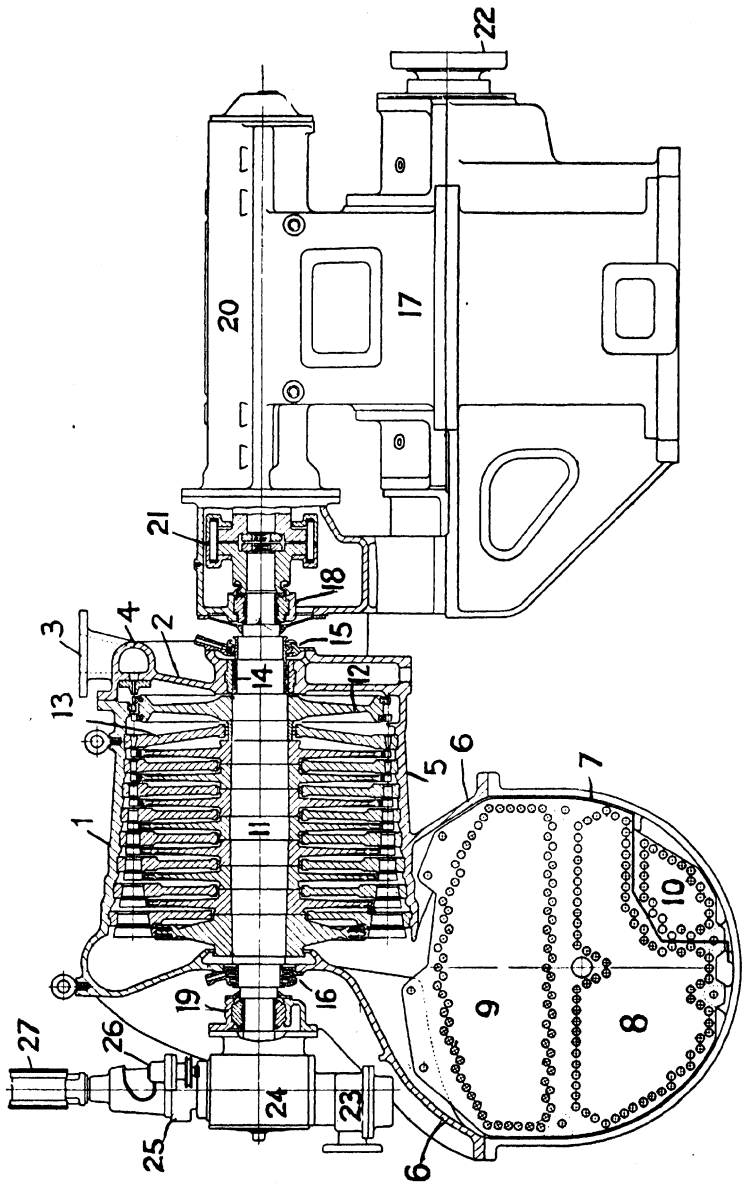



FIG. 404.

impulse stages. All the disc wheels 12 are made of forged steel and machined all over. They are pressed and keyed on the stepped

shaft, which tapers from the middle to each end, so that the steps permit of easy fixture of the discs and also assist in stiffening the shaft. The separating diaphragms 13 are in halves fitted and fixed in grooves machined in the cylinder. They are made of cast-iron, with steel nozzle blades cast in. At the high-pressure end, under exceptional conditions of pressure and temperature, steel centres with machined guide blades are employed. The diaphragms have spigot and socket joints at the sectional faces, and are fitted with labyrinth glands at the inner peripheries to prevent leakage of steam between stages. The simple impulse wheel bosses butt on each other. Each boss has a narrow collar at one side dividing the space between the pair of wheels in two. A ring, dovetailed into the diaphragm boss, carries two copper rings in these spaces. These rings are forked to form two sharp edges, which when new just bear on the wheel hubs. A heavier diaphragm is fitted between the first and second wheels, which are spaced by a sleeve on the shaft, and this diaphragm carries three of these copper fin rings. The blades of the velocity compounded wheel are machined from 5 per cent. nickel steel, with T-shaped roots which fit into corresponding slots in the rim. They are formed with distance spacers integral with the bodies. Each simple impulse wheel has a narrow rib turned on the rim. The blades, in this case, are forked and straddled on this rib and secured to it by rivets.

A steel shroud ring is riveted round the circumference of the blade ring. It is fixed in sections, so as to allow for expansion and contraction under varying temperature conditions. Towards the L.P. end the nickel steel blades are replaced by stainless steel, on account of the wet condition of the steam near exhaust. The blades are all highly polished to reduce friction loss. At the high-pressure end the shaft is fitted with a multi-throttling labyrinth gland 14 of special design, and outside of this a water-sealed gland 15 with an impeller rotating in a chamber containing water. A similar water-sealed gland 16 is fitted at the L.P. end.

The first stage nozzles consist of built-up machined steel blades, and are divided into two groups. The larger group is controlled by a valve and supplies steam sufficient for the economical rating of the machine. The other group is controlled by a second valve which supplies additional steam for an overload. The governing gear, however, is so arranged that the small group can be used alone to take light loads.

**335.** The governing system is an excellent example of one form of oil relay governing, universally used on machines of moderate and large outputs.

It is diagrammatically represented in Fig. 405. The valve 1 supplies the large nozzle group and the valve 2 the small group. They are connected by the rods and links shown, to the ends of a lever 3. This in turn is suspended by the link 4 from another lever 5 pivoted to a fixed point 6. The other end of 5 is jointed to the rod

7 of a power piston 8, operated by oil under pressure. A strong spring 9 pulls on lever 3 at some point 10, to the right of the link 4, and tends to hold the valves in the closed position. If the piston 8 is raised and with it lever 5 the spring will hold valve 2 closed and the link 4 will pull up valve 1. When this is fully opened, further motion of 5 will stretch the spring, and valve 2 will be raised.

The point of spring attachment is variable. If it is shifted to the left of the link 4, the reverse action will obviously take place. The spring will hold valve 1 closed, and valve 2 will be opened, when 8 and 5 are raised on light load. The relay or power piston is controlled in the following manner. The centrifugal governor, consisting of two L-shaped weights 11, is mounted on the spindle 12 driven from the turbine shaft by the worm and wheel 13. These L

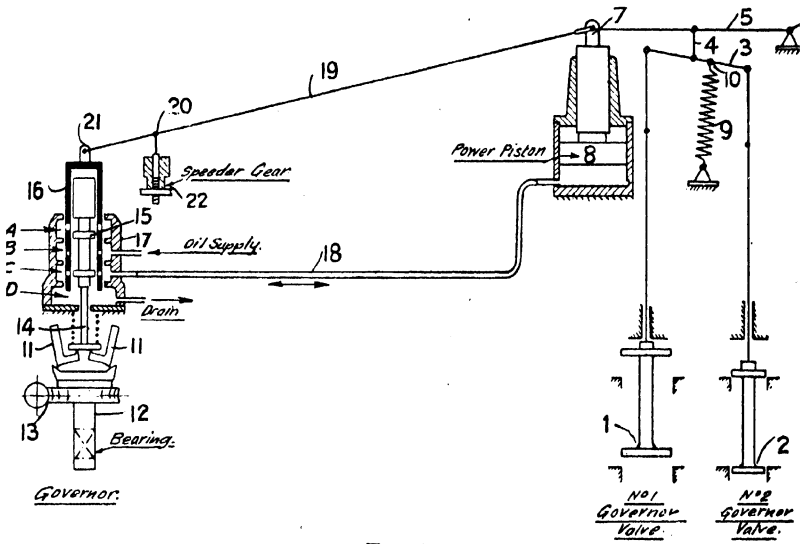



FIG. 405.

arms, at speeed, are thrown outward and raise the spindle 14 against the resistance of the spring shown. The upper part of the spindle carries a control piston 15. This piston works loosely in a sleeve 16 which has three ports. This sleeve can be moved up or down in the outer casing 17, divided into four compartments A, B, C and D. The centre compartment B is connected to the oil pressure supply from a gear wheel pump driven by the governor spindle at the bottom. Compartment C is connected, by the pipe 18, with the relay cylinder. A and D are oil pressure release chambers. A long lever 19, pivoted at an adjustable point 20, is connected to the adjustable sleeve at 21 and the relay piston rod at 7.

When the governor spindle is at rest, the L arms are close in, the piston 15 is in its lowest position, and the lower piston ring is below

the port in the sleeve leading to compartment C. The oil thus passes from B into the sleeve and thence outward again through the lower sleeve part into compartment C and pipe 18, and the relay piston 8 is raised and with it valve 1. The speed increases and piston 15 rises. As the end 7 of link 19 rises the end 21 is lowered and the sleeve is moved down, so that its lower part falls below the edge of the chamber D connected with the drain to the oil receiver. The oil escapes from C to D and the relay piston falls and reduces the opening of the valve 1, and with it the speed. This action, however, raises the sleeve again and gradually reduces the port opening, till the port is closed and equilibrium is established.

When a fall in speed takes place the piston 15 falls and the sleeve is raised, and the port in the sleeve opens between B and C, admitting pressure oil below the relay piston. Adjustment of speed while the machine is running is made by the screw gear 22. The speed is recorded by a tachometer driven from the top end of the governor spindle, and not indicated on the sketch.

It is stated that this gear gives very close regulation and there is no trace of hunting.

**336.** All bearings run under forced lubrication. The oil is drawn through strainers in a tank below the turbine. One discharge connection from the pump leads directly to the governor relay system ; another connection leads to a reducing valve, thence to a cooler, from which the oil is distributed to the pinion and four bearings of the reducing gear in casing 17, Fig. 404, and the two bearings 18 and 19 of the turbine. The latter are in halves and made of cast-iron lined with white metal. The bearing 18 has a Michell thrust ring,* not so much to take end thrust, which is practically negligible with this type of turbine, but for registration of the rotor and adjustment of axial clearances.

The pinion shaft, housed at 20, is connected to the driving turbine shaft 11 by one of the most recent types of coupling, the Bibby flexible coupling 21.

It consists of two flanged discs keyed one to each shaft, a tempered steel spring in segments (indicated in black on the section), bent to form a complete cylindrical grid round the flanges, and a steel cover. This grid spring is the flexible and resilient member. A number of pitched axial grooves are cut on the flanges, and the folds of the spring coil are fitted into these. The grooves widen towards the coupling faces and the spring folds fit close on them only at the outer ends. On overload these springs bear on a larger surface than under normal running conditions.

This coupling compensates for want of alignment and does not set up a cross pull. It is very effective in absorbing shock and damping out vibrations.

* For discussion on Michell bearings, see *Steam Turbines*, by W. J. Goudie, Chapter XII.

The drive from the second motion shaft of the reduction gear is transmitted through the coupling 22.

The oil pump is housed at 23 ; the worm drive and governor at 24 ; the governor valve and sleeve, Fig. 405, at 25. The hand speeder gear is shown at 26, and the tachometer at 27.

The reduction gear consists of the high-speed shaft and pinion machined from high quality alloy steel, and a gear wheel having a cast-iron centre piece with a shrunk on forged steel rim. This wheel is hydraulically forced and keyed on to the mild steel shaft. The teeth are of involute form, and double helical.

**337. B.T.H. Tandem Compound Impulse Turbine.**—For outputs of 15,000 kw. and upward the turbine is usually divided into a high-pressure and low-pressure section, to drive a single alternator on one shaft, the sections necessarily being in tandem. A tandem impulse machine to develop 25,000 kw. at 1500 rev./min., with pressure of 215 lb./in. gauge, temperature 650° Fahr., and vacuum 28·3 in., is shown in longitudinal section in Fig. 406. It was recently installed by the British Thomson Houston Co. in Willesden Power Station, London.

The casing 1 of the H.P. section, on the left, and the first part 2 of the L.P. section are made of cast steel ; the second part 3 of the L.P. is made of cast-iron, integral with the exhaust branches 4, which are splayed out on each side and connected to the condenser inlet, in a pit below the floor. The H.P. diaphragms and the first diaphragm of the L.P. are made of cast steel and checked into the casing, the rest of the L.P. diaphragms are of cast-iron. Each casing is jointed along the horizontal centre line.

The high-grade carbon-steel shafts are of large diameter, so that the critical speed is well above the running speed, and are coupled by a flexible claw coupling 5. The generator and L.P. shafts are also connected by a similar coupling 6.

The shafts are stepped to take the impulse wheels, which are made of steel and machined all over. Each wheel is mounted on a bushing sleeve, which is hydraulically pressed between it and the shaft. The wheels are fixed by the keys shown at the centre line. It will be noted that the first H.P. stage wheel 7 is much larger in diameter than the others. This is on account of the larger pressure drop arranged for in the first set of nozzles.

The nozzle passages in the diaphragms are formed by steel plates cast in. The blades of the H.P. wheels and the two first and two last wheels of the L.P. are of steel. The rest of the L.P. blades are bronze. They are fitted into dovetail grooves in the rims. Each steel blade ring has a shrouding strip riveted round it, and the bronze blades have bronze strips. These strips maintain the tip blade spacing, and help to reduce vibration and spilling of the steam over the tips.

Leakage between the stages is prevented by a restrained type of floating labyrinth gland. This consists of a ring of segments,

serrated at the inner circumference and recessed at the outer circumference to fit over the diaphragm sides. Each segment is held on to the shaft liner by a flat spring, but is capable of moving radially. The shaft packing glands 8, 9, 10, and 25 consist of a series of restrained ring segments serrated at the inner circumferences, which fit into grooves in the packing boxes and are held in place on the shaft bushing by flat springs. Each gland is steam packed, a pressure slightly in excess of atmospheric being maintained in each box. Vent pipes are fitted to carry off leakage steam. The standard method of B.T.H. steam distribution is employed. A nozzle box 11 extends across the front of the H.P. cylinder. A shaft 12 carrying a number of cams is mounted in a casing in front of this box. It is operated by an oscillating type of oil servo-motor, controlled by a pilot valve. This valve is adjusted through a linkage by the governor in the casing 13.

The oil pump 14, supplying the servo-motor and the bearings, is driven from the governor spindle, which in turn is driven by the worm gear 15 at the end of the shaft. As the load on the machine increases the servo-motor swings the cam shaft through a progressive angle, opening in succession the valves supplying groups of nozzles.

As the load decreases the reverse action takes place. An emergency trip governor is provided, to close down the supply valve to the nozzle if the machine seriously over-speeds. The bearings 16, 17, 18, and 19 are cast-iron shells lined with Babbit metal. They are spherically seated and self-aligning and are under forced lubrication. Multi-collar thrust blocks which are adjustable by worm gear are fitted on the shafts at 20 and 21, for the regulation of the axial clearances, between the wheel blades and diaphragm faces.

In order to allow for expansion, the L.P. cylinder is anchored at the foot of the exhaust opening, and the casing is free to expand towards the left. The mid-bearing housing 22 can slide on the sole plate and so can the end bearing housing 23. Instead of one large exhaust branch between the H.P. and L.P., two smaller corrugated branch pipes are used, the pipe on the far side being cased in the insulation and cleading at 24.

A somewhat larger two-cylinder turbine by the English Electric Co. was installed in Valley Road Station, Bradford, in 1926.*

**338.** Compared with the very large American turbines put in commission since 1929 this machine may be regarded as of moderate output.

The demands for electrical power, lighting, and heating on the generating stations of this country are, however, very moderate compared with those on the stations supplying the great American cities. So far the maximum unit output has not reached 50,000 kw.

In a recent extension of Barton Power Station, Manchester, a 41,000 kw. Metropolitan-Vickers impulse tandem machine was

* For complete drawings and description, see *Engineering*, 15 Oct., 1926.

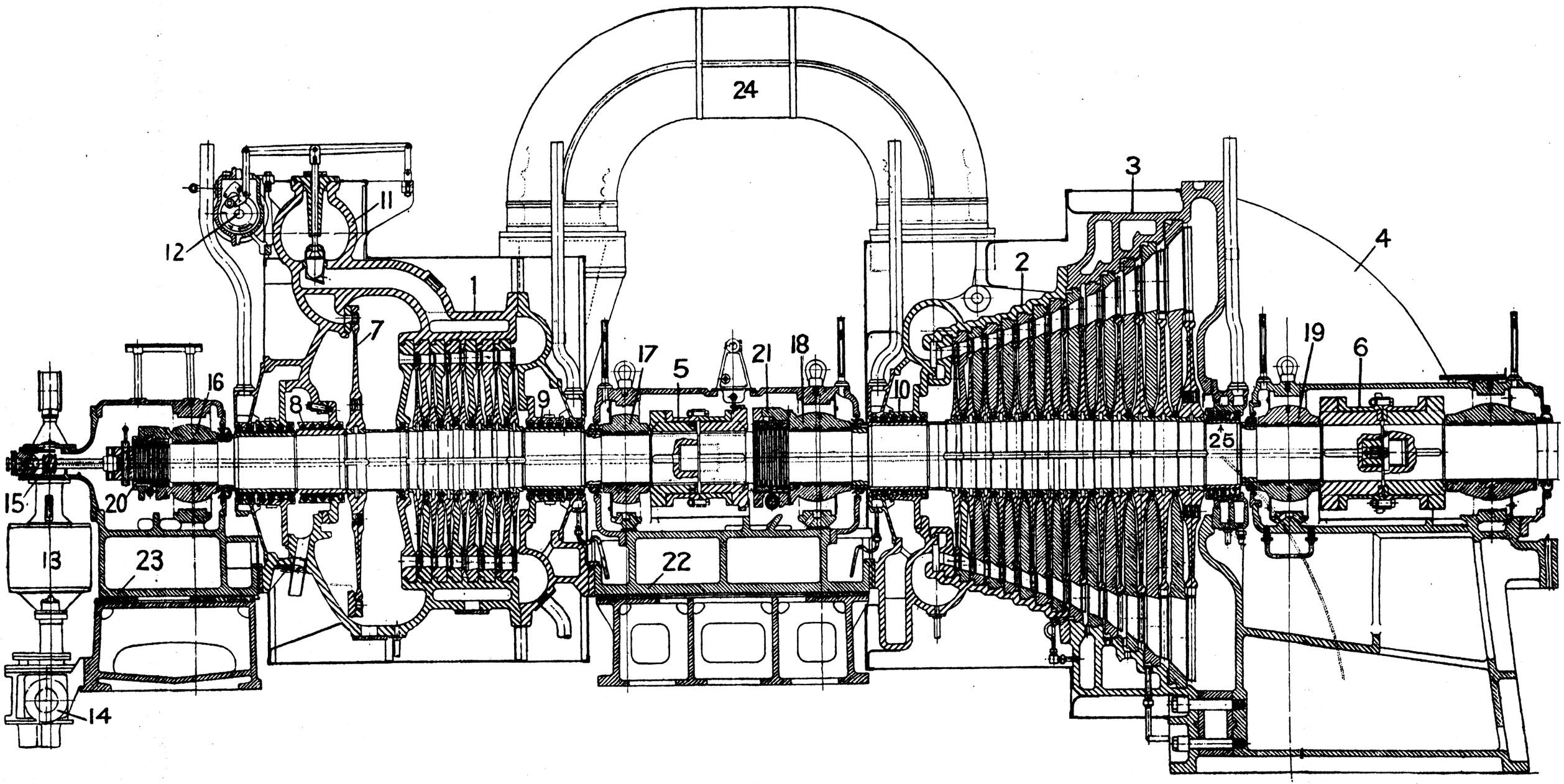



FIG. 406.

[*To face page* 620.

installed.* The steam conditions are 350 lb./in.$^2$, 700° Fahr., and 29 in. vacuum. The machine works on the partial regenerative cycle (see Art. 64).

In the new Dunston " B " Power Station at Newcastle there are to be six 50,000 kw. sets. Three of these, on order, are Parsons (reaction) machines, working on the reheat-regenerative cycle with steam conditions 600 lb./in.$^2$, 800° Fahr., 29 in. vacuum.

In the new Battersea Station of the London Power Co., two 67,200 kw. impulse machines working on the rengerative cycle, one by B.T.H. Co. and one by the M.V. Co., with steam conditions 650 lb./in.$^2$, 900° Fahr., and 29 in. vacuum, are being installed.

In 1928 a big advance in output per unit was made in America. A Westinghouse 110,000 kw. (reaction) machine was installed in Hudson Avenue Station, New York, working on the reheat regenerative cycle with steam conditions 500 lb./in.$^2$, 700° Fahr., and 27·5 vacuum.† In 1929 a Brown Boveri 160,000 kw. (reaction) two-shaft cross compound, and a Westinghouse 160,000 kw. cross compound were installed in Hell Gate Station, New York; while a General Electric 160,000 kw. (impulse) was installed in East River Station, New York, and another G.E.C. 165,000 kw. machine in the Philo Station of the Ohio Power Company.‡

As far as maximum output is concerned the record, to date, is that of the G.E.C. impulse 208,000 kw. three-cylinder cross compound turbine, installed in 1929 in the State Line Station, Hammond, Indiana, on the shore of Lake Michigan.§

In these cases the steam pressure does not exceed 600 lb./in.$^2$ and the temperature runs from 700° Fahr. to 750° Fahr. All these machines may now be classed as " normal " pressure turbines, in comparison with the types which are being evolved to take pressures from 1200 lb./in.$^2$ to 2000 lb./in.$^2$, and steam temperatures 750° Fahr. to 900° Fahr. The latter may be classed as " super-pressure " turbines.‖

It is out of the question, on account of limitation of space, to attempt any detailed description of these recent designs; and the student is referred for full particulars to the articles and papers quoted in the footnotes to this chapter.

**339. G.E.C. Triple Cross Compound Impulse Turbine.**—The State Line G.E.C. 208,000 kw. triple cross compound machine,¶ and G.E.C.

* See paper " Operating Results with Recent Extensions at Barton Power Station," by H. L. Guy and H. C. Lamb, *Proc. Inst. Mech. E.*, March 1930. Also *Engineering*, 20 December, 1929.

† See *Engineering*, 3 May, 1929.

‡ See *Engineering*, 4 January, 1929.

§ For details of the plant of this station, see *Engineering*, 12 February, 4 March, 18 March, 1 April, 1932.

‖ For illustration of super-pressure and other turbines, see paper " Higher Steam Pressures and their Application to the Steam Turbine," by Law and Chittenden, *Proc. Inst. E. E.*, 1927, or *Engineering*, 11 November and 2 and 9 December, 1927. Also *Engineering*, 13 September, 1929.

¶ See *Power*, 29 October, 1929.

super-pressure 110,000 kw. steeple turbine, however, are of special interest and a few general remarks regarding them are desirable.

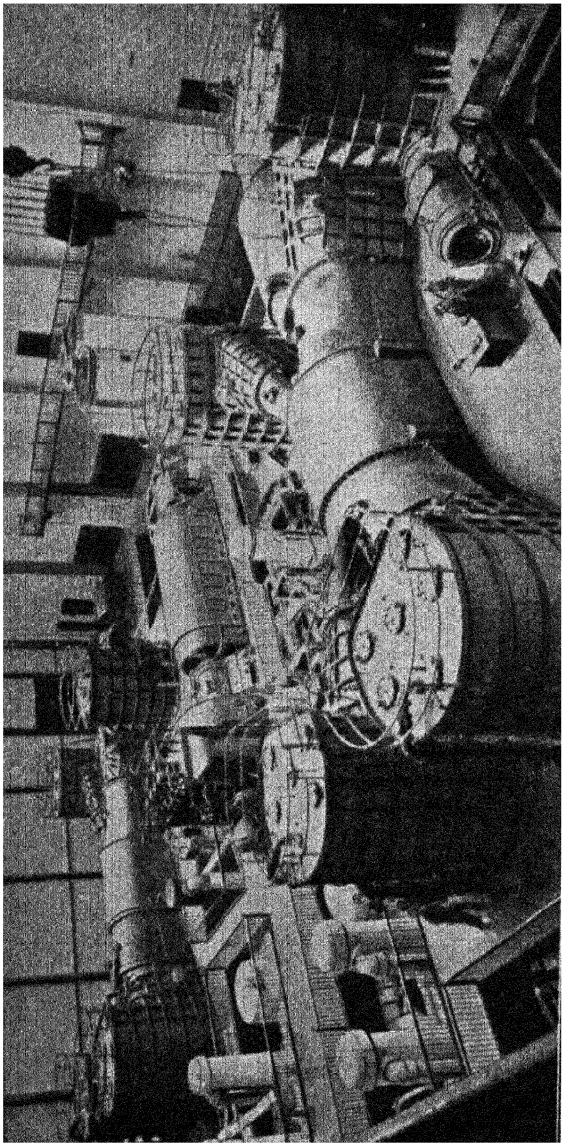


Fig. 407.

The arrangement of the State Line turbine room is shown on the photograph, Fig. 407. The turbine set consists of a H.P. turbine,

in the centre, generating 76,000 kw. and operates between 600 lb./in.$^2$ at 700° Fahr. and 115 lb./in.$^2$ The steam exhausted from this cylinder is reheated to 500° Fahr. and then discharged to two L.P. turbines, one on each side of the H.P., as clearly shown on the photograph. Each L.P. turbine drives a 62,000 kw. generator and a 4000 kw. " house turbine," which supplies current to the station auxiliaries. There are five regenerative heaters, two from the H.P. at 231 lb./in.$^2$ and 140 lb./in.$^2$, one from the H.P. exhaust at 115 lb./in.$^2$, and two from the L.P. turbines at 6 lb./in.$^2$ abs. The water condensate is cascaded through the heaters and returned to the feed line (see Art. 64). Each L.P. element is double flow and exhausts to four vertical single pass condensers, as shown on the photograph at the L.P. turbine in the foreground. This system of vertical single flow condenser was introduced by C. A. Parsons and Co. in the installation of the first 50,000 kw. reaction turbine in Crawford Avenue Station, Chicago, in 1925.

**340. G.E.C. Steeple Compound Impulse Turbine.**—The other G.E.C. machine is a new departure in compound turbine arrangement. It is termed a " steeple " turbine, the high-pressure turbine and generator being superposed on the L.P. A considerable economy of space is thus obtained. A longitudinal view, partly in section, of the steeple compound two-cylinder 110,000 kw. impulse turbine now installed in the Ford Motor Works, Detroit, Michigan, is shown in Fig. 408. The H.P. and L.P. turbines each develop 55,000 kw. at 1800 r.p.m. The first stage of the H.P. turbine, which is supplied with steam at 1200 lb./in.$^2$ and 725° Fahr. is two-velocity compounded, so that the H.P. glands have to pack against a considerably reduced pressure in the first stage chamber. The steam is then expanded in 11 simple impulse stages and exhausted at 85 lb./in.$^2$. It is then reheated, not by return to a flue-gas superheater at the boiler, but by live steam from the supply main at 1200 lb./in.$^2$ and 725° Fahr., and then passed to the double flow L.P. cylinder, in which it is expanded in 10 simple impulse stages and exhausted at the condenser pressure of 29 in. vacuum, by two branches to the surface condenser.

As the whole plant, boiler and turbine installation, is representative of the very latest practice, what is known as the " flow diagram " for it has been drawn out, and is shown in Fig. 409. Starting with the boiler on the right, which is of water-tube type fired by pulverised fuel, the steam, leaving the boiler drum at 1365 lb./in.$^2$ abs., passes through the superheater, dropping in pressure to 1215 lb./in.$^2$ abs., and is superheated to 725° Fahr. As indicated by the directional arrows, part of it is tapped to the reheater, the condensate from which is led at 1215 lb./in.$^2$ abs. to the second last stage of the boiler feed pump. The major supply enters the H.P. cylinder. After expansion the major part of this H.P. supply is exhausted at 85 lb./in.$^2$ abs., through the reheater, and raised to 550° Fahr. A portion is tapped directly, one part of which enters a " raw water "

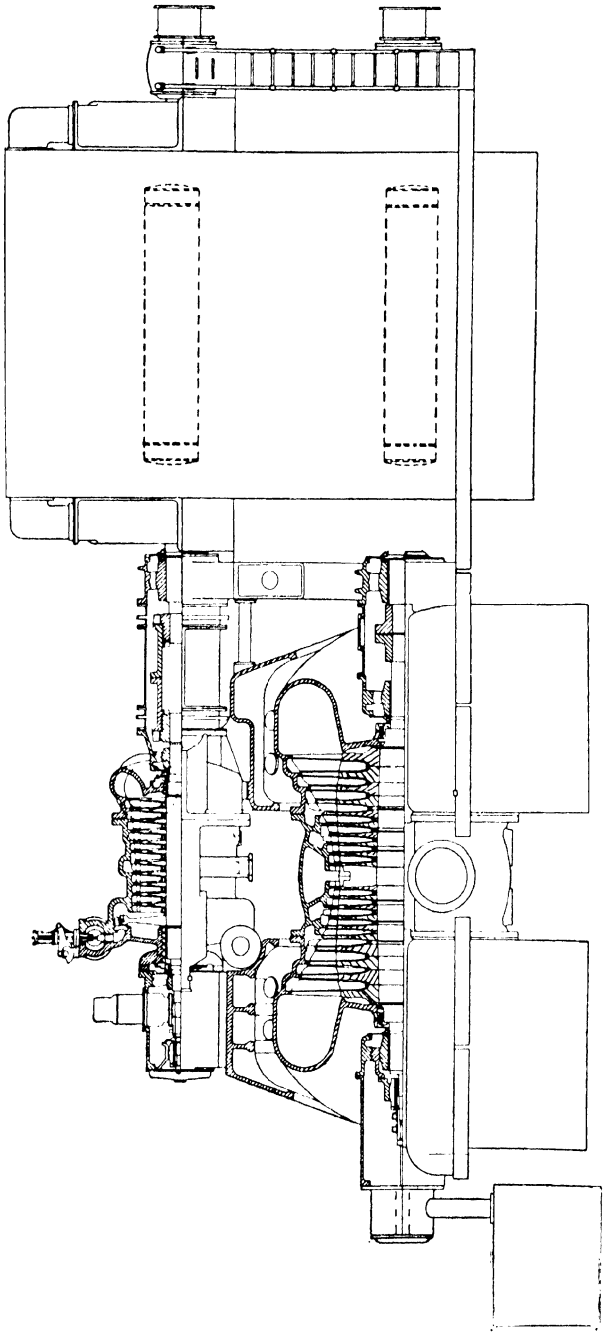


FIG. 408.

evaporator at 81·5 lb./in.$^2$ abs. and the other part enters the third feed heater at 79 lb./in.$^2$ abs. The steam produced in the evaporator at 28·5 lb./in.$^2$ abs. is passed, for make-up feed purposes, to the

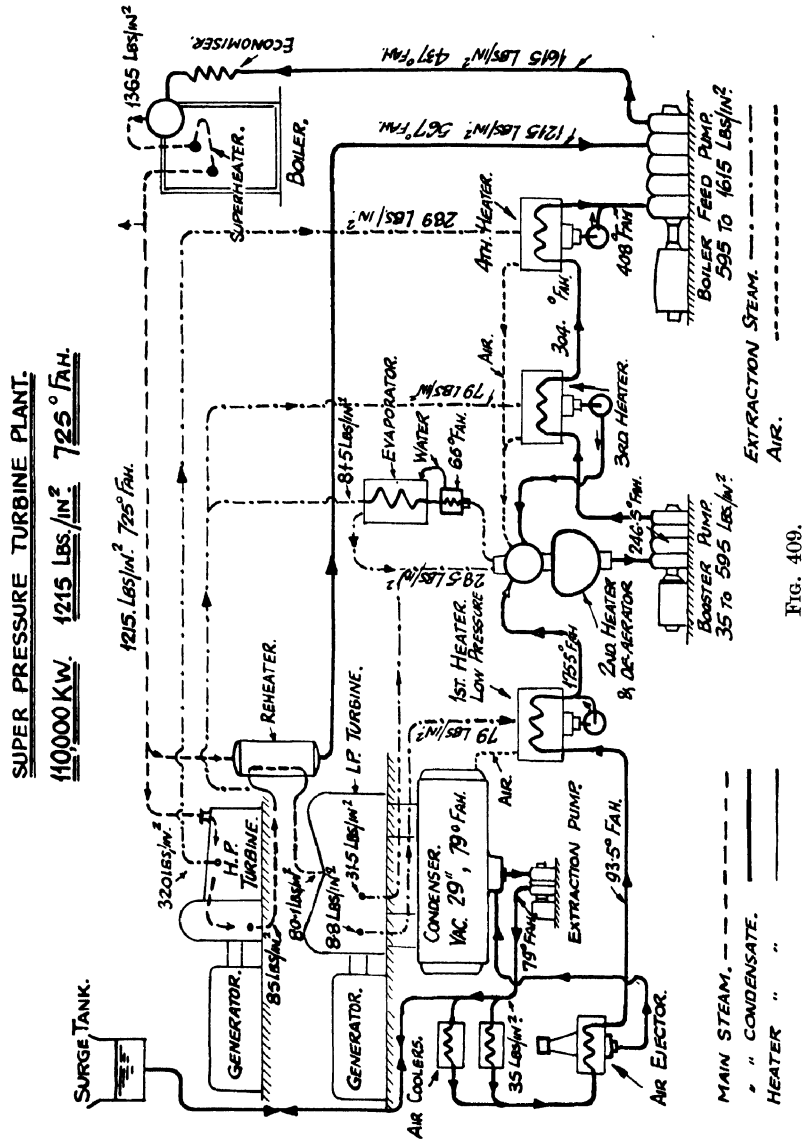



FIG. 409.

second heater and de-aerator ; this is supplemented by steam tapped from the L.P. turbine at 31·5 lb./in.$^2$ abs. Steam is further tapped at the exhaust end of the L.P. at 8·8 lb./in.$^2$ abs. and enters the first

L.P. feed heater. The fourth or H.P. heater is supplied with steam tapped from the H.P. turbine at 320 lb./in.$^2$ abs. The water, fed to the evaporator at 65° Fahr., is preheated by a coil carrying the condensate from the evaporator coils to the second de-aerator heater. The condensate is drawn from the condenser at 29 in. vacuum by the extraction pump, raised in pressure to 35 lb./in.$^2$ abs., and passed through air coolers taking the heated air from the generators, then through a heater taking the exhaust steam of the air ejectors (producing the vacuum). It is raised in the air coolers and ejector heater from 79° Fahr. to 93·5° Fahr. It is then forced through the first or L.P. heater and raised to 175·5° F. It passes through the second heater and de-aerator and is raised to 246·5° Fahr. It is drawn by the four-stage centrifugal pump and raised from 35 lb./in.$^2$ abs. to 595 lb./in.$^2$ abs. It is then discharged through the third heater and raised to 304° Fahr., then through the fourth heater and raised to 408° Fahr. It is raised in pressure from 595 lb./in.$^2$ abs. to 1615 lb./in.$^2$ abs. in a five-stage centrifugal pump, to which the condensate from the reheater is discharged, and is finally discharged to the boiler economiser at 1615 lb./in.$^2$ abs. and 437° Fahr.

The condensate of the first heater is pumped into the feed line, that of the third heater into the second de-aerator heater, and that of the fourth heater to the feed line.

**341. Parsons Reaction Extraction Turbine.** A longitudinal section of a modern design of reaction turbine, by C. A. Parsons and Co., is shown in Fig. 410. It is of the steam extraction type, for use in a paper mill, and is designed for a normal output of 5000 kw. and 95,000 lb. of process steam per hour at 60 lb./in.$^2$ abs. The initial pressure is 300 lb./in.$^2$ gauge, superheat 250° Fahr., and vacuum 29 in.

The forged steel rotor 1 is bored throughout its length, so that it can be examined internally for soundness. The H.P. and L.P. sections are parallel and the mid-length is tapered.

The steel blading on the H.P. section 2 and I.P. section 3 is end-tightened. That on the L.P. section 4 is the usual blading with small radial clearance. It is thinned at the tips, and strengthened against vibration by binding wires, as indicated by the small circles on the blades.

The steam after passing the admission valve, situated outside the casing, and under the control of the speed governor, enters the branch 5 and flows into the H.P. blading. After expansion to the process pressure of 60 lb./in.$^2$ abs. it is exhausted at 6 into the chamber 7 connected with the process steam branch 8. A rotating valve 9 is fitted in the centre of the chamber 7, dividing it into two parts when closed, as shown in the figure. In this position all the steam passing the H.P. and I.P. blading would go to the process main, the L.P. blading running idly in a vacuum, and the turbine would simply function as a back pressure machine. Under normal

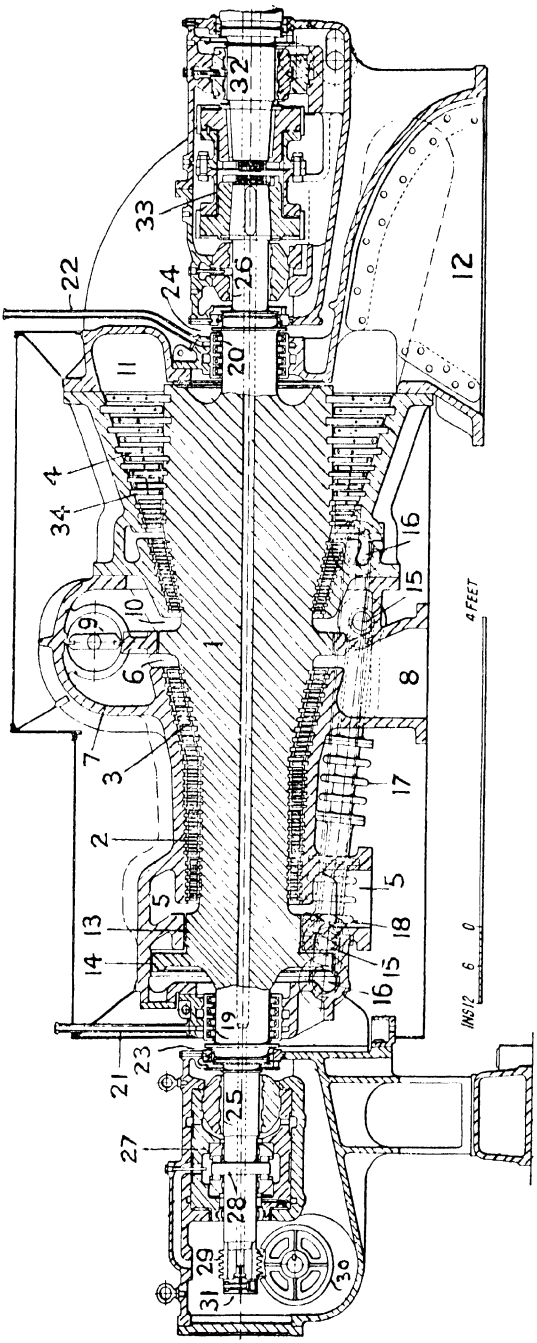

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FIG. 410.

conditions the valve 9 is always more or less open, the opening depending on the demand for process steam. The additional steam is passed into the L.P. blading, expanded between 10 and 11, and exhausted to the condenser through the branch 12. The pass out valve 9 is operated by a servo-motor controlled by the pressure in the process main. The H.P. and I.P. portion of the casing is of cast steel and the L.P. part and the exhaust branch of cast-iron. The H.P. end steam thrust is taken by a dummy piston 13, and the I.P. thrust by the piston 14. The connecting pipes between the backs of the dummies and suitable pressure points in the blading are shown at 15 and 16. An expansion length is provided on each connecting pipe at 17 and 18. The H.P. and I.P. glands 19 and 20

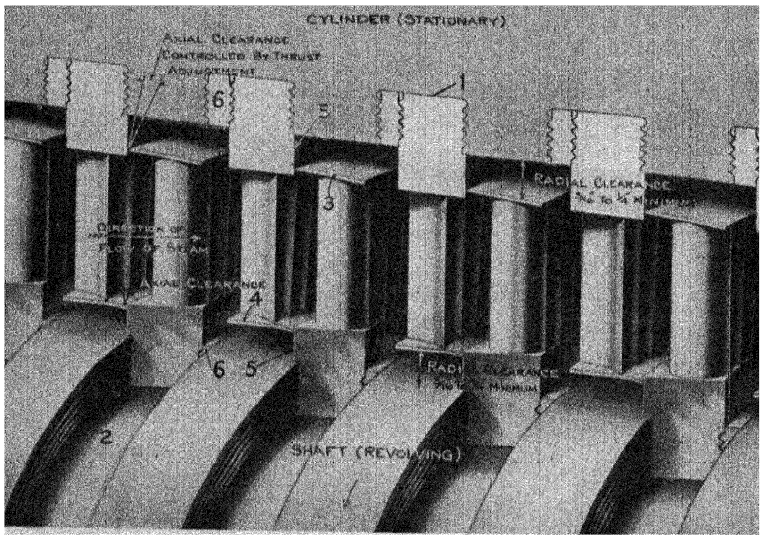



FIG. 411.

consist each of four carbon rings in segments, held on to the shaft surface by garter springs. They are provided with leak-off vents 21 and 22. Fin ring oil catchers 23 and 24 are fitted in front of the glands, to prevent creep of oil from the bearings 25 and 26, which according to standard practice run under forced lubrication. The casting 27 carries a Michell thrust block 28, in addition to the main bearing 25. A steel worm 29 at the end of the shaft drives the worm wheel 30 on the governor shaft. An emergency governor 31 is fitted at the end of the shaft. The outer bearing 32 at the L.P. end takes the generator shaft, which is connected to the turbine shaft by a flexible claw coupling 33. The end-tightened blading extends from 5 to 34.

**342.** A perspective view of this blading is shown in Fig. 411. The

sides of the grooves 1 and 2 in the rotor and stator, into which the blades and distance pieces are fixed, are undercut with V-shaped grooves. The distance pieces 5 between the blades are made narrower than the grooves, so that side locking strips 6 made of copper can be inserted. Each locking strip is rolled with corresponding V serrations on both sides, and is inserted in short lengths (about $\frac{1}{2}$ in. long), pushed circumferentially round the groove, and then hammered tight endwise. There is therefore no top caulking, except where the last three or four packing pieces have to be inserted radially and caulked from above. The modern practice is to use the " group " type of blading. The blades and distance pieces are assembled or strung on a wire into groups of short length, with the roots brazed together, and they are then serrated on each side.

Shroud rings 3 and 4 are riveted round the moving and stationary blade rings. A very small axial clearance is allowed between the edges of these shrouds and the projecting ring faces, formed by the blade distance pieces, while as shown on the photograph, a large radial clearance, $\frac{3}{16}$ in. to $\frac{1}{4}$ in., is given at the casing and rotor surfaces. The pressure of steam and differential expansion of the rotor and casing tend to displace the rotor towards the left, and in order to avoid seizure and blade strip, should any appreciable axial displacement occur, the left side edges of the shrouds of the moving and the right side edges of the shrouds of the stationary blading are thinned down to a fine edge.*

**343. Parsons Three Cylinder Cross Compound 50,000 kw. Reaction Turbine.**—In contrast to this comparatively small single-cylinder industrial machine, the tandem I.P. and L.P. elements of the 50,000 kw. Parsons turbine installed in Crawford Avenue Station, Chicago, in 1925, are shown in sectional elevation in Fig. 412, and a plan of the complete outfit in Fig. 413. This set may be regarded as the pioneer of the large cross compound American turbine outfits already referred to. As seen in the plan, Fig. 413, it consists of a high-pressure element taking steam at 550 lb./in.$^2$ and 750° Fahr., and driving a 15,000 kw. alternator at 1800 r.p.m. It exhausts at 100 lb./in.$^2$ The steam is passed through the vertical pipe shown to the resuperheater at the boiler, from which it is returned at 700° Fahr. by the other vertical pipe to the I.P. element, driving a 30,000 kw. alternator at 1800 r.p.m., and is expanded to 2 lb./in.$^2$ abs., and discharged to the L.P. element through the conical passage. It is then expanded to 29·25 in. vacuum, and discharged through two branches to the twin single-pass surface condensers having a total surface of 56,000 ft.$^2$ The L.P. element drives a 600 kw. alternator at 720 r.p.m.

There are three regenerative heaters, the first and second, supplied with steam extracted from the I.P., and the third with steam from

* For drawings and blade details of a Parsons 40,000 k.w. reaction turbine, see *Engineering*, 15 August, 1930.

the H.P. element. The extracted steam amounts to 22 per cent. of the steam entering the H.P. element, and the feed is heated from 65° Fahr. to 315° Fahr. before it enters the boiler economiser. The conical passage, Fig. 412, between the I.P. and L.P., surrounding the I.P. bearings obviates the use of a large connecting pipe and further conserves the kinetic energy of the I.P. exhaust steam and makes it available in the L.P.

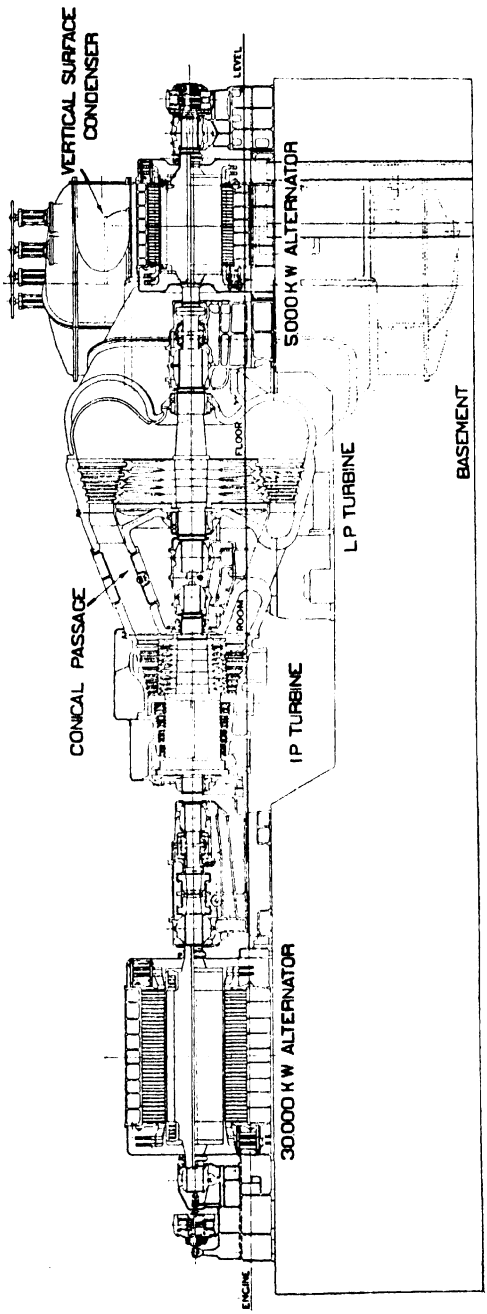



FIG. 412.

As in the case of the impulse type, limitation of space does not permit of the illustration of the various large units ranging from 110,000 to 165,000 kw. recently installed in American power stations. The student is referred, for these, to Prof. A. G. Christie's paper read before the Second World Power Conference in Berlin in 1930.* A paper in German by Prof. A. Stodola dealing with

* " Economic Considerations in the Application of Modern Steam Turbines to Power Generation," reprinted in *Mechanical Engineering* (the Journal of the American Society of Mechanical Engineers) for August 1930. See also *Engineering*, 4 January, 1929.

Continental turbines is also given in the *Transactions* of the Second World Power Conference.

**344. Radial Flow Ljungström Turbine.**—All the foregoing machines are of the purely axial flow type. There is one commercial representative of the radial flow. This is a "double-motion" machine designed in Sweden by the brothers Ljungström in 1912. The nature of the design limited its application to comparatively small outputs, due to the restricted area at exhaust. In recent years the machine has been converted to a combination of radial

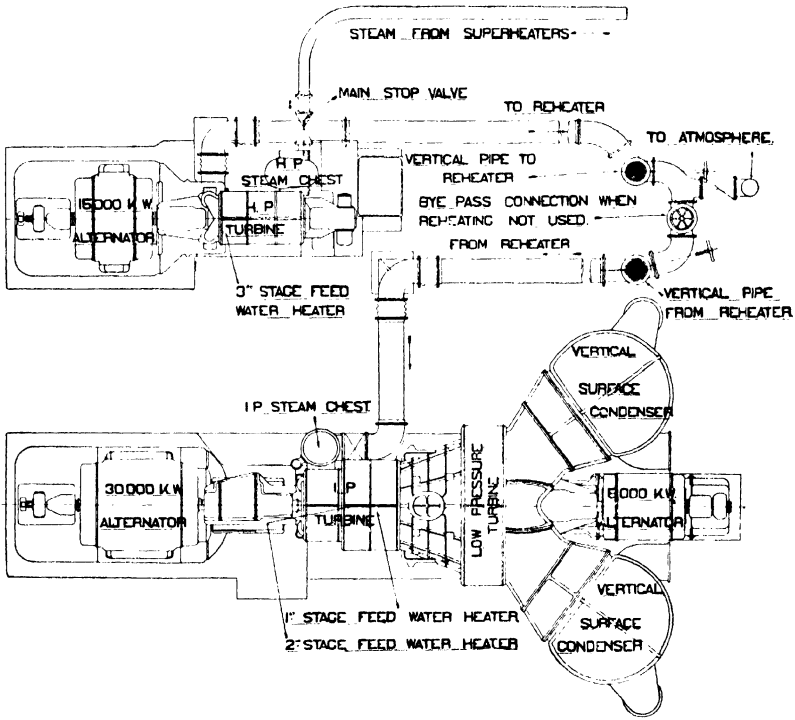



FIG. 413.

and axial flow, by the introduction of two or more axial flow stages at the exhaust end. According to O. A. Winberg of the Stal Co. in Sweden, machines for outputs of 50,000 kw. at 1500 r.p.m. can now be produced, owing to this modification. The largest units, until recently, developed 14,000 kw. at 3000 r.p.m.

A half cross-section of one of these combined radial and axial flow machines,* to develop 14,000 kw. at 3000 r.p.m., with steam

* See "Improvements in the Double Rotation Turbine," by O. A. Winberg, *Power*, 12 November, 1929. For earlier designs, see *Steam Turbines*, by W. J. Goudie.

conditions 200 lb./in.², 600° Fahr., 28 in. vacuum, is shown in Fig. 414. Referring to the right-hand half of the turbine, the steam

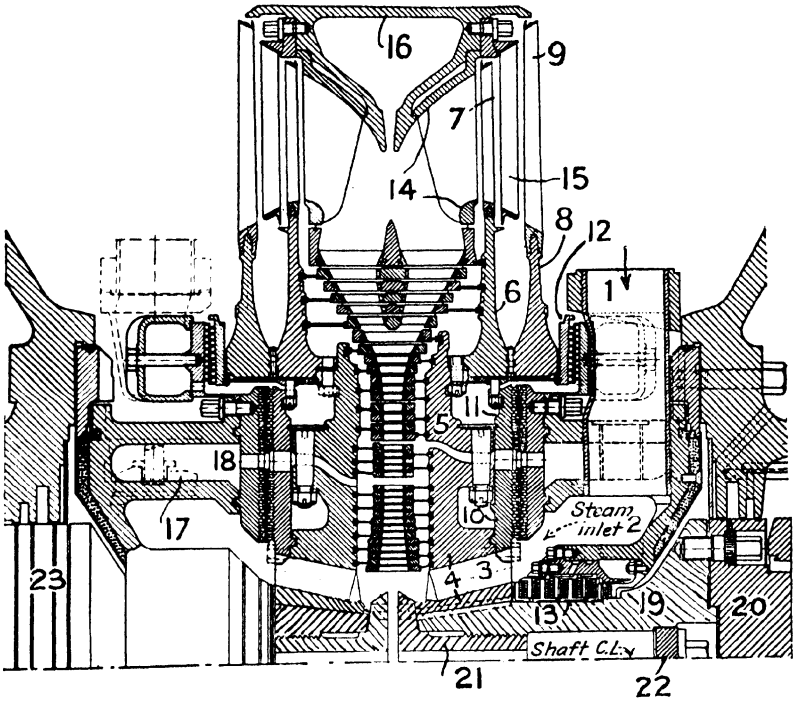



FIG. 414.

is supplied through the passage 1 to an annular belt 2 turned in the steel forging of the body. From this it flows through holes 3, bored in the rotor disc, to the first ring of radial blading, which is mounted on the right-hand disc, the second ring being mounted on the left-hand disc.

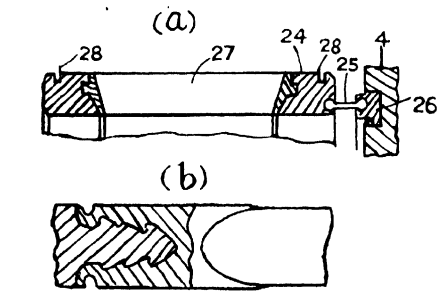



FIG. 415.

The disc is made of forged steel in three parts, 4, 5, and 6, which are flexibly connected together to allow for expansion due to the difference of temperature between inlet and exhaust. Each carrier ring 24 of radial flow blading, Fig. 415 (*a*), is attached to the disc forging 4 by a ring of dumb-bell section 25, caulked into a groove 26 in the disc face, as shown. The blade 27 is made with a dove-

tail root, and the groove in the carrier ring 24 is machined open, and the material is then rolled over, to close it on the dovetail. Originally the blades were welded to the rings. This method precluded the use of alloy steels and non-corrosive metals. The blades are now made as drop forgings of special steel, which ensures much greater strength. Their adoption has reduced the number of sections of the last few rows of radial flow blading from five to two, as shown in Fig. 414. These double-flange blades are carried on the inner disc portion 6 of the rotor, which also carries the inner ring of axial reaction blading 7. The rim is turned with a corrugated ridge, and the forked end of the blade, also corrugated, is sprung over it, then pressed together, and finally caulked at the bottom, as shown on the section at (*b*) Fig. 415. A thin nickel fin 28 is caulked into a groove on the blade carrier ring 24, (*a*) Fig. 415, in order to ensure a very fine radial clearance between the rings without risk of serious seizure if contact takes place. An outer disc 8, registered on disc 6, carries the second L.P. stage axial blading 9.

The rotor parts 4, 5, and 6 carry dummy packing plates 10, 11, and 12 with nickel fin labyrinth packings, which mesh with similar packings on the corresponding plates bolted to the casing. A multi-labyrinth packing 13 is fitted at the shaft to prevent axial leakage of H.P. steam. The castings 14 and 15, forming the reaction guide passages for the axial flow stages, are bolted to the outer casing 16. An overload valve 17 controlled by the governor can admit steam through the passage 18. The shaft 19, which is hollow, is bolted to the flange 20 of the right-hand generator shaft. It is turned conical. The inner portion 4 of the disc is turned to fit and keyed to the shaft. It is held in place axially by the screwed sleeve and flange 21. A screwed plug 22 is used to prevent leakage of steam. This description and reference figures apply equally to the disc of the left-hand side of the turbine, which is bolted to the flange 23 of the left-hand generator shaft. The discs and shafts rotate in opposite directions. The supply of steam is regulated by a balanced double-beat valve operated by an oil relay and servo-motor controlled by a Hartung type of governor, which is driven from the main shaft by a worm gear. An emergency shut-down governor is also provided.*

**345. Marine Turbines.**—The modern marine turbine is either of the single reduction or double reduction type. In the case of the slow-speed cargo vessel having a propeller speed from 70 to 90 r.p.m., double reduction is necessary in order to employ high-speed turbines and obtain a satisfactory turbine efficiency.

The total speed reduction runs from 40 to 1 to 45 to 1. Larger and higher-speed cargo vessels, and high-speed passenger vessels and liners are fitted with single geared turbines. The most recent practice is to group three or four turbines to drive on to a common

* For more detailed description and drawings of this improved type of Ljungström turbine, see *Engineering*, 30 September and 21st October, 1927.

gear wheel and arrange the reductions so that each turbine may be run at its most efficient speed. Single reduction gearing has proved thoroughly satisfactory in service, but many double reduction gears have given trouble. For single reduction the average ratio is 12 to 1, but a maximum value of 26 to 1 has been tried. Both impulse and reaction turbines are in use, but there is a preference for the Parsons type in large cargo and passenger vessels.

**346. M.V. Impulse Marine Turbine Arrangement.**—The plan of a cross compound double reduction set of impulse turbines by

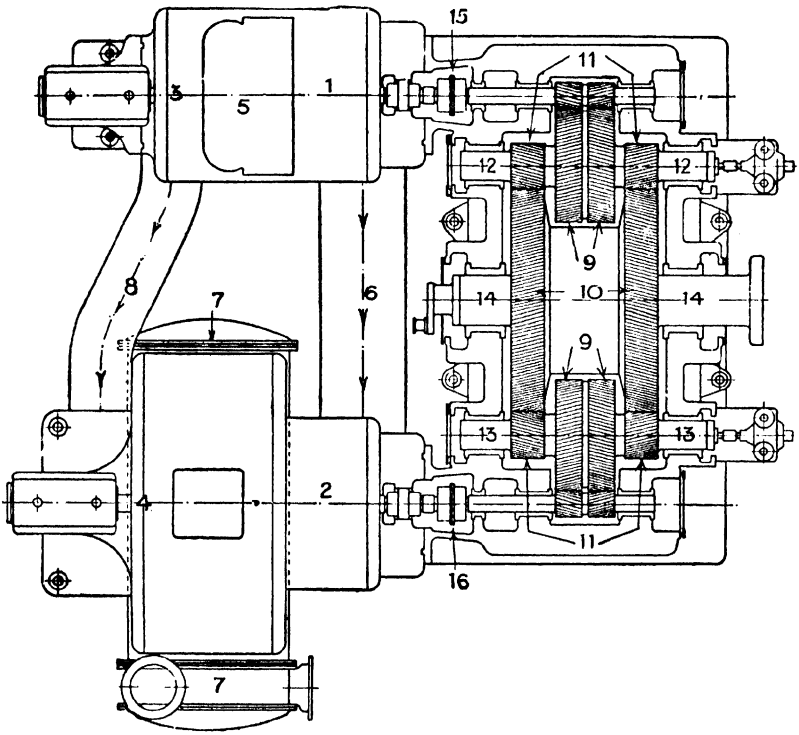



FIG. 416.

Metropolitan-Vickers Co. for a cargo vessel is shown in Fig. 416. It is intended to run with steam conditions 180 lb./in.$^2$ dry and 28 in. vacuum, and to develop 2500 S.H.P. The H.P. turbine 1 has a two-velocity compounded stage and four simple impulse stages. It has also a two-velocity compounded stage at the forward end 3 for astern running. The ahead H.P. turbine exhausts through the pipe 6 to the L.P. ahead turbine 2, which has seven impulse stages and also a two-velocity compounded astern stage at the forward end 4. The H.P. astern stage exhausts to this through the pipe 8. The L.P. turbine exhausts to the surface condenser 7 situated below

it. The steam to the H.P. astern turbine is admitted by a valve underneath the turbine, and to the ahead turbine nozzles, which are in three groups, by three independent stop valves which are situated at the top, below a sheet-metal casing 5. The two astern turbines can develop 70 per cent. of the ahead normal power.

The form of double reduction gear is termed the "interleaved" system. The first reduction wheel 9 is interposed between the halves of the divided second reduction wheel 10. This arrangement enables the whole gearing to be fitted on a single gear box. If the second reduction pinions 11 were not divided and placed aft of the wheels, on an extension of the spindles, to gear with an undivided second reduction wheel, three gear boxes would be required. Although this interleaved arrangement involves a larger gear box it reduces the span of the second reduction H.P. and I.P. pinion shafts, between the bearings 12 and 13, and the second reduction wheel shaft between the bearings 14.

There is thus less liability to bending and faulty gearing of the teeth. The H.P. and L.P. shafts are connected to the first reduction pinion shafts by flexible couplings 15 and 16.

Oil for the bearings and gearing is supplied by two rotary pumps driven from the second reduction gear shaft, which pump it to an overhead tank. It then flows by gravity to the turbine and gear bearings and pinions, and is distributed to the latter by suitably arranged sprayers. Each turbine has an emergency shut-down governor, which can be operated either by hand or foot. The turbine speed is 3000 r.p.m., and the propeller speed 70 r.p.m. The first reduction is 6 to 1 and the second 7·14 to 1, so that the total reduction is 42·84 to 1.

**347. Parsons Cross Compound Marine Turbine Arrangement.**—A plan of the arrangement of a Parsons double reduction geared disc and drum cross compound set is shown in Fig. 417. This set is designed for 2025 S.H.P. maximum, at 3320 r.p.m., with steam conditions 180 lb./in.$^2$ gauge, dry, 28 in. vacuum. The H.P. turbine 1 is placed on the starboard and the L.P. turbine 2 on the port side.

The H.P. turbine rotor carries an ahead two-velocity compounded wheel at the forward end, and a similar wheel for astern running at the after end. The H.P. ahead drum carrying reaction blading is in four steps. The L.P. ahead drum is in two steps and carries eight groups of reaction blading of progressively increasing height. At the L.P. end the drum carries a two-velocity stage of impulse blades and three groups of progressively increasing reaction blades, for astern running. This blading takes the exhaust from the H.P. astern section. The H.P. rotor has no dummy piston, the end thrust being taken by a Michell thrust block. The H.P. forward gland has three carbon rings; the after gland and also the two glands of the L.P. are of the steam-packed labyrinth type. The end thrust

of the L.P. turbine is also taken by a Michell thrust block, at the forward end.

Steam is admitted to the H.P. ahead turbine at 3 and is exhausted through the branch 4 to the L.P. ahead steam inlet 5, as indicated by the directional arrow line. It is exhausted to the condenser through the branch 6 at the bottom. For astern running the steam is admitted to the H.P. at branch 7 and exhausted through branch 8 to the L.P. inlet branch 9, as indicated by the arrow line. It is exhausted to the condenser through the branch 6. Exhaust steam from the auxiliaries can be admitted through branch 10 to the

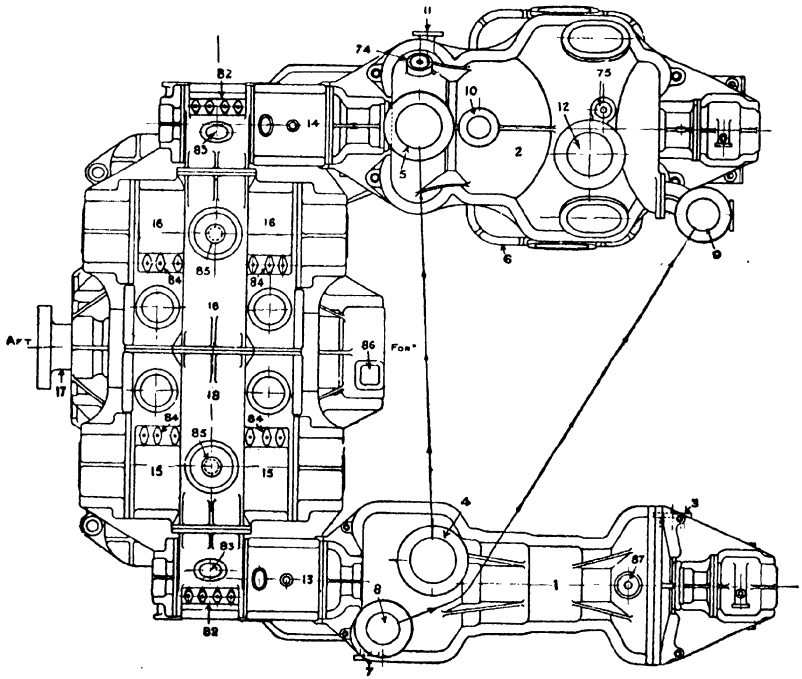



FIG. 417.

reaction blading on the second step of the L.P. ahead drum; and in an emergency reduced steam can be admitted to the L.P. ahead blading through a branch 11 and exhausted through an auxiliary branch 12. Relief valves, to guard against excessive pressure in the L.P. ahead and astern steam chambers, are fitted at 75 and 74. The first reduction wheel and second reduction pinion, on one shaft, run in the casing at 15 and 16, while the second reduction wheel and shaft 17, which is connected to the thrust shaft, run in the central part of the gear case at 18. Oil sprayers are fitted on the first reduction pinion covers of the H.P. and L.P. turbines at 82, two to each half of the divided pinion. Inspection doors are fitted

at 83. Oil sprayers are fitted on each half of the main covers at 84, three to each half of the divided pinion. There are also inspection doors for access to the pinions and sprayers. Vapour pipes open to the atmosphere are fitted at 85. Each turbine shaft is connected to its pinion shaft by a standard Parsons type of claw coupling. The total speed reduction in this case is 44·3 to 1.

**348.** An illustration of the more efficient arrangement of geared marine turbines in which the turbines are grouped round a common gear wheel is shown on the photograph (Fig. 418). This is a three-cylinder or triple-expansion, double-reduction set for a cargo steamer, to develop 3000 S.H.P. at a propeller speed of 80 r.p.m. It consists of a H.P. and an I.P. turbine placed on the port side, and an L.P. turbine placed on the starboard side of the propeller shaft. The axis of the H.P. shaft is in the same horizontal plane as that of

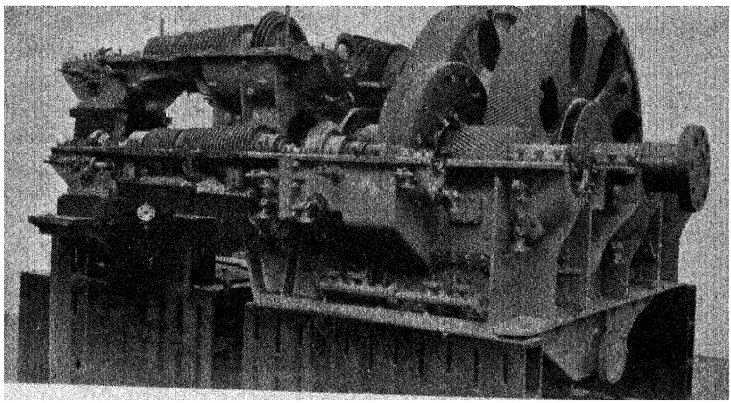


FIG. 418.

the propeller shaft carrying the divided second reduction or main gear wheel. The H.P. pinion drives on the first reduction wheel interleaved with the main wheel. The I.P. turbine is raised above the H.P. with its shaft axis in the same vertical plane as that of the first reduction shaft, and the driving pinion also gears with the first reduction wheel on the top.

The L.P. turbine, just visible on the starboard side, has its shaft axis in the same horizontal plane as that of the I.P., and its pinion gears similarly with a duplicate first motion starboard wheel at the top. Each turbine shaft is coupled to its respective pinion shaft by a standard claw coupling. The pinions and wheel teeth are cut double helical.

The pinions and their shafts are made of nickel steel. The main wheel is of cast-iron with double webs on each half, perforated with eight oval holes. The rim is thickened at each opening, and slotted

through, as clearly shown on the photograph, to relieve it of casting strains. The teeth are cut in a steel ring shrunk on the wheel and locked by tap screws in the sides of the rim. The wheel boss is bored conical to fit the coned end of the second reduction shaft and fixed by a long sunk key and an end nut. Each first reduction wheel is built up of two cast-iron discs clamped together by bolts at the rim and to a deep collar formed on the second reduction pinion shaft.

The teeth are cut on a forged steel ring, shrunk on the rim and locked by side tap screws. The gear reductions are—

| | *1st Reduction.* | *2nd Reduction.* | *Total Reduction.* |
|---|---|---|---|
| H.P. . . | 10·6 | 5·31 | 56·28 |
| I.P. . . | 7·4 | 5·31 | 39·29 |
| L.P. . . | 5·84 | 5·31 | 31·01 |

The H.P. turbine has a three-step solid steel drum carrying four groups of reaction blading ; the I.P. has a four-step solid steel drum also carrying four groups of reaction blading. At its after end it has a high-pressure astern turbine consisting of a two-ring velocity compounded stage ; the L.P. has a composite rotor consisting of a parallel solid steel drum carrying three groups of reaction blading followed by four wheels carrying six groups of reaction blading. The forward part of this rotor carries the L.P. astern turbine blading taking the exhaust of the H.P. astern section. There is a three-velocity compounded impulse stage followed by two wheels carrying three groups of reaction blading.

Dummy pistons have been eliminated in all the turbines ; and a Michell thrust block has been provided at the forward end of each turbine shaft to take the end thrust of the rotor. Also, the H.P. turbine is provided with end-tightened blading. The adoption of this sytem and the elimination of the dummy pistons, through which there is always a certain amount of leakage, appreciably improves the efficiency of the turbines. Carbon glands are fitted to the H.P. and I.P. turbines and steam-packed labyrinth glands to the L.P. The condenser is suspended from the L.P. turbine. An exhaust bend with its expansion troubles is thus avoided, while drainage is facilitated.

**349.** Until 1926 the normal steam pressure used in mercantile vessels was 200 lb./in.$^2$ with from 150° Fahr. to 200° Fahr. superheat. In this year Sir Charles Parsons, in conjunction with Messrs. Denny of Dumbarton and Williamson of Glasgow, made a decided advance to 550 lb./in.$^2$, 700° Fahr., and 29 in. vacuum in the Clyde twin-screw steamer, *King George V.*

This vessel is fitted with hand-fired water-tube boilers and a single-geared turbine installation consisting of a H.P. turbine directly connected to the 1st I.P. turbine, a 2nd I.P. turbine and an L.P. turbine on the port side ; a 1st I.P., 2nd I.P., and L.P. turbine on the starboard side.* Each set is grouped round

* For detailed drawings, see *Engineering*, 10 September, 1926.

the second reduction wheel on the propeller shaft. The H.P. turbine exhaust is discharged to the 1st I.P. port and the 1st I.P. starboard turbine; the I.P. and L.P. turbines of each set are in series. The H.P. and the I.P. turbines run at 6000 r.p.m., and the L.P. turbines at 3000 r.p.m., and the single reduction gears give a propeller shaft speed of 570 r.p.m. The total output is 3500 S.H.P., of which 600 S.H.P. is developed by the H.P. turbine.

End-tightened Monel blading is used in the H.P. and I.P. turbines and the clearance is adjusted by Michell blocks, as in the previous case. The H.P. and I.P. cylinders and nozzle boxes are made of cast steel. An astern section is provided in each first I.P. turbine and exhausts to the astern section of the L.P. turbine. There are two condensers to each turbine, so that one can be opened out for overhaul without stoppage of the plant. The steam pipes are of solid drawn steel with welded joints and large expansion bends.

In this instance the auxiliaries are driven by desuperheated steam reduced to 200 lb./in.$^2$ and 450° Fahr. (slightly superheated). They exhaust at 5 lb./in.$^2$ to a feed heater, and the feed is finally raised from 200° Fahr. to 300° Fahr. by steam extracted from the turbines.

**350.** Sir Charles Parsons' later proposal for an all-steam turbine installation was that of a quadruple screw liner of 160,000 S.H.P., each shaft to be driven by four turbines arranged round the main gear wheel. The auxiliaries to be motor driven from a 3000 kw. turbo-generator of the back pressure type, exhausting to an evaporator, to supply 20,000 lb./hr. of make-up feed, and the surplus steam to be passed to a low-pressure feed heater. The steam conditions proposed were 600 lb./in.$^2$, 600° Fahr., 29 in. vacuum, the steam to be supplied by oil-fired boilers.

Since the experimental outfit in the *King George V* was tried, several of the liners of the C.P.S. fleet have been fitted with the group system of single reduction turbines supplied with moderately high-pressure steam from oil-fired watertube boilers. In the *Duchess* class the total output is 18,000 S.H.P., the steam conditions being 345 lb./in.$^2$, 680° Fahr., 29 in. vacuum. In these cases all the auxiliaries are motor-driven from Diesel generating sets. The make-up feed is obtained by evaporating fresh water (stored in the tank bottom) in Scotch boilers. The steam from these is supplied to the L.P. turbine casing.

Similar installations have been fitted in the high-speed liners *Empress of Japan*, *Empress of Britain*, and the *Statendam*.

In all these modern outfits the feed is heated by exhaust steam and by regenerative heaters to about 300° Fahr., and a considerable proportion of the extracted steam is utilised for ships heating and galley purposes. In the case of the *Statendam* the higher steam conditions are 425 lb./in.$^2$, 650° Fahr., and 29 in. vacuum.

It is probable, in the light of the successful results shown by these recent departures, that much higher pressures will soon become

common in the mercantile marine, and the oil-fired watertube boiler may to a large extent be replaced by the pulverised fuel type. The substitution of pulverised fuel for the more expensive oil would practically reduce the fuel costs about 50 per cent.

**351. Mercury-Steam Turbine Combination.**—The binary vapour cycle has already been discusssed in Chapter IV. The present commercial representative of this type is the 22,900 kw. machine installed in 1928 in the South Meadow Station of the Hartford Electric Light Co., Hartford, Conn. It is the result of fifteen years of experimental work on the part of W. L. R. Emmett of the

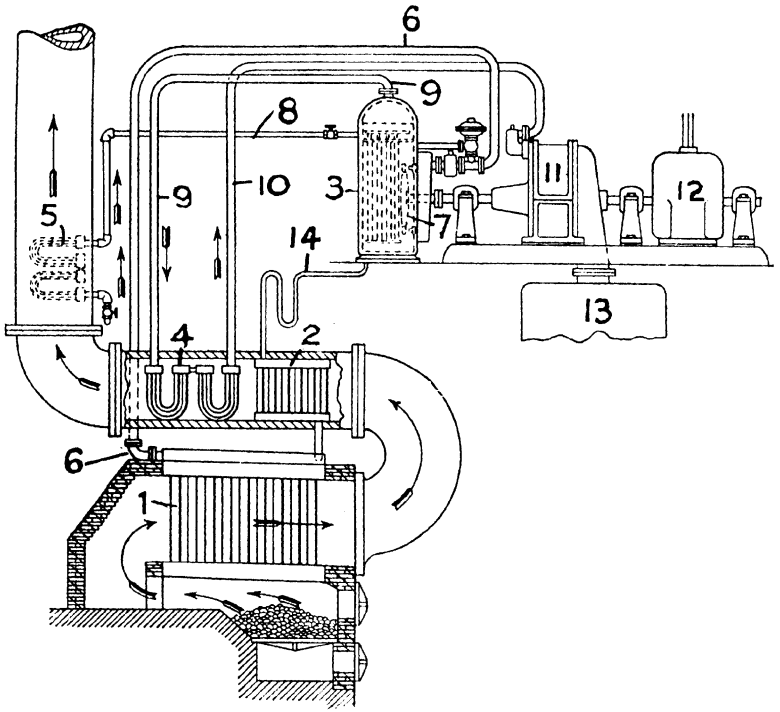



FIG. 419.

G.E.C. Co. A diagrammatic sketch of this interesting plant is shown in Fig. 419, and a section of the mercury turbine in Fig. 420.*

In Fig. 419, 1 is the mercury boiler, which has an air preheater, not indicated here, and is fired with pulverised fuel, 2 is a mercury economiser, in which the liquid mercury is heated in its passage from the mercury condenser (or steam boiler) 3. The flue gases after passing through this mercury heater pass over the steam super-

* See paper *Proc. Inst. Mech. E.*, "The Use of Mercury for Power Production," by F. Samuelson, May 1930. For later particulars and detail drawings, see *Engineering*, 16 January, 1931.

heater coils 4, and finally over the coils of a feed water economiser 5. The mercury vapour at a pressure of 70 lb./in.$^2$ and temperature of 880° Fahr. passes through the pipe 6 to the mercury turbine 7. It is exhausted into the mercury condenser 3 at 28·25 in. vacuum, and is condensed in passing over the heat exchanger tubes, supplied with feed water through the pipe 8. The steam generated at 280 lb./in.$^2$ and amounting to 129,000 lb./hr. is passed through the pipe 9 to the superheater 4 and raised to 735° Fahr., and from this

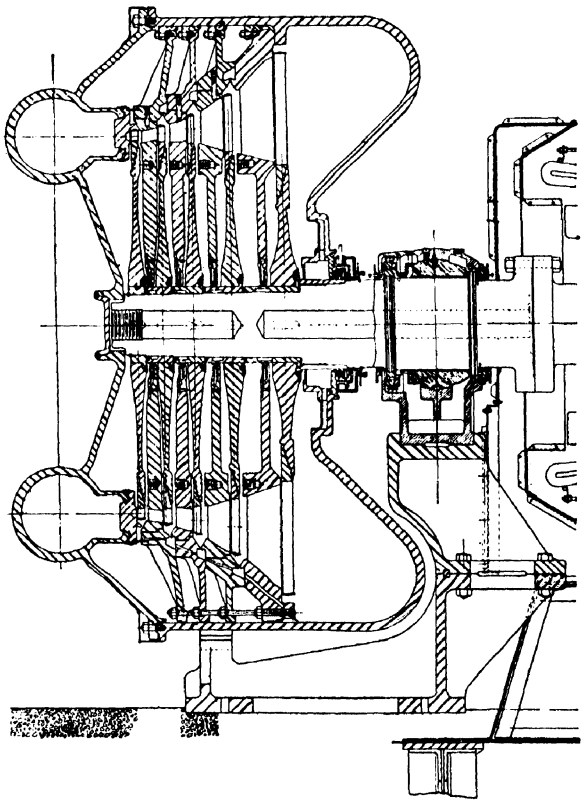


FIG. 420.

by pipe 10 to a normal type of impulse steam turbine 11 coupled to the mercury turbine and driving an electrical generator 12 and exhausting to a condenser 13. The liquid mercury flows by gravity from the condenser to the boiler, through the pipe 14, the head being sufficient to overcome the boiler pressure.

The mercury boiler consists of seven horizontal cylinders each 30 in. diameter by $3\frac{1}{2}$ in. thick by 21 ft. 6 in. long, into which rows of $2\frac{7}{8}$ in. diameter tubes are fixed and project vertically into the furnace. These are of the Field type, with internal circulating tubes

$\frac{5}{8}$ in. diameter. The mercury flows down the central tube and up the annular channel while it is evaporating.

Since mercury vapour is very poisonous, all the joints in the piping and the boiler, etc., are welded.

The turbine, as shown in section, Fig. 420, has five impulse stages, and develops 10,000 kw. at 720 r.p.m. The casing is cast in one piece, and there is no horizontal joint. The only joint is the circumferential one at the end cover. This joint is sealed by a sheet metal cover electrically welded to casing and main cover, and which has to be cut before the joint can be broken. The five disc wheels are carried on the overhung extension of the generator shaft. This arrangement gets rid of a gland at the H.P. end.

At the exhaust end, where the pressure is sub-atmospheric, there is no danger of mercury leakage. A ring of Bakelit fibre is pressed against a flange fixed on the shaft, and is sealed and lubricated by water delivered through flexible connections. The fitting is so arranged that no water can enter the turbine and a perfect vacuum seal is said to be obtained.

Emmett advances the opinion that this mercury steam combination will find its main field of application in the propulsion of ships, reducing the fuel consumption to one half of that now required under present low-pressure conditions, and effecting a large reduction in machinery space. The high first cost of mercury required, however, has to be offset against these advantages.

# CHAPTER XXI

## ENERGY CONVERSION IN TURBINE NOZZLES AND BLADING

In order to arrive at a clear understanding of the conversion of the heat energy of the steam into mechanical work at the shaft of the turbine, it is necessary to consider: (*a*) the expansion of steam in nozzle passages; (*b*) the transformation of the kinetic energy of the high velocity steam jet into mechanical work on the rotor blading.

**352. Equation of Steady Flow of a Fluid.**—The nozzle flow conditions can be easily and quickly deduced from a consideration of the general case of steady flow of any fluid as follows: suppose the fluid flows steadily into a chamber, Fig. 421, at 1, and out at 2, the initial inlet conditions of pressure, volume, temperature and velocity, $p$, $v$, T, V, being denoted by subscript 1 and the final exit conditions by 2. The chamber may contain any appliance for conversion of the energy of the fluid into work. With continuous and constant flow, unit mass must leave at 2 as unit mass enters at 1.

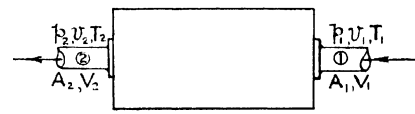


FIG. 421.

At entrance each unit possesses potential or pressure energy $\frac{144}{J}p_1v_1$; internal molecular energy $I_1$, and translational kinetic energy $\frac{V_1^2}{2gJ}$, all these quantities being in heat units.

The total energy at entrance is thus $I_1+\frac{144p_1v_1}{J}+\frac{V_1^2}{2gJ}=H_1+\frac{V_1^2}{2gJ}$.

Similarly at exit the total energy will be $I_2+\frac{144p_2v_2}{J}+\frac{V_2^2}{2gJ}$ $=H_2+\frac{V_2^2}{2gJ}$, since by definition the total heat is $H=I+\frac{144pv}{J}$.

In passing between 1 and 2 heat may be lost, and work may be done in the chamber at the expense of the energy of the fluid. If all these occurrences take place, and $h_e$ is the heat lost while $\frac{E}{J}$ is the heat equivalent of the work done, then by the law of the conservation of energy, the difference of the total energy at

entrance and exit will be the sum of the heat lost and work done. Thus—

$$\left(H_1+\frac{V_1^2}{2gJ}\right)-\left(H_2+\frac{V_2^2}{2gJ}\right)=\frac{E}{J}+h_e \quad . \quad . \quad . \quad (1)$$

or

$$(H_1-H_2)=\frac{V_2^2}{2gJ}-\frac{V_1^2}{2gJ}+\frac{E}{J}+h_e \quad . \quad . \quad . \quad . \quad (2)$$

Now suppose that the chamber is replaced by a length of pipe of varying cross-section, between 1 and 2, and no work is done, then $\frac{E}{J}=0$. Further, if no heat is lost through the wall, that is, if the flow is adiabatic, then $h_e=0$. Hence, for adiabatic flow,

$$(H_1-H_2)=\frac{V_2^2-V_1^2}{2gJ} \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

or the difference of the total heats at 1 and 2 is equal to the change of kinetic energy of the mass flowing. This statement is equally true whether the flow is isentropic or non-isentropic; the sum total of the energy at exit remains the same, only the total heat $H_2$ is increased and the kinetic energy $\frac{V_2^2}{2gJ}$ is proportionately decreased.

**353. Flow of Steam through Nozzle Passages.**—The foregoing condition holds in a nozzle passage, which is simply a short pipe length of varying cross-section in which the steam is expanded between the pressure limits $p_1$ and $p_2$. It may be straight or curved, convergent or convergent-divergent. The primary purpose of such a passage is to convert, between the pressure limits, as much of the entering energy of the fluid as possible into kinetic energy at exit, that is, to obtain a maximum velocity $V_2$ at exit.

In the ideal case the flow is adiabatic and non-resisted, that is, completely insentropic. Actually, in all cases, it is non-isentropic. The adiabatic condition, on account of the rapid passage of the steam through the nozzle, is practically fulfilled, but there is always shock and eddy loss and skin friction loss in the nozzle channel, producing "reheat" or reconversion of kinetic energy to heat. The result is that the total heat of the steam at exit is greater than the value with purely isentropic expansion.

The difference of total heats $(H_1-H_2)$ in the general statement, Equation 2, is defined, as already pointed out in Chapter III, as a "heat drop." For ideal conditions, that is, isentropic expansion, this is termed the "adiabatic" heat drop $H_\phi$ in the nozzle. For the actual condition of adiabatic and resisted expansion it is termed the "net" heat drop, $H_N$.

The general expression can thus be written in the form—

$$H_\phi=\frac{V_2'^2-V_1^2}{2gJ}$$

and

$$V_2'^2=V_1^2+2gJH_\phi \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

where $V'_2$ is the theoretical velocity of exit and $H_\phi$ is the adiabatic heat drop calculated for the given steam conditions, by the methods already given in Chapter IV.

When the heat values are taken in B.Th.U., $2gJ=(223{\cdot}7)^2$, so that the theoretical exit velocity is given by—

$$V_2'=\sqrt{(22{\cdot}37)^2H_\phi+V_1^2}$$
$$=223{\cdot}7\sqrt{H_\phi+\left(\frac{V_1}{223{\cdot}7}\right)^2} \quad . \quad . \quad . \quad . \quad . \quad (5)$$

Due to the frictional effects in the nozzle, an amount of energy $h_f$ is reconverted to heat during the expansion, reducing the quantity of available energy under the root sign of (5); the quality at exit is increased and the exit velocity $V_2$ is less than $V_2'$.

**354. Energy Efficiency of a Nozzle.**—In dealing with the practical design of nozzles, it is customary to express the reheat $h_f$ as a fraction of the theoretical energy at exit under the conditions of adiabatic and non-resisted expansion, that is, as a fraction of the sum of the adiabatic heat drop $H_\phi$ and the initial kinetic energy at entrance $\left(\frac{V_1}{223{\cdot}7}\right)^2$.

Thus, friction reheat is—

$$h_f=f\left[H_\phi+\left(\frac{V_1}{223{\cdot}7}\right)^2\right] \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

where $f$ may be termed the friction factor for the nozzle. The quantity $(1-f)=\eta_n$ is the fraction of the ideally available kinetic energy which is actually obtained at exit.

It is called the energy efficiency of the nozzle, and is given to a very close degree of approximation by—

$$\eta_n=\left(\frac{V_2}{V_2'}\right)^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

The actual velocity of exit is thus given by—

$$V_2=223{\cdot}7\sqrt{\eta_n\left[H_\phi+\left(\frac{V_1}{223{\cdot}7}\right)^2\right]} \quad . \quad . \quad . \quad . \quad (8)$$

This is the general expression. In the majority of cases $V_1$ is negligibly small compared with $V_2$, and the simpler expression—

$$V_2=223{\cdot}7\sqrt{\eta_nH_\phi} \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

can be used, where $\eta_nH_\phi=H_N$, the net heat drop.

The experimental determination of the energy efficiencies, for nozzles of practical form, as used in steam turbines, was the subject of an exhaustive research by a committee of the Institution of Mechanical Engineers. This investigation extended from 1921 to 1930, and the committee issued six reports, dealing with the measurement of nozzle efficiency and "the manner in which it is affected by changes in the shape of the nozzle, and in the velocity pressure

and temperature of the steam flowing through it." The majority of the tests carried out were made on curved nozzles, approximating to commercial types; but various examples of straight-axis nozzles were also tested when such forms proved more convenient for the investigation of particular problems.

A review of the mass of experimental data obtained and the discussions on these six reports cannot be attempted here, owing to limitation of space, but the student can refer to the reports in the Institution Proceedings, quoted in the footnote.*

There is one point, however, arising out of the sixth report which it is advisable to mention, as it has since given rise to considerable discussion.† The experimental method adopted by the Committee was based on the "impulse" principle. The momentum of a steam jet issuing from a fixed nozzle was deduced from the force exerted on a flat plate, placed in the path of the jet, at right angles to it, and the velocity of efflux, per unit mass flowing, was derived from the momentum. The ratio of the efflux velocity to the theoretical velocity, due to adiabatic expansion, is usually defined as the "velocity coefficient" of the nozzle, and this is assumed to be given by the square root of the energy efficiency or by $\sqrt{\eta_n}$. The assumption, however, is true only if the velocity across the section of the issuing jet is the same at all points, that is, if it is uniform. If there is appreciable variation of velocity, then the velocity coefficient, obtained from the overall value deduced from a momentum measurement, is less than the square root of the energy efficiency. The efficiency understood here, is the ratio of the total kinetic energy at exit to the adiabatic energy. Actually the velocity in a jet is never uniform, so that the real energy efficiency is always slightly higher than the square of the (impulse) velocity-coefficient. It would appear, from the paper by Hodkinson and Devey, that the impulse velocity co-efficient may vary from $99{\cdot}7\sqrt{\eta_n}$ to $99{\cdot}1\ \sqrt{\eta_n}$, according to the type of velocity distribution across the jet section.

For a detailed examination of this controversial subject, the student is referred to the two papers and the discussions on them quoted in the footnote.

In order to estimate the actual condition of the steam at exit from a nozzle an extension of the methods of calculation used in Chapter IV is necessary.

* First Report, *Proc. Inst. Mech. E.*, 1923, vol. i, p. 1. Second Report, *ibid.*, 1923, vol. i, p. 311. Third Report, *ibid.*, 1924, vol. i, p. 455; 1924, vol. ii, p. 715. Fourth Report, *ibid.*, 1925, vol. ii, p. 747. Fifth Report, *ibid.*, 1928, vol. i, p. 31. Sixth Report, *ibid.*, 1930, vol. i, p. 215.

† See papers on: "Significance of the Term 'Efficiency' as Applied to Steam Nozzles," by J. C. Oakden, M.A., M.Sc.; "The Amount of Decrease of Nozzle Efficiency caused by Non-Uniformity of Velocity Distribution," by B. Hodkinson, M.Sc., and A. E. Devey, B.Sc., *Proc. Inst. Mech. E.*, May 1931, pp. 603 to 642.

The following cases may arise, steam: (*a*) initially and finally superheated; (*b*) initially superheated and finally dry or wet; (*c*) initially dry or wet and finally wet.

**355.** (*a*) *Steam Initially and Finally Superheated.*—The HΦ

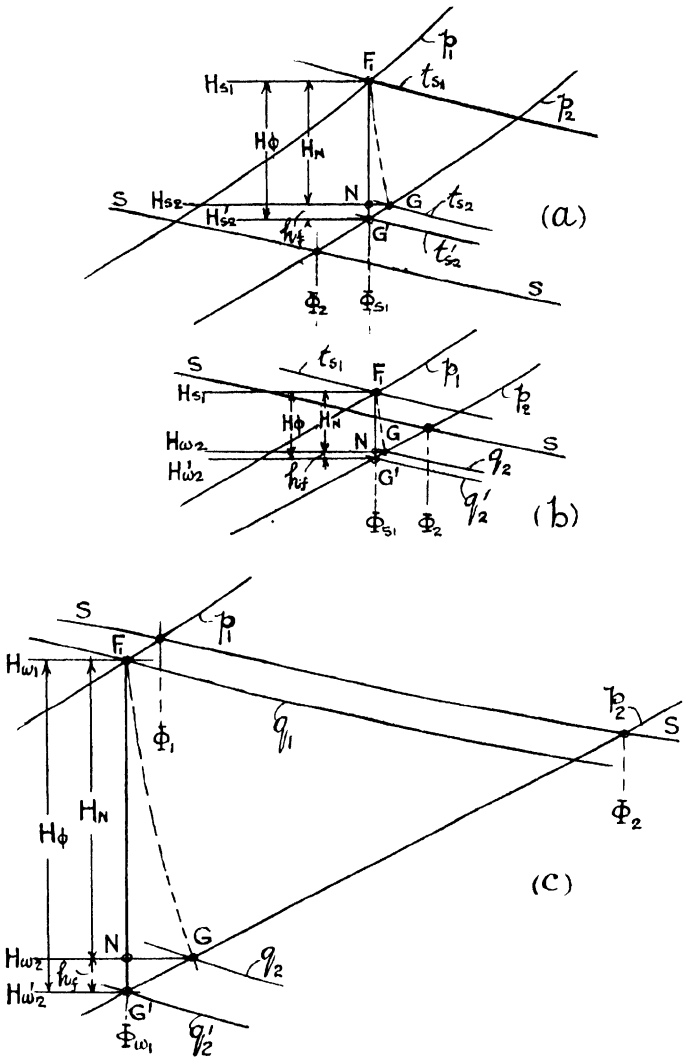



FIG. 422.

diagram for this case is shown in Fig. 422 (*a*), the lettering and notation being that of Chapter III. The initial state point is $F_1$ at $p_1$ and $t_{s1}$; the ideal final state point is G′ at $p_2$ and $t_s'_2$; the actual final state point is G at $p_2$ and $t_{s2}$; the adiabatic heat drop

is $F_1G'=H_\phi=(H_{s1}-H_{s'2})$; the virtual net heat drop is $F_1N=(F_1G'-G'N)=(H_\phi-h_f)=H_N$, where

$$h_f=f\left[H_\phi+\left(\frac{V_1}{223\cdot7}\right)^2\right]=(1-\eta_n)\left[H_\phi+\left(\frac{V_1}{223\cdot7}\right)^2\right] \quad . \quad (10)$$

or
$$H_N=\eta_n\left[H_\phi+\left(\frac{V_1}{223\cdot7}\right)^2\right] \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$$

In order to ascertain, with given initial conditions, whether G' falls in the superheat or saturation field, find first the initial total entropy $\Phi_{s1}$ and the dry steam entropy $\Phi_2$ at $p_2$. If $\Phi_{s1}>\Phi_2$ the steam is finally superheated, and the final ideal temperature is given by—

$$T_{s'2}=T_{s1}\left(\frac{p_2}{p_1}\right)^{\frac{3}{13}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (12)$$

$$t_{s'2}=(T_{s'2}-T_2) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (13)$$

Find $c_{p'2}$ from specific heat chart, and—

$$H_{s'2}=H_2+c_{p'2}t_{s'2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (14)$$

and
$$H_\phi=(H_{s1}-H_{s'2}) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (15)$$

When a large scale HΦ diagram is available $t_{s'2}$ and $H_\phi$ can be ascertained by inspection, with sufficient accuracy.

Given a suitable nozzle efficiency $\eta_n$, the exit velocity is then calculable from Equation 8.

The actual condition of superheat $t_{s2}$, however, is required, to determine the specific volume $v_{s2}$.

If, as a first approximation, $c_{p2}$ is taken the same as $c_{p'2}$, then the actual final temperature is given by—

$$T_{s2}=T_{s'2}+\frac{h_f}{c_{p2}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (16)$$

or
$$T_{s2}=T_{s'2}+\left(\frac{1-\eta_n}{c_{p2}}\right)\left[H_\phi+\left(\frac{V_1}{223\cdot7}\right)^2\right] \quad . \quad . \quad . \quad (17)$$

and
$$t_{s2}=(T_{s2}-T_2) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (18)$$

Alternatively the actual superheat is given by—

$$t_{s2}=\frac{(H_{s1}-H_2)-\eta_n\left[H_\phi+\left(\frac{V_1}{223\cdot7}\right)^2\right]}{c_{p2}} \quad . \quad . \quad . \quad (19)$$

The actual specific volume $v_{s2}$ can then be obtained by interpolation from the steam tables or by the approximate equation—

$$\left.\begin{array}{l} v_{s2}=v_2r \\ \text{or} \quad v_{s2}=v_2(1+0\cdot0015t_s) \end{array}\right\} \quad . \quad . \quad . \quad . \quad . \quad (20)$$

the volume factor $r$ being taken from the curve, Fig. 15, for pressures above 400 lb./in.$^2$ abs.

For a very accurate estimation the Callendar equation may be used, thus—

$$v_{s2}=\frac{1\cdot2464}{p}(H_{s2}-835\cdot2)-0\cdot00216 \quad . \quad . \quad . \quad (21)$$

where $H_{s2}=(H_{s'2}+h_f)$.

Instead of calculation, the value of $G'N=h_f$ can be scaled off on the H$\Phi$ diagram and $t_{s2}$ read directly.

*Example* 1.—Steam at 150 lb./in.$^2$ abs. and superheated 200° Fahr. is expanded in a nozzle to 50 lb./in.$^2$ abs., the nozzle efficiency being 0·92. Calculate the exit velocity and the final quality and specific volume of the steam. Check the results on the H$\Phi$ diagram.

$p_1=150$, $t_{s1}=200$, $T_1=818$, $T_{s1}=1018$, $H_1=1198{\cdot}2$, $c_{p1}=0{\cdot}55$ (chart).

$$H_{s1}=H_1+c_{p1}t_{s1}=1198{\cdot}2+0{\cdot}55\times200=1308{\cdot}2 \text{ B.Th.U./lb.}$$

This checks on the H$\Phi$ diagram.

Also $\Phi_{s1}=\Phi_1+c_{p1}\log_e\dfrac{T_{s1}}{T_1}$

$=1{\cdot}5753+0{\cdot}55\times2{\cdot}3\log 1{\cdot}24$

$=1{\cdot}5753+0{\cdot}55\times2{\cdot}3\times0{\cdot}0969$

$=1{\cdot}5753+0{\cdot}123=1{\cdot}698$

$T_{s1}=1018$, $T_1=818$, $\dfrac{T_{s1}}{T_1}=1{\cdot}24$, $\Phi_1=1{\cdot}5753$

This checks on the H$\Phi$ diagram.

$\Phi_2=1{\cdot}6619$, so that the steam is superheated at $p_2=50$.

$$T_s'_2=T_{s1}\left(\frac{p_2}{p_1}\right)^{\frac{3}{13}}=\frac{1018}{3^{\frac{3}{13}}}=\frac{1018}{1{\cdot}288}=791^\circ \text{ F. abs.} \qquad T_2=741$$

$\therefore t_s'_2=(T_s'_2-T_2)=(791-441)=50^\circ$ Fahr., which again checks on the H$\Phi$ diagram. From chart $c_p'_2=0{\cdot}525$. $H_2=1176{\cdot}4$

$H_s'_2=H_2+c_p'_2t_s'_2$

$=1176{\cdot}4+0{\cdot}525\times50=1176{\cdot}4+26{\cdot}25=1202{\cdot}7$ B.Th.U./lb.

which also checks on the diagram.

Then $H_\phi=(H_{s1}-H_s'_2)=(1308{\cdot}2-1202{\cdot}7)=105{\cdot}5$ B.Th.U./lb.

Exit velocity, $V_2=223{\cdot}7\sqrt{\eta_n\left[H_\phi+\left(\dfrac{V_1}{223{\cdot}7}\right)^2\right]}$ $\eta_n=0{\cdot}92$, $V_1=0$

$=223{\cdot}7\sqrt{0{\cdot}92\times105{\cdot}5}=223{\cdot}7\times9{\cdot}85=2200$ ft./sec.

Final quality $t_{s2}=(T_{s2}-T_2)$

$$T_{s2}=T_{s2}'+\frac{(1-\eta_n)}{c_{p2}}\left[H_\phi+\left(\frac{V_1}{223{\cdot}7}\right)^2\right]$$

$$=791+\frac{0{\cdot}08\times105{\cdot}5}{0{\cdot}525}$$

$=791+16{\cdot}1=807^\circ$ F. abs.$=347^\circ$ F.

$t_{s2}=(807-741)=66^\circ$ F.

which checks closely on the H$\Phi$ diagram.

Specific volume $v_{s2}=v_2(1+0{\cdot}0015t_{s2})$ $v_2=8{\cdot}52$

$=8{\cdot}52(1+0{\cdot}0015\times66)$

$=8{\cdot}52\times1{\cdot}099=9{\cdot}364$ ft.$^3$/lb.

**356.** (*b*) *Steam Superheated and Finally Dry or Wet.*—When the entropy test is made and it is found that $\Phi_{s1}<\Phi_2$, then the steam is finally wet at $p_2$, Fig. 422 (*b*), and

$$H_{w'2}=H_2-(\Phi_2-\Phi_{s1})T_2 \quad . \quad . \quad . \quad . \quad . \quad (22)$$

$$H_\phi=(H_{s1}-H_{w'2}) \quad . \quad . \quad . \quad . \quad . \quad . \quad (23)$$

The actual total heat is

$$H_{w2}=H_{w'2}+h_f=H_{w'2}+(1-\eta_n)\left[H_\phi+\left(\frac{V_1}{223{\cdot}7}\right)^2\right] \quad . \quad (24)$$

and the actual quality is—

$$\left.\begin{aligned} q_2&=1-\frac{H_2-H_{w2}}{L_2}\\ &=1-\frac{H_2-H_{w'2}-(1-\eta_n)\left[H_\phi+\left(\frac{V_1}{223{\cdot}7}\right)^2\right]}{L_2}\end{aligned}\right\} \quad . \quad (25)$$

Alternatively the quality is given by—

$$q_2=\frac{\Phi_{s1}+\frac{h_f}{T_2}-\phi_{w2}}{\phi_{e2}} \quad . \quad . \quad . \quad . \quad . \quad . \quad (26)$$

The actual specific volume is—

$$v_{w2}=q_2v_2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (27)$$

*Example* 2.—At a certain stage of a pressure compounded impulse machine the steam enters the nozzles at 40 lb./in.$^2$ abs. and 30° Fahr. superheat, and is expanded to 20 lb./in.$^2$ abs. The initial velocity of entrance is 300 ft./sec. Calculate the exit velocity, quality, and specific volume of the steam, and check the values on the HΦ diagram. Take $\eta_n=0{\cdot}9$.

$p_1=40$, $t_{s1}=30$, $T_1=727$, $T_{s1}=757$, $\frac{T_{s1}}{T_1}=1{\cdot}04$, $\Phi_1=1{\cdot}6792$. $c_p=0{\cdot}522$, Chart.

$$\therefore \Phi_{s1}=\Phi_1+c_{p1}\log_e\frac{T_{s1}}{T_1}$$

$=1{\cdot}6792+0{\cdot}522\times2{\cdot}3\log 1{\cdot}04$ $\qquad p_2=20$

$=1{\cdot}6792+0{\cdot}522\times2{\cdot}3\times0{\cdot}017$ $\qquad T_2=688$

$=1{\cdot}6792+0{\cdot}204=1{\cdot}6996$ $\qquad H_2=1157{\cdot}1$

Also $\Phi_2=1{\cdot}7333$, so that $\Phi_{s1}<\Phi_2$ and the steam is wet at $p_2$, hence—

$$H_{w'2}=H_2-(\Phi_2-\Phi_{s1})T_2$$

$=1157{\cdot}1-(1{\cdot}7333-1{\cdot}6996)688$

$=1157{\cdot}1-0{\cdot}0337\times688=1157{\cdot}1-23{\cdot}186$

$=1133{\cdot}914$

$H_{s1}=H_1+c_{p1}t_{s1}$ $\qquad H_1=1171{\cdot}7$

$=1171{\cdot}7+0{\cdot}522\times30$

$=1187{\cdot}4$

$\therefore H_\phi=(H_{s1}-H_{w'2})=(1187{\cdot}4-1133{\cdot}9)=53{\cdot}5$ B.Th.U./lb.

Exit velocity—

$$V_2=223{\cdot}7\sqrt{\eta_n\left[H_\phi+\left(\frac{V_1}{223{\cdot}7}\right)^2\right]} \qquad \eta_n=0{\cdot}9,\ V_1=300$$

$$=223{\cdot}7\sqrt{0{\cdot}9\left[53{\cdot}5+\left(\frac{300}{223{\cdot}7}\right)^2\right]}$$

$$=223{\cdot}7\sqrt{0{\cdot}9\times55{\cdot}3}=223{\cdot}7\times7{\cdot}05=1577 \text{ ft./sec.}$$

Actual quality—

$$q_2=1-\frac{H_2-H_{w'2}-(1-\eta_n)\left[H_\phi+\left(\frac{V_1}{223{\cdot}7}\right)^2\right]}{L_2} \qquad H_2=1157{\cdot}1,\ L_2=961$$

$$=1-\frac{1157{\cdot}1-1133{\cdot}914-0{\cdot}1\times55{\cdot}3}{961}$$

$$=1-\frac{177}{961}=(1-0{\cdot}0184)=\underline{0{\cdot}982}$$

All these values check on the HΦ diagram.

Actual specific volume, $v_{w2}=q_2v_2=0{\cdot}982\times20{\cdot}08=19{\cdot}72$ ft.³/lb.

**357.** (*c*) *Steam Initially Dry or Wet and Finally Wet.*—Taking the case with steam initially wet with dryness $q_1$, Fig. 422 (*c*), and total initial entropy $\Phi_{w1}$ and total heat $H_{w1}$.

Then—

$$H_{w'2}=H_{w2}-(\Phi_2-\Phi_{w1})T_2 \quad . \quad . \quad . \quad . \quad (28)$$

$$H_\phi=(H_{w1}-H_{w'2}) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (29)$$

and the actual quality $q_2$ is again obtained by Equations 25 or 26.

*Example* 3.—Steam at 100 lb./in.² abs. and 2 per cent. wet is expanded in a nozzle to 2 lb./in.² abs. Assuming that the total frictional loss is 10·66 per cent., calculate the net heat drop, the exit velocity, the quality and specific volume at exit.

$p_1=100$, $\Phi_1=1{\cdot}6082$, $\Phi_{w1}=\phi_{w1}+q_1\phi_{e1}$
$=0{\cdot}4739+0{\cdot}98\times1{\cdot}1343$
$=0{\cdot}4739+1{\cdot}1116$
$=1{\cdot}5855$

$H_1=1191{\cdot}3$, $L_1=893{\cdot}0$, $\phi_{w1}=0{\cdot}4739$, $\phi_{e1}=1{\cdot}1343$

$H_{w1}=H_1-(1-q_1)L_1$
$=1191{\cdot}3-0{\cdot}02\times893$
$=1191{\cdot}3-17{\cdot}86=1173{\cdot}44$

$H_{w'2}=H_2-(\Phi_2-\Phi_{w1})T_2$
$=1113{\cdot}60-(1{\cdot}9159-1{\cdot}5855)586$
$=(1113{\cdot}60-193{\cdot}61)=920$ B.Th.U./lb.

$p_2=2$, $\Phi_2=1{\cdot}9159$, $T_2=586$, $H_2=1113{\cdot}6$

$H_\phi=(H_{w1}-H_{w'2})=(1173{\cdot}44-920)=253{\cdot}5$ B.Th.U./lb.

$\eta_n=(1-0{\cdot}1066)=0{\cdot}8934$, $V_1=0$

$$\therefore\ V_2=223{\cdot}7\sqrt{\eta_nH_\phi}$$
$$=223{\cdot}7\sqrt{0{\cdot}8934\times253{\cdot}5}$$
$$=223{\cdot}7\times15=3360 \text{ ft./sec.}$$

By Equation 26 the actual quality at $p_2$ is—

$$q_2=\frac{\Phi_{w1}+\frac{h_f}{T_2}-\phi_{w2}}{\phi_{e2}}$$

$$=\frac{1\cdot5855+\frac{27\cdot02}{586}-0\cdot175}{1\cdot7409}$$

$$=\frac{1\cdot4566}{1\cdot7409}=0\cdot837$$

$\phi_{w2}=0\cdot175$
$\phi_{e2}=1\cdot7409$
$\Phi_{w1}=1\cdot5855$
$T_2=586$
$h_f=0\cdot1066\times253\cdot5$
$=27\cdot02$
$v_2=173\cdot5$

Specific volume $v_2=q_2v_2=0\cdot837\times173\cdot5$
$=145\cdot22$ ft.$^3$/lb.

The various values check on the HΦ diagram.

**358. Mass Flow through a Nozzle.**—The function of a nozzle is to pass a given mass of steam and discharge it at as high a velocity of exit as possible. The foregoing equations determine the velocity and quality of the steam for a pressure drop between $p_1$ and $p_2$, on the assumption of complete expansion, between entrance and exit. They give no information as to the mass flowing, or its relation to the pressure drop.

If the narrowest section or "throat" of the nozzle passage is $A_2$ and $p_2$ is the pressure of the steam as it passes this section, then there is a clearly defined and constant ratio of $p_2$ to $p_1$ which corresponds to the maximum mass of steam the nozzle can pass. Let the passage considered have the converging-diverging form shown in Fig. 423, and let the subscripts 1, 2, and 0 denote the characteristic conditions at entrance (1), throat (2), and outlet (0). Also let the nozzle be fixed to a vessel containing the steam at constant pressure $p_1$ from which it enters the section 1 with negligibly small velocity, or $V_1\doteqdot0$. Further, in order to avoid the variation of expansion law pointed out in Art. 58, p. 75, let the steam be initially and finally superheated, so that it can be treated as a gas having a law of expansion $pv^\gamma=C$, where $\gamma$ is a constant adiabatic index. Then, as shown in Chapter I, Art. 16, since—

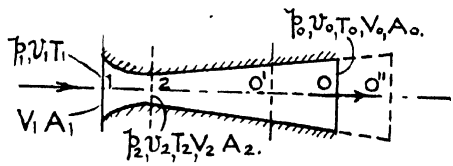


FIG. 423.

$$(p_1v_1-p_2v_2)=p_1v_1\left\{1-\left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}\right\}$$

$$JH_\phi=\frac{V_2'^2}{2g}=144\frac{\gamma}{\gamma-1}p_1v_1\left\{1-\left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}\right\} \quad . \quad . \quad (30)$$

By the equation of continuity the mass passing $A_2$ is—

$$W=\frac{A_2 V_2'}{v_2'} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (31)$$

so that—

$$V_2'^2=\frac{(Wv_2')^2}{A_2^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (32)$$

Substitute for $V_2'^2$ in Equation 30, and—

$$W^2=2g\frac{\gamma}{\gamma-1}\cdot\frac{144}{v_2'^2}p_1v_1\left\{1-\left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}\right\}A_2^2$$

But $v_2'=v_1\left(\frac{p_1}{p_2}\right)^{\frac{1}{\gamma}}$, so that—

$$W^2=2g\times\frac{\gamma}{\gamma-1}\times\frac{144p_1v_1}{v_1^2\left(\frac{p_1}{p_2}\right)^{\frac{2}{\gamma}}}\left\{1-\left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}\right\}A_2^2 \quad . \quad . \quad (33)$$

$$\text{or} \quad W^2=\text{const.}\ \frac{p_1}{v_1}\left\{1-\left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}\right\}\left(\frac{p_2}{p_1}\right)^{\frac{2}{\gamma}}A_2^2$$

$$=\text{const.}\ \left\{\left(\frac{p_2}{p_1}\right)^{\frac{2}{\gamma}}-\left(\frac{p_2}{p_1}\right)^{\frac{\gamma+1}{\gamma}}\right\}A^2{}_2$$

$$\text{or} \quad W=\text{const.}\ A_2\sqrt{(r^{\frac{2}{\gamma}}-r^{\frac{\gamma+1}{\gamma}})} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (34)$$

where $r=\left(\frac{p_2}{p_1}\right)$.

For a given nozzle, $A_2$ is constant, so that W is a maximum when the quantity under the root sign in (34) is a maximum. Differentiating with respect to $r$ and equating to zero—

$$dr\left(r^{\frac{2}{\gamma}}-r^{\frac{\gamma+1}{\gamma}}\right)=0$$

$$\frac{2}{\gamma}r^{\frac{2}{\gamma}-1}-\frac{\gamma+1}{\gamma}r^{\frac{1}{\gamma}}=0$$

$$\text{or} \quad r=\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (35)$$

In the case of superheated steam, $\gamma=1{\cdot}3$, so that the value of the pressure ratio corresponding to maximum discharge is—

$$r=\left(\frac{2}{2{\cdot}3}\right)^{\frac{1{\cdot}3}{0{\cdot}3}}=0{\cdot}545$$

If the steam were initially dry the index would be $\gamma=1{\cdot}3$ during the initial stages of the expansion, but at the Wilson limit (see

Art. 36) it would change to a lower value somewhere between 1·3 and 1·135 before the lower pressure $p_2$ is reached. Taking the older assumption that $\gamma=1{\cdot}135$ for initially dry steam, this gives $r=0{\cdot}575$. Hence for maximum discharge the throat pressure may be in the neighbourhood of 56 per cent. of the initial pressure. This is known as the "critical pressure."

The corresponding throat velocity is obtained as follows:

$$r=\frac{p_2}{p_1}=\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}} \quad \text{so that} \quad \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}=\left(\frac{2}{\gamma+1}\right) \quad \text{or} \quad \left(\frac{p_1}{p_2}\right)^{\frac{\gamma-1}{\gamma}}=\left(\frac{\gamma+1}{2}\right)$$

Also $$p_1v_1^{\gamma}=p_2v_2'^{\gamma} \quad \text{and} \quad \left(\frac{p_2}{p_1}\right)^{\frac{1}{\gamma}}=\frac{v_1}{v_2'}$$

Multiply each side by $\frac{p_1}{p_2}$, and

$$\frac{p_1^{\frac{\gamma-1}{\gamma}}}{p_2^{\frac{\gamma-1}{\gamma}}}=\frac{p_1v_1}{p_2v_2'}=\frac{\gamma+1}{2}$$

$$\therefore\ p_1v_1=p_2v_2'\left(\frac{\gamma+1}{2}\right);$$

substitute for $p_1v_1$ in Equation 30 for the critical or throat velocity—

$$\frac{V_2'^2}{2g}=144\frac{\gamma}{\gamma-1}p_1v_1\left\{1-\left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}\right\}$$

and $$V_2'^2=144\times 2g\frac{\gamma}{\gamma-1}p_2v_2'\left(\frac{\gamma+1}{2}\right)\left\{1-\left(\frac{2}{\gamma+1}\right)\right\}=144g\gamma p_2v_2'$$

$$\therefore\ V_2'=\sqrt{144g\gamma p_2v_2'} \qquad (36)$$

This is the "velocity of sound" in the steam under the throat conditions, and it is therefore called the "acoustic velocity." It must not be confused with the maximum possible velocity that can be generated by further expansion to any pressure $p_0$ below $p_2$.

**359.** From Equation 34, $W=CA_2\sqrt{r^{\frac{2}{\gamma}}-r^{\frac{\gamma+1}{\gamma}}}$, it would appear that with $r=0$ or $p_2$ reduced to zero no flow would take place through the nozzle, an obviously absurd conclusion. This expression, however, does not hold good after the critical pressure is reached. In fact, only when $p_1=p_2$ will no flow take place, that is, $W=0$ for $r=1$.

Experiment has shown that after the critical ratio of about 55 per cent. is reached the rate of discharge remains sensibly at the maximum value, no matter how far below $p_2$ the pressure $p_0$ in the outer zone is lowered. That is, for all values of $r$ below 55 per cent. the discharge is unaffected, with the same constant initial conditions

at entrance. This condition is shown by the actual curve of discharge of a convergent nozzle (Fig. 424).

Where a convergent nozzle discharges steam into a zone of pressure $p_0$, if the pressure ratio $\left(\frac{p_0}{p_1}\right)$ is greater than the critical value $\left(\frac{p_2}{p_1}\right)$, then $p_0$ is to be taken as the lower *exit* limit of expansion. When, however, $\left(\frac{p_0}{p_1}\right)$ is less than $\left(\frac{p_2}{p_1}\right)$, the latter ratio is to be taken for calculation of the mass flow. This is equally the case if the nozzle is convergent-divergent, as the maximum discharge is controlled by the critical throat ratio.

The passage should be made convergent-divergent when the external pressure $p_0$ is less than the critical value $p_2$. This is for the purpose of controlling the jet and directing it as far as possible in the useful or axial direction, so as to prevent undue loss of kinetic energy between extrance 1 and exit 0.

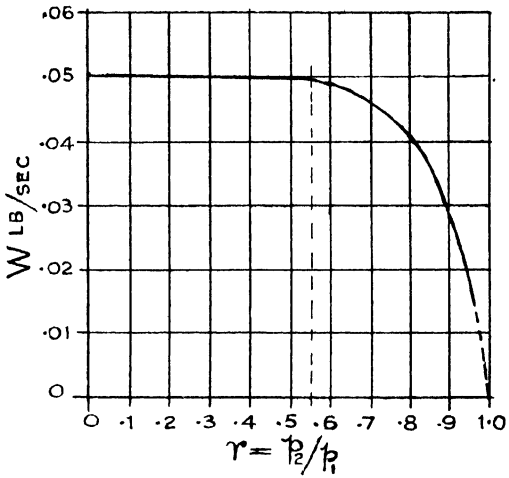



FIG. 424.

**360. Calculation of Nozzle Area for a Given Mass Flow.**—The necessary throat area for the passage of a given mass flow of W lb./sec. is determined from the equation of continuity. Actually, owing to the slight *vena contracta* set up at the throat, even with a well-rounded nozzle entrance the discharge is less than that due to the full area $A_2$, and in calculating this a coefficient of discharge $c_d$ from 0·98 to 0·99 should be used. Thus—

$$W = \frac{c_d V_2' A_2}{v_2'} \text{ lb./sec.} \quad . \quad . \quad . \quad . \quad . \quad (37)$$

where $V_2'$ = the ideal throat velocity in ft./sec.
$v_2'$ = ,, ,, ,, specific volume at $p_2$
$A_2$ = throat area in ft.$^2$

Owing to variation of $\gamma$ with different pressure conditions, $V_2'$ should always be calculated for the critical heat drop $H_{\phi 2}$ and not from Equation 36.

It is usual to express the nozzle areas in inch$^2$, so that, from (37)—

$$A_2 = \frac{144 W v_2'}{c_d V_2'} \text{ in.}^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad (38)$$

When a suitable nozzle efficiency $\eta_n$ is taken, the exit velocity $V_0$ and actual specific volume $v_0$ at $p_0$ are calculated by the methods given in Arts. 35, 36, and 37. As the same mass passes 0 as at the throat 2, then—

$$A_0=\frac{144Wv_0}{V_0} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (39)$$

Alternatively—

$$A_0=A_2\times\frac{c_d V_2' v_0}{v_2' V_0} \quad . \quad . \quad . \quad . \quad . \quad . \quad (40)$$

**361. Under- and Over-Expansion in a Nozzle.**—If the exit area $A_0$, Fig. 423, is calculated as above, the pressure of the steam at this section will just be equal to $p_0$, the pressure in the outer zone. The steam is said to be " completely expanded."

If the nozzle, however, is shortened to 0′ the exit area $A_0'<A_0$ and pressure $p_0'>p_0$, that is, the steam is not fully expanded at 0′, and disorderly flow into the outer zone, with loss of kinetic energy, ensues. The steam is said to be " under expanded."

If the nozzle is lengthened to 0″, as indicated by the dotted outline, then the exit area $A_0''>A_0$. The steam in this case expands below the pressure $p_0$ inside the nozzle and then rises to this value again at exit, that is, there is objectionable recompression with shock and eddy loss. In this case the steam is said to be " over-expanded." The condition of under-expansion is not so objectionable in a turbine as over-expansion, as the clearance space between the nozzle exit and wheel blades is small and the steam continues to expand across this narrow gap and as it enters the blade channels, without much radial distention of the jet. Over-expansion is bad and should not be permitted.

**362.** The method of calculation of mass flow by means of the heat drop is perfectly general; but for the estimation of the maximum discharge in the case of superheated or initially dry steam, this can also be calculated from a function of the initial pressure and volume. Taking $\gamma=1{\cdot}3$ for superheated steam, then, on substitution in Equation 34 with $r=0{\cdot}545$, this becomes, allowing $c_d=0{\cdot}98$—

$$W_s=0{\cdot}3100A_2\sqrt{\frac{p_1}{v_{s1}}}\text{ lb./sec.} \quad . \quad . \quad . \quad . \quad (41)$$

On the assumption that the same law holds for initially dry steam down to the throat pressure $p_2$, then, since $v_{s1}=v_1$ for dry steam, and $v_1=\left(\frac{490}{R}\right)^{\frac{15}{16}}$ (see Art. 28), Equation 41 for initial pressures below 200/in. abs. becomes—

$$W_d=0{\cdot}01695p_1^{\frac{31}{32}}A_2\text{ lb./sec.} \quad . \quad . \quad . \quad . \quad (42)$$

In each case $p$ is in lb./in.$^2$ abs. and $A_2$ in in.$^2$

*Example* 4.—In the case of the nozzles (Example 1) calculate the necessary throat and exit areas for a mass flow of 9 lb./sec.

Throat area. Applying Equation 41—

$$W_s = 0{\cdot}31 A_2 \sqrt{\frac{p_1}{v_{s1}}} \qquad \begin{aligned} W_s &= 9 \\ p_1 &= 150 \\ v_{s1} &= v_1(1+0{\cdot}0015 t_{s1}) \\ &= 3{\cdot}04(1+0{\cdot}0015\times 200) \\ &= 1{\cdot}3\times 3{\cdot}04 = 3{\cdot}95 \end{aligned}$$

$$9 = 0{\cdot}31 A_2 \sqrt{\frac{150}{3{\cdot}95}}$$

$$\therefore\ A_2 = \frac{9}{0{\cdot}31\times 6{\cdot}17} = 4{\cdot}7 \text{ in.}^2$$

Exit area $A_0 = \dfrac{144 W v_0}{V_0}$ $\qquad \begin{aligned} v_0 &= 9{\cdot}364 \\ V_0 &= 2200 \end{aligned}$

$$= \frac{144\times 9\times 9{\cdot}364}{2200} = 5{\cdot}52 \text{ in.}^2$$

and $$\frac{A_0}{A_2} = \frac{5{\cdot}52}{4{\cdot}7} = 1{\cdot}17 = a$$

*Example* 5.—Calculate the throat and exit areas for the nozzles of the turbine stage (Example 2) if the mass flow is 18 lb./sec.

Here $p_1 = 40$, $t_{s1} = 30$, $V_1 = 300$, $p_2 = 20$. Use the heat drop method to obtain the throat area. $p_2 = 0{\cdot}545$, $p_1 = 22$.

From the HΦ diagram $H_{\phi 2} = 47$, and $q_2' = 0{\cdot}98$, $v_2 = 18{\cdot}37$.

$\therefore\ v_w'_2 = 0{\cdot}98 \times 18{\cdot}37 = 18$. The throat energy loss is assumed to be negligible—

$$V_2' = 223{\cdot}7\sqrt{47} = 223{\cdot}7\times 6{\cdot}86 = 1540 \text{ ft./sec.}$$

Then $$A_2 = \frac{144 W v_{w}'_2}{c_d V_2'} \qquad c_d = 0{\cdot}98$$

$$= \frac{144\times 18\times 18}{0{\cdot}98\times 1540} = 31 \text{ in.}^2$$

Also $p_0 = 20$, $V_0 = 1577$ $\qquad \begin{aligned} q_0 &= 0{\cdot}982 \\ v_{w0} &= 0{\cdot}982\times 20{\cdot}08 = 19{\cdot}72 \end{aligned}$

$$\therefore\ A_0 = \frac{144\, v_{w0} W}{V_0}$$

$$= \frac{144\times 19{\cdot}72\times 18}{1577} = 32{\cdot}4 \text{ in.}^2$$

and $$\frac{A_0}{A_2} = \frac{32{\cdot}4}{31} = 1{\cdot}045$$

**363. Nozzle Dimensions.**—The three practicable types of nozzle passages employed on impulse turbines are shown in Figs. 425, 426, 427, 428. The straight convergent-divergent nozzle used on the de Laval simple impulse machines is shown in Fig. 425. The converging part is simply a rounded entrance ; the diverging part has a straight taper, to facilitate manufacture. The length L between throat and exit is controlled by mechanical considerations, as nozzles, to suit varying expansion ratios, may have to be fitted in the same casing.

The nozzle axis is inclined to the plane of the wheel at an angle $\alpha$, which may vary from 18° to 20°. The latter is a standard for the

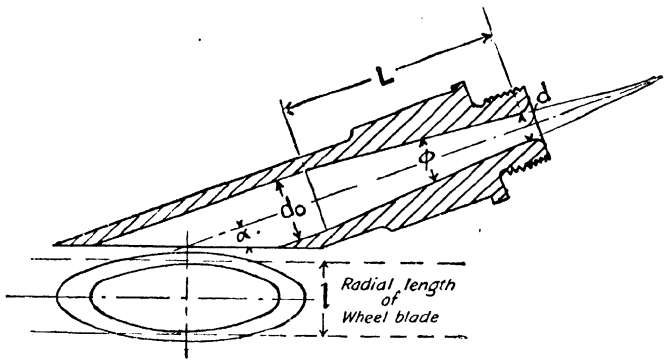



FIG. 425.

simple turbine. The passage is made parallel at the outlet diameter $d_0$. The section on the plane of the wheel is an ellipse 1 as shown

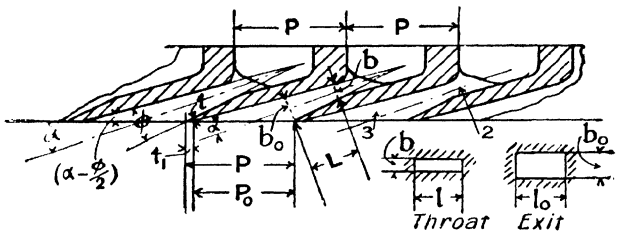



FIG. 426.

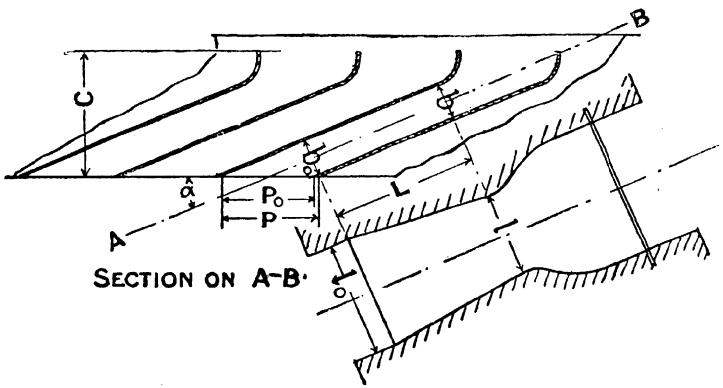


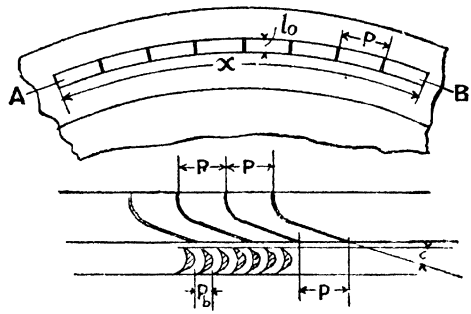

FIG. 427.

in plan, and there is some loss of energy due to the aspiration of dead steam surrounding the nozzle exit, which would be absent

if the exit were rectangular. In terms of the notation on the figure, the length between throat and exit is given by—

$$L=\frac{d_0-d}{2\tan\frac{\phi}{2}} \quad \ldots\ldots\ldots \quad (43)$$

The angle $\phi$ increases with increase of pressure ratio $\left(\frac{p_1}{p_0}\right)$. There is a limit to this angle, however, at which the steam separates from the wall and energy loss occurs. The usual maximum value used in the de Laval nozzles is 12°. It varies from 6° to 12°. In the case of a velocity compounded impulse turbine the passages are usually cast in blocks and have the longitudinal section shown in Fig. 426. The passages are filed smooth to diminish friction loss. The cross-section is rectangular and the radial height $l_0$ is usually kept constant.




DEVELOPED SECTION ON A-B.

FIG. 428.

It has to be arbitrarily fixed with regard to other elements of the design. These nozzle blocks are bolted to the cover and the H.P. diaphragms. At low-pressure stages, where admission is complete or nearly so, the nozzle passages are formed by steel plates cast in the diaphragms, as shown in Fig. 427. The nozzle angle $\alpha$ is usually specified, but the wall angle $\phi$ has to be provisionally fixed. The value of $\phi/2$ may be taken from 2° to 4° for a first approximation to the number of nozzles required.

The net pitch of the nozzle walls is $P_0$ or the clear circumferential width of outlet per nozzle (Fig. 426). The transverse width, at right angles to the nozzles axis, is approximately—

$$b_0=P_0\sin\left(\alpha-\frac{\phi}{2}\right) \quad \ldots\ldots\ldots \quad (44)$$

Also $P_0=(P-t_1)$, where—

$$t_1=\frac{t}{\sin\left(\alpha-\frac{\phi}{2}\right)} \quad \ldots\ldots\ldots \quad (45)$$

$t$ being the thickness of the nozzle wall at exit. It is usually sufficient to take $P_0=cP$, where $c$ is a thickness coefficient varying from 0·94 to 0·98. The exit area $A_0$ being known,

the number of nozzle passages $n$ can be calculated from the expression—

$$A_0 = ncPl_0 \sin\left(a - \frac{\phi}{2}\right) \quad . \quad . \quad . \quad . \quad . \quad (46)$$

Also with $l_0 = l$, $\frac{A_0}{A_2} = \frac{b_0}{b}$, and hence the length between throat and exit is given by—

$$L = \frac{b(a-1)}{2 \tan \frac{\phi}{2}} = \frac{b_0(a-1)}{a \,.\, 2 \tan \frac{\phi}{2}} \quad . \quad . \quad . \quad . \quad . \quad (47)$$

where $a = \frac{A_0}{A_2}$, the nozzle area ratio.

The net arc is $x_0 = ncP$, and the gross arc $x = nP$.

**364.** In the case shown in Fig. 427, in which the nozzle wall plates are parallel in the circumferential direction but the passage walls are inclined in the radial direction—

$$b = b_0 = \frac{A_0}{nl_0} \quad . \quad . \quad . \quad . \quad . \quad (48)$$

and—

$$l = l_0 \frac{A_2}{A_0} \quad . \quad . \quad . \quad . \quad . \quad . \quad (49)$$

and—

$$L = \frac{l(a-1)}{2 \tan \frac{\phi}{2}} \quad . \quad . \quad . \quad . \quad . \quad . \quad (50)$$

where $\phi$ is the inclination of the sides.

In the majority of pressure compounded machines the stage pressure ratio is less than or just equal to the critical, and the passages are slightly convergent or parallel. For the H.P. stages the nozzle blocks are sometimes built up, formed blades being clamped between side plates. Sometimes the passages are formed by simply casting the blades into the diaphragms, as indicated in Fig. 428, which shows a developed section of part of a nozzle arc. Here the gross arc of admission is—

$$x = nP = \frac{A_0}{cl_0 \sin a} \quad . \quad . \quad . \quad . \quad . \quad (51)$$

The band of steam flowing through this arc has to be split up into a number of elemental streams, in order to prevent disorderly flow and loss of energy. This is effected by the set of closely packed nozzle plates. The pitch P is an arbitrary quantity fixed from experience. As a rule it is made from three to four times the pitch $P_b$ of the wheel blades on to which the nozzles discharge.

If $m$=number of blade pitches per nozzle pitch
$c_1$=thickness coefficient for the wheel blades,

then— $$P=m\frac{c_1}{c}P_b \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (52)$$

The exit area $A_0$ being calculated, the radial height $l_0$ and the angle $\alpha$ can be arbitrarily chosen and $x$ calculated from Equation 51.

At high-pressure stages the angle $\alpha$ is kept as low as possible, to obtain a reasonably large value of admission arc and reduce vane friction. It runs from 12° to 20°. At low-pressure stages, much higher values are required, in order to get the steam through the available area with full admission.

*Example* 6.—Calculate the number and dimensions of the nozzles for the case of the stage (Example 4), given the provisional values $l_0$=0·625″ ; pitch P=1·5″ ; nozzle jet angle $\alpha$=18° ; $\frac{\phi}{2}$=2° ; thickness coefficient $c$=0·94. $A_0$=5·52, $A_2$=4·7 ; ring diameter D=5 ft. 9 in.

No. of nozzles $$n=\frac{A_0}{cPl_0\sin\left(\alpha-\frac{\phi}{2}\right)}$$
$$=\frac{5\cdot52}{0\cdot94\times1\cdot5\times0\cdot625\times0\cdot2756}$$
$$=23$$

$P=1\cdot5$
$A_0=5\cdot52$
$c=0\cdot94$
$l_0=0\cdot625$
$\left(\alpha-\frac{\phi}{2}\right)=16°$
$\sin 16°=0\cdot2756$

This number might be carried on two blocks with 12 and 11 passages respectively.

Nozzle arc $x=nP=23\times1\cdot5=34\cdot5$.
Mean ring circumference=$\pi D=\pi\times69=216\cdot74$ in.

Fraction covered by nozzles=$\frac{34\cdot5\times100}{216\cdot74}=16$ per cent.

Radial height at throat $b=\frac{A_2}{nl_0}=\frac{4\cdot7}{23\times0\cdot625}=0\cdot326$ in.

$$\frac{b_0}{b}=a \qquad \therefore\ b_0=0\cdot326\times1\cdot17=0\cdot381$$

The length between throat and exit—

$$L=\frac{b(a-1)}{2\tan\frac{\phi}{2}}=\frac{0\cdot326(1\cdot17-1)}{2\times0\cdot0349}$$
$$=\frac{0\cdot326\times0\cdot17}{0\cdot0698}=0\cdot8 \text{ in.}$$

$\frac{\phi}{2}=2°$
$\tan 2°=0\cdot0349$

This is obviously too small for constructional purposes and L would not be made less than 1 in.

**365. Work done on Turbine Blading.**—When the jet of steam issues from a fixed nozzle passage with a steady flow of W lb./sec. and a constant exit velocity $V_0$ ft./sec. it has a total amount of kinetic energy $\frac{WV_0^2}{2g}$ ft./lb., available for conversion into mechanical work on the blading of the mobile part or rotor, in the turbine casing.

The driving force on the blade surface is dynamical, and is produced by a change of momentum of the fluid, which is given by the product of the mass and change of velocity. This velocity change may involve both a change of magnitude and change of direction.

The change of velocity must not be accompanied by any shock effect on the surface of the blading, or loss of energy will result. The fluid must move smoothly on to and off the blade surface, and the blade has to be formed to obtain this result. The sections of impulse and reaction blading have already been illustrated in Chapter XIX.

The following notation will be found useful for the calculation of the work done on the blading.

Blade angles—

| | |
|---|---|
| Angle of entrace to blade channel | $\theta_1$ |
| ,, exit from blade channel | $\theta_0$ |

Jet angle—

| | |
|---|---|
| Inclination of the jet to the plane of the wheel or to the tangent in a radial machine | entrance $\alpha_1$, exit $\alpha_0$ |

Steam velocities at blading—

| | *Entrance* | *Exit* |
|---|---|---|
| Absolute velocity | $V_{a1}$ | $V_{a0}$ |
| Relative velocity | $V_{r1}$ | $V_{r0}$ |
| Velocity of whirl | $V_{w1}$ | $V_{w0}$ |
| Angular velocity of blading | $\omega$ | |
| Peripheral velocity of blading | $u$ | |
| Work done on blade | $E_b$ | |

The subscripts $a$, $r$, and $w$ denote: absolute, relative, and whirl. The absolute inlet velocity corresponds to the calculated velocity of exit from the nozzle ($V_0$).

A sectional elevation of a ring of blades for a radial flow turbine is shown in Fig. 429, and an elevation and sectional plan of a ring for an axial flow machine in Fig. 430. In each case the direction of motion of the blade is along DI. The steam from the nozzle is discharged into the blade passage with an absolute velocity $V_{a1}$ and at an angle $\alpha_1$ to DI. It enters the blade with a velocity $V_{r1}$ relative to the blade surface, and leaves with a velocity $V_{r0}$ relative to the surface at exit; and it finally leaves the blade ring with an absolute velocity $V_{a0}$ at an angle $\alpha_0$ to HO.

At entrance the component of the absolute velocity of entrance

$V_{a1}$ in the direction DI is $V_{w1}$. This is the velocity of whirl at entrance. Similarly at exit the component $V_{a0}$ of the absolute

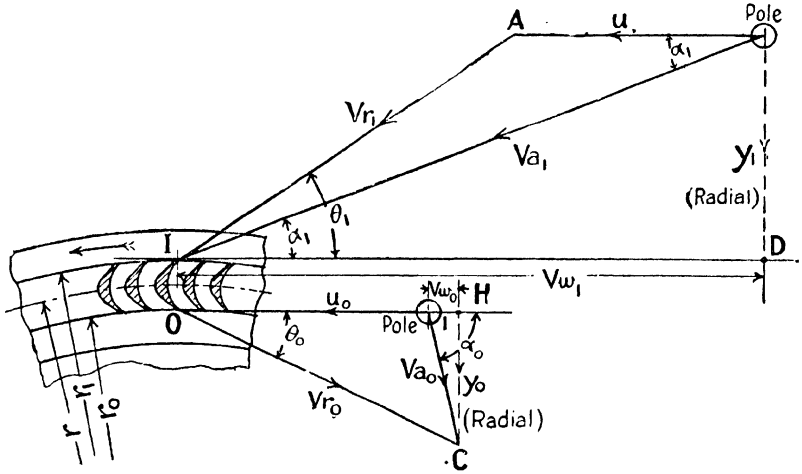



FIG. 429.

exit velocity $V_{a0}$, in the direction $O_1H$, is the velocity of whirl at exit.

A unit mass of fluid entering at I with absolute velocity $V_{a1}$

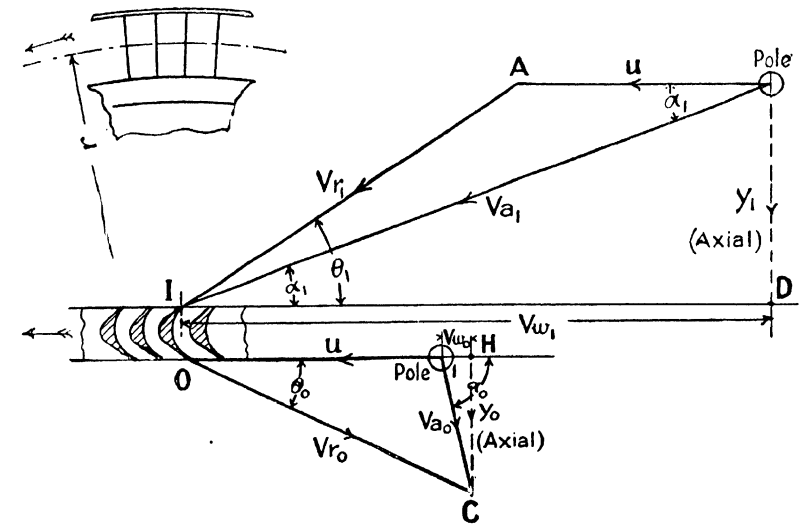



FIG. 430.

has a momentum in direction of motion of the blade of $\frac{1}{g}V_{w1}=\frac{1}{g}V_{a1}\cos a_1$. The moment of this momentum about the

axis of rotation is, for the radial flow case (Fig. 429), $\frac{1}{g}V_{w1}r_1$, where $r_1$ is the radius at entrance I; in the axial machine it is $\frac{1}{g}V_{w1}r$, where $r$ is the mean blade radius (Fig. 430). At exit the fluid has a momentum in the direction of motion of the blade of $\frac{1}{g}V_{w0}$. The corresponding value of the moment of momentum is $\frac{1}{g}V_{w0}r_0=\frac{1}{g}V_{a0}\cos\alpha_0 r_0$, for the radial and $\frac{1}{g}V_{a0}\cos\alpha_0 r$ for the axial. As shown here, the sense of the exit velocity of whirl is negative to that at entrance.

The torque or twisting moment at the shaft axis is given by the change of the moment of momentum at entrance and exit, thus for the radial case—

$$\left.\begin{aligned} &\text{Torque} \quad T=\frac{1}{g}(V_{w1}r_1-V_{w0}r_0)\\ &\text{or} \quad\ T=\frac{1}{g}(V_{a1}\cos\alpha_1 r_1-V_{a0}\cos\alpha_0 r_0)\end{aligned}\right\} \quad . \ . \ . \quad (53)$$

and for the axial case—

$$\left.\begin{aligned} &T=\frac{r}{g}(V_{w1}-V_{w0})\\ &\text{or} \quad T=\frac{r}{g}(V_{a1}\cos\alpha_1-V_{a0}\cos\alpha_0)\end{aligned}\right\} \quad . \ . \ . \quad (54)$$

Equations 53 and 54 are fundamental and equally applicable to impulse and reaction blading.

In a radial flow case, where $r_1$ differs very slightly from $r_0$, it is sufficient to take the mean value $r$ and apply Equation 54.

When the summation of the work done on a series of rings of progressively increasing diameter, as in the case of a Ljungström turbine, has to be made, the exact value of the torque for each ring should be calculated by (53).

When the angular velocity $\omega$ and torque T are known the work done on the unit mass (usually taken as 1 lb.) is calculable from the fundamental equation—

$$E_b=T\omega \quad . \ . \ . \ . \ . \ . \ . \ . \quad (55)$$

where $\omega$ is in radians per second. Substituting for T from (53) or (54), then for the radial flow turbine—

$$E_b=\frac{\omega}{g}(V_{w1}r_1-V_{w0}r_0) \quad . \ . \ . \ . \ . \quad (56)$$

and for the axial flow—

$$E_b=\frac{\omega r}{g}(V_{w1}-V_{w0}) \quad . \ . \ . \ . \ . \ . \quad (57)$$

In the latter case, since the mean blade velocity is $u=\omega r$—

$$\left.\begin{aligned} E_b &= \frac{u}{g}(V_{w1}-V_{w0}) \\ &= \frac{u}{g}V_w \end{aligned}\right\} \quad \ldots\ldots\ldots \quad (58)$$

where $V_w$ represents the vector difference of the velocities of whirl at entrance and exit.

In the case of the radial flow turbine, $\omega r_1=u_1$, and $\omega r_0=u_0$ and—

$$E_b=\frac{1}{g}(V_{w1}u_1-V_{w0}u_0) \quad \ldots\ldots\ldots \quad (59)$$

Except in special cases, it is sufficient to take the mean value $u$ of $u_1$ and $u_0$ and apply Equation 59.

The bracketed term $(V_{w1}-V_{w0})$ represents the vector difference of the velocities of whirl at entrance and exit. If $V_{w1}$ is positive in direction, $V_{w0}$ is usually negative, so that the sum of the numerical quantities is to be taken. This condition is illustrated in Figs. 429 and 430, where $(V_{w1}-V_{w0})=V_w=(DI+O_1H)$. When the graphic method of calculation is used, the value of $V_w$ is given automatically, without reference to the signs of the two components.

**366. Vector Subtraction and Addition.**—The vector difference of two velocities is obtained by choosing a pole and drawing lines from this parallel in direction and equal in magnitude to the two velocities, then joining the extremities of these lines. The closing line gives the vector difference. The essential point of the construction is that each velocity vector must be drawn with the sense arrow pointing *away* from the pole. For example in Fig. 430, $OI=V_{a1}$ and $OA=u$, are the absolute velocities of the steam jet and the wheel blading, having the sense shown by the arrow heads. They are drawn from an arbitrarily chosen pole, or point O, with the sense reading from O. The closing line $AI=V_{r1}$, which is the vector difference $(\overline{OI}-\overline{OA})=(\overline{V_{a1}}-\overline{u})$. The sense of the vector difference always reads from the extremity of the velocity that is subtracted, that is, the velocity having the negative sign in the bracket. In this case it is $u$, and hence the arrow head points from A to I. In a case of vector addition the starting point is the pole $O_1$, but the vector quantities to be summed are drawn consecutively with the sense arrows pointing forward.

The closing line then gives the vector sum. Its sense reads away from the pole O. Thus in Fig. 430 the vector $O_1O=u$, the mean blade velocity vector, is drawn from the pole $O_1$, the sense reading from $O_1$ to O. From O the relative exit velocity vector OC is drawn, the sense reading from O to C. The closing line gives the absolute velocity of exit $O_1C=V_{a0}$, the sense reading from $O_1$ to C.

It can be easily shown that the vector difference of the absolute velocities of two bodies represents the velocity of one body relatively

to the other. In this case in Fig. 430 the vector $AI=V_{r1}$ is the velocity of the steam jet relatively to the wheel blade. It gives, in magnitude and direction, the equivalent absolute velocity with which the steam would enter the blade channel if the blade were at rest. Its inclination to the direction of motion of the blade obviously gives the angle of entrance $\theta_1$ for the blade. The two absolute velocities and the relative entrance velocity determine the velocity triangle OIA at entrance.

Similarly at exit the blade has the mean velocity $u$ and the steam an absolute velocity $V_{a0}$. When the direction of $V_{a0}$ is known, then the same construction gives the velocity triangle $O_1CO$ at exit; the vector difference of $u$ and $V_{a0}$ is $OC=V_{r0}$, the relative velocity of exit of the steam from the blades.

**367. Velocity Diagram for a Simple Impulse Stage.**—In the case

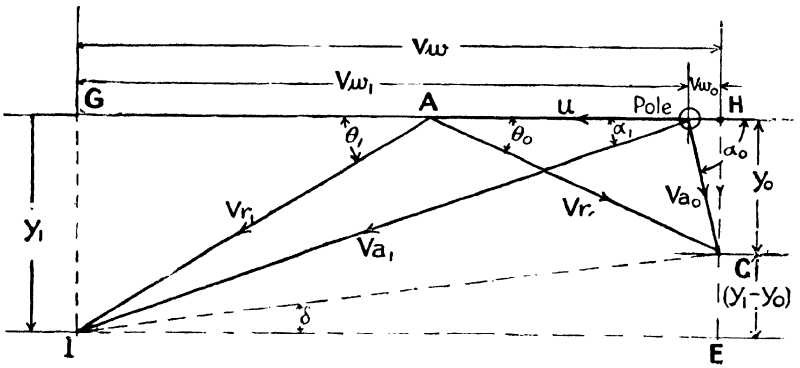



FIG. 431.

of the axial flow machine, in which the blade velocity is the same at entrance and exit, the vector difference of the absolute velocities can be automatically determined by superposing the entrance and exit velocity triangles. The combination is shown in Fig. 431. O is the pole common to both triangles. The perpendicular IG from I to OA produced gives the velocity of whirl $V_{w1}$ at entrance, and the perpendicular CH from C gives the velocity of whirl $V_{w0}$ at exit. The intercept HG is the vector difference $V_w$ of these two quantities. Since H falls to the right of the pole O, the two quantities are added numerically. If the jet exit angle $\alpha_0$ were greater than 90° H would fall to the left of O, and they would be subtracted. Whatever the form of the inlet and outlet triangles, the addition or subtraction is automatically performed in the graphical method.

The change of velocity, from the combined diagram, Fig. 431, is given by $CI=(\overline{OI}-\overline{OC})$. This vector is inclined at an angle $\delta$ to the direction of motion of the blade; and the effective change of velocity between entrance and exit is its component in the direction of motion of the blade, that is, the projection $IE=GH=V_w$.

The practical rule for the construction of the combined velocity diagram and calculation of the work done is as follows.

Choose a pole O, Fig. 431, and draw OA$=u$, the mean blade velocity, and OI$=V_{a1}=V_0$, the exit velocity of the jet from the nozzle, at the jet angle $\alpha_1$ to OA. Join A and I, and AI gives the relative entrance velocity $V_{r1}$, and determines the correct value of the blade entrance angle $\theta_1$. If there were no loss of energy in the blade channel, the relative velocity would remain constant between entrance and exit, or $V_{r1}=V_{r0}$. There is always a loss and $V_{r0}=kV_{r1}$, where $k$ is a velocity coefficient ranging from 0·8 to 0·9. From A draw AC$=V_{r0}$, at the chosen exit angle $\theta_0$, and complete the outlet diagram by joining C and O.

**368. Dynamical Thrust on the Blading.**—The vector CI, Fig. 431, can be resolved into its two components, one in the direction of the blade motion and another in the direction perpendicular to this. The first is $V_w$, the effective velocity change or difference of the velocities of whirl $V_{w1}$ and $V_{w0}$. The second is a non-effective change, given by the difference of the normal components $y_1$ at entrance and $y_0$ at exit, that is, $y_1-y_0$.

This produces an axial thrust on the rotor shaft of—

$$F=\frac{1}{g}CE \quad \text{or} \quad F=\frac{1}{g}(y_1-y_0), \text{ for unit mass} \quad . \quad . \quad (60)$$

As a rule this dynamical thrust is small in an impulse machine, and is easily taken up by the thrust block used to adjust the axial clearances between the fixed and moving surfaces.

**369. Calculation of Work done on a Ring of Blading.**—When the combined velocity diagram has been drawn and $V_w$ determined, the work per unit mass per second can be calculated from Equation 58 for the axial, and from Equation 59, when the separate diagrams are drawn, for the radial case. It can, however, be calculated analytically as follows.

Referring to Fig. 431 for the axial turbine—

$$V_{a1}\cos\alpha_1=V_{r1}\cos\theta_1+u$$

$$\therefore\ V_{r1}=\frac{V_{a1}\cos\alpha_1-u}{\cos\theta_1} \quad . \quad . \quad . \quad . \quad . \quad (61)$$

Also $$V_{r1}\cos\theta_1+V_{r0}\cos\theta_0=V_w$$

or $$V_{r1}(\cos\theta+k\cos\theta_0)=V_w$$

Hence $$V_w=(V_{a1}\cos\alpha_1-u)\left(1+\frac{k\cos\theta_0}{\cos\theta_1}\right) \quad . \quad . \quad . \quad (62)$$

Substituting from (62) in (58), the expression for the work done per lb. on the blades is—

$$E_b=u\left(1+\frac{k\cos\theta_0}{\cos\theta_1}\right)(V_{a1}\cos\alpha_1-u) \text{ ft.-lb. per lb. per sec.} \quad (63)$$

*Denoting the ratio of blade velocity to jet velocity by* $\rho=\frac{u}{V_{a1}}=\frac{u}{V_0}$, this may be written as—

$$E_b=\frac{u^2}{g}c\left(\frac{\cos\alpha_1}{\rho}-1\right) \text{ ft.-lb. per lb. per sec. . . (64)}$$

where $$c=\left(1+\frac{k\cos\theta_0}{\cos\theta_1}\right)$$

To find $c$ for any given case $\theta_0$ is arbitrarily assumed; but $\theta_1$ is determined by $V_{a1}$, $u$, and $\alpha_1$. Thus—

$$\tan\theta_1=\frac{GI}{GA}=\frac{V_{a1}\sin\alpha_1}{V_{a1}\cos\alpha_1-u}$$

or $$\tan\theta_1=\frac{\sin\alpha_1}{\cos\alpha_1-\rho} \quad . . . . . (65)$$

and $\theta_1$ is calculable.

The horse-power expended on the blading by the steam is—

$$H.P.=\frac{E_b}{550}=\frac{u^2c}{550g}\left(\frac{\cos\alpha_1}{\rho}-1\right) \quad . . . . (66)$$

and the heat converted to work on the blades is—

$$h_b=\frac{E_b}{J}=\frac{u^2c}{Jg}\left(\frac{\cos\alpha_1}{\rho}-1\right) \text{ B.Th.U./lb. sec. . . (67)}$$

The axial components of the absolute velocities are—

$$y_1=(V_{a1}\cos\alpha_1-u)\tan\theta_1$$

$$=u\left(\frac{\cos\alpha_1}{\rho}-1\right)\tan\theta_1$$

$$y_0=(u+V_w-V_{a1}\cos\alpha_1)\tan\theta_0$$

Substituting for $V_w$ from (62) and reducing, this becomes—

$$y_0=\frac{k\sin\theta_0}{\cos\theta_1}u\left(\frac{\cos\alpha_1}{\rho}-1\right) . . . . . (68)$$

The values of the jet exit angle $\alpha_0$ and the absolute velocity at exit are obtained as follows:

$$V_{w0}=(V_w-V_{a1}\cos\alpha_1)=c(V_{a1}\cos\alpha_1-u)-V_{a1}\cos\alpha_1$$
$$=V_{a1}\cos\alpha_1(c-1)-cu$$

Then $$\tan\alpha_0=\frac{y_0}{V_{w0}}=\frac{(c-1)(V_{a1}\cos\alpha_1-u)\tan\theta_0}{(c-1)V_{a1}\cos\alpha_1-cu}$$

or $$\tan\alpha_0=\frac{(V_{a1}\cos\alpha_1-u)\tan\theta_0}{V_{a1}\cos\alpha_1-\frac{u}{1-\frac{1}{c}}}$$

$$= \frac{\left(\frac{\cos \alpha_1}{\rho} - 1\right) \tan \theta_0}{\frac{\cos \alpha_1}{\rho} - \frac{1}{1 - \frac{1}{c}}} \quad \ldots \ldots \ldots \quad (69)$$

and $$V_{a0} = \frac{y_0}{\sin \alpha_0} \quad \ldots \ldots \ldots \quad (70)$$

Alternatively it is given by—

$$\left.\begin{aligned} V_{a0}^2 &= V_{r0}^2 + u^2 - 2uV_{r0} \cos \theta_0 \\ &= (kV_{r1})^2 + u^2 - 2ukV_{r1} \cos \theta_0 \end{aligned}\right\} \quad \ldots \quad (71)$$

Equations 61 to 71 enable the work done on the blade ring, the axial thrust, and residual energy to be calculated without the aid of the velocity diagram.

In the majority of cases the graphical method is preferable.

*Example* 7.—The blade ring of the last stage of an impulse turbine has a mean diameter of 54 in. The outlet blade angle is 45°, the inlet jet angle 25° ; the jet velocity at the nozzle is 1410 ft./sec. ; the rotational speed is 2400 rev./min., and $k = 0{\cdot}88$.

Calculate the necessary blade inlet angle, draw the velocity diagram, and find (*a*) the horse-power per lb. expended on the blading, (*b*) the axial thrust, (*c*) the carry over to the condenser. Check the results by direct calculation.

Blade velocity $u = \frac{\pi DN}{60} = \frac{\pi \times 54 \times 2400}{60 \times 12}$ $\quad D = 54$ in. $\quad N = 2400$

$= 565{\cdot}5$ ft./sec.

Speed ratio, $\rho = \frac{u}{V_{a1}} = \frac{565{\cdot}5}{1410} = 0{\cdot}401$ $\quad V_{a1} = 1410$

$\alpha_1 = 25$
$\sin \alpha_1 = 0{\cdot}4226$
$\cos \alpha_1 = 0{\cdot}9063$

$$\tan \theta_1 = \frac{\sin \alpha_1}{\cos \alpha_1 - \rho} = \frac{0{\cdot}4226}{0{\cdot}9063 - 0{\cdot}401} = \frac{0{\cdot}4226}{0{\cdot}5053} = 0{\cdot}837$$

Hence $\theta_1 \doteqdot 40°$

$\theta_1 = 40$, $\quad \theta_0 = 45$, $\quad k = 0{\cdot}88$, $\quad c = 1 + \frac{k \cos \theta_0}{\cos \theta_1}$

$\cos \theta_1 = 0{\cdot}766$ $\quad \cos \theta_0 = 0{\cdot}7071$

$$= 1 + \frac{0{\cdot}88 \times 0{\cdot}7071}{0{\cdot}766}$$

(*a*) Horse-power per lb. $\quad = 1{\cdot}813$

$$\text{H.P.} = \frac{u^2 c}{550g}\left(\frac{\cos \alpha_1}{\rho} - 1\right)$$

$$= \frac{565{\cdot}5^2 \times 1{\cdot}813}{32{\cdot}2 \times 550}\left(\frac{0{\cdot}9063}{0{\cdot}401} - 1\right)$$

$$= \frac{565{\cdot}2^2 \times 1{\cdot}813 \times 1{\cdot}26}{32{\cdot}2 \times 550} = 41{\cdot}3$$

Scaling the diagram, the vector difference of the velocities of whirl is $V_w=1292$ ft./sec.

$$\text{H.P.}=\frac{uV_w}{550g}=\frac{565{\cdot}5\times1292}{550\times32{\cdot}2}=41{\cdot}3$$

(*b*) Normal velocity change at entrance—

$$y_1=u\left(\frac{\cos\alpha_1}{\rho}-1\right)\tan\theta_1$$
$$=565{\cdot}5\times1{\cdot}26\times0{\cdot}839=597{\cdot}8 \text{ ft./sec.}$$

The value scaled from the diagram is 597·5 ft./sec.

$$y_0=\frac{k\sin\theta_0}{\cos\theta_1}u\left(\frac{\cos\alpha_1}{\rho}-1\right) \qquad \sin\theta_0=0{\cdot}707$$
$$=\frac{0{\cdot}88\times0{\cdot}707\times565{\cdot}5\times1{\cdot}26}{0{\cdot}766}=578 \text{ ft./sec.}$$

This checks on the diagram.

$$\text{Axial thrust} \quad F=\frac{1}{g}(y_1-y_0)=\frac{1}{32{\cdot}2}(597{\cdot}8-578)$$
$$=\frac{19{\cdot}8}{32{\cdot}2}=0{\cdot}615 \text{ lb.}$$

The absolute velocity of exit is obtained from—

$$V_{a0}^2=(kV_{r1})^2+u^2-2ukV_{r1}\cos\theta_0$$
$$V_{r1}=\frac{V_{a1}\cos\alpha_1-u}{\cos\theta_1}=\frac{1410\times0{\cdot}9063-565{\cdot}5}{0{\cdot}766}=\frac{1277{\cdot}9-565{\cdot}5}{0{\cdot}766}$$
$$=\frac{712{\cdot}4}{0{\cdot}766}=930 \text{ ft./sec.}$$

This checks on the diagram—

$$V_{a0}^2=(0{\cdot}88\times930)^2+(565{\cdot}5)^2-2\times565{\cdot}5\times0{\cdot}88\times930\times0{\cdot}7071$$
$$=761112+319790-654499=426403$$

$$\text{Carry over} \quad h_e=\frac{V_{a0}^2}{(223{\cdot}7)^2}=\frac{426403}{(223{\cdot}7)^2}=8{\cdot}52 \text{ B.Th.U./lb.}$$

The absolute velocity vector at exit is inclined at nearly 90° to the direction of motion of the blading. By substitution of the necessary values in Equation 69, the value of $\tan\alpha_0$ is found to be 420, which corresponds approximately to the tangent of 89° 50′; and the diagram checks this.

**370. Diagram for Velocity Compounded Impulse Stage.**—The extended velocity diagrams for a two-velocity compounded stage are shown in Fig. 432. In order to distinguish the velocities and angles at the successive rings, subscripts 1 and 2 are added to the left of the symbols, denoting the first and second moving rings.

The guide blade angles are denoted by dashes. The mean blade

speed $u$ is the same for both rings. It will be evident that the absolute velocity of exit, ${}_1V_{a0}$, from the first moving ring is the entrance velocity to the fixed or guide blade. The exit jet angle ${}_1a_0$ is therefore the same as the entrance guide blade angle $\theta_1'$.

The guide blade is given an exit angle $\theta_0'$, at which the steam is discharged into the second moving ring with the absolute entrance velocity ${}_2V_{a1}$. This angle is the same as the second entrance angle ${}_2a_1$ of the jet.

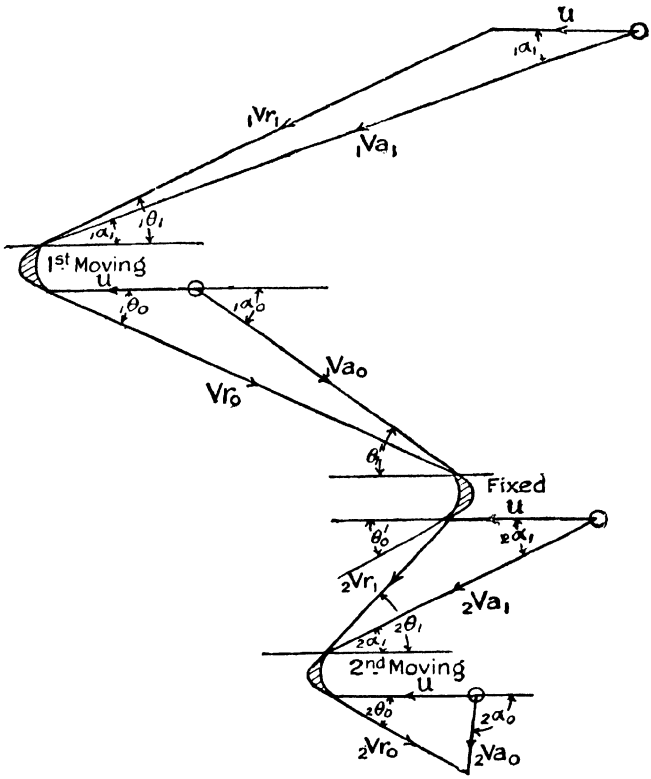



FIG. 432.

The work done on each blade ring can, as before, be obtained by superposing the inlet and outlet triangles. As the mean blade speed is always the same for all rings, the whole series can be superposed to form one diagram, as shown in Fig. 433.

For the sake of clearness the triangles of the second moving ring are dotted. The change of velocity of whirl for the first ring is ${}_1V_w$, and that for the second ${}_2V_w$, and the total work done on the two rings of moving blades is—

$$E_b = \frac{u}{g}({}_1V_w + {}_2V_w) \quad . \quad . \quad . \quad . \quad . \quad . \quad (72)$$

Any number of diagrams can be superposed and dealt with in this way, so that generally the total work done on a series of rings may be expressed by—

$$E_b = \frac{u}{g} \Sigma V_w \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (73)$$

*Example* 8.—The mean blade velocity of a two-velocity compounded wheel is 500 ft./sec., and the speed ratio is 0·2. The jet angle is 20°, and the exit angles of the first moving, fixed, and second moving rings are 22°, 24°, and 35°. Assuming the following values of the velocity coefficient for the blades, first moving $k_1=0{\cdot}84$; fixed $k_f=0{\cdot}87$; second moving $k_2=0{\cdot}9$, draw down the combined velocity diagram and calculate: (*a*) the total horse-power expended on the blade rings; (*b*) the ratio of the power on the second to that on the first ring; (*c*) the proportions of the initial jet energy lost by friction reheat in the first moving, fixed, and second moving rings,

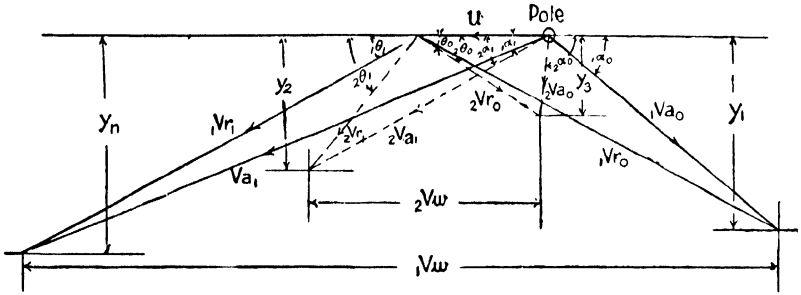



Fig. 433.

and in the carry over. Also find the necessary inlet angles for the three blades.

Here $u=500$, $\rho=0{\cdot}2$, and $V_{a1}=\frac{u}{\rho}=\frac{500}{0{\cdot}2}=2500$ ft./sec.

$_1a_1=20°$, $_2a_1=24°$, $_1\theta_0=22°$, $_2\theta_0=35°$,
$k_1=0{\cdot}84$, $k_f=0{\cdot}87$, $k_2=0{\cdot}9$.

Drawing the two sets of triangles as in Fig. 433 with these values and scaling, the following velocities are obtained.

$_1V_{r1}=2045$, $_1V_{r0}=1718$, $_1V_{a0}=1260$.
$_2V_{a1}=1096$, $_2V_{r1}=670$, $_2V_{r0}=603$, $_2V_{a0}=345$; and the blade inlet angles are: $_1\theta_1=25°$, $_f\theta_1=30°$, $_2\theta_1=42°$.

(*a*) The velocities of whirl are $_1V_w=3430$, $_2V_w=990$.

Horse-power expended on the blading—

$$\text{H.P.}=\frac{u}{550g}(_1V_w+{}_2V_w)$$

$$=\frac{500\times4420}{550\times32{\cdot}2}=125 \text{ per lb. of steam.}$$

(*b*) Ratio of powers $=\left(\frac{{}_2E_b}{{}_1E_b}\right)=\frac{{}_2V_w}{{}_1V_w}=\frac{990}{3430}=0{\cdot}289$

The second ring takes less than a third of the energy usefully expended on the blading of the first ring.

(*c*) Fraction of jet energy in reheat, first moving ring—

$x_1=(1-k_1^2)\left(\frac{{}_1V_{r1}}{{}_1V_{a1}}\right)^2=0{\cdot}294\left(\frac{2045}{2500}\right)^2=0{\cdot}1{\cdot}81$ $\quad (1-k_1^2)=(1-0{\cdot}706)=0{\cdot}294$

Fixed guide ring— $\quad (1-k_f^2)=(1-0{\cdot}757)=0{\cdot}243$

$x_f=(1-k_f^2)\left(\frac{{}_1V_{a0}}{{}_1V_{a1}}\right)^2=0{\cdot}243\left(\frac{1260}{2500}\right)^2=0{\cdot}0618$ $\quad (1-k_2^2)=(1-0{\cdot}81)=0{\cdot}19$

Second moving ring—

$x_2=(1-k_2^2)\left(\frac{{}_2V_{r1}}{{}_1V_{a1}}\right)^2=0{\cdot}19\left(\frac{670}{2500}\right)^2=0{\cdot}0138$

Carry over loss—

$x_e=\left(\frac{{}_2V_{a0}}{{}_1V_{a1}}\right)^2=\left(\frac{345}{2500}\right)^2=0{\cdot}019$

The ratio of the effective work to the initial jet energy is—

$$x_b=\frac{E_b}{\frac{{}_1V_{a1}{}^2}{2g}}=\frac{500\times4420\times2}{2500\times2500}=0{\cdot}7073$$

The sum of these fractional values should be unity, or very nearly unity, if the construction and calculations are correct. Thus—

$(x_1+x_f+x_2+x_e+x_b)=(0{\cdot}1981+0{\cdot}0618+0{\cdot}0138+0{\cdot}019+0{\cdot}7073)=1{\cdot}00.$

**371. Velocity Diagram for a Reaction Stage.**—In a reaction stage there is an increase of relative velocity between entrance and exit, due to the pressure drop in the blade channel. The fixed and moving rings have identical blades, and the heat drop for the "double" stage is divided equally between the rings. Assuming the same friction losses in the identical rings, it follows that, with the same exit angle $\theta=\alpha_1$, the exit and entrance triangles are identical, as shown in Fig. 434. Thus $V_{r0}=V_{a1}$ and $V_{a0}=V_{r1}$.

From the geometry of the figure it will be apparent that the difference of the velocity of whirl is $V_w=(2V_{a1}\cos\theta-u)$. The work done per lb. on the moving ring of the "double" stage is therefore given by—

$$E_b=\frac{u}{g}(2V_{a1}\cos\theta-u)$$

or $$E_b=\frac{u^2}{g}\left(\frac{2\cos\theta}{\rho}-1\right). \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (74)$$

where $\theta$ is the constant blade exit angle and $\rho = \frac{u}{V_{a1}} = \frac{u}{V_{r0}}$. The constant entrance angle for each ring is $\theta_1$, and—

$$\tan \theta_1 = \frac{V_{a1} \sin \theta}{V_{a1} \cos \theta - u}$$

$$\text{or} \quad \tan \theta_1 = \frac{\sin \theta}{\cos \theta - \rho} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (75)$$

In this case, as a rule, the necessary particulars can be more easily obtained by direct calculation from Equations 74 and 75. It should be noted that if the exit blade angle $\theta$ is increased and the blade speed and change of velocity of whirl $V_w$ remain constant, the speed ratio $\rho$ is reduced but the work done on the blades remains the same. This will be apparent from an examination of the dotted triangles, which show increased values of $\theta$ but the same

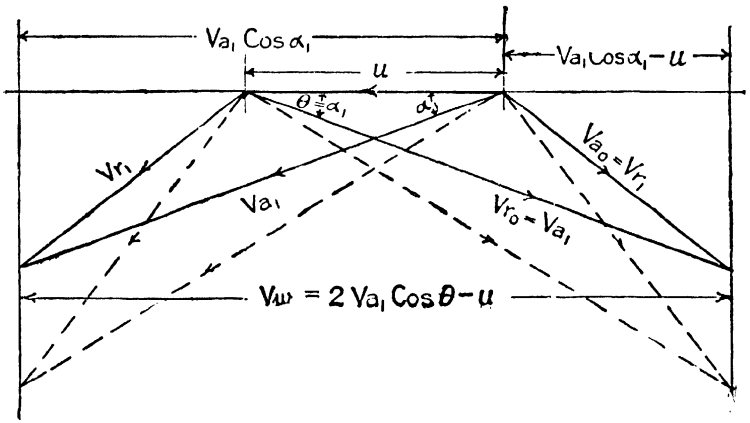



FIG. 434.

value of $V_w$ as the full line diagram. In the first case the value of $\rho$ is 0·5 ; in the second it is 0·455. With the condition of identical velocity triangles for a " double " reaction stage it will be evident that the axial components of the steam velocity are equal, and the dynamical thrust is zero. With the usual construction, however, in which the blade heights are kept constant throughout a group or " expansion," there is a progressive increase of velocity, and consequently a slight thrust. Normally this is too small to be of any account, and it is neglected. There is, however, a statical thrust which does not occur in an impulse turbine. This is due to the difference of pressure between entrance and exit at each moving ring. The cumulative thrust in earlier designs of reaction machines was balanced by means of dummy pistons, but as already indicated in Art. 348, in modern designs the dummy piston is being dispensed with and the thrust is now usually taken up by a Michell block.

*Example* 9.—In a Parsons reaction turbine the last two groups or expansions have a common mean ring diameter of $54\frac{1}{2}$ in. The exit angle at the second last group is $\theta=20°$ and the speed ratio $\rho=0{\cdot}55$. At the last group the angle is increased to 31°. The rotational speed is 1500 rev./min. Calculate the heat equivalent of the work done on each blade ring of the second last group, and the average value of the inlet angle. Also the reduced speed ratio for the last group for the same amount of work per ring.

Mean blade velocity $u=\frac{\pi DN}{60}=\frac{\pi\times 54{\cdot}5\times 1500}{12\times 60}=356$ ft./sec.

$\theta=20°$, $\cos\theta=0{\cdot}94$, $\rho=0{\cdot}55$.

Equation 74—

$$E_b=\frac{u^2}{g}\left(\frac{2\cos\theta}{\rho}-1\right)$$

$$=\frac{356^2}{32{\cdot}2}\left(\frac{2\times 0{\cdot}94}{0{\cdot}55}-1\right)=9521 \text{ ft.-lb./lb. steam.}$$

$$\therefore\ h_b=\frac{E_b}{J}=\frac{9521}{778}=12{\cdot}24 \text{ B.Th.U./lb.}$$

Entrance angle by Equation 75— $\sin 20°=0{\cdot}342$

$$\tan\theta_1=\frac{\sin\theta}{\cos\theta-\rho}=\frac{0{\cdot}342}{0{\cdot}94-0{\cdot}55}=\frac{0{\cdot}342}{0{\cdot}39}=0{\cdot}88$$

$$\therefore\ \theta_1=41°\ 31'$$

Since the work done on the rings of the last group is the same as above, then from Equation 74 it follows that—

$$\frac{\cos\theta}{\rho}=\frac{\cos\theta'}{\rho'}$$

where $\theta'$ and $\rho'$ are the increased angle and decreased speed ratio—

$$\therefore\ \rho'=\rho\frac{\cos\theta'}{\cos\theta}=\frac{0{\cdot}55\times 0{\cdot}8572}{0{\cdot}94}=0{\cdot}502 \qquad \begin{matrix}\theta'=31°\\ \cos\theta'=0{\cdot}8572\end{matrix}$$

**372. Diagram Efficiency Ratio** ($\epsilon_d$).—The kinetic energy of the steam jet entering a ring of turbine blading is $\frac{V_{a1}^2}{2g}$, while the work expended on the ring is $E_b$, or in the case of $m$ moving rings ${}_mE_b$. The ratio of the work done to the jet energy is usually termed the diagram efficiency ratio, since $E_b$ can be directly obtained from the velocity diagram. Thus—

$$\epsilon_d=\frac{E_b}{V_{a1}^2/2g} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (76)$$

For the simple impulse stage it is—

$$\epsilon_d=\frac{2uV_w}{V_{a1}^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (77)$$

For a velocity compounded stage—

$$\epsilon_d = \frac{2u\Sigma V_w}{V_{a1}^2}$$

and for the reaction (double stage)—

$$\epsilon_d = \frac{2u^2}{V_{a1}^2}\left(\frac{2\cos\theta}{\rho} - 1\right)$$

Assuming a sensibly constant friction loss in the blading, then for a given blade ring and jet velocity there will be some value of the speed ratio $\rho$ for which the work done on a blade ring, and hence the diagram efficiency ratio, is a maximum.

Taking the simple impulse stage—

$$E_b = \frac{u^2 c}{g}\left(\frac{\cos\alpha_1}{\rho} - 1\right)$$

and since $u^2 = \rho^2 V_{a1}^2$, it follows that—

$$E_b \propto \rho^2\left(\frac{\cos\alpha_1}{\rho} - 1\right)$$

$$\text{or} \quad E_b \propto \rho\cos\alpha_1 - \rho^2$$

For $E_b$ a maximum $\frac{dE_b}{d\rho} = 0$

Hence $\cos\alpha_1 - 2\rho = 0$ and $\rho = \frac{1}{2}\cos\alpha_1$, so that—

$$u = \tfrac{1}{2}V_{a1}\cos\alpha_1 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (78)$$

In the case of the velocity compounded stage, with the arbitrary variation of velocity coefficients and angles, a general expression for $\Sigma V_w$ cannot be obtained, and no simple relation between $u$ and $V_{a1}$ can be stated. If, however, the blades were made symmetrical, so that a definite progression of angle throughout the set would be obtained, and if the coefficient in each case were taken as unity, an approximation to the value of $\rho$, for maximum diagram efficiency, could be obtained from the resulting combined velocity diagram. This diagram would be a triangle drawn on a base $n$ times the common blade ring velocity $u$, the other sides being the absolute inlet jet velocity ${}_1V_{a1}$ and the absolute exit velocity at the $n$th ring ${}_nV_{a0}$.

In this case the ideal work done on the blading would be—

$$E_b = \frac{{}_1V_{a1}^2 - {}_nV_{a0}^2}{2g}$$

By the geometry of the figure—

$${}_nV_{a0}^2 = {}_1V_{a1}^2 + (2nu)^2 - 2(2nu)_1V_{a1}\cos\alpha_1$$

and

$$E_b = \frac{1}{2g}(4n^2u^2 - 4nu_1V_{a1}\cos\alpha_1)$$

$$= \frac{1}{2g}(4n^2{}_1V_{a1}^2\rho^2 - 4n_1V_{a1}^2\rho\cos\alpha_1)$$

$$\therefore \frac{dE_b}{d\rho}=8n^2\rho-4n\cos\alpha_1=0$$

$$\text{and} \quad \rho=\frac{1}{2}\frac{\cos\alpha_1}{n}$$

$$\text{or} \quad u=\frac{1}{n}\left(\frac{{}_1V_{a1}\cos\alpha_1}{2}\right) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (79)$$

The blade speed for maximum efficiency of the velocity compounded stage, in the actual case, will thus approximate to $\frac{1}{n}$th of the speed for the simple impulse stage.

The exact value for given conditions (such as those specified in Example 8) can be determined by trial and error adjustment of the speed ratio.

In the case of the reaction stage, it follows from Equation 74 that—

$$E_b=\frac{2V_{a1}^2\rho\cos\theta}{g}-\frac{V_{a1}^2\rho^2}{g}$$

Hence $$\frac{dE_b}{d\rho}=2\cos\theta-2\rho=0$$

and $$\rho=\cos\theta$$

$$\text{or} \quad u=V_{a1}\cos\theta \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (80)$$

**373. Stage Efficiency Ratio** ($\epsilon_s$).—The diagram efficiency ratio, while serving for the estimation of the best ratio of jet and blade velocity, does not account for all the disturbing influences affecting each stage of a turbine.

At any stage there is a pressure drop, and the amount of heat available for conversion into kinetic energy for the performance of the stage work is the Rankine cycle heat $h_\phi$ or heat drop between the stage pressure limits. Only a portion of this energy, however, is transmitted through the shaft from the stage in question.

The remainder, in an impulse turbine with few stages, may all be reconverted to heat, through friction in the nozzle and blade channels, disc and vane friction, and in eddying and shock on the chamber walls.

In the multi-stage impulse and reaction machines a proportion of the energy, developed originally in the first stage, is carried forward from stage to stage if the steam flows without sensible obstruction into the consecutive nozzle passages.

The ratio between the net work done on the rotor shaft in the stage and the stage heat drop is, as stated in Chapter IV, the stage efficiency ratio. Denoting the net work by $E_b'$, then—

$$\epsilon_s=\frac{E_b'}{Jh_\phi} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (81)$$

The net work $E_b'$ is the work per lb. of steam ($E_b$) corrected in the case of the impulse machine for disc friction and leakage loss at

the diaphragms, and in the case of a reaction machine for tip leakage loss.

In general the efficiency can be stated as—

$$\epsilon_s=\frac{\gamma E_b}{Jh_\phi}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (82)$$

where $\gamma$ is a correction coefficient. In the impulse machine at the L.P. it is practically unity. At the first few stages it may vary from 0·94 to 0·98. In the axial flow reaction turbine the drum and blade friction are negligible, but the blade tip leakage at the H.P. end may be appreciable where end tightening is not employed. A handy rule for estimating the effect of the blade leakage loss is to regard 1 per cent. of leakage area as representing 2 per cent. loss in stage efficiency.*

The progressive loss throughout the blading of a reaction turbine is thus obtained by taking the progressive leakage area, radial or axial, and doubling it. For instance, if at a certain ring of blading the blade length is 1·5 in. and the radial clearance is 0·026, the stage efficiency loss will approximate to $\frac{2\times0\cdot026}{1\cdot5}=0\cdot034$ or $\gamma=(1-0\cdot034)=0\cdot966$.

In the case of an impulse turbine where there is no carry over and the steam is brought to rest in each stage, with reconversion of kinetic energy to heat, it will be apparent that for the same amount of work per stage a larger heat drop than that with carry over is required, and there is a reduction in the value of the stage efficiency.

The effect, however, depends on the ratio of the residual to the initial jet velocity in each stage. If this is small, the effect on the efficiency is slight.

If the net work expended by the steam on the rotor shaft is expressed as an equivalent heat drop—

$$h_s=\gamma\frac{E_b}{J}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (83)$$

the stage efficiency ratio is also given by—

$$\epsilon_s=\frac{h_s}{h_\phi}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (84)$$

*Example* 10.—Calculate the probable stage efficiency of the last stage of the impulse turbine (Example 7) for which the factor $\gamma$ may be taken as unity and the nozzle efficiency as 0·95.

$u=565\cdot5$
$V_w=1292$
$\eta_n=0\cdot95$
$V_{a1}=1410$

Here $h_s=\frac{\gamma E_b}{J}=\frac{uV_w}{gJ}=\frac{565\times1292}{32\cdot2\times778}=29\cdot2$ B.Th.U./lb.

Heat drop $h_\phi=\left(\frac{V_{a1}}{223\cdot7}\right)^2\times\frac{1}{\eta_n}=\left(\frac{1410}{223\cdot7}\right)^2\times\frac{1}{0\cdot95}$

$=41\cdot8$ B.Th.U./lb.

* Recent experiments have indicated that this figure is low, and 3 per cent. is more likely. See *Engineering*, 4 July, 1929, p. 3.

Then stage efficiency ratio—

$$\epsilon_s=\frac{h_s}{h_\phi}=\frac{29\cdot2}{41\cdot8}=0\cdot7$$

*Example* 11.—Calculate the probable stage efficiency ratio of the two velocity compounded stage of Example 8 on the assumption that $\gamma=0\cdot98$ and $\eta_n=0\cdot92$.

$$h_s=\frac{\gamma E_b}{J}=\frac{0\cdot98\times500\times4420}{32\cdot2\times778}=86\cdot5$$

$$h_\phi=\left(\frac{V_{a1}}{223\cdot7}\right)^2\times\frac{1}{\eta_n}=\left(\frac{2500}{223\cdot7}\right)^2\times\frac{1}{0\cdot92}=136$$

$$\therefore\ \epsilon_s=\frac{h_s}{h_\phi}=\frac{86\cdot5}{136}=0\cdot636$$

$\gamma=0\cdot98$
$E_b=\frac{500\times4420}{32\cdot2}$
$\eta_n=0\cdot92$
$V_{a1}=2500$

It can be shown on the assumption that, for the reaction stage, when the steam is treated as a fluid like water flowing through the blade channel and the frictional loss of energy is proportional to the square of the velocity, the heat drop is given by—

$$h_\phi=\frac{u^2}{2gJ}\left\{\frac{\cos\theta}{\rho}(4-2\phi)+\phi\left(\frac{2}{\rho^2}+1\right)-2\right\}\quad\ldots\quad(85)$$

where $\phi$ is a friction factor for the blade channels varying from 0·2 to 0·3.*

*Example* 12.—Calculate the stage efficiency ratio for the reaction double stage (Example 9) on the assumption that $\gamma=0\cdot93$ and $\phi=0\cdot25$.

In this case $E_b=9521$ ft.-lb./lb. steam, $u=356$, $\rho=0\cdot55$, $\theta=20°$, $\cos\theta=0\cdot94$

$$h_\phi=\frac{356^2}{2\times32\cdot2\times778}\left\{\frac{0\cdot94}{0\cdot55}(4-0\cdot5)+0\cdot25\left(\frac{2}{0\cdot55^2}+1\right)-2\right\}=14\cdot87$$

$$h_s=\frac{\gamma E_b}{J}=\frac{0\cdot93\times9521}{778}=11\cdot38$$

$$\therefore\ \epsilon_s=\frac{h_s}{h_\phi}=\frac{11\cdot38}{14\cdot87}=0\cdot765$$

**374. Experimental Values of Stage Efficiency Ratio.**—It is the practice to express stage efficiency ratios, experimentally determined, in terms of the theoretical speed ratio. This may be denoted by $\rho_t$, and it is obtained by multiplying the actual speed ratio by the square root of the nozzle efficiency. Thus—

$$\rho_t=\rho\sqrt{\eta_n}=\frac{u}{V_0'}\quad\ldots\quad(86)$$

where $V_0'$ is the theoretical velocity of exit from the nozzle passage.

* For deduction of this expression, see *Steam Turbines*, by W. J. Goudie, Chapter IX.

An average set of curves, derived from various sources, for one, two, and three ring impulse stages, and for the reaction stage, is given in Fig. 435.

These curves are intended to apply to blading with what may be termed normal exit angles. For wide-angled blading the values obtained from them are on the high side. In the case of Example 7 the exit blade angle of the last stage ring has the high value of 45°. In this case $\rho_t = 0{\cdot}401 \times \sqrt{0{\cdot}95} = 0{\cdot}39$, and from the curve $\epsilon_s = 0{\cdot}78$,

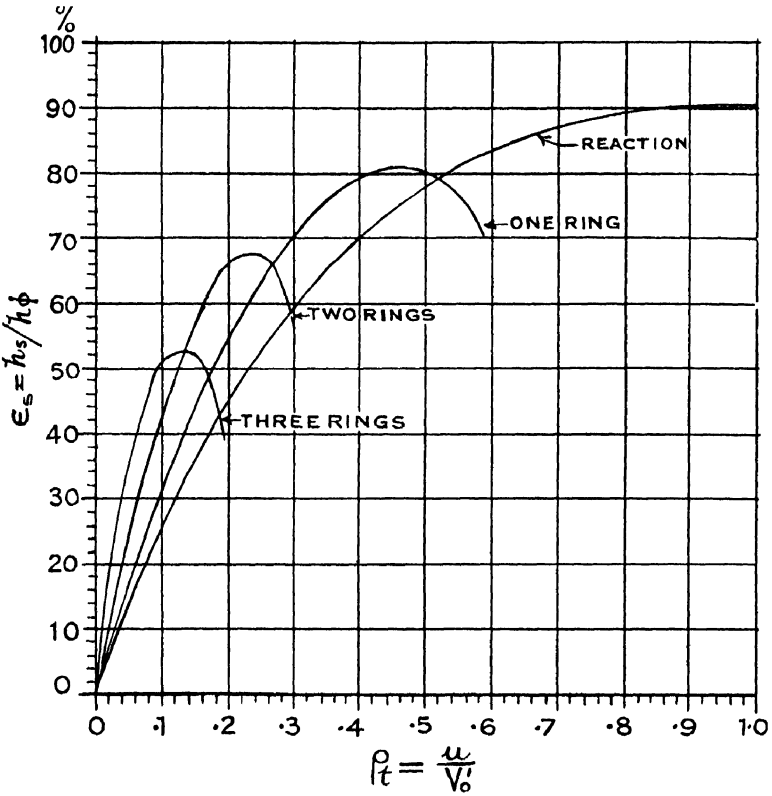



FIG. 435.

while the detailed calculation gives $\epsilon_s = 0{\cdot}7$. In the case of Example 8 with a normal blading arrangement of the two-velocity compounded stage, $\rho_t = 0{\cdot}2 \times \sqrt{0{\cdot}92} = 0{\cdot}192$, and the curve shows $\epsilon_s = 0{\cdot}64$ as against $\epsilon_s = 0{\cdot}636$ by detailed calculation. For the reaction blading (Example 9) if a nozzle efficiency of, say, 0·95 is assumed, then $\rho_t = 0{\cdot}53$ and the curve shows $\epsilon = 0{\cdot}8$ as against 0·765 by the detailed calculation.

**375. Exit Blade Angle $\theta_0$.**—The velocity diagram shows that the smaller this angle the greater the amount of work done on the blade

ring. As the angle decreases the blade height at exit has to increase, in order to obtain the necessary area to pass the steam. When $\theta_0$ becomes very small the blade length becomes impracticable. Also the smaller the angle the greater the channel length and wall surface, a condition that increases the friction loss. Reduction of angle between entrance and exit conforms to the reduction in the circumferential width of the jet which takes place when a fluid flows over a curved surface. The jet tends to spread in the lateral or radial direction, and this condition involves some loss of energy. The steam acquires a velocity in the radial or non-effective direction. This spreading is checked by the shroud ring of the blading.

The choice of the value of $\theta_0$ is arbitrary; but in making this, care has to be taken that for any given case the blade length is not unduly greater than the exit length of the nozzles discharging into the wheel blading. When this occurs, the radial divergence of the blade channel entails too great a spread of the steam with increase of the non-effective component of the velocity. A certain difference of height has to be given, to avoid spilling over the shroud at entrance. The entrance lengths of the wheel blade at the H.P. end may be made about $\frac{1}{8}$ in. greater than the nozzle exit length, and this may increase to $\frac{1}{2}$ in. for the blading at the L.P. end.

In practice it has been found inadvisable to use a blade exit angle smaller than 12°. It usually varies from 12° to 30° at high and intermediate stages. At low-pressure stages, when the ordinary type of uniform impulse blade is used, the practical limitation of blade length may necessitate an increase from 35° to 40°.

In velocity compounded machines at high-pressure stages the series of stage angles may vary from 22° to 30° or 40°. At low-pressure stages slightly higher values may be used, 24° to 45°.

In Parsons axial flow turbines the angle may vary from 15° to 20°, except at the last few L.P. stages, where with uniform blade section "semi-wing," "wing," and "double wing" blades are used. In these cases it may increase from 25° to 50°.

There is, however, a form of multiple-exhaust arrangement, due to Baumann, in which the flow of steam in subdivided, which enables the designer to keep the blade angles of the outer tiers between 20° and 30°, with substantial reduction of carry-over loss.

In two-cylinder machines double flow L.P. sections are often employed to attain the same object.

**376. Maximum Blade Length.**—In determining the maximum length of blade at the low-pressure end of a turbine several conditions have to be considered. In the first place the blade length must be kept a reasonable proportion of the mean or pitch diameter, otherwise the pitching at the tips becomes too large for the proper control of the flow, and disorderly motion of steam in the channel

ensues, with loss of efficiency. Until recently a ratio of blade length to mean diameter of 20 per cent. was considered a desirable limit; but this in some recent designs has been increased to 30 per cent., with special forms of blading.

The important question of blade strength and stiffness has to be considered, and this involves the permissible blade speed Within the past few years blade *tip* speed has been raised from 700 ft./sec. to 1000 ft./sec., and in a few cases speeds of 1150 ft./sec. are being used. One of these machines, by the Westinghouse Co., running at this speed has a final L.P. blade length of 34 in. on a spindle diameter of 78 in. The blades are mounted on a disc wheel forged from $3\frac{1}{2}$ per cent. nickel steel.

**377. Minimum Blade Length.**—In the impulse turbine this figure is controlled by mechanical considerations. The smallest blade length adopted is $\frac{3}{8}$ in. In large machines it runs from 1 in to $1\frac{1}{2}$ in. at the first H.P. stage. In general the minimum value runs from $1\frac{1}{2}$ to 2 per cent. of the mean ring diameter. In a Parsons reaction turbine full peripheral admission is necessary, from the first stage onward, and the minimum length is fixed with reference to tip leakage. It should not be less than 3 per cent. of the mean ring diameter. In recent practice the system of end-tightening has become the rule, and on account of the much finer axial clearances that can be allowed, longer blades can now be fitted.

**378. Radial and Axial Blade Clearance in Impulse Turbines.**—There is no thermodynamic reason for limiting the radial clearances in the impulse machine. In large pressure compounded machines it may vary from 0·18 to 0·25 in. In this type the real clearance difficulty does not arise at the blade tips but at the diaphragm glands. Axial clearance between the nozzles and blades, on the other hand, should be kept as low as possible, especially at the high-pressure end, where there is partial admission. Small clearances help to reduce the friction and eddy loss, where the band of steam flowing between the nozzles and blades cuts across the edge of the zone of dead steam in the clearance space. This axial clearance varies from 0·10 to 0·20 in. and depends on the size of the machine. In the case of the velocity compounded stage it is usual to give a radial increase at the entrance edge of each successive blade, in order to avoid spilling over the ends of the blades. In the de Laval turbine the nozzle exit is kept 0·06 in. clear of the entrance side of the wheel blades.

**379. Radial and Axial Blade Clearance in Parsons Turbines.**—At the high-pressure end when end-tightening is not adopted the radial clearance has to be reduced to the smallest possible value consistent with safe running.

The practical minimum is about 0·02 in. when the turbine is cold. An empirical equation for the axial pitch given by J. M. Newton,*

* "High Speed Turbine Rotor Design and Construction," *Trans. Junior Inst. of Engineers*, 1910.

which is still valid for modern designs running with moderately superheated steam, has the form—

$$P_a = 2B + \tfrac{1}{4} + \frac{l}{16} \quad . \quad . \quad . \quad . \quad . \quad . \quad (87)$$

where $P_a$ = axial pitch in inches
B = blade breadth in inches
$l$ = blade length in inches.

**380. Circumferential Pitch of Blades.**—The number of blades to be provided at any ring of a turbine disc or rotor should just be sufficient to efficiently direct the flow of the steam through them. An unnecessarily large number simply increases the friction loss, through the addition of superfluous surface. The most suitable value of the pitch can only be determined by experiment. It does not appear advisable to reduce it below $\frac{3}{8}$ in. or to increase it above 1 in., when the ordinary form of crescent-shaped impulse blade is used. For pitches between $\frac{3}{8}$ in. and $\frac{3}{4}$ in. the blade efficiency does not appear to vary to any appreciable extent. In practice the pitch in impulse turbines runs from 0·3 to 0·7 in., and in reaction turbines from 0·2 to 0·6 in.

There is a rough practical rule used for impulse blading by which the circumferential pitch $P_b$ is made equal to the radius of curvature of the blade face, and this reduces to—

$$P_b = \frac{b}{2 \sin 2\theta_1} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (88)$$

where $b$ is the axial blade width and $\theta_1$ is the entrance angle. This expression enables a suitable value of the pitch to be determined when the blade width and angle are known.

In the case of the velocity compounded stage, the pitch is fixed with reference to the last blade, which has the greatest entrance angle. As this may run from 40° to 50° the denominator of (88) may vary from 1·96 to 2, so that the pitch may be taken from 0·52 to 0·5 of the axial breadth ($b$).

# CHAPTER XXII

## CONDITION CURVE. REHEAT FACTOR AND EFFICIENCIES HEAT CONSUMPTION RATES, AND DIMENSIONS OF COMPOUND STEAM TURBINES

**381. Condition Curve.**—The quality of the steam, superheat or dryness, on discharge from any stage of a compound turbine, is always higher than that due to adiabatic expansion between the stage pressure limits.

This "reheat," as already indicated, is due to friction losses of kinetic energy in nozzle and blade channels, disc and vane friction, leakage at diaphragm glands in impulse, and at blade tips in reaction turbines.

The progressive effect of reheat on the stage work is best shown by the TΦ diagram. In Fig. 436 the diagram for a four-stage impulse machine, working with steam initially dry and finally wet, is shown. The initial stage pressures are $p_1$, $p_2$, $p_3$, $p_4$, and the exhaust pressure is $p_0$. If there were no "frictional" losses, the qualities of the steam at the successive stages would be given by the location of the points of intersection of the adiabatic CN with the respective pressure horizontals. Due to reheat, however, each state point is moved to the right of the position for adiabatic expansion.

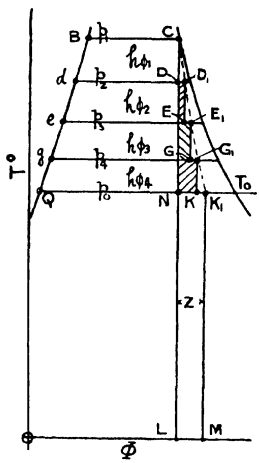


FIG. 436.

In the first stage, between $p_1$ and $p_2$, the Rankine cycle heat or heat drop $h_{\phi 1}$ is given by the area $d$BCD. At $p_2$ the initial pressure of the second stage, the state point is moved from D to $D_1$; the heat drop of the second stage is thus $h_{\phi 2}$, given by the area $cd$D$_1$E. Similarly at the initial pressure $p_3$ of the third stage the state point is moved to $E_1$, and the heat drop is $h_{\phi 3}$ given by the area $ge$E$_1$G. At the initial pressure $p_4$ of the last stage the state point falls at $G_1$, and the heat drop $h_{\phi 4}$ is given by the area Q$g$G$_1$K. Finally the state point at the exhaust pressure $p_0$ is moved from K to $K_1$, which defines its condition on exhaust from the turbine.

The curve $CD_1E_1G_1K_1$ drawn through the initial state points is called the " condition curve " of the compound turbine.

**382. Reheat Factor** R.—The ultimate effect of the frictional losses is to increase the entropy at the exhaust pressure by the amount $NK_1=Z$; and the amount of heat discharged in the exhaust is increased by the area $LNK_1M=T_0Z$.

As shown by the cross-hatched areas, the heat drop in the successive stages is an increasing quantity, and the summation of these heat values gives a larger total area than the area QBCN due to adiabatic expansion between the initial and exhaust pressures $p_1$ and $p_0$.

Thus $h_{\phi 1}+h_{\phi 2}+h_{\phi 3}+h_{\phi 4}>H_\phi$. This relation can be concisely expresssed by—

$$\Sigma h_\phi = RH_\phi. \quad . \quad . \quad . \quad (1)$$

where R is a coefficient called the " reheat factor." Its value varies with the size and type of turbine.

**383. Cumulative Cycle Heat $H_\phi$ for the Straight Rankine Cycle.**—The summation of the heat drops or $\Sigma h_\phi$ is called the cumulative cycle heat of the compound turbine. In the simple case illustrated in Fig. 436 it exceeds $H_\phi$ by the amount of heat represented by the sum of the shaded areas to the right of the adiabatic CN. When the number of stages increases the adiabatic heat drops become smaller and smaller, and in the limit when the turbine has a large number of stages, the stepped curve $CDD_1EE_1GG_1KK_1$ coincides with the condition curve $CK_1$. The heat equivalent of the additional work is given by the area $CNK_1$. The same effect is produced when the steam is initially superheated, as shown in Fig. 437. The initial state point at $p_1$ and superheat $t_{s1}$ is $F_1$, and the final state point at $p_0$ is K.

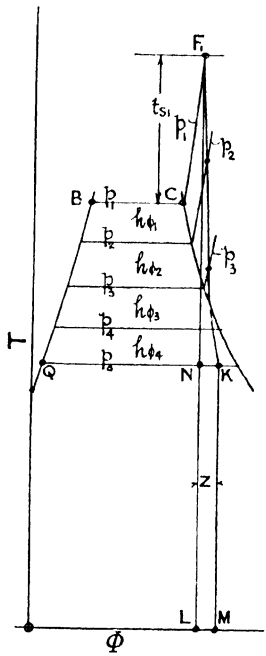



FIG. 437.

The steam is still superheated at $p_2$ and $p_3$, whereas with adiabatic expansion it would be dry at a higher pressure than $p_3$. The triangular area $F_1NK$ represents the heat additional to $H_\phi$, for a large number of stages. For six stages and upward the value of this heat is not sensibly different from that obtained with an infinite number of stages, so that for the pressure compounded impulse and the reaction turbine, the area below the curve $F_1K$ reckoned from the line QK, may be taken. The point K may fall in the wet saturated region, or in the superheat region.

**384. Graphical Location of the Condition Curve on the HΦ**

**Diagram.**—While the TΦ diagram is serviceable in demonstrating the effect of frictional reheat, it is inconvenient for practical purposes, and the HΦ diagram is employed for the estimation of heat drops and qualities. If this diagram, drawn to a *large* scale, is available, the approximate condition curve can be located on it by a step by step graphical construction, when the stage efficiency ratios are known.

In order to get a reasonably accurate curve, small heat drops are used. Unless, however, the work is done very carefully and on the large scale diagram, the curve may be considerably out at the lower end, especially if the total pressure drop is large. It is preferable in all cases to use the calculative method given in Art. 368. The graphical method of location of the condition curve is shown in Fig. 438, for a portion of a condition curve in the superheat and saturation fields of the HΦ diagram.

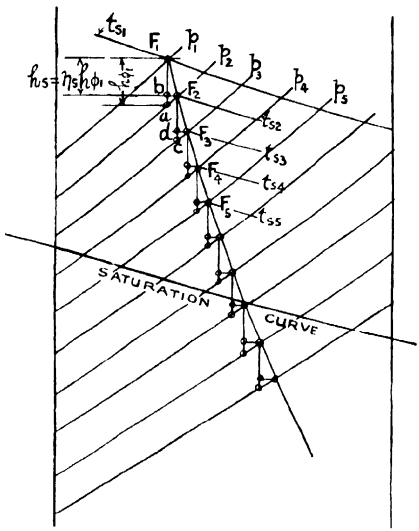



FIG. 438.

Let $h_\phi$ be the heat drop per stage, the stage pressures being chosen, preferably to give equal heat drops; the stage efficiency ratio may be taken as an average value for the range of expansion. The net heat converted to work at the shaft in each stage is thus—

$$h_s = \epsilon_s h_\phi = (1-f)h_\phi \quad . \quad . \quad (2)$$

where $f$ may be termed the stage friction factor. In Fig. 438 draw from the initial state point $F_1$ at $p_1$ and $t_{s1}$, the vertical $F_1a = h_{\phi 1}$, the chosen stage heat drop. The point $a$ falls on the stage exhaust pressure $p_2$. From $F_1$ scale $F_1b = \epsilon_s h_{\phi 1}$ and through $b$ draw $F_2b$, horizontally to cut the $p_2$ curve in $F_2$. This is the initial state point for the second stage. Next draw $F_2c = h_{\phi 2}$, the heat drop in the next stage, and the point $c$ gives the initial pressure $p_3$ for the third stage. Again set off from $F_2$, $F_2d = \epsilon_s h_{\phi 2}$, project horizontally to the $p_3$ curve, and $F_3$ gives the initial state point for the third stage. By a repetition of this process, throughout the complete pressure range of the machine, the series of state points $F_1$, $F_2$, $F_3$, etc., and the qualities $t_{s1}$, $t_{s2}$, $t_{s3}$, etc., are obtained. The condition curve is the locus of these points. A complete curve, traversing the superheat and saturation fields between the initial pressure $p_1$ and the exhaust pressure $p_0$, is shown in Fig. 439.

**385. Internal Efficiency Ratio ($\epsilon_1$) of the Compound Turbine**

**on the Straight Rankine Cycle.**—If in the multi-stage turbine, for which the condition curve $F_1DK$ is shown in Fig. 439, the expansion throughout were adiabatic and frictionless, the final state point at $p_0$ would fall at N. On account of the cumulative frictional reheat it is moved up the $p_0$ curve to K, and the final quality is increased from $q_0'$ to $q_0$. For the ideal case the heat drop is $F_1N=H_\phi$. The actual or net heat drop is $H_i=F_1N_1$, and $NN_1$ the difference of the total heats at N and K is the friction reheat $h_f$ for the whole machine.

The ratio of the net heat transformed to work at the shaft to the heat drop between the initial and exhaust pressures is called the internal efficiency ratio of the turbine.

Thus—

$$\epsilon_1=\frac{F_1N_1}{F_1N}=\frac{H_i}{H_\phi}=\frac{\Sigma h_s}{H_\phi}=\frac{\epsilon_s\Sigma h_\phi}{H_\phi} \quad . \quad (3)$$

or $$\epsilon_1=\frac{\epsilon_s R H_\phi}{H_\phi} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

or $$\epsilon_1=R\epsilon_s \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

Also the cumulative heat is given by—

$$H_c=\frac{H_i}{\epsilon_s} \quad . \quad . \quad . \quad . \quad (6)$$

When the final state point K is located on the $p_0$ curve, either graphically or by calculation, the internal efficiency ratio $\epsilon_1$ and reheat factor R, for the whole machine, are calculable. As a rule the final state point falls in the wet saturation region, the quality being $q_0$, so that the final total heat of the wet steam is—

$$H_{w0}=H_0-(1-q_0)L_0 \quad . \quad . \quad (7)$$

The total net heat drop is—

$$H_i=(H_{s1}-H_{w0}) \quad . \quad . \quad . \quad (8)$$

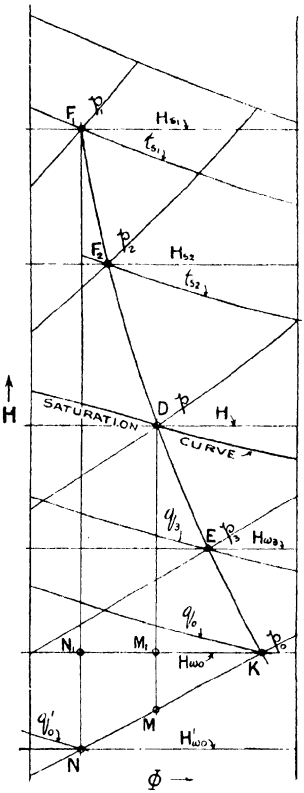



FIG. 439.

where $H_{s1}$ is the total heat of the initially superheated steam. If the final state point falls in the superheat region, the final total heat is $H_{s0}$ and the net heat drop is—

$$H_i=(H_{s1}-H_{s0})$$

If the stage efficiency ratio is assumed to vary in different sections of a turbine, then the calculation has to be carried out in steps, the successive portions of the condition curve being drawn to conform to the altered values of $\epsilon_s$.

Expressed in terms of the initial and final total heats and the adiabatic heat drop, the internal efficiency ratio is given by—

$$\left.\begin{aligned}\epsilon_I&=\frac{H_{s1}-H_{w0}}{H_\phi}\\ \epsilon_I&=\frac{H_{s1}-H_{s0}}{H_\phi}\end{aligned}\right\} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

and the internal thermal efficiency by—

$$\left.\begin{aligned}\eta_I&=\frac{H_{s1}-H_{w0}}{H}\\ \text{or}\qquad \eta_I&=\frac{H_{s1}-H_{s0}}{H}\end{aligned}\right\} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (10)$$

according to whether the final state point falls in the wet or the superheated region. The heat supplied is $H=(H_{s1}-h_0)$.

**386. Location of the Condition Curve by Calculation.**—The portion of the condition curve $F_1F_2C$ lying in the superheat field, Fig. 439, can be expressed by a simple law $pv^n=C$, on the assumption that the superheated steam behaves like a gas during expansion. It can be shown that, with this condition, the index $n$ is a function of the stage efficiency ratio $\epsilon_s$.

It has the form—

$$n=\frac{1}{1-\epsilon_s\left(1-\dfrac{1}{\gamma-1}\right)} \quad . \quad . \quad . \quad . \quad . \quad (11)$$

where $\gamma=1{\cdot}3$ is the adiabatic index of expansion for superheated steam, as deduced by Callendar. On substitution of this value Equation 11 becomes—

$$n=\frac{1}{1-0{\cdot}231\epsilon_s} \quad . \quad . \quad . \quad . \quad . \quad . \quad (12)$$

and the law of expansion can be written—

$$pv^{\frac{1}{1-0{\cdot}231\epsilon_s}}=C \quad . \quad . \quad . \quad . \quad . \quad . \quad (13)$$

Taking the initial volume $v_{s1}$ at $p_1$ and $t_{s1}$, the volume $v_{s2}$ at any chosen lower pressure $p_2$ is calculable from Equation 17, Art. 59; and by Equation 25, Art. 35, the corresponding superheat $t_{s2}$ at $p_2$ between 20 lb./in.$^2$ abs. and 400 lb./in.$^2$ abs., is given by—

$$t_{s2}=\left(\frac{v_{s2}}{v_2}-1\right)\Big/0{\cdot}0015 \quad . \quad . \quad . \quad . \quad . \quad (14)$$

and $t_{s2}$ defines the state point at $p_2$. Several points on the condition curve can thus be located.* The pressure at which the curve crosses

* Equation 25 is not applicable for pressure $p_2$ greater than 400 lb./in.$^2$ abs. See Fig. 15, p. 47. Here $r=\frac{v_{s2}}{v_2}$ and $t_{s2}$ is obtained from the $p_2$ curve.

the saturation curve can, however, be determined from the following expression—

$$p=\left(\frac{333{\cdot}4}{p_1{}^{y}v_{s1}}\right)^{z} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (15)$$

where $y=(1-0{\cdot}231\epsilon_s)$ and $z=\dfrac{1}{(0{\cdot}231\epsilon_s-0{\cdot}0624)}$

It is usually sufficient to determine $p$ and thus the state point D on the saturation curve, then the quality $t_{s2}$ at $F_2$ on some intermediate pressure curve $p_2$, and to draw a fair curve through the three points $F_1$, $F_2$, and D, Fig. 439.

*Example* 1.—The initial pressure of a pressure compounded impulse turbine is 300 lb./in.$^2$ abs. and the superheat range 300° Fahr. (corresponding to a steam temperature of 716° Fahr.). Assuming an average stage efficiency ratio of 0·78, find (*a*) the pressure at which the steam is dry saturated, (*b*) the quality (superheat) at 100 lb./in.$^2$ and draw in the condition curve in the superheat field.

(*a*) $p_1=300$, $t_{s1}=300$. By Equation 25, Art. 35—

$v_{s1}=v_1(1+0{\cdot}0015t_{s1})$ | $v_1=1{\cdot}55$, $\epsilon_s=0{\cdot}78$

$=1{\cdot}55(1+0{\cdot}0015\times300)$

$=1{\cdot}55\times1{\cdot}45=2{\cdot}247$ ft.$^3$/lb.

$y=(1-0{\cdot}231\times0{\cdot}78)=0{\cdot}8198$

$$z=\frac{1}{0{\cdot}231\times0{\cdot}78-0{\cdot}0624}=\frac{1}{0{\cdot}1178}=8{\cdot}509$$

$$p=\left(\frac{333{\cdot}4}{300^{0{\cdot}8198}\times2{\cdot}247}\right)^{8{\cdot}509}=15{\cdot}7 \text{ lb./in.}^2 \text{ abs.}$$

(*b*) $p_2=100$. $n=\dfrac{1}{1-0{\cdot}231\times0{\cdot}78}=\dfrac{1}{0{\cdot}820}=1{\cdot}221$

Then $v_{s2}=v_{s1}\left(\dfrac{p_1}{p_2}\right)^{\frac{1}{n}}=2{\cdot}247\times3^{\frac{1}{1{\cdot}221}}$ $\left(\dfrac{p_1}{p_2}\right)=\dfrac{300}{100}=3$

$=2{\cdot}247\times2{\cdot}476=5{\cdot}51$ ft.$^3$/lb.

then $t_{s2}=\left(\dfrac{v_{s2}}{v_2}-1\right)/0{\cdot}0015=\left(\dfrac{5{\cdot}51}{4{\cdot}428}-1\right)/0{\cdot}0015$ $v_2=4{\cdot}428$

$=\dfrac{0{\cdot}245}{0{\cdot}0015}=\underline{163^\circ \text{ Fahr.}}$

The curve $F_1F_2D$ is shown on Fig. 439, and is continued till the final state point K at the exhaust pressure $p_0$ is reached. The portion DK in the saturation field has to be obtained with the aid of the reheat factor for the expansion between $p$ and $p_0$ and the corresponding internal efficiency ratio for this lower portion of the machine.

Let $H_\phi'$=adiabatic heat drop between $p$ and $p_0$
$H$ =total heat of dry steam at $p$
$h_0$ =total heat of water at $p_0$

$$\text{and} \quad x=\frac{H_\phi'}{2(H-H_\phi'-h_0)}$$

Then the reheat factor for the saturated steam, as deduced by Prof. Callendar, is given by—

$$R'=1+\frac{(1-\epsilon_s)x}{(1+\epsilon_s x)} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (16)$$

Multiply each side by $\epsilon_s$ and reduce, and the internal efficiency ratio of the low-pressure end is—

$$\epsilon_r'=\frac{1+x}{\frac{1}{\epsilon_s}+x} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (17)$$

This is the ratio of $DM_1$ to DM, and the net heat drop in the saturated region is given by—

$$H_i'=\epsilon_r' H_\phi' \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (18)$$

By scaling off $DM_1=H_i'$ from D and projecting horizontally to the $p_0$ curve, the final state point K is obtained.

Other points between D and K on the condition curve can be located by taking several intermediate terminal pressures between $p$ and $p_0$. Usually one point such as E at $p_3$ will be found sufficient. The complete condition curve $F_1DK$ can then be drawn in, the point $N_1$ on the adiabatic vertical $F_1N$ can be located by projection from K; and the internal efficiency ratio for the whole machine is then calculable from the ratio of the intercepts $F_1N_1$ and $F_1N$.

Alternatively the internal efficiency ratio in the saturation region can be obtained from the somewhat simpler expression—

$$\epsilon_r'=\frac{1+a\left(\frac{T-T_0}{T_0}\right)}{\frac{1}{\epsilon_s}+a\left(\frac{T-T_0}{T_0}\right)} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (19)$$

which is seen to be of the same form as Equation 17.

T is the saturation temperature at $p$, and $T_0$ that at $p_0$. An average value of the coefficient $a$ is 0·43.

**387. Reheat Factor and Internal Efficiency and Efficiency Ratio for the Complete Expansion Range.**—The total heat of the wet steam of quality $q_0$ at $p_0$ is—

$$H_{w0}=H-\epsilon_r' H_\phi'$$

and the internal efficiency of the whole machine is—

$$\eta_r=\frac{H_{s1}-H_{w0}}{H} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (20)$$

and the internal efficiency ratio is—

$$\epsilon_1=\frac{H_{s1}-H_{w0}}{H_\phi}=\frac{H_{s1}+\epsilon_1'H_\phi-H}{H_\phi} \quad . \quad . \quad . \quad . \quad (21)$$

The total reheat factor is—

$$R=\frac{\epsilon_I}{\epsilon_s}=\frac{H_{s1}-H_{w0}}{\epsilon_s H_\phi} \quad . \quad . \quad . \quad . \quad . \quad . \quad (22)$$

*Example* 2.—If the expansion, in the case of the turbine of Example 1, is carried down to 0·5 lb./in.$^2$ abs., and the stage efficiency ratio is assumed to remain the same, find the final quality, the internal efficiency and efficiency ratio, and the reheat factor for the machine.

Here $p=15{\cdot}7$ lb./in.$^2$ abs., $\epsilon_s=0{\cdot}78$. By calculation or direct scaling from a large HΦ diagram, the adiabatic heat drop between $p=15{\cdot}7$ and $p_0=0{\cdot}5$ is $H_\phi'=210$; also $H=1152$ and $h_0=47{\cdot}4$.

$$x=\frac{H_\phi'}{2(H-H_\phi'-h_0)}=\frac{210}{2(1152-216-47{\cdot}4)}=\frac{210}{888{\cdot}6}=0{\cdot}1175$$

$$\epsilon_1'=\frac{1+x}{\frac{1}{\epsilon_s}+x}=\frac{1{\cdot}1175}{1{\cdot}282+0{\cdot}1175}=\frac{1{\cdot}1175}{1{\cdot}3995}=0{\cdot}798$$

$$H_i'=\epsilon_1'H_\phi'=0{\cdot}798\times210=167{\cdot}6$$

$$H_{w0}=(H-H_i')=(1152-167{\cdot}6)=984{\cdot}4 \text{ B.Th.U./lb.}$$

This corresponds to $q_0=0{\cdot}897$ at $p_0=0{\cdot}5$.

Internal efficiency ratio for the whole machine is—

$$\epsilon_1=\frac{H_{s1}-H_{w0}}{H_\phi}=\frac{1380-984{\cdot}4}{476}=\frac{395{\cdot}6}{476}=0{\cdot}831$$

From HΦ chart between—
$p_1=300$, $t_{s1}=300$, and $p_0=0{\cdot}5$
$H_{s1}=1380$, and $H_\phi=476$
$h_0=47{\cdot}4$
$H=(H_{s1}-h_0)=1332{\cdot}6$

and the internal thermal efficiency is—

$$\eta_1=\frac{H_{s1}-H_{w0}}{H}=\frac{395{\cdot}6}{1332{\cdot}6}=0{\cdot}297$$

Total reheat factor—

$$R=\frac{\epsilon_I}{\epsilon_s}=\frac{0{\cdot}831}{0{\cdot}78}=1{\cdot}065$$

**388. Net Turbine Efficiency $\eta_B$ and Efficiency Ratio $\epsilon_B$ on the Straight Rankine Cycle.**—In the straight cycle the internal work done per lb. is given by $H_i=\Sigma\epsilon_s h_\phi$. This energy, however, is not all delivered at the driving coupling of the turbine shaft. There are frictional losses at external parts, such as journals, pump gears, governor gears, etc., and a slight amount of carry-over energy is discharged to the condenser. Such "external" losses have no effect on the condition of the steam in the turbine. They can be taken into account by the use of a correction factor analagous to

the mechanical efficiency of the steam engine, or by the deduction of a suitable percentage ($x$) from the gross or internal work value $H_i$, so that in general—

$$H_B = H_i(1-x) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (23)$$

The net or brake thermal efficiency is given by—

$$\eta_B = H_i(1-x) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (24)$$

and when the turbine drives an electric generator the over-all thermal efficiency of the combination is—

$$\eta_0 = \eta_g \eta_B \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (25)$$

where $\eta_g$ is the generator efficiency.

The turbine efficiency ratio is—

$$\epsilon_B = \frac{\eta_B}{\eta} = \frac{H_B}{H_\phi} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (26)$$

and the over-all efficiency ratio is—

$$\epsilon_0 = \eta_g \epsilon_B \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (27)$$

For small machines the external loss ($x$) may run from 4 to 6 per cent., and for large machines from 2 to 4 per cent. In the case of a reaction turbine with dummy pistons where leakage loss takes place the loss may run from 4 to 8 per cent., according to the size.

**389. Effect of Wetness on the Internal Work $H_i$ and Efficiency $\eta_i$.**—In the estimation of the effective work $H_i$ done during the expansion in the saturated region the assumption has been made that for one pound of stuff flowing through the nozzle and blade passages, all the particles of the mass move at any point with the same velocity, that is, the velocity of the entrained moisture is the same as that of the vapour. When the steam is very nearly dry, in passing through a stage, this is approximately the case; but when it is very wet the water particles, which are of considerable magnitude, move at a much lower velocity than the vapour, and at the last stage or two, at high vacuum, this velocity may be extremely low relatively to the steam velocity. The blading at these stages is thus greatly "overspeeded" for the water particles, with the result that the inlet edges of the blades "hit" the slow moving water particles with appreciable force, and the result is "erosion" of the blading at the inlet edges. These slow moving particles perform much less work on the blading than the same mass of higher velocity steam, and the result is a progressive loss of effective work per lb. of steam flowing through the stages in the saturated region. The internal work done in the saturated region, per lb. flowing in the straight Rankine cycle, is in consequence reduced from $H_i'$ to some smaller value $H_{i'w}$, the total loss of effective work due to "wetness" being $h_w = (H_i' - H_{i'w})$. The internal efficiency of the whole machine is reduced to—

$$\eta_{Iw} = \frac{H_{s1} - H_{w0} - h_w}{H} \quad . \quad . \quad . \quad . \quad . \quad . \quad (28)$$

There is some difference of opinion among turbine designers regarding effect of wetness on the internal efficiency or on what is termed the "wetness correction." One assumption made is that at any given stage there is 1 per cent. loss of efficiency for each 1 per cent. mean wetness of the steam in the stage.

On this basis a rough approximation to the value of $h_w$ can be obtained by dividing the cumulative heat for the expansion in the saturation region, that is, between D and K, Fig. 440, into a number of equal heat drops, the drop per stage, $h_\phi$, being chosen to give the nozzle velocity suitable to the type of stage, impulse or reaction, under consideration. With this value of $h_\phi$ the stage pressures can be ascertained from the condition curve DK, and the steam qualities at entrance and exit are then known. The mean quality $q_m$ at each stage is then calculable.

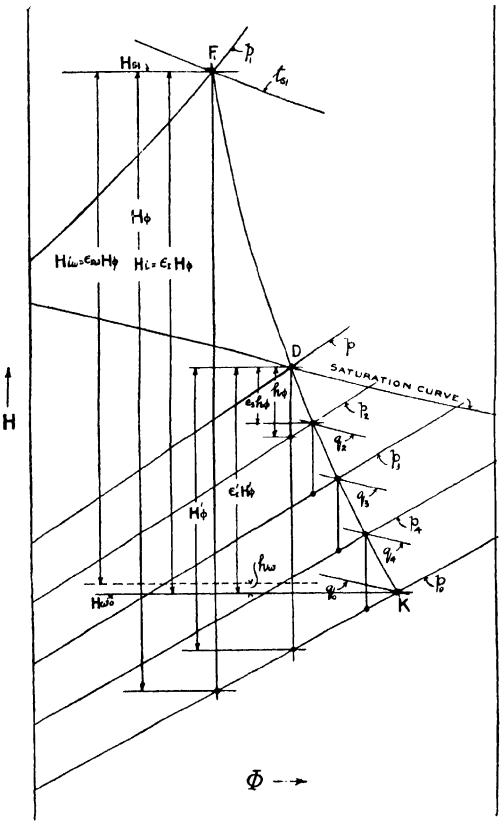



FIG. 440.

If the number of equal stage heat drops is $n$, the approximation to the loss of effective work can be obtained from—

$$h_w = \epsilon_1' H_\phi' \left(1 - \frac{\Sigma q_m}{n}\right) \quad . \quad . \quad . \quad . \quad (29)$$

where $\Sigma q_m = [q_2 + q_3 + q_4 \ldots + \tfrac{1}{2}(1 + q_0)]$ . . . (30)

*Example* 3.—In the case of the impulse turbine on the straight Rankine cycle (Example 2) find the probable loss of effective energy, and the reduced internal efficiency, when a wetness correction is applied.

It may be assumed that the jet velocity is 1590 ft./sec., and the nozzle efficiency $\eta_n = 0{\cdot}92$. The stage efficiency ratio is $\epsilon_s = 0{\cdot}78$.

With the assumed conditions, the cumulative heat being 215 and the stage heat drop 53·5, the number of stage heat drops between $p_1=15{\cdot}7$ and $p_0=0{\cdot}5$ is $n=4$.

Setting off $h_\phi=53{\cdot}5$ from D, Fig. 440, the second pressure $p_2=7{\cdot}3$ and the quality at the point of intersection of the $p_2$ curve and condition curve is $q_2=0{\cdot}971$. Repeating the process the following values are obtained from the points on the condition curve:

| | $p_1$ | $p_2$ | $p_3$ | $p_4$ | $p_0$ |
|---|---|---|---|---|---|
| Pressure lb./ins.$^2$ abs. | 15·7 | 7·8 | 3·3 | 1·4 | 0·5 |
| Quality ($q$) . . . . | 1 | 0·971 | 0·947 | 0·922 | 0·897 |
| Mean quality ($q_m$) . | | 0·99 | 0·96 | 0·934 | 0·91 |

Then $\Sigma q_m=(0{\cdot}99+0{\cdot}96+0{\cdot}934+0{\cdot}91)=3{\cdot}794$; $\dfrac{\Sigma q_m}{n}=\dfrac{3{\cdot}794}{4}=0{\cdot}948$

$$\therefore\ h_w=\epsilon_1'H_\phi'\left(1-\frac{\Sigma h_m}{n}\right) \qquad \epsilon_1'=0{\cdot}798,\ H_\phi'=216$$

$$=0{\cdot}798\times210(1-0{\cdot}948)$$

$$=0{\cdot}798\times210\times0{\cdot}052=8{\cdot}714$$

$H_{s1}=1380$, $h_0=47{\cdot}4$, $H=1332{\cdot}6$

$$\eta_{Iw}=\frac{H_{s1}-H_{w0}-h_w}{H}$$

$$=\frac{1380-984{\cdot}4-8{\cdot}714}{1332{\cdot}6}=\frac{386{\cdot}8}{1332{\cdot}6}=0{\cdot}29$$

$$\eta_I=\frac{395{\cdot}6}{1332{\cdot}6}=0{\cdot}297, \text{ without wetness correction.}$$

The internal efficiency is apparently reduced 2·30 per cent. by the wetness effect.

**390.** Until within recent years, with the normal conditions of 200 lb./in.$^2$ and 700° Fahr., the wetness correction was not a serious item, and the erosion trouble was slight.

The considerable increase of initial pressure without a largely increased temperature has, however, increased the loss of energy, due to excessive wetness in the L.P. stages, to a very appreciable amount, and the erosion accompanying this has become a serious operative trouble. As a conservative estimate the final wetness at the last L.P. stage should not exceed 12 per cent. With an increased temperature of 750° Fahr. the upper pressure limit may be taken as 400 lb./in.$^2$ If it is possible to obtain steam at 850° Fahr., this limit may be raised to 700 lb./in.$^2$ Otherwise, if the wetness condition is to be avoided, reheat after partial expansion must be adopted.

With 750° Fahr. one reheat may serve at an initial pressure of 1400 lb./in.$^2$, and with 900° Fahr. and 2000 lb./in.$^2$ two reheats may be adopted. The student can readily examine the wetness effects for given conditions by drawing condition curves for high initial pressures and the upper temperature limits indicated here.

As regards the mitigation of the erosion effect, which has increased considerably in large turbines running at high blade speed in the L.P. sections, and in which special alloy steels blades have to be used on account of high stresses, chromium plating of the blades has been tried.* Blade wear has been much reduced thereby, but not prevented. The Parsons firm now protect the leading edges of reaction blading with a shield of specially resistant metal.† Partial drainage is also attempted ahead of the last low-pressure turbine stage, where the most serious erosion always occurs.

**391. Internal Thermal Efficiency $\eta_I$ and Efficiency Ratio $\epsilon_I$ of the Compound Turbine on the Reheat Cycle.**—In the actual case the internal thermal efficiency is given by an expression similar to (30) of Art. 61, for the absolute thermal efficiency, in which the net heat drops and the actual reheat are substituted for the ideal values.

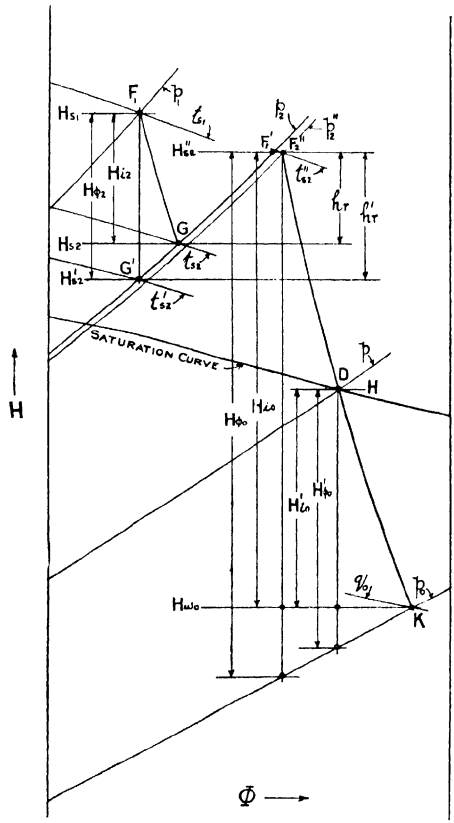



FIG. 441.

Referring to Fig. 441, let $\epsilon_{12}$ be the internal efficiency ratio for the first expansion range between the initial pressure $p_1$ and reheat pressure $p_2$. The adiabatic heat drop is $H_{\phi 2}$, and the net heat available is $H_{i2}=\epsilon_{12}H_{\phi 2}$.

The condition at G, after expansion to $p_2$, is $t_{s2}$, and the total heat is $H_{s2}$. The steam is returned to the boiler and reheated, and passed back to the next turbine section; but in this process there is always a drop of pressure in the heater, piping, and valves of about 7 per cent., and the initial pressure of the steam, re-entering the turbine at the reheat temperature $T_2$, is some lower value, say $p_2''=0{\cdot}93p_2$; the corresponding superheat range at the initial state point $F_2''$ will be $t_{s2}''$, and the total heat $H_{s2}''$. If the

* See "Steam Turbine Blade Wear," by J. L. Ray, *Power*, 26 May, 1931.

† See "Post-War Land Turbine Development," by C. D. Gibb, B.Eng., *Proc. Inst. Mech. Eng.*, April 1931.

internal efficiency ratio for the expansion between $p_2''$ and the final exhaust pressure $p_0$ is $\epsilon_{I0}$, and the adiabatic heat drop is $H_{\phi 0}$, the net heat available for work is $H_{i0}=\epsilon_{I0}H_{\phi 0}$. If $h_0$ is the total heat of the condensate at $p_0$, from the condenser, then the heat supplied from the external source, the boiler, is—

$$H=H_{s1}+h_r-h_0 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (31)$$

where—

$$h_r=(H_{s2}''-H_{s2}) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (32)$$

is the actual reheat. The net internal heat available for work is—

$$H_i=H_{i2}+H_{i0} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (33)$$

Hence the internal thermal efficiency is—

$$\eta_I=\frac{H_i}{H}=\frac{H_{i2}+H_{i0}}{H_{s1}+h_r-h_0} \quad . \quad . \quad . \quad . \quad . \quad (34)$$

In the ideal case the lower and upper state points at the reheat pressure $p_2$ are G′ and $F_2'$, and the ideal reheat is—

$$h_r'=(H_{s2}''-H_{s'2}) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (35)$$

the assumption being made that the total heat value is not altered.

The total adiabatic heat drop is—

$$H_\phi=H_{\phi 2}+H_{\phi 0} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (36)$$

and the ideal heat supply is—

$$H'=(H_{s1}+h_r'-h_0) \quad . \quad . \quad . \quad . \quad . \quad (37)$$

Hence the absolute thermal efficiency is—

$$\eta=\frac{H_\phi}{H'}=\frac{H_{\phi 2}+H_{\phi 0}}{H_{s1}+h_r'-h_0} \quad . \quad . \quad . \quad . \quad . \quad (38)$$

It will be obvious that as the denominators of the fractions are not the same, the internal efficiency ratio is not given, as in the straight Rankine cycle, by the ratio of the net to the adiabatic heat drop or $\left(\frac{H_i}{H_\phi}\right)$, and the ratio of the actual and absolute thermal efficiencies must be taken.

$$\epsilon_1=\frac{\eta_I}{\eta}=\frac{(H_{i2}+H_{i0})(H_{s1}+h_r'-h_0)}{(H_{\phi 2}+H_{\phi 0})(H_{s1}+h_r-h_0)} \quad . \quad . \quad . \quad (39)$$

*Example* 4.—A large reaction turbine is supplied with steam at 500 lb./in.$^2$ abs. and 700° Fahr., and exhausts at 0·5 lb./in.$^2$ abs. There is one boiler reheat at 150 lb./in.$^2$ abs. to 600° Fahr. Calculate the internal thermal efficiency and efficiency ratio, the turbine efficiency ratio, and the rates of heat consumption, assuming the following data. Stage efficiency ratio for expansion, between initial and reheat pressures, $\epsilon_{s1}=0{\cdot}8$; between reheat pressure and saturation condition, $\epsilon_{s2}=0{\cdot}82$; expansion in saturation region, $\epsilon_{s0}=0{\cdot}84$. Reheat pressure drop 7 per cent.; external frictional losses, 3 per cent. of the net internal work.

In Fig. 441, $p_1=500$, $t_{s1}=234$, $H_{s1}=1361{\cdot}69$, and $v_{s1}=1{\cdot}313$. After adiabatic expansion to $p_2=150$, $t_s'_2=60°$ Fahr. and $H_s'_2=1233{\cdot}8$. Adiabatic heat drop $H_{\phi 2}=(1361{\cdot}69-1233{\cdot}8)=127{\cdot}89$ B.Th.U./lb. To determine the actual condition at $p_2=150$—

$$v_{s2}=\left(\frac{p_1}{p_2}\right)^{\frac{1}{n}}v_{s1} \qquad n=\frac{1}{1-0{\cdot}231\epsilon_{s1}}$$

$$=\left(\frac{500}{150}\right)^{\frac{1}{1{\cdot}227}}\times 1{\cdot}313=3{\cdot}505 \qquad =\frac{1}{1-0{\cdot}231\times 0{\cdot}8}=1{\cdot}227$$

Quality at G— $\qquad v_2=3{\cdot}014$

$$t_{s2}=\left(\frac{3{\cdot}505}{3{\cdot}014}-1\right)/0{\cdot}0015$$

$$=\frac{0{\cdot}162}{0{\cdot}0015}=108 \text{ Fahr., and } H_{s2}=1259$$

Allowing 7 per cent. drop, $p_2''=0{\cdot}93\ p_2=0{\cdot}93\times 150=140$ lb./in.$^2$ abs., $T_2=600$, $t_2''=353$. Superheat after reheating, $t_s''_2=(600-353)=247°$ Fahr., and total heat $H_{s2}''=1330$.

Internal heat $H_{i2}=(H_{s1}-H_{s2})$

$=(1\ 61{\cdot}69-1259)=102{\cdot}69$ B.Th.U./lb.

Actual reheat, $h_r=(H_{s2}''-H_{s2})=(1330-1259)=71$ B.Th.U./lb.

Ideal reheat $h_r'=(H_{s2}''-H_s'_2)=(1330-1233{\cdot}8)=96{\cdot}2$ B.Th.U./lb.

To find pressure $p$ at which the steam becomes dry, with $\epsilon_{s2}=0{\cdot}82$, $n=1{\cdot}234$, $z=7{\cdot}884$, and $y=0{\cdot}8106$.

At $p_2''=140$ and $t_s''_2=247$, $v_s''_2=4{\cdot}41$

$$p=\left(\frac{333{\cdot}4}{p_2''^{0{\cdot}8106}\times v_s''_2}\right)=\left(\frac{333{\cdot}4}{140^{0{\cdot}8106}\times 4{\cdot}41}\right)^{7{\cdot}884}=12{\cdot}5 \text{ lb./in.}^2 \text{ abs.}$$

In the saturation region $\epsilon_{s0}=0{\cdot}84$, and—

$$\epsilon_1'_0=\frac{1+0{\cdot}43\frac{T-T_0}{T_0}}{\frac{1}{\epsilon_{s0}}+0{\cdot}43\frac{T-T_0}{T_0}}=\frac{1+\frac{0{\cdot}43\times 124}{540}}{\frac{1}{0{\cdot}84}+\frac{0{\cdot}43\times 124}{540}} \qquad \begin{matrix} T=664 \\ T_0=540 \end{matrix}$$

$$=\frac{1{\cdot}0987}{1{\cdot}2887}=0{\cdot}852$$

Adiabatic heat drop from $p=12{\cdot}5$ to $p_0=0{\cdot}5$ is $H_\phi'_0=196$.

$\therefore\ H_i'_0=\epsilon_1'_0 H_\phi'_0=0{\cdot}852\times 196=167$ B.Th.U./lb.

At $p=12{\cdot}5$, $H=1147$ $\therefore\ H_{w0}=(H-H_i'_0)=(1147-167)=980$ B.Th.U./lb.

Then $H_{i0}=(H_s''_2-H_{w0})=(1330-980)=350$ B.Th.U./lb.

The adiabatic heat drop from $p_2''=140$ to $p_0=0{\cdot}5$ is $H_{\phi 0}=402$, also, $h_0=47{\cdot}4$.

Hence the net internal heat is—

$$H_i=(H_{i2}+H_{i0})=(102{\cdot}69+350)=452{\cdot}69 \text{ B.Th.U./lb.}$$

and $$\eta_1=\frac{H_i}{H}=\frac{H_{i2}+H_{i0}}{H_{s1}+h_r-h_0}=\frac{452{\cdot}69}{1361{\cdot}69+71-47{\cdot}4}=\frac{452{\cdot}69}{1385{\cdot}29}=0{\cdot}326$$

For the absolute thermal efficiency—

$$\eta=\frac{H_{\phi 2}+H_{\phi 0}}{H_{s1}+h_r'-h_0}=\frac{127{\cdot}89+402}{1361{\cdot}69+96{\cdot}2-47{\cdot}4}=\frac{529{\cdot}9}{1410{\cdot}49}=0{\cdot}375$$

$\therefore$ Internal efficiency ratio—

$$\epsilon_I=\frac{\eta_I}{\eta}=\frac{0{\cdot}326}{0{\cdot}375}=0{\cdot}87$$

On the assumption of a 3 per cent. loss of energy on the net work, due to external resistances, the net turbine efficiency is—

$$\eta_B=0{\cdot}97\eta_I=0{\cdot}97\times 326=0{\cdot}316$$

and the turbine efficiency ratio is—

$$\epsilon_B=\frac{\eta_B}{\eta}=\frac{0{\cdot}316}{0{\cdot}375}=0{\cdot}842$$

The rate of heat consumption at the turbine shaft is—

$$Q_B=\frac{3412}{\eta_B}=\frac{3412}{0{\cdot}316}=10{,}797 \text{ B.Th.U./shaft kw.-hr.}$$

The theoretical rate is—

$$Q_\phi=\frac{3412}{\eta}=\frac{3412}{0{\cdot}375}=9100 \text{ B.Th.U./shaft kw.-hr.}$$

The efficiency value is on the high side as wetness effect has been neglected. The correction can be made if desired by the method of Art. 389. The chosen stage efficiency ratio may also be on the high side, so that the real heat consumption on a test would probably be greater than that deduced.

**392. Internal and Net Thermal Efficiency $\eta_I$ and $\eta_B$ of the Compound Turbine on the Partial Regenerative Cycle.**—In the ideal case considered in Art. 64, p. 93, the assumption was made that at any heater the pressure was the same as the turbine stage pressure, and the exit temperature of the feed was the same as the saturation temperature of the steam at the stage pressure. Neither of these conditions hold in the actual case. There is more or less of a drop in pressure between the stage and the heater, depending on the nature of the arrangement. This may run from 5 to 10 per cent. For transmission of heat from steam to water in the heater there has to be a temperature gradient across the coils. This may be taken as 10° Fahr., so that if the steam abstracted at any stage has pressure $p$, and in the heater a lower pressure $p_h$ and saturation temperature $t_h$, the probable temperature of the feed, on discharge from the heater, may be taken as $t_h'=(t_h-10)$, a subscript number being added to denote the particular heater considered.

If radiation effects are neglected, the total heat of the steam in the heater at $p_h$ may be taken sensibly the same as that at the extraction pressure $p$, that is, the total heat as shown by the condition curve at $p$.

In practice the extraction pressures are usually chosen to give practically an equal amount of extraction at each heater.

In dealing with this case the condition curve $F_1DK$, Fig. 442, for 1 lb. of steam can be drawn on the H$\Phi$ field as for a straight Rankine cycle, in accordance with the assumed stage efficiency ratio ($\epsilon_s$) values.

Considering the last or high-pressure heater of a series—

Let $p_2$=stage pressure for the first extraction.

$H_{w2}$ (or $H_{s2}$)=the total heat at $p_2$ (from the condition curve).

$p_{h2}$ =reduced pressure in the heater.

$h_{h2}$ =total heat of the heater condensate at $t_{h2}$.

Then $t_h'_2=(t_{h2}-10)$=feed temperature on discharge to the boiler, and $h_h'_2$=total heat of the feed water at exit.

If the corresponding values for the second last heater are $p_3$, $H_{w3}$, $p_{h3}$, $h_{h3}$, $t_{h3}$, and $h_h'_3$, then the fractional condensation (reckoned per lb. of gross feed entering the HP. end of the turbine) is—

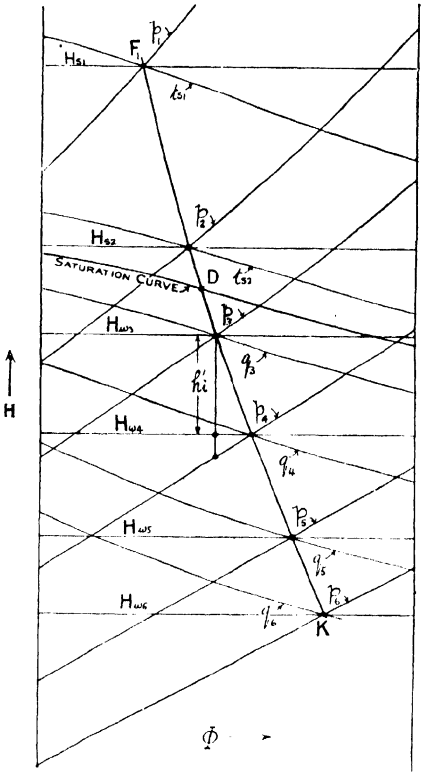



FIG. 442.

$$z_2=\frac{h_h'_2-h_h'_3}{(H_{w2}-h_{h2})} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (40)$$

Denoting the conditions for the next lower heater by the subscript 4, the fractional condensation in the second last heater is given by—

$$z_3=\frac{(h_h'_3-h_h'_4)-z_2(h_{h2}-h_{h3})}{(H_{w3}-h_{h3})} \quad . \quad . \quad . \quad . \quad (41)$$

At the next heater the value will be—

$$z_4=\frac{(h_h'_4-h_h'_5)-(z_2+z_3)(h_{h3}-h_{h4})}{(H_{w4}-h_{h4})} \quad . \quad . \quad . \quad (42)$$

and so on for any other heaters, on the assumption that the condensates are progressively cascaded through the heaters. Taking the general arrangement of four regenerative heaters considered in

the theoretical discussion in Art. 64, let the fractional condensates again be denoted by $z_2$, $z_3$, $z_4$, and $z_5$.

In case (*a*), where the drains are cascaded to the first or low-pressure heater and then discharged to the condenser, the fractional condensate of the L.P. heater is given by—

$$z_5=\frac{(h_h'_5-h_6)-(z_2+z_3+z_4)(h_{h4}-h_{h5})}{(H_{w5}-h_{h5})} \quad . \quad . \quad . \quad (43)$$

In case (*d*), where the drains are cascaded to the second heater and returned to the feed line, while the condensate of the L.P. heater only is discharged to the condenser—

$$z_5=\frac{(h_h'_5-h_6)}{(H_{w5}-h_{h5})} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (44)$$

In case (*a*) the internal net heat converted to work on the turbine shaft is—

$$H_i=(H_{s1}-h_h'_2)-[(1-\Sigma z)(H_{w6}-h_6)+\Sigma z(h_{h5}-h_6)] \quad . \quad (45)$$

and in case (*d*)—

$$H_i=(H_{s1}-h_h'_2)-[(1-\Sigma z)(H_{w6}-h_6)+z_5(h_{h5}-h_6)] \quad . \quad (46)$$

The heat supplied per lb. of gross feed entering the boiler is—

$$H=(H_{s1}-h_h'_2) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (47)$$

In order to obtain an approximate correction of the $H_i$ value for wetness effect the following method may be applied. Scale from the H$\Phi$ diagram (Fig. 442) the net heat value $h_i'$ between any two heater pressures, and note the mean dryness $q_m$ as shown by the condition curve, then the probable amount of heat rendered unavailable through wetness in this section of the turbine will be—

$$h_w'=h_i'(1-\Sigma z)(1-q_m) \quad . \quad . \quad . \quad . \quad . \quad (48)$$

and the total defect is—

$$h_w=\Sigma h_w' \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (49)$$

Then the approximate value of the internal work in heat units is given by—

$$H_{iw}=(H_i-h_w) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (50)$$

As there are various pumps taking energy developed by the turbine, a higher value of outside loss should be taken, say 5 per cent. of the internal work for a moderate or large machine. If $\eta_g$ is the generator efficiency, then the approximate value of the net work or output at the generator terminals will be approximately—

$$H_0=0{\cdot}95H_{iw}\eta_g \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (51)$$

This is the value in heat units per lb. of gross supply at the H.P. end of the turbine. The output at the turbine shaft will be—

$$H_B=0{\cdot}95H_{iw} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (52)$$

The net thermal efficiency of the turbine alone is given by—

$$\eta_B = \frac{H_B}{H} = \frac{0{\cdot}95H_{iw}}{(H_{s1} - h'_2)} \quad . \quad . \quad . \quad . \quad . \quad (53)$$

and the value for the combination turbine and generator by—

$$\eta_0 = \frac{H_0}{H} = \frac{0{\cdot}95H_{iw}\eta_g}{(H_{s1} - h'_2)} \quad . \quad . \quad . \quad . \quad . \quad (54)$$

If it is desired to obtain the turbine efficiency ratio and the over-all efficiency ratio, the absolute thermal efficiency ($\eta$) must be calculated for the ideal case in accordance with the methods of Art. 64, and then—

$$\epsilon_B = \frac{\eta_B}{\eta} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (55)$$

and $$\epsilon_0 = \frac{\eta_0}{\eta} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (56)$$

This figure of merit has now ceased to have much significance, as far as general comparison of turbine performance is concerned. There are differences of opinion as to what should be taken as the theoretical standard condition when a reheat or regenerative or a reheat-regenerative cycle is adopted.* It is becoming the usual practice to compare performance in terms of the rate of heat consumption, which varies inversely as the thermal efficiency—

Thus—

$$Q_B = \frac{3412}{\eta_B} \text{ B.Th.U./shaft kw.-hr.} \quad . \quad . \quad . \quad (57)$$

and $$Q_0 = \frac{3412}{\eta_0} \text{ B.Th.U./kw.-hr.} \quad . \quad . \quad . \quad . \quad . \quad . \quad (58)$$

The latter is the figure usually quoted for turbine performance as " from steam to electricity," to distinguish it from the over-all heat rate of the power station, which includes the effect of boiler plant and auxiliaries.

The following example should serve to emphasise the various points of this discussion.

*Example* 5.—A reaction steam turbine running on the partial regenerative cycle is supplied with steam at 600 lb./in.$^2$ abs. and 700° Fahr., and it exhausts at 0·5 lb./in.$^2$ abs. There are four extraction feed heaters, and the extraction pressures are 110, 38, 10, and 2 lb./in.$^2$ abs. Assuming a heater pressure drop of 8 per cent., a mean stage efficiency ratio of 0·8 throughout, calculate the probable internal work per lb. of gross feed, the internal thermal efficiency, and the rate of heat consumption for both the turbine and turbine and generator. Assume that the heater drains are progressively cascaded, and pumped back to the feed line, but that

* See " Comparison of Steam Turbines," *Engineer*, 5 June, 1931.

the last heater condensate is discharged to the condenser. Take $\eta_g=0{\cdot}97$.

This is the case (*d*) for which the absolute thermal efficiency was worked out in Example 10, Art. 64. Referring to Fig. 442, the condition curve $F_1DK$ is obtained by the methods previously illustrated. It cuts the saturation curve at $p=68$; and at $p_0=p_6=0{\cdot}5$ the quality is $q_6=0{\cdot}847$.

The total heat values, read from the condition curve at the extraction pressures, the chosen reduced heater pressures, and water heats are given in the table below—

| | | | | | | |
|---|---|---|---|---|---|---|
| $p_2=110$ | $p_{h2}=101$ | $t_{h2}=328$ | $t_{h'2}=318$ | $h_{h2}=300$ | $h_{h'2}=288$ | $H_{s2}=1216$ |
| $p_3=38$ | $p_{h3}=36$ | $t_{h3}=261$ | $t_{h'3}=251$ | $h_{h3}=229{\cdot}5$ | $h_{h'3}=220$ | $H_{w3}=1148$ |
| $p_4=10$ | $p_{h4}=9{\cdot}2$ | $t_{h4}=189$ | $t_{h'4}=179$ | $h_{h4}=158$ | $h_{h'4}=147$ | $H_{w4}=1071$ |
| $p_5=2$ | $p_{h5}=1{\cdot}84$ | $t_{h5}=123$ | $t_{h'5}=113$ | $h_{h5}=91$ | $h_{h'5}=81$ | $H_{w5}=992$ |
| $p_6=0{\cdot}5$ | | | | $h_8=47{\cdot}4$ | | $H_{w6}=932$ |

Heater condensates—

$$z_2=\frac{(h_{h'2}-h_{h'3})}{(H_{s2}-h_{h2})}=\frac{288-220}{1216-300}=\frac{68}{916}=0{\cdot}0742$$

$$z_3=\frac{(h_{h'3}-h_{h'4})-z_2(h_{h2}-h_{h3})}{(H_{w3}-h_{h3})}=\frac{(220-147)-0{\cdot}0742(300-229{\cdot}5)}{1148-229{\cdot}5}$$

$$=\frac{73-5{\cdot}23}{918{\cdot}5}=\frac{67{\cdot}77}{918{\cdot}5}=0{\cdot}0737$$

$$z_4=\frac{(h_{h'4}-h_{h'5})-(z_2+z_3)(h_{h3}-h_{h4})}{H_{w4}-h_{h4}}$$

$$=\frac{(147-81)-0{\cdot}1479(229{\cdot}5-158)}{1071-158}=\frac{66-10{\cdot}57}{913}=\frac{55{\cdot}43}{913}=0{\cdot}0607$$

$$z_5=\frac{(h_{h'5}-h_{h6})}{(H_{w5}-h_{h5})}=\frac{(81-47{\cdot}4)}{(992-91)}=\frac{33{\cdot}6}{901}=0{\cdot}0372$$

Then $\Sigma z=(0{\cdot}0742+0{\cdot}0737+0{\cdot}0607+0{\cdot}0372)=0{\cdot}2458$.

Then internal work uncorrected for wetness is—

$$H_i=(H_{s1}-h_{h'2})-[(1-\Sigma z)(H_{w6}-h_6)+z_5(h_{h5}-h_6)]$$
$$=(1355-288)-[(1-0{\cdot}2458)(932-47{\cdot}4)+0{\cdot}0372(91-47{\cdot}4)]$$
$$=(1067-667{\cdot}2)=399{\cdot}8 \text{ B.Th.U./lb.}$$

$H_{s1}=1355$
$H_{w6}=932$
$h_6=47{\cdot}4$
$h_{h'2}=288$
$h_{h5}=91$

With about 15 per cent. final wetness, the wetness correction will be appreciable.

It is obviously unnecessary in this case to consider any extension in the saturation field above the 38 lb./in.$^2$ pressure curve. The following are the net heat (external work) values $h_i'$ scaled from the

chart between 38 and 10, 10 and 2, 2 and 0·5, with the value of $(1-\Sigma z)$ and $(1-q_m)$ in each case—

| *Pressure range* | $h_i'$ | $(1-\Sigma z)$ | $(1-q_m)$ | $h_w'$ |
|---|---|---|---|---|
| 38–10 . . . . . | 77 | (1–0·1479) = 0·8521 | 0·05 | 3·28 |
| 10–2 . . . . . | 79 | (1–0·2086) = 0·7914 | 0·10 | 6.25 |
| 2–0·5 . . . . . | 60 | (1–0·2458) = 0·7542 | 0·14 | 6·33 |
| | | | | $\Sigma h_w' = 15{\cdot}86$ |

This is 3·97 per cent. of the net internal work.

Taking $H_{iw}=(399{\cdot}8-15{\cdot}86)=384$, and $H=(H_{s1}-h_h'_2)=(1355-288)=1067$,

then
$$\eta_I=\frac{H_{iw}}{H}=\frac{384}{1067}=0{\cdot}3598$$

With shaft output at the turbine, $H_B=0{\cdot}95H_{iw}=0{\cdot}95\times 384=364{\cdot}8$, the net thermal efficiency is—

$$\eta_B=\frac{H_B}{H}=\frac{364{\cdot}8}{1067}=0{\cdot}342$$

The over-all thermal efficiency is—

$$\eta_0=\frac{H_0}{H}=\frac{0{\cdot}95H_{iw}\eta_{if}}{H}=\frac{364{\cdot}8\times 0{\cdot}97}{1067}=0{\cdot}332$$

The rates of heat consumption are—

$$Q_B=\frac{3412}{\eta_B}=\frac{3412}{0{\cdot}342}=9{,}976 \text{ B.Th.U./shaft kw.-hr.}$$

$$Q_0=\frac{3412}{\eta_0}=\frac{3412}{0{\cdot}332}=10{,}277 \text{ B.Th.U./kw.-hr.,}$$

steam to electricity. This is about the minimum value obtained so far from large cross compound turbines. If the condition (*a*) of cascade to the L.P. heater and discharge to the condenser is taken, the efficiencies will be reduced and the rates of heat consumption will be increased. This alternative case is left as an exercise for the student. In Example 10 the absolute thermal efficiency case (*d*) was found to be $z=0{\cdot}442$, so that the over-all efficiency ratio, on the theoretical basis chosen, is—

$$\epsilon_0=\frac{\eta_0}{\eta}=\frac{0{\cdot}332}{0{\cdot}442}=0{\cdot}751$$

**393. Internal and Net Thermal Efficiencies $\eta_I$ and $\eta_B$ of the Compound Turbine on the Reheat-regenerative Cycle.**—It will be obvious that this case can be treated by a combination of the methods of Arts. 391 and 392. The HΦ diagram for a case with one reheat and three regenerative heaters is shown in Fig. 443. As in the theoretical discussion, Art. 65, it is assumed here that the first extraction takes place at the reheat pressure $p_2$. The condition curve between initial and reheat pressures is $F_1G$, the point G defining the total heat $H_{s2}$. The initial state point for the second part of the expansion is $F_2''$, on the reduced pressure curve $p_2''$, and the condition curve is $F_2''DK$.

The reheat per lb. is $h_r=(H_{s2}''-H_{s2})$, but since the fraction $z_2$ is abstracted at G before reheat, the actual amount of heat supplied is $(1-z_2)h_r$. As in the previous case of regenerative heating, the water heat of the feed entering the boiler is $h_h'_2$. Hence the heat supplied per lb. of steam entering at the H.P. end is—

$$H=H_{s1}+(1-z_2)h_r-h_h'_2 \quad . \quad . \quad . \quad . \quad . \quad (59)$$

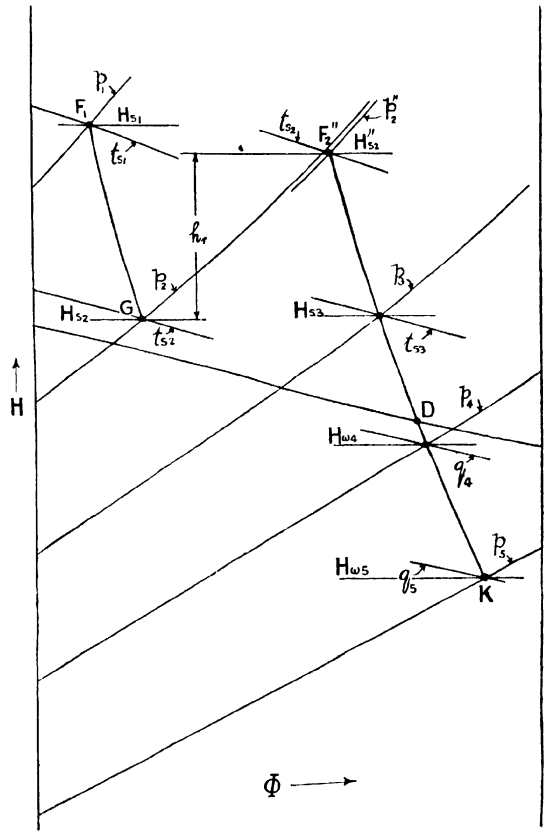



FIG. 443.

The fractional condensate is given by—

$$z_2=\frac{h_h'_2-h_h'_3}{H_{s2}-h_{h2}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (60)$$

and assuming progressive cascade through the heaters and discharge to the condenser as in case (*a*)—

$$z_3=\frac{(h_h'_3-h_h'_4)-z_2(h_{h2}-h_{h3})}{H_{s3}-h_{h3}} \quad . \quad . \quad . \quad . \quad . \quad (61)$$

$$z_4=\frac{(h_h'_4-h_5)-(z_2+z_3)(h_{h3}-h_{h4})}{H_{w4}-h_{h4}} \quad . \quad . \quad . \quad . \quad (62)$$

In this case the heat rejected is—

$$h=(1-\Sigma z)(H_{w5}-h_5)+\Sigma z(h_{h4}-h_5) \quad . \quad . \quad . \quad (63)$$

In the case (*d*) with discharge of the L.P. heater condensate only to the condenser—

$$z_4=\frac{h_h'_4-h_5}{H_{w4}-h_{h4}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (64)$$

and the heat rejected per lb. of gross feed is—

$$h=(1-\Sigma z)(H_{w5}-h_5)+z_5(h_{h4}-h_5) \quad . \quad . \quad . \quad (65)$$

In either case the net or internal work done is—

$$H_i=(H-h) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (66)$$

reckoned per lb. of steam entering the H.P. turbine. This can be corrected for wetness, if necessary, and the probable net output $H_B$ at the turbine coupling can be obtained as in the previous cases.

In any case where the total amount of steam from the H.P. section is reheated and returned, the actual reheat is to be taken as $h_r$ and not $h_r(1-z_2)$.

*Example* 6.—A combination of three turbines in series having one reheat (at the exhaust pressure of the first turbine) and three regenerative heaters is supplied with steam at 600 lb./in.$^2$ abs. and 700° Fahr., and exhausts at 0·5 lb./in.$^2$ abs. The reheat takes place at 100 lb./in.$^2$ abs. to 600° Fahr.

The stage efficiency ratio may be taken as 0·8. The extraction pressures are 100, 20, and 4 lb./in.$^2$ abs. Calculate the internal work done, the thermal efficiency, and the probable rate of heat consumption with a generator efficiency of 0·97. Assume that the heater condensates are cascaded to the L.P. heater and finally discharged to the condenser.

This is the case shown on the HΦ diagram, Fig. 443. Here $p_1=600$, $t_{s1}=214°$ Fahr., $H_{s1}=1355$, $p_2=100$. When the first condition curve $F_1G$ is drawn with $\epsilon_s=0{\cdot}8$, the total heat at G is $H_{s2}=1210$. Allowing, say, 6 per cent. drop of pressure, $p_2''=94$. With reheat to 600° Fahr. at 94 lb./in.$^2$, $t_{s2}''=277$, defining the point $F_2''$, and $H_{s2}''=1333$.

Then the reheat per lb. is—

$$h_r=(H_{s2}''-H_{s2})=(1333-1210)=123.$$

The heater condensates are then calculated from the heat values shown by the condition curve $F_2''DK$, which with $\epsilon_s=0{\cdot}8$, cuts the saturation curve at 5·6 lb./in.$^2$ abs., and the $p_5=0{\cdot}5$ curve at $q_5=0{\cdot}923$. Assuming a pressure drop of 8 per cent. in each heater, the heat values are as tabulated below—

| | | | | | | |
|---|---|---|---|---|---|---|
| $p_2=100$ | $p_{h2}=92$ | $t_{h2}=322$ | $t_h'_2=312$ | $h_{h2}=292$ | $h_h'_2=282$ | $H_{s2}=1210$ |
| $p_3=20$ | $p_{h3}=18{\cdot}4$ | $t_{h3}=223$ | $t_h'_3=213$ | $h_{h3}=192$ | $h_h'_3=181$ | $H_{s3}=1212$ |
| $p_4=4$ | $p_{h4}=3{\cdot}68$ | $t_{h4}=149$ | $t_h'_4=139$ | $h_{h4}=97$ | $h_h'_4=107$ | $H_{w4}=1130$ |
| $p_5=0{\cdot}5$ | | | | $h_5=47{\cdot}4$ | | $H_{w5}=1012$ |

The values of the fractional condensates calculated from those figures are $z_2=0{\cdot}11$ ; $z_3=0{\cdot}0617$ ; $z_4=0{\cdot}0420$.

Hence $\Sigma z=(0{\cdot}11+0{\cdot}0617+0{\cdot}042)=0{\cdot}2137$

Heat supplied—

$$\begin{aligned}H&=H_{s1}+h_r(1-z_2)-h_h'_2\\&=1355+123(1-0{\cdot}11)-282\\&=1355+109{\cdot}4-282=1182{\cdot}4 \text{ B.Th.U./lb.}\end{aligned}$$

The heat rejected is—

$$\begin{aligned}h&=(1-\Sigma z)(H_{w5}-h_5)+\Sigma z(h_{h4}-h_5)\\&=(1-0{\cdot}2137)(1012-47{\cdot}4)+0{\cdot}2137(97-47{\cdot}4)\\&=0{\cdot}7863\times 964{\cdot}6+0{\cdot}2137\times 49{\cdot}6\\&=788{\cdot}465+10{\cdot}6=799{\cdot}065\end{aligned}$$

Hence the internal work done is—

$$H_i=(H-h)=(1182{\cdot}4-799{\cdot}065)=383{\cdot}335 \text{ B.Th.U./lb.}$$

No wetness correction is necessary in this case.

The internal thermal efficiency is—

$$\eta_I=1-\frac{h}{H}=1-\frac{799{\cdot}065}{1182{\cdot}4}=(1-0{\cdot}675)=0{\cdot}325.$$

Allowing, say, 4 per cent. external loss, and $\eta_g=0{\cdot}97$

$$\eta_B=0{\cdot}96\times 0{\cdot}325=0{\cdot}312$$

and $$\eta_0=0{\cdot}97\times 0{\cdot}312=0{\cdot}3026$$

and the probable over-all rate of heat consumption is—

$$Q_0=\frac{3412}{0{\cdot}3026}=11{,}207 \text{ B.Th.U./kw.-hr.}$$

In the calculation of the heat quantities with regenerative feed heaters it has been assumed that the condensate from the main condenser enters the L.P. heater at the vacuum temperature. In practice the vacuum is usually produced by steam ejectors, independently supplied with steam, and this steam is used to raise the feed temperature before it enters the L.P. heater. It is not possible to give any general calculation to take account of this preliminary heating for any particular case. The effect is to reduce the fractional condensate at the L.P. heater.

**394. The Condition Diagram of the Compound Turbine.**—As a preliminary to the detail design of nozzles and blading of the compound impulse turbine, it is necessary to fix provisionally the heat distribution and the approximate values of the successive stage pressures, qualities, and volumes.

In the reaction machine it is necessary to know the heat drop through each section and the variation of volume.

The foregoing information can be obtained from a "condition diagram," consisting of curves of cumulative heat, quality (superheat or dryness), and specific volume, plotted to a base of absolute pressure.

This diagram for the case of the impulse turbine (Examples 1 and 2) is shown in Fig. 444. A number of pressures between the

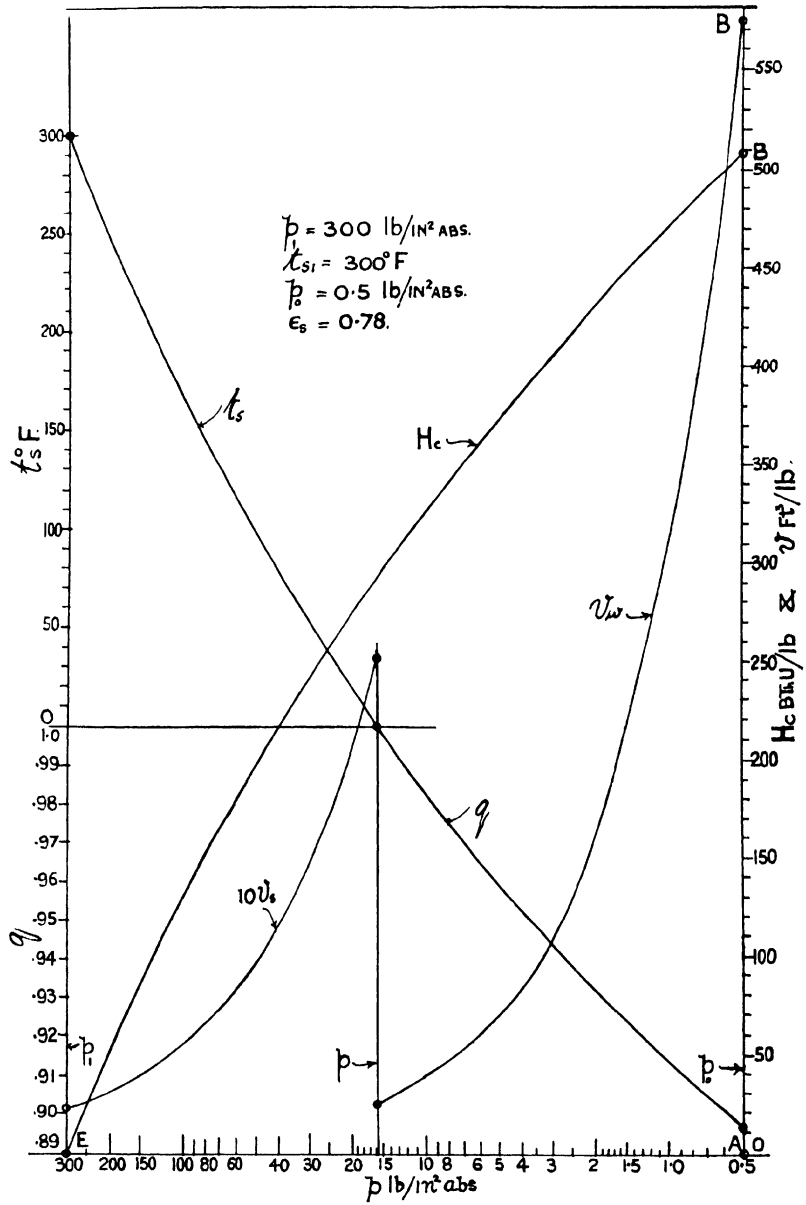



FIG. 444.

initial pressure $p_1$ and exhaust pressure $p_0$ are chosen, and the values of the cumulative heat $h_c=\frac{h_i}{\epsilon_s}$, superheat $t_s$, dryness $q$, and specific volume $v$ are derived from the condition curve on the HΦ diagram.

Owing to the rapid rate of increase of specific volume between atmospheric and exhaust pressures, the volume curve becomes impracticable between these limits if a uniform scale of pressure is used. The defect can be partly overcome by the use of a logarithmic scale of pressure, which may be conveniently obtained from the lower scale of a 10-in. slide rule. With this modification, however, there is difficulty in reading the small volumes at the high-pressure end, and it is advisable to plot the volume curve down to about the atmosphere limit for ten times the actual volume per lb., a tenth of the reading being used in the calculations.

The construction and reading of the diagram is simplified if the same unit length is taken for the heat and volume scales. The end ordinate AB gives the total cumulative heat (per lb.) $H_c$, at the exhaust pressure $p_0$.

It will be obvious that if this value is arbitrarily divided, in accordance with the proposed heat distribution among the stages, projectors from the points to the cumulative heat curve EB will fix the approximate initial stage pressures and the corresponding qualities and volumes. In the case of an axial flow reaction turbine, the initial and final pressures and volumes for successive sections can similarly be determined from the condition diagram.

In the case of a combined impulse type with a two-velocity compounded high-pressure stage, or a disc and drum reaction turbine, where a considerable proportion of the total energy is allocated to the first stage, the arbitrarily chosen heat drop of this stage should be used to determine the first stage pressure limits and stage efficiency ratio. The initial pressure and quality at the second stage should then be made the starting point for the condition curve. In general, the pressure at entrance to the first stage nozzles is less than the stop valve pressure, and in some cases it is advisable to assume a drop of from 10 to 15 lb./in.$^2$ between the stop valve and the nozzles, the lower pressure giving the initial absolute pressure for the condition curve.

The condition diagram can be used for impulse machines of six stages and upward, and for reaction machines; but it is not suitable for the smaller sizes of impulse turbines, having two to five stages. In such a case it is not difficult to obtain the heat drop per stage, and the corresponding stage pressures, directly from the HΦ diagram, by trial. For this purpose a value of the total reheat factor R can be assumed, and the approximate value of the heat drop per stage can be obtained from—

$$h_\phi=\frac{\mathrm{RH}_\phi}{n} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (67)$$

where $n$=number of stages and $H_\phi$=adiabatic heat drop between $p_1$ and $p_0$. For this value of $h_\phi$ the probable value of the first stage exhaust pressure can be obtained on the HΦ diagram and a rough estimation can be made of the average stage efficiency. The initial state point for the second stage can thus be found by the projective method of Art. 384, and the process can be repeated for the successive stages with the same $h_\phi$ and $h_i$ values.

If the final point on the last stage adiabatic falls above or below the exhaust pressure curve, the value of $h_\phi$ has to be slightly modified, and the process is repeated. As a rule two trials are sufficient to give a satisfactory result.

The tentative stage pressures thus determined can be taken for the step by step calculation of nozzle and blade proportions. They may probably have to be modified as the work proceeds, in order to obtain more efficient conditions of operation.

**395. General Dimensions of Impulse Turbines.**—Assume that for a given case the output, speed, and the steam conditions are provisionally settled, then in the case of the pressure compounded impulse, the stage heat drop is usually kept in the neighbourhood of the critical, and may be taken as $h_\phi$=50. With a total cumulative heat value $H_c$ the number of stages may be taken as—

$$n=\frac{H_c}{50} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (68)$$

The mean blade ring diameter may be kept sensibly constant, except at the last stage, where an increase may be necessary to obtain sufficient area for steam flow. Instead of keeping a constant diameter, the successive rings may be progressively stepped, from the first high-pressure to the last low-pressure stage. This arrangement is desirable when the pressure is high at the H.P. end and disc and vane friction may become appreciable.

This condition, however, may also be met by transforming the machine into a combined impulse, with a two-velocity compounded wheel, to absorb a considerable proportion of the available energy, followed by a number of simple impulse wheels.

In the first instance with a common diameter throughout, of D inches, and rotational speed of N rev./min., the blade speed is given by—

$$u=\frac{\pi DN}{12\times 60} \text{ ft./sec.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (69)$$

Also

$$u=\rho_t V_0' \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (70)$$

where $V_0'$ is the absolute velocity due to $h_\phi$, the adiabatic stage heat drop, or $V_0'=223{\cdot}7\sqrt{h_\phi}$.

Substituting for $u$ in (70) and taking $h_\phi$=50—

$$\rho_t\times 223{\cdot}7\sqrt{50}=\frac{\pi DN}{720}$$

and the mean ring diameter is given by—

$$D=\frac{223{\cdot}7\times720\sqrt{50\rho_t}}{\pi N}$$

$$=\frac{362000\rho_t}{N} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (71)$$

The blading velocity may vary from about 550 at the H.P. to 650 at the L.P. end, when D is increased at the L.P. to obtain increased discharge area through the last blade ring. Higher L.P. speeds, however, are in use, values between 700 and 800 ft./sec. occasionally being employed with large output machines, special discs and blading material being used.

**396.** As this tentative method of estimating the ring diameter takes no account of the weight of steam to be passed through the nozzle and blade channels, a check has to be made to ascertain if, at the L.P. end, there is sufficient discharge area. At the H.P. end the difficulty is to provide the necessary area with a sufficiently large nozzle arc to get rid of the partial admission at the second or third stage (where a velocity compounded wheel is not used). The nozzle angles, exit blade angles, and blading heights are kept down to a minimum. The smallest blade length is about $\frac{3}{4}$ in., and the nozzle angle 12°.

An expert designer can, as a rule, fix the rate of steam consumption with considerable accuracy from previous experience. Otherwise the weight to be passed through each stage or W lb./sec. has to be estimated on the over-all thermal efficiency $\eta_0$, after the necessary allowance is made for outside losses and generator efficiency. Thus—

$$W=\frac{3412 \text{ kw.}}{H\eta_0} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (72)$$

where H is the heat per lb. of gross feed, reckoned from the feed temperature and including any reheat, and kw. the output at the generator terminals.

Where regenerative heating is employed, this quantity has to be reduced at the different sections by the amount of extracted steam. Also where divided flow L.P. blading is used, a corresponding division of the total flow has to be made.

Assuming that the value of W at the last stage is ascertained, the mean nozzle ring diameter can be determined from the equation of continuity.

The exit area of the nozzles is given by—

$$A_0=\frac{W\rho_t v_{w0}}{u} \text{ ft.}^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (73)$$

when $v_{w0}$=specific volume of the exhaust steam at $p_0$ in ft.$^3$/lb., $\rho_t$=the theoretical speed ratio, and $u$ the mean blade velocity.

Now if $\alpha$=nozzle angle, the exit area is given by—

$$A_0 \doteqdot \pi D l_n \sin \alpha \quad . \quad . \quad . \quad . \quad . \quad . \quad (74)$$

where $l_n$=the exit length. Also $u=\frac{\pi DN}{60}$

Substituting for $A_0$ and $u$ in Equation 73 and simplifying, the nozzle ring diameter is given approximately by—

$$D=\sqrt[3]{\frac{60\rho_t W v_{w0}}{\pi^2 ZN \sin \alpha}} \text{ ft.} \quad . \quad . \quad . \quad . \quad . \quad (75)$$

where $Z=\frac{l_n}{D}$. It is not desirable under ordinary circumstances to make the value of Z greater than 20 per cent. In some recent and large machines the figure has been advanced to 30 per cent.; but very special alloy steels have been used for the discs and blading in these cases.

When the last ring diameter is calculated for the arbitrarily chosen values, the corresponding mean blade velocity has to be checked. This should not exceed 750 ft./sec. with the uniform type of impulse blade, if the ratio Z is about 20 per cent. It is desirable to keep it below 700 ft./sec., otherwise the factor of safety of the blades is low. When the final stage diameter D is fixed, the velocity diagram for the stage should be drawn down, to determine the final stage efficiency, and to ascertain if the leaving loss or carry over to the condenser is reasonable. In this country the leaving loss at economical load is reckoned at a maximum of 3 per cent.; but in American practice from 5 to 6 per cent is allowed, to obtain larger outputs for given blade stressing, but at a lowered efficiency.

When the general dimensions are provisionally settled the heat drop for the last stage can be estimated. An arbitrary value of the heat drop for the first stage can be chosen, to give a desired first stage pressure. The rest of the cumulative heat can then be divided among the remaining stages and the stage pressure limits ascertained from the condition diagram.

In the stage design, the velocity diagram should be drawn for each stage, with due regard to nozzle and blade angles and lengths; and the nozzle efficiencies can be judged from the qualities shown by the condition diagram. Approximate disc and vane friction losses * can be estimated for the first few stages and the varying values of the stage efficiency ratios can be estimated. The nozzle and blade proportions can then be adjusted if desired.

**397.** In the case of the machine with the two-velocity compounded H.P. stage followed by a series of simple pressure stages with progressively increasing ring diameter, a heat drop of from 25 to 35 per cent. of the total may be chosen. The higher the initial pressure and temperature conditions, the larger the percentage to

* See *Steam Turbines*, by W. J. Goudie, Chapter XII.

be allowed. The main object is to simplify the design constructionally, by the reduction of pressure and temperature, and to improve the thermal efficiency by reduction of disc and vane friction and leakage loss at the H.P. stages. When a definite heat drop has been selected for the first stage, the stage design for nozzles and blades can be carried out, without any reference to the pressure compounded section. The exhaust pressure conditions of the velocity compounded stage are then taken as the initial conditions for this section. Dealing with the pressure compounded part as a separate machine, the condition curve can be drawn on the HΦ diagram for a probable stage efficiency ratio, and the internal thermal efficiency for the whole machine and the probable weight of steam to be passed through the last ring can then be estimated. The diameter of the last stage nozzle and blade rings can then be determined.

The next step involves the consideration of three variables, the diameter, number of stages, and the increment of diameter per stage.

Let $n$ =number of simple impulse stages or one-ring wheels.

$D_1$=mean diameter of the first wheel in inches.

$D_n$=mean diameter of the last or $n$th wheel in inches.

$x$ =constant increment of diameter per stage in inches.

$\rho_t$ =constant theoretical speed ratio.

Then the successive diameters are $D_1$, $(D_1+x)$, $(D_1+2x)$, etc.

At any stage the heat drop is—

$$h_\phi=\left(\frac{V_0'}{223{\cdot}7}\right)^2$$

but since—

$$V_0'=\frac{u}{\rho_t}=\frac{\pi DN}{12\times 60\rho_t},\quad h_\phi=\left(\frac{\pi DN}{720\rho_t}\right)^2\frac{1}{223{\cdot}7^2}=\frac{N^2D^2}{(51270\rho_t)^2} \qquad (76)$$

For a given machine N is constant and $\rho_t$ is taken as a constant, so that $h_\phi$=const. $D^2$, and the total cumulative heat of the pressure compounded section is—

$$H_c=\Sigma h_\phi=\text{const. }\Sigma D^2=\frac{N^2}{(51270\rho_t)^2}\Sigma D^2 \quad . \quad . \quad . \quad (77)$$

$$\Sigma D^2=D_1^2+(D_1+x)^2+(D_1+2x)^2 \ . \ . \ . \ \{D_1+(n-1)x\}^2$$

It can be shown that this summation reduces to—

$$\Sigma D^2=nD_1^2+D_1x(n^2-n)+\frac{x^2}{6}(n^2-n)(2n-1)$$

Let $x=aD_1$, then this sum may be written as—

$$\Sigma D^2=D_1^2\left\{n+a(n^2-n)+\frac{a^2}{6}(n^2-n)(2n-1)\right\}=D_1^2A \qquad (78)$$

$$\text{and}\quad H_c=\frac{N^2D_1^2A}{(51270\rho_t)^2}$$

$$\text{or}\quad A=\frac{(51270\rho_t)^2H_c}{N^2D_1^2} \quad . \quad . \quad . \quad . \quad . \quad (79)$$

By choosing a trial value of the first wheel diameter $D_1$ and substituting this in (79) a value of the coefficient A can be obtained, and the corresponding value of $a$ or $x$ for a proposed number of stages $n$ can be ascertained from (78).

The solution of (78) is greatly facilitated by the use of a set of stage number curves, obtained by plotting A values against the ratio $a=\frac{x}{D}$, as shown in Fig. 445.

If an approximately correct value of the first wheel diameter $D_1$ has been chosen, then the substitution of $D_1$ and $n$ in Equation 80, deduced below, should give the value of $D_n$. As a rule the exact value is not obtained; but an approximation is quite sufficient for the object in view. Any necessary adjustment of the speed ratio $\rho_t$ (assumed constant in this process) can be made afterwards in the stage design. Since an assumption of constant increase of diameter ($x$) is made, the diameters of the successive wheels are in arithmetical progression, hence—

$$D_n=D_1+(n-1)x$$

or

$$D_n=D_1\{1+(n-1)a\} \quad . \quad . \quad . \quad . \quad . \quad (80)$$

If the value of $x$ or $a$, ascertained from the curves (Fig. 445), when substituted in Equation 80 does not give the required value of $D_n$, then $D_1$ must be altered to some value $D_1'$. The value of A to be used to find $a_1$ with a given number $n$ is—

$$A'=A\left(\frac{D_1}{D_1'}\right)^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (81)$$

and the process has to be repeated.

If the final value of $D_1$ which satisfies (80) does not appear to be satisfactory, a change of the stage number $n$ should be made. Owing to the existence of three variables the process is tedious. If a fourth variable, in the form of $\rho_t$, is introduced, the procedure becomes impracticable.

It may be found necessary in a final adjustment to alter the value of the constant $\rho_t$ up to the last stage.

It will be noted in the case of a large machine, where the restricted area at the L.P. end necessitates some system of divided or double flow, and the mean diameter is kept sensibly constant throughout, till the modified staging is reached, that since $x=0$ and $a=0$, Equation 77 reduces to the form—

$$H_c'=\text{const.}\ \Sigma D^2=\frac{nD^2N^2}{(51270\rho_t)^2} \quad . \quad . \quad . \quad . \quad (82)$$

where $H_c'$ is the cumulative heat, up to the modified staging. If $h_c$ is the amount of heat allocated to the divided L.P. section, then—

$$H_c'=(H_c-h_c) \quad . \quad . \quad . \quad . \quad . \quad . \quad (83)$$

It is of interest to note that Equation 78 is applicable to the case

of the radial flow Ljungström reaction turbine, in which there is a progressive increase of the moving ring diameters.

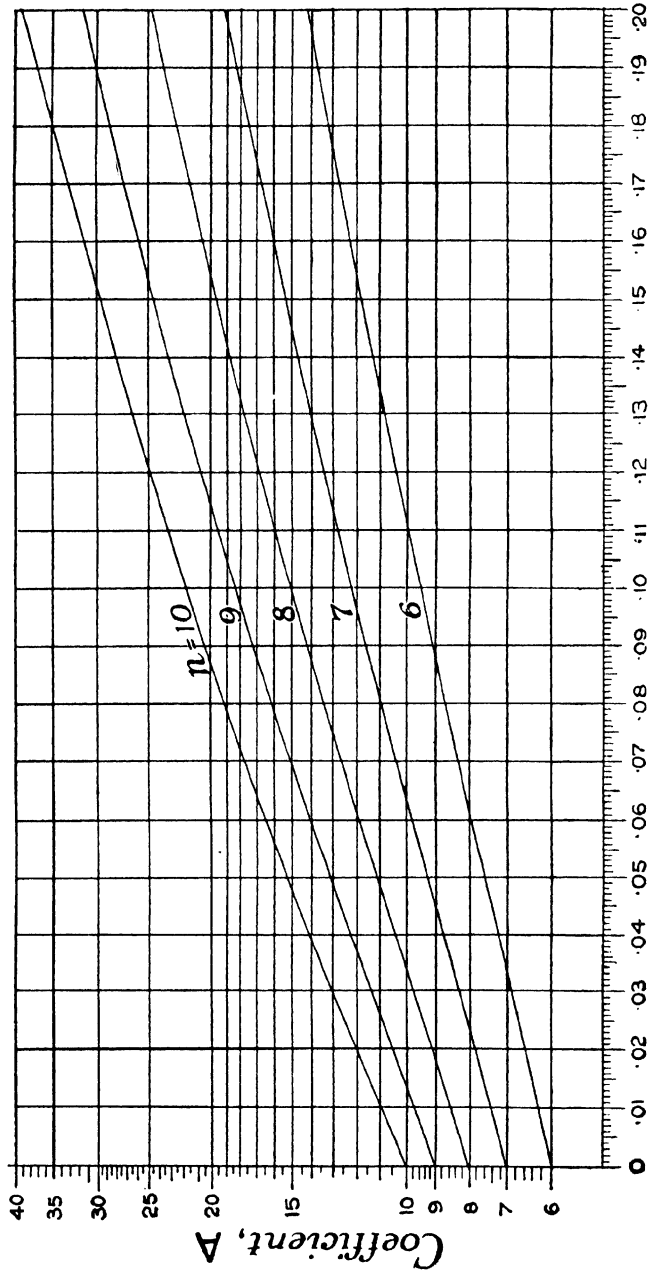



FIG. 445.

**398. General Dimensions of Parsons Reaction Turbines.**—In the last five or six years the reaction turbine has undergone considerable changes in design. Owing to the higher steam conditions and rotational speeds adopted for high outputs, the drum type has practically been replaced by the solid forged rotor. In the earlier practice the drum of the single-cylinder machine was usually formed with three or more steps, each step carrying several groups of blades, with uniform blade length over each group, but with increasing length from group to group. Now the solid rotor is formed in various ways. It may have a parallel H.P. section followed by a tapering I.P. section leading to the parallel L.P. section, instead of the abrupt changes in diameters. In the case of the small geared type from 1000 kw. to 1500 kw. output, a disc and drum rotor may be used, the H.P. section consisting of the two-velocity compounded stage followed by an intermediate section of blading on the tapering rotor, and the L.P. section of blading on the final parallel part of the rotor. In outputs of 15,000 kw. and over the L.P. flow of the single-cylinder machine may be subdivided and the final section of the L.P. blading may be mounted on the rims of several robust disc wheels.

For large outputs, 50,000 kw. and upwards, three or more cylinders may be placed in line, the successive shafts being connected by flexible couplings. In these cases each rotor is usually of the shaft and disc type, the discs being provided with wide rims on which the reaction blading is mounted. The consecutive rims of the H.P. and I.P. discs are equivalent to parallel drum rotors, and the blading is progressively stepped in height from the H.P. to the L.P. end of each drum. The low-pressure cylinder is usually double flow, a disc wheel being employed for each stage.

For very large outputs two and three cylinder cross compound turbines are being adopted, the H.P. and L.P. units usually running at different speeds. In the case of the 165,000 kw. outfit by the Westinghouse Co. in Hell Gate Station, N.Y., the H.P. element has a solid rotor carrying a H.P. velocity compounded stage. Part of the I.P. blading is mounted on the parallel portion of the rotor and on the tapering part, which is followed by a parallel part. The L.P. element is double flow, the blading being mounted on a parallel solid rotor.

On the other hand, the companion 165,000 kw. cross compound turbine in this station, by Brown, Boveri and Co., has a H.P. shaft and disc rotor consisting of five wide-rimmed discs, the equivalent of a parallel drum, carrying sets of reaction blading of progressively increasing diameter. The L.P. element is double flow. Each half of this rotor consists of seven disc wheels, each wheel carrying a ring of reaction blading of progressively increasing diameter.

There are other variations in the rotor design of normal turbines by various Continental makers that might be cited; but the foregoing outline of the diversity of design should be sufficient to

indicate to the student the futility of attempting to formulate a general method for the proportioning of the reaction turbine blading, as each case has obviously to be dealt with in accordance with the special conditions specified. The calculations are further complicated by the provisions for overload, by-pass, reheating, and extraction feed heating.

**399.** The following tentative suggestions regarding the calculation of blading proportions in sections for which assumed conditions are taken may, however, be of some assistance to the student.

In any given case it is possible to assume a probable value of mean stage efficiency ratio $\epsilon_s$ and to draw down the condition curve on the H$\Phi$ diagram, from which the cumulative heat and volume curves can be plotted to a pressure base. For the arbitrarily chosen distribution of the total cumulative heat $H_r$ in the various sections of the single cylinder machine, or the separate cylinders of the tandem or cross compound types, the limiting pressures of the various sections can be determined. Alternatively, as in the case of reheat or regenerative heating, the pressures may first be arbitrarily fixed and the heat values determined.

Instead of using a single condition curve, corresponding to a mean stage efficiency ratio for the whole combination, a curve may be drawn for each arbitrarily selected pressure range corresponding to the mean stage efficiency assumed for the particular section, and the partial condition diagram for this section can be drawn.

Assuming the machine is to be reaction throughout, the mean blade ring diameter at the first H.P. stage, to pass the gross amount of steam, W lb./sec., at normal load under the specified initial conditions $p_1$ and $t_{s1}$, can readily be calculated. Without sensible error the specific volume at first nozzle outlet may be taken as $v_{s1}$. If then the blade exit angle is $\theta$, the speed ratio is $\rho$, corresponding to the assumed $\epsilon_s$ value, the ratio of blade length to mean ring diameter, or $\frac{h_1}{D_1}$, is Z, and the rotational speed is N rev./min., it follows, from the equation of continuity, that—

$$D_1{}^3=\frac{60\rho W v_{s1}}{\pi^2 Z N \sin\theta} \quad . \quad . \quad . \quad . \quad . \quad . \quad (84)$$

With radial clearance, in order to avoid excessive tip leakage, Z should not exceed 0·03. With the modern system of end tightening Z may be taken from 0·06 to 0·12, the value depending on the speed and output. As an alternative the diameter may be provisionally fixed, for an assumed mean blade speed $u$, from 350 to 450 ft./sec. In this case—

$$D_1=\frac{60u}{\pi N} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (85)$$

Assuming that the first section of blading is mounted on a parallel rotor, or equivalent disc rims, the diameter of this rotor is—

$$D_h=D_1(1-Z) \quad . \quad . \quad . \quad . \quad . \quad . \quad (86)$$

The number of stages, and hence the number of moving rings on this H.P. section, is determined by the arbitrary allocation of some proportion of the cumulative heat to the section, and the progression of blade length.

Suppose, as in the most recent practice, that each blade ring progressively increases in diameter. The curve of blade length will approximate in form to the curve of specific volume, as shown on the condition diagram between the pressure limits of the section.

If $v_{s1}$ is the specific volume at $p_1$ and $v_{sm}$ the volume at $p_m$, the final pressure on the section having $m$ moving rings or "double stages," then to a first approximation the $m$th blade length of the group is given by—

$$l_m = l_1 \frac{v_{sm}}{v_{s1}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (87)$$

Using this value provisionally the first approximation to the corresponding mean ring diameter is given by—

$$D_m = (D_h + l_m) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (88)$$

It can be readily shown that, given $D_m$, the blade length $l_m$ is also given by—

$$l_m = \frac{10500 W v_{sm} \rho}{D_m^2 N \sin \theta} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (89)$$

If the value of $l_m$ obtained from (89) does not closely approximate to the value used in (88) to obtain $D_m$, then a modified value has to be used, until there is a coincidence in the results.

The number of moving rings or double stages $m$ can be estimated from the work equation by using the average heat drop per stage, which is $h_\phi = \frac{h_c}{m}$, and correcting by a leakage coefficient $\gamma$ varying from 0·95 to 0·98. The mean value $u$ of the blade velocity which may be taken to correspond to the diameter $\frac{D_1 + D_m}{2}$, is to be used in the work equation.

This is—

$$Jg\epsilon_s \frac{h_c}{m} = \gamma u^2 \left( \frac{2 \cos \theta}{\rho} - 1 \right)$$

so that—

$$m = \frac{Jg\epsilon_s h_c}{\gamma u^2 \left( \frac{2 \cos \theta}{\rho} - 1 \right)} \quad . \quad . \quad . \quad . \quad . \quad . \quad (90)$$

It is then sufficient to assume that the blade lengths are sensibly in geometrical progression so that the common ratio is given by—

$$r = \left( \frac{l_m}{l_1} \right)^{\frac{1}{m-1}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (91)$$

The consecutive lengths are then given by—

$$l_1,\ rl_1,\ r^2l_1,\ r^3l_1, \text{ etc.} \quad . \quad . \quad . \quad . \quad . \quad (92)$$

When these provisional dimensions are fixed the whole section can be examined stage by stage and modification of the blading arrangement made if thought desirable.

**400.** If the older system of mounting groups of blades, each group having a constant blade length, is adopted, the discharge angle $\theta$ may be kept constant throughout. In this case since the mean blade velocity $u$ remains constant the speed ratio $\rho$ increases between the first and last stages of the group. This is due to the progressive increase of steam volume, while the outlet area of the successive blade exits remains constant. If a constant value of $\rho$ is to be maintained, then the blade angles have to be progressively increased. The blades in this case are said to be "gauged."

Usually, however, the angle is kept constant and the velocity ratio is allowed to diminish. The result is that the work done on the series of rings is not quite equally distributed.

The common blade ring diameter, $D_1=D_m$, in this case is obtained from Equation 84, the mean steam volume $\frac{v_{s1}+v_{sm}}{2}$ being used. The number of rings is again obtained from Equation 90.

**401.** At the last L.P. stage the diameter of the blade ring is controlled principally by the permissible stress, and as already pointed out it is advisable to keep the blade-length diameter ratio Z within the 20 per cent. limit, unless very special alloy steel discs and specially formed "conical" steel blades are employed. The difficulty with large outputs can be overcome by splitting up the stream of steam into two streams and passing each into a section of L.P. blading. Owing to the reduction in quantity the volume to be handled in each section is much reduced and the blade length can be made a reasonable value, without excessive increase of angle and increased leaving loss. In the single flow case the blade length limitation involves the case of wide-angled nozzle and moving blades, with angles of 30°, 50°, and 70°, which are designated as "semi-wing," "wing," and "double-wing."

The mean ring diameter $D_l$ at the last L.P. stage is obtained from the equation of continuity, using the provisionally selected values of Z, $\rho$, and $\theta$, and the final steam volume $v_l$.

Thus—

$$D_l^3=\frac{60\rho W v_l}{\pi^2 ZN \sin\theta} \quad . \quad . \quad . \quad . \quad . \quad . \quad (93)$$

The diameter of rotor or disc on which the blades are to be mounted is—

$$D_L=D_l(1-Z) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (94)$$

In the case of a tandem or cross compound turbine with double

flow L.P. cylinder, W is half the gross amount exhausting into this cylinder.

The blading on the low-pressure section, like that on the high-pressure, can be calculated for a chosen amount of cumulative heat, which as before defines the pressures and specific volumes at entrance and exit. The first L.P. ring diameter can be determined by the process of Art. 399, and the number of rings can be determined from the energy equation.

**402.** Between the H.P. and L.P. sections, as already stated in the case of a single cylinder or the H.P. cylinder of a tandem or cross compound, the rotor may be stepped up in different ways. As this stepping and tapering is of an arbitrary nature and as the complete rotor may consist entirely of wheels or a forging plus wheels, it is not possible to consider the estimation of intermediate groups of blading in any definite manner.

On a coned or tapered length of the rotor carrying a group of progressively lengthening blades, assuming the initial and final diameters of this tapered portion are first arbitrarily fixed, then from the limiting pressures and volumes, for the allocated portion of the cumulative heat shown on the condition diagram, the first and last ring diameters can be calculated from the equation of continuity in the manner already described, and the initial and final blade lengths can be ascertained. For the chosen size of blade the axial pitch $P_b$ is determinable, and if the axial length of the tapered part provisionally allocated between the first and $m$th ring is L, then the number of moving rings is given by—

$$m=\left(1+\frac{L}{P_b}\right) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (95)$$

For a given rotational speed N, the blade velocity $u$, at each of the $m$ rings, for which the diameter D has now been provisionally determined, is obtained, and the heat drop of the particular double stage is then given by the energy equation, thus—

$$h_\phi=\frac{\gamma u^2}{Jg\epsilon_s}\left(\frac{2\cos\theta}{\rho}-1\right) \quad . \quad . \quad . \quad . \quad . \quad (96)$$

The summation of these values should approximate to the allocated amount of cumulative heat $h_c$ over the section—

$$\text{or} \qquad \Sigma h_\phi=h_c \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (97)$$

In this way a check on the suitability of the number of rings can be made, and the number or the rotor dimension can be increased or decreased, if necessary, to ensure the absorption of the alloted amount of cumulative heat.

These notes on blading dimensions, etc., refer to the axial flow type of reaction turbine. The procedure in these cases is fairly simple in comparison with that which has to be employed in the case of the radial flow double-motion Ljungström turbine, a section of which has been given in Fig. 414. It is not proposed on account

of space limitations to deal with this type here. The student can refer for a full discussion of this subject to the writer's text-book.*

**403. Coefficient Method of Calculation for Number of Reaction Blade Rings.**—A rapid estimate of the number of rings for a given section of a rotor, to absorb a stated amount of commulative heat, can also be made by the use of a design coefficient ($\lambda_1$).

It can be shown, from first principles, that if in a group of $m$ rings of constant length the diameter is D and the cumulative heat absorbed is $h_c$—

$$\lambda_1 = 1314\delta\rho^2 h_c \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (98)$$

where $\delta$ is a friction coefficient varying from 0·95 to 1·1 according to the exit blade angle, and $\rho$ is the constant speed ratio for the group.

This is also expressed by—

$$\lambda_1 = m\left(\frac{D}{10}\right)^2\left(\frac{N}{100}\right)^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad (99)$$

The numerical value of $\lambda_1$ can be obtained from (98), and for a given diameter and speed the number of rings $m$ on the section can be calculated from (99).

For a complete turbine having various groups of different diameters, $\Sigma\lambda_1 = 1314\delta\rho^2\Sigma h_c$, that is, the coefficient for the whole machine is given by—

$$\lambda = 1314\delta\rho^2 H_c \quad . \quad . \quad . \quad . \quad . \quad . \quad (100)$$

where $H_c$ is the total cumulative heat for the whole machine.

Expressed in terms of the diameters and speed this now becomes—

$$\lambda = \Sigma\left(\frac{D}{10}\right)^2\left(\frac{N}{100}\right)^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad (101)$$

since the diameters vary throughout the machine.

This factor is known as the "Parsons coefficient," and serves as a proportional measure of the internal efficiency ratio of the machine, since $\Sigma D^2N^2 \propto \Sigma u^2$ and with a constant value of speed ratio $\rho$, $\Sigma u^2 \propto \epsilon_I$.

The higher the value of $\lambda$ the greater the value of $\epsilon_I$; and one of the objectives of designers is to obtain as high a value of $\lambda$ as possible from a given design.

It will be evident from what has already been said, with regard to the controlling effect of $\rho$ on the internal efficiency, that this factor will be restricted, if $\rho$ is restricted below the theoretical value for maximum efficiency in any given case.

In early reaction turbine designs with small outputs at relatively low speeds it was seldom that this coefficient exceeded a value of 120,000, the corresponding turbine efficiency ratio ($\epsilon_B$) being in the neighbourhood of 0·7.

On the advent of the famous 25,000 kw. Parsons turbine for the

* See *Steam Turbines* (2nd edition), by W. J. Goudie, Chapter XVII.

Crawford Avenue Station, Chicago, in 1912, the figure rose to 200,000. At the same period the first double-motion Ljungström radial flow turbine of 1000 kw., with its high value of speed ratio, appeared, with a coefficient value of 225 000 and an efficiency ratio in the neighbourhood of 80 per cent. Much higher values are now being obtained from modern high-speed high-output axial flow reaction turbines. For instance, the 165,000 kw. Brown Boveri two-cylinder cross compound turbine, installed in Hell Gate Station, N.Y., in 1929, has a coefficient of 313,000 for the H.P. and of 240,000 for the L.P., giving a total $\lambda$ value of 553,000 with about 83 per cent. efficiency ratio at the turbine couplings. Ljungström turbines for outputs of 15,000 kw. at 3000 r.p.m. are credited with values of 455,000, and for higher outputs of 50,000 kw. at 1500 r.p.m. with the higher value of 680,000.

# CHAPTER XXIII

## CONDENSERS

**404.** THE steam condenser is the third element, the "cooler," of the complete heat engine (Art. 21). Its thermodynamic function is the abstraction of heat.

Practically it serves a wider purpose. It may be operated to condense the exhaust steam, under atmospheric conditions, for use as feed water, or to produce a vacuum and extend the range of expansion of the engine. As a rule it performs both functions.

**Classification of Condensers.**—Condensers may be divided into two classes: (1) Jet condensing. (2) Surface condensing.

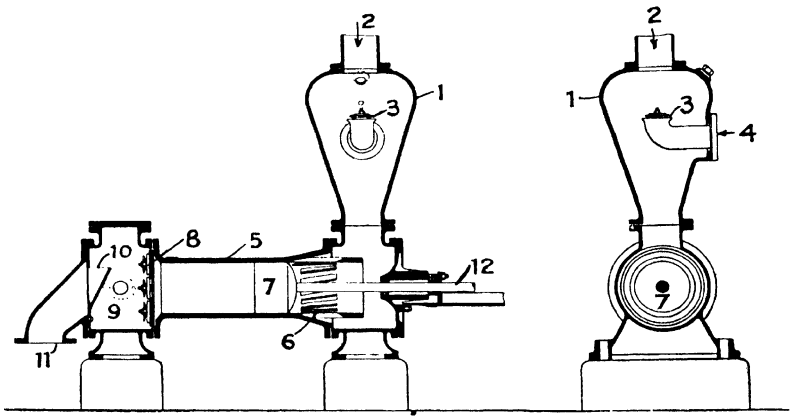



FIG. 446.

Jet condensers may be subdivided into three classes: (*a*) low level; (*b*) high level or barometric; (*c*) ejector condensers.

Surface condensers may be subdivided into two classes: (*d*) Arrangements in which the exhaust steam flows over the surfaces of tubes, through which condensing water is circulated. This form is what is always termed a "surface" condenser. (*e*) Arrangements in which the exhaust steam is passed through a series of tubes exposed to the atmosphere, and over which water is sprayed. The steam in the tubes is condensed by the abstraction of heat to vaporise part of the cooling water and to heat up the air. This type is called an "evaporative" condenser.

**405.** (*a*) **Low-level Jet Condensers.**—When the low-level type

is used in conjunction with a reciprocating engine of small or moderate output, the exhaust steam is usually discharged into the top of the condensing vessel and impinges on a spray of condensing water discharged from a nozzle. The mixture of condensing water, condensate, air and vapour falls to the bottom of the vessel and is drawn therefrom by a reciprocating " air pump."

The standard combination of this type adopted by Marshall, Son and Co., is shown in side and end sectional elevation in Fig. 446. Exhaust steam enters the condenser 1 at 2 and meets a circular water spray from the nozzle 3. The water from the supply pipe

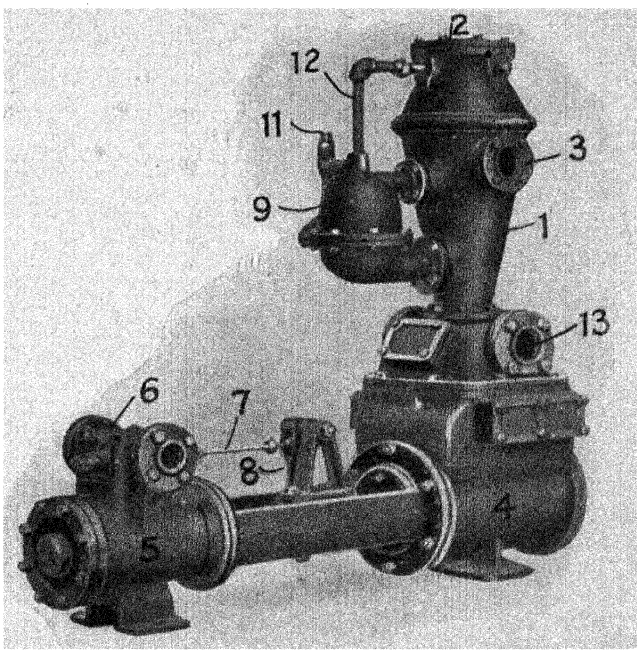



FIG. 447.

is sucked up from the cooling pond, and enters at 4. The pump barrel 5 projects into the lower portion of the condensing vessel, and is provided with a series of ports 6. When the solid piston 7 moves to the extreme right it uncovers these ports, and the mixture rushes in and fills the vacuated pump barrel. It is then discharged through the set of head valves 8 into the hot well 9, during the return stroke, and flows over the weir 10 into the outlet branch 11. The pump rod 12 is connected to the tail rod of the engine piston.

This arrangement dispenses with suction and bucket valves. It is simple in design, efficient in action, and is not liable to get out

of order. Part of the mixture is returned from the hot well as feed to the boiler. When, however, the condensing water is salt, or otherwise unsuitable for boiler feed, a surface type of condenser has to be used.

Another modern arrangement of spray condenser and pump, by Worthington-Simpson, Ltd., and termed the "Spray Pipe Jet" type, which is driven independently of the engine, is shown in Fig. 447. A section through the condenser vessel and pump is shown in Fig. 448. The reference numbers correspond on the two views. 1 is the condenser body, 2 the steam inlet, 3 the injection water inlet, 4 the pump, the piston rod of which is directly connected to the piston of the steam cylinder 5. The steam distributing valve in the casing at 6 is operated by the tappet valve rod 7, driven by a lever 8 from the piston rod. The casing 9 contains a ball float 10 and lever, which operates an atmospheric valve 11. This admits air to the condenser vessel through the pipe 12, and breaks the vacuum. The water cannot, therefore, rise above the level of the float, and flooding back into the engine cylinder is prevented.

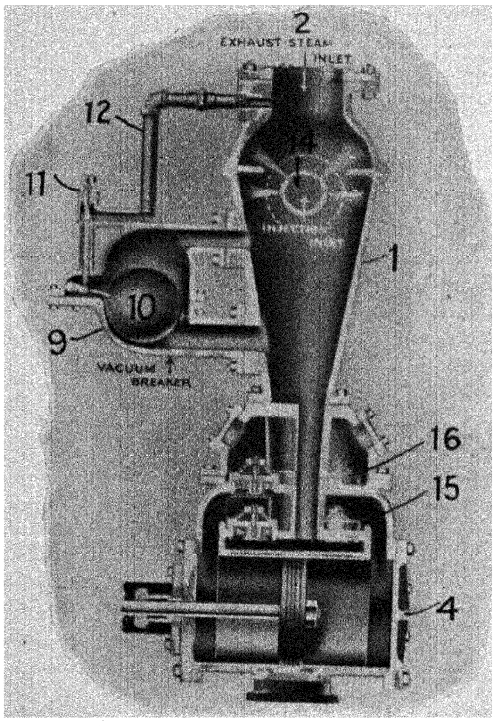



FIG. 448.

In this case the injection pipe 14 projects horizontally across the vessel. It is perforated with several narrow slits through which the water is sprayed, as indicated in Fig. 448. In another type this pipe is replaced by a bend, to discharge the water vertically upward. The outlet is closed by a coned stopper which can be adjusted to any height, by screwed spindle and hand wheel, so as to regulate the thickness of the fan spray and the quantity of condensing water. In this type, Fig. 448, the float is regulated by means of a gate valve on the injection supply pipe. The pump 4 is double acting, with the usual plate suction and discharge valves. The water flows

by gravity through the suction valves 15 and the associated air is drawn by suction as the piston moves out. The water and air are discharged through the valves 16 and branch 13, as the piston makes the return stroke.

**406. Parallel Flow Low-level Jet Condenser.**—Quite a different type of low-level jet condenser is in general use for steam turbines of small and moderate output. It consists of a cylindrical vessel placed vertically, and the condensing water and exhaust steam move through it, either in parallel or contra-flow. The mixture of condensing water and condensate falls to the bottom and is drawn away by a centrifugal water-extraction pump, while the air and

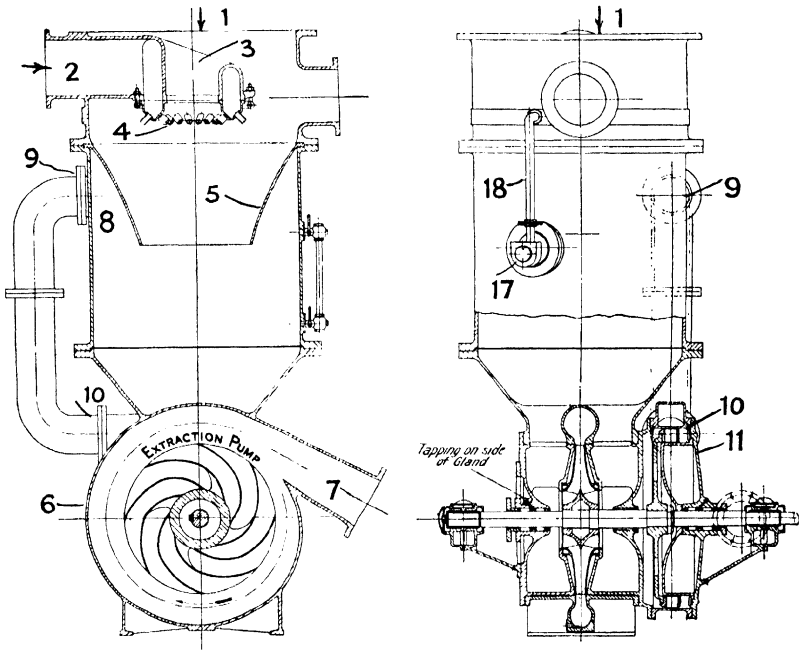



FIG. 449.

vapour are drawn off by some form of "dry" air pump, either at the top of the condenser or about the middle. The separation of the air and vapour from the water is essential for the production of a high vacuum for a steam turbine.

The parallel flow type made by the Metropolitan-Vickers Co. is shown in Fig. 449. The exhaust steam enters at 1 ; the condensing water flows in at 2 to the nozzle box 3, and is discharged through the set of large nozzles 4 with considerable velocity and dispersion, thus ensuring an intimate mixture with the steam and efficient condensation. The mixture is carried into the cone 5, and the incondensible gas is thereby compressed to a higher pressure than

at entry to it. The water falls to the bottom, and is drawn by the motor-driven centrifugal water-extraction pump 6, and discharged through the pump outlet 7, either to a cooling pond or a cooling tower. The mixture of air and vapour passes up underneath the cone at 8, and is drawn away through the branch 9, which is connected to the inlet 10 of the air pump 11, mounted on the same shaft as the extraction pump. The air pump is a Leblanc type of water-jet pump, a cross-section of which is shown in Fig. 450. It consists of a reversed pelton wheel 12 working in conjunction with an ejector 13. Sealing water is admitted to the central casing and discharged, through a fixed nozzle 14, into the blading of the rotating wheel 12. It is discharged from the wheel in the form of thin sheets at high velocity into the ejector cone 15. These sheets trap the air and vapour coming from the condenser through the pump inlet 10. Compression takes place in the cone 15, and the gas is driven into the

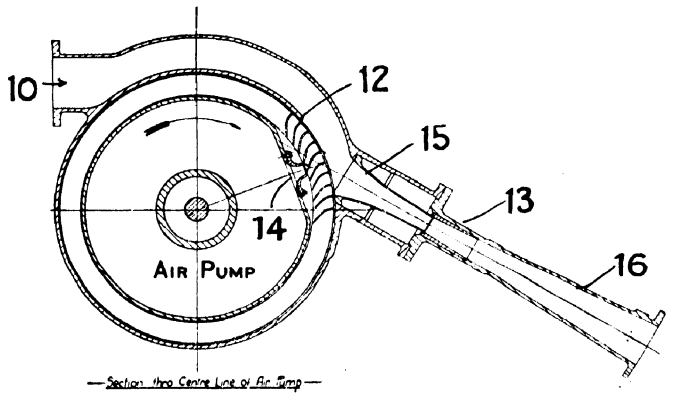



FIG. 450.

diffuser 16, in which sufficient velocity energy is converted to pressure energy to overcome the external atmospheric pressure head.

When a reciprocating pump is used in conjunction with a condenser, the air is extracted at the condenser water temperature; but with this type it is reduced to the temperature of the sealing water, and the volume to be extracted by the ejector is diminished. As already indicated in the previous cases, with this low-level type there is always the risk of flooding into the turbine exhaust, and a float-operated vacuum breaking valve is always fitted. In this case the valve is placed at 17 (Fig. 449) and admits air to the condenser, through the pipe 18, when the float acts.

A section of the type of parallel flow low-level jet condenser made by Worthington-Simpson, Ltd., is shown in Fig. 451. In this case the nozzle box is replaced by a combination of cylinder and cone 3, the cylindrical portion being perforated by two rows

of holes 4. This combination fits into a lower cone 5, so as to form a water belt 6. Injection water is supplied through the branch 2 to the belt and sprays through the holes 4, which are bored large enough to prevent choking by any foreign matter in the condensing water. The exhaust steam discharged into the top of the condenser at 1 is condensed by the water sprays and falls with the condensing water to the bottom, from which the mixture is drawn through the branch 7 by the extraction pump. The air and vapour, compressed in passing through cone 3, flow upward round the outer cone 5 at 8, and are drawn away by the air pump through the branch 9. Flooding is prevented by the float operated vacuum breaker at 10. The level of the water relatively to the suction branch 7 is indicated by the gauge glass 11. Air entrained in the condensing water rises to the top of the water belt, and is drawn away by the air pump through the air release pipe 12. An atmospheric exhaust branch is provided at 13. A vacuum gauge is fitted at 14 and a kenotometer at 15, and a balance from the extraction pump at 16.

The condenser can be drained by the cock at 17, and cleaned out through the manhole at 18. The inlet 19 is provided to take a heater drain connection.

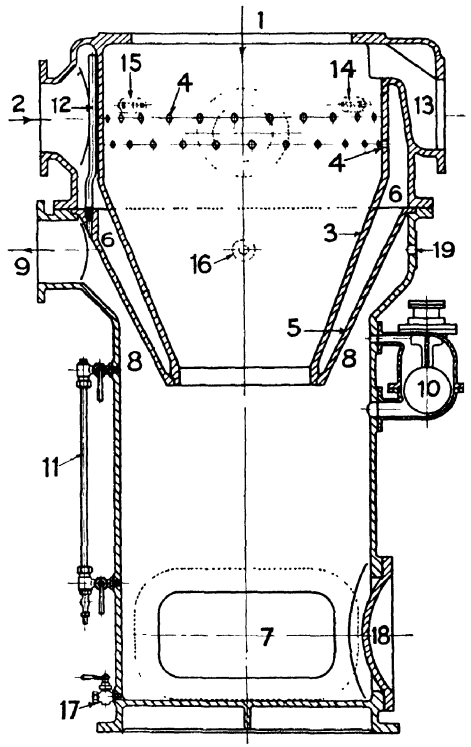



FIG. 451.

In condensers of this type the rotary air pump is often replaced by one or more steam ejector pumps. An outside view of a parallel flow low-level jet condenser with this arrangement, by the Merrlees Watson Co., is shown in Fig. 452. In this case the exhaust enters at 1 and the injection water at 2. It flows into an annular space round the belt 3 and as in the previous case is discharged through holes in an inner cylinder at the top of a compression cone. The water is drawn into the extraction pump 4 and discharged through the outlet branch 5. The air and vapour are drawn up and discharged by the steam-jet ejector 6. The vacuum breaking valve 7

is operated by a float inside the cover 8. When a jet ejector is used it is the usual practice to provide an auxiliary cooling spray to cool the air and vapour below the cone at 8, Figs. 449 and 451, in order to diminish the volume to be handled by the ejector.

**407. Contra-flow Low-level Jet Condenser.**—In this type the

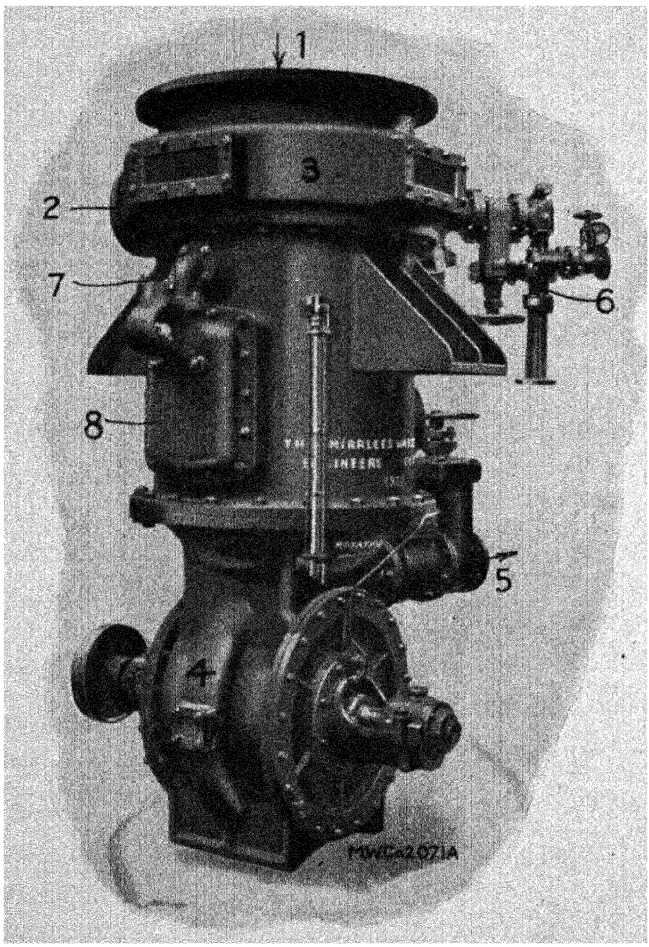



FIG. 452.

injection water is discharged into the open condenser space, not into a nozzle box or water belt, and distributed over a perforated tray, through which it falls under the action of gravity in the form of a rain. It is caught by a lower tray and again discharged downwards from this as a rain. It may be passed through a third and a fourth tray in this way. The exhaust steam is discharged into the

condenser near the bottom, and rises upward through the rain of water. The air and incondensible gas also rise with it, and are drawn away at the top by a dry air pump. The mixture of condensing water and condensate is drawn away at the bottom by a water extraction pump.

A diagrammatic sketch of one form of the contra-flow low-level type is shown in Fig. 453. The injection water enters at 1 and is distributed over the sloping tray 2 provided with several rings of holes. The water, which is maintained at some level, shown by the dotted line, flows through these holes, as indicated by the sloping arrow lines, and is intercepted by the tray 3, also provided with several rings of holes towards the circumference. It flows through these, as indicated by the vertical arrow lines, on to the tray 4 provided with a central passage 5. It then falls into the cone 6 and into the well 7 at the bottom, from which it is drawn away by the extraction pump through the branch 8. The exhaust steam enters at 9, is deflected round the cone 6, and flows across and upward through the water from 4 to the central opening 5, then outward and upward through the water from tray 3, then over the edges of this tray, through the water from tray 2, and the final mixture of air and vapour is drawn away by the air pump through the branch 10. It will be noted that in this case the steam meets the water, which has been considerably raised in temperature and the final temperature of the mixture of condensing water and condensate approximates to the vacuum temperature $t_v$. The difference $(t_v - t_2)$ is thus less than in the case of the parallel flow type. The air-vapour temperature is also much lower, an important condition with regard to the capacity of the dry air pump required.

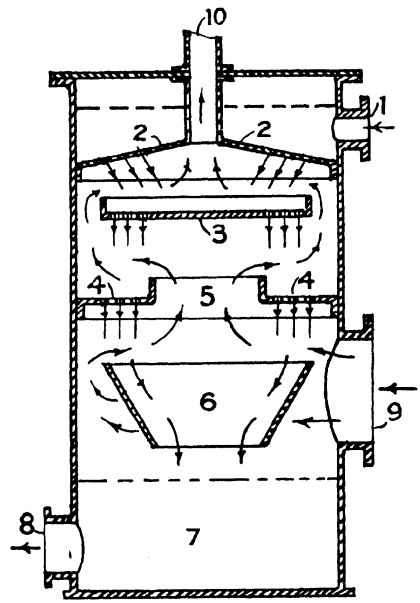


FIG. 453.

**408. (*b*) High-level or Barometric Jet Condenser.**—The low-level type, usually parallel flow, is placed on a level with the engine or below the turbine, the injection water being raised by suction, while the mixture of condensing water and condensate is withdrawn either by an air pump or an extraction pump. The barometric type, on the other hand, is usually contra-flow, and is mounted on a platform in an elevated position, well in excess of the barometric

head. The injection and condensate mixture flows by gravity down a tail pipe or barometric leg to the pond, and no extraction pump is necessary ; but as no suction head is available, the injection water has to be pumped up to the condenser inlet.

A typical modern arrangement of barometric condenser, by Worthington-Simpson, Ltd., is shown in Fig. 454.

The condenser shown in section in Fig. 455 is of the contra-flow type having the same internal tray arrangement as shown in the

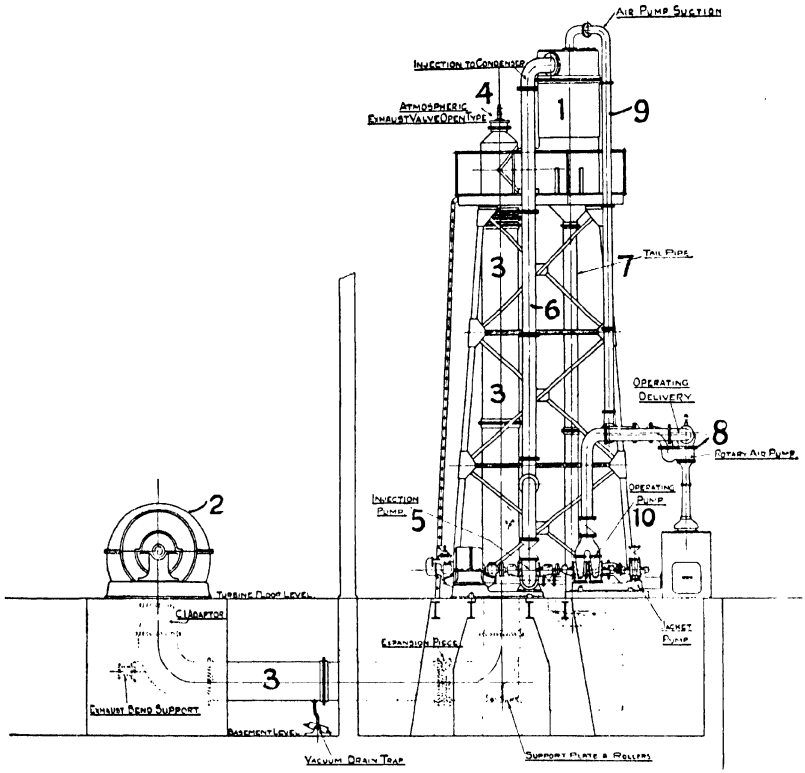



FIG. 454.

contra-flow low-level condenser, Fig. 453. This condenser, however, is formed with a coned bottom to which a barometric leg is attached. The exhaust steam enters at 1, the injection water at 2, the mixture of condensing water and condensate flows downward through 3, while the air and vapour are drawn away at 4.

In Fig. 454, 1 is the condenser, taking the exhaust of a steam turbine 2 through the pipe 3, which is also provided with an atmospheric exhaust valve 4. A motor-driven centrifugal pump 5 discharges the injection water, drawn from the pond, through the

injection water pipe 6, and the water is carried away through the tail pipe 7. The air and vapour are drawn from the top of the condenser by a rotary type of air pump 8, through the air pipe 9. The operating water for the rotary air pump is supplied by a centrifugal pump 10.

**409.** (*c*) **Ejector Condensers.**—The ejector condenser is a condensing apparatus of very simple construction and can be installed at a low cost. Like the simpler forms of jet condenser, it is suited to take the exhaust of high-speed reciprocators which do not require a high vacuum, and it does not require an air pump. Unless,

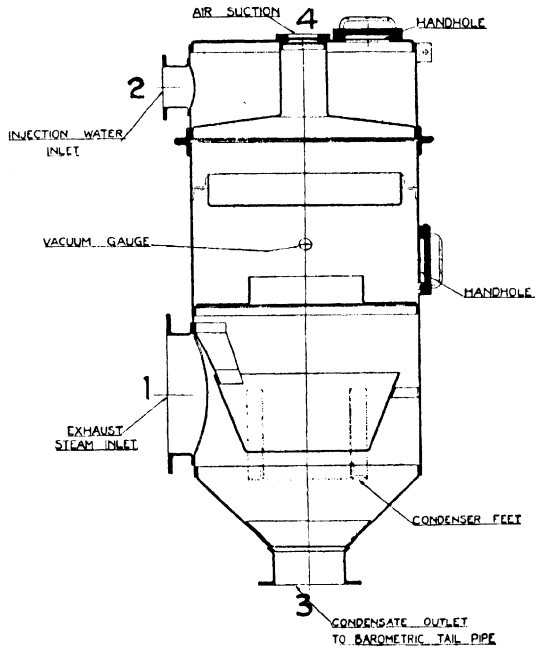



FIG. 455.

however, a natural head of water is available, some form of supply pump is required to provide the condensing water. This is usually a centrifugal pump which can maintain a steady head, from 15 ft. to 20 ft., at the injection nozzle. Where a reciprocating pump is used it is better to raise the water to an overhead tank, from which it can be supplied under a constant head.

A section through a single or central jet type of ejector condenser by Ledward and Beckett is shown in Fig. 456, and the arrangement as fitted to take the exhaust of a reciprocator in Fig. 457. Here the water is supplied from an overhead tank. In more recent designs the water enters directly above the injection nozzle, instead of at

the side as shown in Fig. 456. This water, due to the supply head plus the vacuum head in the condenser, issues from the supply cone with a considerable velocity, and in its downward passage condenses the exhaust steam flowing into the steam cones, as indicated by the arrows. It also entrains and carries the air along with it. After passing through the cones it enters the coned diffuser pipe, in which its pressure is raised, due to the decreased velocity in the expanding cone, to slightly more than atmospheric at exit. It is then discharged to the sump, as indicated in Fig. 457.

This single jet arrangement is at a disadvantage compared with the jet condenser, in which the water is broken up into a spray to expose a large surface to the steam. To remedy this defect the more recent practice is to employ a multi-jet type.

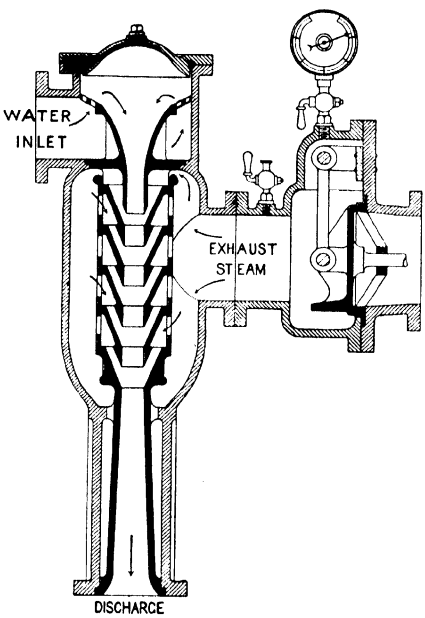


FIG. 456.

A section through one of these improved designs, by Koerting Bros., is shown in Fig. 458. The exhaust steam enters at 1; the condensing water is supplied at 2, and flows into the annular space 3, provided at the bottom with a ring of nozzles 4 having an inward direction. It is obvious that with a series of streams projected from these nozzles, as shown, a much greater water surface is exposed to the steam and air than is the case with an equivalent single jet, and more effective condensation results. The steam cones 5 gradually increase in diameter from the bottom of the condenser upward. In this way the area of the steam passage is proportioned to suit the volume flowing, the steam velocity into the cones is kept sensibly constant from top to bottom; and the vacuum drop between the cones and the exhaust chamber 6 is reduced to a minimum. The series of jets, as indicated, converge into a single jet at the throat 7 of the diffuser 8, in which the pressure is raised to the atmospheric value on discharge at the outlet 9.

An automatic vacuum breaker is bolted to the branch 10. If the water rises in the condenser the float of this apparatus, through a relay valve, operates the air valve, which admits air to the exhaust branch and breaks the vacuum. This prevents flooding of the engine by suction of the water back from the sump.

**410. Quantity of Injection Water for Jet Condensers.—**

Let $W_s$ =wt. of exhaust steam at vacuum pressure $p_v$, in lb./hr.

$q_v$ =dryness fraction of exhaust steam at vacuum pressure $p_v$.

$L_v$ =latent heat of exhaust steam at vaccuum pressure $p_v$, in B.Th.U./lb.

$t_v$ =vacuum temperature and $h_v$=water total heat at $t_v$.

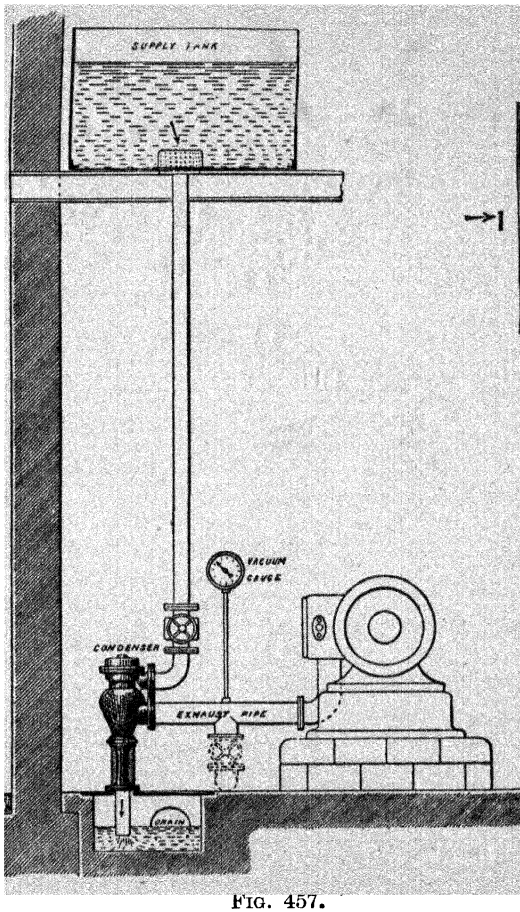



FIG. 457.

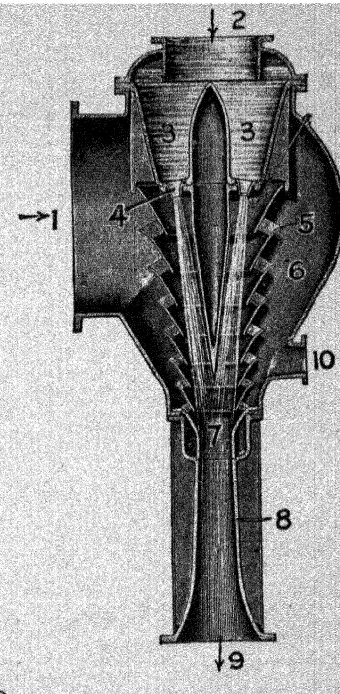



FIG. 458.

$W_w$=weight of injection water in lb./hr.

G =quantity of injection water $\frac{W_w}{600}$ gall./min.

$t_1$ =inlet temperature of injection water, $h_1$=water total heat.

$t_2$ =final temperature of injection water and condensate, $h_2$= water total heat.

Then—

heat from steam=heat to water

$$W_s(h_v+q_vL_v-h_2)=W_w(h_2-h_1)=\frac{G}{600}(h_2-h_1).$$

$$\therefore G=\frac{W_s(h_v+q_vL_v-h_2)}{600(h_2-h_1)} \text{ gall./min.} \quad . \quad . \quad . \quad (1)$$

At the low temperatures obtaining, the temperature values may be substituted for the water heats and (1) may be written—

$$G=\frac{W_s(t_v-t_2+q_vL_v)}{600(t_2-t_1)} \text{ gall./min.} \quad . \quad . \quad . \quad . \quad (2)$$

Obviously for a minimum quantity of injection water, $(t_2-t_1)$ should be kept as great as possible. On account of the operation of Dalton's law of partial pressures, although the total pressure in the condenser is $p_v$, the vapour pressure due to the presence of air is less, and $t_2<t_v$. The quantity $(t_v-t_2)$ in (2) may vary considerably, depending on the entrained air in the injection water and that carried in with the steam. Under normal conditions the value may run from 6° Fahr. to 8° Fahr. with reciprocators, and to 4° Fahr. with turbines.

Another method of estimating the injection water is to first find the heat given up by the exhaust steam, subtracting the heat per lb. converted to work, the radiated heat, and the final water heat of the condensate from the initial total heat $H_{s1}$ at the initial pressure $p_1$ at $t_{s1}$. The radiated heat is small and may be neglected.

If $w_i$=indicated steam consumption in lb./I.H.P. hr.

$w_k$=indicated steam consumption in lb./kw.-hr.

Heat rejected by exhaust steam per lb. is—

$$h=\left(H_{s1}-\frac{2545}{w_i}-h_2\right)$$

or $$h=\left(H_{s1}-\frac{3412}{w_k}-h_2\right)$$

Hence
$$G=\frac{W_s\left(H_{s1}-\frac{2545}{w_i}-h_2\right)}{600(t_2-t_1)} \text{ gall./min.}$$

or
$$G=\frac{W_s\left(H_{s1}-\frac{3412}{w_k}-h_2\right)}{600(t_2-t_1)} \text{ gall./min.} \quad . \quad . \quad . \quad (3)$$

In the case of a contra-flow jet condenser, owing to the vacuum and temperature drop between steam inlet and air outlet, there is a vapour-condensing and air-cooling effect which is not accounted for by Equations 2 and 3, and the value of G is thus on the low side.

The calculation of this additional heat given to the water, which involves the estimation of vapour condensed and air cooled, is

tedious, and as this heat is only a small fraction of the total amount and depends on several assumptions, it is not proposed to detail it at this stage. It is considered later, under surface condensers. Allowance can be made by a slight increase in the calculated value of G.

Under the usual conditions of modern practice with steam turbines the value of the heat given up by the exhaust steam to the water may run from 930 to 950 B.Th.U./lb. In order to allow a sufficient margin for contingencies, as indicated above, it is customary for designers to adopt a standard value of 1000 B.Th.U./lb., and on this basis the approximate value of the amount of injection water is given by the simpler expression—

$$G=\frac{W_s}{0{\cdot}6(t_2-t_1)} \text{ gall./min.} \quad . \quad . \quad . \quad . \quad . \quad (4)$$

*Example* 1.—A turbine developing 1500 kw. on a consumption of 13 lb./kw.-hr. is supplied with steam at 180 lb./in.$^2$ abs. and 480° Fahr. It exhausts to a jet condenser in which a vacuum of 28 in. (30 in. bar.) is to be maintained. The inlet temperature of the injection water is 70° Fahr. and its final temperature is 6° Fahr. lower than the vacuum temperature. Calculate the amount of injection water required.

Here $p_1=180$, $H_{s1}=1264$, $w_k=13$, $\frac{3412}{w_k}=\frac{3412}{13}=262$, $p_v=1$ lb./in.$^2$ abs., $t_v=101°$ Fahr., $t_2=(t_v-6)=95°$ Fahr., $h_2=63{\cdot}8$, $W_s=13\times1500=19{,}500$, $t_1=70$, $(t_2-t_1)=25$.

Hence from (3)—

$$G=\frac{W_s\left(H_{s1}-\frac{3412}{w_k}-h_2\right)}{600(t_2-t_1)}$$

$$=\frac{19500(1264-262-63{\cdot}8)}{600\times25}=\frac{938{\cdot}2\times19500}{15000}=1220 \text{ gall./min.}$$

If a dryness fraction $q_v=0{\cdot}9$ is assumed, then by (2)—

$L_v=1033$
$W_s=19500$

$$G=\frac{(t_v-t_2+q_vL_v)W_s}{600(t_2-t_1)}$$

$$=\frac{(6+0{\cdot}9\times1033)19500}{600\times25}=\frac{935{\cdot}7\times19500}{15000}=1216 \text{ gall./min.}$$

By the approximate rule (4)—

$$G=\frac{W_s}{0{\cdot}6(t_2-t_1)}=\frac{19500}{0{\cdot}6\times25}=1300 \text{ gall./min.}$$

which is 8 per cent. greater than the first value.

**411. Diameter of the Injection Water Pipe of Low-level Condensers.**—The head of water corresponding to the normal barometer of 30 in. or 14·7 lb./in.$^2$ abs. is $2{\cdot}3\times14{\cdot}7=33{\cdot}81$ ft.

Hence if the vacuum in the condenser is $x$ inches of mercury the corresponding vacuum head in feet is—

$$H_v = \frac{33{\cdot}81}{30} x = 1{\cdot}127x \text{ ft.} \quad . \quad . \quad . \quad . \quad . \quad (5)$$

Let H = the statical head or vertical distance from the pond level to the discharge level at the condenser inlet, in ft.

$H_f$ = frictional head in the injection pipe, in ft.

$H_k$ = the pressure head at entrance to the condenser, in ft.

Then $$H_v = H + H_f + H_k \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

By the standard hydraulic formula the friction head is given by—

$$H_f = \frac{4flV^2}{2gd} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

When V = velocity in the injection pipe, in ft./sec.

$d$ = internal diameter of the pipe, in ft.

$l$ = virtual length of the pipe, in ft., or the straight length plus allowances for bends, knees, etc.

Substituting in (6) the pressure head at entry is—

$$H_k = H_v - H - H_f$$
$$= 1{\cdot}127x - H - \frac{4flV^2}{2gd} \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

The virtual length $l$, which may be much greater than the actual length, cannot be ascertained unless $d$ is known, nor can (8) be solved directly.

The most convenient method is to choose a reasonable value of the water velocity (V) and from the given flow (G) to determine the diameter $d$ provisionally.

It is easy to show that if G is in gall./min. and $d$ in inches that $V = \frac{0{\cdot}51G}{d^2}$ ft./sec., so that—

$$d = 0{\cdot}71\sqrt{\frac{G}{V}} \text{ in.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

V may be taken from 4 to 6 ft./sec. It should not exceed 8 ft./sec. $d$ being fixed in ft., $l$ can then be obtained by adding to the straight pipe length from 10 to 15 times the diameter for each easy right-angled bend, and 30 to 36 times for each sharp right-angled bend.*

The value of the coefficient of friction $f$ may be taken as 0·01 for cast-iron pipes.

The value of the head at inlet can be alternatively obtained in terms of the flow G gall./min. The equation has the form—

$$H_k = 1{\cdot}127x - H - \frac{lG^2}{139{\cdot}65 \times 10^6 d^5} \quad . \quad . \quad . \quad . \quad (10)$$

where $l$ and $d$ are in feet. The third term on the right is the friction head ($H_f$). Equation 8 is more the convenient one.

* See *Steam Condensing Plant*, by J. Sim, B.Sc.

In the case of the parallel flow condenser, for a given pressure head $H_k$, the total nozzle area, A in.$^2$, is calculable for a given flow of G gall./min.

Here $$\frac{10G}{60}=62{\cdot}5C_d\frac{A}{144}\sqrt{2gH_k} \quad . \quad . \quad . \quad . \quad . \quad (11)$$

and $$A=\frac{G}{2{\cdot}6C_d\sqrt{2gH_k}} \text{ in.}^2 \quad . \quad . \quad . \quad . \quad . \quad (12)$$

The nozzle coefficient of discharge $C_d$ may be taken between 0·85 and 0·95.

For effective spraying at the nozzles, the head $H_k$ should not be less than 10 ft.

In the case of the contra-flow condenser, in which the water simply flows by gravity through the perforated trays, after discharge into and dispersal over the area of the condenser the major part of this head is dissipated.

*Example* 2.—The injection water supplied to a parallel flow jet condenser is 480 gall./min., and the vacuum is 28 in. The statical head is 10 ft., the length of injection pipe is 50 ft., and there are five easy bends on it. If the velocity is limited to 5 ft./sec. and an allowance of 15 diameters per bend is made, calculate a suitable diameter of injection pipe, the pressure head behind the nozzles, and the total nozzle area required. Take $C_d=0{\cdot}9$.

Pipe diameter—

$$d=0{\cdot}71\sqrt{\frac{G}{V}} \qquad \begin{matrix} G=480 \\ V=5 \end{matrix}$$

$$d=0{\cdot}71\sqrt{\frac{480}{5}}=7 \text{ in.}=0{\cdot}583 \text{ ft.}$$

Then $$l=50+\frac{5\times15\times7}{12}=50+43{\cdot}7=93{\cdot}7 \text{ ft.}$$

$$H_k=1{\cdot}127x-H-\frac{4flV^2}{2gd} \qquad \begin{matrix} x=28 \\ H=10 \end{matrix}$$

$$=1{\cdot}127\times28-10-\frac{4\times0{\cdot}01\times93{\cdot}7\times25}{64{\cdot}4\times0{\cdot}583}$$

$$=31{\cdot}56-10-2{\cdot}3=19{\cdot}26 \text{ ft.}$$

This is a fairly high value, and the static head or height from pond level to condenser inlet, might be increased, say, to 16 ft. if necessary, without adverse effect on the nozzle spray.

Nozzle area—

$$A=\frac{G}{2{\cdot}6Cd\sqrt{2gH_k}} \qquad \begin{matrix} C_d=0{\cdot}9 \\ G=480 \end{matrix}$$

$$=\frac{480}{2{\cdot}6\times0{\cdot}9\sqrt{64{\cdot}4\times19{\cdot}26}}$$

$$=\frac{480}{2{\cdot}6\times0{\cdot}9\times35{\cdot}2}=5{\cdot}85 \text{ in.}^2$$

The nozzle area here is about 16 per cent. of the pipe area. The velocity of flow from the nozzles is 35·2 ft./sec.

*Example* 3.—A contra-flow jet condenser takes the exhaust of a compound engine at a vacuum of 26 in. The lift from the pond level is 12 ft., the length of the injection pipe is 300 ft., and there are five easy bends, for which 15 diameters may be allowed. The injection flow is 455 gall./min. Assuming a water velocity of 6 ft./sec., estimate a suitable pipe diameter, and ascertain the head at entry.

Here $d=0{\cdot}71\sqrt{\frac{G}{V}}=0{\cdot}71\sqrt{\frac{455}{6}}$ $\quad G=455,\ V=6$

$=6{\cdot}16$ in. $=0{\cdot}513$ ft.

$$l=300+\frac{15\times5\times6{\cdot}16}{12}=300+38{\cdot}5=338{\cdot}5$$

$$H_k=1{\cdot}127x-H-\frac{4flV^2}{2gd} \qquad x=26,\ H=12$$

$$=1{\cdot}127\times26-12-\frac{4\times0{\cdot}01\times338{\cdot}5\times36}{64{\cdot}4\times0{\cdot}513}$$

$=29{\cdot}3-12-14{\cdot}7=2{\cdot}6$ ft.

The major part of this head will be expended in eddy formation at entrance to the condenser.

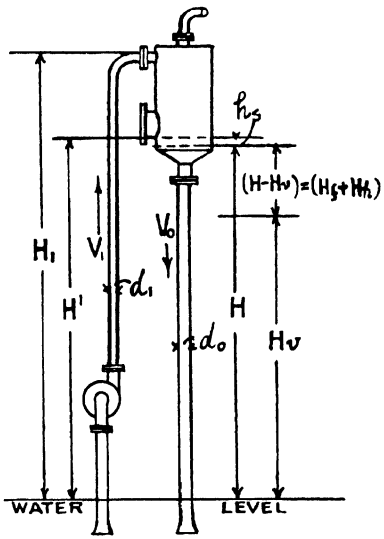



FIG. 459.

**412. Diameter of Tail Pipe of Barometric Condenser.** — Taking the usual case of the contra-flow condenser and referring to Fig. 459.

Let H = distance from pond level to water level in condenser, in ft.

$H_v$ = head due to vacuum, in ft.

$H_f$ = friction head in tail pipe, in ft.

$H_k$ = kinetic head producing downward velocity of water $V_0$, in ft.

$x$ = vacuum in inches of mercury.

$d_0$ = bore of tail pipe, in ft.

$l$ = virtual length of tail pipe, in ft.

Then $$H-H_v=H_f+H_k,$$

or $$H-1{\cdot}127x=\frac{4flV_0^2}{2gd_0}+\frac{V_0^2}{2g} \quad . \quad . \quad . \quad . \quad . \quad (13)$$

As in the case of the parallel flow condenser, a suitable value of

the water velocity $V_0$ can be chosen, from 4 to 6 ft./sec., and the pipe diameter calculated from—

$$d_0=0{\cdot}71\sqrt{\frac{G'}{V_0}} \text{ in.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (14)$$

where G′=combined condensing water and condensate, in gall./min. In order to approximate to the virtual length when allowance is made for any bends necessary, the actual length of the tail pipe may be provisionally assumed equal to the full barometric height of 34 ft. When $l$ and $d_0$ are thus fixed, the value of the height H is calculable from Equation 13. In order to allow for fluctuation of vacuum a safety margin or height ($h_s$) has to be provided, between the low water level and the exhaust pipe inlet; and the height from the pond level to the bottom of the steam inlet branch is—

$$H'=H+h_s \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (15)$$

A margin of from $1\frac{1}{2}$ to 2 ft. may be allowed.

**413. Diameter of the Injection Pipe.**—The water has to be raised by the injection pump a height $H_1$. The effective lift against gravity, however, is only $(H_1-H_v)$, and to this has to be added the friction head $H_f'$ in the injection pipe. If $H_k'$ is the head at entrance, then the total head is—

$$H_p=H_1-H_v+H_f'+H_k' \quad . \quad . \quad . \quad . \quad . \quad (16)$$

With a given value of H′ the height $H_1$ is determined from the condenser dimensions. As before, a provisional value of $V_1$, the injection water velocity, can be chosen and the injection pipe diameter $d_1$ can be obtained from Equation 9.

**Power Required to Drive the Injection Water Pump.**—The total head against which the pump has to work is—

$$H_p=H_1-H_v+H_f'+H_k'$$

The pump horse-power required to raise $W_w$=10G lb. water/min. is—

$$\text{P.H.P.}=\frac{W_wH_p}{33000}=\frac{GH_p}{3300} \quad . \quad . \quad . \quad . \quad . \quad (17)$$

If $\eta_p$ is the efficiency of the pump, the power to be supplied by the electric motor or other driving agent is—

$$\text{M.H.P.}=\frac{W_wH_p}{\eta_p 33000} \quad . \quad . \quad . \quad . \quad . \quad . \quad (18)$$

This is the power under normal vacuum ($x$). When the plant starts up $x$=0, and a considerable overload has to be taken by the driving agent while the vaccuum is being "built up," and also on any occasion on which the vacuum may fall off considerably. Provision for a much larger input than the normal must thus be made when fixing the power of the driving agent.

A similar calculation can be made for the power of the water extraction pump of the low-level type. In this case the head to be overcome is (the vaccum head $H_v$+the static or delivery head H

from the pump to the level at which the water is to be lifted+the friction head $H_f$ in the pipe line), less the small head $h$ above the pump suction in the well 7, Fig. 453.

Thus $$H_p=(H_v+H+H_f-h) \text{ ft.} \quad . \quad . \quad . \quad . \quad . \quad (19)$$

*Example* 4.—In a contra-flow barometric jet condenser the normal vacuum is to be 26 in. It is estimated that the injection water required for the condensation of 134 lb. steam/min. will be 230 gall./min. Allowing a water velocity of 5 ft./sec. and for two easy bends on the tail pipe, find a suitable diameter of pipe and the height from the pond level to the bottom of the steam inlet, so that there may be a safety margin of 2 ft.

Here $G'=230+13{\cdot}4=243{\cdot}4$ gall./min. ; $V_0=5$,

and $d_0=0{\cdot}71\sqrt{\frac{G'}{V_0}}=0{\cdot}71\sqrt{\frac{243{\cdot}4}{5}}=4{\cdot}955$, say 5 in.$=0{\cdot}4127$ ft.

Tail pipe length $l=34+\frac{2\times15\times5}{12}=34+12{\cdot}5=46{\cdot}5$ ft.

$$H=1{\cdot}127x+\frac{4flV_0^2}{2gd_0}+\frac{V_0^2}{2g} \qquad h_s=2 ;\ x=26$$

$$=1{\cdot}127\times26+\frac{4\times0{\cdot}01\times46{\cdot}5\times25}{64{\cdot}4\times0{\cdot}4127}+\frac{25}{64{\cdot}4}$$

$$=29{\cdot}3+1{\cdot}75+0{\cdot}398=31{\cdot}5 \text{ ft.}$$

and $$H'=H+h_s=31{\cdot}5+2=33{\cdot}5 \text{ ft.}$$

*Example* 5.—In the previous example the distance between the bottom of the steam inlet branch and the water level above the injection inlet is 7·0 ft. Allowing an injection water velocity of 5 ft./sec. and for three bends on the pipe having an actual length of 60 ft., calculate a suitable pipe diameter and the power required by the injection water pump : (*a*) under conditions of normal vacuum ; (*b*) when there is no vacuum (under starting-up conditions). Take $\eta_p=0{\cdot}65$.

Here $G=230$ gall./min. and $V_1=5$ ft./sec.

$$d_1=0{\cdot}71\sqrt{\frac{G}{V_1}}=0{\cdot}71\sqrt{\frac{230}{5}}=4{\cdot}81 \text{ in.}=0{\cdot}4 \text{ ft.}$$

$$l=60+\frac{3\times15\times4{\cdot}81}{12}=60+18{\cdot}1=78{\cdot}1 \text{ ft.}$$

(*a*) Pumping head—

$$H_p=H_1-H_v+H_f'+H_k'$$
$$=40{\cdot}5-29{\cdot}3+3+0{\cdot}398$$
$$=14{\cdot}6 \text{ ft.}$$

$$\text{M.H.P.}=\frac{W_w H_p}{\eta_p 33000}=\frac{2300\times14{\cdot}6}{0{\cdot}65\times33000}$$
$$=1{\cdot}56$$

$$H_1=33{\cdot}5+7=40{\cdot}5$$

$$H_f'=\frac{4\times0{\cdot}01\times78{\cdot}1\times25}{64{\cdot}4\times0{\cdot}4}=3$$

$$H_k'=\frac{25}{64{\cdot}4}=0{\cdot}398$$

$$W_w=2300$$

$$\eta_p=0{\cdot}65$$

(*b*) With $x=0$, $H_v=0$
and $H_p=43{\cdot}9$

$$\text{M.H.P.}=\frac{1{\cdot}56\times 43{\cdot}9}{14{\cdot}6}=4{\cdot}7$$

**414. Capacities of Jet Condensers.**—Sim states that in the case of the high-speed reciprocator the condenser capacity may be made about one-third of the volume of the cylinder exhausting into it, and for low speed from one-quarter to one-half. In turbines the capacity can be based on the rate of steam consumption, and taken from 0·008 to 0·001 ft.³/lb. steam hr. The size of the exhaust inlet branch can be determined by the equation of continuity, for exhaust steam velocity from 300 to 400 ft./sec. In the case of the cylindrical parallel or contra-flow condenser, the length is usually about two and a half times the diameter.

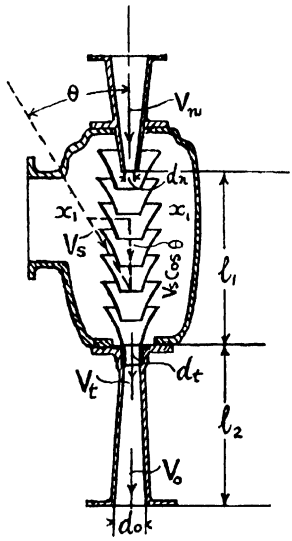


FIG. 460.

**415. Sizes of Injection Nozzle and Diffuser of the Ejector Condenser.**—This calculation is necessarily very tentative, as knowledge of the exact actions occurring in the steam cones is lacking.

Let H=head of water above the nozzle exit in ft. (15 to 20 ft.).
$x_1$=vacuum in the condenser in inches of mercury.
$x$ =vacuum in the steam cones.

The effective head producing flow of the water is—

$$H'=(H+1{\cdot}127x)\text{ ft.} \quad . \quad . \quad . \quad . \quad . \quad (20)$$

If $\eta_w$=the water nozzle efficiency, the velocity of exit is—

$$V_n=\sqrt{2g\eta_w H'}\text{ ft./sec.} \quad . \quad . \quad . \quad . \quad . \quad (21)$$

This may vary from 50 to 60 ft./sec.

Let $W_w$=weight of condensing water in lb./min., then the necessary diameter of nozzle outlet is given by—

$$d_n=0{\cdot}2212\sqrt{\frac{W_w}{V_n}}\text{ ins.} \quad . \quad . \quad . \quad . \quad . \quad (22)$$

Assume that the steam cone passages are proportioned to give a sensibly constant inlet velocity $V_s$, Fig. 460, at the jet angle $\theta$ to the axis and a component $V_s\cos\theta$ axially. The constant heat drop $h_\phi$ corresponding to $V_s$ depends on the difference between the vacuum $x_1$ and the vacuum $x$ produced at the inner edges of the

steam cones. This is not definite, but for the purpose in view it may be taken as 0·5 in. of mercury.

Then $$V_s=223{\cdot}7\sqrt{\eta_n h_\phi} \quad . \quad . \quad . \quad . \quad . \quad . \quad (23)$$

where $\eta_n$ is the nozzle efficiency for the cones (say 0·9). $V_s$ may run from 500 to 900 ft./sec. according to the value of the condenser vacuum $x_1$.

The action occurring between the nozzle exit and the diffuser throat is a complex one, but some degree of approximation to the velocity of the combined mass of condensate, injection water, and air vapour at the diffuser can be obtained by imagining the water to flow as a central core with a smaller quantity of condensate and entrained air moving more rapidly outside it, as a surrounding envelope, and that the two streams mingle, with considerable shock effect, at the entrance to the diffuser throat.

The injection water falls through an additional head $l_1$ so that the total flow head at the throat of the diffuser is $(\eta_w H'+l_1)$ and the velocity is—

$$V_n'=\sqrt{2g(\eta_w H'+l_1)} \quad . \quad . \quad . \quad . \quad . \quad (24)$$

Small portions of the condensate also fall distances from $o$ to $l_1$, and an average distance $l_1/2$ may be assumed for the whole mass of condensate and air weighing W lb. The probable condensate velocity at the diffuser inlet may then be taken as—

$$V_w=\sqrt{(V_s \cos\theta)^2+gl_1} \quad . \quad . \quad . \quad . \quad . \quad (25)$$

Applying the principle of the conservation of momentum, under ideal conditions the theoretical velocity of the mixture entering the throat would be—

$$V=\frac{W_w V_n'+wV_w}{W_w+w} \text{ ft./sec.}$$

or denoting the ratio of water to condensate and air by—

$$Z=\frac{W_w}{w}=\frac{W_w}{w_s}$$

where $w_s$=weight of condensate,

$$V=\frac{(ZV_n'+V_w)}{(Z+1)} \quad . \quad . \quad . \quad . \quad . \quad . \quad (26)$$

This has to be reduced by a shock coefficient of fairly low value, say 0·65 ; and the probable velocity at the throat is taken as—

$$V_t=0{\cdot}65V \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (27)$$

To determine the outlet velocity $V_0$ under atmospheric pressure, the combined mass falls a further distance $l_2$, and if the diffuser efficiency is taken as $\eta_d$ (from 0·8 to 0·9), the available energy for discharge is—

$$\frac{V_0^2}{2g}=\eta_d\left(\frac{V_t^2}{2g}-1{\cdot}127x+l_2\right). \quad . \quad . \quad . \quad . \quad (28)$$

where $x$ is the vacuum at entrance to the throat or inside the cones. Neglecting the weight of entrained air, the diameters at throat and outlet are obtained from Equation 22. Thus—

Throat of diffuser $$d_t=0{\cdot}2212\sqrt{\frac{W_w+w_s}{V_t}} \quad . \quad . \quad . \quad . \quad . \quad (29)$$

Outlet of diffuser $$d_0=0{\cdot}2212\sqrt{\frac{W_w+w_s}{V_0}} \quad . \quad . \quad . \quad . \quad . \quad (30)$$

where $W_w$ and $w_s$ are in lb./min., and—

$$\frac{d_0}{d_t}=\sqrt{\frac{V_t}{V_0}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (31)$$

Equation 2 can be used for the estimation of the injection water $W_w=10G$, the vacuum $x$ in the steam cones being taken. As already stated, the difference between the vacuum and the water temperatures is much greater than in the parallel flow jet condenser. This may run from 15° Fahr. to 25° Fahr. A lower value can be used with a multi-jet arrangement than with the single jet. In determining the nozzle dimensions of a multi-jet condenser, the foregoing calculation is used for an equivalent single jet, and the total throat area of the nozzles is made equal to this value. The small inclination of the jets to the axis may be neglected.

*Example* 6.—An ejector condenser is to maintain a vacuum of 28 in. and condense 25,000 lb. steam/hour. It is to be supplied with water at 60° Fahr. under a 20 ft. head. Assuming a difference of 15° Fahr. between the vacuum and final water temperatures, a height of 4 ft. over the steam cones, and a diffuser length of 3 ft., determine suitable values for the diameters of the injection nozzle and the throat and outlet of the diffuser. Assume the water nozzle, steam cone, and diffuser efficiencies to be 0·9, and steam jet angle $\theta=30°$, and quality at entrance as 0·9.

Condenser vacuum $x_1=28$ in.; steam cone vacuum $x=28{\cdot}5$ in.; $p_{v1}=0{\cdot}98$ and $q_{v1}=0{\cdot}9$; expanding from $p_{v1}$ to $p_v=0{\cdot}735$; $q_v=0{\cdot}89$, and the heat drop is $h_\phi=16$ B.Th.U./lb.

The actual value of $q_v$ will be slightly greater, but this difference may be neglected.

$t_v=92°$ Fahr., $(t_v-t_2)=15$ $\quad\therefore\ t_2=77°$ Fahr. also $t_1=60°$ Fahr.

Then $$G=W_s\frac{(t_v-t_2+q_vL_v)}{600(t_2-t_1)} \qquad \begin{matrix} W_s=25{,}000 \\ L_v=1038 \end{matrix}$$

$$=25000\frac{(15+0{\cdot}89\times1038)}{600\times17}=2300 \text{ gall./min.}$$

$\therefore\ W_w=23{,}000$ lb./min.

$$w_s=\frac{25000}{60}=416{\cdot}6 \text{ lb./min.}$$

$$\therefore\ Z=\frac{W_w}{w_s}=\frac{23000}{416{\cdot}6}=55{\cdot}2$$

Head at nozzle—

$$H' = 1{\cdot}127x + H \qquad H = 20$$
$$= 1{\cdot}127 \times 28{\cdot}5 + 20 = 52{\cdot}2 \text{ ft.} \qquad x = 28{\cdot}5$$

Water velocity—

$$V_n = \sqrt{2g\eta_w H'}$$
$$= \sqrt{64{\cdot}4 \times 0{\cdot}9 \times 52{\cdot}2} = 55 \text{ ft./sec.}$$

and nozzle diameter—

$$d_n = 0{\cdot}2212\sqrt{\frac{23000}{55}} = 4{\cdot}5 \text{ in.}$$

Also $V_n' = \sqrt{64{\cdot}4(0{\cdot}9 \times 52{\cdot}2 + 4)} = 58{\cdot}2$ ft./sec.

$\theta = 30°$, $\cos\theta = 0{\cdot}866$.

$$V_s = 223{\cdot}7\sqrt{\eta_n h_\phi} = 223{\cdot}7\sqrt{0{\cdot}9 \times 16} = 850 \text{ ft./sec.}$$

and $$V_w = \sqrt{(V_s \cos\theta)^2 + gl_1}$$
$$= \sqrt{(850 \times 0{\cdot}866)^2 + 64{\cdot}4 \times 4} = 737 \text{ ft./sec.}$$

$$V = \frac{ZV_n' + V_w}{Z+1} = \frac{55{\cdot}2 \times 58{\cdot}2 + 737}{56{\cdot}2} = \frac{3917}{56{\cdot}2} = 69{\cdot}6 \text{ ft./sec.}$$

and $V_t = 0{\cdot}65 \times V = 0{\cdot}65 \times 69{\cdot}6 = 45{\cdot}2$ ft./sec.

$$\frac{V_0^2}{2g} = \eta_d\left(\frac{V_t^2}{2g} - 1{\cdot}127x + l_2\right) \qquad \begin{matrix} x = 28{\cdot}5 \\ l_2 = 3 \\ \eta_d = 0{\cdot}9 \end{matrix}$$
$$= 0{\cdot}9\left(\frac{(45{\cdot}2)^2}{64{\cdot}4} - 1{\cdot}127 \times 28{\cdot}5 + 3\right) = 0{\cdot}9 \times 2{\cdot}4 = 2{\cdot}16$$

$$\therefore V_0 = \sqrt{64{\cdot}4 \times 2{\cdot}16} = 11{\cdot}8 \text{ ft./sec.}$$

Throat diameter of diffuser—

$$d_t = 0{\cdot}2212\sqrt{\frac{W_w + w_s}{V_t}} = 0{\cdot}2212\sqrt{\frac{23416{\cdot}6}{45{\cdot}2}} = 5 \text{ in.}$$

Outlet diameter of diffuser—

$$d_0 = d_t\sqrt{\frac{V_t}{V_0}} = 0{\cdot}5\sqrt{\frac{45{\cdot}2}{11{\cdot}8}} = 9{\cdot}8 \text{ in.}$$

**416.** (*d*) **Surface Condensers.**—In the case of the surface condenser, the exhaust steam and the condensate are entirely separated from the condensing water.

The water flows through parallel series of brass tubes, averaging from $\frac{5}{8}$ in. to 1 in. diameter, the usual size being $\frac{3}{4}$ in. O.D. with wall thickness of 0·048 in. The tube length may run from 3 ft. to 22 ft. The steam is discharged into the condenser so as to flow transversely through the banks or nests of tubes. The condenser body or shell is made either of cast-iron or steel plating, and carries on its ends the water boxes with the inlet and outlet branches for the circulating water. The cross-section is of varied form.

Some makers adopt a circular, some a shallow cup shape, some a pear-shaped form, and one a form approximately rectangular.

In what until within recent years was regarded as a " normal " type of condenser the tubes were arranged in the " closed " form, that is, without any openings through them between the top and bottom of the condenser. This type of design is still in use for small condensers; but for the very large outfits used in power stations it is being replaced by various tube arrangements to enable the steam to penetrate readily through the tube sections. The reasons for these modifications are given later.

The tubes are fixed in brass tube plates, varying from 1 in. to $1\frac{1}{2}$ in. thick, by a ferrule gland packing which while maintaining a water-tight joint enables the tubes to expand under varying temperature without undue resistance and distortion.

The circulating water is admitted at the bottom of the con-

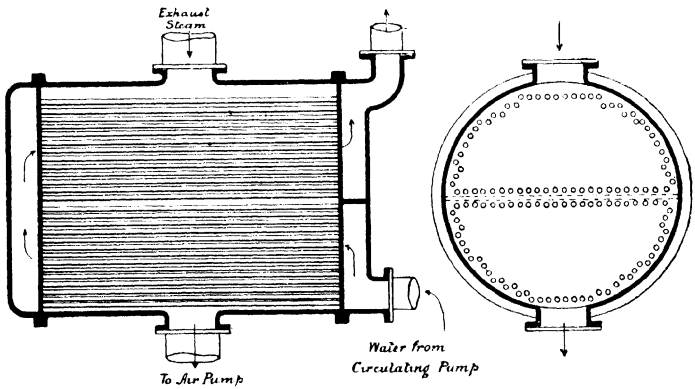



FIG. 461.

denser, flows through a " pass " of tubes to the other end, and is then deflected in the water box into a second pass, flows back through these tubes, and is discharged at the outlet branch, as illustrated in Fig. 461, which is a diagrammatic sketch of an old type of circular two-pass condenser with " closed " tube banks, as indicated on the cross-section. The mixture of air and vapour in this case was drawn away by a " wet " air pump at the bottom. The steam was discharged through an inadequate exhaust opening at the top, flowed transversely through the tubes and was progressively condensed. The lower tubes were not of much use, being drowned in water, and the condensate in contact with the lower pass tubes, containing the circulating water at the inlet temperature, was appreciably under-cooled. Further, there was a considerable vacuum drop between exhaust inlet and air pump suction. As long as this type of condenser was used for the reciprocator, for which a maximum vacuum of 26 in. was found to be the economic limit,

these drawbacks were not serious. The advent of the steam turbine demanding vacua from 29 in. to 29·5 in. ruled out these conditions.

In a modern high vacuum installation the condensate has to be drawn from the condenser at a temperature as close to the vacuum temperature as possible, and in some of the most recent designs this difference has been reduced almost to zero. In one type it is claimed that a condensate temperature one or two degrees in excess of the vacuum temperature (due to kinetic-regenerative effect) can be obtained. In a good normal type of condenser the vacuum drop should not exceed 0·2 in. mercury. In many instances it is as high as 0·5 in. In the latest " regenerative " types this is being reduced to a negligibly small amount.

While the condensate has to be drawn from the condenser at the highest possible temperature, the air, in order to reduce the air pump capacity to a minimum, has to be withdrawn at as low a temperature as possible. Obviously effective separation of the air from the condensate has to be secured, and the air and its associated vapour subjected to a final cooling effect. The condensate, which drains to the bottom of the condenser, is drawn away by an extraction pump, while the air and vapour mixture is extracted at a higher level by either a rotary air pump or steam-jet ejector pump, and on its way to the pump suction is passed over a section of the bottom bank of tubes through which the coldest circulating water passes. This section is blanked off from the main body of the tubes by means of a dividing " baffle " plate (see Fig. 404), and thus virtually acts as an air cooler and devaporiser. The separated steam and water streams in the condenser are in quasi-contra-flow in the normal type with closed tube banks, the steam at entry being condensed by the circulating water of raised temperature. In the regenerative types this condition is only partly operative, as the steam penetrates directly to the tubes of the lower passes.

As in the case of the steam turbine, it is not possible, owing to space limitations, to illustrate the various forms of high vacuum condensers now being manufactured by British, American, and Continental firms, and the descriptive matter is restricted to the designs by three representative British makers. The student can refer, for fuller descriptive information, to the special texts on condensing plant and the articles on steam turbines in the technical journals, quoted in the various footnotes.*

**417. Metropolitan-Vickers Central Flow Surface Condenser.**— A diagrammatic sketch of the cross-section of a normal type of condenser, with closed tube banks and integral cooler, consisting of a section of tubes separated by a diaphragm, is shown in Fig. 462. The exhaust steam entering the top at 1 has to flow through the entire nest of tubes, as shown by the directional arrow lines, and consequently a considerable " drop " in vacuum takes place between

* *Steam Condensing Plant*, by James Sim, B.Sc. *Condensing Plant*, by Kaula and Robinson. *Steam Condensing Plant*, by John Evans, M.Eng.

1 and 2, where the condensate collects, and this is increased at the air pump suction 4. The suction of the air pump connected to this branch draws the air and vapour through the cooler 3 ; part of the vapour is condensed and falls to 2. In order to reduce the drop to a minimum, that is, to maintain the vacuum uniform throughout the condenser, this arrangement is replaced by that shown diagrammatically in Fig. 463. The shell is made much larger in diameter than the tube plate, and a large volute is formed round the nest of tubes. The steam, as shown by the directional arrow lines, has thus access to the whole periphery, and the air and accompanying vapour is drawn away through the central passage 4. The length of steam path is reduced to about half of that in Fig. 462, and the area for steam admission is nearly doubled. This increased area lowers the steam velocity necessary for the flow, and with the shorter

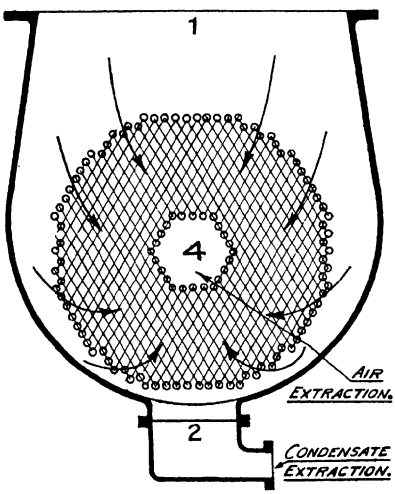



FIG. 463.

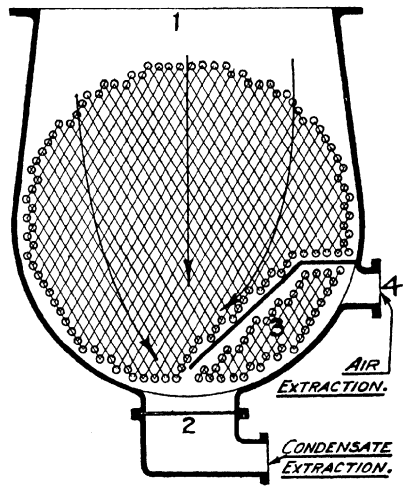



FIG. 462.

path, the drop across the tubes to the air-pump suction is reduced to from one-sixth to one-eighth of what it is in the normal arrangement of Fig. 462.

The practical application of this arrangement is shown in the cross and longitudinal sections of the M.V. Central Flow Condenser (Figs. 464 and 465).

The steam inlet 1 extends over the major part of the length and is slightly wider than the tube bank diameter. An even distribution of steam is thus ensured. The condensate, after collection at the bottom of the condenser, passes to a water-sealed chamber 2, the seal preventing direct flow of steam to the extraction pump. A section 3 of the tubes is isolated by means of the vertical baffle plates shown. These form a passage through which the air and vapour from the centre of the condenser is drawn into the nest of

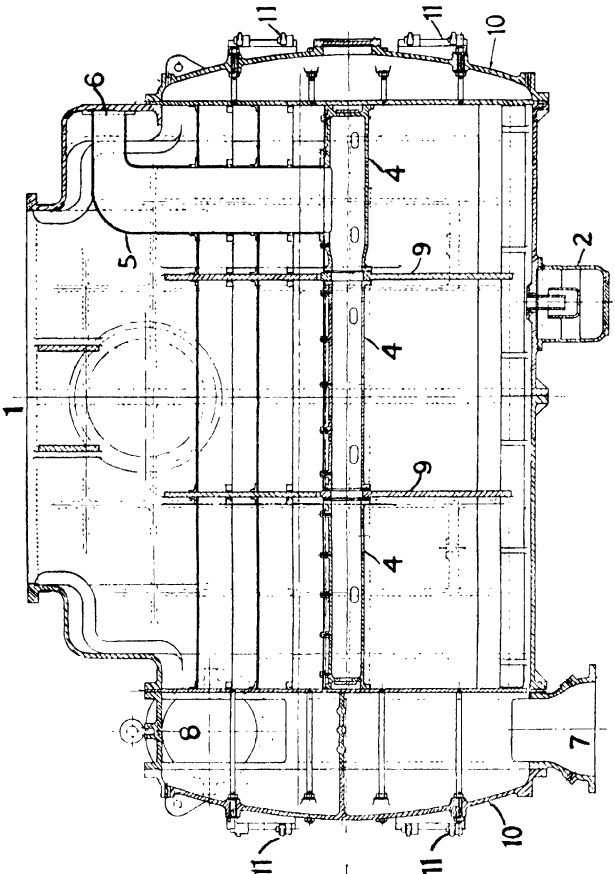



FIG. 465.

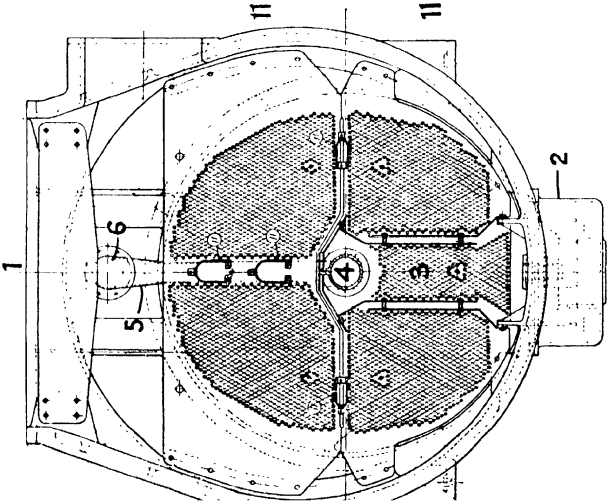



FIG. 464.

cooler tubes, and thence to the air extraction main 4, which is provided with ports and extends the whole length of the condenser. Vapour condensed in the cooler is passed to a further water seal in the chamber 2, so as to prevent re-evaporation, which might occur if the condensate temperature were slightly higher than the temperature corresponding to the pressure in the cooler. The air is drawn at the end of the extraction main through a vertical passage 5 connected with the air-pump suction at 6. The condenser is two pass. The water enters at 7, Fig. 465, flows through the condensing and cooler tubes of the lower pass, then back through the condensing tubes of the upper pass, and is discharged at the top through the branch 8. The tubes are secured at the tube plates by ferrules screwed into the plate, and are packed with linen tape. Sagging plates 9 are fitted to eliminate the possibility of vibration of the tubes, if their period of vibration should correspond to the turbo-generator speed.

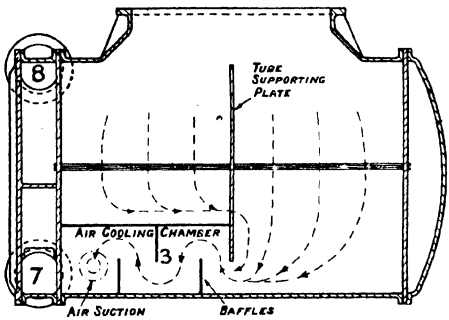



FIG. 467.

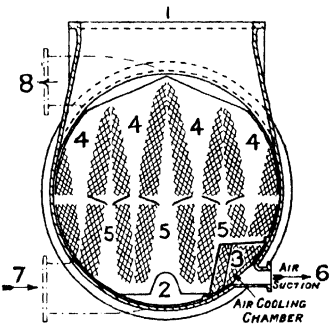



FIG. 466.

The water boxes 10 are of cast-iron and provided with a number of hinged inspection doors, 11.

**418. Mirrlees-Delas-Ginabat Condenser.**—The latest type of high vacuum and regenerative condenser, made by the Mirrlees Watson Co., is shown in skeleton section in Figs. 466 and 467. Here an entirely different arrangement of tubes is employed, as indicated by the cross section in Fig. 466. The tubes are arranged in shallow banks, giving deep V-shaped passages 4 for the flow of the steam, which enters at 1. A large tube area is thus exposed to the incoming steam, and the path through each bank is very short. It is claimed that the kinetic energy of the steam, which in the normal design is absorbed in overcoming the resistance to flow through the tube banks, is finally utilised in compressing the air and raising the temperature of the condensate. The successive rows of tubes are so arranged that the condensate falling from a tube in one row meets the tube immediately below it in the next row, tangentially, and in consequence flows over

only one-quarter of the circumference, and only a very thin film covers the rest of the surface.

The condensate which emerges on the inner side of a tube bank and is deflected by the inclined baffles shown in Fig. 466 has an unrestricted fall through the inverted V-shaped passages 5 to the bottom 2, from which it is drawn by the extraction pump. The air cooling chamber 3 extends for half the length of the condenser, and the paths of the air and vapour, after passing into the V passages, are indicated in Fig. 467 by the directional arrow lines. Baffle plates are provided, to deflect the flow over the cooling tube surface and ensure more effective cooling. Incidentally the higher temperature of the condensate materially assists in de-aeration of the water, and less occluded air is drawn away in the water by the extraction pump. The air is drawn through the branch 6 by a steam air-ejector. As in the previous case, this condenser is two pass, the circulating water entering at 7 and discharging at 8.

**419. Weir Regenerative Condenser.**—A good many years ago Messrs. G. and J. Weir made a departure from the usual condenser practice in the " Uniflux " type of condenser, in which the tubes were progressively pitched (wide at the top and gradually narrowing towards the bottom) and baffle plates were dispensed with. The steam was thus given a much less restricted path, and swept the surface so that no stagnant air pockets were produced. The result was a marked improvement in the condensation rate. The latest development in this direction is the regenerative condenser, an end-view of which, partly in section and partly in elevation, is shown in Fig. 468. The section of the condenser shell is approximately rectangular, the inlet 1 being at the top so that the condenser can be underhung from the turbine exhaust branch. There are two passes, as in the previous cases. Instead of, as in the normal type, the tube banks extending across the width of the condenser, they are divided vertically, and a clear central passage 4 of gradually reducing area is thus formed and extends from the top to the bottom of the condenser. The steam can thus penetrate right down to the bottom, and a large inlet area and surface is obtained on each side. The circulating water enters by the branch corresponding to 7 on the left side and flows first through the lower pass 5, then back through the upper pass 6, and is discharged through the branch corresponding to 8 on the left side. The condenser is really in two parts, which work in parallel. The air cooler 3, partitioned off by the baffle plate 9, in this case includes portions of both the lower and upper pass tubes ; and the air-pump suction 10 is situated at the top of this. With this arrangement the air and vapour flow in the first pass 5 is principally in the upward direction. The relatively cold condensate, in falling from the bottom rows of tubes, in the form of rain, to the bottom at 2, passes through the steam which has penetrated through 4 to the bottom of the condenser, and it is heated by direct contact to a temperature approximately that of

the entering steam. This reheating has the further advantage of practically de-aerating the condensate, which is drawn by the extraction pump from branch 11. It is stated that test results show the oxygen content of the condensate can be reduced to about 0·02 m.$^3$/litre, and zero values for the oxygen have also been obtained. This type is thus suitable for a " closed feed " system without a separate de-aerator. Under all loads the condensate is drawn from the condenser at practically the exhaust steam temperature. This condition is well illustrated by the curves shown in Fig. 469, which are plotted from test results of one of these condensers. The condensate temperatures are plotted to a base of

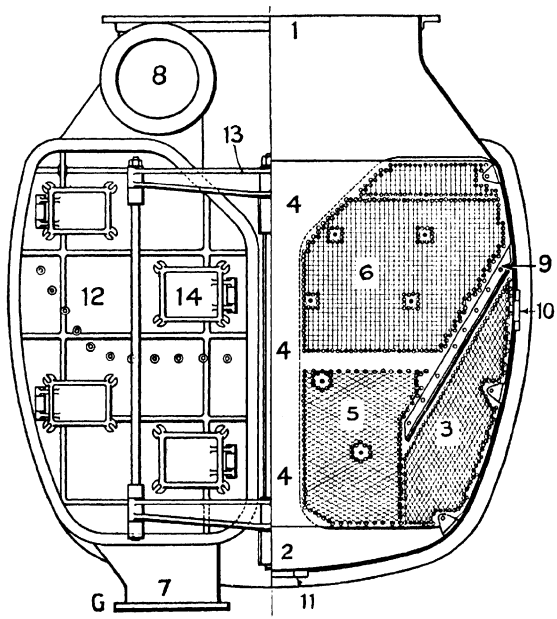



FIG. 468.

load. The upper curve AB is the theoretical or curve of steam temperature corresponding to the vacuum pressures at the various loads, rising from 69° Fahr. at no load to 79° Fahr. at full load. The curve CD gives the actual temperatures recorded, and throughout the range there is a constant difference of only about $\frac{1}{2}$° Fahr. between the temperatures.

Curve EF shows the temperatures usually obtained in a good normal condenser with a variation of from 14° Fahr. at no load to 5° Fahr. at full load. The superiority of this new design will be obvious. The door 12 of each half of the water box (Fig. 468) is carried on a hinge 13 when disconnected, and auxiliary inspection doors 14 are also provided.

If the air leakage is large the major portion of the cooler surface acts as an air cooler, while if it is small only a small portion functions. The maximum surface is thus available for condensation at all variations of air leakage. Also the difference between vacuum and air outlet temperatures increases with the air leakage.

**420. Measurement of Vacuum.**—In modern land installations where high vacua have to be recorded the mercury type of manometer is essential.

The objective is the absolute back pressure or pressure of the

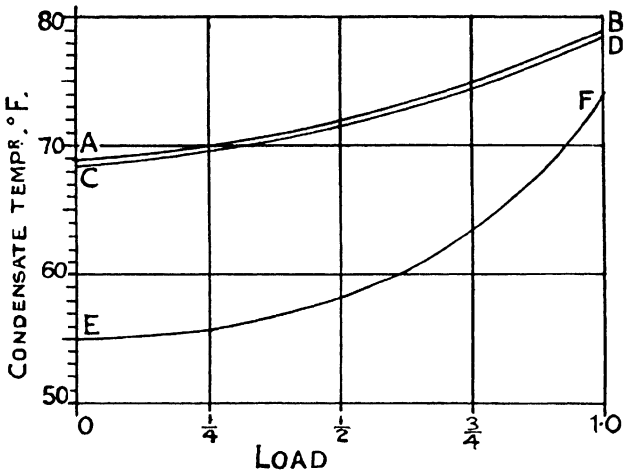



FIG. 469.

exhaust steam $p_v$ corresponding to the vacuum $x$ in. at inlet to the condenser.

Let $h$=barometric height in inches of mercury.

$y$=mercury manometer reading in inches of mercury.

$z$=exhaust pressure reading in inches of mercury.

Then—

$$z=(h-y) \text{ inches of mercury} \quad . \quad . \quad . \quad (32)$$

$$\text{and} \quad p_v=0{\cdot}49z \text{ lb./in.}^2 \text{ abs.} \quad . \quad . \quad . \quad . \quad (33)$$

The barometric height ($h$) which corresponds to the atmospheric pressure ($p_a$) is a variable quantity, due to fluctuating atmospheric conditions, and it also varies with altitude. For purposes of comparison it is necessary to adopt some standard barometric pressure and temperature, to which all readings can be referred. The standard temperature usually selected is 62° Fahr., since this is the value at which the barometric column is 30 in. and equivalent to 14·69 lb./in.$^2$ abs. The vacuum readings in all cases should be referred to the standard of 30 in.

If the temperature of the mercury of the barometer or manometer

is greater or less than 62° Fahr., then for extreme accuracy the readings should first be corrected to the standard temperature. It is rarely this proceeding is necessary. The corrected value can be obtained by multiplying the barometer or manometer reading $h_1$ or $y_1$ at the temperature $t_1$ by the correction factor—

$$[1-101\times10^{-6}(t_1-62)]$$

For instance, the barometer reading at 62° Fahr. would be—

$$h=h_1[1-101\times10^{-6}(t_1-62)]$$

When $h$ and $y$ corrected for temperature are used in (32) to obtain the correct value of $z$, the vacuum, referred to 30 in. bar., is given by—

$$\left.\begin{array}{l} x=(30-z)\text{ in.} \\ \text{or}\quad x=[(30-h)+y]\text{ in.} \end{array}\right\} \quad . \quad . \quad . \quad . \quad . \quad (34)$$

Thus to correct any vacuum reading to standard, subtract the barometric reading from 30 and add the manometer or vacuum gauge reading, all values being expressed in inches of mercury.

Alternatively the vacuum may be expressed as a percentage of the actual barometric pressure thus—

$$\text{Per cent. vacuum}=\frac{100y}{h} \quad . \quad . \quad . \quad . \quad . \quad (35)$$

This figure indicates the effectiveness of the condensation system. It will be apparent that for a given vacuum reading $y$ the effectiveness of the condensation will vary inversely as the barometric pressure. A vacuum gauge reading of 27 in. with a barometric pressure of 29 in. gives 93 per cent. vac., while 27 in. with 30 in. bar. gives 90 per cent. vac.

**421. Quantity of Circulating Water for Surface Condensers.**—As in the case of the jet condenser, this is calculable from the equation—

$$\left.\begin{array}{l} W_w=\dfrac{W_s h}{(h_2-h_1)}\ \text{lb./min.} \\ \text{or}\quad G=\dfrac{W_s h}{600(h_2-h_1)}\ \text{gall./min.} \end{array}\right\} \quad . \quad . \quad . \quad . \quad (36)$$

where $h$ is the heat given up per lb. by the exhaust steam and $h_1$ and $h_2$ are the total heats of the circulating water.

If $t_c$=condensate temperature and $h_c$=total heat of the water, then—

$$\left.\begin{array}{l} h=(h_v-h_c+q_v L_v) \\ \text{or approximately}\quad h=(t_v-t_c+q_v L_v) \end{array}\right\} \quad . \quad . \quad . \quad . \quad . \quad . \quad (37)$$

The difference between the condensate temperature $t_c$ and vacuum temperature in high-class work, as already seen, Art. 416, is very small, and in fact negligible.

The general expressions become—

$$\left.\begin{aligned} W_w &= \frac{W_s(t_v - t_c + q_v L_v)}{(t_2 - t_1)} \text{ lb./min.} \\ \text{or } G &= \frac{W_s(t_v - t_c + q_v L_v)}{600(t_2 - t_1)} \text{ gall./min.} \end{aligned}\right\} \quad . \quad . \quad . \quad (38)$$

Or taking the rough approximation of $h = 1000$—

$$G = \frac{W_s}{0{\cdot}6(t_2 - t_1)} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (39)$$

**422. Temperature of Circulating Water.**—The inlet temperature $t_1$ where there is a plentiful supply of water may run from 32° Fahr. to 60° Fahr. Where water supply is scarce and the same water has to be used again and again, some form of cooling tower is employed, and the temperature may run from 70° Fahr. to 80° Fahr.

Ideally the outlet temperature $t_2$ of the circulating water should be equal to the temperature $t_v$ of the exhaust steam. This is not a possible condition, on account of the temperature gradient across the tube walls, from steam to water. The difference $(t_v - t_2)$ in very good design may fall to about 5° Fahr.; but the more usual range is 8° Fahr. to 12° Fahr. In marine practice much greater differences obtain.

For marine reciprocators it runs from 25° Fahr. to 30° Fahr.
,, ,, turbines ,, ,, 12° Fahr. to 15° Fahr.
In exceptional cases it may run to 50° Fahr.

The possible initial temperature of the circulating water in any given case, together with the quantity available, determines the maximum vacuum that can be obtained. This, however, is not necessarily the most economical vacuum to adopt for a given plant. The pumping outfit has to be considered in relation to the condenser, and the cost of this may not justify the adoption of the maximum vacuum. A low vacuum with a less expensive pumping plant may be a more economic proposition. Where there is a limited water supply and a cooling tower, involving high lift pumping, the temperature difference $(t_v - t_2)$ should be kept low, giving maximum temperature rise with a minimum quantity of water. When there is plenty of water and a low pumping head, $(t_v - t_2)$ may be kept high, as in the case of a marine installation.

A small temperature difference, however, involves a larger amount of condensing surface, with increased first cost; while the larger difference results in a smaller and cheaper condenser. The ratio of steam to circulating water $\left(\frac{W_w}{W_s}\right)$ which it has been found uneconomical to exceed is about 80. If a value of 1,000 B.Th.U. per lb. is taken for $h$, then the limit of the circulating water temperature rise is $\frac{1000}{80} = 12{\cdot}5°$ Fahr.

**423. Transmission of Heat in the Surface Condenser.**—It may be assumed that the circulating water at the first pass enters each tube of that pass with the inlet temperature $t_1$. The mass of water flows through the tubes in parallel streams and is raised at the outlet end of the pass to some higher temperature $t'$, due to condensation of the steam and partial cooling of any air associated with it. The water then flows through the tubes of the upper pass and is similarly raised in temperature from $t'$ to $t_2$. Now if the vacuum $x$ were maintained absolutely constant throughout the whole of the tube banks and if there were no air, the temperature at every point on the length of any tube would be $t_v$. The temperature difference, for all tubes of the first pass, would vary from $(t_v - t_1)$ at entrance to $(t_v - t')$ at exit, and for the second pass from $(t_v - t')$ at entrance to $(t_v - t_2)$ at exit. Owing to the presence of air the actual temperature at any point of any tube is less than $t_v$ on account of the operation of Dalton's Law. The real temperature is $t_s$, corresponding to the partial steam pressure $p_s$, since if $p_a$ is the corresponding partial air pressure, the vacuum pressure is the sum of these, or $p_v = (p_s + p_a)$.

In the ordinary type of close tube bank arrangement, as the steam is progressively condensed the air richness increases, the partial steam pressure $p_s$ and temperature $t_s$ decrease from top to bottom of the condenser. That is, the external temperature of steam, condensate, and air surrounding the tubes varies throughout. In general, with the close tube banks the proportion of air to steam is so small at entrance that $t_s$ may be taken equal to $t_v$ for the upper row of the second pass. For the lowest row of the first pass the steam temperature may be considerably lower. Suppose that for a total or vacuum pressure $p_v$ the partial steam pressure $p_s$ is known along any tube of the lower pass, then the temperature of the air-steam mixture along the tube may be taken as constant and approximately equal to the condensate temperature $t_c$.

At the upper tubes of the first pass and lower tubes of the second pass there will be some value between $t_c$ and $t_v$, and as a first approximation this may be taken to represent a constant temperature $t_s$ for the whole surface. In Fig. 470 OB represents the length of the first pass tubes and BC the length of the second pass, and also on some proportional scale the corresponding surfaces, for a given size of tube. If the water temperatures were registered along the tubes of the first pass and the average value at each vertical section were plotted, a curve of the form DE would be obtained. Similarly, if there were no integral air cooler, a curve EF would be obtained for the second pass, EF simply being a continuation of DE, since the outlet water temperature $t'$ at the first pass would be the same as the inlet temperature at the second pass. Actually, with an air cooler representing from 5 to 8 per cent. of the tube surface, a considerable portion of the water flowing through the first pass emerges at a lower temperature than $t'$ and on mixture with the main body there is a fall of temperature. The curve flattens out to the

right of E and then rises again to the outlet value $t_2$. A full discussion of the conditions arising from the introduction of the air cooler cannot be attempted here, and the student is referred, for this, to an excellent series of articles which have been published in *The Engineer*.*

**424. Mean Temperature Difference.**—If a fluid at temperature $t$ transmits heat, at any point of a surface, through a metal wall to another fluid at temperature $t_s$, and if it is assumed that the amount of heat transmitted is proportional to the difference of temperature $(t_s - t)$, then it can be shown, from first principles, that

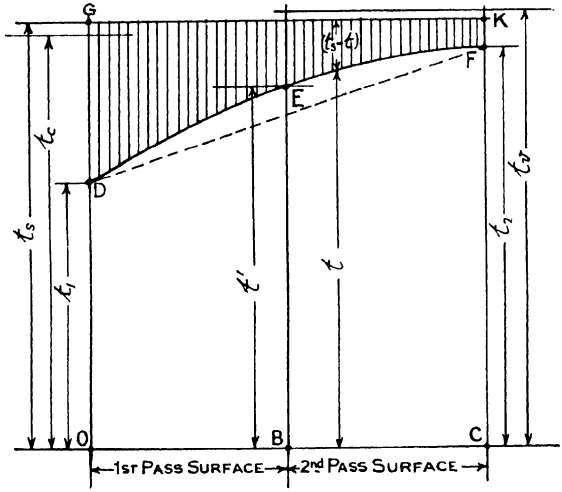



FIG. 470.

the mean temperature difference for the whole surface, or the mean ordinate of the figure DEFKG, Fig. 470, is given by—

$$t_m = \frac{\text{Initial temperature difference—final temperature difference}}{\log_e \dfrac{\text{Initial temperature difference}}{\text{Final temperature difference}}}$$

Thus
$$t_m = \frac{(t_s - t_1) - (t_s - t_2)}{\log_e \dfrac{t_s - t_1}{t_s - t_2}}$$

or
$$t_m = \frac{(t_2 - t_1)}{\log_e \dfrac{t_s - t_1}{t_s - t_2}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (40)$$

In this case, however, $t_s$ is really not a constant for the whole

* "Regenerative Surface Condensers," by Thomas Petty, *The Engineer*, 1, 8, 22, 29 March, 1929.

surface, and it is a usual practice to take the inlet and outlet differences as $(t_c - t_1)$ and $(t_v - t_2)$, so that—

$$t_m = \frac{(t_c - t_1) - (t_v - t_2)}{\log_e \frac{t_c - t_1}{t_v - t_2}} \quad . \quad . \quad . \quad . \quad . \quad . \quad (41)$$

The value being slightly higher than given by (40).

In the case of the modern regenerative type, however, as already shown, $t_c$ and $t_v$ are nearly equal, so that a sufficiently close approximation is given by—

$$t_m = \frac{(t_2 - t_1)}{\log_e \frac{t_v - t_1}{t_v - t_2}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (42)$$

The value of the mean temperature difference given by the foregoing expressions is called the "logarithmic mean temperature difference" to distinguish it from the arithmetical mean of the inlet and outlet differences, that is, the mean ordinate of DFKG, when the straight temperature line DF (dotted) is taken instead of the curve DEF.

**425. Rate of Heat Transmission.**—When the temperature difference $(t_s - i)$ at any element of the tube surface is known, the heat transmitted per unit area of the element per unit time is given by the product of the temperature difference and the rate of heat transmission at the element, expressed in thermal units per unit area per unit time per degree temperature. It is usually taken as B.Th.U./ft.$^2$ hr.° Fahr.

For the whole surface a mean value of K has to be used in conjunction with the mean temperature difference. If S is the total condensing surface in ft.$^2$, the heat transmitted by the surface per unit time is given by—

$$H' = t_m KS \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (43)$$

If $h$ is the heat given up per lb. of exhaust steam and $W_s$ is the weight of steam per hour, then—

$$H' = W_s h = t_m KS$$

and the total surface is given by—

$$S = \frac{W_s h}{t_m K} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (44)$$

The value of K is controlled principally by the velocity of the circulating water through the tubes. The temperature of the water, however, which controls the viscosity, is also a secondary factor. Here again a full discussion of this subject in its various aspects would occupy far too much space, and only one or two salient points can be noted. The student should refer to the texts already quoted in the footnote on p. 746. An interesting discussion will also be found in the paper quoted below.*

* "High Vacuum Condensers," by D. T. McHutchison. Published by the Association of Engineering and Shipbuilding Draughtsmen, Session 1925–26.

The resistance to flow of heat through the tube walls is the reciprocal of the rate of heat transmission. Denoting it by R, then—

$$R=\frac{1}{K} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (45)$$

Under a given temperature "head" or difference, this resistance, in the case of perfectly clean tubes, is the sum of four resistances, and with dirty tubes the sum of five. On the steam side there are the resistances to heat flow offered by the air blanket, the film of condensate, and the film of grease or other impurities which may be carried over by the steam. Then there is the resistance of the tube. On the water side there are resistances due to scale, mud, or any other impurities deposited by the water on the surface, and to a viscuous film of water next the tube.

Let the resistances of the steam side, the metal, and water side be denoted respectively by $R_s$, $R_m$, and $R_w$, then—

$$R=R_s+R_m+R_w=\frac{1}{K} \quad . \quad . \quad . \quad . \quad . \quad (46)$$

As in practice R is an inconveniently small number to handle, it is customary to use $1000R=\frac{1000}{K}$.

In order to get some idea of the relative values of these resistances it may be pointed out that the resistance to flow of heat across a film of any substance having a thickness $x$ in. is given by—

$$r=\frac{1}{k}=\frac{x}{c}$$

where $c$ is the thermal conductivity of the substance in heat units per unit length per unit time. On the basis of—

$$1000r=\frac{1000}{k}=\frac{1000x}{c}$$

the following are average values of the thermal conductivity ($c$) for various substances—

| | $c$ | |
|---|---|---|
| Air film . . . . . | 0·215 | From these figures the effect of a thin film of any of these substances on the resistance will be obvious. |
| Lime scale . . . . . | 17 | |
| Oil film . . . . . | 8·5 | |
| Water film . . . . . | 6·0 | |
| Brass tube . . . . . | 1120 | |

The resistance of the tube itself, however, is practically a negligible quantity. Professor Callendar showed that with perfectly air-free steam the resistance $R_s$ on the steam side would be of the order of 0·333. With the value of $c$ above, for the brass tube of thickness 0·048 in., $R_m=0·044$, so that for a clean tube the steam and tube resistance $(R_s+R_m)=0·377$, or taking the round number, 0·4 may be assumed. For a greasy tube this would have to be

appreciably increased. If such increased value is denoted by $a$, then the resistance may be expressed as—

$$R=R_w+a \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (47)$$

With regard to the resistance on the water side, many researches have been carried out and theories propounded * to arrive at some reliable basis for its calculation; but the design is still dependent to a large extent on experience and practical judgment in the selection of the transmission coefficient for any given case. Sim states that from experiments on actual condensers he has found the resistance on the water side is given by—

$$R_w=\frac{C}{\sqrt{V^{0\cdot82}}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (48)$$

Where C is a function of the kinematic viscosity of the water and the tube diameter and V the velocity of the water in the tube. The viscosity being a function of the water temperature he gives the values of C plotted to a base of mean water temperature, for $\frac{3}{8}$-in., $\frac{3}{4}$-in., $\frac{7}{8}$-in., and 1-in. tubes.† The water velocity is always above the critical, so that turbulent flow, essential for effective convective transmission, is ensured. It runs from 4 to 8 ft./sec. In settling a value of water velocity V, due regard has to be paid to the power required to pump the water at the chosen value through the condenser.

Low velocity reduces power for pumping but increases the size of the condenser, and vice versa.

The value of C varies from 1·4 at 140° Fahr. to 3·4 at 60° Fahr. for $\frac{3}{4}$-in. tube.

The value of C varies from 1·0 at 140° Fahr. to 2·4 at 60° Fahr. for 1-in. tube.

He takes the tube resistance as $R_m=0{\cdot}08$, and the values suggested for $(R_m+R_s)=a$, are—

For reciprocating engine condensers, 1·28 to 1·58
" " turbine " 0·68 to 1·28

The corresponding values of the rate of heat transmission are given by—

$$\text{Sim}\begin{cases} K=\dfrac{1000}{\dfrac{C}{\sqrt{V^{0\cdot82}}}+1\cdot28} \text{ to } K=\dfrac{1000}{\dfrac{C}{\sqrt{V^{0\cdot82}}}+1\cdot58} \text{ for engines} \quad . \quad (49) \\ K=\dfrac{1000}{\dfrac{C}{\sqrt{V^{0\cdot82}}}+0\cdot68} \text{ to } K=\dfrac{1000}{\dfrac{C}{\sqrt{V^{0\cdot82}}}+1\cdot28} \text{ for turbines} \quad . \quad (50) \end{cases}$$

* See "The Theory of the Surface Condenser," *Engineering*, 1 and 9 January, 1914. "Heat Transfer in Surface Condensers," by Dr. A. W. Porter, *Engineering*, 24 and 31 January, 1919. "The Laws of Heat Transfer," by H. M. Martin, *Engineering*, 6 and 20 July, 3 and 24 August, 1923.

† See *Surface Condensing Plant*, p. 10.

It is presumed that the tubes are clean on the water side.

G. A. Orrok, a well-known American engineer, has carried out investigations from which the following expression for the resistance on the water side has been deduced for clean tubes, as—

$$R_w = \frac{1000}{140V + 250} \quad \ldots\ldots\ldots \quad (51)$$

and on this basis—

$$K = \frac{1000}{\left(\frac{1000}{140V + 250} + a\right)} \quad \ldots\ldots \quad (52)$$

In the paper cited Hutchison suggests lower values of $a$ than given by Sim: 1·1 for reciprocators; 0·8 for turbines; 0·6 for high-class power station turbines. The condition of clean tubes is again assumed. There is usually not much difficulty in keeping the tubes clean by periodic overhaul, and there is thus little excuse for reduction of vacuum from this cause.

Generally the value of the rate of heat transmission (K) runs from 400 to 500 B.Th.U./ft.$^2$ hr.° Fahr. under continuous service conditions with normal condensers. With the newer regenerative designs being installed much higher values are being recorded, from 600 to 800 B.Th.U./ft.$^2$ hr. ° Fahr. For example, in a test of a 25,000 kw. Metropolitan-Vickers set supplied to Liverpool Corporation, W. M. Selvey found that the central flow condenser gave a transmission rate (based on the logarithmic mean temperature difference) of 697·7; * while a test of a similar M.V. condenser of a 10,000 kw. set, supplied to the L.M.S. Railway, showed a higher figure of 817 on full load and 824·5 on three-quarter load.†

Still higher values are being quoted from the Mirrlees-Delas-Ginabat regenerative condenser. It is stated that in several installations a transmission rate of 1000 B.Th.U./ft.$^2$ hr. ° Fahr. has been obtained.

**426. Diameter and Length of Tube.**—The diameter of the tube has to be fixed from mechanical considerations, conditioned by the nature of the plant. In land installations where river water or water liable to carry solid matter, wood, leaves, mill refuse, etc., is used, provision for easy and effective cleaning is necessary, and 1-in. diameter tubes are often fitted. In marine work, where clean sea water is available, the tubes are usually $\frac{3}{4}$ in. diameter.

Let G=quantity of circulating water in gall./min.
V=velocity of the water in ft./sec.
S=total condensing surface in ft.$^2$.
$d_i$=internal diameter of tube in in.
$d_o$=external diameter of tube in in.
N=number of passes.

* See *The Engineer*, 19 and 25 October, 2 and 9 November, 1928.
† See *Engineering*, 5 October, 1928.

then it can be shown, from the equation of continuity, that the total number of tubes is given by—

$$n=\frac{0\cdot492GN}{d_i^2V} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (53)$$

and the total length of flow, that is, the sum of the lengths of the passes (not of the number of tubes), is given by—

$$L=\frac{7\cdot76SV}{G}\times\frac{d_i^2}{d_o}\text{ ft.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (54)$$

The length between the tube plates is given by—

$$l=\frac{L}{N}\text{ ft.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (55)$$

This length has to be fixed by considerations of the lay-out of the plant, foundation size, space for tube withdrawal, position of inlet and outlet water branches relatively to the engine or turbine, and also from consideration of the friction head on the pump. The usual number of passes is two, but three or more may be employed. On the other hand, as already stated in Chapter XX, some of the large American installations have vertical condensers of "single pass" type. In any case, an even number of passes is desirable, as the inlet and outlet branches can be placed at one end, with simplification of the pipe arrangement. The designer has to be guided by the ratio of tube face area to length between plates, or in the case of a circular condenser, by the length-diameter ratio. In good practice this runs from 1·5 to 2·5.

*Example* 7.—Calculate the total condensing surface and number and length of tubes for a two-pass condenser on a turbine, from the following provisionally assumed data.

Weight of exhaust steam 100,000 lb./hr.; vacuum 29 in.; quality 0·9; inlet temperature of water 55° Fahr.; difference between outlet temperature and vacuum temperature 8° Fahr.; condensate temperature 5° Fahr. less than vacuum temperature; tubes $\frac{3}{4}$ in. outside diameter, and 0·048 in thick; water velocity 6 ft./sec.

$x=29$, $p_v=0\cdot49$, $t_v=78\cdot82°$ Fahr., $t_2=(t_v-8)=70\cdot82°$ Fahr., $t_c=(78\cdot82-5)=73\cdot82°$ Fahr., $t_1=55°$ Fahr., $q_v=0\cdot9$, $L_v=1044\cdot7$ $d_i=0\cdot702$ in., $d_o=0\cdot75$ in.

Mean temperature difference—

$$t_m=\frac{(t_c-t_1)-(t_v-t_2)}{\log_e\frac{t_c-t_1}{t_v-t_2}} \qquad \begin{matrix}(t_c-t_1)=18\cdot82\\(t_v-t_2)=8\end{matrix}$$

$$=\frac{18\cdot82-8}{2\cdot3\log\frac{18\cdot82}{8}}=\frac{10\cdot82}{2\cdot3\times0\cdot3751}=12\cdot7°\text{ Fahr.}$$

Rate of heat transmission: taking Equation 50, the value of C

from Sim's curve for $\frac{3}{4}$-in. tube, at mean water temperature 63° Fahr., is 3·3; $V^{0·82}=6^{0·82}=3·382$, and $\frac{C}{V^{0·82}}=0·975$. Taking the case for $a=0·68$—

$$K=\frac{1000}{\frac{C}{V^{0·82}}+a}=\frac{1000}{0·975+0·68}=\frac{1000}{1·655}=605$$

Taking Equation 52—

$$K=\frac{1000}{\frac{1000}{140\times 6+250}+a}=\frac{1000}{0·917+a}$$

If $a=0·7$ is chosen, then—

$$K=\frac{1000}{1·617}=620$$

An average of K=610 might be taken. With this value, total condensing surface—

$$S=\frac{W_s h}{t_m K}$$

$$=\frac{100000\times 945·23}{12·7\times 610}=12{,}200 \text{ ft.}^2$$

$W_s=100{,}000$

$h=(t_v-t_c+q_v L_v)$

$=(5+0·9\times 1044·7)$

$=945·23$

Circulating water—

$$G=\frac{W_s h}{600(t_2-t_1)}=\frac{100000\times 945·23}{600\times 15·82}=10{,}000 \text{ gall./min.}$$

Total number of tubes—

$$n=\frac{0·492GN}{d_i^2 V}=\frac{0·492\times 10000\times 2}{0·702^2\times 6}=3332$$

N=2

Flow length—

$$L=\frac{7·76SVd_i^2}{Gd_o}=\frac{7·76\times 12200\times 6\times 0·702^2}{10000\times 0·75}=37·33 \text{ ft.}$$

Tube length—

$$l=\frac{L}{N}=\frac{37·33}{2}=18·66 \text{ ft.}$$

The number of tubes and surface would require to be increased by at least 5 per cent. to provide for effective air cooling.

**427. Air and Vapour Mixture to be extracted by the Dry Air Pump.**—Water drawn from a surface condenser in good condition should be nearly air free, and if it is not unduly exposed to the atmosphere before it is fed to the boiler the air introduced in the feed should be a very small quantity. It is becoming absolutely

essential in the case of high-pressure and high-temperature water-tube boilers that oxygen should be prevented from getting into the boiler except in an exceedingly small amount, in order to prevent serious corrosion troubles; and "closed feed" systems are now being generally applied in new installations. In these arrangements, the feed is drawn from the condenser and discharged to the suction of the feed pump, and is thus prevented from coming into contact with the atmosphere.

In the case of a reciprocator, leakage of air into the exhaust steam can be largely prevented by careful attention to stuffing glands, spindles, relief valves, pipe joints, etc. In the case of the high vacuum condenser used with a turbine set, small pin-hole leaks at the castings may admit a serious amount of air. The capacity of the air extraction plant, rotary pump, or steam ejector depends on the probable leakage to be expected in any given case. There is no definite means of ascertaining this figure, and the probable amount to be dealt with has to be assumed on the basis of previous experience with similar plants.

In high-class condenser work an average value of 4 lb. of air per 10,000 lb. of steam is sometimes allowed.

The B.E.A.M.A. association has fixed the following standard for land plants, to cover leakage and entrained air—

$$w_a = \frac{0{\cdot}5W_s}{1000} + 3 \text{ lb. air/hour} \quad . \quad . \quad . \quad . \quad (56)$$

where $W_s$=steam condensed in lb./hr.

Sim states that from tests on actual plants it appears that half of this amount is normally sufficient.

In order to determine the weight of air-vapour mixture to be handled by the pump for a given air quantity, the Characteristic Law and Dalton's Law have to be applied.

Let $W_a$=weight of air to be extracted per unit of time.

$t_0$ =temperature of air-vapour mixture at the air-pump suction.

$p_{s0}$=partial vapour pressure, corresponding to $t_0$ (obtained from the steam table).

$p_{a0}$=partial air pressure at $t_0$.

$p_{v0}$=the vacuum pressure at the air-pump suction.

$v_{s0}$=specific volume of the vapour at $t_0$ (from table).

$v_{a0}$=specific volume of the air at $t_0$.

Dalton's Law—

$$p_{a0} = (p_{v0} - p_{s0}) \quad . \quad . \quad . \quad . \quad . \quad . \quad (57)$$

Characteristic Law, air volume—

$$v_{a0} = \frac{R_a T_o}{144 p_{a0}}$$

With R=53·3—

$$v_{a0}=\frac{T_0}{2{\cdot}7p_{a0}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (58)$$

The total volume of the air is thus—

$$V_{a0}=W_a v_{a0}=\frac{W_a T_0}{2{\cdot}7p_{a0}} \quad . \quad . \quad . \quad . \quad . \quad (59)$$

Now the air and vapour, according to Dalton, occupy the same volume, or $V_{a0}=V_{s0}$. If $w_{s0}$ is the total weight of vapour associated with the air, then—

$$w_{s0}=\frac{V_{s0}}{v_{s0}}=\frac{W_a v_{a0}}{v_{s0}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (60)$$

Hence the total weight of the mixture to be drawn by the dry air pump is given by—

$$W_{mix.}=W_a\left(1+\frac{v_{a0}}{v_{s0}}\right) \quad . \quad . \quad . \quad . \quad . \quad . \quad (61)$$

**428. Cooling of the Air and Vapour.**—This mass of stuff weighing W lb. is drawn over the tubes of an integral cooler.

Denote the initial temperature at the bottom of the condenser, which is that of the condensate, by $t_2'$, and the vacuum pressure by $p_{v2}$. It may be assumed that the outlet vacuum is sensibly the same, or $p_{v2}\doteq p_{v0}$, then—

$$p_{a2}=(p_{v2}-p_{s2}) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (62)$$

and

$$p_{a0}=p_{v0}-p_{s0} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (63)$$

The weight of vapour at entrance is—

$$w_{s2}=W_a\frac{v_{a2}}{v_{s2}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (64)$$

The weight of vapour at exit is—

$$w_{s0}=W_a\frac{v_{a0}}{v_{s0}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (65)$$

Hence the weight of vapour condensed while the mixture flows through the cooler is—

$$w_s=W_a\left(\frac{v_{a2}}{v_{s2}}-\frac{v_{a0}}{v_{s0}}\right) \quad . \quad . \quad . \quad . \quad . \quad (66)$$

As the major portion of the condensation takes place at entrance to the cooler, it is sufficient to take the heat given up as equal to the latent heat of the dry vapour at $p_{s2}$, that is, as $L_2$. Hence, to a first approximation, heat from vapour to cooling water is—

$$h_s=w_s L_2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (67)$$

The heat given by the air in cooling from $t_2'$ to $t_0$ is—

$$h_a=W_a c_p(t_2'-t_0) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (68)$$

Hence the total amount of heat transmitted to the cooling water in the tubes by the air-vapour mixture is—

$$H=W_a\left\{L_2\left(\frac{v_{a2}}{v_{s2}}-\frac{v_{a0}}{v_{s0}}\right)+c_p(t_2'-t_0)\right\} \quad . \quad . \quad . \quad (69)$$

The cooler surface $S_c$ can be calculated, when suitable values of mean temperature difference $t_{m_c}$ and heat transmission coefficient $K_c$ are estimated.

As the rise in temperature of the circulating water in the cooler is small, it is sufficient to take the inlet temperature difference as $(t_2'-t_1)$ and the outlet difference as $(t_0-t_1)$, so that, approximately—

$$t_{m_c}=\frac{t_2'-t_0}{\log_e\frac{t_2'-t_1}{t_0-t_1}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (70)$$

The value of $K_c$ is a very uncertain figure. A reasonable value of V, from 4 to 6 ft./sec., can be chosen and $K_c$ can be calculated from Equation 52, in which a large value of $a$, say 3, may be used.

Thus

$$K_c\doteqdot\frac{1000}{\frac{1000}{140V+250}+3} \quad . \quad . \quad . \quad . \quad . \quad . \quad (71)$$

The cooler surface is given by—

$$S_c=\frac{H}{t_{m_c}K_c} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (72)$$

and the number of cooler tubes, since $l$ and $d_0$ are known, by—

$$n_c=\frac{12S_c}{\pi d_0 l} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (73)$$

The additional cooling water is given by—

$$G_c=\frac{nVd_i^2}{0{\cdot}492} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (74)$$

the diameters $d_i$ and $d_0$ being in inches, and $l$ in feet.

*Example* 8.—In the previous example, assuming that the air leakage is 4 lb. per 10,000 lb. of steam, and that the drop of vacuum across the condenser is 0·15 in., estimate a suitable value of cooling surface, the additional number of cooler tubes, and quantity of circulating water at a velocity of 6 ft./sec. in order to reduce the outlet temperature of the air-vapour mixture to 65° Fahr.

Here $t_c=t_2'=73{\cdot}82°$ Fahr., $t_1=55°$ Fahr., and $t_0=65°$ Fahr.

then $$t_{m_c}=\frac{t_2'-t_0}{\log_e\frac{t_2'-t_1}{t_0-t_1}}=\frac{18{\cdot}52-10}{2{\cdot}3\log\frac{18{\cdot}52}{10}}=13{\cdot}8°\text{ Fahr.}$$

and $$K_c=\frac{1000}{\frac{1000}{40V+250}+3}=\frac{1000}{3{\cdot}917}=256 \qquad \begin{matrix}V=6\\ a=3\end{matrix}$$

For Vac.=29·15, $p_{v2}\doteqdot p_{v0}$=0·416 lb./in.² abs.,

$t_2'$=73·82, $p_{s2}$=0·412, $v_{s2}$=769·5, $L_2$=1050
$t_0$=65, $p_{s0}$=0·3107, $v_{s0}$=1005 } (Tables.)

$$p_{a2}=(p_{v2}-p_{s2})=(0\cdot416-0\cdot412)=0\cdot004\ ;$$

$$v_{a2}=\frac{T_2}{2\cdot7p_{a2}}=\frac{533\cdot82}{2\cdot7\times0\cdot004}=49500$$

$$p_{a0}=(p_{v0}-p_{s0})=(0\cdot416-0\cdot3107)=0\cdot105\ ;$$

$$v_{a0}=\frac{T_0}{2\cdot7\times p_{a0}}=\frac{525}{2\cdot7\times0\cdot105}=1850$$

$$W_a=\frac{4W_s}{10000}=40\text{ lb.}$$

$$\therefore\ H=W_a\left[L_2\left(\frac{v_{a2}}{v_{s2}}-\frac{v_{a0}}{v_{s0}}\right)+c_p(t_2'-t_0)\right] \qquad c_p=0\cdot238$$

$$=40[1050(64\cdot3-1\cdot85)+0\cdot238(73\cdot82-65)]$$

$$=40\times65522=2{,}620{,}880$$

Cooling surface $S_c=\frac{H}{t_{m_c}K_c}=\frac{2620880}{13\cdot8\times256}=740$ ft.²

then $\frac{S_c}{S}=\frac{740}{12200}=0\cdot06$ $\qquad S=12200$

The calculation is a very tentative one, on account of the various assumptions made; but it would appear that at least 6 per cent. increase of surface is necessary to obtain the required air-vapour temperature at outlet.

The additional number of tubes is—

$$n_c=\frac{12S_c}{\pi d_0 l}=\frac{12\times740}{\pi\times0\cdot75\times18.66}=200 \qquad l=18\cdot66,\ d_0=0\cdot75$$

Additional cooling water—

$$G_c=\frac{nVd_i^2}{0\cdot492}=\frac{200\times6\times0\cdot702^2}{0\cdot492}=1200\text{ gall./min.} \qquad d_i=0\cdot702,\ V=6$$

**429. Extraction of Air and Vapour.**—In the older types of surface condensers used with reciprocators the condensate, air, and vapour were drawn together by a "wet" type of air pump having foot valves, valves in the piston or bucket, and head or discharge valves, the air during the discharge stroke being compressed slightly above and discharged with the water, against the atmospheric pressure, into the "hot well." A typical design of this old form, as applied to a marine engine, is shown in Fig. 471. Rubber valves were used, but these were later superseded by multiple-plate valves. This type is now seldom fitted except on small marine jobs. Incidentally the illustration shows the section of the old form of two-pass marine engine condenser, which was cast integral with the back columns of

the engine. Generally the condenser is now made separately and bolted to the columns, as shown on the modern marine set illustrated in Figs. 374 and 375, Chapter XVII. At a later stage of development the valve bucket air pump was replaced by the valveless bucket type, of which the Edwards pump was, and still is, the chief representative. A section through this design is shown in Fig. 472. The bottom of the casing is coned and the solid piston or bucket is coned to fit it. The liner or barrel, projecting into the bottom of the casing, is provided with ports, which are uncovered by the bucket at the bottom of the stroke. The condensate, as indicated by the arrow line on the left, runs by gravity into the bottom of the casing, and the air is also sucked in during the up-stroke. When the bucket descends it drives the water and air round the paraboloid surface of the casing, as shown by the dotted lines, and through the liner ports on to the top. As soon as the bucket edge closes these ports on the discharge stroke, compression of air and vapour commences, and the mixture is forced, at the top of the stroke, through the multiple-plate head valves, which are sealed by a considerable head of water. The Edwards pump is still being used for reciprocator and small turbo sets.

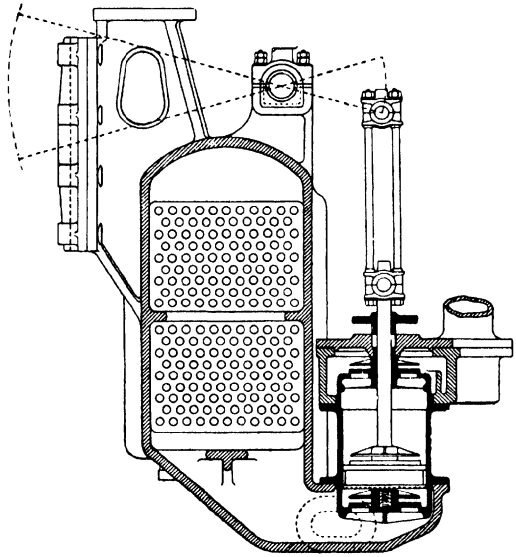


FIG. 471.

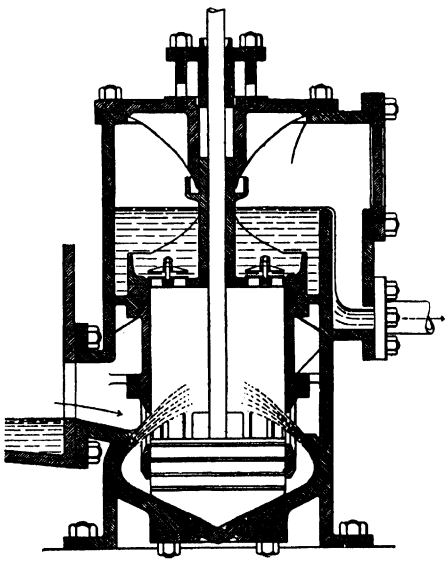


FIG. 472.

As already emphasised, however, separate extraction of air and vapour is an absolute essential for high vacuum turbine plants. The necessity for the highest possible vacuum with the steam turbine was early recognised by Sir Charles Parsons, and his first step towards this end was the introduction of the "vacuum augmenter," in the form of a steam-jet air ejector exhausting into a small auxiliary condenser. A section of the original arrangement is shown in Fig. 473.* The ordinary "wet" air pump was connected to a long pipe, with a dip in it, so as to form a water seal. The condenser was inclined, and this pipe, connected near the lower end, filled with condensate which drained into it. At a higher level the suction branch of the steam ejector was fixed, and the air and vapour were drawn away separately, the steam being condensed in the small condenser, having a surface about 5 per cent. of that

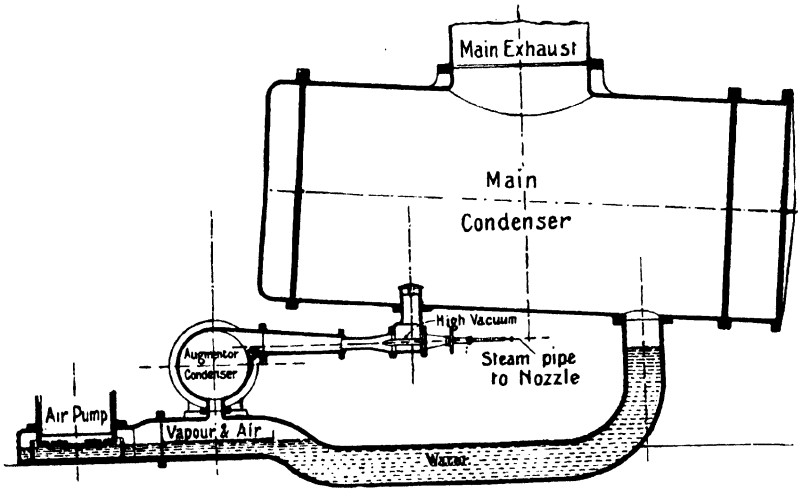



FIG. 473.

of the main condenser. This auxiliary condensate and the air were discharged into the main pipe and handled in the ordinary way by the wet air pump. The steam jet took about $1\frac{1}{2}$ per cent. of the full load steam consumption. Makers of reciprocating air pumps followed suit by providing separate pumps to deal with the condensate and the air and vapour. In some designs a single cylinder was made to serve both as a dry and wet pump.

A diagrammatic section of one of the two cylinder type for marine work, by G. and J. Weir, known as the "dual" air pump, is shown in Fig. 474. A is the wet and B the dry air pump. The condensate flows down the inclined pipe C into the bottom of the wet pump, while the air and vapour are drawn by the dry air pump

* See paper by the Hon. C. A. Parsons, G. Stoney, and C. P. Martin, *Inst. Elect. Engineers*, 12 May, 1904.

through the pipe D. The dry air pump discharges air and vapour through the pipe E against the resistance of a spring-loaded valve F, into the wet air pump, at a point below the wet pump head valves. This arrangement considerably reduces the load on the dry air pump. The air and vapour mixture is cooled by injection water, which enters the suction branch of the dry pump, below the foot

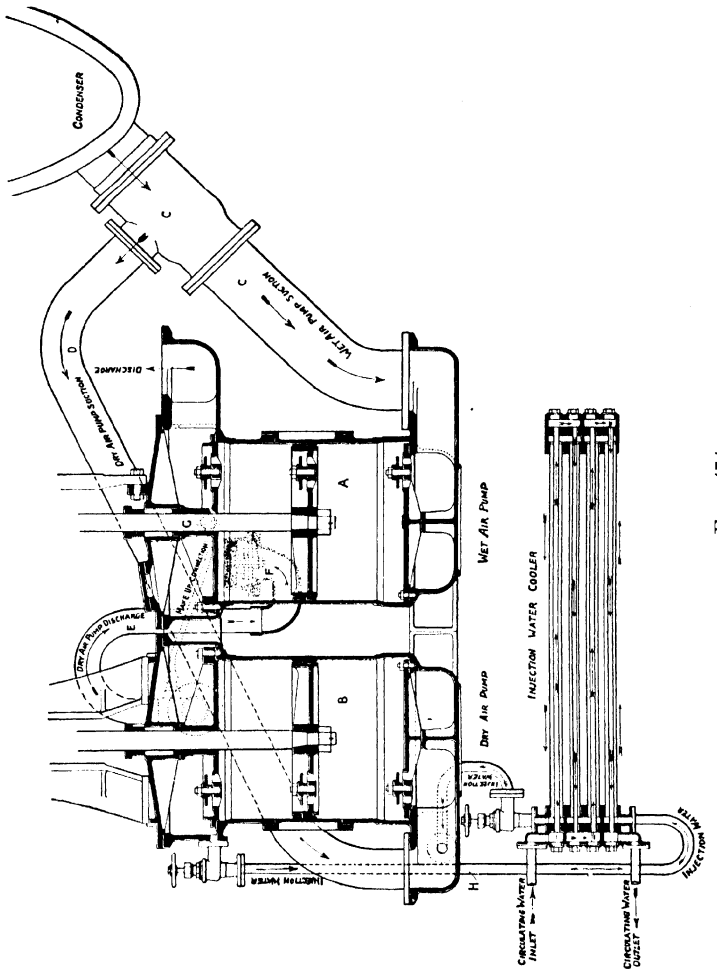



FIG. 474.

valves, from the cooler coils shown underneath. This water is lifted and discharged through the head valves of the dry pump, with the air-vapour mixture, and then flows through the pipe H and the cooler to the suction branch. The cooler is supplied with circulating sea water. In other cases this external cooling arrangement is replaced by the integral cooler in the condenser. This set

is independently and directly driven by an overhead steam cylinder. The steam cylinder and wet air pump pistons are fixed on the same rod. The pump rod of the dry pump is driven from this rod through a rocking lever.

Following Parsons, the French engineer Leblanc developed the rotary form of dry air pump and the steam ejector type, which is now becoming a standard fitting for large power high vacuum installations. The steam operated air ejector is of simple and robust construction, and has no working parts. It occupies a much smaller space than that required for the older forms of air extraction apparatus. The type of design varies with different makers, but the principle of operation is the same in all. A jet of steam is projected at high velocity from a nozzle or nozzles, into a chamber connected to the condenser, entrains the air, vapour, and incondensible gases, and carries them forward into a diffuser in which the pressure is raised, and the whole mass of steam, air, and gases is discharged at higher pressure than that in the entraining chamber. The action is similar to that of the ejector condenser, with steam in place of water.

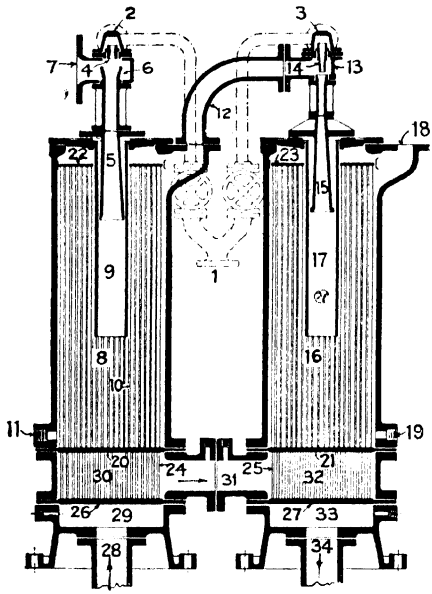



FIG. 475.

In order to obtain a high vacuum it is usual to work two, three, or more of these ejectors in series, and to obtain compression in stages, with improved efficiency. A surface cooler is usually fitted after each stage for condensation of the ejector steam and the cooling of the air. By passing the feed through the coolers on its way to the boiler, these are made to act as surface feed heaters. In fact, they become part of the condensate heating system, which usually includes several regenerative heaters (see Art. 339 and Fig. 409).

A section of the design of two-stage air ejector and surface coolers, made by the Metropolitan-Vickers Co., is shown in Fig. 475. The operating steam is supplied from the pipe 1 through the stop valves (shown dotted) to the nozzle box 2 of the first stage and the nozzle box 3 of the second stage. Referring to the first stage, the steam expands in the nozzle 4, and is discharged at high velocity into the throat of the diffuser 5, which projects upward into the entraining

chamber 6. The branch 7 is connected to the condenser. The air, vapour, and gases, entering at 7, are entrained by the steam jet in crossing the gap above the diffuser. The mixture is compressed and discharged from the diffuser at some pressure between vacuum pressure and atmospheric into the cooler 8 of the stage, and is directed to the bottom by the pipe 9. The steam is condensed by the cooler tubes 10, falls to the bottom, and is drained away at 11, by a syphon pipe, to the condenser. The air and gases pass upward, and are drawn through the pipe 12 into the second stage entraining chamber 13. They are entrained by the steam jet from nozzle 14, compressed in the diffuser 15, to a pressure slightly above atmospheric, and discharged into the cooler 16 of the second stage through the pipe 17. The air is discharged to the atmosphere through the branch 18, and the condensate is drained through the vent 19 to the hot well.

The cooler tubes are of the Field type. The outer tubes 10, closed at the top, are expanded into the tube plates 20 and 21 and are loosely held in place at the top by the guide plates 22 and 23. The inner tubes 24 and 25, open at both ends, are expanded in the tube plates 26 and 27. The condensate from the extraction pump enters through the pipe 28 the first stage lower water box 29, flows upward through the inner tubes, and then downward through the annular space between the tubes to the upper water box 30. It then passes through the connecting pipe 31 to the upper water box 32 of the second stage cooler, flows upward through the annular spaces and down through the inner tubes to the lower water box 33. It is then discharged to the feed line through the pipe 34. The Field tubes, while forming an efficient cooling system, overcome expansion difficulties due to the highly superheated mixture discharged from the diffusers. Both sets of tubes can expand freely, and distortion trouble is avoided. Baffles (not shown) are provided to distribute the mixture after discharge from the central pipes, through the nests of tubes.

A diagrammatic sketch of the arrangement of one of these two-stage ejectors, fitted to a turbine set, is shown in Fig. 476. It will be noted that the first stage syphon drain pipe to the condenser has a long and a short leg, so that the pressure difference between the cooler and condenser may be balanced. When starting up or under light load the feed circulation may be insufficient for cooling, and condensate may be by-passed from the hot well, as indicated by the dotted connection. Also, in order to reduce the time for the creation of the vacuum on starting up, the auxiliary exhauster (shown at the top), which discharges to the atmosphere, is used.

The real interactions between the steam, air, and vapour in this type of extractor are not known with any degree of certainty; and the efficient designs of ejector at present in use have been evolved mainly by experimental trial by the various makers. It is not proposed, therefore, to deal with the "theory" of this apparatus.

The student can refer to the various aspects of this question given in the special texts on condensing plant; and also for detailed information on pumps, de-aerators, feed heaters, evaporators, cooling towers, and other accessories, which cannot be treated here, through lack of space.

**430.** (*e*) **Evaporative Condenser.**—The evaporative type can be used in place of a surface condenser and cooling tower, where the supply of cooling water is restricted. At full load the proportion of heat carried away by the evaporated water and by the air, which may amount to 70 per cent. and 30 per cent. respectively, is about the same as for the cooling tower; but the power required for the circulating water pump is much less (about $\frac{1}{8}$), as the height of the

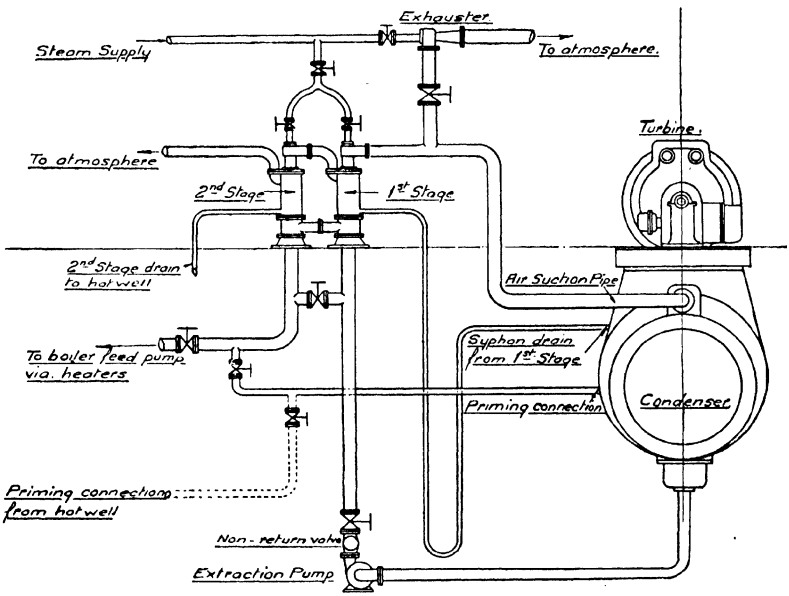



FIG. 476.

lift is about half of that for the tower. Its principal advantage is on light load, when the circulating water supply may be stopped and the condensation accomplished by the cooling action of the atmosphere. In general, however, the vacuum obtainable is lower than that possible with the corresponding surface plant, and it occupies a larger space.

**431. Ledward and Beckett's Evaporative Condenser.**—Side and end elevations of this condenser, as erected at a number of electric light stations, are shown in Figs. 477 and 478. The exhaust steam from the engine or turbine is discharged from the exhaust pipe 1, through a grease separator 2, into the horizontal distributing pipe 3, provided with an automatic relief valve 4. From this main it

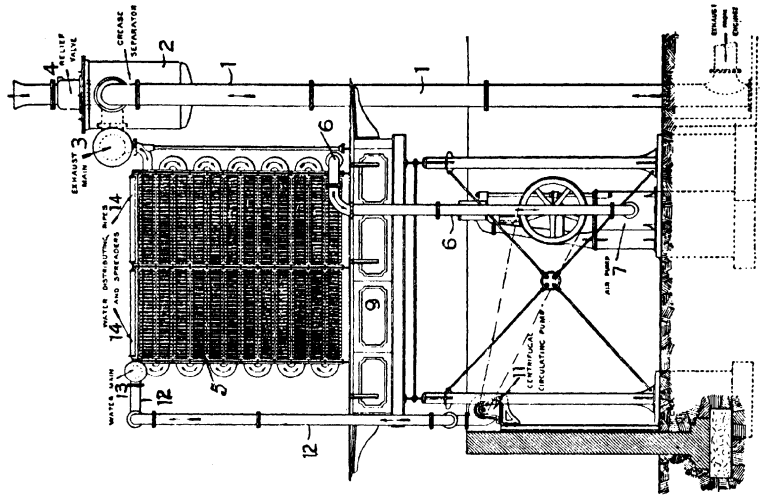



FIG. 478.

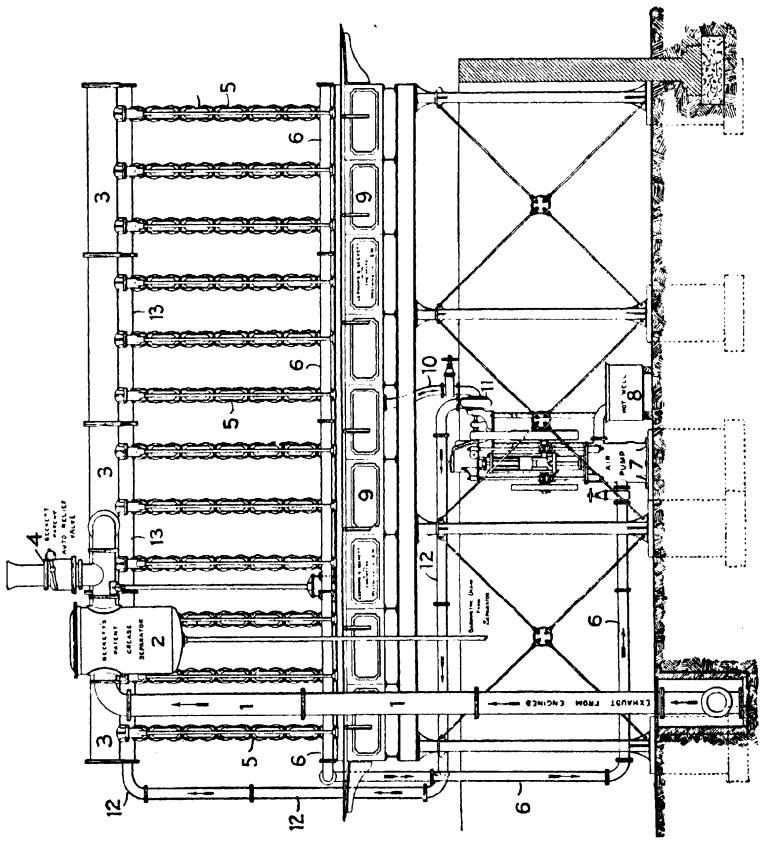



FIG. 477.

enters the upper limbs of a series of flattened coils of corrugated and fluted cast-iron pipes 5. It is progressively condensed as it flows through these, and the final mixture of condensate, air, and vapour is drawn into the suction pipe 6, by the steam-driven wet air pump 7, and discharged to the hot well 8. The condensing water is drawn from a shallow tank 9 through the suction pipe 10, Fig. 477, by a belt-driven centrifugal pump 11, and discharged through the pipe 12 to the water distributing main 13. It passes to water spraying pipes 14 placed right above the corrugated coils. These pipes are provided with deflecting caps having corrugated edges, which distribute the water evenly over the condenser tubes and cover them with a thin film of water. The design of these corrugated tubes, which is the outcome of many years of experience, is such that the water has a perfectly natural flow over the whole tube surface and no part of the surface can become dry, an important condition which is necessary for maximum condensing effect. In falling from the top to the bottom of the condenser a portion of the condensing water is evaporated and carried away with the air. The make-up for this is supplied by a ball cock connected to the outside water service. This ball float keeps the water in the tank at a constant level. In order to obtain a satisfactory air cooling effect the condenser is placed in as exposed a position as possible, preferably higher than the surrounding buildings; and to ensure an effective circulation of the air the corrugated coils are spaced well apart, as shown on Fig. 477. In this case the water tank is placed on I-beams supported on eight braced cast-iron pillars. Where the nature of the load on the prime mover is such that for considerable periods it runs light, a system of stage drainage is adopted. The condensate is trapped from the various coils by special baffle deflectors at different levels, and it is thus prevented from flowing over the lower surfaces of the coils. On light load nearly the whole of the steam is condensed in the upper limbs of the corrugated coils, and by this compartment drainage a higher condensate temperature is obtained without any effect on the vacuum.

In this system in order to prevent any short circuit of the steam, the drainage mains are all connected to a water-sealed vessel from which the water is drawn, either by the wet air pump or a barometric leg.

The makers state that the loss of condensing water by evaporation is equivalent to two-thirds of the total exhaust steam on full load. At lighter load it may fall to 30 per cent.; and at about 30 per cent. full load the water circulation can be stopped and the condensation obtained by air cooling only. This is a valuable feature for some electricity undertakings where there is a light day load and the circulating pump is necessary for the evening load only. It is stated that in some cases a reduction of 70 per cent. in water consumption has been effected.

When a vacuum of 26 in. or over is required, a dry air pump is

used for the air and vapour, and either an extraction pump or a barometric leg is used to transfer the condensate to the hot well. Another merit of this condenser is the low pumping power required. The circulating pump has to handle about one gallon of water per lb. of steam condensed, on a lift of 12 ft. 6 in. (the height from tank water level to the top of the condenser).

The power for the circulating pump represents 0·25 per cent. of the power of the prime mover, and is much less than that required for a cooling tower and surface condenser installation. The circulating pump for a condenser taking the exhaust of, say, a 500 H.P. engine, at 10,000 lb./hr., handles 10,000 gallons/hr. on this height and takes 1·25 H.P. With cooling tower and surface plant, say 40,000 gallons would be pumped at least 25 ft., entailing a pump horse-power of about 10. The additional charge on the working costs in this case will be obvious.

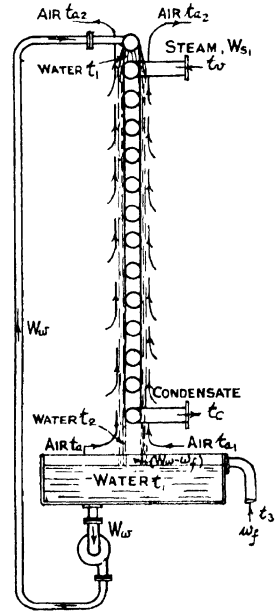


FIG. 479.

**432. Weight of Dry Air, Circulating and Make-up Water for Evaporative Condenser.**—Referring to the diagrammatic sketch, Fig. 479—

Let $W_a$ = weight of dry air passing through the condenser in lb./hr.

$t_{a1}$ = atmospheric temperature ° Fahr. (temperature at which the air comes in contact with the falling water).

$t_{a2}$ = temperature of saturated air at exit from the condenser.

$k$ = relative humidity of atmospheric air.

$p_b$ = atmospheric or barometric pressure = 0·49$h$ lb./in.² abs.

$p_{a1}$ = partial air pressure of atmosphere in lb./in.² abs.

$p_{s1}$ = partial vapour pressure of atmosphere in lb./in.² abs.

Then by Dalton's Law—

$$p_{a1} = (p_b - p_{s1}) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (75)$$

If the air were completely saturated with vapour at $p_b$ and $t_{a1}$, the humidity would be unity or $k$=1, and the partial vapour pressure $p_s'_1$, and from hygrometric laws it follows that the actual partial vapour pressure of the partly saturated air is—

$$p_{s1} = kp_s'_1 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (76)$$

As before, the specific volume of the dry air in the atmosphere at $t_{a1}$ is—

$$v_{a1} = \frac{T_{a1}}{2{\cdot}7p_{a1}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (77)$$

Now with partially saturated air, the vapour, at the low pressures obtaining, is usually slightly superheated and can be treated as a gas. The specific volume of the vapour can then be calculated by using the universal gas constant $\bar{R}=1544$ (see Art. 12, p. 13) and obtaining the lb.-mol. volume $\bar{v}$ at $p_{s1}$ and $t_{a1}$.

$$\text{Thus} \qquad \bar{v}=\frac{1544T_{a1}}{144p_{s1}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (78)$$

Then the specific volume of the vapour is $v_{s1}=\frac{\bar{v}}{m_s}$, where $m_s$ is the molecular weight of the water vapour, or 18.

$$\text{Hence} \qquad v_{s1}=\frac{1544T_{a1}}{18\times 144p_{s1}}=\frac{T_{a1}}{1{\cdot}68p_{s1}} \quad . \quad . \quad . \quad . \quad . \quad (79)$$

The weight of vapour associated with 1 lb. of dry air at $t_{a1}$ is given by—

$$w_{s1}=\frac{v_{a1}}{v_{s1}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (80)$$

The superheat of the vapour is $(t_{a1}-t_{s1})$, where $t_{s1}$ is the saturation temperature at $p_{s1}$ (from tables) and the specific heat may be taken as $c_p=0{\cdot}45$. Hence, reckoned from 32° Fahr., the total heat of $w_{s1}$ lb. of vapour is given by—

$$h_{s1}=w_{s1}[H_1+0{\cdot}45(t_{a1}-t_{s1})] \text{ B.Th.U./lb. dry air} \quad . \quad (81)$$

where $H_1$=total heat of dry vapour at $p_{s1}$.

Reckoning again from 32° Fahr., the heat contained per lb. of dry air passing through the condenser is given by—

$$h_a=0{\cdot}238(t_{a1}-32) \text{ B.Th.U./lb. dry air} \quad . \quad . \quad (82)$$

The total amount of heat entering the condenser, per lb. of dry air, is thus—

$$H_1'=(h_{s1}+h_a)=w_{s1}[H_1+0{\cdot}45(t_{a1}-t_{s1})]+0{\cdot}238(t_{a1}-32) \quad (83)$$

When the air reaches the top or exit from the condenser, after being heated to $t_{a2}$, it may be assumed that it is completely saturated (or $k=1$), and the partial vapour pressure is $p_{s2}$, corresponding to $t_{a2}$ in the tables. The corresponding specific volume $v_{s2}$ is also given by the tables, or calculated approximately from—

$$v_{s2}=\frac{T_{a2}}{1{\cdot}68p_{s2}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (84)$$

$$\text{Again} \qquad p_{a2}=(p_b-p_{s2}) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (85)$$

$$\text{and} \qquad v_{a2}=\frac{T_{a2}}{2{\cdot}7p_{a2}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (86)$$

Then the final weight of vapour, per lb. of dry air at exit, is—

$$w_{s2}=\frac{v_{a2}}{v_{s2}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (87)$$

and the heat carried out per lb. of dry air passing through the condenser is—

$$H_2' = w_{s2}H_2 + 0{\cdot}238(t_{a2} - 32) \quad . \quad . \quad . \quad . \quad (88)$$

where $H_2$=total heat of saturated vapour at $p_{s2}$.

The total amount of heat from the exhaust steam carried away by the saturated air is $H_a' = (H_2' - H_1')$ B.Th.U./lb. dry air,

and $$H_a = W_a H_a' = W_a(H_2' - H_1') \text{ B.Th.U./hour} \quad . \quad (89)$$

Since $w_{s1}$ is carried in and $w_{s2}$ carried out, per lb. of dry air flowing, the amount of water evaporated is—

$$w_w = (w_{s2} - w_{s1}) \text{ lb./lb. of dry air,}$$

and the circulating water feed, or make-up to the tank from the main, is—

$$w_f = W_a w_w = W_a(w_{s2} - w_{s1}) \text{ lb./min.} \quad . \quad . \quad . \quad (90)$$

Let $W_s$=weight of steam condensed in lb./hr.

$t_c$=condensate temperature (10 to 15° Fahr. below vacuum temperature).

$t_v$=vacuum temperature at $p_v$.

$L_v$=latent heat at $p_v$.

$q_v$=quality of exhaust steam at $p_v$.

Then the heat transmitted from the steam to the circulating water and air is—

$$H = W_s[(t_v - t_c) + q_v L_v] \quad . \quad . \quad . \quad . \quad . \quad (91)$$

Let $W_w$=circulating water drawn and discharged by the pump in lb./hr.

$t_1$=temperature of water at pump suction.

$t_2$=temperature of water returned to the tank.

Then weight of water returned is $(W_w - w_f)$ lb./hr., and as this is raised in temperature $(t_2 - t_1)$, the heat from the steam to heat the circulating water is—

$$H_c = (W_w - w_f)(t_2 - t_1) \text{ B.Th.U./hr.} \quad . \quad . \quad . \quad (92)$$

Then $$H = H_a + H_c \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (93)$$

$$W_s[t_v - t_c + q_v L_v] = W_a(H_2' - H_1') + (W_w - w_f)(t_2 - t_1) \quad (94)$$

Now $w_f$ is the amount of make-up to the tank supplied at temperature $t_3$, and it is heated up to $t_1$ by the returned water, so that—

$$(W_w - w_f)(t_2 - t_1) = w_f(t_1 - t_3) \quad . \quad . \quad . \quad . \quad (95)$$

Substituting for $w_f$ from (90)—

$$(W_w - w_f)(t_2 - t_1) = W_a(w_{s2} - w_{s1})(t_1 - t_3) \quad . \quad . \quad (96)$$

Substituting this value in (94) and simplifying—

$$W_a = \frac{W_s[t_v - t_c + q_v L_v]}{(H_2' - H_i') + (w_{s2} - w_{s1})(t_1 - t_3)} \text{ lb. dry air/hr.} \quad (97)$$

The make-up $w_f$ is calculated from Equation 90 and the total water circulated, from (95), is then given by—

$$W_w = w_f\left(1 + \frac{t_1 - t_3}{t_2 - t_1}\right) \quad . \quad . \quad . \quad . \quad . \quad . \quad (98)$$

As already mentioned, the corrugated condenser tubes are the result of many years' experience with this type, and the surface for any given case has to be based on this experience, so that no really reliable method of calculation for the condensing surface can be given.

*Example* 9.—Calculate the weight of dry air, the tank make-up, and the total weight of circulating water of an evaporative condenser working under the following conditions.

Atmospheric pressure 14·7 lb./in.$^2$ abs.; relative humidity of the air 0·85; temperature of tank water 100° Fahr.; temperature of tank feed 50° Fahr. Initial temperature of the air 50° Fahr., and final temperature the same as that of the circulating water at the top of the condenser. Rise of temperature of circulating water in passing from top to bottom, 6° Fahr. Weight of steam condensed 20,000 lb./hr.; vacuum at inlet 26 in.; condensate temperature 10° Fahr. below vacuum temperature; dryness of steam 0·9.

$t_{a1}$ =50° Fahr., $T_{a1}$=510° Fahr., $p_{s1}'$=0·178, $k$=0·85
$p_{s1}=kp_{s1}'=0{\cdot}85\times0{\cdot}178=0{\cdot}151$
$p_b$ =14·7, $p_{a1}=(p_b-p_{s1})=(14{\cdot}7-0{\cdot}151)=14{\cdot}55$

$$v_{a1}=\frac{T_{a1}}{2{\cdot}7p_{a1}}=\frac{510}{2{\cdot}7\times0{\cdot}151}=12{\cdot}95 \text{ ft.}^3\text{/lb.}$$

$$v_{s1}=\frac{T_{a1}}{1{\cdot}68p_{s1}}=\frac{510}{1{\cdot}68\times0{\cdot}15}=2010 \text{ ft.}^3\text{/lb.}$$

$$w_{s1}=\frac{v_{a1}}{v_{s1}}=\frac{12{\cdot}95}{2010}=0{\cdot}00645 \text{ lb. vapour/lb. dry air.}$$

The tank water temperature $t_1$=100° Fahr., so that $t_{a2}$=100 and $p_{s2}$=0·95 lb./in.$^2$ abs., $T_{a2}$=560.

$$p_{a2}=(14{\cdot}7-0{\cdot}95)=13{\cdot}75; \quad v_{a2}=\frac{560}{2{\cdot}7\times13{\cdot}75}=15{\cdot}09,$$

$$v_{s2}=\frac{560}{1{\cdot}68\times0{\cdot}95}=351 \text{ ft.}^3\text{/lb.}$$

$$w_{s2}=\frac{v_{a2}}{v_{s2}}=\frac{15{\cdot}09}{351}=0{\cdot}043 \text{ lb. vapour/lb. dry air}$$

$(w_{s2}-w_{s1})=(0{\cdot}043-0{\cdot}00645)=0{\cdot}03655$ lb./lb. dry air.
$p_{s1}$=0·151, $t_{s1}$=45·56° Fahr., $t_{a1}$=50° Fahr., $H_1$=1075.

$$H_1'=w_{s1}(H_1+0{\cdot}45(t_{a1}-t_{s1})]+0{\cdot}238(t_{a1}-32)$$
$$=0{\cdot}00645[1075+0{\cdot}45\times4{\cdot}5]+0{\cdot}238\times18$$
$$=6{\cdot}89+4{\cdot}284=11{\cdot}174 \text{ B.Th.U.}$$

$p_{s2}$=0·95, $H_2$=1101·6

$$H_2'=w_{s2}H_2+0{\cdot}238(t_{a2}-32)$$
$$=0{\cdot}043\times1101{\cdot}6+0{\cdot}238\times68$$
$$=47{\cdot}369+15{\cdot}64=63 \text{ B.Th.U.}$$

$(H_2'-H_1')=(63-11{\cdot}174)=51{\cdot}826$ B.Th.U.

$$W_a = \frac{W_s[t_v - t_c + q_v L_v]}{(H_2' - H_1') + (w_{s2} - w_{s1})(t_1 - t_3)}$$

$$= \frac{20000[10 + 0{\cdot}9 \times 1021]}{51{\cdot}826 + 0{\cdot}03655 \times 50}$$

$$= \frac{20000 \times 928{\cdot}9}{53{\cdot}65} = 346{,}100 \text{ lb. dry air/hr.}$$

$W_s = 20{,}000$
$t_v = 125{\cdot}4°$ Fahr.
$t_c = 115°$ Fahr.
$q_v = 0{\cdot}9$
$L_v = 1021$
$t_3 = 50°$ Fahr.
$t_1 = 100°$ Fahr.

$$w_f = W_a(w_{s2} - w_{s1})$$
$$= 346{,}100 \times 0{\cdot}03655 = 12{,}650 \text{ lb./hr.}$$

This is $\frac{12650}{20000} = 0{\cdot}632$, or about 63 per cent. of the total steam condensed, which approximates to the value given by the maker.

The total weight of circulating water is—

$$W_w = w_f\left(1 + \frac{t_1 - t_3}{t_2 - t_1}\right)$$
$$= 12650\left(1 + \frac{50}{6}\right)$$
$$= 9{\cdot}33 \times 12650 = 118{,}024 \text{ lb./hr.}$$

$t_2 = 106$
$t_1 = 100$
$t_3 = 50$

The weight ratio of circulating water to steam is thus 5·9.

The temperature of the tank water and the rise of temperature chosen are average values given by the makers.

# CHAPTER XXIV

## STEAM ENGINE AND TURBINE PERFORMANCE

**433.** In the early days of steam prime mover development, the combined boiler and engine or the plant performance was expressed in terms of the fuel burned per horse-power developed by the prime mover. This did not do justice to the engine, as, apart from varying calorific value of the fuel, it involved the performance of the boiler, which might be either good or bad. Thus two engines which might be equally efficient, one supplied by a good and the other by a bad boiler, would show very different plant performances. At a later stage of the development the boiler performance was separated from that of the engine, and the latter was then reckoned in terms of the rate of steam consumption $w_i$ or $w_B$ in lb./H.P. hr. As long as saturated steam from 50 to 150 lb./in.$^2$ gauge was used this figure proved fairly satisfactory, but on the advent of superheated steam and with increasing pressures it ceased to be a valid criterion.

As already pointed out in Chapter IV, the true measure of the prime mover performance is not the rate of steam consumption but the rate of heat consumption Q reckoned in thermal units per unit of power.

This question of engine performance and steam prime mover testing in general has formed the subject of several enquiries by Committees of the Engineering Institutions. These enquiries were the outcome of the classical papers by P. W. Willans in 1888 and 1893 to the Inst. of Civil Engineers on "Engine Testing," and by Captain H. R. Sankey on the "Thermal Efficiency of Steam Engines," in 1896.

The last report on this subject, issued by a joint committee of fourteen engineering and technical societies, in association with the Admiralty, War Office, and Air Ministry, was published in 1927. It comprises sections on Boilers, Steam Engines, and Turbines; Internal Combustion Engines; Calorific Values of Fuels; Measuring Instruments, etc. The section on Steam Boilers, Engines, and Turbines is far too lengthy for reproduction here, and the student is referred to the official publication, entitled *Report on Tabulating the Results of Heat Engine Trials*,* for full details regarding the procedure in testing and tabulating the experimental data, under varying conditions of operation of engines and turbines.

* Published by William Clowes and Sons, Ltd., 94, Jermyn Street, London, S.W.1.

In this final chapter the salient points only of this subject are briefly reviewed.

**434. Methods of obtaining Consumption Test Data.**—For industrial purposes the essential factors for the calculation of performance which have to be ascertained from consumption tests of a reciprocator or turbine are: the steam conditions of pressure, temperature, and vacuum; the indicated and brake horse-power of the reciprocator, or the brake output or electrical output in kilowatts at the shaft or at the generator terminals for a turbo-generator; the steam supply at the H.P. cylinder or the H.P. section of the turbine, as ascertained from the measured condensate. Also in the case of a jacketed engine the weight and temperature of the jacket drain water.

**435. Pressures and Temperatures.**—The gauge pressures before the stop valve, after the throttle valve, at each receiver, and at the exhaust branch close to the engine should be recorded, together with the corresponding temperatures.

All gauges should be calibrated before and after the test, and any necessary corrections made, before the figures are used. Generally the temperatures are recorded by mercury thermometers, although in special cases thermocouples and resistance thermometers are used for highly superheated steam. Thin thermometer pockets, usually of steel, are screwed into the steam pipe or chest or throttle casing, in a vertical position where possible, filled with mercury or lubricating oil, and solid stem theromometers are placed in them. The filling fluid is used to obtain thermal contact between the thermometer bulb and the metal of the pocket. With this system the reading of the thermometer is liable to be on the low side, and it is preferable when possible to place the thermometer in direct contact with the steam. Special glass thermometers of the solid stem type can be obtained to stand temperatures to 750° Fahr. at considerable pressures. For some years the writer has used solid stem thermometers directly for temperatures up to 600° Fahr. under pressures up to 170 lb./in.$^2$ gauge. The thermometer is passed through a deep stuffing box, packed with asbestos cord, with the bulb in the way of the steam. A $\frac{3}{8}$-in. brass tube, with a slot to show the thermometer scale, is screwed on to the stuffing gland and acts both as a guard and an end stopper, to prevent the thermometer from being blown out under the pressure. The top end of the tube is closed by an adjustable screw cap.

The vacuum should, whenever possible, be registered by a mercury gauge manometer. In commercial work when this is not available the ordinary form of Bourdon vacuum gauge is used, and this type requires to be carefully watched. Like the high-pressure gauges, it ought to be checked against a mercury column before and after a test, and the test readings should be corrected in accordance with the calibration.

The vacuum reading should be taken at the exhaust branch, as

close to the engine or turbine as possible. Usually for a reciprocator or a small turbine it is sufficient to screw the nipple of the sampling cock into a hole in the exhaust branch. In the case of a very large turbine installation running under high vacuum this arrangement is not sufficient to give a reliable average of the vacuum pressure across the wide exhaust outlet, and " bus pipes " carrying a set of closed $\frac{1}{4}$-in. sampling tubes, perforated near each end with several $\frac{1}{16}$ in. diameter holes, are placed across the exhaust opening in the plane of the exhaust flange. Readings are also taken from several connections screwed into the wall of the exhaust branch. All readings in these cases are taken on mercury manometers.

**436. Condensate.**—The condensate from the surface condenser, discharged either by a wet air pump or an extraction pump, can be measured either by direct weighing, or in calibrated tanks, or by some form of flow recorder. For small outputs a continuous record can be obtained by placing two tanks on weighing machines and discharging the condensate to each in turn, by means of a swivel pipe, one tank being filled while the other is being emptied into the feed tank, from which the condensate can be drawn by the boiler feed pump. This outfit is expensive and can be replaced by a much simpler and cheaper substitute in the form of two tanks with wooden floats and scale rods, a method originally adopted by Sir A. B. W. Kennedy in the early days of steam engine testing. The floats, made of hard wood, can be well varnished or tarred to prevent absorption of water, and are guided by two vertical wires attached to the bottom of the tank and to a wooden cross-bar at the top through which the square-section scale rod of the float is passed. A brass index plate is fixed on the bar across one face of the rod. The calibration scale is obtained by pouring in definite weights of water and marking the rod positions on each addition against the index plate.

With a uniform tank, preferably circular, the main divisions thus obtained on the rod can then be subdivided as finely as desired. The accuracy of any intermediate reading can be checked by adding the requisite water and noting the scale reading on the rod.

This is a superior arrangement to a tank with a glass gauge, movable index, and scale, which is troublesome to read and liable to get broken. At the time signal the observer, by means of a swivel pipe, switches the discharge on to the measuring tank for which the initial scale reading has been noted. On the next time signal it is switched on to the other tank, which meantime has been emptied and set for its initial reading. The observer then, without hurry, notes the final scale reading, and the difference of the readings gives, directly in lbs., the amount of condensate discharged in the interval of time between the signals.

For turbine installations of large output the continuous weighing or tank measuring methods become impracticable. In the majority of power stations, until the advent of the regenerative and closed

circuit feed system, a continuous record of the steam has been obtained by some form of flow meter, usually a Lea recorder, in which a float in a tank fitted with a V-notch operates, through a special float mechanism, a cylinder with a logarithmic spiral, which records the flow in gallons per hour on the paper of a recording drum. Various forms of steam measuring meters are also in use to directly measure the supply to the turbines.

The adoption in large power stations of the closed circuit with regenerative feed heaters has, within the past ten years, introduced new thermal problems which necessitate special arrangements for the measurement of the condensate. One of the latest developments in this direction has been fully described in a paper to the

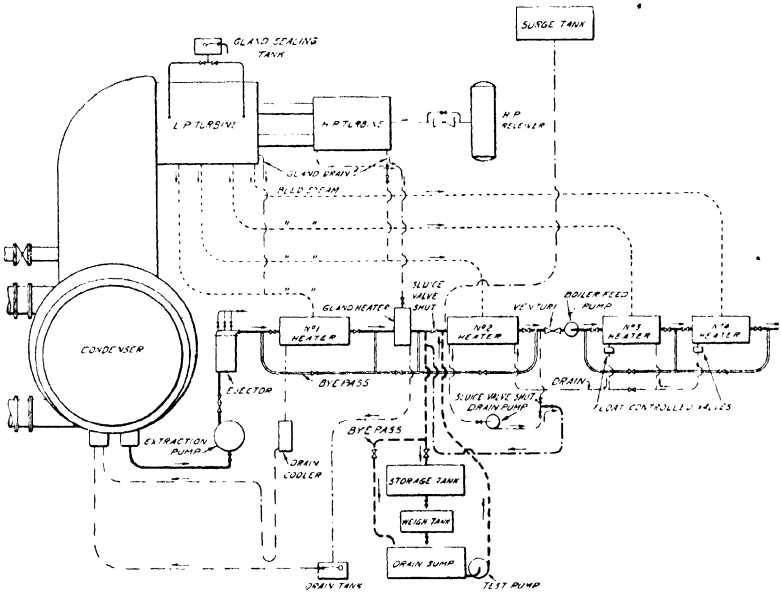



FIG. 480.

Inst. of Mechanical Engineers,* and the diagrammatic arrangement of the scheme, reproduced from this important paper, is shown in Fig. 480. Since the final temperature of the feed leaving the fourth heater was in the neighbourhood of 300° Fahr., or about 100° Fahr. above the boiling temperature at atmospheric pressure, the feed could not be weighed in an open weighing tank. It was therefore intercepted at a point on the feed line between the gland heater and No. 2 regenerative heater, and a shut-off valve was put between the heaters. It was passed down the dotted pipe on the left first

* "Operating Results with the Recent Extensions at Barton Power Station," by H. L. Guy and C. H. Lamb, *Proc. Inst. Mech. E.*, 20 March, 1930.

to a large storage tank, then to a weigh tank having a capacity of about 16 tons. This tank was mounted on a 25-ton Hodgson weighing machine fitted with a ticket-stamping device. After weighing, the condensate was discharged into the sump tank, drawn from this and discharged into the feed line again, through the dotted pipe connection on the right of the shut-off valve. The method of running a weighing test was as follows: prior to the start the feed was passed through the by-pass connection to the sump tank, the inlet valve above the storage tank was kept closed and its outlet valve open. The weight tank outlet valve was closed. At a given signal the by-pass valve was closed and simultaneously the inlet valve of the storage tank was opened and the feed was discharged into this tank and then into the weigh tank. When about 10 tons of water had collected in this tank the outlet valve of the storage tank was shut, the exact weight of the water was read, and the weigh tank was rapidly emptied by opening its outlet valve, to discharge the feed into the sump tank. During this process the feed collected in the storage tank. When the weigh tank was emptied and the outlet valve closed, the outlet valve of the storage tank was again opened and the feed was again discharged into the weigh tank. This process could be continued during any desired interval of time, and the test was terminated at the end of this interval by shutting off the flow to the storage tank, and by-passing the feed again to the sump tank. The water level in the sump tank was accurately observed at the beginning and end of each test.

In marine work consumption tests are sometimes made during trials on the measured mile, the total condensate being measured in tanks. The discharge of the feed pumps is also obtained by calculation from the pump capacity and number of strokes. This value is usually slightly greater than the directly measured quantity, the excess being of the order of about 3 per cent. Manufacturers of steam engine and turbine plant usually have test shop installations for the systematic testing of all engines and turbines for performance before these are sent out to customers. The student will find descriptions of some of the larger works outfits described in the Engineering Societies' Proceedings and in the technical journals.*

In the case of an engine or turbine exhausting into a jet type of condenser, the nearest approach to a direct measurement of the steam consumption is the measurement of the feed to the boiler under steady condition of running, with tight feed pump connections, tight stop and safety valves and pipe joints. Condensation drainage between boiler and stop valve and pump leakage should be subtracted from the feed quantity. The feed can be measured, as before, by drawing alternately from two calibrated tanks. Alternatively by an indirect method, an approximate estimate is obtained by measuring the flow of circulating water and condensate from a tank

* See *Proc. Inst. C.E.*, 1901–02, 1903–04. *Engineering*, 30 September, 1910.

fitted with a V-notch or having a number of orifices in the end, and then equating the heat supply to the steam against the sum of the work done in the engine and the heat taken up by the circulating water. If the common temperature of the mixture is $t_f$, and total heat $h_f$, then by the notation of Chapter XXIII—

$$W_s(H_{s1}-h_f)=2545I+W_w(t_f-t_1)$$

The total flow is $W_T=(W_s+W_w)$, so that—

$$W_s(H_{s1}-h_f)=2545I+(W_T-W_s)(t_f-t_1) \quad . \quad . \quad . \quad (1)$$

and $w_s$ is calculable from (1).

It is not necessary to make any comment on the determination of the power output, as this has been fully dealt with in the earlier chapters.

**437.** When reliable values of the total steam consumption $W_s$ and the power output, either indicated I, or brake B, or electrical K, are obtained from a consumption test, under specified conditions of pressure, temperature, vacuum, and rotational speed, the rates of steam and heat consumption are given by—

| | *Rate of Steam Consumption* | *Rate of Heat Consumption* |
|---|---|---|
| Indicated | $w_i=\frac{W_s}{I}$ lb./I.H.P. hr. | $Q_i=w_i(H_{s1}-h_f)$ B.Th.U./I.H.P. hr. |
| Brake | $w_B=\frac{W_s}{B}$ lb./B.H.P. hr. | $Q_B=w_B(H_{s1}-h_f)$ B.Th.U./B.H.P. hr. |
| Electrical | $w_K=\frac{W_s}{K}$ lb./kw. hr. | $Q_K=w_K(H_{s1}-h_f)$ B.Th.U./kw. hr. |

$t_f$ is taken as the feed temperature on discharge to the boiler and $h_f$ the total heat of the water. For the regenerative cycle $t_{h'2}$ and $h_2'$ are the values (see Art. 392).

**438. Engine Performance under Varying Load.**—A steam prime mover under fluctuating conditions of the external resistance may have to run for considerable periods at reduced load, that is, below the normal load for which it is economically rated. On other occasions it may have to take an overload for a limited period. It is essential, therefore, to ascertain how any given variation of the load affects the performance. The range that is usually of commercial importance is that from half to full load. It is advisable to deal first with the case of the reciprocator, which was systematically investigated by Willans.

**The Willans Line.**—He found that when the total steam consumption of a throttle-governed reciprocator was plotted to a base of indicated mean pressure or indicated horse-power, the locus of the points was a straight line which cut the vertical axis above the origin, that is, above the line of zero consumption. This linear relationship between steam consumption and power has been found to hold generally for saturated and moderately superheated steam, and the throttling consumption line is called the Willans Line, in compliment to Willans, who was the first to draw attention to this

important relationship. It can be shown on theoretical grounds that this linear law should hold good in the case of a throttle-governed engine in which the cut-off, number of expansions, and back pressure remain fixed and only the initial pressure varies. Thus, referring to Fig. 67, p. 135, which shows the theoretical indicator diagram for the engine without clearance—

The steam density can be expressed as a linear function of the pressure, or, $\rho=\alpha+\beta p$. Hence weight per stroke at $p_1$ is

$$W_s'=\rho_1 V_1=V_1(\alpha+\beta p_1)$$

The expansion ratio$=r=$constant.

And M.E.P. $$p_m=p_1\left(\frac{1+\log_e r}{r}\right)-p_b=p_1 X-p_b$$

$$\therefore\ p_1=\frac{p_m+p_b}{X} \quad \text{and} \quad W_s'=V_1\left[\alpha+\beta\frac{p_m+p_b}{X}\right]$$

But $V_1$, $\alpha$, $\beta$, X and $p_b$ are constants.

Hence $$W_s'=a'+b'p_m.$$

Since at constant speed I=const. $p_m$, then at all powers the equation can be written as—

$$W_s=a+bI \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

where $W_s$ is the total steam in lb./hr. and $a$ and $b$ are constants.

This line for a series of trials of a Willans compound central valve engine, using saturated steam, has been replotted in Fig. 481. The referred mean pressure ($p_e$) and the trial indicated horse-power values (I) are set off on the base line OX, and the total steam per hour in lb. ($W_s$) values are plotted against these. The pressures marked on the line BC, are the mean values of the absolute admission pressures at the various loads. The line CB, continued down, cuts the vertical OY at A. The intercept $a$, or the consumption at zero indicated horse-power, Willans defined as the steam required by a frictionless engine doing no work at the revolutions specified. In such a case the indicator diagram *abcde*, Fig. 482, would consist of a positive loop and a negative loop of the same area, so that the net work done on the piston in a revolution would be zero ; but steam would be required to make good the heat loss due to initial condensation, conduction, and radiation, and this quantity is represented by $a$.

The slope of the line BC, Fig. 481, is given by—

$$b=(W_s-a)/I \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

and the indicated rate of steam consumption is—

$$w_i=\frac{W_s}{I}=b+\frac{a}{I} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

In the case illustrated $a=62$ and $b=13{\cdot}061$, and DE, the corre-

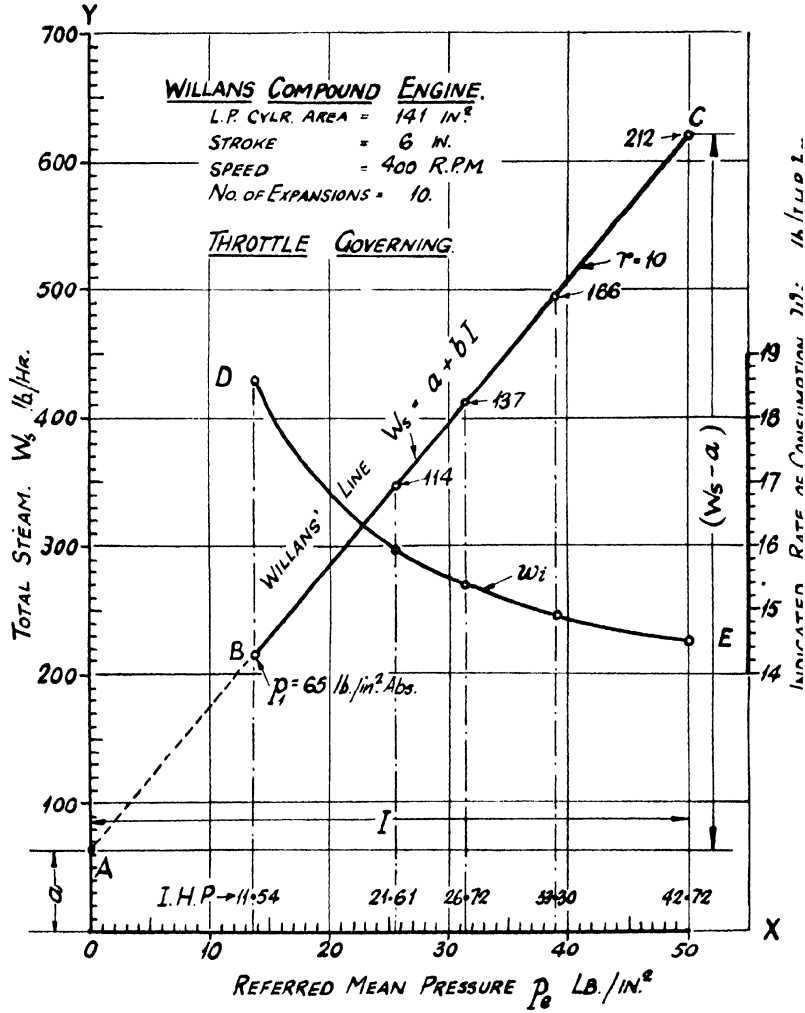



FIG. 481.

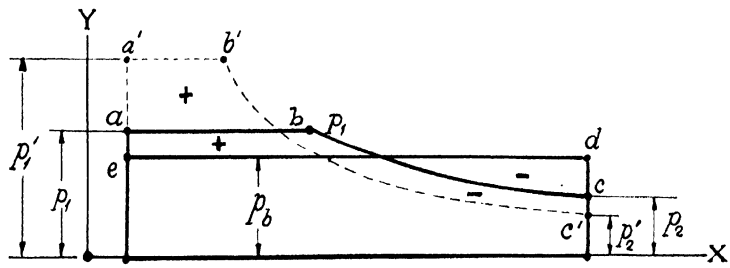



FIG. 482.

sponding curve for $w_i$, is a rectangular hyperbola. It will be obvious from (4) that since $a$ is a constant under throttling conditions, that is, with constant cut-off, the full load is the most economical one at which to run the engine. This fact is also apparent from the consumption rate curve. It should be noted that in graphing the results of a test the mean Willans line should first be drawn to smooth out any slight irregularities, and the consumption rate curve should be derived from the $W_s$ values scaled from the line or from the $w_i$ equation. In this way the true mean curve is obtained. The $w_i$ values from the individual test readings, which

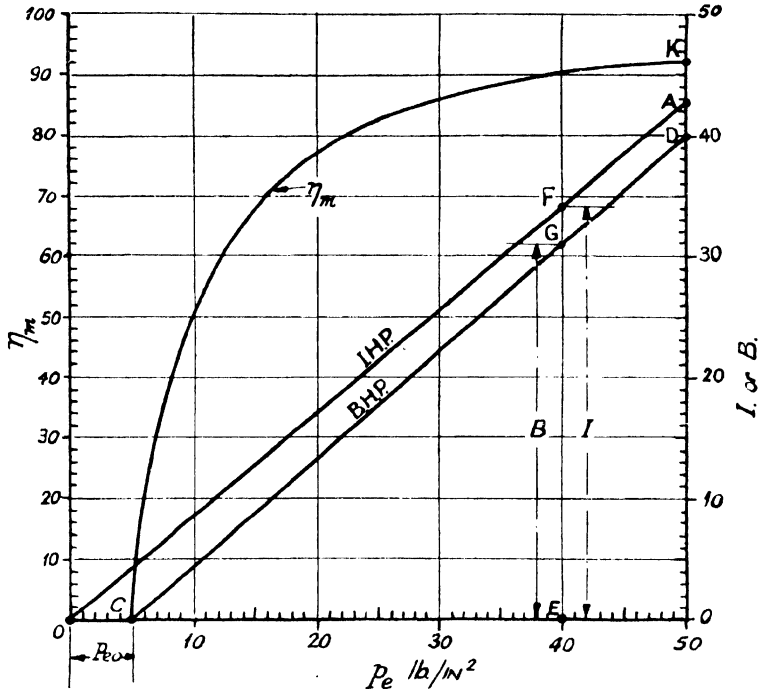



FIG. 483.

may or may not lie exactly on the mean curve, can then be marked on the paper. This method should also be used in plotting curves of heat consumption and thermal efficiency for which the total consumption curve is the basic one. In addition to these thermal curves there is the curve of the mechanical efficiency of the engine, which is dependent on the curve of B.H.P. as the basic reference. It should be derived as follows. Referring to Fig. 483, which was originally drawn by Captain Sankey for the Willans compound engine, set off a scale of mean indicated pressure or its equivalent indicated horse-power on the horizontal axis and a scale of horse-

power to any convenient unit on the right-hand vertical. Set off the I values to this scale, and the graph is the straight line OA passing through the origin O. Then plot the B.H.P. values and draw the mean curve CD through the points. Experience has shown that, in general, the frictional load varies so little between no load and full load that the curve CD does not differ sensibly from a straight line. At any mean pressure, or its equivalent indicated horse-power, the mechanical efficiency is given by the ratio of the intercepts between the base line and the curves CD and OA. Thus at $p_e=40$, $\eta_m=\frac{B}{I}=\frac{EG}{EF}=0{\cdot}905$. Set off a scale of $\eta_m$ on the vertical axis at the left, and to this scale plot the $\eta_m$ values obtained throughout the range of load from the two power curves, and the fair or mean curve CK is obtained. The values of $\eta_m$ calculated directly from the test results can then be plotted.

Owing to the fact that the friction varies only slightly throughout a range of load, the curve of total steam consumption on a B.H.P. base also approximates to a straight line, and it is a common practice to draw the mean Willans line to a base of B.H.P. and a base of kw. output at the generator terminals. Actually the curve is not absolutely straight and tends to turn slightly upward near no load ; but for the relevant range, say from half to full load, it is quite sufficient to take the equation of the mean line to calculate intermediate consumptions and consumption rates.

**439. Effect of Number of Expansions on Steam Consumption.**— When with a given engine, governed on the throttle, a progressive series of tests are run with decreasing cut-off for each series, that is, an increase in the number of expansions or nominal ratio of expansion, the slope of each Willans line decreases with increase in the number of expansions, and in general the value of $a$ increases. These effects are clearly shown by the lines for three sets of Willans trials on his compound central valve engine, Fig. 484. Curve 1 is for 4·8, curve 2 for 10, and curve 3 for 15·5 expansions. The corresponding values of the constants are—

$$a_1=55,\ a_2=62,\ a_3=75\ ;\ b_1=15{\cdot}35,\ b_2=13{\cdot}061,\ b_3=11{\cdot}45.$$

The initial pressures at the full and light loads are again marked on the lines. At full load the pressure has to be raised to 263 lb./in.$^2$ abs. to obtain the same power at 15·5 expansions as against 147 lb./in.$^2$ abs. with 4·8 expansions ; but the indicated rate of steam consumption with the greater expansion ratio is reduced from 16·64 to 13·2 lb./I.H.P. hr. for this particular engine. Some idea of the reason for the increase of $a$ with increased number of expansions can be obtained by considering the theoretical diagrams, Fig. 482, for the frictionless engine doing no work. The indicator diagram with a low rate of expansion, that is, with late cut-off, is shown by *abcde*, the area above the back pressure line *ed* being equal to the area below it. The limits of pressure are $p_1$ and $p_2$. It will

be obvious that for an earlier cut-off a much higher initial pressure $p_1'$ and a lower final pressure $p_2'$ are necessary in order that the plus and minus areas of the dotted diagram $a'b'c'de$ will balance. The wider pressure range will increase radiation, conduction, and cylinder condensation losses.

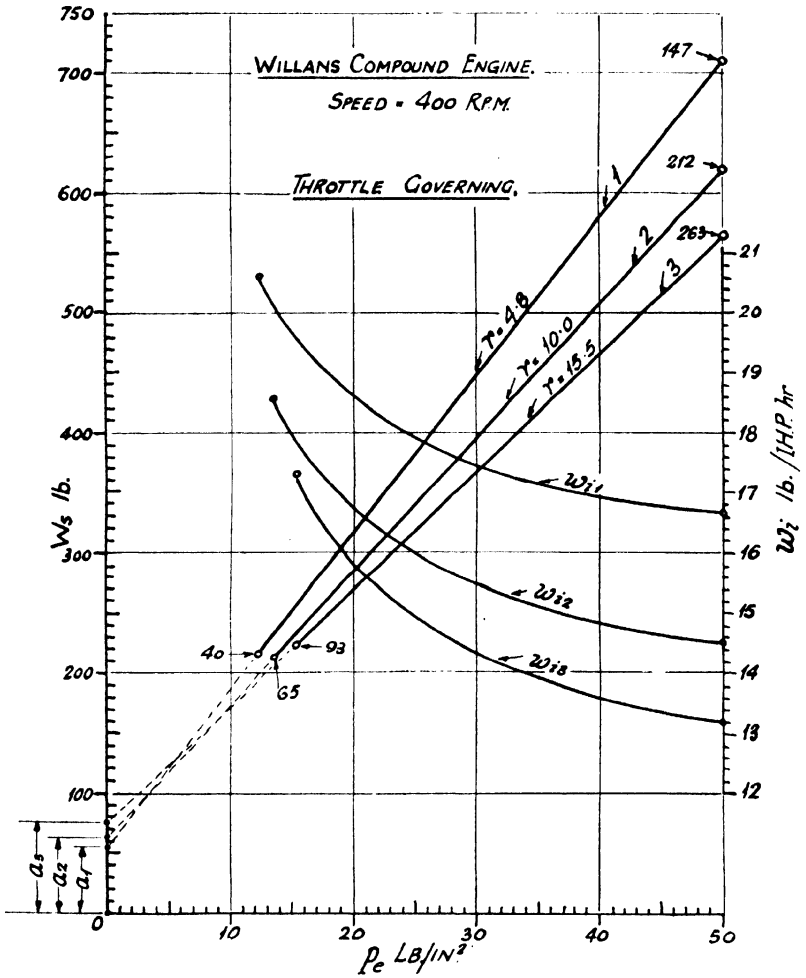



FIG. 484.

**440. Steam Consumption Curve with Expansion Governing.**— In the case illustrated reduced load at constant speed is obtained by throttling or reducing the initial pressure, while the cut-off and number of expansions are kept constant. If the engine is arranged to run with variable cut-off, that is, under expansion governing, the

initial pressure being kept constant at all loads, the graph of the total consumption is changed from a straight line to a flat curve convex to the horizontal axis. This curve can be obtained by drawing the series of Willans lines for several sets of throttling

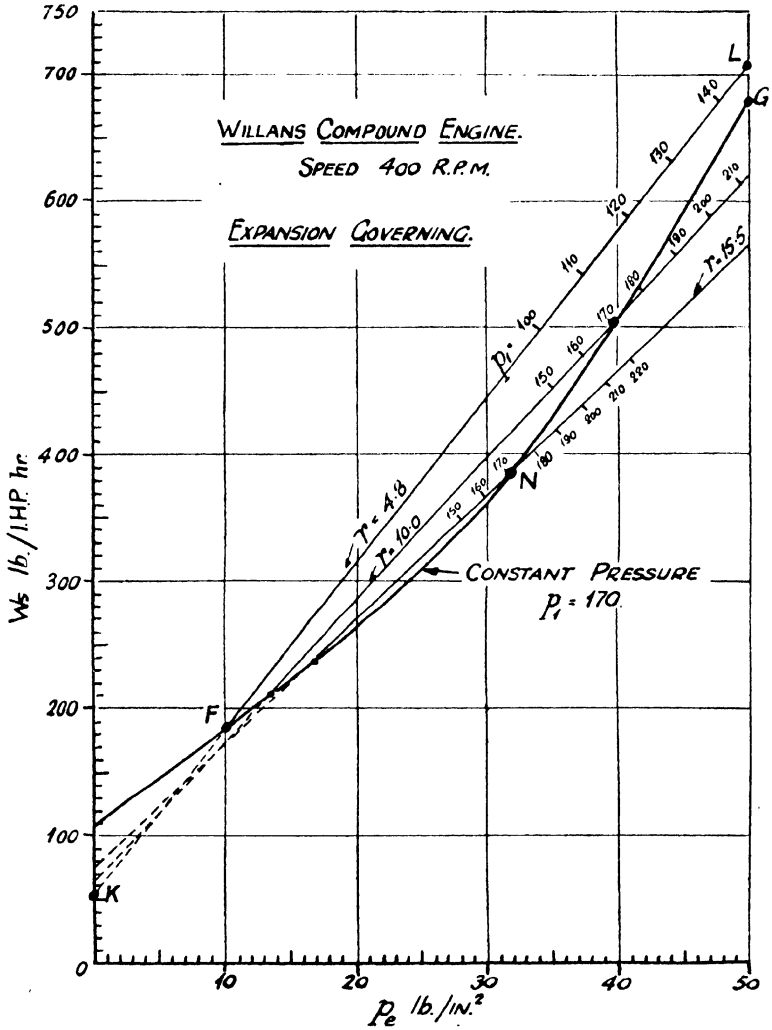



FIG. 485.

trials, corresponding to different expansion ratios, and marking the initial pressures $p_1$ for various loads on each line as shown in Fig. 485, on which the curves of Fig. 484 are reproduced. For any given constant initial pressure $p_1$ at which the engine is run at variable

cut-off, the corresponding consumption curve FNG is obtained by drawing it through the $p_1$ value on each Willans line. In the case illustrated the constant initial pressure is $p_1$=170 lb./in.$^2$ abs. It passes through the 170 value on FL produced. The lower part has to be obtained by drawing lines for greater values of $r$ than 15. This illustration, however, should be sufficient to show the student how to derive the curve from a series of throttling trials, if these are available. Normally the consumptions under cut-off conditions are determined directly on test.

It will be noted that over the range of loads from $p_e$=10 the expansion governing is more economical than the throttle method with $r$=4·8. The vertical intercept between FNG and FL shows the saving at the various loads. The advantage over the throttle conditions for $r$=10 is much less and is confined between the limits of $p_e$=13 and $p_e$=39. In the case of $r$=15·5 the advantage is slight, the limits being roughtly $p_e$=17 and $p_e$=32. Above and below these limiting mean pressures (or equivalent I.H.P.) it is more economical to run this particular engine on the throttle.

**441. Expansion Governing with Compression.**—In the slow-speed Corliss and drop-valve type of engine the variation of cut-off for expansion governing at constant initial pressure, under a fluctuating load, is obtained by the various types of trip gear discussed in Chapter IX, but the compression and release conditions are normally unaffected by this governing. This in the slow-speed type is not a matter of great importance. In the high-speed type, however, as has been shown in Chapter XI, effective cushioning of the piston is necessary, in order to take up the inertia of the reciprocating parts. The control of the piston valve by a shaft governor of the shifting eccentric type accomplishes this end, in addition to the provision of the variation of cut-off. As already shown in Chapter IX, the shifting of the eccentric centre towards the shaft centre reduces the valve travel, with the result that the cut-off, compression, and release all take place earlier. Thus on a light load the high-speed engine controlled by this type has a largely increased compression, which not only is beneficial dynamically but has an important secondary thermal effect, in drying and probably superheating the steam in the clearance. This is bound to reduce initial condensation at the light loads, and some improvement in steam consumption at light load may be expected as a consequence. This effect of compression on the consumption is clearly shown by the set of curves in Fig. 486. The full line total consumption curve was obtained from the Sisson double-crank type of engine illustrated and described in Chapter XVI. The shifting eccentric type of governor used on it is illustrated and described in Chapter X.

The dotted Willans line was obtained from an engine of the same size run on the throttle. The average load on each engine was 50 kw. at the generator, when run non-condensing. The writer has plotted the curves to the equivalent B.H.P. values at the engine

shaft, derived from the data published by the makers. These figures are given in Table I below.

TABLE I

| Load | | Throttling | | Expansion | |
|---|---|---|---|---|---|
| Per cent. | B.H.P. | $W_{ST}$ lb./hr. | $W_{BT}$ lb./B.H.P. hr. | $W_{SE}$ lb./hr. | $W_{BE}$ lb./B.H.P. hr |
| $\frac{1}{4}$ | 22·4 | 1132 | 50·4 | 852 | 38·1 |
| $\frac{1}{2}$ | 39·6 | 1520 | 38·3 | 1132 | 28·5 |
| $\frac{3}{4}$ | 57·5 | 1926 | 33·5 | 1564 | 27·3 |
| 1 | 75 | 2302 | 30·7 | 2100 | 28·1 |
| $1\frac{1}{4}$ | 93·75 | 2670 | 28·5 | 2670 | 28·5 |

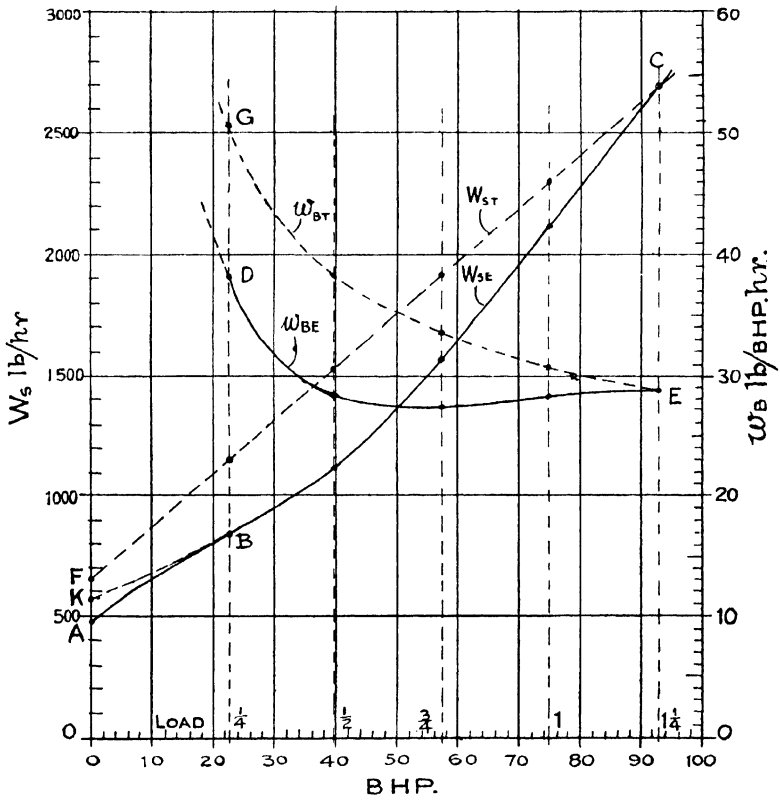



FIG. 486.

The superiority of the expansion governing over the whole range up to 25 per cent. overload is apparent from the intercepts between

the total steam ($W_{SE}$) curve ABC for expansion and the ($W_{ST}$) Willans line FC for the throttling engine. Under normal conditions with an expansion valve gear, such as the Meyer, with constant compression, the total consumption curve $W_{SE}$ would remain convex to the horizontal axis down to the no-load value, as indicated by

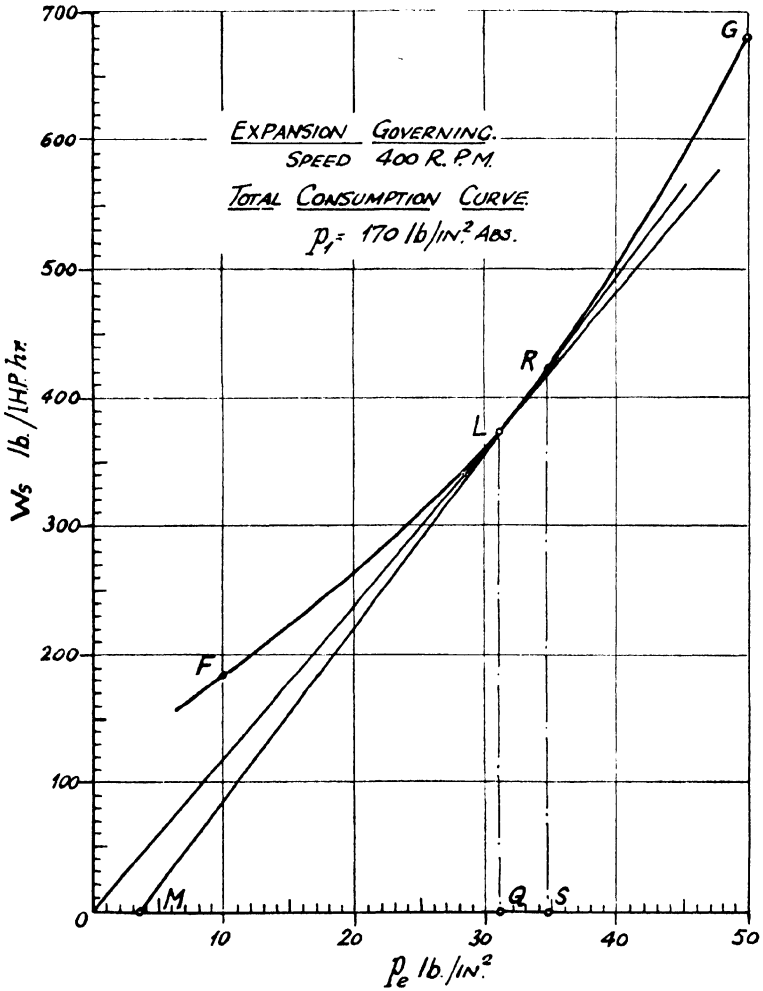



FIG. 487.

the dotted extension BK. The curve obtained with this combined expansion governing and progressive compression, however, changes curvature and becomes concave to the axis at the lower end and is nearly parallel to the Willans line for the throttle-governed engine. The consumption rate curves DE for the expansion-governed and

GE for the throttle-governed engine are derived from the total consumption curves.

In cases of engines in which the load varies largely and unexpectedly, where a heavy overload may be required and the average load is much below the maximum, the expansion and compression governing is much superior to throttle governing, and is essential to secure good economy over a wide range.

**442.** When the total consumption curve FG (Fig. 487) of an expansion-governed engine is given, the mean pressure or the I.H.P. at which the indicated rate of steam consumption is a minimum, can be obtained by drawing a tangent OL from the origin to touch the curve at L. Then OQ is the value of the mean effective pressure

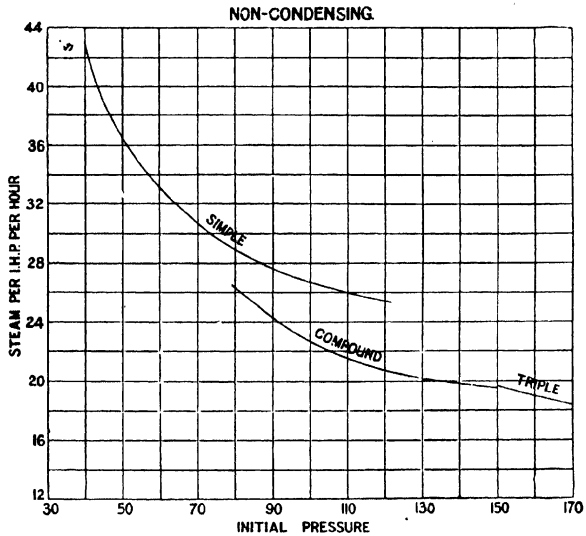



FIG. 488.

(the referred pressure for a multiple-expansion engine). The corresponding value for the brake rate of consumption is obtained by setting off OM equal to the no-load mean effective pressure, or the I.H.P. for the engine running light, and drawing the tangent MR. This fixes the mean pressure OS, referred to the brake or effective horse-power, and gives the minimum consumption rate—

$$w_B = \frac{W_S}{MS}$$

when MS is expressed in horse-power.

**443.** With regard to the general effect of increase of initial pressure on the indicated rates of consumption, this is illustrated for simple, compound, and triple non-condensing engines by the curves of steam rate shown in Fig. 488, and for condensing engines by the

curves of Fig. 489. These curves appeared in the previous editions of this book and are drawn from the results of the trials with saturated steam given in Willans' papers. At each pressure over

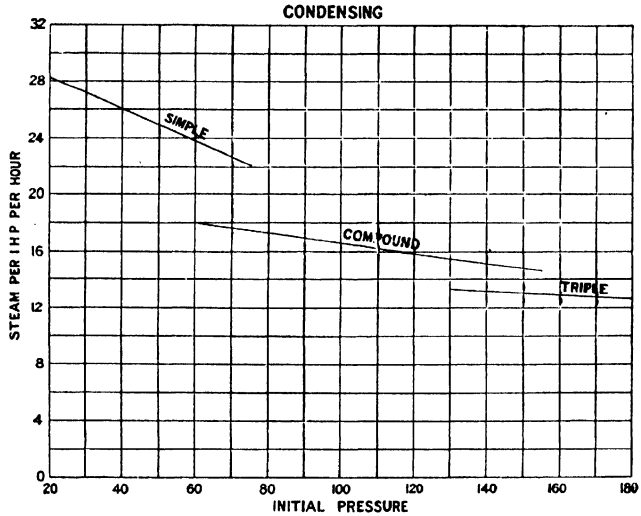



FIG. 489.

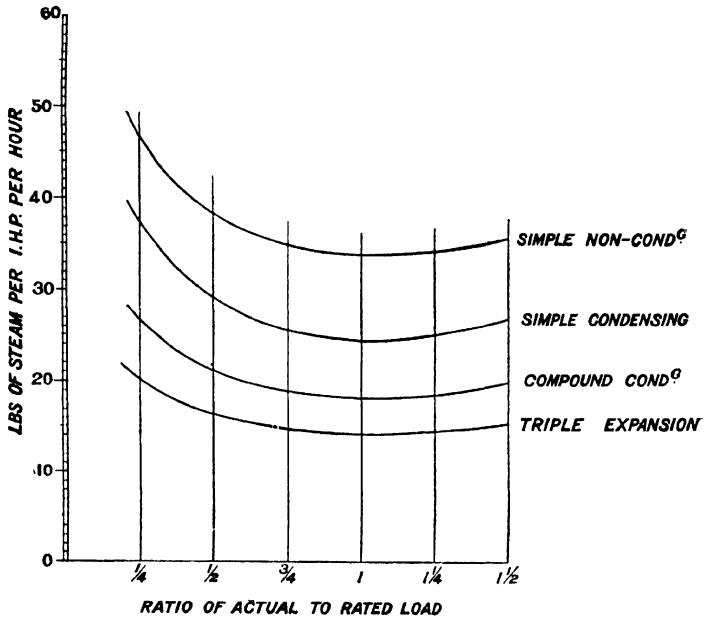



FIG. 490.

the range of the non-condensing tests the number of expansions was kept about the order of $p_1/25$ (see Art. 87) and over the range of the condensing trials about $p_1/14$. The results were all obtained at a constant speed of 400 r.p.m. It will be noted that the general effect of increase of initial pressure is to decrease the steam consumption.

Curves of the average indicated rate of steam consumption for various types using saturated steam, which also appeared in previous editions, are reproduced in Fig. 490. The rated load is taken as the unit of power measurement, and in each case the indicated rate

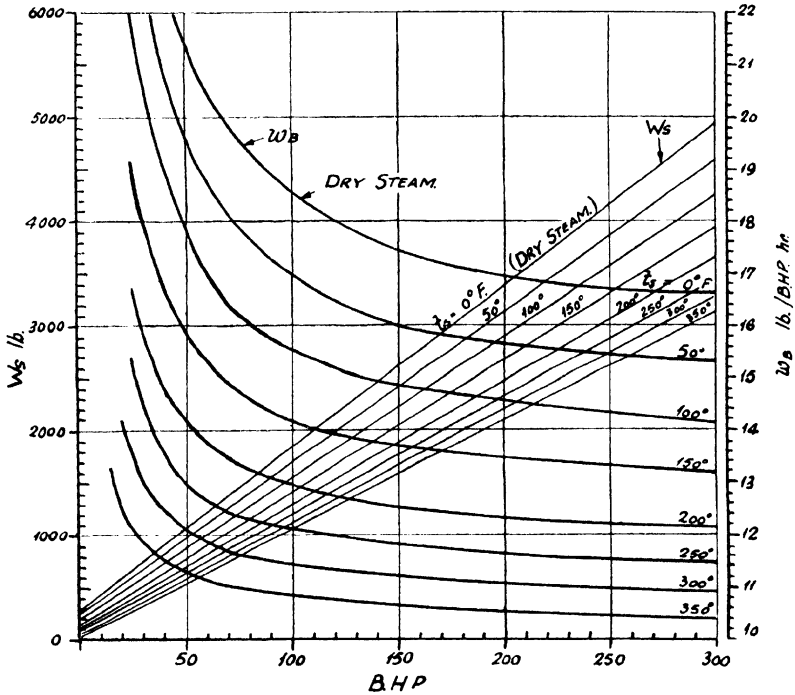



FIG. 491.

of steam consumption has a minimum value at this load. On overload there is, in general, an increase of consumption, as well as on light load.

The general effect of increased rotational speed is to slightly reduce the steam consumption rate, due probably to reduced cylinder condensation and leakage, which are affected by the time element. There is some particular speed at which the consumption reaches a minimum value and above which it increases. This is due to the fact that the loss of power, due to the throttling action caused, balances the reduction of steam per revolution at this speed.

**444. Performance of Engines using Superheated Steam.**—The importance of the use of superheated steam in the reciprocator has already been emphasised in the theoretical discussion in Chapter VI and also in Chapter VII. In order, however, to bring out the general effect of progressive increase of superheat on the performance of a throttle-governed engine, a set of steam consumption curves is shown in Fig. 491. These curves were used to illustrate the remarks by R. T. Smith in the discussion of the First

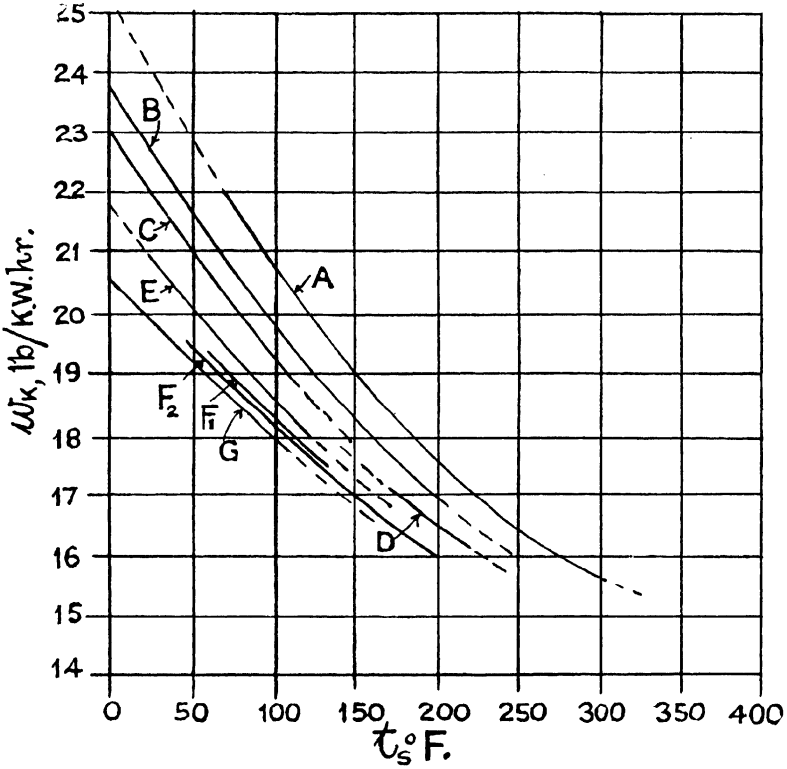



FIG. 492.

Report of the Steam Engine Research Committee of the Inst. of Mech. Eng.* They were obtained from the tests of a non-jacketed high-speed triple-expansion Belliss engine. The Willans lines are plotted to a base of brake horse-power. The number of expansions and the revolutions were kept constant throughout the tests. It will be noted that even on the B.H.P. basis the value of $a$ is a very small proportion of the total consumption at full load. The increasing superheat has the effect of lowering the consumption lines

* *Proc. Inst. Mech. E.*, March 1905, p. 303.

and reducing the slope, thus flattening out the consumption rate curves until at the maximum superheat the consumption rate is nearly constant between half and full load. Taking the steam pressure conditions as 170 lb./in.$^2$ abs. and 26 in. vac. the decrease in steam consumption at 350° Fahr. superheat, over that for saturated steam at full load, is 36·8 per cent. The decrease of heat consumption, however, which is the true measure of improvement, is 31·8 per cent.

Another set of curves (Fig. 492) showing the full load results of a series of seven non-jacketed high-speed triple-expansion engines, taken also from the Inst. of Mech. Eng. *Proceedings*, serves to illustrate the statement made in Chapter VII that all engines of a given type would be equally efficient if steam of very high superheat could be used. In these cases, for which the particulars of the steam conditions are given in Table II, the full load consumption in lb./kw. hr. has been plotted against range of superheat, as the engines drove dynamos directly. In case $F_2$, however, the test was run at three-quarter load. It is obvious that all these curves are converging towards a common minimum value at some higher range of superheat than 400° Fahr. That is, at a sufficiently high value of superheat each engine would have the same consumption and hence the same thermal efficiency.

TABLE II.—NON-JACKETED QUICK REVOLUTION TRIPLE-EXPANSION CONDENSING ENGINES USING SUPERHEATED STEAM. EXPERIMENTS ON BELLISS AND MORCOM'S ENGINES.

| Set | Kw. Output of Generator Coupled to Engine | Load at Test | Stop Valve Steam Pressure, lb./in.$^2$ g. | Vacuum at Engine, Inches of Mercury |
|---|---|---|---|---|
| A | 208 | Full | 155 | 26 |
| B | 220 | ,, | 175 | 25 |
| C | 308 | ,, | 190 | 25 |
| D | 362 | ,, | 162 | 25·8 |
| E | 500 | ,, | 150 | 26 |
| $F_1$ | 700 | ,, | 190 | 27 |
| $F_2$ | 580 | ¾ | 190 | 27 |
| G | 1456 | Full | 183 | 26 |

**445. Methods of Recording Steam Engine Test Data.**—From a complete engine test it should be possible to determine the indicated and brake horse-power; the indicated and brake steam and heat consumptions; the thermal efficiencies and efficiency ratios; mechanical efficiency; and to draw up a heat account.

The following set of log sheets may be used to record the test data required for the calculation of the above figures: I. Pressures and Temperatures. II. Condensate. III. Circulating water. IV. Indicated horse-power. V. Brake horse-power or electrical output.

The items listed will vary with the type of engine and the methods of measurement of the condensate and circulating water employed.

The following are the headings of a set of log sheets used by the writer for a small horizontal compound experimental engine, the condensate and circulating water being measured by the continuous float tank method described in Art. 436, and the output obtained by all-round rope brakes on two water-cooled flywheels. These should serve at least to give the student some idea of the detail procedure of an engine test.

Sheet I.

Trial No.....
Load .....
Date .....

PRESSURES AND TEMPERATURES

| Time | Steam Pressures, lb./in.² g. | | | Steam Temperatures, °F. | | | Vacuum at L.P. Exhaust, ins. | Barometer, ins. |
|---|---|---|---|---|---|---|---|---|
| | Stop Valve | Steam Chest | Receiver | Stop Valve | Steam Chest | Receiver | | |
| | | | | | | | | |
| Total | | | | | | | | |
| Average | | | | | | | | |

Sheet II.

Trial No.....
Load .....
Date .....

CONDENSATE

| Time | Tank A | | Tank B | | Condensate $w_s$ lb./min. | Condensate Temperature $t_c$ Fahr. | Heat in Condensate from 32° Fahr. $H_c$ |
|---|---|---|---|---|---|---|---|
| | $W_A$ lb. | Difference $d$ lb. | $W_B$ lb. | Difference $d$ lb. | | | |
| | | | | | | | |
| | | | | | | Total . | |
| | | | | | | Average | |

Heat left in condensate from 32° Fahr.—
$H_c = W_s(t_c - 32)$ B.Th.U./min.
Weight of condensate from jacket . . . lb./min.
Temperature of condensate from jacket . . ° Fahr.

Sheet III.

Trial No.....
Load .....
Date .....

CIRCULATING WATER

| Time | Tank A | | Tank B | | Water $W_w$ lb./min. | Temperature, °Fahr. | | Heat in Circulating Water $H_w$ |
|---|---|---|---|---|---|---|---|---|
| | $W_A$ lb. | Difference $d$ lb. | $W_B$ lb. | Difference $d$ lb. | | Inlet $t_1$ | Outlet $t_2$ | |
| | | | | | | | | |
| | | | | | | | Total . | |
| Heat carried away in circulating water— $H_w = w_w(t_2 - t_1)$ B.Th.U./min. | | | | | | | Average | |

The rate in each case II and III is obtained by dividing the difference $d$ of the tank reading by the time interval $t$ in minutes.

Sheet IV.

Trial No.....
Load .....
Date .....

INDICATED HORSE-POWER

.....Cylinder

| Time | Counter Reading | Difference $d$ | Speed r.p.m. N | M.E.P. lb./in.² | | | I.H.P. |
|---|---|---|---|---|---|---|---|
| | | | | Head | Crank | Average $p_e$ | |
| | | | | | | | |
| | | | | | | Total . | |
| Cylinder constant $C_c$= I.H.P. = $C_c p_e N$ | | | | | | Average | |

One log sheet is required for the H.P. and another for the L.P. cylinder.

Sheet V.

Trial No.....
Load .....
Date.....

BRAKE HORSE-POWER

| Time | Counter Reading | Difference $d$ | Speed r.p.m. N | Brake Loads | | | | Total Net Load W lb. | B.H.P. |
|---|---|---|---|---|---|---|---|---|---|
| | | | | Left wheel | | Right wheel | | | |
| | | | | $W_L$ lb. | $w_L$ lb. | $W_R$ lb. | $w_R$ lb. | | |
| | | | | | | | | | |
| | | | | | | | | Total . | |
| | | | | | | | | Average | |

Net load on wheels $W=(W_L+W_R)-(w_L+w_R)$.
Brake constant $C_B$=.....
B.H.P.$=C_BWN$

$N=\frac{d}{t}$, where $t$ is the time interval in minutes.

The N values should be the same on IV and V.

A convenient interval between times of reading is ten minutes. All readings should be taken simultaneously on a time signal. An electric gong bell, operated from contacts on a clock face, is most convenient for this purpose.

The results for Sheets II, III, IV, and V should be worked out and entered in the last columns as the trial proceeds. The heat account should show the following items, preferably in the tabular form given below.

| Cr. | B.Th.U./min. | Per cent. | Dr. | B.Th.U./min. | Per cent. |
|---|---|---|---|---|---|
| Heat supplied (from 32° F.) $H_T=w_sH_1$ | | | B.H.P. heat, $H_b=42{\cdot}4B$<br>Friction heat—<br>$H_f=42{\cdot}4(1-B)$<br>Circulating water heat—<br>$H_w=w_w(t_2-t_1)$<br>Condensate heat—<br>$H_s=w_s(t_c-32)$<br>Balance (by difference) . . . . $H_r$<br>Radiation, errors, etc. $H_r$ | | |
| Total . . . | | 100 | Total . . . | | 100 |

$H_1$ is the total heat of dry steam at stop valve pressure $p_1$, or $H_{s1}$ if it is superheated and $H_{w1}$ if wet.

The balance is given by—

$$H_r=H_T-[H_b+H_f+H_s+H_w]$$

When the engine is jacketed, the jacket drain $w$ has to be included in $w_s$, and the additional item $H_J=w_J(t_J-32)$ entered on the debit column, where $t_J$ is the temperature of the jacket drain.

The temperature datum level from which the heat quantities are reckoned is 32° Fahr., as this is the datum for the steam table values.

When a series of progressive loads is run (usually from $\frac{1}{4}$ load to full load) a set of curves on either an I.H.P. or a B.H.P. base should be plotted. Each set should include: total steam consumption in lb./hr.; rate of steam consumption; rate of heat consumption; mechanical efficiency; thermal efficiency; efficiency ratio (on straight Rankine basis). The stop valve and steam chest pressures can also be plotted and the points joined by straight lines.

The following examples refer to tests run on the compound horizontal engine for which the log sheet headings are given above.

This engine, which has cylinders $\frac{5\frac{1}{2}\text{ in.}+9\text{ in.}}{12\text{ in. str.}}$, can develop 23 B.H.P. at normal speed of 150 r.p.m.; with steam conditions 120 lb./in.$^2$ gauge (saturated) and 23 in. vacuum. The H.P. cylinder is fitted with an automatic drop-valve gear controlled by a Proell governor, so that the engine can be expansion governed on varying load, and with a Meyer valve gear on the L.P. cylinder.

A Pickering governor is also fitted, so that the Proell governor can be thrown out of action and the engine can be run on the throttle. For some experimental purposes it has been found convenient to disconnect the Proell and run the engine with the throttle governor in action and to adjust the rod of the drop-valve tappet gear, by hand, until the required initial pressure at the rated speed is obtained with a given load on the brake. In this way a set of expansion trials with constant initial pressure and varying cut-off can be carried out. For throttling trials the H.P. cut-off can be set at any desired value and kept at this throughout the tests. The Meyer valve gear setting is kept constant.

The engine is jacketed on the cylinder barrels and ends, but these tests were run with the jackets out of action.

*Example* 1.—During a full-load non-jacketed test of a compound engine using saturated steam the following average values were given on the log sheets:

| | |
|---|---|
| Stop valve pressure | 140 lb./in.$^2$ g. |
| „ temperature | 358° F. |
| Steam chest pressure | 120 lb./in.$^2$ g. |
| „ temperature | 349° F. |
| Speed | 150 r.p.m. |
| Vacuum at L.P. exhaust | 22½ in. |
| Temperature at L.P. exhaust | 147° F. |
| Barometer | 29·6 in. |
| I.H.P. | 25 |
| B.H.P. | 23·6 |
| Weight of condensate (average) | 9·4 lb./min. |
| Temperature of condensate (average) | 96·5° F. |
| Weight of circulating water (average) | 152 lb./min. |
| Temperature of circulating water, inlet | 59° F. |
| „ „ „ outlet | 120° F. |

Heat left in condensate—
(Average from log sheet II) . . . . . 605 B.Th.U./min.
Heat carried away in circulating water—
(Average from log sheet III) . . . . . 8909 B.Th.U./min.

Calculate: (*a*) the heat supply, from 32° Fahr., on the assumption that the steam in the chest before admission to the H.P. cylinder is 2 per cent. wet; (*b*) the mechanical efficiency; the indicated and brake values of (*c*) steam consumption rate, (*d*) thermal efficiency, (*e*) heat consumption rate, (*f*) efficiency ratio on the Rankine cycle, based on the steam chest pressure.

Draw up the heat account in tabular form.

(*a*) $p_1=134{\cdot}5$ lb./in.$^2$ abs., $H_1=1196{\cdot}3$, $L_1=874{\cdot}5$, $q_1=0{\cdot}98$.

$\therefore H_{w1}=H_1(1-q_1)L_1=1196{\cdot}3-0{\cdot}02\times 874{\cdot}5$
$=1196{\cdot}3-17{\cdot}49=1178{\cdot}8$

$H_T=w_sH_{w1}$ $\qquad w_s=9{\cdot}4$
$=9{\cdot}4\times 1178{\cdot}8=11081$ B.Th.U./min.

(*b*) $\eta_m=\dfrac{B}{I}=\dfrac{23{\cdot}6}{25}=0{\cdot}945$

(*c*) $w_i=\dfrac{60w_s}{I}=\dfrac{60\times 9{\cdot}4}{25}=22{\cdot}6$ lb./I.H.P. hr. $\qquad$ I.H.P.$=25$, B.H.P.$=23{\cdot}6$

$w_B=\dfrac{60w_s}{B}=\dfrac{60\times 9{\cdot}4}{23{\cdot}6}=23{\cdot}9$ lb./B.H.P. hr.

(*d*) $\eta_I=\dfrac{2545}{w_i(H_{w1}-h_2)}$
$=\dfrac{2545}{22{\cdot}6\times 1063{\cdot}8}=0{\cdot}106$

Exhaust temperature—
$t_2=147°$ Fahr. and $h_2=115$
$(H_{w1}-h_2)=(1178{\cdot}8-115)$
$=1063{\cdot}8$

$\eta_B=\dfrac{2545}{w_B(H_{w1}-h_2)}$
$=\dfrac{2545}{23{\cdot}9\times 1063{\cdot}8}=0{\cdot}10$

(*e*) Heat consumption—

$Q_I=w_i(H_{w1}-h_2)=22{\cdot}6\times 1063{\cdot}8=24{,}000$ B.Th.U./I.H.P. hr.

$Q_B=w_B(H_{w1}-h_2)=23{\cdot}9\times 1063{\cdot}8=25{,}450$ B.Th.U./B.H.P. hr.

(*f*) The initial pressure is taken as $p_1=134{\cdot}5$ lb./in.$^2$ abs. and the exhaust pressure as $p_2=3{\cdot}5$ lb./in.$^2$ The adiabatic heat drop between these limits, on the HΦ chart, is $H_\phi=243$.

$\therefore$ Absolute thermal (Rankine) efficiency is—

$$\eta=\frac{H_\phi}{(H_{w1}-h_2)}=\frac{243}{1063{\cdot}8}=0{\cdot}228$$

$$\therefore \epsilon_I=\frac{\eta_I}{\eta}=\frac{0{\cdot}106}{0{\cdot}228}=0{\cdot}463$$

$$\epsilon_B=\frac{\eta_B}{\eta}=\frac{0{\cdot}10}{0{\cdot}228}=0{\cdot}447$$

Heat account items—

The total heat supply from 32° Fahr. is $H_T$=11081 B.Th.U./min.

B.H.P. heat $H_b$=42·4×B=42·4×23·6=1000 B.Th.U./min.

Friction heat $H_f$=42·4(I—B)=42·4×1·4=59 B.Th.U./min.

The average values of the heat in condensate and heat in circulating water are to be taken from the log sheets.

These are $H_s$=606 B.Th.U./min. and $H_w$=8909 B.Th.U./min.

The balance is—

$$H_r=H_T-[H_b+H_f+H_s+H_w]$$
$$=11081-[1000+59+606+8909]$$
$$=(11081-10574)=507 \text{ B.Th.U./min.}$$

HEAT ACCOUNT

| | B.Th.U./min. | Per cent. | | B.Th.U./min. | Per cent. |
|---|---|---|---|---|---|
| Heat supplied to steam from 32° F. | 11081 | 100 | B.H.P. heat | 1000 | 9·03 |
| | | | Friction heat | 59 | 0·53 |
| | | | Heat in condensate | 606 | 5·46 |
| | | | Heat in circulating water | 8909 | 80·40 |
| | | | Balance, radiation, errors, etc. | 507 | 4·58 |
| Total | 11081 | 100 | Total | 11081 | 100 |

*Example* 2.—The compound engine (Example 1) when run at full, three-quarter, and half load, at 150 r.p.m. with constant initial (chest) pressure of 120 lb./in.$^2$ gauge, vacuum 22½ in., and with H.P. cut-off adjusted to suit each load, gave the following results:

| | *Fractional Load* | *Full* | ¾ | ½ |
|---|---|---|---|---|
| I.H.P. | | 25 | 18·3 | 13·2 |
| B.H.P. | | 23·6 | 17·2 | 12·2 |
| $\eta_m$ | | 0·945 | 0·94 | 0·925 |
| $W_s$ | | 564 | 452 | 355 lb./hr. |

Draw the following set of curves on an I.H.P. base. Total steam consumption; rate of steam consumption; rate of heat consumption; thermal efficiency; efficiency ratio; mechanical efficiency.

The values for the three sets of test results taken from the log sheets are as given below—

| I | 25 | 18·3 | 13·2 |
|---|---|---|---|
| $w_i=\frac{W_s}{I}$ | 22·6 | 24·7 | 26·9 lb./I.H.P. hr. |
| $w_B=\frac{W_s}{B}$ | 23·9 | 26·3 | 29·2 lb./B.H.P. hr. |
| $\eta_I=\frac{2545}{w_i H}$ | 0·106 | 0·097 | 0·089; H=1063·8 |
| $\eta_B=\frac{2545}{w_B H}$ | 0·10 | 0·091 | 0·082 |

| I | 25 | 18·3 | 13·2 | |
|---|---|---|---|---|
| $Q_I = \frac{2545}{\eta_I}$ | 24,000 | 26,237 | 28,600 | B.Th.U./I.H.P. hr. |
| $Q_B = \frac{2545}{\eta_B}$ | 25,450 | 27,967 | 31,037 | ,, |
| $\epsilon_I = \frac{\eta_I}{\eta}$ | 0·463 | 0·425 | 0·390 | |
| $\epsilon_B = \frac{\eta_B}{\eta}$ | 0·447 | 0·400 | 0·360 | |

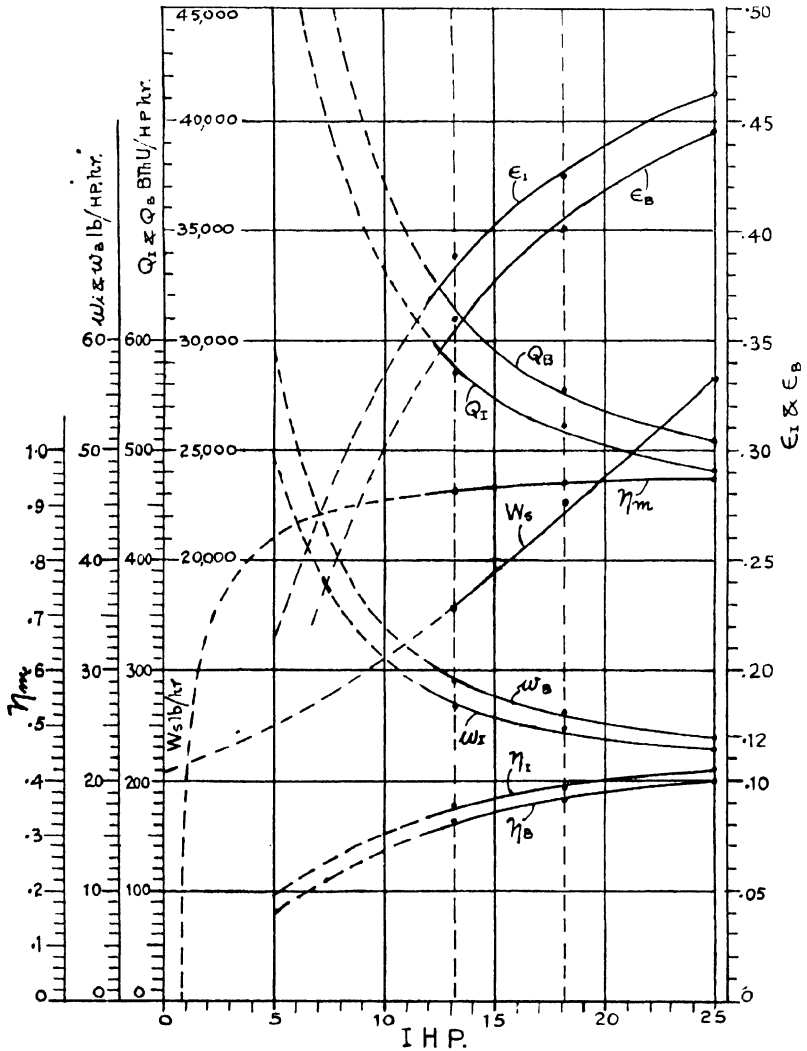



FIG. 493.

The set of curves is shown in Fig. 493. The total steam consumption ($W_s$) curve, according to the methods already described, is drawn first. The three-quarter load consumption seems to be a little out, and the curve has been drawn between the full and half load values and continued (dotted) in the probable direction it would take at lower loads, since the governing is by expansion. From this curve the fair consumption $w_i$ and $w_B$ curves have been obtained. The B.H.P. line, from which the fair mechanical efficiency $\eta_m$ curve has been derived, is omitted in order to avoid confusion with the other curves. The fair curves for efficiency ratio and heat consumption are derived from the consumption rate curves, and the calculated values given above are shown by the dots on the various curves. The student should work out this example and plot the set of curves, and also a second set on a base of B.H.P. It should be noted that the curve of mechanical efficiency, on a base of I.H.P., starts at the "no load" I.H.P. value; when plotted on a B.H.P. base it starts at the origin. This diagram shows the interdependence of the different quantities more clearly than any table of values, and enables any intermediate value between test results to be obtained at once by inspection.

**446. Steam and Heat Consumptions of Various Types of Reciprocators.**—Published data on the performances of the modern types of reciprocating steam engines are scanty, and are often incomplete. The following figures and curves may serve to give some indication of the general trend towards greater heat economy.

In a paper on water-works pumping engines * the following particulars in Table IX of tests on the vertical triple-expansion slow-speed type, with drop valve gears were given. The results were obtained from engines installed between 1923 and 1930, and may thus be taken as representative of the best conditions of operation

TABLE IX

| Year | Size | Pressure, lb./in.² g. | Superheat, °F. | Vac., ins. | Speed, r.p.m. N | P.H.P. | Mechanical Efficiency $\eta_m$ | Steam Consumption $w_B$ | Heat Consumption $Q_B$ | Brake Thermal Efficiency $\eta_B$ |
|---|---|---|---|---|---|---|---|---|---|---|
| 1923 | $\frac{16''+28''+46''}{3'\ 6''\ \text{str.}}$ | 165 | 50 | 28·5 | 25·3 | 181 | 0·9 | 12·91 | 15,150 | 16·8 |
| 1925 | $\frac{24''+44''+66''}{4'\ 0''\ \text{str.}}$ | 181 | 124 | 27·7 | 28·8 | 530 | 0·936 | 10·6 | 12,725 | 20·0 |
| 1927 | $\frac{25''+47''+70''}{5'\ 0''\ \text{str.}}$ | 202 | 100 | 28·1 | 28·1 | 805 | 0·947 | 10·38 | 12,420 | 20·5 |
| 1930 | $\frac{29''+54''+86''}{5'\ 6''\ \text{str.}}$ | 199 | 91 | 28·3 | 21·9 | 762 | 0·974 | 9·89 | 11,800 | 21·6 |

* "The Selection and Operation of Pumping Machinery for Water Works," by F. E. F. Durham, M.I.Mech.E., *Inst. of Water Engineers*, 28 May, 1930, or *Engineering*, 20 June, 1930.

of this type at the present time. The results are all referred to the net output or pump horse-power. The writer has added the heat consumption rates corresponding to the thermal efficiencies given.

The rates of steam and heat consumption and mechanical efficiency, for a considerable number of pumping engines of the foregoing type, working at the standard steam conditions of 200 lb./in.$^2$ gauge, 100° Fahr. and 28 in. vacuum, are shown by the curves of Fig. 494, for outputs ranging from 100 to 1400 P.H.P. It will be noted that at the highest output the heat consumption rate falls

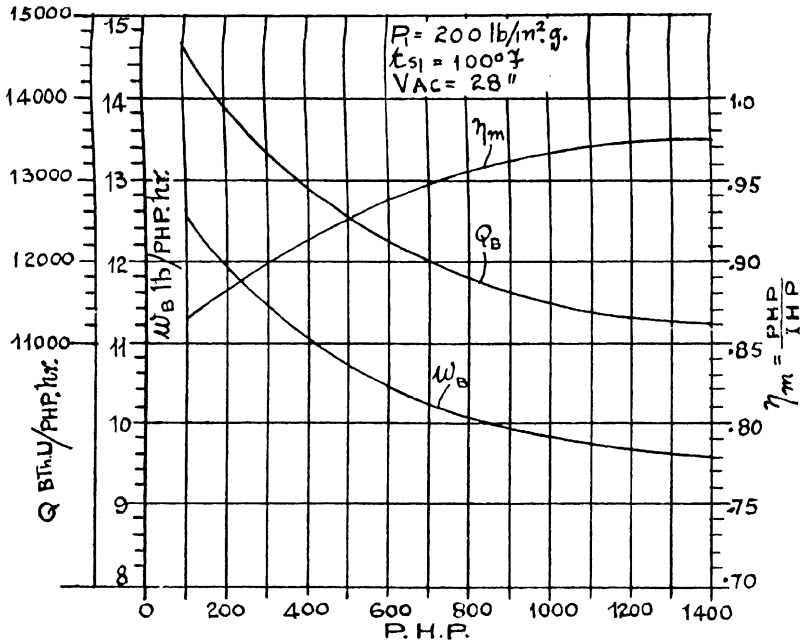



FIG. 494.

to 11,200 B.Th.U./P.H.P. hr. and high mechanical efficiency is obtained.

The following test results are given in Table X for a series of Belliss high-speed triple-expansion engines working with superheated steam (see Chapter VII). The kilowatt output is that at the generator, or the brake horse-power output corrected for the generator efficiency. In calculating the brake rates of heat consumption and thermal efficiency the lower temperature limit ($t_2$) has been taken as that at the engine exhaust.

**447.** With regard to the performance of the horizontal slow-speed type Corliss or drop-valve engine for mill drive, Foster gives * the following limits for steam consumption rates.

* See Kempe's *Engineers' Year Book*, section xxvii.

TABLE X

| Pressure, lb./in.² g. | Super-heat, °F. | Vac., ins. | Speed, r.p.m. | B.H.P. | Kw. | Steam Consumption | | Heat Consumption | | Thermal Efficiency $\eta_B$ | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | | $w_B$ | $w$ | $Q_B$ | $Q_K$ | Brake | Elect. |
| 183 | 66 | 25·6 | 200 | 2050 | 1436 | 13·25 | 18·88 | 15113 | 21534 | 16·8 | 11·82 |
| 175 | 144 | 25·8 | 250 | 1080 | 736 | 11·45 | 16·3 | 13596 | 19355 | 18·71 | 13·14 |
| 181 | 226 | 26·2 | 230 | 1071 | 751 | 10·95 | 15·6 | 14617 | 20824 | 17·4 | 12·22 |
| 169 | 117 | 26·6 | 230 | 865 | 606 | 11·95 | 17·06 | 14089 | 20113 | 18·0 | 12·59 |
| 188 | 98 | 27·5 | 250 | 715 | 500 | 12·3 | 17·5 | 14348 | 20699 | 17·49 | 12·29 |
| 161 | 165 | 25·0 | 343 | 570 | 397 | 11·65 | 16·7 | 13863 | 19873 | 18·35 | 12·50 |
| 160 | 307 | 26·7 | 475 | 300 | 208 | 10·75 | 15·45 | 13824 | 19869 | 18·41 | 12·80 |

Triple condensing, with standard conditions of 180 lb./in.² gauge, 150° Fahr. superheat, 26–27 in. vacuum; indicated rate of steam consumption on normal load, 10·2–11 lb./I.H.P. hr. When run on saturated steam the figure is increased to 12–12·5. Taking the lower rate with 27 in. vacuum, the corresponding heat consumption rate is $Q_i$=12,260 B.Th.U./I.H.P. hr. for the superheated and $Q_i$=13,416 B.Th.U./I.H.P. hr. for the saturated steam. For comparison with the previous sets these figures have to be corrected for the mechanical efficiency, which may be given an average of, say, 0·93. The consumption rates then are $Q_B$=13,200 for the superheated and $Q_B$=14,426 for the saturated steam.

In compound condensing engines with steam conditions 160 lb./in.², 150° Fahr. superheat, 26–27 in. vacuum, $w_i$=10·5–11 lb./I.H.P. hr. With saturated steam the increased figure is $w_i$=13–13·5 lb./I.H.P. hr. The corresponding corrected heat consumptions on the lower figures are $Q_B$=13,525 superheated and $Q_B$=15,600 saturated. With an initial pressure of 120 lb./in.² gauge, other conditions the same, the rates are given as $w_i$=11·5–12·5 with superheated, and $w_i$=14–15 with saturated steam. The corresponding lower values of the brake heat consumption rates are $w_B$=14,740 superheated, $w_B$=16,724 saturated.

The uniflow type is usually run at pressures from 170 lb./in.² to 200 lb./in.² gauge and superheat 160° Fahr. to 200° Fahr.

The following test results are from a three-cylinder compound uniflow marine engine by Gebr. Stork & Co.* This engine has two L.P. uniflow cylinders, and an ordinary contra-flow H.P. cylinder.

| | |
|---|---|
| Steam pressure, stop valve | 173·5 lb./in.² abs. |
| ,, temperature, stop valve | 600° F. (230° F. superheat) |
| Steam pressure in receiver | 59·3 lb./in.² abs. |
| ,, temperature in receiver | 383° F. (90° F. superheat) |
| Vacuum | 27 in. |
| Speed | 117·2 r.p.m. |

* See *Engineering*, 11 October, 1929, for drawings and other particulars.

| | | |
|---|---|---|
| I.H.P. | H.P. | 142·5 |
| | L.P. | 223·5 |
| Total I.H.P. | | 366 |
| Total steam | | 3737 lb./hr. |
| Indicated rate of consumption ($w_i$) | | 10·18 lb./I.H.P. hr. |

According to a speed power diagram given, the mechanical efficiency appears to have the value of 0·82 ; and the corresponding brake or shaft rate of heat consumption is $w_B$=15,443 B.Th.U./S.H.P. hr.

For the land type of purely uniflow engine with steam conditions 170 lb./in.$^2$ gauge, 200° Fahr. superheat, 27 in. vacuum, Sulzer Bros. give the following average figures for the indicated rates of steam consumption. Small engines, 70 to 300 I.H.P., $w_i$=10·5–11 ; medium size, 300 to 1000 I.H.P., $w_i$=10–10·5 ; large, 1000 to 3000 I.H.P., $w_i$=9–10. Reducing as before, with an average mechanical efficiency of 0·93, the corresponding brake or net rates of heat consumption are: small, $w_B$=13,864 ; medium, $w_B$=13,200 ; large, $w_B$=11,900 B.Th.U./B.H.P. hr.

A particular feature of the performance of this type of engine, due to the combination high initial superheat and high compression, is the small variation of the consumption rate from half load up to about 25 per cent. overload.

For the paper on uniflow engines (cited on p. 536) the indicated rates shown by tests on an engine of 700 I.H.P. with steam at 180 lb./in.$^2$ gauge and 180° Fahr. superheat were as shown below—

| *Load* | $\frac{1}{2}$ | $\frac{3}{4}$ | *Full* | 25 *per cent. overload* |
|---|---|---|---|---|
| $w_i$ . . . | 9·9 | 9·7 | 10·2 | 10·6 lb./I.H.P. hr. |

The full load brake rate of heat consumption in this case, taking the same conditions as for the previous sets, figures out at $Q_B$=13,358 B.Th.U./B.H.P. hr.

**448.** In the case of the locomotive, which is a movable power plant, the plant performance is usually stated, that is, the value of coal consumption referred to the drawbar horse-power (D.H.P.). The following records of test figures giving steam consumptions of recent types will enable some comparison to be made with the stationary engine figures given above. The reciprocator labours under the disadvantage of high back-pressure and limited expansion range.

In a paper on " Three-Cylinder High-Pressure Locomotives," * H. N. Gresley has given the average figures listed in Table XI from the results of daily operation of the two and three-cylinder " Atlantic " types of simple locomotive. These were run at average speed of about 45–47 mile/hr., and developed from 450 to 480 D.H.P.

The back pressure has been assumed as 4 lb./in.$^2$ above the atmosphere and the average value of 14,000 B.Th.U./lb. has been taken

* *Proc. Inst. Mech. E.*, July 1925, p. 933.

TABLE XI

| Type | Boiler Pressure, lb./in.² g. | Superheat, °F. | Back Pressure, lb./in.² g. | Steam | | Coal | |
|---|---|---|---|---|---|---|---|
| | | | | $w_B$ lb. / D.H.P. hr. | $Q_B$ lb. / D.H.P. hr. | $w_c$ lb. / D.H.P. hr. | $Q_c$ lb. / D.H.P. hr. |
| 2-Cylinder | 170 | 220 | 4 | 45·2 | 50,985 | 6·15 | 91,000 |
| 3-Cylinder | 170 | 220 | 4 | 37·3 | 42,074 | 4·6 | 64,400 |

as the calorific value of the coal, in the calculation of the heat rates.

In a paper on a "Three-Cylinder Compound Locomotive" (built by the Baldwin Locomotive Works and tested on the Pennsylvania testing plant at Altoona), L. H. Fry * gave the indicated rate of steam consumption, at maximum power and speed of 37½ miles/hr., as 15·15 lb./I.H.P. hr. The steam conditions were 322 lb./in.² gauge, 205° Fahr. superheat, and back pressure 4 lb./in.² gauge. From a curve of D.H.P. given it appears that the mechanical efficiency was about 80 per cent., so that the net steam consumption was $w_B$=19 lb./D.H.P. hr., and this corresponds to a net heat consumption $Q_B$=21,600 B.Th.U./D.H.P. hr. In the discussion on this paper Sir H. Fowler gave a set of test results taken on the road from the Royal Scot Class three-cylinder simple locomotive (Art. 309) at speeds from 50 to 55 mile/hr. The normal steam conditions were 250 lb./in.² gauge, 290° Fahr. superheat. The brake rate of steam consumption varied with the running conditions from $w_B$=27 to $w_B$=22·3 lb./D.H.P. hr., and the rate of coal consumption from 3·2 to 2·66 lb./D.H.P. hr. The net heat consumption for the lower steam figure is $Q_B$=26,514 B.Th.U./D.H.P. hr. For the combination of engine and boiler the over-all rate, taking an average calorific value of 14,000 B.Th.U./lb. for the coal, figures out at $Q_0$=37,240 B.Th.U./D.H.P. hr.

In the same discussion R. P. Wagner gave results from the Schmidt high-pressure compound locomotive (Art. 311). Dynamometer car tests were run on the German State Railway at 50 mile/hr. with trains of varying load. The normal steam conditions were 800 lb./in.² gauge and 233° Fahr. superheat for the high-pressure and 200 lb./in.² gauge and 370° Fahr. superheat for the low-pressure element of the compound boiler, the feed temperature to the low-pressure element being 200° Fahr. The brake rate of steam consumption varied from 22·5 to 16·7 lb./D.H.P. hr., and the rate of coal consumption from 2·88 to 2·46 lb./D.H.P. hr. At the lower steam rate the H.P. element of the boiler supplied 51 per cent. of the total steam used, and for the steam conditions stated the heat supply per lb. passing through the cylinders is 1213 B.Th.U., and the net rate of heat consumption is $Q_B$=20,257 B.Th.U./D.H.P. hr. The calorific value

* *Proc. Inst. Mech. E.*, December 1927, p. 923.

of the coal is given as 12,760 B.Th.U./lb., so that the over-all rate of heat consumption is $Q_0$=31,400 B.Th.U./D.H.P. hr.

Tests of the high-pressure uniflow locomotive built by the Swiss Locomotive and Machine Co. (Art. 312) at maximum of 800 D.H.P. at 50 mile/hr., have shown a coal consumption of 2·25 lb./D.H.P. hr. The calorific value of the coal was 13,500 B.Th.U./lb., so that the over-all rate of heat consumption was $Q_0$=30,375 B.Th.U./D.H.P. hr. A better result might be expected from a larger engine.

Records of the Schwartzkoff-Loeffler high-pressure locomotive (Art. 313) are not available, but it is claimed that a thermal efficiency of 18 per cent. can be obtained, which corresponds to an over-all heat consumption rate of $Q_0$=14,000 B.Th.U./D.H.P. hr.

It is instructive to compare the present locomotive position with that of twenty years ago, as shown in a paper on " Compounding and Superheating in Horwich Locomotives," by G. Hughes.* Average values of the indicated rates of steam consumption, given in the tables for simple engines at speed of about 22 mile/hr., ranged from 23·5 to 24·7 lb./I.H.P. hr., and coal consumption from 3·4 to 3·8 lb./I.H.P. hr. For compounds the values were, for steam 18·5 to 17·7 lb./I.H.P. hr. and for coal 3·1 to 2·6 lb./I.H.P. hr. These results were obtained with saturated steam at 180 lb./in.$^2$ gauge. The values of 17·7 for steam and 2·6 for coal compare favourably with those given by Fry for the American higher pressure and superheat compound. It should, however, be noted that the speed of the modern engine is considerably greater than that of the older compound, in which less " throttling " effect might be expected.

It is not possible to make a definite comparison of the net heat consumptions, as the mechanical efficiency cannot be deduced from the data in the paper.

With regard to the turbo-locomotive, recent road tests of the Krupp-Zoelly engine (Art. 316) at 50 mile/hr., with a maximum of 1250 D.H.P., have shown for steam conditions 200 lb./in.$^2$ gauge, 200° Fahr. superheat, vacuum pressure 1·84 lb./in.$^2$ abs., a net steam rate, $w_B$=15·4 lb./D.H.P. hr. This, for the given steam conditions, corresponds to a net heat rate of $Q_B$=18,865 B.Th.U./D.H.P. hr. The coal consumption rate is 1·68 lb./D.H.P. hr., which is about 60 per cent. of the best value given by the previous reciprocators. The calorific value of the coal is 12,600 B.Th.U./lb., so that the over-all rate of heat consumption is $Q_0$=21,170 B.Th.U./D.H.P. hr., as compared with 37,240 of the Royal Scot and 31,400 of the Schmidt high-pressure compound.

Definite results from the Maffei turbo-locomotive (Art. 317) are not available ; but it is credited with a over-all thermal efficiency of 15·6 per cent., which corresponds to the over-all rate of $Q_0$=16,400 B.Th.U./D.H.P. hr.

Similarly the Ljungström locomotive is credited with an efficiency of 14·7 per cent., which corresponds to $Q_0$=17,300 B.Th.U./D.H.P. hr.

* *Proc. Inst. Mech. E.*, March 1910, p. 399.

**449.** In the case of a back pressure engine the consumption varies with the back pressure. Its performance cannot be compared with the ordinary type, for the reasons discussed in Chapter XV. Some idea of the rate of consumption for medium-sized engines can be obtained from the curve AB in Fig. 495, drawn for a Sulzer engine of 1000 B.H.P. working with initial steam conditions, 215 lb./in.$^2$ gauge, 570° Fahr. (180° Fahr. superheat). The curve CD gives the

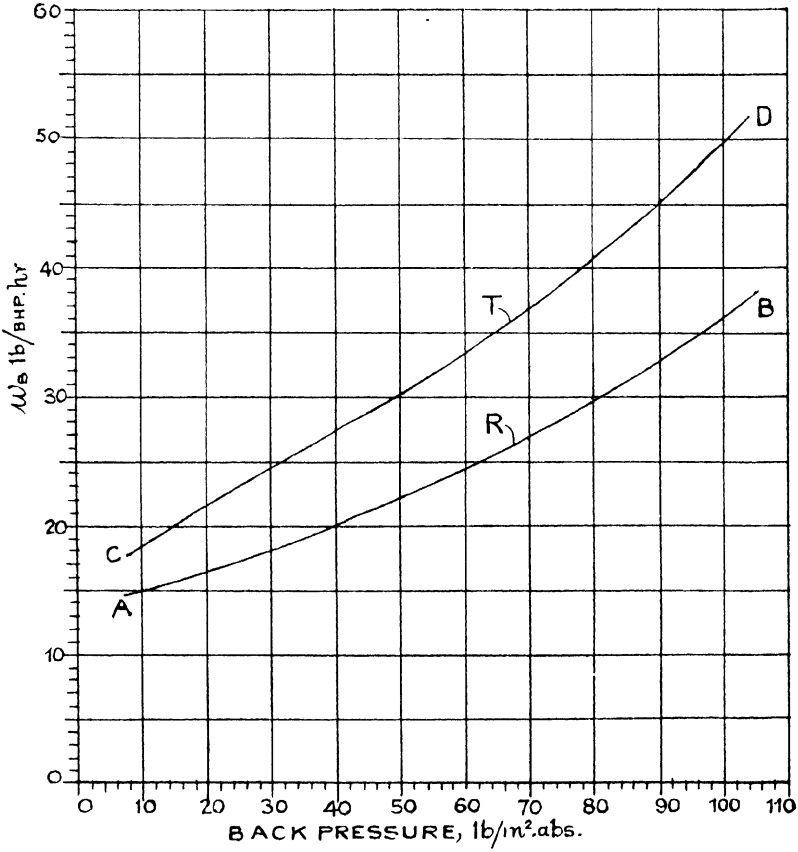



FIG. 495.

corresponding consumption rates for the same size of steam turbine, and, in a general way, indicates the superiority of the reciprocator as a back pressure unit when the back pressure is increased, that is, as the range of expansion is decreased, a point already discussed (Art. 296). The heat consumption rates for any given back pressure can be obtained from $Q_B=(H_{s1}-h_2)w_B$, where $h_2$ is the water heat at the back pressure.

With regard to the use of very high pressure and superheat in the reciprocator, O. H. Hartman, in a paper on "High Pressure Steam; its Present and Future Application on Power and Heating Plants," * gives the results of tests on a high pressure and superheat four-stage Schmidt engine. The steam conditions were 783 lb./in.$^2$, 815° Fahr. (300° Fahr. superheat), and 95 per cent. vacuum. The steam was reheated by live steam between the first I.P. and second I.P. cylinder to 482° Fahr., and between the second I.P. and L.P. cylinders to 428° Fahr.

Including the steam used for reheating, the consumption came out at 5·85 lb./I.H.P. hr. The corresponding rate of heat consumption was $Q_B$=8300 B.Th.U./I.H.P. hr.

**450.** It is difficult to obtain reliable figures of performance for marine reciprocators, but the following may be taken as averages of the consumptions to be expected from the triple reciprocator used in mercantile vessels. The steam pressure is usually 180 lb./in.$^2$ gauge and vacuum 26 in., and both saturated steam and steam superheated about 200° Fahr. is used. Between 500 and 2000 I.H.P. the rates of steam consumptions vary from $w_i$=16·5 to $w_i$=14·5 for saturated, and $w_i$=11·75 to $w_i$=11·25 for superheated steam. The heat consumption rates for the lower values are $Q_i$=16,080 B.Th.U./I.H.P. hr. and $Q_i$=13,354 B.Th.U./I.H.P. hr. To obtain a comparison with the land reciprocators these figures have to be divided by the mechanical efficiency, which varies with the speed of the engine. For the ordinary case a mean value of 0·9 may be used.

Usually the performance is stated in terms of the coal consumption referred to the I.H.P., as this is readily obtained in service. With triples of 2000 I.H.P. and upward it may run from $w_c$=1·6 lb./I.H.P. hr. for saturated and $w_c$=1·4 lb./I.H.P. hr. with superheated steam.

With quadruples, which run at 220 lb./in.$^2$ gauge and higher superheat of 240° Fahr., the values may be taken as $w_c$=1·34 with saturated and $w_c$=1·15 lb./I.H.P. hr. with superheated steam.

The corresponding heat consumption rates (referred to the I.H.P.) are obtained by multiplying these figures by the calorific value of the coal, which in marine service is of too variable a nature to permit of a general comparison of heat consumption rates.

With regard to the modern reciprocator-turbo revival the following results obtained from service records of S.S. *Boniface*,† fitted with the Bauer-Wach system (Art. 305), and SS. *Basil*, with reciprocator only, serve to show the improvement in consumption which may be obtained by the addition of an exhaust steam turbine to the reciprocator. The engines are both triple-expansion and the

* *Trans. First World Power Congress*, 1924, vol. ii, p. 1327.

† For drawings and curves for engines and turbines, see *Engineering*, 29 March, 1929. For particulars of tests in S.S. *Britannia*, see *Engineering*, 5 October, 1928.

boilers are coal fired, the steam conditions being 210 lb./in.$^2$, 190° Fahr. superheat, and the vacuum for reciprocator-turbine $28\frac{1}{2}$ in. For the reciprocator of the *Basil* the vacuum is not given, but it will be considerably less than this figure. The speed of the two vessels, which are identical, was practically the same. The I.H.P. of the combination of the *Boniface* was 2890 and of the *Basil* 2430. The coal consumption rates were 1·04 lb./I.H.P. hr. and 1·4 lb./I.H.P. hr. The calorific values of the coal were 14,125 B.Th.U./lb. and 14,500 B.Th.U./lb., so that the heat consumption rates are $Q_{0i}=1{\cdot}04\times14{,}125=14{,}690$ B.Th.U./I.H.P. hr. for the Bauer-Wach set and $Q_{0i}=1{\cdot}4\times14{,}500=20{,}300$ B.Th.U./I.H.P. hr. for the triple set. The addition of the exhaust turbine thus reduces the indicated rate of heat consumption by 27 per cent. in this case.

It is stated that the thermal efficiency referred to the output at the propeller is 15 per cent., and for this figure the over-all rate of heat consumption for the Bauer-Wach installation is $Q_0=17{,}000$ B.Th.U./S.H.P. hr.

**451. Steam Turbine Performance.**—Like that of the reciprocator, the total steam consumption curve of a turbine, when plotted to a base of B.H.P. or kw., is a straight line. In cases where it is possible to vary the load by cutting out nozzles at the H.P. end, the curve becomes a series of straight line steps, as shown in Fig. 496. This is drawn from the test results of a Brown-Boveri disc and drum turbine, having a velocity compounded stage and groups of nozzles that can be shut off in succession as the load is decreased.

The dotted total steam consumption line AB is the usual Willans line obtained when governing on the throttle, and the corresponding consumption rate curve is *ab*. When the cut-out system is used the curve between 80 per cent. and 100 per cent. full load, or between F and B, coincides with the Willans line AB, as all nozzles are in action. When one group (1) (3 or 4 nozzles per group) is cut out between 60 per cent. and 80 per cent. load the line from E to F falls below the line AB. When two sets (2) are cut out between 40 per cent. and 60 per cent. load the line between D and E falls farther below AB. When three sets (3) are cut out between 25 per cent. and 40 per cent. load the line between C and D falls still farther below AB. At any load below 25 per cent. the turbine is controlled on the throttle. The consumption rate curve, similarly stepped, is shown by *cdefb*. This diagram shows the advantage of the nozzle cut-out over pure throttle governing, between quarter and three-quarter load, for a combination turbine. When this system is applied to a multi-stage turbine having a small pressure drop in the first stage, the reduction of the consumption is much less than in the case of a turbine having a large drop in the first stage, that is, where there is a considerable proportion of the work done on the first stage blading.

The methods of recording the steam engine test data can be equally well applied to the "straight" type of steam turbine test

and it is not necessary to repeat the details, which with slight modification are similar to those given in Art. 445. In the more complicated cases, where reheating, regenerative feed heating, etc., are involved, details of the recording system have to be determined to suit the particular conditions of each case. The various calculations that may be required have been considered and illustrated in Chapter XXII.

**452. Correction of Steam Turbine Consumptions.**—As in the case

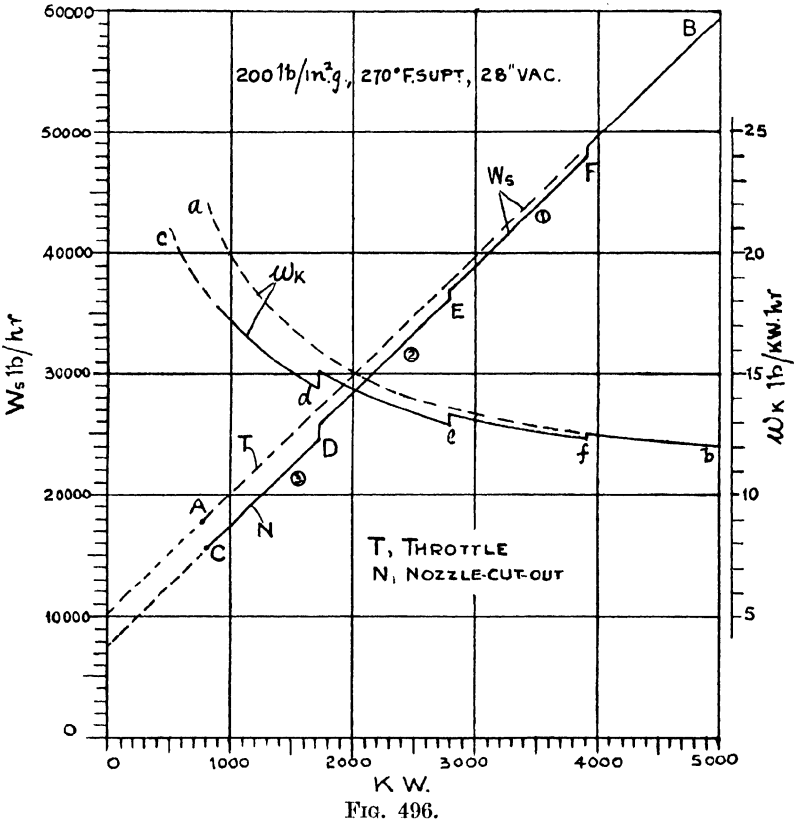



FIG. 496.

of the reciprocator, the steam consumption is affected by variation of initial pressure and superheat, and also, at high vacua, by variation of vacuum.

The general effect of an increase of any of these quantities is a decrease of consumption. It is the practice to ask makers for a guarantee of consumption on official test; and as it is seldom that the test conditions are exactly the same as the guarantee, it is necessary to correct the test figures to guarantee. Correction factors are used for this purpose which have been derived from

experience. The makers and the customer usually agree, before the test, what correction figures are to be applied. Curves of average corrections for pressure, superheat and vacuum are shown in Figs. 497, 498, and 499.

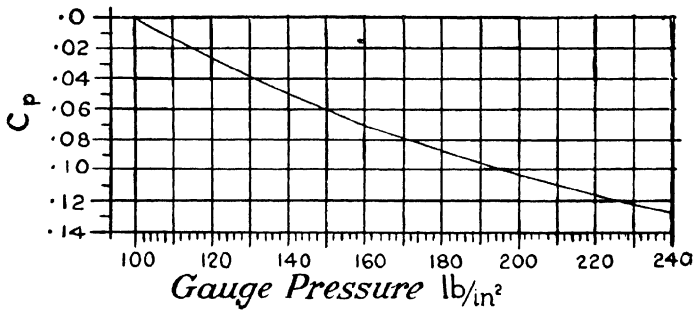



FIG. 497.

In order to obtain the fractional correction for pressure, superheat, or vacuum, read the scale values at the test and guarantee conditions, subtract the guarantee reading from the test reading, and the result

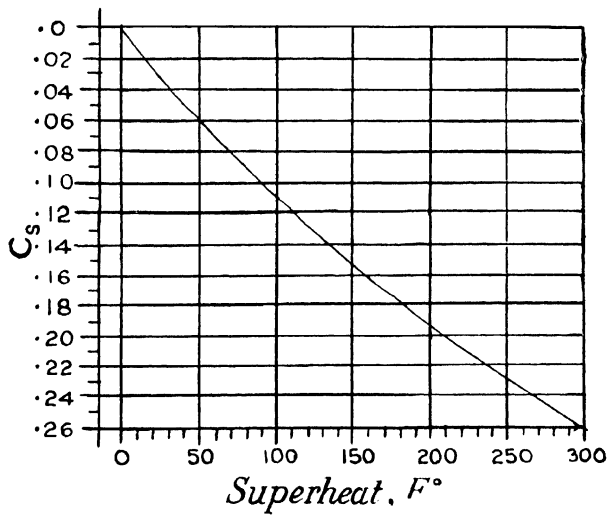



FIG. 498.

is the fractional correction required. These fractional corrections may be either plus or minus.

Let these differences be denoted by $c_p$, $c_s$, and $c_v$, then the total correction is $(1+c_p+c_s+c_v)$, the proper signs being prefixed in the bracket.

If $w_K$=test consumption rate, the corrected "guarantee" rate is—

$$w_{Kg}=w_K(1+c_p+c_s+c_v) \quad . \quad . \quad . \quad . \quad . \quad (5)$$

*Example* 3.—The guarantee steam conditions for a 10,000 kw. combination turbine were 200 lb./in.$^2$ gauge, 650° Fahr. (296° Fahr. superheat), and 28 in. vacuum, and the makers' guarantee consumption was 11·7 lb./kw. hr. The test conditions were 194 lb./in.$^2$ gauge, 634° Fahr. (256° Fahr. superheat), and 28·4 vacuum, and the test consumption was 11·43 lb./kw. hr. Correct to guarantee conditions and find the improvement on the guarantee, on the basis of the corrections given by the curves above.

Scaling, for pressure $c_p$=—0·02, for superheat $c_s$=—0·026, and for vacuum $c_v$=+0·02, $w_K$=11·43.

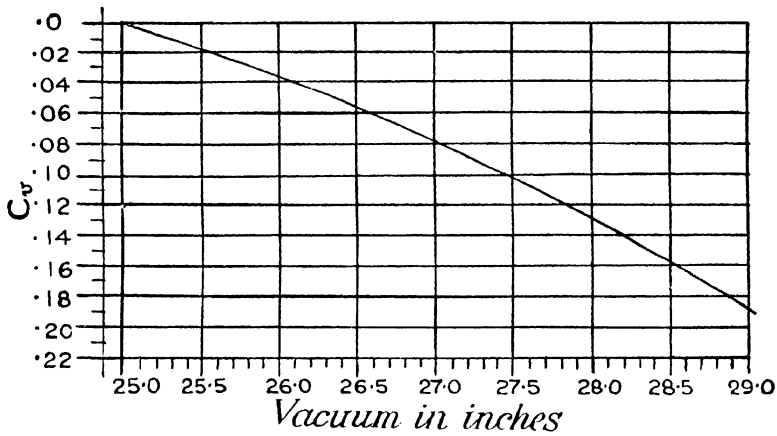



FIG. 499.

$\therefore w_{Kg}=11{\cdot}43(1-0{\cdot}02-0{\cdot}026+0{\cdot}02)=11{\cdot}43\times0{\cdot}974=11{\cdot}13$ lb./kw. hr.

Improvement on guarantee is—

$$\frac{11{\cdot}7-11{\cdot}13}{11{\cdot}7}=0{\cdot}0487 \text{ or } 4{\cdot}87 \text{ per cent.}$$

**453.** The consumptions of the smaller sizes of steam turbine, mostly of the impulse class, run on saturated steam and used for various auxiliary purposes, are much the same as those of the corresponding sizes of reciprocator, and may vary from 40 to 20 lb./B.H.P. hr.

In turbines running on the straight Rankine cycle, that is, without complication such as reheat or feed heat or reheat feed-heat, the steam consumption or the efficiency ratio ($\epsilon$), which is the same as the ratio of ideal consumption to the actual consumption, is a satisfactory figure for comparison of turbine performance provided that it is referred to some standard conditions. Such a standard

is arbitrary. Baumann in 1912 suggested the standard conditions of 180 lb./in.$^2$ gauge (before the governor valve), 150° Fahr. superheat, and 28 in. vacuum (30 in. bar.).

In 1921 * he suggested an advance to 200 lb./in.$^2$ gauge, 200° Fahr. superheat, and 29 in. vacuum, as representing the normal condition of affairs ten years ago. The changes since then in turbine operation and practice, however, have been revolutionary, as has been briefly indicated in Chapter XX, and at present there does not appear to be any clearly defined system of comparison for the new high-output and complex combinations. In the paper to the Inst. Mech. Eng. (already cited, p. 695), Gibb has taken conditions of 250 lb./in.$^2$

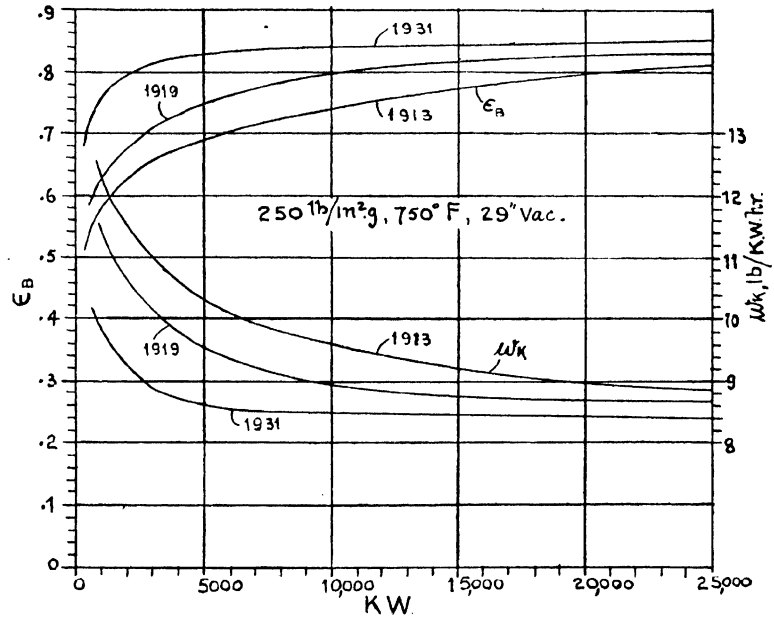



FIG. 500.

gauge, 750° Fahr. (stop valve), and 29 in. vacuum, as a standard of comparison for the efficiency ratios ($\epsilon_B$) of straight turbines.

The three curves from this paper, reproduced in Fig. 500, are instructive in showing how the improvement, principally in the design of nozzles and blading, combined with higher velocity ratios, has raised the figure of merit since 1913. The efficiency ratio of the very small machine remains much the same, and may run between 0·3 and 0·5; but for outputs from 1000 kw. upwards there is a considerable increase up to 5000 kw., as compared with conditions in 1919, and still more so compared with those of 1913, when this

* "Some Recent Developments in Large Steam Turbine Practice," by K. Baumann, *Inst. E.E.*, 15 March, 1921.

output was considered moderate and 25,000 kw. very large. The latter, in fact, was the maximum limit of output then reached, by the Parsons turbine in Fisk Street Station, Chicago.

For the standard conditions of 250 lb./in.$^2$ gauge, 750° Fahr., and 29 in. vacuum the heat input (from the exhaust temperature) is 1352 B.Th.U./lb., the heat drop 477 B.Th.U./lb., and the efficiency of the straight Rankine cycle is 35·27 per cent. With these figures the values of steam consumption $w_K$, efficiency ratio $\epsilon_B$, thermal efficiency $\eta_B$, and heat consumption $Q_K$ have been determined and plotted to a base of kw. in Figs. 500 and 501. These curves show that the efficiency ratio of a 3000 kw. turbine has increased from 65 per cent. in 1913 to 81 per cent. in 1931, that is, an increase of 24·6 per cent. on the 1913 value; from 11 to 8·73 lb./kw. hr. or 19·73 per cent. decrease in steam consumption; and

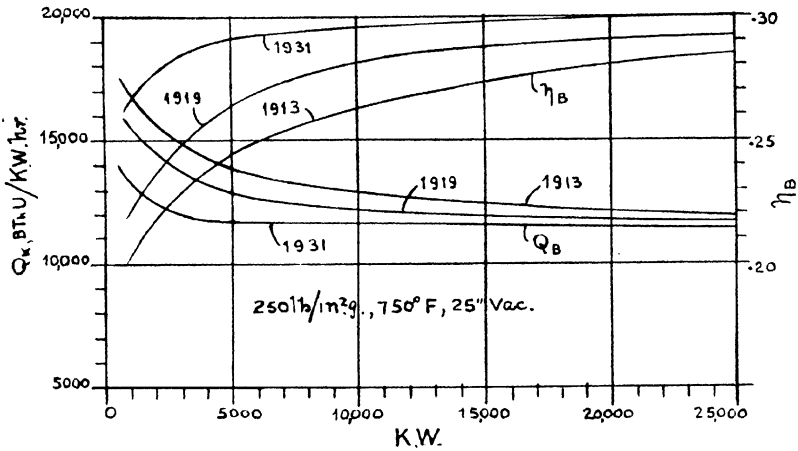



FIG. 501.

from 14,872 to 11,938 B.Th.U./kw. hr., a decrease of 19·73 per cent. in heat consumption. At the upper limit of 25,000 kw. the values are 5 per cent. increase of efficiency ratio and a decrease of 4·8 per cent. in steam and heat consumption. Under the modern conditions the steam or heat consumption of the 3000 kw. machine is now about 5 per cent. less than that of a 25,000 kw. one. The curves show that between 5000 and 25,000 kw. there is little appreciable difference in the heat or steam consumption rate.

**454.** As in the case of the reciprocator, complete test data are difficult to find. A few test results of turbines installed during the past ten years are collected in Table XII. The steam conditions, speed, output, and steam consumption are the actual test values reported; and the writer has added the thermal efficiency and heat consumption values. The efficiency ratios are omitted, as they are misleading. In cases 2 and 5 the heat consumption is based on the

TABLE XII.

| No. | Year | Press., lb./in.$^2$ g. | Temp., °F. | Superheat, °F. | Vac., ins. | Load in kw. | Speed, r.p.m. | Steam Consumption, lb./kw. hr. | Heat Consumption, B.Th.U./kw hr. | Thermal Efficiency | Type | Maker |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1 | 1922 | 205 | 542 | 153 | 27·44 | 59,283 | 1500 | 11·18 | 13,900 | 24·5 | 3-Cylinder Compound | American Westinghouse |
| 2 | 1923 | 250 | 643 | 237 | 29·25 | 18,200 | 1500 | 10·22 | 12,600 | 27·0 | Impulse (one feed heater) | Metropolitan-Vickers |
| 3 | 1923 | 166 | 662 | 290 | 28·95 | 5030 | 3000 | 10·36 | 13,600 | 25·0 | Ljungström Reaction | Brush Electrical Engineering Co. |
| 4 | 1924 | 195 | 620 | 235 | 27·92 | 15,279 | 1500 | 10·95 | 13,900 | 24·5 | Impulse | British Thomson-Houston Co. |
| 5 | 1924 | 194 | 634 | 249 | 28·4 | 8476 | 3000 | 11·43 | 13,600 | 25·0 | Disc and drum (one feed heater) | Richardson, Westgarth & Co. |
| 6 | 1925 | 211 | 548 | 156 | 27·8 | 2470 | 3600 | 12·48 | 15,270 | 22·4 | Impulse | Jas. Howden & Co. |
| 7 | 1926 | 264 | 627 | 217 | 29·33 | 5028 | 3000 | 9·72 | 12,600 | 27·0 | Ljungström reaction | S.T.A.L. Co. |
| 8 | 1927 | 197 | 636 | 250 | 28·77 | 10.975 | 3041 | 10·32 | 13,310 | 25·7 | Impulse | Escher Wyss & Co. |
| 9 | 1928 | 200 | 560 | 173 | 28·6 | 8690 | 1500 | 11·149 | 13,920 | 24·4 | Impulse | Metropolitan-Vickers |
| 10 | 1929 | 217 | 676 | 283 | 28·53 | 11,931 | 3000 | 9·45 | 12,100 | 28·2 | Ljunström Reaction | S.T.A.L. Co. |

given temperature of the feed; in the other cases the exhaust temperature has been taken as the lower limit. It should be borne in mind that although such a table gives a general idea of the consumption rates, the figures are not strictly comparable. All the "straight" cases have to be corrected to the same standard conditions before any valid comparison can be made. This work is left as an exercise to the student. Regenerative, reheat, and reheat-regenerative cases are much more difficult to handle, owing to the uncertainty in fixing the correction factors, caused by the number of variables involved.

Some guidance may be obtained from the data given in a recent paper to the Inst. Mech. Eng.* In this paper Guy has given a table for corrections of the thermal efficiency for increase of pressure between 200 lb./in.$^2$ gauge and 1400 lb./in.$^2$ gauge for a

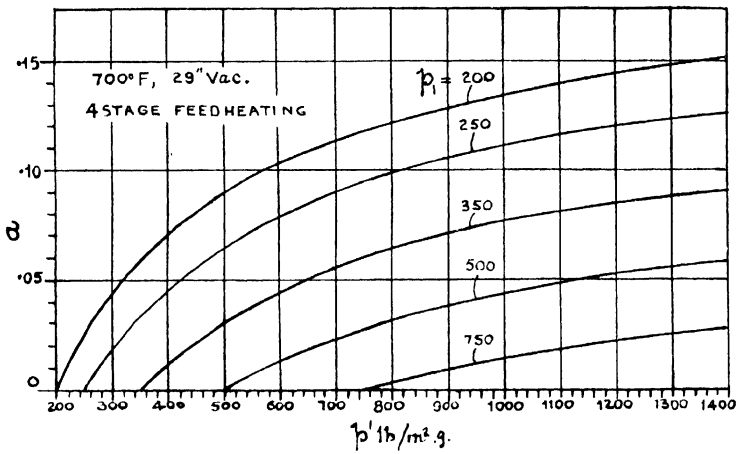



FIG. 502.

standard feed heating turbine working with steam at 700° Fahr. and 29 in. vacuum, and four regenerative heaters.

The figures are stated to be applicable equally to the turbine, with or without boiler reheat to the initial temperature. The writer has deduced from this data the set of curves given in Fig. 502. The percentage increase of thermal efficiency, due to increase of pressure, from the original value $p_1$ to some higher pressure $p'$, the temperature and vacuum remaining as above, is given by the factor $a$, which is defined by the intersection of the vertical through the $p'$ value and the $p_1$ curve.

The correction figures given cover the turbine and condenser and the boiler auxiliaries, and are assumed to apply to outfits of capacity

* "Tendencies in Steam Turbine Development," by H. L. Guy, *Proc. Inst. Mech. E.*, 31 January, 1929.

not less than 40,000 kw. M.C.R. at 1500 r.p.m.; 20,000 kw. M.C.R. at 3000 r.p.m.; 10,000 kw. M.C.R. at 6000 r.p.m.

In general the increased efficiency is given by—

$$\eta_B' = (1+a)\eta_B \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

Gains with pressure for other temperatures between 700° Fahr. and 900° Fahr., and vacua between 29 in. and 28 in. can be obtained by adding corrections for temperature and vacuum.

For increased temperature, the correction may be taken as—

$$b = \frac{0{\cdot}00166}{100}(p' - 750) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

when $p' > 750$.

And for vacuum, from—

$$c = \frac{0{\cdot}003}{100}(750 - p_1) + \frac{0{\cdot}002}{100}(p' - 750) \quad . \quad . \quad . \quad (8)$$

Then $$\eta_B' = (1+a+b+c)\eta_B \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

*Example* 4.—Find the probable increase in thermal efficiency of a turbine with four regenerative heaters running under conditions 350 lb./in.$^2$ gauge, 700° Fahr., 29 in. vacuum, if the pressure is increased to 1200 lb./in.$^2$, temperature to 800° Fahr., and vacuum reduced to 28 in.

$p_1 = 350$, $p' = 1200$. From curves, $a = 0{\cdot}085$.

$$b = \frac{0{\cdot}00166}{100}(p' - 750) = \frac{0{\cdot}00166 \times 450}{100} = 0{\cdot}00747$$

$$c = \frac{0{\cdot}003}{100}(750 - 350) + \frac{0{\cdot}002}{100}(1200 - 750)$$

$$= 0{\cdot}012 + 0{\cdot}009 = 0{\cdot}021$$

$$(a+b+c) = (0{\cdot}085 + 0{\cdot}00747 + 0{\cdot}021) = 0{\cdot}1135$$

$\therefore \eta_B' = 1{\cdot}1135\eta_B$, or the efficiency is increased by 11·35 per cent.

**455. Marine Steam Turbine Performance.**—As in the case of the marine reciprocator, published details of marine turbine performance are very scanty, and consequently this subject can only be dealt with in a general way. In any case, steam consumption has practically ceased to be a suitable figure of comparison under the diverse conditions now obtaining in turbine-engined vessels. Exhaust feed heating has been a long established practice in marine work, and now to this the regenerative feed heating system is being added. There is a series of auxiliary machines, which may be driven either wholly by steam, or by motors from turbo-electric sets, or motors from Diesel engine sets, or directly by mechanical means from the main turbines. The conditions of comparison are further complicated by the growing use of oil instead of coal as fuel. It is now becoming the practice to use either the thermal efficiency or the corresponding rate of heat consumption as the figure of comparison for the " propulsion " part of the whole installation, which includes

boilers, turbines, and all auxiliaries associated with them, and also the steering gear. The rate of fuel consumption per S.H.P./hr. is usually given, and this, in the case of oil fuel, which has a fairly constant calorific value, is satisfactory enough ; but it is not in the case of coal, where the value may run anywhere between 11,000 and 14,000 B.Th.U./lb.

In large passenger liners a very considerable amount of the extracted steam is now used for general ship's " galley " purposes, and the performance estimation is further complicated by the fuel allowance to be made on this score.

The first turbine mercantile vessel, the *King Edward*, was directly driven in 1901 by turbines of 3500 S.H.P. with saturated steam of 150 lb./in.$^2$ gauge, and the steam consumption recorded, for all purposes, was 18 lb./S.H.P. hr. The turbines of the *Lusitania*, which followed in 1906, developed a total of 68,000 S.H.P., with steam of 195 lb./in.$^2$ gauge and a consumption of 13 lb./S.H.P. hr. Then came the *Aquitania*, in 1914, with 60,000 S.H.P. and the same steam conditions, and a consumption of 12 lb./S.H.P. hr. Previous to this, in 1908, Parsons introduced the geared turbine ; and the single-geared outfit of the *Vespasian*, developing the small output of 1050 S.H.P., with steam of 150 lb./in.$^2$, showed a consumption of 13 lb./S.H.P. hr., equal to that of the large directly driven liner outfits. Then superheated steam was applied, and the combination of high-speed turbines (with more efficient velocity ratio) and better thermal conditions further reduced the consumption, and in 1919 for an output of 3200 S.H.P., with steam of 190 lb./in.$^2$ gauge, 200° Fahr. superheat, the double reduction turbines of the S.S. *Cairnross* showed a consumption of 9·0 lb./S.H.P. hr. (for turbines only). In 1924 the single reduction turbines of the S.S. *Orama* developed 19,000 S.H.P. with steam of 215 lb./in.$^2$ gauge, 150° Fahr. superheat, with a consumption of 9·25 lb./S.H.P. hr. The figures for superheated are not directly comparable with those for saturated steam, but the list shows that the marine turbine consumption has practically been halved in the space of twenty years.

Referred to the fuel rate of consumption the figure for this recent liner the *Orama*, run with oil fuel, was 0·785 lb. oil/S.H.P. hr., for all purposes. Taking the calorific value as 19,000 B.Th.U/lb., this corresponds to an over-all rate of heat consumption $Q_0$=14,915 B.Th.U./S.H.P. hr.

As already stated in Art. 349, Sir Charles Parsons again acted as a pioneer in the advancement of marine propulsion, by the adoption in 1926 of the higher boiler pressure of 550 lb./in.$^2$ gauge, 700° Fahr., and 29 in. vacuum, in the *King George*. The results of the official trials with steam of 436 lb./in.$^2$ gauge, 795° Fahr., 29·04 in. vacuum, at the turbines, showed the steam consumption—

| | |
|---|---|
| Turbines . . . . . | 8·01 lb./S.H.P. hr. |
| Auxiliaries . . . . . | 1·66 „ |
| All purposes . . . . . | 9·67 „ |

The auxiliaries thus used 17·3 per cent. of the total steam supplied. The coal consumption was 1·085 lb./S.H.P. hr., and as the calorific value was 13,880 B.Th.U./lb. the rate of heat consumption was $Q_0$=15,059 B.Th.U./S.H.P. hr., and the thermal efficiency 17 per cent. A much better result may be expected from a larger installation.

In the cases of the C.P.S. liners *Duchess of Bedford* and *Duchess of Atholl* already mentioned in Art. 350, the total output of the single-geared turbines is 18,000 S.H.P. and steam conditions 345 lb./in.$^2$, 680° Fahr., 29 in. vacuum.

The full rate on service for "propulsion," which includes all auxiliary machinery associated with the main turbines and steering gear, is 0·57 lb. oil/S.H.P. hr. With oil of 19,000 B.Th.U./lb., the heat consumption rate for propulsion is thus 10,820 B.Th.U./S.H.P. hr., and the thermal efficiency has the higher value of 23·4 per cent. In these installations the main auxiliaries are motor driven, the current being supplied from Diesel generating sets. Also the feed is heated by the main feed pump exhaust and by regenerative feed heating to 300° Fahr. It is of interest, as a comparison, to give the consumption of a sister ship, S.S. *Montclare*, developing 12,500 S.H.P. with steam at 215 lb./in.$^2$ gauge, 550° Fahr. The figure is 0·87 lb./S.H.P. hr., corresponding to a heat consumption rate of 16,500 B.Th.U./S.H.P. hr. and a propulsion thermal efficiency of 15·4 per cent. The adoption of the higher steam conditions and regenerative heating with the efficient Diesel auxiliary drive has thus increased the thermal efficiency about 50 per cent.

In the case of the C.P.S. liner *Empress of Japan* with turbines developing 30,000 S.H.P., working at 375 lb./in.$^2$, 700° Fahr., the still higher efficiency value of 24·3 per cent. has been obtained. This gives an overall heat consumption rate of $Q_0$=10,473 B.Th.U./S.H.P. hr. The turbines of the Holland-America liner *Statendam* are run with steam of 425 lb./in.$^2$ gauge, 650° Fahr., and 37 per cent. of the steam is extracted to heat the feed to 300° Fahr. and to supply the ship's heating and galley system. The service oil consumption is 0·618 lb./S.H.P. hr. for all purposes, giving a heat consumption rate of $Q_0$=11,742 B.Th.U./S.H.P. hr. The actual propulsive rate is appreciably less, owing to the diverted steam, and the apparent efficiency figure of 21·7 per cent. is lower than the real value, which may run from 23 per cent. to 24 per cent.

As regards the smaller types of cargo vessels running under what until the present have been normal conditions of 250 lb./in.$^2$, 650° Fahr., several C.P.S. steamers with 7000 S.H.P. at 15 knots show, with stoker-fired water-tube boilers, a coal consumption rate of 1·1 lb./S.H.P. hr.; and several with hand-fired boilers a rate of 1·15 lb./S.H.P. hr. These are equivalent to $Q_0$=14,850 and $Q_0$=15,525 B.Th.U./S.H.P. hr.

The figures given above may be taken as representative of the latest developments in marine steam turbine practice, and better

performances may be expected when much higher pressures are adopted.

An interesting experiment is being tried in the Hamburg-America liner *Uckermark*. One of the cylindrical boilers has been replaced by a Benson high-pressure water-tube boiler, fired with oil and stated to be capable of generating 20 ton/hr. of steam at the critical pressure of 3120 lb./in.$^2$ An automatically controlled reducing valve lowers the pressure to 1200 lb./in.$^2$ The steam at this pressure is then passed through a superheater, reduced further to 850 lb./in.$^2$, and supplied at 840° Fahr. to an auxiliary two-stage high-pressure turbine running at 6000 r.p.m. The speed is reduced by double reduction gearing to 100 r.p.m. at the

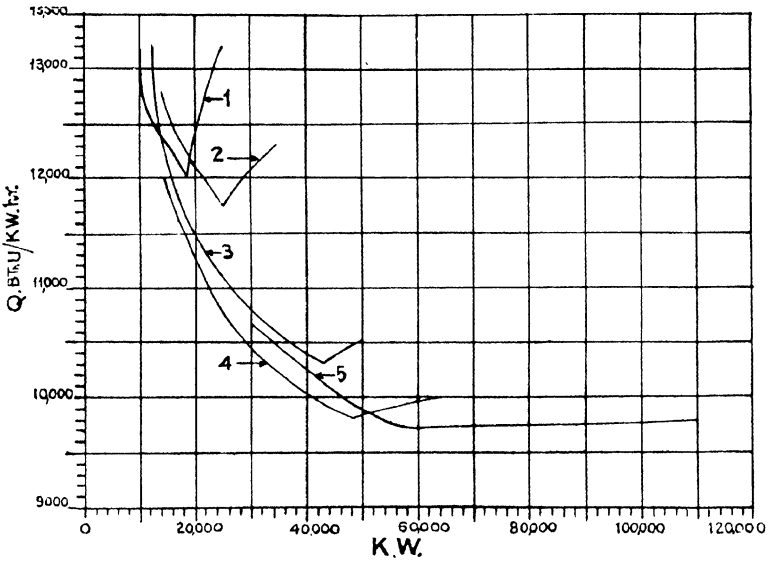



FIG. 503.

propeller shaft. The steam is exhausted from the turbine to the original H.P. turbine of the ship at 120 lb./in.$^2$ It is stated that with this high pressure addition a steam consumption of 6·8 lb./S.H.P. hr. has been obtained.

**456.** With regard to the recent and large turbine outfits in power stations installed during the past five years,* the 50,000 kw. Parsons turbine in Crawford Avenue Station, Chicago, supplied in 1925 (see Art. 343), running with steam conditions 550 lb./in.$^2$ gauge, 750° Fahr., 29·25 in. vacuum, reheat to 700° Fahr., three feed heaters taking 22 per cent. of initial steam supply, has shown a thermal efficiency of 34 per cent. for the turbines alone, or a heat rate

* A list of sixty-three station outfits with leading particulars is given in Gibb's paper (see p. 695). It is too extended for reproduction here.

$Q_B$=10,038 B.Th.U./kw. hr. steam to electricity. The world's largest turbine unit, for the great output of 208,000 kw., installed in the State Line Station, Hammond, Indiana (see Art. 339), with steam conditions, 600 lb./in.$^2$ gauge, 700° Fahr., 29 in. vacuum, reheat to 500° Fahr., five feed heaters, is estimated to give a heat consumption $Q_B$=9850 B.Th.U./kw. hr. for the turbine alone, and $Q_{0B}$=10,500 B.Th.U./kw. hr. for turbine and auxiliaries. The over-all rate for the station, which includes the boiler conditions, is estimated at $Q_0$=13,420 B.Th.U./hr., and corresponds to an over-all thermal efficiency of 25·5 per cent.

The 41,000 kw. Metropolitan-Vickers turbine installed in Barton Power Station, Manchester, in 1928, with steam conditions 350 lb./in.$^2$ gauge, 700° Fahr., 29 in. vacuum, four regenerative heaters, has shown a thermal efficiency for turbine and generator of 31·27 per cent., or a heat consumption $Q_B$=10911 B.Th.U./kw. hr. steam to electricity (see Arts. 338 and 436).

Representative heat consumption curves for five large American turbines have been given in Prof. A. G. Christie's paper,* and are reproduced in Fig. 503. The operative conditions are given in Table XIII. These turbines were installed by the Allis-Chalmers Manufacturing Co., Milwaukee, Wis.

TABLE XIII

| No. | Year | Pressure, lb./in.$^2$ g. | Temperature, °F. | Vac., ins. | Load, kw. | Number of Heaters | Feed Temperature, °F. | Reheat Temperature, °F. | Load factor, per cent. |
|---|---|---|---|---|---|---|---|---|---|
| 1 | 1923 | 350 | 675 | 29 | 25,000 | 2 | 230 | — | 93 |
| 2 | 1925 | 350 | 657 | 29 | 35,000 | 2 | 240 | — | 93 |
| 3 | 1927 | 600 | 725 | 29 | 50,000 | 4 | 320 | — | 85 |
| 4 | 1930 | 600 | 725 | 29 | 65,000 | 4 | 385 | 725 | 85 |
| 5 | 1931 | 625 | 750 | 29 | 115,000 | 4 | 380 | 750 | 95 |

Records of performance for the various very high pressure and high temperature station installations mentioned in Chapter XXII (Art. 338 and footnotes) are not available so far. A good idea, however, of the probable thermal efficiencies to be expected, both for turbines and plant, may be obtained from an inspection of the estimate submitted by Sir Charles Parsons in his paper on "Steam Turbines" to the First World Power Conference in 1924.† This estimate is reproduced in Table XIV.

"The feed water is heated to within about 50° Fahr. of boiler temperature by means of steam drawn from turbine blading at

* "Trends in Steam Turbine Development," *Trans. American Society of Mechanical Engineers*, 11 February, 1931, or *Mechanical Engineering*, April 1931.

† *Trans. First World Power Conference*, vol. ii p. 1495.

TABLE XIV

| No. | Stop Valve Pressure, lb./in.² g. | Stop Valve Temperature, °F. | Superheat, °F. | Assumed Reheat Pressure, lb./in.² abs. | Reheat Temperature, °F. | Restored Superheat, °F. | Exhaust Vac., ins., Bar. 30 ins. | Stage Feed Heating, 65° F. to (°F.) | Thermal Efficiency per cent. (Steam to Electricity) | Assumed Boiler Plant Efficiency (including all Aux.) | Over-all Thermal Efficiency per cent. (Fuel to Electricity) | Reduction in Fuel Consumption, per cent. | Equivalent Oil Fuel Consumption, lb./B.H.P. hr. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| I | 250 | 750 | 344 | 65 | 700 | 402 | 29·25 | 360 | 31·6 | 84·0 | 26·52 | — | 0·501 |
| II | 500 | 750 | 281 | 100 | 700 | 372 | 29·25 | 420 | 33·5 | 83·5 | 27·97 | 5·45 | 0·475 |
| III | 1000 | 750 | 204 | 150 | 700 | 343 | 29·25 | 510 | 35·2 | 83·0 | 29·20 | 10·08 | 0·455 |
| IV | 1500 | 750 | 153 | 250 | 700 | 299 | 29·25 | 550 | 36·6 | 82·5 | 30·20 | 13·82 | 0·440 |
| V | 2000 | 750 | 114 | 400 | 700 | 254·5 | 29·25 | 600 | 37·5 | 82·0 | 30·75 | 15·90 | 0·432 |

several points (regenerative heating). Boilers equipped with economisers or air preheaters or both, so as to recover heat from the flue gases and maintain high thermal efficiency. Dynamo efficiency taken as 96·5 per cent. at full rated output. Gross calorific value of fuel oil taken as 18,500 B.Th.U./lb."

**457.** So far the record of economical working appears to be held by the mercury-steam turbine installation in the South Meadow Station at Hartford, Con. (described in Art. 351). The conditions at the mercury turbine are 70 lb./in.² gauge, 880° Fahr., 28 in. vacuum, and at the steam turbine 280 lb./in.², 735° Fahr., 29 in. vacuum.

A four months' performance of 2023 hours' service, in 1930, showed a net fuel rate, with pulverised fuel, of 0·7146 lb./kw. hr., or a heat consumption rate $Q_0$=9800 B.Th.U./kw. hr., corresponding to an over-all thermal efficiency of 34·5 per cent. This plant, with a comparatively small output, is well in advance of any of the large power station outfits with outputs running into hundreds of thousands of kilowatts.

Judging from the present trend towards the adoption of higher pressures and temperatures, it is probable that in the near future the thermal efficiency of the large power station may reach the value of 30 per cent., and with binary fluid turbines the higher figure from 35 to 40 per cent.

Even under these improved conditions, however, at least 50 to 60 per cent. of the heat of combustion of each pound of fuel will still be thrown away in the exhaust steam, through the medium of the circulating water, to heat uselessly the water of a canal, a river, or a lake, and 5 to 10 per cent. to heat the atmosphere directly through the flue gases of the boiler. The enormous wastage of fuel in a large power station, entailed by the present operative conditions of the steam cycle, is now being more fully appreciated than it was in the past by power station engineers; and various schemes of exhaust heating, industrial and domestic, by which the low temperature heat of the prime mover exhaust may be usefully employed are now being

proposed. The advantage of the use of power-process installations in industrial establishments has been discussed in Chapter XV. In many modern hotels, residential flats, asylums, public institutions, etc., this economical system is being adopted, with good results.

The big waste-heat problem created by the super-power station still awaits a satisfactory solution. A number of papers on heat recovery, combined heat and power supply, district heating, etc., will be found in Vol. IV of the Transactions of the Second World Power Conference, Berlin, 1930.

## Properties of Saturated Steam

(Based on Callendar's (1929) Characteristic Equations)

| Vac. in ins. $x$ | Press. lb./in.$^2$ abs. $p$ | Temp. °F. $t$ | Specific Volume Ft.$^3$/lb. | | Total Heat B.Th.U./lb. | | Latent Heat B.Th.U./lb. | Entropy B.Th.U./lb. °F. | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | | | Water $u$ | Steam $v$ | Water $h$ | Steam H | L | Water $\phi_w$ | Evaporation $\phi_e$ | Total $\Phi$ |
| 29·59 | 0·2 | 53·1 | 0·01603 | 1526 | 21·1 | 1080·7 | 1059·6 | 0·0418 | 2·0686 | 2·1104 |
| 29·38 | 0·3 | 64·2 | 0·01605 | 1040 | 32·2 | 1085·9 | 1053·7 | 0·0636 | 2·0122 | 2·0758 |
| 29·18 | 0·4 | 72·7 | 0·01606 | 792 | 40·7 | 1089·9 | 1049·2 | 0·0793 | 1·9721 | 2·0514 |
| 28·98 | 0·5 | 79·5 | 0·01607 | 640·5 | 47·4 | 1092·2 | 1044·7 | 0·0920 | 1·9379 | 2·0299 |
| 28·88 | 0·55 | 82·4 | 0·01608 | 585 | 50·3 | 1093·5 | 1043·2 | 0·0974 | 1·9247 | 2·0221 |
| 28·78 | 0·60 | 85·2 | 0·01609 | 539·1 | 53·0 | 1094·7 | 1041·7 | 0·1024 | 1·9124 | 2·0148 |
| 28·67 | 0·65 | 87·7 | 0·01609 | 499·8 | 55·5 | 1095·9 | 1040·4 | 0·1069 | 1·9009 | 2·0078 |
| 28·57 | 0·70 | 90·0 | 0·01610 | 466·2 | 57·8 | 1097·0 | 1039·2 | 0·1112 | 1·8906 | 2·0018 |
| 28·47 | 0·75 | 92·2 | 0·01611 | 436·8 | 60·0 | 1098·0 | 1038·0 | 0·1152 | 1·8810 | 1·9962 |
| 28·37 | 0·80 | 94·3 | 0·01612 | 411·1 | 62·2 | 1099·0 | 1036·8 | 0·1192 | 1·8714 | 1·9906 |
| 28·26 | 0·85 | 96·3 | 0·01612 | 388·2 | 64·1 | 1099·9 | 1035·8 | 0·1226 | 1·8632 | 1·9858 |
| 28·16 | 0·90 | 98·2 | 0·01612 | 367·9 | 66·0 | 1100·8 | 1034·8 | 0·1260 | 1·8550 | 1·9810 |
| 28·06 | 0·95 | 100·0 | 0·01613 | 349·6 | 67·8 | 1101·6 | 1033·8 | 0·1292 | 1·8475 | 1·9767 |
| 27·96 | 1·00 | 101·7 | 0·01613 | 333·1 | 69·5 | 1102·4 | 1032·9 | 0·1322 | 1·8402 | 1·9724 |
| 27·85 | 1·05 | 103·3 | 0·01613 | 318·1 | 71·1 | 1103·2 | 1032·1 | 0·1350 | 1·8335 | 1·9685 |
| 27·75 | 1·10 | 104·9 | 0·01614 | 304·5 | 72·7 | 1103·9 | 1031·2 | 0·1378 | 1·8268 | 1·9646 |
| 27·65 | 1·15 | 106·4 | 0·01616 | 292·0 | 74·2 | 1104·6 | 1030·4 | 0·1404 | 1·8196 | 1·9610 |
| 27·55 | 1·20 | 107·9 | 0·01616 | 280·6 | 75·7 | 1105·3 | 1029·6 | 0·1431 | 1·8144 | 1·9575 |
| 27·45 | 1·25 | 109·4 | 0·01616 | 270·0 | 77·2 | 1105·9 | 1028·7 | 0·1458 | 1·8084 | 1·9542 |
| 27·34 | 1·30 | 110·7 | 0·01617 | 260·2 | 78·5 | 1106·5 | 1028·0 | 0·1482 | 1·8027 | 1·9509 |
| 27·24 | 1·35 | 112·0 | 0·01618 | 251·1 | 79·8 | 1107·1 | 1027·3 | 0·1505 | 1·7974 | 1·9479 |
| 27·14 | 1·40 | 113·2 | 0·01618 | 242·7 | 81·8 | 1107·7 | 1026·6 | 0·1526 | 1·7923 | 1·9449 |
| 27·04 | 1·45 | 114·5 | 0·01618 | 234·8 | 82·3 | 1108·3 | 1026·0 | 0·1547 | 1·7873 | 1·9420 |
| 26·94 | 1·50 | 115·7 | 0·01619 | 227·4 | 83·5 | 1108·8 | 1025·3 | 0·1567 | 1·7825 | 1·9392 |
| 26·73 | 1·60 | 118·0 | 0·01620 | 214·0 | 85·8 | 1109·9 | 1024·1 | 0·1608 | 1·7731 | 1·9339 |
| 26·53 | 1·70 | 120·2 | 0·01621 | 202·2 | 88·0 | 1110·9 | 1022·9 | 0·1645 | 1·7645 | 1·9290 |
| 26·32 | 1·80 | 122·2 | 0·01622 | 191·6 | 90·0 | 1111·8 | 1021·8 | 0·1680 | 1·7564 | 1·9244 |
| 26·12 | 1·90 | 124·2 | 0·01623 | 182·1 | 92·1 | 1112·7 | 1020·6 | 0·1716 | 1·7484 | 1·9200 |
| 25·91 | 2·0 | 126·1 | 0·01624 | 173·5 | 93·9 | 1113·6 | 1019·7 | 0·1750 | 1·7409 | 1·9159 |
| 25·50 | 2·2 | 129·6 | 0·01625 | 158·7 | 97·4 | 1115·2 | 1017·8 | 0·1810 | 1·7271 | 1·9081 |
| 25·1 | 2·4 | 132·9 | 0·01627 | 146·4 | 100·7 | 1116·6 | 1015·9 | 0·1864 | 1·7146 | 1·9010 |
| 24·69 | 2·6 | 135·9 | 0·01629 | 135·6 | 103·7 | 1118·0 | 1014·3 | 0·1914 | 1·7033 | 1·8947 |
| 24·28 | 2·8 | 138·8 | 0·01630 | 126·5 | 106·6 | 1119·4 | 1012·8 | 0·1961 | 1·6927 | 1·8888 |
| 23·87 | 3·0 | 141·5 | 0·01631 | 118·6 | 109·3 | 1120·6 | 1011·3 | 0·2006 | 1·6827 | 1·8833 |
| 23·46 | 3·2 | 144·0 | 0·01632 | 111·6 | 111·8 | 1121·7 | 1009·9 | 0·2048 | 1·6732 | 1·8780 |
| 23·06 | 3·4 | 146·4 | 0·01634 | 105·4 | 114·2 | 1122·8 | 1008·6 | 0·2089 | 1·6642 | 1·8731 |
| 22·65 | 3·6 | 148·7 | 0·01635 | 99·93 | 116·5 | 1123·8 | 1007·3 | 0·2126 | 1·6559 | 1·8685 |
| 22·24 | 3·8 | 150·9 | 0·01636 | 95·0 | 118·7 | 1124·8 | 1006·1 | 0·2162 | 1·6479 | 1·8641 |
| 21·83 | 4·0 | 153·0 | 0·01637 | 90·54 | 120·8 | 1125·7 | 1004·9 | 0·2197 | 1·6403 | 1·8600 |
| 21·42 | 4·2 | 155·0 | 0·01638 | 86·5 | 122·8 | 1126·6 | 1003·8 | 0·2229 | 1·6332 | 1·8561 |
| 21·01 | 4·4 | 156·9 | 0·01639 | 82·8 | 124·7 | 1127·4 | 1002·7 | 0·2260 | 1·6264 | 1·8524 |

| Vac. in ins. $x$ | Press lb./in.$^2$ abs. $p$ | Temp. °F. $t$ | Specific Volume Ft.$^3$ | | Total Heat B.Th.U./lb. | | Latent Heat B.Th.U./lb. | Entropy B.Th.U./lb.°F. | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | | | Water $u$ | Steam $v$ | Water $h$ | Steam H | L | Water $\phi_w$ | Evaporation $\phi_e$ | Total $\Phi$ |
| 20·60 | 4·6 | 158·8 | 0·01640 | 79·42 | 126·6 | 1128·2 | 1001·6 | 0·2290 | 1·6199 | 1·8489 |
| 20·20 | 4·8 | 160·6 | 0·01640 | 76·31 | 128·3 | 1129·0 | 1000·7 | 0·2319 | 1·6136 | 1·8455 |
| 19·79 | 5·0 | 162·3 | 0·01641 | 73·44 | 130·0 | 1129·8 | 999·8 | 0·2346 | 1·6076 | 1·8422 |
| 19·38 | 5·2 | 163·9 | 0·01642 | 70·80 | 131·7 | 1130·5 | 998·8 | 0·2372 | 1·6019 | 1·8391 |
| 18·97 | 5·4 | 165·5 | 0·01643 | 68·34 | 133·4 | 1131·2 | 997·8 | 0·2399 | 1·5962 | 1·8361 |
| 18·56 | 5·6 | 167·1 | 0·01643 | 66·05 | 134·9 | 1131·9 | 997·0 | 0·2424 | 1·5907 | 1·8331 |
| 18·16 | 5·8 | 168·6 | 0·01644 | 63·91 | 136·4 | 1132·5 | 996·1 | 0·2448 | 1·5855 | 1·8303 |
| 17·75 | 6·0 | 170·1 | 0·01645 | 61·91 | 137·9 | 1133·2 | 995·3 | 0·2473 | 1·5804 | 1·8277 |
| 16·73 | 6·5 | 173·6 | 0·01646 | 57·44 | 141·4 | 1134·7 | 993·3 | 0·2527 | 1·5687 | 1·8214 |
| 15·70 | 7·0 | 176·9 | 0·01648 | 53·59 | 144·7 | 1136·1 | 991·4 | 0·2580 | 1·5576 | 1·8156 |
| 14·68 | 7·5 | 180·0 | 0·01651 | 50·24 | 147·8 | 1137·4 | 989·6 | 0·2629 | 1·5472 | 1·8101 |
| 13·66 | 8·0 | 182·9 | 0·01652 | 47·30 | 150·8 | 1138·6 | 987·8 | 0·2674 | 1·5375 | 1·8049 |
| 12·64 | 8·5 | 185·7 | 0·01654 | 44·69 | 153·6 | 1139·8 | 986·2 | 0·2717 | 1·5284 | 1·8001 |
| 11·62 | 9·0 | 188·3 | 0·01656 | 42·36 | 156·3 | 1140·9 | 984·6 | 0·2760 | 1·5196 | 1·7956 |
| 10·6 | 9·5 | 190·8 | 0·01657 | 40·27 | 158·7 | 1142·0 | 983·3 | 0·2799 | 1·5115 | 1·7914 |
| 9·58 | 10·0 | 193·2 | 0·01658 | 38·39 | 161·1 | 1143·0 | 981·9 | 0·2834 | 1·5040 | 1·7874 |
| 8·56 | 10·5 | 195·6 | 0·01660 | 36·68 | 163·4 | 1144·0 | 980·6 | 0·2870 | 1·4966 | 1·7836 |
| 7·54 | 11·0 | 197·8 | 0·01662 | 35·11 | 165·7 | 1144·9 | 979·2 | 0·2904 | 1·4895 | 1·7799 |
| 6·51 | 11·5 | 199·9 | 0·01663 | 33·68 | 167·9 | 1145·8 | 977·9 | 0·2937 | 1·4828 | 1·7765 |
| 5·49 | 12·0 | 202·0 | 0·01664 | 32·37 | 169·9 | 1146·6 | 976·7 | 0·2968 | 1·4763 | 1·7731 |
| 4·47 | 12·5 | 204·0 | 0·01666 | 31·15 | 171·9 | 1147·4 | 975·5 | 0·2998 | 1·4701 | 1·7699 |
| 3·45 | 13·0 | 205·9 | 0·01667 | 30·03 | 173·8 | 1148·2 | 974·4 | 0·3026 | 1·4643 | 1·7669 |
| 2·43 | 13·5 | 207·5 | 0·01669 | 28·99 | 175·8 | 1149·0 | 973·2 | 0·3056 | 1·4584 | 1·7640 |
| 1·41 | 14·0 | 209·6 | 0·01670 | 28·02 | 177·6 | 1149·8 | 972·2 | 0·3083 | 1·4528 | 1·7611 |
| 0·39 | 14·5 | 211·3 | 0·01671 | 27·11 | 179·4 | 1150·5 | 971·1 | 0·3109 | 1·4475 | 1·7584 |
| 0 | 14·689 | 212 | 0·01671 | 26·77 | 180·0 | 1150·7 | 970·7 | 0·3199 | 1·4374 | 1·7573 |
| — | 15·0 | 213 | 0·01672 | 26·27 | 181·0 | 1151·2 | 970·2 | 0·3134 | 1·4423 | 1·7557 |
| — | 16 | 216·3 | 0·01674 | 24·73 | 184·4 | 1152·5 | 968·1 | 0·3184 | 1·4322 | 1·7506 |
| — | 17 | 219·5 | 0·01676 | 23·37 | 187·5 | 1153·7 | 966·2 | 0·3229 | 1·4229 | 1·7458 |
| — | 18 | 222·4 | 0·01678 | 22·16 | 190·5 | 1154·9 | 964·4 | 0·3274 | 1·4140 | 1·7414 |
| — | 19 | 225·2 | 0·01680 | 21·06 | 193·4 | 1156·0 | 962·6 | 0·3317 | 1·4056 | 1·7373 |
| — | 20 | 228·0 | 0·01682 | 20·08 | 196·1 | 1157·1 | 961·0 | 0·3356 | 1·3977 | 1·7333 |
| — | 21 | 230·6 | 0·01683 | 19·18 | 198·7 | 1158·1 | 959·4 | 0·3394 | 1·3900 | 1·7294 |
| — | 22 | 233·1 | 0·01685 | 18·37 | 201·3 | 1159·1 | 957·8 | 0·3431 | 1·3827 | 1·7258 |
| — | 23 | 235·5 | 0·01687 | 17·62 | 203·7 | 1160·0 | 956·3 | 0·3466 | 1·3757 | 1·7223 |
| — | 24 | 237·8 | 0·01689 | 16·93 | 206·1 | 1160·9 | 954·8 | 0·3500 | 1·3689 | 1·7189 |
| — | 25 | 240·1 | 0·01691 | 16·29 | 208·4 | 1161·7 | 953·3 | 0·3532 | 1·3625 | 1·7157 |
| — | 26 | 242·2 | 0·01692 | 15·71 | 210·5 | 1162·5 | 952·0 | 0·3563 | 1·3563 | 1·7126 |
| — | 27 | 244·4 | 0·01694 | 15·16 | 212·7 | 1163·4 | 950·7 | 0·3593 | 1·3504 | 1·7097 |
| — | 28 | 246·4 | 0·01695 | 14·66 | 214·8 | 1164·1 | 949·3 | 0·3623 | 1·3446 | 1·7069 |
| — | 29 | 248·4 | 0·01697 | 14·18 | 216·8 | 1164·9 | 948·1 | 0·3652 | 1·3390 | 1·7042 |
| — | 30 | 250·3 | 0·01699 | 13·74 | 218·8 | 1165·6 | 946·8 | 0·3680 | 1·3336 | 1·7016 |
| — | 31 | 252·2 | 0·01700 | 13·33 | 220·7 | 1166·3 | 945·6 | 0·3707 | 1·3284 | 1·6991 |
| — | 32 | 254·0 | 0·01702 | 12·94 | 222·5 | 1166·9 | 944·4 | 0·3733 | 1·3233 | 1·6966 |
| — | 33 | 255·8 | 0·01704 | 12·57 | 224·4 | 1167·6 | 943·2 | 0·3758 | 1·3185 | 1·6943 |
| — | 34 | 257·6 | 0·01705 | 12·22 | 226·1 | 1168·2 | 942·1 | 0·3782 | 1·3137 | 1·6919 |
| — | 35 | 259·2 | 0·01707 | 11·90 | 227·9 | 1168·8 | 940·9 | 0·3806 | 1·3091 | 1·6897 |
| — | 36 | 260·9 | 0·01708 | 11·59 | 229·5 | 1169·4 | 939·9 | 0·3829 | 1·3045 | 1·6874 |
| — | 37 | 262·5 | 0·01709 | 11·29 | 231·2 | 1170·0 | 938·8 | 0·3852 | 1·3000 | 1·6852 |
| — | 38 | 264·1 | 0·01710 | 11·02 | 232·8 | 1170·6 | 937·8 | 0·3874 | 1·2957 | 1·6831 |
| — | 39 | 265·7 | 0·01711 | 10·75 | 234·4 | 1171·2 | 936·8 | 0·3896 | 1·2915 | 1·6811 |
| — | 40 | 267·2 | 0·01713 | 10·50 | 235·9 | 1171·7 | 935·8 | 0·3918 | 1·2874 | 1·6792 |
| — | 42 | 270·2 | 0·01715 | 10·03 | 239·0 | 1172·8 | 933·8 | 0·3960 | 1·2794 | 1·6754 |
| — | 44 | 273·0 | 0·01717 | 9.603 | 241·9 | 1173·7 | 931·8 | 0·4000 | 1·2719 | 1·6719 |

| Press lb./in.$^2$ abs. $p$ | Temp. °F. $t$ | Specific Volume Ft.$^3$/lb. | | Total Heat B.Th.U./lb. | | Latent Heat B.Th.U./lb. | Entropy B.Th.U./lb.°F. | | |
|---|---|---|---|---|---|---|---|---|---|
| | | Water $u$ | Steam $v$ | Water $h$ | Steam H | L | Water $\phi_w$ | Evaporation $\phi_e$ | Total $\Phi$ |
| 46 | 275·7 | 0·01719 | 9·212 | 244·7 | 1774·7 | 930·0 | 0·4038 | 1·2647 | 1·6685 |
| 48 | 278·4 | 0·01723 | 8·853 | 247·4 | 1175·6 | 928·2 | 0·4075 | 1·2576 | 1·6651 |
| 50 | 280·9 | 0·01725 | 8·503 | 250·0 | 1176·4 | 926·4 | 0·4109 | 1·2510 | 1·6619 |
| 52 | 283·3 | 0·01728 | 8·196 | 252·6 | 1177·3 | 924·7 | 0·4144 | 1·2444 | 1·6588 |
| 54 | 285·7 | 0·01730 | 7·911 | 255·0 | 1178·1 | 923·1 | 0·4176 | 1·2384 | 1·6560 |
| 56 | 288·1 | 0·01733 | 7·645 | 257·4 | 1178·7 | 921·3 | 0·4208 | 1·2324 | 1·6532 |
| 58 | 290·3 | 0·01735 | 7·397 | 259·7 | 1179·5 | 919·8 | 0·4239 | 1·2265 | 1·6504 |
| 60 | 292·5 | 0·01737 | 7·166 | 262·0 | 1180·2 | 918·2 | 0·4269 | 1·2208 | 1·6477 |
| 62 | 294·7 | 0·01739 | 6·947 | 264·2 | 1180·9 | 916·7 | 0·4298 | 1·2153 | 1·6451 |
| 64 | 296·7 | 0·01741 | 6·742 | 266·4 | 1181·6 | 915·2 | 0·4327 | 1·2100 | 1·6427 |
| 66 | 298·8 | 0·01743 | 6·552 | 268·4 | 1182·1 | 913·7 | 0·4356 | 1·2047 | 1·6403 |
| 68 | 300·9 | 0·01745 | 6·369 | 270·5 | 1182·8 | 912·3 | 0·4381 | 1·1998 | 1·6379 |
| 70 | 302·7 | 0·01747 | 6·198 | 272·5 | 1183·4 | 910·9 | 0·4408 | 1·1948 | 1·6356 |
| 72 | 304·6 | 0·01749 | 6·036 | 274·4 | 1184·1 | 909·7 | 0·4434 | 1·1900 | 1·6334 |
| 74 | 306·4 | 0·01751 | 5·882 | 276·4 | 1184·5 | 908·1 | 0·4460 | 1·1852 | 1·6312 |
| 76 | 308·2 | 0·01752 | 5·737 | 278·3 | 1185·1 | 906·8 | 0·4484 | 1·1807 | 1·6291 |
| 78 | 310·0 | 0·01754 | 5·597 | 280·1 | 1185·6 | 905·5 | 0·4507 | 1·1764 | 1·6271 |
| 80 | 311·7 | 0·01756 | 5·466 | 281·9 | 1186·2 | 904·3 | 0·4530 | 1·1722 | 1·6252 |
| 82 | 313·4 | 0·01759 | 5·341 | 283·7 | 1186·6 | 902·9 | 0·4552 | 1·1681 | 1·6233 |
| 84 | 315·1 | 0·01761 | 5·22 | 285·4 | 1187·1 | 901·7 | 0·4575 | 1·1639 | 1·6214 |
| 86 | 316·8 | 0·01762 | 5·106 | 287·1 | 1187·6 | 900·5 | 0·4597 | 1·1599 | 1·6196 |
| 88 | 318·4 | 0·01763 | 4·996 | 288·8 | 1188·1 | 899·3 | 0·4618 | 1·1560 | 1·6178 |
| 90 | 320·0 | 0·01764 | 4·891 | 290·4 | 1188·6 | 898·2 | 0·4638 | 1·1522 | 1·6160 |
| 92 | 321·5 | 0·01766 | 4·791 | 292·0 | 1188·9 | 896·9 | 0·4658 | 1·1485 | 1·6143 |
| 94 | 323·1 | 0·01768 | 4·695 | 293·7 | 1189·4 | 895·7 | 0·4679 | 1·1449 | 1·6128 |
| 96 | 324·6 | 0·01769 | 4·602 | 295·2 | 1189·8 | 894·6 | 0·4700 | 1·1410 | 1·6110 |
| 98 | 326·0 | 0·01771 | 4·512 | 296·8 | 1190·2 | 893·4 | 0·4719 | 1·1373 | 1·6092 |
| 100 | 327·4 | 0·01773 | 4·428 | 298·3 | 1190·6 | 892·3 | 0·4739 | 1·1337 | 1·6076 |
| 105 | 330·9 | 0·01777 | 4·228 | 302·0 | 1191·5 | 889·5 | 0·4785 | 1·1253 | 1·6038 |
| 110 | 334·4 | 0·01780 | 4·046 | 305·6 | 1192·5 | 886·9 | 0·4830 | 1·1170 | 1·6000 |
| 115 | 337·1 | 0·01784 | 3·879 | 309·0 | 1193·3 | 884·3 | 0·4874 | 1·1091 | 1·5965 |
| 120 | 340·7 | 0·01788 | 3·726 | 312·3 | 1194·1 | 881·8 | 0·4916 | 1·1014 | 1·5930 |
| 125 | 343·8 | 0·01791 | 3·584 | 315·6 | 1194·8 | 879·2 | 0·4957 | 1·0940 | 1·5897 |
| 130 | 346·8 | 0·01794 | 3·454 | 318·7 | 1195·6 | 876·9 | 0·4996 | 1·0870 | 1·5866 |
| 135 | 349·7 | 0·01798 | 3·332 | 321·8 | 1196·3 | 874·5 | 0·5033 | 1·0803 | 1·5836 |
| 140 | 352·6 | 0·01801 | 3·219 | 324·8 | 1196·9 | 872·1 | 0·5069 | 1·0739 | 1·5808 |
| 145 | 355·2 | 0·01804 | 3·144 | 327·7 | 1197·6 | 869·9 | 0·5104 | 1·0676 | 1·5780 |
| 150 | 357·9 | 0·01807 | 3·014 | 330·5 | 1198·2 | 867·7 | 0·5138 | 1·0615 | 1·5753 |
| 155 | 360·5 | 0·01811 | 2·922 | 333·2 | 1198·9 | 865·7 | 0·5170 | 1·0568 | 1·5738 |
| 160 | 363·0 | 0·01814 | 2·835 | 335·9 | 1199·4 | 863·5 | 0·5202 | 1·0500 | 1·5702 |
| 165 | 365·4 | 0·01817 | 2·753 | 338·5 | 1199·9 | 861·4 | 0·5234 | 1·0443 | 1·5677 |
| 170 | 367·8 | 0·01820 | 2·675 | 341·1 | 1200·4 | 859·3 | 0·5266 | 1·0385 | 1·5651 |
| 175 | 370·1 | 0·01822 | 2·603 | 343·6 | 1200·9 | 857·3 | 0·5297 | 1·0331 | 1·5628 |
| 180 | 372·4 | 0·01825 | 2·534 | 346·1 | 1201·4 | 855·3 | 0·5326 | 1·0278 | 1·5604 |
| 185 | 374·6 | 0·01828 | 2·467 | 348·5 | 1201·9 | 853·4 | 0·5355 | 1·0226 | 1·5581 |
| 190 | 376·8 | 0·01831 | 2·407 | 350·9 | 1202·3 | 851·4 | 0·5383 | 1·0176 | 1·5559 |
| 195 | 379·0 | 0·01834 | 2·347 | 353·2 | 1202·7 | 849·5 | 0·5410 | 1·0129 | 1·5539 |
| 200 | 381·1 | 0·01837 | 2·290 | 355·5 | 1203·1 | 847·6 | 0·5437 | 1·0082 | 1·5519 |
| 205 | 383·2 | 0·01839 | 2·236 | 357·7 | 1203·5 | 845·8 | 0·5465 | 1·0035 | 1·5500 |
| 210 | 385·2 | 0·01842 | 2·196 | 359·9 | 1203·9 | 844·0 | 0·5490 | 0·9991 | 1·5481 |
| 215 | 387·2 | 0·01845 | 2·137 | 362·0 | 1204·2 | 842·2 | 0·5515 | 0·9947 | 1·5462 |
| 220 | 389·2 | 0·01847 | 2·090 | 364·2 | 1204·6 | 840·4 | 0·5539 | 0·9904 | 1·5443 |
| 225 | 391·1 | 0·01850 | 2·046 | 366·3 | 1204·9 | 838·6 | 0·5564 | 0·9860 | 1·5424 |
| 230 | 392·9 | 0·01853 | 2·003 | 368·4 | 1205·2 | 836·8 | 0·5589 | 0·9806 | 1·5405 |

| Press lb./in.$^2$ abs. $p$ | Temp. °F. $t$ | Specific Volume Ft.$^3$/lb. | | Total Heat B.Th.U./lb. | | Latent Heat B.Th.U./lb. | Entropy B.Th.U./lb.°F. | | |
|---|---|---|---|---|---|---|---|---|---|
| | | Water $u$ | Steam $v$ | Water $h$ | Steam H | L | Water $\phi_w$ | Evaporation $\phi_e$ | Total $\Phi$ |
| 235 | 394·8 | 0·01856 | 1·962 | 370·4 | 1205·6 | 835·2 | 0·5612 | 0·9775 | 1·5387 |
| 240 | 396·7 | 0·01858 | 1·923 | 372·4 | 1205·9 | 833·5 | 0·5635 | 0·9735 | 1·5370 |
| 245 | 398·5 | 0·01861 | 1·885 | 374·4 | 1206·2 | 831·8 | 0·5658 | 0·9695 | 1·5353 |
| 250 | 400·2 | 0·01863 | 1·849 | 376·3 | 1206·4 | 830·1 | 0·5680 | 0·9655 | 1·5335 |
| 260 | 403·7 | 0·01869 | 1·779 | 380·1 | 1207·0 | 826·9 | 0·5723 | 0·9580 | 1·5303 |
| 270 | 407·0 | 0·01874 | 1·716 | 383·8 | 1207·5 | 823·7 | 0·5766 | 0·9507 | 1·5273 |
| 280 | 410·3 | 0·01878 | 1·657 | 387·4 | 1207·9 | 820·5 | 0·5808 | 0·9434 | 1·5242 |
| 290 | 413·4 | 0·01883 | 1·601 | 391·0 | 1208·4 | 817·4 | 0·5848 | 0·9364 | 1·5210 |
| 300 | 416·6 | 0·01888 | 1·550 | 394·4 | 1208·8 | 814·4 | 0·5886 | 0·9297 | 1·5183 |
| 310 | 419·5 | 0·01893 | 1·501 | 397·8 | 1209·1 | 811·3 | 0·5925 | 0·9229 | 1·5154 |
| 320 | 422·4 | 0·01898 | 1·455 | 401·1 | 1209·5 | 808·4 | 0·5962 | 0·9165 | 1·5127 |
| 330 | 425·3 | 0·01902 | 1·412 | 403·3 | 1209·9 | 805·6 | 0·5998 | 0·9102 | 1·5100 |
| 340 | 428·1 | 0·01907 | 1·372 | 407·4 | 1210·2 | 802·8 | 0·6033 | 0·9032 | 1·5065 |
| 350 | 430·9 | 0·01912 | 1·334 | 410·6 | 1210·4 | 799·8 | 0·6068 | 0·8989 | 1·5050 |
| 360 | 433·5 | 0·01916 | 1·298 | 413·6 | 1210·7 | 797·1 | 0·6101 | 0·8924 | 1·5025 |
| 370 | 436·2 | 0·01921 | 1·263 | 416·5 | 1210·9 | 794·4 | 0·6134 | 0·8868 | 1·5002 |
| 380 | 438·7 | 0·01925 | 1·231 | 419·5 | 1211·2 | 791·7 | 0·6165 | 0·8814 | 1·4979 |
| 390 | 441·2 | 0·01929 | 1·199 | 422·3 | 1211·3 | 789·0 | 0·6196 | 0·8761 | 1·4957 |
| 400 | 443·7 | 0·01933 | 1·171 | 425·2 | 1211·6 | 786·4 | 0·6227 | 0·8706 | 1·4933 |
| 410 | 446·1 | 0·01938 | 1·142 | 428·0 | 1211·7 | 783·7 | 0·6257 | 0·8654 | 1·4911 |
| 420 | 448·6 | 0·01943 | 1·116 | 430·7 | 1211·8 | 781·1 | 0·6287 | 0·8603 | 1·4890 |
| 430 | 450·9 | 0·01947 | 1·090 | 433·4 | 1212·0 | 778·6 | 0·6317 | 0·8551 | 1·4868 |
| 440 | 453·1 | 0·01951 | 1·065 | 436·0 | 1212·1 | 776·1 | 0·6346 | 0·8502 | 1·4848 |
| 450 | 455·3 | 0·01956 | 1·042 | 438·7 | 1212·2 | 773·5 | 0·6374 | 0·8453 | 1·4827 |
| 460 | 457·6 | 0·01959 | 1·019 | 441·2 | 1212·2 | 771·0 | 0·6403 | 0·8406 | 1·4808 |
| 470 | 459·8 | 0·01964 | 0·998 | 443·8 | 1212·3 | 768·5 | 0·6429 | 0·8359 | 1·4788 |
| 480 | 461·9 | 0·01968 | 0·978 | 446·3 | 1212·3 | 766·0 | 0·6456 | 0·8313 | 1·4769 |
| 490 | 464·0 | 0·01972 | 0·957 | 448·8 | 1212·4 | 763·6 | 0·6483 | 0·8267 | 1·4750 |
| 500 | 466·0 | 0·01977 | 0·938 | 451·2 | 1212·4 | 761·2 | 0·6487 | 0·8245 | 1·4732 |
| 600 | 485·4 | 0·02013 | 0·7805 | 471·6 | 1212·0 | 740·4 | 0·6722 | 0·7837 | 1·4559 |
| 700 | 502·1 | 0·02048 | 0·6672 | 491·6 | 1211·1 | 719·5 | 0·6929 | 0·7478 | 1·4407 |
| 800 | 517·5 | 0·02084 | 0·5804 | 510·1 | 1209·1 | 699·0 | 0·7117 | 0·7152 | 1·4269 |
| 900 | 531·3 | 0·02119 | 0·5120 | 527·4 | 1206·3 | 678·9 | 0·7288 | 0·6854 | 1·4142 |
| 1000 | 544·1 | 0·02152 | 0·4570 | 543·6 | 1203·4 | 659·8 | 0·7443 | 0·6578 | 1·4021 |
| 1200 | 566·9 | 0·02224 | 0·3727 | 573·4 | 1196·1 | 622·7 | 0·7734 | 0·6064 | 1·3798 |
| 1400 | 587·1 | 0·02310 | 0·3111 | 601·3 | 1186·9 | 585·6 | 0·7992 | 0·5595 | 1·3587 |
| 1600 | 605·1 | 0·02390 | 0·2635 | 627·3 | 1176·0 | 548·7 | 0·8230 | 0·5156 | 1·3386 |
| 1800 | 621·5 | 0·02475 | 0·2255 | 652·4 | 1163·8 | 511·4 | 0·8454 | 0·4732 | 1·3186 |
| 2000 | 636·5 | 0·02574 | 0·1936 | 676·8 | 1149·8 | 473·0 | 0·8669 | 0·4316 | 1·2985 |
| 2400 | 663·1 | 0·02820 | 0·1433 | 726·3 | 1115·1 | 388·8 | 0·9098 | 0·3463 | 1·2561 |
| 2800 | 685·4 | 0·03110 | 0·1029 | 780·4 | 1069·2 | 288·8 | 0·9556 | 0·2522 | 1·2078 |
| 3200 | 704·4 | 0·04700 | 0·0629 | 861·7 | 1002·2 | 140·5 | 1·0231 | 0·1207 | 1·1438 |

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