HEAT ENGINES

HEAT ENGINES

Steam, Gas, Steam Turbines
and Their Auxiliaries

BY

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HEAT ENGINES

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Preface to the Fifth Edition

Once more has the progress in the development of heat engines and their auxiliaries made a revision of this text necessary.

Particular attention has been given to the chapters on Boiler Auxiliaries, Steam Turbines, and The Internal-combustion Engine, and the text has been largely rewritten and new illustrations have taken the place of the older ones. Through the courtesy of the authors, Professors Joseph H. Keenan and Frederick G. Keyes, and the publishers, John Wiley & Sons, it has been possible to include in this edition portions of their new Steam Tables. The old chapters on Compound Engines, Valve Gears, and Governors have been abridged and incorporated in the chapters on Steam Engines, and Power and Performance of Steam Engines. A new chapter on Air Compressors and Refrigerating Machinery has been added.

The author wishes to take this opportunity to express his appreciation of the courtesy of those manufacturers who have supplied illustrations and descriptions of their apparatus for use in the text. He also wishes to thank his colleagues on the faculty of the University of Michigan for their assistance in the preparation of this revision. Particular help was given by Professor Axel Marin, who wrote the chapter on Air Compressors and Refrigerating Machinery, Assistant Professor Henry L. Kohler, who rewrote the chapters on The Internal-combustion Engine, Mr. Charles W. Spooner, who revised the chapter on Steam Turbines and several other portions of the book, and to Professors Ransom S. Hawley and Hugh E. Keeler and Assistant Professors Floyd N. Calhoun and Clarence F. Kessler for many valuable suggestions.

JOSEPH A. BURSLEY.

ANN ARBOR, MICH.,
August, 1941.

Preface to the Third Edition

The untimely death of Prof. John R. Allen, whose going was a distinct loss to the whole engineering profession, has thrown upon the other author of this text the responsibility for this edition.

The larger part of the book has been entirely rewritten in order to bring it up to date. The new material included is mainly that which was necessary to make the text conform to current engineering practice.

At the request of a number of the users of the book, several paragraphs on Entropy have been added to the chapter on Thermodynamics. However, as it has always been the intention of the authors to make the text an elementary one to be followed by more advanced courses for those who wish to go into more complete discussion of the subject of Heat Engines, no further reference to entropy has been made anywhere in the book, except in the discussion of the Rankine cycle.

The author wishes to acknowledge his indebtedness to his colleagues in the College of Engineering of the University of Michigan, Prof. J. E. Emswiler, Prof. C. H. Fessenden, Mr. Charles W. Good, and Mr. George G. Brown for their assistance in preparing respectively the chapters on Steam Turbines, Valve Gears and Governors, Internal-combustion Engines, and Fuels and Combustion. He also desires to thank those manufacturers who have cooperated by supplying illustrations of their apparatus.

JOSEPH A. BURSLEY.

ANN ARBOR, MICHIGAN,
June, 1925.

Preface to the First Edition

In preparing this book, it has been the intention of the authors to present an elementary treatise upon the subject of Heat Engines, considering only those engines which are most commonly used in practice. It is written primarily as a text-book, the subject matter having been used in the classes at the University of Michigan for a number of years.

The forms of heat engines discussed include the steam engine with its boiler plant and auxiliaries, the gas engine with its producer, oil engines, and the principal types of steam turbines. Under each division of the next, problems have been worked out in detail to show the application of the subject matter just treated, and, in addition, a large number of problems have been introduced for classroom work. The use of calculus and higher mathematics has been largely avoided, the only place where it is used being in the chapter on thermodynamics, which subject has been treated in its elementary phases only. The matter of the design of engines has been left untouched, as it was felt that that subject did not properly come within the scope of this work.

The authors wish to express their thanks to Professors H. C. Anderson, A. H. Knight, and J. A. Moyer, for their assistance in compiling this work, to Mr. W. R. McKinnon, who made a number of the drawings, and to the various manufacturers who have very kindly furnished illustrations and descriptions of their apparatus.

JOHN R. ALLEN.
JOSEPH A. BURSLEY.

ANN ARBOR, MICHIGAN,
Sept. 1, 1910.

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Symbols for Heat and Thermodynamics[1]

General Principles

Specific heat at constant pressure	c_p
Specific heat at constant volume	c_v
Thermal efficiency	e_t
Mechanical efficiency	e_m
Enthalpy of dry saturated vapor	h_g
Initial pressure	p_1
Final pressure	p_2

General Symbols

Acceleration due to gravity, for most engineering work, where it is not necessary to distinguish between the exact local value and the standard value given below	g
Velocity	v
Revolutions per unit time	n
Length	L
Diameter	D
Radius	r
Area	A
Total volume	V

Symbols for Force, Weight, Work, and Power

Force; total load	F
Pressure; absolute pressure; gage pressure; force per unit area	p
Mean effective pressure	p_m
Total quantity of fluid, water, gas, heat; quantity by volume	Q
Volume per unit time; rate at which quantity of material passes through a machine; quantity of heat per unit time; quantity of heat per unit weight	q
Total weight	W
Weight rate; weight per unit of power; weight per unit of time	w
Mass	m
Total work	W
Work per unit weight	w
Power; work per unit time	P
Efficiency	e

[1] From *Mechanical Engineering*, May, 1930.

Symbols for Properties

Specific volume; volume per unit weight; volume per unit mass . . v
Density; weight per unit volume; mass per unit volume; specific weight . . . . d

$$d = \frac{1}{v}$$

Specific heat c
Specific heat at constant pressure . . . . c_p
Specific heat at constant volume . . . . c_v
Ratio of specific heats . . . . κ (kappa)

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

Symbols for General Thermodynamics

Mechanical equivalent of heat . . . . J
Heat equivalent of work . . . . $\frac{1}{J}$ or A
Ordinary temperature, Fahrenheit (F) or centigrade (C) . . . . t
Absolute temperature, Fahrenheit absolute . . . . T
Gas constant in equation $pv = RT$ . . . . R
Quality of steam, pounds of dry steam per pound of mixture x
Exponent of polytropic expansion, pv^n = const . . . . n

Symbols for the Thermodynamic Quantities

Entropy (the capital should be used for any weight, and the small letter for unit weight) . . . . S or s
Internal energy; intrinsic energy (the capital should be used for any weight and the small letter for unit weight) . . . . U or u
Enthalpy; heat content; total heat (the capital should be used for any weight and the small letter for unit weight) . . . . H or h

$$h = u + Apv$$

Symbols for Saturation, Vaporization, Etc.

Enthalpy of saturated liquid, sometimes called heat of the liquid; heat content of saturated liquid; total heat of saturated liquid . . . . h_f
Enthalpy of dry saturated vapor; heat content of dry saturated vapor; total heat of dry saturated vapor . . . . h_g
Heat of vaporization at constant pressure . . . . h_{fg} or L

Abbreviations

Absolute	abs
Barometer	bar.
Boiler horsepower	boiler hp
Brake horsepower	bhp
Brake horsepower-hour	bhp-hr
British thermal unit	Btu or B
Cent	c or ¢
Cubic foot	cu ft
Cubic centimeter	cc
Degree centigrade	C
Degree Fahrenheit	F
Diameter	diam
Feet per minute	fpm
Feet per second	fps
Foot	ft
Foot-pound	ft-lb
Friction horsepower	fhp
Gallon	gal
Gallons per minute	gpm
High-pressure (adjective)	h-p
Horsepower	hp
Horsepower-hour	hp-hr
Hour	hr
Inch	in.
Indicated horsepower	ihp
Indicated horsepower-hour	ihp-hr
Kilowatt	kw
Kilowatt-hour	kwhr
Logarithm	log
Logarithm (natural)	$\log_e$
Mean effective pressure	mep
Miles per hour	mph
Minute	min
Pound	lb
Pounds per square foot	pst
Pounds per square inch	psi
Pounds per square inch absolute	psia
Revolutions per minute	rpm
Second	sec
Specific gravity	sp gr
Specific heat	sp ht
Square foot	sq ft
Square inch	sq in.
Temperature	temp
Weight	wt

HEAT ENGINES

STEAM—GAS—STEAM TURBINES—AND THEIR AUXILIARIES

CHAPTER I

HEAT

1. Heat.—Since heat is the source of energy for the devices considered in this book, a short discussion of the nature and the more important properties of heat will assist the student to a better understanding of the subject matter of the text. These phenomena will be considered only as they affect perfect gases, steam, and water.

2. Theory of Heat.—The accepted theory of heat at the present time is that it is a motion of the molecules of a body. Physical experiments indicate this to be the fact. The intensity of the heat, or the temperature, is supposed to depend upon the velocity and the amplitude of these vibrations.

Most bodies expand when heated. This expansion is probably due to the increased velocity of the molecules, which forces them farther apart and increases the actual size of the body. The vibration may become so violent that the attraction between the molecules is partly overcome and the body can no longer retain its form. In this case the solid becomes a liquid. If still more heat is added, the attraction of the molecules may be entirely overcome by their violent motion, and the liquid then becomes a gas.

The phenomenon of heat is, then, a form of motion. This is often stated in another way, *viz.*, heat is a form of kinetic energy. As heat is a form of motion, it must be possible to transform heat into mechanical motion. In the following pages, therefore, the most important methods of making this transformation will be discussed.

3. Temperature and Temperature Measurement.—The velocity of the vibration of the molecules of a body determines the intensity of the heat, and this intensity is measured by the temperature. If the molecules of a body move slowly, it is at a low temperature; if they move rapidly, it is at a high temperature. The temperature of a body is, therefore, determined by the rapidity of the motion of its molecules.

Temperature is sometimes defined as *the thermal state of a body considered with reference to its ability to transmit heat to other bodies.* Two bodies are said to be at the same temperature when there is no transmission of heat between them. If there is transmission of heat between them, the one from which the heat is flowing is said to have the higher temperature.

In mechanical engineering work, temperatures are usually measured on the Fahrenheit scale, and in this text, unless otherwise stated, the temperature will be taken on this scale. There is, however, an increasing use of the centigrade scale among engineers, and certain quantities, such as the increase in temperature in a dynamo, are always expressed in centigrade units.

On the Fahrenheit scale the graduations are obtained by noting the position of the top of the mercury column when the bulb of the thermometer is placed in melting ice, and again when it is placed in boiling water under an atmospheric pressure corresponding to sea-level barometer. The distance between these two points is divided into 180 equal parts or degrees. The freezing point is arbitrarily taken as 32 degrees above the zero of the scale, thus making the boiling point 32 degrees + 180 degrees = 212 degrees above zero.

On the centigrade scale the distance between the freezing point and the boiling point is divided into 100 equal parts or degrees, and the freezing point on the scale is marked 0 C. The boiling point is then 100 C.

Since on the Fahrenheit scale there are 180 divisions between the freezing and boiling points and on the centigrade 100 divisions, it follows that $1\text{ F} = \frac{5}{9}\text{ C}$, or $1\text{ C} = \frac{9}{5}\text{ F}$. As, however, the freezing point on the Fahrenheit scale is marked 32 F and on the centigrade scale 0 C, it is necessary to take account of this difference when converting from one scale to the other. If the temperature Fahrenheit be denoted by t_F and the temperature centigrade by t_C, then the conversion from one scale to the other may be made by the following equations:

$$t_F = \tfrac{9}{5}t_C + 32; \tag{1}$$

$$t_C = \tfrac{5}{9}(t_F - 32). \tag{2}$$

The measurement of temperature is not so simple a process as is generally supposed. The mercury of the ordinary glass thermometer does not expand equal amounts for equal increments of heat, and the bore of the thermometer is not absolutely uniform throughout the whole length of the tube. These inaccuracies must be allowed for by accurate calibration. In measuring the temperatures of liquids, the depth to which the thermometer is immersed affects the reading, and it should be calibrated at the depth at which it is to be used. If a thermometer is used to measure the temperature of the air in a room in which there are objects at a higher temperature, its bulb must be protected from the radiant heat of those hot bodies. When accurate temperature measurements are desired, a careful study should be made of the errors of the instrument and the errors in its use.

The ordinary form of mercury thermometer is used for temperatures ranging from −40 F to 500 F. For measuring temperatures below −40 F, thermometers filled with alcohol are used. These are, however, not satisfactory for use at high temperatures. When a mercury thermometer is used for temperatures above 500 F, the space above the mercury is filled with some inert gas, usually nitrogen or carbon dioxide, placed in the thermometer tube under pressure. As the mercury rises, the gas pressure is increased and the temperature of the boiling point of the mercury is raised, so that it is possible to use these thermometers for temperatures as high as 1000 F. This is the limit, however, as the melting point of glass is comparatively low.

For temperatures exceeding 800 F, some form of *pyrometer* is generally used. The simplest of these is the *metallic* or *mechanical pyrometer.* This consists of two metals having different rates of expansion, such as iron and brass, attached to each other at one end and with the other ends free. By a system of levers and gears the expansion of the metals is made to move a hand over a dial graduated in degrees. This should not be used for temperatures exceeding 1000 F to 1200 F.

There are two types of *electrical pyrometers* in use today. In one, the *thermoelectric couple* is employed and the difference in temperatures of the junctions of the two metals forming the couple produces an electric current which is proportional to this difference, and which is measured on a galvanometer calibrated in degrees. By keeping one junction at a known temperature, the other may be computed. This may be used up to 2500 F.

The second type, the *electrical-resistance pyrometer*, depends upon the increase in electrical resistance of metals due to a rise in temperature.

For still higher temperatures, the *optical pyrometer* gives the most satisfactory results. This is based on the results of experiments made by Pouillet, which show that incandescent bodies have for each temperature a definite and fixed color, as follows:

TABLE I.—TEMPERATURE COLORS

Color	Temperature, deg C	Temperature, deg F
Faint red	525	977
Dark red	700	1292
Faint cherry	800	1472
Cherry	900	1652
Bright cherry	1000	1832
Dark orange	1100	2012
Bright orange	1200	2192
White heat	1300	2372
Bright white	1400	2552
Dazzling white	1500 1600	2732 2912

4. Absolute Zero of Temperature.—In considering heat from a theoretical standpoint, it is necessary to have some absolute standard of comparison for the scale of temperature, so that the *absolute scale* is largely used.

A perfect gas under constant pressure contracts $\frac{1}{491.6}$ of its volume at 32 F for each degree that it is reduced in temperature. Hence, if the temperature be lowered to a point 491.6 degrees below 32 F, its volume will become zero. This point is called the *absolute zero* and is manifestly an imaginary one. (The lowest point so far actually reached by experiment is about -458.1 F, *i.e.*, 490.1 degrees below 32 F or 1.5 degrees above absolute zero.) For ordinary usage it is sufficiently accurate to consider absolute zero as 492 degrees below the freezing point on the Fahrenheit scale. In other words, to convert to *absolute temperature,* add 460 to the temperature expressed in degrees Fahrenheit. In this text absolute temperatures will be denoted by T and temperatures in degrees Fahrenheit by t.

On the centigrade scale the absolute zero is 273.1 degrees below the freezing point, and for all practical purposes temperatures on the absolute scale may be found by adding 273 to the thermometer reading expressed in degrees centigrade.

5. Unit of Heat.—Heat is not a substance, and it cannot be measured as water would be measured, in pounds or cubic feet, but it must be measured by the effects which it produces. The unit of heat used in mechanical engineering is the heat required to raise one pound of water one degree Fahrenheit. This unit is called a *British thermal unit,* and is denoted by *Btu.* As the heat necessary to raise a pound of water one degree does not remain the same throughout any great range of temperature, the American Society of Mechanical Engineers (A.S.M.E.) has defined a Btu as $\frac{1}{180}$ *of the heat required to raise one pound of water from* 32 *F to* 212 *F at atmospheric pressure* (14.7 psia).

6. Specific Heat.—If the temperature of a body is raised or lowered a definite amount, a definite amount of heat must either be added to or given up by the body.

Then

$$dQ = cdt, \tag{3}$$

where c is the heat necessary to change the temperature of the body one degree. If the temperature is raised from T_1 to T_2 degrees and at the same time the heat content is increased from Q_1 to Q_2, then the heat Q added to cause this increase in temperature will be found by integrating Eq. (3) between the limits T_1 and T_2, or

$$Q = Q_2 - Q_1 = \int_{T_1}^{T_2} cdt.$$

If the body is of unit weight and c is a constant and equal to the heat necessary to raise the temperature one degree,

$$Q = c\int_{T_1}^{T_2} dt = c(T_2 - T_1). \tag{4}$$

In Eq. (4) c represents the *heat capacity* of the body, or the heat required to raise the temperature of a unit weight of the body one degree.

The heat capacity of any substance compared with that of an equal weight of water is called its *specific heat.*

Expressed in English units, the heat capacity of one pound of water is one Btu, and specific heat may be defined as *the heat necessary to raise the temperature of one pound of a substance one degree Fahrenheit, expressed in British thermal units.* That is, the specific heat of any substance is the *ratio of the heat necessary to raise the temperature of a unit weight of the substance one degree to the heat necessary to raise the same weight of water through the same range of temperature.*

TABLE II.—PROPERTIES OF GASES

Gas	Chemical symbol	Number of atoms	Molecular weight		Weight in pounds of 1 cu ft at atmospheric pressure		Mean specific heat between 32 F and 400 F, expressed in Btu per pound of gas		$R = J(c_p - c_v)$	$\kappa = \frac{c_p}{c_v}$
			Approximate	Exact, $O_2 = 32$	At 62 F	At 32 F	Constant pressure, c_p	Constant volume, c_v		
Air	......	..	29.0	28.95	0.0761	0.0807	0.2410	0.1725	53.33	1.397
Oxygen	O_2	2	32.0	32.00	0.0840	0.0892	0.2182	0.1562	48.25	1.397
Nitrogen	N_2	2	28.0	28.016	0.0737	0.0783	0.2491	0.1783	55.11	1.398
Hydrogen	H_2	2	2.0	2.016	0.00529	0.00562	3.426	2.441	766.89	1.404
Carbon monoxide	CO	2	28.0	28.00	0.0734	0.0780	0.2491	0.1783	55.34	1.398
Carbon dioxide	CO_2	3	44.0	44.00	0.1156	0.1227	0.2080	0.1622	35.66	1.282
Sulphur dioxide	SO_2	3	64.0	64.065	0.1684	0.1786	0.154	0.123	24.14	1.252
Ammonia	NH_3	4	17.0	17.031	0.04483	0.0476	0.523	0.399	96.54	1.311
Acetylene	C_2H_2	4	26.0	26.015	0.0684	0.0725	0.350	0.270	62.29	1.296
Methane	CH_4	5	16.0	16.03	0.0421	0.0447	0.6606	0.5360	97.01	1.232
Ethylene	C_2H_4	6	28.0	28.03	0.0738	0.0780	0.40	0.33	54.50	1.212

In solid and liquid substances it is necessary to consider but one specific heat, as the change in volume when a solid or a liquid substance is heated is so small that its effect may be neglected. In gases the change in volume when the gas is heated is large, and if it is heated under a constant pressure this change is directly proportional to the change in the absolute temperature. If there is a change in volume there must be external work done. On the other hand, when gas is confined and is heated, it cannot expand. If it does not expand, there is no external work done. Therefore, in considering the specific heat of a gas, two cases must be considered—one in which the pressure remains constant and the gas expands when it is heated; and the other where the volume remains constant and the pressure increases when the gas is heated. Hence, for a gas, there are two specific heats, the *specific heat of constant pressure* and the *specific heat of constant volume.* The specific heat of constant volume will be denoted by c_v and the specific heat of constant pressure by c_p, both being expressed in Btu.

7. Radiation.—The heat that passes from a body by radiation may be considered similar to the light that is radiated from a lamp. There is always a transfer of radiant heat from a body of a high temperature to a body of lower temperature. The amount of heat radiated will depend upon the difference in temperature between the bodies and upon the substances of which they are composed. The following table gives the radiating power of different bodies:

TABLE III.—RADIATING POWER OF BODIES

[Radiating power of bodies, expressed in heat units, given off per square foot per hour for a difference of one degree Fahrenheit (Feclet)]

	Btu
Copper, polished	0.0327
Iron, sheet	0.0920
Glass	0.595
Cast iron, rusted	0.648
Building stone, plaster, wood, brick	0.7358
Woolen stuffs, any color	0.7522
Water	1.085

8. Conduction.—The heat transmitted by conduction is the heat transmitted through the body itself. The amount of heat conducted will depend upon the material of which the body is composed and the difference in temperature between the two sides of the body, and is inversely proportional to the thickness of the body. Heat will be conducted from one body to another when they are placed in contact with each other.

The following table gives the conducting power of different bodies:

TABLE IV.—CONDUCTING POWER OF BODIES

[The conducting power of materials, expressed in the quantity of heat units transmitted per sq ft per hr by a plate 1 in. thick, the surfaces on the two sides of the plate differing in temperature by one degree Fahrenheit (Peclet).]

	Btu
Copper	515
Iron	233
Lead	113
Stone	16.7
Glass	6.6
Brickwork	4.8
Plaster	3.8
Pine wood	0.75
Sheep's wool	0.323

9. Convection.—Loss by convection is sometimes called loss by contact of air. When air or other gas comes in contact with a hot body it is heated and rises, carrying away heat from the body. Heat carried off in this manner is said to be lost by convection. The loss by convection is independent of the nature of the surface—wood, stone, or iron losing the same amount—but it is affected by the form and position of the body.

10. Energy, Work, and Power.—*Work* is the overcoming of resistance through space and is measured by the resistance multiplied by the space through which this resistance is overcome. The simplest form of work is the raising of a body against the force of gravity.

Let

m = the mass of the body;
g = the force of gravity;
w = the weight;
l = the distance through which the weight is moved;
W = work.

Then

$$w = mg, \qquad \text{and} \qquad W = wl.$$

If w is expressed in pounds and l in feet, then the unit of work will be the foot-pound (ft-lb).

In considering the work done by a fluid, let the volume be increased from v to $v + \delta v$, and the pressure against which the increase takes place be p; then the work done will be

$$p[(v + \delta v) - v] = p\delta v = \delta W.$$

If a pressure p acts upon an area a through a distance l, then the work

$$W = pla.$$

Work may also be expressed as mass times acceleration times space.

Energy is the capacity for doing work.

Power is the time rate of doing work. The unit of power is the horsepower (hp). *A horsepower is equivalent to raising* 33,000 *pounds one foot in one minute.* This is the unit employed

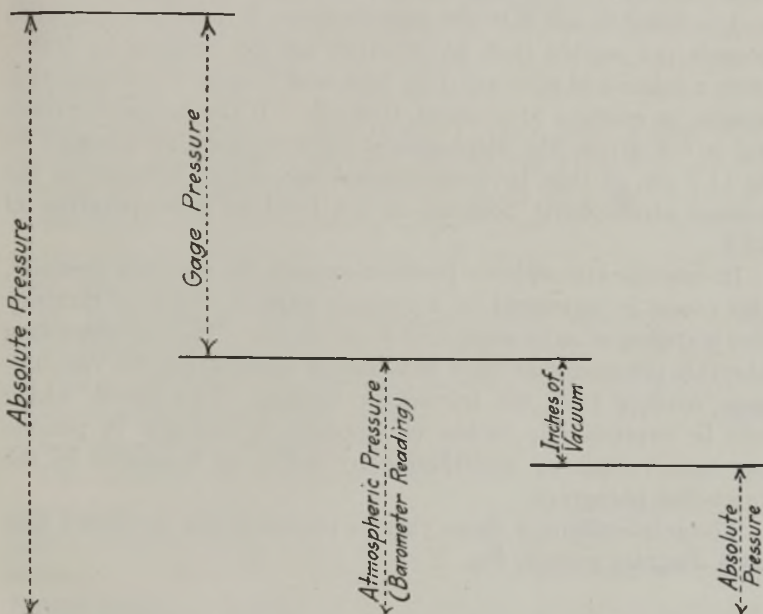


FIG. 1.—Relationship of various kinds of pressure.

in determining the power of a steam engine. If r equals the resistance expressed in pounds, l the distance in feet through which the resistance r is overcome, and t the time in minutes in which the space is passed over, then the horsepower exerted is

$$\frac{l \times r}{33{,}000 \times t}.$$

Power is often expressed in electrical units. This is usually the case where an engine is used to drive a generator. An *ampere* is the unit of current strength or rate of flow. The *volt* is the unit of electromotive force or electrical pressure. The *watt* is the product of the amperes and the volts. One horsepower equals 746 watts, or 1 kw equals 1.34 hp.

11. **Pressure.**—The ordinary pressure gage reads the amount by which the total pressure exerted upon its mechanism exceeds the pressure of the atmosphere. This is called *gage* pressure and is read in pounds per square inch. *Absolute* pressure is found by adding to this gage pressure the *atmospheric* pressure as read on a barometer. The barometer reading is in inches of mercury and, in order to add it to the gage pressure, it must be changed to pounds per square inch by multiplying the reading by 0.491, since a column of mercury 1 in. high and 1 sq in. in cross section weighs, or exerts a pressure of, 0.491 lb. If the barometer reading is not given, the atmospheric pressure may be assumed to be 14.7 psi, as this, by international agreement, is taken as the normal atmospheric pressure at sea level at a temperature of 32 F.

In case the atmospheric pressure exceeds the absolute pressure, this excess is registered on a vacuum gage in inches of mercury and is spoken of as so many *inches of vacuum*. The corresponding absolute pressure may then be found by subtracting the vacuum-gage reading from the barometer reading. The result, which will be expressed in inches of mercury, is changed to pounds per square inch by multiplying by 0.491, as indicated in the preceding paragraph.

The relationship of these various pressures can be shown best by a diagram such as Fig. 1.

CHAPTER II

ELEMENTARY THERMODYNAMICS

12. Thermodynamics.—The Century Dictionary defines thermodynamics as *the general doctrine of the relations of heat and work, involving the consideration of temperature, volume, pressure, and the transformations of energy.* More particularly, as far as the engineer is concerned, it may simply be stated as *that branch of engineering science which deals with the reciprocal conversion of heat and work.* The equipment found in a power plant, such as boilers, steam engines, steam turbines, gas engines, and their auxiliaries, which are dealt with later on in the text, are all devices for bringing about the conversion of heat into work. In the design and development of each of these, an understanding of the laws of thermodynamics and of the properties of perfect gases has been necessary.

13. First Law of Thermodynamics.—*When mechanical energy is produced from heat, a definite quantity of heat goes out of existence for every unit of work done; and, conversely, when heat is produced by the expenditure of mechanical energy, the same definite quantity of heat comes into existence for every unit of work spent.*

The relation between work and heat was first accurately determined by Joule in 1843. Later, Professor Rowland of the Johns Hopkins University redetermined its value with great accuracy. His results showed *one Btu to be equivalent to* 778 *ft-lb.* Recent experiments made at the U. S. Bureau of Standards give 778.57 as the correct figure, although 778 is generally used in engineering work. This factor is often called the *mechanical equivalent of heat,* and is usually denoted by J. The reciprocal of J, or $\frac{1}{J}$, is designated by A and equals $\frac{1}{778.57}$, or 0.00128.

From the first law of thermodynamics and the experiments cited, it is evident that *heat and work are mutually convertible in the definite ratio of* 778 *ft-lb to* 1 *Btu.*

14. Second Law of Thermodynamics.—The second law of thermodynamics is stated in different ways by various authors.

For the purpose of this text, the following statement by Clausius is selected: *It is impossible for a self-acting machine, unaided by any external agency, to convey heat from one body to another of higher temperature.* The second law is not capable of proof, but is axiomatic. All the experiments with heat engines go to show that this law is true.

It follows from the second law that no heat engine can convert more than a small fraction of the heat given to it into work. From this law the following expression for the efficiency of a heat engine is derived:

$$e_t = \frac{\text{heat added} - \text{heat rejected}}{\text{heat added}}.$$

15. Laws of Perfect Gases.—There are two laws expressing the relation of pressure, volume, and temperature in a perfect gas—the law of Boyle and the law of Charles.

Boyle's Law.—The volume of a given mass of gas varies inversely as the absolute pressure, provided the temperature remains constant.

If p_1 = the pressure, and v_1 = the volume of the initial condition of the gas, and p and v any other condition of the same gas, then

$$p_1v_1 = pv = \text{a constant.}$$

Charles' Law.—Under constant pressure equal volumes of different gases increase equally for the same increment of temperature. Also, *if the gas be heated under constant pressure, equal increments of its volume correspond to equal increments of temperature on the absolute scale.*

This law may also be stated as follows: *When a gas receives heat at a constant volume, the absolute pressure varies directly as the absolute temperature,* or *when a gas receives heat at a constant pressure, the volume varies directly as the absolute temperature.*

Let a gas receive heat at a constant volume v_1, the absolute pressure and absolute temperature varying from p_1, T_1 to p, T'. Then

$$\frac{p}{p_1} = \frac{T'}{T_1}.$$

If the gas now receives heat at this pressure p, the volume and temperature changing to v and T, then

$$\frac{v}{v_1} = \frac{T}{T'}.$$

These two laws of Boyle and Charles may be represented diagrammatically as shown in Figs. 2 and 3.

In Fig. 2, the pressure p_2 is two-thirds pressure p_1. Therefore, volume v_2 will be three-halves times volume v_1. Similarly, since p_3 is one-half p_2, v_3 will be twice v_2, and since p_4 is one-half p_3, v_4 will be twice v_3.

In Fig. 3, the pressure is assumed to be constant and, therefore, according to Charles' law the absolute temperature T_2 will be twice the absolute temperature T_1, since volume v_2 is twice volume v_1.

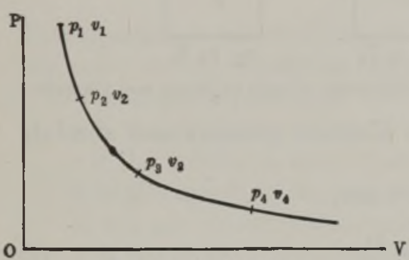


FIG. 2.—Boyle's law.

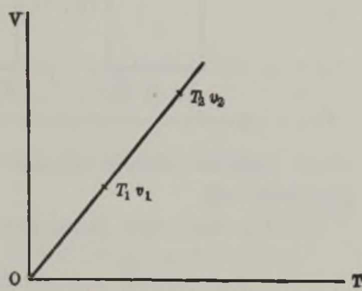


FIG. 3.—Charles' law.

Charles' law can be expressed graphically with the volume kept constant instead of the pressure. In this case the coordinates will be absolute pressure and absolute temperature in place of volume and absolute temperature as is the case in Fig. 3.

16. Equation of a Perfect Gas.—A *perfect gas* is sometimes defined as a gas that fulfills the laws of Boyle and Charles. It is probably better to define it as *a gas in which no internal work is done*, or, in other words, *a gas in which there is no friction between the molecules under change of conditions.* There is no actual gas that fulfills these conditions, but under ordinary temperatures air, nitrogen, hydrogen, and oxygen so nearly do that they may be considered as approximately perfect. These laws were first determined for air, and they hold true for all perfect gases.

The equation of a perfect gas is found by combining the laws of Boyle and Charles.

Let the original condition of absolute pressure, volume, and absolute temperature of a perfect gas be

$$p_1,\ v_1,\ \text{and}\ T_1,$$

as shown in Fig. 4.

It is desired to change the volume, pressure, and temperature of this gas to some new condition where these values are all different, as in 2, Fig. 4.

First let the gas be changed from state *1* to an intermediate state x, Fig. 4, at constant temperature, both the volume and absolute pressure changing; then, from state x to state *2* at constant volume with a change in absolute pressure and absolute temperature.

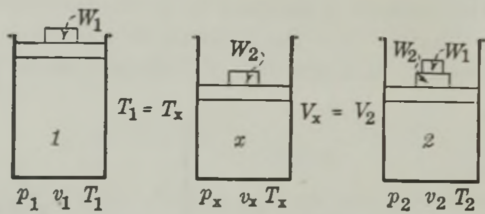


Fig. 4.—Graphical representation combination of laws of Boyle and Charles.

For the first step, from Boyle's law,

$$\frac{p_x}{p_1} = \frac{v_1}{v_x}. \tag{1}$$

For the second step, from Charles' law,

$$\frac{p_x}{p_2} = \frac{T_x}{T_2}. \tag{2}$$

But since

$$v_x = v_2 \qquad \text{and} \qquad T_x = T_1,$$

Eq. (1) becomes

$$\frac{p_x}{p_1} = \frac{v_1}{v_2},$$

or

$$p_x = \frac{p_1 v_1}{v_2}, \tag{3}$$

and Eq. (2) becomes

$$\frac{p_x}{p_2} = \frac{T_1}{T_2},$$

or

$$p_x = \frac{p_2 T_1}{T_2}. \tag{4}$$

Combining Eqs. (3) and (4),

$$\frac{p_1 v_1}{v_2} = \frac{p_2 T_1}{T_2}.$$

Hence,

$$\frac{p_1v_1}{T_1} = \frac{p_2v_2}{T_2} = \frac{pv}{T} = \text{a constant.} \tag{5}$$

Denoting this constant by R, then,

$$pv = RT, \qquad p_1v_1 = RT_1, \qquad \text{and} \qquad p_2v_2 = RT_2. \tag{6}$$

The value of R given in this equation is for 1 lb of the gas. To state this law for other than 1 lb, let w equal the weight of the gas, and the law becomes

$$pv = wRT. \tag{7}$$

This equation is called the *equation of the gas* and holds true for any point on any expansion line of any perfect gas.

In the above expressions,

p is the *absolute* pressure in pounds per square foot;
v is the volume in cubic feet;
w is the weight of the gas in pounds;
R is a gas constant (foot-pounds per pound per degree);
T is the *absolute* temperature in degrees Fahrenheit.

The value of R for any given substance may be determined, provided the volume of one pound for any given condition of pressure and temperature is known. For example, it has been found by experiment that for air under a pressure of 14.696 psia and at a temperature of 32 F, the volume of 1 lb is 12.39 cu ft. Substituting these values in Eq. (7),

$$\begin{aligned} R &= \frac{pv}{wT}, \\ &= \frac{14.696 \times 144 \times 12.39}{1 \times (32 + 459.6)} \\ &= 53.34 \text{ (compare with the value of } R \text{ for air given in Table II, page 6).} \end{aligned} \tag{8}$$

Therefore, for 1 lb of air with units taken,

$$pv = 53.34T, \tag{9}$$

or for w lb,

$$pv = 53.34wT. \tag{10}$$

This equation is always true for air at all times and under all conditions, as long as it remains a gas.

It must be kept in mind that 53.34 is the value for R for air only and varies with different gases (see Table II, page 6).

Example.—A tank contains 5 lb of air at 75 F, under a pressure of 100 psi. Find the volume of the air.

Solution.

$$pv = wRT$$
$$p = (100 + 14.7)144 = 114.7 \times 144 \text{ psfa}$$
$$T = 75 + 460 = 535 \text{ F abs.}$$

Therefore, substituting in the equation of the gas,

$$114.7 \times 144 \times v = 5 \times 53.34 \times 535$$
$$v = \frac{142{,}760}{16{,}520}$$
$$v = 8.64 \text{ cu ft.}$$

Example.—Ten pounds of air under a pressure of 50 psi occupy a volume of 10 cu ft. Find the temperature.

Solution.

$$pv = wRT$$
$$p = (50 + 14.7)144 = 64.7 \times 144 \text{ psfa.}$$

Therefore,

$$64.7 \times 144 \times 10 = 10 \times 53.34 \times T$$
$$T = \frac{93{,}200}{533.4}$$
$$T = 174.5 \text{ abs}$$
$$T = 174.5 - 460 = -285.5 \text{ F.}$$

NOTE.—The conditions of this problem could exist only in case the temperature were kept as low as −285.5 F.

17. Absorption of Heat.—When a substance receives heat, this heat may be dissipated in one or all of three ways, *viz.*, by *increasing its temperature*, by *doing internal work*, or by *doing external work.*

Let dQ denote the heat absorbed, dU the heat used in increasing the temperature, dI the heat used in doing internal work, and dW the heat equivalent of the external work done. Then

$$dQ = dU + dI + dW. \tag{11}$$

The heat equivalent of the *total work* done is represented by $dI + dW$, and $dU + dI$ represents the heat utilized in changing the *internal energy* or the energy that the gas possesses by virtue of the motions and relative positions of its molecules. From this it is seen that the

Heat absorbed = *the heat utilized in increasing the internal energy* + *the heat equivalent of the external work done.* (12)

By "internal work" is meant work done in overcoming changes in the physical state of the substance, and in overcoming the attraction of the molecules for each other, thus changing the potential energy of the body. Since no internal work is done in heating a perfect gas, the second term in Eq. (11) becomes zero and all the heat absorbed goes either to increasing the temperature or doing external work. Therefore, in a perfect gas,

$$dQ = dU + dW$$

and

$$\int_{Q_1}^{Q_2} dQ = \int_{U_1}^{U_2} dU + \int_{v_1}^{v_2} dW.$$

Integrating,

$$Q_2 - Q_1 = U_2 - U_1 + \int_{v_1}^{v_2} pdv.$$

Let

$$Q = Q_2 - Q_1, \qquad U = U_2 - U_1, \qquad \text{and} \qquad W = \int_{1}^{v_2} pdv.$$

Then

$$Q = U + W. \tag{13}$$

The A.S.M.E. recommends that the sum of the internal energy of any fluid plus its pressure volume product [$U + PV$, *not* $U + W$, which is $U + P(V_2 - V_1)$] be known as its *enthalpy*, H, (accent on second syllable). Since the absolute quantity of internal energy of a substance cannot be measured, its absolute enthalpy cannot be measured. However, this makes no difference as it is *change* of internal energy and *change* of enthalpy that are the factors used in actual practice and these changes can be measured against any assumed zero. The *zero of enthalpy* for water is arbitrarily selected as saturated liquid at 32 F; and, for ammonia and Freon -12, it is taken as saturated liquid at -40 F.

When heat is added to a perfect gas, the internal energy will undergo a change, which is equal to the difference between the total heat supplied and the heat equivalent of the work done by the gas during the change. In other words,

$$dU = dQ - dW,$$

or

$$U = Q - W. \tag{14}$$

18. Joule's Law.—*When a perfect gas expands without doing external work and without taking in or giving out any heat, its temperature remains unchanged and there is no change in its internal energy.* This law was established by the following experiment performed by Joule:

Two vessels a and b (Fig. 5) connected by a tube containing a stopcock c were placed in a water bath. One vessel contained air compressed to a pressure of 22 atmospheres, while a vacuum was established and maintained in the other. After the vessels

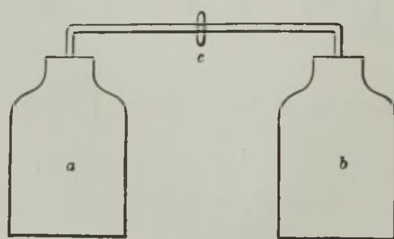


FIG. 5.—Joule's apparatus.

had remained in the bath long enough so that the air and water were at the same temperature and there could, therefore, be no further flow of heat from one to the other, the stopcock c was opened and the air allowed to flow from one vessel to the other until the pressure in each was 11 atmospheres. The temperatures of the air and water were then read again and found to be unchanged. From the conditions of the apparatus no work external to the two vessels could have been done. As the gas had done no work and had neither gained nor lost any heat, its internal energy must have remained unchanged. Although the pressure and the volume of the gas had changed, the temperature had not, thus proving that a change in internal energy depends upon a change in temperature only.

19. Relation of Specific Heats.—If w lb of a perfect gas are heated at a *constant pressure* from a temperature T_1 to a temperature T_2, and the volume is changed from a volume v_1 to a volume

v_2, the heat absorbed,

$$Q = wc_p(T_2 - T_1), \text{ in Btu} \tag{15}$$

and the work done in foot-pounds is

$$W = \int p\,dv.$$

Integrating between limits,

$$W = \int_{v_1}^{v_2} p\,dv = p\int_{v_1}^{v_2} dv = p(v_2 - v_1). \tag{16}$$

From the equation of a perfect gas

$$pv_2 = wRT_2, \qquad \text{and} \qquad pv_1 = wRT_1.$$

Substituting these values in Eq. (16),

$$W = wR(T_2 - T_1), \text{ in ft-lb} \tag{17}$$

$$= w\frac{R}{J}(T_2 - T_1), \text{ in Btu.} \tag{18}$$

Since from Eq. (13), $U = Q - W$, then the difference between Eqs. (15) and (18) would be the heat which goes to increasing the temperature, or

$$U = w\left(c_p - \frac{R}{J}\right)(T_2 - T_1). \tag{19}$$

If the gas is heated at a *constant volume* from a temperature T_1 to a temperature T_2, then the total heat added,

$$Q = wc_v(T_2 - T_1), \text{ in Btu} \tag{20}$$

and, since no external work is done, $W = 0$ and this heat all goes to increasing the temperature, or

$$wc_v(T_2 - T_1) = U. \tag{21}$$

As both Eq. (19) and Eq. (21) represent the heat going to increase the temperature, or U, these equations are equal to each other and

$$w\left(c_p - \frac{R}{J}\right)(T_2 - T_1) = wc_v(T_2 - T_1).$$

Therefore,

$$c_p - \frac{R}{J} = c_v, \tag{22}$$

$$c_p - c_v = \frac{R}{J}, \tag{23}$$

or

$$R = J(c_p - c_v). \tag{24}$$

Since from Eq. (23)

$$c_p - c_v = \frac{R}{J},$$

then, dividing both sides by c_v,

$$\frac{c_p}{c_v} - 1 = \frac{R}{Jc_v},$$

or, denoting the ratio of the two specific heats, *i.e.*, $\frac{c_p}{c_v}$, by κ,

$$\kappa - 1 = \frac{R}{Jc_v}, \tag{25}$$

and hence

$$c_v = \frac{R}{J(\kappa - 1)}. \tag{26}$$

Since, $\frac{c_p}{c_v} = \kappa$, then $c_v\kappa = c_p$, and multiplying Eq. (26) through by κ,

$$c_p = \frac{R\kappa}{J(\kappa - 1)}. \tag{27}$$

If, in Eq. (17), w is taken as one pound of gas, and $(T_2 - T_1)$ as 1 degree, then,

$$W = R$$

or,

R is the number of foot-pounds of external work done when one pound of gas is heated one degree Fahrenheit at constant pressure.

For air,

$$\kappa = \frac{c_p}{c_v} = \frac{0.2410}{0.1725} = 1.397, \tag{28}$$

and, from Eq. (24),

$$R = 778.57(0.2410 - 0.1725) = 53.33 \text{ [compare Eq. (8)]}.$$

20. Expansions in General.—When air, steam, or any other gas is used as the working substance in an engine, the gas is allowed to expand, doing work for a portion of the working stroke. The variation in pressure and volume during this expansion may be represented graphically by a mathematical curve on the pressure-volume plane. The same is true in the compression of these gases. On this plane the ordinates of any curve represent pressures and the abscissas represent volumes.

Almost all the expansion or compression curves ordinarily occurring in steam or gas engines, or in the various forms of compressors, can be represented by the equation

$$pv^n = \text{a constant.} \qquad . \quad (29)$$

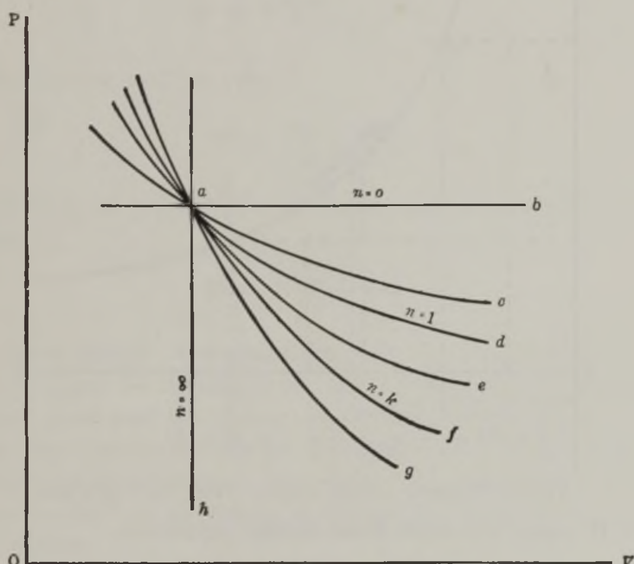


Fig. 6.—Paths of an expanding gas.

This is called the *equation of the path of the gas* and must be distinguished from the *equation of the gas* as given by Eq. (7), page 15.

During expansion, or compression, n in Eq. (29) may have any value between zero and infinity, but is constant for any given curve. Figure 6 shows how the path of a gas will vary during expansion depending upon the value of n.

21. Work of Expansion.—The curve ab in Fig. 7 represents graphically the relation between pressure and volume during expansion. Let the equation of this curve be

$$pv^n = \text{a constant.}$$

The gas expands from a point a, where the pressure is p_1 and the volume v_1, to the point b, where the pressure is p_2 and the volume v_2. The area $abcd$ represents the work done during this expansion.

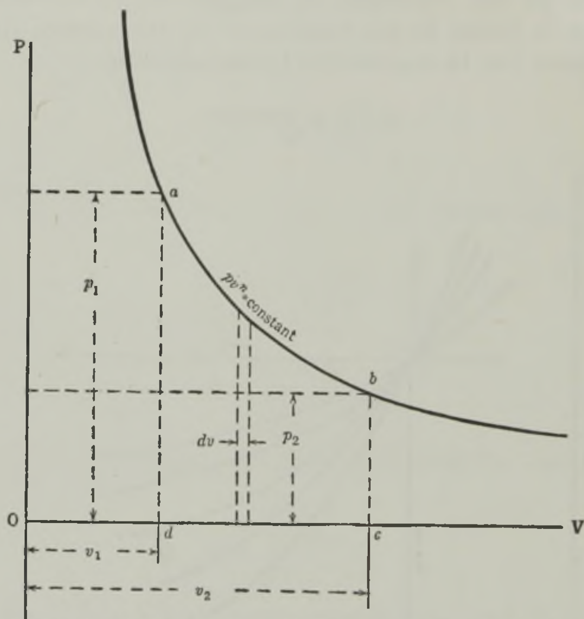


FIG. 7.—Pressure-volume diagram of an expanding gas.

Let W equal the work done during expansion.
Then

$$W = \int_{v_1}^{v_2} pdv. \tag{30}$$

Since every point in the curve must fulfill the original conditions for tbe equation of the curve,

$$pv^n = p_1v_1^n = p_2v_2^n.$$

Hence

$$p = \frac{p_1v_1^n}{v^n}. \tag{31}$$

Substituting this expression in Eq. (30),

$$W = \int_{v_1}^{v_2} \frac{p_1 v_1^n dv}{v^n}$$

or, removing the constant from under the radical sign,

$$W = p_1 v_1^n \int_{v_1}^{v_2} \frac{dv}{v^n}. \tag{32}$$

Integrating,

$$W = p_1 v_1^n \frac{(v_2^{1-n} - v_1^{1-n})}{1 - n}. \tag{33}$$

Multiplying out the parenthesis,

$$W = \frac{p_1 v_1^n v_2^{1-n} - p_1 v_1}{1 - n},$$

or, substituting $p_2 v_2^n$ for $p_1 v_1^n$,

$$W = \frac{p_2 v_2 - p_1 v_1}{1 - n}. \tag{34}$$

Substituting for pv its value in terms of w, R, and T, Eq. (34) becomes

$$W = \frac{wR(T_2 - T_1)}{1 - n}. \tag{35}$$

22. Heat Added—General Case.—In any expansion, the heat added is equal to the algebraic sum of the heat equivalent of the work done and the change in internal energy [see Eq. (12)]. As has been previously shown, the change in internal energy of a perfect gas depends upon the change in temperature only and is equal to the heat necessary to change the temperature at constant volume.

Therefore, in a perfect gas,

$$Q = U + W,$$

$$= wJc_v(T_2 - T_1) + \frac{p_2 v_2 - p_1 v_1}{1 - n}, \tag{36}$$

$$= \frac{wR}{\kappa - 1}(T_2 - T_1) + \frac{p_2 v_2 - p_1 v_1}{1 - n}, \tag{37}$$

$$= \frac{p_2 v_2 - p_1 v_1}{\kappa - 1} + \frac{p_2 v_2 - p_1 v_1}{1 - n}. \tag{38}$$

This result will be expressed in foot-pounds, since p_1 and p_2 are expressed in pounds per square foot, and v_1 and v_2 are expressed in cubic feet. To find the equivalent Btu, divide Eq. (38) by 778.

Equation (37) may also be changed to read as follows:

$$Q = \frac{wR(T_2 - T_1)}{\kappa - 1} + \frac{wR(T_2 - T_1)}{1 - n}. \tag{39}$$

In Eqs. (38) and (39), p_1, v_1, and T_1 refer to the original state of the gas, and p_2, v_2, and T_2 to the final state.

Example.—Five cubic feet of air under a pressure of 75 psi are expanded to a pressure of 25 psi along a curve, the equation of which is $pv^{1.2}$ = a constant.

(*a*) Find the final volume of the air. (*b*) Find the work in foot-pounds done during the expansion. (*c*) Find the heat in Btu, added during the expansion.

Solution.—(*a*) From Eq. (29)

$$p_1v_1^n = p_2v_2^n,$$

or

$$v_2^n = \frac{p_1}{p_2}v_1^n.$$

Therefore,

$$\begin{aligned}
v_2^{1.2} &= \frac{(75 + 14.7) \times 144}{(25 + 14.7) \times 144} \times 5^{1.2} \\
&= 2.26 \times 5^{1.2} \\
1.2 \log v_2 &= \log 2.26 + 1.2 \log 5 \\
&= 0.354 + 1.2 \times 0.699 = 0.354 + 0.839 \\
&= 1.193 \\
\log v_2 &= 0.994 \\
v_2 &= 9.86 \text{ cu ft.}
\end{aligned}$$

(*b*) From Eq. (34)

$$\begin{aligned}
W &= \frac{p_2v_2 - p_1v_1}{1 - n} \\
&= \frac{144(39.7 \times 9.86 - 89.7 \times 5)}{1 - 1.2} \\
&= \frac{144(391.44 - 448.50)}{-0.2} = \frac{144(-57.06)}{-0.2} = \frac{-8216.6}{-0.2} \\
&= 41{,}083 \text{ ft-lb.}
\end{aligned}$$

(*c*) From Eq. (38)

$$\begin{aligned}
Q &= \frac{p_2v_2 - p_1v_1}{\kappa - 1} + \frac{p_2v_2 - p_1v_1}{1 - n} \\
&= \frac{144(39.7 \times 9.86 - 89.7 \times 5)}{(1.4 - 1)} + \frac{144(39.7 \times 9.86 - 89.7 \times 5)}{1 - 1.2}
\end{aligned}$$

$$= \frac{144(-57.06)}{0.4} + \frac{144(-57.06)}{-0.2}$$
$$= -20{,}541 + 41{,}083 = 20{,}542 \text{ ft-lb}$$
$$= 26.4 \text{ Btu.}$$

23. Expressions for Heat Added at Constant Volume and at Constant Pressure.—Since, when the gas is heated at constant

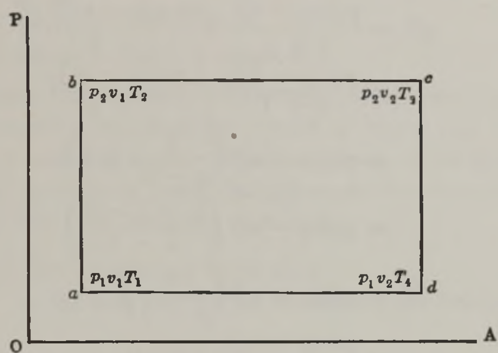


FIG. 8.—Pressure-volume diagram when heat is added to a gas at constant volume and when added at constant pressure.

volume, as from a to b in Fig. 8, no external work is done, W in the general equation $Q = U + W$ becomes zero and

$$Q = U.$$

Equation (38) then becomes

$$Q = \frac{p_2v_2 - p_1v_1}{\kappa - 1};$$

but since by hypothesis the volume remains constant, $v_2 = v_1$, and

$$_aQ_b = \frac{p_2v_1 - p_1v_1}{\kappa - 1},$$
$$= \frac{v_1(p_2 - p_1)}{\kappa - 1}, \text{ in ft-lb,} \quad (40)$$
$$= \frac{v_1(p_2 - p_1)}{778(\kappa - 1)}, \text{ in Btu,} \quad (41)$$

where $_aQ_b$ represents the heat added along the constant-volume line ab.

If the gas is heated at constant pressure, as from b to c (Fig. 8),

$$p_b = p_c,$$

that is, p_2 remains unchanged and Eq. (38) becomes

$$_bQ_c = \frac{p_2v_2 - p_2v_1}{\kappa - 1} + \frac{p_2v_2 - p_2v_1}{1 - n}. \tag{42}$$

Since, when a gas expands along a constant-pressure line $n = 0$, Eq. (42) becomes

$$\begin{aligned}
_bQ_c &= \frac{p_2(v_2 - v_1)}{\kappa - 1} + \frac{p_2(v_2 - v_1)}{1 - 0}, \\
&= \frac{p_2(v_2 - v_1)}{\kappa - 1} + \frac{p_2(v_2 - v_1)}{1}, \\
&= p_2(v_2 - v_1)\left(\frac{1}{\kappa - 1} + 1\right), \\
&= p_2(v_2 - v_1)\left(\frac{1 + \kappa - 1}{\kappa - 1}\right), \\
&= p_2(v_2 - v_1)\left(\frac{\kappa}{\kappa - 1}\right), \\
&= \frac{p_2(v_2 - v_1)\kappa}{\kappa - 1}, \text{ in ft-lb}, && (43) \\
&= \frac{p_2(v_2 - v_1)\kappa}{778(\kappa - 1)}, \text{ in Btu}. && (44)
\end{aligned}$$

Example.—Suppose that, in Fig. 8, $p_1 = 15$ psia, $p_2 = 75$ psia, $v_1 = 5$ cu ft, and $v_2 = 25$ cu ft. (*a*) Find the heat added in Btu. (*b*) Find the heat rejected in Btu.

Solution.—(*a*) Heat added $= Q_1 = {_aQ_b} + {_bQ_c}$.

From Eq. (41)

$$\begin{aligned}
_aQ_b &= \frac{v_1(p_2 - p_1)}{(\kappa - 1) \times 778} \\
&= \frac{5 \times 144(75 - 15)}{(1.4 - 1) \times 778} = \frac{5 \times 144 \times 60}{0.4 \times 778} = \frac{43{,}200}{311.2} \\
&= 138.81 \text{ Btu}.
\end{aligned}$$

From Eq. (44)

$$\begin{aligned}
_bQ_c &= \frac{p_2(v_2 - v_1)\kappa}{(\kappa - 1)778} \\
&= \frac{75 \times 144(25 - 5) \times 1.4}{(1.4 - 1) \times 778} = \frac{75 \times 144 \times 20 \times 1.4}{0.4 \times 778} \\
&= \frac{302{,}400}{311.2} = 971.72 \text{ Btu}.
\end{aligned}$$

$$Q_1 = 138.81 + 971.72 = 1{,}110.53 \text{ Btu}.$$

(*b*) Heat rejected $= Q_2 = {_cQ_d} + {_dQ_a}$.

$$_cQ_d = \frac{v_2(p_2 - p_1)}{(\kappa - 1)778}$$

$$= \frac{25 \times 144(75 - 15)}{(1.4 - 1) \times 778} = \frac{25 \times 144 \times 60}{0.4 \times 778} = \frac{216,000}{311.2}$$
$$= 694.08 \text{ Btu.}$$
$$Q_a = \frac{p_1(v_2 - v_1)\kappa}{(\kappa - 1) \times 778}$$
$$= \frac{15 \times 144(25 - 5) \times 1.4}{(1.4 - 1) \times 778} = \frac{15 \times 144 \times 20 \times 1.4}{0.4 \times 778}$$
$$= \frac{60,480}{311.2} = 194.34 \text{ Btu.}$$
$$Q_2 = 694.08 + 194.34 = 888.42 \text{ Btu.}$$

24. Adiabatic Expansion.—*Adiabatic expansion is one in which the expanding gas does not receive or reject any heat except in the form of external work.* That is, there is no radiation or conduction of heat to or from the gas and, therefore, the "heat absorbed" in Eq. (12) becomes zero and

The heat equivalent of external work done =
— the heat utilized in increasing the internal energy,

or

$$W = -U.$$

In other words, the external work done is done at the expense of the internal energy in the gas. That is, all the heat lost due to a change in temperature goes to doing work. If compressed adiabatically, the work done on the gas goes to increasing its internal energy. (It must be understood that the negative sign before U does not mean negative work, but does mean a decrease in internal energy.)

Since any change in the internal energy of a gas depends upon a change in temperature, it is impossible to have an increase in the internal energy without an increase in temperature, or a decrease in internal energy without a decrease in temperature.

Adiabatic expansion could only be produced in a cylinder made of a perfectly nonconducting material with the working fluid itself undergoing no chemical change. In actual engines, or compressors, this is never the case, and adiabatic expansion is only approximated.

Since Eq. (35),

$$W = \frac{wR(T_2 - T_1)}{1 - n},$$

is the expression for the work done during any expansion, it is now necessary to find the value of n for adiabatic expansion.

In Art. 19 it was shown that the change in internal energy due to a change of temperature equals

$$wc_v(T_2 - T_1),\text{ in Btu,}$$

or

$$U = wJc_v(T_2 - T_1),\text{ in ft-lb.}$$

Since, as already shown, in adiabatic expansion

$$W = -U,$$

then

$$W = wJc_v(T_1 - T_2). \tag{45}$$

But, from Eq. (26),

$$Jc_v = \frac{R}{\kappa - 1}$$

and hence, substituting this value in Eq. (45), and changing signs

$$W = \frac{wR(T_2 - T_1)}{1 - \kappa}, \tag{46}$$

$$= \frac{p_2v_2 - p_1v_1}{1 - \kappa}. \tag{47}$$

Comparing Eqs. (35) and (46), both of which express the value for work done in adiabatic expansion, $n = \kappa$. Therefore, the equation for adiabatic expansion is

$$pv^\kappa = p_1v_1^\kappa = p_2v_2^\kappa = \text{a constant.} \tag{48}$$

Example.—Five cubic feet of air under a pressure of 75 psi are expanded adiabatically until the pressure is 25 psi.

(*a*) Find the final volume of the air. (*b*) Find the work done during the expansion. (Compare example, Art. 22.)

Solution.—(*a*) From Eq. (48)

$$p_1v_1^\kappa = p_2v_2^\kappa.$$

or

$$v_2^\kappa = \frac{p_1}{p_2}v_1^\kappa.$$

Therefore,

$$v_2^{1.4} = \frac{(75 + 14.7) \times 144 \times 5^{1.4}}{(25 + 14.7) \times 144} = 2.26 \times 5^{1.4}.$$

$$\begin{aligned}
1.4 \log v_2 &= \log 2.26 + 1.4 \log 5 \\
&= 0.354 + 1.4 \times 0.699 = 0.354 + 0.979. \\
&= 1.333 \\
\log v_2 &= 0.952 \\
v_2 &= 8.95.
\end{aligned}$$

(*b*) From Eq. (47)

$$
\begin{aligned}
W &= \frac{p_2v_2 - p_1v_1}{1 - \kappa} \\
&= \frac{144(39.7 \times 8.95 - 89.7 \times 5)}{1 - 1.4} \\
&= \frac{144 \times (-93.18)}{-0.4} = \frac{-13{,}418}{-0.4} \\
&= 33{,}545 \text{ ft-lb.}
\end{aligned}
$$

25. Isothermal Expansion.—*A gas expands or contracts isothermally when its temperature remains constant during a change of volume.* Since the temperature remains constant during isothermal expansion, no heat is absorbed in increasing the temperature, and, in a perfect gas, U in Eq. (13) becomes zero, and Q equals W, or all the heat absorbed during isothermal expansion of a perfect gas goes to doing external work.

As in isothermal expansion, there is no change in temperature, Eq. (6) becomes

$$pv = \text{a constant} \tag{49}$$

(which is the equation of a rectangular hyperbola). Equation (49) is of the same form as Eq. (29), and the exponent n is in this case equal to 1. By substituting 1 for the value of n in Eq. (34), or Eq. (35), an indeterminate expression is derived. Therefore, in order to derive an expression for the work done in isothermal expansion, it is necessary to proceed differently.

Assume the curve ab (Fig. 7) to be an isothermal curve, or a rectangular hyperbola. The work done by the gas in expanding isothermally from volume v_1, represented at the point a, to the volume v_2, represented at the point b, is the area $abcd$; or

$$W = \int_{v_1}^{v_2} pdv. \tag{50}$$

To integrate this expression, the pressure must be expressed in terms of volume. From Eq. (49)

$$pv = p_1v_1 = p_2v_2,$$

hence,

$$p = \frac{p_1v_1}{v}. \tag{51}$$

Substituting Eq. (51) in Eq. (50), and removing the constant from under the radical sign,

$$W = p_1v_1 \int_{v_1}^{v_2} \frac{dv}{v}.$$

Integrating,

$$W = p_1v_1(\log_e v_2 - \log_e v_1);$$

hence,

$$W = p_1v_1 \log_e \frac{v_2}{v_1}. \tag{52}$$

Since

$$p_1v_1 = wRT,$$

then

$$W = wRT \log_e \frac{v_2}{v_1}, \tag{53}$$

but $\frac{v_2}{v_1} = r$, the ratio of expansion, and $wRT = pv$;
hence,

$$W = wRT \log_e r \tag{54}$$
$$= pv \log_e r. \tag{55}$$

During isothermal expansion there is no change in the internal energy, since the temperature remains constant. Hence the gas takes in, during isothermal expansion, an amount of heat equivalent to the work done during the expansion. Equations (54) and (55), then, *represent not only the work done, but the equivalent amount of heat taken in or rejected during isothermal expansion or compression.*

In actual practice, when gas is suddenly compressed, the compression curve is approximately an adiabatic, and when slowly compressed may be approximately isothermal.

Example.—If in the example given in Art. 22, the air expands isothermally, find (*a*) the final volume of the air; (*b*) the work done in foot-pounds; (*c*) the heat added in Btu.

Solution.—(*a*) From Eq. (49)

$$p_1v_1 = p_2v_2$$

or

$$v_2 = \frac{p_1v_1}{p_2}.$$

Therefore,

$$v_2 = \frac{89.7 \times 144}{39.7 \times 144} \times 5 = 2.26 \times 5$$
$$v_2 = 11.30 \text{ cu ft.}$$

(b) From Eq. (55)

$$W = p_1 v_1 \log_e r,$$

but

$$r = \frac{v_2}{v_1} = \frac{11.30}{5} = 2.26.$$

Therefore,

$$\begin{aligned} W &= 89.7 \times 144 \times 5 \log_e 2.26 \\ &= 89.7 \times 144 \times 5 \times 2.3^* \times 0.354 \\ &= 52{,}600 \text{ ft-lb.} \end{aligned}$$

(c) Heat added $= \dfrac{52{,}600}{778}$

$= 67.6$ Btu.

26. Values of n.—As indicated in the preceding discussions, the value of n in the equation $pv^n =$ a constant will determine whether heat is added, rejected, or remains constant, and whether the temperature rises, falls, or does not change during any given expansion or compression of a gas. This value may vary from zero to infinity.

For example, if $n = 0$, the equation becomes

$$p = \text{a constant.}$$

From Charles' law, it is known that, the pressure remaining constant, heat must be added and the temperature will rise as the volume increases.

If, on the other hand, $n = \infty$, the equation becomes

$$v = \text{a constant.}$$

Here, again, from Charles' law it is known that, the volume remaining constant, heat will be added and the temperature rise as the pressure increases, or heat will be rejected and the temperature fall as the pressure decreases.

These varying changes in heat and in temperature as determined by the value of n during the expansion of a perfect gas are clearly shown in Table V, and the corresponding changes in pressure and volume appear in Fig. 6.

During isothermal expansion (line ad, Fig. 6), where, as proved in Art. 25, $n = 1$, there is, by definition, no change in tempera-

* The Napierian logarithm of a number equals 2.3026 times the common logarithm of that number or

$$\log_e n = 2.3 \times \log_{10} n.$$

ture, and $H = W$. In other words, heat will be added as the gas expands doing external work, and the heat added will just equal the work done.

For any path lying between *ad* and *af*, Fig. 6, heat is added to the system and yet the temperature falls, since the heat supplied is not sufficient to keep the temperature constant. In this case the heat equivalent of the work done is greater than the heat supplied, the difference coming from the decrease in internal energy which accompanies the fall in temperature.

Finally, it was proved in Art. 24 that during adiabatic expansion (line *af*, Fig 6), the value of n equals κ. By definition, no heat is added or rejected during this expansion, but all the work done is done at the expense of the internal energy, and there is, consequently, a fall in the temperature of the gas (see Art. 18).

In case the gas is being compressed instead of expanding, the changes in heat and in temperature will be just the opposite of those shown in the table.

TABLE V.—HEAT AND TEMPERATURE CHANGES DEPENDENT UPON VALUE OF n DURING EXPANSION

Value of n	Equation of path of gas	Path as shown in Fig. 6, page 21	Heat	Temperature
$n = 0$	p = constant	*ab*	Added	Rises
$n > 0$ and < 1	pv^n = constant	*ac*	Added	Rises
$n = 1$	pv = constant	*ad*	Added	Constant
$n > 1$ and $< \kappa$	pv^n = constant	*ae*	Added	Falls
$n = \kappa$	pv^κ = constant	*af*	Constant[1]	Falls
$n > \kappa$ and $< \infty$	pv^n = constant	*ag*	Rejected	Falls
$n = \infty$	v = constant	*ah*	Rejected	Falls

[1] Neither added nor rejected except in form of external work.

27. Relation between p, v, and T during Expansion or Compression.—Since the equation of the path of a gas during expansion or compression is

$$pv^n = \text{a constant [see Eq. (29)]},$$

then

$$p_1v_1^n = p_2v_2^n, \tag{56}$$

$$\frac{p_1}{p_2} = \left(\frac{v_2}{v_1}\right)^n, \tag{57}$$

and

$$\frac{v_2}{v_1} = \left(\frac{p_1}{p_2}\right)^{\frac{1}{n}}. \tag{58}$$

From the equation of a perfect gas,

$$\frac{p_1v_1}{T_1} = \frac{p_2v_2}{T_2},$$

or

$$\frac{T_1}{T_2} = \frac{p_1v_1}{p_2v_2}. \tag{59}$$

Substituting in Eq. (59), the value of $\frac{p_1}{p_2}$ as shown in Eq. (57),

$$\frac{T_1}{T_2} = \frac{v_2^n v_1}{v_1^n v_2}.$$

Hence,

$$\frac{T_1}{T_2} = \left(\frac{v_2}{v_1}\right)^{n-1}. \tag{60}$$

Substituting in Eq. (60) the value of $\frac{v_2}{v_1}$ in terms of p_1 and p_2 from Eq. (58),

$$\frac{T_1}{T_2} = \left(\frac{p_1}{p_2}\right)^{\frac{n-1}{n}}. \tag{61}$$

Equations (57), (58), (60), and (61) give the relations between pressure, volume, and temperature in any expansion or compression.

In adiabatic expansion or compression, the value of n in these equations becomes κ (see Art. 24).

Example.—Five cubic feet of air under a pressure of 75 psi and at 60 F are expanded adiabatically until the pressure is 25 psi (compare example, Art. 22). Find the temperature at the end of expansion.

Solution.—Since the expansion is adiabatic, Eq. (61) becomes

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\kappa-1}{\kappa}}.$$

$$T_2 = T_1\left(\frac{p_2}{p_1}\right)^{\frac{\kappa-1}{\kappa}}$$

$$= (60 + 460)\left(\frac{39.7 \times 144}{89.7 \times 144}\right)^{\frac{1.4-1}{1.4}} = 520 \times 0.442^{0.286}.$$

$$\log T_2 = \log 520 + 0.286 \log 0.442$$
$$= 2.716 + 0.286 \times \bar{1}.645 = 2.716 - 0.102$$
$$= 2.614.$$
$$T_2 = 411 \text{ abs}$$
$$= 411 - 460 = -49 \text{ F}.$$

28. Entropy.—In solving many thermodynamic problems there is recourse to a ratio or abstract quantity called *entropy* (accent on first syllable). This is not a physical property and, consequently, it is difficult to give a comprehensive definition of it. For the purposes of this book, it must suffice to say that *for any reversible operation an infinitesimal change of entropy is equal to an infinitesimal change in the quantity of heat added to a body divided by the absolute temperature at which the change in heat occurs.* In other words, if dQ is a small amount of heat added to a body and T is the *absolute* temperature at which this addition takes place, then the change in entropy, ds, is $\frac{dQ}{T}$, or

$$ds = \frac{dQ}{T}, \tag{62}$$

and

$$s = \int_{T_1}^{T_2} \frac{dQ}{T}. \tag{63}$$

It should be noted that s in Eq. (63) is *change in entropy* rather than absolute entropy, because it is not possible to compute absolute entropy, since it is impossible to determine the quantity of heat necessary to bring a substance from a temperature of absolute zero to a given state, as the specific heats of substances at very low temperature are unknown. However, any arbitrary temperature may be taken above which entropy is calculated and it is customary to refer to a change in entropy as so many units of entropy. For water the temperature taken is 32 F (see Art. 42).

If the equation

$$ds = \frac{dQ}{T}$$

is rewritten in the form

$$dQ = Tds,$$

then

$$Q = \int_{s_1}^{s_2} Tds, \tag{64}$$

and the result is an expression for heat in terms of absolute temperature and the change of entropy. This latter quantity may be calculated, or obtained from tables or charts.

In adiabatic expansion no heat is added or rejected, and Eq. (62) becomes

$$ds = 0, \tag{65}$$

or, in other words, there is no change in entropy during adiabatic expansion. Conversely, *entropy is that property of a substance which changes when heat is added to or subtracted from it.*

Observation of the equation

$$ds = \frac{dQ}{T}$$

shows that a change in entropy is directly proportional to the change in heat content and inversely proportional to the absolute temperature at which this change takes place.

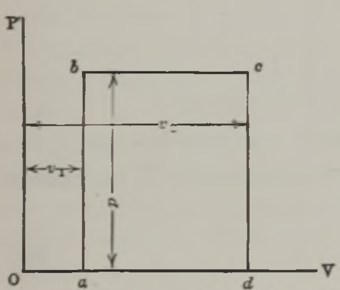


FIG. 9.—Pressure-volume diagram.

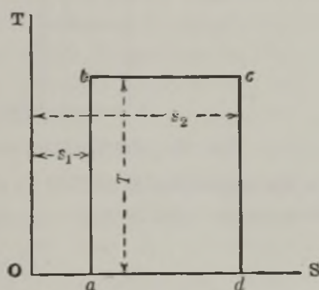


FIG. 10.—Temperature-entropy diagram.

When heat is added to a gas at constant pressure, the work done equals the pressure times the change in volume, or

$$W = p(v_2 - v_1). \tag{66}$$

This may be expressed graphically by Fig. 9.

Here the area *abcd* represents the work done during a change in volume, $v_2 - v_1$, the pressure p remaining constant. If the pressure is changed, as from a to b, the volume remaining constant at v_1, no work is done, as there is no area measured above the axis OV.

Figure 10 is similar in shape to Fig. 9 but has different coordinates. As in Fig. 9 the area *abcd* represents the *work done,* so in Fig. 10 let the area *abcd* represent *heat added,* the ordinates representing absolute temperatures, and the abscissas, a quantity which changes in proportion to the heat added to or subtracted from the gas. By previous definition this property is called *entropy.*

From Fig. 10 it is seen that if there is a change in temperature without the addition or rejection of heat, as along the line *ab*, there is no change in entropy. This is the condition in adiabatic expansion or compression. On the other hand, if heat equal

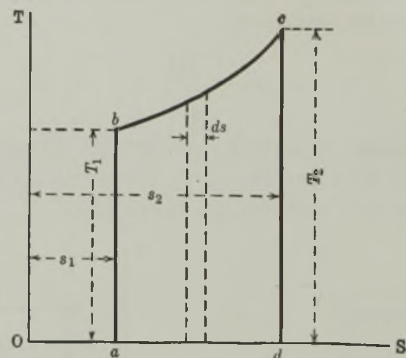


FIG. 11.—Temperature-entropy diagram of an expanding gas.

to the area *abcd* is added at a constant temperature T, the change in entropy will be $s_2 - s_1$, or, in other words,

$$Q = T(s_2 - s_1). \tag{67}$$

If, in the pressure-volume diagram, both the pressure and the volume are changing as in Fig. 7, the expression for the work done is

$$W = \int_{v_1}^{v_2} pdv. \tag{68}$$

Similarly, if, in the temperature-entropy diagram, the temperature is changed during the addition of heat, then

$$Q = \int_{s_1}^{s_2} Tds \text{ [compare Eq. (64)].} \tag{69}$$

This is shown graphically in Fig. 11.

29. Heat Engine.—Any device used to convert heat into work is called a *heat engine.* The ideally perfect heat engine would convert all the heat which it receives into useful work, but this can never be the case, since as stated under the second law of thermodynamics, not all the heat which is received by the engine can be converted into useful work. In fact, a major portion of it is rejected. The ratio of the useful work done to the heat received is called the *thermal efficiency* of the engine, or

$$\text{Thermal efficiency} = \frac{\text{heat equivalent of the work done}}{\text{the heat taken in by the engine}}. \quad (70)$$

In every heat engine there must be a working medium for transferring the heat. The working substance may be solid, liquid, or gaseous. In all the commercial heat engines now in use, the working substance is a gas. In the theoretical engine the working substance is supposed to go through a cycle of changes, returning to its original condition at the end of the cycle. Each working cycle involves (1) the taking in of heat by the working substance; (2) the doing of work by the working substance; and (3) the rejection of heat by the working substance. For example, take a condensing steam plant, including the boiler. Water is fed into the boiler from the hot well of the condenser. In the boiler the water receives heat from the coal and is transformed into steam. The steam carries the heat to the engine, part of which heat is used in doing useful work, the balance being lost when the steam is condensed in the condenser. The condensed steam is discharged into the hot well and the cycle is completed. In this cycle of operations the following equation must hold good:

$$\text{Heat taken in} - \text{heat rejected} = \text{heat equivalent of work done.} \quad (71)$$

30. Carnot Cycle.—The most efficient means for converting heat into work for any given difference in the temperatures of the *heat taken in* and *heat rejected* was first described by the French engineer, Sadi Carnot, in 1824.

In the cycle as described by him, the gas first expands isothermally, as from A to B in Fig. 12, then expands adiabatically from B to C, is then compressed isothermally from C to D, and is finally compressed adiabatically from D to A.

To understand more clearly the action of an engine working in this cycle, imagine a hot body H (Fig. 14) as an infinite source of heat at the temperature T_1; a cold body C at a temperature T_2 which is lower than T_1, and with an infinite capacity for absorbing heat without any change in temperature; a nonconducting cover N; and a cylinder covered, except on the outer end,

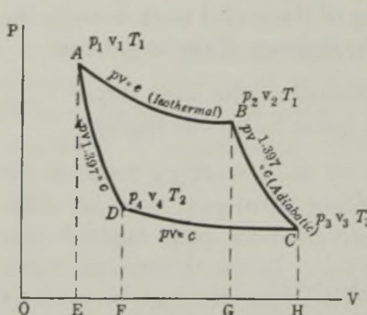


FIG. 12.—Carnot cycle—pressure-volume plane.

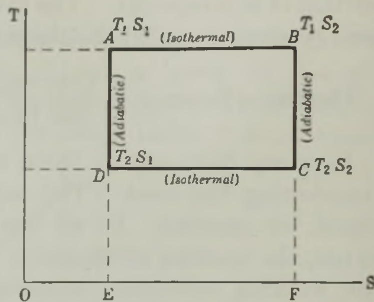


FIG. 13.—Carnot cycle—temperature-entropy plane.

with a perfectly nonconducting material and containing a nonconducting, frictionless piston. The outer end of the cylinder is assumed to be a perfect conductor.

The cylinder containing v_1 cu ft of a perfect gas under a pressure p_1 is first placed so that the conducting end is in contact with the hot body H and the gas allowed to expand to a volume

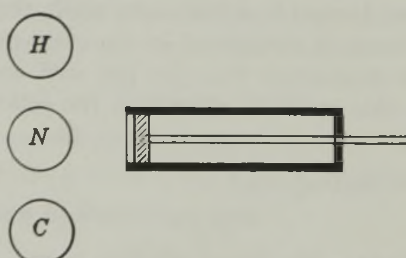


FIG. 14.—Engine working in Carnot cycle.

v_2 and pressure p_2. Since the supply of heat is infinite, the temperature will remain constant and the expansion will be isothermal. The cylinder is next placed against the nonconducting cover N, and the gas allowed to expand adiabatically to a volume v_3 and pressure p_3. At the same time the temperature falls to T_2. Then the cold body C is placed in contact with the cylinder head, and the gas is compressed, rejecting heat to C, the temperature of which remains constant at T_2 so that the compression is isothermal. Finally, the nonconducting cover is again placed on the cylinder head and the gas compressed adiabatically to the original conditions of pressure, volume, and temperature. The third step, or isothermal compression, is carried to such a point D (Fig. 12) that the adiabatic through D will pass through A.

The heat absorbed along the isothermal AB (Fig. 12) is equal to $\frac{p_1v_1}{J}\log_e\frac{v_2}{v_1}$, and the heat rejected along CD is equal to $\frac{p_3v_3}{J}\log_e\frac{v_3}{v_4}$. As BC and DA are adiabatics, there will be no heat received or rejected along these lines, and all the heat will be received along AB and all the heat rejected along CD.

Since the temperature along AB is T_1, and along CD is T_2, then,

$$\frac{T_1}{T_2} = \left(\frac{v_3}{v_2}\right)^{\kappa-1} = \left(\frac{v_4}{v_1}\right)^{\kappa-1}.$$

Hence,

$$\frac{v_3}{v_4} = \frac{v_2}{v_1}.$$

Let Q_1 equal the heat added, and Q_2 the heat rejected, and let $\frac{W}{J}$ equal the heat equivalent of the work done. Then, from Eq. (71),

$$\frac{W}{J} = Q_1 - Q_2,$$

and the thermal efficiency is

$$e_t = \frac{W}{JH_1} = \frac{Q_1 - Q_2}{Q_1}. \tag{72}$$

Substituting in this expression the expression for the heat absorbed and the heat rejected as given above, the result is the efficiency

$$e_t = \frac{\frac{p_1v_1}{J}\log_e\frac{v_2}{v_1} - \frac{p_3v_3}{J}\log_e\frac{v_3}{v_4}}{\frac{p_1v_1}{J}\log_e\frac{v_2}{v_1}}.$$

Substituting for $\frac{v_3}{v_4}$ its value in terms of v_2 and v_1, and simplifying the expression,

$$e_t = \frac{p_1v_1 - p_3v_3}{p_1v_1}. \tag{73}$$

From the equation of a perfect gas,

$$p_1v_1 = wRT_1, \qquad \text{and} \qquad p_3v_3 = wRT_2.$$

Substituting in Eq. (73),

$$e_t = \frac{T_1 - T_2}{T_1}. \tag{74}$$

Figure 13 shows the Carnot cycle using temperature-entropy coordinates. The heat added is represented by the area *ABFE* and the heat rejected by the area *CDEF*.

Since

$$\text{Area } ABFE - \text{area } CDEF = \text{area } ABCD,$$

then,

$$\text{Area } ABCD = \text{heat equivalent of work done,}$$

as, from Eq. (71), the heat added minus the heat rejected equals the heat equivalent of the work done.

From Eq. 72

$$e_t = \frac{Q_1 - Q_2}{Q_1};$$

then

$$e_t = \frac{T_1(s_2 - s_1) - T_2(s_2 - s_1)}{T_1(s_2 - s_1)}$$
$$= \frac{T_1 - T_2}{T_1} \text{ [compare Eq. (74)]}.$$

This expression for efficiency is general for all engines operating in a Carnot cycle and using perfect gases, as in deriving the expression no special conditions dependent upon the nature of the gas have been assumed.

Equation (74) can be unity only when $T_2 = 0$, *i.e.*, when the temperature of the condenser, or cold body, is absolute zero. The nearer unity Eq. (74) becomes, the higher the efficiency of the engine. In order to obtain this result, $T_1 - T_2$ must be made as large as possible. This can be attained only by making T_1 larger, or T_2 smaller. In actual practice there are limits to the values of T_1 and T_2 which may be available in the different forms of engines.

It may also be shown by the following demonstration that, in any working medium in which equal increments of temperature represent equal increments of heat, the expression for efficiency applies.

Assume a scale of temperature so that each degree on the temperature scale represents one heat unit, then heat and temperature would be represented by the same quantity numerically. From Eq. (72)

$$e_t = \frac{Q_1 - Q_2}{Q_1};$$

but on the assumed temperature scale

$$Q_1 = T_1, \qquad \text{and} \qquad Q_2 = T_2,$$

hence,

$$e_t = \frac{T_1 - T_2}{T_1} \text{ [compare Eq. (74)]}.$$

All experience in testing engines using either perfect or imperfect gases as their working medium goes to show that this law applies to all forms of engines operating in a Carnot cycle no matter what the working medium may be.

31. Reversibility of Carnot Cycle.—The Carnot cycle is a reversible one, as the gas may be considered first to expand adiabatically along AD, Fig. 12, and then isothermally along DC, then to be compressed adiabatically along CB, and finally compressed isothermally along BA. It is thus possible to work around the cycle in the reverse direction.

Having proved that the Carnot cycle is reversible and that its efficiency is equal to $\frac{T_1 - T_2}{T_1}$, it is now necessary to show that no cycle can be more efficient than a reversible one, and that no reversible cycle can have a greater efficiency than that of the Carnot cycle.

Assume a nonreversible engine A and a Carnot engine B, both working between the same limits in temperature. Engine A takes Q_A heat units from the hot body and rejects Q'_A heat units to the cold body, while engine B takes Q_B heat units from the hot body and rejects Q'_B heat units to the cold body.

If engine A is more efficient than engine B it must take less heat from the hot body and reject less to the cold body, or, in other words,

$$Q_A < Q_B$$

and

$$Q'_A < Q'_B.$$

Now assume that B is to run in the reverse direction and that A is to drive B, which acts as a heat pump. Since B is a reversible engine, it will reject to the hot body, when running in a reverse direction, the same amount of heat that it takes from that body when running direct. Therefore, the combined unit of $A - B$ will, in each cycle, take from the hot body the quantity of heat Q_A and reject to the hot body the quantity of heat Q_B. But

$$Q_B > Q_A,$$

which means that this "self-acting machine unaided by any external agency" is transferring heat from a body of lower to one of higher temperature. This is contrary to the Second Law of Thermodynamics. It is, therefore, impossible for engine A to be more efficient than engine B. As these represent *any* engines of these particular types, no nonreversible engine can be more efficient than a reversible one working in the Carnot cycle.

Now assume engine A to be a reversible engine also. It can be similarly proved that it cannot be *more* efficient than the Carnot engine.

The conclusion is therefore reached that no cycle can be *more* efficient than the Carnot cycle.

It can also be proved that this cycle is the *most* efficient cycle that any engine can follow when working between any given temperature limits. This necessitates, however, a more thorough exposition of the principles of thermodynamics than it is deemed wise to include in this text, and therefore will be omitted.

PROBLEMS[1]

1. One pound of air under a pressure of 100 psia occupies 0.3 cu ft in volume. What is its temperature in degrees Fahrenheit?

[1] NOTE TO INSTRUCTOR.—In place of air, any other perfect gas may be used in these problems. The values of c_p, c_v, R, and κ for the various gases may be obtained from Table II, p. 6.

2. A quantity of air at a temperature of 60 F under a pressure of 14.7 psia has a volume of 5 cu ft. What is the volume of the same air when its temperature is changed to 120 F at constant pressure?

3. A tank contains 200 cu ft of air at a temperature of 60 F and under a pressure of 200 psia. (*a*) What is the weight of the air? (*b*) How many cubic feet will the air occupy at atmospheric pressure?

4. A tank containing 1,000 cu ft is half full of air and half full of water. The pressure in the tank is 60 psia and the temperature is 60 F. If half the water is withdrawn from the tank, what will be the resulting pressure, assuming the temperature to remain constant?

5. A tank is 4 ft in diameter and 10 ft long. Two-thirds of its volume is filled with water and one-third with air. The pressure in the tank is 100 psia. How much water must be withdrawn from the tank to reduce the pressure to 40 psia?

6. The volume of a quantity of air at 70 F under a pressure of 15 psia is 20 cu ft. (*a*) What is the temperature of the air when the volume becomes 5 cu ft and the pressure is 80 psia? (*b*) What is the weight of air?

7. A compressed-air pipe transmission is 1 mile long. The pressure at entrance is 1,000 psia; at the exit, 500 lb. The velocity at the entrance to the pipe, which is 12 in. in diameter, is 100 fps. (*a*) What must be the diameter of the pipe at the exit end to have the same velocity as at the entrance, the temperature of the air in the pipe remaining constant? (*b*) What, if the velocity at the exit is to be 90 fps?

8. A streetcar has an air-storage tank for its air brakes with a volume of 400 cu ft. The pressure in the tank at starting is 200 psia and the temperature is 60 F. The air-brake cylinders take air at 40 psia and have a volume of 2 cu ft. How many times can the brakes be operated on one tank of air, assuming the temperature of the air to remain constant?

9. The compressed-air tank on a streetcar has a volume of 250 cu ft. The pressure in the tank is 250 psi and the temperature is 60 F. There are two air cylinders each 8 in. by 10 in. The brakes take air at 60 F and a pressure of 40 psi. How many times will the tank operate the brakes?

10. An automobile dealer has an air-storage tank from which he obtains air to inflate tires. This tank is 4 ft in diameter and 8 ft long and contains air at a pressure of 200 psia. If the thermometer reads 70 F and the barometer 29 in. Hg, how many new 36-in. by 5-in. tires can he inflate to a pressure of 80 psia? The tires are taken directly from stock and the dimensions are extreme outside measurements. The tire casing is $\frac{1}{2}$ in. thick. Assume the tires to be filled with air at atmospheric pressure to start with. 1 in. Hg = 0.491 lb.

11. A tank filled with 200 cu ft of air at atmospheric pressure and at 60 F is heated to 150 F. What will be the resulting air pressure in the tank and how many Btu will be required to heat the air?

12. A tank contains 100 cu ft of air at 60 F under a pressure of 50 psia. If the air in the tank receives 100 Btu of heat, what will be the resulting temperature and pressure?

13. Ten pounds of air enclosed in a tank at 60 F under a pressure of 100 psia are heated to 100 F. (*a*) What is the volume of the air? (*b*)

What will be the final pressure? (*c*) How many Btu will be required to heat it?

14. A certain auditorium will seat 5,000 people. If each person is supplied with 2,000 cu ft of air per hr for ventilation, the outside temperature being 0 F and that in the hall being 70 F, how many pounds of air will be admitted per hour, and how many Btu will be required to heat it? Weight of 1 cu ft at 0 F is 0.0863 lb; at 70 F is 0.075 lb.

15. Four pounds of air having a volume of 12 cu ft under a pressure of 80 psia are expanded isothermally to a pressure of 15 psia. What external work is done and how much heat has been added during expansion?

16. An air-compressor cylinder has a volume of 2 cu ft. It takes in air at a pressure of 15 psia and 70 F, and compresses it isothermally to a pressure of 100 psia. (*a*) Find the weight of the air in the cylinder at the beginning of the compression stroke. (*b*) Find the final volume of the air. (*c*) Find the foot-pounds of work done upon the air during compression.

17. An air compressor handles 1.3 cu ft of compressed air per min. Discharge pressure is 95 psi. The atmospheric pressure is 14.4 psia. The temperature of the air in the engine room and in the discharge pipe is practically the same. Determine the foot-pounds of work done per minute, assuming isothermal compression.

18. An air compressor takes in air at atmospheric pressure and compresses it isothermally to a pressure of 100 psia. The compressor is 8 in. by 12 in. At what point in the stroke will the compressor begin to discharge air from the cylinder? (Neglect clearance.)

19. Two cubic feet of air at 60 F and an initial pressure of 1 atmosphere abs are compressed in a cylinder to 5 atm gage pressure. If there is no transference of heat, find the final temperature and volume.

20. A cylinder with a base area of 1 sq ft and with the upper end open to the atmosphere contains 2 cu ft of air at 60 F when compressed by a frictionless piston weighing 2,000 lb resting upon it. Find the temperature and the volume of the air if the vessel be inverted, there being no transmission of air or heat.

21. A cylinder 20 in. in diameter, with one end open to the atmosphere, is fitted with a piston that requires a total force of 50 lb to overcome its friction. It contains 5 cu ft of air under a pressure of 12 psia and at 60 F. The barometer reads 30 in. Hg. How much work is required to move the piston outward a distance of 2 ft, if no heat enters or leaves the cylinder?

22. Two cubic feet of air at 540 F under a pressure of 100 psia expand adiabatically until the temperature is 40 F. (*a*) How much heat has been added? (*b*) How much work has been done?

23. Determine the temperature of the air in the exhaust of an air engine when the air is supplied to the engine at 150 psia and at a temperature of 80 F. The expansion is adiabatic and the exhaust pressure is 15 psia.

24. Three cubic feet of air at 60 F are under a pressure of 45 psia. (*a*) Find the volume and the temperature of this air after it has expanded adiabatically until its pressure is 15 psia. (*b*) What is the work done during the expansion? (*c*) What is the heat in Btu converted into work?

25. Two cubic feet of air at 60 F are under a pressure of 80 psia. (*a*) What is the weight of the air? (*b*) What will be the final temperature and pressure if the air is expanded adiabatically until its volume is 8 cu. ft? (*c*) How much work will be done during this expansion? (*d*) How much work will be done if the air is expanded isothermally until its volume is 8 cu ft?

26. A gas expands along a path the equation of which is $pv^{1.35} = C$. During the expansion, 125 Btu are supplied to the gas and 778,000 ft-lb of work are done. Find the value of k for the gas.

27. (*a*) Find the exponent for an equation to show the relation between p and v of air compressed to one-fifth of its original volume, and rising from 13 psia to 68 psia. If the initial volume is 4 cu ft and the temperature 100 F, (*b*) find the work done; and (*c*) the heat added or rejected, and state which it is.

28. Eight hundred twenty-nine thousandths of a pound of air occupy a volume of 2 cu ft under a pressure of 80 psia. It is allowed to expand to a volume of 8 cu ft and a pressure of 11.5 psia. (*a*) How much work is done during this expansion; and (*b*) how much would have been done if the final pressure had been 20 psia?

29. (*a*) Is heat added or rejected when 4 cu ft of air under a pressure of 100 psia are expanded to a pressure of 15 psia and a volume of 14.2 cu ft along a line, equation of which is $pv^n = c$? (*b*) How much is added or rejected? (*c*) How much work is done?

30. Ten pounds of air at a pressure of 200 psia and a temperature of 150 F expand to a pressure of 15 psia, the equation of the path of the gas being $pv^{0.9}$ = constant. (*a*) Find the temperature at the end of the expansion. (*b*) How much work is done during the expansion, expressed in foot-pounds? (*c*) Is heat added or rejected during the expansion and how much, expressed in Btu?

31. (*a*) Is heat added, or rejected, when a given weight of air expands from a pressure of 100 psia and a volume of 2 cu ft to a pressure of 25 psia and a volume of 4 cu ft? (*b*) How much in Btu? (*c*) If the final volume is 5.38 cu ft instead of 4, how much heat has been added, or rejected? (*d*) How much work has been done by the gas during this expansion? (*e*) If the gas expands from the original conditions to a final pressure of 25 psia along a line the equation of which is $pv^1 = c$, what will be the final temperature, if the original temperature is 70 F? (*f*) If the expanding gas is not air, but is one in which $c_p = 0.1569$ and $c_v = 0.131$, would more or less heat be added, or rejected, than in question (*c*); and (*g*), how much?

32. (*a*) Would heat be added, or rejected, when a given weight of air is compressed from a pressure of 12 psia and a volume of 9 cu ft to a pressure of 33 psia and a volume of 6 cu ft along a curve the equation of which is $pv^n = c$? (*b*) How much, expressed in Btu? (*c*) How many foot-pounds of work would be done on, or by, the gas during this compression, and (*d*) which would it be? (*e*) Will the temperature rise, or fall? (*f*) Would more, or less, heat be added, and (*g*) how much, if the gas were compressed adiabatically from the original pressure and volume to a final pressure of 33 psia?

(*h*) How much work would be done on, or by, the gas in this case; and (*i*) which would it be? (*j*) If the gas were compressed from the original conditions to a final pressure of 33 psia along a curve the equation of which is $pv^1 = c$, what would be the final temperature if the original temperature was 70 F?

33. Given a quantity of air whose volume is 10 cu ft at 60 F under a pressure of 20 psia. Heat is added at constant volume until its pressure is 200 psia; then added at constant pressure until its volume is 40 cu ft; then rejected at constant volume until its pressure is 20 psia; and then rejected at constant pressure until its volume is the same as at the beginning of the cycle. Find (*a*) the temperature at the end of the first step; (*b*) the temperature at the end of the second step; (*c*) the temperature at the end of the third step; (*d*) the total heat added in Btu; (*e*) the total heat rejected in Btu; (*f*) the work done in foot-pounds; (*g*) the efficiency of the cycle.

34. Two cubic feet of air at a pressure of 15 psia are heated at constant volume to 100 psia; then expanded isothermally until the pressure is 15 psia; then compressed at constant pressure to its original volume. Find (*a*) the heat added in Btu; (*b*) the work done in foot-pounds; (*c*) the efficiency of the engine.

35. Given a quantity of air whose volume is 20 cu ft at 60 F under a pressure of 6 psi. The pressure of the air is caused to increase at constant volume until it reaches 192.3 psi; then the air is allowed to expand adiabatically until its pressure is 6 psi; and then it is compressed at constant pressure until its volume, pressure, and temperature are the same as at the beginning of the cycle. Find (*a*) the temperature at the end of the first change of pressure; (*b*) the temperature at the end of the adiabatic expansion; (*c*) the volume at the end of the adiabatic expansion; (*d*) the total heat added in Btu; (*e*) the total heat rejected in Btu.

36. One-half pound of air under a pressure of 20 psia and a temperature of 40 F is heated at constant volume to a pressure of 95 psia; then expanded to a volume of 10 cu ft and a pressure of 20 psia; then compressed at constant pressure to the original volume. (*a*) What is the total heat added in Btu? (*b*) What is the work done in foot-pounds? (*c*) What is the efficiency of the cycle?

37. One and three-tenths cubic feet of air under a pressure of 15 psia are heated at constant volume to a pressure of 80 psia; then expanded adiabatically to a pressure of 15 psia; then compressed at a constant pressure to the original volume. (*a*) What is the total heat added in Btu? (*b*) What is the work done in foot-pounds? (*c*) What is the efficiency of the cycle?

38. Ten cubic feet of air at a pressure of 15 psia are heated at constant volume, increasing the pressure to 100 psia; then expanded to the original pressure according to the equation $pv^{1.2} = c$, and then compressed along a constant-pressure line to its original volume. Find (*a*) the heat added; (*b*) the heat rejected; (*c*) the work done; and (*d*) the efficiency.

39. A quantity of air whose volume is 10 cu ft at 60 F under a pressure of 20 psia has heat added to it at constant volume until its pressure is 150 psia; it is then expanded according to the equation $pv^n = c$ to a pressure of 20 psia and a volume of 75 cu ft; it is finally compressed at constant pressure to the original condition. Find (*a*) the value of n; (*b*) the temperatures at the end of each step; (*c*) the heat added; (*d*) the heat rejected; (*e*) the work done; (*f*) the efficiency of the cycle.

40. Two pounds of air are carried through the following cycle: Starting at point a, where the volume is 1.04 cu ft and the pressure 100 psia, the air expands to point b, where the volume is 6.93 cu ft and the pressure 15 psia. It is then compressed to point c, where the volume is 4.025 cu ft and the pressure 15 psia. From point c it is compressed to point a. Determine (*a*) the heat added in Btu; (*b*) the heat rejected in Btu; (*c*) the work done in foot-pounds; (*d*) the efficiency of the cycle.

41. Air having a pressure of 20 psia and a volume of 1.0 cu ft is compressed along the path $pv^{1.3}$ = a constant until the pressure is 200 psia. It then expands isothermally until the pressure is 20 psia. It then follows a constant-pressure path back to the original volume. Find (*a*) the volume at the end of compression; (*b*) the volume at the end of expansion; (*c*) the total heat added in Btu; (*d*) the total heat rejected in Btu; (*e*) the efficiency of the cycle.

42. A cylinder contains 10 cu ft of air at 15 psia and 70 F. The air is first compressed isothermally to 1 cu ft. It then receives 200 Btu at constant volume, after which it expands to the original pressure and volume. Calculate (*a*) the weight of air in the cylinder; (*b*) the increase or decrease (state which) in internal energy for each part of the cycle; (*c*) the work done on or by (state which) the air for each part of the cycle; (*d*) the heat added or rejected (state which) for each part of the cycle; (*e*) the efficiency of the cycle.

43. Given the following cycle using CO_2 as the working substance: $p_1 = 300$ psia; $v_1 = 1$ cu ft; $p_2 = 30$ psia; $v_2 = 5.8$ cu ft; $p_3 = 15$ psia; $v_3 = 5.8$ cu ft. Determine (*a*) the heat added in Btu; (*b*) the heat rejected in Btu; (*c*) the work done in foot-pounds; and (*d*) the efficiency of the cycle.

44. Given the following cycle using a gas whose specific heat at constant pressure is 0.26 and at constant volume is 0.20. Four cubic feet of the gas at 14 psia are compressed along a path whose equation is $pv^n = c$ to a pressure of 224 psia and a volume of 0.473 cu ft. It is then expanded to a pressure of 26.5 psia and a volume of 4 cu ft from which point it follows a constant-volume path to the starting point. (*a*) If the working medium had been air, would the heat added or rejected during compression have been greater or less than with the above gas and how much? (*b*) Is the net work done for this cycle greater or less with the above gas than it would be if air were used?

45. The cycle of operations for a certain heat engine is as follows: Expansion from point a ($v_a = 1.04$ cu ft, $p_a = 100$ psia) to point b ($v_b = 6.93$ cu ft, $p_b = 15$ psia); then compression to point c ($v_c = 4.025$ cu ft, $p_c = 15$ psia); and finally compression back to point a. (*a*) Will there be any difference between the amount of heat added during this cycle when the working medium used is CO_2 and when it is air? (*b*) If there is a difference in the heat added, in which case will it be larger and how much in Btu? (*c*) Is there any difference in the amount of net work done during this cycle when using CO_2 than when using air as the working medium? If so, how much? (*d*) Will there be any difference in the efficiency for the above cycle when CO_2 is used than when air is used? If so, how much?

46. Two pounds of air are carried through the following cycle: Starting at point *a*, where the volume is 1 cu ft and the pressure 100 psia, the air expands to point *b*, where the volume is 7 cu ft and the pressure 6.57 psia; it is then compressed to point *c*, where the volume is 2 cu ft and the pressure 23 psia; from point *c* it is compressed to point *a*. Find (*a*) the heat added in Btu; (*b*) the heat rejected in Btu; (*c*) the work done in foot-pounds; (*d*) the efficiency of the cycle; (*e*) how the temperature at *c* compares with that at *b*.

47. Two pounds of air under a pressure of 100 psia and occupying a volume of 1 cu ft are expanded to a point where the pressure is 100 psia and the volume is 6 cu ft, then changed at constant volume to a point where the pressure is 10 psia, and then compressed to the original condition of pressure and volume. Find (*a*) the heat added in Btu; (*b*) the heat rejected in Btu; (*c*) the work done in foot-pounds; (*d*) the efficiency of cycle. (*e*) If in the second step the air were allowed to expand from a pressure of 100 psia to a pressure of 10 psia along a line the equation of which was $pv^1 = c$, instead of having the pressure changed at constant volume, what would be the final volume? (*f*) What would be the final temperature if the temperature at the beginning of the second step were 70 F?

48. A perfect gas, whose $c_p = 0.391$, $c_v = 0.301$, and $R = 70$, is confined in a cylinder of 1 cu ft volume at a pressure of 15 psia. This gas is compressed to 75 psia and a volume of 0.20 cu ft, then expanded at constant pressure to a volume of 0.291 cu ft and finally expanded to the original conditions. Compute (*a*) the total heat added; (*b*) the total heat rejected; (*c*) the net work; (*d*) the cycle efficiency.

49. One cubic foot of air at 15 psia and 70 F is compressed adiabatically to 375 psia; 10 Btu of heat is then added at constant pressure, after which the air is expanded to its original volume. The removal of 8 Btu of heat at constant volume returns the air to its initial conditions. (*a*) What quantity of heat is added in Btu? (*b*) What quantity of heat is rejected in Btu? (*c*) How much net work, in foot-pounds, is produced per cycle? (*d*) What is the efficiency of the cycle?

50. Two-tenths of a pound of air at a pressure of 15 psia and at a temperature of 70 F. is compressed adiabatically until the pressure is 150 psia and the temperature is 560 F. It expands at constant-pressure until the temperature is 900 F. It then expands adiabatically until the pressure is 15 psia and the temperature is 247 F. It follows a constant-pressure line until it reaches the original conditions. Find (*a*) the initial volume of the air; (*b*) the heat added in Btu; (*c*) the heat rejected in Btu; (*d*) the net work done in foot-pounds; (*e*) the efficiency of the cycle.

51. Find the work done, per pound of air, on a Carnot cycle when the upper temperature is 500 F, the lower temperature is 70 F, and the ratio of expansion, $\frac{v_2}{v_1} = 2$.

52. One pound of air passes through the following cycle: Heat is added along an isothermal path at 400 F, the volume increasing from 2 cu ft to 4 cu ft. It is then expanded adiabatically until its temperature becomes 70 F. From this point it is compressed isothermally until its volume becomes 6.55 cu ft, and is then compressed adiabatically until its volume becomes 2 cu ft, and its temperature 400 F. Find (*a*) the initial pressure of the air in pounds per square inch absolute; (*b*) the heat added in Btu; (*c*) the heat rejected in Btu; (*d*) the net work done in foot-pounds; (*e*) the efficiency of the cycle.

53. Two cubic feet of air under a pressure of 15 psia are heated at constant volume to a pressure of 100 psia; then heated at constant pressure to a volume of 4 cu ft; then expanded to the original pressure; and finally compressed at constant pressure to the original volume. The equation of the path of the gas during expansion is $pv^{1.2}$ = a constant. Find (*a*) the heat added in Btu; (*b*) the heat rejected in Btu; (*c*) the work done in foot-pounds; (*d*) the efficiency of the cycle.

54. Given the following cycle: Ten cubic feet of air at a pressure of 100 psia and a temperature of 70 F are expanded isothermally to a pressure of 50 psia. The air is then expanded to a pressure of 15 psia along a path, the equation of which is $pv^{1.6}$ = a constant. It then undergoes a reduction in volume along a constant-pressure line, and is finally compressed adiabatically back to its initial condition. Determine (*a*) the heat added in Btu; (*b*) the heat rejected in Btu; (*c*) the work done in foot-pounds; (*d*) the efficiency of the cycle.

55. One cubic foot of air under a pressure of 100 psia is expanded along a constant-pressure line until the volume is 3 cu ft. It is then expanded until the pressure is 15 psia and the volume is 6 cu ft. The air is then compressed along an isothermal and an adiabatic back to its original pressure and volume. Determine (*a*) the heat added in Btu; (*b*) the heat rejected in Btu; (*c*) the net work of the cycle in foot-pounds; (*d*) the efficiency of the cycle.

56. Five cubic feet of air pass through the following cycle: Starting at a pressure of 100 psia and a temperature of 60 F, it is expanded isothermally to a pressure of 60 psia and then expanded to a pressure of 15 psia along a path, the equation of which is $pv^{1.6} = c$. It then follows a path of constant pressure and finally is compressed adiabatically to its initial condition. Determine (*a*) the heat added in Btu; (*b*) the heat rejected in Btu; (*c*) the efficiency of the cycle.

57. Given a quantity of air whose volume is 0.5 cu ft under a pressure of 200 psia. Heat is added at constant pressure until its volume is 2 cu ft. The air is then expanded according to the equation $pv^n = c$ to a pressure of 15 psia and a volume of 14 cu ft. It is then compressed according to the equation $pv^n = c$ to a pressure of 90 psia and a volume of 0.5 cu ft. It is then compressed along a constant-volume line to its original starting point. Find (*a*) the heat added in Btu; (*b*) the heat rejected in Btu; (*c*) the net work done in foot-pounds; (*d*) the efficiency of the cycle.

58. Air, used as a working substance in a cycle, has an initial pressure of 15 psia and an initial volume of 0.5 cu ft. The air is compressed isothermally to a pressure of 75 psia. Heat is then added at constant volume until the pressure becomes 225 psia. This is followed by adiabatic expansion to a pressure of 15 psia, after which the air follows a constant-pressure path until the volume becomes 0.5 cu ft. Find (*a*) the heat added in Btu; (*b*) the heat rejected in Btu; (*c*) the net work done in foot-pounds; (*d*) the efficiency of the cycle.

59. An engine using air as the working substance operates on the following cycle: 1 cu ft of air at a pressure of 40 psia is heated at constant volume until the pressure is 150 psia; the air is then expanded along a path whose equation is $pv^n = c$ to a pressure of 20 psia and a volume of 4.22 cu ft; heat is then rejected at constant pressure until the volume is 2 cu ft; the air is then compressed along a path whose equation is $pv^n = c$ back to its original starting point. Find (*a*) the net work done in foot-pounds; (*b*) the efficiency of the cycle.

60. Given a closed cycle using air as the working substance in which p_1 is 15 psia, and v_1 is 1 cu ft. The gas is compressed isothermally to a pressure of 75 psia, the volume being reduced to 0.2 cu ft. Heat is then added at constant volume until the pressure rises to 225 psia. This is followed by adiabatic expansion to a pressure of 15 psia, after which the heat is rejected at constant pressure until the volume is reduced to 1 cu ft. Find (*a*) the heat added in Btu; (*b*) the heat rejected in Btu; (*c*) the work done per cycle in foot-pounds; (*d*) the efficiency of the cycle.

61. An engine is using a mixture of air and gas, for which c_p is 0.26 and c_v is 0.20. The engine takes 0.25 cu ft of this mixture at a temperature of 100 F and at a pressure of 14 psia. The gas is compressed adiabatically until the temperature is 450 F. Heat is then added and the temperature rises to 1500 F, the volume remaining constant. Adiabatic expansion follows until the pressure is 30.85 psia and the volume is 0.25 cu ft. The gas is then cooled at constant volume until the original values of temperature and pressure are reached. Find (*a*) the weight of mixture used per cycle; (*b*) the heat added in Btu; (*c*) the heat rejected in Btu; (*d*) the efficiency of the cycle. (*e*) What horsepower is being developed, if the engine is operated so that there are 1,000 cycles per min?

62. An engine uses for its working medium a mixture of gas and air for which the values of c_p and c_v are 0.351 and 0.27, respectively. The mixture is compressed from a pressure of 14 psia and a volume of 3 cu ft to a pressure of 560 psia and a volume equal to 5.85 per cent of the original volume. It is then expanded along a line the equation of which is pv^0 = a constant to a volume of 0.702 cu ft. From this point it expands adiabatically to the original volume, and then follows a constant-volume line to the original pressure. (*a*) What kind of cycle is this? (*b*) Find the heat added in Btu. (*c*) Find the heat rejected in Btu. (*d*) Find the work done in foot-pounds. (*e*) Find the efficiency of the cycle.

63. The following cycle is followed by a heat engine using a gas mixture for which $c_p = 0.25$ and $c_v = 0.19$. Two-tenths of a cubic foot of the mixture is taken in at 14 psia pressure and 140 F temperature. It is compressed adiabatically to a temperature of 490 F. Heat is added at constant volume of 0.0466 cu ft, until the temperature rises to 1640 F. The gas is then expanded adiabatically to a volume of 0.2 cu ft. It is then cooled at constant volume until the original temperatures are reached. Determine for the cycle (*a*) the heat added; (*b*) the heat rejected; (*c*) the efficiency; (*d*) the horsepower delivered to the engine piston if the engine makes 1,000 cycles per min.

64. Twenty-three hundredths of a pound of air with a pressure of 14.7 psia, a temperature of 61 F, and a volume of 3.02 cu ft passes through the following cycle: It is compressed adiabatically to a pressure of 500 psia and a temperature of 968 F; then expanded along a constant-pressure line to a volume of 0.753 cu ft. Then it is expanded adiabatically to a pressure of 71.6 psia, and a volume of 3.02 cu ft, and finally returned to the original condition along a constant-volume path. (*a*) What is the volume after being compressed to a pressure of 500 psia and a temperature of 968 F? (*b*) What is the total heat added in Btu? (*c*) What is the total heat rejected in Btu? (*d*) Find the net work done in foot-pounds. (*e*) What is the efficiency of the cycle?

65. An engine using air as the working substance operates on the following cycle: 2 cu ft of air at a pressure of 14 psia and 600 F are compressed adiabatically until the pressure rises to 500 psia; isothermal expansion then takes place until the volume is trebled; the air is then expanded adiabatically to the original volume; heat is then rejected at constant volume until the air returns to its original condition. Find (*a*) the work done per cycle in foot-pounds; (*b*) the efficiency of the cycle.

66. The cycle of an air engine is bounded by the paths *AB* (constant pressure), *BC* (adiabatic expansion), *CD* (constant volume), *DE* (constant pressure), and *EA* (constant volume), where $v_a = v_e = 0$; $p_a = p_b = 180$ psia; $v_b = 1$ cu ft; $v_c = v_d = 5$ cu ft; and $p_d = p_e = 15$ psia. By what percentage would the work be increased or decreased, if the expansion curve were $pv^{1.3} =$ constant?

67. Two cubic feet of air under a pressure of 100 psia are expanded to a volume of 4 cu ft along a line the equation of which is $pv^0 = c$; then the gas is allowed to expand to a pressure of 15 psia and a volume of 14.2 cu ft; then heat is added along a line the equation of which is $pv^{\infty} = c$, until the pressure is 30 psia; then the gas is compressed at constant pressure to a volume of 4.46 cu ft; and, finally, compressed to its original volume and pressure. Find (*a*) the heat added in Btu; (*b*) the heat rejected in Btu; (*c*) the work done in foot-pounds; (*d*) the efficiency of the cycle.

CHAPTER III

PROPERTIES OF STEAM

32. Formation of Steam.—In order to understand the operation of a steam engine it is necessary to study the nature and the properties of steam. Steam as produced in the ordinary boiler is a vapor, and often contains a certain amount of water in suspension, as does the atmosphere in foggy weather.

Assume that a boiler is partly filled with cold water and that heat is applied to its external shell. As the water in the boiler is heated, its temperature slowly rises. This increase of temperature continues from the initial temperature of the water until the temperature of the boiling point is reached, this latter temperature depending upon the pressure in the boiler. When the boiling point is reached, small particles of water are changed into steam. They rise through the mass of water and escape to the surface. The water is then said to *boil*. *The temperature at which the water boils depends entirely on the pressure in the boiler*. The steam produced from the boiling water is at the *same temperature* as the water, and under this condition the steam is said to be *saturated*. Saturated steam, then, is steam at the temperature of the boiling point corresponding to the absolute pressure at which it is formed. It may be either *wet* or *dry*, depending upon whether or not it carries moisture in suspension. The term *saturated* means merely that the steam is at this particular temperature and does not in any way indicate its moisture content. If the application of heat to the water in the boiler is continued, the pressure remaining the same, the temperature of the steam and the water will *remain constant* until all the water is evaporated. If more heat is added after all the water is converted into steam, the pressure still being kept unchanged, the temperature will rise. Steam under this condition is said to be *superheated*.

In the formation of steam the heat used is divided into three parts:

1. The heat that goes to raising the temperature of the water from its original temperature to the temperature of the boiling point which represents a *change of enthalpy of the liquid.*

2. The heat that goes to changing the water at the temperature of the boiling point into steam at the temperature of the boiling point, called *enthalpy of vaporization.*

3. The heat that goes to changing the saturated steam at the temperature of the boiling point into steam at a higher temperature but at the same pressure, called *enthalpy of superheat.*

33. Dry Saturated Steam.—*Saturated steam always exists at the temperature of the boiling point corresponding to the pressure.* If this saturated steam contains no moisture in the form of water, it is said to be *dry saturated steam;* in other words, *dry saturated steam is steam at the temperature of the boiling point and containing no water in suspension.* Water so contained is often called *entrained moisture.* If heat is added to dry saturated steam not in the presence of water, it will become superheated. If heat is taken away from dry saturated steam, it will become wet steam. Dry saturated steam is not a perfect gas, and the relation of pressure, volume, and temperature for such steam does not follow any simple law, but has been determined by experiment.

The properties of dry saturated steam were originally determined by Regnault between 1840 and 1850, and so carefully was his work done that no errors in his results were apparent until within very recent years, when the great difficulty in obtaining steam which is exactly dry and saturated became appreciated, and new experiments by various scientists proved that Regnault's results were slightly high at some pressures and slightly low at others. The steam tables given in this book are based upon the most recent experiments, and are probably correct to a fraction of 1 per cent.

34. Quality of Steam.—Saturated steam may or may not contain entrained moisture. By *quality* of steam is meant *the per cent of dry saturated steam in the sample.* Saturated steam containing no entrained moisture has a quality of 100 per cent.

The *percentage of moisture* is found by subtracting the quality from 100.

35. Wet Steam.—*Wet steam is saturated steam which contains entrained moisture.* When saturated steam is used in a steam engine, it almost always contains moisture in the form of water, so that the substance used by the engine as a working fluid is a mixture of steam and water called *wet steam.* The steam and the water in this case are at the same temperature.

36. Superheated Steam.—*Superheated steam is steam at a temperature higher than the temperature of the boiling point corresponding to the pressure.* It is sometimes called *steam gas.* If water were to be mixed with superheated steam, this water would be evaporated as long as the steam remains superheated. Superheated steam can have *any temperature higher* than that of the boiling point. When raised to any considerable temperature above the temperature of the boiling point, it follows very closely the laws of a perfect gas and may be treated as a perfect gas. The equation for superheated steam, considered as a perfect gas, is

$$pv = 85.5t, \text{ approximately.} \tag{1}$$

The value of κ for superheated steam is approximately 1.3.

The difference between the actual temperature of the steam and the temperature of the boiling point corresponding to the pressure gives the number of *degrees of superheat* in the steam, and this quantity multiplied by the specific heat of superheated steam, c_p, gives the *amount* of superheat, measured in Btu.

The specific heat of superheated steam is a variable and depends upon the pressure of the steam and the temperature to which the steam is superheated. Table VI gives the *mean specific heat* of superheated steam between the saturation temperatures and certain higher temperatures. This table should be used for approximate calculations only. In general, it is better to obtain the enthalpy of superheated steam by the use of Table VIII, or the complete Keenan and Keyes' "Thermodynamic Properties of Steam."

37. Enthalpy.—This term (from the Greek word *enthalpo* meaning *heat within*) was selected by the A.S.M.E., as announced in the July, 1936, issue of *Mechanical Engineering,* "to designate the thermodynamic function defined as the internal energy plus the pressure volume product of any working substance." It is used in place of the old terms "total heat," "heat content," and "heat of the liquid" which were discarded as confusing and misleading. In the report of the committee, published in the

August, 1936, issue, various reasons are given as being responsible for this change in terminology.

For example, "the total heat of saturated steam" was formerly defined as "the heat of the liquid" plus the latent heat of vaporization at the given pressure. This definition becomes clear only when a precise meaning is applied to the term "the heat of the liquid," which has been variously defined. The term "total heat" of saturated steam has usually meant the quantity of heat required to heat the liquid from 32 F to the boiling point, and then vaporize it all at *constant pressure;* but the restriction to constant pressure is too often forgotten and the resulting confusion has been very great. Furthermore, with the properties of compressed liquid demanding more recognition as higher and higher pressures enter our engineering calculations, it becomes very important to recognize that the initial state of the liquid cannot be specified in terms of its temperature alone; hence the initial reference state of the liquid must vary with the pressure if a precise meaning is to be attached to the old term, "total heat."

The term "heat content" is fundamentally unsatisfactory because the name itself is misleading when used to represent the quantity, *internal energy plus the pressure times volume.* Furthermore, the term "heat content" is, unfortunately, sometimes used to represent the heating value of a fuel; and another meaning is occasionally given to it when it is incorrectly interpreted to mean internal energy.

38. Enthalpy of the Liquid.—As stated in Art. 32, the first heat added to water which is being changed into steam goes to raising the temperature of the water from its original temperature to the temperature of the boiling point and is called *enthalpy of the liquid.* In order to show in steam tables definite quantities of heat required for this purpose, a fixed weight of water and a definite initial temperature must be agreed upon. One pound has been taken as the weight and 32 F as the starting point above which the quantity of heat taken in is calculated.

The enthalpy of the liquid may, therefore, be defined as *the heat necessary to raise one pound of water from 32 F to the temperature of the boiling point, the pressure remaining constant.* This may be expressed numerically as follows: let c be the specific heat of the water, t the temperature of the boiling point, and h_f the enthalpy of the liquid; then

$$h_f = c(t - 32). \qquad (2)$$

For approximate results c may be taken as 1, but where great accuracy is required the enthalpy of the liquid should be taken from the steam tables as shown in Table VII. During this operation the change in the volume of the water is extremely small, and the amount of external work done may be neglected and all the enthalpy of the liquid may be considered as going to increasing the heat energy of the water.

39. Enthalpy of Vaporization or Latent Heat of Steam.—When water has reached the boiling point, more heat must be added to convert this water into steam. Although the temperature remains the same during this change in state, a large amount of heat is required to produce this vaporization. *The heat necessary to convert one pound of water at the temperature of the boiling point into steam at the same temperature is called the enthalpy of vaporization or latent heat of vaporization, or simply the latent heat, and is designated by h_{fg} or L.* The values of h_{fg} may be obtained from Table VII. It should be noted that the enthalpy of vaporization decreases as the temperature of the boiling point increases.

When steam condenses, the same amount of heat is given up as was required to produce it.

40. Total Enthalpy of Steam.—The total enthalpy of dry saturated steam is *the heat necessary to change one pound of water at 32 F into one pound of steam at the temperature of the boiling point.* It is equal to the sum of the enthalpy of the liquid and the enthalpy of vaporization, and is designated by h_g. Therefore,

$$h_g = h_f + h_{fg}. \tag{3}$$

The values of the total enthalpy of dry saturated steam are given in Table VII.

If x represents the percentage of dry steam in a mixture of steam and water, *i.e.*, if the mixture has a quality of x per cent, then the enthalpy of vaporization of one pound of wet steam equals

$$xh_{fg}, \tag{4}$$

and the total enthalpy of one pound of wet steam equals

$$h_f + xh_{fg}. \tag{5}$$

It should be noted that the quality affects the enthalpy of vaporization only and must not be applied to the enthalpy of the liquid since the whole 100 per cent of the mixture must have enthalpy of the liquid added to it or, in other words, must be brought up to the temperature of the boiling point corresponding to the pressure before any of it will begin to evaporate, or change into steam.

The total enthalpy of superheated steam exceeds that of dry saturated steam at the same pressure by the amount of heat required to raise the steam from the boiling point to the actual temperature. In other words, it is equal to

$$h_f + h_{fg} + c_p(t_{\text{sup.}} - t_{\text{sat.}}), \tag{6}$$

where $t_{\text{sup.}}$ is the actual temperature of the superheated steam, $t_{\text{sat.}}$ is the temperature of saturation, or the boiling point, corresponding to the pressure, and c_p is the specific heat of superheated steam at constant pressure. (The condition of heating or cooling at constant pressure is the one encountered in practically all engineering problems.)

41. External Work of Evaporation.—When water is changed into steam, the volume is greatly increased, so that a portion of the enthalpy of vaporization goes to doing external work. This *external work of vaporization* is a definite quantity, since for any given temperature, the pressure and the change in volume from water to steam are both definite quantities. If p equals the pressure in pounds per square inch at which the steam is formed, v_g equals the volume of one pound of the steam, and v_f the volume of one pound of the water, both expressed in cubic feet, then the external work done equals

$$\frac{144p(v_g - v_f)}{778}, \text{ expressed in Btu.} \tag{7}$$

The volume of one pound of water under these conditions may be taken as approximately 0.017 cu ft. At 212 F the external work done in producing one pound of steam is equivalent to 73 Btu, or about one-thirteenth of the latent heat.

42. Internal Energy.—Only a small part of the enthalpy of vaporization goes to external work. The balance goes to increase the internal energy of the steam and is called *internal energy of vaporization*, I, and expressed in Btu; or

$$I = h_{fg} - \frac{144p(v_g - v_f)}{778}. \tag{8}$$

The total *internal energy of the steam, E,* is equal to the internal energy of evaporation plus the enthalpy of the liquid, or

$$E = I + h_f. \tag{9}$$

It will be seen that it has been assumed here that all the enthalpy of the liquid, h_f, goes to increasing the internal energy of the steam. This is practically true since, as stated in Art. 38, the change in the volume of the water when it is heated from 32 F to the temperature of the boiling point is so small as to be negligible; consequently, the external work done during this rise in temperature is not considered.

The internal energy of the steam may also be expressed as the total enthalpy of the steam less the heat equivalent of the external work done, or

$$E = h_g - \frac{144p(v_g - v_f)}{778}. \tag{10}$$

Therefore,

$$h_g = E + \frac{144p(v_g - v_f)}{778}, \tag{11}$$

or the total enthalpy equals the internal energy of the steam plus the external work done [see Eq. (12), Chap. II].

43. Entropy of Steam.—The change in entropy when water at 32 F is heated to the boiling point corresponding to the pressure is called the *entropy of the liquid*, s_f. Therefore,

$$s_f = \int_{T_1}^{T_2} \frac{dh}{T} \text{ [compare Eq. (63), Chap. II]} \tag{12}$$

where T_1 is the absolute temperature at 32 F and T_2 the absolute temperature of the boiling point.

Assuming the specific heat of water to be 1,

$$s_f = \int_{T_1}^{T_2} \frac{dT}{T} \tag{13}$$

$$= \log_e T_2 - \log_e T_1,$$

$$= \log_e \frac{T_2}{T_1}. \tag{14}$$

When water at the boiling point is converted into steam, the latent heat added is added at a constant temperature. The gain in entropy during this process is equal to the latent heat divided by the absolute temperature of the boiling point. This is called the *entropy of vaporization* and is designated by $\frac{h_{fg}}{T}$. (15)

The *total entropy of dry saturated steam*, s_g, is the sum of the entropy of the liquid and the entropy of vaporization, or

$$s_g = s_f + \frac{h_{fg}}{T} \tag{16}$$

and the total entropy of one pound of wet steam with a quality of x per cent equals

$$s_f + \frac{xh_{fg}}{T}. \tag{17}$$

During superheating, the entropy changes occur with increasing temperatures and varying specific heats. This is similar to the condition that exists when water is heated from 32 F up to the boiling point, with the difference that the specific heat of superheated steam is much more variable than the specific heat of water.

Letting $s_{\text{sup.}}$ represent the entropy of superheat, the total entropy of one pound of superheated steam, s, will be

$$s = s_f + \frac{h_{fg}}{T} + s_{\text{sup.}}. \tag{18}$$

The numerical values for s_f and $\frac{h_{fg}}{T}$ will be found in Table VII and the values of s are given in Table VIII. For more accurate work consult Keenan and Keyes' "Thermodynamic Properties of Steam."

Figure 15 shows the change in entropy in 1 lb of water as heat is added to it converting it first into dry saturated and then into superheated steam.

Beginning at a where the temperature of the water is 32 F, an examination of the steam tables shows the entropy of the liquid, s_f, to be 0 and the entropy of vaporization, $\frac{h_{fg}}{T}$, to be 2.1877. Laying this latter figure out to the right from a gives the point i. Then ai represents the entropy of vaporization and also the total entropy of dry saturated steam with a boiling point of 32 F.

If the water is evaporated at 212 F, the entropy of the liquid is found to be 0.3120 and the entropy of vaporization, 1.4446, represented in the figure by *mb* and *bh*. The total entropy will equal the sum of these two or *mh*, which equals 1.7566.

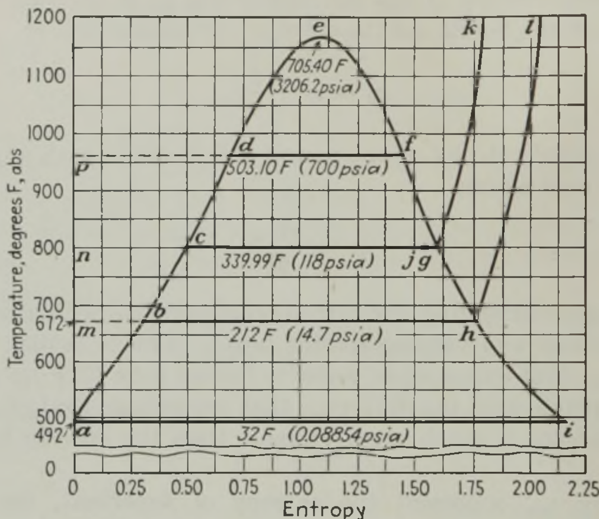


FIG. 15.—Temperature-entropy diagram for steam.

In a similar manner the points *c*, *g*, *d*, and *f* are plotted. Curves drawn through the points *a*, *b*, *c*, and *d*, and points *i*, *h*, *g*, and *f* give what are known as the *water* or *liquid line*, and the *saturation line*. If these lines are continued, they will meet at the point *e* forming the "steam dome." At this point the temperature of vaporization is 705.40 F, the entropy of vaporization becomes zero, and the total entropy equals the entropy of the liquid. This is called the *critical point* and is the place where water would pass direct from a liquid to a vapor without the addition of any latent heat, *i.e.*, the enthalpy of steam would consist of the enthalpy of the liquid only.

In case the steam is wet, having a quality *x*, the entropy of vaporization must be multiplied by *x*, so that at 118 psia, for example, the entropy of vaporization will be *cj* instead of *cg*, and the total entropy *nj* instead of *ng*. In this case the quality of the steam is represented by $\frac{cj}{cg}$ and the moisture by $\frac{jg}{cg}$.

When the saturated steam is superheated, the change of entropy is represented by such lines as *hf* and *gk* which are similar to the water line *ae*. Since the specific heat of superheated steam varies much more than the specific heat of water, there will be a particular superheat line on the temperature-entropy diagram for each particular temperature of evaporation.

44. The Mollier Diagram.—An enthalpy-entropy chart (Fig. 16), published by Dr. Mollier in 1904 and known as the *Mollier diagram* furnishes a quick and convenient means of solving many problems involving the properties of steam. On this diagram the total enthalpies above 32 F are plotted as ordinates and total entropies as abscissas. Three sets of intersecting curves are drawn on the diagram: one for constant pressure, one for constant temperature, and one for constant quality or superheat.

If drawn to a large enough scale, the diagram may be used instead of the steam tables for most commercial work. Figure 16 is drawn to too small a scale for accurate work. The diagram accompanying the steam tables should be used for that.

Example.—Find the total enthalpy and total entropy in one pound of steam at a pressure of 120 psia, (*a*) when dry and saturated; (*b*) if the quality is 96 per cent; and (*c*) when it contains 50 degrees of superheat.

Solution.—(*a*) On the diagram from Keenan and Keyes' "Thermodynamic Properties of Steam" locate the intersection of the 120-lb pressure line and the saturation line. Read along the horizontal line to find the total enthalpy, 1,190 Btu, and along the vertical line to find the total entropy, 1.588.

(*b*) From the intersection of the 120-lb pressure line with the 4 per cent moisture line follow the horizontal to find the total enthalpy, 1,154.5 Btu, and the vertical line to find the total entropy, 1.542.

(*c*) From the intersection of the 120-lb pressure line with the 50-degree superheat line follow the horizontal to find the total enthalpy, 1,219 Btu, and the vertical to find the total entropy, 1.623.

45. Steam Tables.—Table VII showing the properties of saturated steam and Table VIII showing those of superheated steam are abstracted from "Thermodynamic Properties of Steam" by Professor J. H. Keenan and Professor F. G. Keyes, 1936 Edition, by permission of the publisher, John Wiley & Sons, Inc. The headings of the various columns are such that further explanation is unnecessary. It must always be kept in mind that these tables are based on *absolute* pressures, and, if a gage pressure is given, the atmospheric pressure, as shown by the

(*Text continues on p.* 72)

Absolute pressure, in Hg.	Saturation temperature, deg. fahr.	Absolute pressure, lb. per sq. in.	Saturation temperature, deg. fahr.	Absolute pressure, lb. per sq. in.	Saturation temperature, deg. fahr.
1	79.1	55	287.1	800	518.2
1½	91.8	60	292.7	850	525.2
2	101.2	70	302.9	900	532.0
Lb. per		80	312.0	950	538.4
sq. in.		90	320.3	1000	544.6
1.5	115.7	100	327.8	1100	556.3
2	126.1	110	334.8	1200	567.2
3	141.5	120	341.3	1300	577.3
4	153.0	130	347.3	1400	587.0
5	162.3	140	353.0	1500	596.1
6	170.1	150	358.4	1600	604.7
7	176.9	160	363.6	1700	613.0
8	182.9	180	373.1	1800	620.9
9	188.3	200	381.8	1900	628.4
10	193.2	220	389.9	2000	635.6
12	202.0	240	397.4	2100	642.6
14	209.6	260	404.4	2200	649.3
16	216.3	280	411.1	2300	655.7
18	222.4	300	417.3	2400	661.9
20	228.0	350	431.7	2500	668.0
22	233.1	400	444.6	2600	673.8
24	237.8	450	456.3	2700	679.5
26	242.3	500	467.0	2800	684.9
28	246.4	550	476.9	2900	690.2
30	250.3	600	486.2	3000	695.2
35	259.3	650	494.9	3100	700.2
40	267.2	700	503.0	3200	704.9
45	274.5	750	510.8	3226	706.1
50	281.0	...	.....	.....	.....

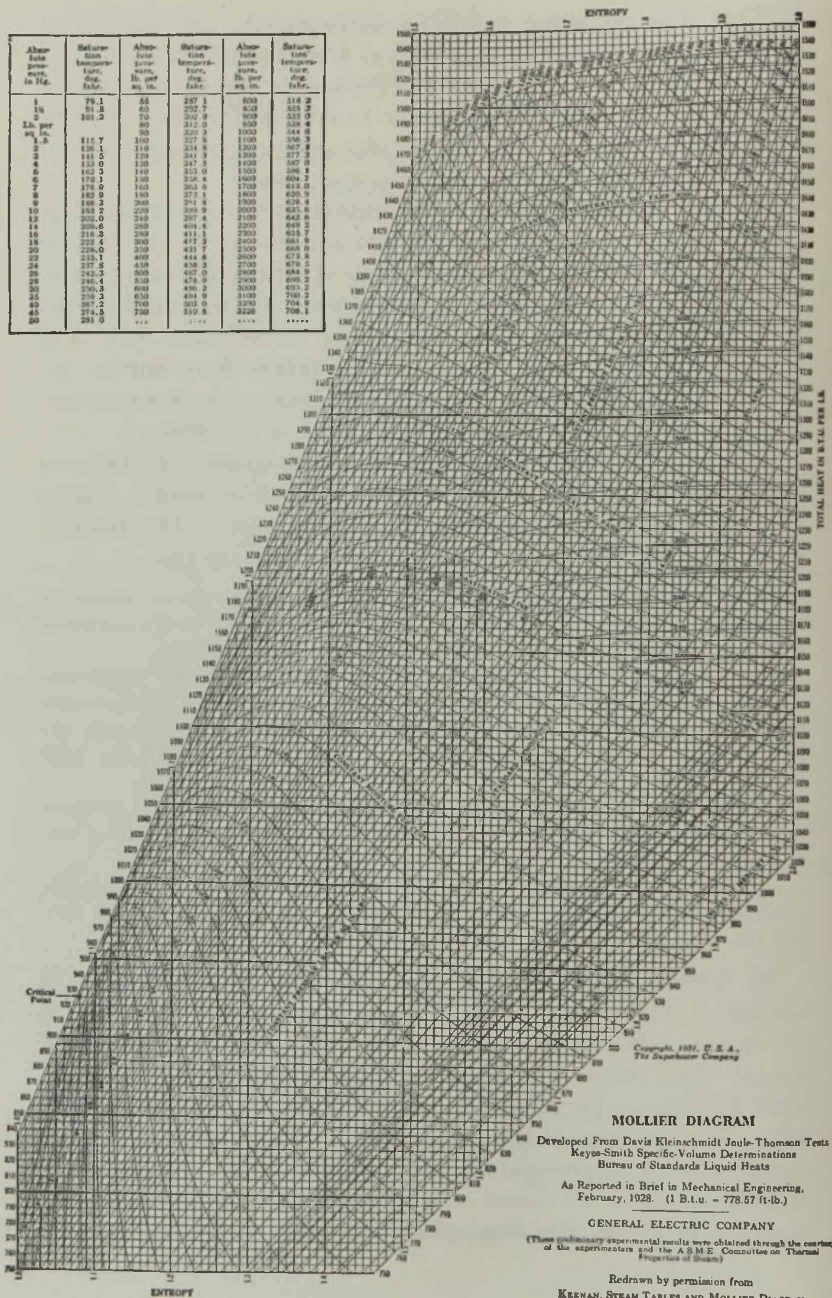


FIG. 16.—Mollier diagram. Reproduced by permission from a drawing made by The Superheater Company from the Mollier Diagram published by the A.S.M.E. in 1930 in "Steam Tables and Mollier Diagram" by Keenan.

TABLE VI.—MEAN SPECIFIC HEATS OF SUPERHEATED STEAM

Absolute pressure, psia	14.7	25.0	50.0	75.0	100.0	150.0	200.0	250.0	300.0	350.0	400.0	450.0	500.0	550.0	600.0	700.0	800.0	900.0	1000.0	Absolute pressure, psia
Temperature of boiling point, deg F	212.0	240.1	281.0	307.6	327.8	358.4	381.8	401.1	417.3	431.7	444.6	456.3	467.0	476.9	486.2	503.0	518.2	532.0	544.6	Temperature of boiling point, deg F
Actual temperature of steam, deg F																				Actual temperature of steam, deg F
250	0.48	0.52	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	**250**
275	0.48	0.50	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	**275**
300	0.48	0.50	0.55	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	**300**
325	0.48	0.49	0.53	0.57	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	**325**
350	0.47	0.49	0.52	0.56	0.59	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	**350**
375	0.47	0.49	0.52	0.55	0.57	0.63	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	**375**
400	0.47	0.48	0.51	0.54	0.56	0.61	0.66	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	**400**
425	0.47	0.48	0.51	0.53	0.55	0.59	0.64	0.69	0.73	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	**425**
450	0.47	0.48	0.50	0.52	0.55	0.58	0.62	0.66	0.70	0.74	0.78	.....	.....	.....	.....	.....	.....	.....	.....	**450**
475	0.47	0.48	0.50	0.52	0.54	0.57	0.61	0.64	0.68	0.71	0.75	0.80	0.85	.....	.....	.....	.....	.....	.....	**475**
500	0.47	0.48	0.50	0.52	0.53	0.56	0.59	0.63	0.66	0.69	0.73	0.77	0.81	0.86	0.91	.....	.....	.....	.....	**500**
525	0.47	0.48	0.50	0.51	0.53	0.56	0.58	0.61	0.64	0.67	0.70	0.74	0.78	0.82	0.86	0.96	1.09	.....	.....	**525**
550	0.47	0.48	0.50	0.51	0.52	0.55	0.58	0.60	0.63	0.65	0.68	0.71	0.75	0.78	0.82	0.91	1.01	1.13	1.28	**550**
600	0.47	0.48	0.49	0.51	0.52	0.54	0.56	0.59	0.61	0.63	0.65	0.68	0.70	0.73	0.76	0.83	0.90	0.98	1.07	**600**
650	0.47	0.48	0.49	0.51	0.52	0.54	0.56	0.58	0.60	0.61	0.63	0.66	0.68	0.70	0.72	0.77	0.83	0.88	0.95	**650**
700	0.48	0.48	0.49	0.50	0.51	0.53	0.55	0.57	0.59	0.60	0.62	0.64	0.66	0.68	0.70	0.74	0.78	0.82	0.87	**700**
750	0.48	0.48	0.49	0.50	0.51	0.53	0.55	0.56	0.58	0.59	0.61	0.63	0.64	0.66	0.68	0.71	0.75	0.78	0.82	**750**
800	0.48	0.49	0.50	0.51	0.51	0.53	0.54	0.56	0.57	0.59	0.60	0.62	0.62	0.65	0.67	0.69	0.72	0.76	0.79	**800**
850	0.48	0.49	0.50	0.51	0.51	0.53	0.54	0.56	0.57	0.58	0.59	0.61	0.63	0.64	0.65	0.68	0.71	0.74	0.76	**850**
900	0.48	0.49	0.50	0.51	0.51	0.53	0.54	0.55	0.56	0.57	0.59	0.60	0.61	0.63	0.64	0.66	0.69	0.72	0.74	**900**
950	0.49	0.49	0.50	0.51	0.51	0.53	0.54	0.55	0.56	0.57	0.58	0.60	0.61	0.62	0.63	0.65	0.67	0.70	0.73	**950**
1000	0.49	0.49	0.50	0.51	0.52	0.53	0.54	0.55	0.56	0.57	0.58	0.59	0.60	0.61	0.62	0.64	0.67	0.69	0.71	**1000**

TABLE VII.—PROPERTIES OF SATURATED STEAM
(English units)

Absolute pressure, in. Hg	Temperature	Specific volume		Enthalpy			Entropy			Internal energy			Absolute pressure, in. Hg
		Saturated liquid	Saturated vapor	Saturated liquid	Evaporation	Saturated vapor	Saturated liquid	Evaporation	Saturated vapor	Saturated liquid	Evaporation	Saturated vapor	
p	t	v_f	v_g	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g	u_f	u_{fg}	u_g	p
0.25	40.23	0.01602	2423.7	8.28	1071.1	1079.4	0.0166	2.1423	2.1589	8.28	1016.0	1024.3	0.25
0.50	58.80	0.01604	1256.4	26.86	1060.6	1087.5	0.0532	2.0453	2.0985	26.86	1003.5	1030.4	0.50
1.00	79.03	0.01608	652.3	47.05	1049.3	1096.3	0.0914	1.9473	2.0387	47.05	990.0	1037.0	1.00
1.5	91.72	0.01611	444.9	59.71	1042.0	1101.7	0.1147	1.8894	2.0041	59.71	981.4	1041.1	1.5
2.0	101.14	0.01614	339.2	69.10	1036.6	1105.7	0.1316	1.8481	1.9797	69.10	974.9	1044.0	2.0
3.0	115.06	0.01618	231.6	82.99	1028.6	1111.6	0.1560	1.7896	1.9456	82.99	965.5	1048.5	3.0
10	161.49	0.01640	74.76	129.38	1001.4	1130.8	0.2335	1.6121	1.8456	129.37	933.4	1062.8	10
20	192.37	0.01658	39.07	160.33	982.7	1143.0	0.2822	1.5069	1.7891	160.30	911.7	1072.0	20
30	212.13	0.01672	26.74	180.19	970.3	1150.5	0.3122	1.4442	1.7564	180.14	897.5	1077.6	30
psia													psia
1.0	101.74	0.01614	333.6	69.70	1036.3	1106.0	0.1326	1.8456	1.9782	69.70	974.6	1044.3	1.0
2.0	126.08	0.01623	173.73	93.99	1022.2	1116.2	0.1749	1.7451	1.9200	93.98	957.9	1051.9	2.0
3.0	141.48	0.01630	118.71	109.37	1013.2	1122.6	0.2008	1.6855	1.8863	109.36	947.3	1056.7	3.0
4.0	152.97	0.01636	90.63	120.86	1006.4	1127.3	0.2198	1.6427	1.8625	120.85	939.3	1060.2	4.0
5.0	162.24	0.01640	73.52	130.13	1001.0	1131.1	0.2347	1.6094	1.8441	130.12	933.0	1063.1	5.0
6.0	170.06	0.01645	61.98	137.96	996.2	1134.2	0.2472	1.5820	1.8292	137.94	927.5	1065.4	6.0
7.0	176.85	0.01649	53.64	144.76	992.1	1136.9	0.2581	1.5586	1.8167	144.74	922.7	1067.4	7.0
8.0	182.86	0.01653	47.34	150.79	988.5	1139.3	0.2674	1.5383	1.8057	150.77	918.4	1069.2	8.0
9.0	188.28	0.01656	42.40	156.22	985.2	1141.4	0.2759	1.5203	1.7962	156.19	914.6	1070.8	9.0
10	193.21	0.01659	38.42	161.17	982.1	1143.3	0.2835	1.5041	1.7876	161.14	911.1	1072.2	10
12	201.96	0.01665	32.40	169.96	976.6	1146.6	0.2967	1.4763	1.7730	169.92	904.8	1074.7	12
14	209.56	0.01670	28.04	177.61	971.9	1149.5	0.3083	1.4522	1.7605	177.57	899.3	1076.9	14
14.696	212.00	0.01672	26.80	180.07	970.3	1150.4	0.3120	1.4446	1.7566	180.02	897.5	1077.5	14.696
16	216.32	0.01674	24.75	184.42	967.6	1152.0	0.3184	1.4313	1.7497	184.37	804.3	1078.7	16
18	222.41	0.01679	22.17	190.56	963.6	1154.2	0.3275	1.4128	1.7403	190.50	889.9	1080.4	18
20	227.96	0.01683	20.089	196.16	960.1	1156.3	0.3356	1.3962	1.7319	196.10	885.8	1081.9	20
22	233.07	0.01687	18.375	201.33	956.8	1158.1	0.3431	1.3811	1.7242	201.26	882.0	1083.3	22
24	237.82	0.01691	16.938	206.14	953.7	1159.8	0.3500	1.3672	1.7172	206.07	878.5	1084.6	24
26	242.25	0.01694	15.715	210.62	950.7	1161.3	0.3564	1.3544	1.7108	210.54	875.2	1085.7	26
28	246.41	0.01698	14.663	214.83	947.9	1162.7	0.3623	1.3425	1.7048	214.74	872.1	1086.8	28

30	250.33	0.01701	13.746	218.82	945.3	1164.1	0.3680	1.3313	1.6993	218.1	869.1	1087.8	30
32	254.05	0.01704	12.940	222.59	942.8	1165.4	0.3733	1.3209	1.6941	222.49	866.3	1088.7	32
34	257.58	0.01707	12.226	226.18	940.3	1166.5	0.3783	1.3110	1.6893	226.07	863.5	1089.6	34
36	260.95	0.01709	11.588	229.60	938.0	1167.6	0.3831	1.3017	1.6848	229.0	861.0	1090.5	36
38	264.16	0.01712	11.015	232.89	935.8	1168.7	0.3876	1.2929	1.6805	232.77	858.5	1091.3	38
40	267.25	0.01715	10.498	236.03	933.7	1169.7	0.3919	1.2844	1.6763	235.90	856.1	1092.0	40
45	274.44	0.01721	9.401	243.36	928.6	1172.0	0.4019	1.2650	1.6669	243.22	850.5	1093.7	45
50	281.01	0.01727	8.515	250.09	924.0	1174.1	0.4110	1.2474	1.6585	249.93	845.4	1095.3	50
55	287.07	0.01732	7.787	256.30	919.6	1175.9	0.4193	1.2316	1.6509	256.12	840.6	1096.7	55
60	292.71	0.01738	7.175	262.09	915.5	1177.6	0.4270	1.2168	1.6438	261.90	836.0	1097.9	60
65	297.97	0.01743	6.655	267.50	911.6	1179.1	0.4342	1.2032	1.6374	267.29	831.8	1099.1	65
70	302.92	0.01748	6.206	272.61	907.9	1180.6	0.4409	1.1906	1.6315	272.38	827.8	1100.2	70
75	307.60	0.01753	5.816	277.43	904.5	1181.9	0.4472	1.1787	1.6259	277.19	824.0	1101.2	75
80	312.03	0.01757	5.472	282.02	901.1	1183.1	0.4531	1.1676	1.6207	281.76	820.3	1102.1	80
85	316.25	0.01761	5.168	286.39	897.8	1184.2	0.4587	1.1571	1.6158	286.11	816.8	1102.9	85
90	320.27	0.01766	4.896	290.56	894.7	1185.3	0.4641	1.1471	1.6112	290.27	813.4	1103.7	90
95	324.12	0.01770	4.652	294.56	891.7	1186.2	0.4692	1.1376	1.6068	294.25	810.2	1104.5	95
100	327.81	0.01774	4.432	298.40	888.8	1187.2	0.4740	1.1286	1.6026	298.08	807.1	1105.2	100
105	331.36	0.01778	4.232	302.10	886.0	1188.1	0.4787	1.1199	1.5986	301.75	804.1	1105.9	105
110	334.77	0.01782	4.049	305.66	883.2	1188.9	0.4832	1.1117	1.5948	305.30	801.2	1106.5	110
114	337.42	0.01784	3.914	308.43	881.1	1189.5	0.4866	1.1053	1.5919	308.05	798.9	1106.9	114
115	338.07	0.01785	3.882	309.11	880.6	1189.7	0.4875	1.1037	1.5912	308.73	798.4	1107.1	115
120	341.25	0.01789	3.728	312.44	877.9	1190.4	0.4916	1.0962	1.5878	312.05	795.6	1107.6	120
125	344.33	0.01792	3.587	315.68	875.4	1191.1	0.4956	1.0888	1.5844	315.26	792.8	1108.1	125
130	347.32	0.01796	3.455	318.81	872.9	1191.7	0.4995	1.0817	1.5812	318.38	790.2	1108.6	130
135	350.21	0.01800	3.333	321.85	870.6	1192.4	0.5032	1.0749	1.5781	321.40	787.7	1109.1	135
140	353.02	0.01802	3.220	324.82	868.2	1193.0	0.5069	1.0682	1.5751	324.35	785.2	1109.6	140
145	355.76	0.01806	3.114	327.70	865.8	1193.5	0.5104	1.0618	1.5722	327.22	782.8	1110.0	145
150	358.42	0.01809	3.015	330.51	863.6	1194.1	0.5138	1.0556	1.5694	330.01	780.5	1110.5	150
160	363.53	0.01815	2.834	335.93	859.2	1195.1	0.5204	1.0436	1.5640	335.39	775.8	1111.2	160
170	368.41	0.01822	2.675	341.09	854.9	1196.0	0.5266	1.0324	1.5590	340.52	771.4	1111.9	170
180	373.06	0.01827	2.532	346.03	850.8	1196.9	0.5325	1.0217	1.5542	345.42	767.1	1112.5	180
190	377.51	0.01833	2.404	350.79	846.8	1197.6	0.5381	1.0116	1.5497	350.15	763.0	1113.1	190
200	381.79	0.01839	2.288	355.36	843.0	1198.4	0.5435	1.0018	1.5453	354.68	759.0	1113.7	200
210	385.90	0.01844	2.183	259.77	839.2	1199.0	0.5487	0.9925	1.5412	359.05	755.2	1114.2	210
220	389.86	0.01850	2.087	364.02	835.6	1199.6	0.5537	0.9835	1.5372	363.27	751.3	1114.6	220
225	391.79	0.01852	2.0422	366.09	833.8	1199.9	0.5561	0.9792	1.5353	365.32	749.5	1114.8	225
230	393.68	0.01854	1.9992	368.13	832.0	1200.1	0.5585	0.9750	1.5334	367.34	747.7	1115.0	230
240	397.37	0.01860	1.9183	372.12	828.5	1200.6	0.5631	0.9667	1.5298	371.29	744.1	1115.4	240
250	400.95	0.01865	1.8438	376.00	825.1	1201.1	0.5675	0.9588	1.5263	375.14	740.7	1115.8	250

TABLE VII.—PROPERTIES OF SATURATED STEAM.—(*Continued*)

Absolute pressure, psia	Temperature	Specific volume		Enthalpy			Entropy			Internal energy			Absolute pressure, psia
		Saturated liquid	Saturated vapor	Saturated liquid	Evaporation	Saturated vapor	Saturated liquid	Evaporation	Saturated vapor	Saturated liquid	Evaporation	Saturated vapor	
p	t	v_f	v_g	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g	u_f	u_{fg}	u_g	p
260	404.42	0.01870	1.7748	379.76	821.8	1201.5	0.5719	0.9510	1.5229	378.86	737.3	1116.1	260
270	407.78	0.01875	1.7107	383.42	818.5	1201.9	0.5760	0.9436	1.5196	382.48	733.9	1116.4	270
275	409.43	0.01878	1.6804	385.21	816.9	1202.1	0.5781	0.9399	1.5180	384.26	732.3	1116.6	275
280	411.05	0.01880	1.6511	386.98	815.3	1202.3	0.5801	0.9363	1.5164	386.01	730.7	1116.7	280
290	414.23	0.01885	1.5954	390.46	812.1	1202.6	0.5841	0.9292	1.5133	389.45	727.5	1116.9	290
300	417.33	0.01890	1.5433	393.84	809.0	1202.8	0.5879	0.9225	1.5104	392.79	724.3	1117.1	300
310	420.35	0.01894	1.4944	397.15	806.0	1203.1	0.5916	0.9159	1.5075	396.06	721.3	1117.4	310
320	423.29	0.01899	1.4485	400.39	803.0	1203.4	0.5952	0.9094	1.5046	399.26	718.3	1117.6	320
330	426.16	0.01904	1.4053	403.56	800.0	1203.6	0.5988	0.9031	1.5019	402.40	715.4	1117.8	330
340	428.97	0.01908	1.3645	406.66	797.1	1203.7	0.6022	0.8970	1.4992	405.46	712.4	1117.9	340
350	431.72	0.01913	1.3260	409.69	794.2	1203.9	0.6056	0.8910	1.4966	408.45	709.6	1118.0	350
360	434.40	0.01917	1.2895	412.67	791.4	1204.1	0.6090	0.8851	1.4941	411.39	706.8	1118.2	360
370	437.03	0.01921	1.2550	415.59	788.6	1204.2	0.6122	0.8794	1.4916	414.27	704.0	1118.3	370
380	439.60	0.01925	1.2222	418.45	785.8	1204.3	0.6153	0.8738	1.4891	417.10	701.3	1118.4	380
390	442.12	0.01930	1.1910	421.27	783.1	1204.4	0.6184	0.8683	1.4867	419.88	698.6	1118.5	390
400	444.59	0.0193	1.1613	424.0	780.5	1204.5	0.6214	0.8630	1.4844	422.6	695.9	1118.5	400
420	449.39	0.0194	1.1061	429.4	775.2	1204.6	0.6272	0.8527	1.4799	427.9	690.8	1118.7	420
440	454.02	0.0195	1.0556	434.6	770.0	1204.6	0.6329	0.8426	1.4755	433.0	685.7	1118.7	440
460	458.50	0.0196	1.0094	439.7	764.9	1204.6	0.6383	0.8330	1.4713	438.0	680.7	1118.7	460
480	462.82	0.0197	0.9670	444.6	759.9	1204.5	0.6436	0.8237	1.4673	442.9	675.7	1118.6	480
500	467.01	0.0197	0.9278	449.4	755.0	1204.4	0.6487	0.8147	1.4634	447.6	671.0	1118.6	500
520	471.07	0.0198	0.8915	454.1	750.1	1204.2	0.6536	0.8060	1.4596	452.2	666.2	1118.4	520
540	475.01	0.0199	0.8578	458.6	745.4	1204.0	0.6584	0.7976	1.4560	456.6	661.7	1118.3	540
560	478.85	0.0200	0.8265	463.0	740.8	1203.8	0.6631	0.7893	1.4524	460.9	657.3	1118.2	560
580	482.58	0.0201	0.7973	467.4	736.1	1203.5	0.6676	0.7813	1.4489	465.2	652.8	1118.0	580
600	486.21	0.0201	0.7698	471.6	731.6	1203.2	0.6720	0.7734	1.4454	469.4	648.3	1117.7	600
620	489.75	0.0202	0.7440	475.7	727.2	1202.9	0.6763	0.7658	1.4421	473.4	644.1	1117.5	620
640	493.21	0.0203	0.7198	479.8	722.7	1202.5	0.6805	0.7584	1.4389	477.4	639.9	1117.3	640
660	496.58	0.0204	0.6971	483.8	718.3	1202.1	0.6846	0.7512	1.4358	481.3	635.7	1117.0	660
680	499.88	0.0204	0.6757	487.7	714.0	1201.7	0.6886	0.7441	1.4327	485.1	631.6	1116.7	680
700	503.10	0.0205	0.6554	491.5	709.7	1201.2	0.6925	0.7371	1.4296	488.8	627.5	1116.3	700
720	506.25	0.0206	0.6362	495.3	705.4	1200.7	0.6963	0.7303	1.4266	492.5	623.5	1116.0	720
740	509.34	0.0207	0.6180	499.0	701.2	1200.2	0.7001	0.7237	1.4237	496.2	619.4	1115.6	740
760	512.36	0.0207	0.6007	502.6	697.1	1199.7	0.7037	0.7172	1.4209	499.7	615.5	1115.2	760
780	515.33	0.0208	0.5843	506.2	692.9	1199.1	0.7073	0.7108	1.4181	503.2	611.6	1114.8	780

800	518.23	0.0209	0.5687	509.7	688.9	1198.6	0.7108	0.7045	1.4153	506.6	607.8	1114.4	800
820	521.08	0.0209	0.5538	513.2	684.4	1198.0	0.7143	0.6983	1.4126	510.0	604.0	1114.0	820
840	523.88	0.0210	0.5396	516.6	680.8	1197.4	0.7177	0.6922	1.4099	513.3	600.3	1113.6	840
860	526.63	0.0211	0.5260	520.0	676.8	1196.8	0.7210	0.6862	1.4072	516.6	596.5	1113.1	860
880	529.33	0.0212	0.5130	523.3	672.8	1196.1	0.7243	0.6803	1.4046	519.9	592.7	1112.6	880
900	531.98	0.0212	0.5006	526.6	668.8	1195.4	0.7275	0.6744	1.4020	523.1	589.0	1112.1	900
920	534.59	0.0213	0.4886	529.8	664.9	1194.7	0.7307	0.6687	1.3995	526.2	585.3	1111.5	920
940	537.16	0.0214	0.4772	533.0	661.0	1194.0	0.7339	0.6631	1.3970	529.3	581.7	1111.0	940
960	539.68	0.0214	0.4663	536.2	657.1	1193.3	0.7370	0.6576	1.3945	532.4	578.1	1110.5	960
980	542.17	0.0215	0.4557	539.3	653.3	1192.6	0.7400	0.6521	1.3921	535.4	574.6	1110.0	980
1000	544.61	0.0216	0.4456	542.4	649.4	1191.8	0.7430	0.6467	1.3897	538.4	571.0	1109.4	1000
1050	550.57	0.0218	0.4218	550.0	639.9	1189.9	0.7504	0.6334	1.3838	545.8	562.2	1108.0	1050
1100	556.31	0.0220	0.4001	557.4	630.4	1187.8	0.7575	0.6205	1.3780	552.9	553.5	1106.4	1100
1150	561.86	0.0221	0.3802	564.6	621.0	1185.6	0.7644	0.6079	1.3723	559.9	544.8	1104.7	1150
1200	567.22	0.0223	0.3619	571.7	611.7	1183.4	0.7711	0.5956	1.3667	566.7	536.3	1103.0	1200
1250	572.42	0.0225	0.3450	578.6	602.4	1181.0	0.7776	0.5836	1.3612	573.4	527.8	1101.2	1250
1300	577.46	0.0227	0.3293	585.4	593.2	1178.6	0.7840	0.5719	1.3559	580.0	519.4	1099.4	1300
1350	582.35	0.0229	0.3148	592.1	584.0	1176.1	0.7902	0.5604	1.3506	586.4	511.1	1097.5	1350
1400	587.10	0.0231	0.3012	598.7	574.7	1173.4	0.7963	0.5491	1.3454	592.7	502.7	1095.4	1400
1450	591.73	0.0233	0.2884	605.2	565.5	1170.7	0.8023	0.5379	1.3402	599.0	494.3	1093.3	1450
1500	596.23	0.0235	0.2765	611.6	556.3	1167.9	0.8082	0.5269	1.3351	605.1	486.1	1091.2	1500
1600	604.90	0.0239	0.2548	624.1	538.0	1162.1	0.8196	0.5053	1.3249	617.0	469.7	1086.7	1600
1700	613.15	0.0243	0.2354	636.3	519.6	1155.9	0.8306	0.4843	1.3149	628.7	453.1	1081.8	1700
1800	621.03	0.0247	0.2179	648.3	501.1	1149.4	0.8412	0.4637	1.3049	640.1	436.7	1076.8	1800
1900	628.58	0.0252	0.2021	660.1	482.4	1142.4	0.8516	0.4433	1.2949	651.2	420.2	1071.4	1900
2000	635.82	0.0257	0.1878	671.7	463.4	1135.1	0.8619	0.4230	1.2849	662.2	403.4	1065.6	2000
2100	642.77	0.0262	0.1746	683.3	444.1	1127.4	0.8721	0.4027	1.2748	673.1	386.5	1059.6	2100
2200	649.46	0.0268	0.1625	694.8	424.4	1119.2	0.8820	0.3826	1.2646	683.9	369.2	1053.1	2200
2300	655.91	0.0274	0.1513	706.5	403.9	1110.4	0.8921	0.3621	1.2541	694.8	351.2	1046.0	2300
2400	662.12	0.0280	0.1407	718.4	382.7	1101.1	0.9023	0.3411	1.2434	706.0	332.0	1038.6	2400
2500	668.13	0.0287	0.1307	730.6	360.5	1091.1	0.9126	0.3197	1.2322	717.3	313.3	1030.6	2500
2600	673.94	0.0295	0.1213	743.0	337.2	1080.2	0.9232	0.2973	1.2205	728.8	293.1	1021.9	2600
2700	679.55	0.0305	0.1123	756.2	312.1	1068.3	0.9342	0.2740	1.2082	741.0	271.3	1012.3	2700
2800	684.99	0.0315	0.1035	770.1	284.7	1054.8	0.9459	0.2487	1.1946	753.8	247.4	1001.2	2800
2900	690.26	0.0329	0.0947	785.4	253.6	1039.0	0.9587	0.2205	1.1792	767.7	220.5	988.2	2900
3000	695.36	0.0346	0.0858	802.5	217.8	1020.3	0.9731	0.1885	1.1615	783.4	189.3	972.7	3000
3100	700.31	0.0371	0.0753	825.0	168.1	994.1	0.9919	0.1449	1.1368	803.7	146.2	949.9	3100
3200	705.11	0.0444	0.0580	872.4	62.0	934.4	1.0320	0.0532	1.0852	846.0	52.4	898.4	3200
3206.2	705.40	0.0503	0.0503	902.7	0	902.7	1.0580	0	1.0580	872.9	0.0	872.9	3206.2

TABLE VIII.—PROPERTIES
(English

Absolute pressure, psia (Saturation temperature)		Saturated liquid	Saturated vapor	220 F	240 F	260 F	280 F	300 F	320 F	340 F	360 F	380 F
	v	0.02	333.6	404.5	416.5	428.4	440.4	452.3	464.2	476.2	488.1	500.0
1	*h*	69.7	1106.0	1159.5	1168.5	1177.6	1186.7	1195.8	1204.9	1214.1	1223.3	1232.5
(101.74)	*s*	0.1326	1.9782	2.0647	2.0779	2.0907	2.1031	2.1153	2.1271	2.1387	2.1501	2.1612
	v	0.02	26.80	27.15	28.00	28.85	29.70	30.53	31.37	32.20	33.03	33.85
14.696	*h*	180.1	1150.4	1154.4	1164.2	1173.8	1183.3	1192.8	1202.3	1211.7	1221.1	1230.5
(212.00)	*s*	0.3120	1.7566	1.7624	1.7766	1.7902	1.8033	1.8160	1.8283	1.8402	1.8518	1.8631
	v	0.02	20.09	......	20.48	21.11	21.74	22.36	22.98	23.60	24.21	24.82
20	*h*	196.2	1156.3	......	1162.3	1172.2	1182.0	1191.6	1201.2	1210.8	1220.3	1229.7
(227.96)	*s*	0.3356	1.7319	......	1.7405	1.7545	1.7679	1.7808	1.7932	1.8053	1.8170	1.8285
	v	0.017	10.498	......	......	......	10.711	11.040	11.364	11.684	12.001	12.315
40	*h*	236.0	1169.7	......	......	......	1176.5	1186.8	1197.0	1207.0	1216.9	1226.7
(267.25)	*s*	0.3919	1.6763	......	......	......	1.6855	1.6994	1.7126	1.7252	1.7375	1.7493
	v	0.017	7.175	......	......	......	......	7.259	7.486	7.708	7.927	8.143
60	*h*	262.1	1177.6	......	......	......	......	1181.6	1192.5	1203.0	1213.4	1223.6
(292.71)	*s*	0.4270	1.6438	......	......	......	......	1.6492	1.6633	1.6766	1.6894	1.7017
	v	0.018	5.472	......	......	......	......	......	5.543	5.718	5.888	6.055
80	*h*	282.0	1183.1	......	......	......	......	......	1187.6	1198.8	1209.7	1220.3
(312.03)	*s*	0.4531	1.6207	......	......	......	......	......	1.6266	1.6407	1.6541	1.6669
	v	0.018	4.432	......	......	......	......	......	......	4.521	4.663	4.801
100	*h*	298.4	1187.2	......	......	......	......	......	......	1194.3	1205.7	1216.8
(327.81)	*s*	0.4740	1.6026	......	......	......	......	......	......	1.6117	1.6258	1.6391
	v	0.018	3.728	......	......	......	......	......	......	......	3.844	3.964
120	*h*	312.4	1190.4	......	......	......	......	......	......	......	1201.6	1213.2
(341.25)	*s*	0.4916	1.5878	......	......	......	......	......	......	......	1.6017	1.6156
	v	0.018	3.220	......	......	......	......	......	......	......	3.258	3.365
140	*h*	324.8	1193.0	......	......	......	......	......	......	......	1197.3	1209.4
(353.02)	*s*	0.5069	1.5751	......	......	......	......	......	......	......	1.5804	1.5950
	v	0.018	2.834	......	......	......	......	......	......	......	......	2.914
160	*h*	335.9	1195.1	......	......	......	......	......	......	......	......	1205.5
(363.53)	*s*	0.5204	1.5640	......	......	......	......	......	......	......	......	1.5766
	v	0.018	2.532	......	......	......	......	......	......	......	......	2.563
180	*h*	346.0	1196.9	......	......	......	......	......	......	......	......	1201.4
(373.06)	*s*	0.5325	1.5542	......	......	......	......	......	......	......	......	1.5596
	v	0.018	2.288	......	......	......	......	......	......	......	......	......
200	*h*	355.4	1198.4	......	......	......	......	......	......	......	......	......
(381.79)	*s*	0.5435	1.5453	......	......	......	......	......	......	......	......	......
	v	0.019	2.087	......	......	......	......	......	......	......	......	......
220	*h*	364.0	1199.6	......	......	......	......	......	......	......	......	......
(389.86)	*s*	0.5537	1.5372	......	......	......	......	......	......	......	......	......
	v	0.0186	1.9183	......	......	......	......	......	......	......	......	......
240	*h*	372.1	1200.6	......	......	......	......	......	......	......	......	......
(397.37)	*s*	0.5631	1.5298	......	......	......	......	......	......	......	......	......
	v	0.0187	1.7748	......	......	......	......	......	......	......	......	......
260	*h*	379.8	1201.5	......	......	......	......	......	......	......	......	......
(404.42)	*s*	0.5719	1.5229	......	......	......	......	......	......	......	......	......
	v	0.0188	1.6511	......	......	......	......	......	......	......	......	......
280	*h*	387.0	1202.3	......	......	......	......	......	......	......	......	......
(411.05)	*s*	0.5801	1.5164	......	......	......	......	......	......	......	......	......
	v	0.0189	1.5433	......	......	......	......	......	......	......	......	......
300	*h*	393.8	1202.8	......	......	......	......	......	......	......	......	......
(417.33)	*s*	0.5879	1.5104	......	......	......	......	......	......	......	......	......
	v	0.0191	1.3260	......	......	......	......	......	......	......	......	......
350	*h*	409.7	1203.9	......	......	......	......	......	......	......	......	......
(431.72)	*s*	0.6056	1.4966	......	......	......	......	......	......	......	......	......

OF SUPERHEATED STEAM
units)

400 F	420 F	440 F	460 F	480 F	500 F	600 F	700 F	800 F	900 F	1000 F		Absolute pressure, psia (Saturation temperature)
512.0	523.9	535.8	547.7	559.7	571.6	631.2	690.8	750.4	809.9	869.5	v	
1241.7	1251.0	1260.3	1269.6	1278.9	1288.3	1335.7	1383.8	1432.8	1482.7	1533.5	h	**1**
2.1720	2.1827	2.1931	2.2034	2.2134	2.2233	2.2702	2.3137	2.3542	2.3923	2.4283	s	(101.74)
34.68	35.50	36.32	37.14	37.96	38.78	42.86	46.94	51.00	55.07	59.13	v	
1239.9	1249.3	1258.8	1268.2	1277.6	1287.1	1334.8	1383.2	1432.3	1482.3	1533.1	h	**14.696**
1.8743	1.8850	1.8956	1.9060	1.9162	1.9261	1.9734	2.0170	2.0576	2.0958	2.1319	s	(212.00)
25.43	26.04	26.65	27.25	27.86	28.46	31.47	34.47	37.46	40.45	43.44	v	
1239.2	1248.7	1258.2	1267.6	1277.1	1286.6	1334.4	1382.9	1432.1	1482.1	1533.0	h	**20**
1.8396	1.8505	1.8612	1.8716	1.8818	1.8918	1.9392	1.9829	2.0235	2.0618	2.0978	s	(227.96)
12.628	12.938	13.247	13.555	13.826	14.168	15.688	17.198	18.702	20.20	21.70	v	
1236.5	1246.2	1255.9	1265.5	1275.2	1284.8	1333.1	1381.9	1431.3	1481.4	1532.4	h	**40**
1.7608	1.7719	1.7828	1.7934	1.8038	1.8140	1.8619	1.9058	1.9467	1.9850	2.0212	s	(267.25)
8.357	8.569	8.779	8.988	9.196	9.403	10.427	11.441	12.449	13.452	14.454	v	
1233.6	1243.6	1253.5	1263.4	1273.2	1283.0	1331.8	1380.9	1430.5	1480.8	1531.9	h	**60**
1.7135	1.7250	1.7361	1.7470	1.7575	1.7678	1.8162	1.8605	1.9015	1.9400	1.9762	s	(292.71)
6.220	6.383	6.544	6.704	6.862	7.020	7.797	8.562	9.322	10.077	10.830	v	
1230.7	1240.9	1251.1	1261.1	1271.1	1281.1	1330.5	1379.9	1429.7	1480.1	1531.3	h	**80**
1.6791	1.6909	1.7023	1.7134	1.7242	1.7346	1.7836	1.8281	1.8694	1.9079	1.9442	s	(312.03)
4.937	5.071	5.202	5.333	5.462	5.589	6.218	6.835	7.446	8.052	8.656	v	
1227.6	1238.1	1248.6	1258.8	1269.0	1279.1	1329.1	1378.9	1428.9	1479.5	1530.8	h	**100**
1.6518	1.6639	1.6756	1.6869	1.6979	1.7085	1.7581	1.8029	1.8443	1.8829	1.9193	s	(327.81)
4.081	4.195	4.307	4.418	4.527	4.636	5.165	5.683	6.195	6.702	7.207	v	
1224.4	1235.3	1246.0	1256.5	1266.9	1277.2	1327.7	1377.8	1428.1	1478.8	1530.2	h	**120**
1.6287	1.6413	1.6533	1.6649	1.6760	1.6869	1.7370	1.7822	1.8237	1.8625	1.8990	s	(341.25)
3.468	3.569	3.667	3.764	3.860	3.954	4.413	4.861	5.301	5.738	6.172	v	
1221.1	1232.3	1243.3	1254.1	1264.7	1275.2	1326.4	1376.8	1427.3	1478.2	1529.7	h	**140**
1.6087	1.6217	1.6340	1.6458	1.6573	1.6683	1.7190	1.7645	1.8063	1.8451	1.8817	s	(353.02)
3.008	3.098	3.187	3.273	3.359	3.443	3.849	4.244	4.631	5.015	5.396	v	
1217.6	1229.3	1240.6	1251.6	1262.4	1273.1	1325.0	1375.7	1426.4	1477.5	1529.1	h	**160**
1.5908	1.6042	1.6169	1.6291	1.6407	1.6519	1.7033	1.7491	1.7911	1.8301	1.8667	s	(363.53)
2.649	2.732	2.813	2.891	2.969	3.044	3.411	3.764	4.110	4.452	4.792	v	
1214.0	1226.1	1237.8	1249.1	1260.2	1271.0	1323.5	1374.7	1425.6	1476.8	1528.6	h	**180**
1.5745	1.5884	1.6015	1.6139	1.6258	1.6373	1.6894	1.7355	1.7776	1.8167	1.8534	s	(373.06)
2.361	2.438	2.513	2.585	2.656	2.726	3.060	3.380	3.693	4.002	4.309	v	
1210.3	1222.9	1234.9	1246.5	1257.8	1268.9	1322.1	1373.6	1424.8	1476.2	1528.0	h	**200**
1.5594	1.5738	1.5873	1.6001	1.6123	1.6240	1.6767	1.7232	1.7655	1.8048	1.8415	s	(381.79)
2.125	2.198	2.267	2.335	2.400	2.465	2.772	3.066	3.352	3.634	3.913	v	
1206.5	1219.5	1231.9	1243.8	1255.4	1266.7	1320.7	1372.6	1424.0	1475.5	1527.5	h	**220**
1.5453	1.5603	1.5742	1.5874	1.5998	1.6117	1.6652	1.7120	1.7545	1.7939	1.8308	s	(389.86)
1.9276	1.9964	2.062	2.126	2.187	2.247	2.533	2.804	3.068	3.327	3.584	v	
1202.5	1216.0	1228.8	1241.1	1253.0	1264.5	1319.2	1371.5	1423.2	1474.8	1526.9	h	**240**
1.5319	1.5475	1.5619	1.5754	1.5882	1.6003	1.6546	1.7017	1.7444	1.7839	1.8209	s	(397.37)
......	1.8257	1.8882	1.9483	2.006	2.063	2.330	2.582	2.827	3.067	3.305	v	
......	1212.4	1225.7	1238.3	1250.5	1262.3	1317.7	1370.4	1422.3	1474.2	1526.3	h	**260**
......	1.5354	1.5503	1.5642	1.5773	1.5897	1.6447	1.6922	1.7352	1.7748	1.8118	s	(404.42)
......	1.6789	1.7388	1.7960	1.8512	1.9047	2.156	2.392	2.621	2.845	3.066	v	
......	1208.7	1222.4	1235.4	1247.9	1260.0	1316.2	1369.4	1421.5	1473.5	1525.8	h	**280**
......	1.5238	1.5392	1.5536	1.5670	1.5796	1.6354	1.6834	1.7265	1.7662	1.8033	s	(411.05)
......	1.5513	1.6090	1.6638	1.7165	1.7675	2.005	2.227	2.442	2.652	2.859	v	
......	1204.8	1219.1	1232.5	1245.3	1257.6	1314.7	1368.3	1420.6	1472.8	1525.2	h	**300**
......	1.5126	1.5286	1.5434	1.5572	1.5701	1.6268	1.6751	1.7184	1.7582	1.7954	s	(417.33)
......	......	1.3478	1.3984	1.4463	1.4923	1.7036	1.8980	2.084	2.266	2.455	v	
......	......	1210.3	1224.8	1238.5	1251.5	1310.9	1365.5	1418.5	1471.1	1523.8	h	**350**
......	......	1.5037	1.5197	1.5344	1.5481	1.6070	1.6563	1.7002	1.7403	1.7777	s	(431.72)

TABLE VIII.—PROPERTIES OF

Absolute pressure, psia (Saturation temperature)		Saturated liquid	Saturated vapor	460 F	480 F	500 F	520 F	540 F	560 F	580 F	600 F	620 F
	v	0.0193	1.1613	1.1978	1.2426	1.2851	1.3259	1.3652	1.4034	1.4406	1.4770	1.5128
400	h	424.0	1204.5	1216.5	1231.3	1245.1	1258.3	1271.0	1283.3	1295.2	1306.9	1318.3
(444.59)	s	0.6214	1.4844	1.4977	1.5135	1.5281	1.5417	1.5546	1.5667	1.5783	1.5894	1.6001
	v	0.0195	1.0320	1.0403	1.0830	1.1231	1.1612	1.1977	1.2330	1.2672	1.3005	1.3332
450	h	437.2	1204.6	1207.7	1223.6	1238.4	1252.3	1265.6	1278.4	1290.8	1302.8	1314.6
(456.28)	s	0.6356	1.4734	1.4768	1.4939	1.5095	1.5239	1.5373	1.5500	1.5620	1.5735	1.5845
	v	0.0197	0.9278	.....	0.9543	0.9927	1.0289	1.0633	1.0963	1.1282	1.1591	1.1893
500	h	449.4	1204.4	.....	1215.4	1231.3	1246.0	1260.0	1273.4	1286.2	1298.6	1310.7
(467.01)	s	0.6487	1.4634	.....	1.4752	1.4919	1.5072	1.5213	1.5345	1.5470	1.5588	1.5701
	v	0.0199	0.8419	.....	0.8478	0.8852	0.9199	0.9528	0.9840	1.0141	1.0431	1.0714
550	h	460.8	1203.9	.....	1206.7	1223.7	1239.4	1254.2	1268.1	1281.5	1294.3	1306.8
(476.94)	s	0.6608	1.4542	.....	1.4571	1.4751	1.4913	1.5061	1.5199	1.5329	1.5451	1.5568
	v	0.0201	0.7698	.....	.....	0.7947	0.8286	0.8602	0.8901	0.9187	0.9463	0.9729
600	h	471.6	1203.2	.....	.....	1215.7	1232.5	1248.1	1262.7	1276.6	1289.9	1302.7
(486.21)	s	0.6720	1.4454	.....	.....	1.4586	1.4759	1.4916	1.5061	1.5196	1.5323	1.5443
	v	0.0203	0.7083	.....	.....	0.7173	0.7506	0.7814	0.8103	0.8378	0.8641	0.8894
650	h	481.8	1202.3	.....	.....	1207.2	1225.2	1241.7	1257.0	1271.5	1285.3	1298.6
(494.90)	s	0.6825	1.4373	.....	.....	1.4424	1.4610	1.4776	1.4928	1.5069	1.5201	1.5325
	v	0.0205	0.6554	.....	.....	.....	0.6832	0.7134	0.7416	0.7681	0.7934	0.8177
700	h	491.5	1201.2	.....	.....	.....	1217.5	1235.0	1251.1	1266.3	1280.6	1294.3
(503.10)	s	0.6925	1.4296	.....	.....	.....	1.4463	1.4640	1.4800	1.4947	1.5084	1.5212
	v	0.0209	0.5687	.....	.....	.....	0.5715	0.6015	0.6288	0.6541	0.6779	0.7006
800	h	509.7	1198.6	.....	.....	.....	1200.5	1220.5	1238.6	1255.2	1270.7	1285.4
(518.23)	s	0.7108	1.4153	.....	.....	.....	1.4172	1.4375	1.4553	1.4715	1.4863	1.5000
	v	0.0212	0.5006	.....	.....	.....	.....	0.5124	0.5396	0.5644	0.5873	0.6080
900	h	526.6	1195.4	.....	.....	.....	.....	1204.4	1224.9	1243.2	1260.1	1275.9
(531.98)	s	0.7275	1.4020	.....	.....	.....	.....	1.4111	1.4313	1.4492	1.4653	1.4800
	v	0.0216	0.4456	.....	.....	.....	.....	.....	0.4668	0.4915	0.5140	0.5350
1000	h	542.4	1191.8	.....	.....	.....	.....	.....	1209.7	1230.3	1248.8	1265.9
(544.61)	s	0.7430	1.3897	.....	.....	.....	.....	.....	1.4073	1.4364	1.4450	1.4610
	v	0.0223	0.3619	.....	.....	.....	.....	.....	.....	0.3786	0.4016	0.4222
1200	h	571.7	1183.4	.....	.....	.....	.....	.....	.....	1200.4	1223.5	1243.9
(567.22)	s	0.7711	1.3667	.....	.....	.....	.....	.....	.....	1.3832	1.4052	1.4243
	v	0.0231	0.3012	.....	.....	.....	.....	.....	.....	.....	0.3174	0.3390
1400	h	598.7	1173.4	.....	.....	.....	.....	.....	.....	.....	1193.0	1218.4
(587.10)	s	0.7963	1.3454	.....	.....	.....	.....	.....	.....	.....	1.3639	1.3877
	v	0.0239	0.2548	.....	.....	.....	.....	.....	.....	.....	.....	0.2733
1600	h	624.1	1162.1	.....	.....	.....	.....	.....	.....	.....	.....	1187.8
(604.90)	s	0.8196	1.3249	.....	.....	.....	.....	.....	.....	.....	.....	1.3489
	v	0.0247	0.2179	.....	.....	.....	.....	.....	.....	.....	.....	.....
1800	h	648.3	1149.4	.....	.....	.....	.....	.....	.....	.....	.....	.....
(621.03)	s	0.8412	1.3049	.....	.....	.....	.....	.....	.....	.....	.....	.....
	v	0.0257	0.1878	.....	.....	.....	.....	.....	.....	.....	.....	.....
2000	h	671.7	1135.1	.....	.....	.....	.....	.....	.....	.....	.....	.....
(635.82)	s	0.8619	1.2849	.....	.....	.....	.....	.....	.....	.....	.....	.....
	v	0.0287	0.1307	.....	.....	.....	.....	.....	.....	.....	.....	.....
2500	h	730.6	1091.1	.....	.....	.....	.....	.....	.....	.....	.....	.....
(668.13)	s	0.9126	1.2322	.....	.....	.....	.....	.....	.....	.....	.....	.....
	v	0.0346	0.0858	.....	.....	.....	.....	.....	.....	.....	.....	.....
3000	h	802.5	1020.3	.....	.....	.....	.....	.....	.....	.....	.....	.....
(695.36)	s	0.9731	1.1615	.....	.....	.....	.....	.....	.....	.....	.....	.....
	v	0.0503	0.0503	.....	.....	.....	.....	.....	.....	.....	.....	.....
3206.2	h	902.7	902.7	.....	.....	.....	.....	.....	.....	.....	.....	.....
(705.40)	s	1.0580	1.0580	.....	.....	.....	.....	.....	.....	.....	.....	.....

Superheated Steam.—*(Concluded)*

640 F	660 F	680 F	700 F	720 F	760 F	800 F	850 F	900 F	950 F	1000 F		Absolute pressure, psia (Saturation temperature)
1.5480	1.5827	1.6169	1.6508	1.6844	1.7507	1.8161	1.8968	1.9767	2.056	2.134	v	
1329.6	1240.8	1351.8	1362.7	1373.6	1395.1	1416.4	1442.9	1469.4	1495.9	1522.4	h	**400**
1.6105	1.6205	1.6303	1.6398	1.6491	1.6670	1.6842	1.7049	1.7247	1.7438	1.7623	s	(444.59)
1.3652	1.3967	1.4278	1.4584	1.4888	1.5486	1.6074	1.6800	1.7516	1.8225	1.8928	v	
1326.4	1337.5	1348.8	1359.9	1370.9	1392.7	1414.3	1441.0	1467.7	1494.3	1521.0	h	**450**
1.5951	1.6054	1.6153	1.6250	1.6344	1.6525	1.6699	1.6908	1.7108	1.7300	1.7486	s	(456.28)
1.2188	1.2478	1.2763	1.3044	1.3322	1.3868	1.4405	1.5065	1.5715	1.6358	1.6996	v	
1322.6	1334.2	1345.7	1357.0	1368.2	1390.3	1412.1	1439.1	1466.0	1492.8	1519.6	h	**500**
1.5810	1.5915	1.6016	1.6115	1.6210	1.6395	1.6571	1.6781	1.6982	1.7176	1.7633	s	(467.01)
1.0989	1.1259	1.1523	1.1783	1.2040	1.2544	1.3038	1.3644	1.4241	1.4830	1.5414	v	
1318.9	1330.8	1342.5	1354.0	1365.4	1387.8	1409.9	1437.2	1464.3	1491.2	1518.2	h	**550**
1.5680	1.5787	1.5890	1.5991	1.6088	1.6274	1.6452	1.6665	1.6868	1.7063	1.7250	s	(476.94)
0.9988	1.0241	1.0489	1.0732	1.0971	1.1440	1.1899	1.2460	1.3013	1.3557	1.4096	v	
1315.2	1327.4	1339.3	1351.1	1362.6	1385.3	1407.7	1435.2	1462.5	1489.7	1516.7	h	**600**
1.5558	1.5667	1.5773	1.5875	1.5974	1.6163	1.6343	1.6558	1.6762	1.6958	1.7147	s	(486.21)
0.9140	0.9379	0.9612	0.9841	1.0065	1.0505	1.0934	1.1459	1.1973	1.2479	1.2981	v	
1311.4	1323.9	1336.1	1348.0	1359.8	1382.8	1405.4	1433.3	1460.8	1488.1	1515.3	h	**650**
1.5443	1.5555	1.5663	1.5767	1.5867	1.6060	1.6242	1.6458	1.6665	1.6862	1.7052	s	(494.90)
0.8411	0.8639	0.8860	0.9077	0.9289	0.9704	1.0108	1.0600	1.1082	1.1556	1.2024	v	
1307.5	1320.3	1332.8	1345.0	1356.9	1380.3	1403.2	1431.3	1459.0	1486.5	1513.9	h	**700**
1.5333	1.5449	1.5559	1.5665	1.5767	1.5962	1.6147	1.6366	1.6573	1.6771	1.6963	s	(503.10)
0.7223	0.7433	0.7635	0.7833	0.8026	0.8400	0.8763	......	0.9633	......	1.0470	v	
1299.4	1312.9	1325.9	1338.6	1351.0	1375.2	1398.6	......	1455.4	......	1511.0	h	**800**
1.5129	1.5250	1.5366	1.5476	1.5582	1.5783	1.5972	......	1.6407	......	1.6801	s	(518.23)
0.6294	0.6491	0.6680	0.6863	0.7041	0.7385	0.7716	......	0.8506	......	0.9262	v	
1290.9	1305.1	1318.8	1332.1	1345.0	1369.9	1393.9	......	1451.8	......	1508.1	h	**900**
1.4938	1.5066	1.5187	1.5303	1.5413	1.5620	1.5814	......	1.6275	......	1.6656	s	(531.98)
0.5546	0.5733	0.5912	0.6084	0.6251	0.6571	0.6878	......	0.7604	......	0.8294	v	
1281.9	1297.0	1311.4	1325.3	1338.7	1364.4	1389.2	......	1448.2	......	1505.1	h	**1000**
1.4757	1.4893	1.5021	1.5141	1.5256	1.5470	1.5670	......	1.6121	......	1.6525	s	(544.61)
0.4410	0.4586	0.4752	0.4909	0.5060	0.5347	0.5617	......	0.6250	......	0.6843	v	
1262.4	1279.6	1295.7	1311.0	1325.6	1353.2	1379.3	......	1440.7	......	1499.2	h	**1200**
1.4413	1.4568	1.4710	1.4843	1.4968	1.5198	1.5409	......	1.5879	......	1.6293	s	(567.22)
0.3580	0.3753	0.3912	0.4062	0.4203	0.4468	0.4714	......	0.5281	0.5547	0.5805	v	
1240.4	1260.3	1278.5	1295.5	1311.5	1341.3	1369.1	......	1433.1	1463.5	1493.2	h	**1400**
1.4079	1.4258	1.4419	1.4567	1.4704	1.4953	1.5177	......	1.5666	1.5886	1.6093	s	(587.10)
0.2936	0.3112	0.3271	0.3417	0.3553	0.3804	0.4034	......	0.4553	0.4794	0.5027	v	
1215.2	1238.7	1259.6	1278.7	1296.4	1328.8	1358.4	......	1425.3	1456.6	1487.0	h	**1600**
1.3741	1.3952	1.4137	1.4303	1.4455	1.4725	1.4964	......	1.5476	1.5702	1.5914	s	(604.90)
0.2407	0.2597	0.2760	0.2907	0.3041	0.3284	0.3502	......	0.3986	0.4208	0.4421	v	
1185.1	1214.0	1238.5	1260.3	1280.1	1315.5	1347.2	......	1417.4	1449.7	1480.8	h	**1800**
1.3377	1.3638	1.3855	1.4044	1.4213	1.4509	1.4765	......	1.5301	1.5534	1.5752	s	(621.03)
0.1936	0.2161	0.2337	0.2489	0.2624	0.2863	0.3074	......	0.3532	0.3738	0.3935	v	
1145.6	1184.9	1214.8	1240.0	1262.3	1301.4	1335.5	......	1409.2	1442.5	1474.5	h	**2000**
1.2945	1.3300	1.3564	1.3783	1.3974	1.4300	1.4576	......	1.5139	1.5380	1.5603	s	(635.82)
......	......	0.1484	0.1686	0.1841	0.2090	0.2294	......	0.2710	0.2891	0.3061	v	
......	......	1132.3	1176.8	1210.0	1261.8	1303.6	......	1387.8	1424.2	1458.4	h	**2500**
......	......	1.2687	1.3073	1.3357	1.3789	1.4127	......	1.4772	1.5034	1.5273	s	(668.13)
......	......	......	0.0984	0.1251	0.1548	0.1760	......	0.2159	......	0.2476	v	
......	......	......	1060.7	1136.1	1213.8	1267.2	......	1365.0	......	1441.8	h	**3000**
......	......	......	1.1966	1.2611	1.3259	1.3690	......	1.4439	......	1.4984	s	(695.36)
......	......	......	......	0.1020	0.1363	0.1583	......	0.1981	......	0.2288	v	
......	......	......	......	1090.7	1190.6	1250.5	......	1355.2	......	1434.7	h	**3206.2**
......	......	......	......	1.2190	1.3024	1.3508	......	1.4309	......	1.4874	s	(705.40)

barometer, must be added to it, and the result used when consulting the tables for the corresponding properties of the steam. (In case the barometer reading is not given, 14.7 lb may be added to the gage reading in order to find the absolute pressure.)

PROBLEMS

1. If, in the only steam tables available, the entropy columns were obliterated, how would you determine the entropy of vaporization at 100 psia? Show calculations.

2. How much external work is done when 1 lb of water is heated from 32 F through the vapor phase to dry, saturated steam at atmospheric pressure?

3. A pound of saturated water at 300 psia is turned into dry and saturated steam by the addition of a certain amount of heat. (*a*) What is that required amount of heat? (*b*) What is the change in entropy? (*c*) What is the specific volume of the steam? Compute the external work done during the boiling process.

4. An open-ended, vertical cylinder 10 in. in diam and 22 in. high is filled with dry saturated steam. The pressure of the steam is just sufficient to support a piston carrying a load of 15,394 lb (including its own weight). What are the weight, temperature, and pressure of the steam in the cylinder? The barometer reads 28.53 in. Hg.

5. Ten gallons of water per minute is to be heated from 55 F to 212 F by passing through a coil surrounded by steam at a pressure of 120 psi which has been superheated 150 degrees. How much steam is required per minute if the condensed steam is not reduced in temperature?

6. One hundred gallons of water per minute are to be heated from 60 F to 200 F in a closed feed-water heater (water passes through coils surrounded by steam). The exhaust steam passing around the heater coil is at a pressure of 15 psia and contains 10 per cent moisture. How much steam is required per hour? (Steam is condensed but does not drop in temperature.)

7. One hundred cubic feet of air per minute at atmospheric pressure and 60 F are compressed isothermally to a pressure of 90 psi. The air then passes around a series of steam coils and is heated to 150 F at constant pressure. Dry steam enters the coil at a pressure of 50 psi and leaves at atmospheric pressure and 96 per cent quality. How many pounds of steam per minute will be required to heat the air?

8. At a certain stage in the process in an engine, steam at a pressure of 56 psia has a quality of 72 per cent. What is its internal energy?

9. (*a*) Find the external work of evaporation per pound, when 1 cu ft of water at a temperature of 400.95 F, weighing 53.62 lb, is converted into steam at a pressure of 250 psia. (*b*) Find the internal energy of evaporation per pound, under the foregoing conditions. (*c*) Find the latent heat of evaporation for the whole cubic foot of water.

10. How much heat is required to change 50 lb of water at 70 F into steam at a pressure of 149.3 psi and containing 2 per cent moisture? Barometer, 29.93 in. Hg.

11. How many Btu must be added to 1,000 lb of water at 90 F to convert it into steam at a pressure of 200 psia and a temperature of 400 F?

12. If water at 337.9 F and a pressure of 100 psi is allowed to pass through a valve into the atmosphere, what per cent of the water will be changed to steam?

13. How many pounds of water will 10 lb of dry steam heat from 50 F to 150 F, if the steam pressure is 100 psi?

14. If 10 lb of steam at a pressure of 100 psi raises 93 lb of water from 50 F to 140 F, what per cent of moisture is in the steam, radiation being zero?

15. A tank contained 400 lb of water at a temperature of 50 F. Into this water, steam at a pressure of 125 psia was admitted until the temperature of the water and condensed steam in the tank became 100 F. The weight of water in the tank was then 418.5 lb. Determine the quality of the steam.

16. Water amounting to 100 lb per min is to be heated from 55 F to 200 F by blowing into it a jet of steam at a pressure of 100 psia, 90 per cent dry. What is the minimum amount of steam required per hour?

17. At a pressure of 100 psia, 10 lb of wet steam occupy a volume of 25 cu ft. What is the quality of the steam? One pound of water occupies a volume of 0.017 cu ft.

18. A pound of steam and water occupies 3 cu ft at a pressure of 110 psia. What is the quality of the steam?

19. One pound of steam containing 2 per cent moisture expands adiabatically from a pressure of 150 psia to a pressure of $1\frac{1}{2}$ in. Hg abs. (*a*) Find from the Mollier diagram the quality at the end of the expansion. (*b*) Check this answer by calculating the quality.

20. Represent diagrammatically the formation of steam from 1 lb of water at 32 F and a pressure of 100 psia into steam at a pressure of 100 psia and 400 F. Use temperatures as ordinates and heat units as abscissas, and place values on the diagram.

21. Plot a curve using temperatures as ordinates and enthalpies as abscissas to illustrate the process of changing 1 lb of ice at 20 F into superheated steam at 534 F under a constant pressure of 205 psia. (Consider the enthalpy of water at 32 F to be zero.) Indicate on the diagram the values at all important points.

CHAPTER IV

CALORIMETERS AND MECHANICAL MIXTURES

46. Calorimeters.—As has already been seen, steam may be either wet, dry and saturated, or superheated. If it is wet, the quality (see Art. 34), or per cent, of dry saturated steam in the sample may be determined by means of an instrument called a *steam calorimeter.* The two types of these calorimeters in general use are:

1. Throttling or superheating calorimeters.
2. Separating calorimeters.

There are several makes of each type, but it will suffice to describe only one or two of each.

In order to obtain a fair sample of the steam that is to be tested, the A.S.M.E. recommend the use of a *sampling nozzle,* or *calorimeter nipple.* This collecting nipple is a piece of $\frac{1}{2}$-in. standard-weight black-iron pipe extending nearly across the steam main, as shown in Fig. 18, with a cap on the end and a series of not less than twenty $\frac{1}{8}$-in. holes along and around its cylindrical surface. As the steam to be tested must enter the calorimeter through this nipple, a fair sample of the steam is ensured. The sampling nozzle should be inserted in the steam main at a point where the entrained moisture is likely to be most thoroughly mixed.

It is necessary that both the calorimeter and the connection to it be well insulated in order to prevent loss of heat by radiation and consequent condensation of steam and increase in the moisture content in the sample, so that it has not the same quality as when drawn from the main. The connection from the sampling nozzle to the calorimeter should be as short as possible for the same reason.

47. Throttling or Superheating Calorimeter.—This type of calorimeter was invented by Professor C. H. Peabody and is the form most commonly used in engineering practice.

The principle of its operation is as follows: A pound of dry saturated steam at a high pressure contains more heat than a pound of dry saturated steam at a lower pressure. When steam at a high pressure flows through an orifice into a region of lower pressure, some of the heat of this steam is given up. If this drop in pressure takes place under conditions such that no external work is done, this heat will all be reabsorbed by the steam itself and be dissipated in doing internal work and raising the temperature of the steam. If this steam contained some moisture at the higher pressure, part of the heat liberated when the pressure is lowered will go to evaporating this moisture, and the excess will go to superheating the steam.

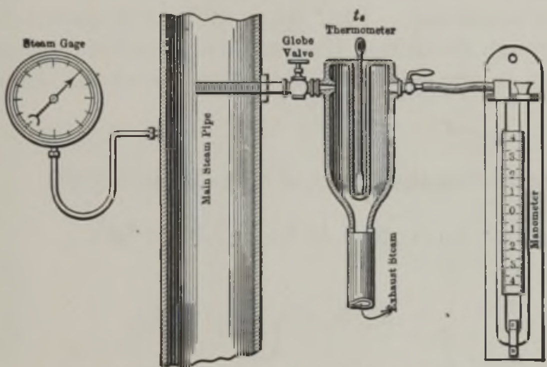


Fig. 17.—Carpenter's throttling calorimeter.

Let x = the quality of the steam;

t_1 = the temperature of the wet steam before passing through the orifice;

p_1 = the absolute pressure of the wet steam in the main;

t_2 = the temperature corresponding to the absolute pressure on the low-pressure side of the orifice;

t_s = the actual temperature of the steam on the low-pressure side of the orifice as shown by the thermometer;

h_{f1} and h_{fg1} = enthalpy of liquid and enthalpy of vaporization corresponding to the temperature t_1, or the absolute pressure p_1;

h_{f2} and h_{fg2} = enthalpy of liquid and enthalpy of vaporization corresponding to the pressure on the low-pressure side of the orifice;

h_1 = enthalpy in the steam before entering the orifice, *i.e.*, at a temperature t_1, or an absolute pressure p_1;

h_2 = enthalpy in the steam after leaving the orifice.

The heat contained in one pound of the mixture of steam and water at temperature t_1, or pressure p_1, would be

$$h_1 = h_{f1} + xh_{fg1}, \tag{1}$$

and the heat contained in one pound of the steam on the low-pressure side of the orifice after expansion would be

$$h_2 = h_{f2} + h_{fg2} + c_p(t_s - t_2) = h_{g2} + c_p(t_s - t_2), \tag{2}$$

where c_p is the specific heat of superheated steam (0.48 in all calorimeter problems). But, since no heat is gained or lost in passing through the orifice, the heat in a pound of the steam must be the same on one side of the orifice as it is on the other, or

$$h_1 = h_2. \tag{3}$$

Hence, substituting the values of h_1 and h_2 in Eq. (3),

$$h_{f1} + xh_{fg1} = h_{g2} + c_p(t_s - t_2). \tag{4}$$

Solving for x,

$$x = \frac{h_{g2} + c_p(t_s - t_2) - h_{f1}}{h_{fg1}}. \tag{5}$$

The *percentage of moisture* equals $1 - x$. (6)

The expression $h_{g2} + c_p(t_s - t_2)$ in Eqs. (2), (4), and (5) represents the total heat of superheated steam at the absolute pressure on the low-pressure side of the orifice and a temperature t_s, and may be found directly from the Tables for Superheated Steam and substituted in Eq. (5) when solving for x; or the proper values for h_{g2}, c_p, t_s, and t_2 may be substituted in the same equation in order to find x.

Ordinarily, t_2 is found from the tables by looking up the temperature corresponding to the absolute pressure in the calorimeter, *i.e.*, the sum of the atmospheric pressure and the pressure shown by the manometer. This practice, however, is not permitted by the A.S.M.E. rules for finding the quality of steam, since t_s is taken with a thermometer that has part of its stem exposed, and is thus subject to radiation, nor does it take account of the radiation from the calorimeter itself, which may be considerable even though well covered. Therefore, for accurate work it is necessary to take a *normal reading* of the thermometer, as described in the A.S.M.E. *Transactions*, Vol. 21, page 43, and Vol. 24, page 740, to correct for these errors.

The calorimeter shown in Fig. 18 differs from the one shown in Fig. 17 in that the *temperature* of the steam being admitted to the calorimeter is observed instead of the *pressure*. In other words, h_{f1} and h_{fg1} correspond to the temperature t_1 rather than to the absolute pressure p_1. Another difference is that in the

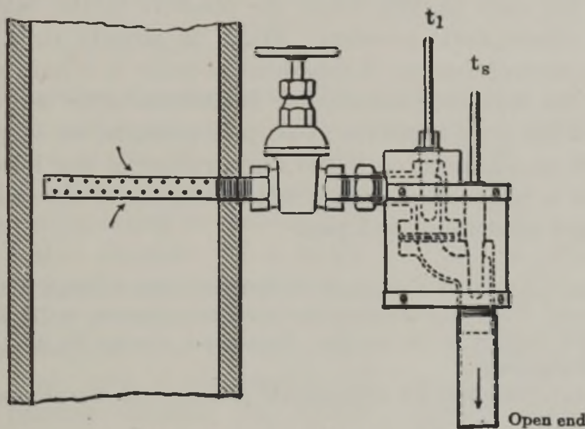


FIG. 18.—Barrus throttling calorimeter.

Barrus calorimeter the exhaust is made very free and the pressure p_2 on the lower side of the orifice is assumed to be atmospheric. A long exhaust pipe will cause a back pressure in the calorimeter where it has been assumed that the pressure is atmospheric. In the Barrus calorimeter the orifice is $\frac{3}{32}$ in. in diameter.

In case the atmospheric pressure is not known, it can be assumed as 14.7 psi. If the barometer reading is given, however, it should always be used. This reading, as well as that of the manometer giving the pressure in the calorimeter, will be given in inches of mercury. To change this to pounds per square inch, multiply the inches of mercury by 0.491.

Although the throttling calorimeter is the most accurate form where it can be used, it has certain limitations. It is obvious that if the entering steam contains too much moisture there may not be enough heat on the lower side of the orifice to evaporate this moisture and superheat the leaving steam. Unless there are at least 8 to 10 degrees of superheat in this steam, the accuracy of the results is uncertain.

In general, it may be said that the working limits of this form of calorimeter vary with the initial steam pressure. The calorimeter ceases to superheat when the moisture exceeds about 2 per cent at a pressure of 40 psia, 5 per cent at a pressure of 150 psia, and $6\frac{1}{4}$ per cent at a pressure of 250 psia, assuming atmospheric pressure on the lower side of the orifice in each case. These figures will vary slightly when the pressure in the calorimeter exceeds atmospheric pressure. Since, as already stated, there must be several degrees of superheat in order to obtain accurate results, the throttling calorimeter is unsuitable for use, if there is over 2 per cent moisture at 50 psia pressure, or $4\frac{1}{2}$ per cent moisture at 150 psia, or if the temperature of the lower thermometer is below 220 F, which will be the case if the steam is at a very low pressure (4 or 5 psi).

Example.—Steam at a pressure of 100 psi blows through a throttling calorimeter. The temperature of the lower thermometer is 275 F and the manometer reading is 5.6 in. Hg. Barometer reading 29 in. Find the quality of the steam.

Solution.—First find the atmospheric pressure and the pressure in the calorimeter.

$$\text{Atmospheric pressure} = 0.491 \times 29 = 14.25 \text{ lb.}$$
$$\text{Pressure in calorimeter} = 0.491 \times 5.6 = 2.75 \text{ lb.}$$

Now from the saturated steam tables find h_{f1} and h_{fg1} corresponding to the pressure in the main, 114.25 psia, and from the superheated steam tables, the enthalpy in superheated steam at a pressure of 17 psia and a temperature of 275 F.

Then from Eq. (5),

$$x = \frac{h_{g2} + c_p(t_s - t_2) - h_{f1}}{h_{fg1}}$$
$$= \frac{1180.30 - 308.60}{880.98} = \frac{871.70}{880.98} = 0.989.$$

Ans. 98.9 per cent.

Or, from the saturated steam tables find h_{f1} and h_{fg1} corresponding to the pressure in the main, 114.25 psia, and also h_{g2} and t_2 corresponding to the pressure in the calorimeter, 17 psia.

Then from Eq. (5),

$$x = \frac{h_{g2} + c_p(t_s - t_2) - h_{f1}}{h_{fg1}}$$

$$= \frac{1{,}153.1 + 0.48(275 - 219.4) - 308.60}{880.98}$$

$$= \frac{1{,}153.1 + 0.48 \times 55.6 - 308.60}{880.98} = \frac{871.2}{880.98} = 0.989.$$

Ans. 98.9 per cent.

Example.—Find the quality in the same problem using the Mollier diagram instead of the steam tables.

Solution.—Locate the intersection of the 17-lb pressure line with the 275 F temperature line and then, as the total heat remains constant, follow horizontally to the left to the intersection with the 114.25-lb pressure line. At this point read the quality, 99 per cent.

48. Separating Calorimeters.—*The weight of the dry steam that will pass through a given size of orifice in a given time depends upon the pressure on the two sides of the orifice.* This law holds true until the lower pressure equals or exceeds 0.58 of the higher pressure. If a is the area of the orifice in square inches, p the absolute pressure of the steam entering the orifice in pounds per square inch, and w the pounds of steam passing through the orifice per second, then

$$w = \frac{pa}{70} \text{ (Napier's empirical rule).} \quad (7)$$

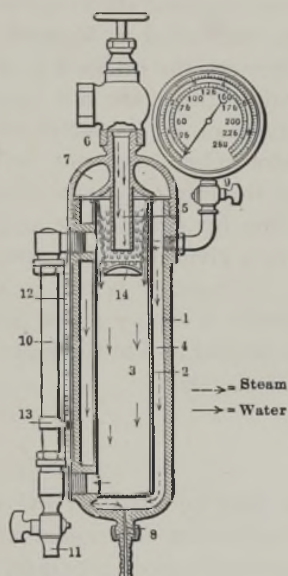


FIG. 19.—Carpenter's improved separating calorimeter.

From Napier's rule the weight of steam flowing through an orifice of known area is proportional to the absolute steam pressure.

The amount of steam flowing through any orifice may, therefore, be determined. Professor R. C. Carpenter has a calorimeter based upon this principle. Wet steam enters the calorimeter (Fig. 19) through the pipe 6, and is projected against the cup 14. The steam and water are then turned through an angle of 180 deg, which causes the water to be thrown outward by inertia through the meshes in the cup into the inner chamber 3. Forcing the steam to strike the cup, instead of allowing it to flow directly into the chamber 3, prevents any moisture already thrown out from being picked up again and carried on. The steam after leaving the cup passes upward and enters the top of the outer chamber 7. It then flows down around the inner chamber in the annular space 4 and is discharged through the orifice 8. The area of this orifice, which is known, is so small that there is no loss in pressure of the steam as it flows through the calorimeter. The pressure in the two chambers being the same, the temperature is the same, and there is no loss of heat from the inner chamber by radiation. The gage glass 10, connected with the inner chamber, has a scale 12 graduated in *hundredths of pounds*, so that the weight of moisture separated from the steam can be read directly. The pressure gage 9 has two scales, the inner one showing the steam pressure and the outer one being so calibrated as to read directly the number of *pounds* flowing through the orifice 8 in a given time (generally 10 min). These calibrations are not proportional to the pressure readings on the gage, for the latter are proportional to the pressures above the atmosphere and not to the absolute pressures. The accuracy of the results obtained by using the gage may be checked at any time by condensing and weighing the discharge from orifice 8 for a given period of time.

If, now, w is the weight of dry steam discharged from the orifice 8 in any given period of time, W the weight of moisture collected in 3 in the same period of time, and x the quality of the steam, then

$$x = \frac{w}{w + W}. \tag{8}$$

w may be obtained either from the reading of the gage 9, or by condensing the steam and weighing it. This latter method is generally more accurate, as the orifice 8 may become clogged up with dirt or rust, or the gage may give incorrect readings for other reasons. The weight of moisture, W, is found by taking the difference between the readings on scale 12 at the beginning and end of the test.

Example.—(*a*) Find the quality of the steam as shown by a separating calorimeter, if the data are as follows: weight of dry steam escaping through orifice, 4.5 lb; weight of moisture collected, 0.05 lb; steam pressure entering calorimeter, 114.25 psia.

(*b*) Find the diameter of the orifice if the length of the run is 20 min.

Solution.—(*a*) From Eq. (8),

$$x = \frac{w}{w + W}$$
$$= \frac{4.5}{4.5 + 0.05} = \frac{4.5}{4.55} = 0.989.$$

(*b*) Find the weight of the steam flowing through the orifice per second, and call it w'

Then

$$w' = \frac{4.5}{20 \times 60} = \frac{4.5}{1{,}200} = 0.00375 \text{ lb per sec.}$$

From Eq. (7),

$$w' = \frac{pa}{70}.$$
$$a = \frac{70w'}{p} = \frac{70 \times 0.00375}{100 + 0.491 \times 29} = \frac{0.26250}{114.25}$$
$$= 0.0023 = \pi r^2$$
$$r^2 = \frac{0.0023}{3.1416} = 0.000732$$
$$r = 0.02705$$
$$d = 0.0541$$

PROBLEMS

1. Steam is throttled in a calorimeter from a pressure of 200 psia to a pressure of 15 psia, its temperature becoming 235.5 F. What was its initial quality?

2. The temperature of steam in a steam main is 390 F. The steam after passing through a throttling calorimeter has a temperature of 260 F. The steam discharges into the atmosphere where the barometer reads 28.53 in. Hg. Find the quality of steam in the main.

3. Steam at a pressure of 150 psia blows through a throttling calorimeter. The temperature of the steam on the low-pressure side of the orifice is 250 F and the manometer reading is 5.6 in. Hg; barometer, 29 in. Hg. Find the quality of the steam.

4. The low-pressure side of a throttling calorimeter is attached to a condenser. The vacuum in the condenser is 27.86 in. Hg; the temperature of the steam before passing through the calorimeter, 250.3 F; the temperature on the low-pressure side, 179.4 F. Find the quality of the steam.

5. Steam passes from the high-pressure header in a boiler plant through a pressure-reducing valve into the heating system, without loss of heat. If the pressure in the header is 125 psia, the pressure in the heating system is 20 psia, and the temperature of the steam immediately after passing the pressure-reducing valve is 264 F, what is the quality of the steam in the header?

6. Steam at a pressure of 110 psi blows through an orifice into the atmosphere. The temperature of the steam after passing through this orifice is 240 F. What per cent of moisture is in the original steam?

7. Five pounds of a mixture of steam and water containing 2 per cent moisture at a pressure of 150 psia expands through an orifice to a pressure of 14.7 psia. What will be the temperature of the steam at the lower pressure?

8. What pressure must be maintained on the low-pressure side of a throttling calorimeter in order to measure quality as low as 92 per cent in a steam line where the pressure is 100 psia? The degrees of superheat on the low-pressure side of the calorimeter are to be not less than 10.

9. Steam at a pressure of 200 psi and having a quality of 94 per cent passes through a throttling calorimeter. At what pressure will the exhaust steam become just dry and saturated?

10. Steam at a pressure of 150 psia passes through a throttling calorimeter. Assuming that the lowest condition in the calorimeter for measuring the quality is 10 deg superheat and that the pressure in the calorimeter is 15 psia, what is the largest percentage of moisture the calorimeter is capable of measuring under the above conditions?

11. Determine the condition of the steam in a line where the pressure is 208 psia, when the following readings are obtained from a throttling calorimeter attached to the line: the pressure in the low-pressure chamber of the calorimeter is 15 psia and the temperature of the steam is 400 F.

12. Determine the quality of the steam in a line where the pressure is 130 psia, when the following readings are obtained from a throttling calorimeter attached to this line: the pressure in the low-pressure chamber is 16 psia and the temperature of the steam is 280 F.

13. A steam line develops a leak, the area of which is 0.1 sq in. The line pressure is 150 psi; barometer, 30 in. Hg. The pressure in the chamber into which the steam discharges is 25 psi. Determine the weight of steam loss in pounds per hour.

14. Find the quality of the steam if, when tested with a separating calorimeter, 4.5 lb of dry steam blows through the orifice while 1.5 lb of moisture are separated out. If the run is 30 min and the steam pressure is 100 psia, determine the diameter of the orifice.

15. Steam at a pressure of 100 psia blows through a separating calorimeter. The run is 45 min, 10.5 lb of dry steam flow through the orifice, and 0.5 lb of moisture is collected. Find the quality of the steam and the area of the orifice.

16. A separating calorimeter has an outlet orifice whose area is 0.014 sq in.; initial steam pressure is 100 psia; the weight of moisture collected in 10 min is 0.84 lb. Determine the quality of the steam, assuming that the flow through the orifice follows Napier's rule.

17. A boiler discharges 1,200 lb of steam per hour at 140 psia into a steam main. A steam separator in the main removes 60 lb of water per hr and the following readings were observed on a throttling calorimeter, downstream from the separator: pressure in calorimeter = 14.7 psia; temperature = 260 F. Assume no loss due to radiation from the pipe line. What is the quality of the steam leaving the boiler when the room temperature is 87 F?

49. Mechanical Mixtures.—Problems involving the resulting temperature and final condition when various substances are mixed mechanically are often met. They are best treated by first determining the heat in Btu that would be available for use if the temperature of all the substances were brought to 32 F, and then using this heat (positive or negative) to raise (or lower) the total weight of mixture to its final temperature and condition.

If one pound of dry saturated steam at a temperature t_1 is condensed and then the temperature of the condensed steam is lowered to a temperature t_2, the amount of heat h' given off would be

$$h' = h_{fg1} + c(t_1 - t_2), \tag{9}$$

where h_{fg1} is the enthalpy of vaporization corresponding to the temperature t_1.

If the steam was condensed only, the heat given off would be

$$h' = h_{fg1} \tag{10}$$

and the temperature of the mixture is the temperature corresponding to the pressure.

If the steam is only partly condensed, let x equal the per cent of steam condensed. Then

$$h' = xh_{fg1} \tag{11}$$

and the temperature of the mixture is the temperature corresponding to the pressure.

The general laws of thermodynamics do not apply in the case of mixtures, as the equations become discontinuous. The general expression for heat absorbed in passing from a solid to a gaseous state may be stated as follows:

Let c_1 be the specific heat in the solid, c_2 in the liquid, and c_3 in the gaseous state, w the weight of the substance, t the initial temperature, t_1 the temperature of the melting point, t_2 the temperature of the boiling point, t_3 the final temperature, h the enthalpy of the liquid, and h_{fg1} the enthalpy of vaporization.

$$h' = w[c_1(t_1 - t) + h + c_2(t_2 - t_1) + h_{fg} + c_3(t_3 - t_2)]. \tag{12}$$

TABLE IX.—SPECIFIC HEATS OF LIQUIDS AND SOLIDS

Substances	Specific heat, c
Mercury	0.0333
Alcohol	0.615
Turpentine	0.462
Wrought iron	0.114
Cast iron	0.129
Copper	0.095
Ice	0.504
Spermaceti	0.320
Sulphur	0.177
Glass	0.187
Graphite	0.200

Latent heat of fusion of ice = 144 Btu.

Example.—Find the final temperature and condition of the mixture after mixing 10 lb of ice at 20 F, 20 lb of water at 50 F, and 2 lb of steam at atmospheric pressure. Mixture takes place at the pressure of the steam.

Solution.

Heat to raise ice to 32 F = $10 \times 0.5(32 - 20)$	=	60
Heat to melt ice = 10×144	=	1,440
Total heat necessary to change the ice to water at 32 F	=	1,500 Btu
Heat given up by water when temperature is lowered to 32 F = $20 \times (50 - 32)$	=	360
Heat in steam above 32 F (from tables) = $2 \times 1{,}150.4$	=	2,300.8
Total heat given up in lowering water and steam to 32 F	=	2,660.8 Btu
Heat available for use = $2{,}660.8 - 1{,}500$	=	1,160.8 Btu
Degrees this heat will raise the mixture	=	$\frac{1{,}160.8}{32} = 36.3.$

$\therefore$ Final temperature of mixture = $36.3 + 32 = 68.3$ F.

Ans. 32 lb water at 68.3 F.

Example.—Find the resulting temperature and condition after mixing 10 lb of ice at 20 F, 20 lb of water at 50 F, 40 lb of air at 82 F, and 20 lb of steam at a pressure of 100.3 psi and containing 2 per cent moisture. Mixture takes place at the pressure of the steam.

Solution.

$$10 \times 0.5(32 - 20) = 60$$
$$10 \times 144 = 1{,}440$$
$$1{,}500 \text{ Btu} = \text{heat to raise ice to water at } 32 \text{ F}$$

$$
\begin{array}{lrl}
20 \times (50 - 32) & = \quad 360 & \\
40 \times 0.241(82 - 32) & = \quad 482 & \\
20(309.11 + 0.98 \times 880.6) & = \underline{23{,}440} & \\
 & 24{,}282 \text{ Btu} & = \text{heat given up by air, water, and steam} \\
 & \underline{1{,}500} & \\
 & 22{,}782 \text{ Btu} & = \text{heat available} \\
40 \times 0.241(338.07 - 32) & = \underline{2{,}950} \text{ Btu} & = \text{heat to raise air to 338.07 F} \\
 & 19{,}832 \text{ Btu} & = \text{heat available to raise the water} \\
50 \times 309.11 & = \underline{15{,}456} \text{ Btu} & = \text{heat to raise water to 338.07 F} \\
 & 4{,}376 \text{ Btu} & = \text{heat available to evaporate water}
\end{array}
$$

$$\frac{4{,}376}{880.6} = 4.97 \text{ lb steam}$$

Ans. $\left.\begin{array}{l}\text{40 lb air} \\ \text{45.03 lb water} \\ \text{4.97 lb dry saturated steam}\end{array}\right\}$ at 338.07 F.

Example.—Find the resulting temperature and condition after mixing 10 lb of ice at 20 F, 20 lb of water at 50 F, and 30 lb of steam at a pressure of 101.3 psi and 400 F temperature. The mixture takes place at a pressure of 25.3 psi.

Solution.

$$
\begin{array}{lrl}
10 \times 0.5(32 - 20) & = \quad 60 & \\
10 \times 144 & = \underline{1{,}440} & \\
 & 1{,}500 \text{ Btu} & = \text{heat to raise ice to water at 32 F} \\
20 \times (50 - 32) & = \quad 360 & \\
30 \times 1{,}225.0 \text{ (from Table VIII)} & = \underline{36{,}750} & \\
 & 37{,}110 \text{ Btu} & = \text{heat given up by water and steam} \\
 & \underline{1{,}500} & \\
 & 35{,}610 \text{ Btu} & = \text{heat available} \\
60 \times 236.03 & = \underline{14{,}162} \text{ Btu} & = \text{heat to raise water to 267.25 F} \\
 & 21{,}448 \text{ Btu} & = \text{heat available to evaporate water}
\end{array}
$$

$$\frac{21{,}448}{933.7} = 22.97 \text{ lb steam.}$$

Ans. $\left.\begin{array}{l}\text{37.03 lb water} \\ \text{22.97 lb dry saturated steam}\end{array}\right\}$ at 267.25 F.

PROBLEMS

1. A tank contains 400 lb of water at 50 F. Steam at a pressure of 100 psia is admitted until the temperature rises to 100 F. The weight of the water was then 418.5 lb. Determine the quality of the steam.

2. An open feed-water heater is supplied with 1,000 lb of 90 per cent quality exhaust steam per hr. How many pounds of cold water can be heated with this steam, if the water enters at 70 F and leaves at 190 F? The pressure of the exhaust steam is 15 psia.

3. Water enters an open feed-water heater at 75 F and leaves at 190 F. Determine the amount of steam necessary to furnish 1,000 lb of hot water, if the steam is available at 20 psia and 95 per cent quality.

4. Three pounds of a solid substance have 75 Btu supplied to the mass to change the temperature 100 F. What is the specific heat?

5. Ten pounds of steam at 212 F are mixed with 50 lb of water at 60 F and 2 lb of ice at 32 F. What will be the resulting temperature and the condition of the mixture?

6. Five pounds of steam at atmospheric pressure, 10 lb of water at 60 F, and 2 lb of ice at 20 F are mixed at atmospheric pressure. What will be the resulting temperature?

7. Find the final temperature and the condition of the mixture after mixing 8 lb of ice at 12 F, 25 lb of water at 40 F, and 3 lb of steam at a pressure of 14.7 psia.

8. Determine the resulting temperature and condition of a mixture of 8 lb of ice at 20 F, 12 lb of water at 75 F, 10 lb of air at 190 F, and 5 lb of steam at a temperature of 588 F and atmospheric pressure. The mixture takes place at atmospheric pressure.

9. Ten pounds of steam at atmospheric pressure, 95 per cent dry, and 10 lb of ice at 32 F are mixed together. (*a*) What is the final temperature of the mixture? (*b*) What is the condition of the mixture? (*c*) If all the steam is not condensed, determine the per cent that is condensed.

10. Find the final temperature and condition of the mixture after mixing at the pressure of the steam 12 lb of ice at 15 F, 20 lb of water at 60 F, and 4 lb of steam at atmospheric pressure and 90 per cent quality.

11. Find the resulting temperature and condition of a mixture of 10 lb steam at a pressure of 150 psia and a temperature of 400 F, 15 lb water at 60 F, 30 lb ice at 20 F and 50 lb air at 112 F. The mixture takes place at a pressure of 40 psia.

12. Find the temperature and condition of the mixture after mixing 10 lb of steam at a pressure of 30 psia and a temperature of 250.3 F, 2 lb of ice at 10 F, and 20 lb of water at 40 F. The mixture takes place at the pressure of the steam.

13. Mix the following: 3,740 lb of steam at a pressure of 22 psia having a quality of 90 per cent, and 20,470 lb of water at 46 F. Determine the final temperature, if the mixture takes place at the pressure of the steam.

14. Five pounds of steam at a pressure of 120 psia having a quality of 95 per cent, 5 lb of ice at −48 F, and 10 lb water at 70 F are mixed at atmospheric pressure. Find (*a*) the condition of the mixture; (*b*) the temperature of the mixture; (*c*) the per cent of steam condensed.

15. Five pounds of steam at a pressure of 120 psia and containing 2 per cent moisture, 2 lb of ice at 20 F, and 70 lb of water at 60 F are mixed in an open tank. (*a*) What will be the temperature of the mixture? (*b*) What will be the weight of water, ice, and steam in the mixture?

16. Two pounds of steam at a pressure of 150 psia and a temperature of 400 F, 5 lb of ice at 22 F, and 10 lb of water at 60 F are mixed at atmospheric pressure. Find the final temperature and condition of the mixture.

17. Eight pounds of steam at a pressure of 150 psi, having a quality of 98 per cent, are mixed at a pressure of 15 psi with 11 lb of water at 89 F, and 6 lb of ice at 20 F. What are the temperature and condition of the mixture?

18. Find the resulting temperature and condition after mixing at a pressure of 25 psia, 20 lb of ice at 0 F, 30 lb of water at 50 F, 40 lb of air at 80 F, and 10 lb of steam at a pressure of 150 psia and a temperature of 400 F.

19. Find the resulting temperature and condition of mixture after mixing 10 lb of ice at 30 F, 50 lb of water at 50 F, 2 lb of steam at a pressure of 20 psia having a quality of 95 per cent, 4 lb of steam at a pressure of 120 psia with a quality of 100 per cent, and 6 lb of steam at a pressure of 165 psia and a temperature of 466 F. The mixture takes place at atmospheric pressure.

20. Steam at a pressure of 95 psi blows through an orifice into a chamber where the pressure is 8.2 in. Hg above the atmosphere. The temperature on the lower side of the orifice is 248 F. Barometer, 29.8 in. Hg. Five pounds of steam from the same line are mixed with 40 lb of ice at a temperature of −8 F, 60 lb water at 50 F, and 30 lb air at 20 F, the mixture taking place at a pressure of 5 psi. Find the temperature and condition of mixture.

21. A heating system is supplied with steam at a pressure of 25 psia Exhaust steam from a turbine at a pressure of 25 psia, containing 15 per cent moisture, comes into the system at the rate of 1,000 lb per min, and is mixed with steam throttled from the high-pressure mains, in which the pressure is 150 psia and the quality 98 per cent. This throttled steam is supplied at the rate of 500 lb per min. What is the quality of the steam immediately after mixing?

22. A certain artificial ice skating rink has a floor 100 ft by 200 ft. The ice on this floor is 6 in. thick and has a temperature of 20 F. It becomes necessary for the owner to melt this ice and drain the basin and he runs onto it 600,000 gal of water at 47 F. He is then prepared to turn onto it steam at a pressure of 5 psi from his heating system. Will it be necessary for him to use any of this steam and if so, how many pounds must he use in order to have the mixture discharge into the sewer at 36 F? The steam contains 2 per cent moisture. Barometer, 30 in. Hg. One gallon of water weighs $8\frac{1}{3}$ lb. One cubic foot of ice weighs $57\frac{1}{2}$ lb.

CHAPTER V

FUELS AND COMBUSTION

50. Fuels.—The source of heat that is used to produce steam in a boiler is the fuel, which may be in a *solid, liquid,* or *gaseous* form. The principal solid fuels are wood, peat, lignite, and coal. The liquid fuels are usually some of the mineral oils, generally unrefined petroleum. In some gas plants liquid tar is used. The most commonly used gaseous fuel is natural gas, but many plants use gas which is a waste product from a manufacturing operation. In the steel mills the "downcomer" gases from the blast furnaces are often used as a fuel for the steam boilers. Coke-oven gases are similarly used. In some cases the coal is distilled in a gas producer, and this producer gas used as a fuel.

Another classification is that of *natural fuels, prepared fuels,* and *by-products,* or waste, from the industries. Among the fuels included in the first group of this latter classification are wood, peat, coal, crude oil, and natural gas; in the second, powdered coal, briquettes, charcoal, coke, petroleum distillates (such as kerosene and gasoline), alcohol, producer gas, and water gas; and in the third, sawdust, tanbark, paper-mill refuse, bagasse (or the refuse of sugar cane), blast-furnace gas, and coke-oven gas. The principal ingredients of all these fuels are carbon and hydrogen.

The fuel most commonly used for supplying heat to boilers is bituminous coal, although in many parts of the world the use of fuel oil for this purpose is rapidly increasing.

51. Coal Analysis.—In order to compare the products of various mines and to determine the relative values of these fuels for different uses, coal is subjected to an *ultimate* and a *proximate* analysis, and it is tested in a coal calorimeter to ascertain its *calorific* or *heating* value.

In the *ultimate,* or *chemical, analysis* the proportions of carbon, hydrogen, oxygen, nitrogen, sulphur, and ash are determined.

In the *proximate*,[1] or *physical*, analysis determinations are made of the amounts of moisture, volatile matter, fixed carbon, and ash. The sulphur content is usually reported with the proximate analysis, but it is determined separately.

The *moisture* in the coal consists of two parts, that which is inherent in the coal itself, and the moisture due to exposure to weather. Coal from which this latter moisture has been evaporated so that it is in equilibrium with the surrounding air is said to be *air dried.*

The *volatile matter* is that part of the coal, moisture excepted, which is driven off when a sample is heated to a temperature of about 1750 F, and consists of hydrocarbons, such as marsh gas or methane (CH_4), olefiant gas or ethylene (C_2H_4), pitch, tar, and naphtha. All of these must be distilled from the fuel before being burned. The remainder consists of the *fixed (solid) carbon* and *ash.* The sum of the volatile matter and the fixed carbon is the *total combustible*, and the *ash* is the incombustible material that remains after the fuel has been completely burned.

The U. S. Bureau of Mines *Bulletin* 22 says,

> For the commercial valuation of coals a proximate analysis and a calorific-value determination are usually sufficient. Moisture and ash are of importance; they not only displace their own weights of combustible matter, but the evaporation of the moisture wastes heat. A high percentage of ash increases the cost of handling coal in a power plant and decreases the efficiency of the furnace. The ratio of the volatile matter to the fixed carbon indicates in a way the type of furnace best adapted for burning a coal with maximum efficiency. The smokeless combustion of coal containing a low percentage of volatile matter is not difficult in furnaces of ordinary types, but to burn a high-volatile coal without smoke requires a suitably designed furnace. A high percentage of sulphur is undesirable in coal used for the manufacture of coke and gas. For ordinary steaming purposes sulphur is not a serious drawback unless associated with elements, such as iron or lime, that promote clinkering.

The coal analyses may be given on either a *dry*, or *as fired* basis. If given on a dry basis and it is desired to change to the other, *multiply* each percentage by $\left(1 - \frac{\text{per cent of moisture}}{100}\right)$.

[1] It should be noted that the term *proximate* does not mean that the analysis is only "approximate," the facts being actually to the contrary.

To change from a wet, or as fired, to a dry basis, *divide* each percentage by this same factor.

Sometimes it is desirable to know the analysis, or the calorific value of the fuel, on a "moisture- and ash-free" basis. In this case divide each percentage, or the heating value, as given on a wet-coal basis by $\left(1 - \frac{\text{per cent of moisture} + \text{per cent of ash}}{100}\right)$.

52. Classification of Coals.—Various methods of classification to designate the differences in coals have been suggested, some based on the carbon-hydrogen ratio, others on the ratio between the volatile matter and either the fixed carbon or the total combustible present. The U. S. Geological Survey uses the word *rank* to indicate differences due to the progressive change from lignite to anthracite, this change being accompanied by a decrease in moisture, oxygen, and volatile matter, and generally by an increase in fixed carbon, sulphur, and ash. The word *grade* is used to distinguish one coal from another on a basis of ash, or sulphur content.

Although there is no difference of opinion as to the general classes, or groups into which coal is divided, it is difficult to define each group exactly. Figure 20, taken from the U. S. Bureau of Mines *Circular* 6094, shows the ranks of coal recognized by the bureau together with their physical compositions and heating values. It must not be supposed that all coals of the various ranks contain exactly the same percentages of moisture, volatile matter, and fixed carbon as those shown in the chart. In fact, these themselves do not agree. The figures are based, however, upon actual analyses of typical coals of the different ranks.

The following comments on Fig. 20 are taken from the U. S. Bureau of Mines *Circular* 6094:

The upper diagram shows the progressive increase in the calorific value from peat to low-rank semibituminous coal. At this rank the calorific value attains a maximum and then decreases with further increase in rank due to the elimination of hydrogen that has taken place in the anthracitization of the coal by temperature and pressure in the earth's crust.

The upper diagram also shows that the calorific value of the coal as occurring in the bed but calculated free from ash indicates rather definitely the proper position in the scale of rank for those coals which rank below semibituminous. This change is due to the progressive increase of moisture with decrease in rank, as shown in the lower diagram.

From these diagrams it is seen that the percentage of fixed carbon is the outstanding factor which shows the relative rank. Metamorphism of peat to coal consists of the gradual elimination of water and volatile matter. In the early stages, water elimination predominates, and in the later stages volatile matter removal is the principal factor.

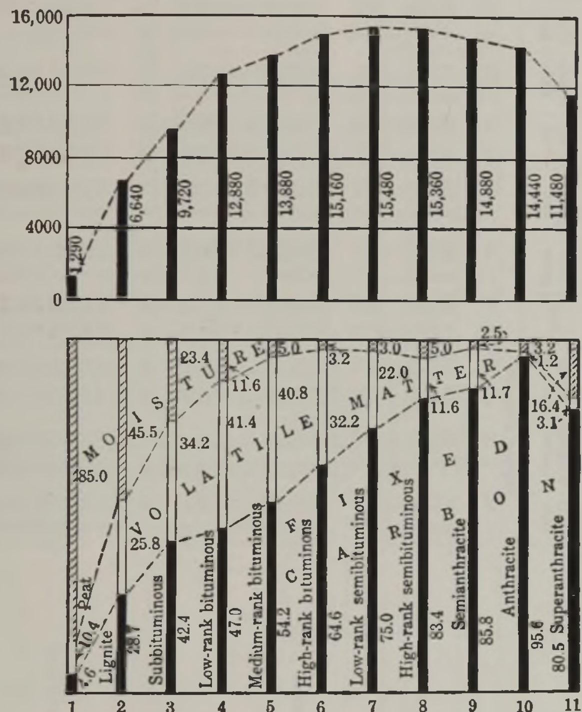


FIG. 20.—Diagrams showing the average analysis of the various ranks of coal. The lower diagram shows the ash-free proximate analysis, and the upper diagram shows the moisture- and ash-free heating values expressed in Btu. (*After M. R. Campbell.*)

The term "fuel ratio"—that is, the ratio of fixed carbon to volatile matter expressed by the symbol $\frac{FC}{VM}$—is generally used by American geologists in setting the boundaries of the ranks of coal. This ratio applies very well to coals of medium-bituminous and higher ranks, but fails in the lower range due to the fact that the volatile-matter factor decreases with decrease in rank. A glance at the lower diagram shows that the percentage of fixed carbon in the coal as mined but calculated

TABLE X.—ANALYSES OF COALS

Location		Kind of sample	Proximate analysis coal as received				Ultimate analysis coal as received					Fuel ratio	Calorific value, Btu per pound	
State	County		Moisture	Volatile matter	Fixed carbon	Ash	Sulphur	Hydrogen	Carbon	Nitrogen	Oxygen		As received	Dry
			Superanthracite											
Rhode Island	Newport	Mine	13.26	2.56	65.30	18.88	0.30	1.88	64.23	0.22	14.49	25.50	9,313	10,737
			Anthracite											
Colorado	Gunnison	Mine	2.7	3.32	88.15	5.83	0.80	3.28	85.38	1.12	3.59	26.55	14,099	14,490
Pennsylvania	Lackawanna	Car culm	5.41	7.02	71.79	15.78	0.74	3.10	72.65	0.77	6.96	10.25	12,047	12,737
Pennsylvania	Schuylkill	Mine	2.80	1.16	88.21	7.83	0.89	1.89	84.36	0.63	4.40	76.00	13,298	13,682
			Semianthracite											
Arkansas	Pope	Mine	2.07	9.81	78.82	9.30	1.74	3.62	80.28	1.47	3.59	8.03	13,702	13,991
Pennsylvania	Sullivan	Mine	3.40	9.34	75.58	11.68	0.81	3.64	77.85	0.95	5.07	8.11	13,120	13,581
			Semibituminous											
Arkansas	Logan	Mine	2.77	14.69	73.47	9.07	2.79	4.02	78.71	1.46	3.95	5.00	13,774	14,167
Colorado	Pitkin	Car run of mine	0.96	21.49	68.93	8.62	0.52	4.66	79.61	1.83	4.76	3.20	14,330	14,468
Maryland	Allegany	Mine	3.0	14.5	75.6	6.9	0.85	4.52	80.37	1.82	5.52	5.21	14,240	14,670
Oklahoma	Haskell	Mine	2.37	19.26	69.54	8.83	1.03	4.57	79.39	1.55	4.63	3.09	13,840	14,177
Pennsylvania	Cambria	Mine	3.3	12.5	77.9	6.33	1.04	4.46	81.65	1.27	5.25	6.23	14,340	14,820
Pennsylvania	Somerset	Mine	2.4	13.5	77.8	6.31	1.26	4.44	82.62	1.31	4.06	5.77	14,370	14,730
Virginia	Tazewell	Mine	3.8	15.5	77.8	2.92	0.63	4.92	84.70	1.16	5.67	5.02	14,860	15,440
West Virginia	Fayette	Mine	3.49	14.81	76.2	5.5	0.76	4.91	82.35	1.44	5.04	5.14	14,368	14,888
West Virginia	McDowell	Mine	2.8	12.5	80.4	4.31	0.66	4.37	83.55	1.01	6.10	6.43	14,660	15,080
			Cannel											
Kentucky	Johnson	Mine	2.20	50.64	36.70	10.46	0.99	6.57	72.01	1.17	8.80	0.725	13,748	14,058
			Bituminous—high rank											
Alabama	Shelby	Mine	3.83	32.03	58.66	5.48	0.97	5.29	77.26	1.25	9.75	1.83	13,799	14,348
Colorado	Las Animas	Mine	2.28	29.81	58.75	9.16	0.50	5.22	74.24	1.08	9.80	1.97	13,781	14,101
Illinois	Williamson	Mine	8.80	29.85	53.83	7.52	1.13	5.08	68.70	1.33	16.24	1.81	12,222	13,401
Kansas	Cherokee	Car (lump and nut)	4.10	31.65	53.71	10.54	3.77	5.10	70.25	1.06	9.28	1.69	12,895	13,448
Kentucky	Pike	Mine	3.73	30.01	59.42	6.84	0.56	5.07	76.30	1.06	10.17	1.98	13,649	14,177
New Mexico	Colfax	Mine	2.45	27.54	52.61	17.40	0.77	4.82	68.38	0.90	7.73	1.91	12,200	12,506
Ohio	Hocking	Mine	9.72	32.44	53.41	4.43	0.54	5.70	69.50	1.25	18.58	1.65	12,247	13,567
Oklahoma	Pittsburg	Mine	3.10	32.07	59.34	5.49	0.89	5.13	73.52	1.89	13.08	1.85	13,495	13,927
Pennsylvania	Allegheny	Mine	2.60	32.67	59.41	5.32	0.77	5.39	78.16	1.45	8.91	1.82	14,085	14,461
Tennessee	Cumberland	Car run of mine	3.89	27.61	54.07	14.43	0.78	4.84	70.04	0.98	8.93	1.96	12,514	13,021
Virginia	Wise	Mine	2.48	31.71	60.30	5.51	0.52	5.59	79.69	1.56	7.13	1.90	14,252	14,614
Washington	Pierce	Mine	2.79	33.78	53.87	9.56	1.01	5.44	73.88	2.00	8.11	1.60	13,451	13,837
West Virginia	Mingo	Car run of mine	2.86	33.23	58.08	5.83	0.67	5.01	78.38	1.43	8.68	1.75	14,105	14,521

		Bituminous—medium rank												
Illinois	Williamson	Car, mixed run of mine and lump	8.86	31.25	48.23	11.66	2.46	5.24	64.29	1.29	15.06	1.54	11,702	12,839
Indiana	Greene	Car run of mine	13.58	32.07	46.20	8.15	0.91	5.65	63.53	1.42	20.34	1.44	11,419	13,214
Kansas	Linn	Car (lump, over 1¼-in. screen)	9.04	29.69	45.55	15.72	3.72	5.01	60.99	1.06	13.50	1.53	11,142	12,249
Kentucky	Bell	Car run of mine	3.10	36.12	56.39	4.39	1.22	5.43	77.37	1.83	9.76	1.56	14,148	14,602
Michigan	Saginaw	Mine	11.91	31.50	49.75	6.84	1.24	5.84	66.56	1.19	18.33	1.58	11,781	13,374
Missouri	Lafayette	Mine	15.31	32.46	42.42	9.81	3.12	5.80	59.19	1.04	21.04	1.31	10,757	12,700
Ohio	Jefferson	Car (over ¾-in. screen)	3.53	37.45	49.90	9.12	3.47	5.15	71.66	1.31	9.29	1.33	13,072	13,550
		Bituminous—low rank												
Illinois	Clinton	Car (lump, over 1¾-in. screen)	11.35	34.62	40.63	13.40	4.76	5.41	57.36	1.05	18.02	1.16	10,733	12,107
Indiana	Pike	Car run of mine	10.57	35.03	42.75	11.65	3.87	5.40	61.11	1.17	16.80	1.22	11,266	12,598
Missouri	Adair	Mine	16.07	32.13	39.15	12.65	3.31	5.70	55.76	1.00	21.58	1.22	10,235	12,195
New Mexico	Colfax	Mine	5.02	36.78	46.20	12.00	0.56	5.15	66.01	1.28	15.00	1.25	12,064	12,703
Utah	Iron	Mine	10.35	36.33	43.70	9.62	5.82	5.13	61.24	0.95	17.24	1.20	10,874	12,128
		Subbituminous												
Colorado	Boulder	Car run of mine	18.68	34.88	40.45	5.99	0.55	6.07	57.46	1.15	28.78	1.12	10,143	12,472
Montana	Carbon	Mine	11.69	36.14	40.19	11.98	1.05	5.26	55.46	1.20	25.05	1.11	9,787	11,083
New Mexico	McKinley	Car slack	10.79	33.82	36.73	18.66	1.26	5.22	55.07	0.95	18.84	1.09	9,907	11,106
Washington	King	Mine	13.8	32.5	36.0	17.7	0.49	5.64	51.24	0.95	23.98	1.11	9,135	10,600
		Lignite												
Montana	Dawson	Mine	34.55	35.34	22.91	7.20	1.10	6.60	42.40	0.57	42.13	0.65	7,090	10,833
North Dakota	McLean	Car lump *	35.96	31.92	24.37	7.75	1.15	6.54	41.43	1.21	41.92	0.76	7,069	11,038
Texas	Houston	Car (over ½-in. screen)	34.70	32.23	21.87	11.20	0.79	6.93	39.25	0.72	41.11	0.68	7,056	10,805
Washington	Lewis	Mine	30.50	34.94	29.61	4.95	1.25	6.89	45.48	0.75	40.68	0.85	7,934	11,417
Wyoming	Sweetwater	Mine	31.37	29.60	28.91	10.12	1.27	5.43	37.03	0.79	45.36	0.98	5,634	8,210

free from ash may be preferable to the fuel ratio for indicating rank. It may be applied throughout the entire range of coals.

The Geological Survey system of classification by rank on the basis of proximate analysis and physical characteristics is practical and easily applied. In common with any system it has the disadvantage that the boundary lines between ranks are difficult to choose; there are no sharp boundaries—one rank merges into another. The boundaries must be arbitrary to a considerable degree. It is also true that not all coals with the same proximate characteristics are identical in properties.

53. Characteristics of Coals.—The chief characteristics of the various kinds of coal are briefly as follows:

Anthracite, or *hard* coal, ignites very slowly and burns at a high temperature. It is black in color and has a shiny, metallic luster. Its principal component is fixed carbon. Consequently, it gives off almost no smoke and the flame is very short. Owing to its smokeless burning, it is almost all consumed for domestic purposes. Nearly all anthracite used in this country comes from eastern Pennsylvania. An anthracite coal should contain not less than 92 per cent of fixed carbon as compared with the volatile matter.

Semianthracite is softer than anthracite and does not have its metallic luster. It kindles more easily, as it contains more hydrocarbons. It should not contain more than 12.5 per cent of volatile matter as compared with the fixed carbon.

Semibituminous is a still softer coal than semianthracite and has lost most of the luster, but in appearance it looks more like anthracite than bituminous coal. It is lighter than anthracite and burns more rapidly, and is a valuable coal where it is necessary to keep an intense heat. It comes mainly from the northeast section of the Appalachian field. A semibituminous coal should not contain, usually, more than 25 per cent volatile matter as compared with the fixed carbon.

Bituminous Coal.—Coals that contain over 25 per cent volatile matter are usually classed as *bituminous*, or *soft* coals. The principal bituminous coal mines are in Pennsylvania, West Virginia, Ohio, Illinois, and Indiana. Bituminous coals are divided into coking, noncoking, and cannel coals.

Coking coal is a term used in reference to coals that fuse together on being heated and become pasty. These coals are used in gas manufacture and are very rich in hydrocarbons.

Noncoking coals are free-burning and the lumps do not fuse together on being heated. "Jackson Hill" is an example of this kind of coal. Cannel coal is very rich in volatile matter, is generally high in hydrogen, and consequently burns with great heat. It ignites readily and burns with a bright flame. It is very homogeneous, breaks like glass without any definite line of fracture, and has a dull resinous luster. It is very valuable as a gas coal, so that it is little used for steaming purposes.

Subbituminous coal, sometimes called *black lignite*, ranks between bituminous and lignite. It differs from the former in that it contains a greater percentage of moisture and breaks down or "slacks" when subjected to alternate wetting and drying, and from the latter in its color and its freedom from a distinctly woody texture and structure.

Lignite is coal of very recent formation and in analysis is somewhat similar to peat. It usually resembles wood in appearance, is uneven of fracture, and in the better grades has a dull luster resembling anthracite. Owing to its brownish color it is sometimes called *brown coal.* It is found generally only west of the Mississippi River. As it comes from the mine, it usually contains from 30 to 40 per cent or more of moisture. Its heating value is consequently low and the consumer cannot afford to pay freight on so much water for any great distance.

54. Properties of Various Coals.—The heating values and analyses of coals given in Table X are taken from the U. S. Bureau of Mines *Bulletin* 22. These analyses were made of samples of coal as received in the laboratory. As the samples were sealed in airtight receptacles immediately after being cut in the mines, they reached the laboratory in the same condition as when cut.

55. Sizes of Coal.—Coal as taken from the mine varies in size from large lumps to fine dust. Although the heating value of a pound of combustible is the same regardless of the size of the coal, the heating value of a pound of the coal itself is generally greater in the larger sizes, as the small sizes contain a greater number of impurities. For this reason and also because the smaller sizes require special apparatus to burn them, the larger sizes command better prices. This is particularly true with anthracite.

Coal is graded into sizes by screening through standard openings. The grading for anthracite recommended by the

A.S.M.E. in Vol. 37, page 1283, of the *Transactions* is given in Table XI.

TABLE XI.—ANTHRACITE COAL SIZES

Name	Diameter of opening through or over which coal will pass, inches	
	Through	Over
Broken	$4\frac{1}{2}$	$3\frac{1}{4}$
Egg	$3\frac{1}{4}$	$2\frac{5}{16}$
Stove	$2\frac{5}{16}$	$1\frac{5}{8}$
Chestnut	$1\frac{5}{8}$	$\frac{7}{8}$
Pea	$\frac{7}{8}$	$\frac{9}{16}$
No. 1 buckwheat	$\frac{9}{16}$	$\frac{5}{16}$
No. 2 buckwheat (rice)	$\frac{5}{16}$	$\frac{3}{16}$
No. 3 buckwheat (barley)	$\frac{3}{16}$	$\frac{3}{32}$
Culm	$\frac{3}{32}$	

No classification for sizes of bituminous coal holds good for all localities. The A.S.M.E. recommendation as given in Vol. 37, page 1411, of the *Transactions* is as follows:

EASTERN BITUMINOUS COAL

a. Run-of-mine coal; the unscreened coal taken from the mine.

b. Lump coal; that which passes over a bar screen with openings $1\frac{1}{4}$ in. wide.

c. Nut coal; that which passes through a bar screen with $1\frac{1}{4}$-in. openings and over one with $\frac{3}{4}$-in. openings.

d. Slack coal; that which passes through a bar screen with $\frac{3}{4}$-in. openings.

WESTERN BITUMINOUS COAL

e. Run-of-mine coal; the unscreened coal taken from the mine.

f. Lump coal; divided into 6-in., 3-in., and $1\frac{1}{4}$-in. lump, according to diameter of the circular openings over which the respective grades pass; also 6-in. by 3-in. lump and 3-in. by $1\frac{1}{4}$-in. lump, according as the coal passes through a circular opening having the diameter of the larger figure and over one of the smaller diameter.

g. Nut coal; divided into 3-in. steam nut, which passes through an opening 3 in. in diameter and over an opening $1\frac{1}{4}$ in. in diameter; $1\frac{1}{4}$-in.

nut, which passes through a $1\frac{1}{4}$-in. diameter opening and over a $\frac{3}{4}$-in. diameter opening; and $\frac{3}{4}$-in. nut, which passes through a $\frac{3}{4}$-in. diameter opening and over a $\frac{5}{8}$-in. diameter opening.

h. Screenings; that which passes through a $1\frac{1}{4}$-in. diameter opening.

56. Weathering of Coal.—Within the last few years many manufacturing concerns and central power stations have found it necessary, or at least very desirable, to provide for the storage of large quantities of coal, in order to take advantage of market conditions, and to avoid the danger of a shortage of fuel due to crowded transportation facilities and to strikes at the mines or on the railroads.

The chief difficulties incurred in storing coal are disintegration and spontaneous combustion. Anthracite coal is practically free from disintegration and does not ignite spontaneously. It is, therefore, the ideal coal for storing, but on account of its higher cost, and for other reasons, it is not ordinarily used by large consumers.

On the other hand, nearly all bituminous coals disintegrate more or less, and will ignite if stored in large enough piles. In order to overcome the loss due to disintegration, the coal stored should be of a larger size than that to be used. As thorough ventilation is a safeguard against spontaneous combustion, the storage of larger sized coal will assist in overcoming this loss also.

The only certain preventive of spontaneous ignition is storage of the coal under water, which also minimizes the disintegration loss.

57. Clinker.—Clinker trouble is one of the most annoying problems met in power-plant operation. In a paper read before the Ohio Society of Mechanical, Electrical, and Steam Engineers, E. G. Bailey says,

> This requirement (non-clinkering) is of equal and often greater importance than the demand for a given number of British thermal units in a pound of fuel, for the formation of clinkers may retard and prevent development of heat, as clinkers affect both the capacity and the efficiency of the plant as well as the repairs to the furnace and its equipment.

Experiments made by John P. Sparrow established the fact that coals whose ash fused at a temperature above 2400 F produced no objectionable clinkers. In other words, they might be called nonclinkering coals. Whether the clinkering coals, *i.e.*, coals whose ash fused at a temperature below 2400 F, produced objectionable clinkers or not depended upon the method of firing. If the ash forming in the fuel bed is subjected to a temperature above that of the fusion point, clinkers are formed. Such an arrangement must therefore be avoided, if clinkers are to be eliminated when using clinkering coal.

58. Firing Characteristics.—Other characteristics of coals affecting their behavior in commercial service are the moisture content, the tendency to coke, and the percentage of ash.

High- or low-moisture content in coal has much to do with its ignition and rate of combustion. Some coals contain so much moisture as to be practically unusable from a commercial standpoint. Any high-moisture coal (20 per cent or more) must be given careful consideration in furnace design if proper rates of combustion are to be obtained.

The division into coking or non-coking coals may be made by observing their characteristics of disintegration between the time they are first introduced into a furnace—subjected to heat—and that time when they ignite and burn. Professor S. W. Parr of the University of Illinois states that coking coals are those in which the tars of the coal fuse and run at a temperature lower than that at which they volatilize, or are driven off as a gas. In this event, the freed tars permeating the fuel bed induce the formation of coke masses by closure of fuel particles and exclusion of air.

Conversely, if the tars of the coal are of such composition that they volatilize and are driven off as a gas before they fuse and run through the fuel bed, the coal is then said to be a non-coking coal—a free-burning coal.

To permit complete combustion of coking coals the forming of coke masses must be avoided and the fuel bed should be maintained in a fragmentary condition until the volatilization of the gas has been completed. But non-coking coals should not be agitated during this period, as they burn best with an undisturbed fuel bed.

Coals containing over 10 per cent ash are generally known as high-ash coals, and those with less than 10 per cent ash as low-ash coals. The problem of the removal of the ash from the furnace is one that must be given serious consideration when using coals with a high-ash content. The U. S. Bureau of Mines says,

The essential requirement for economical burning of high-ash fuels is a continuous and automatic removal of ash from the furnace, so that the operation of the boiler need not be interfered with by frequent cleaning of fires.

59. Other Solid Fuels.—*Peat* is an intermediate between wood and coal. It is formed from the immense quantity of rushes, sedges, and mosses that grow in the swampy regions of the temperate zone. These in the presence of heat and moisture are subject to a chemical change which leaves behind the hydrocarbons, fixed carbon, and 70 to 80 per cent of moisture. It is usually cut in blocks and air dried. Good air-dried peat contains about 60 per cent of carbon, 6 per cent of hydrogen, 31 per cent of oxygen and nitrogen, and 3 per cent of ash. The following table gives the heat value of some of the different peats:

TABLE XII.—CALORIFIC VALUE OF PEATS

Location	Fixed carbon	Volatile matter	Ash	Btu per pound combustible
Maine	29.7	66.1	4.2	8,900
Southern Michigan	29.0	68.5	2.3	9,500
New York	29.2	65.6	8.25	10,200
Wisconsin	27.6	60.5	11.8	8,250

Woods may be divided into two general classes: soft and hard. The commonest hardwoods are oak, hickory, maple, beech, and walnut. The commonest softwoods are pine, elm, birch, poplar, and willow. When first cut, wood contains about 50 per cent of moisture, but after being dried this is reduced from 10 to 20 per cent. The following table gives the chemical composition and the heat value of some of the more common woods (from Poole's "Calorific Value of Fuels"):

TABLE XIII.—CALORIFIC VALUE OF WOODS

Name	C	H	O	N	Ash	Btu per pound combustible
Ash	49.2	6.3	43.9	0.07	0.57	8,480
Beech	49.0	6.1	44.2	0.09	0.57	8,590
Birch	48.9	6.0	44.7	0.10	0.29	8,590
Elm	48.9	6.2	44.3	0.06	0.50	8,510
Oak	50.2	6.0	43.4	0.09	0.37	8,320
Pine	50.3	6.2	43.1	0.04	0.37	9,150

In boiler tests a pound of wood is usually assumed as equal to 0.4 lb of coal, or a pound of coal equals $2\frac{1}{2}$ lb of wood.

Bagasse, or the refuse of sugar cane after the juice has been extracted, is very commonly used as fuel for boilers on sugar plantations.

The fuel value of bagasse depends upon the amount of woody fiber it contains and the amount of combustible matter, such as glucose and sucrose, in the liquid remaining in the fuel.

Wet bagasse contains from 35 to 50 per cent of woody fiber, less than 10 per cent of glucose, sucrose, etc., and from 40 to 55 per cent of water.

The heating value of dry bagasse is about 8,300 Btu per lb, and of the fuel as fired, 4,000 Btu or less, due to the moisture content.

60. Powdered Coal.—Although powdered coal has been used extensively in the cement industry and in metallurgical processes for the past 35 or 40 years, it is only since about 1920 that furnaces have been designed to permit its successful use under steam boilers. At the present time, it is common practice in the largest and most recent central power stations to use the lower grades of coal in the powdered form.

When coal is burned in fuel beds on grates, clinker trouble is liable to occur. The use of powdered coal has greatly reduced this difficulty by eliminating fuel beds. Furthermore, the use of this form of fuel has made possible an increase in the variety of coals that can be burned in any given plant and in the capacity and flexibility of the furnace.

The capacity of a furnace in which the combustion of the fuel takes place on grates is limited by the area of the grates, whereas the capacity of the pulverized coal unit is limited only by the combustion space which may be increased by increasing the area of the horizontal cross section of the boiler, or by making the furnace higher or deeper.

In smaller units using pulverized coal, a system of dry-ash removal is employed. This is also the case in larger furnaces where relatively high-fusing ash is available.

Among the advantages then of this fuel are

1. Reduction in clinker trouble.
2. Ability to use coal of any grade.
3. Increased capacity of furnace.
4. Greater flexibility.
5. Ease in removing ash.

6. Absence of moisture in coal as fired.
7. Smaller stack gas loss due to reduction in excess air.
8. Ease with which the fuel may be changed from coal to oil or gas, or vice versa.

Against these advantages must be weighed

1. The cost of pulverizing and preparing the coal for use, including both maintenance and operating charges of the pulverizing machinery.
2. Furnace depreciation.
3. The nuisance caused by the fine dust.
4. Increased liability of trouble with ash and slag.

61. Liquid Fuels.—Of the various liquid fuels, the one in general use for firing boilers is petroleum, either crude or refined. This is found in large quantities in the United States, Mexico, and Russia, and in smaller amounts in various other countries. It has been used in Russia in locomotives and mârine boilers for years, the oil fields at Baku on the Caspian Sea being among the most famous in the world.

In this country, the railroads running to southern California were early users of oil as a fuel, and its use in marine work is increasing tremendously every year. At the present time many of the largest vessels afloat are oil burners.

Crude oil, or petroleum, is roughly classified as *paraffin base*, or *asphalt base*, depending upon the residue left after distillation.

Paraffin-base oil is found chiefly in the central and eastern portions of the United States. Owing to improved methods of refining, the number of heavy distillates from this oil have been so reduced that their price as fuel for power boilers has become almost prohibitive.

The oils found in the southwestern and Pacific Coast states and in Mexico and Russia are largely asphalt base. They are heavier and darker in color than those with a paraffin base and provide most of the fuel oil used for steam-boiler plants.

Some of the advantages of oil over coal as a fuel are

1. Simplification in methods of handling and storing.
2. Elimination of dust and ashes, and reduction of dirt in furnace room to a minimum.
3. Reduction of labor and of firing tools or equipment.
4. Easier control, and therefore lowering, of fuel waste.
5. Greater flexibility of operation.

6. Higher efficiency.
7. Great reduction in stand-by losses.
8. Possibility of keeping rate of steaming uniform, there being no loss due to cleaning fires, etc.
9. Reduced cost of maintenance. Fewer repairs on boiler due to uniform temperature in furnace and combustion chamber.

The demand for oil is so great that it is not possible to meet it; therefore, the cost as compared with that of coal is considerably higher for the same number of heat units. In spite of this fact, the saving in the cost of handling, firing, etc., makes oil the cheaper fuel in the end in many cases.

The efficiency of the average oil-fired plant is much higher than that of a hand-fired coal-burning plant, but about the same as that of a modern stoker-fired plant.

Recent results obtained with pulverized coal indicate that just as high efficiencies can be obtained with that fuel as with oil.

Petroleum and its distillates consist largely of hydrocarbon compounds together with some moisture, sulphur, and other impurities. Inspection of Table XIV, taken from *Power*, shows that in spite of the widely scattered source of the oils, the carbon content of these compounds varies only from about 82 to 85 per cent and the hydrogen from about 12 to 15 per cent. It is not stated how the computed calorific values were found.

TABLE XIV.—PROPERTIES OF AMERICAN OILS

Kind of oil	Composition by weight, per cent				Specific gravity	Pounds per gallon	Btu per pound	
	Carbon	Hydrogen	Sulphur	Oxygen			By test	Computed
Ohio	83.4	14.7	0.6	1.3	0.800	6.68	19,580	19,718
Pennsylvania, light	82.0	14.8	1.0	2.2	0.816	6.80	19,930	19,519
Pennsylvania, heavy	84.9	13.7	...	1.4	0.886	7.40	19,210	19,385
West Virginia, light	84.3	14.1	0.3	1.3	0.841	7.02	18,400	18,527
West Virginia, heavy	83.5	13.3	0.8	2.4	0.873	7.28	18,324	18,860
Texas	84.0	13.2	1.0	1.8	0.925	7.71	19,100	18,928
California	85.2	12.4	0.5	1.9	0.959	8.00	18,500	18,656
Average	83.9	13.9	0.7	1.8	0.871	7.27	19,006	19,086

In general, the calorific value of oil used for firing boilers may be assumed as running from 18,000 Btu to 19,500 Btu per lb, no matter whether it is crude or refined. On the other hand, the calorific value of coal varies greatly, as was seen in Table X.

62. Gaseous Fuels.—These fuels are of two general classes: *natural* and *artificial.* The artificial gases are blast-furnace gas and coke-oven gas, both of which are by-products; producer gas; coal gas; and water gas. Both the natural and the manufactured gas are ideal fuels on account of their cleanliness and high efficiency and because of the simplicity with which they may be controlled and the saving in labor, but they are too valuable for general use as a fuel for steam-boiler plants. They are used more frequently in internal-combustion engines where a higher thermal efficiency is obtained than in steam plants. A more extended discussion of fuels for internal-combustion engines will be found in Chap. XIV.

63. Heat of Combustion.—The term *combustion* as applied here refers to the rapid chemical union of oxygen with some other substance producing heat. The perfect combustion of ordinary fuel should result in carbon dioxide, nitrogen, water vapor, and a trace of sulphur dioxide.

The calorific power, or heating value, of a fuel is the total amount of heat developed by the complete combustion of a unit weight, or volume, of fuel. The heating value of solid and liquid fuels is expressed in Btu per pound, while that of gaseous fuels is expressed in Btu per cubic feet of gas when measured at a pressure of 30 in. Hg and at a temperature of 60 F.

When a fuel is burned, water vapor is formed, which will be condensed only when the temperature of the products of combustion falls below the boiling point corresponding to the pressure. Owing to the low partial pressure at which this vapor exists in the mixture of stack gases, the temperature at which the vapor will condense is considerably below 212 F. As long as this water remains in the form of vapor, the heat necessary to maintain it as such, *i.e.*, the enthalpy of vaporization of steam at the partial pressure of the water vapor in the gas times the weight of vapor, is unavailable for use. The difference between the *higher heating value,* which is that as determined by a calorimeter, and this unavailable heat is called the *lower heating value.* This is the "available calorific value" in nearly all cases. For example, in a boiler plant the temperature of the stack gases, and in a gas engine the temperature of the exhaust, are both above the temperature of the boiling point of water; therefore, the heat actually available for use in either case is the lower heating value of the fuel. In Europe the lower heating value is generally used as the basis for comparison of boiler and gas-engine performance. However, the A.S.M.E. Power Test Committee recommends the use of the higher heating values in comparing efficiencies and other performance figures of heat engines "in order that all heat apparatus shall be charged with heat supplied, on the same basis."

When a coal is analyzed, the percentage of hydrogen shown includes not only the free hydrogen in the sample but also that which existed in combination with the oxygen in the form of water. As 16 parts by weight of oxygen unite with 2 parts of hydrogen, the weight of hydrogen which was in combination with the oxygen will be equal to one-eighth the total weight of oxygen. The balance of the hydrogen is available for producing heat, and in determining the heat value of a fuel the number of Btu in the coal may be found from the ultimate analysis by the following formula.

Heat value of fuel in Btu per pound

$$= 14{,}600C + 62{,}000\left(H - \frac{O}{8}\right) + 4{,}000S, \quad (1)^1$$

where the symbols C, H, O, and S represent the weights of carbon, hydrogen, oxygen, and sulphur in one pound of the fuel. This is called Du Long's formula.

As will be seen by reference to Table XV, page 114, the figures 14,600, 62,000, and 4,000 in Eq. (1) are the number of Btu given off when one pound each of carbon, hydrogen, and sulphur are burned, respectively, to CO_2, H_2O, and SO_2. If the combustion of C is incomplete and CO is formed instead of CO_2, only 4,430 Btu are liberated per pound. This is the result if an insufficient amount of air is supplied, or if the air and the volatile gases are not thoroughly mixed. This process in a furnace is an inefficient one, as much heat is lost which would have been available under proper conditions. If an excess of air is supplied, there is a

[1] These are the values recommended by the A.S.M.E. in the Power Test Code. The U. S. Bureau of Mines states the formula as

$$14{,}544C + 62{,}028\left(H - \frac{O}{8}\right) + 4{,}050S.$$

resultant heat loss up the stack due to the necessity of raising the temperature of this air up to the stack temperature.

The heat value obtained from Eq. (1) is *only an approximate result;* where greater accuracy is desired, it is necessary actually to test the coal experimentally in a *coal calorimeter.*

64. Fuel Calorimeters.—One form of calorimeter very commonly used for determining the heating value of solid and liquid fuels is the Burgess-Parr bomb calorimeter (Fig. 21). This

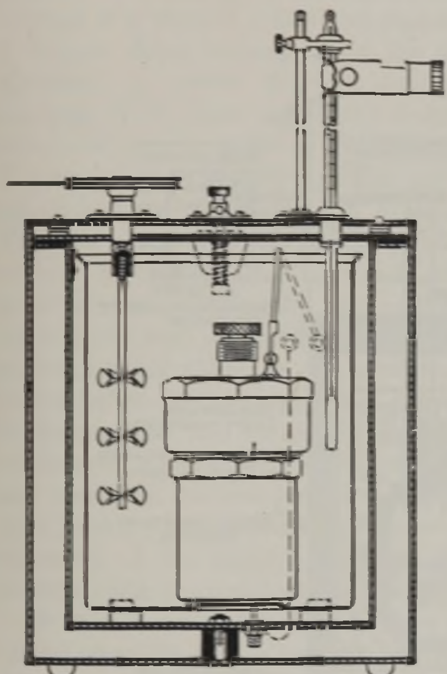


FIG. 21.—Burgess-Parr oxygen bomb calorimeter.

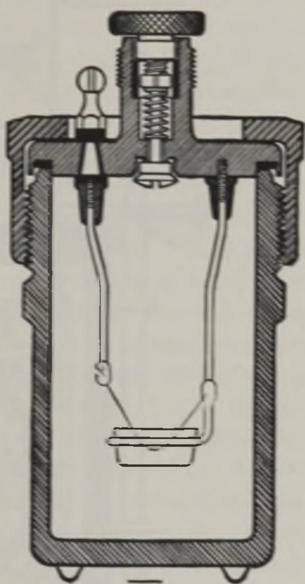


FIG. 22.—Bomb in Burgess-Parr calorimeter.

consists of a strong steel vessel into which a known weight (usually 1 g) of the liquid fuel, or of finely powdered, air-dried coal is introduced. This coal is placed in a cup or dish suspended from the cover of the bomb by a platinum-wire electrode. Another wire passes through the cover, although well insulated from it, and extends down into the fuel. The cover is then screwed down tight and the bomb charged with oxygen to a pressure of from 300 psi to 400 psi. This allows a considerable excess of oxygen over that theoretically required for the combustion of the fuel. After the bomb has been charged, it is placed in a vessel containing a known weight of water and an electric current is passed through the wire electrodes, igniting

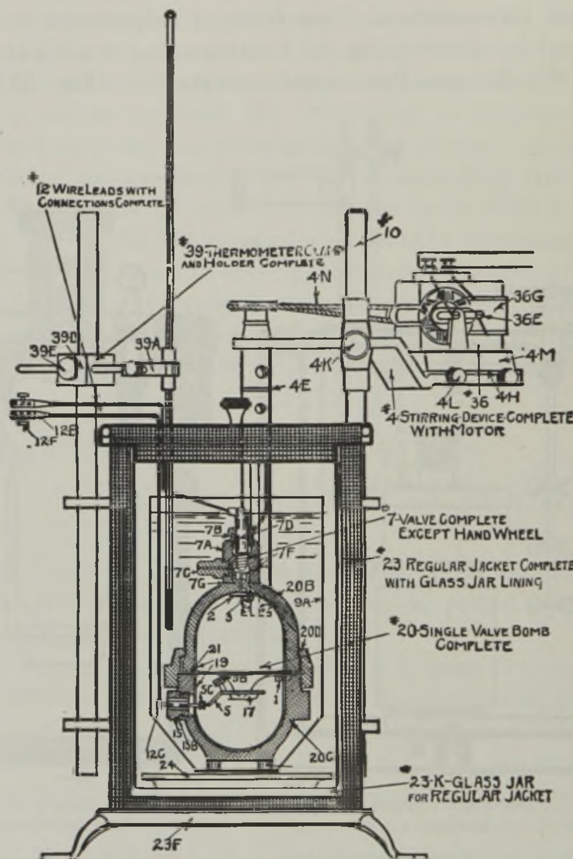


FIG. 23.—Emerson fuel calorimeter.

the fuel. While the combustion is going on, the water in the containing vessel is kept thoroughly stirred by the apparatus.

The rise in temperature of the water is carefully noted, and, after making allowances for radiation, the heat generated by the electric current, etc., the heating value of the fuel can be computed, since the heat gained by the water must equal the heat given up by the fuel (after the allowances just mentioned have been made).

Another calorimeter very largely used for the determination of the heat value of coal, coke, fuel oil, gasoline, etc., is the Emerson (Fig. 23). The method followed in finding the calorific values of fuels is the same in this as in the Burgess-Parr calorimeter.

For measuring the calorific value of gaseous fuels, a continuous-flow calorimeter such as the Junker must be used. This consists of a vertical cylindrical copper shell with a water jacket. A Bunsen flame burning inside the shell supplies heat which is absorbed by water flowing through the jacket. Frequent readings of the inlet and outlet temperatures of the water are taken and from them the average rise in temperature is computed. The weight of water and the cubic feet of gas flowing through the calorimeter in a given time are both accurately measured. The heat given up by the gas is equal to the heat gained by the water. By multiplying the weight of water by its rise in temperature and dividing by the cubic feet of gas, the calorific value of the fuel per cubic foot may be determined. This is the higher heating value, as the products of combustion have been cooled below the temperature of the boiling point corresponding to the pressure, and the water vapor which was formed has been condensed. The lower heating value may be found by collecting and weighing this condensed vapor and multiplying it by the latent heat. The product will be the heat liberated by the condensation of the vapor. If this is subtracted from the higher heating value the result will be the lower heating value of the gas.

65. Air Theoretically Required for Combustion.—The oxygen furnished to the fuel in order to burn it is obtained from the air. Air is a mechanical mixture containing by weight 23.15 per cent oxygen and 76.85 per cent nitrogen, and by volume 20.9 per cent oxygen and 79.1 per cent nitrogen. The oxygen only is used in the combustion of the fuel, the nitrogen being an inert gas and having no chemical effect upon the combustion.

When hydrogen is burned to water, H_2O, $16 \div 2 = 8$ lb of oxygen are required for the complete combustion of 1 lb of hydrogen, since the molecular weight of oxygen is 16 and of hydrogen 1; and for the complete combustion of 1 lb of carbon to carbon dioxide, CO_2, $32 \div 12 = 2.667$ lb of oxygen are required.

For each pound of hydrogen there will be required $\frac{8}{0.2315} = 34.56$ lb of air, and for each pound of carbon $\frac{2.667}{0.2315} = 11.52$ lb of air to produce complete combustion. If the combustion of the carbon is incomplete so that carbon monoxide, CO, is formed instead of carbon dioxide, the oxygen required will be

$$16 \div 12 = 1.333 \text{ lb per lb of carbon,}$$

and the air necessary will be $\frac{1.333}{0.2315} = 5.76$ lb.

In case the fuel contains sulphur, there will be required $32 \div 32 = 1$ lb of oxygen for the complete combustion to SO_2. The air necessary in this case will be $\frac{1}{0.2315} = 4.32$ lb.

As has already been stated, the oxygen in the fuel unites with its equivalent of hydrogen to form water. In determining the weight of air theoretically required for combustion this hydrogen should be disregarded. The air theoretically required for the *complete* combustion of any fuel may then be found from its analysis by the following expression:

Weight of air per pound of fuel,

$$W_t = 11.52\text{C} + 34.56\left(\text{H} - \frac{\text{O}}{8}\right) + 4.32\text{S}. \tag{2}$$

The pounds of air required per pound of fuel when the combustion is incomplete and CO is formed will be

$$11.52\text{C}_1 + 5.76\text{C}_2 + 34.56\left(\text{H} - \frac{\text{O}}{8}\right) + 4.32\text{S}, \tag{3}$$

where C_1 and C_2 represent respectively the weights of carbon burned to CO_2 and CO for each pound of fuel.

Since two volumes of hydrogen combine with one volume of oxygen, $\frac{1}{2}$ cu ft of oxygen will be required for the complete combustion of 1 cu ft of hydrogen, or for each *cubic foot* of H there will be required $\frac{0.5}{0.209} = 2.39$ cu ft of air; and since 12 lb of C unite with 378 cu ft of O_2 to form 378 cu ft CO_2, 378 being the number of standard cubic feet (*i.e.*, at 60 F and under a pressure of 30 in. Hg) in a pound molecular weight, 1 *lb* of C will require $\frac{378}{12} = 31.5$ standard cu ft O_2, or $\frac{31.5}{0.209} = 150.7$ cu ft of air, to produce complete combustion.

In Eqs. (1) and (2) it has been assumed that there was perfect combustion, by which is meant the complete burning of the fuel with the exact chemical requirement of oxygen, in which case the products of combustion would contain neither combustible material nor free oxygen. In actual practice this condition does not exist.

However, complete combustion can be accomplished if more than the chemical requirement of oxygen is furnished; the products of this process would contain no combustible material, but would contain free oxygen. Whether the attainment of complete combustion is the ideal to strive for in the operation of a particular furnace depends upon the conditions peculiar to that furnace. A simple statement that would be generally applicable cannot be made concerning this question.

The basic requirements for the attainment of complete combustion are these:

1. Enough air must be supplied so that every particle of fuel may receive its quota of oxygen.
2. The fuel and air must be mixed so thoroughly that each particle of fuel is made to come in contact with enough oxygen to consume it.
3. The temperature of the fuel and air must be maintained at a level above the ignition temperature of the fuel particles for a long enough time to permit the burning to be completed.

These statements might be condensed into the form "Enough air, enough mixing, enough temperature, and enough time." These requirements are general, applicable to all fuels and all furnaces; their simplicity does not reveal the complexities that arise in their application to furnace operation.

How much is "enough air"? Or, more to the point, how can one judge whether the air supply is the optimum for a particular furnace and fuel? Human senses, unaided by instruments or records, are not capable of rendering a satisfactory answer to the question. Instruments capable of analyzing the products of combustion must be used. Of these, the Orsat analyzer, described in Art. 67, is probably the most satisfactory for studying the combustion conditions. When properly handled, it will permit the determining of the amount of carbon dioxide, oxygen, and carbon monoxide in the gaseous products of combustion with sufficient exactness; it does not permit the measuring of the amount of water vapor resulting from combustion, nor does it detect free hydrogen or hydrocarbons if they are present in the gaseous products.

Free oxygen should be found in the gases emerging from the furnace, but the presence of oxygen here is not, of itself, a guarantee that enough air is being supplied in the combustion zone. If insufficient mixing occurs, oxygen may pass through the furnace without being used and appear in the flue gases, even though the total supply of oxygen is inadequate for complete combustion. If cooling of the gaseous materials is permitted to occur too early, the combustion process may be terminated before completion, and the unused oxygen may appear in the flue gases, even though an insufficient supply of air is being furnished. Again, if leaks permit the entrance of air into the furnace or gas passages, but outside of the combustion zone, its oxygen will appear in the analysis from the Orsat. It is evident that supplementary information is needed in judging the sufficiency of the air supply.

If the gas analysis indicates that carbon monoxide is emerging from the furnace, one can be certain that combustion is not complete. However, the failure to burn the carbon to carbon dioxide may be due to insufficient air, or to insufficient mixing of air and fuel, or to premature cooling of the burning gaseous material, or to a combination of these.

It is evident, then, that the Orsat analyzer can indicate whether the combustion is complete or incomplete; if it is incomplete, the apparatus does not point out the cause of the difficulty directly, except for this case: the presence of carbon monoxide and little or no free oxygen in the gaseous products is a quite definite indication that the air supply is insufficient for complete combustion.

If there is reason to suspect that the mixing of fuel and air is not vigorous enough, experimenting with the furnace structure or plan of operation will be most likely to yield a convincing answer. Make some change calculated to increase the turbulence of flow of the gaseous materials, and observe whether the change results in an improvement in combustion. Probably the simplest changes that might be suggested would involve changing the location, and perhaps the sizes, of the air admission openings, and increasing the velocity of the air at the point of admission. Turbulence is of most importance in the regions where the distillation of volatile material from the fuel is occurring, and where carbon monoxide is emerging from the reducing zone of a fire bed. Increasing the turbulence in the cases of powdered fuel, liquid fuel, or gas would involve alterations to the burners themselves or changes in their location. It is to be noted that increased turbulence means increased friction losses in moving the gases, and hence more draft will be needed, whether furnished by a chimney or by mechanical draft equipment. The possibility that the cost of the improvements will exceed the value of the results obtained needs to be inquired into.

If premature cooling of the gaseous materials is suspected as being the cause of incomplete combustion, experimenting is again the procedure most likely to yield a conclusive answer. However, changes that will increase the time allowed for burning are likely to be rather expensive, and it may well be that the improvement in combustion would not justify the expenditures for making such changes.

The time available for the burning of the gaseous materials depends upon their velocity and upon the length of travel from the fire bed to the water-cooled boiler surfaces; this length of travel is difficult to change after the boiler has been erected. The needed distance also depends upon the kind of fuel being burned, increasing as the amount of volatile material in the fuel increases. Note that the important furnace dimension here is not volume, but length measured along the path of the burning gases. Burners that introduce powdered fuel, gas, or oil with rotary motion provide a length of travel for the fuel particles and air which is considerably greater than the straight-line length of the flame.

How hot should the fuel and air mixture be kept in order to ensure the completion of combustion? The ignition temperature of the gaseous combustible materials that are likely to be encountered in the furnace will fall in the range 900 F to 1200 F; the glow-point of the usual solid fuels will fall within the range 850 F to 1150 F. These seem rather low compared to combustion chamber temperatures of 2000 deg and more, and suggest the thought that there ought not to be any difficulty in holding the temperatures high enough in any furnace to permit completion of combustion. But the temperature of the air, even if preheated, at the entrance to the combustion chamber is quite low compared with the ignition temperatures of the fuel components, and heat not only must be available, but must be applied to heating the air and fuel above the ignition temperature. Furthermore, the water-cooled surfaces of a boiler are really cold in comparison with the temperature of the flame, and loss of heat from the burning mixture becomes quite rapid as approach to the cool surfaces is made. A familiar example that may offer some illumination on these points is the blowing out of a lighted match. The principal action here is the chilling of the distilling and burning gases by the flood of cool air directed at the flame; the other factor entering into the action is the propelling of the gases away from their source at a higher velocity than that with which the flame can travel and hence stopping the distillation of the gases by removing the heat supply. Another example is that of a coke fire which has been checked too severely by shutting off the draft; an attempt to revive it by merely opening the draft and restoring the normal air supply is usually followed by complete extinguishment of the fire; it is "frozen to death" by the air supply.

When more air is supplied than is needed to obtain complete combustion, the tendency is for the temperature of the gases leaving the combustion zone to decrease; the quantity of heat wasted in the flue gases tends to increase. These considerations suggest that the amount of air supplied should not exceed that needed for complete combustion. However, since *total* cost of operation of the furnace is usually the criterion, other factors may dictate the use of more or less air than the minimum which will result in complete combustion.

Theoretically, most coals require for complete combustion, approximately 12 lb, or 155 standard cu ft of air. In actually burning coal under a boiler with natural draft, it is found that the coal requires about 24 lb, or 310 standard cu ft of air per lb of coal. For forced draft there are usually required about 18 lb, or 232 standard cu ft of air per lb of coal.

In the actual operation of a boiler plant, one of the most important considerations is the admission of a proper quantity of air to the fire. As will be seen later, the less the quantity of air given to the fire the better the efficiency of combustion, provided enough air enters so that all the carbon is burned to CO_2.

Excess air in a furnace is a source of waste, since it lowers the temperature of combustion by dilution, retards the heat absorption by decreasing the difference in temperature between the hot gases and the boiler content, and increases the volume of the products of combustion so that the chimney gases carry off more heat.

The heat given off per pound by the elements ordinarily met with in fuels, together with the air required for combustion and the combining volumes and weights, is shown in Table XV.

66. Smoke.—Smoke is unburned carbon in a finely divided state. The amount of carbon carried away by the smoke is usually small, not exceeding 1 per cent of the total carbon in the coal. Its presence, however, often indicates improper handling of the boiler, which may result in a large waste of fuel. Smoke is produced in a boiler when the incandescent particles of carbon are cooled before coming into contact with sufficient oxygen to unite with them. It is necessary that the carbon be in an incandescent condition before it will unite with the oxygen. Any condition of the furnace that results in the cooling of carbon below the point of incandescence before sufficient oxygen has been furnished to unite with it will result in smoke. Smoke once formed is very difficult to ignite and the boiler furnace must be handled so as not to produce smoke. Fuels rich in hydrocarbons are most apt to produce smoke. When the hydrocarbon gas liberated from the coal is kept above the temperature of ignition and sufficient oxygen for its combustion added, it burns with a red, yellow, or white flame. The slower the combustion, the larger the flame. When the flame is chilled because the cold heating surfaces near it take away heat by radiation, combustion may be incomplete, and part of the gas and smoke pass off unburned. If the boiler is raised so high above the grate as to give room for the volatile matter to burn and not strike the boiler surface at once, both the amount of smoke given off and the quantity of coal used will be reduced.

67. Analysis of Flue Gases.—In all large powerhouses and carefully conducted power plants the flue gases leaving the

TABLE XV.—COMBUSTION PROPERTIES OF ELEMENTS

Combustion	Combustion formula	Combining volumes	Combining weights	Higher heating value per pound, Btu	Higher heating value per standard cubic foot, Btu	Air required per pound, pounds
H to H_2O (water)	$2H_2 + O_2 = 2H_2O$	2 vol H_2 + 1 vol O_2 = 2 vol H_2O	2 lb H_2 + 16 lb O_2 = 18 lb H_2O 1 lb H_2 + 8 lb O_2 = 9 lb H_2O	62,000	328	34.56
C to CO_2 (carbon dioxide)	$C + O_2 = CO_2$	1 vol C + 1 vol O_2 = 1 vol CO_2 12 lb C + 378 cu ft O_2 = 378 cu ft CO_2 1 lb C + 31.5 cu ft O_2 = 31.5 cu ft CO_2	12 lb C + 32 lb O_2 = 44 lb CO_2 1 lb C + 2.66 lb O_2 = 3.66 lb CO_2	14,600	.....	11.52
C to CO (carbon monoxide)	$2C + O_2 = 2CO$	2 vol C + 1 vol O_2 = 2 vol CO 24 lb C + 378 cu ft O_2 = 756 cu ft CO 1 lb C + 15.75 cu ft O_2 = 31.5 cu ft CO	12 lb C + 16 lb O_2 = 28 lb CO 1 lb C + 1.33 lb O_2 = 2.33 lb CO	4,430	.....	5.76
CO to CO_2	$2CO + O_2 = 2CO_2$	2 vol CO + 1 vol O_2 = 2 vol CO_2	28 lb CO + 16 lb O_2 = 44 lb CO_2 2.33 lb CO + 1.33 lb O_2 = 3.66 lb CO_2 1 lb CO + 0.57 lb O_2 = 1.57 lb CO_2	$\left(\frac{10,170}{2.33} = 4,365\right)$	324	2.46
S to SO_2 (sulphur dioxide)	$S + O_2 = SO_2$	1 vol S + 1 vol O_2 = 1 vol SO_2 32 lb S + 378 cu ft O_2 = 378 cu ft SO_2 1 lb S + 11.8 cu ft O_2 = 11.8 cu ft SO_2	32 lb S + 32 lb O_2 = 64 lb SO_2 1 lb S + 1 lb O_2 = 2 lb SO_2	4,000	.....	4.32
CH_4 (methane or marsh gas)	$CH_4 + 2O_2 = CO_2 + 2H_2O$	1 vol CH_4 + 2 vol O_2 = 1 vol CO_2 + 2 vol H_2O	16 lb CH_4 + 64 lb O_2 = 44 lb CO_2 + 36 lb H_2O 1 lb CH_4 + 4 lb O_2 = $2\frac{3}{4}$ lb CO_2 + $2\frac{1}{4}$ lb H_2O	23,840	1,009	17.28
C_2H_2 (acetylene)	$2C_2H_2 + 5O_2 = 4CO_2 + 2H_2O$	2 vol C_2H_2 + 5 vol O_2 = 4 vol CO_2 + 2 vol H_2O	52 lb C_2H_2 + 160 lb O_2 = 176 lb CO_2 + 36 lb H_2O 1 lb C_2H_2 + 3.08 lb O_2 = 3.38 lb CO_2 + 0.7 lb H_2O	21,830	1,531	13.30
C_2H_4 (ethylene or olefiant gas)	$C_2H_4 + 3O_2 = 2CO_2 + 2H_2O$	1 vol C_2H_4 + 3 vol O_2 = 2 vol CO_2 + 2 vol H_2O	28 lb C_2H_4 + 96 lb O_2 = 88 lb CO_2 + 36 lb H_2O 1 lb C_2H_4 + 3.43 lb O_2 = 3.14 lb CO_2 + 1.29 lb H_2O	21,830	1,620	14.82

boilers are analyzed from time to time. In some cases records are kept, by an automatic device, of the percentage of carbon dioxide in the flue gases. In analyzing the flue gases it is customary to use some modification of the Orsat analyzer. This consists of three pipettes, a measuring tube, and a leveling bottle, as shown in Fig. 24. The first pipette *D* contains a saturated solution of potassium or sodium hydrate and absorbs CO_2;

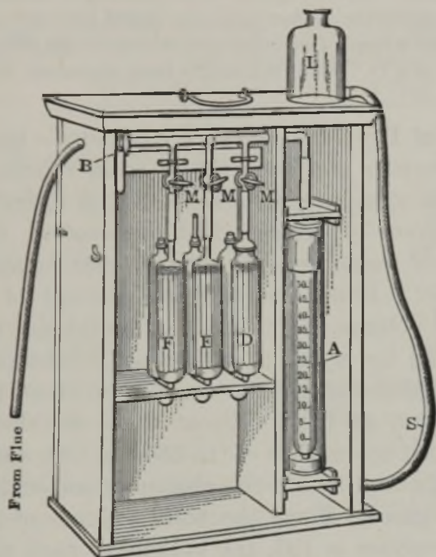


FIG. 24.—Orsat analyzer.

the second pipette *E* contains potassium pyrogallate and absorbs O_2; and the third pipette *F* contains cuprous chloride and absorbs CO. The gas is passed through the pipettes in the order named, and the remainder is assumed to be nitrogen. The readings obtained from this apparatus give the percentage composition of the gases by volume.

The following directions will show how the reagents used in the Orsat apparatus are prepared:

Potassium Hydrate.—1. For the determination of CO_2, dissolve 500 grams of the commercial hydrate in 1 liter of water. One cubic centimeter of this solution will absorb 40 cc of CO_2.

2. For the preparation of potassium pyrogallate for use in case the percentage of oxygen is high, dissolve 120 grams of the commercial hydrate in 100 cc of water.

Potassium Pyrogallate.—Put 5 grams of the solid pyrogallic acid in a funnel placed in the neck of the pipette *E*, and pour over this 100 cc of potassium hydrate, solution 1 or 2. Solution 1 may be used in case there is not more than 25 per cent of O_2 in the gas. Otherwise, solution 2 must be used or CO may be given up. One cubic centimeter of this solution absorbs 2 cc of O_2.

Cuprous Chloride.—Pour from $\frac{1}{4}$ in. to $\frac{1}{2}$ in. of copper scale into a 2-liter bottle and also place in the bottle a number of long pieces of copper wire. Then fill the bottle with hydrochloric acid of 1.10 sp gr (1 part muriatic acid to 1 part water). Let the bottle stand, shaking it occasionally until the solution becomes colorless. Then pour the liquid into the pipette *F*, which is filled with copper wires. One cubic centimeter of this solution will absorb from 1 cc to 2 cc of CO. After 0.2 cc has been absorbed, the absorption is incomplete.

68. Weight of Dry Flue Gases.—Avogadro's law states that *at the same pressure and temperature equal volumes of different gases contain the same number of molecules;* in other words, *under the same conditions of pressure and temperature, the weights of equal volumes of gases must be proportional to their atomic or molecular weight.* From this law the concept of a molecular volume, or *mol* volume, is derived. A pound mol volume is the volume occupied by a pound molecular weight of any gas at standard conditions (approximately 378 cu ft at 14.7 psia and 60 F). Therefore, 44 lb of CO_2 at these standard conditions occupies 378 cu ft as do 32 lb of O_2, 28 lb of CO, and 28 lb of N_2, where 44, 32, 28, and 28 are the atomic or molecular weights of the respective gases. Since the total volume of gas analyzed by an Orsat analyzer is 100, the volume of each constituent as shown by the apparatus is in effect the percentage by volume of each gas going to make up the total.

Each cubic foot of CO_2 weighs $\frac{44}{378}$ lb, so a 100-cu ft sample of gas having 10 per cent CO_2 will have $10 \times \frac{44}{378}$ lb of CO_2 in it. From this it can be seen that the total weight of 100 cu ft of dry flue gas will be

$$\frac{44CO_2}{378} + \frac{32O_2}{378} + \frac{28CO}{378} + \frac{28N_2}{378}, \tag{4}$$

providing the several constituents exist under standard conditions and CO_2, O_2, CO and N_2 are per cent by volume.

The weight of C in a pound of CO_2 is $\frac{12}{44}$ lb, and in a pound of CO there is $\frac{12}{28}$ lb of C; therefore, the total weight of C in the 100 cu ft of gas will be

$$\frac{12}{44} \times \frac{44}{378}CO_2 + \frac{12}{28} \times \frac{28}{378}CO. \tag{5}$$

The ratio of total weight of 100 cu ft of gas to the weight of C in that gas will be

$$\frac{\dfrac{44CO_2}{378} + \dfrac{32O_2}{378} + \dfrac{28CO}{378} + \dfrac{28N_2}{378}}{\frac{12}{44} \times \frac{44}{378}CO_2 + \frac{12}{28} \times \frac{28}{378}CO}. \tag{6}$$

Then, W_1, the weight of dry flue gas formed per pound of carbon burned is

$$W_1 = \frac{44CO_2 + 32O_2 + 28CO + 28N_2}{12CO_2 + 12CO} \tag{7}$$

$$= \frac{11CO_2 + 8O_2 + 7(CO + N_2)}{3(CO_2 + CO)}. \tag{8}$$

Equation (8) may also be expressed as

$$W_1 = \frac{11CO_2 + 8O_2 + 7[(100 - CO_2 - O_2 - N_2) + N_2]}{3(CO_2 + CO)},$$

$$= \frac{11CO_2 + 8O_2 + 700 - 7CO_2 - 7O_2 - 7N_2 + 7N_2}{3(CO_2 + CO)},$$

$$= \frac{4CO_2 + O_2 + 700}{3(CO_2 + CO)}. \tag{9}$$

This equation is independent of the condition of the gas, providing all constituents are at the same condition, and it is also independent of the size of sample.

In a pound of fuel ordinarily there will be less than a pound of C; also under most conditions some of this carbon will not be burned.

The weight of carbon burned per pound of fuel fired will be

$$C_b = \frac{W_f C_f - W_a C_a}{W_f \times 100}, \tag{10}$$

where C_b = weight of carbon burned per pound fuel fired;
W_f = weight of fuel fired in pounds;
C_f = per cent of carbon in fuel fired;
W_a = weight of ash and refuse in pounds;
C_a = per cent of carbon in refuse.

Then, W_2, the weight of dry flue gas formed per pound of dry fuel fired, will be

$$W_2 = C_b\left(\frac{44CO_2 + 32O_2 + 28CO + 28N_2}{12CO_2 + 12CO}\right), \tag{11}$$

$$= \left[\frac{W_f C_f - W_a C_a}{W_f \times 100}\right]\left[\frac{11CO_2 + 8O_2 + 7(CO + N_2)}{3(CO_2 + CO)}\right]. \tag{12}$$

or

$$W_2 = \left[\frac{W_fC_f - W_aC_a}{W_f \times 100}\right]\left[\frac{4CO_2 + O_2 + 700}{3(CO_2 + CO)}\right]. \tag{13}$$

In case all the carbon in the coal is burned (and this may be assumed if no carbon content of the refuse is given), C_b will equal $\frac{C_f}{100}$ and Eqs. (12) and (13) become, respectively,

$$W_2 = \frac{C_f}{100}\left[\frac{11CO_2 + 8O_2 + 7(CO + N_2)}{3(CO_2 + CO)}\right] \tag{14}$$

and

$$W_2 = \frac{C_f}{100}\left[\frac{4CO_2 + O_2 + 700}{3(CO_2 + CO)}\right]. \tag{15}$$

69. Weight of Air Actually Used.—In Art. 65 the weight of air theoretically required for combustion was determined. As stated there, the weight actually used is in excess of the theoretical requirement. Per pound of coal fired, it may be found in a manner similar to that used in finding the weight of dry flue gases in Art. 68.

Most solid and liquid fuels have a very small amount of nitrogen so that for all practical purposes the nitrogen appearing in the products of combustion may be considered as coming only from the air supplied for combustion.

From Eqs. (4), (5), and (6) the weight of nitrogen in the stack gases per pound of carbon burned is

$$\frac{28N_2}{12(CO_2 + CO)}. \tag{16}$$

Since one pound of air contains 0.7685 lb of N, the weight of air supplied, A_1, per pound of carbon burned is

$$A_1 = \frac{\frac{28N_2}{0.7685}}{12(CO_2 + CO)} = \frac{28N_2}{12(CO_2 + CO) \times 0.7685}, \tag{17}$$

or

$$A_1 = \frac{3.036N_2}{CO_2 + CO}.$$

The weight of air supplied, A_2, per pound of dry fuel fired will be

$$A_2 = \left[\frac{W_fC_f - W_aC_a}{W_f \times 100}\right]\left[\frac{28N_2}{12(CO_2 + CO) \times 0.7685}\right], \tag{18}$$

or

$$A_2 = \left[\frac{W_fC_f - W_aC_a}{W_f \times 100}\right]\left[\frac{3.036N_2}{CO_2 + CO}\right]. \tag{19}$$

If there is no unburned carbon in the refuse, Eqs. (18) and (19) become

$$A_2 = \frac{C_f}{100}\left[\frac{28N_2}{12(CO_2 + CO) \times 0.7685}\right] \tag{20}$$

and

$$A_2 = \frac{C_f}{100}\left[\frac{3.036N_2}{CO_2 + CO}\right]. \tag{21}$$

In case the stack-gas analysis is known, the percentage of excess air used may be found in the way illustrated by the following problem:

Example.—Find the percentage of excess air used if the stack-gas analysis is 11 per cent CO_2, 9 per cent O_2, 80 per cent N_2.

Solution.—With 80 cu ft of N_2, $\frac{20.9}{79.1} \times 80 = 21.14$ cu ft of O_2 were supplied by the air. Of this 21.14 cu ft of O_2, 9 cu ft were not used to burn the coal but were taken in excess, and 21.14 − 9, or 12.14 cu ft of O_2, were actually used to burn the coal. Then the percentage of excess air, which is the same as the percentage of excess oxygen, is

$$\frac{9 \text{ (excess)}}{12.14 \text{ (used)}} = 74.1 \text{ per cent.}$$

Another method of determining the air used is shown by the following problem:

Example.—A stack gas shows the following analysis: CO_2, 12 per cent; CO, 1 per cent; O, 7 per cent; N, 80 per cent. Find the air used in burning 1 lb of coal, if the coal contains C, 80 per cent; H, 4 per cent; O, 2 per cent

Solution.

	Vol. in 100 cu ft		Dens		Wt
Carbon dioxide (CO_2)................	12	×	0.1164	=	1.396
Carbon monoxide (CO)..............	1	×	0.0741	=	0.074
Oxygen (O)........................	7	×	0.0846	=	0.592

where 0.1164, 0.0741, and 0.0846 are the densities of CO_2, CO, and O_2 at 60 F.

One pound of carbon dioxide contains $\frac{8}{11}$ lb of oxygen, and 1 lb of carbon monoxide contains $\frac{4}{7}$ lb. Therefore, the weight of the oxygen in 100 cu ft of the flue gases would be

In carbon dioxide	$\frac{8}{11} \times 1.396$	$= 1.015$
In carbon monoxide	$\frac{4}{7} \times 0.074$	$= 0.042$
Free oxygen		$= 0.592$
Total weight of oxygen		$= 1.649$ lb

and the weight of the carbon would be

In carbon dioxide	$\frac{3}{11} \times 1.396$	$= 0.381$
In carbon monoxide	$\frac{3}{7} \times 0.074$	$= 0.032$
Total weight of carbon		$= 0.413$ lb.

Air contains 23.15 per cent of oxygen by weight; hence the pounds of air required to burn 0.413 lb of carbon would be

$$1.649 \div 0.2315 = 7.12,$$

and the pounds of air to burn 1 lb of carbon under the conditions of the flue gases would be

$$7.12 \div 0.413 = 17.25.$$

The pounds of air used to burn a pound of coal of the given analysis would be

$$17.25\mathrm{C} + 34.56\left(\mathrm{H} - \frac{\mathrm{O}}{8}\right)$$
$$= 17.25 \times 0.80 + 34.56\left(0.04 - \frac{0.02}{8}\right) = 13.8 + 1.3$$
$$= 15.1 \text{ lb.}$$

It should be noted here that in this solution the weight of air *theoretically* required to burn the hydrogen has been added to the weight *actually* required to burn the carbon as shown by the stack-gas analysis. Although this is, of course, not exactly correct, it is approximately so, and the error is slight, as the amount of air used to burn the hydrogen is small as compared with the total amount required.

The above results are such as might be expected in a boiler plant using induced draft.

70. Material Balance.—It should be pointed out that the following material balance may be written for the combustion process:

$$1 \text{ lb of fuel} + W_a = W_g + \text{water vapor} + \text{refuse}, \qquad (22)$$

where W_a = weight of air actually used per pound of fuel;
W_g = weight of *dry* gas per pound of fuel.

The water vapor comes from three sources: (1) combustion of H in the fuel, (2) moisture in the fuel, (3) moisture in the combustion air. Refuse includes ash and unburned carbon.

71. Excess Air and Dilution Coefficient.—The excess air present during the combustion process = $W_a - W_t$.

W_t = theoretical air necessary for complete combustion from Eq. (2).

$$\text{Percentage of excess air} = \frac{W_a - W_t}{W_t} \times 100. \tag{23}$$

The term *dilution coefficient* is sometimes used. It is

$$\frac{W_a}{W_t}. \tag{24}$$

TABLE XVI.—MEAN SPECIFIC HEATS OF GASES AT CONSTANT PRESSURE, EXPRESSED IN BTU PER CUBIC FOOT, AT 60 F AND 30 IN. HG, CALCULATED FOR THE INTERVAL 60 – T, DEG F

T, deg F	Carbon dioxide	Water vapor	Nitrogen, oxygen, and other diatomic gases
200	0.0237	0.0220	0.0174
400	0.0246	0.0220	0.0175
600	0.0253	0.0221	0.0177
800	0.0260	0.0222	0.0178
1000	0.0268	0.0224	0.0180
1200	0.0275	0.0226	0.0181
1400	0.0282	0.0229	0.0183
1600	0.0287	0.0232	0.0184
1800	0.0292	0.0236	0.0186
2000	0.0298	0.0240	0.0187
2200	0.0302	0.0245	0.0189
2400	0.0306	0.0250	0.0190
2600	0.0309	0.0256	0.0192
2800	0.0312	0.0263	0.0194
3000	0.0314	0.0270	0.0196
3500	0.0317	0.0288	0.0201
4000	0.0319	0.0312	0.0206

72. Theoretical Temperature of Combustion.—If the total heat available to the gases and the specific heats of the products of combustion from a given coal are known, the temperature that might result from their combustion may be approximately calculated. The calculated temperatures of combustion are often higher than those obtained in practice, since the gases are cooled by radiation before their maximum temperature is reached and also since the CO_2 partially dissociates to CO and O_2 at temperatures above 1800 F.

Tables XVI and XVII give the mean specific heats of various gases at constant pressure (c_p) over the interval from room temperature (60 F) to any other temperature (T F).

Example.—Assume the following composition of coal: C, 75 per cent; H, 5 per cent; O, 3 per cent; N, 2 per cent; the ash and S may be disregarded. Find the theoretical and actual rise in temperature of the products of combustion.

TABLE XVII.—MEAN SPECIFIC HEATS OF GASES AT CONSTANT PRESSURE EXPRESSED IN BTU PER POUND, CALCULATED FOR THE INTERVAL 60 − T, DEG F

T, deg F	Carbon dioxide	Water vapor	Nitrogen
200	0.2067	0.4653	0.2365
400	0.2143	0.4657	0.2386
600	0.2216	0.4673	0.2407
800	0.2285	0.4698	0.2428
1000	0.2348	0.4735	0.2449
1200	0.2406	0.4782	0.2470
1400	0.2462	0.4841	0.2491
1600	0.2512	0.4910	0.2512
1800	0.2559	0.4990	0.2534
2000	0.2601	0.5081	0.2555
2200	0.2638	0.5182	0.2576
2400	0.2670	0.5294	0.2597
2600	0.2698	0.5420	0.2618
2800	0.2722	0.5557	0.2639
3000	0.2742	0.5702	0.2660
3500	0.2770	0.6093	0.2707
4000	0.2790	0.6599	0.2755

Solution.—A coal of the above composition has a heat value of 13,820 Btu. The theoretical amount of air required to burn 1 lb of it is 10.24 lb. In 10.24 lb of air there are 10.24 × 0.7685 = 7.87 lb N, to which must be added the 0.02 lb of N in the coal, giving a total of 7.89 lb N.

$$\left.\begin{array}{l}\text{Total } CO_2 \text{ formed} = 0.75 \times 3.66 = 2.745 \text{ lb}\\ \text{Total } H_2O \text{ formed} = 0.05 \times 9 \quad = 0.45 \text{ lb}\end{array}\right\}\text{(see Table XV).}$$

Assuming a final temperature of 2500 F, the thermal units required to raise the products of combustion through 1 degree would be

		Specific Heat	Btu
Carbon dioxide	2.75 ×	0.268 =	0.737
Water vapor	0.45 ×	0.536 =	0.241
Nitrogen	7.89 ×	0.261 =	2.059
Total			3.037.

The theoretical rise in temperature of the products of combustion would be

$$13{,}820 \div 3.037 = 4550 \text{ deg.}$$

In the actual operation of a boiler it is found necessary to add 50 to 100 per cent more air than is required for combustion. This additional air, as the following calculation shows, materially reduces the theoretical temperature of combustion. Assuming 100 per cent more to be required, there would then be added 10.24 additional lb of air. From Table XVII the heat to raise this one degree would be

$10.24 \times 0.261 =$	2.683 Btu
Add for undiluted products	3.037
Total Btu per degree	5.720.

The actual rise in temperature would be, then,

$$13{,}820 \div 5.720 = 2415 \text{ deg.}$$

This is more nearly the temperature obtained in a boiler plant with hand firing.

If the temperature of the boiler room is given, the final temperature of the products of combustion may be found by adding to this temperature the rise in temperature as found above, the assumption being made that the temperature of the coal is the same as that of the boiler room.

In boilers operated by automatic stokers, temperatures of the fire of over 3000 F have been observed. Such temperatures are usually obtained when the boilers are being crowded to their full capacity and their operation is being given careful attention, especially with reference to the amount of air admitted to the furnace.

Example.—A coal containing 70 per cent C and 10 per cent H_2O, and having a heating value of 13,000 Btu per lb, is burned so that the stack gases analyze 15 per cent CO_2, 5 per cent O_2, and 80 per cent N_2. The temperature of air entering the furnace is 60 F. What is the temperature of combustion?

Solution.—Since the stack-gas analysis shows 15 cu ft CO_2 and 5 cu ft O_2 for 80 cu ft N_2, for 79 cu ft N_2 there will be $15 \times \frac{79}{80} = 14.8$ cu ft CO_2, and $5 \times \frac{79}{80} = 4.94$ cu ft O_2. As it takes 14.8 cu ft O_2 to form 14.8 cu ft CO_2, the O_2 accounted for is only $14.8 + 4.94 = 19.74$ cu ft. The difference between this amount and the 21 cu ft of O_2 which was supplied with the 79 cu ft N_2, or $21 - 19.74 = 1.26$ cu ft O_2, united with the H in the coal to form 2.52 cu ft H_2O and did not show in the gas analysis.

The coal contains 0.70 lb C per lb; hence 1 lb of the coal will give off

$$31.5 \times 0.70 = 22.05 \text{ cu ft } CO_2.$$

From the stack-gas analysis, the ratio between the CO_2 and N_2 is as 15:80 and between the CO_2 and O_2 as 15:5. Therefore, for 22.05 cu ft CO_2 there will be

$$22.05 \times \frac{80}{15} = 117.6 \text{ cu ft } N_2$$

and

$$22.05 \times \frac{5}{15} = 7.35 \text{ cu ft } O_2.$$

For 79 cu ft N_2 there are 2.52 cu ft H_2O formed by the combustion of the H in the coal. Hence, for 117.6 cu ft N_2 there will be

$$2.52 \times \frac{117.6}{79} = 3.75 \text{ cu ft } H_2O.$$

The 10 per cent water contained in the coal represents

$$0.10 \times \frac{378}{18} = 2.1 \text{ cu ft } H_2O \text{ vapor.}$$

100 Cu Ft Dry Stack Gas	100 Cu Ft Dry Air		Cubic Feet Stack Gases per Pound of Coal
80 N_2	79	= 79 N_2	117.6 N_2
15 CO_2	14.8		22.05 CO_2
5 O_2	4.94	= 21 O_2	7.35 O_2
	1.26	= 2.52 H_2O	3.75 H_2O
100	100.00		2.1 H_2O

The heat required to evaporate 5.85 cu ft, or $5.85 \times \frac{18}{378} = 0.279$ lb, of H_2O at 60 F is $0.279 \times 1{,}057.8 = 295$ Btu, where 1,057.8 is the heat necessary to evaporate one pound of water from and at 60 F (press. 0.2562 lb). The heat available to raise temperature of gases is

$$13{,}000 - 295 = 12{,}705 \text{ Btu.}$$

The number of Btu required to raise the gases one degree Fahrenheit over the range 60 F to 3000 F is

		Mean Specific Heat, 60 F to 3000 F	Btu
$N_2 + O_2$	$(117.6 + 7.35) \times$	0.0196	= 2.457
CO_2	$22.05 \times$	0.0314	= 0.692
H_2O	$5.85 \times$	0.0270	= 0.158
			3.307

The rise in temperature of the gases is, then,

$$\frac{12{,}705}{3.307} = 3840 \text{ deg,}$$

and the temperature of combustion is

$$3840 + 60 = 3900 \text{ F.}$$

To obtain a more correct result, the rise in temperature should be recalculated, using mean specific heats 60 F to 3800 F instead of 60 F to 3000 F. But this is usually unnecessary for boiler-room computations.

The foregoing example gives an extremely high temperature because of very little excess air, as shown by the stack-gas analysis. With 80 cu ft N_2, $21 \times \frac{80}{79} = 21.3$ cu ft O_2 were supplied by the air. Of this amount, 5 cu ft were not used to burn coal, but were taken in excess, leaving $21.3 - 5 = 16.3$ cu ft as the O_2 actually used. The percentage of excess air is, therefore, $5 \div 16.3 = 30.7$ per cent. The excess is usually 50 per cent or more, which means that more nitrogen and oxygen must be heated, thereby lowering the temperature of combustion.

73. Efficiency of Fuels.—The commercial value of a fuel is determined by the number of pounds of water it will evaporate into steam per hour from and at 212 F. This, however, involves the efficiency of the boiler, so that to compare fuels in actual use, they should be burned in the same boiler. In practice, the value of a fuel in any given plant is affected by the form and character of the furnace, the amount of air supplied, and the intensity of the draft. There are, in fact, so many variables entering into the problem that it is difficult to make an accurate comparison of the value of the different coals.

It is easy to burn either anthracite or semibituminous coal in almost any boiler. For bituminous coals containing less than 40 per cent volatile matter, plain grate bars with a firebrick arch over the fire give very good results. With coals containing over 40 per cent volatile matter, it is desirable to use some form of furnace arranged so that the gases are mixed with warm air, and with these a large combustion chamber should be provided.

The commercial results obtained from a given coal are usually determined by the cost to evaporate 1,000 lb of water into steam from and at 212 F. Where the principal cost of the coal is in the freight rate, it is usually more economical to burn a good grade of coal than a cheap grade.

PROBLEMS

1. A coal contains 3 per cent moisture and 8 per cent ash, based on coal as received. The heating value of the combustible portion is 14,000 Btu per lb. What is the heating value of the dry coal in Btu per pound?

2. A bituminous coal has a total moisture content of 15.71 per cent. The proximate analysis percentages on a dry basis are: volatile matter, 37.30; fixed carbon, 42.90; and ash, 19.80. Express the analysis on an as-received basis.

3. The heat value of the coal in Prob. 2 is 11,312 Btu per lb of dry coal by calorimeter determination. What is the heating value per pound as received? Also per pound of combustible?

4. A coal contains 5 per cent moisture and 10 per cent ash based on coal as received. The heating value of the combustible portion is 15,300 Btu per lb. What is the heating value of the dry coal in Btu per pound?

5. A coal as received contains 3.10 per cent moisture and 6.90 per cent ash. The heating value of the combustible portion is 15,630 Btu per lb. Calculate the heating value of the coal as received and of the dry coal.

6. There is a rule that theoretically it requires 7.5 lb of air per 10,000 Btu to burn coal. How nearly does this rule check for a coal having the following ultimate analysis: 83 per cent carbon; 4.7 per cent hydrogen; 5.6 per cent oxygen; and 6.7 per cent ash?

7. A liquid fuel averaging $C_{12}H_{26}$ (84.7 per cent carbon and 15.3 per cent hydrogen by weight) is to be burned in air. What ratio, pounds of air per pounds of fuel, should give the maximum percentage by volume of CO_2 in the products?

8. A West Virginia bituminous coal has the following composition: C, 82 per cent; H_2, 4.24 per cent; O_2, 2.4 per cent; N_2, 1.2 per cent and the balance ash. If 90 per cent of the carbon is burned to CO_2, and the balance to CO, what would be the heat loss per pound? How much air would be used?

9. A coal has the following composition: C, 82 per cent; H, 4.8 per cent; O, 5 per cent. If 25 per cent of the carbon is burned to CO and the remainder to CO_2, how much heat would be lost? How much air is needed per pound of coal for such a condition to occur?

10. In a boiler plant the coal burned contains CH_4, 20 per cent; C, 60 per cent; O, 6 per cent. The efficiency of combustion is 70 per cent. How many pounds of water at 100 F will be evaporated per pound of coal into steam at a pressure of 150 psi? Neglect heat used in decomposing the CH_4.

11. A flue-gas analysis shows the following composition: CO_2, 11.36 per cent; CO, 0 per cent; O, 7.81 per cent; N, 80.83 per cent. An analysis of the coal shows C, 74 per cent; H, 5.1 per cent; O, 6.12 per cent. What is the heating value of the coal and how many pounds of air are used to burn it?

12. A flue gas shows the following composition, as determined by an Orsat apparatus: CO_2, 8 per cent; CO, 0 per cent; O, 12 per cent; N, 80 per cent. The ultimate analysis of the coal used gives C, 80 per cent; H, 5 per cent; O, 3 per cent; N, 1 per cent. How much excess air was used, in per cent?

13. Some fan builders design forced-draft fans to have a volume capacity of 300 cu ft per lb of coal burned. If the air is at atmospheric pressure and has a temperature of 70 F, what percentage of excess air would this estimate give if the coal had the following ultimate analysis: 76.51 per cent carbon; 4.27 per cent hydrogen; 6.59 per cent oxygen; 1.00 per cent nitrogen; 0.51 per cent sulphur; 11.12 per cent ash? Weight of 1 cu ft of air at 70 F = 0.075 lb.

14. A coal has the following ultimate analysis: C, 76.59 per cent; H, 5.32 per cent; N, 1.24 per cent; O, 10.92 per cent; S, 1.20 per cent; and the balance ash. A forced-draft fan for supplying air for this coal has a capacity of

200 cu ft of air per lb of coal burned. The air is at 70 F and atmospheric pressure. How much excess air can the fan supply?

15. The stack gases leaving a boiler setting analyze 6 per cent CO_2 and 14 per cent O_2 by the Orsat apparatus, the balance being N_2. What is the percentage of excess air supplied?

16. An analysis of the flue gas from a boiler showed 12.2 per cent CO_2; 7.3 per cent O_2, and the balance N_2. The coal contains 81.6 per cent carbon. Find (*a*) pounds of air supplied per pound of coal; (*b*) per cent of excess air; (*c*) the pounds of dry stack gas formed per pound of coal.

17. What would be the increase in the per cent of excess air when burning coal that gave the following flue-gas analysis: 12 per cent CO_2; 8 per cent O_2 and 80 per cent N_2, if the fan speed were changed and the following analysis were obtained: 9 per cent CO_2; 11 per cent O_2; and 80 per cent N_2?

18. The ultimate analysis, as fired, of a certain coal is: carbon 78.4 per cent; hydrogen 5.1 per cent; oxygen 8.8 per cent; nitrogen 1.4 per cent; sulphur 0.7 per cent; ash 5.6 per cent. This coal is burned completely under such conditions that 17 lb of dry gas and 0.459 lb of water vapor are evolved from each pound of coal fired. (*a*) What per cent of excess air is being furnished? (*b*) What per cent of the heat value of the coal is lost in the dry flue gases if the stack temperature is 490 F and the air and fuel enter the furnace at 100 F? The specific heat of dry gases may be taken as 0.24.

19. The stack-gas analysis from a boiler test shows CO_2, 11 per cent; O_2, 8.9 per cent; and CO, 0.1 per cent. The total carbon in the coal is 76 per cent by weight and the hydrogen is 6 per cent. The per cent of moisture in the coal is 4 per cent. The per cent of oxygen is 6 per cent. Preheated air is supplied for combustion purposes at a rate of 316 cu ft per lb of coal, under a pressure of 15 psia and at a temperature of 310 F. Determine, (*a*) the per cent of excess air; (*b*) the weight of air used per pound of coal as fired; (*c*) the heat loss in per cent due to stack gas if stack temperature is 570 F, room temperature is 70 F, and heating value of coal is 13,000 Btu per lb as fired. The specific heat of gas is 0.24.

20. A certain powerplant used 40 tons of coal per day for 300 days per yr. Analysis of the coal shows C, 80 per cent; H, 5 per cent; O, 8 per cent. By improper firing methods, 10 per cent of the carbon was burned to CO with a use of 60 per cent more air than is theoretically needed in the combustion. If, by proper methods of firing, all the carbon were burned to CO_2, what would be the saving per year if the coal cost $4 per ton?

21. A coal contains C, 90 per cent; H, 1 per cent; and O, 2 per cent. If on account of improper firing methods, 15 per cent of the C was burned to CO instead of CO_2 in a plant burning 50 tons of coal per hr, how much would be lost in the cost of fuel, if the plant operated 24 hr a day, 360 days in the year, and the coal cost $5 per ton?

22. Suppose a plant operated as stated in Prob. 8 and used 70 tons of coal each day throughout the year. What would be the saving in dollars per year if the coal were burned completely to CO_2? Coal costs $4 per ton.

23. Calculate the heat loss in the refuse per pound of coal fired if the coal as fired has an ash content of 12 per cent and the combustible in the dry refuse is 40 per cent of the dry refuse.

24. Calculate the heat loss in the refuse per pound of coal fired if the coal as fired has an ash content of 15 per cent and the combustible in the dry refuse is 20 per cent of the dry refuse. Calorific value of the combustible in the ash is 14,600 Btu per lb.

25. Given the following stack-gas analysis from a boiler test: CO_2, 15.68 per cent; O_2, 2.56 per cent; CO, 0 per cent; and N_2, 81.76 per cent. Carbon burned per pound coal, 0.76 lb. The boiler-room temperature is 70 F and the stack temperature is 500 F; sp ht of the stack gas is 0.24. Heat value of coal as fired is 14,000 Btu. Determine the heat loss due to dry stack gas in per cent.

26. When burning a bituminous coal, the dry flue-gas analysis, per cent by volume, showed CO_2, 11.7; O_2, 7.04; CO, 0.71; N_2, 80.55. The refuse from the ashpit contains 32 per cent combustible matter. Coal contains 80.2 per cent carbon and 4.3 per cent ash. Find (*a*) the weight of air supplied per pound of coal; (*b*) the weight of dry stack gases per pound of coal; (*c*) the per cent of excess air; (*d*) the heat lost due to combustible in refuse.

27. A stoker-fired steam generating unit has an economizer in which the flue gases are cooled from 657 F to 357 F. The ultimate analysis of the coal is as follows: C, 79.8 per cent; H, 4.2 per cent; O, 3.4 per cent; S, 1.0 per cent; N, 0.6 per cent; ash, 11.0 per cent. On a test the combustible (assumed to be carbon) lost in the refuse is 0.021 lb per lb of coal fired, and the flue-gas analysis by volume is CO_2, 15.0 per cent; O_2, 3.8 per cent; CO, 0.0 per cent; N_2, 81.2 per cent. The specific heat of the dry flue gas is 0.24 and that of the steam at low pressure is 0.455. How much heat does the flue gas lose to the economizer per pound of coal fired?

28. A boiler plant has the following stack-gas analysis: CO_2, 5 per cent; O, 12 per cent. The coal contains C, 80 per cent; H, 4 per cent; and O, 6 per cent. The plant burns 30 tons of coal per day costing $4.50 per ton and operates 300 days of the year. The temperature of the stack is 500 F and temperature of the boiler room 70 F. How much money can be saved per year if the plant is changed so as to have the following stack-gas analysis: CO_2, 12 per cent; O, 5 per cent?

29. A boiler plant burns coal containing C, 75 per cent; H, 5 per cent; O, 8 per cent. If twice as much air is used as is theoretically required, what will be the analysis of the stack gases as shown by an Orsat apparatus? The moisture formed by the combustion does not appear in the Orsat apparatus.

30. A coal gives the following analysis: C, 75 per cent; H, 6 per cent; O, 4 per cent; and N, 2 per cent. Seventy-five per cent of excess air is used in burning it. What is the theoretical rise in temperature of the gases?

31. Coal containing 80 per cent C and 4 per cent H_2O is burned in a boiler furnace. The stack gases analyze 6 per cent CO_2, 14 per cent O_2, and 80 per cent N_2. How many standard cubic feet of dry air are being supplied to burn 1 lb of coal?

32. A flue gas is 12 per cent CO_2, 7.5 per cent O_2, the balance N_2. Find the percentage of excess air, and the amount of air theoretically needed to burn 1 lb of the coal which contains 68 per cent C.

33. The flue gases leaving a boiler are 14 per cent CO_2, 2 per cent CO, 5.3 per cent O_2, the balance N_2. Find the temperature of combustion if the coal used is 75 per cent C, 6 per cent H_2O, and has a heating value of 13,800 Btu.

34. A coal 72 per cent C and 3 per cent H_2O gives stack gases of 9.5 per cent CO_2, 10.5 per cent O_2, and 80 per cent N_2, which leave the boiler at 600 F. Find the heat carried up the stack as latent heat by the water and as sensible heat by all the gases.

35. A coal 68 per cent C and 4.5 per cent H_2O gives flue gases of 12 per cent CO_2, 8.5 per cent O_2, the balance N_2, which leave the boiler at 760 F. Find the heat carried up the stack as latent heat by the water and as sensible heat by all the gases.

36. A coal 80 per cent C and 3 per cent H_2O gives flue gases 10 per cent CO_2, 2 per cent CO, 8.5 per cent O_2, the balance N_2, which leave the boiler at 800 F. Find the heat carried up the stack as latent heat by water, in unburned gases, and as sensible heat by all the gases.

CHAPTER VI

BOILERS OR STEAM GENERATORS

74. Definition and Classification.—The A.S.M.E. defines a *steam-generating unit* as "a combination of apparatus for producing, furnishing, or recovering heat, together with apparatus for transferring the heat so made available to the fluid being heated and vaporized." Such a unit may include "boiler, waterwalls, water floor, water screen, superheater, reheater, economizer, air heater, furnace, and fuel-burning equipment. Economizer and air heater are not included when the heat absorbed is not returned to the unit."

Boilers may be divided, from the path taken by the fire, into *fire-tube* or *tubular* boilers and *water-tube* or *tubulous* boilers. In the fire-tube boiler the hot gases from the fire pass *through* the tubes, and in the water-tube boiler they pass *outside* the tubes. Either type may be constructed as a *vertical* or as a *horizontal* boiler, depending upon whether the shell is set vertically or horizontally.

Fire-tube boilers are divided also into two classes, depending on the position of the fire; these are known as *externally fired* and *internally fired* boilers.

In the externally fired boiler, the fire is entirely external to the boiler and is usually confined in a brick chamber. These boilers are used largely for stationary plants.

The internally fired boiler is most commonly used for locomotive and marine boilers. The fire is entirely closed in the steel shell of the boiler and no brick setting is necessary. These boilers are more expensive per horsepower than the ordinary forms of stationary boilers. Vertical fire-tube boilers are usually internally fired.

The various forms of boilers under proper operating conditions give essentially the same economical results.

75. Horizontal Return Tubular Boilers.—Figure 25 shows the plan and elevation of the setting of an externally fired tubular boiler of the return-flue type. This is what is known as a *horizontal return tubular boiler.* The coal burns upon the grates, which rest upon the front of the boiler setting and upon the bridge wall. The flames pass under and along the boiler shell, then turn in the back combustion chamber Y and pass through the tubes of the boiler, then out through the smoke nozzle A' and through the breeching to the chimney. The smoke nozzle is shown at the front of the boiler setting.

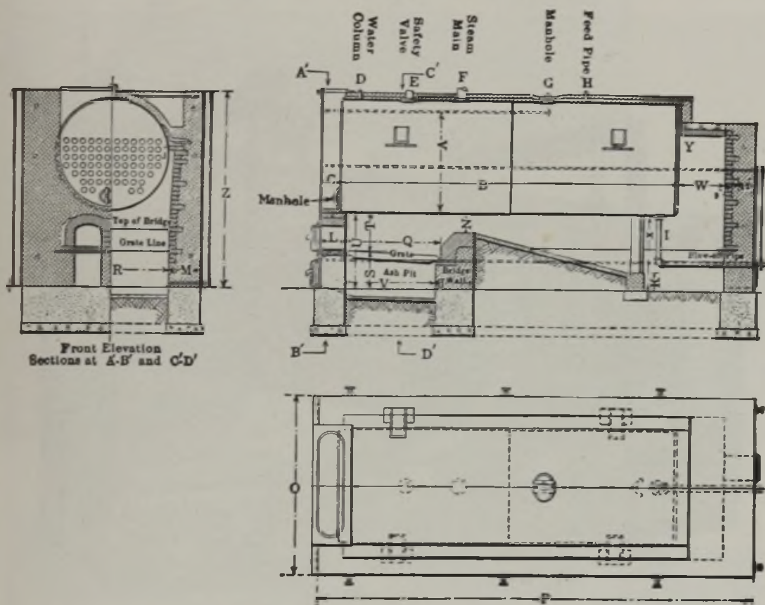


Fig. 25.—Return tubular boiler.

There are usually two manholes in the boiler, one in front under the tubes and one in the top of the boiler. These openings are reinforced with flanged-steel reinforcements. The shells are made of boiler steel having a tensile strength of 55,000 lb to 66,000 lb. The shell of the boiler is rolled to form and riveted together. The heads of the boiler which form the *tube sheets* and into which the tubes are fastened are made of flanged steel of about 55,000-lb tensile strength. The tubes are made of steel, usually lap welded. Charcoal iron tubes are the best, but are difficult to get, so that most manufacturers use a hot-rolled, lap-welded steel tube.

In accordance with the A.S.M.E. Boiler Code, horizontal return tubular boilers are built in standard sizes varying from 36 in. in diameter and 10 ft long to 84 in. in diameter and 20 ft long. They are set in brick settings, and in all brick-set boilers great care should be taken in building the setting. Air leaks in the brickwork should be carefully avoided, as they cause serious loss in economy. All brick should be set with full-flush mortar joints so as to make the setting strong and avoid leakage.

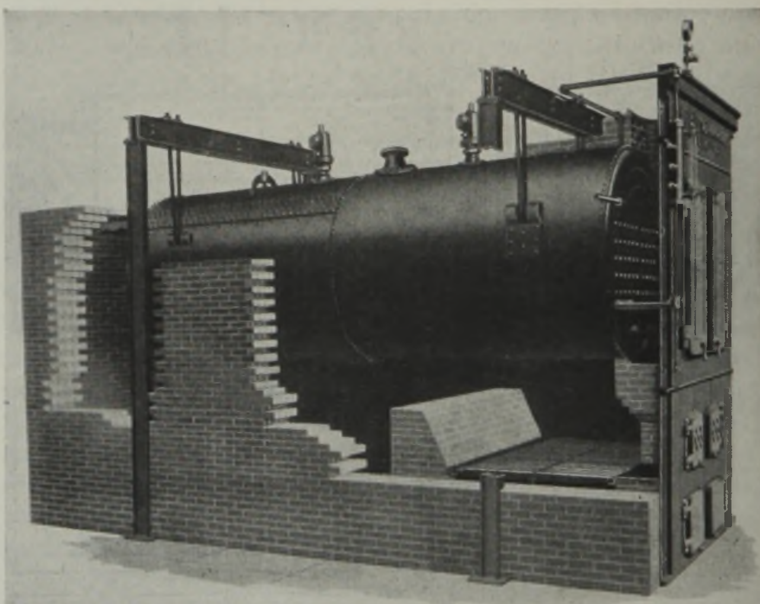


Fig. 26.—Brick setting for fire-tube boiler with full-front setting.

Figure 26 shows a boiler with a solid brick setting. Some engineers prefer a setting having a 2-in. air space in the center of the wall. The brick walls enclosing a fire-tube boiler are made very heavy so as to give good heat insulation, preventing an excessive loss of heat from the boiler, and also to prevent the filtration of air through the setting and the consequent cooling of the hot gases passing away from the fire.

Figures 25 and 26 show the boiler resting upon the brickwork. Boilers of this type over 72 in. in diameter are required by the A.S.M.E. Code to be suspended from a steel framework, as shown in Fig. 27. This method is preferable, as the brick setting of a boiler has very little strength and this arrangement leaves the boiler independent of the setting and the setting free from all strain due to the weight of the boiler.

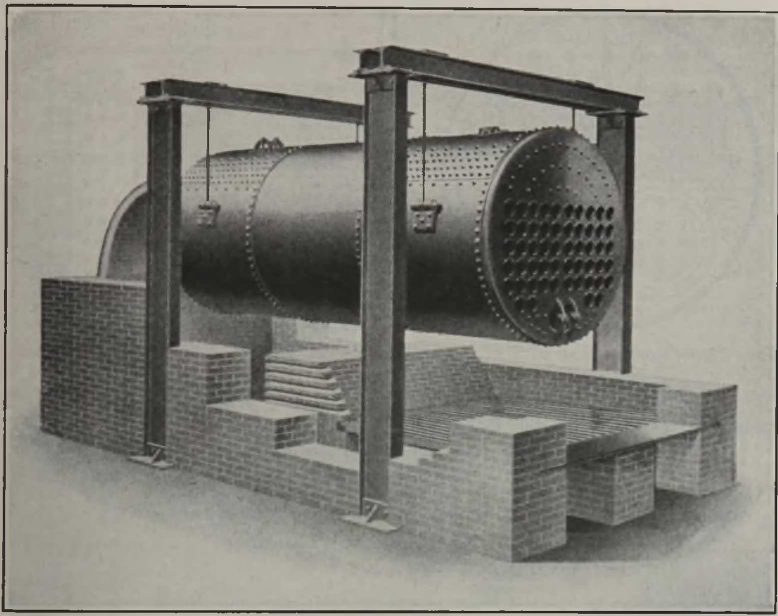


Fig. 27.—Steel-frame boiler suspension.

In earlier boiler construction, it was customary to place a steam dome on all boilers, the object being to provide dry steam. Most engineers have discarded the use of steam domes on high-pressure boilers, as they weaken the boiler shell and add to the

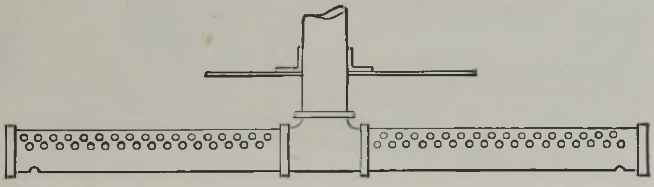


Fig. 28.—Dry pipe.

expense of the boiler construction. To avoid getting wet steam from the boiler a *dry pipe* is provided as shown in Fig. 28.

76. Internally Fired Boilers.—Another large class of return tubular boilers are the internally fired boilers. These boilers can be built in large sizes, are very compact, and thus are particularly suitable for marine work. Figure 29 shows an internally fired Scotch marine boiler with two furnaces. In large boilers of this type there are often three or even four furnaces; in the

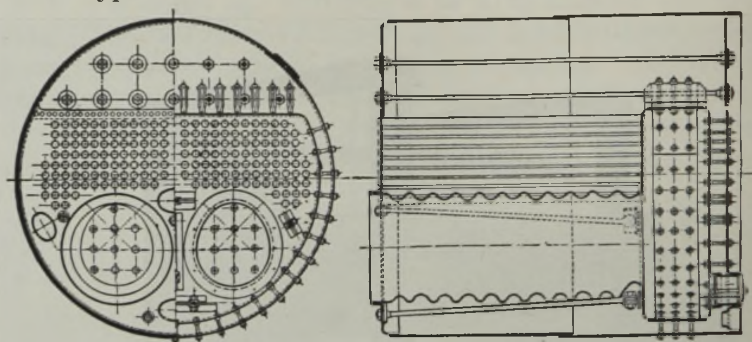


FIG. 29.—Cross section of Scotch marine boiler with steel combustion chamber.

largest ones, furnaces are provided at each end opening into a common combustion chamber in the middle of the boiler.

In recent years the Scotch marine boiler has been largely replaced by the marine type of water-tube boiler, which is lighter in weight and has greater capacity for the space occupied.

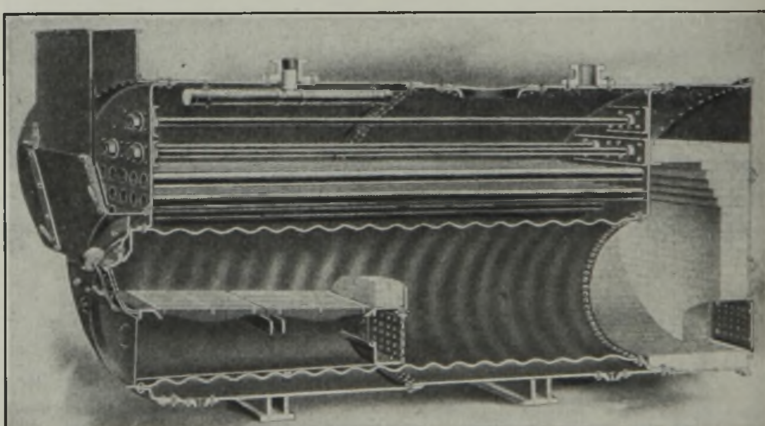


FIG. 30.—Scotch marine boiler with brick combustion chamber.

Figure 30 shows a Scotch marine boiler built for stationary use. The steel-back combustion chamber used in marine work, shown in Fig. 29, is replaced by brick construction in Fig. 30.

77. Locomotive Type of Boiler.—A special type of internally fired tubular boiler is used on locomotives. In this boiler the walls of the furnace, or firebox, and the ashpit are made double with a space between. This space is connected with the water in the shell and forms what are called *water legs*. The gases pass directly from the fire through the tubes and up the stack. As in other forms of internally fired boilers, the hot gases do not come in contact with the shell. This permits of the use of higher pressures in these boilers—sometimes as high as 450 lb. The horsepower rating is ordinarily from 300 to 1,000, although larger sizes have been built recently.

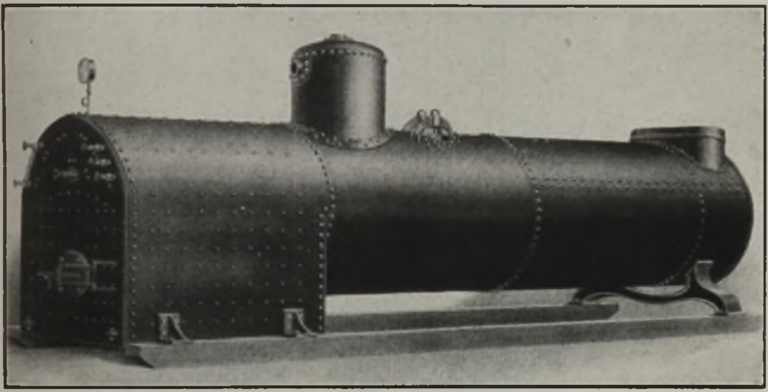


FIG. 31.—Locomotive-type boiler for stationary use.

Modifications of this type in sizes generally under 150 hp are used for threshing, oil drilling, and other work where portable boilers are necessary or desirable. They are also sometimes used for stationary purposes, particularly for heating where a compact form of boiler is desirable. Figure 31 shows the side elevation of a boiler of this class designed for stationary use and having an open-bottom firebox. Another design of this boiler has a closed-bottom firebox. Figure 32 shows sectional views of a locomotive type of boiler for use in the oil country. The advantages of this boiler are portability and large capacity for their size. The disadvantages are large flat surfaces requiring much staying, poor circulation, and high explosive risk.

Figure 33 shows a phantom view of a locomotive built by the Lima Locomotive Works.

78. Vertical Fire-tube Boilers.—This is another form of internally fired boiler. It consists of a vertical cylindrical shell with the furnace and ashpit in the lower end. The firebox is

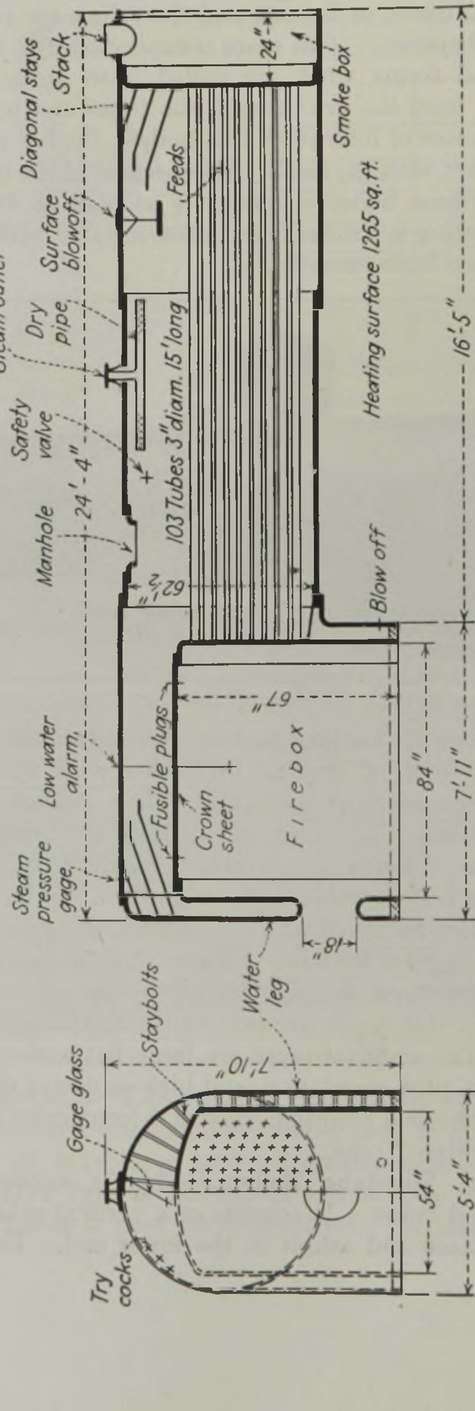


Fig. 32.—Locomotive-type (oil-country) boiler for stationary service.

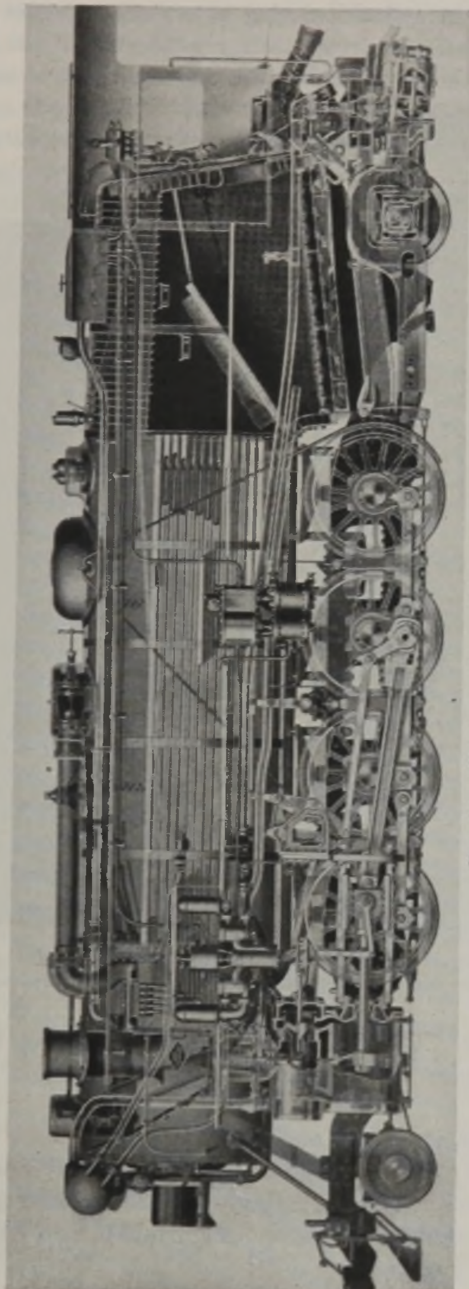


FIG. 33.—Phantom view of Lima locomotive.

surrounded by a cylindrical ring of water, or water leg. The tubes extend from the *crown sheet* over the top of the furnace to a *tube sheet* in the upper end of the boiler and act to some extent as stays, or braces, for the crown and tube sheets. The tube sheet, or head, may or may not be submerged, depending upon the

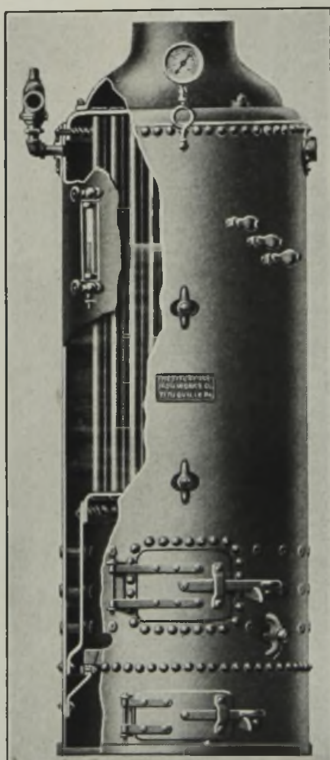


FIG. 34*a*.—Vertical fire-tube boiler with exposed tubes and extended shell ashpit.

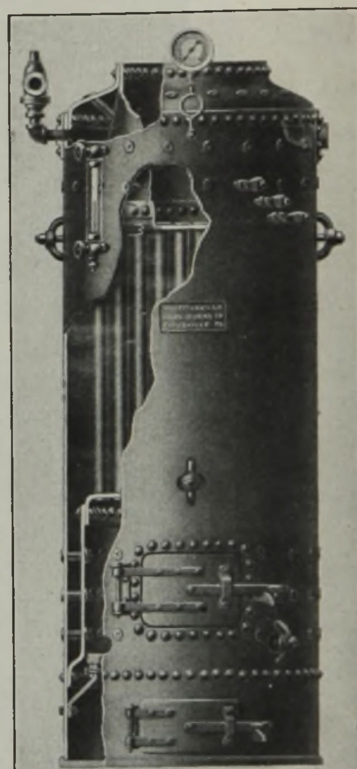


FIG. 34*b*.—Vertical fire-tube boiler with submerged tubes and extended shell ashpit.

type of boiler. The hot gases pass directly from the furnace through the tubes to the upper part of the shell and then out to the stack.

Although inexpensive and requiring little floor space, the ordinary type of small upright boiler is generally uneconomical, has small capacity, has insufficient space for the disengagement of steam, and is difficult to clean. Figures 34*a* and 34*b* show two boilers of this type.

As these vertical tubular boilers are internally fired, the shell is not subjected to the deteriorating effect which direct contact with the flame would have. Consequently, the metal in the shell may be made as thick as desired, thus making possible the higher working pressures demanded by recent practice.

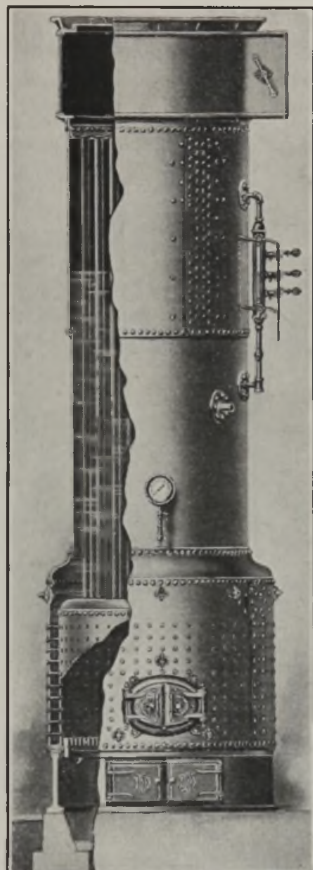


FIG. 35.—Manning upright boiler.

Figure 35 shows a Manning upright boiler, in which the water line is carried 3 ft or 4 ft below the upper head. The upper ends of the tubes thus become superheating surface. These larger boilers have the advantage of minimum floor space per boiler horsepower and at the same time, owing to the enlarged grate area and increased steam space, do not have the disadvantages of the small vertical boiler. Their ability to meet the modern requirements of high pressure and superheated steam is undoubtedly a reason for their increased use.

79. Use of Tubular Boilers.—The fire-tube boiler, as shown in Fig. 25, has certain limitations in use. Its construction is such that hot gases pass outside the shell, with cold water on the inside. This produces a large difference of temperature on the two sides of the shell, and a strain is produced in the metal, owing to this difference of temperature. The thicker the metal of the shell, the greater the difference in temperature between the two sides, and the greater the difficulty in rolling this metal to a uniform quality. In practice, it is found that for best results the thickness of the metal should not exceed $\frac{1}{2}$ in. to $\frac{3}{4}$ in. This limitation in the thickness of the shell limits the diameter of the boiler and the pressure that the boiler can carry. In general, this class of boilers is used where the gage pressure does not exceed 125 psi to 150 psi and the size is not larger than 200 boiler hp, although many of the more recent boilers are built in sizes up to 300 hp or 400 hp and some have been built with a rating as high as 500 hp.

Since a majority of the modern plants are being operated at over 150-lb pressure, a fire-tube boiler cannot be used. In addition, the horsepower of each boiler unit is so small that a

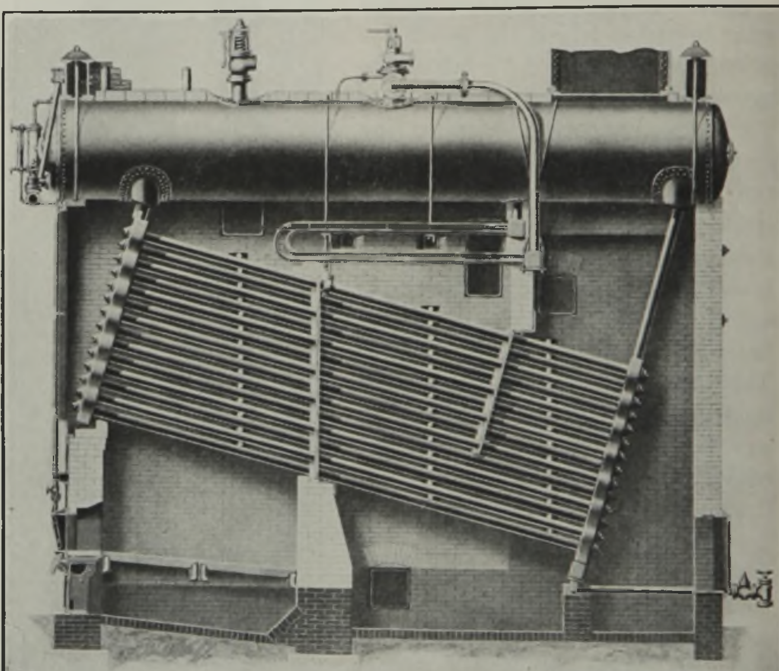


FIG. 36.—Babcock and Wilcox wrought-steel inclined-header longitudinal-drum boiler with Babcock and Wilcox superheater, and arranged for hand firing.

very large number of boiler units would be necessary. In a power plant of, say, 50,000 hp, if this type of boiler were used, there would be required 200 to 300 boilers and the space needed for this number of units would make it almost impossible to install such a plant.

The internally fired boiler is not so limited in the pressure that it can carry as is the return fire-tube type, since the fire does not come in contact with the boiler shell and the shell can be made thicker. The increased thickness of shell permits the building of larger boilers of this type than of the return fire-tube. They have not been much used for stationary purposes owing to their first cost and the cost of repairs where conditions are not favorable to their use.

80. Water-tube Boilers.—The demand for increased pressure and for larger sized boiler units has led to the introduction of

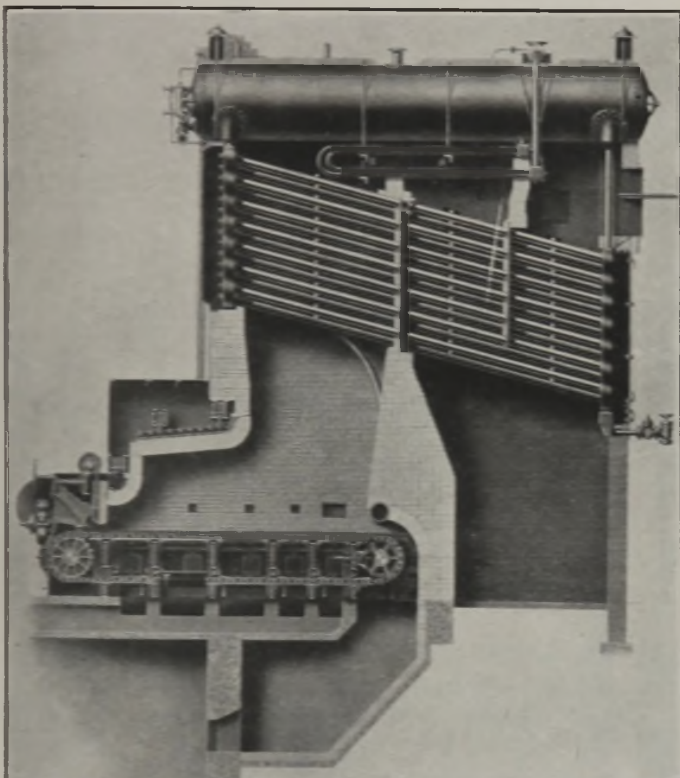


FIG 37.—Babcock and Wilcox longitudinal-drum boiler with Babcock and Wilcox superheater and Babcock and Wilcox blast chain-grate stoker.

water-tube boilers, and all the larger power stations today are using such boilers almost exclusively. The principal reasons for using the water-tube boilers in large power stations are adaptability to high pressure, reduced space taken by the boiler, and greater safety in operation. There are many different makes of water-tube boilers on the market of various types, both vertical and horizontal.

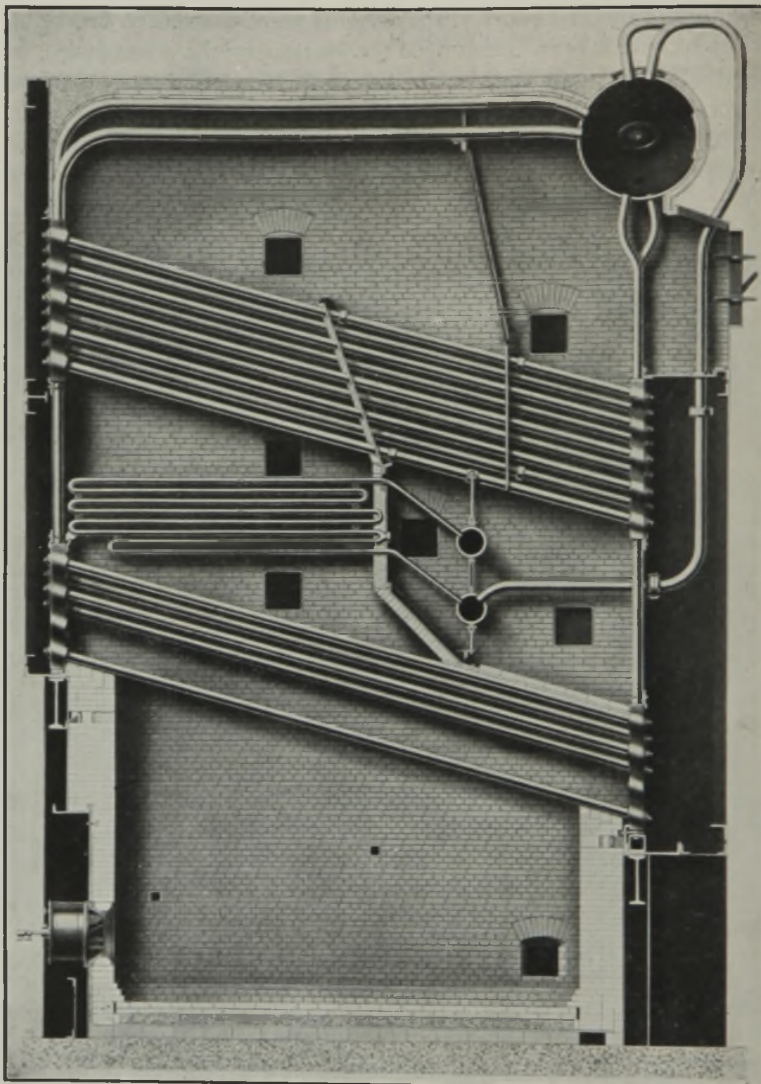


FIG. 38.—Babcock and Wilcox water-tube boiler of the forged-steel, sectional-header, double-deck, cross-drum type, with interdeck superheater and oil-burning furnace.

81. Horizontal Water-tube Boilers.—Figures 36, 37, and 38 show side views of three types of Babcock and Wilcox boilers. Gases from the fire pass up around the tubes, being deflected vertically by a baffle wall located between the tubes and directly above the bridge wall. They then pass down around the tubes to the space back of the bridge wall, being deflected by another baffle, then up between the tubes and out through the smoke opening which is in the rear of the boiler setting and above the tubes.

As it is heated, the water in the tubes tends to rise toward their upper, or front, end, then rises through the front header and connection into the steam and water drum, where the steam separates from the water, and the latter flows back in the drum and down through the rear header. The feed water enters the boiler through a pipe passing through the front end of the drum and extending back about one-third its length.

Fig. 39.—Wrought-steel inclined header in Babcock and Wilcox boiler.

This class of boiler gives very satisfactory service for high-pressure work, having large disengaging surfaces for the steam to leave the water, and ample steam space.

Figure 39 shows a wrought-steel inclined header in a Babcock and Wilcox boiler and indicates clearly the way the tubes are "staggered."

As already shown, Babcock and Wilcox boilers are built in two general types: the longitudinal-drum type (Figs. 36 and 37), and the cross-drum type (Fig. 38). Either type may be constructed with vertical headers, or with inclined headers perpendicular to the tubes. The headers are of either cast iron or wrought steel depending upon the pressure which the boiler is to carry. For pressures in excess of 160 psi, wrought-steel headers are used.

Figure 40 shows a cross section of a Heine water-tube boiler with inclined longitudinal drum. In this boiler the gases of combustion pass over the bridge wall into the combustion chamber, where they are completely burned. They then pass upward back of the lower baffle wall (which consists of a row of tiling) and then forward around the tubes, and parallel to them, to the front of the boiler, where they turn up in front of the forward end of the upper baffle wall and then pass back along and parallel to the shell, and finally up around the shell to the opening to the breeching.

The feed water enters the boiler through the pipe shown, passing into the front end of the internal mud drum. It then

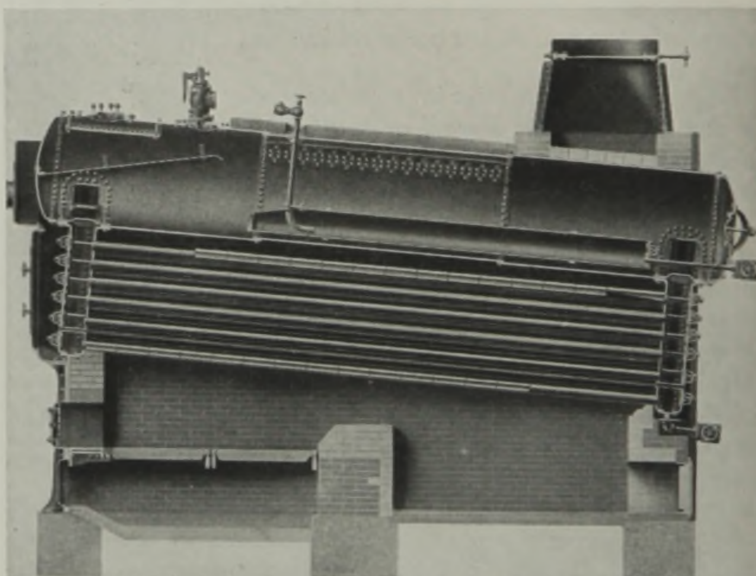


FIG. 40.—Inclined, longitudinal-drum, longitudinally baffled Heine boiler with gas outlet at top.

flows slowly back along the bottom of this drum depositing the dirt and sediment. As the water becomes heated, it rises and flows forward along the top of the mud drum and out of it at the front end. From here the circulation is toward the back of the boiler, down the rear water leg, forward through the tubes, and up the front water leg into the boiler again. The steam, which is formed very largely in the tubes, is carried along with the water and discharged into the boiler from the front water leg.

Figure 41 shows a side elevation of the box headers, drums, and tubes in a Heine boiler with cross baffles. It should be noted that in this boiler the header at the downtake end is set some distance below the drums and connected with them by bent tubes. This makes it possible to set the drums horizontally and at the same time have the tubes incline upward from the downtake to the uptake end. In the boiler shown in Fig. 40, both headers are directly attached to the drums, the tubes are parallel to the drums, and both drums and tubes are set in an inclined position.

Heine boilers, which are built by the Combustion Engineering Co., are made also in the cross-drum type (Fig. 42). In both this

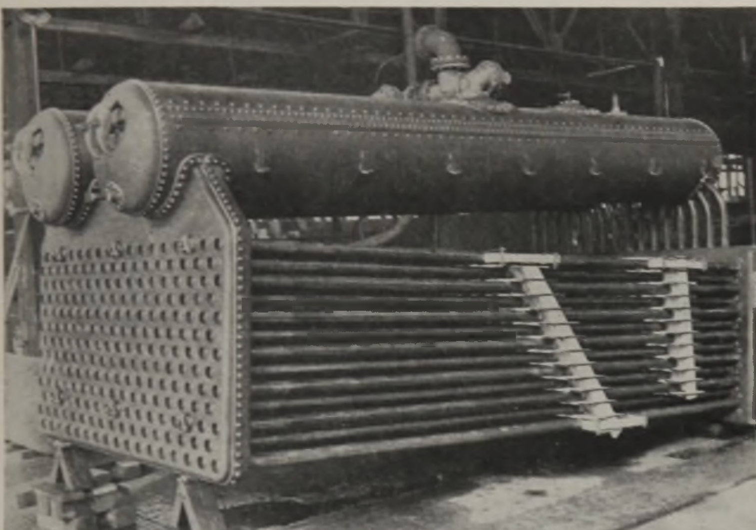


FIG. 41.—Shop view of Heine horizontal, longitudinal-drum, cross-baffled boiler taken from uptake end.

and the longitudinal-drum type, the baffles may be horizontal as shown in Fig. 40 or the boiler may be cross-baffled as in Fig. 41.

In some cases a combination of horizontal and cross baffles is used. Such an arrangement, shown in Fig. 42, exposes the entire lower rows of tubes to radiant heat besides being in contact with some proportion of the hot gases. These gases rise in the first pass at the back of the boiler "turning forward and downward into the second pass, under the front baffle, thence upward into the space beneath the horizontal circulating tubes into the uptake located just back of the drum."

Figure 43 shows a sectional side elevation of a Stirling boiler. This consists of three transverse steam and water drums set parallel and at the same level and connected to one mud drum by water tubes so curved that their ends enter the tube sheets at right angles to the surface. This curvature of the tubes gives ample and efficient provision for expansion and contraction. The three upper drums are interconnected by curved steam-equalizing tubes above the water line, while the front and middle drums only are connected by curved water-circulating tubes below the water line.

Feed water enters the upper rear drum and is delivered into a trough by which it is distributed along the whole length of the

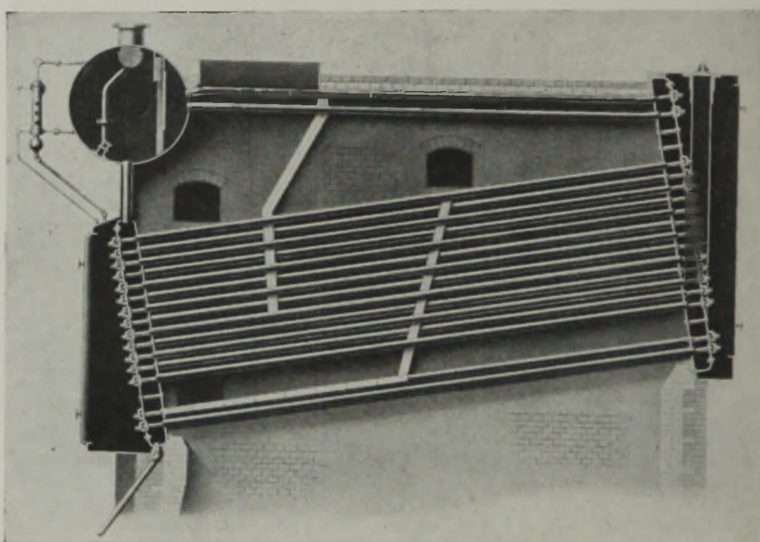


FIG. 42.—Sectional side elevation of C-E Heine cross-drum boiler.

drum and then passes downward through the rear bank of tubes to the lower drum, thence upward through the front bank to the forward steam and water drum. The steam formed during the passage upward through the front bank of tubes becomes separated from the water in the front drum, and passes through the upper row of cross tubes into the rear drum. The water from the front drum passes through the lower cross tubes into the middle drum, and thence downward through the middle bank of tubes to the lower drum, from which it is again drawn up the front bank. Steam generated in the boiler is collected at the top of the middle drum and passes from there into the superheater header.

In its passage down the rear bank of tubes the feed water is heated so that much of the scale-forming matter is precipitated and gathers in the rear bank of tubes and in the mud drum, where it is protected from high temperatures. The blowoff

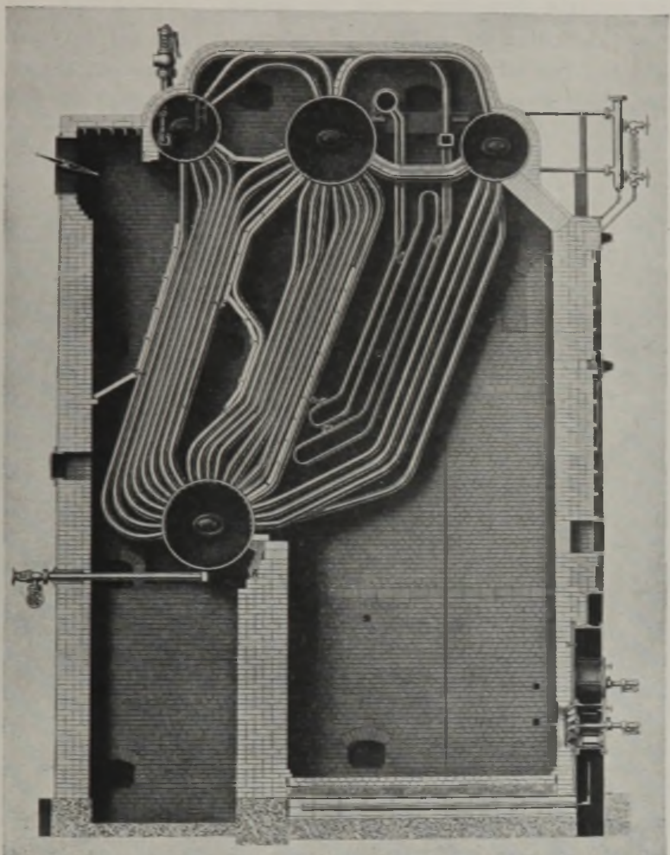


FIG. 43.—Stirling boiler with Babcock and Wilcox superheater and mechanical-atomizing oil burners.

connection is attached to the bottom of the mud drum at the center and passes out through a sleeve in the rear wall, just outside of which the blowoff valve is located. The water column, located at one side of the front of the boiler, is connected to one head of the center steam and water drum. The safety valves are located on the top of the rear drum.

This boiler represents the ideal circulation as far as the paths of the water and gases are concerned, *i.e.*, the coldest gases come in contact with the coldest water in the boiler, and the hottest gases come in contact with the hottest water. The hot gases circulate in the reverse direction from the water. On leaving the fire, they are deflected by baffle walls so as to pass up between and around the front bank of tubes and the superheater, then down around the tubes from the middle drum, and again up between

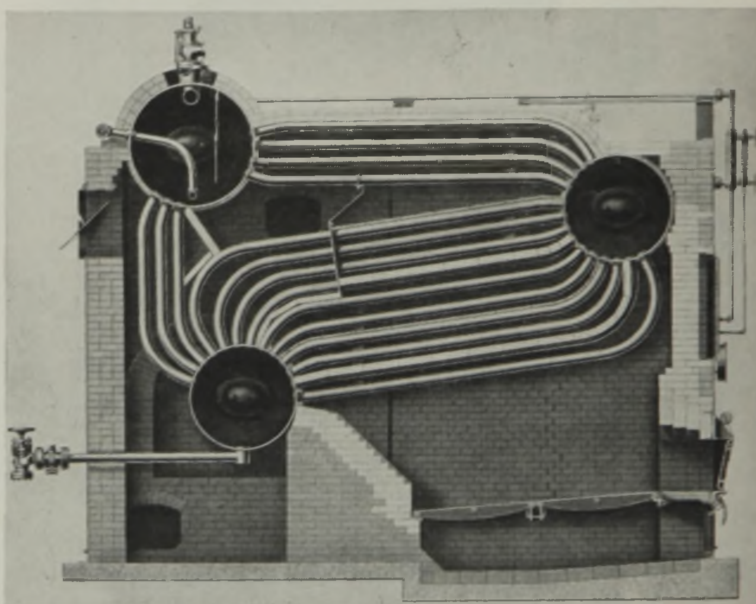


FIG. 44.—Babcock and Wilcox Type H Stirling boiler arranged for hand firing. T-tiles extend throughout the length of some of the tubes in the first bank.

the rear bank of tubes toward the rear drum. The burned gases leave the boiler at the rear near the upper end of the last bank of tubes. The drums with their connecting tubes are supported by a steel frame built into the brickwork of the boiler. The brick setting serves only to enclose the gases and is under no strain due to the weight of the boiler. There is a manhole in one end of each of the four drums; by removal of the manhole plates the drums may be entered.

The Babcock and Wilcox Type H Stirling boiler (Fig. 44) is a relatively small, water-tube unit designed primarily for installation where head-room is limited, and is particularly suitable for industrial power plants, for process steam requirements, for the smaller central stations, and for heating purposes. It is also admirably adapted to the modernization of smaller power plants. It can be built for any fuel or method of firing, and may be set singly or in battery.

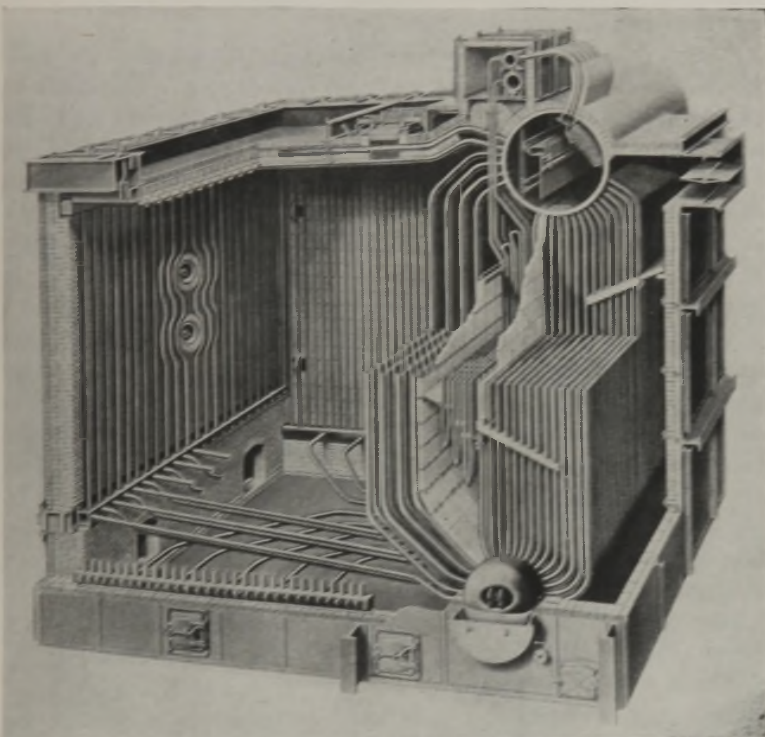


FIG. 45.—C-E Steam Generator Type VU.

The gas outlet may be at the front, rear (Fig. 44), or top of the boiler, whichever is most convenient in the particular installation.

The method of baffling with any arrangement of gas outlet assures the most advantageous flow of gases over the heating surfaces, resulting in high rates of heat transfer with correspondingly high efficiencies.

A recent development in the industrial boiler is the integral furnace type. These boilers are compact, fired with gas, oil, or pulverized coal, and are waterwalled for protection of refractory. They may be used with or without air preheaters, or economizers, and have high efficiency and high availability.

The C-E Steam Generator Type VU (Fig. 45) is one form of integral furnace boiler. It is designed for economy of space as well as performance, and is built in a number of sizes ranging from 20,000 lb to 250,000 lb of steam per hour for varying requirements of pressure and temperature.

The furnace is water-cooled on top, bottom, and sides with tubes 3 in. in diameter, in circuit with the boiler. The tubes in the front bank, subjected to a high rate of heat absorption by both radiation and convection, are 3 in. in diameter. The first tube of the second bank, which serves as a baffle support, is 3 in. in diameter. The rest of the tubes in the second bank, entirely shielded from furnace radiation and subjected to comparatively low gas temperature, are 2 in. in diameter. The superheater is installed in the space between the two banks of tubes.

The feed water is distributed throughout the length of the upper drum by means of a perforated feed-water pipe. The major portion of the steam generated, *i.e.*, the discharge from the furnace wall and roof tubes and from the front bank of boiler tubes, enters the steam drum above the water level which is approximately the center line of the drum. This feature, in conjunction with the effective removal of any entrained water by the baffles and screens in the drum, assures dry steam at all ratings.

One of the most recent types of integral furnace is the S-A boiler (Fig. 46) built by the Foster-Wheeler Corporation.

The S-A boiler is made up of three drums and two banks of water tubes. The large upper drum is connected to each of the smaller lower drums by widely separated banks of tubes making a large furnace between the banks.

Since the tubes of the two banks form the side waterwalls of the furnace, no separate water-wall construction, steel work, headers, or circulators are required for oil or gas firing—a baffle behind the first or second row of boiler tubes is all that is necessary. This permits firing from either end, firing from both ends, or reverse firing. In Fig. 46 the reverse method of firing is used, *i.e.*, the gases of combustion flow from the burner end of the furnace toward the opposite end, turn, and return close to the front wall before leaving the furnace through tube banks at each side.

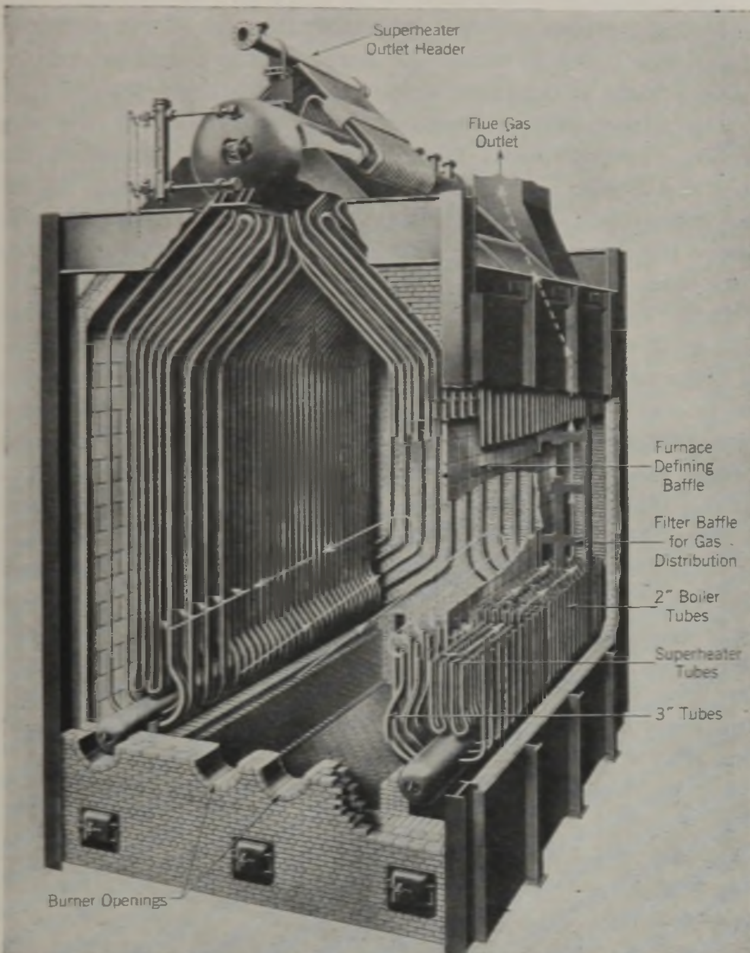


FIG. 46.—Foster-Wheeler "S-A" boiler arranged for reverse firing.

When gases leave the furnace at each side they make a single straight pass lengthwise of the unit through the two parallel banks of boiler tubes.

Every S-A boiler normally has a water-cooled rear wall formed by bringing the end row of tubes straight down from the upper drum to the bottom of the furnace and then sideways to the water drums.

Two sizes of tubes are used—3 in. and 2 in. All furnace tubes, including the front and rear waterwall tubes, are 3 in. in diameter. Tubes not exposed to radiant heat—all the tubes behind the furnace baffle in the tube banks—are 2 in. in diameter.

The superheater is installed immediately behind the slag screen, the continuous loop elements being expanded at the inlet end directly into the top of the steam drum and at the outlet end into an outlet header. The single superheater outlet header above the drum is common to the superheater sections in each tube bank.

In the S-A boiler there are two refractory baffles, one on either side of the furnace immediately behind the first or second row of boiler tubes and extending from the lower drums to the upper drum. Longitudinally, the baffles extend from one end of the furnace about four-fifths of the furnace length, thereby leaving enough space at the other end to permit gases to enter the tube bank. This entrance space is at the front of the boiler when reverse firing is used and at the rear of the boiler when straight firing is used. The only other baffles in the S-A boiler are a pair of steel plates, one in each tube bank, perforated with large rectangular openings. They are placed at right angles to the center line of the drums and extend the entire distance between the upper and lower drums. These *leaky*, or *filter*, baffles as they are frequently called, assure equal distribution of the flue gases throughout the height of the boiler-tube bank.

Figure 47 shows a Ladd boiler with three banks of tubes and three drums. The feed water enters the feedbox in the lower drum and passes up through the rear bank of tubes, down through the middle bank, and up in the front bank. These boilers have been designed to meet the requirements of heavy duty and sustained overloads, and have been installed in some of the largest industrial plants and central stations.

82. Twin-furnace Boiler.—In the design of large steam generators, the furnace presents one of the major problems, especially if conservative heat liberation is to be maintained. Most designers are limited to definite rates of heat liberation per unit of volume, regardless of furnace size. As the linear dimensions of the furnace grow, the volume increases faster than

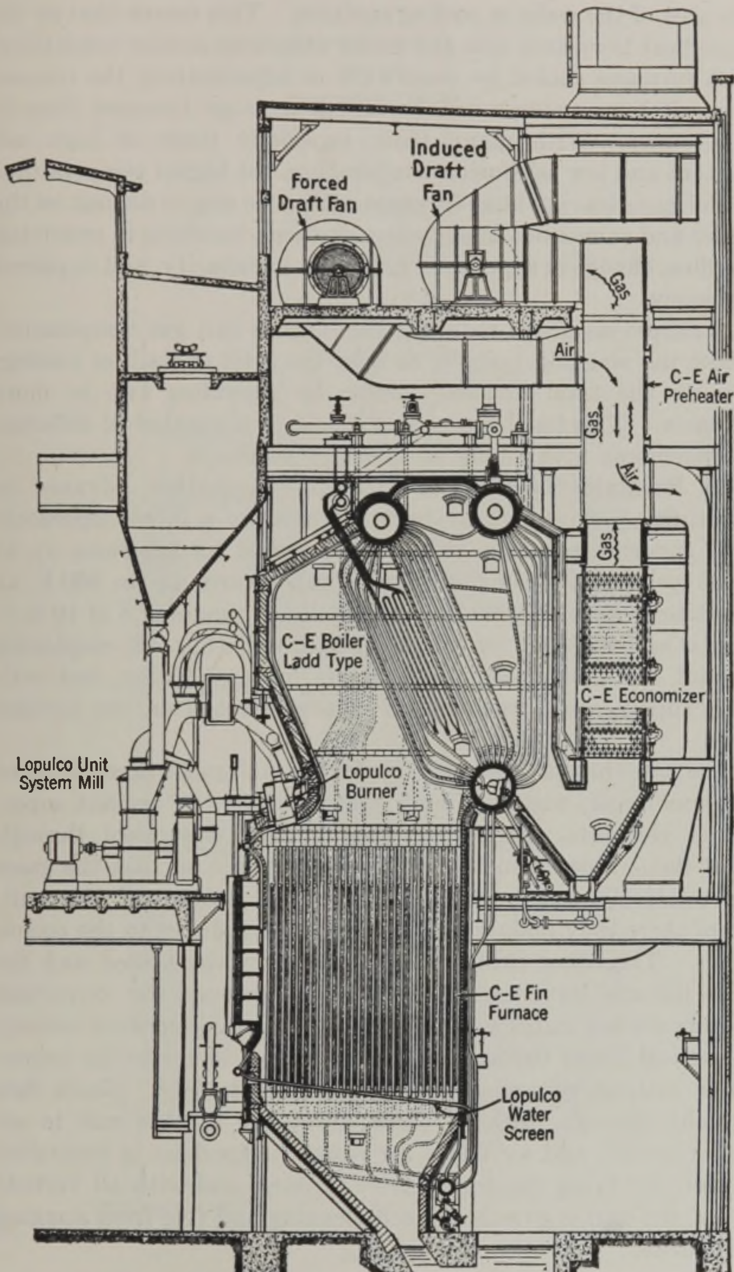


FIG. 47.—1,695-hp Ladd boiler designed for 1,400-lb pressure.

the area of the walls or cooling surfaces. This means that at the same heat liberation rate and under otherwise similar conditions, with furnaces cooled by waterwalls or superheaters, the furnace gas exit temperatures will be higher in large furnaces than in small ones. With many fuels, especially those of high ash content and low ash-fusion temperature, the higher temperatures of the gases leaving large furnaces will cause slag to deposit on the boiler and convection superheater surfaces, resulting in restricted gas flow, change in superheat, reduction in capacity, and impaired efficiency.

A simple means of reducing the furnace exit gas temperature below the slagging point is to raise the ratio of wall, or cooling, area to the total furnace volume by providing two or more furnaces. This has been accomplished in a number of different arrangements with highly satisfactory results.

83. Separate Superheater Furnace.—A further advance in multifurnace development is represented by a large, separately fired superheater steam generator designed for pressures up to 1,550 psi and for constant final temperatures up to 950 F, at capacities, up to 500,000 lb per hr and load ranges of 8 or 10 to 1. This is accomplished by a twin-furnace unit (Fig. 48) employing radiant and convection superheaters in combination, but with both superheaters receiving all of their heat from one furnace only.

The superheater furnace is cooled by waterwall tubes in three walls and roof, with the inner wall consisting of radiant superheater elements. Pulverized coal is fired downward through short-flame burners in the roof of the furnace so that the gases are relatively cool when they enter the bottom of the unit. From there they make a single pass upward direct to the economizer. The other furnace is completely water-cooled and the gases likewise leave near the bottom. However, the convection heating surface in the gas pass from this furnace consists entirely of vertical boiler tubes, and the gases flow direct to the economizer without sweeping any superheating surface. Gases flow straight through, without turns, from the furnace exit to air heater outlet, and air ducts are short. Superheat is controlled entirely by firing the superheater furnace, and with all vertical tubes, the unit is practically self-cleaning and free from slagging difficulties.

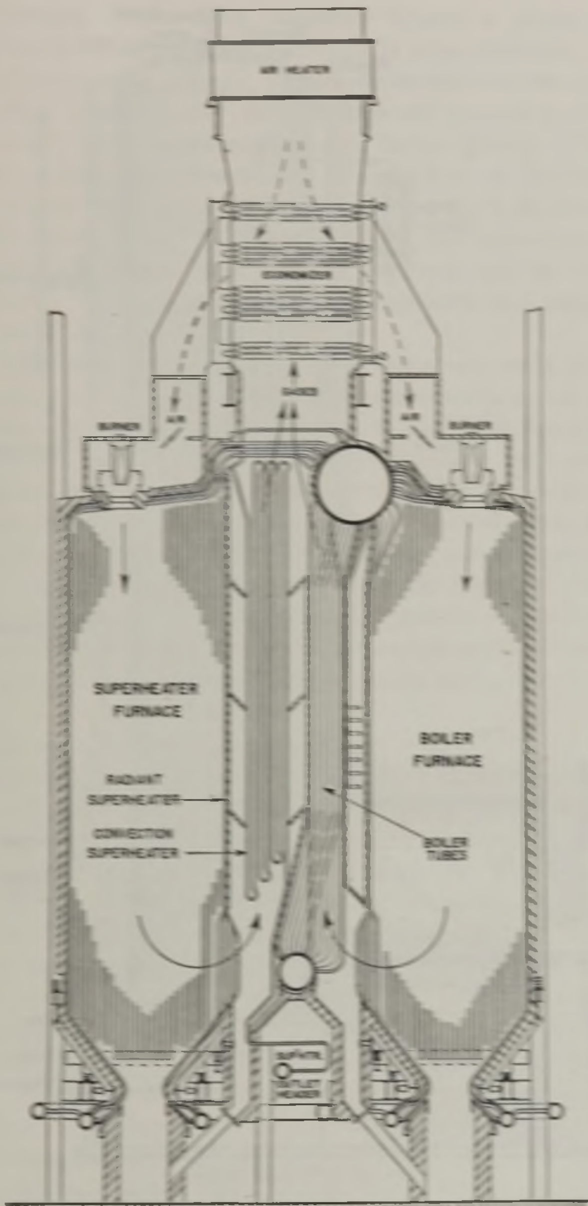


FIG. 45.—Foster-Wheeler steam-generating unit with separately fired superheater.

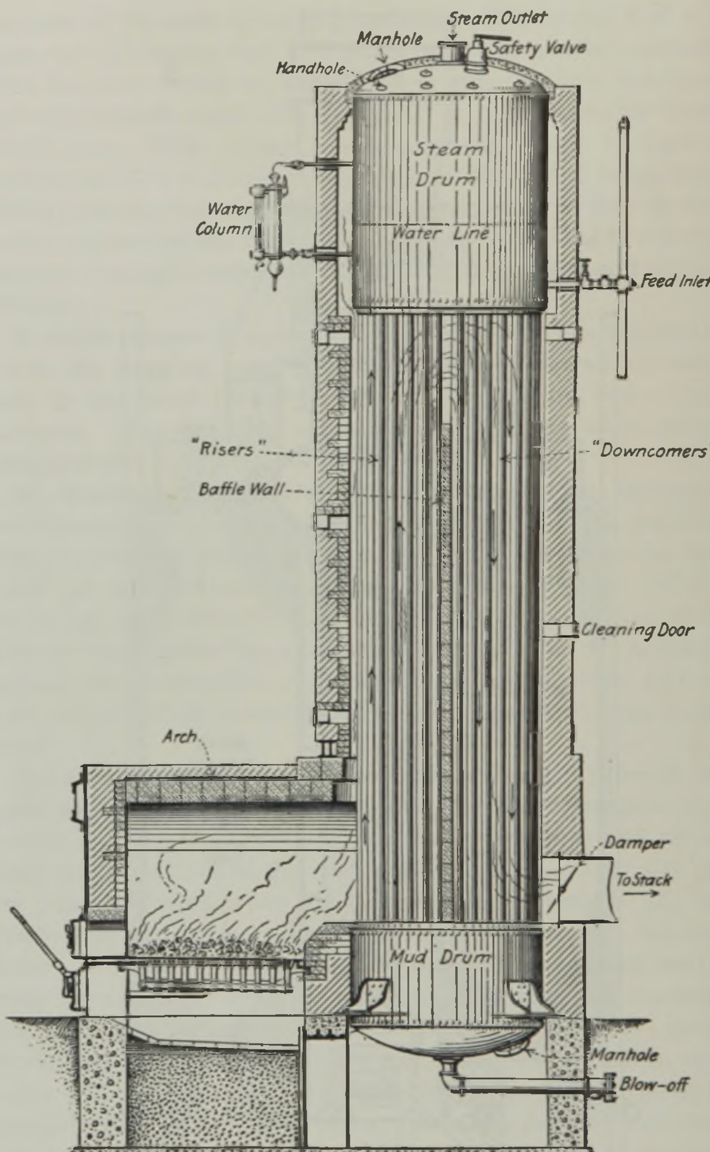


FIG. 49.—Wickes boiler.

84. Vertical Water-tube Boilers.—Where a plant is very limited in the floor space available, it is often desirable to use a vertical water-tube boiler. Figure 49 shows a cross section of the Wickes vertical boiler. The grates are located in a "Dutch oven" front built out from the main boiler setting. The gases pass up around the tubes in the forward half of the boiler and down around them in the rear half, leaving the boiler in the rear near the lower drum. The water inside the tubes flows in the same direction as the gases, in both the front and the rear compartments. These boilers are quick steamers and occupy relatively small floor space.

85. High-pressure Boilers.—Beginning about 1920, the design and construction of steam boilers for pressures in excess of about 400 psi to 450 psi have progressed rapidly and many high-pressure installations have been made. The pressure still commonly used in small power plants does not exceed 100 psi to 200 psi. However, the majority of the small installations of the past five or six years have pressures between 200 psi and 400 psi and even in quite moderate-sized power plants a boiler pressure of 450 psi is not uncommon. This is particularly true in paper-mill power plants and others where there is complete use made of the exhaust steam from the prime movers for heating and process work, or both.

In the central-station field where it is necessary to generate electrical energy at the lowest possible cost, boiler pressures have risen until there are now many plants operating at pressures in the range of 1,200 psi to 1,400 psi.

In order to prevent excessive moisture in the exhaust of a steam turbine (12 per cent is about the usual limit), it has been found that with an exhaust pressure of from 1 in. to 2 in. of mercury absolute, it is necessary to increase the temperature of the superheated steam with an increase of boiler pressure. With a pressure of 450 psi the steam temperature should be 725 F; with 650 psi, 825 F; with 850 psi, 925 F; and with 1,400 psi, 940 F. Thus, total steam temperatures have increased, since 1920, from a maximum of about 725 F to 1000 F in one experimental installation. Several plants are now operating, or are being installed, where the steam leaves the superheaters at temperatures ranging from 900 F to 960 F.

It is stated by Howe that, when a body is at a temperature of 878 F, it has a red color which is visible in the dark; when it has a temperature of 887 F, it has a red color which is visible in daylight. It is therefore evident that steam which is in the range 900 F to 950 F is "red hot."

With the present limitation of steam temperature at approximately 950 F because of the alloys available, the highest pressure that is found in use with single-barreled turbines is about 1,400 psi. Above that pressure it is necessary to reheat the steam after it has been partially expanded.

There is under construction at the present time at the Twin Branch Station of the American Gas & Electric Co. at Mishawaka, Ind., a unit to operate at a pressure of 2,500 psi in the saturated steam drums, using a regenerative-reheat cycle. The boiler will deliver steam from the superheaters at about 2,400 psi and 940 F and will reheat at about 450 psi to substantially this same temperature by passing the steam through a reheat superheater located in the boiler setting. This 2,400 psi pressure is about as high as it is possible to use in a boiler that depends upon convection currents for water circulation. Above that, it would appear that forced circulation is necessary as the density of the steam approaches that of water near the critical pressure of 3,206.2 psia.

These high-pressure units are, in general, located in public-utility plants where the high load factor makes high efficiency imperative. However, an increasing number of industrial plants are installing high-pressure units as a means of cutting the cost of power.

The forced circulation or series type of boiler, such as the Benson boiler, has not enjoyed any degree of popularity in this country, as yet, its use being limited to a single experimental unit at Purdue University. In Europe and in Great Britain some Benson boilers have been installed where the pressure is 3,206.2 psia, which is the critical pressure of steam. In fact, some of these boilers have been operated at pressures considerably above the critical pressure. One of the outstanding features of the Benson boiler, or any boiler for that matter which operates at or above the critical pressure, is the fact that no enthalpy of evaporation has to be supplied. As a result of this no steam-releasing surface is required and the boilers may be constructed without steam and water drums of the usual types.

Another important and highly interesting new high-pressure boiler development is the Velox boiler. This boiler is a high-capacity, high-pressure type. It is usually oil fired and has air for combustion supplied by turbocompressors at pressures ranging from about 25 psi to about 35 psi; the oil is also supplied under pressure and the pressure in the combustion chamber is about 25 psi to 35 psi. The products of combustion, after they have been greatly reduced in temperature, but still under pressure, are passed into a waste-heat turbine. This turbine drives the turbocompressor for supplying the combustion air and also an electric generator which supplies power for any desired uses. The steam output from the Velox boiler is generally used to drive a turboalternator set. The heat release per cubic foot of combustion space is about 900,000 Btu per cu ft per hr which is far above that obtained in ordinary practice where a heat release of 40,000 Btu per cu ft per hr represents good practice in modern central stations.

The first boilers for a pressure of 1,200 psi were very costly as the drums had to be forged from large steel ingots and many costly forging and reheating operations were required. Progress was slow until fusion welding was developed and successfully applied to the fabrication of large steam and water drums. Progress was also retarded until X-ray apparatus was developed which made it possible to examine the entire interior structure of welds and determine their character. When these two important developments had progressed to a satisfactory degree, high-pressure boiler installations mounted steadily in number and their cost decreased markedly.

A noteworthy high-pressure boiler installation is at the No. 1 Power Plant of the Ford Motor Company at Dearborn, Mich.

This unit is designated as boiler No. 7 and at the present time is the largest high-pressure boiler in the world. The following interesting data are published by the Ford Motor Company:

> The steam-generating unit will generate 900,000 lb of steam per hour at a pressure of 1,400 psi. The total steam temperature is 925 F, and the equivalent boiler horsepower is 26,500. The heating surface of the boiler proper equals 24,410 sq ft; the heating surface of the waterwalls, 9,500 sq ft; the heating surface of the superheater, 24,019 sq ft; the heating surface of the economizer, 25,200 sq ft; and the heating surface of the air preheater, 86,016 sq ft. The sum of all of the above heating surfaces equals 169,145 sq ft.

Some other interesting facts are given in the following:

The furnace is 25 ft 8 in. by 32 ft by 35 ft high and has a volume of 29,000 cu ft. There are 24,470 boiler tubes and the total length of the tubes is 12 miles. The total length of the superheater tubes is 8.5 miles. The guaranteed efficiency is 87.8 per cent. The total weight of the steam-generating unit is 3,631,200 lb. The material in the boiler amounts to 50 carloads. One upper steam drum has an internal diameter of 40 in., a thickness of $3\frac{11}{16}$ in., and a length of 28 ft 1 in. Two upper steam drums each have an internal diameter of 48 in., a thickness of $6\frac{1}{4}$ in., and a length of 35 ft 7 in. Two lower drums each have an internal diameter of 40 in., a thickness of $5\frac{1}{4}$ in., and a length of 31 ft. Forty-six tons of coal are consumed per hour at full load.

Table XVIII taken from the September, 1940, issue of *Power* gives a list of some of the most recent high-pressure installations.

For the higher pressures and temperatures and the larger capacities demanded under present-day conditions, the trend in boiler design is changing from the three or four standardized types. The new designs vary with the conditions but, in general, consist of one or two drums with water-cooled furnace walls, superheater (radiant or convection, or both), economizer, and air preheater. These changes are made necessary in order to accommodate larger and more efficient stokers, pulverized-coal equipment, water-cooled furnaces, higher steam pressures, higher steam temperatures, superheat control, economizers, air preheaters, greater steam output capacities, clean steam, long periods of operations at high load factors, and maximum efficiency.

What the future holds in store in the matter of boiler design, no one can foretell, as a few years ago boiler units of the size and power of those now in common use would have been thought impossible.

86. Heating Surface, Grate Surface, and Breeching.—*The water-heating surface in a boiler is that part of the boiler which has water on one side and hot gases on the other. Superheating surface has steam on one side and hot gases on the other.* In both cases the *side in contact with the hot gases* is the one to be measured.

The A.S.M.E. Code on Definitions and Values contains the following statements on Heating Surface:

The heating surface for boilers comprises the total area of all boiler elements forming part of the circulating system of the boiler proper which are in actual contact with hot gases. Any surface that is not in the boiler circulating system and is not connected to the steam space of the boiler should be considered as preheater or integral economizer surface, and not as heating surface of the boiler.

TABLE XVIII.—HIGH-PRESSURE POWER STATIONS[1]
(Steam pressure, 1,000 psi and above)

Name of plant	Year	Steam conditions at throttle		Kw, total (high-pressure unit)	Company and location
		Pressure, psi	Temperature, deg F		
Somerset	1941	1,950	960	20,000	Montaup Electric Co., Somerset, Mass.
Sherman Creek	1942	1,600	950	50,000	Consolidated Edison Co., New York, N. Y.
Oswego	1941	1,500	950	80,000	Central New York Power Corporation, Oswego, N. Y.
Waterside	1941	1,485	925	65,000	Consolidated Edison Co., New York, N. Y.
Burlington	1940	1,475	950	100,000	Public Service Electric & Gas Co., Burlington, N. J.
Atlantic City	1941	1,385	955	25,000	Atlantic City Electric Co., Atlantic City, N. J.
Philo	1941	1,375	950	85,000	Ohio Power Co., Philo, Ohio
Chester	1941	1,350	935	50,000	Philadelphia Electric Co., Chester, Pa.
Manchester St.	1941	1,350	915	40,000	Narragansett Electric Co. Providence, R. I.
Northwest	1940	1,325	935	50,000	Commonwealth Edison Co., Chicago, Ill.
Westport	1940	1,325	915	25,000	Consolidated Gas, Electric Light & Power Co., Baltimore, Md.
L Street	1939	1,300	910	25,000	Boston Edison Co., Boston, Mass.
Windsor	1941	1,300	925	60,000	West Penn Power Co., Power W. Va.

[1] From *Power*, September, 1940.

Superheater surface shall comprise total area of all elements in contact with the hot gases on one side and steam only on the other. Superheater, boiler, and preheater surfaces shall be separately stated. Since the gas side of the surface offers the controlling resistance to heat transmission, the heating surface of water-tube boilers shall be figured on the outside diameter of the tubes, and the heating surface of fire-tube boilers on the inside diameter of the tubes. Preheater and superheater surfaces shall be figured on outside diameters.

Boiler heating surface shall consist of that portion of the surface of the heat-transfer apparatus in contact with the fluid being heated on one side and the gas or refractory being cooled on the other, in which the fluid being heated forms part of the circulating system; this surface shall be measured on the side receiving heat. This includes the boiler, waterwalls, water screens, and water floor.

The proportion of grate surface to heating surface depends upon the kind of fuel and the intensity of the draft. In small boilers such as are used for heating purposes, with light draft and hard coal, it is usual to allow 1 sq ft of grate to from 20 sq ft to 30 sq ft of heating surface. In large-power boilers the ratio of grate surface to heating surface varies from 1:50 to 1:70. In locomotive boilers with forced draft the ratio is from 1:50 to 1:100.

The rate of combustion varies with the kind of coal and with the draft, and is expressed in pounds of fuel burned (*a*) per square foot of projected grate surface per hour, or (*b*) per cubic foot of furnace volume per hour. With anthracite coal and moderate draft, not exceeding $\frac{5}{10}$ in. of water, it is from 12 lb to 15 lb per sq ft of grate surface per hr, and with bituminous coal from 15 lb to 20 lb. With mechanical stokers and natural draft the rate averages from 20 lb to 40 lb per sq ft per hr; with mechanical draft from 40 lb to 60 lb per sq ft per hr. Rates as high as 200 lb per sq ft of grate surface per hr have been obtained with locomotive boilers with very high induced draft.

Sufficient combustion space, or furnace volume, must be provided in order to ensure thorough mixture of the gases given off by the fuel with the required amount of air. Furnaces equipped with chain-grate stokers require a volume of about $\frac{1}{2}$ cu ft for each pound of coal burned per hour; with underfeed stokers $\frac{1}{3}$ cu ft is necessary. With pulverized-coal furnaces, volumes of from $\frac{3}{4}$ to 11 cu ft. per pound of powdered fuel burned per hour are found to give the best results.

The air opening in the grate depends upon the kind and size of coal and usually varies from 25 to 50 per cent of the grate area.

Anthracite and the better grades of bituminous coal require less air opening than the poorer grades of coal. The total grate area depends upon the rate of combustion and the heating value of the fuel.

The following rule is used for determining the heating surface of a horizontal return-flue fire-tube boiler: *The heating surface is equal to one-half the external cylindrical surface of the shell, plus the internal area of all the tubes, plus two-thirds the area of both tube sheets, minus twice the combined external cross-sectional area of all the tubes, all expressed in square feet.*

The connection for carrying the hot gases from the boiler to the chimney is called the *breeching*. The area of the breeching is from one-sixth to one-eighth of the area of the grates, depending on the strength of the draft. The breeching is usually made of sheet steel well braced, and should be provided with a door for cleaning and inspection.

TABLE XIX.—DIAMETERS OF BOILER TUBES

Outside		Inside	
Inches	Feet	Inches	Feet
2	0.167	1.80	0.150
2½	0.208	2.28	0.190
3	0.250	2.78	0.232
3½	0.292	3.26	0.272
4	0.333	3.74	0.312
4½	0.375	4.24	0.353
5	0.417	4.72	0.393

87. Boiler Capacity.—Until recent years the unit in universal use for measuring the capacity of a boiler was the "boiler horsepower," and this is still the term commonly employed, although it is a misnomer to use the word "horsepower" in this connection. As applied to the rating of a boiler, the term is used to indicate the unit time rate of heat transfer. In order to give it a definite meaning when applied to the performance of boilers, the A.S.M.E. adopted the following standard of measurement: *A boiler horsepower is 34.5 lb of water evaporated per hr from and at 212 F into dry and saturated steam, or the equivalent in heating effect.* This is equal to $34.5 \times 970.3 = 33{,}475$ Btu per hr.

The Power Test Committee of the A.S.M.E. recommends that boiler performance, or output, be stated in terms of 1,000 Btu per hr instead of boiler horsepower. This *unit of evaporation* is called a kilo Btu (kB) and a boiler develops one of these units when it absorbs 1,000 Btu in the generation of steam. The unit represents $1{,}000 \div 970.3 = 1.03$ times the heat necessary to evaporate one pound of dry and saturated steam from and at 212 F. Although this measure of performance is frequently employed when stating the output of large power plants, "boiler horsepower" is still the unit in general use throughout this country in spite of its limitations.

When the term "boiler horsepower" (boiler hp) was originally selected, a 1-hp boiler would supply the steam required by a 1-hp engine, but the economy of engines has so increased that at the present time *there is no constant relation between a boiler horsepower and an engine horsepower.* The boiler horsepower required to supply steam for a given engine horsepower will be determined by the number of pounds of steam the engine requires to develop a horsepower. The steam required per horsepower-hour (hp-hr) varies through a wide range in the different types of engines, as for example from 8 lb per boiler hp-hr in large condensing turbines to 50 lb or 60 lb in small noncondensing units.

The ability of a boiler to make steam depends upon the number of square feet of heating surface in it. Although in rating boilers it has been customary to allow 10 sq ft of heating surface per boiler horsepower, this figure is arbitrary and the ratings arrived at by its use do not in any way necessarily indicate the relative capacities of the boilers. Some manufacturers rate their boilers on a basis less than 10 sq ft of heating surface per boiler horsepower. All that any of these ratings mean is a claim by the manufacturer that his product, under ordinary operating conditions, or conditions which may be specified, will evaporate 34.5 lb of water per hr from and at 212 F per a certain definite amount of heating surface.

Owing to the improvements in boiler and furnace design, most modern boilers are able to run continuously at from 150 per cent to 250 per cent over their rating, and for short periods at 400 per cent or even 600 per cent over. (There is no longer much significance attached to the term "rated boiler horsepower.") Therefore, in order to have definite standards by which to measure boiler capacity it is customary, in writing specifications for boilers, to state *the number of square feet the boiler is to contain and the number of pounds of water it is to evaporate under given conditions*, rather than the boiler horsepower. The rating of boilers *for size only*, in square feet of heating surface, is recommended by the A.S.M.E.

88. Boiler Economy.—The economy of a boiler is usually expressed as the number of pounds of water evaporated by the boiler per pound of coal fired.

In order to compare boilers working under different conditions of feed temperature and steam pressure and with different coals, it is best to reduce them all to the same conditions, and the *economy* may then be expressed as *the number of pounds of equivalent evaporation from and at* 212 *F per pound of combustible burned. By "equivalent evaporation from and at* 212 *F" is meant the number of pounds of water that would be evaporated from a feed temperature of* 212 *F into dry and saturated steam at* 212 *F by the expenditure of the same amount of heat as is actually used under the given conditions.*

The "factor of evaporation" is that factor by which the water evaporated must be multiplied in order to get the equivalent evaporation. It is equal to the heat necessary to make one pound of steam under the given conditions divided by the heat necessary to make one pound of dry and saturated steam from and at 212 F, *viz.*, 970.3 Btu.

Originally, in the ordinary form of boiler, the maximum economy was obtained where the rate of evaporation did not exceed 3 lb of water per hr per sq ft of heating surface. However, in modern boilers, this rate of evaporation has been greatly exceeded without material loss in efficiency. In large power stations carrying variable loads and using stokers, or pulverized fuel, and water-cooled furnaces, the rate has been materially increased to from 8 lb to 12 lb, the loss in economy being more than offset by the reduced investment for boilers and power-house space. The rate of evaporation will depend upon the amount of heat generated in the furnace and the amount that is transmitted to the water. Overloading, or forcing, a boiler does not injure it, since, as long as it is transmitting heat to the water, no harm is done. There may be, however, danger to the furnace lining, unless it is protected by waterwalls.

It must be kept in mind that these higher rates cannot be obtained with all types of boilers, the design of the boiler having much to do with it. The amount of heat given off by the fuel will depend upon the kind of fuel used, the area of the grate, the amount of draft, and the method of firing. A very rapid rate of combustion, much in excess of the designed rate, usually results in a large escape of heat to the stack and reduced economy.

89. Efficiency of Steam Boilers.—The efficiency of *boiler, furnace, and grate* is the ratio of the heat absorbed per pound of *coal fired* to the heating value of a pound of *coal*, or it may be stated as *the ratio of the heat absorbed by the boiler to the heat supplied to the grates.*

The efficiency of *boiler and furnace* is the ratio of the heat absorbed per pound of *combustible burned* to the heating value of a pound of *combustible*, or it is *the ratio of the heat absorbed by the boiler to the heat supplied to the furnace* (equals the heat given off by the grates).

The efficiency of the *grate* is the ratio of *combustible burned* to the *combustible supplied* to the grates.

The "heat absorbed" per pound of dry coal, or combustible, is found by multiplying the equivalent evaporation from and at 212 F per pound of dry coal, or combustible, by 970.3.

The "dry coal fired" is found by deducting the moisture in the coal from the total weight of coal supplied to the grates.

The "combustible burned" is determined by deducting from the weight of coal fired, the weight of moisture in the coal plus the weight of ash and refuse taken from the ashpit plus "the weight of dust and soot, if any, withdrawn from the tubes, flues, and combustion chamber, including ash carried away in the gases, if any, determined from the analysis of coal and ash."

The "heating value of a pound of combustible" is equal to the heating value of one pound of dry coal divided by 1 minus the percentage of ash in the dry coal as shown by analysis.

Actual tests of various boilers show that the efficiency of small, hand-fired plants may run as low as 40 per cent to 60 per cent. Stoker-fired plants under ordinary working conditions show an efficiency of from 60 per cent to 80 per cent. Large modern central stations using pulverized coal and with economizers or air preheaters, or both, have developed efficiencies of from 80 per cent to 85 per cent and, under exceptional conditions, nearly 92 per cent.

90. Losses in Boiler.—The principal losses in a boiler are the heat that is carried away by the flue gases, the loss due to water vapor in the products of combustion formed by burning the hydrogen in the coal, the loss through the grates, and the loss by radiation. Of these, the largest is the heat carried up the chimney by the stack gases.

The heat carried away by the chimney gases depends upon the amount of air admitted to the fire and upon the temperature at which the gases leave the boiler. In a properly operated plant, the gross loss of heat up the chimney should not exceed 20 per cent. It is often much more than this, caused by the fireman's admitting too much air to the coal—more than is necessary for its complete combustion. This excess air is heated from the temperature of the boiler room to the temperature of the stack gases, and all the heat used for this purpose passes up the chimney and is wasted. It is, therefore, very important that the amount of air admitted to the fire should not be more than is absolutely necessary. Whether or not the correct amount is being supplied will be shown by the stack-gas analysis.

91. Heat Balance.—A statement of the relative proportions of the losses in a boiler plant is termed a *heat balance* and may be computed from the analysis of the coal and the stack gases. Such a balance gives an accurate idea of the efficiency of combustion and greatly aids in improving boiler efficiency. It is generally based upon the heating value of 1 lb of dry coal considered as 100 per cent.

The method of determining the heat balance, or distribution of the heating value of the fuel based upon dry coal, as outlined in the A.S.M.E. Test Code (short form) for Stationary Steam Generating Units, is shown in Table XX.

Example.—Compute a heat balance from the following data taken from a boiler test: Temperature of wet-bulb thermometer, 67 F; temperature of boiler room, 93 F; temperature of exit gases, 575 F. Ultimate analysis of coal as fired: C, 76.92 per cent; H_2, 5.46 per cent; O_2, 8.09 per cent; N_2, 1.07 per cent; S, 0.98 per cent; ash, 5.57 per cent. Moisture in coal as fired, 1.91 per cent; Btu per pound of coal as fired, 13,733; ash and refuse, 6.9 per cent of coal as fired; unconsumed carbon in ash, 19.2 per cent. Flue-gas analysis: CO_2, 14 per cent; O_2, 5.5 per cent; CO, 0.42 per cent; N_2, 80.08 per cent. Evaporation from and at 212 F per pound of coal as fired : 10.91 lb.

From the wet- and dry-bulb thermometer readings and psychrometric tables, the weight of moisture in the air per pound of dry air supplied is found to be 0.0127 lb.

TABLE XX.—HEAT BALANCE OR DISTRIBUTION OF THE HEATING VALUE OF THE FUEL BASED UPON DRY COAL

	Btu	Per cent
1. Heat absorbed by boiler = evaporation from and at 212 F per lb coal as fired × 970.3		
2. Loss due to moisture in coal = $\frac{\text{per cent moisture in coal as fired}}{100} \times (1{,}089 + 0.46t_g - t_c)$* where t_g = temperature of flue gases and t_c = temperature of coal		
3. Loss due to water formed by the burning of hydrogen $= \frac{\text{per cent hydrogen in coal as fired}}{100} \times 9 \times (1{,}089 + 0.46t_g - t_c)$		
4. Loss due to moisture in air = air supplied per lb coal as fired × moisture in air (lb per lb air) × 0.46 $(t_g - t_1)$ where t_1 = temperature of air for combustion		
5. Loss due to heat carried away in dry chimney gases = weight of gas per lb coal as fired† × $0.24(t_g - t_1)$		
6. Loss due to incomplete combustion of carbon = $\frac{CO}{CO_2 + CO}$ × carbon burned per lb coal as fired × 10,160‡		
7. Loss due to unconsumed combustible in refuse = $\frac{\text{refuse, per cent coal as fired} - \text{per cent ash by analysis}}{100} \times 14{,}600$		
8. Loss due to unconsumed hydrogen and hydrocarbons, to radiation, and unaccounted for = heating value per lb fuel as fired − (items 1 to 7 inclusive)		
9. Total		100.00

* The expression $(1{,}089 + 0.46t_g - t_c)$ represents the heat required to raise 1 lb of moisture in the coal from the temperature of the coal up to the boiling point, and then to superheat it to the temperature of the flue gases. Because of the low vapor pressure exerted by the water in the flue gas, it is assumed that the evaporation of the moisture takes place at 150 F, in which case the expression for the enthalpy of the superheated steam is $1126.1 + 0.46(t_g - 150)$. From this must be subtracted $(t_c - 32)$ or the heat in the moisture in the fuel as it enters the furnace at boilerroom temperature.

† The weight of gas per pound of carbon burned may be calculated from the gas analys is as follows:

Dry gas per pound carbon $= \frac{11CO_2 + 8O_2 + 7(CO + N)}{3(CO_2 + CO)}$ in which, CO, CO_2, O_2, and N are the percentages by volume of the several gases.

The weight of the dry gas per pound of coal as received is found by multiplying the dry gas per pound of carbon by the percentage of carbon in the coal and dividing by 100.

‡ CO_2 and CO are, respectively, the percentage by volume of carbon dioxide and carbon monoxide in the flue gases. The quantity 10,160 = number of heat units generated by burning to carbon dioxide, 1 lb of carbon contained in carbon monoxide.

Solution.

Item 1.

Heat absorbed by boiler = evaporation from and at 212 F per pound of coal as fired × 970.3 ÷ Btu per pound of coal as fired.

$10.91 \times 970.3 \div 13{,}733 = 10{,}585 \div 13{,}733 = 0.7708$, or 77.08 per cent.

Item 2.

$$\text{Loss due to moisture in coal} = \frac{\text{per cent of moisture in coal as fired}}{100} \times$$

$(1{,}089 + 0.46t_g - t_c) \div$ Btu per pound of coal as fired.

Assume the temperature of the coal to be the same as the boiler-room temperature.

$$\frac{1.91}{100} \times (1{,}089 + 0.46 \times 575 - 93) \div 13{,}733 = 24 \div 13{,}733 = 0.0018, \text{ or}$$

0.18 per cent.

Item 3.

Loss due to water formed by the burning of hydrogen =

$$\frac{\text{Per cent of hydrogen in coal as fired}}{100} \times 9 \times (1{,}089 + 0.46t_g - t_c) \div$$

Btu per pound of coal as fired.

$$\frac{5.46}{100} \times 9 \times (1{,}089 + 0.46 \times 575 - 93) \div 13{,}733 = 620 \div 13{,}733 =$$

0.0452, or 4.52 per cent.

Item 4.

Loss due to moisture in air = Air supplied per pound of coal as fired × moisture in air × $0.46(t_g - t_1)$ ÷ Btu per pound of coal as fired, where t_1 = temperature of air in boiler room.

$$\text{Dry gas per pound of carbon burned} = \frac{11CO_2 + 8O_2 + 7(CO + N_2)}{3(CO_2 + CO)}$$

$$= \frac{11 \times 14 + 8 \times 5.5 + 7(0.42 + 80.08)}{3(14 + 0.42)} = \frac{761.5}{43.26} = 17.603 \text{ lb.}$$

The carbon burned per pound of coal fired =

$$\frac{\text{per cent of carbon in coal from ultimate analysis}}{100} - \frac{\text{per cent of refuse* } \times \text{ per cent combustible in refuse}}{10{,}000}$$

$$\frac{76.92}{100} - \frac{6.9 \times 19.2}{10{,}000} = 0.7692 - 0.0132 = 0.756 \text{ lb.}$$

Dry gas per pound of coal as fired = $17.603 \times 0.756 = 13.308$ lb.

Air supplied per pound of coal as fired = dry gas per pound of coal as fired + $\dfrac{9 \times \text{per cent of hydrogen in coal as fired}}{100}$

$$- \frac{100 - \text{per cent of refuse}}{100} = 13.308 + \frac{9 \times 5.46}{100} - \frac{100 - 6.9}{100} =$$

$$13.308 + 0.4914 - 0.931 = 12.868 \text{ lb.}$$

* When the per cent of refuse is not given, it may be computed as follows:

$$\frac{\text{Ash by analysis per pound of coal}}{1 - \text{combustible in refuse per pound of refuse}} \times 100.$$

The loss due to moisture = 12.868 × 0.0127 × 0.46(575 − 93) ÷ 13,733 = 36.3 ÷ 13,733 = 0.0026, or 0.26 per cent.

Item 5.

Loss due to heat carried away in dry chimney gases = weight of gas per pound of coal as fired × $0.24(t_g - t_1)$ ÷ Btu per pound of coal as fired. 13.308 × 0.24(575 − 93) ÷ 13,733 = 1,539 ÷ 13,733 = 0.1121, or 11.21 per cent.

Item 6.

Loss due to incomplete combustion of carbon = $\frac{CO}{CO_2 + CO}$ × carbon burned per pound of coal as fired × 10,160 ÷ Btu per pound of coal as fired.

$\frac{0.42}{14 + 0.42}$ × 0.756 × 10,160 ÷ 13,733 = 223.7 ÷ 13,733 = 0.0163 or 1.63 per cent.

Item 7.

Loss due to unconsumed combustible in refuse =

$\frac{\text{refuse, per cent of coal as fired} - \text{per cent of ash by analysis}}{100}$ × 14,600 ÷ Btu per pound of coal as fired.

$\frac{6.9 - 5.57}{100}$ × 14,600 ÷ 13,733 = 194.2 ÷ 13,733 = .0141, or 1.41 per cent.

Item 8.

Loss due to unconsumed hydrogen and hydrocarbons, to radiation, and unaccounted for = 100 − (sum of items 1, 2, 3, 4, 5, 6, 7).

100 − (77.08 + 0.18 + 4.69 + 0.26 + 11.21 + 1.63 + 1.41) = 100 − 96.46 = 3.54 per cent.

Ans.

TABLE XXI.—HEAT BALANCE IN BOILER PLANT

Distribution of heat in coal as fired	Btu	Per cent
1. Heat absorbed by the boiler	10,585	77.08
2. Loss due to evaporation of moisture in the coal	24	0.18
3. Loss due to heat carried away by steam formed by the burning of hydrogen in the coal	620	4.52
4. Loss due to heating moisture in the air	36	0.26
5. Loss due to heat carried away in dry chimney gases	1,539	11.21
6. Loss due to carbon monoxide	224	1.63
7. Loss due to combustible in ash and refuse	194	1.41
8. Loss due to unconsumed hydrogen and hydrocarbons, to radiation, to sensible heat in ash, and unaccounted for	511	3.71
Total	13,733	100.00

The following comments on several of the items in Table XX should be noted:

3. Moisture in the Coal.—Besides the loss in heat, moisture exercises a loss in another way. By reducing the weight of actual coal purchased by the weight of water present, you must pay for the water at the same rate per ton as you pay for the coal. High heat-value coals contain approximately 3 to 4 per cent moisture while many coals carry 10 to 14 per cent and some as high as 25 to 40 per cent moisture. Moisture loss is approximately 1 per cent for every 10 per cent of moisture present in the coal.

4. Loss Due to Moisture in the Air.—This loss is usually small and is frequently included in Item 5.

5. Loss Due to Heat Carried Away in the Dry Chimney Gases.—The loss of heat in the chimney gases, as determined by the formula in Table XX, Item 5, is dependent upon the amount of air admitted to the boiler setting, both through the grates and by leakage through cracks or holes in the brickwork, open or loose fitting doors, and around steel to brickwork connections, and the temperature of the gases leaving the boiler.

This loss can be kept as low in everyday practice (as shown by Col. 3, Table XXIII, Item 5) as 10 per cent. It is frequently found to be 25 per cent and possibly more.

6. Loss Due to Incomplete Combustion of Carbon.—This loss includes the loss caused by the escape of visible smoke and a small amount of combustible gas. Although in most plants this loss is small on account of the large excess of air, in some cases it may reach an alarming figure. Such high losses occur only in small furnaces where no provision is made for effective mixing of volatile combustible and air, and where heavy firings are made at long intervals.

7. Loss Due to Unconsumed Combustible in Refuse.—This loss computed from the formula in Table XX, Item 7, may be, in carelessly handled plants, a very large one, since all of the combustible matter which has fallen to the ashpit, also that carried into boiler setting and stack by draft, is a direct loss. It is recommended in the A.S.M.E. Test Code for Stationary Steam Generating Units that the combustible in the refuse be computed at 14,600 Btu per lb. This loss should not exceed approximately 3 per cent (see Table XXIII, Col. 3), but has been known to reach 15 per cent and over.

8. Loss Due to Unconsumed Hydrogen, Hydrocarbons, Radiation, and Unaccounted for.—Included in Item 8 is the loss due to unconsumed hydrogen and hydrocarbons. In the presence of a CO loss (see Item 6), these gases may form a considerable loss, in fact one of greater magnitude than the CO loss.

When the unaccounted-for loss, as is the case in many tests, exceeds 10 per cent the test should be repeated, as there are undoubtedly errors somewhere in the work.

From the heat balance formulas described in Table XX, Table XXIII has been computed, using the data given. The results are fairly indicative of the conditions heading each column. Column 1 is a theoretically perfect condition which cannot be realized in

TABLE XXII.—HEAT BALANCE

	1	2	3	4	5	6
	Theoretically perfect	Best test practice	Best every day practice	Good every day practice	Average every day practice	Poor practice
Heat balance, based on per cent of calorific value of coal "as fired"						
1. Heat absorbed by boiler	90.24	82.00	76.20	70.80	61.70	47.50
2. Loss due to moisture in coal	0.16	0.17	0.37	0.58	0.81	1.70
3. Loss due to water from combustion of hydrogen	4.95	5.06	5.15	5.20	5.38	5.57
4. Loss due to moisture in air	0.17	0.26	0.37	0.53	0.78	1.05
5. Loss due to dry chimney gas	4.48	7.22	10.04	14.16	20.81	27.70
6. Loss due to incomplete combustion	0	0.58	0.88	0	0	0
7. Loss due to unconsumed combustible in refuse	0	1.88	3.23	4.05	5.02	7.56
8. Loss due to unconsumed hydrogen and hydrocarbons, radiation and unaccounted for	0	2.83	3.76	4.68	5.50	8.92
Total	100.00	100.00	100.00	100.00	100.00	100.00
Data from which heat balance is calculated						
CO_2	21.0	15.0	12.0	9.0	7.5	6.5
O_2	0	3.7	6.7	10.0	12.0	13.5
CO	0	0.16	0.20	0	0	0
N	79.0	81.1	81.1	81.1	80.5	80.0
Pounds gas per lb carbon	12.4	16.7	20.6	27.5	32.9	37.9
Pounds gas per lb wet coal	9.5	12.6	14.9	19.3	22.3	22.9
Temperature boiler room, deg F	92	92	92	92	92	92
Temperature gases leaving boiler	365.88	425	475	500	600	700
Temperature steam due to 150 lb pressure	365.88	365.88	365.88	365.88	365.88	365.88
Pounds moistuie in dry air per lb	0.02	0.02	0.02	0.02	0.02	0.02
Percentage combustible in refuse	0	20	30	35	40	50
Ultimate analysis of coal as fired						
Moisture	1.90	1.90	4.00	6.00	8.00	15.00
Ash	7.21	7.21	7.05	6.90	6.76	6.25
Carbon	77.04	77.04	75.39	73.82	72.25	66.75
Hydrogen	6.31	6.31	6.17	6.04	5.91	5.47
Oxygen	5.04	5.04	4.92	4.82	4.72	4.36
Nitrogen	1.53	1.53	1.50	1.47	1.44	1.33
Sulphur	0.97	0.97	0.96	0.94	0.92	0.85
Total	100.00	100.00	100.00	100.00	100.00	100.00
Btu per pound	13,928	13,928	13,630	13,346	13,062	12,068
Evaporation "from and at" 212 F per lb wet coal	12.95	11.17	10.70	9.74	8.31	5.91

Ultimate analysis per pound combustible:

C—84.75 per cent; H—5.58 per cent; O—6.92 per cent; N—1.68 per cent; S—1.07 per cent; Btu 15,324.

TABLE XXIII.—COMPARISON OF HEAT BALANCES FOR WEST VIRGINIA COAL (BITUMINOUS)
("Standard" conditions: flue-gas temp. 365 F; air to F.D. fans 80 F and 60 per cent relative humidity; 30 per cent excess air supplied; 12 per cent combustible in refuse)
(Deviation from "standard" conditions)

	1		2		3		4		5		6	
	Standard		70 per cent excess air		400 F flue-gas temp.		5 per cent increased surface moisture		20 per cent combustible in refuse		1 per cent CO flue gas	
	Btu	Per cent	Btu	Per cent	Btu	Per cent	Btu	Per cent	Btu	Per cent	Btu	Per cent
1. Heat value of coal	13,150	100.00	Same		Same		Same		Same		Same	
Heat loss:												
2. Moisture in coal	36.5	0.28	Same		37.0	0.28	95.3	0.72	Same		Same	
3. Water from hydrogen in coal	521.1	3.96	Same		527.8	4.01	Same		Same		Same	
4. Moisture in air	22.3	0.17	29.1	0.22	25.0	0.19	Same		Same		Same	
5. Dry chimney gases	913.1	6.94	1,185.3	9.01	1,025.3	7.80	Same		Same		Same	
6. Incomplete combustion (CO)	0	0	Same		Same		Same		Same		Same	
7. Combustible in refuse	179.5	1.36	Same		Same		Same		328.5	2.50	537.3	4.08
8. Radiation	125.0	0.95	Same		Same		Same		Same		Same	
9. Unaccounted for	197.2	1.50	Same		Same		Same		Same		Same	
10. Total losses	1,994.7	15.16	2,273.7	17.28	2,116.8	16.10	2,053.5	15.60	2,143.7	16.30	2,532.0	19.25
11. Boiler unit efficiency (heat absorbed)	11,155.3	84.84	10,876.3	82.72	11,033.2	83.90	11,096.5	84.40	11,006.3	83.70	10,618.0	80.75
12. Nitrogen	81.3		80.8		81.3		81.3		81.3		80.8	
13. Carbon dioxide	13.8		10.5		13.8		13.8		13.6		12.8	
14. Carbon monoxide	0		0		0		0		0		1.0	
15. Oxygen	4.9		8.7		4.9		4.9		5.1		5.4	

practice even under the most skillful handling. Column 2 represents the best results to be obtained under test condition when handled by a very skilled engineer. Column 3 represents the best everyday practice in large plants under the best supervision. Column 4 represents what can be done by any very careful operator, with a good boiler plant. Column 5 represents usual practice in what are reputed to be fairly well-handled plants. Column 6 is poor practice and represents a heavy loss.

Table XXIV.—Effect of Increased Load
West Virginia Coal (Bituminous)
("Standard" conditions: flue-gas temp. 365 F; air to F.D. fans 80 F and 60 per cent relative humidity; 30 per cent excess air supplied; 12 per cent combustible in refuse)
(2-hr. maximum conditions: flue-gas temp. 397 F; 25 per cent excess air supplied; air to F.D. fans 80 F and 60 per cent relative humidity; 20 per cent combustible in refuse)

	"Standard" (normal steaming rate)		2-hr. maximum steaming rate	
	Btu	Per cent	Btu	Per cent
1. Heat value of coal	13,150	100.00	13,150	100.00
Heat loss:				
2. Moisture in coal	36.5	0.28	36.9	0.28
3. Water from hydrogen in coal	521.1	3.96	575.2	4.37
4. Moisture in air	22.3	0.17	23.8	0.18
5. Dry chimney gases	913.1	6.94	976.9	7.43
6. Incomplete combustion (CO)	0	0	0	0
7. Combustible in refuse	179.5	1.36	328.5	2.50
8. Radiation	125.0	0.95	98.6	0.75
9. Unaccounted for	197.2	1.50	197.2	1.50
10. Total losses	1,994.7	15.16	2,237.1	17.01
11. Boiler unit efficiency (heat absorbed)	11,155.3	84.84	10,912.9	82.99
12. Nitrogen	81.3		81.4	
13. Carbon dioxide	13.8		14.2	
14. Carbon monoxide	0		0	
15. Oxygen	4.9		4.4	

Table XXIII, showing a comparison of heat balances, has been computed from a typical analysis of a West Virginia bituminous coal.

"Standard" operating conditions are assumed as follows: excess air, 30 per cent; flue-gas temperature, 365 F; combustion air temperature, 80 F; relative humidity, 60 per cent; combustible in refuse, 12 per cent. The heat balance for these conditions is in Col. 1. The other heat balances have the same operating conditions except for that one listed at the top of the column. Column 2 shows what happens when the air-fuel ratio is too high; Col. 3 is for the condition of dirty heating surfaces; Col. 4 is for watered coal; Col. 5 shows what occurs when the refuse is discharged with a rather large amount of combustible in it; Col. 6 portrays the effects of an uneven fire bed.

The reduction in efficiency of the boiler unit is a rather small value in per cent; considered over a year's time in tons of coal wasted, it becomes very appreciable in a large unit.

When the load on a boiler unit is changed, usually several items on the heat balance are influenced. Table XXIV shows the effect of increased load on a unit as compared with the "standard" operating conditions.

The excess air has decreased to 25 per cent, which is a characteristic of the stoker and furnace, the flue-gas temperature has increased to 397 because of the larger gas flow through the tube nest, and the carbon in the refuse has increased to 20 per cent, which is a characteristic of the stoker. The radiation loss decreases as the load increases, because the heat release increases much faster than does the loss.

Example.—A 48-in. by 12-ft return-flue fire-tube boiler has thirty 4-in. tubes. It evaporates 1,600 lb of water per hr from a feed temperature of 120 F into steam at 100 psi. What per cent of its rating is the boiler developing?

Solution.—First find the heating surface from the rule in Art. 86.

Heating surface of cylindrical portion of shell

$$= \tfrac{1}{2} \times 3.1416 \times 4 \times 12 = 75.4 \text{ sq ft.}$$

Heating surface of tubes $= 30 \times 3.1416 \times \frac{3.74}{12} \times 12 = 352.5$ sq ft.

Heating surface of tube sheets

$$= 2\left[\tfrac{2}{3}\,(3.1416 \times 2 \times 2) - 30 \times 3.1416 \times \tfrac{1}{6} \times \tfrac{1}{6}\right]$$

$$= 2(8.38 - 2.62) = 11.5 \text{ sq ft.}$$

$$= 439.4 \text{ sq ft.}$$

Total heating surface

From Art 87, the rated horsepower $= \dfrac{439.4}{10} = 43.94.$

Now find the actual horsepower developed.

The heat actually used in evaporating 1 lb of water is equal to the total heat in 1 lb of steam at a given pressure minus the heat already in the feed water.

The heat used in evaporating water under actual conditions

$$= 1,600[1,189.6 - (120 - 32)] = 1,762,560 \text{ Btu.}$$

From Art. 88, the equivalent evaporation from and at 212 F

$$= \frac{1,762,560}{970.3} = 1,816 \text{ lb per hr}$$

and from Art. 87, the boiler horsepower

$$= \frac{1,816}{34.5} = 52.6$$

$$\frac{52.6}{43.9} = 1.20, \text{ or } 120 \text{ per cent.}$$

Ans. The boiler is developing 20 per cent overload.

Example.—A boiler evaporates 8.23 lb of water per pound of coal fired. The feed temperature is 120 F; steam pressure, 100 psi. The coal as fired contains 2 per cent moisture. Dry coal contains 5 per cent ash and has a heating value of 12,800 Btu per lb. Twelve per cent of the coal fired is taken from the ashpit in the form of ash and refuse. (*a*) Find the efficiency of the boiler, furnace, and grates combined. (*b*) Find the efficiency of the boiler and furnace.

Solution.—(*a*) Heat necessary to evaporate 1 lb of water

$$= 1,189.6 - (120 - 32) = 1,101.6 \text{ Btu.}$$

Water evaporated per pound of dry coal fired

$$= \frac{8.23}{1.00 - 0.02} = \frac{8.23}{0.98} = 8.4 \text{ lb.}$$

Heat utilized per pound of dry coal fired

$$= 8.4 \times 1,101.6 = 9,253 \text{ Btu.}$$

Efficiency of boiler, furnace, and grates combined

$$= \frac{\text{heat utilized per pound of dry coal fired}}{\text{heating value of 1 lb of dry coal}}$$

$$= \frac{9,253}{12,800} = 0.7229, \text{ or } 72.29 \text{ per cent.}$$

(*b*) Heating value of 1 lb of combustible

$$= \frac{12,800}{1.00 - 0.05} = \frac{12,800}{0.95} = 13,474 \text{ Btu.}$$

Water evaporated per pound of combustible burned

$$= \frac{8.23}{1.00 - (0.02 + 0.12)} = \frac{8.23}{0.86} = 9.571 \text{ lb.}$$

Heat utilized per pound of combustible burned

$$= 9.571 \times 1,101.6 = 10,543 \text{ Btu.}$$

Efficiency of boiler and furnace

$$= \frac{\text{heat utilized per pound of combustible burned}}{\text{heating value of 1 lb of combustible}}$$

$$= \frac{10{,}543}{13{,}474} = 0.7825, \text{ or } 78.25 \text{ per cent.}^{1}$$

Ans. $\begin{cases} (a) & 72.29 \text{ per cent.} \\ (b) & 78.25 \text{ per cent.} \end{cases}$

Example.—If 26 lb of. air are used to burn 1 lb of coal containing 13,500 Btu, and the temperature of the stack gases is 550 F, what per cent of heat is lost up the stack in dry stack gases, if the temperature of the boiler room is 70 F?

Solution.—If there were no ash in the coal, each pound burned would give off 1 lb of gas and the total weight of stack gas per pound of coal fired would be 26 + 1 = 27 lb. This, however, is never the case, as there are always some ash and unburned coal, and hence the actual weight of stack gas per pound of coal is something a little less than 27 lb. The average of the specific heats of the various components of the stack gases is a little higher than that of air, 0.2410. To be absolutely correct it would be necessary to multiply the weight of each of the various gases in the stack gas by its particular specific heat, and then add these products to get the Btu necessary to raise the products of combustion one degree. The method commonly used, however, is to assume the specific heat of the stack gases to be the same as that of air, 0.2410, although really it is slightly higher, and to assume that 1 lb of gas is given off from 1 lb of coal, although in reality it is a little less. Thus, one assumption practically offsets the other, and the result is approximately correct.

Hence, the heat necessary to raise the products of combustion one degree

$$= 0.241(26 + 1) = 6.51 \text{ Btu.}$$

Rise in temperature of the stack gases

$$= 550 - 70 = 480 \text{ deg.}$$

Heat necessary to raise the stack gases 480 deg

$$= 480 \times 6.51 = 3{,}125 \text{ Btu.}$$

Per cent of heat lost up the stack in dry stack gases

$$= \frac{3{,}125}{13{,}500} = 0.2315, \text{ or } 23.15 \text{ per cent.}$$

[1] This answer may be checked as follows:

Efficiency of grate alone

$$= \frac{\text{combustible burned per pound of coal fired}}{\text{combustible fired per pound of coal fired}}$$

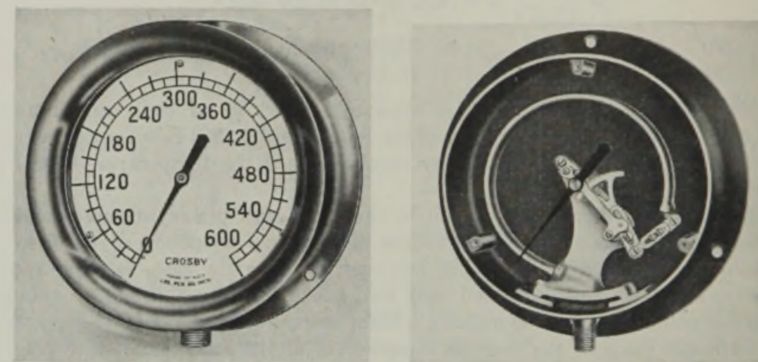
$$= \frac{1.00 - (0.02 + 0.12)}{1.00 - [0.02 + 0.05(1.00 - 0.02)]} = \frac{0.86}{0.931} = 0.9237, \text{ or } 92.37 \text{ per cent.}$$

Efficiency of boiler and furnace

$$= \frac{\text{efficiency of boiler, furnace, and grate}}{\text{efficiency of grate alone}}$$

$$= \frac{0.7229}{0.9237} = 0.7826, \text{ or } 78.26 \text{ per cent.}$$

92. Boiler Accessories.—In order to determine the physical condition of the steam and water in a boiler, all boilers are provided with a *steam gage* showing the pressure per square



Elevation. Interior mechanism.

Fig. 50.—Bourdon-tube pressure gage.

inch in the boiler, a *gage glass* to indicate the water level in the boiler, and a *safety valve* which automatically relieves the pressure in the boiler should it exceed the safety point. The *blowoff cock* is attached to the lowest point of the boiler and drains the water from the boiler. This is usually opened from time to time to blow the mud and settlings out of the boiler. Another accessory is a *water regulator* for regulating the level of the water in the boiler.

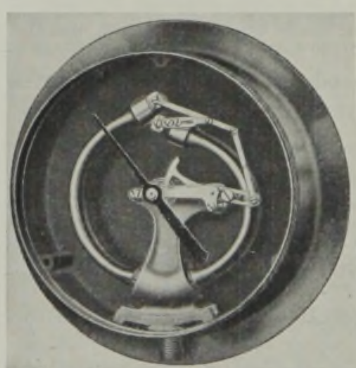
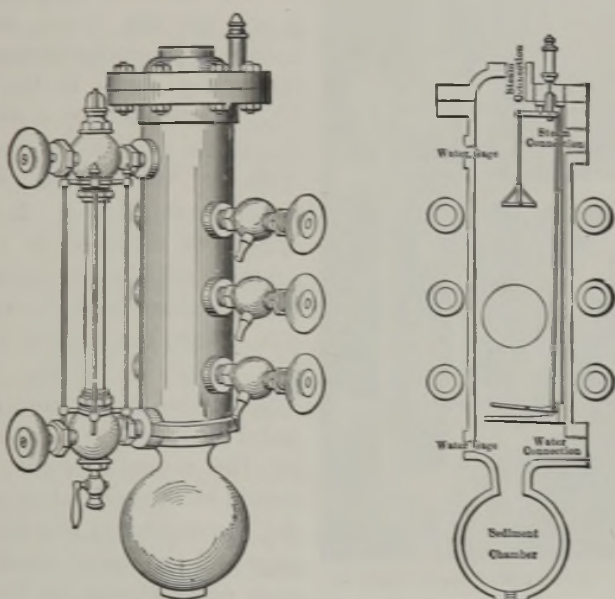


Fig. 51.—Double Bourdon-tube gage.

The ordinary form of pressure gage is the *Bourdon-tube* gage, shown in Figs. 50 and 51. The operating portion of the gage consists of a flattened bronze tube bent in a circle and closed at the end. One end is fixed (Fig. 50) or, as shown in Fig. 51, there are two such tubes. When fluid pressure is applied to the inside of the tube, its cross section tends to assume a circular form and the tube tends to straighten. The greater the pressure the more the straightening of the tube. By proper mechanism this change of form due to pressure is registered on a dial, which, when properly calibrated, shows the pressure in the boiler. The zero of the scale is always set at atmospheric pressure.

In another type of gage, not so commonly used, the steam pressure is exerted against a corrugated diaphragm instead of a tube. The movement of the diaphragm is transmitted to the hand in a manner similar to the one described for the Bourdon gage.



Elevation. Cross-section.

Fig. 52.—Water column.

Pressure gages should be placed at a convenient point for easy observation, and the piping should be as short as possible. *The gage should always be provided with a siphon containing water so that the hot steam cannot enter the gage.* (If hot steam enters the gage, it changes the calibration of the instrument.) It should also have a gage cock and union, so that it may be easily removed.

Figure 52 shows the elevation and the cross section of a *water column* with its gage glass. The section shows the float so arranged that it will blow a whistle when the water in the boiler is too high or too low. This is called a *high- and low-water alarm.*

The water gage and the water column to which it is attached are important accessories in boiler operation. The water column acts as a receiver to steady the agitation in the boiler for easier reading of the water gage, which in turn gives visual indication of the boiler water level. On the water column are placed try cocks, or gage cocks, used as a check upon the water gage since the water gage is sometimes clogged with dirt. The top of the water column should be connected with the steam space, so that it will get dry steam, and the bottom of the water column, with the water space in the boiler. The length of the water gage should be such as to cover the ordinary fluctuations of the water in the boiler. It should always be attached to a water column and the lowest point in the gage glass should be set about 3 in. above the highest point of the tubes in tubular boilers. The position of the gage glass in water tube boilers is determined by the manufacturers. There should be blowoff valves on both the water column and the water gage and the water columns and gage cocks should be blown off frequently. Figure 53 shows the ordinary arrangement of water column, water gage, and try cocks.

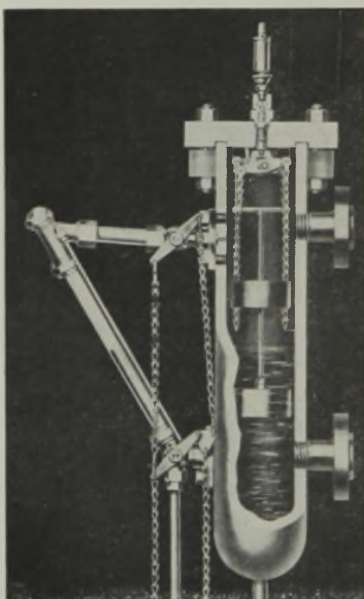


Fig. 53.—Yarway water column.

In the Yarway water column (Fig. 53) the alarm whistle, which calls attention to incorrect water levels, is actuated by the displacement of water by solid weights suspended from opposite ends of two levers, instead of by floats. When the water level is normal, the apparatus is in equilibrium and the whistle valve is closed. As the level becomes too high or too low, the equilibrium is changed, the valve at the top of the column is opened, and the whistle sounds the alarm.

Safety valves are constructed in many forms, but, in general, they consist of a valve opening outward and held in place by a spring; in the old forms, by a lever and an adjustable weight. Figure 54 shows the construction of the ordinary safety valve. The size of the safety valve may be determined in a variety of ways. A formula recommended by P. G. Darling in the A.S.M.E. *Proceedings* for 1909 is the following:

Let

E = pounds steam discharged per hour;

L = lift of valve vertically in inches;

D = valve diameter in inches.

Then, for flat-seated valves,

$$D = 0.0067 \times \frac{E}{L \times P};$$

and for valves with 45-deg seats

$$D = 0.0095 \times \frac{E}{L \times P}.$$

The average lift for a safety valve is about 0.1 in.

Some authorities allow in spring-loaded safety valves 1 sq in. of safety valve for every 3 sq ft of grate surface. More exact results may be obtained by reference to Mr. Darling's paper on this subject.

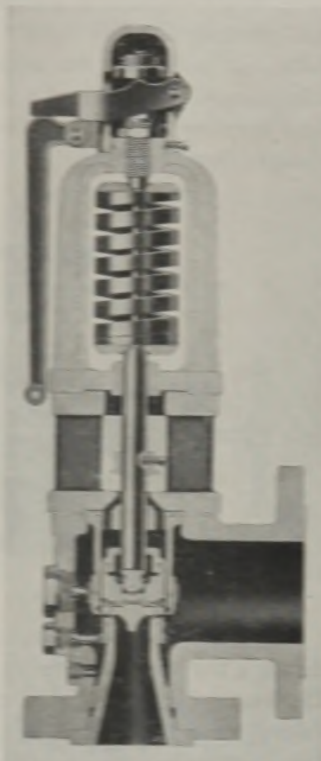


FIG. 54.—Safety valve.

Formerly, the lever safety valve was the type most used, but it was easily tampered with. At the present time the pop safety valve is almost universally used. Safety valves are adjusted so as to blow at one pressure, and seat at a pressure usually 2 lb less than that at which they open. The safety valve on the boiler should be tried at least once a day to see that it is in working condition.

The feed pipe to the boiler is always provided with a valve and check valve. In case of accident to the feed valve the check valve will close and prevent the water from leaving the boiler.

It sometimes happens that a boiler shell may become overheated, and a boiler explosion results. Such accidents are avoided by screwing into the boiler a plug consisting of a brass bushing filled with a metal that melts before any damage can be done to the boiler. These are called *fusible plugs* and are often used.

Figure 55 shows two views of a Yarway Unit-Tandem *blowoff valve,* which consists of a horizontal hard-seat valve and a vertical seatless valve. The latter consists of a hollow plunger with ports, or openings, on opposite sides. The plunger is raised and lowered by means of a hand wheel and nonrising screw stem, thus opening and closing the ports. Slotted guides prevent the plunger from turning.

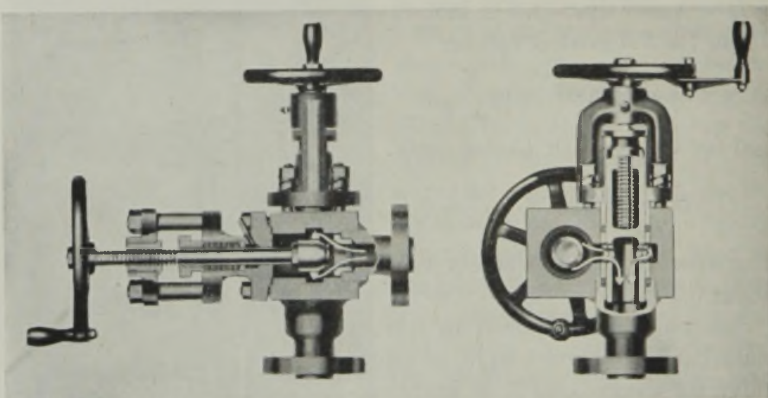


FIG. 55.—Yarway Unit-Tandem blowoff valve.

When the combination valve is open, the blowoff discharges through the hard-seat valve into and through the seatless valve. The vertical seatless valve, which functions as the outside seating valve, should be opened first and closed last; the horizontal hard-seat valve, which is installed nearest the boiler, should be opened last and closed first. Both valves should be opened rapidly and fully. Never blow down the boiler with valves partly open. A good blowoff valve should open easily and close tightly and should freely discharge water heavily laden with mud and sediment, without injuring the valve.

The Bailey Thermo-hydraulic Feed Water Regulator (Fig. 56) consists of a generator, metal bellows, regulator valve, and connecting piping. The generator consists of an outer tube A, having fins for cooling, and an inner tube B, the inner tube being connected to the boiler drum with one end above and one below the water level in the drum; hence the water level in the tube is the same as that in the drum. The annular space between tubes

A and *B* is connected with the metal bellows by the copper tubing, thus forming a closed system.

When the equipment is cold, the closed system is filled with water, but when steam from the boiler enters the upper portion of tube *B*, the surrounding water in the annular space flashes into steam. The pressure thus formed forces water out of this space

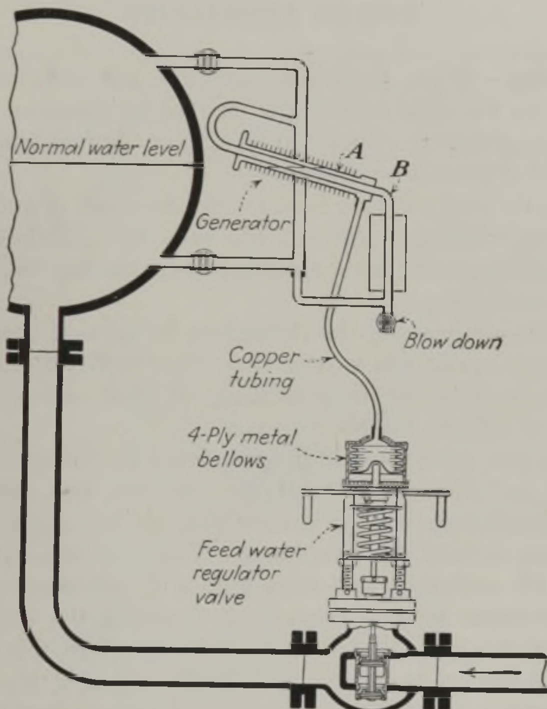


Fig. 56.—Diagrammatic installation of Bailey thermo-hydraulic feed-water regulator.

until the water level in it is the same as that in the inner tube (and hence the same as that in the boiler).

Water forced out of the annular space passes through the connecting tubing to the metal bellows, which expands and opens the regulating valve, thus admitting water to the boiler. As the level in the boiler rises, cold water from the water storage leg immediately above the blowdown line rises up into tube *B*. This cold water plus radiation from the fins on tube *A* condenses the steam in the annular space, reducing the pressure on the metal bellows and allowing the regulating valve to close.

CHAPTER VII

BOILER AUXILIARIES

93. Firing.—When firing a boiler, it is not sufficient simply to open the fire door and throw on coal at irregular intervals. In order to obtain economic combustion of the fuel, some system or method of firing is necessary.

Anthracite coal must be fired evenly, in small quantities, and at frequent intervals. After it has been fired, it should be let alone, the best results being obtained where the fire tools are used as infrequently as possible.

With bituminous coal the difficulties in burning economically and without smoke are greater than with anthracite, and these difficulties increase as the percentage of fixed carbon decreases and that of volatile matter increases.

The various methods used in hand firing are the *spreading*, the *alternate*, and the *coking* methods. In the first, the coal is fired in small quantities at a time and spread evenly over the entire grate surface in a thin layer. This system is satisfactory for use with anthracite and some grades of bituminous coals.

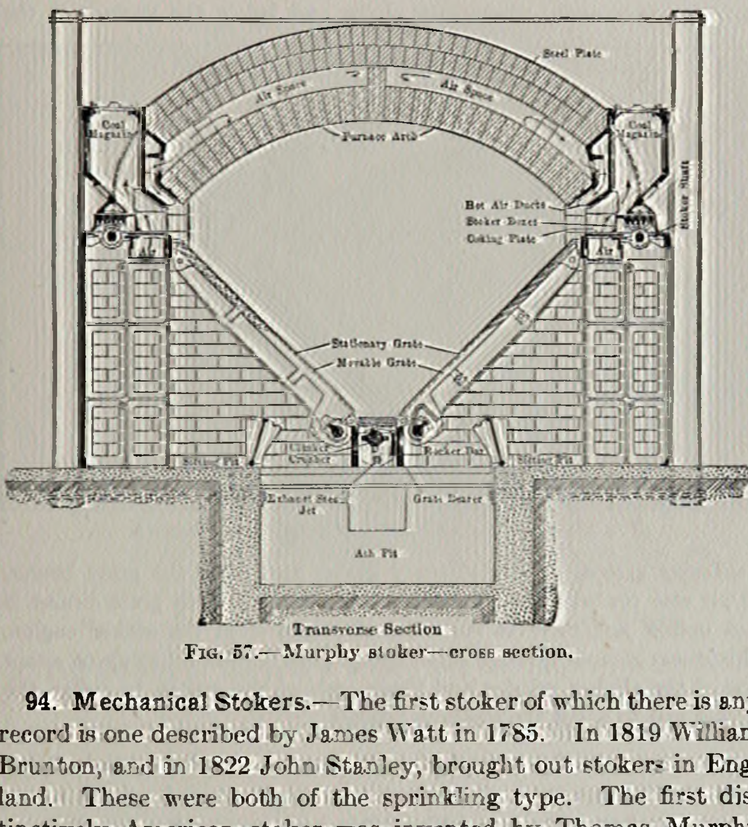
The alternate method consists in throwing the coal first on one side of the grate and then on the other. In this way, but one-half of the fire is covered with green coal at a time, and the volatile gases coming off from the fresh charge are burned as they pass over the incandescent bed of fuel on the other half of the grate.

In the coking method the fresh fuel is thrown on the front part of the grate. After the volatile matter has been driven off, the coked remainder is broken up and pushed back to the hotter part of the fire and another charge of fuel added at the front. This system is used for highly volatile and smoky coals.

Modern practice tends to working bituminous as well as anthracite coal as little as possible after it has once been fired.

In firing a boiler the best results are obtained by firing the coal in small quantities, or by progressive burning of the coal.

With any of the hand-firing methods, these results are difficult to obtain. Most firemen prefer to shovel the coal into the furnace in relatively large amounts and then rest. It is difficult to get them to give the proper attention to the handling of their fires. With mechanical stokers it is possible to introduce small quantities of coal frequently, or so to arrange the stoker that there may be progressive burning of the coal.



Transverse Section

FIG. 57.—Murphy stoker—cross section.

94. Mechanical Stokers.—The first stoker of which there is any record is one described by James Watt in 1785. In 1819 William Brunton, and in 1822 John Stanley, brought out stokers in England. These were both of the sprinkling type. The first distinctively American stoker was invented by Thomas Murphy of Detroit, Mich., in 1878.

Stokers may be divided into four principal classes: the *overfeed*, the *traveling grate*, the *underfeed*, and the *spreader* type.

95. Overfeed Stokers.—The Murphy stoker, shown in Figs. 57 and 58, is an example of the overfeed stoker, known as the *side-feed, double-inclined*, or V type.

At either side of the furnace, extending from front to rear, is a coal magazine, into which the coal may be introduced either mechanically from conveyors, or by hand. At the bottom of this magazine is the coking plate, against which the inclined grates rest at their upper ends. The stoker boxes, operated by segment gear shaft and racks, push the coal out over the coking plate and onto the grates.

The grates are made in pairs—one fixed, the other movable. The movable grates, pinioned at their upper ends, are moved by a rocker bar at their lower ends, alternately above and below the surface of the

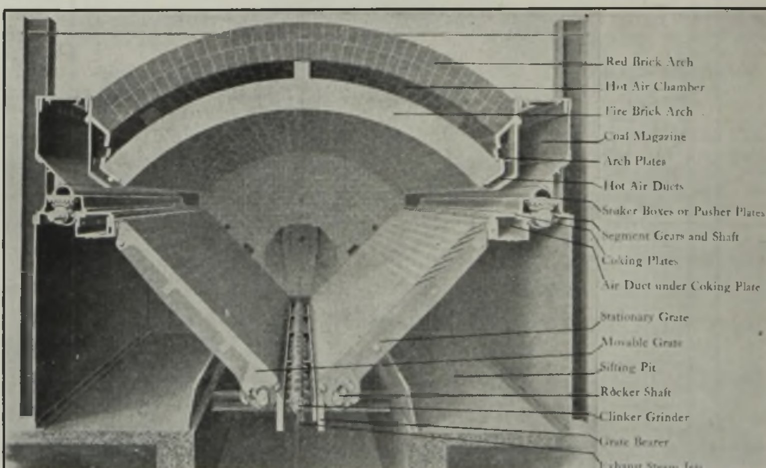


FIG. 58.—Murphy stoker—view from rear of furnace.

stationary grates. The stationary grates rest upon the grate bearer, which also contains the clinker or ash grinder. This grate bearer is cast hollow and receives the exhaust steam from the stoker engine. This steam escapes through small openings at regular intervals on either side of the clinker grinder and lower ends of the grates, to soften the clinker and to assist the cleaning process.

The grates in the Murphy stoker are set at an angle of 45 deg with the horizontal. Grates at this angle are adaptable to all grades of bituminous and semibituminous coal, but are not suited to the use of lignite or anthracite. The Murphy stoker is particularly adapted to small or medium-sized boilers carrying a constant load.

Either the flush front, or the Dutch oven or extension setting, (see Fig. 49) may be used with the side-feed, double-inclined stoker. The latter is more generally used and is commonly regarded as the most effective, both as to efficiency and as to the elimination of smoke.

Another form of overfeed stoker is the *front-feed* or *single-inclined* type. The Huber (Fig. 59) is an example. In this stoker, which is used with natural draft only, the coal is fed into a hopper in front of the boiler and is pushed upon an inclined grate from the front.

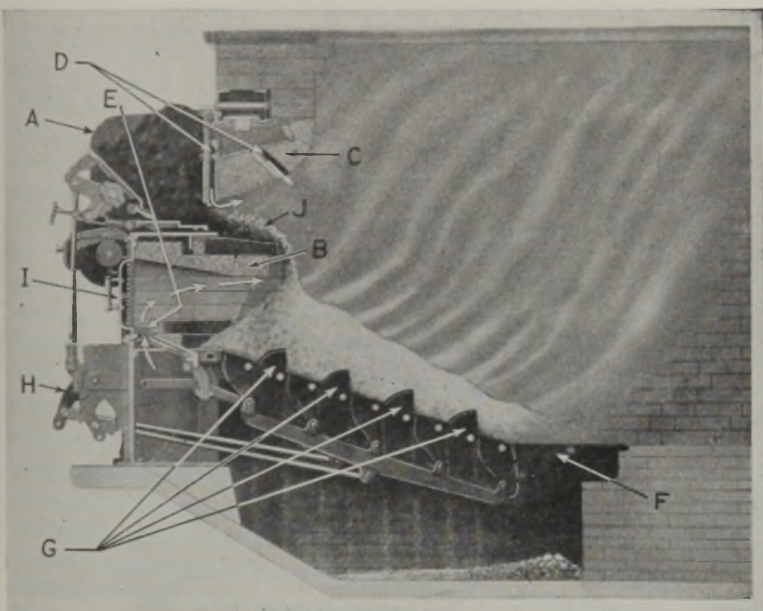


Fig. 59.—Huber full automatic, overfeed, natural-draft stoker.

"The feeding of the coal is a slow and continuous process. As it leaves the hoppers *A*, it passes over a coking shelf *B*, which extends the full width of the furnace and likewise passes directly underneath the ignition arch *C*. In doing so, the coal is subjected to the radiant heat of the furnace, becomes ignited and partially coked. This releases the light volatile matter which, when mixed with the secondary air supplied for this purpose through the air ducts *D* in the hopper back, is consumed in the hot furnace.

"As this ignited, partially coked fuel drops onto the stoker directly underneath the coking shelf *B*, it is subjected to an even more intense heat and the coking process is completed.

The heavier volatile matter is released and consumed in the furnace as it becomes mixed with a further supply of secondary air introduced through the dead plate shutters *E*, below.

"This fuel, which is now coke or fixed carbon, is subjected to the full heat of the furnace and, by the action of the alternate-movement stoker bars *G*, is gradually moved back toward the rear of the furnace.

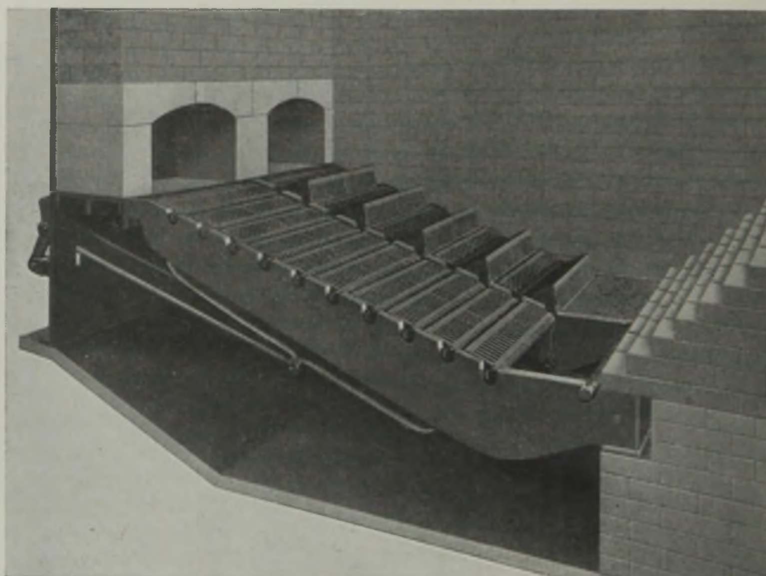


Fig. 60.—Side view of Huber hand-fired stoker.

"The stoker bar action also keeps the fuel bed continually broken up, shaking out the finer ash and finally depositing the clinkers on the drop bars *H* at the extreme rear of the furnace, where they can be easily dropped into the ashpit at any intervals necessary."

Figure 60 shows a side view of the Huber Hand Stoker in its setting. With this stoker the operation is manual throughout, the coal is fired through the doors, and the stoker bars are operated by hand.

The overfeed stoker has given excellent satisfaction, particularly in using diversified coals. In conditions of excessive loads this is probably not so smokeless as some other forms of stoker, but when carefully operated it is one of the most satisfactory forms, particularly for small, or medium-sized, boilers.

96. Traveling Grates.—As stated in a previous paragraph, the best results with a furnace can be obtained where there is progressive combustion of the fuel. By this is meant a continuous movement of the fuel through the surface, the rate and manner of feeding this fuel and the control of the air supply being so arranged as to provide a definite time and place for the burning of each constituent of the fuel.

Traveling grates operate on this principle. The moisture is driven off as the coal leaves the hopper and falls on the grate.

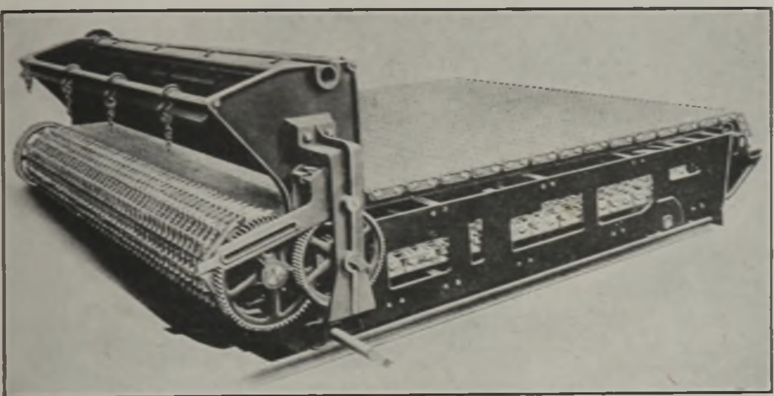


Fig. 61.—Green natural-draft chain grate.

Next, the volatile matter is driven off and burned, and, finally, the fixed carbon is consumed. The residue, or ash, drops off the back of the grate into the ashpit. This form of grate gives excellent satisfaction with noncoking coals and uniform loads on the boiler, and will be almost smokeless under proper conditions of operation. It is not generally adapted to the use of semibituminous, or anthracite, coal. The greatest difficulty is improper installation, which will permit of the passing of an excess of air through the grate. This, however, may be avoided by careful setting. In installing these grates, provision should be made for the easy removal of the ashes.

Figure 61 shows a Green natural-draft chain grate. The coal is fed into a hopper, the bottom of which is open to the chain grate, composed of a series of flexible links rotating upon two cylinders, one at each end of the grate. The grate is driven by a small engine, the speed of which can be adjusted to the particular form of fuel burned and the load on the boiler. This speed should be regulated so that the fuel is completely burned just as it reaches the back of the grate. If the speed is too fast, unburned coal will be carried over the back of the grate into the

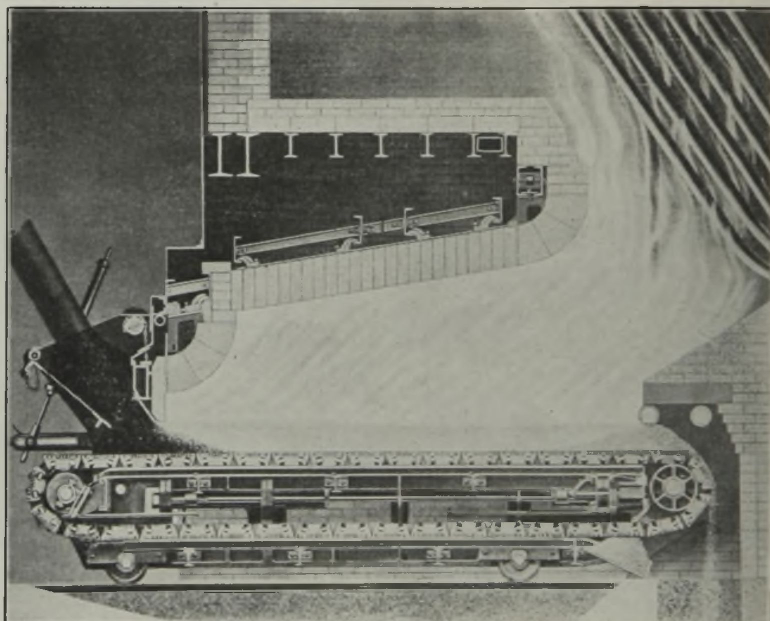


FIG. 62.—Section through Green natural-draft stoker under actual operating conditions.

ashpit; if too slow, there will be holes in the fire toward the rear, allowing an excess of air to pass through the grate. The coal drops from the hopper upon this slowly moving grate, the thickness of the bed of coal being adjusted by an apron at the front of the boiler. Figure 62 shows a cross section of this stoker installed under a boiler.

With the development of modern conditions calling for greater capacities from boilers, it became necessary to construct chain-grate stokers using forced instead of natural draft. At the present time practically all central stations using chain grates are installing the forced-draft type.

Whereas 200 per cent of boiler rating represents approximately the maximum capacity obtained with natural-draft chain grates, with forced-draft stokers of this type over 300 per cent of boiler rating has been obtained. The combustion per square foot of grate surface has reached 55 lb to 60 lb of coal per hr with the forced-draft chain grate.

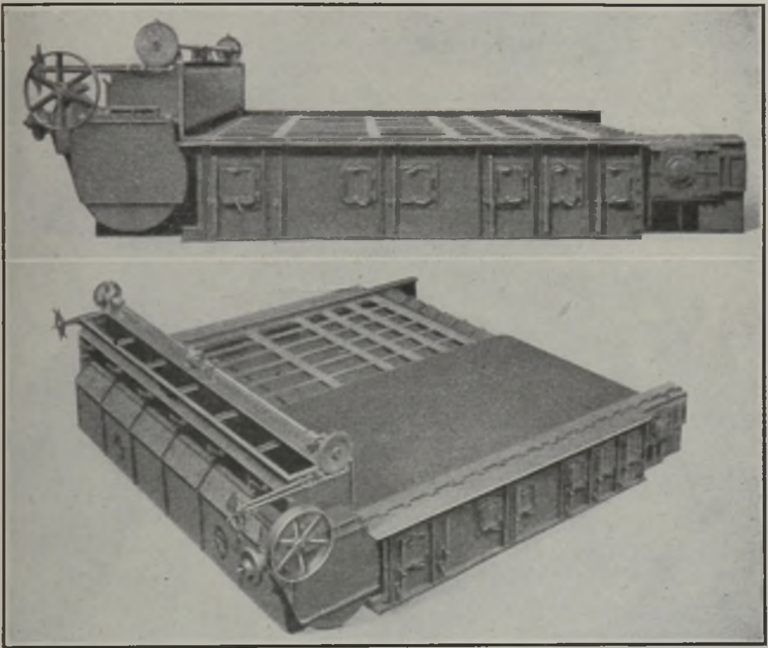


FIG. 63.—Side and top of Babcock and Wilcox blast chain-grate stoker.

Figure 63 shows a Babcock and Wilcox forced-draft, or blast chain-grate stoker. In this stoker

. . . the side frames consist of two side-wall blast boxes which are divided into five or six compartments depending upon the length of the stoker. These two blast boxes are connected by transverse I-beams which divide the space between the upper and lower runs of chain into compartments corresponding to those of the side boxes. All of the side-wall blast boxes are connected to a common blast duct, the air pressure in the different compartments being regulated by an adjustment of the individual compartment dampers. The air pressure, which rarely exceeds 2 in. in any compartment, is usually carried highest in the second and third compartments from the front, over which ignition has been completed and the maximum rate of combustion takes place.

Although the original chain-grate stokers were built for use with natural draft and to burn bituminous coal, a forced-draft stoker of this type was designed and built by E. B. Coxe in 1893 to

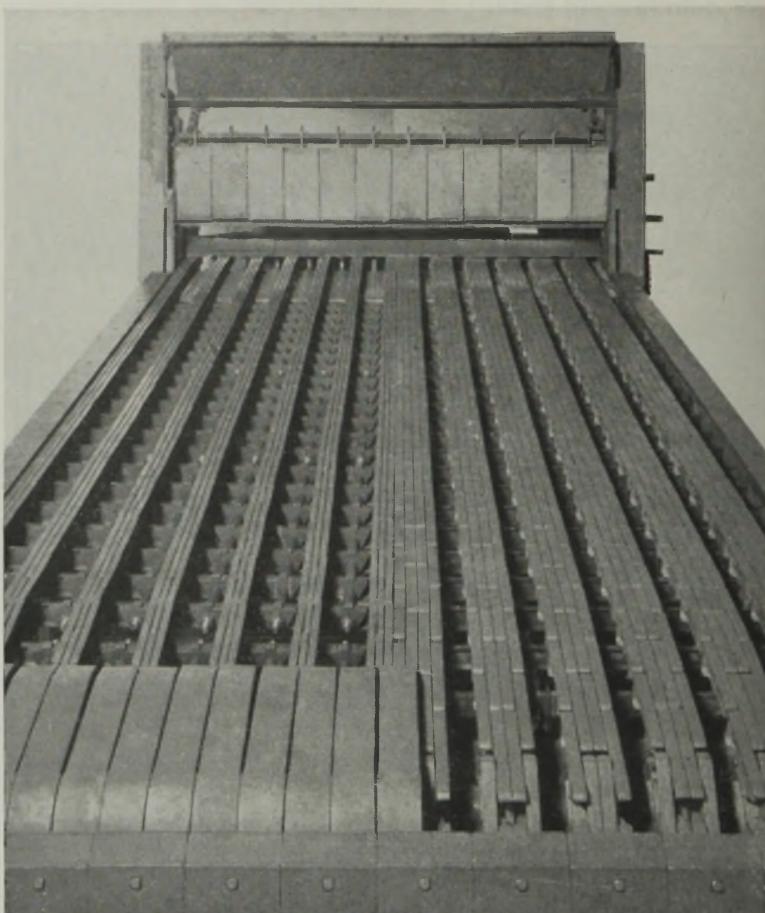


FIG. 64.—Stowe Stoker showing (at left) the stationary grate sections and skid plates and (at right) the traveling chains.

burn anthracite coal. Later it was modified so as to handle coke breeze. The first successful installation of a forced-draft chain grate for burning bituminous coal was not made until 1918 or 1919.

Another form of traveling-grate stoker recently developed is the Stowe. In this the grates (Fig. 64) "are constructed from cast-iron segments assembled into stationary sections that extend from the feed gate to the rear end assembly—alternating across the width of the stoker with continuous traveling chains." "Tuyères, or air passages, are maintained around the entire circumference of each segment," thus forcing the air upward from beneath the fuel bed.

"Incoming fuel is carried into the furnace by the traveling chains and struck off evenly by the feed gate," thereby securing

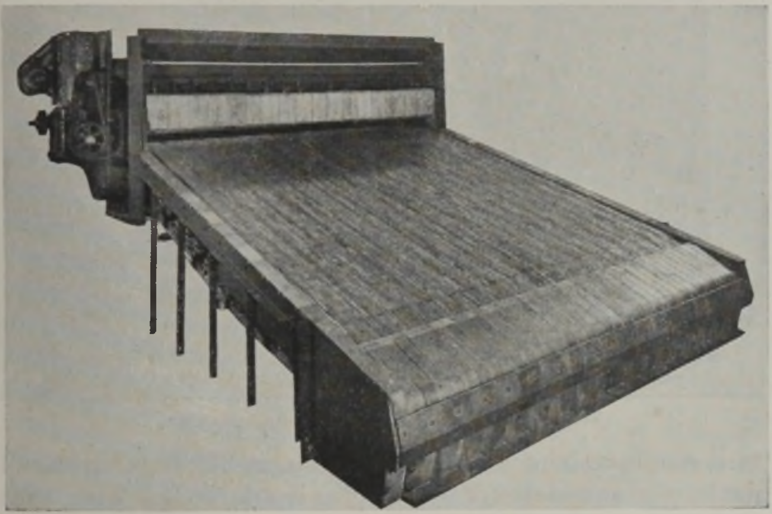


FIG. 65.—Stowe Stoker fully assembled.

an even fuel bed to start with. The stationary grates retard the movement of the burning coal and so keep the fuel bed tightly compressed, thus avoiding cracks or fissures in it. The reduced volume of fuel resulting from the burning of the combustible material is compensated for by this retarding and compressing action since "the fuel bed moves more slowly across the grates instead of becoming thinner and thinner." This result is accentuated by the cover plates at the rear end of the stoker, "as it requires considerable pressure on the part of the fuel bed to push the ashes across these plates and into the ash hopper." Figure 65 shows another view of the Stowe Stoker grates.

97. Underfeed Stokers.—Underfeed stokers are of two general types: *single retort* and *multiple retort.* In both, the principle of operation is to force the green coal in beneath the burning fuel. As the coal leaves the retort, it begins to coke, the volatile matter being driven off and passing up through the incandescent bed on top. By the time the coal reaches the top of the fire it is completely coked and the surface of the fire consists of this layer of incandescent coke. The ash accumulates on the surface of the fire and from there is gradually floated onto the dead plates, or dump plates. Since the ash is raised to a high temperature, coal containing ash that is high in sulphur and iron should not be used in a stoker of this type, as it will produce large, hard clinkers.

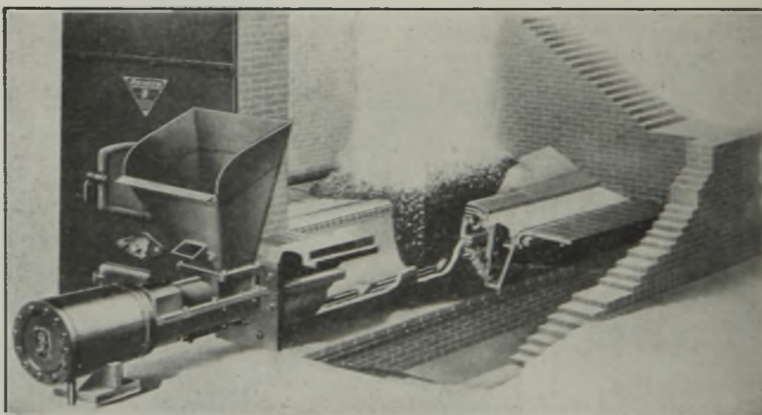


FIG. 66.—Jones "standard" side-dump stoker.

The underfeed stoker produces a very intense heat directly above the fire, thus making possible practically complete combustion of the volatile matter and the most smokeless operation of all types of stoker.

Underfeed stokers cannot be successfully operated with natural draft, the resistance of the fuel bed to the passage of air being so great that it is necessary to use a blower to force the air through this bed. This blower is usually driven by a steam engine, and the excessive amount of steam used is one of the objections offered to their use.

One of the earliest and simplest forms of underfeed stokers is the Jones, shown in Fig. 66. This is one of the single-retort type; in it coal is dropped down from hoppers in front of a piston at regular intervals, depending upon the load. This piston moves forward, forcing the coal into the retort. The wedge-shaped pushers, which are reciprocated horizontally, carry the coal on in the retort under the burning fuel. In this way coal is always introduced under the fire, and all the gases are passed through the incandescent fuel. Air is forced into the sealed ashpit by a blower, and from the ashpit passes into the furnace through the tuyère openings. These openings are at a point

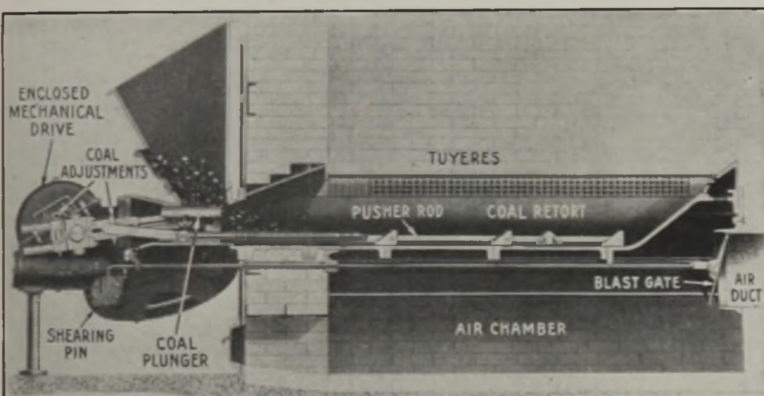


FIG. 67.—Longitudinal section through Detroit single-retort stoker.

above the green coal in the retort but below the fire. The ash forms on top of the fire and falls over onto the dead plates at the sides, dropping the ashes into a shallow ashpit from which they can be easily removed.

The Jones stokers are especially suitable for use in smaller installations.

The Detroit single-retort stoker is shown in longitudinal section in Fig. 67; Fig. 68 shows a furnace view of the same. A plunger forces the coal into the retort from which it overflows onto the tuyères, grates, and ash dump (see Fig. 68). The fuel is first coked and then completely burned, the refuse passing to the dumping sections. A sealed air chamber extends beneath the fuel bed for its entire length and air passes up through every part of the bed of fuel including that resting on the dumping sections. In this way the combustible is all burned before it is dumped.

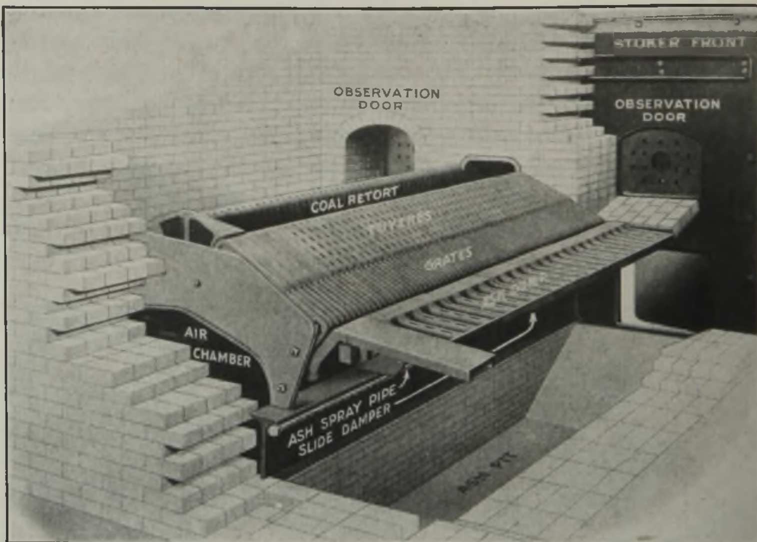


Fig. 68.—Furnace view of Detroit stoker.

Fig. 69.—Front view of Detroit unistoker.

This stoker is made in a multiple-retort type also.

The Detroit unistoker (Figs. 69, 70 and 71) combines the motor, blower, and fuel-feeding mechanism into one unit mounted on

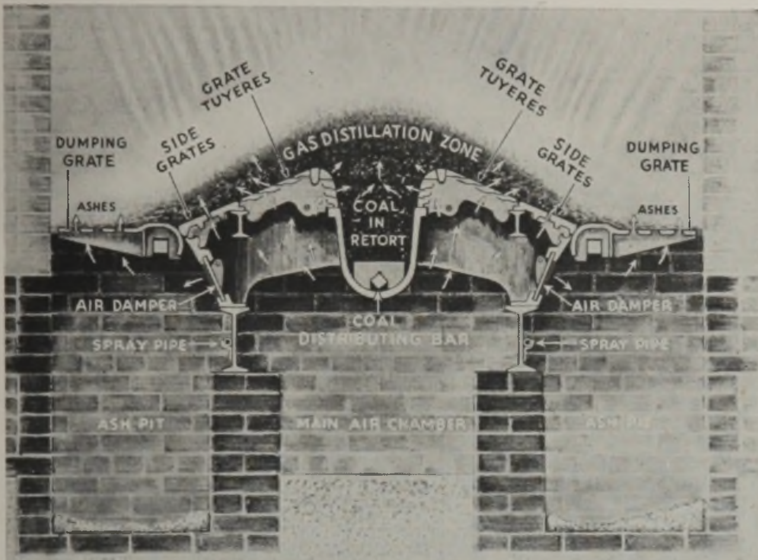


Fig. 70.—Front sectional elevation of Detroit unistoker.

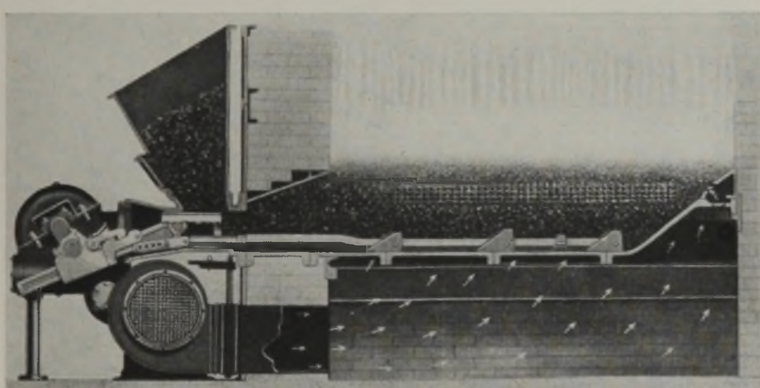


Fig. 71.—Longitudinal section through Detroit unistoker.

the front of the stoker. The blower discharges directly into the air chamber, eliminating long runs of piping and air ducts.

The Type E stoker built by the Combustion Engineering Corporation is constructed in both the single- and the multiple-

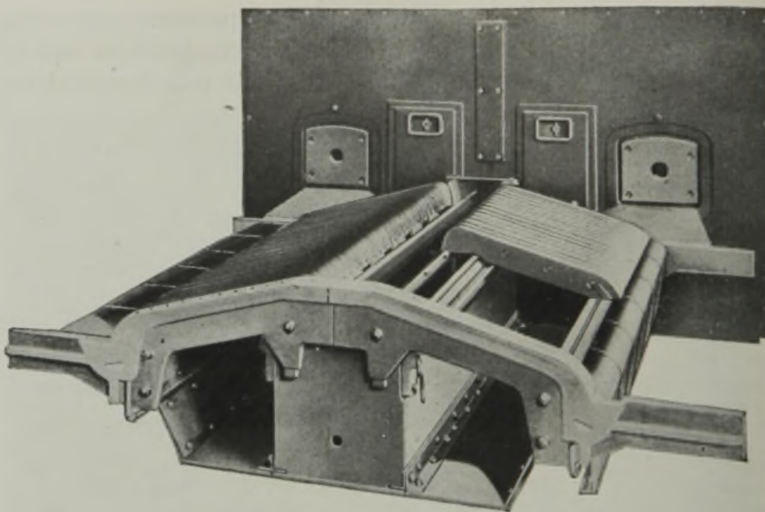


FIG. 72.—Rear view of Type E underfeed stoker.

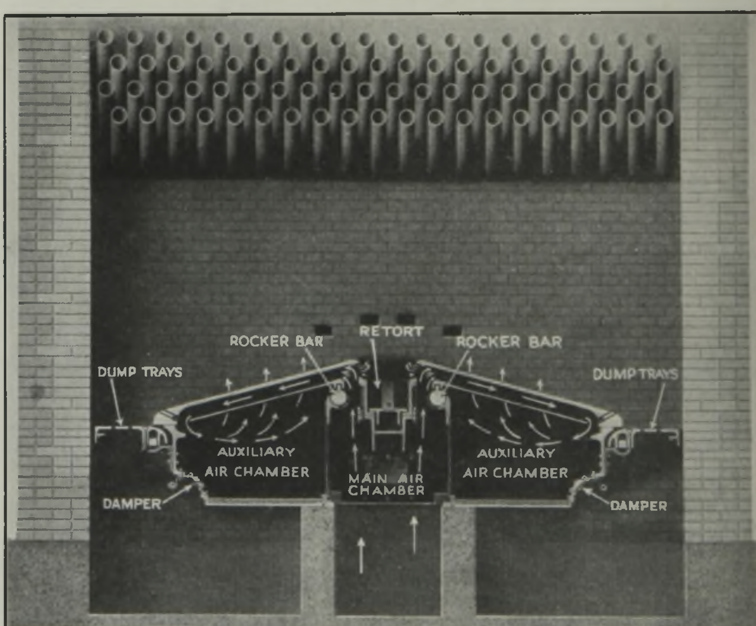


FIG. 73.—Cross section of Type E stoker.

retort types. Figure 72 shows a rear view of a single-retort stoker of this make and Fig. 73 a cross section of the same with arrows indicating the distribution of air under the grate and through the grate bars.

In the longitudinal section of this stoker (Fig. 74), the method of conveying the fuel from the front to the rear of the fire is shown. The coal is fed from the hopper onto the sliding bottom

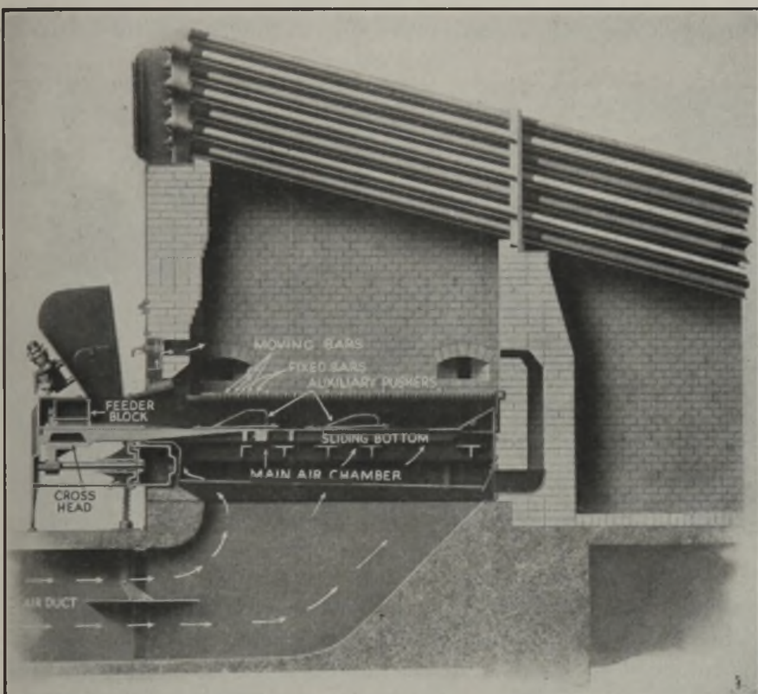


Fig. 74.—Type E stoker—longitudinal section showing coal-feeding mechanism and path of entering air.

of the retort. This bottom extends the full length of the grate and has a reciprocating motion imparted to it by the piston in the steam cylinder of the stoker. The coal is carried from front to rear by the wedge-shaped pushers attached to the moving bottom of the retort and, as it rises to the top of the fire, is distributed to the sides of the furnace by the moving grate bars. These bars are arranged in pairs, alternately moving and fixed, the moving ones having a horizontal motion transverse to the retort.

This motion is from $\frac{1}{2}$ in. to 1 in. in extent. Besides acting to distribute the coal over the grate, the movement of these bars conveys the ash and clinkers to the dump plates at the sides.

The Taylor (Figs. 75 and 76) is a multiple-retort inclined underfeed stoker operating on the gravity principle. This stoker is made up of a series of sloping troughs, or retorts, which extend from the front furnace wall to a point within 3 ft to 6 ft of the rear furnace wall, depending upon the particular installation. The retorts have their maximum depth at the front of the furnace,

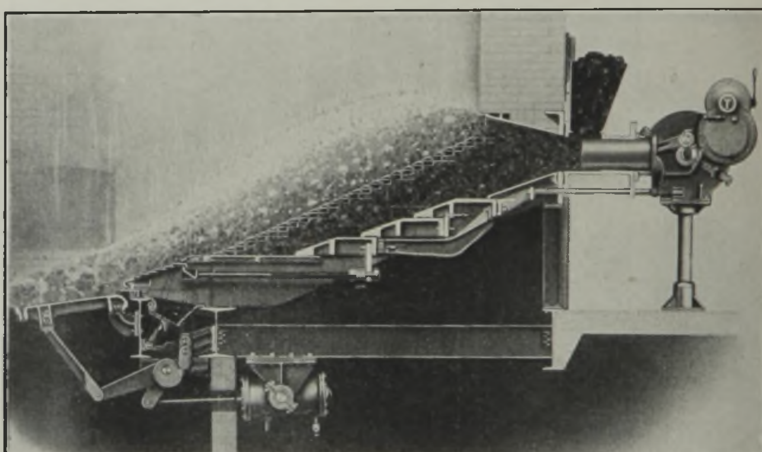


Fig. 75.—Section through fire in Taylor stoker.

gradually tapering to none at the rear. Each retort is equipped with a reciprocating coal feeding ram located in the bottom of the retort at the front wall end, and a series of auxiliary reciprocating pushers separated by stationary plates at the bottom of the retort within the furnace.

The entire width of the furnace just rearward from the lower end of the retorts and tuyère rows is equipped with an overfeed section. This section terminates at the ash-eliminating portion of the stoker which may be a dumping plate, a continuous discharging mechanism, or an ashpit equipped with clinker grinder rolls. The sides of the retorts are formed by hollow boxes with covers of perforated iron blocks called *tuyères*. A blower forces air for combustion into the boxes and out through the tuyères into the fuel bed.

As the coal drops from the hopper, the reciprocating rams force it into the upper or front end of the retorts, crowding downward and outward some of the coal previously introduced. The green coal is then transported rearward by the auxiliary distributing pushers in the bottom of the retort, aided by gravity. The movement of the distributing pushers keeps the fuel bed open, porous, and free from clinker. Since the depth of the retort gradually decreases owing to its tapered construction, each movement of the coal mass toward the rear exposes a new top layer of coal to the combustion process.

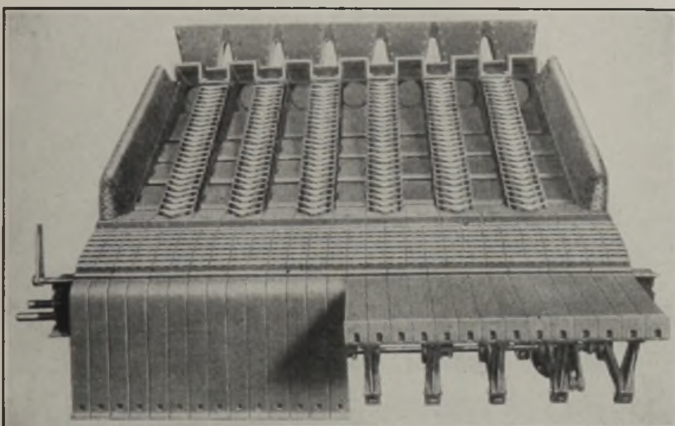


FIG. 76.—Rear view of Taylor stoker.

The fuel bed over the tuyères is from 2 ft to 4 ft thick. As the green coal is introduced to the fire from beneath, it works up through the fire and downward toward the extension grates. During this process the volatile gases are driven off and the coal is slowly coked. The ignition of the fuel does not take place until it has become coke and passed beyond the tuyère lines. Although the underfeed retorts distill the volatile matter in the coal, they cannot finish the process of combustion except at very low ratings. Final combustion of the coke takes place on the extension grates, so that but little hot ash reaches the dump plates. A damper is provided for admitting air beneath the fire on the extension grates, if necessary, in order to burn any combustible remaining in the ash before dumping.

The dumping mechanism may be operated by power, as is the case in the stokers shown in Figs. 75 and 76, or by hand. These plates are dumped periodically as the conditions of service may require.

Figure 77 is a view of a Taylor stoker showing the construction within the furnace, from the ash plate toward the furnace face of the side and front walls. The cutaway section shows coal-distributing pushers within the retort, showing variation in retort depth from the front wall where green coal is introduced, to the overfeed section or end of the tuyère section where the final

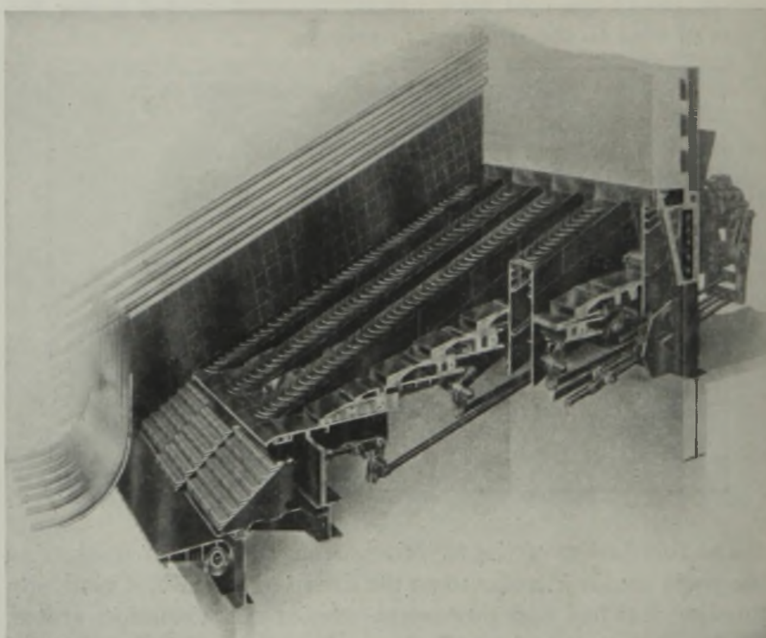


Fig. 77.—Side view of Taylor stoker.

combustion of the fuel not completely consumed is prepared for discharge from the furnace. The pushers and stepped overfeed section reciprocate and the movement can be increased or decreased to suit the kind of coal being burned to provide the proper fuel bed. The rear water-cooled wall and inclined side waterwalls with cast-iron blocks on the lower section are shown.

Figure 78 is a cross section of a bent-tube boiler showing the upper front drum and lower drum with water-cooled furnace walls and water-cooled Taylor stoker. Water-cooled tubes are extended from a header below the ash plate of the stoker along the tuyère line and lower retort section up the front wall to a header in the front wall and along the roof of the furnace to the front boiler drum forming a completely water-cooled furnace. The water supply to these cooling tubes is by connection from

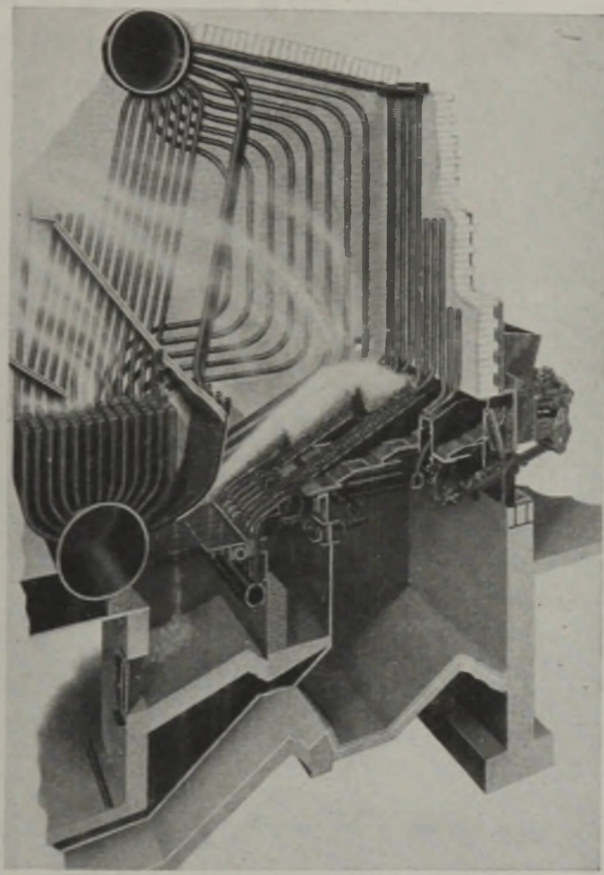


FIG. 78.—Cross section of bent-tube boiler and water-cooled Taylor stoker.

the lower boiler drum. This water cooling of the stoker results in low maintenance, high reliability, continuity of service, and ability to burn low-grade, low-fusing ash coals with good results.

Figure 79 is a shop view of a water-cooled Taylor stoker showing water-cooling tubes along the tuyère line and at the lower portion of the retort section. The view is from the ash plate looking toward the main coal-feeding rams and coal-distributing pushers in the bottom of the retort.

Figure 80 is a view taken from the ashpit of a Taylor stoker showing stoker extension grates and stoker ashpit wall on the right, the stoker ash grinder rolls in the bottom of the pit with

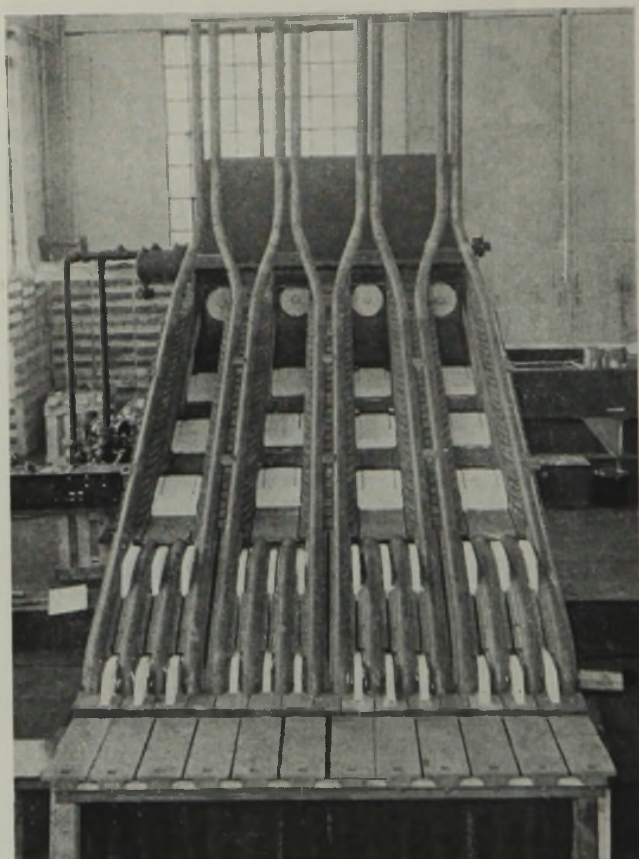


FIG. 79.—View of water-cooled Taylor stoker showing water-cooling tubes.

grinder teeth, and the rear water-cooled wall on the left. The lower portion of the rear wall tubes is armored with cast-iron blocks with bare tubes above. The side wall at the end of the pit is partly brick and partly inclined waterwalls. The lower portion of the side waterwalls is armored with cast-iron blocks along the fuel line and bare tubes above set to form a slag drip.

98. Spreader Stokers.—The *spreader stoker* consists of coal-feeding units placed above the grate line, on the front wall of the furnace. These units consist of a variable feeding device, that conveys the coal into a revolving rotor with blades mounted on it.

Fig. 80.—View of lower end of Taylor stoker and of ashpit.

These revolving blades strike the incoming fuel and throw and sprinkle it uniformly over the entire surface of a substantially horizontal forced-draft grate.

The dust of the coal is burned in suspension, and the coarse coal is burned on the grate. The latter is of pinhole design, made to give uniform air distribution and to admit the air at high velocity to create turbulence and rapid burning. The fresh

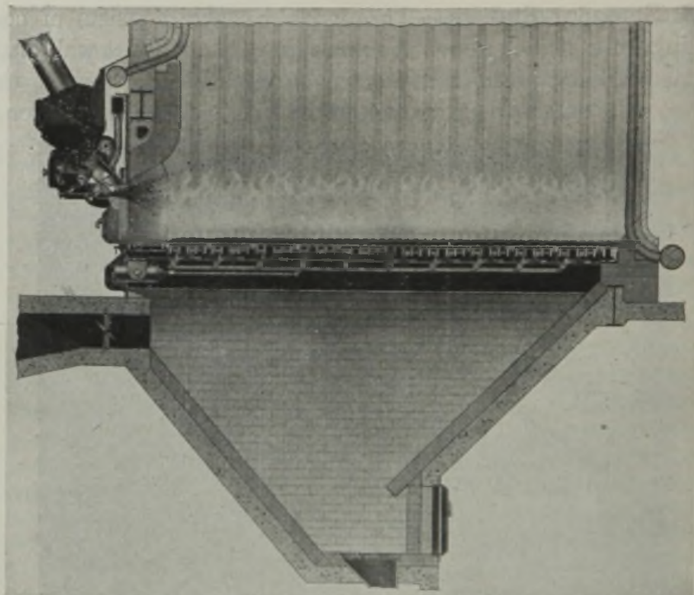


FIG. 81a.—Hoffman Firite dumping grate stoker—longitudinal section.

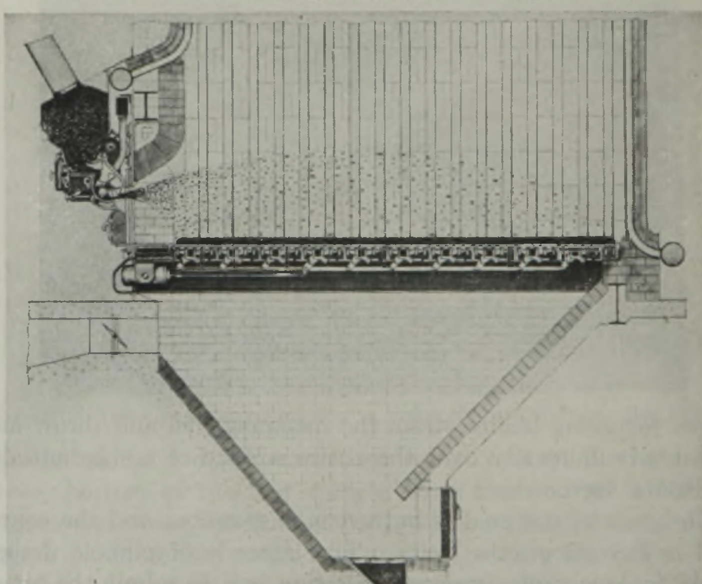


FIG. 81b.—Hoffman Firite dumping grate stoker-line drawing.

coal is thrown on top of the fuel bed. Ignition is instantaneous, and the fuel is burned practically as fast as it is thrown into the furnace.

Figures 81*a* and 81*b* show a cross section of a Hoffman Firite Stoker. Directly beneath the coal hopper is the feed plate, which has a reciprocating motion that is imparted to it by means of a cam

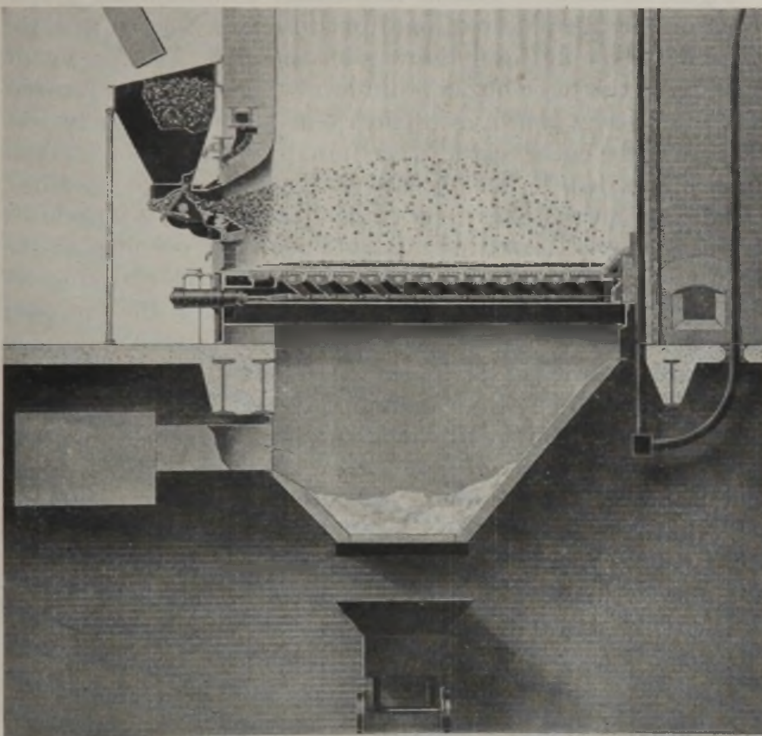


FIG. 82.—Detroit RotoStoker—power-operated dumping grate type.

arrangement driven through enclosed steel cut gears. The coal that is on the feed plate is driven over the edge of the plate by the reciprocating motion and is deposited on the curved adjustable tray or throat piece, whence it is picked up by the revolving distributor blades of special design and spread evenly and continuously and lightly out over the entire grate area.

Extending beneath the fuel bed is the blast chamber. Air from the blower enters the chamber and, as required for combustion, is forced up through the steel tubes that extend downward from the tuyère plates. As the air passes through these tubes, it is preheated and thence delivered through the fuel bed via an infinite number of angularly disposed openings on the face of the tuyère plates, causing crisscrossing of air jets and promoting an even distribution of air over the entire grate area.

Figure 82 shows a Detroit RotoStoker with dumping grates. Uniform fuel distribution from front to rear is secured with the RotoStoker by an "over throw rotor action." The fuel hit by the tips of the rotor blades is thrown farthest into the furnace. The rotor blades are curved to distribute the fuel uniformly sideways over the entire width of the furnace. The air for combustion is supplied to the furnace through the grate openings. Overfire air in the proper ratio to the undergrate air is introduced through tuyères located just beneath the rotor openings at the front. The grate surface is built in sections, each served by an individual rotor or feeder. When cleaning fires, the fuel feed and air supply to one section of the grate are temporarily discontinued. Although the fire on that section is allowed to burn down, the steam pressure is uniformly maintained by continuing the operation of the adjoining sections. The power dumping type of grate is operated by either steam or compressed-air cylinders located at the front. The grates tilt and discharge the ash into the pit below.

This stoker is also built with a hand-operated dumping grate and with a stationary grate. It has no moving parts in the furnace and may be fired by hand continuously and, in an emergency, may be operated with natural draft.

99. Grate Surface in Stokers.—The grate surface in a stoker with an inclined grate is taken as the area of the horizontal projection of the grates, and is termed *projected area.* The ratio of projected grate area in the stoker to the heating surface in the boiler varies from 1:55 to 1:65.

100. Advantages and Disadvantages.—The advantages in the use of the mechanical stoker over hand firing are, in the order of their importance, as follows: (1) saving in fuel cost, due primarily to a better control of the supply of air, but also to the more uniform and continuous supply of fuel, these conditions bringing about a higher furnace efficiency and making possible the use of poorer and cheaper grades of coal; (2) in larger plants, a material saving in the cost of labor; (3) increased output of boiler; (4) ability to meet rapid fluctuations in the load on the boiler; (5) more nearly smokeless operation of the furnace.

The disadvantages are: (1) high initial cost; (2) large maintenance cost; (3) cost of operation—most stokers using from $\frac{1}{2}$ to 3 per cent of the steam generated, and those of the blast type requiring from 3 to 5 per cent of the steam generated.

In small plants where coal- and ash-handling machinery is not provided, stokers will not reduce the labor charge; even in these cases, the growing need for conservation of fuel, the difficulty of obtaining grades of coal satisfactory for hand firing, and the increasing stringency in smoke regulation make necessary a serious consideration of the adoption of stokers.

With the larger plants, there is no room for questioning the advisability of a stoker, the only point to be considered being the particular type that will best fit the conditions. One man can handle about 500 boiler hp when firing by hand; with stoker firing only one attendant is required for every 5,000 to 10,000 boiler hp.

Modern stokers are operated at rates of firing which liberate from 40,000 to 50,000 Btu per hr per cu ft of furnace volume. "Turbulence" within and above the fuel bed occurs with both the chain-grate and underfeed stokers when forced draft is used. Among the serious problems affecting capacity limitation are the slagging of the boiler tubes by the particles of coke and molten ash which are thrown up from the fuel bed, and the erosion of the brickwork due to the flowing slag.

101. Powdered-coal Equipment.—Since 1920 the increase in the use of powdered coal as a fuel in boiler plants has been very marked, and the decision as to whether to use powdered coal or stokers in a given plant which is to be mechanically fired should not be made without careful consideration of all the factors involved.

There are two systems used to prepare and burn pulverized (or powdered) coal: (1) the *bin-and-feeder* system and (2) the *unit* system. The former was the one first adopted, and the two systems give equally good operating results, but recently the unit, or direct-firing, system has been gaining in popularity on account of its simplicity and lower first cost and cost of operation and maintenance.

In the *bin-and-feeder*, or storage, system the fuel is pulverized in a central plant and stored in bins. From there it is delivered to the boilers through ducts by a screw conveyor, or air blast. In this system the typical equipment consists of a drier for removing moisture from the coal, a mill for grinding or pulverizing the fuel to the required fineness, a screw conveyor, or blower system, for transporting it from the mill to the boiler room, a blower for feeding the fuel to the burners, and, finally, the burners themselves. Some coals are free enough from moisture so that they can be pulverized and fed to the boiler without drying, in which cases the drier can be omitted from the equipment needed. In most of the newer plants separate driers have been abandoned in favor of drying by the use of preheated air in the pulverizer.

"When the bin-and-feeder system was used for burning practically all pulverized coal, the specifications for a pulverizer merely required that it have a capacity of approximately so many tons of dried coal ground to a reasonable fineness with the least power consumption and maintenance.

"Now the trend of pulverized-coal firing is almost completely toward direct firing with turbulent burners and water-cooled furnaces. With this system, the pulverizer must do infinitely more than pulverize coal—it must handle coal of any moisture content and with any reasonable variation in size and grindability, and not only pulverize it to a higher degree of fineness than heretofore required, but also maintain this fineness over wide ranges of output during the life of the grinding elements of the pulverizer. The pulverizer must also maintain the proper ratio of primary air to fuel over a wide range of load requirements, and must respond instantly to hand control or to automatic combustion control."

In the *unit* system a complete pulverizing equipment is furnished for each boiler, but this equipment is reduced to a minimum. The coal is used as soon as it is pulverized, thus avoiding any necessity for storage; the transportation of the powdered fuel is simplified as the mills are located near the boilers, and the use of preheated air through the pulverizers eliminates the necessity for separate driers. The unit system permits of the use of coal with a higher moisture content, as there is not the danger of clogging bins and supply pipes which is present when the central storage system is used.

In both systems the coal is first crushed to a regulation size of lumps which will pass through a 1-in. ring. It is then passed over a magnetic separator to take out any "tramp" iron. From here it goes to the pulverizing mills, and then either to a storage bin in the bin-and-feeder system, or directly to the boiler in the unit system.

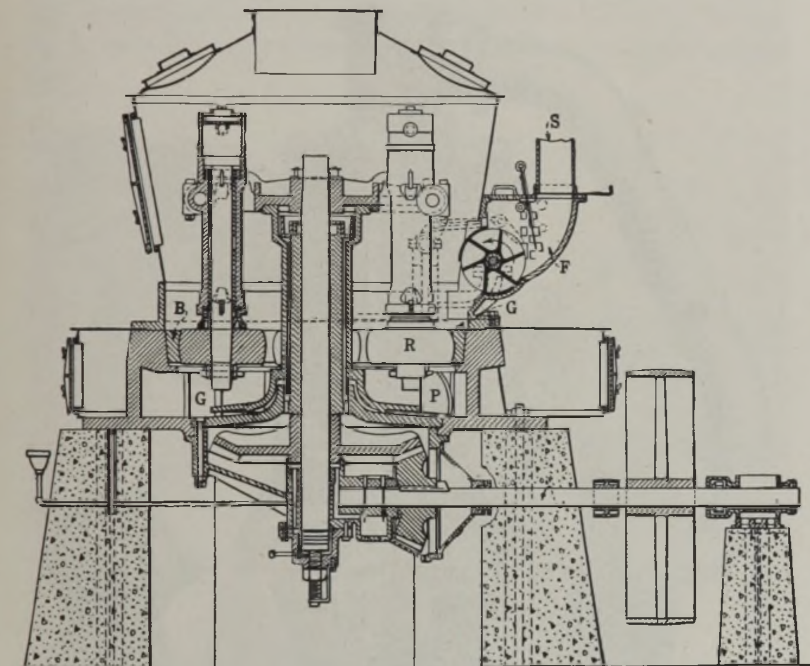


FIG. 83.—Cross section of Raymond roller mill.

There are three general types of mills used for pulverizing coal, as follows:

1. Roller.
2. Impact.
3. Ball.

The Raymond Bowl Mill (Figs. 83 and 84) is an example of the *roller* type. In it the crushed and dried coal is fed "to the mill from a storage bin through the spout *S* (Fig. 83) into the automatic feeding mechanism *F* which delivers it in proper quantities to the grinding chamber *G*. Here it is caught by the manganese-steel plows *P* and thrown up between the rollers *R* and the pulverizing surface or bull ring *B*. One of these plows is located just ahead of each roller so that a constant stream of material is forced between the rolls and the ring.

"The separation of the finely ground material from the coarser particles is accomplished by the constant stream of air drawn

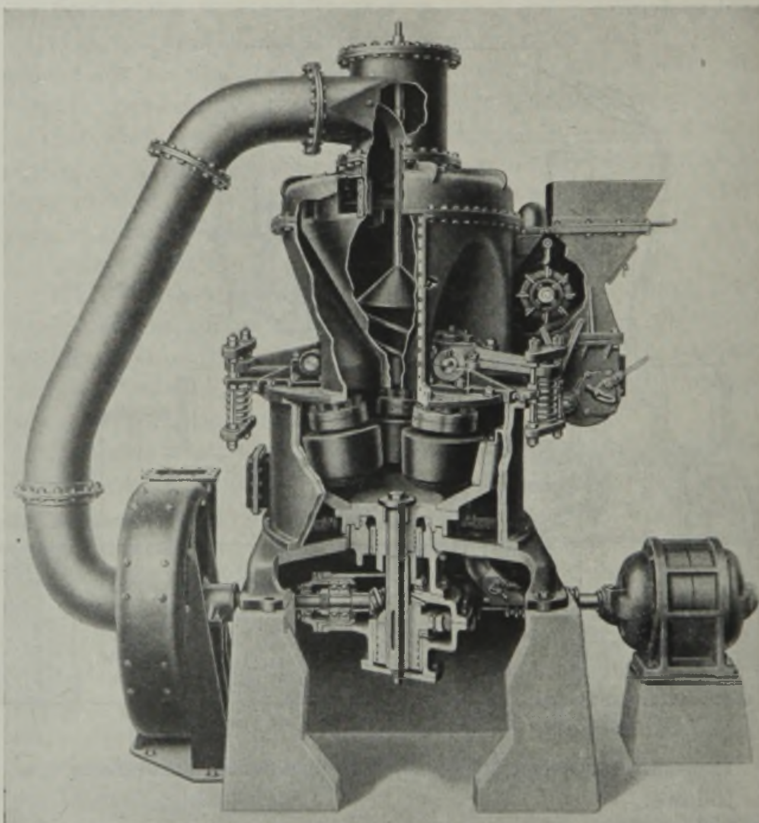


FIG. 84.—View showing interior construction of C-E Raymond bowl mill.

through the mill by the exhaust fan. The stream of air enters the mill through a series of tangential openings around the base of the grinding chamber and passes upward around the rollers *R* and the bull ring *B*. In passing up it carries with it the finely pulverized material from the grinding chamber into the separator. The coarser and heavier particles fall back and are thrown up by the plows *P* to be reground until reduced to the desired fineness.

No fine material remains in the grinding chamber to clog the mill and prevent continuous operation on coarse material."

"Tramp iron and other foreign matter, although supposed to be removed by screens and magnetic separator, sometimes pass through the feeder to the mill. Any tramp iron that does reach the mill is ordinarily discharged by centrifugal force over the rim of the bowl into the air chamber where revolving sweeps discharge it through an opening to a spout with a counterweighted door."

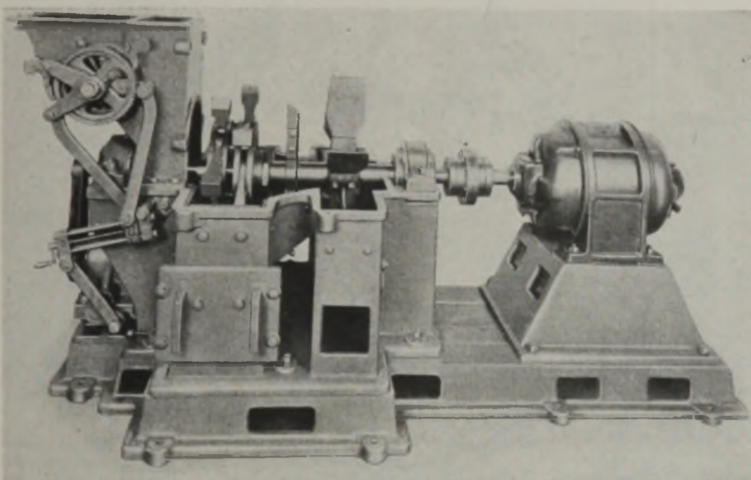


Fig. 85.—View of C-E Raymond impact mill with cover removed to show swinging hammers, fineness regulator and steel plate fan.

In the Raymond bowl mill shown in Fig. 84, "the mixture of pulverized coal and air passes through openings in the outer part of the top plate of the grinding chamber into the classifier where it passes through openings equipped with deflectors or vanes. These are adjustable from the outside for variation of the fineness. A spiral on the inner cone of the classifier facilitates the return of oversized particles to the bowl through an opening in the bottom of the classifier. A cone suspended above this opening tends to build up pressure at this point and aids in rejecting the coarse particles.

"The pulverized coal is blown through the burner supply pipes by the exhauster, which is a steel plate fan enclosed in a cast-iron housing with replaceable liners. It may be coupled to the opposite end of the horizontal worm shaft from the drive unit, or it may be separately driven in the most convenient location. Where there is more than one burner per mill, distributors are required to equalize the flow of coal."

Figures 85 and 86 show a Raymond Impact Mill. In this type "the coal is pulverized by the impact of hammers which are

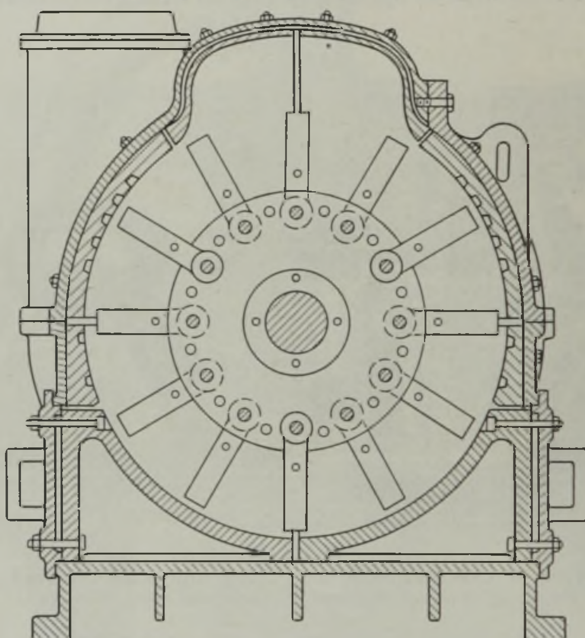


FIG. 86.—Section through C-E Raymond impact mill showing arrangement of rotor with swinging hammers.

pivoted to disks mounted on the shaft. The swinging hammers allow for the passage of foreign matter into the tramp iron pocket at the bottom of the grinding chamber without damage to mill parts.

"For the control of fineness, the smaller mills have an adjustable regulator consisting of revolving blades attached to the shaft between the grinding chamber and the exhauster. The larger sizes have a classifier, similar to the type used on the bowl mill, attached to the top of the mill housing. The action of either the adjustable blades or the classifier causes the oversized particles of coal to be returned to the mill for further grinding. This mill operates on the air separation principle, being equipped with a steel plate exhaust fan which blows the coal to the burners."

In the Aero-unit pulverizer (Fig. 87) the coal is reduced to the proper fineness through progressive stages of pulverization. Air, preferably preheated, carries the particles of coal from one stage to another only after they have been reduced to a size which the

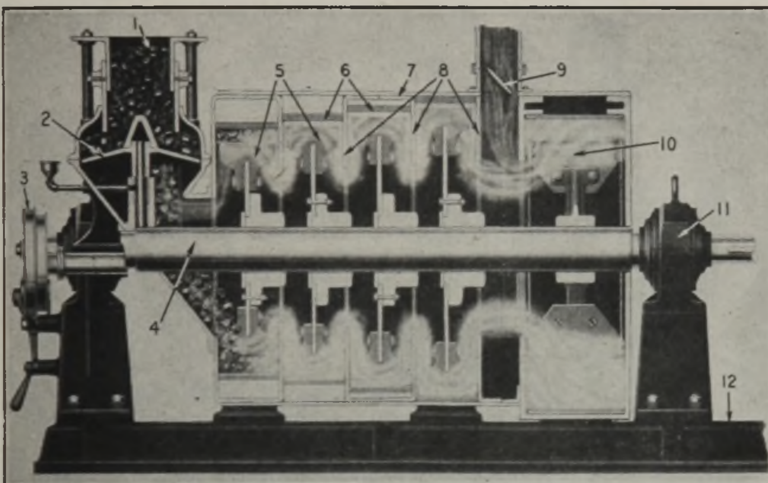


FIG. 87.—Diagrammatic cross section of the Aero-unit pulverizer. (1) Coal inlet. (2) Rotating feed table. (3) Drive for feed mechanism. (4) Extra-heavy alloy-steel shaft. (5) Alloy-steel paddles. (6) Alloy-steel liners. (7) Steel plate casing. (8) Steel diaphragms. (9) Adjustable air intake damper. (10) Fan. (11) Self-aligning double roller bearings. (12) Bed plate.

air velocity is able to move in opposition to the centrifugal force. The fineness of the coal is controlled by the velocity of air through the machine.

The Raymond Impact Mill, the Aero-pulverizer, and other impact-type mills are used in the smaller units. Larger mills are of the roller type such as the Raymond bowl mill (Fig. 84), or the ball mill of which the Babcock and Wilcox Type E Pulverizer (Fig. 88) is an example. This mill, of which Fig. 89 is a cross section, is built in capacities up to 26,000 lb of coal per hr. Another machine (Fig. 90) built by the same company is available in capacities ranging from 1,000 lb to 92,000 lb per hr.

"In the Type B Pulverizer, of intermediate to large capacities, the upper and the lower rings are stationary. The two rows of balls are driven by the intermediate ring, which floats on and is driven by the main shaft. Pressure on the grinding elements is applied by the externally adjusted springs.

"Pulverized coal is removed from the grinding zone by air passing between the balls of the lower row and through ports in the rotating intermediate ring. Some oversize particles drop out of the swirling air stream immediately and drop into the

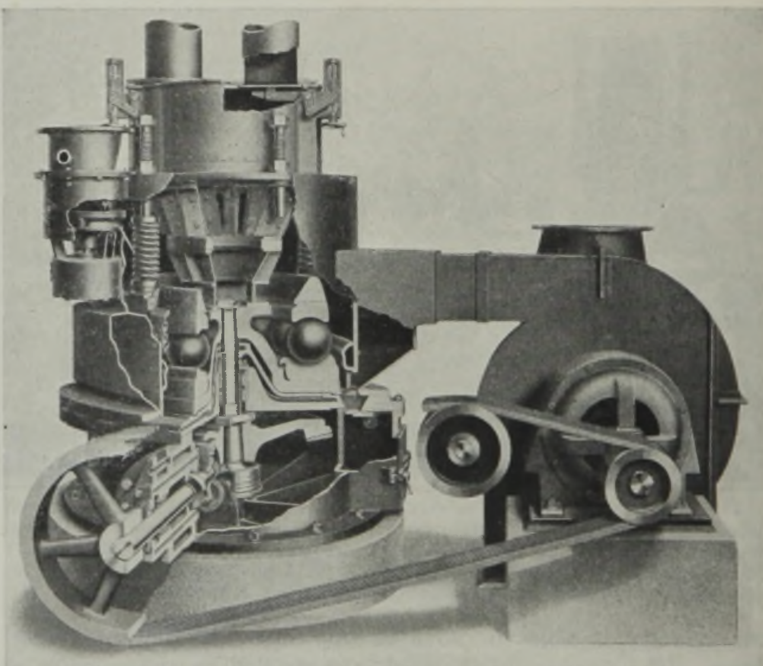


Fig. 88.—Babcock and Wilcox Type E pulverizer.

grinding zone for further pulverization. Further separation of coarse material from fines is effected at the top of the pulverizer by the rotating classifier through which the coal-air stream passes. The coarse material returns to the grinding zone for further grinding, and a homogeneous mixture of air and coal of the proper fineness passes to the burners through pipes attached to the top plate of the pulverizer. This method of classification results in the removal of particles of coal as soon as they become fine enough to be carried out of the pulverizer by the controlled flow of air."

Another type of large ball mill is the Hardinge (Fig. 91). In it the coal enters the conical drum at the left and is crushed by the cast-iron balls as the drum rotates. Air enters through the duct at the right end of the drum and is drawn out by the fan at the extreme right of the mill, carrying with it the pulverized fuel. After passing through a separator, the coarser particles of coal are returned to the mill for further grinding, and the fine material is carried on to the burners.

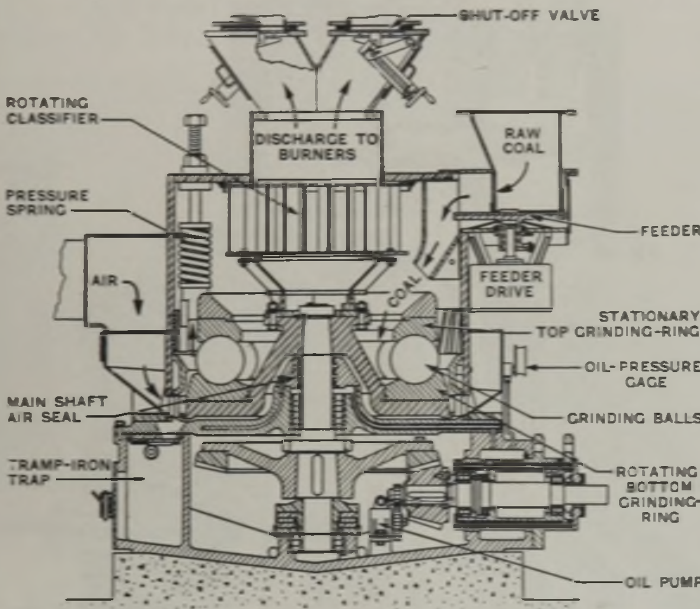


FIG. 89.—Cross-section of Babcock and Wilcox Type E pulverizer.

102. Burners.—There are three methods of firing pulverized-fuel furnaces: *tangential* or *corner, vertical*, and *horizontal*. The proper one to use in a particular installation depends upon a variety of factors and can be determined only by a careful study of these factors, which include such items as plant layout, type of boiler, size and proportions of furnace, availability of preheated air, and others.

"In pulverized-coal firing, turbulence throughout the furnace is essential (1) to bring oxygen and combustible into continuous contact, (2) to scrub away the ash from the surfaces of the particles, and (3) to cause the gases to sweep the water-heating surface in the furnace. Without intensive mixing of coal and air, efficient combustion is impossible. Scrubbing of the coal particles assures contact between the combustible and oxygen, thus promoting rapid combustion and reducing carbon loss.

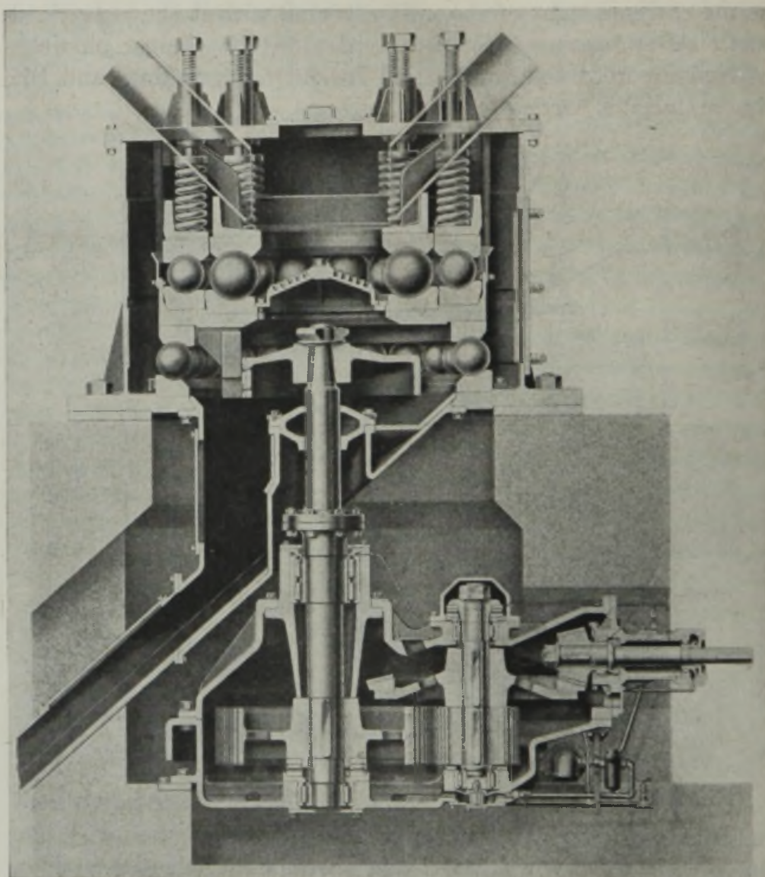


FIG. 90.—Sectional View of Babcock and Wilcox Type B pulverizer with two rows of balls for intermediate capacities.

Sweeping of the water-heating surface in the furnace by the gases increases the evaporation rate for the furnace heating surface.

"Turbulence may be obtained either by variations in design or arrangement of burners. The most effective method is an arrangement of simple nozzles which provides turbulence by impingement of one flame on another. Corner (or tangential) firing is an example of this principle. It is generally conceded that where it can be applied, this type is preferable for thorough mixing of coal and air. These burners have simple nozzles arranged in a casing with forced-draft air ports. The construction is indicated in Fig. 92, and an interior view of a completely water-cooled, dry-bottom furnace, equipped for corner firing, is shown in Fig. 93. The tangential method of firing is well adapted to use where high capacities are desired and where floor space is at a premium. Burners are located in each of the four corners of the furnace, and the fuel nozzles are set within, or

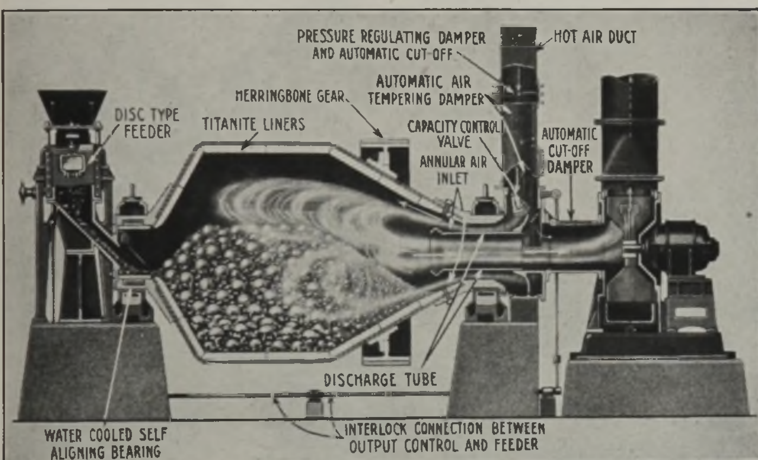


FIG. 91.—Longitudinal section through typical Hardinge ball mill.

placed adjacent to, air inlets. From one to three burners, stacked upon each other, are located in each corner. Placing the burners in the corners of the furnace simplifies plant layout, particularly with respect to the arrangement of air ducts and feed pipes to the burners. Generally, the fuel and air streams are directed tangentially and in a horizontal plane, toward a small imaginary circle in the center of the furnace. This action imparts a rotating motion to the flame body, which spreads out and fills the furnace area. Thorough mixing is assured by the initial impact of the fuel and air streams, and by the manner in which these streams cut through the rotating body of flame.

"However, in some cases the limitations for the layout may prevent an arrangement for flame impingement. For these cases, the straight horizontal or vertical types of burners with internal arrangements for mixing are used. In all burners, the objective is to operate with the minimum amount of primary air to obtain quick ignition and to provide for the addition of secondary air beyond the point of ignition.

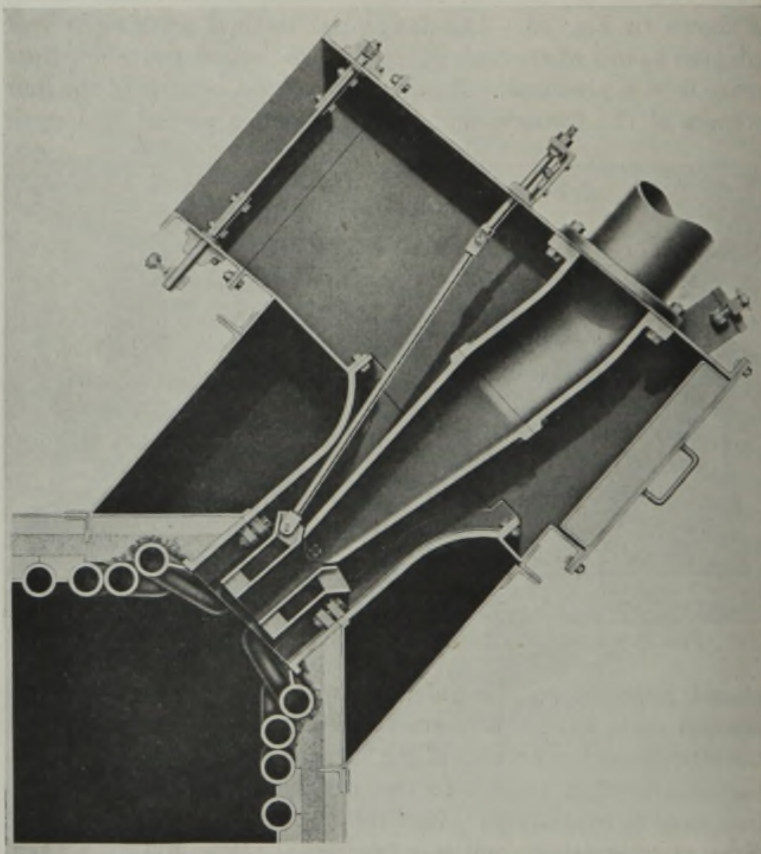


FIG. 92.—Plan view of Combustion Engineering Co. tangential burner.

"In the vertical burners, the primary air carries the fuel and enters the furnace through the burner nozzle, the secondary air is admitted through the front wall of the furnace, and the tertiary air, which may be forced or natural draft at boiler-room temperature or preheated, is admitted through dampers in the burner body. In the horizontal and tangential burners, the primary air carries the fuel and enters the furnace through the burner nozzle; the secondary air enters either through dampers in the burner body or through the casing surrounding the burner; there is no tertiary air supply to the horizontal and tangential burners.

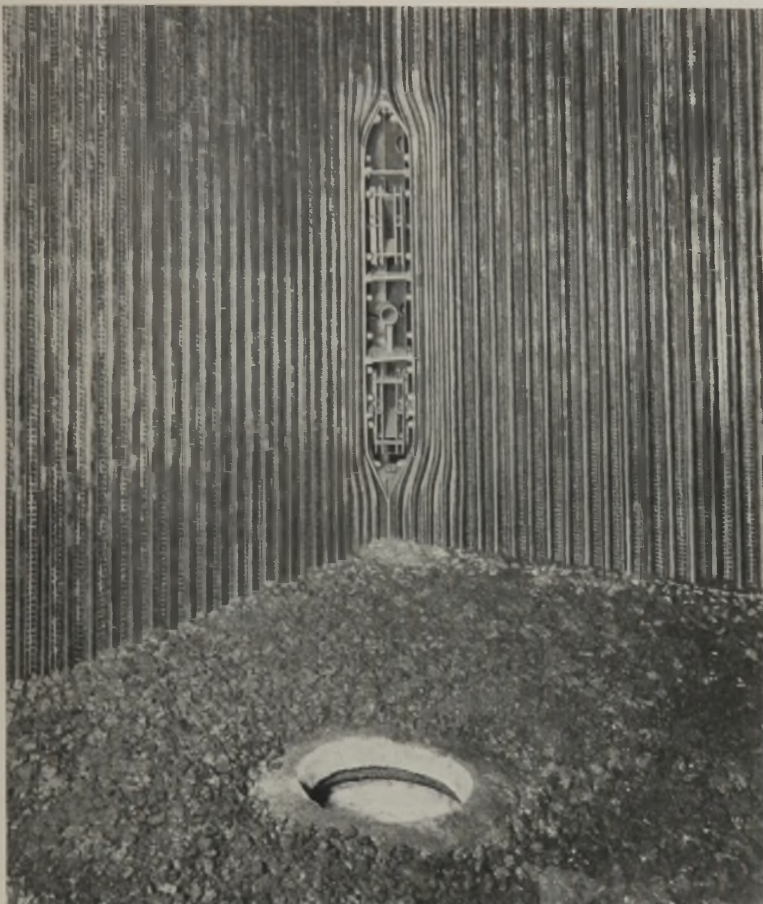


FIG. 93.—Installation view of corner-fired furnace.

"Although the greatest field for application of vertical firing is with the storage system, it may be advantageously used with the direct fired system when burning low volatile coal which requires long flame travel, or in connection with narrow furnaces where the fantail flame is most suitable. A burner for vertical firing is shown in Fig. 94. The coal and air mixture enters through the flat interior nozzle around which air is admitted. Either natural or forced draft may be used. Secondary air is admitted through air ports in the front wall. Preheated air may be used in any type of forced draft burner. In addition to increasing the effi-

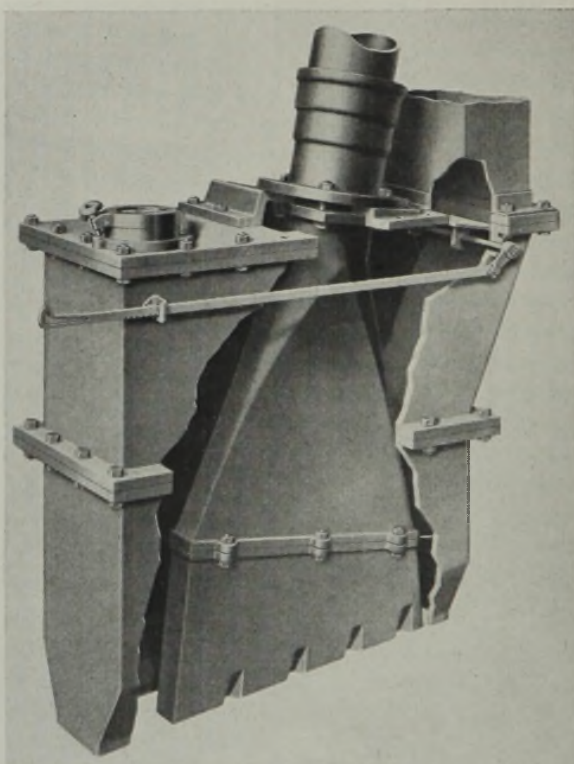


Fig. 94.—Arrangement of Combustion Engineering Co. Type L burner for vertical firing.

ciency of the unit, greater rapidity of ignition may be secured, which, in connection with turbulence and mixing, is essential for high rates of heat release."

103. Problems in Use of Powdered Coal.—Ash and slag from the burning of coal have long been recognized as important factors in boiler-furnace operation. Hand-fired boilers were often limited in output by clinkers in the fuel bed, with resulting excess burden on the firemen. The high efficiency obtained from the use of powdered coal is due to the low excess air. This means high furnace temperatures and consequent fusion of ash and erosion of furnace lining. In the early stages of the use of this fuel much of the molten ash was sprayed against the furnace walls and then ran down into the ashpit, carrying part of the brick with it. When it cooled, it was in such a hard mass as to be almost impossible to remove. The most successful way of counteracting this trouble is through the use of a radiant superheater, or by water-cooling the entire furnace with water-walls, or rows of water tubes lining the walls of the combustion chamber, and a water screen placed across the bottom of the furnace, thus cooling the ash as it drops through the screen into the ashpit and preventing its fusing. All these tubes are connected with the boiler forming part of the water-heating surface.

Another method of reducing the erosion of the furnace walls is by the use of hollow-wall construction. Here the walls are built with a space between the furnace lining and the outer walls; through this space is passed a large part of the air required for combustion. Water-cooling the walls has proved more effective than air-cooling them if the furnace is being operated at a high rate. Admitting excess air and giving up 1 or 2 per cent efficiency will do much to eliminate the erosion of the side walls.

These same difficulties due to high temperatures have arisen in stoker-fired plants and are being met there with the same remedies, *viz.*, water- or air-cooled walls.

Although larger furnaces are required for the use of pulverized coal than were formerly built for stoker installation, the modern stoker furnace is not much smaller than that required for the successful burning of powdered fuel. In some of the latest stoker plants over 30 ft have been allowed between the stoker and the tubes.

Powdered-coal furnaces must be large in order to allow of complete combustion of the fuel, and also to avoid impingement of the flame against the furnace walls, particularly if these walls are lined with refractory materials.

Another problem of the powdered-fuel plant which caused much trouble in former years is that of the very fine powder or ash which passed up the stack and then spread over the surrounding neighborhood like a gray pall, causing as much protest from the communities as black smoke. Various methods have been used with generally satisfactory results to reduce this nuisance. These include collecting the larger particles by water spraying and the fine fly ash by the use of mechanical separators or electrical precipitators.[1]

104. Waterwalls and Screens.—Up to within a few years the walls of all furnaces were built of refractory materials. With the development of the use of pulverized fuel and of high-capacity

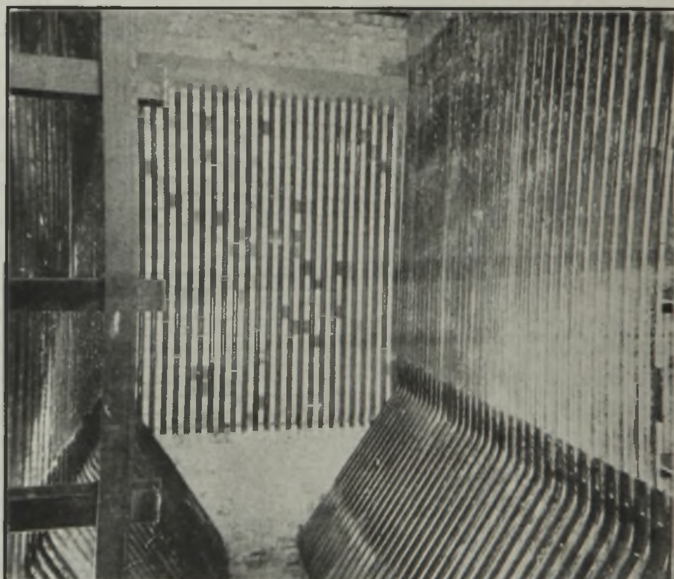


FIG. 95.—Pulverized-fuel furnace with waterwalls on three sides.

boilers some other form of furnace construction became necessary. The air-cooled type of wall was the next step. An air space was left between the firebrick lining of the furnace and the outside wall, and air was circulated through this, heated, and admitted to the furnace. Further experiments showed that better results could be obtained by lining the walls with water tubes and putting a *water screen* at the top of the ashpit. This water screen consists of rows of horizontal water tubes spaced far enough apart so that ash or slag from the furnace can readily drop between them. As the drops of molten slag come near these

[1] See Art. 60 for a previous consideration of pulverized coal and the advantages and disadvantages in its use.

tubes, they radiate their heat away to them and are chilled and solidified before they land in the ashpit, thus preventing slagging.

The water tubes going to make up the walls and water screen are sometimes connected with the boiler circulation and sometimes have an independent circulation. In either case they serve to add to the heat-absorbing capacity of the boiler and are cheaper to maintain than refractory walls. At the present time refractories are eliminated as far as possible and waterwalls

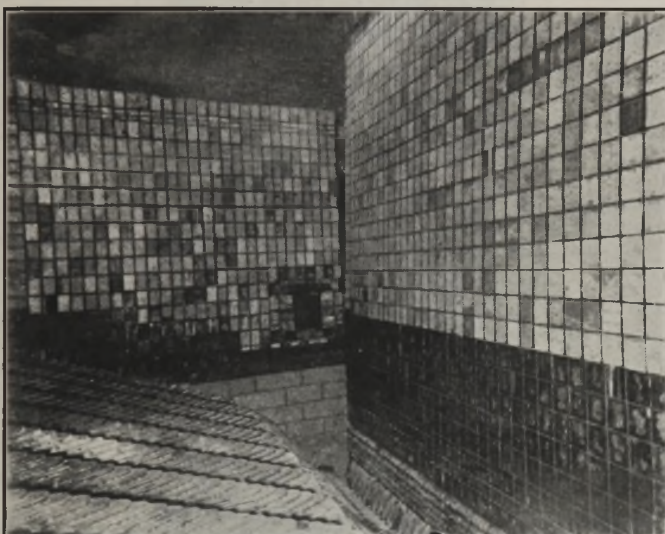


FIG. 96.—Stoker-fired furnace with waterwalls.

are used almost exclusively for high-capacity boilers. In some, all four walls are water-cooled; in others radiant superheaters are placed in one or more of the walls and the others are lined with water tubes. These water-cooled furnaces have become main water evaporation surfaces in high-pressure installations, the remainder of the boiler surface evaporating less than 20 per cent of the water. With modern water-cooled furnaces the rate of combustion is from 2.5 lb to 3 lb of coal per cu ft of furnace volume per hr, as against 1 lb to 1.5 lb with the old refractory lined surface. In both instances the upper limit is set by the slagging of ash on the boiler and superheating surfaces.

Figure 95 shows a pulverized-fuel furnace with Foster-Wheeler waterwalls on three sides; Fig. 96 shows the application of a

Bailey water-cooled furnace to a stoker-fired furnace. In the latter, the vertical water-cooling tubes are protected from localized overheating by metal blocks tightly clamped to the tubes ensuring constant conductivity. This furnace serves a 1,658 hp Babcock and Wilcox boiler which generates 200,000 lb of steam per hr at 375 psi.

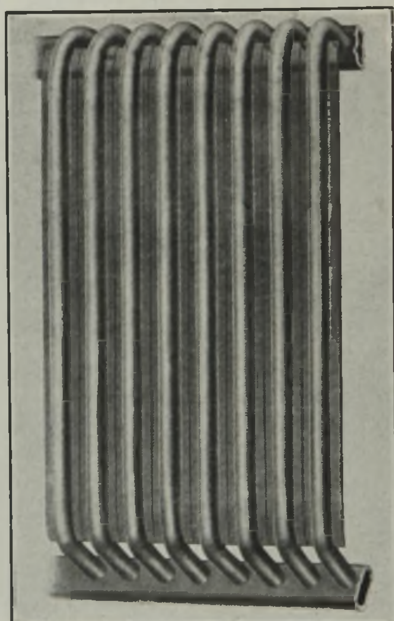


Fig. 97.—View of section of C-E fin-furnace wall from inside of furnace.

In Fig. 124 is shown a furnace with a Foster-Wheeler radiant superheater in the center, waterwalls of the Combustion Engineering fin-tube type at the left, and a water, or slag, screen at the bottom of the picture.

A view of a section of the C-E fin-furnace walls from inside the furnace is shown in Fig. 97. The tubes are bent through the walls at the top and bottom, and the headers are located outside the walls.

105. Boiler Feed Water.—The modern tendency toward large boiler units and high overloads has necessitated more careful consideration of the purity, or impurity, of the feed water. Most natural waters contain certain impurities which, when introduced into a boiler plant, may cause trouble of one kind or another. In case the water supply available is only slightly impure, and so causes little difficulty, the question as to whether the installation of a water-softening or purifying device is war-

TABLE XXV.—APPROXIMATE CLASSIFICATION OF IMPURITIES FOUND IN FEED WATERS, THEIR EFFECT, AND ORDINARY METHODS OF RELIEF

Difficulty resulting from presence of	Nature of difficulty	Ordinary method of overcoming or relieving
Sediment, mud, etc.	Incrustation	Settling tanks, filtration, blowing down.
Readily soluble salts	Incrustation and priming	Blowing down.
Bicarbonates of lime, magnesia, etc.	Incrustation	Heating feed. Treatment by addition of lime or of lime and soda. Caustic soda and barium hydrate.
Sulphate of lime	Incrustation	Treatment by addition of soda. Barium carbonate.
Chloride and sulphate of magnesium.	Corrosion	Treatment by addition of carbonate of soda.
Acid	Corrosion	Alkali.
Dissolved carbonic acid and oxygen.	Corrosion	Heating feed. Keeping air from feed. Addition of caustic soda or slacked lime.
Grease	Corrosion	Filter. Iron alum as coagulant. Neutralization by carbonate of soda. Use of best hydrocarbon oils.
Organic matter	Corrosion	Filter. Use of coagulant.
Organic matter (sewage)	Foaming	Settling tanks. Filter in connection with coagulant.
Carbonate of soda in large quantities.	Priming	New feed supply. If from treatment, change.

ranted must be met and answered. Although, in general, the saving due to the increased economy and the lengthened life of the boiler plant will more than offset the cost of the treatment of the water, it must be remembered that there are certain waters which are absolutely unfit to be used in a boiler, and which no treatment will render usable.

Table XXV, taken from Babcock and Wilcox's *Steam*, lists the principal ill effects in boiler operation occasioned by the use of impure feed, and indicates the causes and methods of relief. These ills are (1) *incrustation* or *scale*, (2) *corrosion*, (3) *foaming*, and (4) *priming*.

One of the commonest troubles caused by improper feed water is boiler scale. Table XXVI, prepared by Professor E. C. Schmidt of the University of Illinois, gives an idea of the loss due to the presence of scale in a boiler.

TABLE XXVI.—LOSS DUE TO BOILER SCALE

Thickness of Scale, In.	Loss of Efficiency, Per Cent
$\frac{1}{50}$	5.4
$\frac{1}{32}$	7.2
$\frac{1}{25}$	9.3
$\frac{1}{20}$	11.1
$\frac{1}{16}$	12.6
$\frac{1}{11}$	15.0
$\frac{1}{9}$	15.9

If, after proper investigation, it is decided that some form of water treatment is desirable, the selection of the method to be followed must be determined. In any case the best results are obtained if the water is purified before it enters the boiler. The principal objects of the purification are the removal of solids held in solution or suspension, and of gases. The various methods used to accomplish these ends are suggested in the last column of Table XXV. If sea water is the source of supply, the only way of treating it is by distillation, which results in practically pure water.

106. Evaporators.—In order to avoid the necessity for treating such large quantities of feed water, the condensate from the condensers is returned to the boilers in plants where surface condensers are used. This condensate is distilled water and its use largely eliminates feed-water difficulties.

As, during the cycle from the boiler to the engine or turbine and return, there are always losses of water, or steam, in one way or another, it is necessary to supply a certain amount of *make-up* water. If distilled water is to be used for this purpose, and it usually is in modern power plants, some form of *evaporator* is necessary. This is a piece of boiler-room equipment designed to use steam to supply the heat necessary for vaporization. There are various types available, in most of which the steam passes through a series of tubes, either surrounded by water, or over which water is allowed to pass in a thin film. In either case evaporation takes place and the vapor is then conveyed directly to the feed-water heater, or to evaporator condensers.

Figure 98 shows a Foster-Wheeler submerged type of evaporator containing 1,850 sq ft of surface and receiving superheated steam at 252 psi. Two of these evaporators operating as a

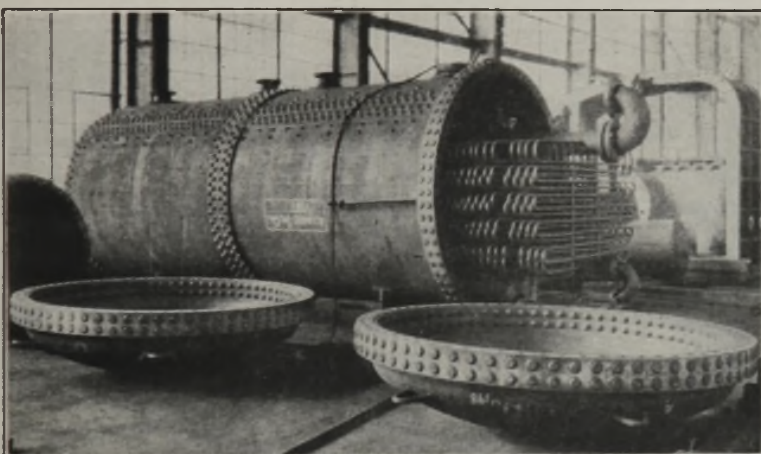


FIG. 98.—Foster-Wheeler submerged type evaporator.

single unit in the power plant of the Illinois Steel Company produce 100,000 lb of make-up per hr in the form of steam at 150 psi, superheated about 60 deg to a final temperature of 430 F. Connected with the evaporators is a reheat desuperheater which desuperheats the steam to the evaporators and superheats the vapor from them. There are also provided two heat exchangers which use the condensate from the evaporator coils to preheat the raw feed to the evaporators. Each heat exchanger contains 800 sq ft of surface and the two together will heat the 100,000 lb of raw water per hr to 255 F while cooling the condensate.

Figure 99 shows a multiple-section horizontal-type Griscom-Russel evaporator with the cover removed and one section partially withdrawn.

107. Deaerators.—As pure water absorbs oxygen very readily, and oxygen carried into the boiler attacks the drum and tubes, especially at the higher temperatures, it becomes necessary to remove the oxygen from the feed water before it enters the boiler. The devices used to accomplish this removal are called *deaerators* and are built in various types. In all of them the water is boiled in the deaerator and the oxygen, air, and other gases driven off from the vapor. Steam is used to supply the heat necessary to vaporize the water. In some cases an open-feed water heater will act as a deaerator, but in order to be effective the

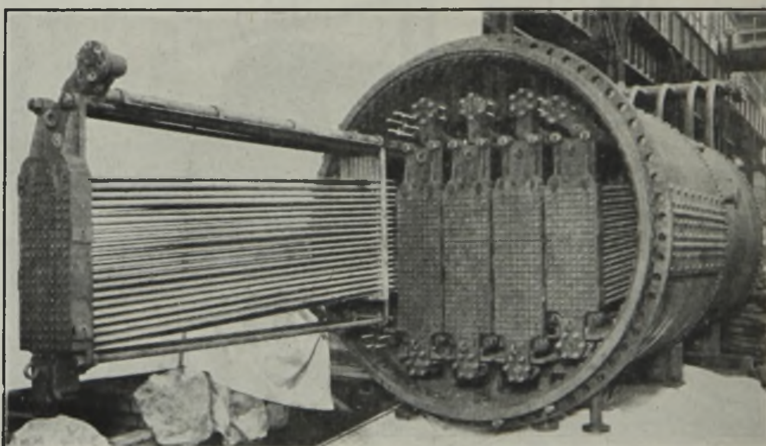


Fig. 99.—Large Griscom-Russel bent-tube evaporator, multiple-section horizontal type.

water must be heated to a temperature high enough to drive off the gases.

108. Boiler Feed Pumps.—A boiler auxiliary which must, of necessity, be absolutely positive and reliable under all conditions is the device used to force the feed water into the boiler. This may be either a *feed pump* or an *injector*. The feed pump may, in turn, be (1) a direct-acting pump driven by its own steam cylinders; (2) a reciprocating pump belted to the machinery, or driven by an independent motor; or (3) a centrifugal pump driven by either a turbine or an electric motor.

The direct-acting pump driven by its own steam cylinders is used in many plants of less than 1,500 hp. It has the advantage over the belted pump of being independent of the operation of the main engine; in addition, its speed can be adjusted so as to give uniform feeding. Its principal disadvantage is in its large steam consumption. The smaller pumps use from 150 lb to 300 lb of steam per ihp-hr; larger ones from 80 lb to 150 lb; and

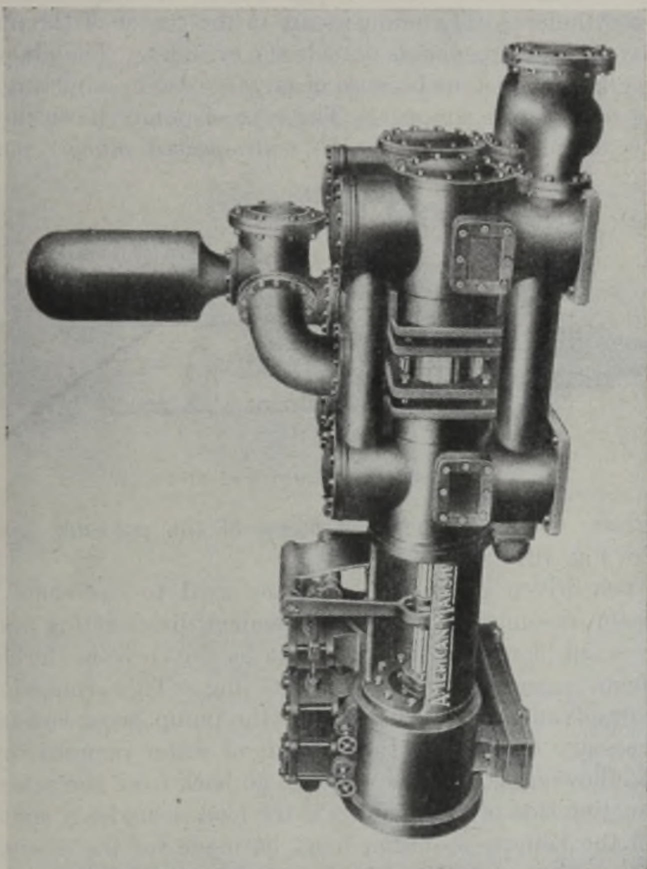


FIG. 100.—Duplex, outside, center-packed, turret type.

compound condensing feed pumps of the direct-acting type from 60 lb to 75 lb. The mechanical efficiency of these pumps is about 80 per cent.

In direct-acting pumps both the steam and water pistons are on the same piston rod, and both the steam and water cylinders are double acting. The steam is admitted to the cylinder at boiler pressure for the full stroke and is not used expansively since, if it were, the pump would stop when the pressure in the steam cylinder dropped to the pressure against the piston in the water cylinder.

Figure 100 shows a direct-acting feed pump having four single-acting water cylinders. This pump has two plungers working in these cylinders. The plungers are in the center of the pump and have the packing glands outside the cylinder. The plungers are preferable to pistons because of greater ease in adjusting the packing to minimize slippage. The type of pump shown in this figure is called a *duplex, outside, center-packed plunger pump.*

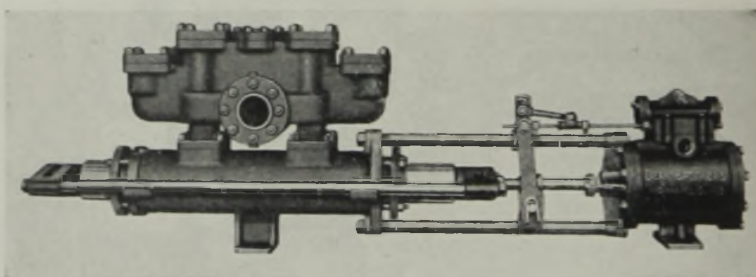


FIG. 101.—Outside, end-packed, pot-valve type.

An *outside, end-packed plunger pump* of the *pot-valve type* is shown in Fig. 101.

The belt-driven pump is sometimes used to overcome the high steam consumption of the independent direct-acting pump. In very small plants these pumps may be driven from the shaft of the main engine, or from the line shafting. This arrangement has its disadvantages. The speed of the pump being constant, it is necessary to regulate the amount of water pumped, by a by-pass, allowing part of the water to go back from the pressure to the suction side of the pump. If the feed is suddenly shut off from all the boilers, provision must be made for the discharge from the pump to be turned back to the suction automatically. It is not possible to use a pump so driven except when the engine is running, and an auxiliary feeding device must be provided that can be operated when the main engine is shut down.

In the larger plants using belted pumps, the power is supplied by an electric motor. This gives many of the advantages of the independent pump without its disadvantage of high steam consumption.

In plants of over 1,500 hp to 2,000 hp, centrifugal feed pumps are in almost universal use. They are built in several stages, depending upon the pressure in the boiler to be fed. Centrifugal pumps supply a continuous flow of water without the pulsating strains incident to reciprocating pumps. Although, with their drives, their initial cost is greater than that of the direct-acting pumps, their cost of maintenance is less, owing to the absence of valves. They may be either turbine- or motor-driven, although the latter is the method used in the majority of recent installations.

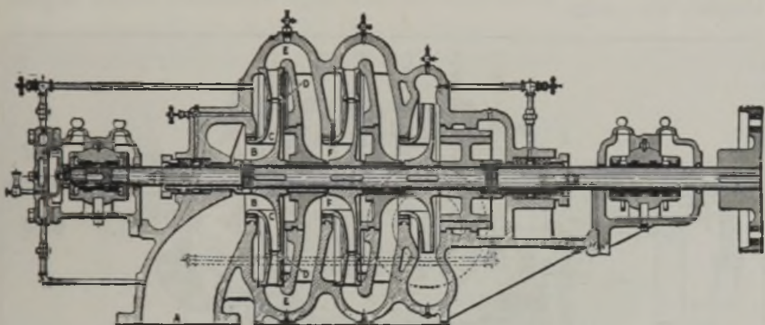


FIG. 102.—Cross section of Cameron three-stage centrifugal turbine pump.

Figure 102 shows a cross section of a Cameron centrifugal turbine pump.

Water enters the pump at point *A* and proceeds to the first impeller, which it enters at points *B*. This impeller is revolving at high speed and has curved radial passages, or vanes *C*, which connect with openings *B* in the hub of the impeller. Immediately on entering the impeller, the water comes under the influence of the centrifugal force resulting from the rotation of the impeller, and is moved to the periphery at a gradually increasing speed, finally leaving the impeller at high velocity and immediately entering the diffusion vane at point *D*.

The water enters *D* at high velocity, which velocity is to be changed into useful pressure, and also reduced so that the turn of 180 degrees at point *E* shall be accomplished without serious friction loss, and without the water striking the surface at a velocity that might be destructive, even if very pure water were being handled. The diffusion vanes accomplish both these results. They form part of a stationary concentric ring completely surrounding the periphery of the impeller, and contain a number of openings or passageways formed by vanes, these passageways gradually increasing in area so that at point *E* the velocity has been very much reduced and the energy represented by the velocity now exists as pressure.

The water is now flowing at a reduced velocity, and at the pressure gained at the first stage, through the channel ring into the second stage at point F, where the process is repeated until the last stage is reached, each impeller adding its increment of pressure. In the final stage the diffusion vane is omitted and a volute chamber substituted for the diffusion vane. The vane is omitted here, because the water does not have to make the turn as in the preceding stages, but passes instead directly into the discharge line. The omission of this vane, therefore, tends to simplify the pump.

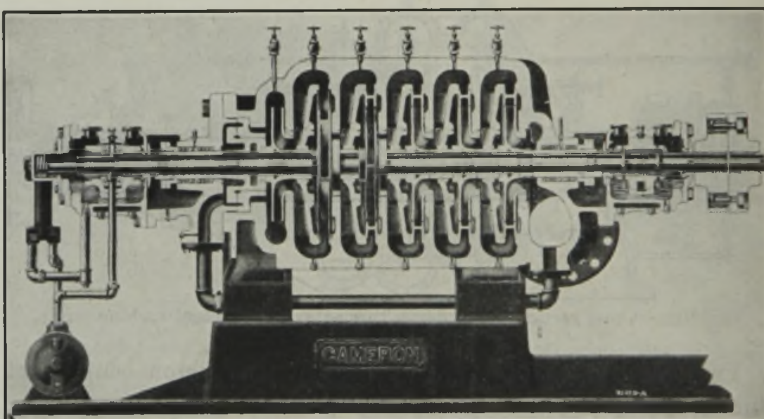


FIG. 103.—Cross section of six-stage Cameron class HMT pump equipped with force-feed lubrication.

Figure 103 shows a Cameron six-stage centrifugal boiler feed pump for pressures from 700 psi to 800 psi.

109. Steam Injectors.—The injector, invented about 1858 by Giffard, a French engineer, is used as an auxiliary device for feeding stationary boilers, and is in general use for feeding locomotives.

Figure 104 shows a cutaway view and Fig. 105 a cross section of a Sellers self-acting injector of the lifting type. Should the water supply to this injector be broken, the injector will restart when the supply is reestablished.

Steam from the boiler enters the injector through 19, Fig. 105. To start the injector, the handle 33 is pulled to the left a short distance, admitting steam from 20 through the small diagonal drilled holes 4, and discharging it through 7 into the overflow chamber 25 and thence out through the overflow valve 30 to the waste pipe 29. This steam in passing through the annular nozzle surrounding the steam nozzle carries the air along with it, creating a vacuum in the space 6 and raising water from the supply through 10. This water is carried along by the steam through opening 7 into the space 25 and issues from the waste pipe 29. When water appears here, the handle 33 is drawn farther to the left, allowing steam to flow through the central portion of nozzle 3. This steam, coming in contact with the ring of water, combines with it in the combining tube 2 and carries it on to the delivery tube 1. By the time the mixture has reached the throat of this nozzle, the steam has all been condensed and the

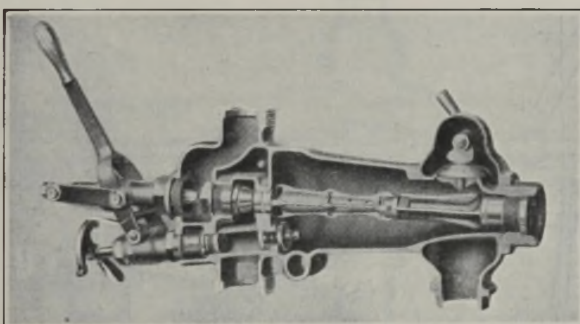


FIG. 104.—Cutaway view of Sellers injector.

water is traveling at a high velocity imparted to it by the steam. As the delivery tube, or nozzle, increases in cross section, the velocity of the water is decreased and the pressure increased. Consequently, the head is sufficient to open the check valve 12 and force the water to enter the boiler.

If the supply of water is cut off, the steam passing through the small openings 4 will continue to produce a vacuum in 6, so that, as soon as the water supply is reestablished, the injector will automatically start to operate again. When the injector is in operation, the vacuum in the overflow chamber 25 draws an additional supply of water through the automatic inlet valve 309. This water enters the combining tube 2 through the small holes 7, and is forced into the boiler, increasing the capacity about 20 per cent. As the pressure drops, the vacuum in 6 falls and less water is raised to enter the combining tube and boiler. The water-regulating valve 40 is used only to regulate the capacity to suit the needs of the boiler. When the injector is used as a heater, the handle 34 is raised so as to hold the overflow valve 30 closed.

The injector is mechanically a very inefficient pump for general pumping purposes. It is installed, however, as an auxiliary method of feeding the boiler in case of accident to the regular feed pump. As a boiler feeder it has a thermal efficiency of almost 100 per cent, since all the heat of the steam used by the injector, except that lost by radiation, goes into the feed water. It

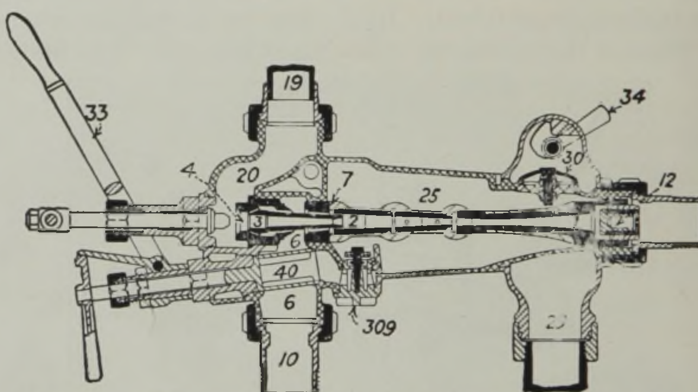


Fig. 105.—Cross section of Sellers self-acting injector—lifting type.

cannot be used with water having a temperature of over 100 F to 120 F.

There are many forms of injectors made for different conditions. In locomotives, injectors are commonly used for feeding the boiler, as they take very little space and furnish the only means of warming the feed water, the exhaust steam being used to produce draft. Each locomotive is usually provided with two injectors. In recent years special boiler feed pumps for use in locomotives have been developed.

110. Pump Connection.—When a pump or injector is handling cold water, the lift on the suction side should not exceed 22 ft. Most engineers try to install pumping apparatus with a suction head of not more than 15 ft.

Hot water, *i.e.*, water exceeding 120 F in temperature, cannot be raised by a pump, as the lowering of the pressure in the suction pipe lowers the temperature of the boiling point of the water in the pipe, the water in the suction boils, and all the pump draws is steam.

When hot water is to be handled, the pump should be below the level of the water on the suction side. Where pumps are installed handling hot water from a feed-water heater, the level of water in the heater should be 5 ft above the center line of the pump cylinders if possible.

Injectors are seldom used to handle hot water, as they are very difficult to start with water exceeding 100 F.

111. Feed-water Heating.—In every modern boiler plant some means is provided for heating the feed water before it enters the boilers. In most cases the water used has been previously distilled and is, therefore, free from the scale-forming matter which otherwise would be precipitated when this water is heated. The feed-water heating is usually accomplished in one of two ways: (1) by utilizing the heat in the exhaust from the pumps, fan engines, or other auxiliaries, or by using heat from live or bleeder steam; (2) by the use of the waste gases from the furnace. Devices for using steam for heating the water are called *feed-water heaters,* and those for using the gases from the furnace are termed *economizers.*

In the larger plants, particularly those designed for high-pressure steam, the feed-water heating is done in steps, or stages, using first either the exhaust steam, or the flue gases, to heat the water up to a certain temperature, and then raising the temperature further by the use of steam drawn off from various stages of the turbines. This practice is known as *interstage heating;* in it the steam is passed through closed feed-water heaters.

The principal advantages of heating the feed water are: (1) the saving in Btu and consequent saving in fuel consumed, due to the increase in the temperature of the feed; (2) the saving in wear and tear on the boiler due to introducing hot instead of cold water, thereby reducing the strain on the boiler; (3) the saving in the time required for the conversion of water into steam and the consequent increase in the capacity of the boiler; and, (4) the reduction, by stage bleeding to feed-water heaters, of the volume of low-pressure steam that would otherwise have to pass through the last stages of a turbine. A heater which increases the temperature of the feed water from 70 F to 200 F may save about 12 per cent of the fuel, or 1 per cent for each 11-deg rise in temperature, and often will pay for itself in a few months.

112. Types of Feed-water Heaters.—There are two general types of heaters: the *open* and the *closed.* The open feed-water heater (Fig. 106) consists of a cast- or wrought-iron shell into which the steam is led. The cold water is admitted at the top of the heater, and is allowed to pass through the steam in streams or sheets. In this type of heater the feed water and the

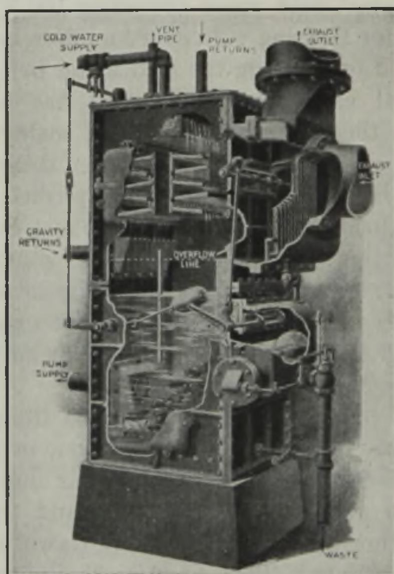


Fig. 106.—Cochrane open feed-water heater.

steam come into direct contact with each other. The water usually passes over pans, or trays, upon which any scale-producing matter can be deposited. When it is desired to clean the heater, it is only necessary to take out these pans and clean them. When exhaust steam is used, it should be passed through an oil separator before entering the heater. The hot feed water is usually passed through some form of filter before flowing to the feed pumps. The feed-water heater should be located at a sufficient height above the feed pump so that the water will enter at a pressure. This distance should be 5 ft or more. The heater may also be used as a receptacle for the hot water which is drained from the steam mains, and for other hot condensed steam which does not contain oil. A uniform water level is maintained in the heater by a float valve, which automatically allows water to enter the heater when the level gets below a certain point.

The closed heater consists of a cylindrical shell of cast iron, or steel, containing tubes extending from a header at one end of the heater to a header at the other end; or tubes in the form of coils of pipe (Fig. 107). Steam enters the heater on the

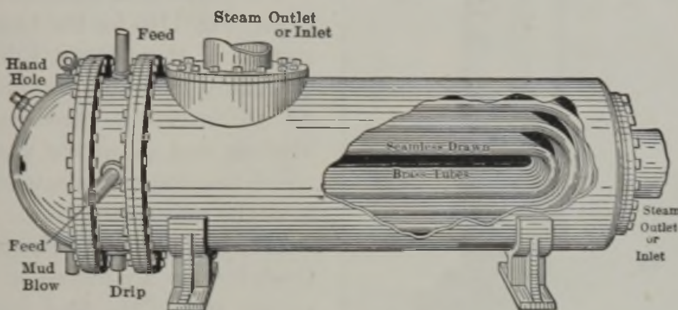


FIG. 107.—Closed feed-water heater.

outside of the tubes and the feed water is pumped through them. In a closed heater the feed water and the steam used do not come in contact with each other. The closed heaters are usually used where it is desired to pass the water through the heaters under pressure. They are more expensive than the open heaters and are more difficult to clean, but their use is increasing owing to the growing practice of interstage heating of the feed water in plants using high-pressure steam, by extracting steam from successive stages of the turbine.

Figure 108 shows a Reilly closed type of feed-water heater, which is made in sizes from 50 to 4,000 boiler hp.

The growth of high-pressure steam plants has resulted in the development of a new design of feed-water heater in which the parts are all made much heavier, as pressures as high as 1,400 lb to 1,600 lb are becoming not uncommon in these heaters.

Figure 109 shows a diagram of a high-pressure feed-water heater with floating head installed at the Hell Gate Power Plant of the United Electric Light and Power Company of New York. This heater

. . . is designed to heat 1,500,000 lb of feed water per hr at a pressure of 1,500 lb per sq in. The tube sheets and cover plates are made of steel

6 in. thick for the stationary water box, and 5 in. for the floating water box. At maximum rating the steam used for heating is superheated 200 degrees Fahrenheit, and at low ratings the steam will contain more than 350 degrees superheat.

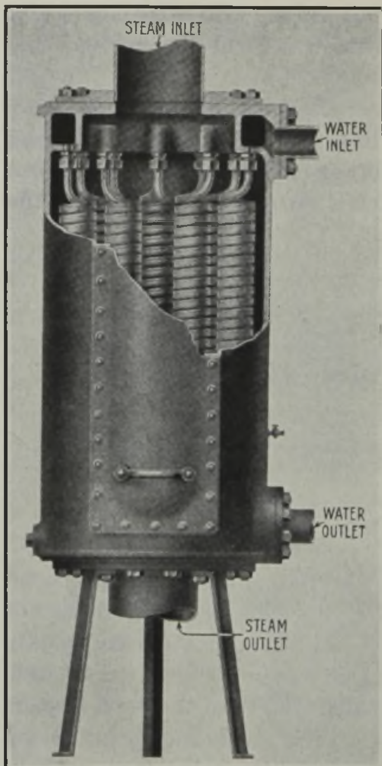


Fig. 108.—Reilly closed feed-water heater.

The plant heat-balance diagram (Fig. 110) for the Delray station of the Detroit Edison Company indicates the method of interstage feed-water heating, and shows the way in which the evaporator is used to heat the make-up water.

113. Installation of Heaters. Open heaters are placed on the *suction* side of the feed pump. Closed heaters handling water not previously heated are generally placed on the *discharge* side of the pump. The center of the feed pump cylinder should be at least 5 ft below the level of the water in an open heater, or a closed heater on the suction side of the pump, as a feed pump cannot lift hot water.

Injectors are never used with an open heater as they cannot use hot water, but they may be used with closed heaters.

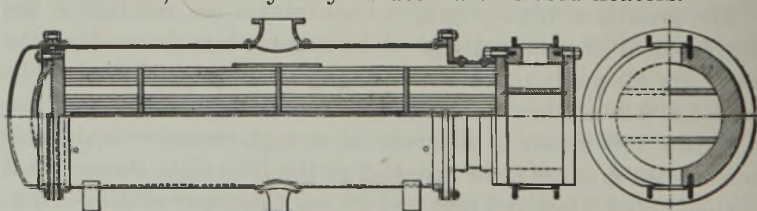


Fig. 109.—Diagram of high-pressure feed-water heater.

114. Economizers.—Any device that heats the feed water by means of the heat in the gases that leave the boiler is termed an *economizer*. Figure 111 shows the elevations of a Green economizer and Fig. 112 its location in the breeching, or flue, which carries the gases from the boiler to the chimney. The cold

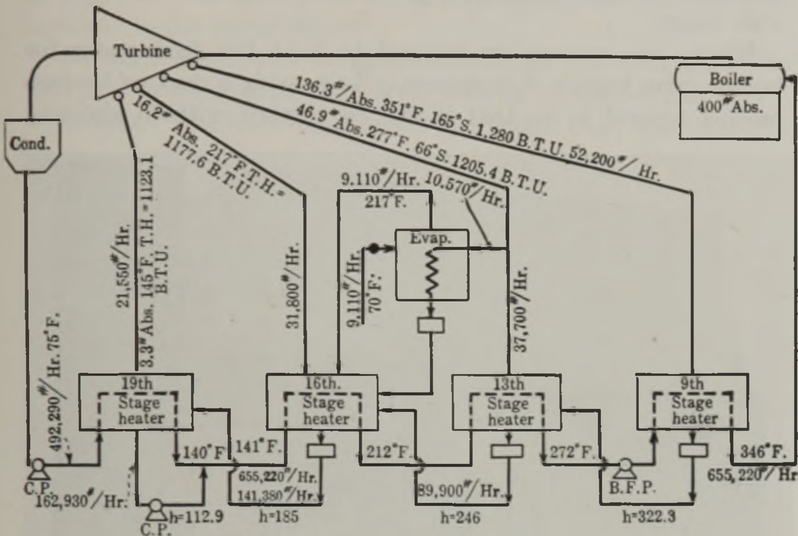
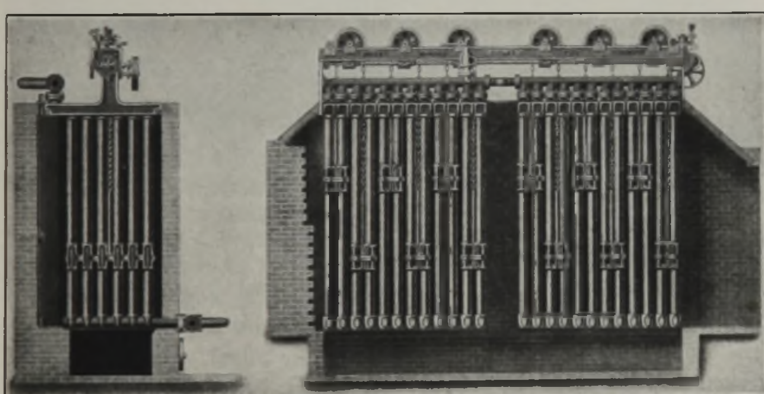


FIG. 110.—Diagram showing the plant heat-balance at Delray station of Detroit Edison Company.



End elevation Side elevation

FIG. 111.—Green's economizer.

water is pumped into the lower header, and, after being heated, passes out from the upper header to the boiler. The flue gases from the furnace pass around the tubes and headers containing the feed water. The tubes, as shown in the cut, are provided with scrapers operated from time to time to remove the soot. Other types of economizers use air or steam soot blowers to clean the external tube surfaces; some types are cleaned by washing with water.

Where one economizer is used to heat the feed water for two or more boilers, it is necessary to provide a duct or by-pass passing around it, so that it can be cleaned without shutting

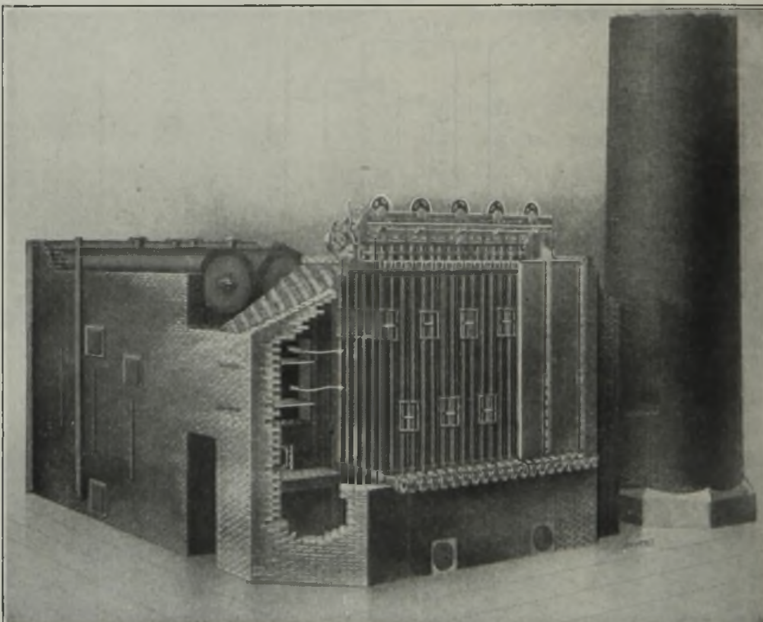


FIG. 112.—Green's economizer, showing location in breeching.

down the plant. Present practice tends toward the use of individual economizers for each boiler without any provision for by-passing the gases. There is ordinarily no necessity for shutting down such a unit to clean the economizer at other periods than those in which the boiler with which it is set is taken out of service. The omission of a by-pass duct reduces to a minimum the liability of air leakage.

The Babcock and Wilcox *return bend economizer* is shown in Figs. 113 and 114. Figure 113 shows the return bends on the seamless steel tubes, the end doors that can be opened for cleaning, inspection, and repair, and the opening at the bottom of the side, out of which the stack gases flow. The soot hoppers in the form of rectangular truncated pyramids are shown at the bottom of Fig. 114. At the top of the figure are seen the multiple connections to the boiler drum.

The stack gases enter the economizer at the top of the back side and flow down around the tubes, leaving at the bottom of the front side. The feed water enters the forged steel header shown

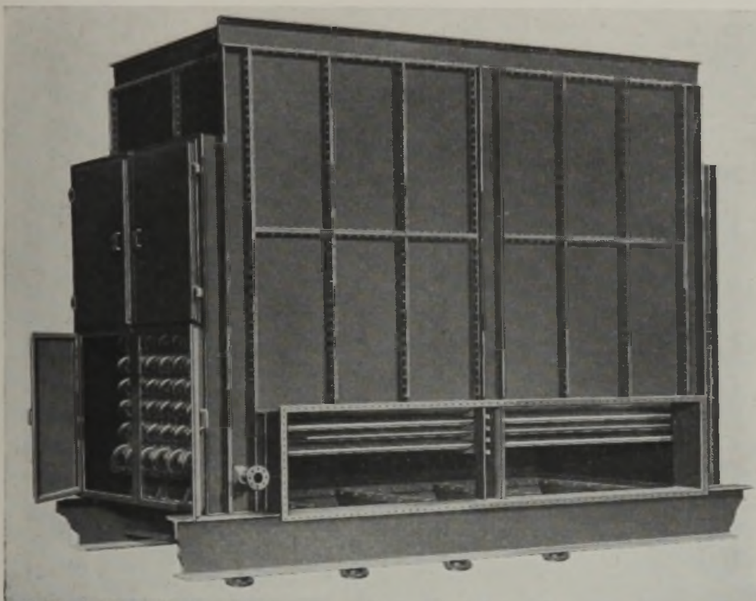


FIG. 113.—Babcock and Wilcox return bend economizer.

in the lower left corner of both Figs. 113 and 114 and flows up through the tubes to their connection with the boiler drum.

As is shown in Fig. 115, the Elesco *fin tube economizer* has fins welded to the tubes in a vertical position, thus increasing the heating surface and hence decreasing the length of tubing required for any given performance condition. This economizer is made in two types: one for installation where internal cleaning of the tubes is necessary, and one (Fig. 115) where feed-water conditions are such as to make access to the internal heating surface of the economizer seldom, if ever, necessary. If occasional internal inspection seems desirable, a few tube units can be supplied with standard flanged return bends.

Figure 116 shows the setting of a Foster-Wheeler economizer in an installation which includes a Heine boiler, a Foster-Wheeler superheater, and a chain grate.

Economizers are installed so as to make use of the heat in the gases leaving a boiler and thus reduce the waste in heat going up the stack. They may be installed also to increase the capacity of a boiler plant which is too small for its services.

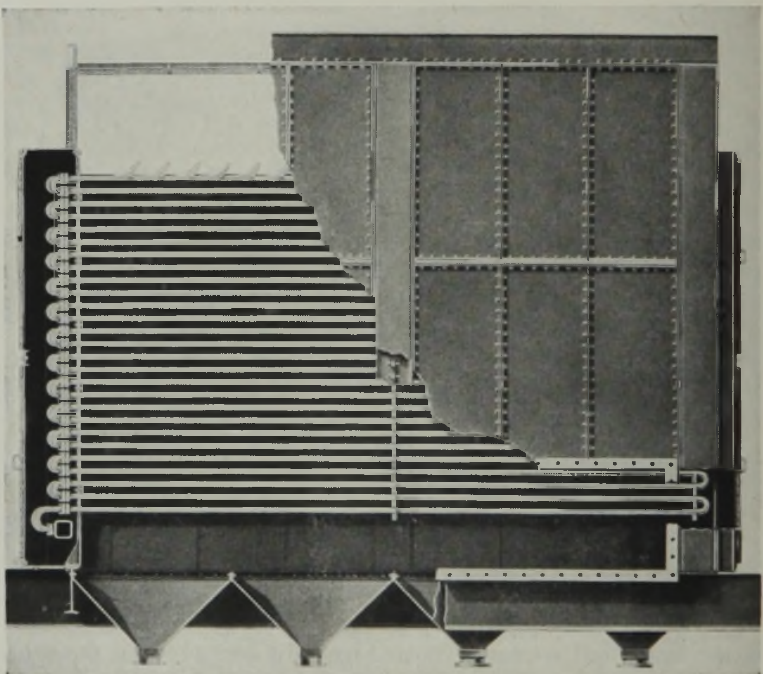


Fig. 114.—Cutaway view of Babcock and Wilcox return bend economizer.

They deliver the water to the boiler at a high temperature, reducing the strain and the leakage caused by the admission of cold water, and they precipitate in the economizer many impurities in the feed water, thus keeping them out of the boilers. Their disadvantages are (1) high first cost and cost of repairs; (2) reduction in the strength of the draft, owing to the additional friction caused by the economizer; (3) increased liability of air leakage; and (4) increased complexity of installation and operation. From 4 sq ft to 6 sq ft of economizer surface should be provided per boiler horsepower developed under average conditions.

It must be remembered that the economizer return as represented by the rise in temperature of the feed water, or fall in temperature of the exhaust gases, is the gross and not the net gain. In all economizer installations an induced-draft fan is necessary which, unless the plant has a very high stack, must be operated the larger part of the time. The power consumption

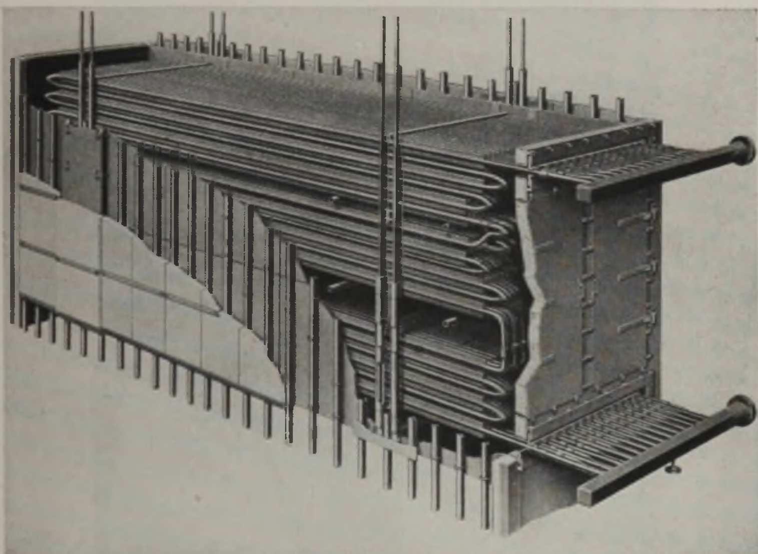


Fig. 115.—Arrangement of Type C Elesco economizer as installed within the boiler setting.

of this fan should then be deducted from the gross increased efficiency due to the economizer, in order to get the real net gain.

The use of economizers has been increased during recent years by three factors: (1) the increased cost of fuel; (2) the higher boiler capacities required; and (3) the tendency toward increased boiler pressure. The greater the fuel cost, the larger will be the warranted outlay for equipment or devices that will decrease fuel consumption. The higher boiler capacities are inevitably accompanied by an increase in the temperature of the stack gases, and a consequent increase in the heat which it is possible for a given economizer to absorb.

115. Air Preheaters.—As has been stated in the previous chapter, the largest loss of heat in a boiler plant is that which goes up the stack in the escaping flue gases. The economizer has absorbed a certain amount of this loss but has certain inherent disadvantages that make its installation impractical and uneco-

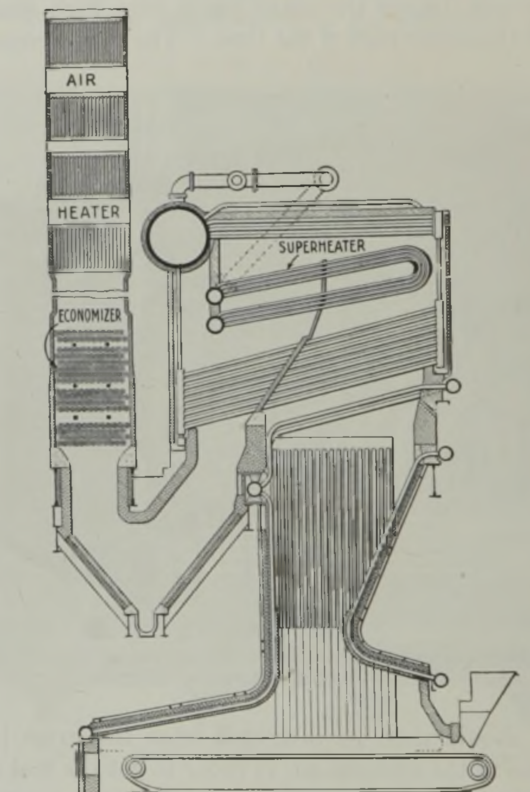


FIG. 116.—Installation at Kansas City Light and Power Company.

nomical in many cases. In addition, the practice of turbine bleeding for heating the feed water is replacing the use of the economizer in many modern plants.

These facts together with the trend toward higher pressures and higher rates of evaporation resulting in higher flue-gas temperatures have led to the development and the use of another piece of equipment to reduce the flue-gas loss. This is the *air* *preheater*, a device using the heat of the flue gases to preheat the air used in combustion.

There are three general classes of air preheaters: the *plate*, the *tubular*—both of which operate on the recuperative or convection principle—and the *regenerative*.

In the plate type (Fig. 117) the air flows through the elements or units going to make up the preheater, and the hot gases flow

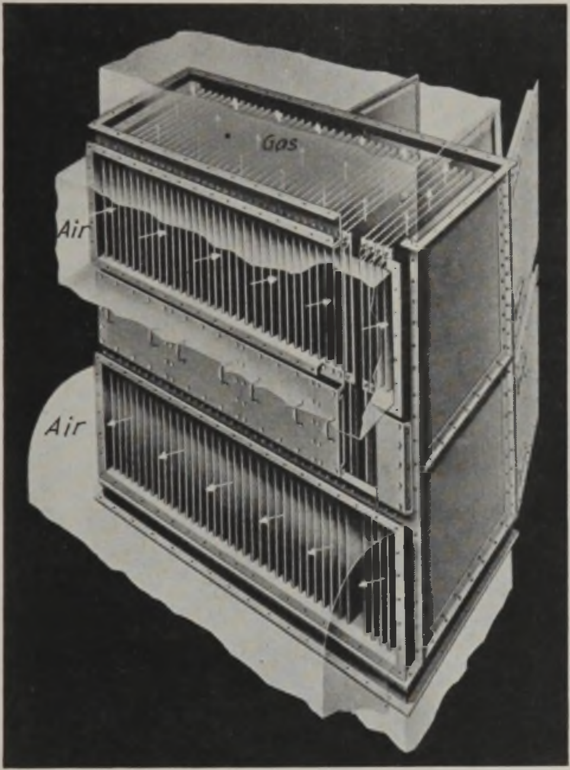


FIG. 117.—Sturtevant plate-type air preheater.

between them. The cold air enters the heater near the top of the front face in Fig. 117 and leaves at the bottom opening of the front face. The hot gases enter at the bottom and pass vertically upward, leaving at the top. The principle of counter-flow is followed; thus the hottest gases are giving up heat to the hottest air, and the colder entering air receives heat from the cooler gases.

The tubular type (Fig. 118) acts on the same principle as the plate type. In this preheater the gas usually flows through the tubes and the air circulates around them, but in some cases the opposite arrangement is made.

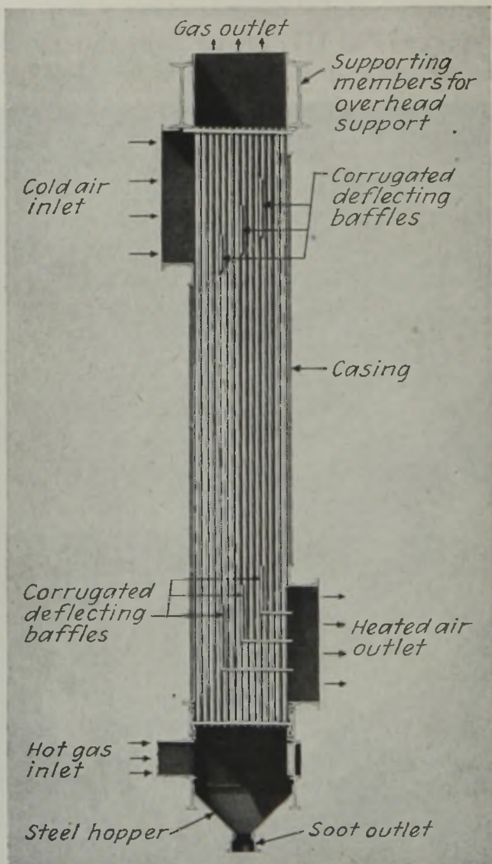


Fig. 118.—Babcock and Wilcox tubular-type air heater.

The regenerative type is illustrated by the Ljungstrom air preheater (Fig. 119). Here a metal rotor is alternately heated by the gases and cooled by the air. Flue gases enter the lower left chamber as indicated and, after passing vertically upward through the passages of the slowly moving rotor, are drawn to the stack by the induced-draft fan. At the same time cold air enters the upper right chamber and is forced downward through the rotor into a duct leading to the furnace. The flow of both flue gases and air is continuous.

The hot flue gas gives up heat to the metal surfaces in the rotor. This heat remains stored up in this metal until the travel of the rotor brings the hot surface to the opposite side of the preheater where it comes in contact with the stream of air which in turn

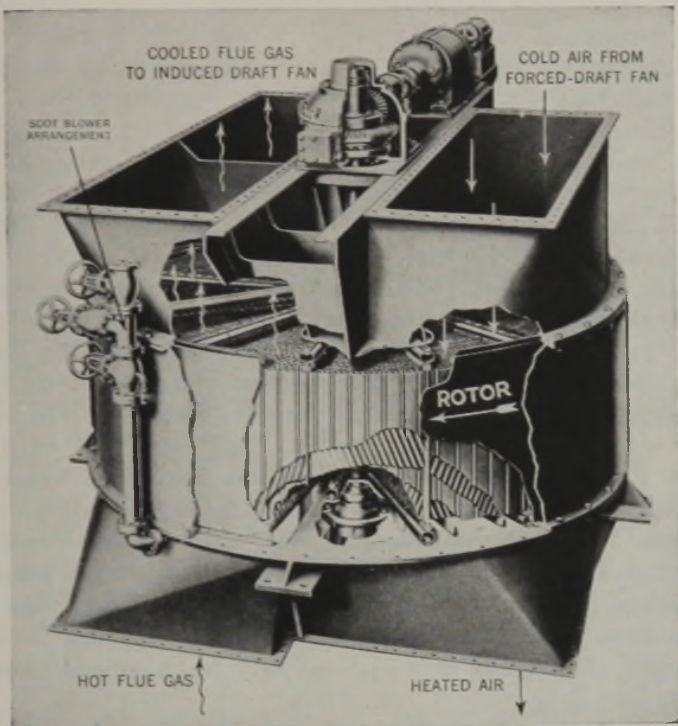


FIG. 119.—Ljungstrom regenerative air preheater.

absorbs the heat from the rotor. As this process is continuous, there is a constant transfer of heat from the flue gas to the air.

There are limitations, particularly with stoker-fired furnaces, to the air temperatures that can be used without damage to the equipment. Maximum temperatures of 400 F to 500 F have been used under properly designed conditions and with a coal of high ash-fusion temperature. With a poorer grade of coal, the air temperature should be lower—probably not over 350 F—in order to avoid clinkering or slagging troubles.

The growth in the use of pulverized fuel has had the effect of increasing the use of air preheaters, as preheated air adapts itself to powdered fuel even more readily than to stokers. In case preheated air is used with pulverized fuel, it is necessary to have a considerable portion of the furnace walls water-cooled. Where preheated air is used with refractory walls, there will be excessive maintenance costs and frequent shutdowns.

The result of preheating the air is not only to reduce the heat loss up the stack, but also to increase the boiler efficiency due both to the smaller amount of heat required to raise the temperature of the air after it enters the furnace, and to the lower combustible in the ash due to the improved furnace conditions with the preheated air.

116. Superheaters.—In the past few years the use of superheated steam with both reciprocating engines and turbines has become very general, and in the modern power station superheaters are almost universally used. The benefits derived are many. The steam remains in a dry condition until all the superheat is lost. The heat lost by the steam, while passing through the piping from the superheater to the place where it is to be used, does not cause condensation, as it is simply superheat which is given up. In reciprocating engines the initial condensation loss is greatly reduced, or entirely eliminated, depending upon the amount of superheat in the steam. In turbines the use of superheated steam is especially advantageous not only on account of increased steam economy but also because the absence of moisture is particularly desirable, as the water coming in contact with the blading at a high velocity has a serious eroding effect.

Two general types of superheaters are used: the *convection* and the *radiant*. The convection superheater is located out of the direct light of the fire and receives its heat by contact with the hot gases flowing to the stack; the radiant superheater is in the direct light of the fire and receives its heat by radiation from the burning fuel. The convection superheater is subdivided into the *independently fired* and the *built-in*, or *attached*, type. The built-in type is much more common, as it costs less to install, is cheaper to operate, and requires less space, since it is located in the boiler setting and receives its heat from the same grates that supply heat to the boiler itself. As its name implies, the independently fired heater is entirely independent of the boiler in its setting and operation. It has not been widely used, its chief

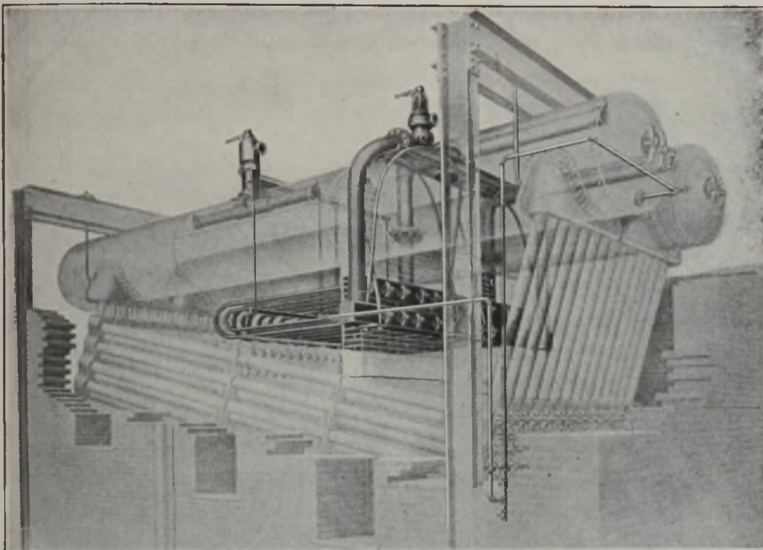


Fig. 120.—Superheating coil in Babcock and Wilcox boiler.

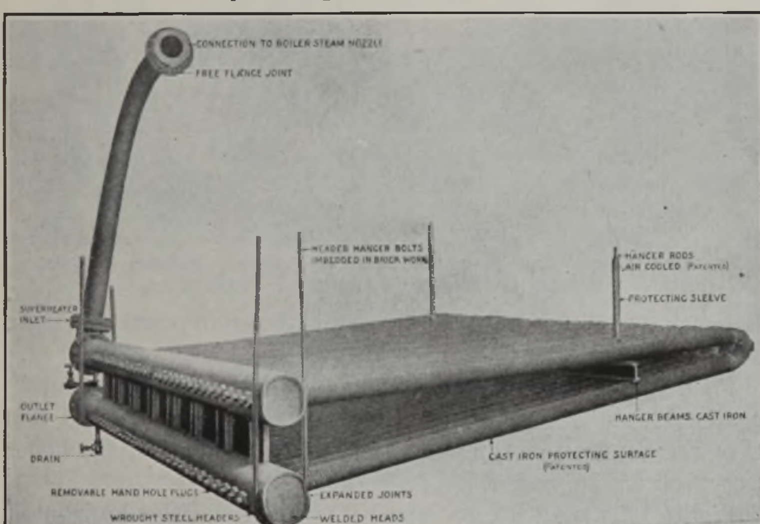


Fig. 121.—Foster-Wheeler convection superheater.

advantage being that, with it, it is possible to obtain higher temperatures than with the built-in type.

The ordinary form of the built-in, convection superheater is illustrated in Fig. 120. As the figure shows, it is located in the path of the flue gases through the boiler, and absorbs heat from these gases as they pass between and around its tubes. This is a

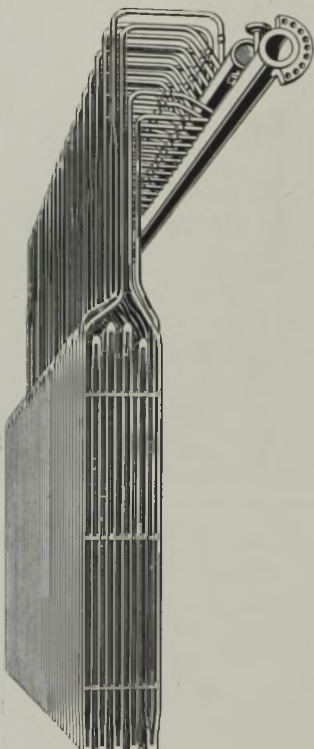


FIG. 122a.—Elesco superheater.

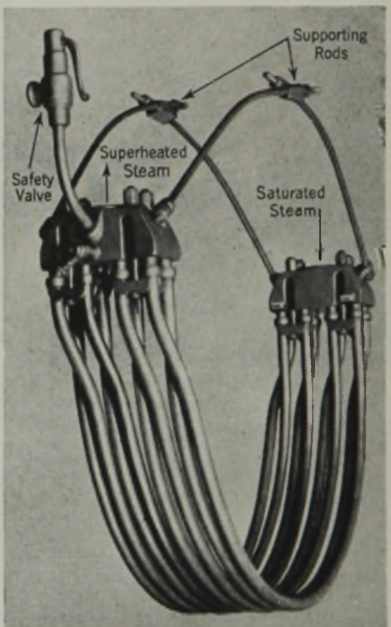


FIG. 122b.—Elesco girth-type superheater.

Babcock and Wilcox superheater and is shown in position in Figs. 36 and 37.

Figure 122*a* shows a multiple-loop, single-pass Elesco superheater. This, like the superheaters shown in Figs. 120 and 121, is of the convection type and is similarly located in the boiler (see Figs. 45, 46, and 47).

Figure 122*b* is an Elesco superheater for use with a horizontal, return-flue, fire-tube boiler. It is placed around the outside of the boiler shell in direct contact with the hot gases.

A recent development has been the incorporation of superheaters as a part of the boiler proper—not merely as a convection bank of tubes, but as involving a separately fired furnace. Such an arrangement is described in Art. 81 and shown in Fig. 48.

The radiant superheater has been developed as a result of the demand for high superheats. One type consists of flat-faced cast, or forged, steel sections assembled so as to form a practically continuous flat metal surface in one or more walls of the furnace

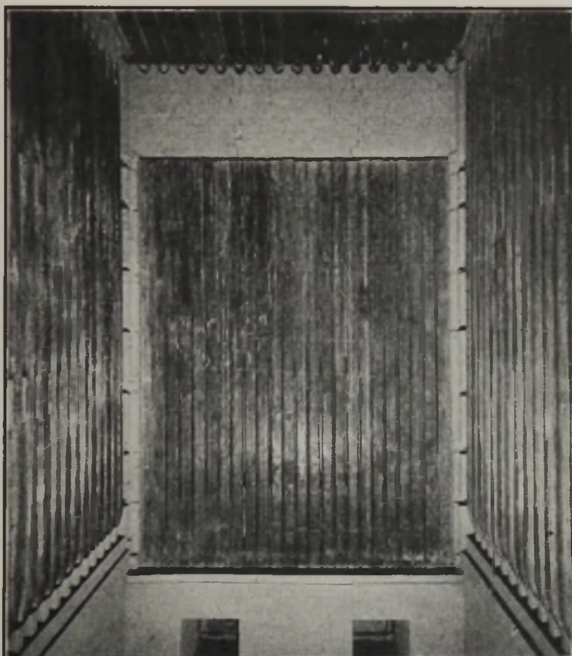


FIG. 123.—Interior of furnace with Foster-Wheeler radiant superheater with flat-faced elements forming rear wall, and Foster-Wheeler water-cooled side walls.

(Fig. 123). This surface, being exposed to the fire, absorbs heat by direct radiation.

In a later development of the radiant superheater, round tubes widely spaced both from each other and from the furnace wall behind them have been substituted for the flat-faced elements. The round-tube type is lighter in construction than the earlier type and as the tubes are

. . . placed 8 in. or more away from the furnace wall and spaced on wide centers they receive heat on all sides—at the front by radiation from the furnace and at the back by radiation from the hot brickwork behind them. A considerable amount of heat is also absorbed by convection from furnace gases which engage with the superheating surface on all sides.

Figure 124 shows a round-tube, high-pressure radiant superheater installed in a furnace of 2,850 hp operating at a pressure of 1,400 psi and raising the temperature of 250,000 lb of steam an

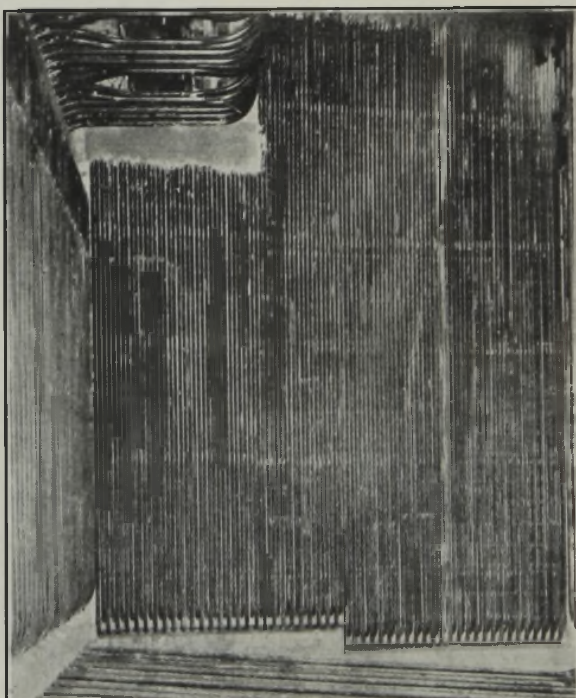


FIG. 124.—High-pressure round-tube Foster-Wheeler radiant superheater.

hour to 750 F. The tubes are built closer together than in the superheater for normal pressure just described, since it would be impossible to obtain sufficient surface with the tubes widely spaced. The amount of radiant heating is unusually large in this case, since there is no convection superheater in series with the radiant superheater.

When steam at 1,200 psi pressure and 750 F temperature is expanded adiabatically it will become saturated at a pressure of about 300 psi. This means that the economy of a turbine using this steam would be impaired considerably from that point down to back pressure; therefore, the majority of high-pressure plants now being built include a reheat cycle. The high-pressure superheated steam is expanded to a pressure where most of the superheat has been used and it is then circulated through a resuperheater and raised to its initial temperature and then led back to the turbines again. In this way the efficiencies of the machines are kept higher than would be possible if the steam were expanded through them without superheating.

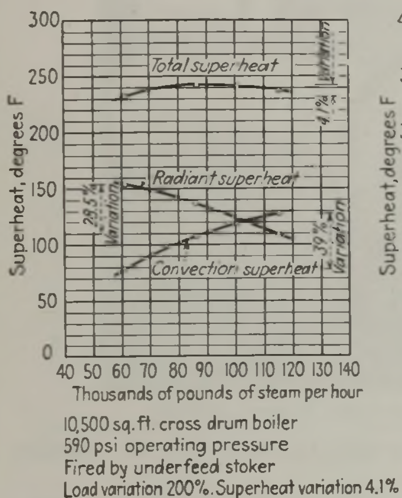


FIG. 125.—Temperature curves.

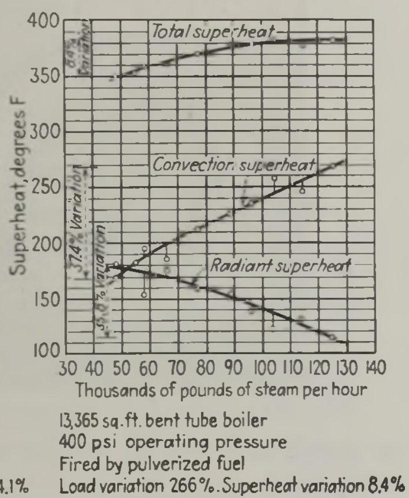


FIG. 126.—Temperature curves.

In some plants both the original superheating and the reheating take place in radiant superheaters, but recent experience has shown that in many cases the most efficient operation is obtained in high-pressure reheat cycles when the high-pressure steam is superheated in radiant superheaters and the reheating occurs in convection superheaters.

The outstanding advantage of combination superheaters is the flat, final steam temperature curve. Typical superheat rating curves of radiant and convection superheaters have opposite characteristics, the temperature of the steam from a convection superheater increasing with an increase in boiler capacity; with a radiant superheater, the superheat decreases as the boiler output increases. When radiant and convection superheaters are combined their opposite characteristics compensate each other to give a flat final curve (see Figs. 125 and 126). The final steam temperature from a combination superheater is more constant than can be obtained from either a radiant or convection superheater under the same operating conditions.

The flat curve permits the use of maximum superheat over a wide range of loads to provide high over-all power generating efficiency. Indirect advantages are reduced expansion strains

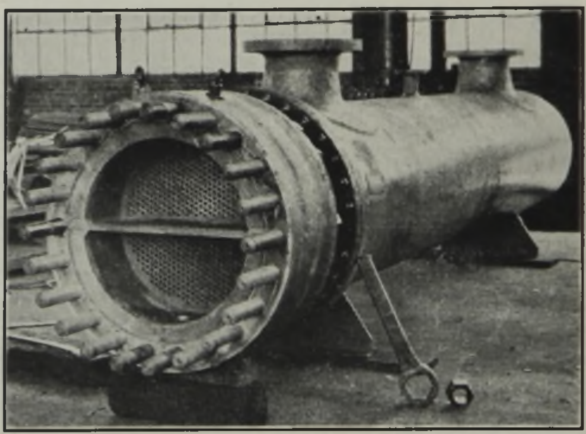


Fig. 127.—Foster-Wheeler steam-to-steam reheater.

on piping and reduced turbine maintenance, due to constant temperature.

A third type is the *steam-to-steam reheat superheater* (Fig. 127). In this particular piece of equipment, which is installed as a part of the high-pressure steam generator at the San Antonio Public Service Company's plant,

. . . the high-pressure steam header of the reheater is built for a working pressure of 1,450 psi, and is made from a solid steel plate machined out to provide space for the high-pressure steam entering the tubes and the condensate leaving the tubes. The heating surface consists of 600 tubes having a total surface of 2,060 sq ft. The high-pressure steam will make two passes through these tubes. Steam exhausted from the high-pressure turbine at the rate of 196,000 lb per hr and at a pressure of 190 psi will pass through the shell flowing across the tube bundle several times. This steam will be superheated about 60 deg to a final temperature of 550 F. Condensate from the high-pressure steam header of the reheater drains by gravity into the boiler drum.

Figure 128 is a diagram of the complete unit at the San Antonio plant, and Fig. 129 that of the plant at the Philo station of the

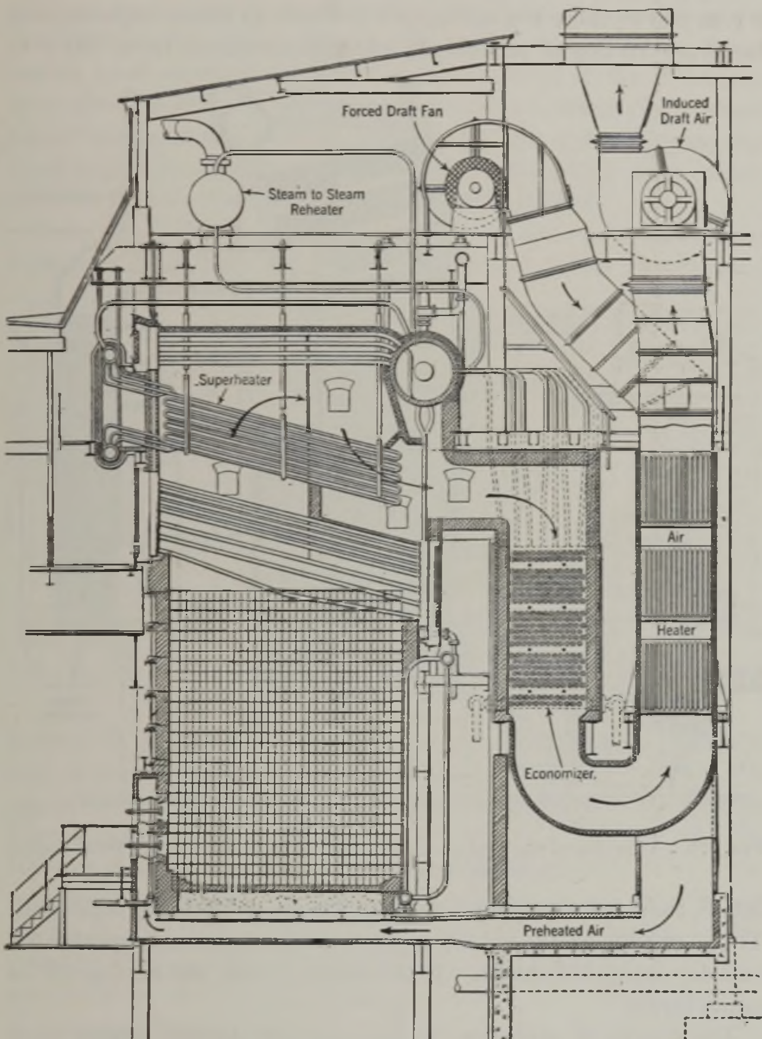


FIG. 128.—Diagram of the steam-generating unit at San Antonio Public Service Company's Station B.

Ohio Power Company. In this latter plant "each unit contains 21,550 sq ft of surface in the boiler and 58,600 sq ft of heating surface in the air heater. Ash is tapped from the water-cooled furnace in liquid form."

Experiments have shown that when steam is superheated from 0 F to 100 F, there is a saving of 1 per cent in steam consumption for every 10 deg of suoerheat; when superheated from 100 F to

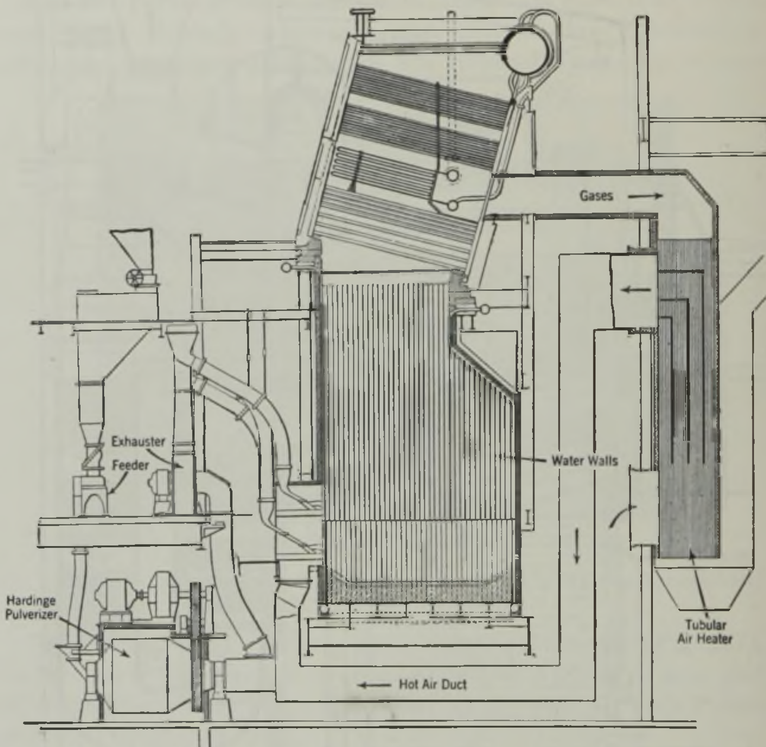


Fig. 129.—Diagram of the new steam-generating equipment at the Philo station of the Ohio Power Company.

200 F, there is a saving of 1 per cent for every 12 deg of superheat. These results are based on a comparison between superheated and dry saturated steam. If the steam is wet, the saving will be much larger.

The degree of superheat to be used will depend largely upon the conditions. It has been found that for engines using slide valves, or Corliss valves, the final temperature of the steam should not exceed about 475 F. Engines of the piston-valve, or poppet-valve, type may utilize much higher degrees of superheat with corresponding increase in economy. In these cases steam having a temperature of from 500 F to 600 F may be used to advantage. Where steam turbines are installed, present practice tends toward much higher pressures and superheat, as described in Art. 83 on High-pressure Boilers. Here, as stated before, until recently the limiting temperature of the steam has been about 750 F, on account of the deteriorating effect which higher temperatures have on the metal of the pipes, fittings, and apparatus. Recent developments in new alloys have made possible the use of temperatures of 900 F to 1000 F.

It is impossible to state accurately the saving in water consumption that may be expected through using superheated

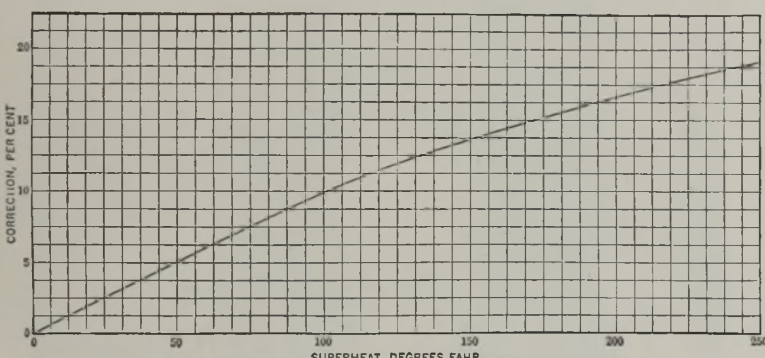


FIG. 130.—Correction or saving in water rate of steam turbines when superheated instead of saturated steam is used.

instead of saturated steam with reciprocating engines. In general, it may be said that it varies from 4 to 8 per cent for 100 deg of superheat in large economical plants, where the engines have a high ratio of expansion, to from 15 to 30 per cent for 100 deg of superheat in simple engines and steam pumps. The corresponding saving in the water rate of steam turbines is shown in Fig. 130. The saving in steam consumption anticipated by the use of superheated steam must be balanced against a consideration of the pipes, fittings, etc., the first cost and upkeep of the superheater, the efficiency of the superheater, and the nature of the service. The cost of superheaters varies with the type and with the degree of superheat desired.

117. Desuperheaters.—Most modern industrial, as well as power, plants superheat all the heat produced, but in many industrial processes saturated steam, or steam with a pressure and temperature lower than that at which it is generated, is required. Under these conditions it becomes necessary to extract some of the heat from this superheated steam, or in other

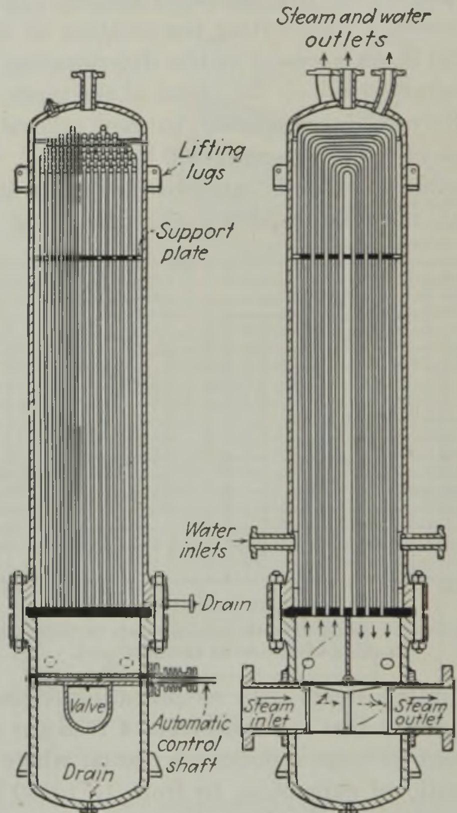


Fig. 131.—Babcock and Wilcox surface-type desuperheater.

words, to desuperheat as much of it as is required for the process or equipment in question.

Figure 131 shows a Babcock and Wilcox *surface-type desuperheater*. In this, boiler water is circulated around the tubes while enough of the steam is passed through them to secure the desired temperature reduction.

A second type of desuperheater is the *spray* or *atomizing* type in which superheated steam instantly vaporizes the water that is sprayed into it. This operation involves the evaporation of a sufficient amount of water to accomplish the reduction of the superheat in any predetermined amount. The additional steam formed by the evaporation of the water offsets the reduction in superheat.

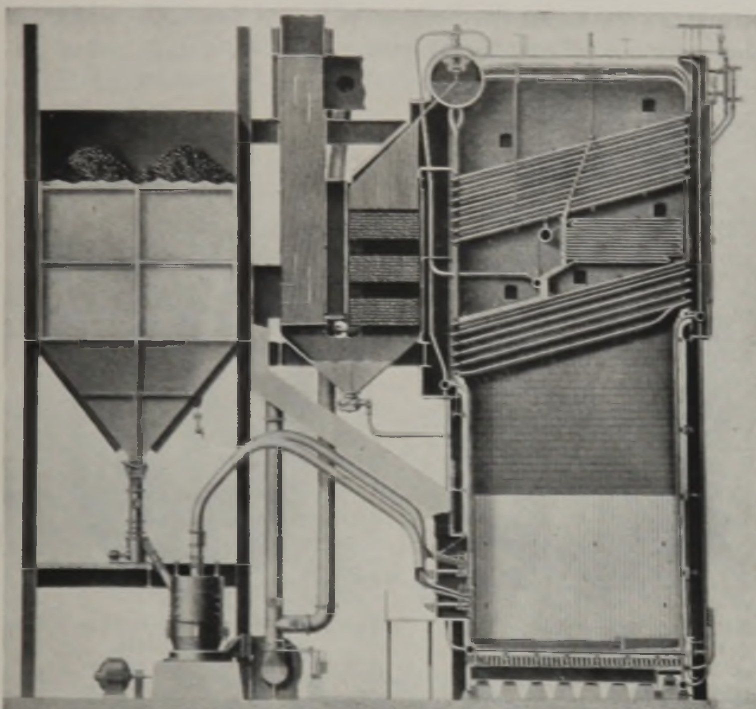


Fig. 132.—Complete Babcock and Wilcox boiler unit including boiler, superheater, economizer, and air heater with Bailey slag-tap furnace fired by a Babcock and Wilcox direct-firing pulverized-coal system.

118. Steam Separators.—Steam flowing from the boiler to the engine or turbine, unless it is superheated, will carry along with it a certain amount of water, or entrained moisture. This water comes from priming in the boiler and condensation in the steam mains. Owing to the small clearance in reciprocating engines, water entering with the steam may knock out the cylinder head and will cause increased initial condensation. In steam turbines it will cause erosion of the blades; in both engine and turbine it will result in increased steam consumption, or lowered economy.

Steam separators are devices placed in the steam line just ahead of the engine or turbine to remove, or separate, the water from the steam. These separators operate on the principle of centrifugal force, reverse flow, baffle plates, or mesh separation. The first two types, as the names indicate, function by suddenly changing or reversing the direction of flow of steam, thus throwing the water out by centrifugal force.

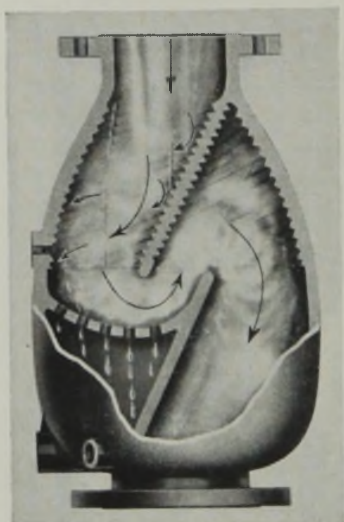


Fig. 133.—Wright-Austin Type A vertical separator.

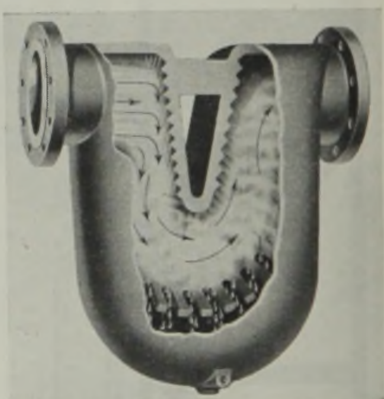


Fig. 134.—Wright-Austin Type B horizontal separator.

The Wright-Austin Type A separator (Fig. 133) "is designed for installation in vertical steam lines, usually just above the throttle of the engine. It is of the baffle type. In it the baffle plate is not set at right angles to the entering steam current, but is set so that when the incoming steam is impinged against it and rebounds to the opposite wall of the separator, the particles of moisture are driven down the deep, slanting corrugations on walls and baffle, entirely out of the course of the steam, and into the well below where they are drained off. Passing around the lower edge of the baffle, the flow of the steam is completely reversed by a quick, sharp turn upward. This sudden reversal whips out the final trace of moisture, which continues downward and is caught in the open baffle over the well below."

The type B horizontal separator (Fig. 134) is designed especially for use where there "is limited head or side room. No part of the separator projects beyond the outside diameter of the pipe flange, except the body which hangs directly underneath. It can be installed in a horizontal pipe line that is tight up against the ceiling, or close up to parallel pipes or wall on either side.

"The flow of steam may be passed through the Type B separator in either direction with equal efficiency. Upon striking the

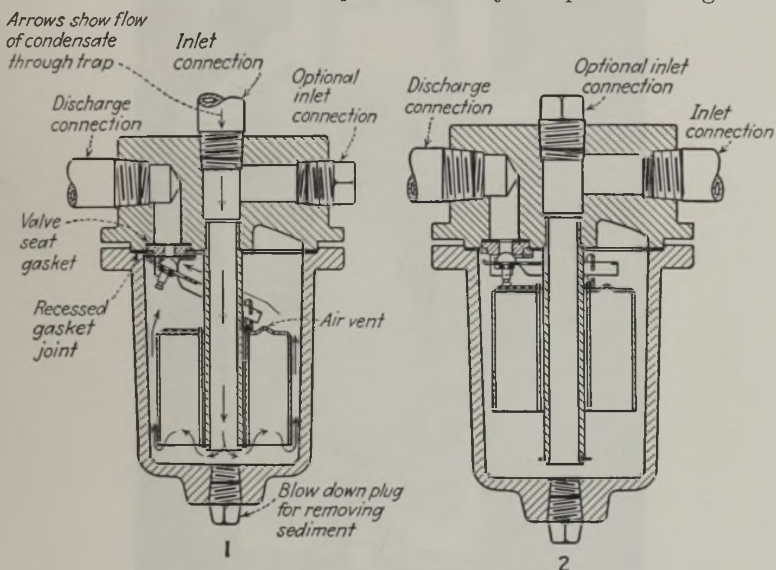


FIG. 135.—Anderson Super-Silvertop trap.

inclined baffle, the condensation is driven down the deep, slanting corrugations out of the path of the steam current. Any moisture not caught by the upper baffle and by the inner wall grooves is finally separated by additional baffles located just over the well or receiver of the separator.

"Every separator should be drained automatically by a steam trap. The efficiency and even the usefulness of the separator depends upon the instant removal of the water collected. A good automatic steam trap is the best practical method of accomplishing this. The automatic draining is necessary because the amount of water collected in a separator varies greatly, and must not be allowed to accumulate. Drainage by hand is not successful, and should not be attempted."

119. Steam Traps.—In order to remove the water collected in a separator without loss of any steam, recourse is had to what is known as a *steam trap*, the purpose of which is to draw condensation or air from steam-using equipment and to vent air, at the same time preventing the escape of live steam.

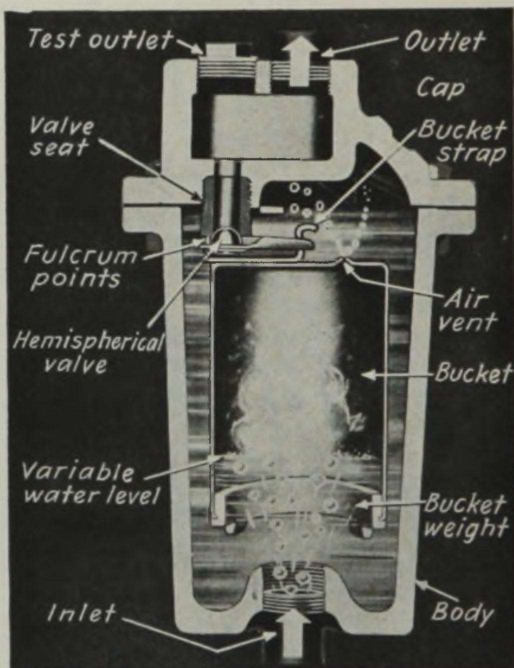
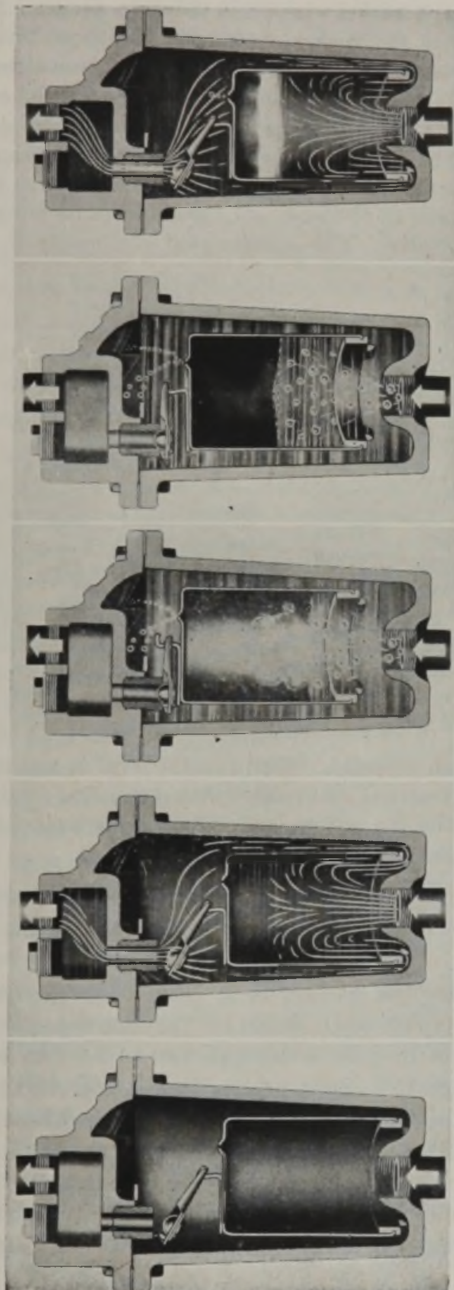


Fig. 136.—Armstrong Steam Trap.

Figure 135 shows two views of an Anderson Super-Silvertop trap. When the steam is shut off, the bucket is down and the outlet valve is open. As the steam is turned on, water enters the trap filling the body and flowing out through the trap valve.

The bucket remains in the down position with the valve open until gases, which are lighter than water, rise to the top of the bucket displacing the water therein until the bucket becomes buoyant at which time it floats and closes the valve. If the gas in the bucket is air, it will pass through the small vent hole in the bucket and displace the water in the case. Without this water, the bucket then drops to the bottom. In dropping it opens the valve and the accumulated air escapes through the valve orifice. This action continues until all air has been eliminated.



1 *Steam off. Bucket down. Valve open. Inlet bottom. Outlet top. When steam* 2 *is turned on, water fills body. Excess water escapes through orifice until* 3 *steam reaches trap and fills bucket which floats and closes orifice. When* 4 *steam condenses, water enters. Bucket loses buoyancy, pulls on lever.* 5 *When weight of bucket times leverage equals pressure on valve, trap opens.*

FIG. 137.—Five views of Armstrong Steam Trap.

When steam enters a bucket which is in the down position, this steam displaces the water in the bucket, causing it to become buoyant and rise. This steam condenses a little at a time as it seeps through the vent hole in the bucket. But enough steam constantly enters the bucket to just equal the condensed steam. Thus the trap will remain closed until the supply of steam to the trap is shut off due to an accumulation of condensate in the inlet line. When this happens the steam in the bucket continues to condense, the bucket loses its buoyancy and drops, opening the valve. The accumulated condensate is then dis-

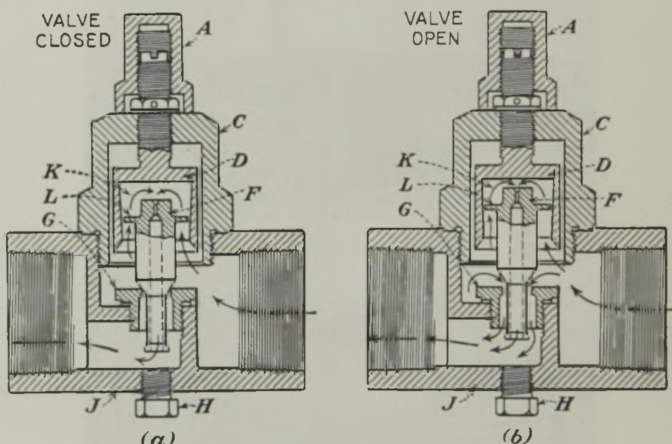


Fig. 138.—Yarway Impulse Steam Trap.

charged by the steam pressure. With the discharge of this condensate the passages to the trap are again clear, thus permitting steam to enter the trap, be caught by the bucket, again causing the bucket to become buoyant and rise, shutting off the valve.

The captions under the various views of the Armstrong Steam Trap (Fig. 137) explain clearly the operation of this piece of equipment.

The Yarway Impulse Steam Trap (Fig. 138) operates on an entirely new principle: when fluids of varying densities (due to changes in temperature) flow through two orifices in series, the changing characteristics cause pressure variations in the control chamber *K* located between the two orifices. These pressure variations are utilized to open and close the valve *F*.

The first orifice is the clearance around the control disk *L*; the second orifice is the small hole through the top of the valve *F*. A small portion of the condensate entering the trap is by-passed through the first orifice, control chamber, and second orifice into the discharge side of the trap. This is called the *control flow.*

The pressure tending to lift valve F and open the trap for discharging the condensate is the line pressure in the trap inlet; the pressure tending to close the valve after the condensate has been discharged is the pressure in control chamber K. When the valve is open and discharging cold condensate and entrained air (see Fig. 138*b*), it remains in this position until the condensate nearly reaches steam temperature. At this point the control flow of hot condensate which is passing through the control chamber as well as through the main valve seat flashes into steam as it leaves the second orifice. This chokes the orifice, causing an increase in the pressure in the control chamber, thereby pushing the valve down and closing the trap so that steam cannot escape (see Fig. 138*a*). In other words, high-temperature condensate which is near steam temperature causes heavy flashing which closes the valve to prevent steam discharge.

The trap adapts itself to light, medium, or heavy condensate loads by opening and closing the valve F as necessary, to get rid of the condensate.

120. Draft.—*Draft* is the difference between the pressure of the gas within a boiler unit and that of the air outside.

In order that the proper amount of air for combustion shall enter the furnace of any power plant, draft is necessary. It may be produced by natural or mechanical means.

121. Natural Draft.—Natural draft is obtained by the use of a chimney; in a plant depending upon this method of bringing the air for combustion into the furnace and through the bed of fuel, the operation of the plant depends upon the draft and capacity of the chimney.

The draft in a chimney is produced by the difference in weight between the column of hot gases inside the chimney and a column of gases of the same dimensions outside the chimney. The hot gases, being light, are forced up the chimney by the cold gases coming through the grates.

The height of the chimney, then, determines the intensity of the draft. This intensity of draft, or difference in pressure, is usually measured in inches of water, and for a given height of stack may be found from the following equation:

$$F' = Hw^{\circ}\left(1 - \frac{T^{\circ}}{T'}\right), \tag{1}$$

where F' = the force of the draft in pounds per square foot;
H = the height of the chimney above the grate in feet;
T° = the absolute temperature of the gases outside the chimney;
T' = the absolute temperature of the gases inside the chimney;
w° = the weight of 1 cu ft of air at a temperature T°.

To reduce Eq. (1) to inches of water it must be multiplied by 0.192.[1] Hence the force of the draft in inches of water,

$$F = 0.192Hw^{\circ}\left(1 - \frac{T^{\circ}}{T'}\right). \tag{2}$$

As shown in Eq. (2), the intensity of the draft is determined by the height of the chimney and the temperatures inside and outside of it.

122. Chimney Capacity.—The capacity of a chimney is the quantity of gases that it will pass per hour; upon the capacity depends the number of pounds of coal that the plant will burn.

If A = area of the chimney in square feet, and W° = the total weight of the gases passing up the chimney per second, then

$$W^{\circ} = 0.30A\sqrt{H}, \text{ in pounds per second.} \tag{3}$$

or, in pounds per hour,

$$W_1^{\circ} = 3{,}600 \times 0.3A\sqrt{H}. \tag{4}$$

This assumes the efficiency of a chimney to be 1, but experience shows the average efficiency to be about 35 per cent, so that the actual weight of air passed per hour is

$$W_a^{\circ} = 3{,}600 \times 0.35 \times 0.3A\sqrt{H} = 378A\sqrt{H}. \tag{5}$$

[1] A column of water 1 sq ft in cross section and 1 ft high weighs 62.4 lb.

A column of water 1 sq ft in cross section and 1 in. high weighs $\frac{62.4}{12} = 5.2$ lb.

A column of water 1 sq ft in cross section and $\frac{1}{5.2} = 0.192$ in. high weighs 1 lb.

If each pound of coal is supplied with 24 lb of air to burn it, and each boiler horsepower uses about 5 lb of coal, the boiler horsepower of a chimney is

$$\text{Boiler hp} = \frac{378}{24 \times 5} A \sqrt{H} = 3.15A \sqrt{H}. \qquad (6)$$

Various authors give values of the constant in this expression varying from 3.5 to 3.0.

123. Height of a Chimney.—The height of a chimney is always measured from the level of the grate and, in any given case, depends upon the amount of frictional resistance to flow offered by the gas passages through the unit and upon the acceleration desired. It depends too upon the kind of fuel that is to be burned under the boiler, upon the nuisance that the gases and fly-ash may cause in the neighborhood, and upon local ordinances.

The following table gives the minimum height of chimney for various kinds of fuels:

Table XXVII.—Chimney Heights

	Feet
For straw or wood	35
For bituminous lump, free-burning	100
For ordinary slack	100
For ordinary bituminous coal	115
For small slack or anthracite	125
For anthracite pea coal	150

The height of the chimney should not be too short for its diameter. A very large diameter of chimney in proportion to the height may show reduced capacity. As an example, a chimney 100 ft high should not exceed 6.5 ft in diameter. In general, the inside diameter of a chimney should not exceed 8 per cent of its height. The size of stack necessary for a given installation can be determined by means of empirical formulas available in engineering handbooks.

124. Materials Used.—Brick, hollow tile, or concrete is used in building chimneys where permanency is desired and few changes are expected. The life of a brick chimney is probably 40 or 50 years. In plants where the station is not expected to remain more than 20 or 25 years, without extensive changes, the expense of a brick or concrete chimney is not warranted.

For temporary use, the unlined sheet-steel chimney is very commonly used. It is necessary to brace these chimneys with steel guy wires. The life of these chimneys is short, at the best not more than 10 years, and where the coal contains much sulphur, not more than 5 years.

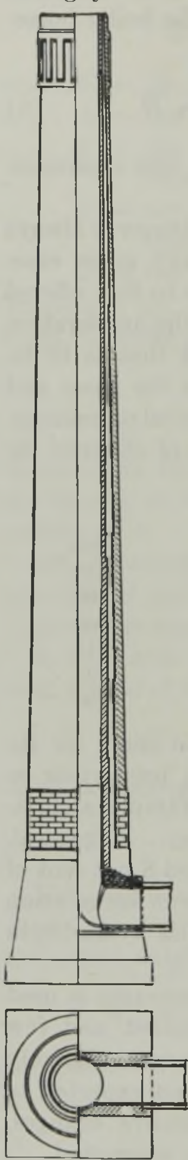


FIG. 139.—Brick chimney.

125. Brick, Tile, and Concrete Chimneys.—Brick chimneys, as shown in Fig. 139, are built in two parts, an outer shell and an inner shell, usually lined with firebrick which forms a flue for the burned gases. There should be an air space between the outer and the inner shells, so that the inner shell is free to expand. Brick chimneys are expensive to erect but very permanent in character.

The radial brick chimney is constructed of hollow tile and has no lining. These chimneys are much lighter than the solid brick chimney. They are much less expensive than the brick and cost but little more than a self-sustaining steel chimney.

Reinforced-concrete stacks have the advantage over brick stacks of an absence of joints, of lighter weight, and of economy in space required.

Care should be taken in investigating the ground which is to support a chimney, as unequal or excessive settlement may endanger the stack.

126. Steel Chimneys.—Steel chimneys of the self-sustaining type are built of boiler plates riveted together. They are supported on ample foundations, to which they are bolted by very heavy anchor bolts. The pressure of the wind against the chimney is carried to the foundation by these bolts, and the foundation must be of sufficient size and weight to prevent overturning. Chimneys of this type are lined with firebrick usually for their full length. Steel stacks must be kept painted to avoid rust and corrosion.

127. Mechanical Draft.—In many cases conditions will not permit of the construction of a tall chimney;

in other cases, the draft required is more than the ordinary chimney will give. It is then necessary to resort to some form of forced or mechanical draft.

Mechanical draft is entirely independent of the temperature inside, or outside, of the chimney. Where economizers, or air preheaters, are used, the temperature of the gases in the chimney may be lowered to such an extent, and the resistance to the flow of the gases offered by these auxiliaries so great, as to require mechanical draft in order to avoid chimneys of excessive size or cost.

128. Systems of Mechanical Draft.—Mechanical draft may be classified as *forced* draft or *induced* draft, depending upon the system used to produce it. In the case of *forced* draft the ashpit must be closed and the air is forced into it and through the fire. One of two methods may be used to force the air into the ashpit: (1) a steam jet; (2) a fan. Most stoker-fired plants use forced draft.

In a pulverized-fuel-fired unit, it is necessary to use forced draft, because of the manner of burning coal.

With *induced* draft a fan is placed in the breeching, or, as in locomotives, the cylinders exhaust directly into the stack, and air is drawn through the fire. The action in this case is analogous to the action of a chimney. Induced draft is generally used with economizers and air preheaters and the fan is placed between them and the stack.

In many cases both forced and induced draft are used, the air being supplied under stokers or grates, or through pulverized-fuel burners under pressure, and the induced-draft fan being used to remove the products of combustion and discharge them into the chimney. Sometimes there is natural draft enough so that the induced-draft equipment is not necessary. When the combination of forced and induced draft is such as to result in approximately atmospheric pressure above the grates, the draft is said to be *balanced*.

Under ordinary conditions the rate of combustion may be taken as from 15 lb to 50 lb or 60 lb of coal per sq ft of projected grate area per hr with mechanical draft. With modern stokers the air required per pound of coal may be lower than 15 lb. The forced-draft pressures for underfeed stokers may run as high as 10 in. of water and, for traveling-grate stokers, may be 5 in. or

6 in. of water. The induced draft may run up as high as 10 in. of water where exceptionally high percentage ratings are used. In large units automatic combustion control, operated by steam pressure and quantity, is used. This control in some units operates all fan speeds and coal feeds, in others only the induced-draft fan.

PROBLEMS

1. The barometer reading is 29.3 in. Hg, and a boiler is operating at a vacuum of 18 in. Hg. (*a*) What is the absolute pressure per square inch in the boiler? (*b*) What would be the temperature of the dry steam produced?

2. A boiler is producing steam having a quality of 98.5 per cent and a temperature of 182.9 F. Barometer reading is 29.3 in. Hg. Under what vacuum (inches of Hg) is the boiler operating?

3. A boiler evaporates 8 lb of water per lb of coal; pressure, 100 psi; feed-water temperature, 100 F. (*a*) What will it evaporate if the pressure is 80 psi and feed-water temperature 200 F? (*b*) What will it evaporate from and at 212 F?

4. (*a*) One hundred pounds of coal containing 13,000 Btu per lb will evaporate how many pounds of water at 200 F into steam at a pressure of 100 psi? (*b*) What will it evaporate from and at 212 F? The efficiency of the boiler, furnace, and grate is 70 per cent.

5. A coal containing C, 80 per cent; H, 5 per cent; O, 4 per cent is burned in a boiler where the efficiency of boiler, furnace, and grate combined is 70 per cent. If the boiler is developing 300 hp, (*a*) how many pounds of water are being evaporated per hour from a feed of 100 F into steam at a pressure of 100 psi and containing 2 per cent moisture. (*b*) How many pounds of coal are burned per hour?

6. If 40 per cent of the heat of combustion of coal containing 12,750 Btu per lb is lost, how many pounds of coal will be required to evaporate 5,650 lb of water from an initial temperature of 130 F and under a pressure of 80 psi?

7. If the temperature of the boiler room is 70 F and the temperature of the stack gases is 500 F and 2.75 times as much air is used as is theoretically required, what percentage of heat is lost up the stack, if the coal contains C, 75 per cent; H, 5 per cent; O, 4 per cent?

8. A boiler is reported to evaporate 12.5 lb of water per lb of coal. Coal contains 13,000 Btu and uses 24 lb of air per lb to burn it. The temperature of the boiler room is 70 F, and of the stack gases, 550 F; steam pressure, 100 psi; feed-water temperature, 70 F. (*a*) Would this result be possible? (*b*) If not, how many pounds of water could the boiler evaporate per pound of coal?

9. An engine uses 30 lb of steam per ihp-hr. The feed-water temperature is 120 F; steam pressure, 120 psi. The boiler evaporates 9 lb of water per lb of coal. How many pounds of coal are required per indicated horsepower per hour?

10. A boiler evaporates 7.5 lb of water per lb of coal. The steam pressure is 150 psi; feed-water temperature, 200 F. Coal costs $2.50 per ton. What is the cost to evaporate 1,000 lb of water from and at 212 F?

11. A 72-in. return tubular boiler 18 ft long has seventy 4-in. tubes. Find the heating surface and rated boiler horsepower.

12. A certain power plant develops 2,500 boiler hp. The chimney is 150 ft high. What should be its inside diameter?

13. A power plant develops 300 boiler hp. It has a stack 100 ft high and 4 ft in diameter. If the horsepower of the plant is increased to 450, what will have to be added to the height of the stack to carry the additional load?

14. A plant burns 1,500 lb of coal per hr. The height of the stack is 130 ft. The temperature of the boiler room is 70 F, and of the stack gases, 500 F, and 24 lb of air are used to burn 1 lb of coal. Coal contains 12,000 Btu per lb. (*a*) What would be the area of the stack? (*b*) What percentage of the heat is lost up the stack? (*c*) What is the pressure of the draft in tenths of inches of water?

15. A heating boiler is evaporating 2,000 lb of water per hr from a feed temperature of 166.5 F into steam at 96 per cent quality and under a vacuum of 6 in. Hg. The barometer reading is 28.6 in. Hg, the flue-gas temperature is 490 F; the specific heat of the flue gas is 0.24. 230 cu ft of air at 60 F and atmospheric pressure are supplied for each pound of coal burned. Evaporation is 8 lb of water per lb of coal. (*a*) What is the factor of evaporation? (*b*) What is the steam temperature? (*c*) What is the equivalent evaporation from it at 212 F? (*d*) What boiler horsepower is being developed? (*e*) What is the stack-gas loss in Btu per pound of coal?

16. The exhaust steam from a boiler feed pump contains 10 per cent moisture at 15 psia and goes into an open feed-water heater where it mingles with the incoming water at 70 F. The feed pump uses 3.0 per cent of the steam generated by the boiler. What will be the temperature of the water leaving the heater if all of the feed water to the boiler is taken from the heater?

17. Water at a temperature of 40 F enters on open feed-water heater where it is heated with exhaust steam at a pressure of 14.7 psia, and at a quality of 90 per cent. This heater is required to supply feed water at 212 F to boilers delivering dry saturated steam at a pressure of 200 psia and developing 1,000 boiler hp. How many pounds of exhaust steam will the heater require per hour?

18. A closed feed-water heater is supplied with 10,000 lb per hr of steam bled from the tenth stage of a steam turbine at 20 psia and 230 F, and also 1,000 lb per hr of exhaust steam from a small turbine at 20 psia and containing 5 per cent moisture. The steam is all condensed in the heater and leaves as water at 200 F. The feed water enters the heater at 190 F and leaves at 222 F. How many pounds of water will be heated per hour?

19. Water enters a closed feed-water heater at 60 F and leaves at 200 F. It then enters an economizer where it is heated to 320 F. From the economizer the water enters a boiler where it is converted into steam of 98 per cent quality at 275 psia. The steam from the boiler passes into a superheater where it is superheated at constant pressure to a total temperature of

600 F. Of the total amount of heat added from the time the water enters the heater until it leaves the superheater as steam, what per cent is furnished (*a*) by the heater; (*b*) by the economizer; (*c*) by the boiler; and (*d*) by the superheater?

20. A return fire-tube boiler is 60 in. in diameter, 16 ft long, and has 52 4-in. tubes. It evaporates 4,000 lb of water per hr. The steam pressure is 100 psi; feed-water temperature, 150 F. Is it working above or below its rated horsepower and how much?

21. A boiler plant burns 1,500 lb of coal per hr containing C, 85 per cent; H, 5 per cent; O, 4 per cent. The temperature of feed water is 150 F; steam pressure in boiler, 100 psi. The throttling calorimeter shows a temperature on the lower thermometer of 230 F at atmospheric pressure. The efficiency of boiler, furnace, and grate is 70 per cent. How many horsepower is the boiler actually developing?

22. A 30,000-kw steam turbine and auxiliaries require 12 lb of steam per kw-hr. The initial pressure is 250 psi; barometer, 29.53 in. Hg; superheat, 250 degrees; feed-water temperature, 180 F. (*a*) Find the boiler horsepower necessary to furnish turbine and auxiliaries with steam. (*b*) If the boilers are operated at 250 per cent of rating, determine the rated boiler horsepower. (*c*) The coal used has a heating value of 13,250 Btu per lb. The efficiency of boiler, furnace, and grate is 68.4 per cent. If coal costs $5 per ton, and the plant operates 24 hr per day, 365 days in a year, find the annual cost of fuel.

23. A water-tube boiler containing 10,000 sq ft of heating surface is operating under the following conditions: barometer reading, 28.53 in. Hg; steam pressure, 211 psi; steam temperature, 600 F; feed-water temperature, 180 F; actual evaporation under these conditions, 50,000 lb of steam per hr. Find (*a*) the rated boiler horsepower; (*b*) the actual boiler horsepower developed; and (*c*) the per cent of rating the boiler is developing. (*d*) If the same amount of heat is absorbed per hour, how many pounds of steam would the boiler deliver per hour if the conditions were changed so that it operated at a vacuum of 10.20 in. Hg; feed-water temperature the same (180 F); quality of steam, 100 per cent; barometer the same (28.53 in. Hg)? (*e*) What would be the actual boiler horsepower developed under the changed condition?

24. A boiler has an over-all efficiency of 70 per cent, when burning coal at the rate of 500 lb per hr. The coal has a heating value of 12,000 Btu per lb as fired and contains 6 per cent moisture. The boiler pressure is 150 psia and the temperature of the steam leaving is 460 F. The feed water enters the boiler at a temperature of 202 F. Find (*a*) the pounds of steam discharged per hour; (*b*) the boiler horsepower developed; (*c*) the equivalent evaporation per pound of dry coal; (*d*) the fuel cost for evaporating 1,000 lb of steam from and at 212 F, if coal costs $4 a ton.

25. A steam-boiler unit (including a superheater) has an over-all efficiency of 75 per cent. Coal is supplied to this boiler at the rate of 2,000 lb per hr. The coal contains 6 per cent moisture and has a heating value of 12,000 Btu per lb of dry coal. The boiler pressure is 200 psia and the temperature of the steam discharged is 460 F. The feed-water temperature is 200 F. (*a*)

How many pounds of steam are discharged per hour? (*b*) How many pounds of water are evaporated per pound of dry coal fired? (*c*) If the coal is purchased at $4 per ton (delivered), find the coal cost per 1,000 lb of steam.

26. A boiler evaporates 7.5 lb of water per lb of coal. Coal contains 13,000 Btu. The steam pressure is 100 psi; feed-water temperature, 150 F. What is the combined efficiency of the boiler, furnace, and grate?

27. A boiler is supplied with 10,000 lb of water per hr at a temperature of 100 F. The steam pressure is 135.3 psi; barometer, 29.92 in. Hg. The quality of the steam was determined by a throttling calorimeter, the temperature of the steam after passing the orifice being 270 F, and the pressure atmospheric. Dry coal fired per hour, 1,250 lb; heating value, 13,000 Btu per lb. Determine (*a*) the boiler horsepower being developed; and (*b*) the efficiency of boiler, furnace, and grate.

28. A boiler equipped with a superheater delivers 300,000 lb of steam per hr at a pressure of 450 psia and at 700 F. Feed water enters at a temperature of 180 F. The weight of coal fired per hour is 33,000 lb and the heating value is 13,750 Btu per lb. Find (*a*) the boiler horsepower developed; (*b*) the efficiency of the boiler, furnace, and grate; (*c*) the diameter of steam main necessary for the boiler if the steam velocity is to be limited to 10,000 fpm.

29. A boiler uses 1 lb of dry coal containing 13,000 Btu to evaporate 9 lb of water. The steam pressure is 100 psi; feed-water temperature, 100 F. (*a*) What is the efficiency of the boiler plant? (*b*) What will be the efficiency of the plant if a heater is added which heats the feed water to 200 F? (*c*) What will be the evaporation per pound of coal after the feed-water heater is installed?

30. A boiler evaporates 9 lb of water per lb of coal fired. The feed-water temperature is 70 F; steam pressure, 150 psi. Coal as fired contains 3 per cent moisture. Dry coal contains 14,000 Btu per lb and shows 6 per cent ash by analysis. Twelve per cent of the coal fired is taken from the ashpit in the form of ash and refuse. (*a*) What is the efficiency of the boiler, furnace, and grates combined? (*b*) What is the efficiency of the boiler and furnace?

31. A boiler burns coal containing 13,500 Btu per lb dry coal, 5 per cent moisture, and 3 per cent ash by analysis. Ten per cent ash and refuse is taken from the ashpit. The steam pressure is 150 psi; feed-water temperature, 180 F; the steam contains 2 per cent moisture. The amount of water fed to the boiler per pound of coal fired is 8 lb. (*a*) What is the evaporation from and at 212 F per pound of dry coal? (*b*) What is the efficiency of the boiler, furnace, and grate? (*c*) What is the efficiency of the boiler and furnace?

32. A boiler evaporates 10,000 lb of water per hr. The boiler operates under a pressure of 150 psia, the quality of the steam is $98\frac{1}{2}$ per cent, and the feed-water temperature is 100 F. Twelve hundred and fifty pounds of dry coal are fired per hour, the coal having a heating value of 13,000 Btu per lb. The percentage of ash by analysis is 3 and the weight of ash and refuse taken from the ashpit is 125 lb per hr. Determine (*a*) the equivalent evaporation per pound of dry coal; (*b*) the boiler horsepower being developed; (*c*) the efficiency of boiler, furnace, and grate; and (*d*) the efficiency of boiler and furnace.

33. A boiler, using three different fuels simultaneously, delivers 300,000 lb of steam per hr at 400 psia and 680 F. Feed water is supplied at 282 F. 150,000 standard cu ft of oil refinery waste gas are fired per hour; heating value 1,000 Btu per standard cu ft. Acid sludge, having a heating value of 11,000 Btu per lb and weighing 9.091 lb per gal, is fired at the rate of 1,000 gal per hr. Fuel oil, having a heating value of 18,340 Btu per lb and weighing 7.25 lb per gal, is fired at the rate of 1,130 gal per hr. (*a*) What is the factor of evaporation? (*b*) What boiler horsepower is being developed? (*c*) What is the over-all efficiency of this unit?

34. A boiler evaporates 20,000 lb of water per hr from a feed-water temperature of 180 F into dry saturated steam at a pressure of 115 psia. Coal contains 4 per cent ash by analysis and 13,000 Btu per lb of dry coal. Ten per cent ash and refuse is taken from the ashpit. Actual evaporation is 9 lb of water per lb of dry coal. Find (*a*) the horsepower the boiler is developing; (*b*) the efficiency of the boiler, furnace, and grate; (*c*) the efficiency of the boiler and furnace; (*d*) the efficiency of the grate.

35. Given, a boiler plant developing 600 boiler hp; steam pressure, 150 psia; quality, 97 per cent; temperature of feed water, 200 F. The boiler uses 2,064 lb of dry coal per hr with a heating value of 13,000 Btu per lb, and 310 lb of ash and refuse are taken from the ashpit per hour. Dry coal contains 5 per cent ash by analysis. Determine (*a*) the efficiency of the boiler, furnace, and grate; (*b*) the efficiency of the boiler and furnace; (*c*) the weight of water fed to the boiler per hour.

36. A boiler plant burns 15,000 lb per hr of a certain Ohio coal containing 4 per cent moisture. The dry coal shows 7 per cent ash, and 14,000 Btu per lb by analysis. Twelve per cent of the weight of coal fired is removed from the ashpit in the form of ash and refuse. The steam pressure is 150 psia; feed-water temperature, 200 F; temperature of steam leaving boiler, 410 F; 136,800 lb of water are fed to the boiler per hour. Find (*a*) efficiency of boiler, furnace, and grate; (*b*) the efficiency of the boiler and furnace; (*c*) the horsepower developed by the boiler.

37. The following data are taken from a boiler trial: water fed to the boiler per hour, 14,350 lb; coal as fired per hour, 1,560 lb; dry refuse removed from ashpit per hour, 200 lb; steam pressure, 175.5 psi; barometer, 14.5 psia; quality of steam, 98 per cent; feed-water temperature, 192 F; heating value of coal as fired, 13,150 Btu per lb; heating value of dry refuse from ashpit, 3,260 Btu per lb. Determine (*a*) the boiler horsepower developed; (*b*) the boiler, furnace, and grate efficiency in per cent; (*c*) the boiler and furnace efficiency in per cent.

38. In testing a water-tube boiler with a heating surface of 4,000 sq ft the following data were obtained: length of test, 24 hr; weight of water evaporated, 582,300 lb; average steam pressure, 145.7 psi; average barometer reading, 29.19 in. Hg; average temperature of feed water, 110 F; temperature of steam leaving superheater, which is to be considered a part of the boiler, 460 F; weight of coal fired, 73,810 lb; weight of dry coal fired, 70,520 lb; weight of ash and refuse taken from ashpit, 11,360 lb; percentage of ash in coal, by analysis, 11.3; heating value of coal as fired, 12,860 Btu per lb. Determine (*a*) the boiler horsepower; (*b*) the rated horsepower; (*c*) the percentage of rated capacity developed; (*d*) the evaporation from and at 212 F per pound of dry coal; (*e*) the efficiency of boiler and furnace; (*f*) the efficiency of the grate.

39. Given the following data from a boiler test: duration of test, 24 hr; weight of water evaporated, 1,165,200 lb; average steam pressure, 197.7 psi; average barometer, 29.19 in. Hg; temperature of feed water entering boiler, 171.2 F; temperature of steam leaving superheater, which is to be considered a part of the boiler, 465 F; weight of coal fired, 148,150 lb; weight of dry coal fired, 140,600 lb; weight of ash and refuse (dry), 24,480 lb; proximate analysis of coal—moisture, 5.07 per cent; volatile matter, 39.59 per cent; fixed carbon, 43.05 per cent; ash, 12.29 per cent; heating value of coal as fired, 12,170 Btu per lb. Determine (*a*) the boiler horsepower being developed; (*b*) the evaporation from and at 212 F per pound dry coal; (*c*) the combined efficiency of the boiler, furnace, and grate; (*d*) the efficiency of the boiler and furnace; (*e*) the heating and (*f*) grate surfaces necessary in this boiler.

40. Given the following information on a large boiler installed at the Ford Motor Company: type of boiler, water tube with superheater, economizer, waterwalls, and air preheater, and using powdered coal:

Water heating surface of boiler	24,410 sq ft
Waterwall heating surface	9,500 sq ft
Superheating surface	24,019 sq ft
Economizer heating surface	25,200 sq ft
Air heater heating surface	86,016 sq ft
Steam pressure	1,400 psia
Average temperature of feed water entering economizer	300 F
Average temperature of water entering boiler	407 F
Average temperature of steam leaving superheater	925 F
Weight of water fed per hour	900,000 lb
Weight of coal fired per hour	46 tons
Ash in coal by analysis	8 per cent
Heating value of coal as fired	13,200 Btu

Find (*a*) the rated horsepower of the boiler, waterwalls, and economizer; (*b*) the horsepower developed by the unit (*i.e.*, the boiler, waterwalls, economizer, and superheater); (*c*) the equivalent evaporation from and at 212 F per lb of coal as fired; (*d*) the efficiency of the unit.

41. Given the following information from a test of a boiler and superheater:

Water evaporated per hour = 100,000 lb
Pressure of steam leaving superheater = 585.5 psi
Barometer = 29.5 in. Hg
Temperature of steam leaving superheater = 740 F
Temperature of water entering boiler = 306 F
Coal fired per hour = 10,200 lb

Moisture in coal = 2 per cent
Ash by analysis = 6 per cent
Weight of refuse per hour = 796 lb
Combustible in refuse = 23 per cent of refuse
Heating value of coal = 13,000 Btu as fired

Determine (*a*) the factor of evaporation for boiler and superheater; (*b*) the equivalent evaporation from and at 212 F per pound coal as fired; (*c*) the efficiency of the boiler, furnace, powdered-fuel burner and superheater. (*d*) What per cent of heat is lost due to combustible in the refuse?

42. A boiler unit (boiler, superheater, and economizer) burning pulverized coal generates steam at 250 psia pressure and 600 F total temperature from feed water at 182 F. The unit delivers 10 lb of steam per lb of coal fired. The coal has a heating value of 13,700 Btu per lb as fired. (*a*) What is the factor of evaporation in this case? (*b*) How many pounds of steam from this boiler are equivalent to 1 boiler hp-hr? (*c*) What is the over-all efficiency of this unit? (*d*) If the unburned carbon collected from the furnace, etc., amounts to 2 per cent of the weight of coal fired, what is the efficiency of the burners? The heating value of carbon is 14,600 Btu per lb.

43. A boiler, economizer, and superheater are installed as a single unit. The weight of water fed to the boiler per hour is 90,000 lb. The water enters the economizer at 180 F and leaves at 360 F. The steam leaves the water and steam drum with a quality of 97 per cent, and leaves the superheater with 200 deg of superheat. The steam pressure is 400 psia. The weight of coal fired per hour is 10,000 lb containing 5 per cent ash, 4 per cent moisture, and 13,100 Btu per lb. The ash and refuse removed from the ash hopper contain 20 per cent combustible material and 10 per cent moisture. The ash hopper is equipped with a water-sprinkling system. Find (*a*) the boiler horsepower developed; (*b*) the efficiency of the boiler plant as a unit; (*c*) the efficiency of the boiler and furnace alone.

44. A modernization program of a small power plant included the installation of a separately fired superheater using gas as a fuel. The boiler, which was unchanged, takes feed water at 150 F and produces 50,000 lb per hr of steam at 98 per cent quality and 120 psia with 82 per cent efficiency of boiler, furnace, and grate. The coal fired has a calorific value of 13,800 Btu per lb. The superheater is fired with methane, CH_4 (1,009 Btu per cu ft) and requires 5,700 cu ft per hr to superheat all the boiler steam to 500 F. (*a*) How much coal (pounds per hour) is used in the boiler? (*b*) What is the efficiency of the superheater unit? (*c*) What is the percentage increase in boiler horsepower with the superheater?

45. A water-tube boiler and superheater are supplied with 144,000 lb of water per hr, at a temperature of 294 F. The pressure of the steam leaving the superheater is 415 psia and the temperature is 720 F. The coal fired per hour is 12,600 lb and has a heat value of 14,200 Btu per lb. Determine (*a*) the factor of evaporation; (*b*) the equivalent evaporation per pound of coal as fired; (*c*) the efficiency of the boiler, furnace, and burner. (*d*) If the feed water enters the economizer at 94 F and leaves at 294 F, what per cent of the heat in each pound of coal is absorbed by the economizer?

46. Given a Stirling boiler equipped with an economizer and built-in superheater operating at a pressure of 250.5 psi. The weight of water fed to the boiler per hour is 60,000 lb. The temperature of the water entering the economizer is 180 F. The temperature of the water leaving the economizer is 260 F. The quality of the steam leaving the steam and water drum is 98 per cent and the temperature of the steam leaving the superheater is 500 F. The weight of the coal fired is 6,000 lb per hr. The coal as fired contains 3 per cent moisture and has a heating value of 14,000 Btu per lb. Barometer reads 29.53 in. Hg. Find (*a*) the boiler horsepower being developed; (*b*) the combined efficiency of the boiler, furnace, grate, economizer, and superheater. (*c*) What percentage of the capacity of the unit is developed by the economizer; (*d*) what by the superheater?

47. The following data were obtained from a test of a boiler equipped with an economizer and a superheater:

Weight of water fed to boiler per hour	360,000 lb
Weight of coal fired per hour	36,800 lb
Steam pressure	400 psia
Temperature of water entering economizer	180 F
Temperature of water leaving economizer	360 F
Temperature of steam leaving superheater	700 F

Steam leaves the water and steam drum with a quality of 98 per cent. The heating value of the coal as fired is 13,500 Btu. Find (*a*) the boiler horsepower of the unit; (*b*) the per cent of the capacity developed by the economizer, and (*c*) by the superheater; (*d*) the efficiency of the unit.

48. Given the following information from a boiler test: type of boiler, water tube with superheater; water heating surface of boiler, 6,132 sq ft; superheating surface, 1,200 sq ft; grate surface, 91 sq ft; steam pressure, 137.3 psi; barometer reading, 29.92 in. Hg; average temperature of feed water, 184 F; average temperature of steam leaving superheater, 460 F; weight of water fed per hour, 38,880 lb; weight of dry coal fired per hour, 4,860 lb; weight of ash and refuse taken from ashpit per hour, 583 lb; percentage of ash in dry coal by analysis, 8.0; heating value per pound of dry coal, 13,220 Btu. Find (*a*) the rated horsepower not including the superheater; (*b*) the horsepower developed by the boiler and superheater; (*c*) the equivalent evaporation from and at 212 F per pound of dry coal; (*d*) the efficiency of the boiler, furnace, grate, and superheater; (*e*) the efficiency of the boiler, furnace, and superheater, based on combustible burned.

49. A boiler and superheater are installed as a single unit. The boiler delivers steam to the superheater at a pressure of 210 psia and a quality of 97 per cent. The pressure through the superheater remains constant. The steam leaving the superheater has a temperature of 536 F. The feed water has a temperature of 205 F. Twenty-four hundred pounds of coal, having a heating value of 13,500 Btu per lb, and 20,000 lb of water are used per hr. (*a*) What part of the capacity of the unit is developed by the boiler and what part by the superheater? (*b*) What is the efficiency of the unit?

50. The indicated horsepower of the main steam engines of a power plant is 5,000. These engines use 19 lb steam per ihp-hp. The auxiliaries of the plant total 12 per cent as much power as the main engines and use 90 lb steam per ihp-hr. The water is fed to the boiler at 182 F. Steam containing 1.5 per cent moisture is delivered to the engines at a pressure of 165.5 psi. The boiler uses coal containing 13,000 Btu per lb. The efficiency of the boiler, furnace, and grate is 70 per cent. The barometer reads 29.53 in. Hg. Find (*a*) the boiler horsepower necessary to run the plant; (*b*) the pounds of coal used per hour; (*c*) the equivalent evaporation from and at 212 F per pound of coal.

51. Given the following data from a Stirling boiler equipped with a superheater: steam pressure 226 psia; temperature of steam leaving superheater 560 F; temperature of feed water 194 F. The boiler delivers 144,000 lb of steam per hr and consumes 16,000 lb of coal per hr. The coal fired has a heating value of 13,780 Btu per lb and contains 2 per cent moisture. Determine (*a*) the boiler horsepower developed by the unit; (*b*) the efficiency of the boiler, furnace, and grate; (*c*) the heat loss due to moisture in the coal if the room temperature is 70 F and the stack temperature is 500 F.

52. A steam turbine operating at a load of 30,000 kw requires 15 lb of steam per kwh delivered at the switchboard. The steam is supplied at 200 psia and at a temperature of 500 F. The temperature of the feed water entering the boiler is 184 F and the efficiency of boiler, furnace, and grate is 78 per cent. (*a*) How many pounds of coal are required per kilowatt-hour if the coal has a heating value of 13,500 Btu per lb? (*b*) How many boilers would be necessary to carry the load, if each of the boilers supplying steam has 20,000 sq ft of heating surface and is operating at 150 per cent rating? (*c*) What would be the over-all efficiency of the plant?

53. A boiler rated at 1,000 hp is operating at 150 per cent of its rating. The boiler supplies superheated steam at a temperature of 720 F and at a pressure of 305 psia. Water is fed to the boiler at 150 F. The boiler operates at an efficiency of 80 per cent when using coal having a heating value of 13,000 Btu per lb. Determine (*a*) the pounds of steam delivered per hour; (*b*) the pounds of coal used per hour. (*c*) What would be the percentage of saving in fuel if an economizer were added and the temperature of the feed water raised to 300 F?

54. The following data are taken from a boiler test: dry saturated steam is delivered at a pressure of 180 psia; feed temperature is 180 F; water fed to the boiler per hour is 114,000 lb; dry coal fired per hour is 12,000 lb; refuse removed from the ashpit per hour is 890 lb. The ultimate analysis of the coal, per cent by weight, is C = 77.37; H = 5.43; O = 9.76; N = 1.83; S = 1.22; ash = 4.39. Heating value of the dry coal is 13,950 Btu per lb. Flue-gas analysis, per cent by volume: CO_2 = 12.00; CO = 0.94; O_2 = 6.06; N_2 = 81.00. The room temperature is 70 F; stack temperature is 570 F. specific heat of stack gases = 0.24. Calculate (*a*) the boiler horsepower being developed; (*b*) the efficiency of the boiler, furnace, and grate; (*c*) the efficiency of the boiler and furnace; (*d*) the heat carried away by the dry products of combustion in Btu per pound of coal; (*e*) the per cent excess air.

55. A coal-fired boiler evaporates 2,000 lb of steam per hr at 100 psia and a quality of 98 per cent. 250 lb of coal are burned per hour. The flue-gas analysis by Orsat is CO_2, 12 per cent; O_2, 8 per cent; CO, 0.5 per cent.

Analysis shows that the refuse contains 50 per cent carbon (carbon has a heating value of 14,600 Btu per lb). The ultimate analysis of coal is as follows: C, 74.4; H, 5.2; O, 9.60; N, 1.3; S, 1.0; ash 8.5; Btu per pound is 13,400. The flue-gas temperature is 500 F, room temperature is 70 F and feed-water temperature is 150 F. Calculate (*a*) the per cent of heat loss due to dry flue gas; (*b*) the per cent of heat loss due to carbon in the ashpit; (*c*) the efficiency of the steam generator.

56. A water-tube boiler having a heating surface of 5,080 sq ft was tested and the following data taken during a $16\frac{1}{4}$-hr run: total weight of coal as fired, 39,670 lb; heating value of the coal as fired, 12,000 Btu per lb; total water fed to boiler, 336,200 lb; steam condition, 200 psia and 138.2 deg of superheat; feed-water temperature 50 F; cost of coal \$3.75 per ton at boiler room. Calculate (*a*) the factor of evaporation; (*b*) the equivalent economy per pound of moist coal; (*c*) the efficiency of the boiler, furnace, and grate; (*d*) the boiler horsepower developed; (*e*) the percentage of rated horsepower developed; (*f*) the coal cost of steam per 1,000 lb delivered to header; (*g*) the coal cost per 1,000,000 Btu of steam delivered to header.

57. A boiler received 12,000 lb water per hr at 112 F; steam pressure, 150 psia; quality of steam, $98\frac{1}{2}$ per cent; dry coal fired per hour, 1,500 lb, each pound containing 13,500 Btu; percentage of ash by analysis, 3; ash and refuse taken from ashpit per hour, 150 lb. Coal costs \$9 per ton. The plant runs 10 hr per day, 300 days a year. (*a*) What horsepower is the boiler developing? (*b*) What is the efficiency of the boiler, furnace, and grate combined? (*c*) What is the efficiency of the boiler and furnace alone? (*d*) If the interest and depreciation are 12 per cent and a feed-water heater costs \$700, what would be saved per year and, (*e*) what per cent profit would be made by installing one to bring the feed temperature up to 206 F?

58. A boiler evaporates 12,000 lb of water per hr from a feed temperature of 110 F into steam at a pressure of 140 psia with a quality of 98 per cent. The weight of dry coal fired is 1,500 lb per hr, each pound containing 13,500 Btu and $3\frac{1}{2}$ per cent ash, by analysis. Nine per cent ash and refuse is taken from the ashpit. The coal costs \$4 per ton and the plant runs 10 hr per day, 300 days per yr. Find (*a*) the boiler horsepower; (*b*) the efficiency of the boiler and furnace; (*c*) the efficiency of the grate. (*d*) What could be paid for a heater which raised the feed-water temperature to 200 F, allowing a profit of 10 per cent over and above a 12 per cent allowance for depreciation and interest.

59. It requires a 750-kw turbine to furnish electric current for a factory. The plant operates 24 hr a day for 300 days a year. The average steam consumption is 20 lb of steam per kwhr. The average load for the year is 50 per cent of the rated capacity. Steam pressure, 150 psi; temperature of feed water, 130 F; heating value of dry coal, 13,000 Btu per lb; coal as fired contains 2 per cent moisture; average efficiency of boiler plant, 60 per cent. Determine (*a*) the coal cost per year if coal as fired can be bought for \$2 per ton; (*b*) the amount saved per year by a feed-water heater that would increase the feed temperature to 200 F; (*c*) the net return (per cent) on the investment if the heater cost \$350, allowing 5 per cent for interest and 7 per cent for depreciation and repairs.

60. An economizer raises the temperature of the feed water in a power plant from 200 F to 300 F. The economizer costs $5.50 per boiler hp to install, and entails an operating expense of $1 per boiler hp per yr. Steam pressure, 100 psi. Coal costs $3 per ton, and has a heating value of 13,000 Btu per lb. The efficiency of the boiler, furnace, and grate without the economizer is 70 per cent. The plant operates 10 hr per day, 360 days per yr. (*a*) Assuming 6 per cent interest and 8 per cent depreciation on the economizer, would it pay to install it? (*b*) If so, what would be the net return (per cent) on the investment?

61. A boiler plant develops 500 hp and uses 4 lb of coal per hp-hr. Coal contains 13,000 Btu per lb; steam pressure, 150 psi; feed-water temperature, 120 F. A feed-water heater is added, raising the temperature of the water to 195 F; heater costs $650; plant operates 10 hr a day, 300 days a year. The cost of coal is $5.50 per ton. Allow 7 per cent depreciation and repairs, 5 per cent interest, and 3 per cent insurance. (*a*) How much would the owner make by installing the heater? (*b*) What per cent profit? (*c*) If later an economizer is added which raises the feed water from 195 F to 300 F, and the economizer costs $5,000, allowing the same rates, how much more could be saved? (*d*) What would be the efficiency of the plant under the last condition?

62. A boiler plant runs 24 hr day for 300 days in the year. It burns 30 tons of coal per day, costing $3 per ton. The analysis of the stack gases is CO_2, 5 per cent; O_2, 15 per cent; N, 80 per cent. The coal contains C, 80 per cent; H, 6 per cent; and O, 4 per cent. The plant is changed so that the stack-gas analysis is CO_2, 14 per cent; O_2, 6 per cent; N, 80 per cent. (*a*) What will be the saving in dollars per year? Stack-gas temperature, 600 F. Boiler-room temperature, 70 F. Boiler radiation loss, 4 per cent.

After this change is made, an economizer is installed which reduces the temperature of the stack gases from 600 F to 400 F. The evaporation is 9 lb of water per lb of coal. The feed-water temperature is 120 F. (*b*) What will be the final temperature of the feed water? (*c*) What will be the saving in dollars per year after this second change is made?

63. Compute the heat balance from the following data obtained during a test on a water-tube boiler: heating surface, 5,080 sq ft; water fed to boiler, 29,200 lb per hr; coal used, 4,300 lb per hr; dry refuse from ashpit, 840 lb per hr; combustible in dry refuse, 43.7 per cent; feed-water temperature, 144 F; boiler-room temperature, 62 F; flue-gas temperature, 510 F; relative humidity of air, 55 per cent; steam pressure, 120 psi, dry steam; barometer, 29.39 in. Hg. Analysis of coal as fired: C, 65 per cent; H, 3 per cent; N_2, 1 per cent; O_2, 7 per cent; moisture, 13 per cent; ash, 11 per cent; Btu per pound, 10,900. Flue-gas analysis: CO_2, 14 per cent; CO, 0.8 per cent; O_2, 5.5 per cent; and N_2, 79.7 per cent (by difference). (From the relative humidity and psychrometric tables, the weight of moisture in the air per pound of dry air supplied is 0.01233 lb.)

64. Given the following data, compute a heat balance based on calorific value of coal as fired: temperature of boiler room, 92 F; temperature of gases leaving boiler, 475 F; steam pressure, 150 psi. Analysis of coal as fired: C, 75.39 per cent; H, 6.17 per cent; O, 4.92 per cent; N, 1.5 per cent; S, 0.96 per cent; moisture, 4 per cent; ash, 7.05 per cent. Combustible in refuse, 30 per cent; moisture in dry air per pound 0.02 lb; flue-gas analysis: CO_2, 12 per cent; O_2, 6.7 per cent; CO, 0.2 per cent; N, 81.1 per cent (by difference); gas given off per pound of wet coal fired, 14.9 lb; gas given off per pound of carbon burned, 20.6 lb; evaporation from and at 212 F per pound of wet coal, 10.7 lb; heating value of coal as fired, 13,630 Btu per lb.

65. Calculate values for the blank spaces given below. Use the calorimeter heating value of coal in the calculations. Duration of test, 24 hr.

Temperatures, Deg F.		Total Quantities, Etc.	
Feed water	173.0	Coal as fired	337,100 lb
Flue gases	506.0	Water fed to boiler	2,819,700 lb
Boiler room	73.0	Dry refuse	48,500 lb
Steam	592.0	Steam pressure	280.0 abs
Coal Analysis (as Received)	Per Cent	Dry Flue-gas Analysis (by Volume)	Per Cent
Carbon	70.05	Carbon dioxide	12.0
Hydrogen	4.90	Oxygen	6.3
Oxygen	8.00	Carbon monoxide	0.6
Nitrogen	1.04	Nitrogen	81.1
Sulphur	1.15		100.0
Ash	11.20		
Free moisture	3.66		
	100.00	Btu as fired, by calorimeter	12,946

HEAT BALANCE, BASED ON COAL AS FIRED

Item	Btu	Per Cent
Heat absorbed by boiler	______	______
Loss due to moisture in coal	______	______
Loss due to hydrogen in fuel	______	______
Heat carried away to dry flue gases	______	______
Loss due to incomplete combustion (CO)	______	______
Loss due to combustible in dry refuse	______	______
Radiation and unaccounted-for losses	______	______
Total	12,946.00	100.00

CHAPTER VIII

STEAM ENGINES

129. Classification.—A fundamental classification of engines, based upon the number of working strokes per revolution, groups them as follows:

1. Single-acting engines.
2. Double-acting engines.

Single-acting engines are those in which the steam pressure is exerted against the piston every other stroke only, or once per revolution; in double-acting engines the steam acts on the piston every stroke, or twice per revolution. In a single-acting engine, the inertia of the flywheel carries the engine along from one working stroke to the next.

Engines may be classified, according to whether they exhaust into the atmosphere or into a condenser, as:

1. Noncondensing engines.
2. Condensing engines.

In the noncondensing engine the exhaust passes directly to the atmosphere. In condensing engines the exhaust steam passes into a cold chamber where, by means of a cooling medium, the steam is changed to water. This produces a vacuum so that the exhaust occurs at a pressure lower than that of the atmosphere. The condensed steam is removed and the vacuum is sustained by means of an air pump.

Another classification may be made according to the way in which their speed is governed, as:

1. Throttling engines.
2. Automatic cutoff engines.

In the throttling engines the speed of the engine is controlled by means of a valve in the steam pipe which regulates the pressure of the steam entering the engine. In the automatic cutoff engine the pressure of the entering steam remains constant and the governor controls the amount of steam admitted to the cylinder.

Engines may also be classified according to the number of cylinders in which the steam is allowed to expand successively, as:

1. Simple engines.
2. Compound engines.
3. Triple-expansion engines.
4. Quadruple-expansion engines.

In a simple engine the steam expands in but one cylinder and is then allowed to exhaust. In a compound engine a portion of the expansion occurs in the high-pressure cylinder; from there the steam passes to the low-pressure cylinder, where it is further expanded to a pressure approximating the exhaust pressure. In the triple-expansion engine the steam expands successively in three cylinders, and in the quadruple-expansion in four.

A fourth classification depends upon the position of the cylinder, as:

1. Vertical engines.
2. Horizontal engines.

A classification frequently made is on a basis of location, as:

1. Stationary engines.
2. Marine engines.
3. Locomotive, or portable, engines.

Finally, engines may be classified as:

1. Side-crank engines.
2. Center-crank engines.

Side-crank engines have only one bearing on the engine frame. The end of the shaft away from the crank is supported by a bearing separate from the engine frame, often called the *outboard* bearing. When this bearing is on the right side, looking from the cylinder toward the flywheel, the engine is said to be "right hand"; when on the left side, to be "left hand."

A center-crank engine has both bearings on the main frame of the engine.

130. Plain Slide Valve Engine.—The simplest form of engine is the plain *D*-slide valve engine, as shown in Fig. 140.

The valve is shown in its normal position in the steam chest. A cross section of a valve of this type showing the steam ports is shown in Fig. 158.

This type of engine is used where high economy is not necessary. It requires little attention, and is easily repaired and adjusted. These engines are governed by a throttling governor of the flyball type, which controls the speed of the engine by changing the pressure of the steam in the steam chest.

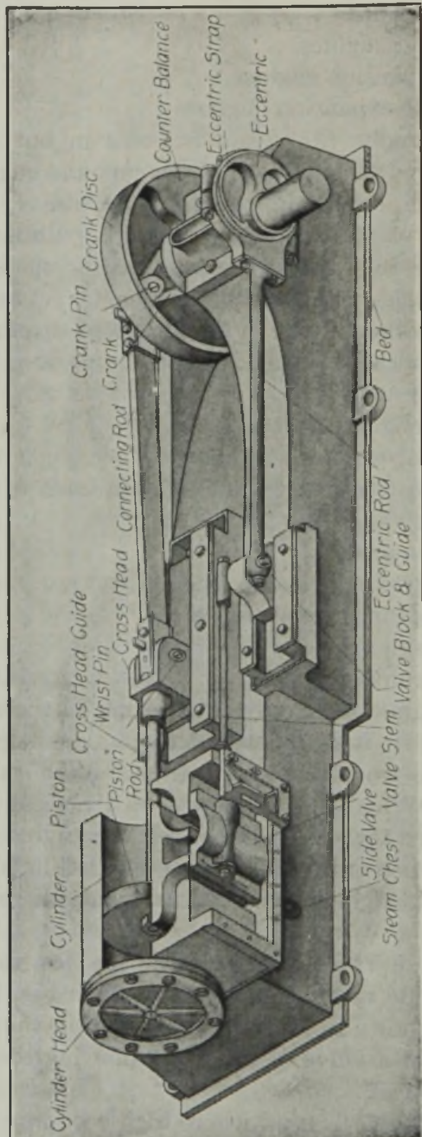


FIG. 140.—Block plan of steam engine.

131. Automatic Cutoff High-speed Engine.—This class of engines was developed rapidly after the introduction of electrical lighting machinery, and was designed primarily for the direct driving of electric generators. The engines have balanced slide valves such as are shown in Fig. 142. The governors in this class of engines control the valve directly, and it is necessary that the valve be balanced so that it may be moved easily by the governor.

Engines of this class are well adapted to a high rotative speed. The stroke of these engines is usually short, so that the average

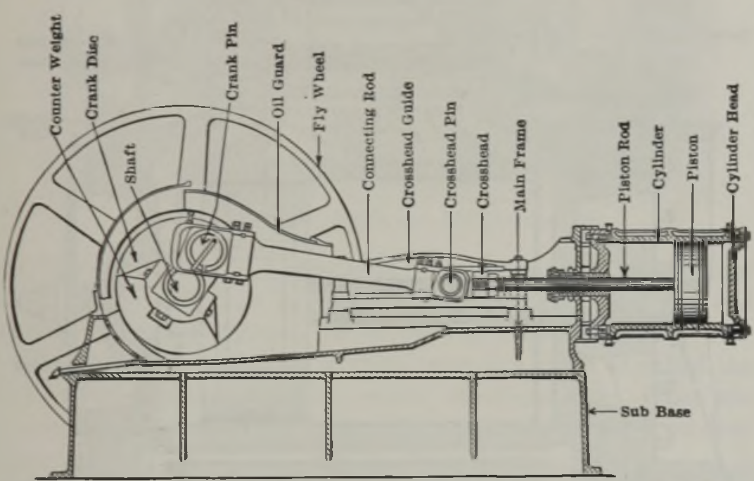


FIG. 141.—Vertical section of Skinner engine.

piston speed may exceed 600 fpm when the engine runs at a large number of revolutions per minute.

Most engines of this class are of the center-crank type, so that all parts of the engine are supported on one casting.

132. The Simple Steam Engine.—A simple form of stationary steam engine and one in general use is shown in Fig. 141. It is a small direct double-acting engine with a balanced slide valve and a cast-iron cylinder closed at its ends by cylinder heads bolted on. The engine has no steam jacket and is surrounded on the outside by nonconducting material and cast-iron lagging. Figure 142 shows the steam chest containing the valves and the ports leading from the steam chest to the cylinder. The steam is admitted and exhausted through these ports. The piston is made a loose fit in the cylinder. The spring rings shown in the piston serve to prevent leakage from one side of the piston to the other. The piston rod is usually fastened into the piston head by a taper-ended rod and nut, and is then carried through the cylinder head, the gland and packing serving to make a steamtight joint. The other end of the piston rod is connected with the crosshead. The power is communicated from the connecting rod to the crank, which is attached to the

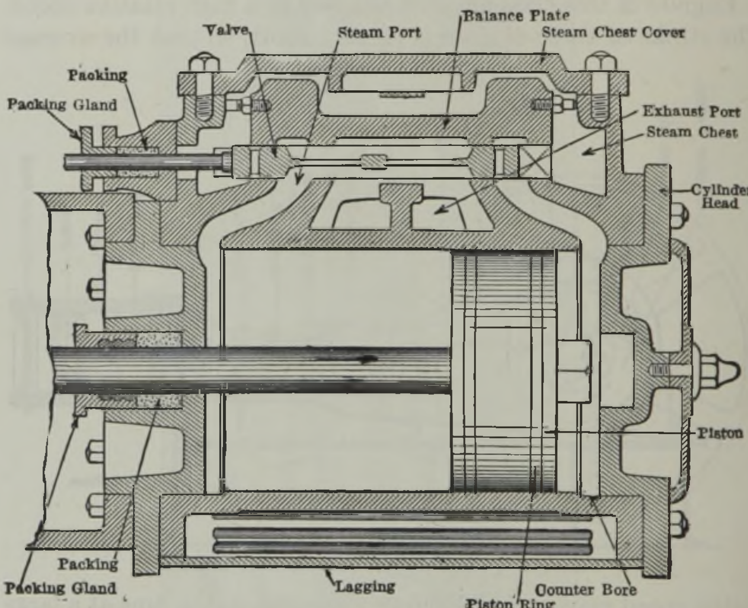


FIG. 142.—Section through steam-engine cylinder and valve.

main shaft. To this main shaft the eccentric is fastened by means of set screws. The valve of the engine is driven by the eccentric through the agency of the eccentric rod and the valve stem. The valve stem passes through the steam chest, being made tight by the glands and packing, as in the case of the piston rod, and is fastened by lock nuts to the valve. The function of this valve is to admit the steam surrounding the valve to each end of the cylinder alternately. On the opposite stroke, the valve opens up the ends of the cylinder to the exhaust space in the center of the valve, this space being connected to the exhaust pipe of the engine, and the space outside of the valve being connected to the steam pipe admitting the steam to the engine.

Figure 142 shows the slide valve in a position admitting steam to the head end of the cylinder. On the crank end, the cylinder is open to exhaust. As the steam enters behind the piston, the steam in the space on the opposite side of the piston is forced out through the space under the valve and out of the exhaust port. When the piston reaches the opposite end of the stroke,

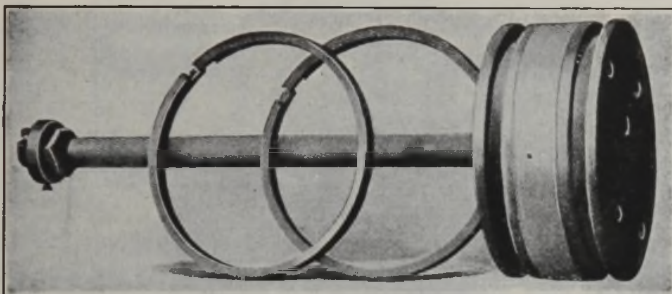


FIG. 143.—Piston, piston rod, and piston rings.

the valve will have been moved to a similar position at the other end. Steam will then be admitted at that end, and the end previously receiving steam will be open to exhaust.

The end of the cylinder farthest from the crank is called the *head end* and that nearest the crank, the *crank end.* When the piston is at the end of its stroke, the engine is said to be on *dead*

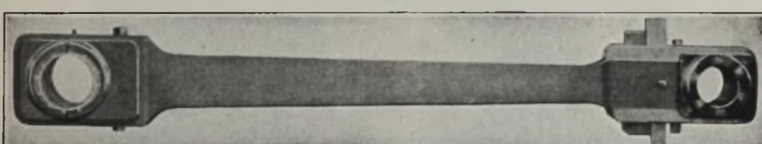


FIG. 144.—Strap-end connecting rod.

center. The volume swept through by the piston per stroke is called the *piston displacement.*

133. Engine Details.—Figure 143 shows a piston, piston rod, and piston rings. The piston is turned a little smaller than the cylinder, and is made tight in the cylinder by means of spring rings. In the figure, these rings are shown leaning against the piston rod. They are made of cast iron and are so constructed that they have to be compressed in order to get them into the cylinder; when the piston is in place, the rings bear firmly against the cylinder walls.

Figure 144 shows a strap-end connecting rod. These rods are usually made of forged steel. The bearings that enclose the pin are made of bronze, or are bronze lined with a babbitt bearing metal, and fitted into the ends of the rods. These bearings,

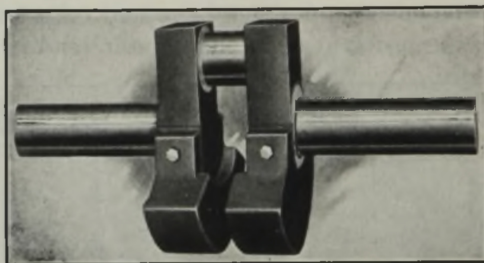


Fig. 145.—Counterbalanced crank.

or *brasses*, are held in place by steel straps that encircle them. These straps are fastened to the body of the connecting rod by means of a taper key and a cotter. The brasses in this rod are shown lined with babbitt metal, which is much softer than the steel pins themselves.

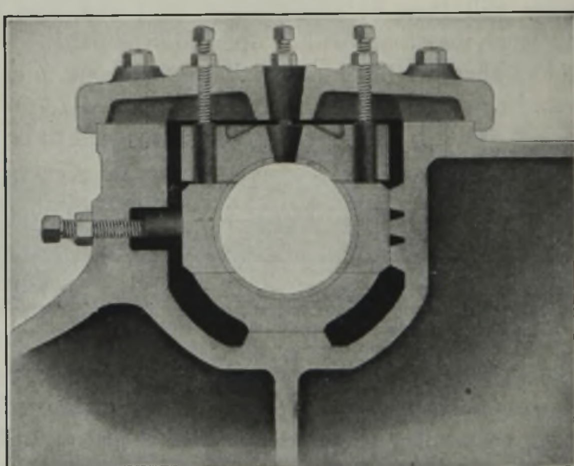


Fig. 146.—Four-part main bearing.

Figure 145 shows the crankshaft and its counterbalance weights which are bolted to the crank. The crankshaft is a solid forging of open-hearth steel. The counterweights are made of cast iron. The crankshaft shown in the figure is designed for a center-crank engine.

Figure 146 shows one of the main bearings for the crankshaft. The figure shows what is called a four-part bearing. The bearing proper is made up of four pieces. The two side pieces, or brasses, take up most of the wear in the bearing and are adjusted by means of set screws fastened with lock nuts. The upper part of the brasses is adjusted by a screw in the top of the bearing. The brasses are supported by the main frame of

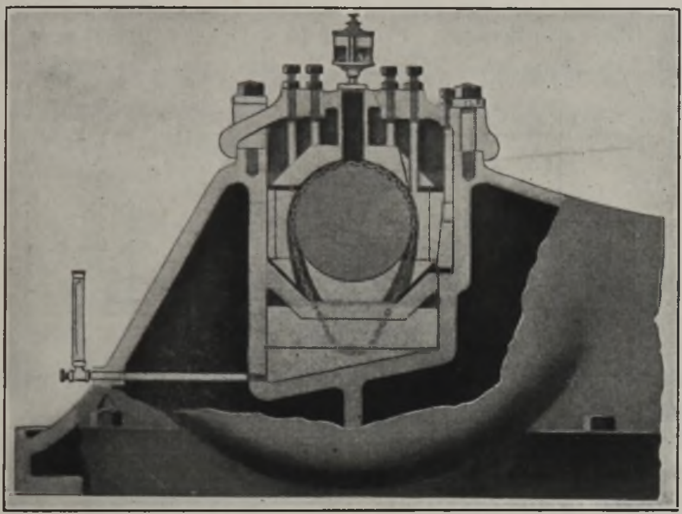


FIG. 147.—Main engine bearing with oil well—cross section.

the engine and are held down by a main bearing cap bolted to the main frame of the engine.

Figure 147 shows a main engine bearing having an oil well. The lower part of the well is filled with oil, which is carried up onto the bearing by means of a chain which hangs over the shaft and dips into the well. The chain is moved by the rotation of the shaft, bringing the oil up on to the shaft.

134. Steam Cycles.—Although a number of cycles may be performed by the working fluid in a steam engine, only two will be considered here—the Carnot and the Rankine.

135. Carnot Cycle for Steam.—As shown in Arts. 30 and 31, the Carnot cycle gives the maximum efficiency obtainable for an engine working between any given temperature limits. The Carnot cycle for a steam engine using dry and saturated steam is shown in Fig. 148. A volume of water *a* at the boiling point corresponding to the pressure a has heat added to it along the isothermal (and isobaric) line ab. At b the water has all been changed to dry and saturated steam and this steam expands adiabatically and isentropically from b to c, some of it being condensed during the expansion. When the expansion has reached the point c where the temperature is that of the cooler, compression begins and continues along the isothermal (and isobaric) line cd to the point d, further condensation occurring during this compression. From d to a the compression is continued, but along an adiabatic. By the time a is reached the

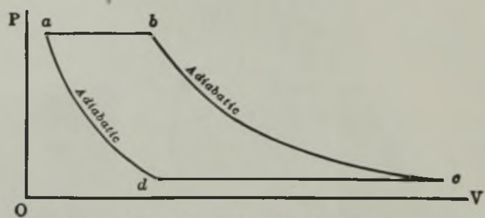


FIG. 148.—Carnot cycle for steam.

steam has all been condensed and the water raised to the original temperature.

The efficiency of this cycle depends entirely upon the absolute temperatures along the lines ab and cd, and, as shown in Art. 30, is expressed by the equation

$$e_t = \frac{T_1 - T_2}{T_1}. \tag{1}$$

136. Rankine Cycle.—In a steam engine operating along a Carnot cycle it would be necessary to evaporate the water in the engine cylinder instead of a separate boiler, and to condense the steam in the cylinder instead of rejecting it to the air or a condenser. Such an arrangement is manifestly impossible and, therefore, it has been found necessary to select some other cycle the efficiency of which may be used as the standard with which to compare the performances of actual steam engines.

The cycle which has been adopted as this standard is the Rankine, since it is the most efficient steam cycle in which the working fluid is not evaporated and condensed in the engine cylinder, or, in other words, is the most efficient practical steam cycle.

In this cycle it is assumed that there is no clearance; that there is no heat loss from the cylinder walls, *i.e.*, that the expansion is adiabatic; and that the valves operate instantaneously and there is no wiredrawing. These assumptions can never be fully realized. Clearance is necessary for cushioning the piston and bringing the reciprocating parts to rest; adiabatic expansion is an aim or ideal that is never reached, as with the best lagging some heat is given off from the cylinder; and wiredrawing is certain to occur to some extent even with the greatest care in operation and design. This cycle, then, is a standard that can never be attained, but can be approached as the imperfections in an engine are eliminated.

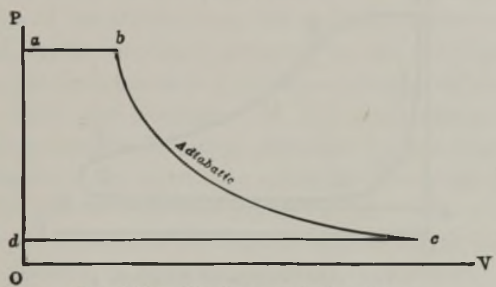


Fig. 149.—Rankine cycle for steam.

Figure 149 shows a Rankine cycle. Steam from the boiler is admitted to the cylinder along the isothermal (and isobaric) line *ab*. At *b* the supply of steam is cut off and adiabatic or isentropic (constant entropy) expansion begins, continuing to *c*, where the pressure and the temperature are those of the exhaust. During this step some of the steam is condensed. At *c* the exhaust valve opens and from *c* to *d* the steam is exhausted along a line of constant temperature and pressure. At *d* the exhaust valve closes, the admission valve opens, and the pressure and the temperature rise along *da* to *a*, the volume remaining constant.

137. Action of the Steam in the Steam Engine.—In the simplest form of steam engine, the steam is admitted for the full stroke of the piston and, when the valve opens the cylinder to exhaust, the steam is exhausted at nearly full boiler pressure. This action of the engine is, of course, very uneconomical, and early in the development of the engine it was found desirable to allow the steam to expand in the cylinder. This is accomplished by having the valve close the entrance port before the piston has reached the end of its stroke. Then for the balance of the stroke, as the piston is forced out, the steam pressure in the cylinder gradually falls, owing to the increasing volume of the steam.

Figure 150 shows graphically the action that goes on in the cylinder. The ordinates of the diagram represent the steam pressure, and the abscissas represent cylinder volumes as the piston moves out. The steam enters along the line *CDE*, the pressure at *D* being a little below boiler pressure. At the point *E*, known as the *point of cutoff*, the valve closes. The steam expands from the point *E* to *F*, along the expansion line *EF*. At

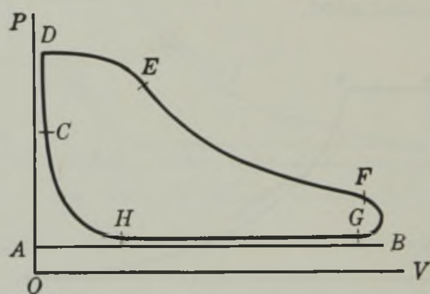


Fig. 150.—Indicator diagram.

the point *F*, called the *point of release*, the valve opens, and from the point *F* to the point *H* the exhaust occurs. At the point *H* the valve closes the exhaust port and compression of the steam left in the cylinder begins, continuing along the line *HC* to the point *C*. At this point steam is again admitted to the cylinder. A similar action occurs on the opposite end of the cylinder, so, while the steam is being admitted at one side, at the opposite side of the piston there is exhaust pressure. Such a diagram is termed an *indicator diagram* and may be graphically produced by an instrument known as the *indicator*.

138. Losses in a Steam Engine.—The action of the steam in the steam engine is different from that which has been assumed as the ideal action. The action of the ideal engine is useful, however, as a basis of comparison for the action of the steam in actual engines. In the actual engine, the steam is never expanded completely, and has at the end of the expansion a higher pressure than the back pressure in the exhaust pipe. It is not advisable to give the steam complete expansion, as there will be no added work due to such expansion of this steam, the pressure being insufficient to overcome the friction of the engine. Because complete expansion does not occur, it is necessary to open the exhaust valve before the end of the stroke so as to bring the pressure at the end of the stroke down to the back pressure. Comparing the ideal diagram (Fig. 187) with the actual diagram (Fig. 150) it will be noticed that during admission in the actual diagram the steam does not remain at full boiler pressure, but that there is a reduction of the pressure due to *wiredrawing* of the steam through the ports of the valve. In the ideal engine there is no transmission of heat to the steam except in the boiler, but in the actual engine there is a transfer of heat from the steam to the cylinder walls during a portion of the stroke, and during other portions of the stroke from the cylinder walls to the steam. In an actual engine the back pressure in the cylinder is always greater than the vacuum in the condenser, owing to the resistance of exhaust valve and passage. In the ideal engine the whole volume of the cylinder is swept through by the piston, and in the actual engine there must be a space at the end of the cylinder to prevent the piston from striking the head.

The principal losses of heat from an engine are given as follows, as nearly as possible in the order of their importance:

1. Heat lost in the exhaust. This loss may be as much as 70 per cent or more of the entire heat admitted to the engine.
2. Initial condensation.
3. Wiredrawing at admission and exhaust.
4. Condensation in the clearance space during compression.
5. Incomplete expansion.
6. Radiation and convection from the cylinder.
7. Leakage past the piston and valves.

139. Heat Lost in the Exhaust.—Most of the heat brought to the engine by the steam is rejected by the engine in the exhaust. This loss varies from 70 per cent of the heat of the steam in the best engines to over 90 per cent in the poorer types. In many steam plants this heat is partly recovered by using the exhaust for heating or manufacturing purposes. The steam leaving the exhaust of an engine usually contains from 10 to 20 per cent of water.

140. Initial Condensation and Reevaporation.—Early experimenters in steam-engine economy found that the surfaces of the cylinder wall and steam ports played a very important part in the economy of the steam engine. The inner surfaces exposed to the action of the steam in the engine naturally have a temperature very close to that of the steam itself. When the steam enters the cylinder, it comes in contact with the walls of the cylinder which have just been exposed to exhaust steam and are, necessarily, at a lower temperature than this entering steam. A part of this steam will, therefore, be condensed in warming the walls; as the piston moves out, more of the walls will be exposed, so that condensation increases to a point even beyond the point of cutoff. After the point of cutoff the steam expands, the pressure falls, and the temperature drops until a point is reached where the temperature of the cylinder walls is the same as the temperature of the steam in the cylinder. Condensation ceases at this point and the cylinder walls are by this time covered with a film of moisture. If the expansion of the steam is continued still further, the temperature in the cylinder corresponding to the steam pressure will be less than the temperature of the cylinder walls, and this film of moisture on the surface will begin to reevaporate. Usually the amount of reevaporation during expansion is very much smaller than the initial condensation and the cylinder walls are still wet when the exhaust valves open. This reevaporation also continues during the exhaust stroke. It is very desirable that all the moisture on the surface of the cylinder be evaporated before the end of the exhaust. If it is not evaporated, the cylinder walls will be wet when steam is again admitted to the cylinder and the initial condensation will be greatly increased. The transfer of heat from the steam to the walls of the cylinder is always increased by the presence of moisture.

In the average noncondensing engine, initial condensation is from 15 to 20 per cent, in small reciprocating steam pumps an initial condensation as high as 75 per cent sometimes occurs, and in the most perfect engines it is from 10 to 12 per cent.

141. Factors Affecting Initial Condensation.—Initial condensation is always increased by increasing the *range of temperature* in the cylinders.

It also increases as the *ratio of the area of the cylinder walls to the volume of the cylinder* increases. The greater this ratio the less the economy, as the more wall there is exposed the more heat the wall will take up. This accounts for the large consumption of steam shown by most rotary engines.

Time is also important, and, other conditions being the same, the slower the speed of the engine the greater the initial condensation, as the whole action depends upon the time during which the heat has an opportunity to be taken up or given off by the cylinder walls. As the element of time during which the steam is in contact with the walls of the cylinder increases, the initial condensation increases. The changes of temperature affect only the inner surfaces of the cylinder, and the greater the time the greater the depth of cylinder walls that will be affected.

Initial condensation increases as the *ratio of expansion* is increased, *i.e.*, as the cutoff becomes shorter. This is easily explained; as the cutoff is shortened, the weight of steam admitted to the cylinder becomes less and, since the amount of heat taken up by the cylinder walls remains substantially the same, the proportion of steam condensed increases. With very short cutoffs this initial condensation becomes very large. When the cutoff is reduced below a certain point, the increased initial condensation offsets the increase in economy due to longer expansion. If the cutoff is shortened to less than this point, the steam consumption of the engine will be increased. The point of greatest economy in most single-cylinder engines is from one-quarter to one-fifth stroke. In an engine having a short cutoff and using a high steam pressure, the economy may often be increased by reducing the steam pressure and increasing the cutoff.

The three methods commonly used to minimize initial condensation are *steam jacketing*, *superheating*, and *compounding*. Another method that has come into use in recent years is the *Unaflow* system. Sometimes two of these methods are combined, the best results up to the present time being obtained by the use of steam jackets together with superheated steam.

142. Steam Jacket.—The action of initial condensation is increased by the loss of heat through the cylinder wall by conduction. This may be reduced by surrounding the cylinder with steam at boiler pressure. Such an arrangement is called a *steam jacket*. The effect of the steam jacket is to reduce initial condensation and to increase the reevaporation. The steam used by the steam jacket is always charged to the engine as though it had been used in the cylinder. Engines with jackets show increased economy, particularly when operated at slow speed. The higher the speed of the engine the less is the element of time during which the jacket can affect the steam in the cylinder and the less effective the jacket becomes. In slow-speed engines with large ratios of expansion, the use of the jacket will show a saving of from 10 to 20 per cent.

143. Superheating.—Superheating the steam previous to its admission to the engine is used as a means of reducing initial condensation. A sufficient amount of superheat should be given to the steam so that, on admission of steam to the cylinder, the cylinder walls take up this superheat instead of condensing the steam. The effect of this is to leave the cylinder walls entirely dry, reducing the amount of heat that would be conducted to the walls, as dry gas is one of the best nonconductors of heat. The experiments of Professor Gutermuth show that with sufficient superheat the economy of a simple noncondensing engine may be made to equal that of a compound condensing engine.

144. Compound Expansion.—By increasing the steam pressure and using a longer range of expansion, the range of temperatures in the cylinder of a steam engine is much increased, thereby increasing the initial condensation. In order to reduce this condensation in the cylinder, it has been found economical to expand the steam partially in one cylinder and then exhaust into a second cylinder in which the expansion is completed. By this means the range of temperature in each cylinder is reduced and the initial condensation reduced. Compound engines are used only when the steam pressure is sufficiently high so that the initial condensation would be excessive if the steam were expanded in one cylinder. They are seldom used with steam pressures less than 100 psi, and it is not necessary to use them for less than 125 psi unless the ratio of expansion is very large.

145. Wiredrawing.—The resistance offered by the valves, ports, and passages lowers the pressure of the steam in the cylinder during admission and raises the pressure during exhaust. As the valves do not close instantly when the valve nears the point of closing, or cutoff, the pressure is reduced, owing to the small port opening. This is shown by the rounded corners at cutoff and release. This resistance is often called *throttling* or *wiredrawing*. The effect of this throttling of the steam is to dry it slightly; if it were absolutely dry to start with, there would be a small amount of superheating.

It will be noticed in the indicator diagram (Fig. 150) that the initial line DE is not absolutely horizontal, but that there is a gradual reduction of pressure from D to E. The initial pressure line is always lower than the boiler pressure, owing to the resistance of the passages between the boiler and the cylinder.

The steam in passing through the piping leading to the engine loses a certain quantity of heat, with a corresponding condensation. It is customary to place a separator in the main just before it reaches the engine, so that this water of condensation can be removed from the steam.

146. Clearance and Compression.—In order that the piston may not strike the end of the cylinder, it is necessary to leave a small space between the piston and the cylinder head. In addition, there is always some space in the steam ports between the valve and the cylinder. The volume of the ports between the valves and the cylinder, together with the space between the piston at the end of its stroke and the cylinder head, is called the *clearance volume.* It is usually determined by placing the piston at the extreme end of its stroke and filling the clearance space with water. Knowing the weight and the temperature of the water put into the clearance space, the volume of the water may be determined. Dividing the volume of the clearance by the volume of the piston displacement gives the *per cent of clearance.* The clearance in engines varies from 1 to 10 per cent. The steam in the clearance affects the expansion curve of the engine.

In Fig. 151, ED represents the piston displacement, and AB represents the volume of the steam admitted to the cylinder. The apparent ratio of expansion is

$$\frac{ED}{AB}. \tag{2}$$

Actually, however, the steam expanding includes not only the steam admitted from the boiler, but also the steam left in the clearance, so that the real ratio of expansion is

$$\frac{ED + AF}{AB + AF}. \tag{3}$$

Changing the clearance volume of an engine will alter the amount of steam consumed per stroke. In order to reduce the amount of live steam necessary to fill the clearance at each stroke, the exhaust valves of the engine are closed before the end of the stroke; for the balance of the stroke the steam is compressed. This compression of the steam serves to fill the clearance space with steam at a higher pressure than the exhaust pressure. In addition, compression of steam in the clearance space serves to retard the reciprocating masses in the engine and bring them back to rest at the end of the stroke. If an engine is operated with too little compression, it will be found to pound at the end of the stroke. The effect of compression, or the cushioning of the

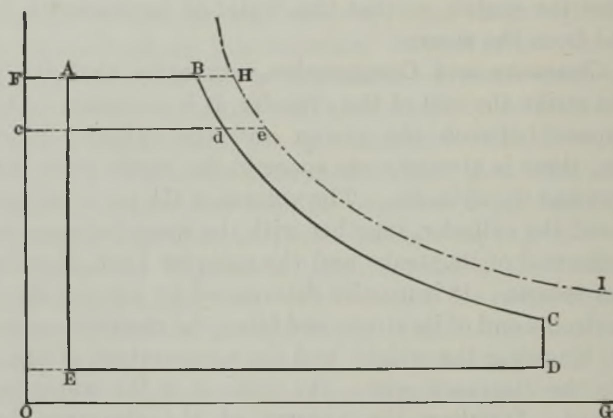


Fig. 151.—Theoretical indicator diagram, showing clearance.

piston, is materially increased by the *lead* of the engine. The lead is the amount the valve is open when the piston reaches the end of its stroke. In order to have lead, it is necessary to open the valves before the end of the stroke, and this steam admitted before the end of the stroke serves to assist in cushioning the piston and reciprocating parts.

The steam occupying the cylinder may be considered to be composed of two parts: the part that has been left in the clearance, which is called *cushion steam;* and the part that has been admitted from the boiler, which is called *cylinder feed.* The sum of the cushion steam and the cylinder feed gives the total weight of steam in the engine cylinder during expansion, *i.e.*, from B to C (Fig. 151).

The compression of the steam in the clearance space always involves a loss. Just previous to compression, the cylinder walls have been exposed to the exhaust steam. During compression the steam compressed has its temperature increased, and when the temperature of the compressed steam exceeds the temperature of the walls, condensation begins to occur. The action is similar to initial condensation.

147. Valves.—An essential part of every steam engine is the valve. The function of the valve is to admit steam to the cylinder at the proper time in the stroke, and on the return stroke to open the cylinder to the exhaust and let the steam escape either to the atmosphere or to the condenser. The proper action of the engine depends very largely upon the proper distribution of steam in the cylinder.

In a *single-acting* engine, steam is admitted to one side of the piston only; in the *double-acting* engine, it is admitted alternately, first to one side and then to the other. Most steam engines in common use are double acting.

In the simpler forms of steam engines, only one valve is used, which is so arranged that it admits steam to either end of the cylinder and also controls the exhaust.

148. Lap, Lead, Angular Advance, and Eccentricity.—Consider a valve such as is shown in Fig. 152. This valve is constructed so that it just covers the steam ports A and B. If the valve is moved to the right, or to the left, steam will be admitted to the cylinder at one end or the other, and exhausted from the opposite end. A valve constructed as shown will admit steam to the end of the stroke and permit the exhaust to continue to the end of the stroke at the opposite side of the piston. There would then be no expansion of the steam on the working stroke and no compression of steam on the exhaust stroke. The indicator diagram for a valve such as is shown in Fig. 152 is a simple rectangle.

In direct-acting steam pumps the valves have no lap or lead and the indicator diagrams are rectangular in general form. The economy of such pumps is very poor.

In order partially to expand the steam and to have compression at the end of the exhaust stroke, it is necessary that the valve be lengthened as shown in Fig. 153. The lengthening of the valve on the steam side causes the port to be closed before the end of the stroke, and for the balance of the stroke the steam expands. The increased length of the valve on the steam side of the valve is called the *steam lap*. The steam lap in Fig. 153 is the distance S. Steam lap may be defined as *the distance that the valve, when in its mid-position, extends beyond the edge of the steam port toward that side from which it takes steam.* It is equal to the distance the valve must move from its mid-position before steam is admitted to the cylinder. The lap is not always the same for the two ends of the cylinder.

In order to have compression at the end of the exhaust stroke, the valve must extend beyond the exhaust port by an amount called the *exhaust lap* E (Fig. 153), which may be defined as follows: *Exhaust lap is the distance that the valve, when in its mid-*

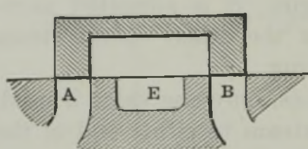


FIG. 152.—Simple valve without lap.

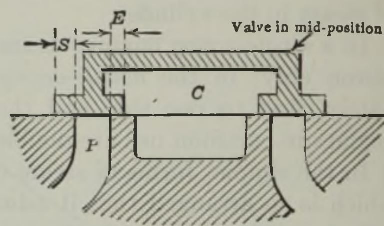


FIG. 153.—Valve with steam and exhaust lap.

position, extends beyond the edge of the steam port toward that side into which it exhausts. It is equal to the distance the valve must move from its mid-position before exhaust begins. The exhaust lap may be *negative,* in which case it is equal to the amount the port P (Fig. 153) would be open to the exhaust chamber C when the valve is in its mid-position.

If the valve did not begin to admit steam until just as the engine was on the dead center, full steam pressure in the cylinder would not be attained until the piston had traveled some distance on the next stroke. In order to have full steam pressure in the cylinder at the beginning of each stroke, it is necessary for the valve to open just before the piston reaches the end of the previous return stroke, thus causing *preadmission.* This opening before the end of the stroke is called the *lead.*

Lead is the amount the steam port is open when the piston is at the end of its stroke, and the engine is on dead center.

The valve is driven by the eccentric (Fig. 157) and if it were to be constructed as shown in Fig. 152, the eccentric would be set exactly 90 deg in advance of the position of the crank. But with the valve having both lap and lead, it is necessary to set the eccentric ahead of the crank an angle greater than 90 deg by an amount sufficient to move the valve a distance equal to the lap plus the lead. This angle is called the *angle of advance.*

The angle of advance is the angle which the perpendicular to the line of motion of the piston makes with the center line of the eccentric

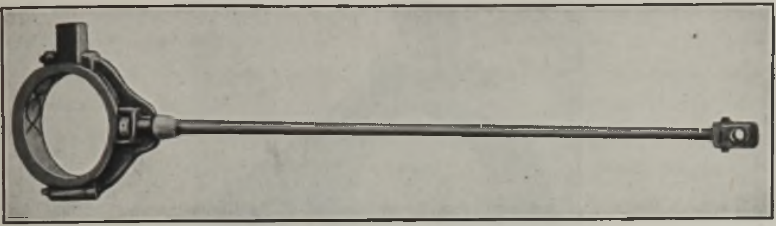


Fig. 154.—Eccentric strap and eccentric rod.

when the engine is on the dead center; or, it is the angle between the center lines of the eccentric and the crank minus 90 *degrees.*

Eccentricity is the distance between the center line of the shaft and the center of the eccentric.

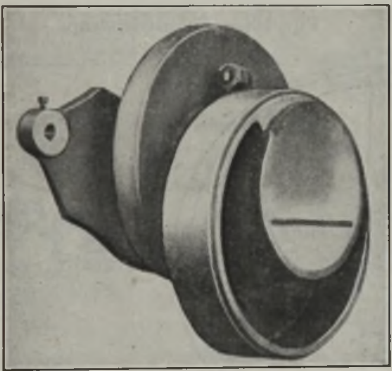


Fig. 155.—Eccentric sheave for shaft governor.

The throw of the eccentric is equal to the travel of the valve, or to twice the eccentricity.

Figure 154 shows an eccentric strap and eccentric rod. The eccentric strap is driven by an eccentric sheave, the position of which is determined by the governor. Figure 155 shows the eccentric sheave.

Figure 156 shows the eccentric strap more in detail. The strap is made in two parts and bolted together so that it can be placed over the sheave.

Figure 157 shows a governor, eccentric rod, rocker shaft, valve stem and valve.

149. Plain D-slide Valves.—Figure 158 shows an engine cylinder with a plain *D*-slide valve, so called from its longitudinal

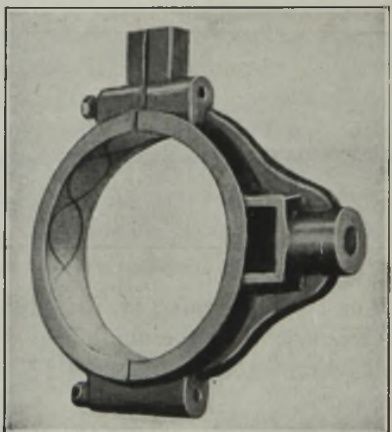


FIG. 156.—Eccentric strap.

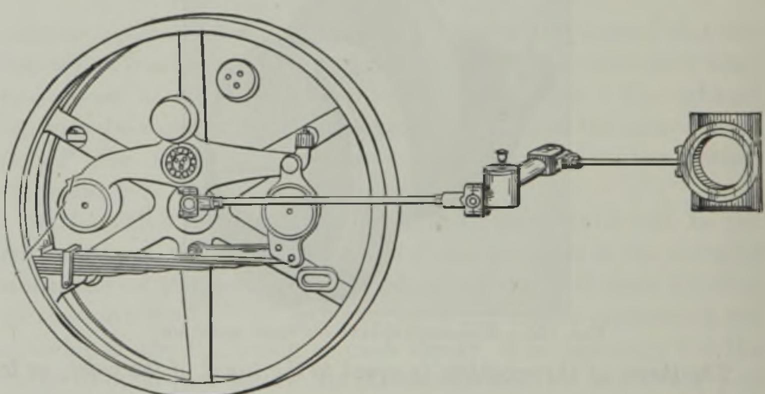


FIG. 157.—Governor, eccentric rod, rocker shaft, valve stem and valve.

cross section. The space D is filled with live steam under pressure, and the space C is open to the exhaust. In the position shown, steam has just ceased flowing from the space D, through the steam port A, into the cylinder. On the other side of the piston, steam is exhausting through the steam port B into the exhaust space C. The valve is moving to the left and the piston to the right, and the point of cutoff has just been reached. The steam will now expand in the head end of the cylinder until the valve has moved far enough to the left to uncover port A, placing it in communication with the exhaust port C, when exhaust from the head end will begin. Compression in the right end of the cylinder will begin when the valve has moved far enough to the left to cover port B. When it has moved still farther to the left, port B will again be uncovered and steam will be admitted to the right end of the cylinder, driving the piston toward the left. A plain D-slide valve will, therefore, if given a proper reciprocating motion, control the admission and the

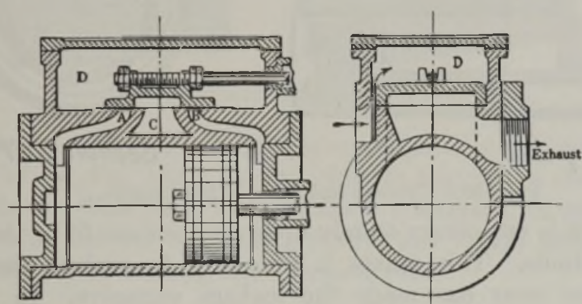


FIG. 158.—D-slide valve.

exhaust of the steam so that the piston will be given a reciprocating motion.

150. Piston Valves and Other Balanced Valves.—The plain D-slide valve, although entirely satisfactory under certain conditions, has a number of inherent faults which preclude its use in many cases. Prominent among these is the amount of resistance to movement which it offers when used with high-pressure steam. An examination of Fig. 158 shows that the entire back of the valve is exposed to live steam, with the result that it is pressed against its seat with great force and, in consequence, a large frictional resistance must be overcome in moving it.

By using a piston valve, as illustrated in Figs. 159[1] and 160,[1] this difficulty is overcome, and, as it is commonly expressed, the valve is perfectly "balanced," since the pressure upon the valve acts radially around its entire circumference. In the plain D-slide valve, leakage of steam past the valve is prevented by the

[1] From Fessenden's "Valve Gears."

fact that it is held tightly against its seat by the steam pressure. In the piston valve no such force is present, and in stationary

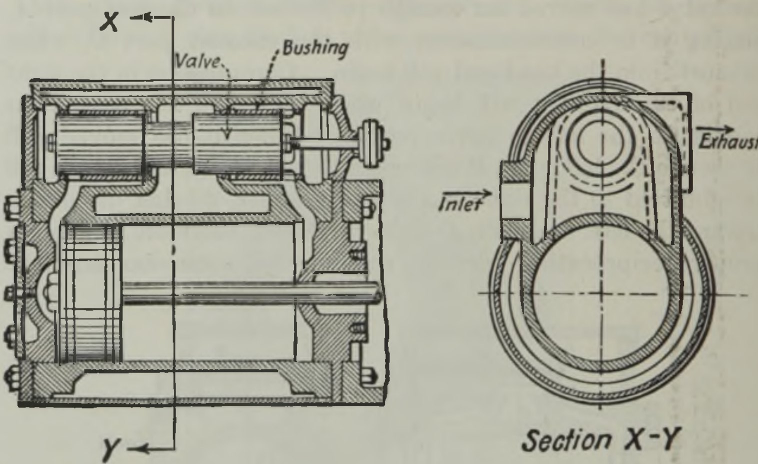


FIG. 159.—Cylinder and valve of Ideal engine.

engines it is customary to rely upon an accurate fit of the valve for tightness. This makes it necessary to replace the valve when the wear has made the leakage excessive. In marine

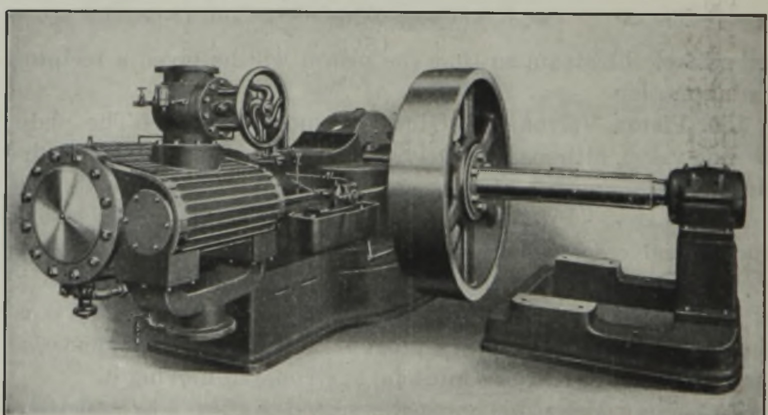


FIG. 160.—Ideal engine.

practice, tightness is obtained by the use of spring rings similar to those used on a piston.

The piston valve is the exact equivalent of the plain D-valve, since it may be considered to be formed by rolling the flat working surface of the plain D-valve into a cylindrical form. The piston valve is used extensively in marine engines, compound locomotives, and also in a number of types of high-speed stationary engines.

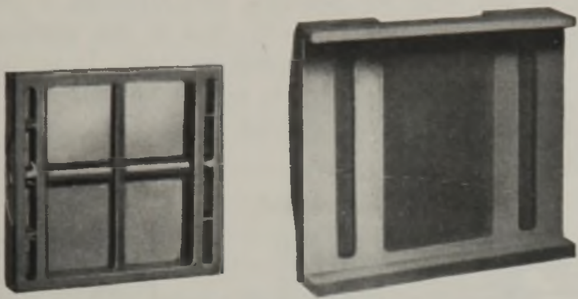


Fig. 161.—Double-ported valve with cover plate.

The valve often used in high-speed engines is the one shown on the left in Fig. 161. To the right in Fig. 161 is shown the cover plate. This cover plate is made a scraping fit when it is placed over the valve. This prevents any steam pressure on top of the valve, making it a balanced valve.

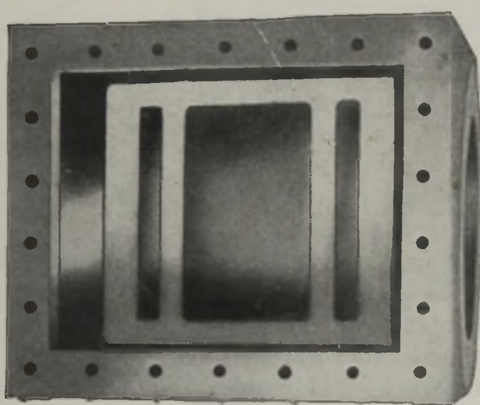


Fig. 162.—Steam chest showing valve seat.

Figure 162 shows the valve seat in the steam chest. The valve and its cover plate are fitted to this seat. The whole arrangement is shown in cross section in Fig. 142. The steam ports are at the ends of the steam chest, and the exhaust port between them.

151. Double-ported Valves.—In cases where the cutoff in the cylinder is at one-fourth stroke, the results obtained will show that so short a cutoff in the *D*-slide valve is a practical impossibility. The eccentricity and steam lap for one-fourth cutoff are entirely too large for practical use, although a one-fourth cutoff is not extraordinarily early, but is the working cutoff used in the majority of high-speed engines. It is thus quite evident that a simple *D*-valve is not at all suitable for early cutoffs and some modification must be made to obtain a satisfactory form.

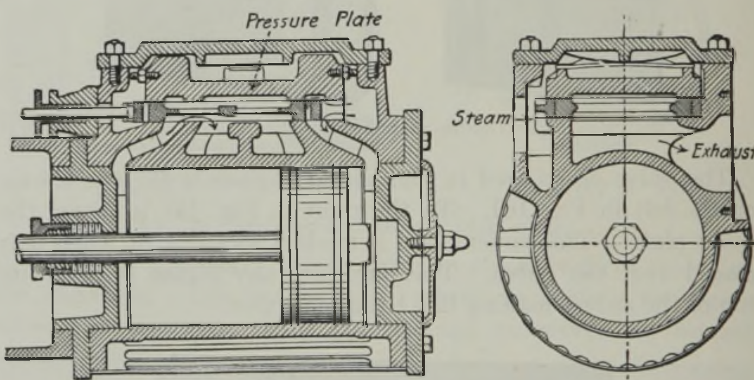


Fig. 163.—Modern high-speed engine valve.

Figure 163[1] shows a modern high-speed engine valve suitable for early cutoff. At the right end of the valve, admission is begun and the steam is entering the port past the end of the valve in the ordinary manner. At the same time a flow of steam past the upper corner is taking place. This steam passes through a port in the valve and enters the cylinder with the steam coming directly past the lower corner. The advantage of this arrangement lies in the fact that for any given movement of the valve, the port opening is twice as large as for the simple *D*-valve, since there are two ports instead of one. In other words, for a double-ported valve with a given cutoff, port opening, and lead, the eccentricity and the steam lap are one-half as large as for a plain *D*-slide valve giving the same steam distribution.

Another important feature of this valve is the pressure plate which extends over the entire back of the valve, thus relieving

[1] From Fessenden's "Valve Gears."

it from the action of the steam pressure, and consequently reducing wear and friction. The pressure plate extends around the side of the valve and rests upon the valve seat. It is held in its place by a flat spring at the back when there is no steam present in the steam chest. In case a large quantity of water is present in the cylinder during compression, the spring allows the pressure plate and the valve to lift from its seat, thus permitting the water to escape instead of bursting the cylinder as it would otherwise do. This form of valve may be restored to a tight condition when worn by planing off the faces of the pressure plate that bear against the valve seat, thus reducing the clearance between the valve and the pressure plate.

The governor usually fitted in engines having this type of valve controls the speed by altering the point of cutoff as the load changes. This variation in cutoff is effected by changing the position of the eccentric center in one of three ways:

1. By revolving the eccentric around the shaft, thus keeping the eccentricity constant while varying the angle of advance.

2. By moving the eccentric in a straight line at right angles to the crank, thus altering both the eccentricity and the angle of advance, but keeping the lead constant.

3. By moving the eccentric center in a circular arc, the center of which is on the opposite side of the arc from the shaft center, and very frequently directly opposite the crank. In this case the angle of advance and the eccentricity are both varied, but not in the same way as in the second type.

Of these three forms, the third is the most common, largely because it is the most convenient. With the eccentric swung from a point opposite the crank, as the cutoff is shortened the lead is reduced, while the points of release and compression are made earlier. For cutoff as early as one-fourth stroke, the points of release and compression are very much too early for low-speed engines, but not objectionably so for the high-speed engines in which this valve gear is used.

152. Relative Position of Valve and Piston.—In order to study the action of the valve, it is necessary to know its exact position for each position of the piston. As has been stated, the valve is driven by an eccentric, which is really a crank in which the crankpin is enlarged until it includes the shaft. As the size of the crankpin has nothing to do with the motion produced in the rod attached to it, any two cranks having the same arm, or distance between the shaft center and the crankpin center, will produce the same motion. In the eccentric the arm is called the *eccentricity*. As the eccentric is equivalent to a crank, the problem consists in finding the simultaneous positions of two reciprocating pieces, driven by two cranks upon the same shaft.

In Fig. 164 let OC represent any position of the crank, OD the center line of the eccentric, α the angle between the two, and BC the connecting rod. Drawing the arc CI with B as a center, it is found that the piston has moved a distance EI from its extreme position at the left, or its distance from its mid-position is OI. Similarly, by dropping a perpendicular from D upon EH, the

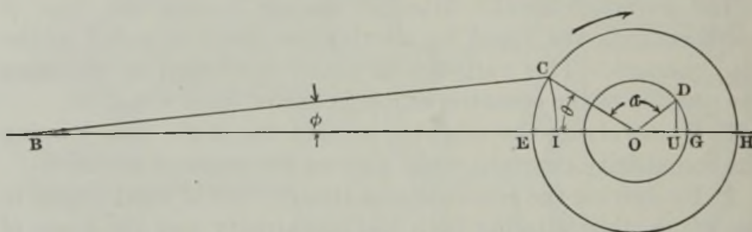


FIG. 164.—Relative position of valve and piston.

valve is found to be at the distance UG from its extreme right position, or distance OU from its mid-position.

To be absolutely correct, an arc should be struck through D with a radius equal to the length of the eccentric rod and with the center on the line OB, or OB extended to the left, and the point U found as the intersection of this arc with the line EH, rather than as the foot of the perpendicular dropped from D. However, as the ratio of the length of the eccentric rod to the length of the eccentric arm, or eccentricity, is so great, the error caused by using the perpendicular instead of the arc is negligible.

153. Valve Diagrams.—The method indicated in Fig. 164, although the most apparent way of attacking problems involving the analysis of valve movement, is inconvenient in practice, and simpler constructions, known as *valve diagrams*, are commonly used. Of the many forms of valve diagrams which have been devised, those due to Zeuner and Bilgram are perhaps the best known. Neither will be discussed here as they go into valve analysis in detail more than is felt necessary for this text.

154. Changing the Direction of Rotation.—Horizontal engines rotating in a clockwise direction with the cylinder at the left of the shaft, or, in other words, taking steam in the head end of the cylinder while the crank passes through the upper half of its path, are said to "run over." To produce rotation in the opposite direction, or to make the engine "run under," it is only necessary to lay off the angle α in the opposite direction from the crank. That is, to set the eccentric at an angle of 90 deg + δ (the angle of advance) from the crank, measured in a counterclockwise direction.

For many purposes, engines are required that can be reversed, or made to run in either direction, at the will of the operator.

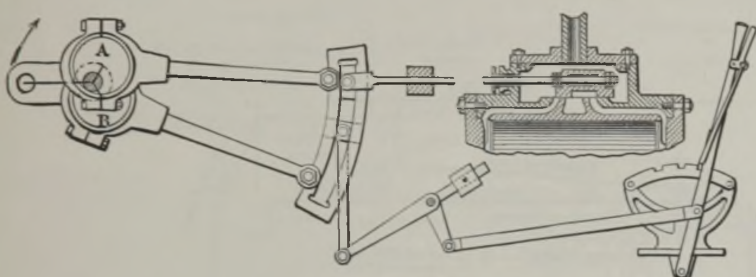


Fig. 165.—Stephenson link motion.

By arranging the eccentric so that it could be revolved through an angle of 180 deg − 2δ, the engine would be made reversible. This arrangement has actually been used, though it is now practically obsolete. Instead of this construction, mechanisms known as *reversing gears* are used, which, besides making the engine reversible, permit a variation in the point of cutoff.

155. Stephenson Link Motion.—In 1842, Robert Stephenson and Company applied to their locomotives a form of reversing gear which has received the name of the Stephenson link motion. This has been more widely used than any other type of reversing gear. This gear, as shown in Fig. 165, has as its essential feature a curved piece, or link, connected at its ends to the rods of the two eccentrics. On the end of the valve stem is a block, fitted to slide in the link and free to turn on a pin carried by the valve stem. By means of a bell crank and suspension rods connecting it to the link, it is possible to raise or lower the link, and so cause the valve to take its motion from any desired point along the arc of the link. One end of the link is connected to an eccentric for the "go-ahead" position, and the other end of the link to an eccentric set for the "back-up" position. When the block is thrown to the end controlled by the go-ahead eccentric, the valve is moved so as to drive the engine forward; when thrown to the opposite end, the engine reverses. As the block is moved nearer the middle of the link, both eccentrics affect the motion of the valve, and the cutoff is shortened. When the middle of the link is reached, admission and cutoff are found to occur at equal crank angles on either side of the dead-center position and the engine has no motion. Beyond the mid-position, the motion of the engine is in the opposite direction.

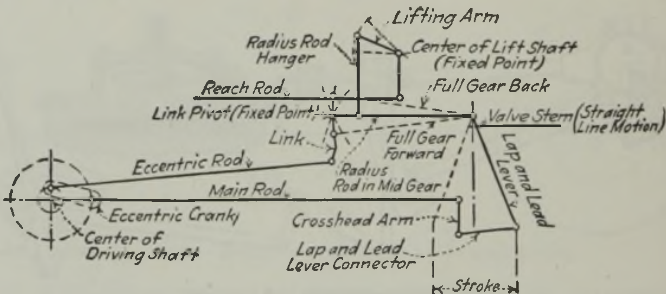


FIG. 166.—Diagram of Walschaert valve gear.

In American locomotives a rocker arm is always placed between the link block and the valve stem. This arrangement causes the valve and the link block to move in opposite directions. For this reason each of the eccentrics is placed at an angle of 180 deg from the position shown in Fig. 165. In marine practice, the link block is usually carried on the end of the valve stem, as shown in the figure.

156. Radial Gears.—In addition to the Stephenson link motion, a number of other types of reversing gear are in more or less common use. One class of these, known as *radial gears*, have either one eccentric and derive part of their motion from the connecting rod, or are entirely without an eccentric and derive their entire motion from the connecting rod. The most important of these is the Walschaert gear, which is commonly used on locomotives. A diagrammatic sketch of this gear is shown in Fig. 166. On the outer end of the crankpin, a second crank is carried, which is connected with the link in such a way as to cause it to oscillate about its point of support. The valve stem is connected to the vertical lever which derives its motion both from the block carried on the link, and from the crosshead of the engine. By setting the block at different points along the link, the cutoff may be varied or the engine reversed. With the Walschaert gear, the lead remains the same for all cutoffs, instead of increasing when the cutoff is made earlier, as in the Stephenson gear.

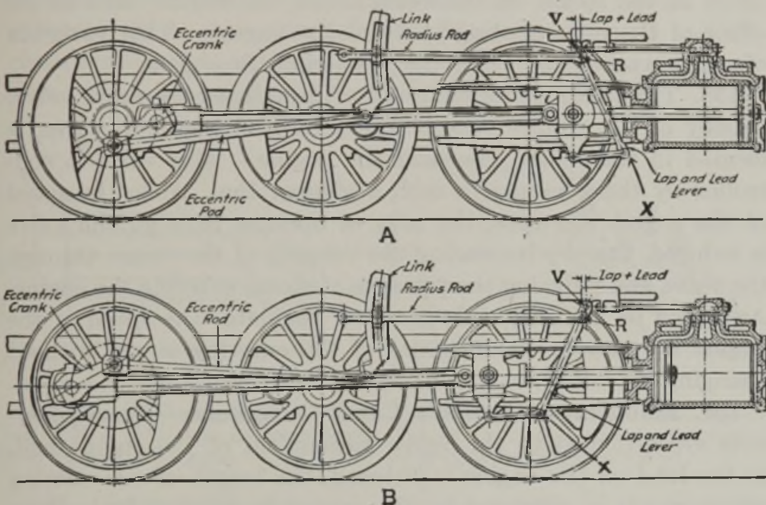


Fig. 167.—Walschaert valve gear as applied to a locomotive.

Figure 167[1] shows a Walschaert valve gear as applied to a locomotive. Two positions are shown: *A* when steam is being admitted to the forward end, and *B* when it is being admitted to the rear end of the cylinder.

Another method which may be used for reversing engines having a balanced slide valve is to change, by means of a three-way cock, the steam ports into exhaust ports and the exhaust ports into steam ports. This method should be used only for engines under 10-in. bore and under 200 rpm.

157. Governors.—In stationary-engine practice it is essential that the engine operate at a uniform speed irrespective of the power that it develops. In most cases the load on the engine is continually varying, requiring a constant change in the amount of

[1] From Fessenden's "Valve Gears."

power given by the engine. Two general forms of governors are used for this purpose: the *throttling* governor, which regulates the pressure of steam entering the engine, the cutoff remaining constant; and the *automatic cutoff* governor, which regulates the volume of steam admitted but does not change the pressure.

In addition to the changes of speed brought about by the change of external load on the engine, there is also a change of speed during each revolution of the engine due to the variable effort of the steam on the crankpin of the engine, and to the effect of the reciprocating parts of the engine. This variation of speed is taken care of by the *flywheel* of the engine.

158. Throttling Governors.—In a throttling governor a valve, usually of the poppet type or other form of balanced valve, is located in the steam pipe near the engine. This valve is controlled by the governor in such a manner that, when the speed of the engine increases, the area of opening through the valve is reduced, thereby increasing the velocity of the steam through the valve and reducing the pressure of steam entering the engine. As stated in Art. 157, this governor regulates the speed of the engine by varying the pressure of the entering steam without changing the cutoff.

159. Automatic or Variable Cutoff Governors.—These governors are attached to the valve mechanism of the engine and, as the load on the engine is reduced, the length of time during which steam is admitted to the engine is reduced by making the cutoff come earlier. Thus, as the load becomes less, less steam is admitted to the engine, but the pressure of the steam remains unchanged.

160. Relative Economy.—The indicator diagrams shown in Fig. 168 are taken from an engine using a throttling governor. This figure shows a number of diagrams taken at different loads. Under a light load, owing to the action of the governor, the steam pressure is very low; under a heavy load, the diagram shows high pressure. At the light load the steam is expanded almost to atmospheric pressure, but at the heavy load, the cutoff being kept the same, there is a very small expansion. This condition is not favorable to economical operation.

Figure 169 shows a diagram similar to Fig. 168, but taken from an automatic engine. In this form of governing the initial pressure remains the same for all loads and the cutoff varies. This enables the engineer to select a load giving a cutoff at which an engine using a given steam pressure will show maximum economy. In most engines this is found to be about one-fourth stroke; therefore, an automatic engine should be operated with a load requiring the governor to maintain the cutoff as nearly as possible at this point.

Under most conditions this form of governor is more economical in its operation than the throttling governor. Actual experiment with an engine having both an automatic cutoff and a throttling governor shows the automatic cutoff governor to give a steam consumption of about 75 per cent of the steam consumption of the same engine operated with a throttling governor.

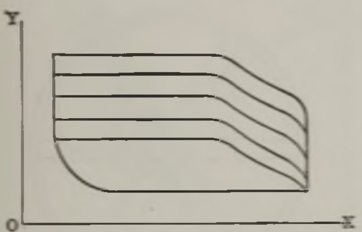


Fig. 168.—Indicator diagram, showing effect of throttling governor when load on engine is varied.

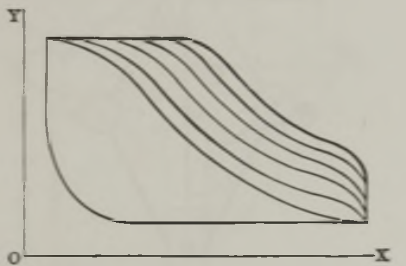


Fig. 169.—Indicator diagram, showing effect of automatic cut-off governor when load on engine is varied.

161. Governor Mechanism.—The mechanism of the governor which is to maintain the speed of the engine uniform must be such that the change of speed will cause a change in the position of the parts of the governor. There are two general types of mechanism used for this purpose. The *flyball* governor is the first type and consists of two balls fastened to pivoted arms and rotated by the engine, and, as the speed of the engine increases, the balls move out and change either the throttle valve or the valve mechanism.

The second type, the *shaft* governor, is fastened to the flywheel of the engine. It usually consists of two weights attached to the flywheel by arms. Since these arms are pivoted, as the engine speed increases, the governor weights move out against the resistance of a spring. The governor arms are attached to the eccentric, and as the weights move out, the position of the center of the eccentric changes.

162. Flyball Governors.—Figure 170 shows a line diagram of a flyball governor. B,B are the balls of the governor. These balls are suspended by arms AB, and are also attached to the weight W by the arms BC. The arms and balls of the governor rotate around the vertical spindle AC and are pivoted at the point A. The weight W is free to move in a vertical direction along the axis AC. As the speed of the engine increases, the balls of the governor move out into the dotted positions B',B'.

Through a connection not shown in the figure, the weight W is attached to the valve, or valve mechanism, and as the balls

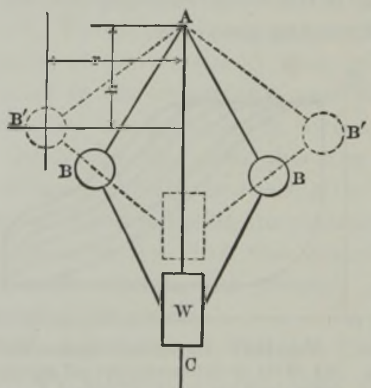


FIG. 170.—Line diagram of a flyball governor.

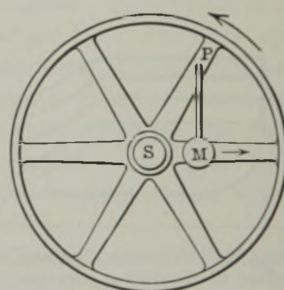


FIG. 171.—Elementary centrifugal governor.

move outward raising this weight, the steam pressure is throttled down, or the cutoff brought earlier, depending upon the type of governor being used.

Theoretically, the action of the governor is independent of the weight of the balls. Practically, there is more or less friction in the mechanism of the governor, and the balls must have considerable weight in order easily to overcome the friction of the governor.

163. Shaft Governors.—There are two forces either of which may be used to actuate the mechanism of a shaft governor. In the earlier forms of governors of this type, *centrifugal force* was commonly employed.

In a centrifugal governor (Fig. 171), the governor weight is so suspended that it moves approximately in a radial direction owing to the action of centrifugal force. As the speed of the wheel increases, the centrifugal force increases and the weight M moves out against the resistance of the spring. In the actual

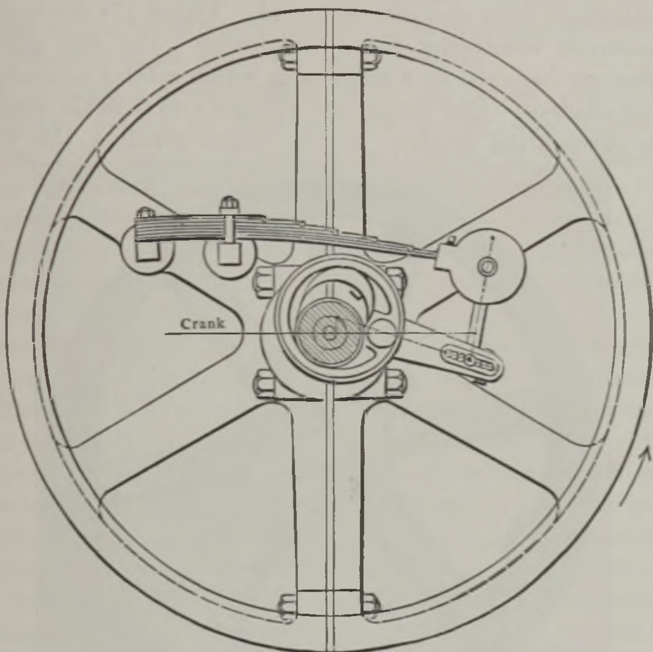


FIG. 172.—Actual construction of centrifugal governor.

construction of the governor as shown in Fig. 172, the centrifugal force acts against the resistance of a spring.

The governor in this case regulates the position of the eccentric. The angular advance and eccentricity are changed at the same time, leaving the lead almost constant for all positions of the governor.

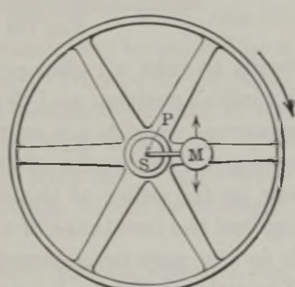


FIG. 173.—Elementary inertia governor.

In Fig. 173 the weight M is fastened so that centrifugal force has no effect upon its movement but produces only a stress in the arm SM. But, if the wheel were suddenly stopped, the weight would continue to move, owing to the *inertia*, and exert a force upon a spring (not shown) against the resistance of which the governor ball acts. The motion of this weight is arranged to change the position of the valve. Inertia alone is not used as the actuating force, but a combination of centrifugal force and inertia is used. Figure 174 shows a form of governor combining these two forces. The two governor weights are fastened to a single arm which rotates around a pin (shown shaded). One weight has a longer arm than the other and is the dominating weight. As the engine revolves, this weight

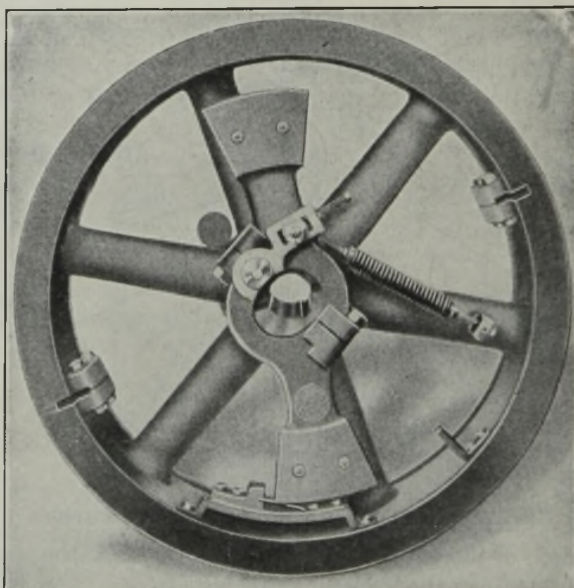


Fig. 174.—Actual inertia governor.

tends to take a radial position. This action gives the governor its initial position and determines the position of the valve. The governor weights are suspended so that if the speed of the engine changes, the inertia of the weights moves the governor against one or the other of the stops shown. The governor weights act against the resistance of a spring. The speed at which the engine is to run may be changed by changing the tension of this spring. The valve is driven by a pin fastened to the governor arm.

164. Isochronism.—An *isochronous* governor is one in which the balls (Fig. 170) are in equilibrium at one speed and only at one. Any variation from this speed will send the balls to the limit of their travel in one direction or the other. Friction makes it impossible for a governor to be perfectly isochronous, but approximate isochronism is obtained by using crossed arms, so that the governor balls have a parabolic path, and the height h will remain approximately constant. In some forms of governors the balls are guided in a parabolic guide, so that their motion is an exact parabola which gives h a uniform value.

165. Hunting.—Oversensitive governors often exhibit the phenomena known as *hunting*. No matter how quickly a governor may change its position in response to a demand for more or less steam, the engine does not respond instantly. This is in consequence of the energy stored in the moving parts of the engine, and in the element of time that must elapse between the moment when the steam is admitted by the governor and the time that it acts on the piston. Therefore, when a sudden demand for power is made on an engine in which the governor is too sensitive, or too nearly isochronous, the drop in speed will be sufficient to force the governor into a position of overcontrol, so that too much steam is admitted. This causes the revolutions to increase beyond the desired point and the same overcontrol is exercised in the opposite direction. In other words, the governor balls, or weights, fly first in one direction and then in the other, "hunting" for the position of equilibrium. The effect is to make the speed of the engine change rapidly, first having an excess of speed, and then a speed below the normal. This trouble may be overcome by adding a small weight to one of the governor balls, or changing the tension of the governor spring, or both.

166. Practical Considerations.—When a properly designed engine does not govern properly, the trouble is often due to undue friction in the valve mechanism, which may be caused by a tightening of the glands or the journals, or by friction in the dash-pot and springs. It may also be due to excessive leakage in the valve, unbalancing it, or by the valve's being too tight. The governor should also be examined to see that the weights have not been changed. The tension of the springs should be uniform, if more than one spring is used.

If the engine operates at a lower speed than that desired, the tension of the governor spring should be increased. If this tension has been increased to the limit of the spring, additional weight should be placed in the governor balls.

In all forms of governors it is necessary that the friction of the valve mechanism be made as small as possible, and it should, if possible, be a constant quantity. It is better to have balanced valves, where they are directly operated by the governor, and the valves should have small travel. In the D-slide type of valve, small travel is obtained by using a double-ported valve.

In direct-connected engines, 2 per cent variation in speed is the maximum allowable, and most specifications require the variation to be less than 1 per cent. In mill engines a variation of 5 per cent is sometimes allowed.

167. Flywheel.—The governor of an engine confines the speed variation within certa n limits by controlling the action of the valve. It takes a few revolutions, however, to bring the governor into action. In addition to this variation in the number of revolutions per minute, an engine has fluctuations of speed that occur in the fraction of a revolution. These fluctuations must be controlled by a *flywheel*, and are due to three principal causes:

1. The pressure of steam is not the same at all points of the stroke.

2. The motion of the piston is carried to the shaft by the connecting rod and crank. This means of changing reciprocating into rotary motion causes a turning effort, which varies from zero to a maximum.

3. The reciprocating motion of the engine piston and other parts necessitates these parts being brought to rest and started again twice each revolution. The overcoming of the inertia effect, caused by the action described, causes a variable force to be transmitted to the crank.

The flywheel is fastened to the main shaft of the engine and its inertia serves to carry the engine over those portions of the stroke where the piston is not giving sufficient power to the shaft to carry the load.

The effectiveness of the flywheel depends upon the energy stored in it. As most of the weight of the wheel is in the rim, the action of the rim may be considered, for an approximation, as giving the flywheel effect. If W is the weight of the flywheel rim in pounds, and R is the average radius in feet, and the wheel makes N rpm, then the energy of the rim

$$= \frac{1}{2} mv^2$$

$$= \frac{W}{2g}\left(\frac{2\pi RN}{60}\right)^2$$

$$= \frac{WR^2N^2}{5,874} \text{ ft-lb.} \qquad (4)$$

The expression shows that the effectiveness of a flywheel depends upon the weight of the rim, the square of the radius of the wheel, and the square of the number of revolutions that it makes.

168. Lubrication.—Although strictly not parts of the engine, the various devices used for oiling, or lubricating, are so closely associated with it that it seems appropriate to describe them here.

The object of all lubrication is to separate by a film of oil the wearing surfaces of the machine in question. Originally, the oil was supplied to these surfaces by hand, but the development of machine design has been followed by an evolution in various methods of automatic lubrication.

The following general classification of automatic lubrication is one published by the Texas Company in their bulletin, *Lubrication:*

1. Waste-pad lubrication.
2. Individual lubricating devices involving sight-feed drip cups, bull's-eye lubricators, and compression grease cups.
3. Wick-feed oiling devices.
4. Self-oiling systems, which include ring, chain, and collar oilers.
5. Gravity-feed systems.
6. Force-feed lubricating systems.
7. Mechanical force-feed lubricators.
8. Lubrication by means of splash methods.
9. Hydrostatic lubricators.

Of these systems, all except systems 7 and 9 are generally adaptable only to external, or bearing, lubrication. A main bearing with a chain oiler (system 4) is shown in Fig. 147. The splash system of lubrication (8) is commonly used in automobile and tractor engines, as well as certain types of steam engines. In it the crank case is closed and is partly filled with oil. The crank dips into this oil at each revolution and throws it over the various wearing surfaces within the case.

In addition to the external lubrication already referred to, engines must be lubricated internally, *i.e.*, oil must be supplied to the rubbing or sliding surfaces in the steam chest and cylinder. This means that the valves, valve rods, valve seats, cylinder walls, pistons, and piston rods are involved. The ordinary method of supplying oil to these parts is by feeding it into the steam line between the throttle valve and the steam chest. The steam atomizes the oil and carries it onto the moving parts within the steam chest and cylinder.

Two general types of lubricators are used for steam-cylinder lubrication: the *mechanical force-feed lubricator* (system 7) and the *hydrostatic lubricator* (system 9).

169. Mechanical Force-feed Lubricators.—These lubricators consist of a reservoir with one or more pumps located within this reservoir, each pump controlling the particular oil-feed line to which it is attached. The pumps are operated by connection with some moving part of the engine, the rate of oil feed being regulated by changing the stroke of the pump plunger, or by changing the rate at which it is driven. This type of lubricator functions only when the engine is running, so that there is little chance of wasted oil or flooding of the system. It is entirely positive in its action and the flow of oil is not affected by the change in viscosity accompanying a temperature change, as it is in the hydrostatic lubricator.

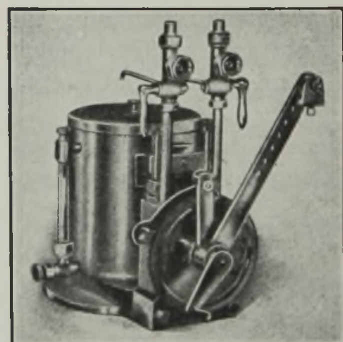


Fig. 175.—Ratchet-drive model Rochester automatic lubricator.

One form of force-feed lubricator is shown in Fig. 175.

When using either the force-feed or the hydrostatic lubricator, care must be taken to see that the oil is thoroughly atomized before it enters the steam chest. This is accomplished by installing a properly designed atomizer at the end of the lubricator delivery pipe. These atomizers are merely nipples inserted in the steam pipe and cut or drilled with holes, in such a way as to cause the oil to leave the nipple in a finely divided state, so that it will be uniformly distributed throughout the steam by the time the latter reaches the wearing surfaces in the steam chest and cylinder.

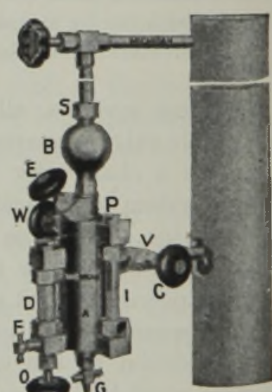


Fig. 176.—Connection of lubricator to steam main.

170. Hydrostatic Lubricators.—Figure 176 shows the connection of a hydrostatic sight-feed lubricator to the steam main, and Fig. 177 shows a cross section of such a lubricator. The operation of this lubricator does not depend upon the steam pressure in any way, as both connections are attached to the same main and receive the same pressure. The force that does operate it is

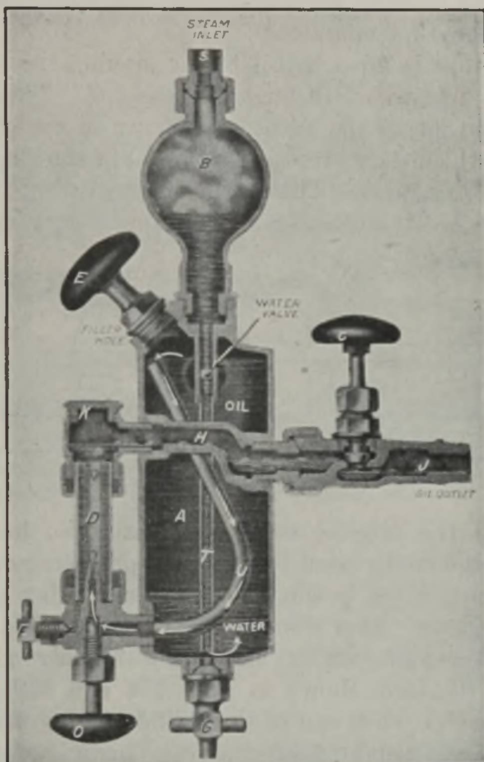


Fig. 177.—Cross-section of Michigan hydrostatic lubricator.

the weight, or pressure, of the water formed by condensation of the steam in the condenser *B* (Figs. 176 and 177) and the vertical connection above the lubricator (Fig. 176). This connection should be from 15 in. to 24 in. long. The water, being heavier than the oil, flows down past the water valve and through the pipe *T* (Fig. 177) into the bottom of the chamber *A*, forcing the oil to the top of the chamber and thence down through the tube *U*, past the feed valve *O*, out through the nozzle into the sight-feed glass *D*. As this glass is filled with water, the drop of oil from the end of the nozzle floats up through it to the top and then passes through the passage *H* and the support shank *J* and is discharged into the steam main and then carried on into the cylinder by the steam.

The water in the sight-feed glass is formed by condensation of steam entering through the oil outlet shank when the lubricator is being prepared for operation.

The lubricator is filled with oil by removing the plug *E*, and the water is drained off through the valve *G*. There is a gage glass *I* on the side of the lubricator (shown in Fig. 176, but not in Fig. 177) which shows the amount of oil in the chamber *A*.

171. Corliss Engines.—The Corliss engine, invented by George H. Corliss in 1849, and in its more recent forms varying only

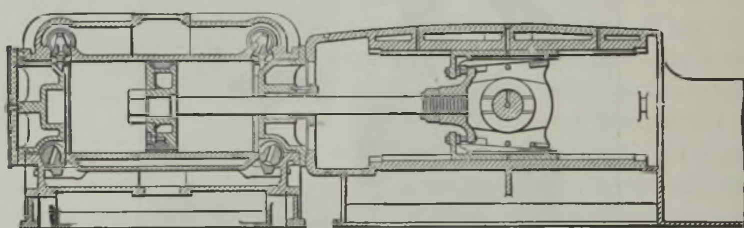


FIG. 178.—Corliss engine—cylinder and frame section.

slightly from the original engine of this type, has been one of the most commonly used forms of reciprocating engines, particularly in large sizes, in the United States. They give as high an economy as any other form of engine made. The distinctive features of this engine are the valves and the valve gear.

Valves of the form shown in Figs. 178 and 179 are used in the Corliss engine, each end of the cylinder being provided with separate admission and exhaust valves. Instead of sliding upon their seats with a straight-line motion like a common slide valve, these valves have an oscillatory motion about the common axis of the cylindrical seat and valve. In horizontal cylinders the admission, or steam, valves are placed above with their axes at right angles to the axis of the cylinder; the exhaust valves are similarly placed below. All four valves have spindles which extend through stuffing boxes to the outside of the cylinder, where they are rigidly connected to short cranks called *valve arms*. As shown in Fig. 179, these valve arms all derive their motion from the *wrist plate*, which is, in turn, oscillated by the eccentric rod. Valve rods permanently connect the arms of the exhaust valves to the wrist plate, but for the steam valves a trip gear is provided, which disengages the valve arm at the point of cutoff and allows the valve to close with a rapid motion. This sudden closure of the valve is due to its connection to the *dashpot* piston. As the valve opens, the dashpot piston is raised, producing a partial vacuum in its cylinder, so that, as soon as the trip gear releases the valve arm from its connection with the

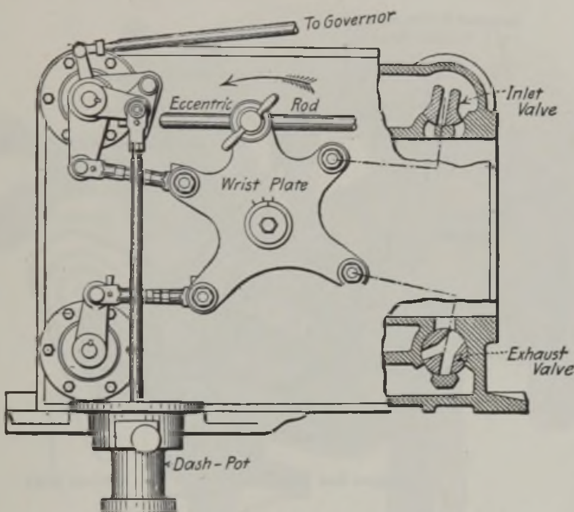


Fig. 179.—Corliss engine, showing arrangement of valves.

wrist plate, atmospheric pressure forces the dashpot piston down and closes the valve.

Figures 180 and 181 show the trip gear for the steam valve. The steam arm is keyed to the valve stem; as the outer end of the arm is raised or lowered, the valve is turned on its seat. The knockoff cam lever and the bell-crank lever are both free to oscillate about the valve stem as an axis. The valve rod connects one end of the bell-crank lever with the wrist plate, and, as the wrist plate oscillates back and forth, the bell crank is given a rocking motion about the valve stem. The other arm of the bell-crank lever carries a steam hook, the inner leg of which is kept in close contact with the knockoff cam lever by a spring. In the position shown in Fig. 180 the steam hook has engaged with a block on the outer end of the steam arm, and, as the valve rod is moved to the left, the steam hook is raised, pulling up with it the outer end of the steam arm and turning the valve on its seat, opening it. When the bell-crank lever has been turned about its axis until the point is reached where the inner leg of the steam hook strikes the knockoff cam, the outer leg will be forced to the right, releasing the steam arm, which will suddenly be pulled downward by the dashpot rod which is attached to it. This sudden movement of the steam arm closes the valve and

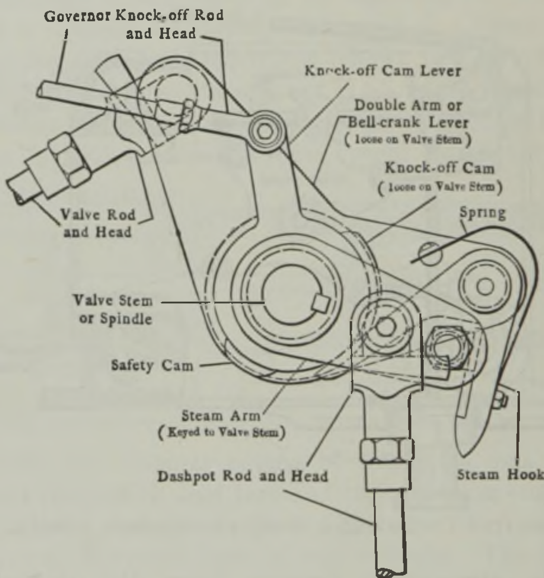


FIG. 180.—Line diagram of Corliss trip mechanism.

gives a sharp cutoff. The governor controls the position of the knockoff cam, thus determining the point at which the steam hook releases the valve arm and cutoff takes place. A safety cam is provided so that, in case the governor belt breaks, the dropping of the governor balls will rotate the safety cam in a counterclockwise direction, causing cutoff to occur so early that the engine will stop.

An analysis of the motion of a properly designed Corliss valve reveals two important points:

1. That the valve is moving at nearly its greatest velocity when the edge of the valve crosses the edge of the port.

2. That during the period when the valve is closed its motion is very slight.

The first of these features reduces the wiredrawing effect and makes the corners of the indicator card more sharply defined than is the case with simple slide valves. The second reduces the friction and the wear, since the valve is pressed against its seat by the full steam pressure during the large part of the period when the port is closed. The use of the trip gear makes the cut-

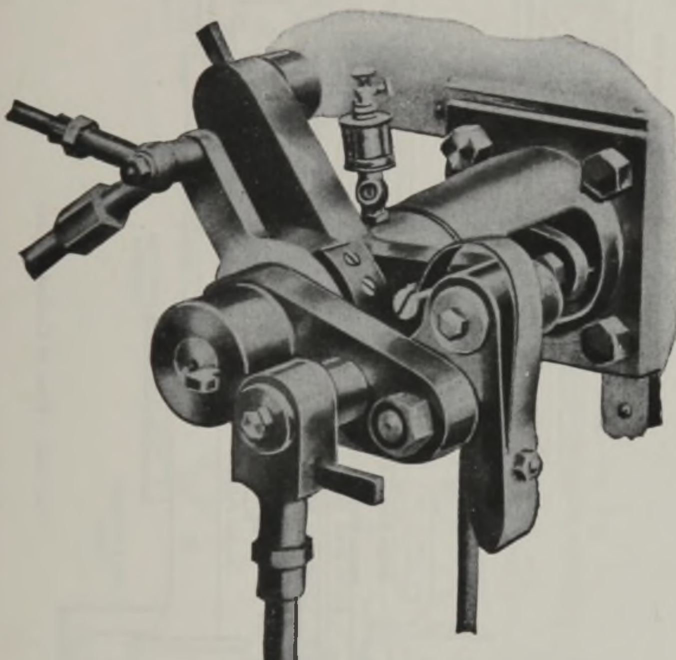


Fig. 181.—Corliss trip mechanism for steam valve.

off independent of all the other events, and consequently the lead and the points of compression and release remain the same for all loads. With the Corliss valve gear the combination of excellent steam distribution, slight leakage, and wiredrawing, with a minimum amount of clearance, is obtained, resulting in a high degree of economy.

Figure 182 shows a plan and Fig. 183 an elevation of a Corliss engine.

172. The Uniflow Engine.—In 1908, Professor Stumpf of Charlottenburg (Berlin), Germany, brought out what he called a

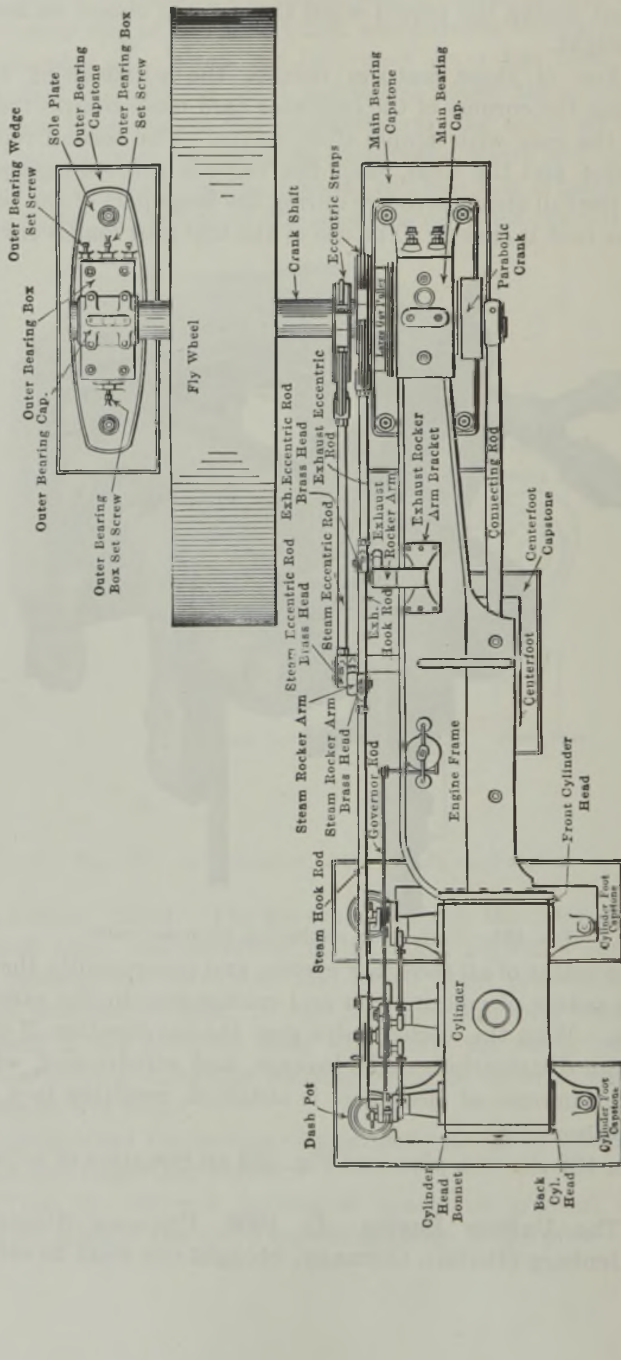


FIG. 182.—Plan of Corliss engine.

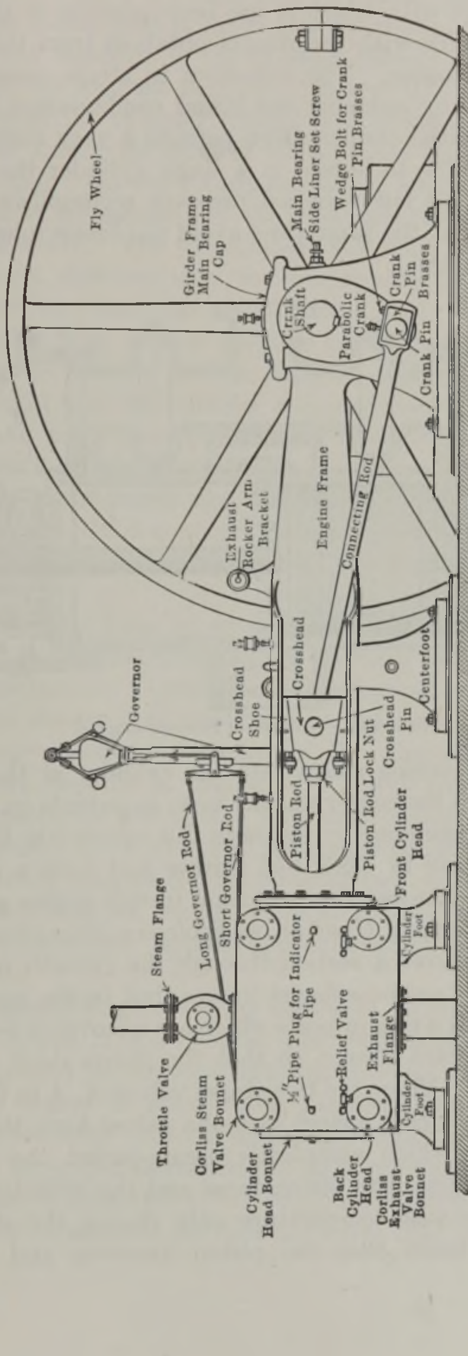


FIG. 183.—Elevation of Corliss engine.

Unaflow engine with which it has been possible to obtain economies comparable with the results obtained from the best type of compound engine. The reduction in steam consumption is brought about by reducing the initial condensation loss. This reduction of initial condensation permits a very early cutoff, so that it is possible to obtain in a single cylinder the high ratio of expansion and low terminal pressure accomplished by compounding, and at the same time avoid the losses inherent in the compound engine.

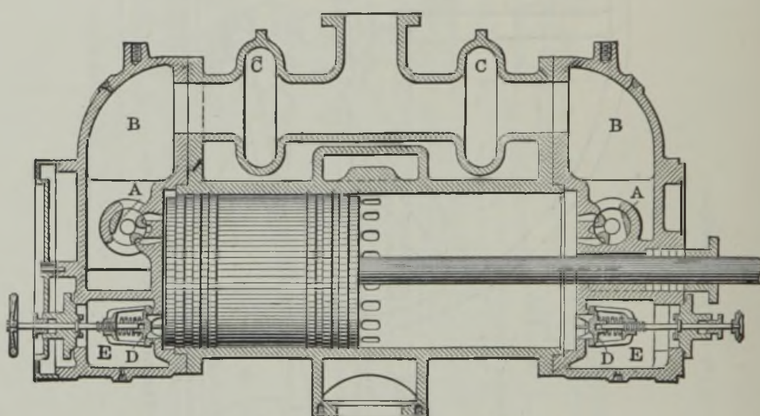


Fig. 184.—Section of Stumpf engine.

The idea of taking steam into the cylinder at the ends and exhausting it at the center was not new, as patents on this principle had been taken out by Eaton in this country in 1872 and by Todd in England in 1885, but neither had been a commercial success. The one-way flow of steam in this engine gave rise to the name *uniflow*, or *Unaflow*, as Professor Stumpf called it.

Figure 184 shows a section through the cylinder of a Stumpf engine. There are no exhaust valves, but in the middle of the cylinder there is a ring of ports which are uncovered by the piston at the end of each stroke, so that the piston itself acts as the exhaust valve. There are two steam valves A, A in the cylinder heads, and the steam spaces over the valves have the clearance pockets B, B which completely steam-jacket the heads. In the uniflow engine, the piston faces and the cylinder heads are exposed to exhaust temperature only during the short period between the times that the piston uncovers and covers the exhaust ports. Owing to the fact that the inlet valves are steam-jacketed, the coolest part in the engine cylinder is the face of the piston and any moisture in the cylinder collects on or near this face. When the exhaust ports are uncovered, this moisture is swept out from the cylinder, rather than being left in it to be reevaporated during the exhaust stroke, as is the case with the ordinary engine. On the return stroke the steam remaining in the cylinder is compressed in the clearance spaces up to the admission pressure. The temperature also increases in the compression space, due not only to compression but also to the absorption of heat from the jacketed head.

The cylinder (Fig. 184) is a simple cylindrical casting with a belt cast in the middle for the exhaust passage. The steam chest is integral with the cylinder and provided with two drums *C*, *C* to take up expansion without distorting the cylinder. Each cylinder head has a large valve *D* opening into a pocket *E*. This valve opens automatically to serve as a relief valve to let out entrained water. It also serves as extra clearance to prevent excessive compression pressure when the engine is operating noncondensing.

In the particular form of uniflow engine described, the inlet valves are of the Corliss type and are operated by the usual Corliss-valve mechanism. Other forms use poppet valves. These engines have shown very low steam consumption, particularly with superheated steam. In addition, they have a flat economy curve and are capable of taking very heavy overloads. This overload capacity is due to the fact that ordinarily the cutoff occurs very early, *i.e.*, at about 10 per cent of the stroke, while the maximum cutoff possible is from 60 to 65 per cent of the stroke. In most uniflow engines, release occurs at about 90 per cent of the stroke and compression begins at approximately 10 per cent of the return stroke.

173. Compound Engines.—Any engine in which the expansion of steam is begun in one cylinder and continued in another is called a *compound engine*, although this term as commonly used refers to an engine in which the expansion takes place in *two* cylinders *successively*. A *triple-expansion* engine is one in which the steam is expanded *successively* in *three* cylinders.

When steam is expanded in two or more cylinders successively, the number of expansions per cylinder is less than when only one is used; therefore, the range of temperature in each cylinder is less. Reducing the range of temperature in the cylinder reduces the condensation losses. *The principal object of compounding is to reduce the amount of steam used per horsepower per hour*, and, under proper conditions, compounding accomplishes this, owing to the reduction of initial condensation. The radiation losses from a compound[1] engine are usually larger than from a simple engine, and very often the mechanical losses are increased by compounding.

The tendency, then, in a compound[1] engine, is to increase the radiation loss and to increase the mechanical losses. On the other hand, compounding decreases the thermodynamic losses by decreasing the range of temperature in each cylinder. With low pressure and a small number of expansions (*i.e.*, a small ratio of expansion), a single-cylinder engine is more economical than a compound[1] engine, but with high-pressure steam and a large number of expansions the reverse is the case. The higher the pressure and the larger the number of expansions, the greater the economy of the compound[1] engine.

For pressures under 100 psi, the single-cylinder condensing engine is more economical than the compound engine. But for pressures above 100 psi the compound engine is usually more economical. In the noncondensing engine, the compound engine does not show any economical advantage until the pressure reaches 150 psi. The compound condensing engine becomes less economical than the triple-expansion engine for pressures greater than 150 psi.

The single-cylinder engine is more economical than the compound engine when the number of expansions of the steam is less than four. When there are from 4 to 6 expansions, there is very little difference in the economy. With from 6 to 15 expansions the compound engine is more economical. When the number of expansions exceeds 15 it is customary to use a triple-expansion engine.

174. Tandem-compound Engines.—A *tandem-compound engine* (Fig. 185) is one in which the two cylinders are placed one in front of the other. The pistons of the two cylinders are attached to the same piston rod, and there is but one connecting rod

[1] The term *compound* as here used includes triple expansion, quadruple expansion, etc.

and crank. The steam flows directly from the high-pressure cylinder into the low-pressure cylinder, and the connecting pipes are relatively small, there being no receiver except the piping between the cylinders. The tandem-compound engine occupies less space than the cross-compound. The principal objection to this form of engine is the difficulty of getting at the cylinder which is nearest the crankshaft. This is the earliest form of compound engine used.

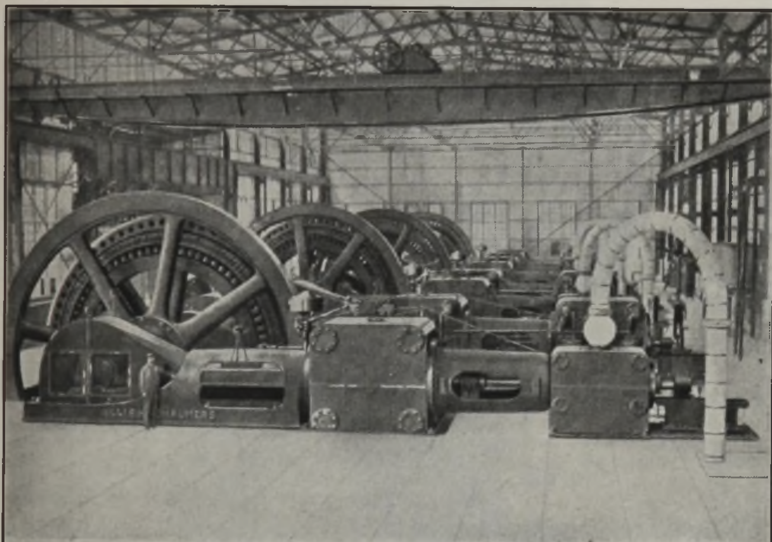


FIG. 185.—Tandem arrangement of cylinders.

175. Cross-compound Engine.—In the *cross-compound engine* (Fig. 186) the two cylinders are placed side by side, and each cylinder has its separate piston rod, connecting rod, and crank. The steam, after leaving the high-pressure cylinder, usually enters a steam reservoir called a *receiver;* from this receiver the low-pressure cylinder takes its steam. The cranks in a cross-compound engine are usually set 90 deg apart, so that when the high-pressure cylinder is at the beginning of its stroke the low-pressure cylinder is at mid-stroke. A cross-compound engine with cranks at 90 deg must always be provided with a receiver, as the low-pressure cylinder may need to take steam when the high-pressure cylinder is not exhausting. The cross-compound engine occupies a much larger space than the tandem

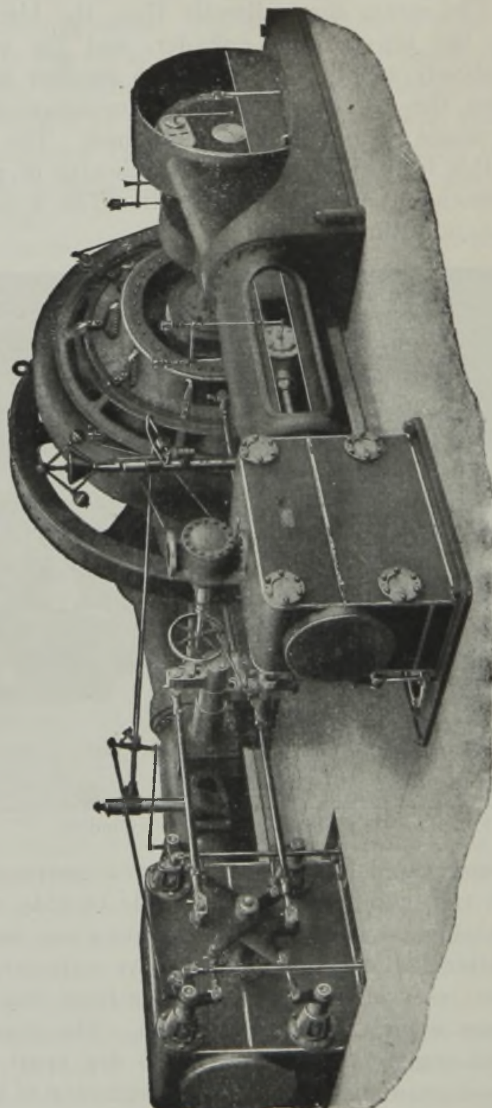


Fig. 186.—Cross-compound engine.

engine, but the parts are lighter. Each piston, crosshead, connecting rod, and crank does only approximately one-half the work that it would do in a tandem engine. The turning effort on the crankshaft is made more uniform by placing the cranks 90 deg apart. This reduces the size of the flywheel necessary to overcome the fluctuation in the speed of the engine, and also assists the governing.

A vertical cross-compound engine is often termed a *fore-and-aft compound.*

176. Ratio of Cylinders in the Compound Engine.—In the compound engine the strokes of the two cylinders are usually the same. If the ratio of the volumes of the two cylinders is represented by L, the diameter of the high-pressure cylinder by d, and that of the low-pressure cylinder by D, then

$$L = \frac{D^2}{d^2}. \tag{5}$$

The value of L should be such that the fall in pressure, termed *drop*, between the exhaust pressure in the high-pressure cylinder and the admission pressure in the low-pressure cylinder is small when the work of the engine is equally distributed in the various cylinders. The value of L varies from $2\frac{1}{4}$ to 4 for automatic cutoff high-speed engines, and from 3 to $4\frac{1}{2}$ for engines of the Corliss type. *L is equal to the quotient of the number of times the steam is expanded in the engine divided by the number of expansions in the high-pressure cylinder.*

The *ratio of expansion, r,* in a compound engine is equal to the *ratio of the total volume of the low-pressure cylinder, or cylinders, to that of the high up to the point of cutoff.* That is, it is, as in a single-cylinder engine, the ratio of the *final* to the *initial* volume occupied by the steam while in the engine.

This ratio r may be varied in an engine by varying the point of cutoff in the high-pressure cylinder. It is customary to proportion an engine and so set the valves that each cylinder does an equal amount of work. This, however, is not always the case, some engines being designed to give equal ranges of temperatures in the cylinders. Theoretically, this gives the best economy.

The proportion of work that is done by each cylinder may be adjusted by changing the low-pressure cutoff. The shorter the cutoff in the low-pressure cylinder, the less the steam taken from the receiver and the higher the pressure in the receiver. Increasing the pressure in the receiver causes a higher back pressure for the high-pressure cylinder and, consequently, less work done by that cylinder. Increasing the low-pressure cutoff will decrease the work done by the low-pressure cylinder. Theoretically, *changing the cutoff in the low-pressure cylinder does not change the gross horsepower developed by the engine;* in actual practice this does not hold absolutely true, although the change is very slight. The equalization of the work in the two cylinders cannot be accomplished in most engines, as in equalizing the work at different loads an excessive drop may be produced between the cylinders.

CHAPTER IX

POWER AND PERFORMANCE OF STEAM ENGINES

177. Power.—The power of an engine may be stated in terms of *indicated* horsepower (ihp), or *brake* horsepower (bhp). The indicated horsepower is the power developed in the engine cylinder, or the *input* of the engine; the brake horsepower is the power taken off the flywheel, or it is the *output* of the engine. The indicated horsepower is always larger than the brake horse-

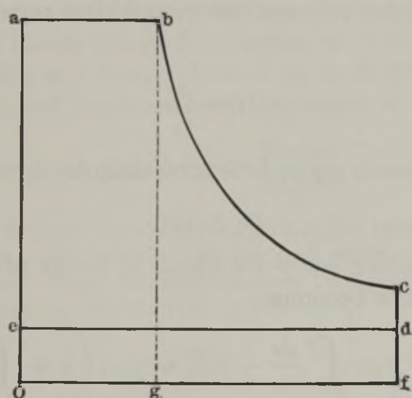


Fig. 187.—Theoretical indicator diagram.

power, the difference between the two being called the *friction* horsepower, as it is the power necessary to overcome friction, windage, and other mechanical losses.

178. Theoretical or Rated Indicated Horsepower of a Steam Engine.—In determining the theoretical horsepower of a steam engine it is assumed that there is no clearance, that the full pressure of steam is maintained during admission, that the cutoff and release occur instantly, and that the engine acts without compression. Then the indicator diagram of the engine would be as shown in Fig. 187. The curve of expansion is assumed to be a rectangular hyperbola, the equation of which is pv = a constant, as this is the curve that coincides most nearly with the actual expansion curve in a simple noncondensing engine.

Let the pressure at the point of cutoff b be p_1, and the volume, v_1; and let the pressure at the point d be p_2, and the volume, v_2. The amount of work is represented by the area

$$abcde = oabg + gbcf - oedf.$$

Area $oabg = p_1v_1$. Area $gbcf = \int_{v_1}^{v_2} pdv$. Area $oedf = p_2v_2$

Substituting these values in the previous equation, the area of work,

$$abcde = p_1v_1 + \int_{v_1}^{v_2} pdv - p_2v_2. \tag{1}$$

As v_1 and v_2 are the volumes before and after expansion, the ratio of expansion,

$$r = \frac{v_2}{v_1}. \tag{2}$$

Since the expansion curve bc is a rectangular hyperbola,

$$pv = p_1v_1.$$

Hence, substituting for p its value in terms of p_1 and v_1, the equation for work becomes

$$abcde = p_1v_1 + p_1v_1 \int_{v_1}^{v_2} \frac{dv}{v} - p_2v_2 = p_1v_1 \left(1 + \int_{v_1}^{v_2} \frac{dv}{v}\right) - p_2v_2.$$

Integrating, and substituting r for $\frac{v_2}{v_1}$,

$$abcde = p_1v_1(1 + log_e\, r) - p_2v_2. \tag{3}$$

The average pressure shown by the diagram, which is termed the *mean effective pressure* (mep), is found by dividing the area by the length of the diagram, v_2, or

$$\text{mep} = \frac{p_1}{r}(1 + log_e\, r) - p_2 \tag{4}$$

In practice, however, the assumptions made are not fulfilled, and the actual mean effective pressure is less than the theoretical, the proportion borne by the actual to the theoretical being termed the *diagram factor, e.* (*Transactions* A.S.M.E., Vol. 24, page 751.)

The actual mean effective pressure is

$$\text{mep} = e\left[\frac{p_1(1 + \log_e r)}{r} - p_2\right]. \tag{5}$$

This diagram factor is found by experiment and varies from 70 to 90 per cent, depending upon the type of engine.

To determine the theoretical or *rated indicated horsepower* of a steam engine, it is necessary to find the work theoretically done in the engine cylinder. Assume the engine to have a cylinder a square inches in cross section and l feet long, that it is double acting and makes n revolutions per minute (rpm), and that the mean effective pressure, determined from Eq. (5), acting on the piston is p pounds per square inch. Then the total pressure against the piston will be pa pounds and the space traveled per minute by the piston will be $2\,ln$; hence, the foot-pounds of work done per minute is $2\,plan$. Since 1 hp = 33,000 ft-lb per min, the rated indicated horsepower of the engine is

$$\text{Rated ihp} = \frac{2\,plan}{33{,}000}. \tag{6}$$

Example.—A 12-in. by 15-in. double-acting engine runs 200 rpm. Cutoff, one-fourth stroke; steam pressure, 100 psi; back pressure, 2 psia. Diagram factor, 80 per cent. Find the rated indicated horsepower of the engine.

Solution.—From Eq. (2), the ratio of expansion,

$$r = \frac{v_2}{v_1} = \frac{1}{\frac{1}{4}} = 4,$$

and from Eq. (5) the

$$\begin{aligned}
\text{Mep} &= e\left[\frac{p_1}{r}(1 + \log_e r) - p_2\right] \\
&= 0.80\left[\frac{114.7}{4}(1 + \log_e 4) - 2\right] = 0.80[28.7(1 + 1.386) - 2] \\
&= 0.80(68.5 - 2) = 0.80 \times 66.5 \\
&= 53.2 \text{ lb.}
\end{aligned}$$

The cross-sectional area of the cylinder,

$$\begin{aligned}
a &= \pi\frac{d^2}{4} = 3.1416 \times \frac{12^2}{4} \\
&= 113.3 \text{ sq in.}
\end{aligned}$$

From Eq. (6), the

$$\text{Rated ihp} = \frac{2\,plan}{33{,}000} = \frac{2 \times 53.2 \times 1.25 \times 113.3 \times 200}{33{,}000} = 91.4.$$

Ans. 91.4 rated ihp.

179. Actual Horsepower.—In order to find the actual horsepower developed in an engine cylinder, it is necessary to obtain the mean effective pressure (mep) from an actual indicator diagram taken with an instrument called an *indicator*, instead of from a theoretical diagram, as in Art. 178.

The theoretical, or rated, horsepower of an engine is the power developed at the point of maximum economy, or minimum steam consumption, per horsepower per hour; the actual power developed at any time may be greater or less than this.

180. The Indicator.—The indicator is a device by which the pressure of the steam for each point in the stroke of the engine is graphically recorded. It was first invented by James Watt and has since reached a high state of perfection.

There are three principal things which may be determined by an indicator diagram:

1. The actual average pressure of the steam acting against the piston, which is the actual *mean effective pressure* (mep).

2. The distribution of the steam in the engine, *i.e.*, the point at which the valves of the engine are opened and closed. By the use of the indicator it is possible to determine whether or not the engine has a proper distribution of steam.

3. From the indicator the actual weight of dry steam that is being worked in the engine cylinder may be determined. The indicator makes possible a complete analysis of the action of the steam engine.

Figure 188 shows a cross section of a Crosby steam-engine indicator. This instrument is attached to the engine cylinder, and the space under the piston 8 is in direct communication with the engine cylinder. The pressure of the steam acts against the piston 8, compressing a spring above it. The pressure of the steam raises an arm 16, and the attached pencil at 23. The drum 24 is covered with a sheet of paper; a cord passing over a pulley 34 is attached to the engine crosshead through a reducing motion, so that with each stroke of the engine the drum makes almost a complete revolution. The movement of the drum corresponds to the movement of the piston, and the upward movement of the pencil corresponds to the pressure in the cylinder. The result, therefore, is a diagram of the pressure in the cylinder for each point in the stroke of the engine. The springs used above the piston are of various strengths. What is termed a 40-lb spring

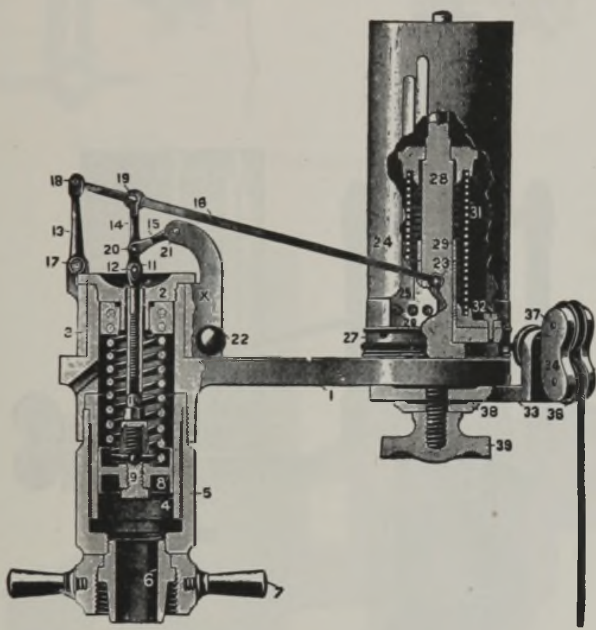


Fig. 188.—Crosby indicator.

would be one of such strength that a pressure of 40 psi under the piston would raise the pencil 1 in. These springs are carefully calibrated so that certain movements of the piston give a corresponding movement of the pencil on the paper.

A brass stylus is sometimes used in place of a pencil. This has the advantage of keeping a sharp point longer than a pencil does. It dulls in time, but can be sharpened again and again until too short to use. The indicator diagrams are taken on a specially prepared paper with a metallic surface, as no mark would be made on ordinary paper. One disadvantage of the use of the stylus and metal-surfaced paper is that frequently the outline traced by the brass point is not permanent, but fades out in a comparatively short time.

Figure 189 shows a similar indicator with the spring external to the indicator cylinder. The temperature of the spring in this indicator is independent of the steam pressure, and the spring may be changed easily without removing the indicator piston.

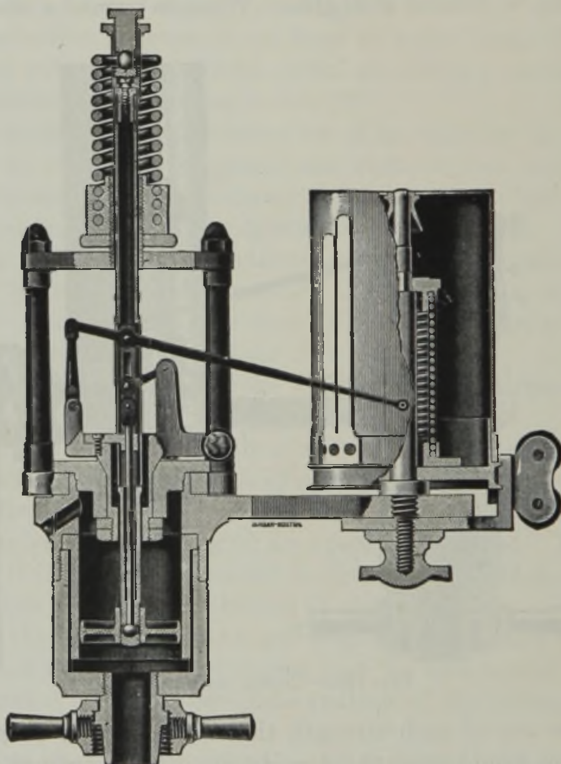


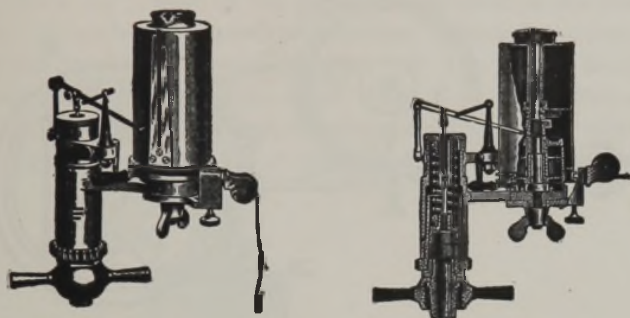
FIG. 189.—Crosby indicator with outside spring.

This form is particularly adapted for indicator work where great accuracy is desired.

Figure 190 shows the elevation and cross section of the Thompson indicator. This form of indicator is particularly well adapted to hard service.

The Maihak indicator shown in Figs. 191 and 192 is similar in action to those already described. Its parts are slightly more rugged and the piston is lighter than in some others.

For high-speed work the Maihak high-speed indicator (Figs. 193 and 194) may be used. This is built for speeds up to 2,400 rpm and pressures up to 6,000 lb. One feature peculiar to



Elevation. Cross section.

Fig. 190.—Thompson indicator.

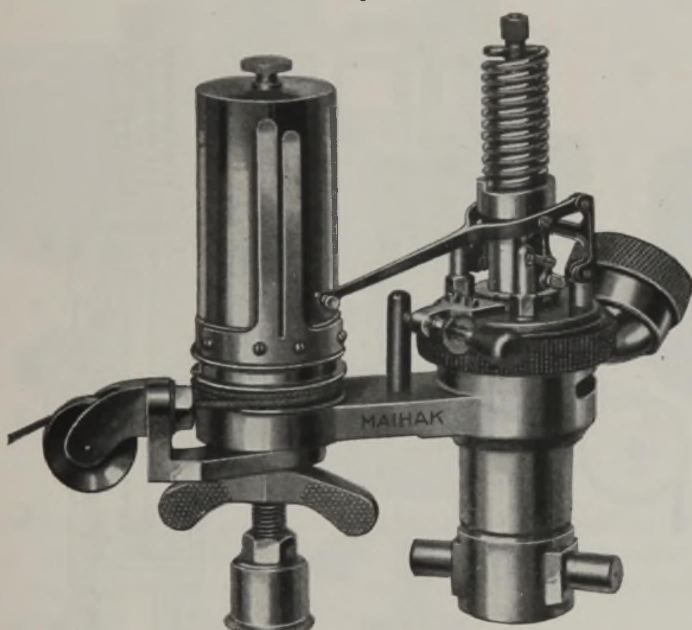


Fig. 191.—Maihak standard indicator.

this indicator is the cantilever spring (12). The period of natural vibration of this spring is high and gives rigidity to the recording system and reduces the inertia effects much below those accompanying helical springs. In this indicator the piston rod is very short and is made in one piece with the piston. This gives a combination of lightness and strength.

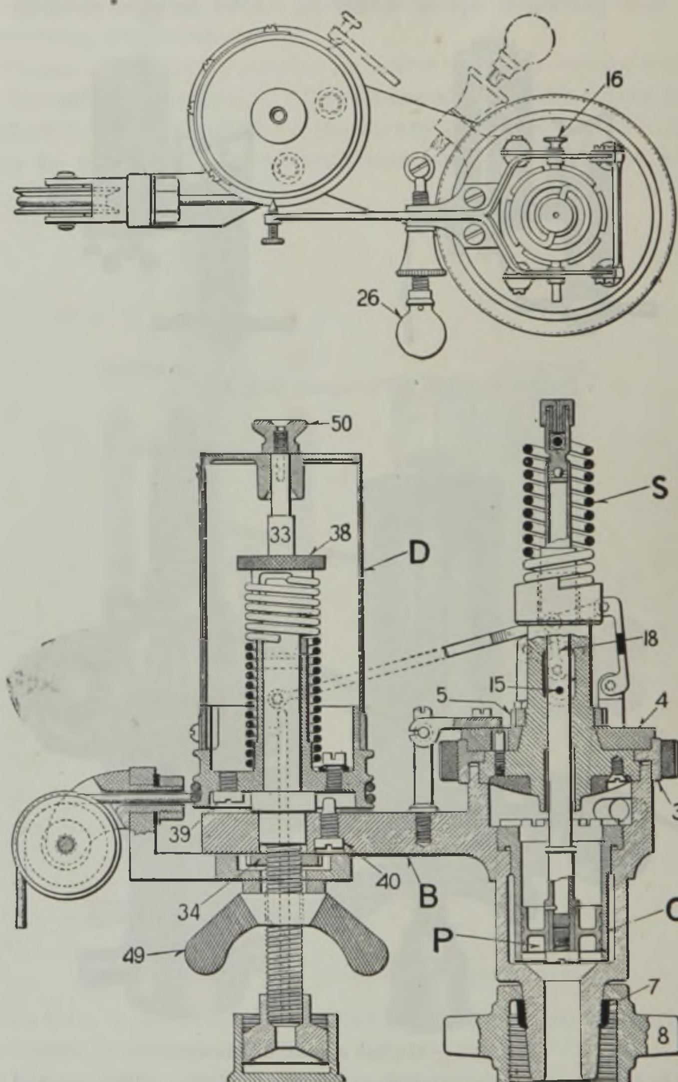


FIG. 192.—Sectional view of Maihak standard indicator.

181. Use of Indicator.—The accuracy of an indicator depends upon the accuracy with which the pressure in the cylinder is recorded on the indicator drum, and also upon the accuracy with which the motion of the piston is conveyed to the indicator drum. In order to have the pressure recorded properly, the following conditions should be observed: The piping leading to the indicator should not be more than 18 in. long, and should be $\frac{1}{2}$ in. in diameter; the indicator should never be connected to a pipe through which a current of steam is passing; the

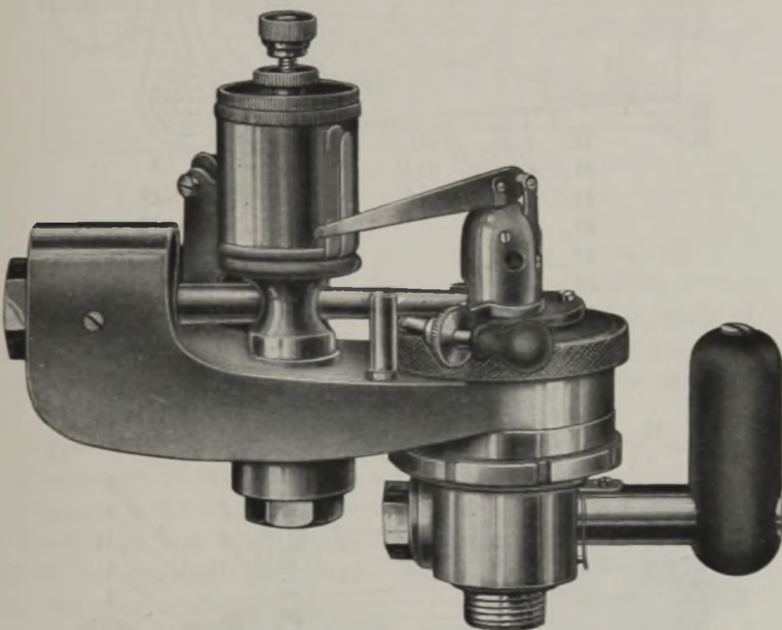


Fig. 193.—Maihak high-speed indicator.

holes connecting the indicator with the cylinder should be drilled into the clearance space so that the piston will not cover the opening; the indicator should, if possible, be placed in a vertical position.

Where great accuracy is desired, the indicator spring should be calibrated before and after the test.

The motion of the drum may be taken from any part of the engine which has the same relative motion as the engine piston. The movement of the drum, which is usually taken from the crosshead, must be reduced to the length of the indicator diagram by some form of mechanism which makes the reduced motion an exact ratio to the movement of the engine piston. The indicator drum is then connected with this reduced motion of the piston by means of a cord. A reducing lever with segment is one of the

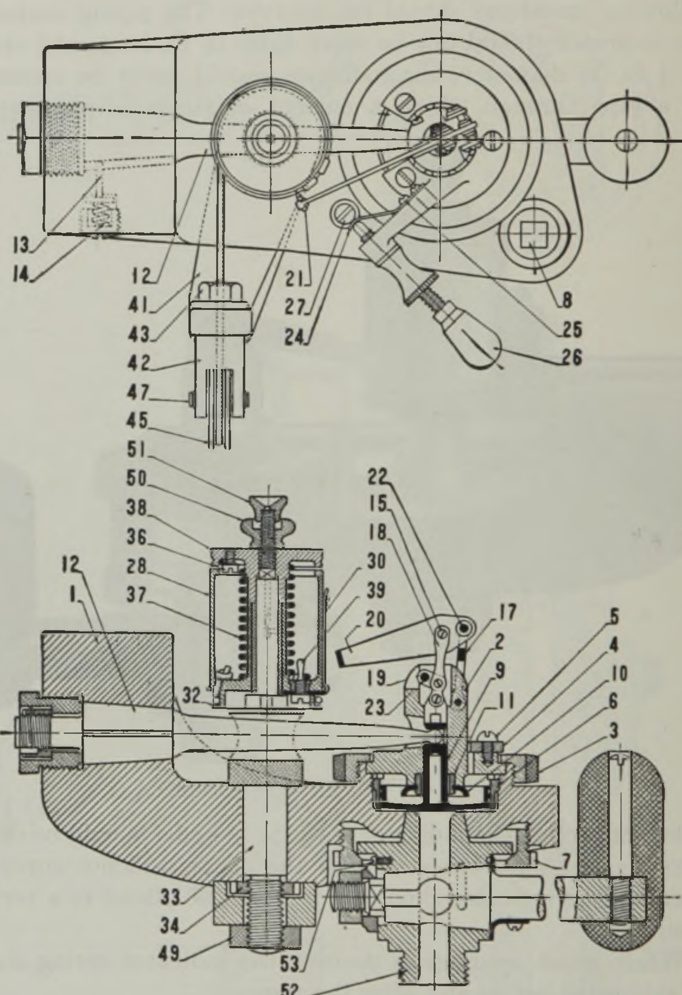


Fig. 194.—Sectional view of Maihak high-speed indicator.

commonest means used to accomplish this reduction. There are also on the market various forms of reducing wheels which make the reduction by means of gearing and pulleys. These reducing motions are more satisfactory when they are provided with a clutch so that the drum may be disengaged without removing the cord connection from the reducing motion to the engine.

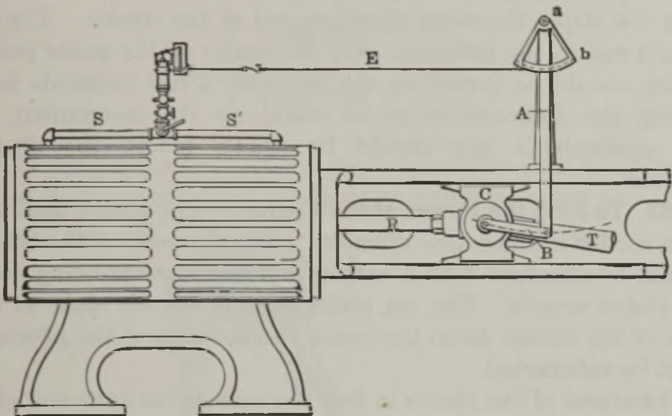


FIG. 195.—Reducing motion, showing method of attachment.

Figure 195 shows a simple form of reducing motion made of hardwood splines and a brass segment. It is better to use a segment of a circle at the point b, so that ab is the same distance for every point of the stroke.

Figure 196 shows a reducing wheel having a clutch, so that it is not necessary to disconnect the motion from the crosshead when the paper on the drum is replaced.

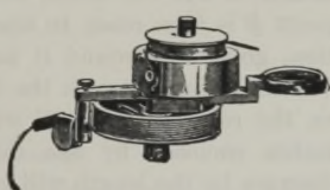


FIG. 196.—Reducing wheel.

Cord that has been stretched should be used on the indicator and reducing motion, so that the give of the cord will not reduce the length of the diagram. Wherever very long cords are found necessary, it is better to replace them with piano wire.

182. Taking an Indicator Diagram.—Before attaching the indicator, oil the parts of the mechanism with watch oil and the piston with cylinder oil. Be sure the piston is working freely in the cylinder. The piston should drop by gravity in the cylinder when the spring is removed. The pencil should have a smooth, fine point. Be sure there is no lost motion in the instrument.

The reducing motion should be adjusted so that the length of the diagram is from $2\frac{1}{2}$ in. to 3 in. The higher the speed, the shorter should be the diagram. The tension of the indicator drum spring should be just sufficient to prevent slackness in the cord. Before taking a diagram, try the indicator and see that it does not strike the stops at either end of the stroke. The cord should run to the indicator over the center of the guide pulleys. Steam should be turned on the indicator a few moments before taking the diagram, so as to warm up the instrument, and the atmospheric line should be drawn before the diagram is taken.

183. To Find the Power of the Engine.—The piston area is the cross section of the engine cylinder. The diameter of the cylinder should be obtained with a caliper and the corresponding area is the piston area a. The net piston area is not the same at both ends of the stroke, as on the crank end the area of the piston rod must be subtracted.

The travel of the piston in feet per minute for each end of the stroke is found by multiplying the length of the stroke by the revolutions of the crankshaft per minute.

The mean effective pressure is obtained from the indicator diagram. The usual method is to measure the area of the diagram with an instrument called a *planimeter*.

Figure 197 is a standard form of planimeter. In using it the point B is placed on a point on the indicator diagram to be measured and the initial reading taken on the vernier E. The point B is then made to trace the diagram in a clockwise direction, going all around it and returning to the starting point. The difference between the final and initial readings of the scale on the rotating wheel C will then give the number of square inches enclosed by the diagram. Dividing the area of the diagram by the length will give the average height in inches, and this multiplied by the scale of the spring used gives the *mean effective pressure* (mep). The mean effective pressure should be determined for each end of the cylinder separately.

The mean ordinate from the diagram may also be obtained by dividing the diagram into 10 spaces by vertical lines drawn equal distances apart. Then measure the distance from the back-pressure line to the forward-pressure line at the center of each space. The average of these lengths will be approximately the mean ordinate.

Let p_h be the mean effective pressure for the head end, and p_c for the crank end; a_h the cross-sectional area of the piston in square inches for the head end, and a_c for the crank end; l the length of the stroke in feet; and n the number of revolutions per minute. Then the indicated horsepower actually developed in the engine cylinder will be

$$\text{Ihp}\begin{cases}\text{head end} = \dfrac{p_h l a_h n}{33{,}000}, & (7)\\[2ex] \text{crank end} = \dfrac{p_c l a_c n}{33{,}000}, & (8)\end{cases}$$

and the total actual indicated horsepower of the engine will be the sum of the horsepower for the head end and the crank end.

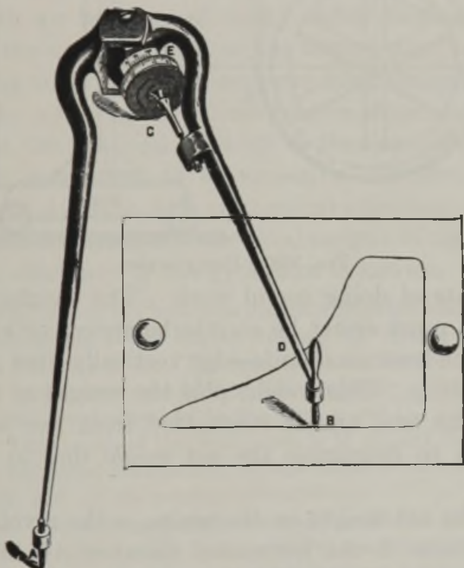


Fig. 197.—Polar planimeter.

184. Brake Horsepower.—The indicated horsepower of an engine does not represent the actual useful work that can be obtained from the engine, as part of this power must be used in overcoming the friction of the engine itself. The actual power of the engine delivered upon the flywheel is usually measured by a Prony brake or some similar device. The horsepower obtained at the brake is termed the *brake*, or effective, horsepower.

The brake used to determine the brake horsepower usually consists of an adjustable strap which encircles the rim of the brake wheel that is fastened to the crankshaft of the engine.

The brake wheel should be provided with internal flanges for holding water to keep the rim cooled. To the strap encircling the brake wheel is rigidly fastened an arm which rests on a platform scales. The friction of the strap DE (Fig. 198) tends to carry the arm FK in the direction of rotation of the wheel. The force tending to depress the arm FK is measured on the scales. The net force on the scales times the distance AC is the moment of friction, and this multiplied by the angular velocity

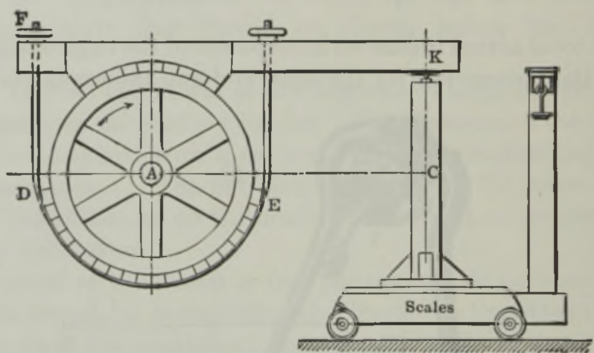


Fig. 198.—Prony brake.

equals the rate of doing useful work. The weight of the lever on the scales must either be counterbalanced, or else found by suspending the lever on a knife-edge vertically over A and noting the scale reading. This weight plus the weight of the standard C is called the *tare*, and is subtracted from the weight shown on the scales to determine the *net weight* due to the force of friction.

Let w = the net weight on the scales, n the revolutions of the shaft per minute, L the horizontal distance AC in feet, or the brake arm, and bhp, the brake horsepower. Then

$$\text{Bhp} = \frac{2\pi Lwn}{33{,}000}. \tag{9}$$

185. Friction Horsepower and Mechanical Efficiency.—The indicated horsepower minus the brake horsepower is called the *friction horsepower* (fhp) and the brake horsepower divided by the indicated horsepower is the *mechanical efficiency* of the engine. The mechanical efficiency of an engine is usually about 85 per cent; in well-built engines it may be as high as 90 per cent and over.

In large engines it is not possible to obtain the brake horsepower, as such an engine would require a very elaborate brake. In such cases it is customary to obtain the horsepower lost in friction, approximately, by what is termed a *friction diagram*. A friction diagram is obtained by removing all the load from the engine, so that the only load acting upon the engine is the friction of the engine itself. An indicator diagram is taken from the engine under these conditions, and the horsepower shown by this diagram is called the *friction horsepower*. A diagram so taken does not give the actual friction of the engine, as the friction increases with an increase of load. After finding the friction horsepower, the actual output of the engine may be determined by subtracting this friction horsepower from the indicated horsepower. If the power taken by the friction diagram is more than 10 per cent of the full-load capacity of the engine, the friction of the engine is considered to be excessive. Where an engine is used to drive a dynamo, the mechanical efficiency of the engine may be determined from the electrical output of the generator, if the electrical efficiency of the generator is known.

Example.—The area of the indicator diagram from the head end of an 8-in. by 12-in. double-acting steam engine running 227 rpm is 1.17 sq in., and from the crank end 1.34 sq in. The length of each diagram is 2.91 in., and the scale of the spring used was 60 lb. The diameter of the piston rod is $1\frac{1}{2}$ in. A Prony brake was attached to the engine and the gross weight on it was 103.5 lb. The length of the brake arm is 54 in. and the tare 28.5 lb.

Find (*a*) the indicated horsepower, (*b*) the brake horsepower, (*c*) the friction horsepower, and (*d*) the mechanical efficiency.

Solution.—(*a*) The average height, or mean ordinate, of the diagram is equal to the area divided by the length, and this multiplied by the scale of the spring used will give the mean effective pressure. Hence, the

$$\text{Mep}\begin{cases}\text{head end} = \dfrac{1.17}{2.91} \times 60 = 24.1 \text{ lb.}\\[2ex] \text{crank end} = \dfrac{1.34}{2.91} \times 60 = 27.6 \text{ lb.}\end{cases}$$

$$\text{Area}\begin{cases}\text{head end} = 3.1416 \times 4 \times 4 = 50.26 \text{ sq in.}\\ \text{crank end} = (3.1416 \times 4 \times 4) - (3.1416 \times 0.75 \times 0.75)\\ \qquad\qquad = 48.50 \text{ sq in.}\end{cases}$$

The indicated horsepower for each end $= \dfrac{plan}{33{,}000}$;

hence, the

$$\text{Ihp}\begin{cases}\text{head end} = \dfrac{24.1 \times 1 \times 50.26 \times 227}{33{,}000} = 8.34.\\[2ex] \text{crank end} = \dfrac{27.6 \times 1 \times 48.5 \times 227}{33{,}000} = 9.22.\end{cases}$$

Total indicated horsepower $= 8.34 + 9.22 = 17.56$.

(*b*) Net weight on brake $= 103.5 - 28.5 = 75$ lb.

Length of brake arm $= \frac{54}{12} = 4.5$ ft.

$$\text{Bhp} = \frac{2\pi Lnw}{33{,}000} = \frac{2 \times 3.1416 \times 4.5 \times 227 \times 75}{33{,}000} = 14.6.$$

(*c*) Fhp = ihp − bhp $= 17.56 - 14.6 = 2.96$.

(*d*) Mechanical efficiency $= \frac{\text{bhp}}{\text{ihp}} = \frac{14.6}{17.56} = 0.832$ or 83.2 per cent.

186. Horsepower of a Compound[1] Engine.—In determining the horsepower that a compound[1] engine ought to develop, it is necessary to know the absolute initial steam pressure, the total number of expansions of steam, the number of strokes per minute, the length of the stroke, the diameters of the high- and low-pressure cylinders, and the diagram factor.

The horsepower is then determined as though there were but one cylinder, and that one the size of the low-pressure cylinder, and the total expansion of steam took place in that cylinder. The reason for this is apparent when it is considered that the power of any engine per stroke depends on the weight of steam admitted and its ratio of expansion, and that all the power of the compound[1] engine could be developed in its low-pressure cylinder, if there were admitted into that cylinder the same weight of steam as was admitted to the high-pressure cylinder, if the steam in this cylinder were expanded the same number of times as it was expanded in the whole engine, and if it were exhausted against the same back pressure. If the horsepower obtained by assuming all the work done in the low-pressure cylinder is multiplied by a diagram factor, the result will be equal to the horsepower of the engine. This may be expressed mathematically as follows:

Let

D = the diameter of the low-pressure cylinder;
d = the diameter of the high-pressure cylinder;
A = the area of the low-pressure cylinder in square inches;
l = the length of stroke of the engine in feet;
p = the mean effective pressure for the whole engine;
n = number of revolutions per minute;
x = the proportion of the stroke to the point of cutoff in the high-pressure cylinder;
r = ratio of expansion for the whole engine;

[1] See note on p. 332.

e = the diagram factor;
p_1 = initial pressure steam entering the engine;
p_2 = pressure of the exhaust.

Then

$$r = \frac{\pi \frac{D^2}{4}}{x\pi \frac{d^2}{4}} = \frac{D^2}{xd^2}, \tag{10}$$

and

$$p = e\left[\frac{p_1(1 + \log_e r)}{r} - p_2\right]. \tag{11}$$

$$\text{Rated ihp} = \frac{2plAn}{33{,}000}. \tag{12}$$

The value of the factor e depends upon the type of the engine, and varies from 0.70 to 0.80 for automatic high-speed engines, and from 0.75 to 0.85 for a Corliss engine.

In determining the actual horsepower of a compound engine from the indicator diagrams, the diagram from each end of each cylinder is worked up and the horsepower calculated for each; the sum of the horsepowers determined from each diagram will be the horsepower of the engine.

Example.—A 15-in. and 24-in. and 36-in. by 30-in. engine runs 100 rpm. Cutoff in the high-pressure cylinder, three-eighths stroke; in the intermediate cylinder, three-eighths stroke; in the low-pressure cylinder, one-half stroke. The steam pressure is 225 psi. The engine exhausts into a condenser having a vacuum of 26 in. The barometer reading is 28.65 in. Hg. Assume a diagram factor of 0.80.

Indicator diagrams were taken from the engine with the following areas: high-pressure cylinder, head end, 1.32 sq in.; crank end, 1.35 sq in.; intermediate cylinder, head end, 1.8 sq in.; crank end, 1.71 sq in.; low-pressure cylinder, head end, 2.01 sq in.; crank end, 2.04 sq in. Length of all diagrams, 3 in. A 160-lb spring was used on the high-pressure cylinder, a 50-lb spring on the intermediate, and a 20-lb spring on the low-pressure. The diameters of the piston rods were as follows: high-pressure cylinder, 2 in.; intermediate cylinder, 2½ in.; low-pressure cylinder, 3 in.

(*a*) What is the rated indicated horsepower of the engine?
(*b*) What per cent of the rated horsepower is being developed?

Solution.
(*a*) Atmospheric pressure

$$= 28.65 \times 0.491 = 14.07 \text{ lb.}$$

Exhaust pressure $p_2 = (28.65 - 26) \times 0.491 = 1.3$ lb.

$$r = \frac{D^2}{xd^2} = \frac{36 \times 36}{\frac{3}{8} \times 15 \times 15} = \frac{8 \times 36 \times 36}{3 \times 15 \times 15} = 15.35.$$

$$\text{Mep} = e\left[\frac{p_1}{r}(1 + \log_e r) - p_2\right].$$

$$\text{Mep} = 0.8\left[\frac{239.07}{15.35}(1 + \log_e 15.35) - 1.3\right] = 0.8(58.1 - 1.3) = 45.44 \text{ lb.}$$

Area low-pressure cylinder = 3.1416 × 18 × 18 = 1,018 sq in.

$$\text{Rated ihp} = \frac{2 \times 45.44 \times 2.5 \times 1{,}018 \times 100}{33{,}000} = 701.$$

Practically a 700-hp engine.

(b) Mep

Mep	high-pressure cylinder, head end,	$\frac{1.32}{3} \times 160 = 70.4$ lb.
	high-pressure cylinder, crank end,	$\frac{1.35}{3} \times 160 = 72.0$ lb.
	intermediate-pressure cylinder, head end,	$\frac{1.8}{3} \times 50 = 30.0$ lb.
	intermediate-pressure cylinder, crank end,	$\frac{1.71}{3} \times 50 = 28.5$ lb.
	low-pressure cylinder, head end,	$\frac{2.01}{3} \times 20 = 13.4$ lb.
	low-pressure cylinder, crank end,	$\frac{2.04}{3} \times 20 = 13.6$ lb.

$$\text{Area}\begin{cases} \pi 7.5^2 & = 176.7 \text{ sq in.} \\ \pi(7.5^2 - 1^2) & = 173.6 \text{ sq in.} \\ \pi 12^2 & = 452 \text{ sq in.} \\ \pi(12^2 - 1.25^2) & = 447 \text{ sq in.} \\ \pi 18^2 & = 1{,}018 \text{ sq in.} \\ \pi(18^2 - 1.5^2) & = 1{,}011 \text{ sq in.} \end{cases}$$

$$\text{Constant} = \frac{ln}{33{,}000} = \frac{2.5 \times 100}{33{,}000} = 0.007575.$$

Ihp	high-pressure cylinder, head end =	70.4 × 176.7 × 0.007575 = 94.1
	high-pressure cylinder, crank end =	72 × 173.6 × 0.007575 = 94.7
	intermediate-pressure cylinder, head end =	30 × 452 × 0.007575 = 102.7
	intermediate-pressure cylinder, crank end =	28.5 × 447 × 0.007575 = 96.5
	low-pressure cylinder, head end =	13.4 × 1,018 × 0.007575 = 103.3
	low-pressure cylinder, crank end =	13.6 × 1,011 × 0.007575 = 104.2
		Total = 595.5

Per cent of rated horsepower developed

$$= \frac{595.5}{700} = 0.851 \text{ or } 85.1 \text{ per cent.}$$

Ans. (*a*) 700 hp.
(*b*) 85.1 per cent.

187. Combined Indicator Diagrams.—The combined diagram of a compound engine is a hypothetical figure which would be obtained if the admission, expansion, and exhaust all took place in one cylinder, and that the low-pressure cylinder of the engine. It is a diagram on which may be measured the pressure of the steam at any point in the stroke of any of the cylinders, and the volume of that steam. In it the indicator diagram from each cylinder appears in its true proportion.

When combining the diagrams from a compound engine, it is first necessary to reduce them all to the same scales of pressures and volumes. It is generally more convenient to use the diagram from the low-pressure cylinder as the basis to which to change the other diagrams.

On each of the diagrams lay off a vertical line back of the admission line a distance equal to the clearance volume for that particular cylinder. These lines represent the lines of zero volume.

Divide the high and intermediate diagrams into any convenient number of parts by vertical lines spaced an equal distance apart. Multiply the distance from the atmospheric line of each of the points where these vertical lines cross the diagrams by the ratio of the scale of the spring used in the cylinder being considered to that used in the low-pressure cylinder. The results will be the ordinates of the points to be plotted when drawing the combined diagram. (The pressure in each cylinder at any point in the stroke may be determined from the indicator diagram for that cylinder, knowing the value of the spring used and the position of the atmospheric line.) Multiply the horizontal distances from the zero-volume line to the points of intersection of the vertical lines and the diagram by the ratio of the volume of the cylinder under consideration to the volume of the low-pressure cylinder. The results obtained will be the abscissas of the points on the combined diagram.

Now plot the results, using the atmospheric line and line of zero volume on the low-pressure diagram as the horizontal and

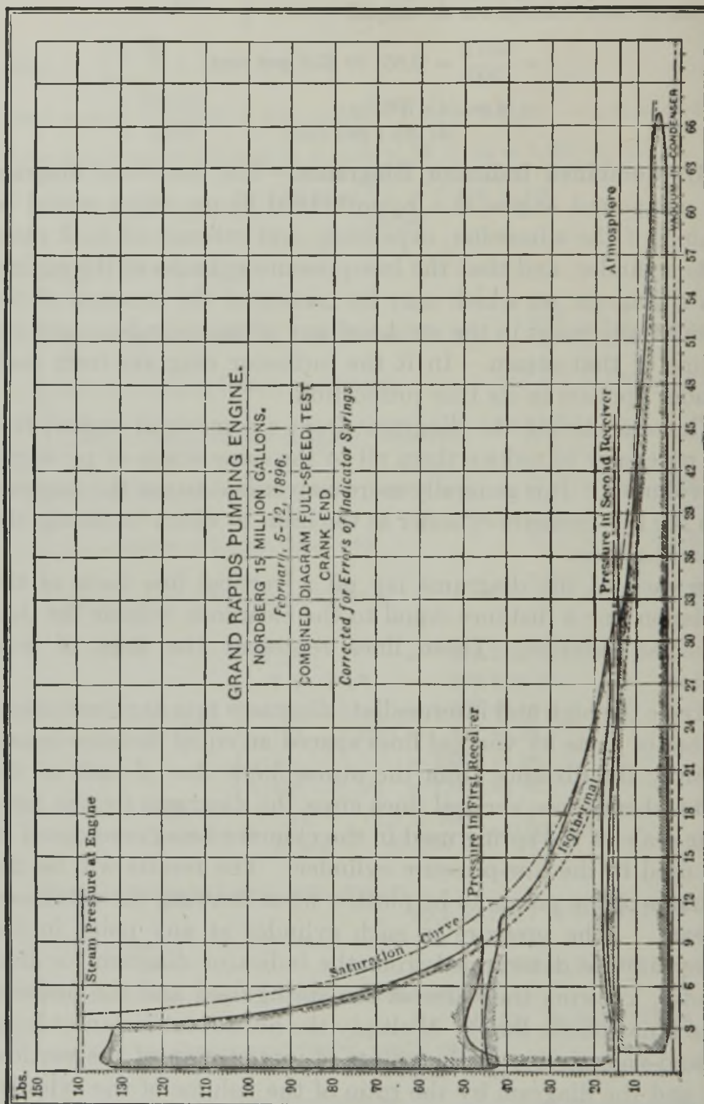


FIG. 199.—Combined indicator diagrams for triple-expansion pumping engine.

vertical axes from which to measure the ordinates and abscissas. Through the points so plotted, draw the combined diagram.

Figure 199 shows the combined diagram from a triple-expansion pumping engine. The diagrams for the high-, intermediate-, and low-pressure cylinders have all been reduced to the same scale of volumes and pressures. The ordinates in this combined diagram are absolute pressures and the abscissas are volumes. The diagram from each cylinder was divided into an equal number of parts and the pressure and the volume at each of these points were computed and plotted in the figure. The diagram for each cylinder, it will be noticed, does not begin at the zero-volume line, the difference between zero volume of the diagram and the zero of volumes representing the volume of the clearance. The dotted saturation curve shows what the curve of expansion would have been if the actual weight of steam expanding in the engine had remained saturated.

188. Rankine Cycle Efficiency.—*The efficiency of the Rankine cycle, e_R, is equal to the heat available for work divided by the heat chargeable.* The *heat available* is equal to the difference between the enthalpy of the steam admitted, h_1, and the enthalpy of the steam exhausted, h_2. The *heat chargeable* is equal to the enthalpy of the steam admitted less the enthalpy of the liquid in the exhaust steam h_{f2}. Therefore,

$$e_R = \frac{h_1 - h_2}{h_1 - h_{f2}}. \tag{13}$$

h_1 and h_2 must not be confused with h_g, the enthalpy of dry and saturated steam as given in the steam tables, since the steam admitted may be wet, dry and saturated, or superheated, and h_1 will vary accordingly; the steam exhausted will nearly always be wet and h_2 will equal $h_{f2} + x_2 h_{fg2}$.

When finding the efficiency of the Rankine cycle for any given case, it will be necessary first to determine the quality x_2 of the exhaust steam. This determination is based on the fact that the entropy of the steam at the beginning and end of the expansion along the line bc (Fig. 149) is the same, since this expansion is adiabatic, and there is consequently no change in entropy.

189. Actual Thermal Efficiency.—*The actual thermal efficiency of an engine is the heat equivalent of one horsepower per hour* *divided by the number of heat units chargeable to the engine per horsepower-hour, either indicated or brake.*

Since a horsepower is 33,000 ft-lb per min, then the heat equivalent of 1 hp-hr is

$$\frac{33{,}000 \times 60}{778} = 2{,}545 \text{ Btu.} \tag{14}$$

Let w equal the steam consumption of an engine per horsepower-hour, h_1 the enthalpy of the entering steam, and h_{f2} the enthalpy of the liquid in the exhaust steam. Then the actual thermal efficiency, e_t, will be

$$e_t = \frac{2{,}545}{w(h_1 - h_{f2})}. \tag{15}$$

It must be remembered that h_1 is not necessarily h_{g1}, as the steam entering the engine may be wet, dry and saturated, or superheated.

190. Engine Efficiency.—*The ratio of the heat converted into work in an engine to the heat available for conversion into work by the cycle employed is called the engine efficiency, e_e.*

But

$$e_t = \frac{\text{heat converted into work}}{\text{heat chargeable to engine}},$$

and

$$e_R = \frac{\text{heat available for conversion into work}}{\text{heat chargeable to engine}}.$$

Then

$$e_e = \frac{e_t}{e_R} = \frac{\text{heat converted into work}}{\text{heat available for conversion into work}}.$$

Hence, by definition, engine efficiency equals thermal efficiency divided by cycle efficiency.

Therefore,

$$e_e = \frac{\dfrac{2{,}545}{w(h_1 - h_{f2})}}{\dfrac{h_1 - h_2}{h_1 - h_{f2}}} = \frac{2{,}545}{w(h_1 - h_2)} \tag{16}$$

Example.—An engine uses 25 lb of steam per ihp-hr at a pressure of 160 psia and a quality of 95 per cent. The exhaust pressure is 5 psia. Find (*a*) the thermal efficiency of the Rankine cycle; (*b*) the actual thermal efficiency of the engine; and (*c*) the engine efficiency.

Solution.—(*a*) The entropy of the entering steam will be

$$\begin{aligned} s_{g1} &= s_{f1} + x_1 s_{fg1} \\ &= 0.5204 + 0.95 \times 1.0436 \\ &= 0.5204 + 0.9914 \\ &= 1.5118. \end{aligned}$$

The entropy of the exhaust steam will be the same as the entropy of the entering steam, and the entropy of evaporation of the exhaust steam will be

$$\begin{aligned} s_{g2} - s_{f2} &= 1.5118 - 0.2347 \\ &= 1.2771. \end{aligned}$$

Since the entropy of evaporation of dry and saturated steam at 5 psia is 1.6094, the quality of the exhaust steam is

$$\begin{aligned} x_2 &= \frac{1.2771}{1.6094} = 0.794 \\ &= 79.4 \text{ per cent.} \\ h_1 &= h_{f1} + x_1 h_{fg1} \\ &= 335.9 + 0.95 \times 859.2 = 335.9 + 816.2 \\ &= 1{,}152.1 \text{ Btu.} \\ h_2 &= h_{f2} + x_2 h_{fg2} \\ &= 130.1 + 0.794 \times 1{,}001.0 = 130.1 + 794.3 \\ &= 924.4 \text{ Btu.} \end{aligned}$$

$$\begin{aligned} e_R &= \frac{h_1 - h_2}{h_1 - h_{f2}} \\ &= \frac{1{,}152.1 - 924.9}{1{,}152.1 - 130.1} = \frac{227.2}{1{,}022.0} = 0.2223 \\ &= 22.23 \text{ per cent.} \end{aligned}$$

(*b*) Actual thermal efficiency

$$\begin{aligned} &= \frac{2{,}545}{w(h_1 - h_{f2})} \\ &= \frac{2{,}545}{25(1{,}152.1 - 130.1)} = \frac{2{,}545}{25 \times 1{,}022} \\ &= \frac{2{,}545}{25{,}555} = 0.0996 \\ &= 9.96 \text{ per cent.} \end{aligned}$$

(*c*) Engine efficiency

$$\begin{aligned} &= \frac{2{,}545}{w(h_1 - h_2)} \\ &= \frac{2{,}545}{25(1{,}152.1 - 924.9)} = \frac{2{,}545}{25 \times 227.2} \\ &= \frac{2{,}545}{56{,}800} = 0.4481 \\ &= 44.81 \text{ per cent.} \end{aligned}$$

191. Determination of Steam Consumption.—The theoretical steam rate of an engine in pounds per horsepower per hour is

$$w = \frac{2,545}{h_1 - h_3}, \tag{17}$$

where w and h_1 represent the same quantities as in Eq. (16), and h_3 is the *actual* enthalpy of the exhaust steam as determined by test.

The *steam rate* of an engine, turbine, or complete steam plant is the pounds of steam used per hour under actual conditions, per unit of output.

When an engine is used with a surface condenser, the steam consumption may be determined by weighing the steam condensed. It is seldom, however, that this can be done; usually it is necessary to measure the amount of feed water going to the boiler which supplies steam to the engine to be tested. When this is done, great care should be taken to see that all the steam produced from this feed water goes to the engine. If all the steam does not go to the engine, the amount going to other purposes should be measured and deducted from the total feed, the difference being the engine feed. Tests of this character should be at least 10 hr in length, and still better 24 hr, so as to allow for the effect of varying conditions such as the level of water in the boiler. The engine should be credited with the moisture in the steam and should be operated for some time before the test begins, so that the heat conditions may be uniform. During the test the engine should be run as nearly as possible at a uniform load. Indicator diagrams are usually taken every 10 min or 15 min, and the average horsepower shown by the diagrams is taken as the average horsepower developed during the test. As has already been stated, to determine the number of pounds of steam used by a steam engine per horsepower per hour, the water entering the boiler is weighed and the quality of the steam entering the engine determined, and all the dry steam that actually goes to the engine is charged to it. This weight of water reduced to pounds per hour is divided by the average horsepower developed by the engine; the result is the number of pounds of steam used by the engine per horsepower per hour. The A.S.M.E. has adopted a standard method of testing steam engines, which is described in their "Power Test Codes."

The number of pounds of steam used by the various forms of engines are summarized in the following table. These results are very general for the various classes of engines.

TABLE XXVIII.—STEAM CONSUMPTION OF VARIOUS CLASSES OF ENGINES

	Pounds per Ihp-hr
Simple throttling engine, noncondensing	44 to 45
Simple automatic engine, noncondensing	30 to 35
Simple Corliss engine, noncondensing	26 to 28
Simple automatic engine, condensing	22 to 26
Simple Corliss engine, condensing	22 to 24
Compound automatic engine, noncondensing	25 to 30
Compound automatic engine, condensing	18 to 20
Compound Corliss engine, condensing	14 to 16
Triple Corliss engine, condensing	12.25 to 13
Uniflow engine, simple condensing, superheat	11.25 to 12

192. Variation of Steam Consumption.—Since most engines work at a varying load, it is important to know the steam consumption of the engine at the different loads. Figure 200 shows the variation of steam consumption in a uniflow engine at various loads. The upper curve shows the steam consumption when the engine was running noncondensing, and the lower curve when it was running condensing.

In these curves the ordinates represent the steam consumption per kilowatt per hour, and the abscissas represent the kilowatt output.

193. Duty.—The economy of pumping engines is usually expressed not as the number of pounds of steam per indicated horsepower per hour, but in terms of *duty.*

In the earlier history of pumping engines, duty was defined as the number of foot-pounds of work done in the pump cylinder per 100 lb of coal burned in the boiler. The objection to this definition was that it included both boiler and engine economy. In purchasing a pumping engine it was necessary to allow the contractor to furnish the boilers also.

To meet this objection, it was found better to define it as the number of foot-pounds of work obtained in the pump cylinders per 1,000 lb of steam furnished to the engine. The specifications stated at what pressure this steam must be furnished.

The best definition of duty and the one in general use at the present time is the *number of foot-pounds of work done in the pump cylinders per* 1,000,000 *Btu chargeable to the engine,* as this eliminates all considerations of the steam pressure. Engines working under widely different conditions may be compared when their duty is based on foot-pounds developed in the pump cylinder per 1,000,000 Btu chargeable to the engine.

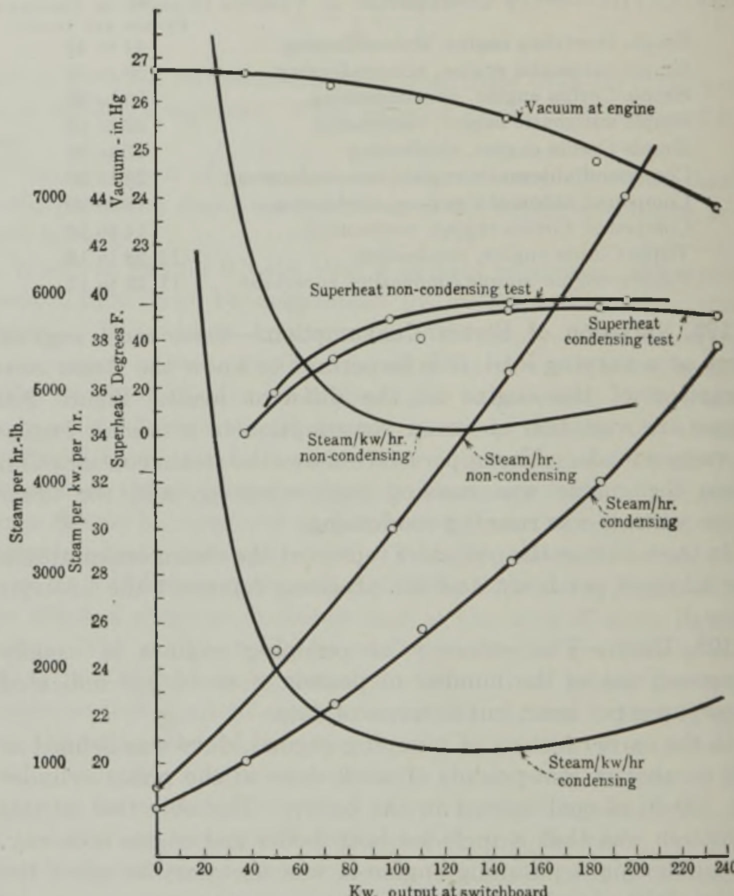


FIG. 200.—Curves showing steam consumption.

The amount of "work done" is equal to the weight of water pumped times the *head* pumped against. The total head is made up of the pressure shown by the gage on the discharge line plus that on the suction line, both reduced to feet, plus the vertical distance between the center of the pressure gage and the point of attachment of the suction gage to the main. If the suction line is under a pressure instead of a vacuum, the correction for the difference in the level of the gages is the vertical distance between the centers of the gages, and the reading of the suction gage is to be subtracted from that of the pressure gage.

The *capacity* of a pump is *the number of gallons pumped in 24 hr.*

The duty that may be obtained in the various forms of pumping engines is given in the following table:

TABLE XXIX.—DUTY OF VARIOUS FORMS OF PUMPS

	Foot-pounds
Injectors	2,000,000
Small duplex noncondensing pumps	10,000,000
Large duplex noncondensing pumps	25,000,000
Small simple flywheel pumps, condensing	50,000,000
Large simple flywheel pumps, condensing	65,000,000
Small compound flywheel pumps, condensing	85,000,000
Large compound flywheel pumps, condensing	120,000,000
Large triple-expansion flywheel pumps, condensing	165,000,000
Large triple-expansion pumps, condensing, of exceptional economy	180,000,000

Example.—A 500-hp engine pumps 16,000,000 gal of water in 24 hr against a total head of 70 psi. The steam consumption is 15 lb per ihp-hr; steam pressure, 100 psi; exhaust pressure, 2 psia. (*a*) What is the duty per 1,000 lb of steam? (*b*) What is the duty per 1,000,000 Btu?

Solution.[1]—(*a*) Weight of water pumped in 24 hr

$$
\begin{aligned}
& = 8\tfrac{1}{3} \times 16{,}000{,}000 \\
& = 133{,}333{,}333 \text{ lb.} \\
\text{Head pumped against} \quad & = 70 \times 2.31 = 161.7 \text{ ft.} \\
\text{Work done in 24 hr} \quad & = 133{,}333{,}333 \times 161.7 \\
& = 21{,}560{,}000{,}000 \text{ ft-lb.} \\
\text{Work done per hour} \quad & = 21{,}560{,}000{,}000 \div 24 \\
& = 898{,}333{,}333 \text{ ft-lb.} \\
\text{Steam used per hour} \quad & = 500 \times 15 = 7{,}500 \text{ lb.} \\
\text{Duty per 1,000 lb of steam} \quad & = 898{,}333{,}333 \div 7.5 \\
& = 119{,}777{,}000 \text{ ft-lb.}
\end{aligned}
$$

(*b*) Net heat chargeable to engine per pound of steam

$$= 1{,}189.6 - 94 = 1{,}095.6 \text{ Btu.}$$

[1] For approximate results, the weight of 1 gal of water may be taken as $8\frac{1}{3}$ lb and a water pressure of 1 psi as equal to a head of 2.31 ft. These figures are correct for water at 65 F. If the temperature is above, or below, this and greater accuracy is required, it will be necessary to change these amounts to correspond with the actual weight of 1 cu ft of water at the given temperature.

One inch of mercury equals a pressure of 0.491 lb.

Total heat charged to engine per hour

$$= 1{,}095.6 \times 7{,}500 = 8{,}217{,}000 \text{ Btu.}$$

$$\text{Duty per } 1{,}000{,}000 \text{ Btu} = 898{,}333{,}333 \div 8.217$$
$$= 109{,}326{,}000 \text{ ft-lb.}$$

194. Graphical Determination of Initial Condensation.—Initial condensation may be determined graphically from the indicator diagram. In determining the amount of steam working in the engine cylinder, the amount supplied to the engine per stroke is determined either by weighing the water entering the boiler, which passes over as steam into the engine, or by weighing the steam condensed in a condenser attached to the exhaust of the

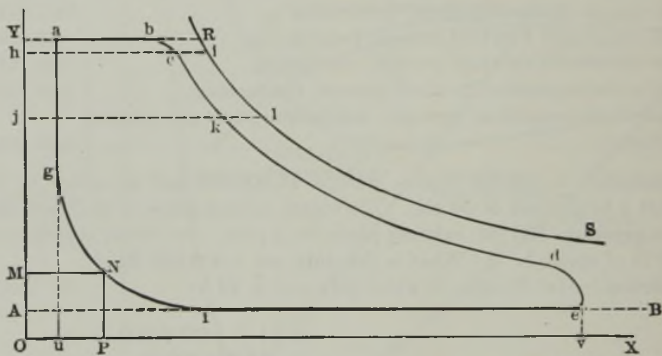


FIG. 201.—Indicator diagram and saturation curve, showing effect of initial condensation.

engine. This total quantity of steam used by the engine is then reduced to the amount of steam used per stroke, and is called the *cylinder feed*.

To determine the amount of *cushion steam*, an average indicator diagram is selected, and at a point after compression has begun and it is certain that the valve is closed, the pressure is measured and the volume determined. This volume must include the volume of the clearance. From this pressure and volume, by reference to the steam tables, the weight of the cushion steam may then be calculated, assuming the steam to be saturated. The total steam in the cylinder during expansion is then found by adding this cushion steam to the cylinder feed. A curve of saturation for this total quantity of steam can then be drawn upon the indicator diagram, and this curve will represent the volume the steam would have occupied at each point of the stroke if no initial condensation had occurred.

Figure 201 shows a saturation curve RS, constructed on an indicator diagram. YR represents the volume of the steam as used in the engine per stroke, or, in other words, it represents the volume of the total steam in the cylinder at boiler pressure, assuming no initial condensation. The curve RS represents the volume of this same weight of steam for the varying pressure of expansion. The difference in the volume between this theoretical expansion line and the actual expansion line represents the loss in volume due to condensation. The precentage of initial condensation at the point of cutoff will be $\frac{ci}{hi}$, and at any other point, such as k, will be $\frac{kl}{jl}$.

Example.—An 8-in. by 12-in. double-acting engine runs 230 rpm and uses 700 lb steam per hr. The steam pressure is 100 psi; exhaust atmospheric; clearance, 10 per cent; scale of indicator spring, 60 lb. Find the total weight of steam in the cylinder during expansion.

Solution.—First find the cylinder feed, or amount of steam supplied by the boiler to the engine per stroke.

$$\text{Strokes per hour} = 230 \times 2 \times 60 = 27{,}600.$$
$$\text{Cylinder feed} = 700 \div 27{,}600 = 0.02536 \text{ lb.}$$

To find the amount of cushion steam, first lay off from u (Fig. 201), the distance uO equal to 10 per cent of uv, since the clearance is 10 per cent and uv represents the volume swept through by the piston per stroke. If the length uv of the card is 2.9 in., the total length Ov is 3.2 in.

The volume swept through by the piston is $3.1416 \times 4 \times 4 \times 12 = 603.2$ cu in. The clearance volume is then 60.3 cu in., and the total volume 663.5 cu in. In other words, each inch of length of the line Ov represents $663.5 \div 3.2 = 207$ cu in.

Now take a point on the compression curve after the exhaust valve has closed, such as N. The coordinates of this point measured from the axes OX and OY are, $p = 34.8$ psia, and $v = 124.2$ cu in. $= 0.07187$ cu ft. From the steam tables it is found that 1 lb of dry saturated steam at a pressure of 34.8 psia has a volume of 11.964 cu ft.

The weight of 0.07187 cu ft, or the cushion steam, will then equal

$$0.07187 \div 11.964 = 0.006 \text{ lb.}$$

The total weight of steam in the cylinder during expansion is, therefore,

$$0.0254 + 0.006 = 0.0314 \text{ lb.}$$

Finally, plot the curve of saturation for 0.0314 lb of steam. To do this, take any pressure, such as 80 psia, and from the steam tables find the volume of 1 lb of steam at that pressure. This equals 5.47 cu ft. The volume of 0.0314 lb would then be

$$0.0314 \times 5.47 = 0.1718 \text{ cu ft} = 297 \text{ cu in.}$$

Hence the ordinates of this point will be

$$p = \tfrac{80}{60} = 1.33 \text{ in.}$$

and

$$v = \tfrac{297}{207} = 1.43 \text{ in.}$$

This point is then plotted, and others are found and plotted in the same way. A curve drawn through these points will be the saturation curve.

195. Indicator Diagrams.—The indicator is very often used to determine the setting of the valve and the distribution of steam in the cylinder. Figure 202 shows a typical indicator diagram

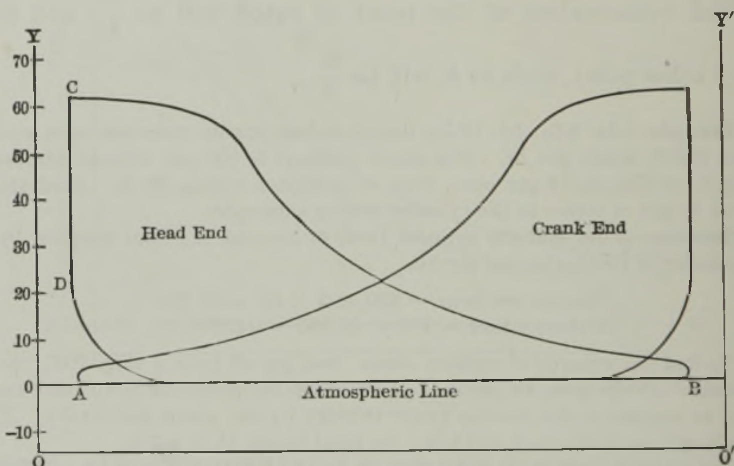


FIG. 202.—Indicator diagram from noncondensing engine.

from a high-speed engine running noncondensing. AB is the atmospheric line, and OO' the line of absolute vacuum, or zero pressure absolute. OY is the line of no volume for the head end, and $O'Y'$ for the crank end of the cylinder. The horizontal distance between the lines OY and CD represents the clearance volume for the head end of the cylinder. The clearance on the crank end is similarly shown.

196. Setting the Valve by Measurement.—In setting a valve, the first step is to place the engine on *dead center*, *i.e.*, the piston at the extreme end of its stroke. To do this, proceed in the following way: Place the engine near the center and turn it away from the center about 15 deg. Measure with a tram from a fixed point on the frame to the flywheel and mark the wheel. While in the same position, mark a line across the crosshead and the crosshead guide. Now turn the engine past the center until the lines on the crosshead and the crosshead guide again coincide. From the same point on the frame, mark the flywheel again with the tram. Bisect the distance between the tram marks, and turn the flywheel until this point of bisection is just the length of the tram from the fixed point on the frame. The engine will now be on center. The opposite center can be determined in the same way.

The next step is properly to place the valve on the valve stem. The engine being on center, move the eccentric on the shaft

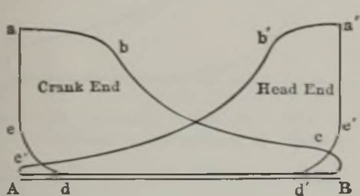


Fig. 203.—Indicator diagram, showing unequal distribution of work in the two ends of the cylinder.

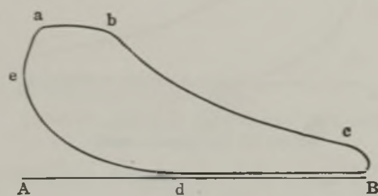


Fig. 204.—Indicator diagram, showing effect of insufficient lead.

until the valve has a slight lead. Measure this lead very carefully. Now place the engine on the opposite center, and again measure the lead. If the lead is not the same, move the valve on the stem one-half of the difference. Then repeat the operation until the lead at both ends is the same. The valve is now traveling equally over both steam ports. Now move the eccentric on the shaft, the engine being kept on the center, until the port is just closed, and then move it ahead to the amount of the lead desired. The lead is set anywhere from "line and line" to $\frac{3}{16}$ in., depending upon the speed and size of the engine.

197. Setting the Valve by the Indicator.—It is difficult to set the valve exactly by measurement. After the valve has been set by measurement, it is best to check the setting with an indicator.

When the valve is not set in the proper position on the stem, the steam admission at the two ends of the cylinder will not be alike, and the indicator diagram will appear as shown in Fig. 203. The objection to this diagram is that one end of the cylinder is doing more work than the other. In single-valve engines, this condition may be remedied by changing either the position of the valve on the stem, or the length of the valve stem. In the Corliss engine, it is changed by varying the relative length of the governor rods to the two admission valves.

Figure 204 shows an indicator diagram taken on an engine where the valve has insufficient lead. This diagram can usually be corrected by changing the position of the eccentric on the shaft. The eccentric should be changed in position until the line *ea* is a vertical line.

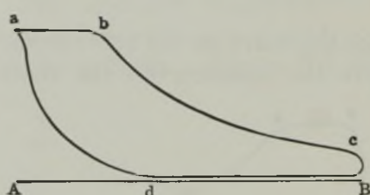


FIG. 205.—Indicator diagram, showing effect of too much lead.

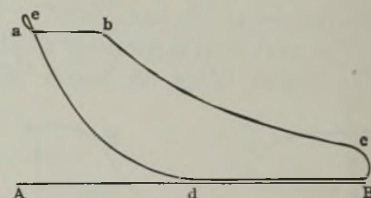


FIG. 206.—Indicator diagram, showing effect of too much compression.

Figure 205 shows an indicator diagram with too much lead. As before, this diagram may be corrected by changing the eccentric.

Figure 206 shows an indicator diagram with too much compression. In single-valve engines with automatic governors this often occurs at light load. In Corliss engines it may be corrected by changing the length of the rod connecting the valve and wrist plate.

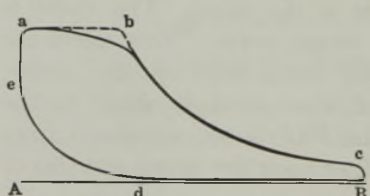


FIG. 207.—Indicator diagram, showing effect of wiredrawing.

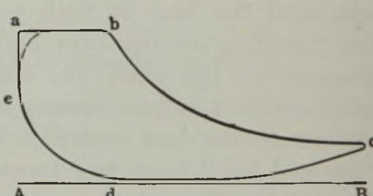


FIG. 208.—Indicator diagram, showing effect of insufficient exhaust lead.

Figure 207 shows an indicator diagram in which the admission line *ab* is a falling line. This is due to friction in the admission valve, which is usually caused by the valves opening slowly. With rapidly opening valves, such as a Corliss valve, the admission line will have the dotted position.

The indicator diagram shown in Fig. 208 has insufficient exhaust lead, *i.e.*, the point of release is too late. With a single-valve engine this condition of exhaust lead will usually be accompanied by insufficient steam lead, and the admission line will be as shown in the dotted position. Correcting the steam lead will correct the exhaust lead. In a four-valve engine, the lead of the exhaust valve should be increased.

When an engine is operated with a very light load, the cutoff may be so short that the steam will be expanded below atmospheric pressure before the valve opens to exhaust. As shown

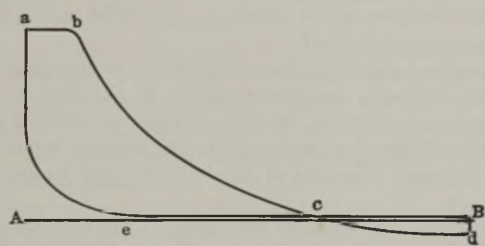


FIG. 209.—Indicator diagram, showing effect of too short cutoff.

in Fig. 209 this gives a loop of negative work from c to d, and shows an uneconomical condition of operation. When this occurs regularly, the engine is too large for the work it has to do. The best way to correct it is by reducing the steam pressure until the cutoff is long enough so that expansion is not carried below atmospheric pressure.

PROBLEMS

1. An engine has zero lead and the angle of advance is 45 deg. At what per cent of the stroke will cutoff occur if the connecting rod is considered infinitely long?

2. A certain valve has a travel of 5 in. With the crank on dead center, this valve is displaced 1 in. from its mid-position. If the steam lead is $\frac{1}{16}$ in., what is the steam lap equal to? The width of the port is 1.25 in.

3. (*a*) Given a valve with a travel of 3 in. The maximum port opening when taking steam is $\frac{7}{8}$ in. Determine the steam lap necessary to give this port opening. (*b*) If the exhaust lap on the same valve is $\frac{1}{8}$ in., what would the width of the steam port have to be so as to prevent overtravel of the valve? Show by a sketch the reasons for your answer.

4. An electrical plant runs a factory having five 10-hp motors, two 20-hp motors, four 30-hp motors. Efficiency of the motors, 80 per cent; of the transmission, 80 per cent; of the engine and dynamo combined, 80 per cent. What should be the rated indicated horsepower of the engine and kilowatts of the generator?

5. A streetcar plant uses 10 cars, each requiring an average horsepower of 75 at the wheels. The efficiency of the car is 60 per cent; of transmission

75 per cent; of the substations, 75 per cent; and of the main engines and dynamo, 75 per cent; mep of the engine, 40 lb; rpm, 150. The plant has two engines of equal size. What are the dimensions of their cylinders? Assume 600 fpm piston speed.

6. Assume the mep to be 40 lb, the number of revolutions to be 75 per min, and the length of the stroke to be 42 in., and determine the diameter of the cylinder of a double-acting engine which will develop 200 hp.

7. An engine is to develop 600 ihp at a piston speed of 600 fpm. The initial steam pressure is 100 psi; exhaust pressure, 1 psi; cutoff at one-fourth stroke; speed, 100 rpm; diagram factor, 85 per cent. (*a*) What should be the stroke and the diameter of the cylinder? (*b*) What should be the diameter if the back pressure is 2 psia?

8. It is desired to build a double-acting steam engine that will develop 150 ihp under the following conditions: cutoff, 22 per cent of stroke; clearance, 8 per cent; steam pressure, 150 psia; exhaust pressure, 2 psia; average piston speed, 600 fpm; speed of engine, 240 rpm; diagram factor, 80 per cent. Determine the required diameter of the cylinder and length of stroke in inches.

9. A double-acting steam engine operating at 240 rpm takes steam at a pressure of 160 psia and exhausts into a vacuum of 25.46 in. Hg. Cutoff, one-fourth stroke; mean total force on the piston, 5,336 lb; average piston speed, 720 fpm; clearance, 8 per cent; barometer, 29.54 in. Hg; diagram factor 80 per cent. Determine the cylinder dimensions and the indicated horsepower being developed.

10. An engine is 18 in. by 30 in.; cutoff, one-fourth stroke. It runs 100 rpm. The initial steam pressure is 80 psi; exhaust, atmospheric. What would be the increase of horsepower if the cutoff was increased to one-half stroke and initial pressure to 150 psi? Diagram factor, 80 per cent.

11. An 18-in. by 30-in. engine runs 100 rpm with an initial pressure of 150 psi; atmospheric exhaust. A condenser is added bringing the exhaust pressure down to 2 psia. In both cases cutoff occurs at one-fourth stroke. The diagram factor is 80 per cent. (*a*) How much is the rated indicated horsepower of the engine increased? (*b*) If the power is sold for $1\frac{1}{2}$¢ per hp-hr and the plant runs 10 hr a day, 300 days per year, how much more would be earned per year? (*c*) If the condenser costs $2,000, what per cent profit would be realized? Allow 15 per cent for fixed charges.

12. The cylinders of a locomotive are 19 in. in diameter and the stroke is 24 in. The driving wheels are 7 ft in diameter, and the mean back pressure against which the piston works is 19 psia. Determine the indicated horsepower theoretically developed by the locomotive when taking steam at a pressure of 150 psi and cutting off at five-eighth stroke, while traveling at a speed of 40 mph. Diagram factor, 75 per cent.

13. A tank contains 2,000 cu ft of air at a pressure 1,000 psia and a temperature of 60 F. The tank operates a double-acting air engine 6 in. by 10 in. The engine admits air for one-fourth of the stroke and the air expands in the cylinder for the remainder. The engine runs 200 rpm. The pressure of the air entering the engine is 100 psia. How long will the engine run to bring the pressure in the tank down to 500 psia?

14. A double-acting compressed-air locomotive has two air tanks 3 ft in diameter and 12 ft long. These tanks supply two 8-in. by 12-in. cylinders. The cylinders take their air through a reducing pressure valve at 100 psi, and the original pressure in the tanks was 1,100 psi. Assuming the air to act in the cycle at 60 F and the expansion in the engine to be isothermal, (*a*) how long will the tanks run the engine with one-fourth cutoff in the cylinders and 150 rpm, and (*b*) how many horsepower will be developed in the cylinders? Diagram factor, 90 per cent.

15. An engine has a clearance volume which is 0.08 of the volume swept through by the piston per stroke. If the steam be cut off at one-fifth stroke, what will be the ratio of expansion?

16. A 9-in. by 16-in. engine has a clearance volume of 10 per cent. The engine runs 220 rpm, cuts off at one-fourth stroke, has an initial steam pressure of 120 psi, and exhausts into a vacuum of 22 in. Hg. The barometer reading is 28.5 in. Hg; diagram factor, 80 per cent. Determine (*a*) the ratio of expansion, and (*b*) the rated indicated horsepower.

17. The exhaust steam from an engine is to be used to heat a group of buildings which will require 15,000,000 Btu per hr. The engine receives dry steam at 190 psia and exhausts at 18 psia, and uses 25 lb of steam per ihp-hr. Five per cent of the heat in the exhaust steam is lost by radiation before reaching the heating system. All of the exhaust steam is condensed in the heating system, the condensate being returned to the boilers at 180 F. What load must be carried on the engine (indicated horsepower) so that there will be sufficient steam for heating purposes?

18. A 12-in. by 14-in. double-acting engine runs 260 rpm and cuts off at three-eighth stroke. The steam pressure is 70 psi; back pressure. 2.7 psia; diagram factor, $77\frac{1}{2}$ per cent. What is the weight of steam actually used per horsepower per hour, assuming that one-fourth of that theoretically required is lost through radiation, condensation, etc.

19. A 10-in. by 12-in. double-acting, noncondensing steam engine runs 250 rpm; cutoff, one-fourth stroke; steam pressure, 80.3 psi. (*a*) What is the rated indicated horsepower if the diagram factor is 90 per cent? (*b*) Find the steam consumption per horsepower-hour, allowing 20 per cent initial condensation. (*c*) Find the size of pipe necessary to supply steam to engine, assuming a steam velocity of 5,000 fpm.

20. What is the efficiency of a Rankine cycle where steam at a pressure of 200 psia superheated 200 deg goes through a cycle, exhausting at a pressure of 1 psia?

21. Two pounds of steam at a pressure of 130 psia and having a volume of 8.538 cu ft pass through a Rankine cycle in which the exhaust pressure is 25 psia. What is the efficiency of the cycle?

22. A 16-in. by 18-in. engine runs 200 rpm; initial steam pressure, 120 psi; area of indicator diagram from head end, 1.4 sq in.; from crank end, 1.45 sq in.; length of diagram, 3 in.; scale of spring, 60 lb; diameter of piston rod, 3 in.; mechanical efficiency of engine, 90 per cent. A Prony brake on the engine has an arm of 4 ft. The tare of the brake is 25 lb. What will be the weight on the scales?

23. Given a direct-current generator having a full-load rating of 100 kw, and an electrical efficiency of 90 per cent at full load. The speed of the generator is 240 rpm. A simple high-speed engine is to be chosen for direct connection to this generator. The piston speed of the engine is to be 750 fpm. The mechanical efficiency of the engine is 90 per cent at full load. (*a*) What brake horsepower must be available at the engine shaft? (*b*) What must be the indicated horsepower of the engine? (*c*) Determine the stroke of the engine in inches? (*d*) Determine the bore of the engine cylinder in inches. Assume the engine double-acting, neglect the area of the piston rod, and assume that the mean effective pressure is 40 psi on each end.

24. The area of the indicator diagram from the head end of an 8-in. by 12-in. double-acting engine running at 227 rpm is 1.30 sq in. and the area of the diagram from the crank end is 1.21 sq in. The length of each diagram is 2.90 in. A 60-lb spring was used in the indicator. The diameter of the piston rod is $1\frac{1}{2}$ in. A Prony brake was applied to the engine and the gross weight on the scales was 106.0 lb. The length of the brake arm is 54 in. and the tare is 28.5 lb. The engine used steam at the rate of 650 lb per hr, steam being supplied at a pressure of 135.5 psi with a quality of 98 per cent. The pressure of the exhaust was 0.5 psi. The barometer reading was 29.6 in. Hg. Find (*a*) the indicated horsepower; (*b*) the brake horsepower; (*c*) the friction horsepower; (*d*) the mechanical efficiency; (*e*) the actual thermal efficiency based on the indicated horsepower.

25. The following data were obtained during a test of an 8-in. by 12-in. steam engine: rpm 230; steam used per hour, 610 lb; area of head-end indicator diagram, 1.21 sq in.; area of crank-end diagram, 1.30 sq in.; length of diagram, 2.9 in.; scale of indicator spring, 60 lb; tare of Prony brake, 40.5 lb; gross load on brake scales, 106 lb; length of brake arm, 54 in.; steam pressure, 135.5 psi, and quality, 98 per cent; vacuum in condenser, 27.0 in. Hg; barometer, 29.5 in. Hg; diameter of piston rod 1.5 in. Find (*a*) the indicated horsepower; (*b*) the mechanical efficiency; (*c*) the thermal efficiency based on the indicated horsepower; (*d*) the Rankine cycle efficiency for the engine.

26. A 10-in. by 14-in. reciprocating steam engine is direct connected to a reciprocating pump. Steam is taken into the engine at a pressure of 100 psia during the entire stroke and exhausts at a pressure of 16 psia during the entire return stroke. The engine is double acting and makes 120 strokes per min. The piston rod is 2 in. in diameter. Neglect radiation and leaks. Find (*a*) the indicated horsepower being developed; (*b*) the steam consumption per indicated horsepower per hour; (*c*) the actual thermal efficiency based on the indicated horsepower.

27. Given a single-acting steam engine whose bore is 11 in.; piston speed, 480 fpm; cutoff at 30 per cent of stroke; line pressure, 140 psi; vacuum at engine, 22.43 in. Hg; barometer, 29.93 in. Hg; clearance, 11 per cent; diagram factor, 85 per cent. (*a*) Determine the indicated horsepower being developed. (*b*) If the steam consumption of the engine is 16 lb per bhp-hr, what will be the over-all thermal efficiency?

28. A steam engine operates with a thermal efficiency of 9 per cent based on brake horsepower, when using 1,230 lb of dry steam per hr at a pressure of 155 psia and exhausting against a back pressure of 2 psia. The engine was loaded by means of a Prony brake having a lever arm 63 in. long. The engine ran at 220 rpm. The friction horsepower was measured as 5.5. (*a*) What was the net load registered on the scales? (*b*) What was the thermal efficiency based on the indicated horsepower?

29. How many pounds of dry steam per hour are necessary to develop one horsepower if an engine, operating on the Rankine cycle, takes steam at 140 psia and exhausts at 2 psia?

30. An engine uses 20 lb of steam per ihp-hr at a pressure of 150 psia and containing 2 per cent moisture, and exhausts at a pressure of 2 psia. Find (*a*) the thermal efficiency of the Rankine cycle; (*b*) the actual thermal efficiency based on the indicated horsepower.

31. A steam engine is supplied with 848 lb of steam per hr at 135.8 ps and 500 F and exhausts into a condenser at 24 in. Hg vacuum. The barometer reading is 29.00 in. Hg. The engine develops 50 bhp and has a mechanical efficiency of 83.3 per cent. Calculate (*a*) the indicated horsepower; (*b*) the Rankine cycle efficiency; (*c*) the thermal efficiency based on brake hp; (*d*) the engine efficiency.

32. A large engine, operating condensing, is supplied with steam at 200 psia and a temperature of 420 F and exhausts against a back pressure of 3 in. Hg abs. The heat chargeable per horsepower-hour is 18,000 Btu. Calculate (*a*) the steam rate in pounds per horsepower-hour; (*b*) the actual thermal efficiency of the engine; (*c*) the Rankine cycle efficiency of the engine.

33. A steam engine is supplied with steam at 150 psia and containing 1 per cent moisture. The steam is exhausted at 2 psia. (*a*) Determine the Rankine cycle efficiency. (*b*) If the engine drives a generator and uses 18 lb of steam per kwhr, what is its over-all thermal efficiency?

34. Given a 500-kw generating set; mechanical and electrical efficiency of engine and generator, 85 per cent; steam pressure, 150 psia; exhaust temperature, 180 F. The engine uses 20 lb of steam per ihp-hr. Evaporation from and at 212 F is 10 lb of water per lb of dry coal. Coal contains 13,000 Btu per lb. (*a*) What is the heat efficiency of plant based on the indicated horsepower? (*b*) Using a velocity of 5,000 fpm for steam, calculate the diameter of the pipe necessary to supply the steam to the engine.

35. A steam plant burns 6,000 lb of coal per hr; coal contains 13,500 Btu per lb; efficiency of boiler, furnace, and grate, 70 per cent; heat lost in steam pipes, 5 per cent; heat efficiency of engine, 15 per cent; mechanical efficiency of engine, 90 per cent; electrical efficiency of generator driven by engine, 88 per cent. (*a*) What is the indicated horsepower developed by engine? (*b*) What is the kilowatt output of the generator?

36. A 9-in. by 15-in. engine takes steam at a pressure of 100 psi and runs 200 rpm. It uses 1,128 lb steam per hr at one-fourth cutoff; clearance, 10 per cent. A point on the compression curve at 88 per cent of the return stroke has an absolute pressure of 20 psi. Find the percentage of initial condensation at cutoff.

37. Given a feed-water heater working under the following conditions: Pounds of water entering heater per hour, 20,469; temperature of water entering heater, 46.3 F; temperature of water leaving heater, 209.4 F; pressure in heater, 22.3 psia; pressure of steam supplied to engine, 120 psi; barometer reading, 29.5 in. Hg; quality of steam entering engine, 99.2 per cent; pressure of exhaust from engine, 22.3 psia. The engine uses 22.25 lb of steam per ihp-hr. Determine (*a*) the heat used by the engine per pound of steam; (*b*) the quality of the exhaust steam; (*c*) the pounds of steam used by the heater per hour.

38. A double-acting engine is 12.65 in. by 15 in. and has a $2\frac{1}{4}$-in. piston rod. The area of the head-end diagram is 0.87 sq in. and of crank-end diagram, 0.8 sq in.; length of each, 3 in.; scale of spring, 80 lb. A Barrus calorimeter is attached to the steam line and the temperature of steam in the main is 326 F, and after passing the calorimeter orifice is 231 F. The barometer reading is 29.1 in. Hg. The engine running condensing and making 262 rpm used 3,150 lb of steam per hr and the temperature of exhaust was 152.5 F. The temperature of the feed water was 94 F. A Prony brake was attached to the engine and carried a net load of 190 lb. The length of the brake arm is 60.25 in. and the tare 55 lb. The boiler evaporates 9 lb of water per lb of coal and the coal costs $2.25 per ton. The feed water costs 7¢ per 1,000 gal. Find (*a*) the indicated horsepower; (*b*) the brake horsepower; (*c*) the cost of coal per indicated horsepower per hour; (*d*) the cost per brake horsepower per hour; (*e*) the cost of boiler feed per indicated horsepower per hour; (*f*) per brake horsepower per hour; (*g*) the total heat supplied to steam above temperature of boiler feed per hour; (*h*) the thermal efficiency of the Rankine cycle; (*i*) the actual thermal efficiency of the engine based on the indicated horsepower; and (*j*) the boiler horsepower necessary to supply the steam for the engine running under the given load.

39. A double compound pumping engine (two high-pressure and two low-pressure steam cylinders), 12 in. and 22 in. by 24 in., runs at 60 rpm. Steam is supplied at 75 psia containing 2 per cent moisture and is exhausted into a condenser in which a vacuum of 27.5 in. Hg is maintained. The barometer reads 29.50 in. Hg. Cutoff in the high-pressure cylinders occurs at 80 per cent of the stroke. The diagram factor is 0.90. (*a*) What is the rated indicated horsepower of the steam cylinders for these conditions? (*b*) If the horsepower appearing as useful work of pumping water amounts to 80 per cent of the rated indicated horsepower and the total head against which the pump operates is 100 ft, how many gallons per 24 hr does the pump deliver?

40. A 15-in. and 24-in. and 36-in. by 30-in. compound engine runs at 100 rpm. Cutoff in the high-pressure cylinder is at three-eighths stroke; in the intermediate cylinder at 39 per cent; and in the low-pressure cylinder at 44 per cent of the stroke. Clearance in each cylinder is 10 per cent. Determine the actual ratio of expansion for the engine.

41. What will the cutoff have to be in a 24-in. by 30-in. simple engine running 180 rpm, in order to develop the same horsepower as is developed by a 16-in. and 24-in. by 30-in. compound engine running at the same speed and having a cutoff at one-quarter stroke in the high-pressure cylinder and at one-half stroke in the low-pressure cylinder? The initial pressure is 160 psia; back pressure 2 psia; diagram factor 85 per cent in both cases.

42. An engine is 9 in. and 15 in. by 9 in. It runs 200 rpm and the cutoff in the high-pressure cylinder is one-fourth stroke. The initial pressure is 150 psia; and back pressure, 2 psia; diagram factor, 80 per cent. What should be the diameter of the cylinder of a simple engine having the same length of stroke, to develop the same indicated horsepower when running at the same speed, the initial and back pressures and ratio of expansion being the same as with the compound engine?

43. What is the rated indicated horsepower of a double-acting, triple-expansion steam engine, 18 in. and 28 in. and 36 in. by 30 in.? The engine exhausts into a condenser in which a vacuum of 25.46 in. Hg is maintained. The barometer reading is 29.53 in. Hg. The engine runs 100 rpm under a steam pressure of 185.5 psi. Assume a diagram factor of 80 per cent, cutoff in the high-pressure cylinder at one-fourth stroke, in the intermediate cylinder at 0.3 stroke, and in the low-pressure cylinder at 0.35 stroke.

44. It is desired to build a compound engine with its low- and high-pressure cylinders double acting. This engine is to develop 600 ihp when using steam having an initial pressure of 150 psia and back pressure of 2 psia. The speed of the engine is to be 150 rpm and the piston speed in each cylinder 750 fpm. If the ratio of the cross-sectional areas of the low-pressure to the high-pressure cylinder is 4 and the total ratio of expansion is 12, find the dimensions of the engine assuming a diagram factor of 0.8.

45. A single-acting engine is 9 in. and 15 in. by 9 in. and runs 160 rpm Cutoff in high-pressure cylinder, one-fourth stroke; in low-pressure cylinder, one-half stroke; diagram factor for both cylinders, 85 per cent. The steam pressure is 125 psia; back pressure, 3 psia; area of indicator diagram from high-pressure cylinder, 0.9 sq in.; from low-pressure cylinder, 1.15 sq in.; length of each, 2.35 in. An 80-lb spring is used in the indicator on the high-pressure cylinder and a 40-lb spring in indicator on the low-pressure cylinder. (*a*) What is the rated indicated horsepower of the engine? (*b*) What indicated horsepower is being developed in the cylinders?

46. An engine is 9 in. and 15 in. by 9 in. and runs 320 rpm, the cutoff in the high-pressure cylinder being one-fourth stroke. The steam pressure is 125 psi; back pressure, 3 psia. The engine is single acting and the area of the indicator diagram from the high-pressure cylinder is 0.9 sq in.; from the low-pressure cylinder, 1.1 sq in. The length of each is 2.35 in. An 83-lb spring is used in the indicator on the high-pressure cylinder and a 40-lb spring in the indicator on the low-pressure cylinder. The engine is fitted with a Prony brake carrying a gross weight of 120 lb. The tare of the brake is 20 lb and the length of the brake arm is 51 in. Find (*a*) the indicated horsepower; (*b*) the brake horsepower; (*c*) the friction horsepower; and (*d*) the mechanical efficiency.

47. The following information was obtained during a test of a Buckeye cross-compound engine: cylinder dimensions, 13 in. and 25 in. by 16 in.; diameter of piston rods, $2\frac{1}{2}$ in.; speed, 190 rpm; steam pressure at throttle, 130 psi; barometer, 29.53 in. Hg; quality of steam, 99 per cent; steam pressure in receiver, 10.82 psi; vacuum in condenser, 21.5 in. Hg; steam used per indicated horsepower per hour, 16.35 lb; steam used per brake horsepower per hour, 18.2 lb; cutoff in high-pressure cylinder, one-fourth stroke, diagram factor, 85 per cent.

Mep, high-pressure cylinder, head end, 54.5 psi.
Mep, high-pressure cylinder, crank end, 60.3 psi.
Mep, low-pressure cylinder, head end, 12.0 psi.
Mep, low-pressure cylinder, crank end, 14.63 psi.

Determine (*a*) the rated indicated horsepower of engine; (*b*) the indicated horsepower being developed; (*c*) the actual heat efficiency based on the brake horsepower; (*d*) the Rankine-cycle efficiency.

48. Given the following information:

A triple-expansion, 28-in. and 54-in. and 80-in. by 60-in. pumping engine is cross-connected to a 30.5-in. by 60-in., three-cylinder, single-acting, plunger-type pump; diameter of piston rod, 6.5 in.; steam pressure at throttle, 150 psi; barometer reading, 28.95 in. Hg; vacuum gage on condenser shows 26 in. vacuum. The engine runs 20 rpm and uses 12 lb of steam per ihp-hr.

Average mep, high-pressure cylinder, head end, 59.5 psi.
Average mep, high-pressure cylinder, crank end, 56.5 psi.
Average mep, intermediate cylinder, head end, 14.0 psi.
Average mep, intermediate cylinder, crank end, 13.5 psi.
Average mep, low-pressure cylinder, head end, 4.7 psi.
Average mep, low-pressure cylinder, crank end, 5.0 psi.

The center of the gage on the discharge pipe is 15 ft above its point of attachment; the center of the suction gage is 5 ft above its point of attachment to the suction pipe and 10 ft below the center of the discharge gage. The discharge gage indicates a pressure of 60 psi and the suction gage indicates 20 in. Hg vacuum. The pump delivers 15,000,000 gal in 24 hr. Determine (*a*) the mechanical efficiency of the unit; (*b*) the actual thermal efficiency based on the output; and (*c*) the duty per million heat units.

49. A small power plant contains a double-acting, compound condensing engine, a pumping engine, and the necessary boilers operating under the following conditions: steam pressure, 125 psi; feed-water temperature, 140 F; quality of steam, 98 per cent. Boiler evaporation is equivalent to 10 lb of steam from and at 212 F per pound of coal fired. Coal costs $5 per ton (2,000 lb) and contains 82 per cent C, 7 per cent O, 4 per cent H, and 6 per cent ash by analysis; ash and refuse taken from ashpit, 13 per cent. The plant runs 24 hr per day, 365 days per year.

The compound engine is 9 in. and 15 in. by 9 in., double-acting, and runs 300 rpm. The back pressure is 3 psia; steam consumption, 18 lb per ihp-hr; diameter of piston rods, 2 in.

Area indicator diagram from high-pressure cylinder, head end, 1.00 sq in.
Area indicator diagram from high-pressure cylinder, crank end, 0.90 sq in.
Area indicator diagram from low-pressure cylinder, head end, 1.00 sq in.
Area indicator diagram from low-pressure cylinder, crank end, 0.90 sq in.

Length of all diagrams 2.35 in. An 85-lb spring is used in the indicator on the high-pressure cylinder. A 40-lb spring is used in the indicator on the low-pressure cylinder. The engine is fitted with a Prony brake carrying a gross weight of 240 lb. The tare is 20 lb and the length of the brake arm 51 in.

The pumping engine has a capacity of 20,000,000 gal of water per 24 hr, pumping against a head of 70 lb and has a duty of 120,000,000 ft-lb per 1,000,000 Btu. The back pressure is 3 psia.

Determine (*a*) the indicated horsepower; (*b*) the brake horsepower and (*c*) the mechanical efficiency of the compound engine; (*d*) the heat efficiency of the pumping engine; (*e*) the probable boiler horsepower necessary to operate the plant; (*f*) the coal cost per year; and (*g*) the heat efficiency of the plant.

50. A 500-hp engine pumps 20,000,000 gal of water in 24 hr. Steam used by the engine is 12 lb per hp-hr; steam pressure, 200 psia; quality of steam, 100 per cent; exhaust pressure, 3 psia; barometer, 29 in. Hg; duty of pump, 160,000,000 ft-lb per 1,000,000 Btu. (*a*) Find the total head in feet. (*b*) What vacuum in inches of Hg is being maintained in the condenser?

51. An engine developing 60 ihp supplies power to a pump delivering 6,000 gal of water in 5 min. The suction gage reads 15 in. Hg vacuum and is attached 4 ft below the center line of the pressure gage. Determine (*a*) the discharge pressure in pounds per square inches, if the over-all efficiency is 75 per cent; (*b*) the duty per 1,000,000 Btu, if the engine uses 20 lb of steam per ihp-hr; steam pressure, 160 psia; exhaust pressure, 4 psia.

52. A centrifugal pump delivers 1,000 gal of water per min against a discharge pressure of 75 psi, while the suction gage reads 10 in. Hg vacuum referred to a 29.5 in. barometer. The temperature of the water pumped is 70 F. The diameter of the suction pipe is 10 in. and the diameter of the discharge pipe is 8 in. Find the total head pumped against in feet and the water horsepower developed.

53. A steam-driven pumping unit delivers 60,000 gal of water per hr against a discharge pressure of 90 psi, while the suction gage reads 8 in. Hg. vacuum. The barometer reads 29.2 in. Hg. Dry steam is supplied to the engine at 150 psia and the exhaust pressure is 2 psia. The duty of the unit is 120,000,000 ft-lb per 1,000,000 Btu and the mechanical efficiency is 80 per cent. Determine (*a*) the water horsepower of the pump; (*b*) the steam consumption in pounds per water horsepower per hour.

54. A direct-acting boiler feed pump has a single steam cylinder and a single water cylinder. The pump makes 50 strokes (25 each way) per min. The steam cylinder has a bore of $7\frac{1}{2}$ in. and a stroke of 10 in., and takes steam at full stroke. The heat losses in the cylinder increase the apparent steam consumption $33\frac{1}{3}$ per cent. The pump delivers 45 gal of water per min against a total head of 100 psi. (*a*) What is the indicated horsepower of the steam cylinder if the mechanical efficiency of the unit is 60 per cent? (*b*) What is the steam consumption in pounds per hour?

55. A pumping engine delivers 30,000,000 gal of water in 24 hr. The pressure shown by the gage on the discharge side is 50 psi. The vertical distance between the center line of the discharge gage and the point of attachment of the suction gage is 11.55 ft, the discharge gage being the higher. The suction gage shows a reading of 20.37 in. Hg. The barometer reads 28 in. Hg. The engine develops 900 ihp with a steam consumption of 12 lb dry steam per ihp-hr. (*a*) What is the total head pumped against? (*b*) What horsepower is delivered by the pump cylinders? (*c*) What is the duty per 1,000 lb of steam? (*d*) What is the mechanical efficiency in per cent? (*e*) If the engine is supplied with dry saturated steam at a pressure of 125 psi, and the exhaust pressure is 1 psia, what is the duty per 1,000,000 Btu?

56. A turbine-driven centrifugal pump delivers 28,000,000 gal of water in 24 hr against a total head equivalent to 125 psi. The turbine is supplied with steam at a pressure of 175.3 psi and having a temperature of 500 F. The barometer indicates 29.92 in. Hg. The turbine exhausts into a condenser carrying a vacuum of 27.92 in. Hg. If the duty per 1,000 lb of steam is 112,000,000 ft-lb, what is the duty per 1,000,000 Btu?

57. A pumping engine having a duty of 120,000,000 ft-lb per 1,000,000 Btu develops 500 ihp and uses 13 lb of steam per ihp-hr. The steam pressure is 180 psi, and the exhaust pressure, 2 psia. The barometer reading is 29.4 in. Hg. Superheat is 50 deg. The pressure on the delivery side of pump is 120 psi. The vacuum gage reads 20 in. Hg. The distance between the center of pressure gage and the point of attachment of the vacuum gage is 2 ft. (*a*) How many pounds of water will this pump deliver per hour? (*b*) What will be the capacity in gallons per 24 hr?

58. A steam-driven pump delivers 1,000,000 gal of water per hr against a discharge pressure of 70 psi while the suction gage reads 10 in. Hg vacuum. The center line of the discharge gage and the point of attachment of the suction gage are at the center line of the pump. Dry steam is supplied to the engine at 200 psi and the engine exhausts into a condenser where the pressure is 2 psia. The duty of the unit is 100,000,000 ft-lb per 1,000,000 Btu. (*a*) Determine the pounds of steam used per hour. (*b*) What portion of the energy available on the basis of the Rankine cycle is utilized in the engine cylinder if the over-all mechanical efficiency of the pump and engine is 85 per cent?

59. A 24,000,000 gal pumping engine has a duty of 150,000,000 ft-lb per million Btu. Steam is supplied at a pressure of 200 psia and a temperature of 460 F and exhausts into a condenser carrying a vacuum of 28 in. Hg. The barometer reads 30 in. Hg. The pressure gage on the discharge line reads 100 psi and the suction gage reads a suction lift of 12 in. Hg. The suction gage is attached to the line 6 ft below the center of the discharge gage. The over-all mechanical efficiency of the unit is 90 per cent. (*a*) What is the steam consumption per indicated horsepower per hour? (*b*) If the temperature of the boiler feed water is 101 F, what boiler horsepower is required to supply the engine? (*c*) What is the efficiency of the Rankine cycle? (*d*) What per cent of the Rankine cycle efficiency is actually obtained in the engine cylinder?

60. The duty of a 12,000,000-gal pumping engine is 160,000,000 ft-lb per 1,000,000 Btu. The steam pressure is 180 psia; exhaust pressure, 2 psia; feed-water temperature, 120 F; reading of the pressure gage, 60 psi; reading of the suction gage, 20 in. Hg; distance between center of pressure gage and point of attachment of the suction gage, 10 ft. (*a*) What boiler horsepower will be required to operate the plant? (*b*) If the mechancal efficiency of the unit is 90 per cent, what will be the steam consumption per indicated horsepower per hour? (*c*) What is the actual thermal efficiency of the unit based on the output of the pump? (*d*) If the boiler efficiency is 70 per cent and the coal contains 13,000 Btu per lb, and costs $7.50 per ton, what will be the coal cost per year, if the plant operates 24 hr a day for 365 days per year?

61. A 30,000,000-gal pumping engine has a duty of 140,000,000 ft-lb per 1,000,000 Btu. The steam pressure is 150 psia; temperature of feed water, 130 F; back pressure, 3 psia. The pressure gage reads 70 psi, and the level of the water supply is 10 ft below center of the gage. (*a*) What boiler horsepower will be required to operate the pump, if the steam contains 2 per cent moisture? (*b*) If the mechanical efficiency of the pumping engine is 85 per cent, how many pounds of dry steam will it use per indicated horsepower per hour? (*c*) If the coal contains 13,000 Btu per lb of dry coal and 6 per cent moisture, the temperature of the boiler room is 70 F and of the stack gases 500 F, and the losses in the boiler equal 22 per cent, what will be the heat efficiency of the entire plant?

62. A 50,000,000-gal pumping engine has a duty of 160,000,000 ft-lb per 1,000,000 Btu. Steam is supplied at a pressure of 150 psi and contains $1\frac{1}{2}$ per cent moisture. The temperature of the exhaust steam is 120 F. The pressure gage on the discharge pipe indicates 70 psi, and the vacuum gage on the suction pipe indicates 14 in. Hg. The center of the pressure gage is 10 ft above the point of attachment of the suction gage. The barometer is 29.2 in. Hg. The mechanical efficiency of the unit is 88 per cent. (*a*) What is the steam consumption per indicated horsepower per hour? (*b*) If water is fed to the boiler supplying steam to this unit at a temperature of 120 F, what boiler horsepower is being developed? (*c*) If the coal contains 13,000 Btu per lb, and 2,200 lb are fired per hr, what is the efficiency of the boiler, furnace, and grate?

63. A 60,000,000-gal pumping engine has a duty of 160,000,00 ft-lb per 1,000,000 Btu; steam pressure, 150 psia; temperature of feed water, 140 F; back pressure, 5 psia. The pressure gage reads 70 psi; the vacuum gage reads 12 in. Hg; the distance between the center line of the pressure gage and the point of attachment of the suction gage is 10 ft. (*a*) What boiler horsepower will be required to operate the pump? (*b*) If the coal contains 13,000 Btu per lb and costs $3 per ton of dry coal, and the efficiency of the boiler, furnace, and grate is 70 per cent, what will be the cost of coal per year if the plant operates 24 hr per day and 365 days per year? (*c*) How much could be saved per year by installing a feed-water heater costing $800 that would increase the feed temperature to 200 F, allowing 10 per cent interest and depreciation?

64. A pumping engine delivers 18,000,000 gal of water in 24 hr against a discharge pressure of 60 psi. The gage attached to the suction pipe indicates 16 in. of Hg vacuum, and the distance between the center of the discharge gage and the point of attachment of the suction gage is 5 ft. The indicated horsepower being developed in the cylinder is 600 and the engine is using 15 lb of steam per ihp-hr. The steam pressure is 100 psi; barometer 29.92 in. Hg; quality of steam, $99\frac{1}{2}$ per cent. The engine exhausts into a surface condenser, the pressure in which is 0.95 psia. (*a*) Determine the duty per 1,000 lb of steam. (*b*) Determine the duty per 1,000,000 Btu. (*c*) The water supplied to the boilers passes first through an economizer. The temperature of the water entering the economizer is 80 F, and leaving economizer is 210 F. Water enters the boilers at a temperature of 200 F. What boiler horsepower is being developed? (*d*) What is the efficiency of the boiler plant, including the economizer, if the actual evaporation is 8 lb of water per lb of dry coal? The heating value of the coal is 13,000 Btu per lb. (*e*) What is the over-all efficiency of the plant, including the pumping engine?

65. The test of a compound pumping engine shows the following: water pumped in 24 hr, 14,400,000 gal; average steam pressure, 210.5 psi; degrees superheat in steam, 58.2; average vacuum in condenser, 25.44 in. Hg; average barometer reading, 29.52 in. Hg; water pressure, 80 psi; water level in reservoir is 15 ft below center of discharge gage; weight of steam used per hour, 6.757 lb. (*a*) Determine the duty in foot-pounds per 1,000,000 Btu. (*b*) As electrical energy is available at a low figure, the installation of a motor-driven centrifugal pump is being considered. This new unit will deliver the same quantity of water against the same total head. If the combined mechanical and electrical efficiency of the unit is 75 per cent, what will the power cost at 5¢ per kwhr?

CHAPTER X

STEAM TURBINES

198. Turbine vs. Piston Engine.—The steam turbine is a kinetic type of prime mover which depends entirely upon changes in velocity of the steam flowing through it to produce a turning force upon the rotor. On the other hand, the shaft of a piston engine is turned by the direct pressure of steam confined in a cylinder and acting upon a piston whose motion is transmitted to the crank.

Steam flows continuously through a turbine, in some places with low velocity and in others with high velocity. The turning force is produced by these variations in steam speed. In a piston engine, the steam flows intermittently, being admitted in measured amounts to the cylinder, where it performs its work and is then released.

The power developed by any prime mover, *i.e.*, its rate of doing work, is expressed by the relation

$$\text{Power} = F \times S,$$

where F = mean turning force on the rotor, and S = speed (linear velocity) of the point of application of the force. The fundamental distinction between the turbine and the piston type of engine lies in the character or origin of F. In the turbine type, F is derived entirely from velocity effects; in the piston type, F is derived directly from the fluid pressure of the working substance.

The initial impetus for the development of the practical steam turbine was given by the large manufacturers of electrical equipment in their search for a more satisfactory prime mover to drive electric generating machinery than the piston engine. The advantages of the turbine as compared with the reciprocating engine are (1) higher rotative speed and hence smaller physical size of machine for the same power output, (2) constant turning moment on the shaft so that little flywheel effect is needed, (3) essentially perfect balance of the moving parts since their motion is purely rotary.

199. Kinetic Force of a Jet or Fluid Stream.—The expression for the kinetic force of a jet may be derived directly from the laws of motion of individual particles. A small object such as a block of wood can be moved over the level surface of a table either by the direct pressure of the hand or by means of balls thrown against it in rapid succession, the impact of which will produce a series of intermittent forces whose average is equal to that necessary to move the block along by continuous pressure.

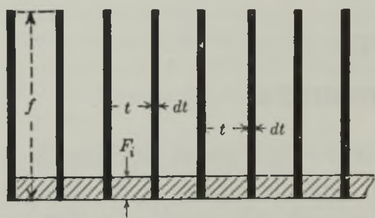


FIG. 210.—Relation between force of impulse, f, of individual projectiles, and average continous impulse force, F_i.

Let

w = weight of each ball in pounds;
V_1 = velocity of the balls in feet per second;
U = velocity of the block in feet per second;
t = time interval between the successive balls in seconds;
dt = time required to stop a ball, or rather to change its velocity from V_1 to U (assuming that the face of the block is covered with some plastic substance in which the balls stick and so do not bounce back);
f = force produced by each ball, acting for a time, dt;
F_i = average or equivalent continuous impulse force produced by the stream of balls.

The situation is represented by Fig. 210, and it is desired to derive an expression for F_i.

$$f = \text{mass} \times \text{acceleration} = \frac{w}{g} \times \frac{V_1 - U}{dt}. \tag{1}$$

From Fig. 210,

$$F_i = f\frac{dt}{t} = \frac{w}{gt}(V_1 - U). \tag{2}$$

But $\frac{w}{t}$ = weight of projectiles hurled against the block in *pounds per second,* and $(V_1 - U)$ = *change of velocity* in the direction under consideration. Designating these two quantities by W and V, respectively, we get

$$F_i = \frac{WV}{g} \qquad (3)$$

which is independent of the *number* of individual particles involved, and is therefore applicable to the case of *fluid flow.*

200. Reaction vs. Impulse Force.*—An *impulse force* results whenever a jet is bent from its original course or is diminished in velocity. A *reaction force* appears whenever velocity is generated. The reaction force F_r, accompanying the acceleration of a fluid stream from zero to V_1 velocity, must necessarily equal the impulse force F_i that would result in decelerating the jet from V_1 to zero velocity.

Hence,

$$F_r = F_i = \frac{WV}{g} \qquad (4)$$

where W = weight of fluid flowing per second, V = total change of velocity in the direction under consideration, and $g = 32.2$.

As an illustration, when water is accelerated from zero to V_1 fps velocity in a nozzle, there is an accompanying reaction force F_r tending to move the vessel containing the nozzle in a direction opposite to that of the jet produced, as represented in Fig. 211. Should the jet thus formed be directed against a stationary flat plate as shown, so that every vestige of the original velocity is destroyed, an impulse force F_i will be exerted upon the plate. By Eq. (4), if $V_1 = 100$ fps and $W = 40$ lb per sec, then.

$$F_i = F_r = \frac{40}{32.2}(100 - 0) = 124.2 \text{ lb.}$$

Should the jet be directed tangentially into a semicircular cut in a block, as illustrated in Fig. 212, the stream will be turned

* The definition of impulse force in the Newtonian system of mechanics is in terms of the effect it produces on the rate of change of momentum with respect to time.

Let

m = mass;
v = velocity;
t = time.

Then

$$F = \frac{d(mv)}{dt}. \quad \text{[Compare Eq. (1)]}$$

through 180 deg in direction. If the block is stationary and frictionless flow is assumed, the leaving velocity V_2 will equal V_1, and the total change in velocity is $V_1 + V_2$. Again taking

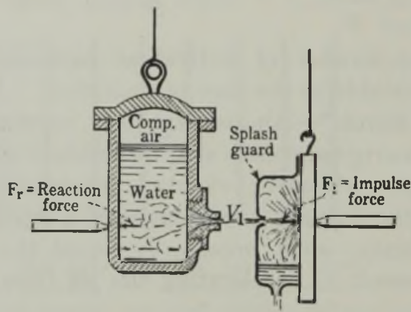


FIG. 211.

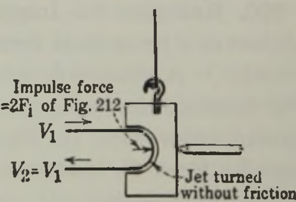


FIG. 212.

FIGS. 211 and 212.—Diagrams illustrating forces of impulse and reaction.

$V_1 = 100$ fps and $W = 40$ lb per sec, the impulse force upon the block of Fig. 212 is

$$F_i = \frac{40}{32.2}(100 + 100) = 248.4 \text{ lb.}$$

The case of a stationary bucket turning a jet through an angle of less than 180 deg is illustrated in Fig. 213, where the impulse

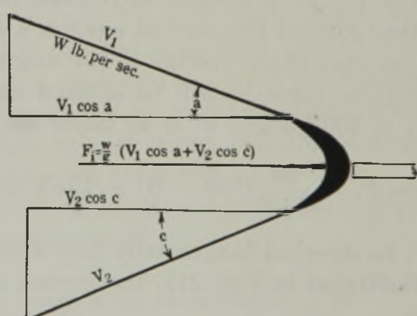


FIG. 213.—Diagram illustrating effect of bucket angles on force of impulse, in case of frictionless flow.

force F_i, *in the horizontal direction,* is determined by the total change of velocity in respect to that direction. The actual case of a turbine bucket in motion will be taken up later.

201. Impulse and Reaction Turbines.—Turbines are classified primarily as *impulse* or *reaction,* depending upon whether the force of impulse or that of reaction is employed to turn the rotor. A *simple impulse steam turbine* is illustrated in Fig. 214, where the motive force is clearly seen to be derived from the impulse of the jet or jets of steam acting upon the buckets of the wheel. The usual concept of the simplest form of *pure reaction turbine* is illustrated in Fig. 215. Here, the reaction force of the fluid,

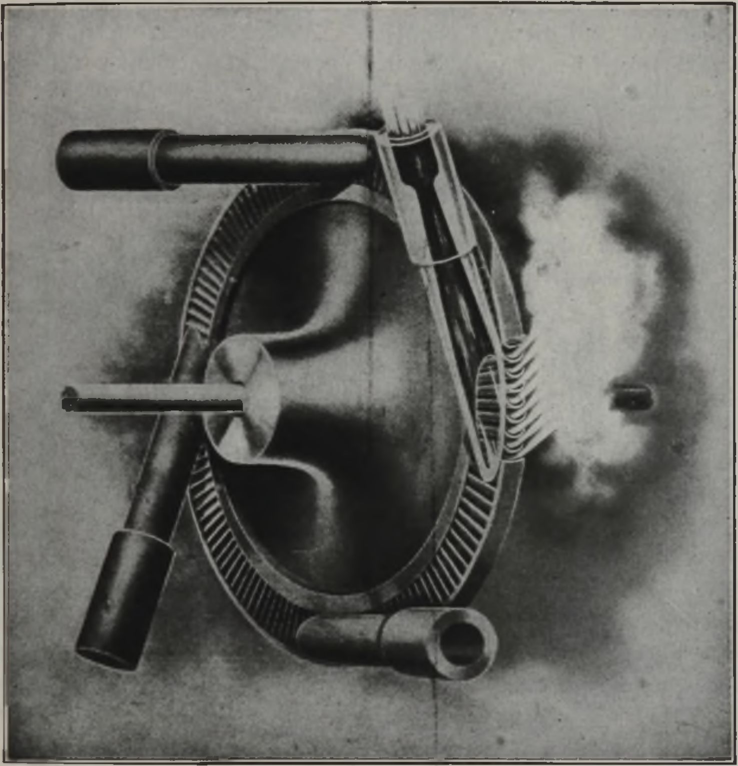


FIG. 214.—De Laval turbine wheel and nozzles.

as it is accelerated through the velocity generating element, acts directly to turn the rotor. Commercial turbines of the pure reaction type have never been developed, owing largely to the difficulty of transferring steam from a stationary steam chest into nozzles on the rim of the rotor where the force of reaction can be utilized. The reaction principle is, however, successfully employed in conjunction with impulse in the *Parsons turbine,* which will be described later (Art. 207).

202. Essential Elements of a Turbine.—There are two distinct functions to be performed in a steam turbine:

1. The transformation of available heat in the steam, or potential energy, into kinetic energy.

2. The utilization of steam velocity to produce turning force upon a rotor.

In the pure *impulse turbine*, the two functions are performed in separate and distinct elements. Available heat of the steam is first transformed into kinetic energy in stationary elements

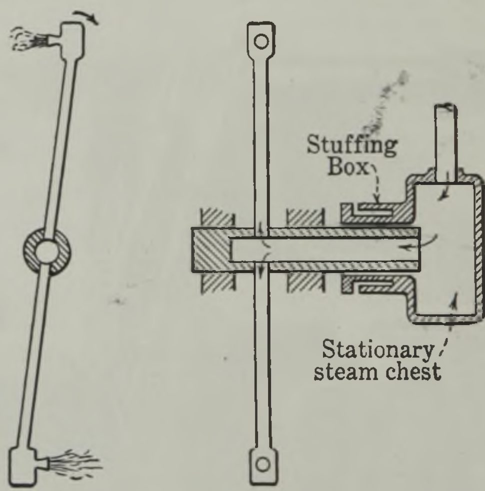


FIG. 215.—Simple concept of pure reaction turbine.

called *nozzles.* The resulting steam velocity is then utilized in *buckets* to produce an impulse turning force upon the rotor. Reaction force is necessarily created by the action of the nozzles, but the force is exerted upon stationary elements and hence performs no work.

In the pure *reaction turbine*, both functions of generation and utilization of velocity are performed in one and the same element, as illustrated plainly in Fig. 215.

203. Classification of Impulse Turbines.—Impulse turbines may be arranged so that the entire available heat of the steam between supply and exhaust pressures is transferred into kinetic energy:

1. In a single step or stage (*the single-pressure-stage type*).

2. In a succession of steps or stages (*the multipressure-stage type*).

The velocity developed in the single-pressure-stage turbine, or that developed in any step of a multipressure-stage turbine may be utilized:

1. In a single row of moving buckets (*the single-velocity-stage type*).

2. In two or more rows of moving buckets (*the multivelocity-stage type*).

204. The Simple Impulse Steam Turbine.—A diagrammatic sketch of a single-pressure-stage, single-velocity-stage turbine is shown in Fig. 216. Section *A* is on a plane passing through the turbine shaft; *B* is on a plane parallel to the shaft and tangent to the wheel at the pitch radius. Curves *I* and *II* show the variation of pressure and velocity respectively as the steam passes through the turbine. It should be noted that the velocity at the entrance to the nozzle is almost zero and the pressure a maximum. When the steam leaves the nozzle, the velocity is a maximum and the pressure a minimum.

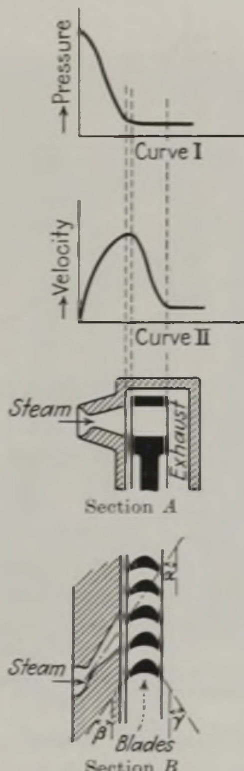


Fig. 216.—Variation of velocity and pressure in a De Laval turbine.

This is termed a *simple impulse* turbine because the steam expands from the supply pressure to the exhaust pressure in a single series of stationary nozzles (Fig. 214), and the velocity and direction of the steam are changed by a single row of moving buckets. Because of the expansion of the steam from initial to final pressure in a single series of nozzles, the velocity of the steam issuing is apt to be very high, in the order of 3,500 fps in some cases. For good economy the bucket velocity should be about one-half the steam velocity. However, the practical maximum for this bucket velocity is about 1,400 fps so that the residual velocity of the steam leaving the buckets is large. The kinetic energy of this steam leaving the buckets performs no useful work and so is wasted. For example, take the initial jet velocity as 1.00. If the residual velocity is 0.3, then the wasted energy which is proportional to the square of the velocity, is 9 per cent of the original kinetic energy of the jet.

The flow of steam in this turbine is parallel to the axis of the shaft and the machine is called an *axial flow turbine.*

The *single-pressure-stage, single-velocity-stage* principle is illustrated by Fig. 214. This arrangement is generally referred to as the *De Laval type* because it was first successfully developed by De Laval. It is characterized by very high rotative speed, the smaller machines running at 30,000 rpm. This speed is too high for the ordinary power utilizer and a speed reduction gear with a ratio of about 10 to 1 is used. Because of the high residual energy loss these turbines are built only in small sizes. Turbines embodying both the multipressure and multivelocity principles in various combinations are now manufactured by the De Laval Steam Turbine Company.

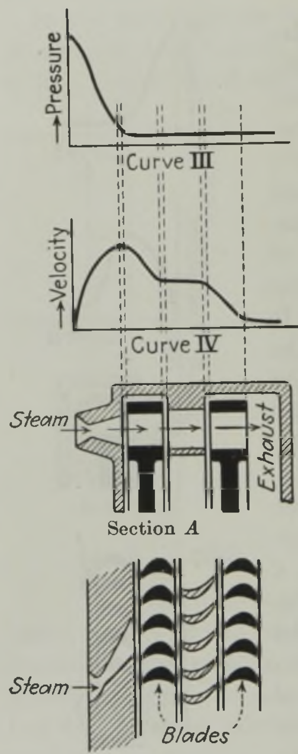


FIG. 217.—Variation of velocity and pressure in a single-stage Curtis turbine.

205. The Simple Velocity Staged Impulse Turbine.—Figure 217 is a diagrammatic sketch of the variation of velocity and pressure in this turbine which is of the *single-pressure-stage, multivelocity-stage* type. In the *simple velocity staged impulse* turbine the steam expands from the supply pressure to the exhaust pressure in one series of nozzles which results in a high velocity of the steam leaving the nozzles. This velocity energy is then utilized in two or more rows of moving buckets with stationary redirectioning buckets between the rows of moving buckets. It is to be noted that these redirectioning buckets are not nozzles in that there is no drop in pressure across them.

This too is an axial flow turbine and is called the Curtis type. The Curtis turbine is built also in the *multipressure stage, multi-*

velocity-stage type, in which case the steam expands in two or more series of nozzles, the velocity generated in each series being absorbed or utilized by two or more rows of moving buckets.

Another plan (besides the Curtis) for embodying the multi-velocity-stage principle, is represented in Fig. 218. Turbines operating upon this plan are variously referred to as of the *reversed-flow, reentry,* or *helical-flow* type. The nozzle A, quite small in bore in comparison with the width of the wheel, discharges steam with a high velocity into one side of the buckets B, which are cut as semicircular pockets in the rim of the wheel. The jet, turned through 180 deg, shoots out of the bucket still at high velocity, but less than that with which it entered, on account of the absorption of energy by the moving bucket, and of friction. The outshooting stream is caught in a stationary bucket (Fig. 219), turned through 180 deg, and redirected upon the single wheel again. At each turn of the steam in its corkscrew path, the velocity of the jet is reduced until its energy is finally spent,

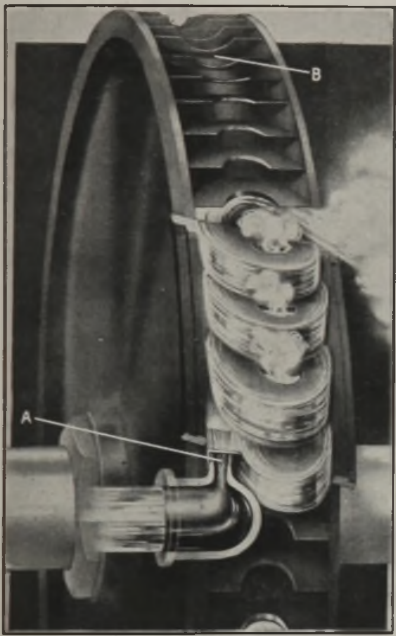


FIG. 218.—Terry reentry type of impulse turbine.

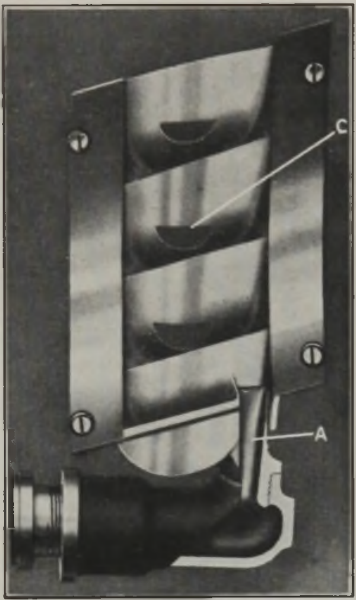


FIG. 219.—Stationary elements of Terry turbine.

and it finds its way out through the clearance space between wheel and stator, the holes C (Fig. 219) provided in the stationary buckets, to the exhaust. By this ingenious scheme, multiple velocity staging is accomplished with only one wheel. Turbines employing this plan are built in relatively small sizes, and usually operated noncondensing, and serve extensively for driving fans, blowers, pumps, exciters, and even belt-driven apparatus.

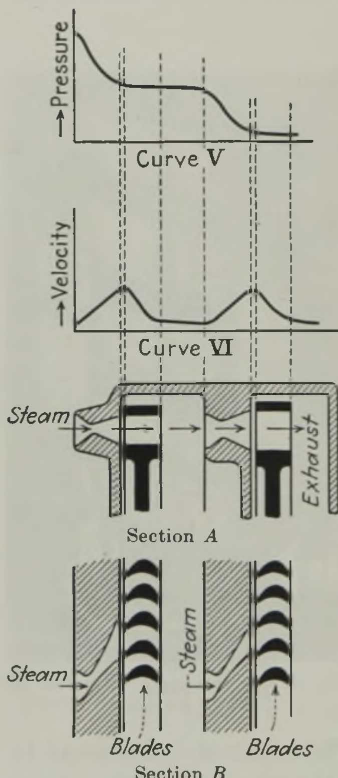


FIG. 220.—Variation of velocity and pressure in a two-stage Rateau turbine.

206. The Pressure Staged Impulse Turbine.—Figure 220 is a diagrammatic representation of a *multipressure-stage, single-velocity-stage* turbine, known for short as a *pressure staged* turbine. If the steam is expanded serially in several pressure steps or stages and the velocity resulting from each expansion is utilized in a moving row of buckets, the steam velocity at the exit of each nozzle will be less than for a single-pressure-stage turbine. Because of this the bucket velocity for maximum economy will be reduced. In addition, the residual velocity from each stage can be utilized in the succeeding stage. The residual energy from the last stage will, of course, still be wasted, but because of the lower bucket velocity this wasted energy will be a small percentage of the total available energy.

This turbine is essentially several simple impulse turbines in series and is known as a *Rateau* type. It may have from 2 or 3 to 20 or more stages and is one of the most efficient forms of prime mover, especially in the large sizes, but it is an expensive machine to build. Since there is a pressure difference between the entrance and exit of each nozzle, provision must be made to prevent steam by-passing these nozzles through leakage along the shaft. There is no tendency for steam to short-circuit the *moving buckets* as there is no pressure difference here.

207. Impulse-reaction Turbine.—In the impulse-reaction turbine a combination of impulse and reaction forces is used to produce the turning force on the rotor. The peculiar feature of this turbine in the employment of the force of reaction is the manner in which steam under pressure is transferred from a stationary compartment into rapidly moving nozzles on the rim of a rotor. Steam is literally thrown or tossed into the traveling nozzles, and must be given a velocity at least as great as that of the moving element. To do this, some of the steam's own available energy is used to accelerate the steam itself through stationary nozzles appropriately directed to "land" the steam in the entrances of the moving nozzles. With its remaining available energy, the steam then expands in the moving nozzles, and the reaction force produced thereby exerts the motive push upon the rotor.

If the initial velocity is just equal to that of the moving element, then this latter experiences no impulse effect whatever, and the rotor carrying the moving nozzles is acted upon by *reaction* force only, and a turbine operating in this manner would be a *pure reaction turbine.* In the *impulse-reaction* turbine, the velocity of the steam delivered by the stationary nozzles is greater than that of the moving nozzles; in receiving this stream, the latter experiences an *impulse force,* which, combined with the *reaction force* developed by further expansion, operates to turn the rotor. Figure 221 illustrates the change of velocity and pressure in an impulse-reaction type of turbine.

Turbines using the impulse-reaction principle of operation are always multiple staged.

In any multiple-stage turbine, steam progresses axially through the machine in a kind of annular passage composed of the alternate rings of stationary and moving elements. On account of the use of the reaction principle in a Parsons turbine, steam always has a higher pressure upon entering *any* row of blades than it has upon exit therefrom, and because of this pressure difference it will tend to force itself through every blade opening in the ring. Consequently, it is useless to try to confine the flow to a fraction of the periphery. In an impulse turbine, where the pressure drop all occurs in stationary nozzles, there is no pressure difference (axially) across a row of moving buckets. Steam shoots across the clearance space with only the energy represented by its inertia and has no tendency to spread. Therefore, in impulse turbines, the axial movement of the steam may be confined to any fraction of the periphery desired up to the full 360

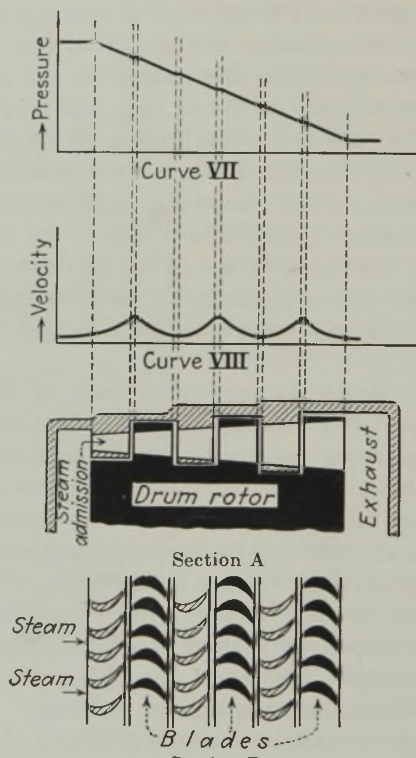


FIG. 221.—Variation of velocity and pressure in a Parsons turbine.

deg, and the size of the annulus which the buckets occupy adjusted in size to accommodate the steam flow, whereas in the reaction turbine full annular flow must be provided for.

At the high-pressure end of a Parsons turbine where the specific volume of steam is small, this requirement of full annular flow calls for a blade ring, or annulus, of relatively small area to accommodate the total volume. In order that the blades may not be unreasonably short, the peripheral length of the annulus must be small, which means small diameter, low energy-absorbing capacity per stage, and hence a larger number of stages in the high-pressure range of expansion. As the volume of the steam increases with expansion, the diameter of the rotor can be stepped up. In turbines of larger capacity, with greater total volumes of steam to be handled, the difficulties at the high-pressure end diminish. On the other hand, it becomes impractical to build

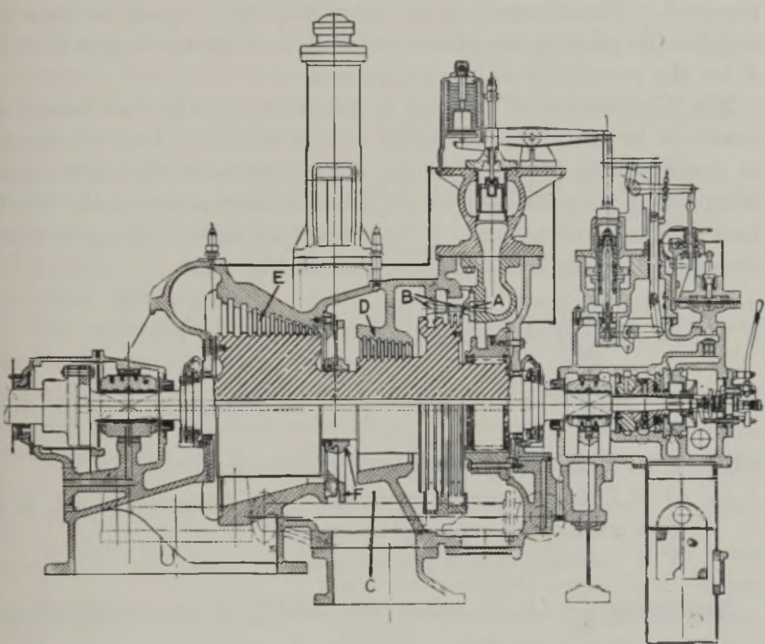


Fig. 222.—Westinghouse combination impulse and reaction turbine.

full impulse-reaction turbines in capacities of less than 300 or 400 kw, whereas impulse turbines of any small capacity may be built, because this latter type is not limited to the requirement of full annular flow.

The axial flow impulse-reaction turbine goes by the name of its inventor, Sir Charles A. Parsons. A radial flow adaptation of the same principle is known as the Ljungstrom type.

The Parsons turbine is one of the most efficient types, but is complicated and expensive. One construction difficulty encountered in this type is the necessity of preventing steam leakage around the *moving* blades as well as around the stationary nozzles.

The problem is especially difficult in the high-pressure end of the turbine where the pressure differences are of considerable magnitude.

Figure 222 illustrates a Parsons turbine in which expansion in the higher pressure range takes place in a two-row impulse wheel *B* of the Curtis type, which replaces the many stages of the Parsons type at the high-pressure end that would otherwise be required. The steam volume after this first expansion step is sufficiently great to be easily handled by Parsons stages *D* and *E* for the remainder of the expansion.

208. Generation of Velocity in the Nozzle.—The function of a nozzle is to transform as much of the available heat of steam as possible into kinetic energy. The process of transformation is represented by Eq. (5) which is but an expression of the First Law of Thermodynamics. The subscripts refer to the points in the steam's progress along the nozzle of Fig. 223.

$$E = \left(Jh_1 + \frac{V_1^2}{2g}\right) = \left(Jh_t + \frac{V_t^2}{2g}\right) = \left(Jh_x + \frac{V_x^2}{2g}\right)$$
$$= \left(Jh_e + \frac{V_e^2}{2g}\right), \quad (5)$$

where E = total energy per pound of steam;
J = 778 (foot-pound equivalent of 1 Btu);
h = *enthalpy* per pound of steam;
V = velocity in feet per second.

Neglecting $\frac{V_1^2}{2g}$ (V_1 is called the *velocity of approach*), which is usually small, the kinetic energy at the discharge end of the nozzle is, of course, equal to the drop in enthalpy of the steam. Then

$$Jh_1 = Jh_e + \frac{V_e^2}{2g}$$
$$\frac{V_e^2}{2g} = J(h_1 - h_e) \quad (6)$$

and

$$V_e = \sqrt{2gJ}\,\sqrt{h_1 - h_e} = 223.7\,\sqrt{h_1 - h_e}. \quad (7)$$

Example.—Let p_1 (Fig. 223) = 150 psia; initial moisture content = 1.5 per cent; and p_e = 15 psia. Assume the nozzle to be perfect, in which

case expansion from p_1 to p_e is adiabatic. From the Mollier diagram, $h_1 = 1{,}181.2$ Btu, and (for constant entropy) $h_e = 1{,}014.9$ Btu.
Whence

$$h_1 - h_e = 166.3 \text{ Btu}$$

and

$$V_e \text{ [from Eq. (7)]} = 2{,}879 \text{ fps}.$$

209. Form of Nozzle for Proper Expansion.—In order to approach ideal expansion, a nozzle must conform to certain requirements in its proportions. In a general way, the proper form of nozzle for an elastic substance (a gas) is one with a *convergent-divergent* contour in longitudinal section, such as represented by Fig. 214 or Fig. 223. Practically, it is found that the convergent part may be very short and usually consists of only a "well-rounded" entrance.

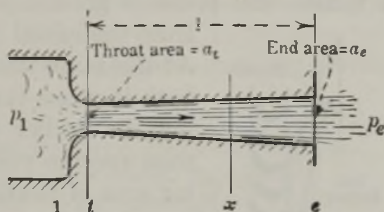


FIG. 223.—Practical form for steam nozzle when back pressure is considerably less than critical pressure.

The divergent part of a nozzle is necessary only under certain conditions which are determined by the relation of the back pressure p_e to the initial pressure p_1. When p_e is less than $0.58p_1$, then, theoretically, the nozzle should have a divergent part, as in Fig. 223, the amount of divergence, *i.e.*, the area ratios $\frac{a_e}{a_t}$ depending upon the relation of p_e to p_1 as will be explained presently. When p_e is greater than $0.58p_1$, the nozzle needs no divergent part and takes the form of Fig. 224, or more usually Fig. 225, which has a short passage of uniform area merely for the purpose of directing the jet. The expansion or energy transformation, however, takes place entirely in the convergent part. Practically it has been shown by experiment that the simple *straight or nondivergent* nozzle of Fig. 225 may be used with advantage for all back pressures above about $0.35p_1$. For values of p_e less than $0.35p_1$, a divergent passage must be used to give proper expansion, and the area ratio $\frac{a_e}{a_t}$ (Fig. 223) must be different for each

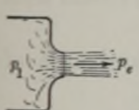


FIG. 224.

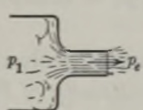


FIG. 225.

FIGS. 224 and 225.—Non-expanding nozzles. Appropriate for steam, theoretically when $p_e \gtreqless 0.58p_1$, and practically when $p_e \gtreqless 0.35\,p_1$.

different ratio $\frac{p_e}{p_1}$. The straight nozzle is the correct form of *any* back pressure above $0.35p_1$.

For side delivery upon a turbine wheel, the nozzle takes the form shown in Fig. 226. The end section of the nozzle is the last section in which the steam is completely confined within the nozzle passage, and expansion should be completed by the time it has reached this point.

210. The Critical Pressure of Steam.—As steam flows through a nozzle, the pressure gradually (though very quickly) drops from the initial value p_1 to the back pressure p_e. At the same time, specific volume and velocity both increase throughout the expansion. During the earlier part, velocity increases faster than specific volume, which calls for a decreasing area of nozzle section; during the later part, specific volume increases faster than velocity, which calls for an increasing area. The pressure, in the course of expansion, at which this peculiar reversal takes place, is called the *critical pressure,* and it occurs at the *throat,* or place of smallest cross section, provided the back pressure is lower than this critical value itself.

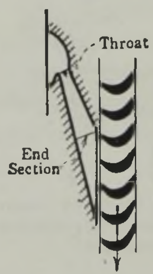


FIG. 226.—Nozzle for turbine wheel.

The critical pressure p_c bears a definite ratio to the initial pressure p_1. For steam initially wet, $p_c = 0.58p_1$; for steam with initial superheat, $p_c = 0.54p_1$.

The relation between velocity, specific volume, and area of nozzle is given by the equation of *Continuity of Flow,*

$$W = \frac{A_1V_1}{v_1} = \frac{A_tV_t}{v_t} = \frac{A_xV_x}{v_x} = \cdots \frac{A_eV_e}{v_e}, \tag{8}$$

where W = weight of steam flowing per second—constant for all sections;

A = area of sections designated by subscripts in square feet;

V = velocity at the designated sections in feet per second;

v = specific volume at the various sections in cubic feet.

Subscripts refer to sections designated in Fig. 223.

Equation (8) may be written

$$W = A_1\left(\frac{V_1}{v_1}\right) = A_t\left(\frac{V_t}{v_t}\right) = A_x\left(\frac{V_x}{v_x}\right) = \cdots A_e\left(\frac{V_e}{v_e}\right) \quad (9)$$

which differentiates A on the one hand, which is a dimensional quantity of the nozzle, from V and v on the other, which are properties of the steam.

The variation in value of the interesting ratio $\frac{V}{v}$ for a particular case is shown in column 8 of Table XXX. The pressures of column 1 are chosen as desired. Thus for steam expanding in a nozzle from 150 psia (1.5 per cent moisture), to 15 psia, the pressure must be, say, 60 lb at *some* point along the nozzle.

TABLE XXX.—VELOCITIES, SPECIFIC VOLUMES, AND AREAS CORRESPONDING TO CHOSEN PRESSURES BETWEEN ENTRANCE AND DISCHARGE END OF A NOZZLE, FOR CONSTANT ENTROPY EXPANSION
(h_1 = enthalpy of steam at 150 psia, 1.5 per cent moisture = 1,181.2)

1	2	3	4	5	6	7	8	9
Pressure, psia (chosen), p	Enthalpy after adiabatic expansion, h	Available heat Btu, $h_1 - h$	Velocity, fps, V	Per cent quality after adiabatic expansion (Diagram)	Specific volume of dry steam (Tables)	Actual specific volume (5) × (6), v	Ratio, $\frac{V}{v}$	Area of nozzle, sq. in. W = 1 lb per sec, a
150	1,181.2	0.0	0	98.5	3.010	2.96	0.0	
135	1,172.0	8.6	655	97.7	3.329	3.25	201.5	0.714
120	1,161.9	18.7	965	96.9	3.725	3.61	267.3	0.538
105	1,151.7	28.9	1,205	96.0	4.226	4.06	296.3	0.486
90	1,139.8	40.8	1,430	95.0	4.892	4.65	307.5	0.469
75	1,125.7	54.9	1,660	93.9	5.813	5.46	304.0	0.474
60	1,109.2	71.4	1,890	92.6	7.172	6.64	284.6	0.506
45	1,088.3	92.3	2,150	91.0	9.399	8.55	251.5	0.573
30	1,060.2	120.4	2,450	89.0	13.745	12.23	200.3	0.717
15	1,014.9	165.7	2,880	86.0	26.31	22.63	127.3	1.132

Assuming adiabatic expansion, the enthalpy and moisture content of steam at that point can be ascertained by use of the Mollier diagram, whence available heat, velocity, and specific volume can be readily determined. The curve of Fig. 227 shows that the maximum value of $\frac{V}{v}$ occurs at a pressure of about 87 psia, which is 58 per cent of 150 lb.

211. Rate of Steam Flow through Nozzles.—The rate of steam flow through a nozzle is given by the general expression

$$W = \frac{A_t V_t}{v_t}, \tag{10}$$

where W = pounds per second;
A_t = area of throat section in square feet;
V_t = velocity at throat in feet per second;
v_t = specific volume at throat in cubic feet.

If the back pressure p_e is less than the critical pressure, the throat pressure p_t is a direct function of p_1, and, through p_t, V_t and v_t [of Eq. (10)] are directly dependent upon p_1. From

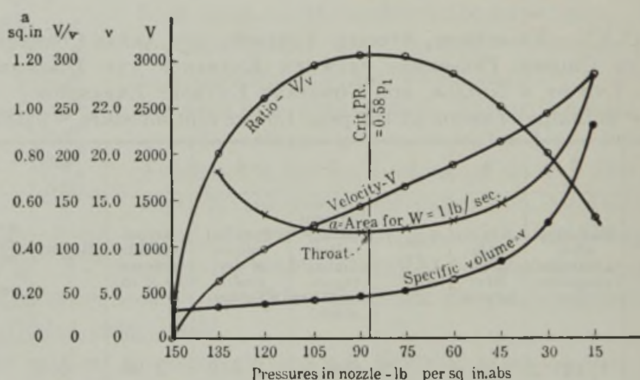


FIG. 227.—Curves showing relation of velocity, specific volume, and area of section to pressure, for steam flowing through a nozzle.

this relationship, Eqs. (11) and (12) can be derived by processes too complex for this text.

For steam initially saturated,

$$W = 0.3002 a_t \sqrt{\frac{p_1}{v_1}}. \tag{11}$$

For steam initially superheated,

$$W = 0.3155 a_t \sqrt{\frac{p_1}{v_1}}, \tag{12}$$

where W = pounds per second;
a_t = area of *throat* of nozzle in *square inches;*
p_1 = initial pressure in *pounds per square inch absolute;*
v_1 = initial specific volume, from steam tables.

Also, for conditions of back pressure less than the critical, Napier's simple empirical rule is very easy to apply and gives fairly good results.

$$W = \frac{a_t p_1}{70} f. \tag{13}$$

The symbols W, a_t, and p_1 are defined as above. f is a corrective factor to be applied when the *steam supplied* is wet or superheated. For wet steam, $f = \frac{1}{1 - 0.012m}$, where m is the percentage of moisture. For superheated steam, $f = \frac{1}{1 + 0.00065D}$, where D = degrees (F) superheat.

When the back pressure p_e is greater than the critical pressure, the throat pressure p_t is equal to p_e, and the velocity can be determined by the procedure of Art. 208. This value, together with the specific volume obtained from the steam tables, substituted in Eq. (10) will enable W to be calculated.

212. Nozzle Calculations.—The calculations necessary to determine a nozzle of nearly correct proportions are relatively simple. The quantities usually specified, or known to begin with, are initial pressure p_1, initial state (moisture or superheat), final or exhaust (back) pressure p_e, and weight of steam per hour. The quantities to be determined are area at throat, a_t sq in.; area at the discharge end section, a_e; and length l from throat to discharge end (Fig. 223).

The convergent part of the nozzle takes care of itself, with merely a good fillet or rounded entrance (Fig. 214). Whether the nozzle is circular, square, or rectangular in section, it has been found experimentally that a divergent part with straight-line elements joining throat and end sections is as efficient as any more complex form, and of course much simpler to make. The nozzle will take the form of Fig. 223, if p_e is less than $0.35p_1$, and that of Fig. 225 in case p_e is greater than $0.35p_1$.

In nozzles of the divergent form, the length l from throat to end is made such that the total angle of divergence between nozzle walls will not exceed about 12 deg. In *straight nozzles*, the length of the straight section may be calculated by the rule $l = \sqrt{15a_t}$, where l = length of nozzle from throat to end in inches, and a_t = throat area in square inches.

Example.—Let $p_1 = 150$ psia, initial moisture content = 1.5 per cent, and back pressure $p_e = 15$ psia. The flow is to be 1,800 lb per hr, or 0.5 ($= W$) lb per sec. Determine (*a*) the area at the nozzle at the throat; (*b*) the area at the discharge end section; and (*c*) the length from the throat to the discharge end.

Solution.—Using Napier's simple formula [Art. 211, Eq. (13)],

$$a_t \text{ (area at throat)} = \frac{70 \times 0.5}{150} \times \frac{(1 - 0.012 \times 1.5)}{1} = 0.229 \text{ sq in.}$$

The end area a_e is related to the throat area by Eq. (8)

$$\frac{a_t V_t}{v_t} = \frac{a_e V_e}{v_e}.$$

The pressure at the throat is $p_t = 0.58 \times 150 = 87$ psia. The enthalpy of steam at 150 psia and 1.5 per cent moisture = 1,181.2 Btu, and at 87 lb after adiabatic expansion (Mollier diagram or calculation) = 1,137.2 Btu = h_t, with a moisture content of 5.2 per cent. From Eq. (7), the velocity at the throat is $V_t = 223.7\sqrt{1{,}181.2 - 1{,}137.2} = 1{,}475$ fps. The specific volume is $v_t = 5.055$ (dry steam at 87 lb) $\times$ 0.948 = 4.79 cu ft.

At the discharge end of the nozzle, the enthalpy after adiabatic expansion to 15 psia is found to be 1,014.9 Btu, whence

$$\text{the available heat} = 1{,}181.2 - 1{,}014.9 = 166.3 \text{ Btu.}$$

Assume that the nozzle turns only 95 per cent of the available heat into kinetic energy, in which case $V_e = 223.7\sqrt{0.95 \times 166.3} = 2{,}805$ fps. The theoretical enthalpy at the end of the expansion, 1,014.9 Btu, will be enriched by the 5 per cent loss of available heat ($0.05 \times 166.3 = 8.3$ Btu), so that the steam actually leaves the nozzle at a pressure of 15 psia and with an enthalpy of 1,023.2 Btu, at which state the specific volume v_e is found to be 22.86 cu ft.

We now have all the necessary factors to introduce into Eq. (8).

$$\frac{(0.229 \times 1{,}475)}{4.79} = \frac{(a_e \times 2{,}805)}{22.86},$$

whence $a_e = 0.575$ sq in. and ratio $\frac{a_e}{a_t} = 2.51$. If the nozzle is made circular in section, it is easily shown that the length l from throat to end must be 1.5 in. in order that the total divergence shall not exceed 12 deg.

213. Supersaturation.—Steam-turbine nozzles are short in length (only a few inches at most), and the time required for expansion to take place, with all its accompanying changes, is of the order of $\frac{1}{5{,}000}$ sec. Under conditions of more leisurely expansion, as in an engine cylinder, steam if initially dry and saturated, as at C in Fig. 228, will *begin to condense* as soon as expansion starts along the line *CG*. If initially superheated as at *D*, the steam expands from *D* to *H* as a superheated vapor, and condensation will occur during the remainder of the expansion *HF*.

In the extremely rapid expansion of steam in a nozzle, there is not sufficient time for condensation to take place as it normally should, and as a consequence it "overruns" point *H* and continues to behave as a dry or superheated vapor throughout a part or all of the remainder of its expansion range. It seems probable that steam is always supersaturated at the throat of a nozzle. In the supersaturated state, steam is in a condition of unstable

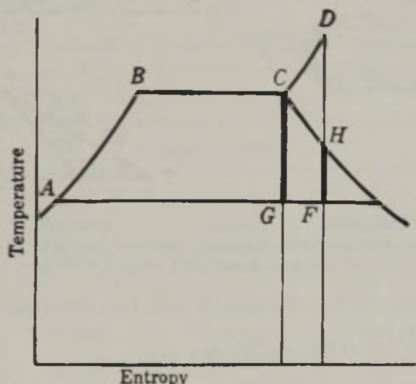


Fig. 228.—Supersaturation may occur in nozzles in expansion ranges such as *CG* and *HF*.

thermal equilibrium, and its temperature may be far below that corresponding to its pressure as given in the steam tables.

Supersaturation introduces an element of irreversibility (entirely aside from friction) into the expansion of steam and is therefore a cause of loss. This is one reason why superheated steam is universally used in large turbines. The loss occasioned by supersaturation is avoided if the entire working range of steam from admission to exhaust is kept in the superheat region, either by the use of very high initial superheat or by reheating (see Art. 220, Fig. 259).

214. Utilization of Velocity in the Buckets.—In order to utilize kinetic energy effectively, a bucket must travel at some optimum speed in relation to the jet speed. In Fig. 229, a nozzle stream of V_1 = 2,000 fps *absolute velocity* acts upon a 180-deg frictionless bucket traveling at a speed of $U = 500$ fps. The *relative velocity* of the jet as it enters the bucket is then $R_1 = 1{,}500$ fps, and the relative velocity leaving (R_2), is also 1,500 fps, since frictionless flow is specified. The jet jumps off the bucket with an absolute velocity $V_2 = 1{,}500 - 500 = 1{,}000$ fps. The bucket has operated to reduce the velocity of the steam from 2,000 to 1,000 fps, and has extracted $\frac{(2{,}000^2)}{2g} - \frac{(1{,}000^2)}{2g}$ ft-lb of energy

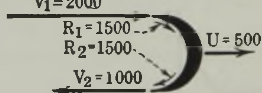


Fig. 229.

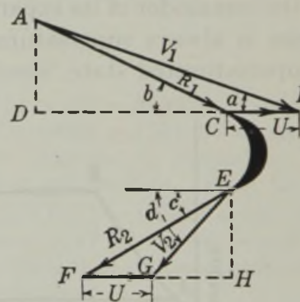


Fig. 230.

Figs. 229 and 230.—Elementary impulse turbine buckets illustrating relation between bucket speed and steam velocity.

per lb of steam. The efficiency of the bucket is, therefore, $\frac{(2{,}000^2 - 1{,}000^2)}{2{,}000^2} = 0.75$, or 75 per cent.

The efficiency may also be arrived at by determining the kinetic force. Using Eq. (4),

$$F_i = \left(\frac{W}{g}\right)(V_1 + V_2), \qquad \text{or} \qquad F_i = \left(\frac{W}{g}\right)(R_1 + R_2), \quad (14)$$

whence the force per pound of steam flowing per second is

$$F = \frac{1}{32.2}(2{,}000 + 1{,}000) \qquad \text{or} \qquad \frac{1}{32.2}(1{,}500 + 1{,}500)$$
$$= 93.17 \text{ lb.}$$

The *work done* on the rotor per pound of steam is

$$F \times U = 93.17 \times 500 = 46{,}585 \text{ ft-lb},$$

and the *kinetic energy* supplied in the jet per pound of steam is

$$\frac{2{,}000^2}{2g} = 62{,}110 \text{ ft-lb.}$$

The *efficiency* is

$$\frac{46,585}{62,110} = 0.75 \text{ or } 75 \text{ per cent as before.}$$

Other values of U may be chosen, and similar calculations made, as in Table XXXI, where it will be observed that the kinetic force decreases with bucket speed as a straight-line function, while the efficiency rises to a maximum at the value $U = \frac{1}{2}V_1$, and decreases to zero at "runaway" speed.

The case of steam discharged upon a bucket wheel at an angle (the usual case) is illustrated in Fig. 230. Here the component of V_1 in the direction of U is $BD = V_1 \cos a$. Subtracting U

TABLE XXXI.—CALCULATED VALUES OF RELATIVE VELOCITY, RESIDUAL ABSOLUTE VELOCITY, KINETIC FORCE, AND EFFICIENCY FOR DIFFERENT BUCKET SPEEDS

(For one pound of steam per second, V_1 = 2,000 fps, and a 180-deg frictionless bucket)

Bucket speed (chosen), U	Relative velocity of steam, $R_1 = R_2$	Residual absolute velocity, V_2	Kinetic force on bucket, F	Bucket efficiency, per cent
0	2,000	2,000	124.2	0
200	1,800	1,600	111.8	36
400	1,600	1,200	99.4	64
600	1,400	800	87.0	84
800	1,200	400	74.5	96
1,000	1,000	0	62.1	100
1,200	800	− 400	49.7	96
1,400	600	− 800	37.3	84
1,600	400	−1,200	24.8	64
1,800	200	−1,600	12.4	36
2,000	0	−2,000	0	0

from BD gives DC, the effective component of relative velocity R_1. AD, the perpendicular or ineffective component of V_1, is also that of the relative velocity R_1. Hence the value of R_1 (vector AC) can be readily determined. The angle b (= ACD) is the correct "entrance angle" for the bucket. Assuming an effect for friction, the magnitude of R_2 is something less than R_1 and is represented by the vector EF, whose angle c is the specified "exit angle" of buckets. Subtracting U from FH, the effective component of FE, leaves GH as the effective component of the absolute leaving velocity V_2 of the steam. With EH and GH (the vertical and horizontal components) known, the magnitude and direction of V_2 can be readily determined.

The efficiency of the bucket may be considered from two different angles:

$$(a)\ \text{Efficiency} = \frac{\text{energy supplied} - \text{total losses}}{\text{energy supplied}}. \tag{15}$$

$$(b)\ \text{Efficiency} = \frac{\text{useful work}}{\text{energy supplied}}. \tag{16}$$

The two expressions really mean the same thing but suggest two different ways of going about a solution. Applying expression (a) to the case represented by Fig. 230,

Energy supplied $= \dfrac{V_1^2}{2g}$ per pound of steam.

Total losses = friction loss + residual velocity loss

$$= \left(\frac{R_1^2}{2g} - \frac{R_2^2}{2g}\right) + \frac{V_2^2}{2g}.$$

or

$$\text{Efficiency} = \frac{\dfrac{V_1^2}{2g} - \left[\dfrac{V_2^2}{2g} + \left(\dfrac{R_1^2}{2g} - \dfrac{R_2^2}{2g}\right)\right]}{\dfrac{V_1^2}{2g}}$$

$$= \frac{V_1^2 - V_2^2 - (R_1^2 - R_2^2)}{V_1^2} \tag{17}$$

Applying Eq. (b), energy supplied $= \dfrac{V_1^2}{2g}$ as before, and useful work = force × speed. The speed of the bucket in Fig. 230 is U fps and the force acting in the direction of its motion per pound of steam flowing per second is given by either of the expressions,

$$F = \frac{1}{g}(V_1 \cos a + V_2 \cos d), \quad \text{or} \quad F = \frac{1}{g}(R_1 \cos b + R_2 \cos c). \tag{18}$$

The turning force calculated in the manner indicated above is that *applied at the buckets* on the rim of a wheel, and is not the net turning force on the shaft of the turbine because of intervening windage loss. Likewise the efficiencies of Table XXXI are *bucket efficiencies* and not those of the turbine. On account of rotation losses (windage, and bearing and gland friction) the maximum *shaft efficiency*, for a single-row turbine wheel, occurs at a bucket speed of about 45 per cent of the jet speed.

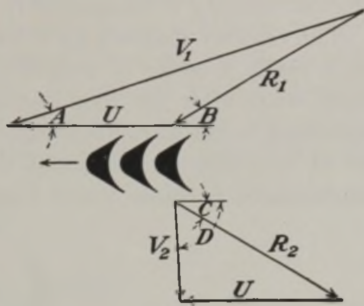


Fig. 231.—Velocity diagram for De Laval turbine blading.

215. Velocity Diagrams.—Figure 231 is a velocity diagram for a simple impulse or De Laval turbine. The steam velocity is approximately twice the bucket velocity.

Figure 232 is a diagram for a simple velocity staged or Curtis turbine. The determination of impulse forces on the moving

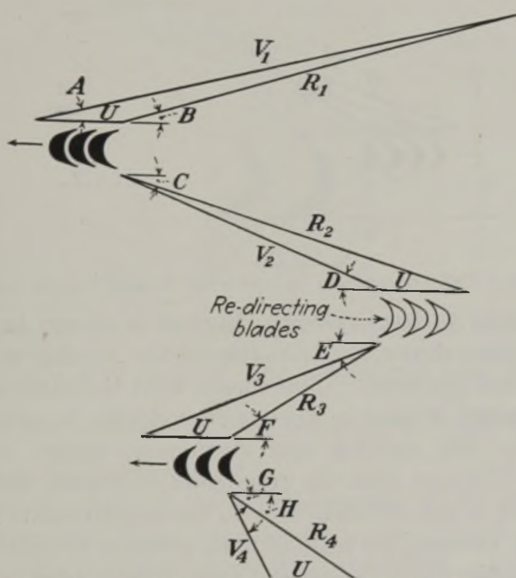


Fig. 232.—Velocity diagram for two-row Curtis turbine blading.

rows of buckets is made in the same manner as for the simple impulse turbine. Since the kinetic energy of a jet is proportional to the velocity squared, the energy-absorbing capacity of a two-row Curtis turbine is four times the capacity of a De Laval turbine with the *same bucket velocity.* It is to be noted that the initial steam velocity V_1 (Fig. 232) is more than four times the bucket velocity U because of friction.

The velocity diagram for a Rateau turbine is shown in Fig. 233. It is really a series of simple impulse turbines. The nozzles of each stage after the first one have their entrance angles corresponding to the absolute steam velocity from the preceding stage so as to utilize this residual velocity. This is the case for all multistaged turbines where high efficiency is desired.

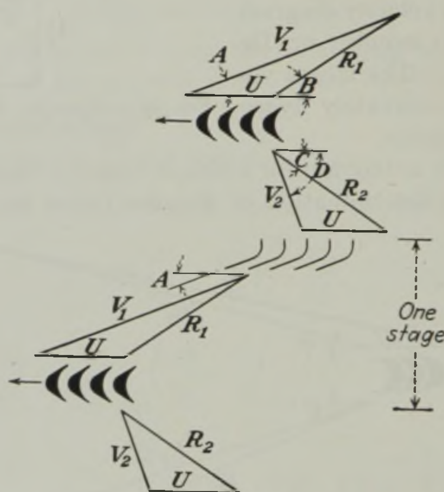


Fig. 233.—Velocity diagram for two-stage Rateau turbine blading.

The Parsons turbine velocity diagram is shown in Fig. 234. As this diagram shows, the triangles for the moving and stationary blades are the same. This means that the same amount of available energy is used in generating velocity in each element; consequently the moving and stationary blades are almost identical. The fact that the passageway between the blades is nondivergent is not contradictory to the requirements for proper nozzle form, because the ratio of back pressure to initial pressure is always higher than the critical ratio 0.58, owing to the large number of stages.

A *stage* of a Parsons turbine consists of a row of moving blades and a row of stationary blades. The velocity triangles of a Parsons stage "X," consisting of the rows B and C, are shown in Fig. 234. In order to represent the state of the steam as it enters the stage under consideration, attention must be given to the row of moving blades A, just preceding. The relative velocity of steam entering blades A is R_1. Within the blades, the pressure drops from p_1 to p_2, releasing some available energy, which increases the relative velocity from R_1 to R_2, which, combined with the blade velocity U_1, results in the absolute residual velocity V_2 representing the kinetic energy that blades A failed to make use of. The steam, therefore, approaches the

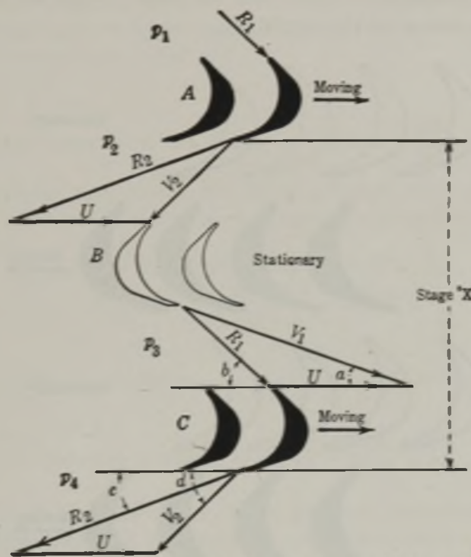


FIG. 234.—Velocity diagram for Parsons turbine blading.

stationary row B of stage X with a potential heat energy represented by the pressure p_2, and a legacy of $\frac{V_2^2}{2g}$ ft-lb of kinetic energy from the preceding stage.

In order to conserve the kinetic energy of V_2, the blades B have an entrance angle appropriate to the direction of V_2, so that the steam is guided into them without loss. In this respect, the stationary blades are like the stationary buckets of a Curtis stage. Within the passages of blades B, the steam is accelerated from V_2 to V_1 fps by the further expansion of steam from pressure p_2 to p_3, the blade passages functioning as nozzles in this respect. The absolute velocity V_1, combined vectorially with the blade speed U of moving row C, results in the stream entering the blades with the relative velocity R_1. In this respect, these moving blades of the Parsons turbine are like those of an impulse turbine, receiving steam with a velocity generated in a stationary element. However, in C, as in A, there is a further expansion from p_3 to p_4, and an acceleration of relative velocity from R_1 to R_2, so that the blade passages of the moving row C partake of the functions of nozzles, and sustain a reaction force therefrom which cooperates with the impulse force at entrance, to help turn the rotor of the turbine.

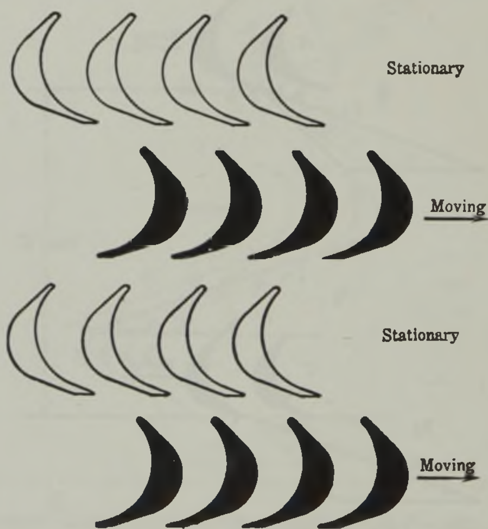


FIG. 235.—Parsons blades.

It will thus be seen that the action of the steam in both the stationary and moving rows of the stage is exactly alike. If the blade forms of the two rows are alike, it can be shown that the velocity triangles are also alike, and the velocities are therefore the same within the passages of both rows. As a matter of fact the stationary and moving blades of the Parsons turbine are usually identical in form, as illustrated by Fig. 235. The fact that the channel passage between blades is nondivergent in form is not contradictory to the requirements of a proper nozzle form, because the ratio of back pressure to initial pressure is always higher than the critical ratio of 0.58, owing to the large number of stages.

216. Energy Distribution.—The number of stages that a multipressure turbine should contain must be such that the energy-absorbing capacity of the rotor is equal to the available energy of the steam. Energy-absorbing capacity is determined by aggregate wheel speed of the rotor, which depends in turn upon diameter, turning speed, and number and kind of stages.

Example.—Assume the available energy of steam working between limits specified to be 400 Btu per lb. A turbine of the Rateau type is to be considered in which the pitch diameter of the wheels of all stages is to be 3.5 ft and the turning speed is 3,600 rpm. Approximately, how many pressure stages should the turbine have?

Solution.

$$\text{Peripheral speed of wheel} = 3.5 \times 3.1416 \times \frac{3{,}600}{60} = 660 \text{ fps.}$$

The steam velocity generated in the nozzles of each stage should be about twice this or 1,320 fps. Assuming the nozzles to be 95 per cent efficient,

$$1{,}320 = 223.7 \sqrt{AEs \times 0.95} \quad \text{(Eq. 7).}$$

$$0.95AEs = \left(\frac{1{,}320}{223.7}\right)^2 = 34.81.$$

Whence

$$AEs \text{ (available energy per stage)} = 36.6 \text{ Btu,}$$

and

$$\text{Number of stages required} = \frac{400}{36.6} = 11.$$

If the first stage is to be of the Curtis type and the others Rateau, with 3.5 ft pitch diameter for all stages, the steam velocity to be generated in the nozzles of the Curtis stage must be *four times* the wheel speed, or 2,640 fps, which, with a nozzle efficiency of 95 per cent, calls for 146.5 Btu available energy. The remaining energy to be absorbed in the Rateau stages is $400 - 146.5 = 253.5$ Btu. As before, each Rateau stage absorbs 36.6 Btu, whence the number of such stages is $\frac{253.5}{36.6} = 7$. The turbine will, therefore, consist of one Curtis and seven Rateau stages.

In case it is desired to have a rotor with stages in groups of different diameter, or with continuously varying diameter, the approximate number of stages can readily be determined by the principles illustrated above, once the plan of diameter variation is decided upon.

The allotment of available heat, or quota per stage having been determined, it is necessary to know *how* such division of energy is realized in the actual turbine. This is accomplished entirely by making the aggregate nozzle area in the several diaphragms, or partition plates J, K, etc., of Fig. 236, appropriate to the head or pressure drop that will result in the release of the predetermined quota of available heat. A diaphragm acts as a dam or obstruction to the flow of steam and its restrictive effect or resistance may be made great or small as desired by making the aggregate area of nozzles in the diaphragm small or large, and thus causing the head or pressure drop to be large or small, as desired.

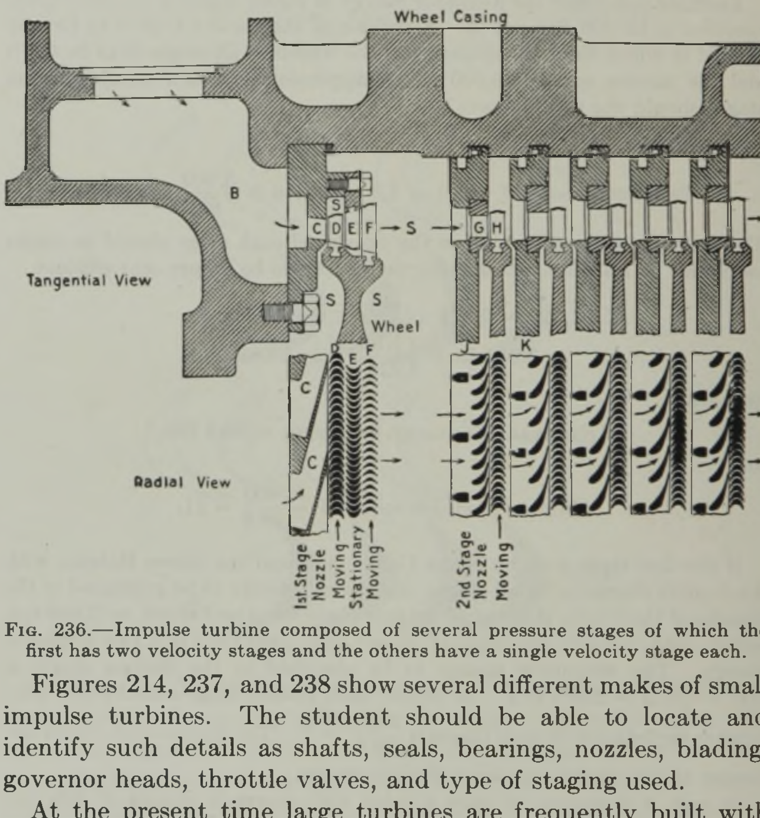


FIG. 236.—Impulse turbine composed of several pressure stages of which the first has two velocity stages and the others have a single velocity stage each.

Figures 214, 237, and 238 show several different makes of small impulse turbines. The student should be able to locate and identify such details as shafts, seals, bearings, nozzles, blading, governor heads, throttle valves, and type of staging used.

At the present time large turbines are frequently built with more than one type of staging. Thus in Figs. 239, 240, and 241 are seen the following combinations: Curtis-Rateau and Curtis-Parsons. The almost universal use of the Curtis first stage is made because it limits the portion of the turbine subjected to initial high-pressure and high-temperature steam to the manifolds, valves, and first-stage nozzles, and thus the remainder of the casing is lighter and less expensive to build.

A turbine employing both the Curtis and the Rateau principles is illustrated in Fig. 236. The velocity developed in the first-stage nozzles C is utilized in two rows of moving buckets D and F. The buckets D absorb about one-half the velocity and three-quarters of the energy, and the steam escaping from them is turned by *stationary* buckets E, and directed into the second row of moving buckets F, which absorbs the remainder of the kinetic

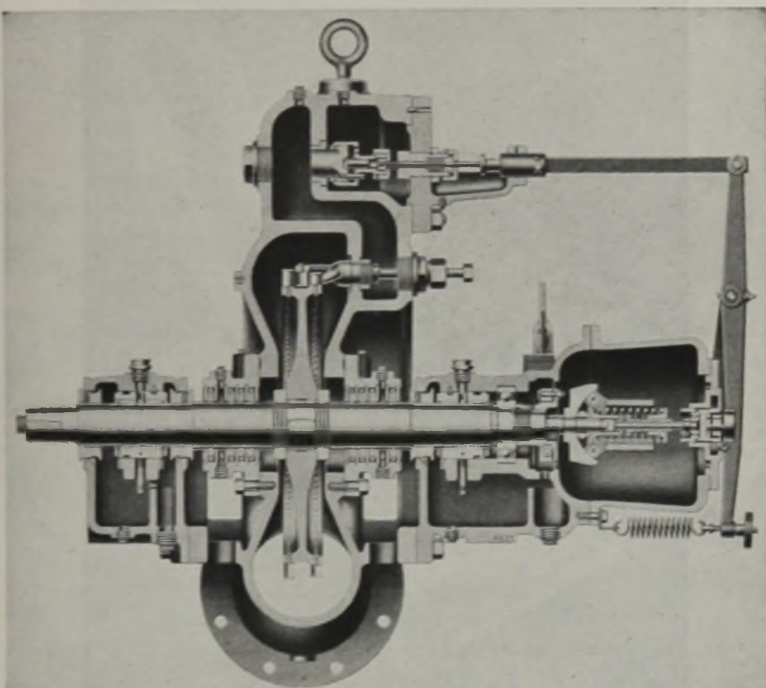


Fig. 237.—Axial section of De Laval velocity stage turbine with vertical governor and oil pump.

energy. The steam then passes on to the other section of the turbine which consists of *several pressure stages* with but *one row of buckets each*, or *one velocity stage per pressure stage*. Each set of nozzles has an allotment of so much available heat, which when transformed into velocity results in a jet speed approximately twice the bucket speed, so that the kinetic energy is absorbed by the single row of buckets.

The multivelocity absorption of energy of the nozzle discharge, as exemplified in the first stage of Fig. 236, is commonly known as the *Curtis principle;* the following stages embody the *Rateau principle.* A Rateau turbine, or section of a turbine, consists essentially of a number of De Laval type turbines in series.

In the turbine of Fig. 236, the wheels of all the stages shown are of the same diameter, so that bucket speed is necessarily equal in all stages. The steam velocity generated in Rateau-

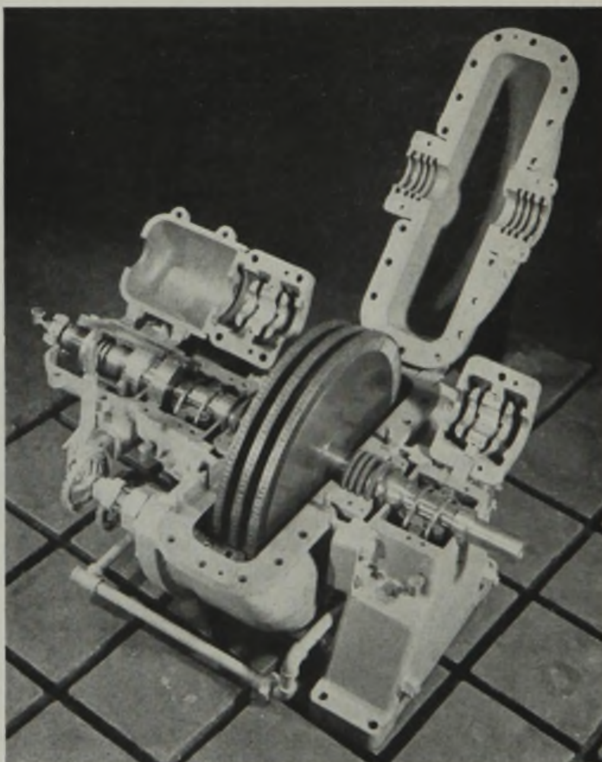


FIG. 238.—Elliott CY Curtis type turbine for mechanical drive.

stage nozzles is therefore only about one-half that generated in the Curtis-stage nozzles, C, because there are two velocity stages per nozzle in the Curtis and only one in the Rateau; consequently the available heat consumed in a Rateau stage is only one-fourth of that consumed in a Curtis two-wheel stage of the same diameter. In other words, it requires theoretically four Rateau stages in the turbine of Fig. 236 to develop the same amount of work as the one Curtis stage. Curtis stages of three rows of

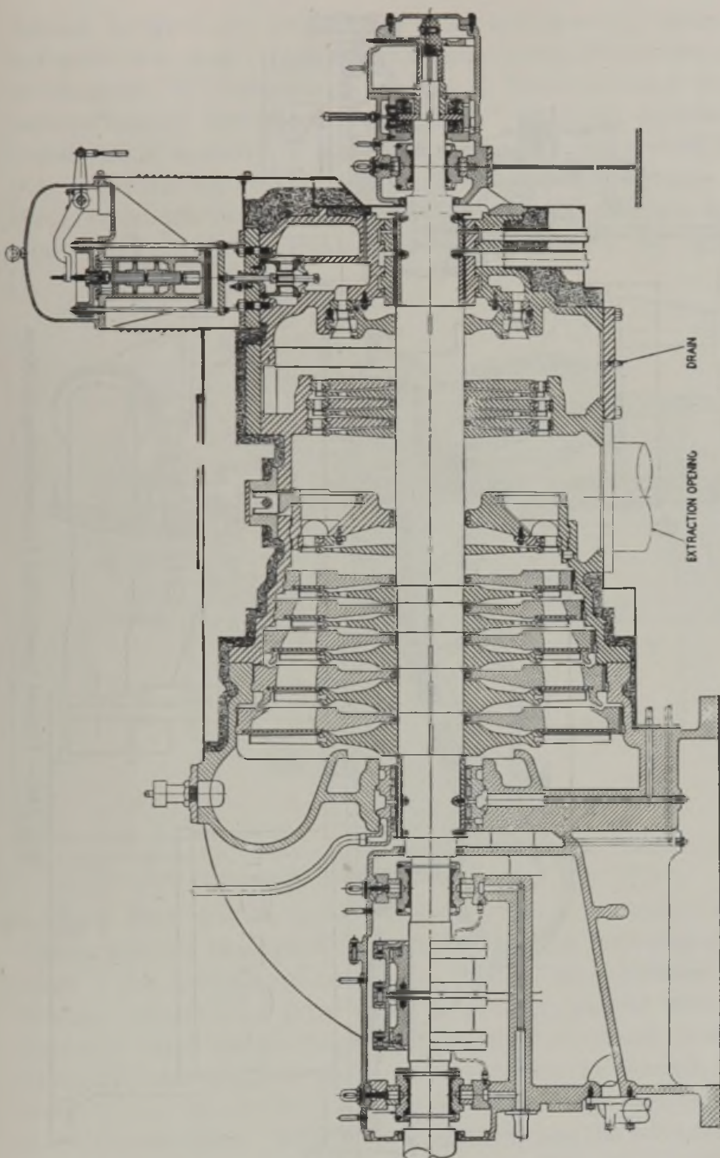


Fig. 239.—Bleeder turbine.

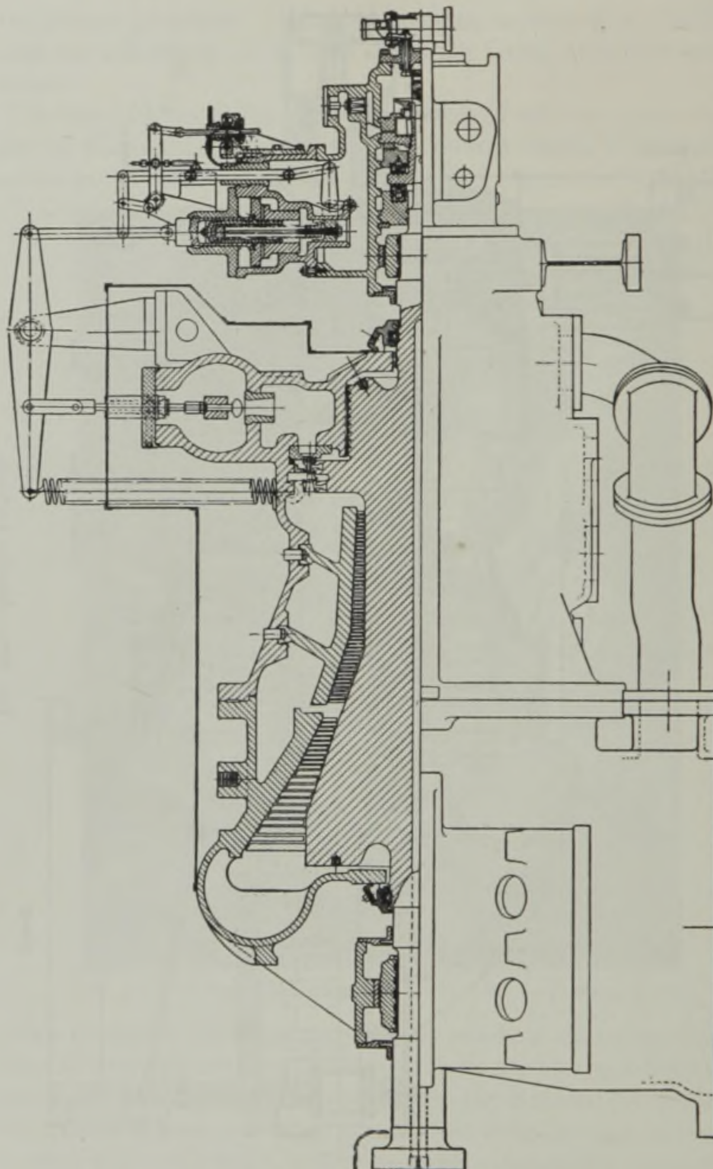


Fig. 240.—Typical construction for condensing turbines.

moving buckets are occasionally used, but generally there are not more than two. It is evident that a complete turbine may be composed of Curtis stages or Rateau stages in other combinations besides that shown in Fig. 236. Although the energy-transforming capacity of the Curtis stage is much greater than that of the Rateau (which points to fewer wheels and stages, and cheaper construction), nevertheless the Rateau principle is the more efficient in the transformation, and nearly all impulse

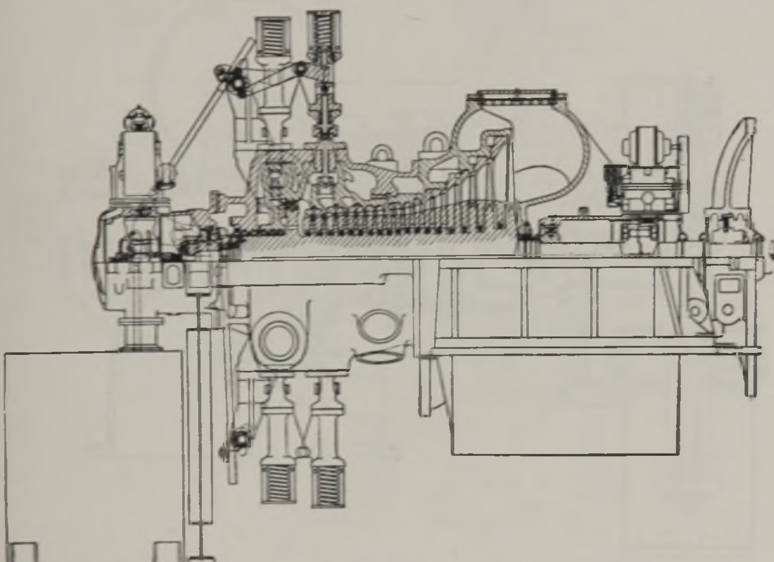


FIG. 241.—25,000-kw, 3,600-rpm condensing double-shell General Electric turbine, designed for 1,350 psi, 1000 F initial temperature.

turbines that are designed for good efficiency employ either Rateau staging throughout, or Rateau staging following a first stage of the Curtis type, as in Fig. 236. This particular combination results in a large drop in pressure in the first-stage nozzles *C* and the consequent admission of steam into the relatively large shell or casing of the turbine at a much lower pressure than would have been the case if the first stage had been of the Rateau type. Such an arrangement has the advantage of greatly simplifying the problem of design of the shell and separating diaphragms.

A cross section of the complete turbine, which Fig. 236 shows only in part is presented in Fig. 242, from which it is seen that the entire machine consists of one Curtis stage, seven Rateau stages of constant wheel diameter, and six Rateau stages of enlarging diameter. The enormous volume of the steam after expansion to low exhaust pressure is provided for here by making the diameter of the last stage and radial width of nozzle, or length of bucket, a maximum consistent with safety, to provide an annulus of the greatest length and width possible. At the same time, the higher bucket speed calls for a higher steam velocity, which, associated with the greater area of flow, is sufficient to handle the volume.

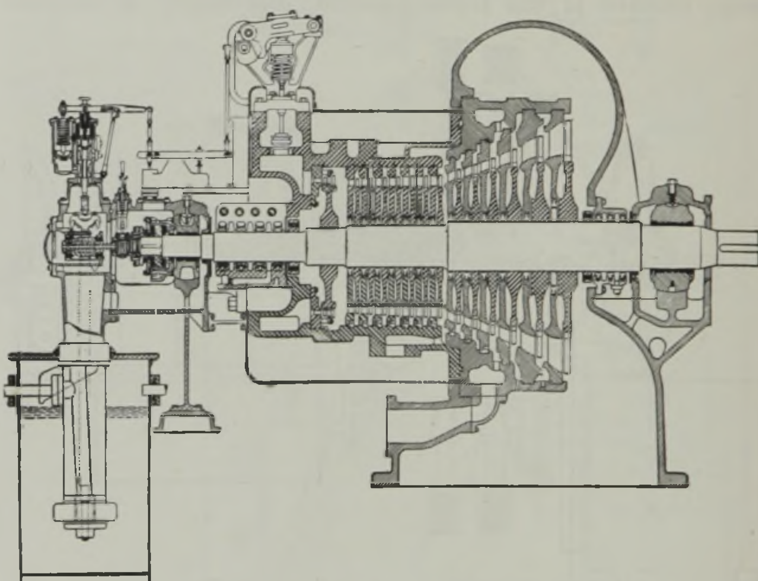


FIG. 242.—Sectional view of a Curtis-Rateau turbine of 3,000-hp rating, showing the arrangement of wheels, diaphragms, bearings, packing, and other features.

217. Details.—Steam leakage from one part of a turbine to another and to the atmosphere must be held to a minimum if the turbine is to be economical in the use of steam. Also air leakage into the low-pressure sections must be prevented if condensers are used. In small turbines, closely fitting carbon rings are placed around the shaft. In larger turbines "labyrinth packing" (Figs. 243 and 244) is used. This packing consists of a series of strips of soft metal with small running clearance between the moving parts. A small amount of steam leaks continuously

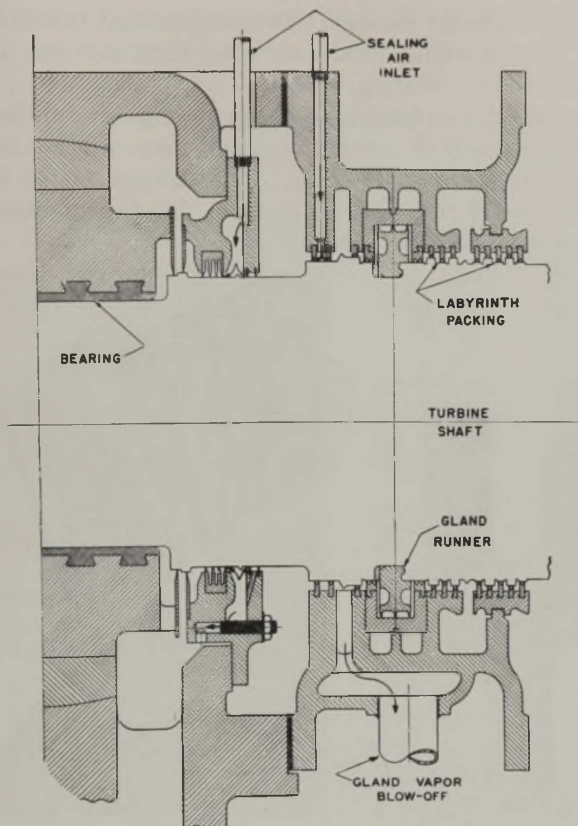


FIG. 243.—Air seal for bearing oil rings and gland in Allis-Chalmers turbine

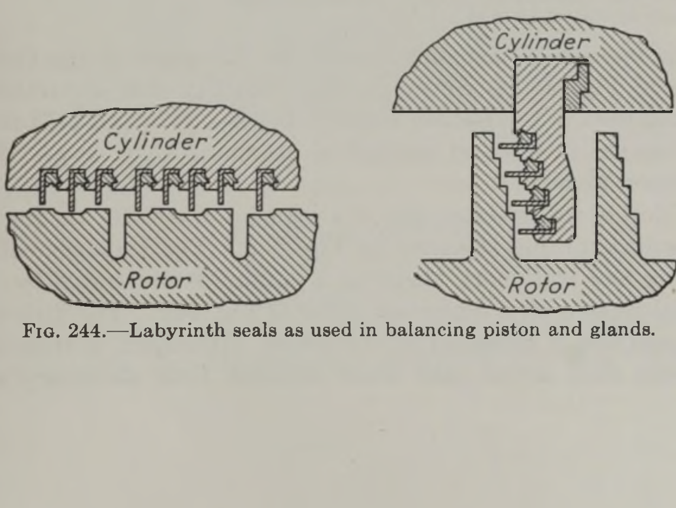


FIG. 244.—Labyrinth seals as used in balancing piston and glands.

through this tortuous passage. It is impractical to build packing with zero running clearance with high rubbing velocity because of localized heating.

Turbine buckets, or blades, are made by forging or milling alloy steel into the special shapes required and these are then fastened to the rotor by various methods. The stresses on the roots of these blades are of two kinds: (1) tensile, due to centrifugal force and (2) bending, due to steam forces.

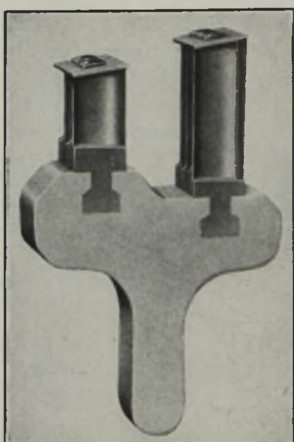


Fig. 245.—Segment cut from a Curtis turbine wheel, showing the dovetail attachment of the buckets to the wheel disk.

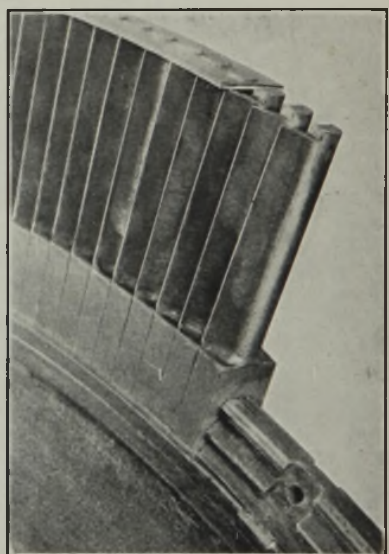


Fig. 246.—A method of fastening single-row buckets to wheels used in Rateau turbine.

The manner of fastening buckets to the wheel of the Curtis stage of Fig. 236 is shown in Fig. 245; Fig. 246 illustrates a method used in the Rateau stages. Figures 247*a* and 247*b* show the impulse blades and method of attaching them in an Allis-Chalmers turbine.

A view of the entrance side of a segment of spindle blading for a Parsons turbine is shown in Fig. 248. The arrangement of moving and stationary blades in a turbine is illustrated by the partial section on a diametral plane of Fig. 249. The direction of steam flow is indicated by the arrow. Inasmuch as there is a pressure drop across each blade annulus, both stationary and moving, steam will tend to by-pass the blades from D to E, E to F, F to G, etc., down through the turbine, and it is necessary to make the clearances at the blade tips at A, B, C, etc., as small as

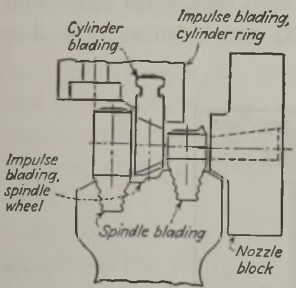


Fig. 247a.—Impulse blades and method of attaching them in Allis-Chalmers turbine.

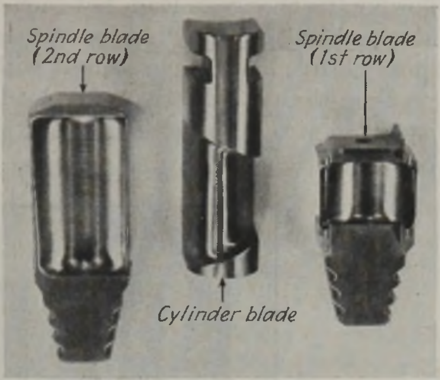


Fig. 247b.—Impulse blading for Allis-Chalmers turbine.

consistent with safe operation in order to reduce leakage to a minimum.

218. Methods of Governing Steam Turbines.—The function of the governor of any prime mover is to adjust the rate of energy

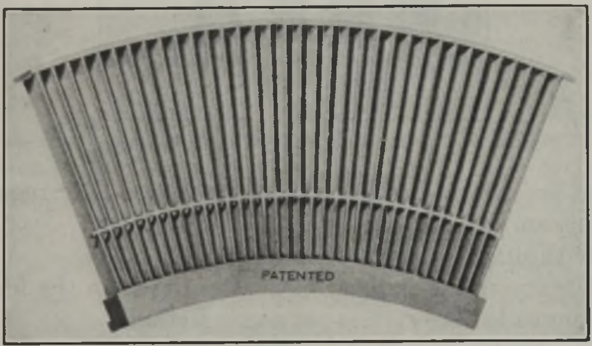


Fig. 248.—Segment of spindle blading for Allis-Chalmers Parsons turbine.

inflow to the load, which is accomplished by controlling the rate of flow of the working fluid. In most engines it is necessary to maintain a speed within slight variations from constancy, but no self-contained governing system can maintain absolute constancy, since it is only through a change of speed that a governor can sense an unbalance between load demand and energy supply, and take the necessary steps to restore equilibrium. This does not mean a mere momentary departure from constant speed following the change in load, but each different load is associated with a particular speed. At light loads, the unit runs faster than at heavy loads. A turbine's speed, within the range of governing, is an almost exact indication of its load, although total variations of speed from zero to maximum load may not be more than 3 or 4 per cent.

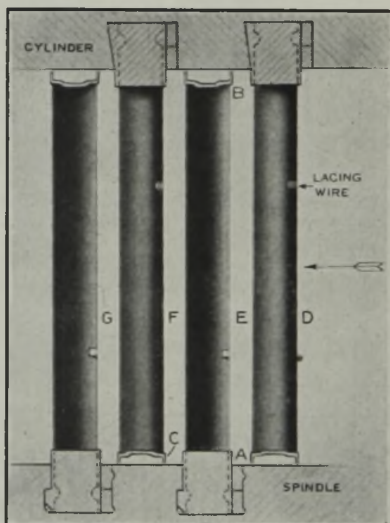


Fig. 249.—Two stages of Allis-Chalmers Parsons turbine.

There are two means by which a governor may regulate the flow of steam to a turbine:

1. By throttling.
2. By varying the number of nozzles in use in the first stage, sometimes called the *cutting-out nozzle* method.

In the *throttling method,* a valve under the control of the governor restricts the flow of steam to the necessary degree for regulation. As a result, the steam enters the turbine with less *available heat* than it had before throttling, which represents a loss.

With the *variable-nozzle method,* the rate of flow is controlled by varying the *number of first-stage nozzles* in use. Steam, therefore, enters the turbine with full pressure whether the number of nozzles open be many or few. Although this method involves no loss of available heat *before* steam enters the turbine, losses occasioned by disturbance of stage pressures *within* the turbine (assuming multipressure-stage type) are somewhat greater than those resulting from throttling.

Either method of steam control may be used in connection with impulse turbines, depending upon the design. However, the Parsons turbine, partaking as it does of the reaction principle, *must have full annular flow* throughout and is, therefore, limited to the use of the throttling method only. In a combination turbine whose first stage is of the impulse type, and remaining stages Parsons (see Fig. 222), either method of governing may be employed.

At *normal load*, which is that load requiring the flow of steam for which the turbine is designed, the method of governing will have no influence upon efficiency in the use of steam. But at *fractional loads*, the imposition of a necessary control upon steam flow causes losses in energy that do not exist at normal loads. For turbines operating through wide ranges in load, the variable-nozzle method is considered to be somewhat more economical of steam, but under conditions of less deviation from normal, the throttling method is as good if not better.

With the throttling method, the maximum rate of flow through the complete turbine occurs at normal load, when the throttling valve is wide open. In order to carry an *overload* a greater mass rate of flow must be provided for, which is accomplished by *by-passing* steam, either around the first stage or, in later designs by by-passing steam around a few stages after passing through the first stage, this latter reducing the temperature inside the casing itself. The by-passing is accomplished by the governor opening the by-pass valve whatever amount necessary. The maximum rate of flow takes place, and hence the maximum load, when the by-pass valve is wide open, in which case the by-passed stages do no work. Thus, somewhat paradoxically it may seem, the turbine carries a heavier load with fewer stages. Its efficiency is less, however, because the number of stages is then insufficient to handle the steam economically. With the variable-nozzle method, overload is provided for by additional nozzles in the first stage, above the number required for normal load, which are brought into use by the governor.

Figure 250 shows one arrangement of steam chest for variable nozzle control, and Fig. 251 is a common type of valve for regulating the extraction pressure of a turbine.

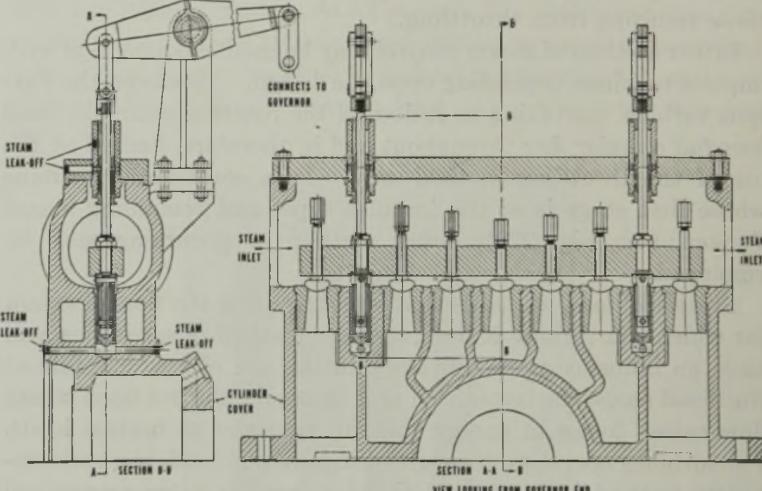


FIG. 250.—Multivalve steam chest, Westinghouse turbine.

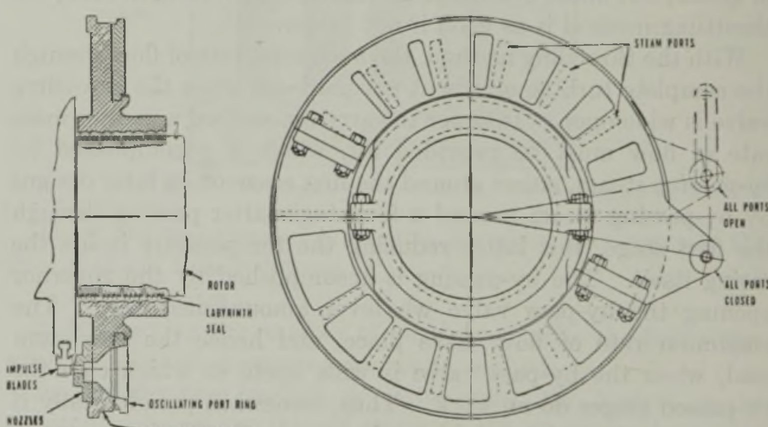


FIG. 251.—Grid type extraction valve, Westinghouse turbine.

219. Compounding Steam Turbines.—The term *compounding*, formerly employed to describe what we have called multiple staging of velocity or pressure, is now generally used to denote an arrangement in which the working elements of a turbine are housed in more than one casing or "cylinder." Classified upon this basis, turbines may be:

1. Single cylinder.
2. Tandem compound.
3. Cross-compound.
4. Vertical compound.

A large single-cylinder turbine is illustrated in Fig. 252. The external part of the wheel casing is included between the points marked *A* and *B*, although several of the low-pressure stages with their casing extend into the exhaust shell *C*. (See similar

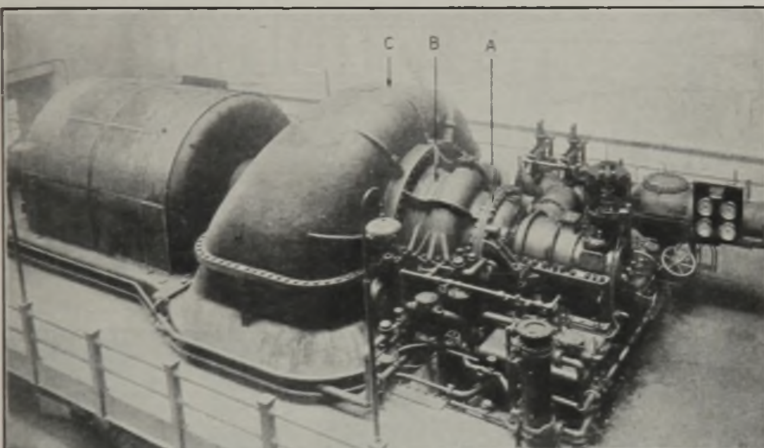


Fig. 252.—The 35,000-kw, 1,500-rpm, single-cylinder General Electric turbine in Boston Elevated Railway Company station.

arrangement in Fig. 242.) The turbine proper is rather dwarfed by the great exhaust shell and the multitude of control and auxiliary apparatus necessary for operation.

A section view of the 160,000-kw turbine of the East River station in New York is seen in Fig. 253. The high-pressure section is *single flow*, while the low-pressure is *double flow*, steam entering at the center and dividing, half passing in either direction and completing its expansion in six stages. With double flow, only half the enormous total volume of steam is handled in each of the last stages. The energy of this turbine is absorbed by a single electric generator. Cooling air is circulated through the generator by fans in a closed system and the air is cooled by water coils located in the housings at the sides.

A photograph of a tandem-compound Parsons turbine with single-flow high-pressure section and double-flow, low-pressure,

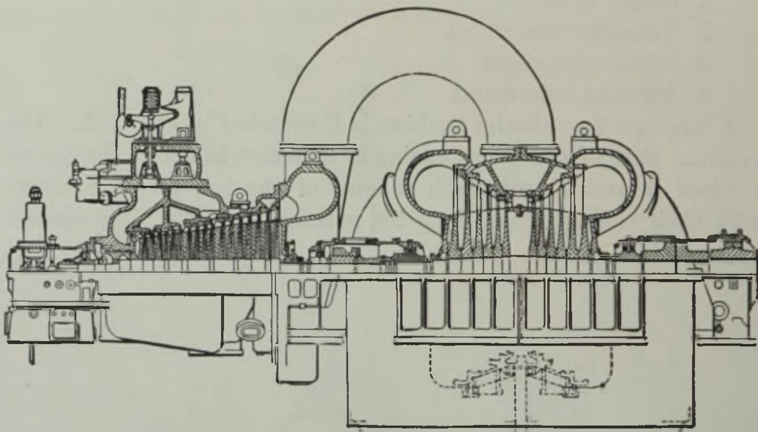


Fig. 253.—160,000-kw, 1,500-rpm, 24-stage tandem-compound double-flow steam turbine for New York Edison Company.

is shown in Fig. 254. Figure 255 illustrates the rotor of the low-pressure section of this turbine.

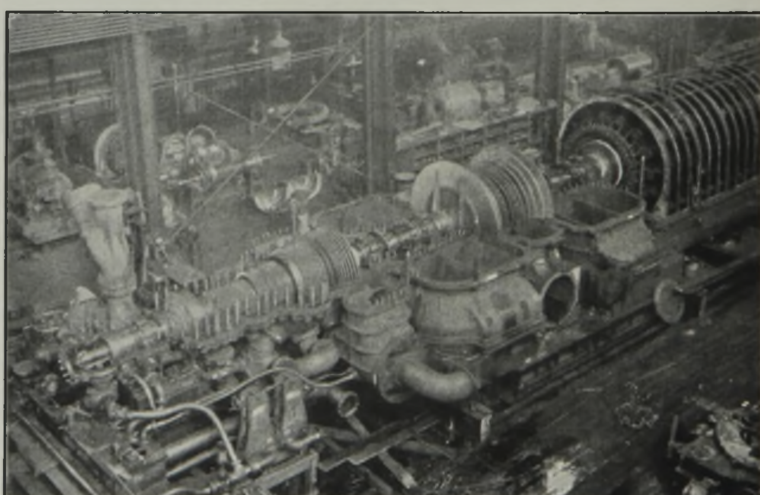


Fig. 254.—A 65,000-kw, 1,800-rpm Allis-Chalmers tandem-compound turbine.

A drawing of a *vertical-compound* arrangement is shown in Fig. 256, in which the high-pressure turbine with its generator is placed on top of the low-pressure element. In this turbine, the steam entering the high-pressure unit has a pressure of 1,200 psi

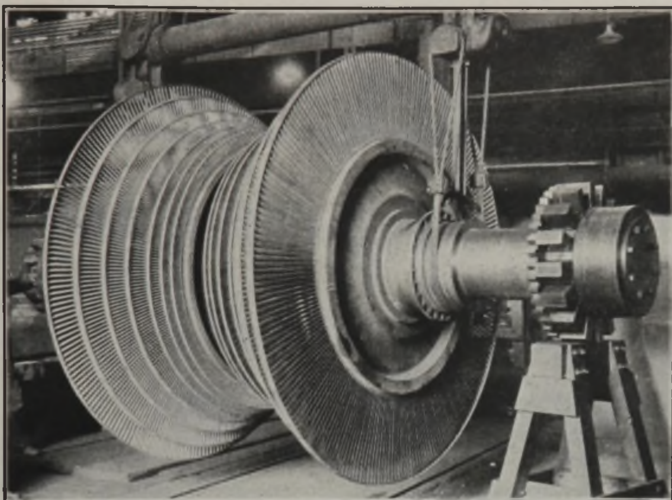


FIG. 255.—Rotor, or spindle, 65,000-kw Allis-Chalmers turbine.

and a temperature of 750 F. After passing through this element, it is reheated to 750 F and then expands through the low-pressure unit to an exhaust pressure of $1\frac{1}{4}$ in. Hg.

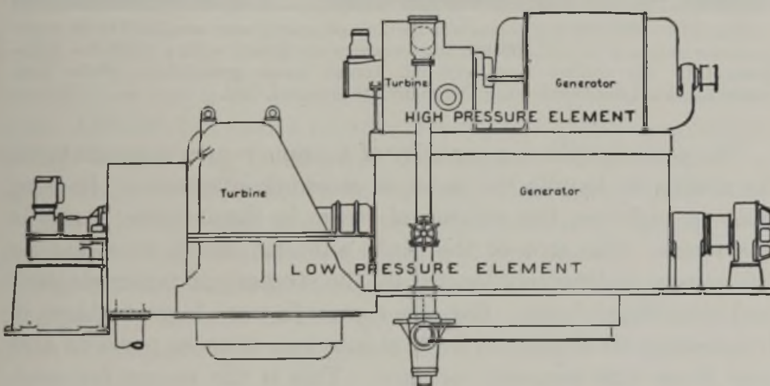


FIG. 256.—50,000-kw, 1,200-lb, vertical-compound turbine generator set. The 3,600-rpm, high-pressure element is mounted on top of generator of low-pressure element.

A photograph of the great 208,000-kw turbogenerator unit of State Line, Ind., is shown in Fig. 257. The entire steam flow passes through the single-flow high-pressure turbine, near the center of the picture, the exhaust from which is divided, half going to each of the low-pressure turbines located on either side. The steam is further divided at each end of the low-pressure turbines in the Y-branches, the total exhaust being disposed of in eight vertical condensers. The generator of the low-pressure turbine in the foreground can just be seen in the fork of the farther Y.

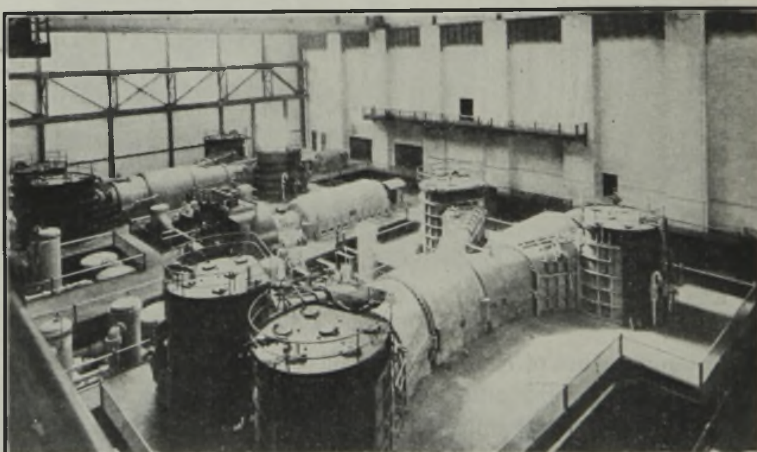


FIG. 257.—208,000-kw, three-unit turbine-generator set—one 76,000-kw high-pressure unit and two 62,000-kw, low-pressure units each with a 4,000-kw house generator. Air coolers used with all except house generators. State Line station, State Line Generating Company, Hammond, Ind.

The power-producing capacity of a prime mover depends upon its ability to handle the mass of working substance. In condensing turbines, the volume of steam in the last few stages is enormous. The area of the blade annulus, which must handle this steam, is definitely limited by the strength of materials used and centrifugal force. For this reason in very large turbines it is necessary to divide the flow up into two or more portions and pass these into separate turbines. This is the reason for most compounding. In some units it is desirable to *reheat* the steam after it is partly expanded; it is then more convenient to pass this reheated steam into a separate turbine.

220. Turbines for High Pressure and High Temperature.—From Carnot's deductions, it has been shown that of the total amount of heat supplied to a gas, the fraction $\frac{(T_1 - T_2)}{T_1}$ is *available* to a heat engine for doing useful work, but the expression applies only when *all of the heat* is supplied at the absolute temperature T_1, and all of the unavailable heat is rejected at the absolute temperature T_2. If steam is formed at a pressure of 350 psia from water at 80 F, and superheated to 750 F, the supply of heat takes place along the path $ABCD$ (Fig. 258), which is far from a constant-temperature line. Superheating

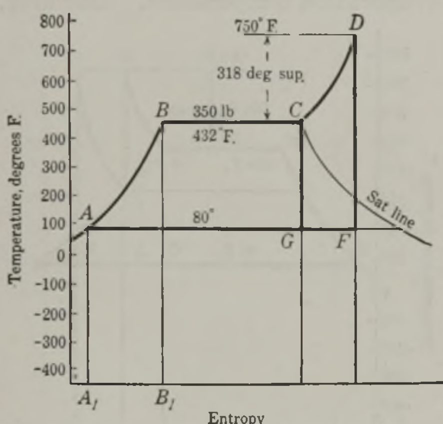


FIG. 258.—Temperature-entropy diagram for steam at a pressure of 350 psia and a temperature of 750 F.

contributes only a small amount to the efficiency of the Rankine cycle $ABCDF$, but from a practical standpoint it is of the greatest benefit in reducing moisture in the turbine. The moisture is detrimental mechanically to buckets and nozzles and reduces the stage efficiency. Operation in the wet or saturated region is also conducive to supersaturation. When steam expands from C to G, Fig. 258, the whole operation occurs in the wet region, whereas in expansion from D to F, a good share of the operation takes place in the superheat region, and the end point F is well toward the dry state.

If fuel cost is such as to make economical the use of higher pressure to increase the cycle efficiency, it also becomes necessary to increase the steam temperature at the entrance to the turbine to limit the moisture content of the steam at exhaust to a practical maximum of 12 per cent. Another standard pressure that is used is 650 psi and the corresponding temperature is 825 F. Other pressures that are used without reheating of the steam are 850 psi and 900 F and 1,200 psi to 1,400 psi and 950 F to 1000 F. This last temperature is the practical maximum that is permitted in the present stage of the art of turbine and superheater building.

The theoretical efficiency of a higher pressure Rankine cycle *AMNRH* (Fig. 259) is markedly greater than the cycle *ABCDF* (Fig. 258), because a much larger part of the heat is supplied at the higher evaporation temperature. However, the "ceiling"

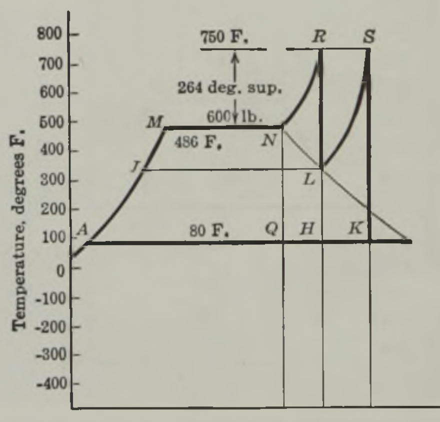


FIG. 259.—Temperature-entropy diagram for steam at a pressure of 600 psia and a temperature of 750 F and reheated to 750 F.

temperature of 950 F to 1000 F for superheating must be observed on account of the present limitations of engineering materials, and this displaces the expansion line *RH* (Fig. 259) to the left of *DF* (Fig. 258), so that the resulting practical loss from higher moisture content in the low-pressure stages of the turbine will largely discount the theoretical gain from higher pressure. In order to conserve this gain, it becomes necessary to *resuperheat* in connection with pressures above 1400 psia. After expanding in the higher pressure stage of the turbine (*RL*, Fig. 259) until it reaches the saturation state at *L*, the steam is withdrawn from the turbine and superheated again at the lower pressure (*JLS*) to some temperature *S* which approaches or even equals the initial superheat temperature. It is then returned to the turbine to complete its expansion along *SK*, whose end point *K* is not far inside the saturation line. The Rankine cycle for high pressure and resuperheating is represented by *AMNRLSK.*

One 76,500-kw unit, being installed in Indiana, is to operate on 2,400 psi and 940 F steam. It is necessary to resuperheat the steam to 900 F after it is partly expanded in order to limit the moisture percentage in the exhaust steam. The ideal cycle for this machine is *OHJKLMN*, Fig. 260. Superimposed on this reheat cycle is the Rankine cycle *OABCD* for 1,200 psi and 950 F.

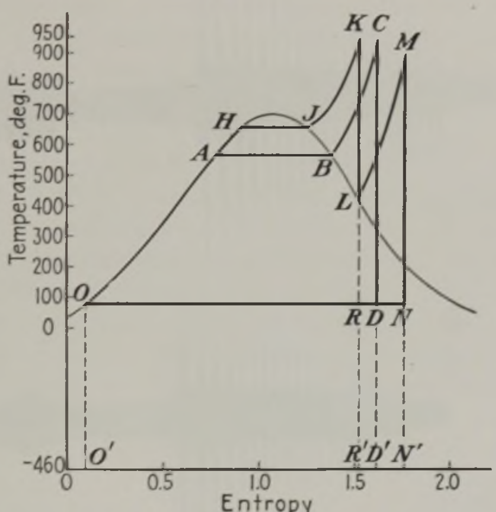


FIG. 260.—Temperature-entropy diagram for a 1,200 psi Rankine cycle *OABCD* and for 2,400 psi reheat cycle *OHJKLMN.*

Since heat is added at a higher temperature during a large part of the reheat cycle, the ideal cycle efficiency is higher. The same exhaust pressure, 1 in. Hg abs, is used for both cycles.

Since energy absorption in steam-turbine buckets is increased when bucket speeds are increased, the trend in recent years has been to increase the rotative speeds of the larger turbines and thus reduce the physical size of machine necessary for a given capacity. Higher steam pressures have increased the available energy per pound of steam and hence reduced the quantity of steam used for a given capacity of machine. This also permits a reduction in physical size of the machines.

Figure 261 from an article by G. B. Warren in the January, 1941, issue of the *Transactions* of the A.S.M.E. shows turbine rotors for three 50,000-kw turbines. The 3,600-rpm machine is a tandem-compound machine with the low-pressure element made double flow. This is made necessary by the high speed since stresses in the roots of the blades limit their length.

221. Superposed Prime Mover.—A superposition, or superposed prime mover, is one that is superposed upon a prime mover

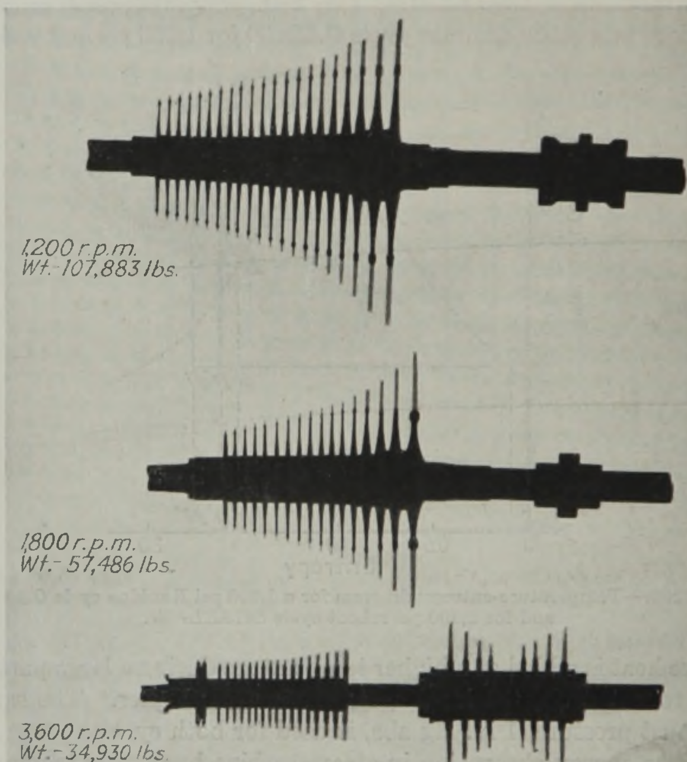


FIG. 261.—Three rotors for 50,000-kw General Electric condensing-steam turbines. Longitudinal sections.

working at lower pressure. For example, to rehabilitate an existing power station operating with 300 psi or 400 psi by installing new boilers operating with 1,000 psi to 1,500 psi and expanding the high-pressure steam in superposed turbines down to the pressure required by the low-pressure turbines or that of the low-pressure boilers if any remain in operation. Another name commonly given to this type of turbine is *topping turbine.*

222. Partial Energy Turbines.—In most manufacturing plants, low-pressure steam is required to heat buildings in the winter time; in many industries, large amounts of heat at relatively low temperatures are required in various manufacturing processes, of which paper and textile mills, rubber factories, woodworking and chemical plants are examples. Under these conditions, it is often economical to generate steam at high pressure (which requires but little more heat than to make steam at low pressure), and to *divide the total available heat*, assigning only a part to a turbine for power generation and utilizing the remainder to effect some purely thermal process. Turbines for use in such a combination are classified as:

1. Bleeder turbines.
2. Back-pressure turbines.

The *bleeder turbine* is built as a full-energy turbine, with the necessary number of stages to utilize the total available heat of the steam expanding from initial conditions to condenser back pressure, but has an outlet in the casing at an appropriate point in the steam's progress through the turbine at which steam may be drawn off or "bled," with sufficient available heat or thermal head to accomplish the purpose desired. In the bleeder turbine the entire amount of steam passes through the high-pressure stages, after which it divides, a part passing out through the bleeder line, and the remainder going on through the low-pressure section of the turbine. With this arrangement, both the demand for steam for thermal processes and that for power upon the turbine may be accommodated within limits. The bleeder turbine offers a wide opportunity for effecting industrial power-plant economies, but each particular case is a problem in itself requiring careful study. A bleeder turbine is illustrated in Fig. 222, where A indicates the nozzles and B the rotor of the high-pressure double-row impulse stage. After expanding in this first stage and the Parsons section D, a part of the steam passes out through the opening C to the bleeder line; the remainder goes on through the turbine doing work in the Parsons section E and exhausting into the condenser. F is a ring under the control of a governor, by means of which the flow of steam to the low-pressure section of the turbine can be dammed in sufficient amount to maintain a constant pressure at C. Other bleeder turbines are illustrated in Figs. 242 and 262.

A *back-pressure turbine* consists of the high-pressure section only of a full-energy turbine, and all of the steam exhausting from it is used for heating. The exhaust pressure is that corresponding to the temperature at which it is desired to use the steam, and may be only a few pounds above atmospheric, or very much higher. The rate at which the turbine develops work energy and that at which heat is supplied to thermal processes are rigidly related. If more heating steam is demanded, the turbine *must* use more steam and consequently develop more

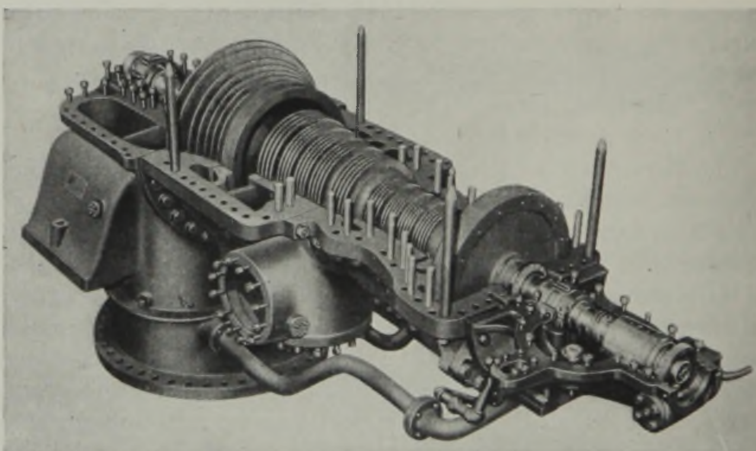


FIG. 262.—Top view (open) 1,500-kw Allis-Chalmers bleeder turbine.

power. The back-pressure turbine must be either one unit of a power system, with others free to respond to variations in power demand; or the turbine's exhaust must be but a part of the total supply of heating steam, so that other sources may be controlled to respond to variations in the total demand for heat. Owing to this inflexibility, the back-pressure turbine is rather limited in its applications.

In some cases, a turbine may share the total available heat of the steam with a piston engine. Turbines for this kind of service fall into two classes:

3. The mixed-pressure or mixed-flow turbine.
4. The low-pressure turbine.

The mixed-pressure turbine, like the bleeder type, is built as a full-energy turbine but with an *inlet* at an appropriate point in the casing for receiving exhaust steam from a piston engine, usually at about atmospheric pressure, from which the low-pressure stages complete the expansion to condenser exhaust pressure. The high-pressure end of the turbine is connected to a steam source, but as long as the exhaust from the engine is sufficient in amount to develop the power demanded, no high-pressure steam enters the turbine and the wheels in that section run idly. In case the supply of low-pressure steam becomes inadequate, the governor admits high-pressure steam to make up the deficiency; in the event of zero supply of low-pressure steam, the turbine operates as a complete or full-energy unit. The mixed-pressure turbine finds application wherever it is necessary or desirable to employ a piston engine, as for example to operate steam hammers, drive rolling mills, or transmit power by belt or ropes. The turbine develops electrical energy for other uses in the plant.

A low-pressure turbine is like a mixed-pressure turbine except that it has no high-pressure section and depends solely for its steam supply upon the exhaust of a piston engine. The entire combination is employed to develop electrical energy, the engine and turbine each with its own generator but tied together electrically, so that the whole becomes a single cross-compound unit. The low-pressure turbine is now seldom used but, during the period of transition when the turbine was replacing the piston engine for electrical generation, it had a considerable application where, by its installation in connection with already existing high-grade reciprocating engines, too good to be scrapped, the capacity and efficiency of the plant could be greatly increased.

223. Feed-water Heating by Extraction Steam.—In a Rankine cycle, such as $ABCDF$ (Fig. 258), it is possible by the use of an economizer to supply a part of the enthalpy of the liquid, A_1ABB_1, from furnace gases that have previously given up such an amount of heat to the boiler, or boiler and superheater, that by the time they reach the economizer they are not hot enough to raise the temperature of the water all the way from A up to the boiling point B, and it is thermodynamically wasteful to use high-temperature gases to make up this deficiency. It is better to employ these hot gases to evaporate the water at a high temperature or to superheat the steam, and to warm the condensate on its way back from the condenser to the boiler, by the use, step by step, of the latent heat of a part of the working steam extracted from the turbine at a series of points (usually one to four) at successively lower pressures. In this way, all of the high-temperature heat of the furnace gases is used to evaporate or superheat steam, the main body of which (perhaps 75 or 80 per cent) expands all the way through the turbine. The remainder expands only partway therein, a portion being drawn off at each point of extraction. Each portion or quota of such steam must be appropriate in amount so that its latent heat will bring the feed water up to or near its own temperature from the next lower temperature level.

Extraction heating exemplifies the very important thermodynamic principle that, to conserve availability, when heat is passed from one body to another, the transfer should occur with the least possible difference of temperature. The theoretical gain from the use of extracted or bled steam in the manner described, is verified by practical results, and all later turbines of large size (which must, of course, belong to the multipressure-stage class) are now designed to operate upon this plan.

224. The Mercury-steam Plant.—In the Emmet mercury-steam system as developed by engineers of the General Electric Company in the plant of the Hartford (Conn.) Electric Light Company, mercury vapor, generated at a pressure of about 70 psi and 884 F temperature is utilized in a turbine where the vapor is expanded to about 28 in. vacuum, at which pressure the temperature is 458 F. Latent heat of the exhaust mercury is then transferred to water at approximately 435 F to make steam at a pressure of 350 psi in a heat exchanger which is at once a mercury condenser and a steam boiler. The steam thus formed, after being superheated to 700 F, is utilized in a steam turbine, expanding therein to condenser exhaust pressure.

In Fig. 263, the mercury cycle is $MGHN$ and the steam cycle, $ABCDE$. Heat received by the water along the line JBC (J being the temperature of the feed water) is equal to that given up by the condensing mercury along NM. The total amount of primary heat given to the mercury is M_1MGHH_1, of which the major part G_1GHH_1 (heat of vaporization) is absorbed at *the maximum temperature of the cycle.*

The point F, where liquid and saturation lines merge, is called the *critical point* of steam. With each increase of pressure, the amount of heat that can be absorbed as latent heat becomes less and less, until at the critical point the latent heat is zero. At the same time, the use of higher initial pressure brings the starting point of expansion farther to the left on the temperature-entropy diagram, so that a greater part of the expansion line lies in the wet region and the difficulties with moisture in low-pressure stages of the turbine are thereby aggravated. Steam therefore, although possessing admirable properties as a working substance at moderate temperatures of evaporation, has its limitations at

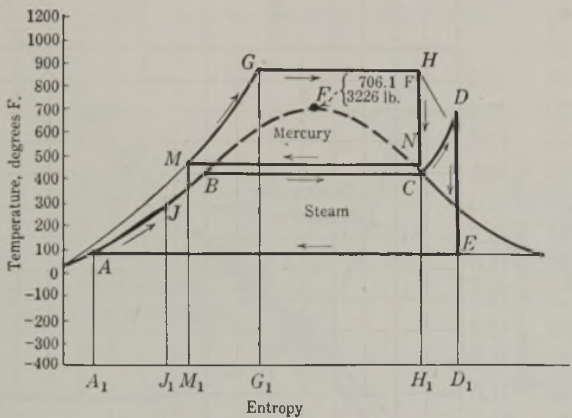


Fig. 263.—Mercury-steam cycle.

high temperatures. If some substance could be found whose critical temperature was say 1200 F to 1500 F and which still offered the desirable properties of steam in the low-temperature range, the efficiency of the vapor heat engine could be made much higher than that developed by the steam plant. Mercury is a substance that has excellent thermodynamic properties in the high-temperature range, as indicated by the length of evaporation line *GH* of Fig. 263, and also by the fact that the high evaporation temperature of 884 deg is accompanied by the very moderate pressure of 70 psi. On the other hand, mercury is unsuitable in the low-temperature range because of the impossibly high vacuum that would be necessary to bring the expansion down to a temperature near that of condenser cooling water. The mercury-steam cycle of Fig. 263 represents a combination of these two working substances in one system, with the mercury cycle utilizing heat in the high-temperature range, for which it is well suited, after which the heat, still at high temperature, is transferred to water and utilized in the remaining temperature range, to which its properties are well adapted. Such a scheme is called a *binary vapor system.*

There is no essential thermodynamic virtue in the mere use of two working substances, except as an expedient to utilize heat through a wide temperature range in the absence of some single substance with suitable properties throughout that range.

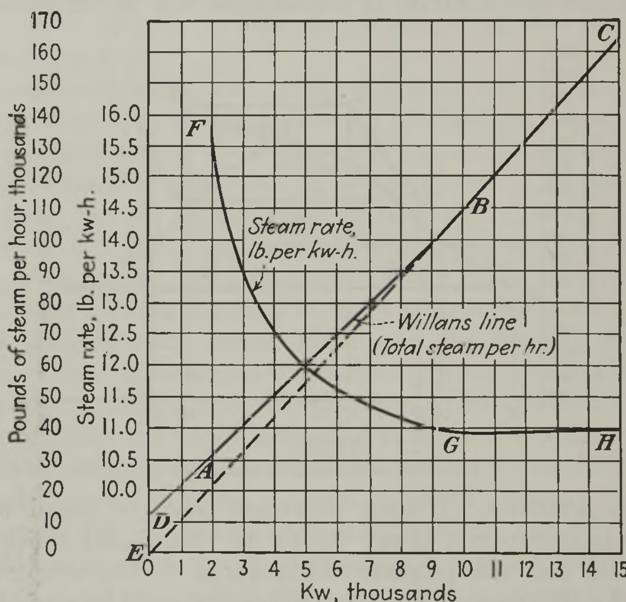


FIG. 264.—Willans line and steam-rate curve for a 15,000-kw turbogenerator.

225. Steam-turbine Performance.—Steam-turbine performance may be expressed in four different ways:

1. Total steam consumption, in pounds per hour.
2. Steam rate in pounds per kilowatt-hour, or pounds per horsepower-hour.
3. Thermal efficiency.
4. Ratio of actual thermal efficiency to ideal cycle efficiency.

Total steam consumption is a misnomer because a turbine does not "consume" steam; it removes some of the heat from the steam. It is desirable to know the pounds of steam used per hour by a prime mover so that the size of piping, boilers, pumps, etc., may be determined. The plot of total steam consumption per hour vs. load is known as a *Willans line*, the peculiar characteristic of which is that it is straight. Such a line is shown in Fig. 264.

Steam rate is the number of pounds of steam used per hour divided by the output of the unit in horsepower or kilowatts. The steam rate of a turbine varies with its load, and is lowest at that load for which the turbine is designed to give its best efficiency

$$\text{Steam rate} = \frac{\text{lb of steam per hr}}{\text{hp output}}, \tag{19}$$

or

$$= \frac{\text{lb of steam per hr}}{\text{kw output}}. \tag{20}$$

Since a statement of either the total pounds of steam used per hour, or the steam rate, tells nothing about the *heat supplied* to the steam used, such statements should always be accompanied by a statement of the initial steam pressure and temperature (or quality), and the exhaust pressure.

From Eq. (15), page 358, the thermal efficiency of a turbine is the heat equivalent of a horsepower-hour, or kilowatt-hour, divided by the heat chargeable to the turbine per horsepower-hour, or kilowatt-hour. Hence in the form of an equation

$$e_t = \frac{2{,}545}{\text{lb steam per hp-hr} \times \text{heat chargeable per lb of steam}},$$

or

$$e_t = \frac{3{,}412}{\text{lb steam per kwhr} \times \text{heat chargeable per lb of steam}}.$$

The heat chargeable is equal to the total enthalpy of the steam entering the turbine less the enthalpy of the liquid at the pressure of the exhaust. For example, take the case of a turbine generator using 10.26 lb of steam when operating with an initial pressure of 220 psia, 200 deg of superheat and 1 in. Hg exhaust pressure. The total enthalpy of steam at a pressure of 220 psia and containing 200 deg of superheat is 1,315.4 Btu and the enthalpy of the liquid corresponding to an exhaust pressure of 1 in. Hg is 47.1 Btu. Therefore, the heat chargeable per pound of steam is

1,315.4 − 47.1 = 1,268.3 Btu, and the thermal efficiency of the unit is

$$\frac{3,412}{10.26 \times 1,268.3} = 0.2622, \text{ or } 26.22 \text{ per cent.}$$

The ratio of actual thermal efficiency to ideal cycle efficiency is a statement of how well the actual machine performs compared to

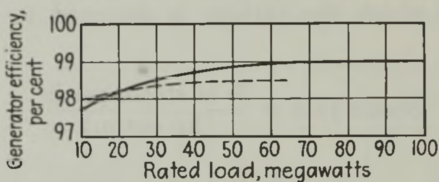


FIG. 265.—Generator efficiencies for Fig. 266.

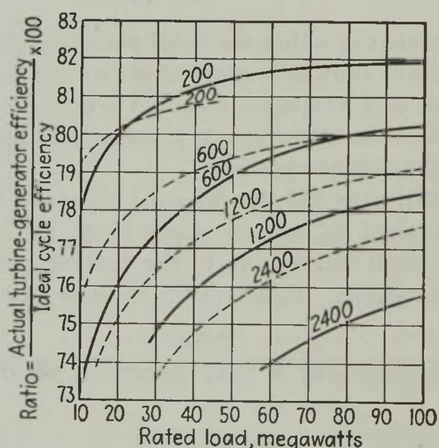


FIG. 266.—Variation of engine efficiency with pressure, size, and rpm for 300 F superheat. Dashed curves, 3,600 rpm; solid curves, 1,800 rpm; numbers on curves indicate supply pressure, pounds per square inch. Four per cent exhaust loss and 1.25 per cent mechanical loss.

a perfect machine operating under identical steam pressure and temperature conditions. This ratio is always less than one. It is sometimes expressed in per cent and called by the misleading term *engine efficiency.* This term may be defined also as the ratio of the number of Btu out of each pound of steam transformed into useful work, to the *available heat* of a pound of steam working between the limits specified. Taking the same example as above, the turbogenerator requires 10.26 lb steam per kwhr, so that each pound of steam contributes $\frac{3,412}{10.26} = 332.6$ Btu to useful work, and the available heat per pound of steam from a pressure of 220 psi and 200 deg superheat to 1 in. Hg back pressure is found, from the Mollier diagram, to be 422.5 Btu, whence the engine efficiency $= \frac{332.6}{422.5} = 0.7872$, or 78.72 per cent.

The engine efficiency is always less than unity because stage efficiencies are less than 100 per cent and because of windage, bearing friction, radiation, and other similar losses. A turbine is designed for a certain steam velocity and wheel speed, and when the governor regulates the steam supply in accordance with load changes, these velocity relations change while the nozzle and

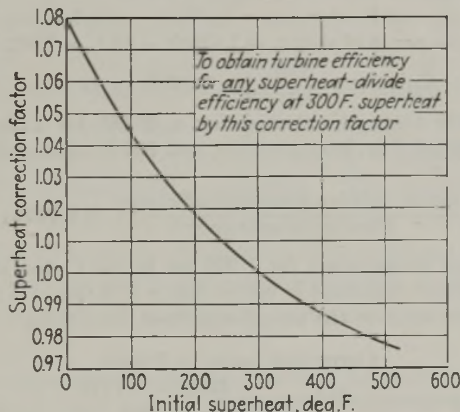


FIG. 267.—Superheat correction factor for Fig. 266.

bucket angles remain fixed. This results in turbulence and decreased stage efficiencies. At low loads the quantity of steam leakage through packings becomes a larger part of the total, and the bearing friction, windage, and radiation are larger in comparison with the power output. Hence, *any deviation* from design conditions results in a lower ratio of actual thermal efficiency to ideal cycle efficiency. This in turn is reflected in a higher *steam rate.*

The ratio of actual thermal efficiency to ideal cycle efficiency under *design conditions* is dependent upon initial steam pressure, and temperature, exhaust pressure, revolutions per minute, and size of unit. Figures 265, 266, and 267 from an article by Warren and Knowlton in the A.S.M.E. *Transactions* for February, 1941, show these relations for large condensing turbine-generator sets built recently by the General Electric Company. In Fig. 266 the generator losses are included with the various turbine losses to obtain the over-all ratio. Variation of the ratio with initial steam temperature is shown in Fig. 267. Individual machines may be expected to deviate from the curve values.

Example.—A 50,000-kw turbine generator runs at 3,600 rpm and is supplied with steam at 1,200 psia and 950 F. The back pressure is 1 in. Hg abs. The turbine operates on the Rankine cycle. Determine (*a*) the actual thermal efficiency; (*b*) the steam rate; (*c*) the total steam consumption; and (*d*) the moisture in the exhaust steam.

Solution.—(*a*) Enthalpy of entering steam = 1,470.0 Btu per lb

Quality of exhaust steam (Mollier diagram) = 77.9 per cent

Enthalpy of exhaust steam = 864.5 Btu per lb

Available energy per lb of steam = 1,470.0 − 864.5 = 605.5 Btu per lb

Heat chargeable per lb of steam = 1,470.0 − 47.1 = 1,422.9 Btu per lb

$$\text{Rankine cycle efficiency} = \frac{605.5}{1,422.9} = 0.4255, \text{ or } 42.55 \text{ per cent}$$

From Fig. 266 there is obtained for a 50,000-kw turbine generator, 3,600 rpm, 1,200 psi, 1 in. back pressure, and 300 F superheat, the following ratio:

$$\frac{\text{Actual turbine-generator efficiency}}{\text{Ideal cycle efficiency}} = 0.7775.$$

The saturation temperature for 1,200 psi is 567 F; the initial steam is 950 F; so the actual superheat is 950 − 567 = 383 deg. It is necessary to correct the above ratio to the actual superheat conditions. From curve *B*

$$\text{Correction factor} = 0.9885.$$

$$\text{Actual turbine-generator efficiency} = \frac{42.55 \times 0.7775}{0.9885}$$

$$= 0.3347, \text{ or } 33.47 \text{ per cent.}$$

(*b*) The actual turbine-generator set converts 33.47 per cent of the heat chargeable into electrical work at the generator terminals.

Heat converted to electrical work per lb of steam $= 0.3347 \times 1,422.9 = 476.2$ Btu per lb.

$$\text{Steam rate} = \frac{3,412}{476.2} = 7.17 \text{ lb per kwhr.}$$

(*c*) Total steam consumption $= 50,000 \times 7.17 = 358,500$ lb per hr.

(*d*) To determine the quality of the exhaust steam it is necessary to take account of those losses external to the turbine casing.

Losses external to turbine casing = 1.55 per cent generator (from Fig. 265)
1.25 per cent mechanical (bearings, etc.)
2.80 per cent

Hence, of all the work done by the turbine 2.80 per cent is wasted, and the heat converted to mechanical work on the turbine shaft

$$= \frac{476.2}{1 - 0.028} = \frac{476.2}{0.9720} = 490 \text{ Btu per lb of steam.}$$

Enthalpy of exhaust of actual turbine $= 1{,}470.0 - 490 = 980.0$ Btu per lb.

Quality of exhaust of actual turbine $= \dfrac{980.0 - 47.05}{1{,}049.2} = \dfrac{932.95}{1{,}049.2}$

$= 0.8864$, or 88.64 per cent.

Moisture in exhaust of actual turbine $= 11.36$ per cent.

226. Development and Application of the Steam Turbine.—DeLaval in Sweden and Parsons in England began their developments between 1880 and 1890, and Rateau in France was engaged upon his improvements somewhat later. By 1900, the Westinghouse Electric and Manufacturing Company in this country was building steam turbines under Parsons license, and the General Electric Company was in the process of developing a line of impulse turbines under the patents of Curtis, an American inventor. Both of these companies, it will be noted, were primarily manufacturers of electrical apparatus. The search for a more suitable prime mover for electrical generation was the incentive that led to the successful commercial development of the steam turbine, and reciprocally the steam turbine has made possible the tremendous output of electricity now demanded by modern civilization.

The *power capacity* of a prime mover is measured by the *weight* of working fluid that can be handled per unit of time, and this in turn is limited by the *volumetric capacity* of the machine to take care of the fluid after expansion to the final pressure. Steam flows continuously through a turbine and does its work while flowing, whereas in a piston engine the flow lasts for a fraction of the cycle only, and then must be interrupted to allow the steam to complete its work. The steam turbine dominated its early rival, the piston engine, chiefly because of its ability to handle enormously greater volumes of steam than could ever be utilized in an intermittent-flow "containing" type of unit. The volume of steam discharged by a 50,000-kw turbine exhausting at 29 in. vacuum, is of the order of 30,000,000 cu ft per hr, to handle which the last wheel of the rotor must be some 10 ft in diameter with buckets nearly 3 ft long.

Although its principal field of application is that of electrical generation, the steam turbine is now used for the propulsion of nearly all large vessels, both mercantile and naval. The screw propeller being a relatively slow-speed element, the speed of the turbine must be reduced accordingly, which is accomplished by mechanical gears or electrical reduction, although in earlier installations it was done by the multiplication of stages only. Something of the turbine's inherently desirable qualities of small size and weight and low cost must be sacrificed whenever its speed is reduced, no matter by what means it is done. A turbine is nonreversible as to direction of rotation, so that in marine work, a separate reversing turbine, of small size but large torque, must be provided, which runs idly during forward operation.

Steam turbines are now used extensively in smaller sizes, to operate direct-connected centrifugal compressors, and to drive fans and centrifugal pumps, usually through mechanical gears. Small geared units provided with pulley for belt drive are also available.

PROBLEMS

1. A jet of water is delivered from a nozzle 1 in. in diameter with a velocity of 100 fps and is directed upon a wheel with buckets that turn the stream through 180 deg. If the buckets are traveling at 40 fps, what impulse force will be exerted upon the buckets, (*a*) assuming frictionless flow within the buckets; (*b*) considering a frictional effect that makes the relative leaving velocity 90 per cent of the relative entering velocity?

2. What reaction force will be exerted upon the compartment containing the nozzle of Prob. 1?

3. What velocity would be generated in a perfect nozzle expanding steam from a pressure of 160 psia and 400 F temperature to a pressure of 1 psia?

4. Steam is supplied to a nozzle at 140 psia and 500 F; the back pressure is 20 psia. Determine the theoretical velocity of discharge.

5. A reciprocating engine receives steam at 185 psia and a temperature of 400 F and exhausts at a pressure of 16 psia. If a turbine is added and the condenser pressure is 2 in. Hg abs, what is the increase in the available heat? (Expansion is adiabatic.)

6. What weight of steam, in pounds per hour, will be delivered by a nozzle whose throat area is 0.5 sq in., when steam is supplied at a pressure of 250 psia and 600 F temperature, and the back pressure is 140 psia?

7. An impulse turbine having a single-pressure stage develops 500 hp when utilizing 70 per cent of the available energy. The turbine is supplied with dry steam at a pressure of 190 psia and exhausts at a pressure of 16 psia. (*a*) Find the pounds of steam used per horsepower per hour. (*b*) The turbine is equipped with three converging-diverging nozzles each discharging the same amount of steam. Calculate the throat and exit area of one nozzle. (Neglect nozzle friction.)

8. Steam at 400 psia and a temperature of 700 F expands through a frictionless nozzle to a pressure of 50 psia. The exit area of the nozzle is 1.50 sq in. (*a*) Assuming the initial velocity of the steam to be zero, what weight of steam will pass through the nozzle per second? (*b*) What is the throat velocity in feet per second? (*c*) What is the throat area in square inches?

9. A Rateau turbine, to run at 3,600 rpm, is furnished with steam at a pressure of 160 psia and exhausts at 1 psia, for which conditions the energy available for work is about 320 Btu per lb of steam. The pitch diameter for all stages is to be 33 in. Making the approximate assumption that nozzle and bucket angles are zero, determine about how many stages this turbine should have.

10. A large steam turbogenerator uses 10 lb of steam per kwhr delivered at the generator terminals. The steam is supplied at a pressure of 390 psia and at a temperature of 720 F. The exhaust pressure is 1.5 in. Hg abs. (*a*) What is the actual thermal efficiency of the unit based on its delivered power? (*b*) What is the quality of the exhaust steam? (Neglect radiation loss.)

11. A turbine-generator set develops 10,000 kw at the switchboard and uses 10.5 lb steam per kwhr at the switchboard. Steam is admitted at a pressure of 405.5 psi and 600 F and exhausts into a condenser at 27 in. Hg vacuum. If the mechanical, radiation, and electrical losses are 10 per cent based on the output and the barometer reads 29.5 in. Hg, find (*a*) the thermal efficiency of the unit. (*b*) What per cent of the available heat is converted into useful work at the generator terminals?

12. A steam turbine operates with a throttle pressure of 250 psia and a total steam temperature of 600 F and exhausts into a condenser where the pressure is 1 in. Hg abs. Eighty per cent of the energy available on the basis of the Rankine cycle is delivered to the turbine shaft. The turbine shaft is directly connected to the shaft of an alternating-current generator. The efficiency of the generator is 95 per cent. Determine (*a*) the pounds of steam required to deliver one kilowatt-hour at the generator terminals; (*b*) the actual thermal efficiency of the entire unit based on the electrical output at the generator terminals.

13. A steam turbine receives steam at a pressure of 1,250 psia and 760 F, and exhausts at 350 psia into an evaporator. (*a*) What is the available energy per pound of steam? (*b*) What is the Rankine cycle efficiency of the turbine? (*c*) If the turbine utilizes 75 per cent of the available energy as shaft output, and the efficiency of the generator it drives is 95 per cent, what should be the water rate in pounds per kilowatt-hour? (*d*) If 20 per cent of the available energy appears in the exhaust steam, in addition to the unavailable energy, what is the condition of the exhaust steam?

14. A 50-hp turbine is supplied with low-pressure dry saturated steam at a pressure of 15 psia and exhausts into a condenser at 3 in. Hg abs. The turbine at full load uses 1,500 lb of steam per hr. A jet condenser is used with this machine and 60,000 lb of cooling water are supplied to the condenser per hr at 70 F. Determine, (*a*) the available energy of the steam; (*b*) the per cent of the available energy converted to work, on the turbine rotor; (*c*) the efficiency of the Rankine cycle; (*d*) the actual thermal efficiency; (*e*) the temperature of the water discharged from the condenser hot well.

15. Calculate the throat and discharge-end areas respectively, and length from the throat to the end of a nozzle to deliver 2,160 lb of steam per hr. Steam is supplied at a pressure of 140 psia and 1.0 per cent moisture, and the back pressure is 2 in. Hg abs. (Neglect nozzle loss.)

16. Steam at a pressure of 200 psia and with 138.21 deg superheat enters a nozzle and expands to 3 in. Hg abs. The flow is 2,400 lb per hr. (*a*) If 8 per cent of the available energy is lost through friction in the nozzles, what is the area of the exit? (*b*) What would be the throat area if there were no losses up to that point?

17. A jet of steam is delivered by a nozzle with a velocity of 2,000 fps upon a single row of impulse buckets traveling at 900 fps. The center line of the nozzle makes an angle of 18 deg with the plane of the wheel, and the discharge angle of the buckets is 27 deg. Friction in the buckets causes a loss of 12 per cent in relative velocity. (*a*) Determine either graphically or analytically, with illustrative sketches, the proper entrance angle for buckets, relative velocities, and absolute velocity and direction of steam leaving the buckets. (*b*) Calculate the impulse force per pound of steam acting upon the buckets from absolute velocities and also from relative velocities. (*c*) Calculate the bucket efficiency by both equations (*a*) and (*b*) of Art. 214.

18. Consider a two-row Curtis stage of a turbine, and for simplicity assume 180-deg buckets in both the moving and stationary rows. Neglect friction in buckets. The nozzle steam velocity is 2,000 fps. (*a*) At what speed must the buckets travel to give maximum efficiency? (*b*) What are the absolute and relative velocities throughout? (*c*) What impulse force per pound of steam is exerted upon each of the two moving rows of buckets?

19. Steam having a velocity of 2,500 fps enters a turbine blade moving at 1,000 fps in the same direction. The blade turns the steam through an angle of 180 deg. Owing to friction, the steam loses 15 per cent of its relative velocity while passing through the blade. (*a*) What is the residual absolute velocity of the steam? (*b*) How much work is done on the blade per second, if the steam is flowing at the rate of 1 lb per sec? (*c*) What is the efficiency of the blade?

20. For a steam pressure of 200 psia, 150 deg superheat, and 1 in. Hg abs back pressure, determine approximately the number of stages required in each of the following cases: (*a*) all Rateau stages with uniform-pitch diameter of 36 in., 3,600 rpm; (*b*) first-stage two-row Curtis, others Rateau, all having pitch diameter of 36 in., 3,600 rpm; (*c*) all Rateau with stages in two groups—first group having 36-in. pitch diameter, and second 48 in., 1,800 rpm, one-third of total energy to be absorbed in first group; (*d*) first-stage two-row Curtis, 48-in. pitch diameter, others Rateau, 42-in. pitch diameter, 1,800 rpm.

21. A 10,000-kw turbine with 18 pressure stages is supplied with steam at 280 psia and a temperature of 460 F. It is constructed with the first stage of a Curtis element (two velocity stages) and 17 Rateau stages (one velocity stage per pressure stage) of varying diameters. The back pressure on the Curtis element is 130 psia. The exhaust pressure of the turbine is 1.5 in. Hg abs; rpm, 3,600; available energy lost through friction in nozzles, 6 per cent; steam consumption, 130,000 lb per hr. The radiation loss from the turbine is 1,040,000 Btu per hr. (*a*) Assuming that the steam makes a 180-deg turn in the blades, what should be the pitch diameter of the Curtis element? (*b*) What is the actual thermal efficiency? (*c*) What is the heat content of the steam entering the condenser in Btu per pound?

22. It is proposed to construct a 20,000-kw turbine to operate on a pressure of 450 psia and 740 F temperature of steam and with a back pressure of 1 in. Hg abs. This gives an available heat of 500 Btu per lb of steam. The turbine will turn 3,600 rpm and will consist of one 2-row Curtis first stage and 16 Rateau stages, all with the same pitch diameter. An engine efficiency of 72 per cent of Rankine cycle efficiency is expected. The buckets will reverse the flow 180 deg; consider them frictionless. Determine (*a*) the pitch diameter of the buckets; (*b*) the pounds of steam used per hour; (*c*) the area of the discharge end of a first-stage nozzle if each nozzle carries 1 lb of steam per sec. (Consider the nozzle frictionless.)

23. Given a single-pressure, single-velocity stage turbine operating on 150 psia and 500 F temperature; back pressure atmospheric. Calculate the approximate pitch diameters of the wheel with rotative speeds of 10,000 and 3,600 rpm. (Assume 180-deg reversal of flow and frictionless buckets and nozzles.)

24. Referring to Fig. 258, determine the percentage of moisture in the steam at the adiabatic end points G and F, either by the Mollier chart or by calculation.

25. Referring to Fig. 259, determine the percentage of moisture in the steam at the adiabatic end points Q, H, and K.

26. Steam is supplied to a turbine at a pressure of 350 psia and 700 F temperature, with an exhaust pressure of 1.5 in. Hg abs. At a certain load, the governor throttles the steam to a pressure of 250 psia at entrance to the turbine. What percentage of the available energy of the steam is lost by throttling?

27. Given a single-pressure, single-velocity stage De Laval steam turbine, gear-connected to a generator. The steam pressure is 130 psia; quality of steam, 100 per cent; vacuum in condenser, 22 in. Hg; load at switchboard, 30 kw; rpm, 22,000; barometer, 29.5 in. Hg; steam consumed per hour, 1,000 lb. (*a*) Determine the actual thermal efficiency of the unit, based on output of the generator. (*b*) Assuming that 3 kw are required to overcome friction in the turbine, gears, and generator, and neglecting radiation losses, determine the quality of the exhaust steam. (*c*) If the turbine has four nozzles and steam line pressure acts on the entrance side, what should be the throat area of the nozzles for this load?

28. Steam leaves a nozzle at 700 fps at an angle of 20 deg to the blade wheel and leaves the blades at an angle of 30 deg to the direction of rotation.

The wheel is 36 in. in diameter; rpm, 1,800; steam consumption, 7,200 lb per hr. The steam loses 10 per cent of its relative velocity in the blades due to friction. Determine (*a*) the force on the blades in pounds; (*b*) the hp developed by the wheel.

	20 deg	30 deg
sin	.34202	.5000
cos	.93969	.86603
tan	.36397	.57735

29. A certain power plant consists of two 5,000-kw turbine-generator units, each of which uses 12 lb of steam per kwhr at full load. The steam pressure and temperature are 600 psia and 800 F; exhaust pressure 1 in. Hg abs. Various auxiliaries about the plant require 450 kw. Determine the cycle efficiency for the same net plant output (*a*) if these auxiliaries are electrically driven from the main generators (assume no electrical losses and a feed-water temperature of 70 F); (*b*) if these auxiliaries are steam-driven and the exhaust steam is used for feed-water heating. The water rate for these auxiliary engines is 25 lb of steam per kwhr and they exhaust into an open feed-water heater at atmospheric pressure. The feed-water temperature out of the heater is 186 F.

30. Assume the following data for a regenerative feed-heating cycle: steam supply pressure, 450 psia; temperature, 720 F; first bleeder pressure, 90 psia; second bleeder pressure, 15 psia; exhaust pressure, 1 in. Hg abs; condensate temperature, 70 F; water temperature out of first heater, 210 F; feed temperature out of second heater, 315 F. (Assume constant-entropy expansion of steam.) For 1 lb of condensate (*i.e.*, steam to condenser) determine (*a*) the pounds of steam bled to the first heater; (*b*) the pounds of steam bled to the second heater; (*c*) the total steam to the turbine; (*d*) the available heat of each portion of the steam supplied; (*e*) the total available heat per pound of steam supplied for regenerative cycle; (*f*) the cycle efficiency; (*g*) the Rankine-cycle efficiency for the same supply and exhaust pressures.

31. A steam plant operates on the reheat cycle using high-pressure saturated steam as the reheating agent. Steam is supplied to the high-pressure turbine at 615 psia and 730 F and is exhausted at 115 psia. It is then reheated to 485 F at constant pressure and expanded in the low-pressure turbine to 1 in. Hg abs. In each turbine 79 per cent of the available energy is converted to work on the shaft. (*a*) Determine the moisture content of the exhaust and the cycle efficiency. (*b*) If the reheater were to be by-passed so that no reheat occurred, what would the moisture content and cycle efficiency be?

CHAPTER XI

CONDENSERS

227. Condensers.—The primary function of condensers in steam-power plants is to reduce the back pressure at the exhaust of steam engines and steam turbines. Frequently the recovery of the condensed steam is equally important. Condensation of the exhaust steam is accomplished by the use of cooling water, which is either brought into contact with the exhaust steam, as in *jet condensers,* or is kept separate from the steam by metallic walls, as in *surface condensers.* Condensation of the steam results in a decrease in temperature and pressure, and a great decrease in its volume, thus producing the required partial vacuum. This partial vacuum makes it possible to expand the steam still further, thus increasing the power developed per pound of steam and reducing the pounds of steam required per brake horsepower per hour, per indicated horsepower per hour, or per kilowatt-hour. Vacuum is most conveniently and generally measured in inches of mercury.

228. Jet Condensers.—There are two principal types of jet condensers: *low-level jet condensers* and *barometric condensers.* In the low-level jet condensers, the condensing chamber is at a low level and the condensed steam, water, air and noncondensable gases are removed by a pump, or by the kinetic energy of water jets. In the latter case the condenser is known as an *ejector* or *siphon condenser.* In the barometric type, the condenser is placed high enough above the point of water discharge so that the steam, water, and air are removed by gravity. Both the low-level and the barometric condensers may be built as *low-vacuum* or as *high-vacuum* condensers. In the low-level, low-vacuum type the water, condensate, and noncondensable gases are removed by a single pump; if the noncondensable gases are removed by a separate pump, higher vacuums may be obtained and the condenser is known as a low-level, high vacuum type.

Low-vacuum condensers are designed to produce ordinarily vacuums not in excess of 26 in. of mercury and are generally used with reciprocating steam engines. High-vacuum condensers are designed ordinarily to produce vacuums from 26 in. of mercury up to the highest obtainable and are always used in the highest class of steam-turbine installations.

Figure 268 shows a low-level, low-vacuum jet condenser. The exhaust steam from the engine or turbine enters at A, and the

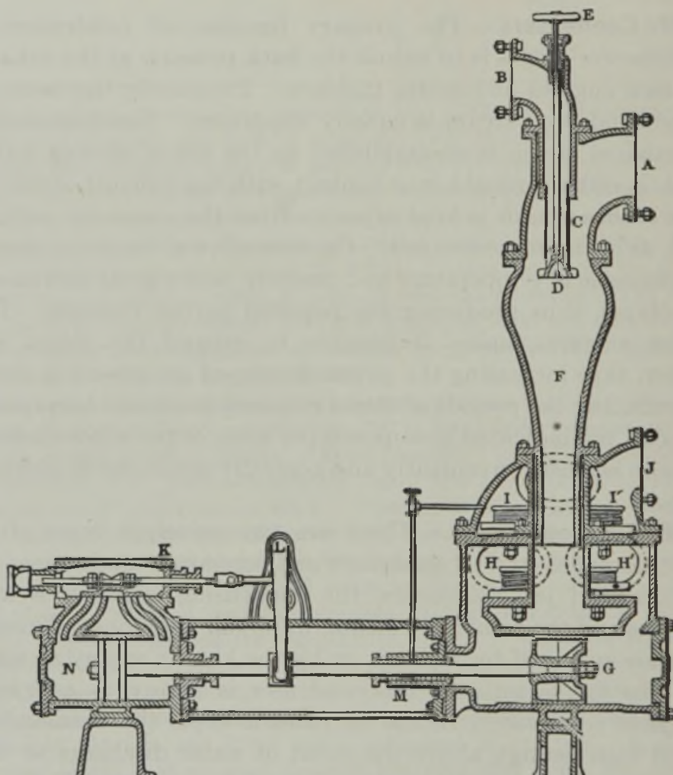


FIG. 268.—Low-level, low-vacuum jet condenser.

injection or cooling water at B. These are mixed in the combining chamber F. The cooling water enters the combining chamber in the form of a spray which is produced by a spray nozzle D on the end of the injection pipe. The condensation of the steam reduces its volume many times, and this reduction of volume produces a vacuum in the chamber F. This vacuum is continuously maintained by the pump G, which removes the condensing water and the condensed steam together with air which is always present.

The condenser, illustrated in Fig. 268, has a direct-acting, double-acting steam cylinder N, direct connected to a direct-acting water and air cylinder G. In the position shown, both pistons are assumed to be moving toward the left, thus forcing the water in the crank end (left-hand end) of the water and air cylinder out through valve I into the discharge pipe J and holding valve H closed. At the same time the suction in the head end (right-hand end) of the water and air cylinder closes valve I' and opens valve H', drawing in the water and the air from the combining chamber F. If the source of the condensing water is not more than about 15 ft below point B, it is possible to draw the water into the condenser by the vacuum in the chamber F and produce a satisfactory spray; otherwise a water supply pump must be used.

When this type of condenser is designed as a low-level, high-vacuum-type jet condenser, a separate air pump is supplied for withdrawing the air from the condensing chamber F through a suitable opening, the condensing water and condensed steam being removed by a pump such as is shown in Fig. 268, or by a centrifugal pump driven in any manner.

A low-level multijet condenser is shown in Fig. 269. It consists of a closed cylindrical body in the upper end of which is a water-nozzle case containing the nozzle plate and water nozzles. Below the nozzle case is a combining tube consisting of several sets of tapering rings joined together by equally spaced ribs. The lower end of the combining tube, or of a tail pipe connected to it, is immersed in a hot well.

"The injection water is delivered under pressure into the nozzle case of the condenser and passed through the nozzles, which are accurately designed to discharge a specified amount of water according to the conditions of flow determined by the water pressure and the guaranteed vacuum. As the water jets emerge from the nozzles, they are directed through the combining tube and into the throat of the tail diffuser, where they unite to form a single jet. The exhaust steam enters the condenser chamber through the exhaust-steam inlet, flows through the annular passes of the combining tube, comes into direct contact with the converging water jets, and is condensed. Owing to the combined effect of the external water pressure, the vacuum existing in the condenser, and gravity, the water jets attain a velocity sufficient to entrain the condensed steam, air, and noncondensable gases, and to discharge them into the hot well against the pressure of the atmosphere. *The water jets create*

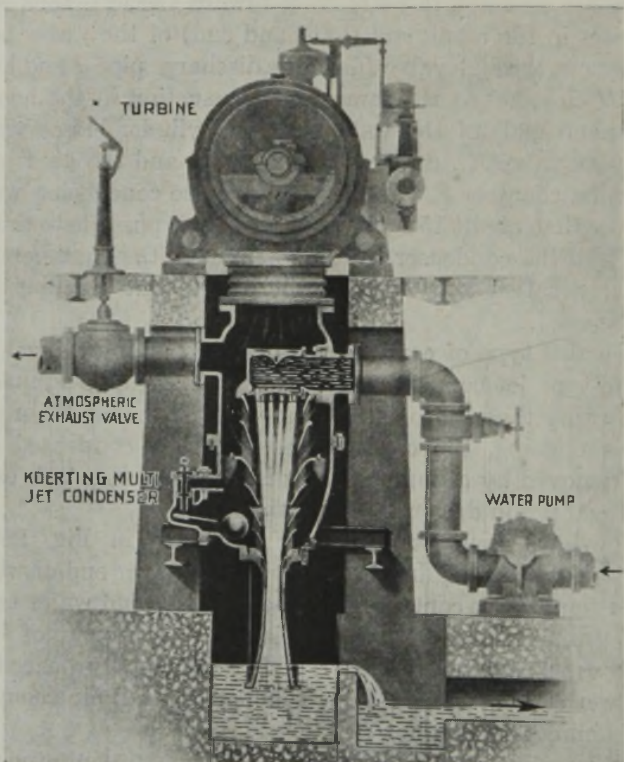


FIG. 269.—Schutte and Koerting multijet ejector condenser installed under a turbine.

the vacuum by condensing the steam, and maintain it by entraining and removing the air and noncondensable gases. No separate vacuum pumps are needed.

"A float-operated vacuum breaker is provided to admit air to the vacuum space when there is a back flow of water from the hot well caused by shutting down the injection water. This breaker gives ample protection against flooding turbine casings or engine exhaust lines."

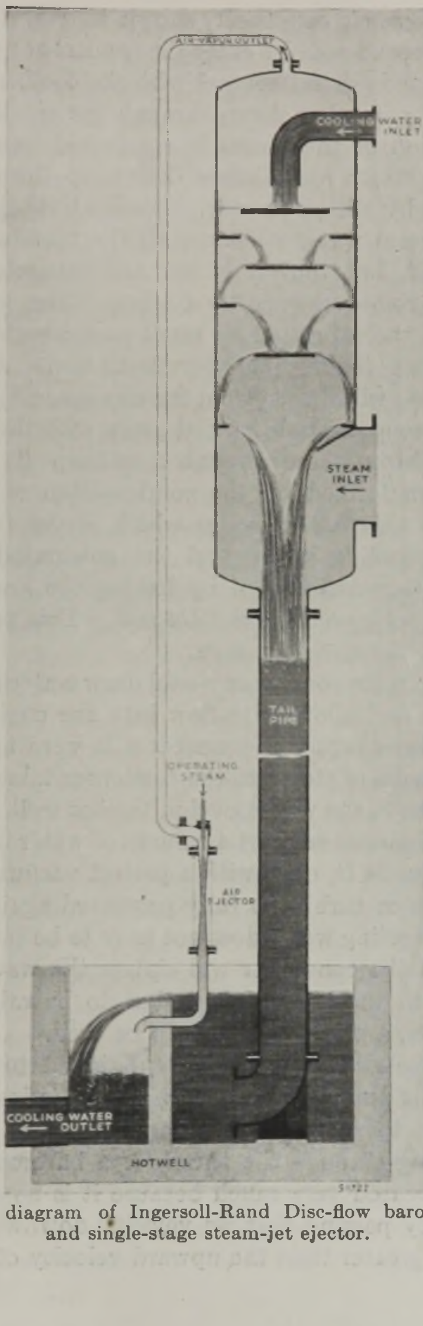


FIG. 270.—Flow diagram of Ingersoll-Rand Disc-flow barometric condenser and single-stage steam-jet ejector.

"In the *barometric* condenser, shown in Fig. 270, steam and air or other noncondensables enter the condenser near the bottom where they come in direct contact with the cooling water flowing by directed gravity flow down through the condenser. A considerable portion of the steam is condensed immediately, and the remaining steam and air are drawn up through successive water curtains by the air ejector. By the time the mixture has penetrated several water curtains, all the remaining steam has been condensed, but the air is hot and saturated with vapor which can be removed only by cooling. The upper curtains, through which the saturated air must pass, are designed to give the best possible cooling and devaporization. After thorough cooling, the air is withdrawn from the condenser by the steam-jet ejector. The condensate is carried away with the cooling water and drains to atmosphere through a tail pipe." The vacuum, which has been formed by the condensation of the steam, is maintained by this fall of water which serves to remove continuously the cooling water and the condensed steam. The vacuum is further maintained by having the lower end of the discharge pipe submerged in the *hot well.* This prevents admission of air into the discharge pipe.

The vacuum in the condenser would draw water up through the discharge pipe and allow it to flow into the engine cylinder or turbine with disastrous consequences if it were not for the fact that the admission of steam to the condenser takes place at least 35 ft to 40 ft above the water level in the hot well. Since atmospheric pressure cannot support a column of water in the discharge pipe higher than 34 ft, even with a perfect vacuum, it is evident that the engine or turbine is fully protected against this occurrence. If the cooling water does not have to be raised over 18 ft, the vacuum in the condenser will siphon the water in and mix it properly with the steam. If it has to be raised through a greater distance, a pump will have to be used.

When barometric condensers are designed to produce high vacuums, an air pump for removing the air is installed close to the point where the steam and cooling water mix. The quantity of water passing through the throat of a barometric condenser cannot be decreased very much because it is necessary to have such a quantity passing that its velocity downward in the discharge pipe is greater than the upward velocity of the entrained air bubbles. If this condition is not maintained, it is impossible to maintain a vacuum. For this reason the barometric condenser is not adapted to variable loads and requires much more cooling water at light loads on the engines, or turbine, than other types.

Jet condensers are relatively low in first cost and require less cooling water than surface condensers. The most advanced designs are capable of producing the highest vacuums attainable in steam-power-plant practice. Jet condensers operate more satisfactorily with dirty cooling water than do surface condensers. They have the disadvantage of mixing the condensed steam with the cooling water, thus making it impossible to reclaim the condensate as a source of boiler feed water.

229. Surface Condensers.—In a surface condenser (Fig. 271) the steam to be condensed and the condensing water do not come in direct contact with each other. The water is circulated on one side of a series of tubes by means of the circulating pump and the steam is condensed by coming in contact with the other side of the tubes. In the "standard" type of surface condenser, which is the form in which the vast majority are built, the condensing water passes through the tubes. In the "water-works" type, ordinarily found only in water-works pumping stations, the condensing water flows around the outside surface of the tubes and the steam is condensed within them.

The tubes in a surface condenser are always of small diameter and as thin as possible. They are usually made of special brass or suitable alloys. Surface condensers are used where the cooling water is not suitable for boiler feed, and where it is desirable to reclaim the condensed exhaust steam for this purpose. Condensed steam, being distilled water, contains no scale-forming material and thus is an excellent boiler feed water. When surface condensers are used with reciprocating steam engines, it is necessary to see that none of the lubricating oil in the exhaust steam is allowed to go back to the boilers. With surface condensers, the character of the cooling water is relatively immaterial, provided it is reasonably clean and noncorrosive, as none of it will be used for feed water. Surface condensers are always used in salt-water marine practice and nearly always in land service where salt water is used for condensing.

Surface condensers may operate on the *wet-vacuum* system or on the *dry-vacuum* system.

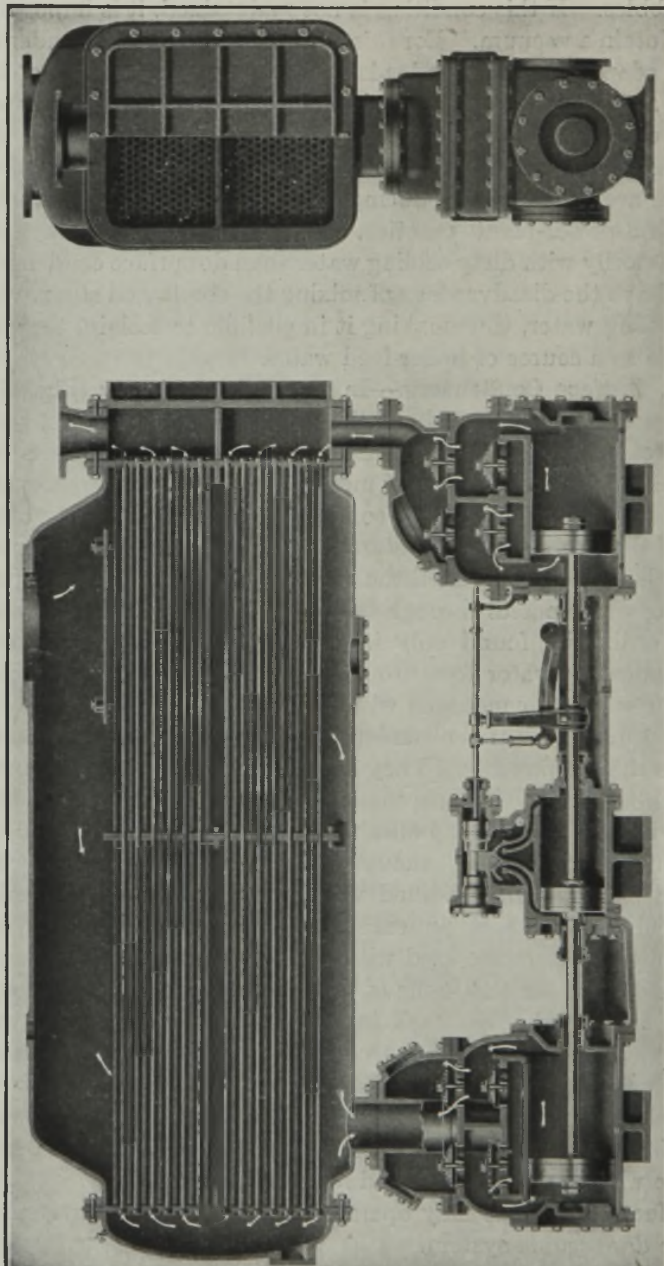


FIG. 271.—Surface condenser operating on wet-vacuum system.

For operation on the wet-vacuum system two pumps are required: a circulating or cooling-water pump, and a so-called *wet-air pump* which removes both the air and the condensed steam from the condenser. For operation on the dry-vacuum system three pumps are required: a circulating or cooling-water pump, a condensate pump for removing the condensed steam, and a *dry-air pump* for removing the air. Circulating water pumps may be of the piston or plunger type, but are nearly always of the centrifugal type and are usually driven by electric motors or steam turbines. Wet-air pumps may be of the steam-driven, direct-acting, reciprocating piston type, or special rotary pumps. Dry-air pumps may be of the reciprocating piston type, rotary type, hydraulic-entrainment type, or they may be of the steam-jet type, single or multistage. Any form of drive may be used for the above noted pumps with the exception of the steam-jet pumps. Steam-jet pumps are now almost universally used because of their low first cost, absence of moving parts, light weight, large air-handling capacity, and ability to produce and maintain extremely high vacuums. When supplied with inter- and after-surface condensers nearly all of the heat in the steam used by the jets is returned to the boiler feed water, thus assuring economical operation.

The wet-vacuum type of surface condenser, shown in Fig. 271, has a reciprocating piston type of circulating water pump, shown at the right. The cylinder at the left is also of the reciprocating piston type and is the wet-air pump and handles both the condensed steam and the air. The middle cylinder is the steam cylinder for operating both pumps.

The wet-vacuum system of operation is satisfactory for vacuums not exceeding about 26 in. of mercury but for the highest vacuums the dry-vacuum system is far preferable because the enormous volumes of air at extremely low pressures are much more easily and satisfactorily removed by the separate air pump.

Figure 272 shows a 20,000-sq-ft two-pass Foster-Wheeler surface condenser; Fig. 273 is a Foster-Wheeler surface condenser which incorporates the cross-flow condenser principle in which the steam flow through the tube banks is generally horizontal. It offers the advantages of large inlet area, shallow tube bank, and low pressure drop, giving uniform depth of penetration along the entire tube length and uniform distribution of work. Figure

274 is a 27,000-sq-ft two-pass condenser built by the C. H. Wheeler Manufacturing Company.

230. Air Pumps.—One of the greatest obstacles to successful condenser performance is the leakage of air into the vacuum space. Some of this air is dissolved in the feed water and is carried through with the steam; a large quantity is added by leaks around the piston rod and valve rod in the low-pressure cylinder when reciprocating engines are used, and past the shaft

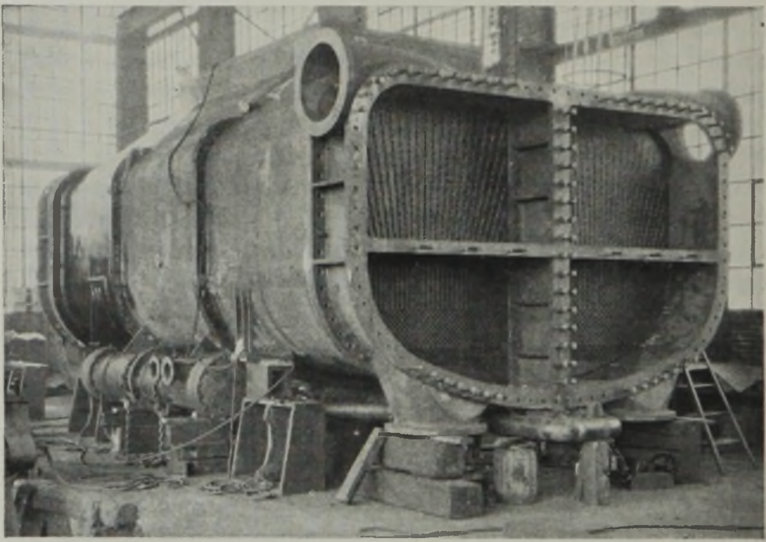


Fig. 272.—20,000-sq-ft Foster-Wheeler two-pass surface condenser with divided water box.

seals of turbines; some air enters through leaks in the exhaust-line connections and joints.

In jet condensers an additional source of air, larger than any of the others, is the cooling water. This always contains a considerable quantity of absorbed air, much of which is released when the cooling water is sprayed into the evacuated condensing chamber. Owing to the low pressure, this released air expands to very large volumes and, since air is noncondensable, the air pressure would build up in the condenser and the vacuum would soon be lost. Consequently, this air must be removed continuously by means of air pumps. It should be borne in mind that the vacuum is formed by the condensation of the steam.

whereby a large volume of steam is converted into a relatively very small volume of water. All that the air pump does is to remove the air that is continually coming in and tending to destroy the vacuum. Since the presence of air in a condenser reduces its efficiency and requires the expenditure of work to remove it, it is extremely desirable to eliminate all leaks as far as

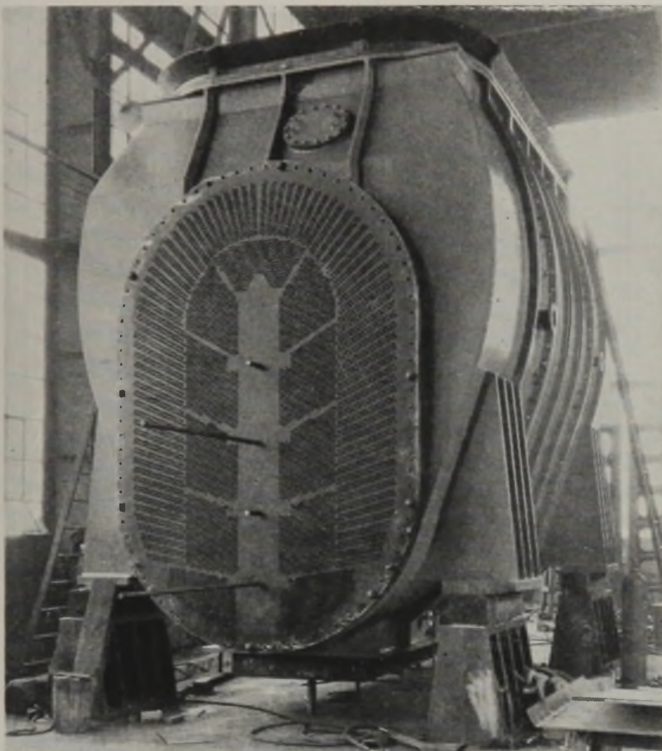


FIG. 273.—Foster-Wheeler 32,500-sq-ft surface condenser.

possible. Entry of air by way of the steam and the cooling water cannot be avoided.

The capacity of the air pump depends upon the temperature of the air, or the mixture of air and water, which it must withdraw from the condenser, as well as upon the amount of air leakage into the condenser. For this reason it is desirable to provide for the air removal at the point where the air has the lowest temperature, and consequently the greatest density and smallest volume.

On the other hand, since, when a surface condenser is used, one of the objects is to recover the condensate for use as boiler feed water, it is, in such installations, most economical to keep the temperature of this condensate as high as possible. Where *wet-air* pumps are used, this necessitates a compromise between cold air and hot condensate; where a *dry-air* pump is part of the equipment, this is not necessary. In the latter case the

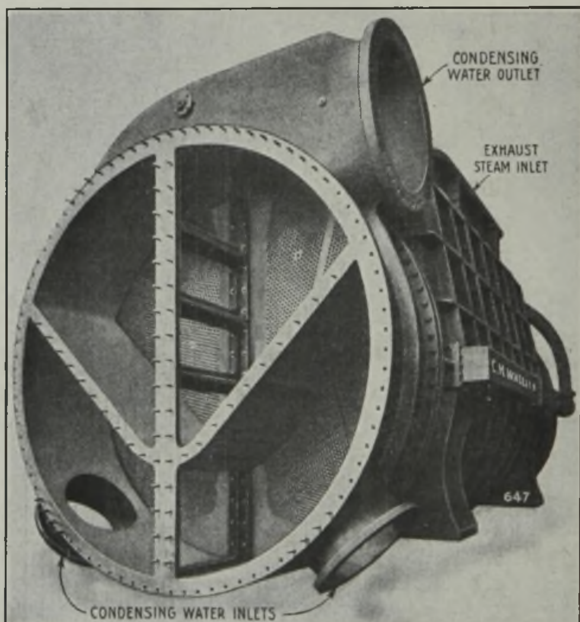


Fig. 274.—C. H. Wheeler Manufacturing Company two-pass condenser.

wet pump handles the condensate only; the dry pump handles the air that has been "refrigerated" by being exposed to the condensing water before this water has become heated by the steam.

With surface condensers the condensed steam as *condensate* is usually discharged into a *hot well* and then pumped from there to feed-water heating devices. In the case of jet condensers the condensed steam is discharged with the cooling water.

231. Amount of Cooling Water.—The amount of cooling water required in a condenser depends upon the temperature of the water and the degree of vacuum desired. If the temperature in the condenser is too high, low vacuum cannot be obtained, as the pressure in the condenser cannot be less than the pressure corresponding to the temperature of the boiling point in the condenser. If the temperature of the water in the hot well is 120 F, the corresponding pressure as given in the steam tables is 1.7 lb, which is the lowest possible pressure that can be obtained. If a lower vacuum is desired, the temperature of the water leaving the condenser must be lowered. This can be done in two ways: by increasing the amount, or by decreasing the temperature, of the cooling water.

Let

t_1 = the initial temperature of the cooling water;
t_2 = the final temperature of the cooling water (and, in a jet condenser, of the condensed steam also);
t_3 = the temperature of the condensed steam leaving a surface condenser;
h = heat (above 32 F) in each pound of steam entering the condenser;
W = the weight in pounds of cooling water entering per minute;
w = the weight in pounds of steam condensed per minute.

Then the heat given up by the steam in a *jet* condenser

$$= w[h - (t_2 - 32)] \tag{1}$$

and the heat received by the water

$$= W(t_2 - t_1). \tag{2}$$

But these two expressions must be equal; by equating and solving

$$W = \frac{w[h - (t_2 - 32)]}{t_2 - t_1}. \tag{3}$$

Substituting t_3 for t_2 in Eq. (1), Eq. (3) becomes

$$W = \frac{w[h - (t_3 - 32)]}{t_2 - t_1}, \tag{4}$$

which is the expression for the weight of cooling water used to condense w lb of steam per minute in a *surface* condenser.

The amount of cooling water per pound of steam entering a surface condenser is larger than that used in a jet condenser, as the temperature of the condensed steam is lower than the temperature of the cooling water leaving the condenser.

In ordinary stationary practice, 1 sq ft of cooling surface is allowed for every 10 lb of steam condensed per hr, except in turbines using high vacuum, where 1 sq ft is allowed for every 4 lb to 8 lb of steam. In navy practice, from 1 sq ft to $1\frac{1}{4}$ sq ft of surface are allowed for every indicated horsepower.

Example.—The initial temperature of the cooling water entering a jet condenser is 55 F and the temperature of the discharge from the condenser is 105 F. The engine takes steam at 140 psia and exhausts into the condenser where the vacuum is 25.8 in. Hg. The barometer reading is 29.9 in. Hg. The engine developing 300 ihp has an economy of 20 lb per ihp-hr. Assuming no loss due to radiation, find (*a*) the number of gallons of cooling water flowing through the condenser per hour; and (*b*) the quality of the steam entering the condenser.

Solution.—(*a*) The heat given up by steam doing work in the engine

$$= \frac{2{,}545}{20} = 127.25 \text{ Btu per lb.}$$

The heat in the steam leaving the engine and entering the condenser

$$= 1{,}193.0 - 127.25 = 1{,}065.75 \text{ Btu per lb.}$$

The total steam used by the engine per hour

$$= 300 \times 20 = 6{,}000 \text{ lb.}$$

From Eq. (3) the weight of cooling water used by the condenser per hour

$$= \frac{6{,}000[1{,}065.75 - (105 - 32)]}{105 - 55} = \frac{6{,}000 \times 992.75}{50}$$
$$= 119{,}130 \text{ lb.}$$

The water used by the condenser

$$= \frac{119{,}130}{8\frac{1}{3}} = 14{,}295 \text{ gal per hr.}$$

(*b*) The heat in the steam entering the condenser

$$= h_f + xL = 1{,}065.75 \text{ Btu per lb,}$$

where h_f and L correspond to the pressure in the condenser. The absolute pressure in the condenser

$$= (29.9 - 25.8) \times 0.491$$
$$= 2.01 \text{ lb.}$$

Therefore,

$$x = \frac{1{,}065.75 - 94.0}{1{,}022.2} = \frac{971.75}{1{,}022.2} = 0.951$$

or 95.1 per cent.

232. Condensers for Reciprocating Engines.—Condensing the exhaust steam from a reciprocating engine diminishes the back pressure and consequently increases the mean effective pressure. In the expression for mean effective pressure,

$$\text{mep} = e\left[p_1\left(\frac{1 + \log_e r}{r}\right) - p_2\right],$$

the quantity affected by the vacuum is the term p_2. In a noncondensing engine this is usually about 15 psia, but in a condensing engine the effect of adding the condenser is to lower p_2 to about 2 psia. With a single-cylinder engine taking steam at a pressure of 100 psia and cutting off at one-fourth stroke, this means an increase in the mean effective pressure of from about 40 lb to about 53 lb, or an increase of approximately 30 per cent in the indicated horsepower developed by the engine. In engines formerly operated noncondensing and later connected up so as to exhaust at atmospheric pressure into a mixed- or low-pressure turbine, and there expanded to 28 in. of vacuum or more, 75 per cent or more has been added to the power derived from the engine.

It is not practical to handle in a reciprocating engine the large volumes assumed by steam at the lower pressures; furthermore, after a certain point, the increased initial condensation losses accompanying a further decrease in back pressure may more than offset any gain from additional expansion, and a vacuum of about 26 in. is all that is ordinarily desired. With the uniflow engines, somewhat higher vacuums can be used advantageously, as in these engines the initial condensation loss is practically eliminated, and exhaust ports of very large area can be provided.

233. Condensers for Steam Turbines.—In turbine plants an increase in the vacuum increases the economy of the turbine materially; consequently, every means is used to get the highest possible vacuum. There is no alternate heating and cooling of the wall in a turbine, and reducing the pressure and the temperature of the exhaust does not increase the condensation. Furthermore, the turbine can take care of the large volume of steam accompanying low pressures. The result has been that many recent turbine plants operate with a vacuum of 28 in. or 29 in., which means an absolute pressure in the condenser of 1 in. Hg or less.

In most steam-turbine plants, surface condensers are used, principally for the reason that the exhaust from the steam turbine does not contain oil, and when condensed is an ideal feed water, as it contains no scale-producing matter. It is also possible in a surface condenser to use very large quantities of circulating water, and thus reduce the temperature of the condenser.

234. Cooling Towers.—"Power plants operating condensing and process plants that use water for cooling require relatively large quantities of water, ranging from 20 lb to 60 lb per lb of

Fig. 275.—Spray pond.

steam condensed. If the water is thrown away after it is used, a continuous supply of cheap and fresh water is indispensable.

"Where plants are located near a natural supply of water, the cost of the water consumed is merely that of pumping. For many plants, however, the only source of supply is the local city water main. The expenditure involved in the purchase of this water is prohibitive; it may in fact exceed the saving effected by the use of the condensing apparatus.

"Hence it is necessary in many plants to recool a fixed supply of fresh water, and to use it over and over again. Thus the benefits of operating engines and turbines with condensers can be economically obtained in places where a continuous supply of fresh water would be inadequate or its cost excessive, where river water is impure, or where water from sources near the ocean is brackish or affected by the tides." The process of recooling the water is accomplished generally by means of cooling ponds or cooling towers.

The method first used was to discharge the hot water into a pond of sufficient area so that the water would be cooled by the air passing over the surface of the pond. This meant ponds of

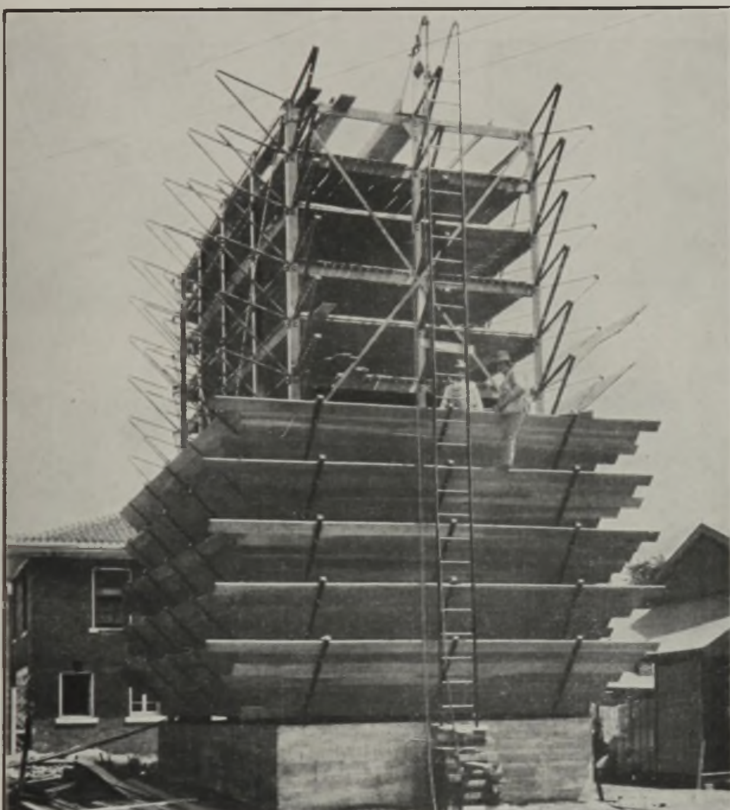


FIG. 276.—Atmospheric cooling tower in process of erection.

large size, and, in order to reduce the area necessary, the water was sprayed into the air over the pond (Fig. 275), thus increasing the rate of cooling.

The impossibility of avoiding certain objectionable features inherent in the spray pond led to the development of the cooling tower. In this device, the water is delivered to the top of the tower and then allowed to drop to a tank below, the water being broken into spray during the fall.

Three types of cooling towers are in common use: the *atmospheric-draft* (Fig. 276), the *natural-draft* (Fig. 277), and the *forced-draft* (Fig. 278) tower. In atmospheric-draft towers the air circulates through the tower due to atmospheric air currents; in natural-draft towers the circulation is brought about by the difference in pressure between the air inside and

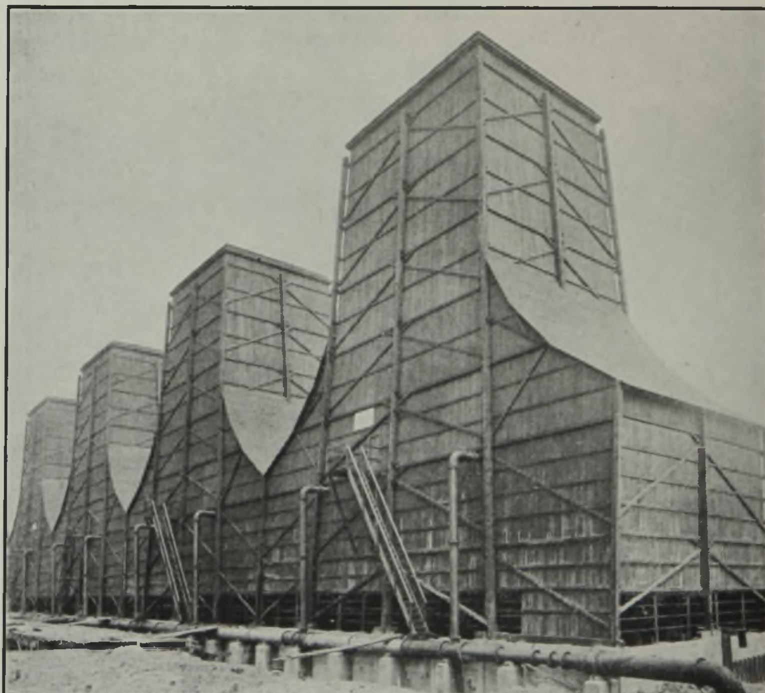


Fig. 277.—Large Foster-Wheeler natural-draft cooling-tower installation with a capacity to cool 25,000,000 gal of water per day.

outside the stack just as it is in the case of a chimney; and forced-draft towers depend upon fans to create the air circulation.

The atmospheric tower should be located where it will receive the benefit of all breezes, as for example on the roof of the plant, or in a place where it is not closely surrounded by buildings.

In a natural-draft tower, the sides are solid walls, instead of being open as in the atmospheric towers, but the bottom is left open to allow the air to enter. The top is in the shape of a high stack and the warm air inside the tower rises, thus providing a draft. This type of tower is used chiefly in large sizes.

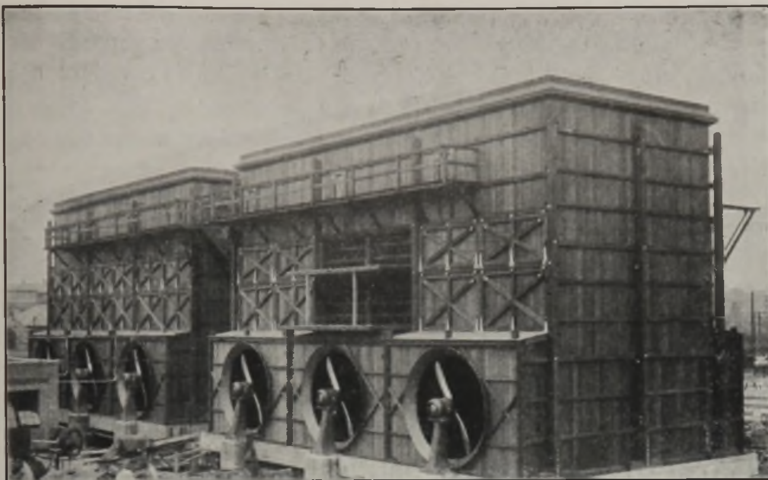


FIG. 278.—Forced-draft cooling tower at a plant of the Procter & Gamble Company.

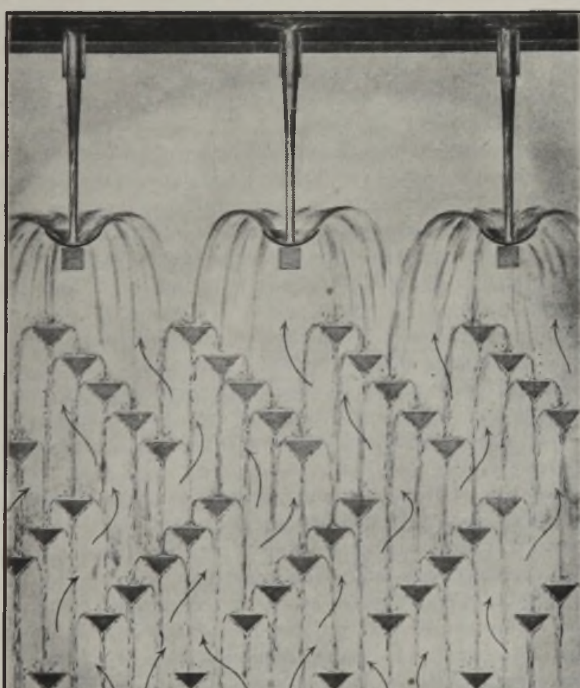


FIG. 279.—Cooling water distribution and tower filling.

The forced-draft tower also has solid sides, of either wood or metal, but has a fan at the bottom for forcing air through the tower. It is the most compact and, relative to its capacity, the lightest of the three types. A modification of this type is the induced-draft tower in which the fan is placed at the top and draws air through the tower.

The water distribution in the Foster-Wheeler towers consists of open troughs from which water falls through pipe nipples of brass splash plates, as shown in Fig. 279. Beneath the splash plates is a filling used to retard the passage of the water through the tower. In the atmospheric tower, where the air flow is horizontal, this filling consists of closely spaced splash boards, as shown in Fig. 276; in the enclosed types of tower, in which the air flow is vertical, the filling consists of triangular members.

PROBLEMS

1. A 700-ihp triple-expansion engine using superheated steam at a pressure of 150 psia exhausts into a jet condenser having a vacuum of 23 in. Hg. The cooling water enters at 70 F and leaves at 100 F. (*a*) How much cooling water in gallons per hour would be required, if the engine has an economy of 15 lb of dry steam per ihp-hr and the steam entering the condenser is dry and saturated? (*b*) How many degrees of superheat are in the steam entering the engine?

2. A steam turbine shows a thermal efficiency of 16.8 per cent when supplied with steam at a pressure of 250 psia and superheated to a total temperature of 480 F. The vacuum in the surface condenser is 28.79 in. Hg referred to a 30-in. Hg barometer. If the condensate is 3 deg lower than the exhaust-steam temperature, how many pounds of cooling water will be required per pound of steam if the rise in temperature of the cooling water is 12 deg?

3. A 300-hp engine uses 18.18 lb of steam per hp-hr when running 150 rpm and exhausting into a condenser where the vacuum is 26.46 in. Hg. The barometer reads 29.5 in. Hg. Steam enters the engine at a pressure of 150 psia and containing 100 deg of superheat. The cooling water enters the condenser at 60 F and leaves at 105 F. The temperature of the condensed steam is 95 F. Assume no loss due to radiation. Find (*a*) the gallons of cooling water used per hour; (*b*) the quality of the exhaust.

4. A 500-ihp compound Corliss engine uses 14.97 lb of steam per ihp-hr when exhausting into a surface condenser which maintains a vacuum of 27.5 in. Hg. Steam enters the engine at a pressure of 160 psia and a temperature of 470 F. Cooling water enters the condenser at 55 F and leaves at 100 F. The temperature of the condensed steam is 92 F, the barometer is 30 in. Hg. Assume that there is no radiation loss. Find (*a*) the capacity of the cooling-water pump in gallons per hour; and (*b*) the quality of the exhaust steam.

5. In a jet condenser, the initial temperature of the cooling water is 60 F, and the final temperature 100 F. Dry steam enters the engine at a pressure of 150 psia and the condenser at a pressure of 2 psia. The engine develops 200 ihp and uses 26 lb of steam per ihp-hr. Assuming no loss due to radiation, find (*a*) how many gallons of condensing water must be used per hour; (*b*) the quality of the exhaust steam; and (*c*) the actual thermal efficiency of the engine based on indicated horsepower.

6. The following data were taken from the test of a condenser attached to a 30,000-kw turbine: duration of test, 3 hr; steam condensed per hour, 317,000 lb; steam pressure, 300 psia; superheat, 142.67 deg; vacuum, 28.17 in. Hg; initial temperature of cooling water, 76.5 F; final temperature of cooling water, 86.9 F; temperature of condensate, 91.3 F; barometer, 29.93 in. Hg. Assuming no loss due to radiation, find (*a*) the quality of steam in the exhaust; (*b*) the capacity of the circulating pump in gallons per hour. (Caution: Remember that steam in passing through a turbine does work.)

7. A condensing engine developing 250 ihp uses 20 lb steam per ihp-hr. The vacuum in the surface condenser is 26 in. Hg; the barometer reading is 30 in. Hg. The cooling water enters the condenser at a temperature of 50 F and leaves at 100 F. The temperature of the condensate is 120 F. Find the capacity, in gallons per hour, of the pump that supplies the cooling water (*a*) assuming the quality of the exhaust to be 100 per cent; and (*b*) assuming that the steam entering the engine at a pressure of 150 psia is dry and saturated. (*c*) In case (*b*), what is the quality of the exhaust? Assume that there is no radiation loss in either case.

8. A 10,000-kw turbogenerator uses 13 lb of steam per kwhr delivered at the generator terminals when operating at full load. Steam is supplied at a pressure of 275 psia and at a temperature of 600 F. The vacuum in the condenser is 28.2 in. Hg when the barometer reads 29.7 in. Hg. Cooling water enters the condenser at 60 F and leaves at 72 F. Condensate leaves at 75 F. The mechanical efficiency of the unit is 92 per cent. Determine (*a*) the number of Btu per pound of steam, that are converted into work; (*b*) the quality of the exhaust steam entering the condenser (radiation losses are to be neglected); (*c*) the number of pounds of condenser cooling water per pound of steam.

9. A 10,000-kw turbine requires 14 lb of steam per kwhr at rated load. Steam is supplied at a pressure of 200 psi and 540 F temperature. Feed water is supplied to the boiler at 180 F. The turbine exhausts into a condenser in which the vacuum is 27.03 in. Hg. The barometer reading is 28.53 in. Hg. Cooling water for the condenser enters at 58 F and leaves at 78 F. The temperature of the condensed steam is 4 deg below the temperature corresponding to the vacuum. The coal contains 13,000 Btu per lb as fired. The over-all efficiency of the boiler unit is 78 per cent. Find (*a*) the pounds of cooling water per pound of steam condensed; (*b*) the pounds of coal per kilowatt-hour at rated load; (*c*) the boiler horsepower at rated turbine load.

10. A 10,000-kw turbine is connected to a surface condenser. It uses 15 lb of steam per kwhr. Steam is supplied at a pressure of 190 psia and superheated 100 deg. The radiation losses from the turbine amount to

20 per cent of the heat chargeable to the turbine. The pressure in the condenser is 2 in. Hg abs. The cooling water enters at 60 F and leaves at 90 F. The temperature of the condensate is 96 F. Determine the amount of cooling water used in gallons per minute.

11. How many pounds of cooling water would be required per hour by a surface condenser that serves a reciprocating engine delivering 45 bhp? Steam is supplied at a pressure of 130 psia and at a quality of 98 per cent. The back pressure is 1 psia. The mechanical efficiency of the engine is 90 per cent. Heat lost by radiation from the cylinder is 20,000 Btu per hr. The engine consumes 1,000 lb of steam per hr. The temperature of the condensate is 96 F. The cooling water enters the condenser at 55 F and leaves at 95 F.

12. Twenty-five hundred pounds of steam per hour with an enthalpy of 1,050 Btu per lb are exhausted from a steam engine into a low-level, barometric, jet condenser. Cooling water is supplied at 72 F and leaves at 92 F. A velocity of water in the tail pipe of 10 fps is necessary to ensure that the air bubbles are carried out with the water instead of rising to the condenser head. What is the diameter of the tail pipe?

13. A 24-in. and 36-in. by 36-in. cross-compound engine drives a generator. This unit is operated on a pressure of 125 psi and exhausts into the heating system carrying a pressure of 8 psi. With one-fourth cutoff in the high-pressure cylinder and the engine making 125 rpm and the barometer at 29.5 in. Hg, (*a*) what is the rated horsepower, with a card factor of 0.80? (*b*) If this engine was equipped with a surface condenser carrying a vacuum of 27.14 in. Hg, and, at an actual indicated horsepower of 600, used 25 lb of steam per ihp-hr, how many pounds of cooling water would be necessary per hour? The temperature of the condensed steam leaving the condenser is 72 F, and the cooling water enters at 50 F and leaves at 80 F. Neglect radiation loss. (*c*) Find the quality of the exhaust.

14. An 8,000-kw turbine is supplied with steam at a pressure of 450 psia and 550 F temperature. When operating at full load, the back pressure on the first-stage nozzles is 200 psia. The exhaust steam enters the condenser at 3 psia and with a heat content of 1,000 Btu per lb. Cooling water enters the condenser at 50 F and leaves at 70 F. The condensate temperature is 68 F. Neglect heat losses due to radiation and bearing friction. Find (*a*) the weight of steam used per hour; (*b*) the velocity of the steam entering the first-stage blades, if the efficiency of the nozzles is 95 per cent; (*c*) the actual quality of the exhaust steam; (*d*) the Rankine cycle efficiency of the turbine; (*e*) the cooling water required, expressed in gallons per minute.

15. Assuming a mean value for the coefficient of heat transmission, Btu per hour, per degree difference of temperature, per square foot of cooling surface to be 600; and assuming that the enthalpy of exhaust steam at condenser pressure is 1,000 Btu per lb with the following data calculate the amount of cooling surface necessary for the condenser; temperature of exhaust steam, 140 F; temperature of circulation water at inlet, 75 F; temperature of circulating water at outlet, 105 F; temperature of condensed steam, 100 F; weight of steam per hour, 25,000 lb.

CHAPTER XII

AIR COMPRESSORS AND REFRIGERATING MACHINERY

235. Compressed Air.—Where power is required at some distance from the source, compressed air is particularly valuable because the supply pressure can be maintained nearly constant, there is no trouble with condensation as in the case of steam, and the exhaust air may be discharged directly into the atmosphere. A practical use of compressed air is found in the operation of pneumatic tools, such as, hammers and drills in mining operation, air-brake service, air hoists, sand blasting, spraying paint, and air engines.

236. Types of Compressors.—Compressors may be classified as *rotary, centrifugal,* or *reciprocating.* The rotary blower, shown in Fig. 280, has two impellers mounted on parallel shafts that rotate in opposite directions. The air is drawn into the space between the impellers and the casing, and forced to the outlet under a positive pressure. Leakage of air past the impellers is very small because of the close-fitting parts. There is no actual contact between the impellers and the casing or between the impellers themselves.

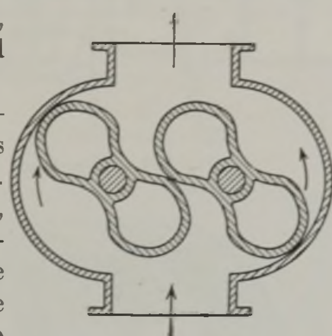


FIG. 280.—Rotary blower.

The centrifugal compressor, or turbo compressor, is built in single-stage units for pressures up to 20 psi and multistage units for pressures up to 125 psi. The discharge pressure obtained depends upon the speed of the impellers and the number of stages used. Figure 281 shows a four-stage centrifugal compressor. The air enters the casing at the right and in passing through the first stage has its velocity increased. The velocity energy is changed to potential energy in passing through the enlarged passages from the first to the second stage. The remaining impellers each add an increment to the pressure in the same manner until the air is finally discharged at the left. For multistage

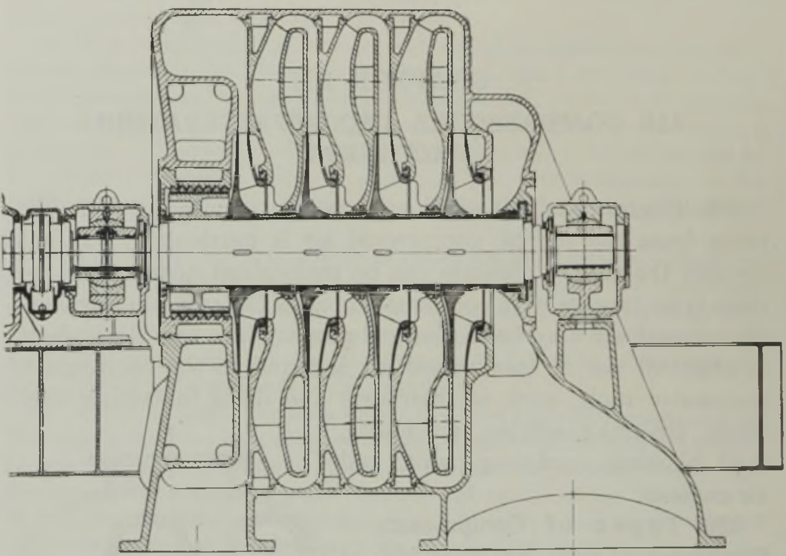


FIG. 281.—Four-stage De Laval centrifugal compressor.

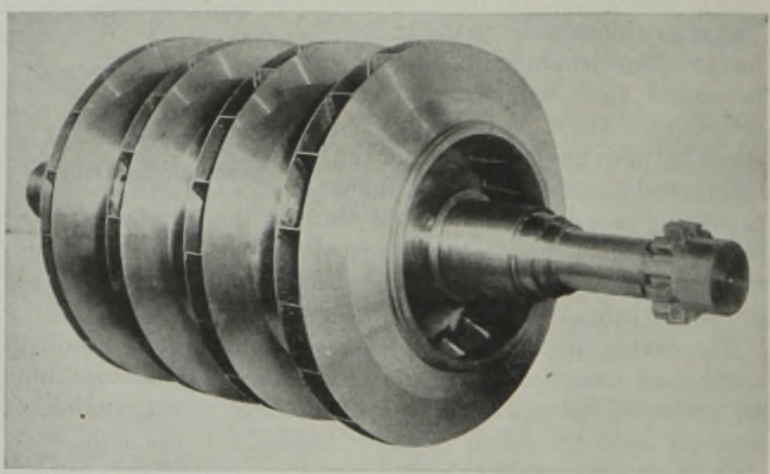


FIG. 282.—75,000 cfm 30 lb pressure turbine-driven blast furnace blower rotor; inlet end view.

centrifugal compressors that operate at higher pressures, water cooling is generally employed in the spaces surrounding the outer portion of the impeller casing to remove some of the heat of compression.

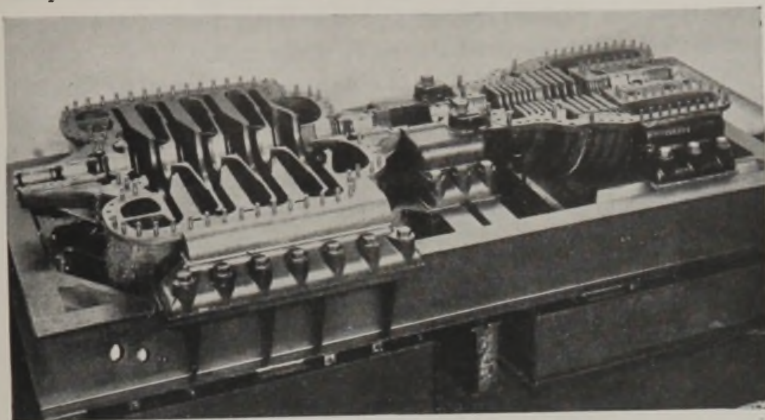


FIG. 283.—75,000 cfm, 30 lb pressure, turbine-driven blast furnace blower with covers and rotors removed.

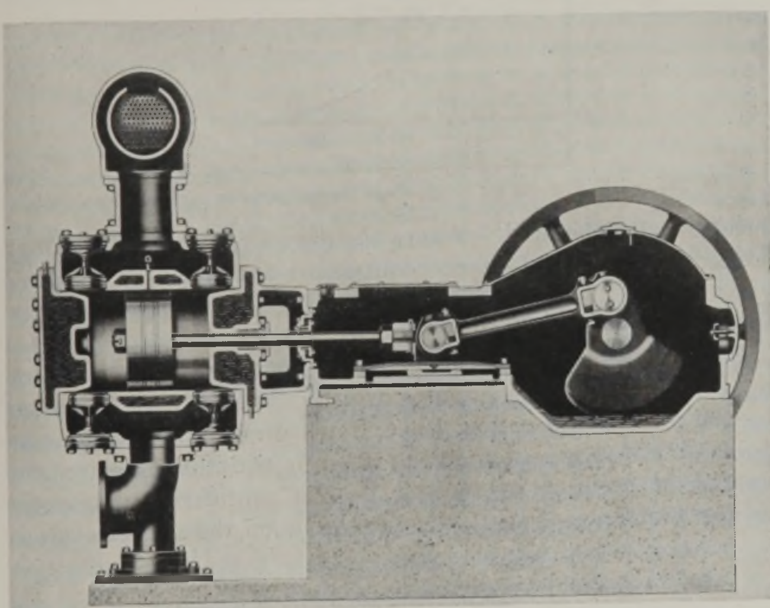


FIG. 284.—Double-acting reciprocating compressor.

Figure 282 shows the rotor of the four-stage centrifugal compressor removed from the casing. Note the inlet passageway for the first stage at the right.

Figure 283 shows the lower half of the casing of the compressor with the rotor and the upper half of the casing removed. The compressor is turbine-driven and the lower half of the steam-turbine casing is shown at the right of the figure.

A double-acting piston type of compressor is shown in Fig. 284. The construction is similar to that of a steam engine except that the compressor has an inlet and a discharge valve in each end of the cylinder, thus reducing the clearance volume. The valves may be spring loaded and operated by the air or they may be

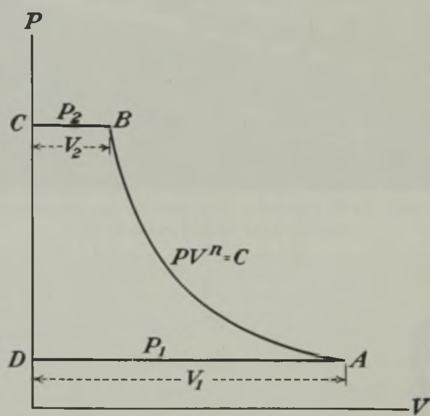


Fig. 285.—*PV* diagram for single-stage air compressor without clearance.

mechanically operated. Where the discharge pressure does not exceed 100 psi, single-stage compressors are used; for air pressures above 100 psi, two or more stages are employed with interstage cooling.

237. Work of Compressor without Clearance.—The net work done upon the air by a compressor without clearance is indicated by the area *ABCD* in Fig. 285. If the pressure and volume at point *A* are represented by P_1 and V_1 and the pressure and volume at point *B* by P_2 and V_2 and considering polytropic compression from *A* to *B*, the work done on the air per cycle is

$$W = \frac{P_2V_2 - P_1V_1}{(n-1)} + P_2V_2 - P_1V_1. \tag{1}$$

Expanding and combining terms

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

$$W = \frac{n}{(n-1)}(P_2V_2 - P_1V_1), \tag{2}$$

or

$$W = \frac{n}{(n-1)} P_1V_1\left(\frac{P_2V_2}{P_1V_1} - 1\right); \tag{3}$$

but

$$\frac{V_2}{V_1} = \left(\frac{P_1}{P_2}\right)^{\frac{1}{n}}.$$

Substituting,

$$W = \frac{n}{n-1} P_1V_1\left[\left(\frac{P_2}{P_1}\right)\left(\frac{P_1}{P_2}\right)^{\frac{1}{n}} - 1\right], \tag{4}$$

or

$$W = \frac{n}{n-1} P_1V_1\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right]. \tag{5}$$

Equation (5) is the general expression for the work done on air in a single-stage compressor when the compression is polytropic and when the clearance is zero.

The net work done on air in a single-stage compressor, when the compression is isothermal and when the clearance is zero, is

$$W = P_1V_1 \log e \frac{V_1}{V_2} + P_2V_2 - P_1V_1, \tag{6}$$

but

$$P_2V_2 = P_1V_1 \qquad \text{and} \qquad \frac{V_1}{V_2} = \frac{P_2}{P_1}.$$

Thus

$$W = P_1V_1 \log e \frac{P_2}{P_1}, \tag{7}$$

which is the general expression for work done on air by a single-stage compressor with isothermal compression and with zero clearance.

238. Capacity of Compressor.—The output, or capacity, of a compressor is usually expressed in cubic feet of *free air* per minute. Free air is air at the atmospheric pressure and temperature prevailing where the compressor is operating. In Fig. 285, the free air taken in per cycle is designated by V_1, the volume at point A.

239. Work of Compressor with Clearance.—An actual compressor must have a certain amount of linear clearance between the cylinder head and the piston when at the extreme end of its travel for safe operation. The *clearance volume* is the volume enclosed between the cylinder head and the piston at dead center plus the volume of the ports up to the valves, and is represented in Fig. 286 by V_c. The air remaining in the clearance space expands on the return stroke of the piston reducing the volume of new air taken in during the suction stroke. In Fig. 286 the clearance air expands to volume V_d before the intake valve opens. V_1 represents the volume of new air taken in per cycle and V_p the piston displacement.

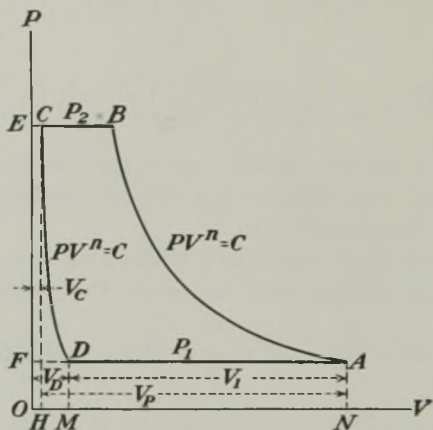


FIG. 286.—*PV* diagram for single-stage air compressor with clearance.

The ratio of the new air actually taken into the cylinder to the volume displaced by the piston is called the *volumetric efficiency* of the compressor, or

$$e_v = \frac{V_1}{V_p}. \tag{8}$$

The *volumetric clearance* of a compressor is usually expressed as a ratio of the clearance volume to the piston displacement and may vary from 1 per cent to 5 per cent depending upon the design and size of compressor. In Fig. 286,

$$\frac{V_c}{V_p} = C \text{ (volumetric clearance)}, \tag{9}$$

or

$$V_c = CV_p$$

and

$$V_1 = V_p + V_c - V_d, \tag{10}$$

but

$$V_d = V_c\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} = CV_p\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}. \tag{11}$$

Substituting

$$V_1 = V_p + CV_p - CV_p\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} \tag{12}$$

or

$$V_1 = V_p\left[1 + C - C\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}\right]. \tag{13}$$

Therefore,

$$e_v = \frac{V_1}{V_p} = \left[1 + C - C\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}\right]. \tag{14}$$

If the value n for the reexpansion path CD is the same as for the compression path AB, the net work done on the air per cycle is the same as for the compressor without clearance; *i.e.*,

$$W = \frac{n}{n-1}P_1V_1\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right], \tag{15}$$

or

$$W = \frac{n}{n-1}P_1V_p\left[1 + C - C\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}\right]\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right]. \tag{16}$$

Example.—A double-acting air compressor that has a bore of 8 in. and a stroke of 10 in. runs at 150 rpm. The clearance is 3 per cent and the value of n for the compression and the reexpansion path is 1.3. The suction pressure is 14 psia and the temperature is 60 F. The discharge pressure is 112 psia. Find (*a*) the volumetric efficiency of the compressor; (*b*) the capacity of the compressor in cubic feet per minute (free air); (*c*) the air horsepower.

Solution.—(*a*) From Eq. (14),

$$\begin{aligned} e_v &= 1 + C - C\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} \\ &= 1 + 0.03 - 0.03\left(\frac{112}{14}\right)^{\frac{1}{1.3}} \\ &= 1 + 0.03 - 0.148 = 0.882, \text{ or } 88.2 \text{ per cent.} \end{aligned}$$

(*b*) From Eq. (8),

$$\begin{aligned} V_1 &= e_vV_p \\ &= \frac{0.882 \times 2 \times 3.1416 \times 16 \times 10 \times 150}{1{,}728} = 77 \text{ cu ft per min.} \end{aligned}$$

(c) From Eq. (15),

$$\text{Air hp} = \frac{W}{33{,}000} = \frac{n}{(n-1)} \times \frac{P_1V_1}{33{,}000}\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right]$$

$$= \frac{1.3 \times 14 \times 144 \times 77}{(1.3-1) \times 33{,}000}\left[\left(\frac{112}{14}\right)^{\frac{1.3-1}{1.3}} - 1\right]$$

$$= \frac{1.3 \times 14 \times 144 \times 77}{0.3 \times 33{,}000}\left[(8)^{\frac{1}{4.33}} - 1\right]$$

$$= 20.38(1.616 - 1) = 12.55 \text{ hp}.$$

240. Multistage Compression.—Figure 287 shows the theoretical PV diagram for two-stage compression with clearance.

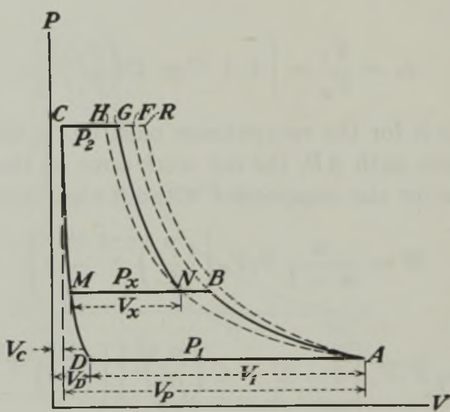


FIG. 287.—PV diagram for two-stage air compressor with clearance.

Starting at point A, the air is compressed in the low-pressure cylinder along the path AB to the receiver pressure at B. BN represents the cooling of the air at constant pressure in the receiver to the initial temperature T_1, point N lying on the isothermal path ANH. The compression in the second stage follows the path NG to the discharge pressure P_2. The reexpansion of the clearance air for both cylinders is considered to be a continuous path and represented by the line CMD.

The work done upon the air in the low-pressure cylinder is represented in Fig. 287 by the area $ABNMD$, and the work in the high-pressure cylinder by the area $NGCM$. The work done by a single-stage compressor operating between the same pressure limits, *i.e.*, P_1 to P_2, would be the area $ABFCD$. The saving of work by two-stage compression with perfect intercooling of the air over single-stage compression is represented by the area

BNGF. The path *AR* represents an adiabatic compression from P_1 to P_2 for a single-stage compressor. The saving in work effected by two-stage compression with perfect intercooling over that of single-stage adiabatic compression is represented by the area *ABNGR*.

From Eq. (15) the work done upon the air in the low-pressure cylinder is

$$W_1 = \frac{n}{n-1} P_1 V_1 \left[\left(\frac{P_x}{P_1} \right)^{\frac{n-1}{n}} - 1 \right] \tag{17}$$

and the work done in the high-pressure cylinder is

$$W_2 = \frac{n}{n-1} P_x V_x \left[\left(\frac{P_2}{P_x} \right)^{\frac{n-1}{n}} - 1 \right]. \tag{18}$$

Assuming the value of n for the reexpansion of the clearance air to be the same as the compression path and with point N on the isothermal path ANH, then $P_x V_x = P_1 V_1$ and

$$W_2 = \frac{n}{n-1} P_1 V_1 \left[\left(\frac{P_2}{P_x} \right)^{\frac{n-1}{n}} - 1 \right]. \tag{19}$$

The suction pressure (P_1) and the discharge pressure (P_2) are fixed quantities in any given installation, but the receiver pressure (P_x) may be varied at will. For minimum work, it can be shown that the work of compression will be divided equally between the two cylinders of a compound compressor. Thus, when W_1 equals W_2, the ratio of compression pressures will be equal and

$$\frac{P_2}{P_x} = \frac{P_x}{P_1}, \qquad \text{or} \qquad P_x = \sqrt{P_1 P_2}. \tag{20}$$

Hence, substituting P_x its value from Eq. (20),

$$\frac{P_2}{P_x} = \frac{P_x}{P_1} = \left(\frac{P_2}{P_1} \right)^{\frac{1}{2}}. \tag{21}$$

By substituting Eq. (21) in Eq. (17) and Eq. (19) and adding, it is seen that the total work of a two-stage air compressor, with perfect intercooling and clearance, is

$$W = \frac{2nP_1V_1}{n-1} \left[\left(\frac{P_2}{P_1} \right)^{\frac{n-1}{2n}} - 1 \right]. \tag{22}$$

241. Mechanical Refrigeration.—In mechanical refrigeration, a compressor, similar to an air compressor, is employed to compress the refrigerant to a pressure sufficiently high so that its saturation temperature will be above that of the natural cooling agent. The cooling agent may be air at atmospheric temperature, or water. The refrigerating machine extracts heat from

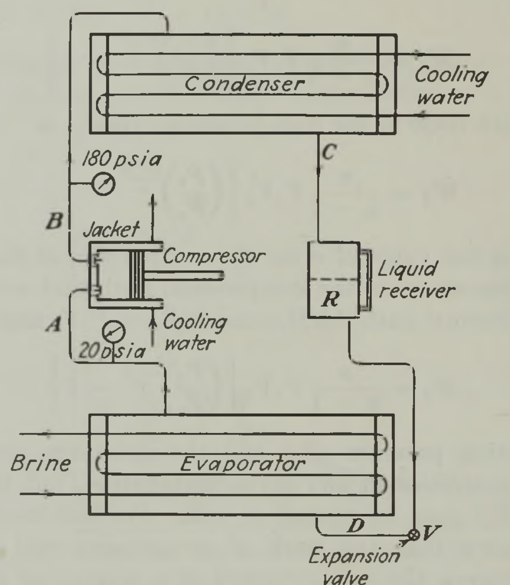


FIG. 288a.—Vapor compression refrigerating machine.

one level and raises it to a higher level in the same manner that a pump elevates water from one level to another of higher elevation.

242. The Vapor Compression Refrigerating Machine.—Figure 288*a* represents diagrammatically a vapor compression refrigeration cycle. The vapor, ammonia in this case, enters the compressor at point A at a pressure of 20 psia and in a dry saturated state, temperature -16.64 F. The ammonia is compressed adiabatically to a pressure of 180 psia and leaves the compressor at a temperature of 271.6 F, point B, in a superheated state. The saturation temperature at 180 psia is 89.78 F. The ammonia enters the condenser where it gives up first its superheat, then its latent heat, and a part of its heat of the liquid to the cooling water and emerges at point C as a liquid at a temperature of 75 F. The heat rejection takes place at substantially constant pressure. The condenser may be of the double-tube type with water flowing through the inner tube and the refrigerant through the outer one, or it may be of the shell-and-tube type where the water flows through the tubes and the ammonia occupies the space in the shell around the tubes.

The liquid ammonia coming from the condenser is collected in the receiver R and is then throttled through the expansion valve V to the suction pressure, in this case 20 psia. The expansion valve is nothing more than a pressure-reducing valve

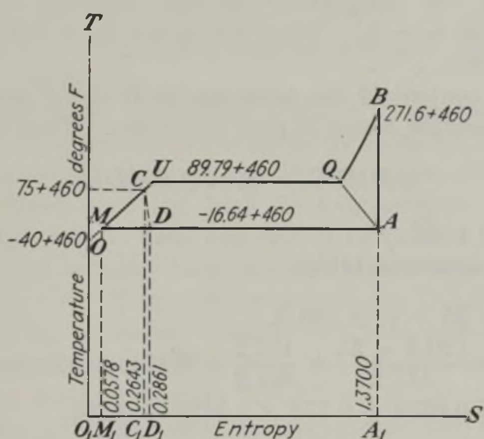


Fig. 288*b*.—Temperature-entropy diagram of ammonia cycle.

and may be manually or thermostatically controlled. The valve is a direct means of regulating the flow oı refrigerant into the evaporator and, as such, controls the evaporator pressure and the amount of cooling or refrigeration produced.

The ammonia enters the evaporator or refrigeration coils at point D at a temperature of -16.64 F, saturation temperature corresponding to 20 psia. The enthalpy at point D is the same as that of the liquid before passing the expansion valve, the operation being a pure throttling one. The condition or quality will be determined later. In passing through the refrigeration coils, the ammonia receives heat from the space being refrigerated and emerges at point A in a dry saturated state to start its cycle again.

243. Temperature-entropy Diagram of Ammonia Cycle.—Figure 288*b* shows the refrigeration cycle on a temperature entropy plane with lettering to correspond to Fig. 288*a* and with 1 lb of ammonia circulated.

At point *A* the ammonia is dry saturated at a pressure of 20 psia; from the ammonia tables the entropy is seen to be 1.3700. It is then compressed adiabatically and isentropically to *B* where the pressure is 180 psia and the tables show the temperature to be 271.6 F for an entropy of 1.3700.

The ammonia enters the condenser at state point *B* and leaves at state *C*. The heat given up by the ammonia is represented on the temperature-entropy diagram by the area A_1BQUCC_1 or $h_B - h_C$. From the tables, $h_B = 748.94$ Btu and $h_C = 126.2$ Btu.

The heat content of the ammonia at *D* is the same as at *C*, since the operation is one of pure throttling. Thus area

$$O_1OMCC_1 = \text{area } O_1OMDD_1$$

and $h_c = h_m + q_dL$.

From the ammonia tables,

$$126.2 = 25 + q_d \times 581.2.$$

$$q_d = \frac{126.2 - 25}{581.2} = \frac{101.2}{581.2} = 0.174, \text{ or } 17.4 \text{ per cent.}$$

$$S_d = 0.0578 + 0.174 \times 1.3122 = 0.2861$$

The ammonia enters the refrigerating coils at a quality of 17.4 per cent and leaves in a dry saturated state. The refrigerative effect per pound of ammonia circulated is seen to be $h_A - h_D$ and is represented by the area A_1ADD_1.

$$h_A - h_D = 606.2 - 126.2 = 480 \text{ Btu per lb.}$$

244. One ton of refrigeration is defined as the removal of the latent heat of fusion of one ton of water at 32 F to ice at 32 F in a period of 24 hr. Thus, the rate of heat removal per ton of refrigeration is $2{,}000 \times 144 = 288{,}000$ Btu per 24 hr, or 12,000 Btu per hr, or 200 Btu per min.

The amount of ammonia circulated per hour to produce one ton of refrigeration for the illustrated problem would then be $12{,}000 \div 480 = 25$ lb per hr.

245. Coefficient of Performance.—A common method of expressing the output of a compression system of refrigeration in terms of the input is to give its coefficient of performance (Cop). The output is the heat extracted by the refrigerant. In Fig. 288*b* it is represented by the area D_1DAA_1 per pound of ammonia circulated. The input is the heat equivalent of the work done upon the substance in the compressor, and is represented by the area $MCUQBADM$ per pound.

$$\text{Cop} = \frac{\text{refrigeration}}{\text{work} \times \frac{1}{778}}. \tag{23}$$

Another widely used and favored method of expressing the performance of a refrigerating machine is to give the horsepower per ton of refrigeration.

246. Refrigerants.—Ammonia, NH_3, is by far the most commonly used refrigerant in the artificial ice industry, owing to the moderate pressures that prevail in the cycle, the moderate specific volumes, and the high heat absorption per pound. Ammonia, however, is toxic and cannot be used in hospitals, on board ship, and in many other places where leakage would be a real danger to life. Dichlorodifluoromethane, CCl_2F_2 (known as Freon or F-12), or carbon dioxide, CO_2, are preferred in such places because they are neither toxic nor inflammable. Sulphur dioxide, SO_2, methyl chloride, CH_3Cl, ethyl chloride, C_2H_5Cl, butane, C_4H_{10}, and many other refrigerants are used in household refrigerators. For a more detailed list consult the handbook of the American Society of Refrigerating Engineers.

TABLE XXXII.—COMPARISON OF COEFFICIENT OF PERFORMANCE AND HORSEPOWER PER TON OF A FEW REFRIGERANTS REFERRED TO STANDARD OPERATING TEMPERATURES

Standard conditions, +5 F to 86 F	Coefficient of performance	Work, hp per ton	Pressure, psia	
			Suction	Discharge
Ammonia, NH_3	4.85	0.973	34.27	169.2
Carbon dioxide, CO_2	2.56	1.840	331.95	1043.0
Freon, CCl_2F_2	4.72	0.997	11.81	93.2
Methyl chloride, CH_3Cl	4.67	1.007	21.0	95.5
Sulphur dioxide, SO_2	4.74	0.995	11.81	66.45

PROBLEMS

1. An ideal air compressor without clearance is to compress 80 cu ft of air per min from a pressure of 14 psia to a pressure of 90 psia. Calculate the net work done upon the air (*a*) if the compression is isothermal; (*b*) if it is polytropic with $n = 1.3$.

2. An air compressor delivers 10 lb of air per min. The suction pressure is 14.5 psia and the initial temperature of the air is 70 F. The equation of the compression path is $pv^{1.3} = c$ and the final temperature is 400 F. Determine (*a*) the discharge pressure; (*b*) the air horsepower required to compress the air.

3. A 10-in. by 12-in. double-acting compressor with 4 per cent clearance runs at 150 rpm. The suction pressure and temperature are 14.4 psia and 75 F. The equation of the compression and the reexpansion curves is $pv^{1.3} = c$. The discharge pressure is 115.2 psia. Determine (*a*) the volumetric efficiency; (*b*) the cfm of free air handled; (*c*) the air horsepower required to compress the air.

4. A double-acting air compressor delivers 29.78 cfm of free air per min while running at 100 rpm. The suction pressure and temperature are 14 psia and 70 F. The compressor has 3 per cent clearance and the compression and reexpansion paths follow the equation $pv^{1.3} = c$. The discharge pressure is 84 psia. If the cylinder diameter is 6 in., what is the stroke of the compressor?

5. Determine the tons of refrigerative effect produced by a cooling coil that is furnished with 3 gal of water per min. The water enters the coil at 50 F and leaves at 60 F.

6. An ideal refrigerating machine using ammonia as the refrigerant operates with a suction pressure of 15 psia and a discharge pressure of 160 psia. The temperature of the ammonia on the high side of the expansion valve is 70 F. Determine the pounds of ammonia that must be furnished to the evaporator per hour per ton of refrigerative effect, if the ammonia leaves in a dry saturated state.

7. In Prob. 6, if the ammonia is compressed isentropically, determine (*a*) the heat rejected to the condenser cooling water per ton of refrigeration; and (*b*) the indicated horsepower of the compressor per ton.

8. A refrigerator cools 10 tons of air per hour. The air enters the cooler at 70 F and leaves at 50 F at atmospheric pressure. The refrigerator is driven by an electric motor which develops 8 hp. Determine (*a*) the cfm of warm air handled by the fan; (*b*) the tons of refrigeration produced; (*c*) the over-all coefficient of performance.

9. Determine the displacement of a compressor, without clearance, in cfm for a 25-ton refrigerating capacity where the refrigerant is: (*a*) NH_3; (*b*) CO_2; (*c*) F-12 (Freon); (*d*) SO_2.

The temperature of the liquid entering the expansion valve is 80 F, and the temperature of the evaporator is 20 F. The vapor leaves the evaporator and enters the compressor in a dry and saturated state.

CHAPTER XIII

THE INTERNAL-COMBUSTION ENGINE

247. Definition.—An internal-combustion engine is one in which the working medium is composed of the gases resulting from the combustion of the fuel that supplies the heat energy. In the steam cycle the heat of the fuel burned under the boiler is transferred to water and steam in the boiler and this working medium, after rejection from the engine or turbine, is usually condensed and recovered; hence the cycle may be repeated with the same physical medium. In the internal-combustion engine, the products of combustion as a working medium perform the same function as steam in the cylinder, but after the work of the cycle has been completed the medium is rejected and replaced by an identical new charge. Unlike the steam cycle, there is no possibility of reuse of the rejected charge.

Internal-combustion engines are almost all of the reciprocating type. A few successful gas turbines have been built, but the continuous exposure of metal parts to extremely high temperatures in the first stages of expansion constitutes a problem not easily solved.

248. Development.—Historical records show that an internal-combustion engine operating on coal gas was built by John Barber in England as early as 1791, preceded only by Huygens' impractical "gunpowder" engine (1680). It was not until 1862, however, that Beau de Rochas, a Frenchman, outlined the principles of the internal-combustion engine cycle as used today. In 1876 Otto built and sold the first successful engines embodying de Rochas' principles of compression, ignition, and expansion of the charge, and this sequence has come to be called the *Otto cycle.* The early engines were heavy, inefficient, and very uncertain in operation and regulation, weighing over 1,000 lb per bhp and having a thermal efficiency of about 4 per cent. Modern aircraft engines weigh less than 1 lb per bhp and efficiencies of nearly 40 per cent have been obtained. Most modern engines may be operated for hours at a time with little or no attention; stationary plants, given proper care, operate continuously for months and even years. Starting is seldom a problem even under severe climatic conditions, and trouble in starting can usually be traced to improper maintenance or operation. Regulating devices have been developed that permit the use of internal-combustion engines for power purposes requiring very close control, for instance as prime movers with parallel-connected alternators.

In 1898 Rudolf Diesel, after experimenting with an internal-combustion engine using powdered coal as fuel, developed the compression-ignition cycle that bears his name. The powdered-coal engines were not then successful, but with the use of oil as fuel a Diesel engine was run with an efficiency much higher than any previous engine, and the modern counterparts of that engine are still the most efficient prime movers known.

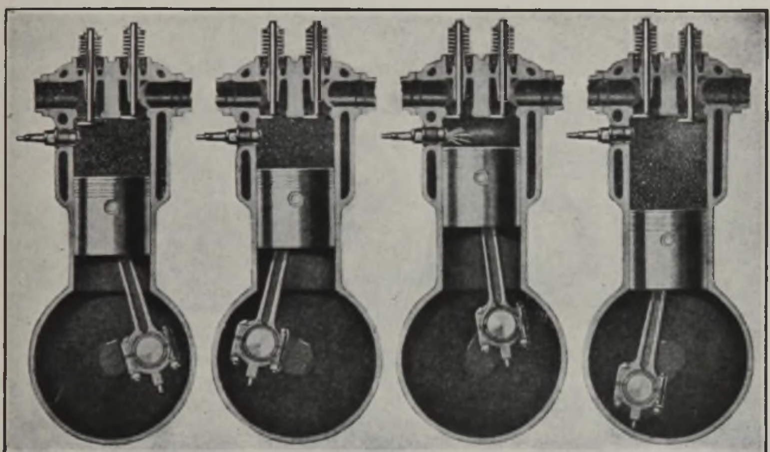
During the first few years of the development of the Otto cycle engine, speeds were low and single units developed no more than 10 hp. Today racing-car engines are run at speeds in excess of 8,000 rpm; Diesel engines of 24,000 hp are in regular operation. Wherever high speed and economy in transportation are necessary, the internal-combustion engine has proved to be the most satisfactory source of power. The science of flying in heavier-than-air ships would be impractical without it, and its intensive development in the automobile has brought about refinements in materials and production methods that have affected all of industry. There are over 31,000,000 automotive vehicles in use in this country and 92 per cent of the total installed horsepower is in internal-combustion engine units. In marine installations, of the 66,800,000 tons of shipping registered with Lloyds in 1938–1939, some 15,200,000 were Diesel powered, as compared with 8,000,000 Diesel tonnage in 1930–1931 when 68,000,000 total was registered. New construction in 1939 was about 58 per cent Diesel powered.

249. Classification.—Internal-combustion engines are classified according to their various construction and operating features. Since no single system of classification serves for all engines, it is usually necessary to refer to several in order to describe an engine. The various classifications are as follows:

1. Cycle of operation.
2. Cycle of combustion.

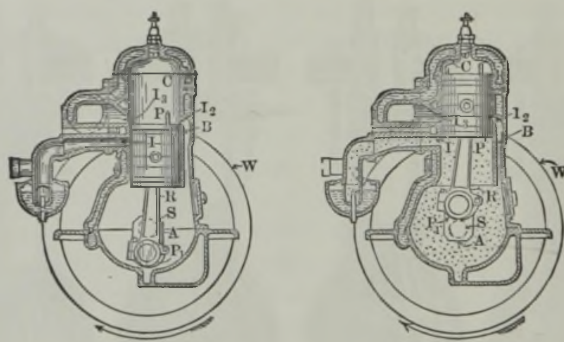
3. Number of cylinders.
4. Arrangement of cylinders.
5. Cooling system.
6. Intake system.
7. Function.
8. Fuel used.

250. Cycles of Operation.—When the term *two-* or *four-cycle engine* is used, it is meant that two or four strokes of the piston, respectively, are required for the completion of one cycle of operation. Correctly speaking then, it is a two-*stroke* cycle or four-*stroke* cycle engine.



Suction. Compression. Power Exhaust.
FIG. 289.—Cycle of operation in four-cycle (Buick) engine.

Figure 289 illustrates the operation of a four-stroke cycle engine, or one in which the completion of one cycle of operation is accomplished in four strokes of the piston. With the crank turning in a clockwise direction the piston is shown first at the left moving downward and drawing a charge of fuel vapor and air into the cylinder through the open intake valve with the exhaust valve closed. This is the suction stroke. On the succeeding upward stroke both valves are closed and the charge is compressed. This is the compression stroke. As the piston nears the top of the compression stroke the charge is ignited by a spark that raises the temperature of a portion of the combustible mixture to a point where combustion begins. The flame of combustion is propagated in every direction away from the point of ignition until the whole charge is burned. The heat energy thus released raises the temperature and pressure of the products of combustion and the piston is driven down on the power stroke by the expanding gases. During most of this expansion, or power stroke, both valves remain closed, but near the bottom of the stroke, the exhaust valve is opened and the following upward (exhaust) stroke of the piston pushes most of the burned gases from the cylinder. Near the top center position of the crank the exhaust valve is closed and the intake opened to repeat the cycle. When it is considered that this series of events occurs more than 30 times per second in a modern

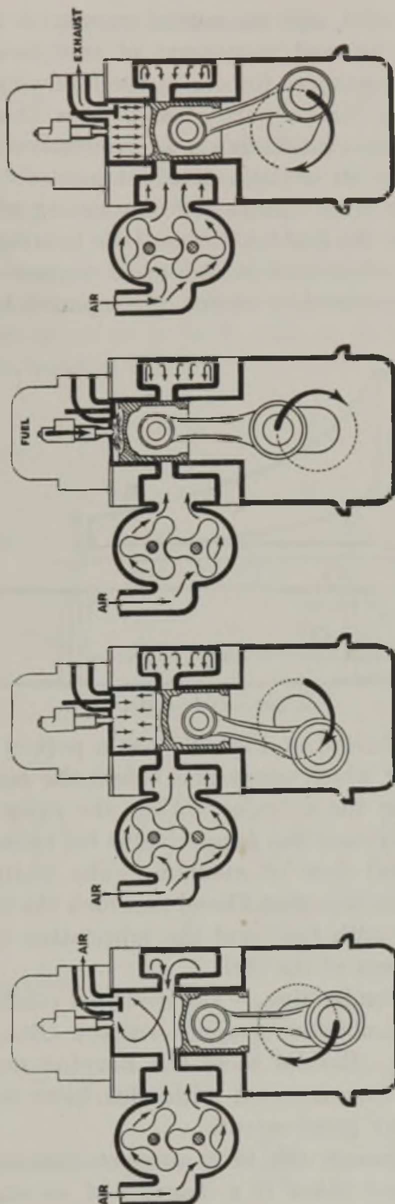


1. 2.

Fig. 290.—Crosssection of two-cycle engine.

engine at full power, the problems involved in high-speed engine operation can be more fully appreciated.

Even more problems are introduced when the full cycle occurs in two strokes of the piston. Figure 290 illustrates the "crankcase compression" method commonly used. In an engine of this type the piston itself acts as a valve and uncovers intake and exhaust ports at the proper time. In position 1 the piston is shown after completing the power stroke, permitting exhaust gases to pass out through the exhaust ports 1_3 which are opened first, and then admitting a fresh mixture through the intake port 1_2. This mixture is forced into the cylinder by the partial compression of the charge trapped in the crankcase. Thus the exhaust and intake events occur while the piston is near the bottom of its stroke. As the crank reaches top center on the compression stroke, the lower end of the piston uncovers



Exhaust and intake Compression Combustion Power

FIG. 291.—Two-cycle engine equipped with an auxiliary blower for scavenging and air induction.

the carbureting port 1, and the partial vacuum in the crankcase produced by the upward movement of the piston draws air and fuel into the crankcase for use in the following cycle. The downward (power) stroke partly compresses this charge and when the exhaust and intake ports are successively opened, the cycle is repeated. An obvious disadvantage of this system is the pollution of the fresh charge with lubricating oil, which must be introduced with the fuel to lubricate the bearings and piston. This difficulty is eliminated in two-cycle engines in which the fresh charge is introduced by means of a separate blower.

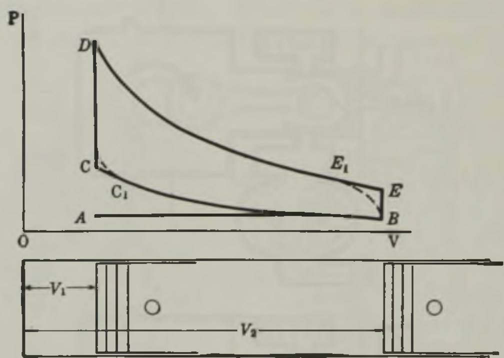


FIG. 292.—Otto cycle.

The engine illustrated in Fig. 291 has a poppet valve at the top of the cylinder which opens just before the piston uncovers the intake ports in the cylinder. Thus the cylinder ports are used for intake only, and the poppet valve for exhaust only, giving a unidirectional flow of medium. The particular engine illustrated is a solid-injection Diesel in which the blower pumps only air, unmixed with fuel, and the lubrication system of the engine is independent of the fuel.

251. Cycles of Combustion.—The principal combustion cycles used in internal-combustion engines are the Otto, Diesel, and semi-Diesel cycles. Besides these the Brayton and the Lenoir cycles have been experimented with, but have not been used commercially to any great extent.

Figure 292 represents the ideal pressure-volume diagram of the Otto engine and below it a diagram of an engine cylinder showing the piston positions at upper and lower dead centers. V_1 is the volume at the end of the compression stroke and at the beginning of the expansion stroke. V_2 is the volume at the beginning of the compression stroke and at the end of the expansion stroke. $\frac{V_2}{V_1}$ is then not only the *compression* ratio, but also the *expansion* ratio, and is usually designated as r.

The lines BC and DE are assumed to be adiabatics. All the heat must then be absorbed along the line CD and all rejected along EB. Let the heat received along CD be represented by H_1, and the absolute temperature at C by T_c, and at D by T_d; let the heat rejected along EB be represented by H_2, and the absolute temperature at E be T_e and at B by T_b; and let the weight of the charge $= w$.

Then

$$H_1 = wc_v(T_d - T_c)$$

and

$$H_2 = wc_v(T_e - T_b).$$

The work done $= H_1 - H_2 = wc_v(T_d - T_c) - wc_v(T_e - T_b)$.

$$\text{Efficiency} = \frac{H_1 - H_2}{H_1} = \frac{wc_v(T_d - T_c) - wc_v(T_e - T_b)}{wc_v(T_d - T_c)},$$

$$= \frac{(T_d - T_c) - (T_e - T_b)}{T_d - T_c},$$

$$= 1 - \frac{T_e - T_b}{T_d - T_c}. \tag{1}$$

Both curves are adiabatic, hence

$$\left(\frac{T_e}{T_d}\right) = \left(\frac{V_d}{V_e}\right)^{\kappa-1} = \left(\frac{V_c}{V_b}\right)^{\kappa-1} = \left(\frac{T_b}{T_c}\right).$$

Therefore,

$$\frac{T_e}{T_d} = \frac{T_b}{T_c}, \tag{2}$$

and, by subtraction,

$$\frac{T_e}{T_d} = \frac{T_e - T_b}{T_d - T_c} = \frac{T_b}{T_c}. \tag{3}$$

Substituting Eq. (3) in Eq. (1),

$$\text{Efficiency} = 1 - \frac{T_b}{T_c} = 1 - \frac{T_e}{T_d}. \tag{4}$$

This is the most important expression in connection with the gas engine. It shows that *the efficiency of a gas engine working* *in the Otto cycle depends upon the temperature before and after compression.* The knowledge of this fact, first demonstrated by Dougal Clerk, has led to the production of the modern high-efficiency engine.

The efficiency of the Otto cycle may also be expressed in terms of volume and pressure.

Since the compression curve is an adiabatic,

$$\frac{T_b}{T_c} = \left(\frac{V_c}{V_b}\right)^{\kappa-1}.$$

Substituting this value of $\frac{T_b}{T_c}$ in Eq. (4),

$$\text{Efficiency} = 1 - \left(\frac{V_c}{V_b}\right)^{\kappa-1} = 1 - \left(\frac{1}{r}\right)^{\kappa-1}. \tag{5}$$

Equation (5) shows that the efficiency *depends upon and varies with V_c, the clearance volume.* That is to say, it varies with the *compression* and *expansion* ratio r.

Finally, since BC is an adiabatic,

$$\frac{T_b}{T_c} = \left(\frac{P_b}{P_c}\right)^{\frac{\kappa-1}{\kappa}}.$$

Substituting this value in Eq. (4),

$$\text{Efficiency} = 1 - \left(\frac{P_b}{P_c}\right)^{\frac{\kappa-1}{\kappa}}. \tag{6}$$

From Eq. (6) it is seen that the *efficiency increases as the compression pressure P_c increases,* since the pressure P_b will remain nearly constant.

The dotted lines in Fig. 292 show the deviation of the actual cycle from the theoretical. Because combustion is not instantaneous, it is necessary that ignition should occur at a point C_1 before the piston reaches the end of the compression stroke. If the pressure is not relieved before the end of the expansion stroke, the piston must be forced back against this pressure. For this reason the exhaust valve is opened at E_1, so that the pressure in the cylinder is practically atmospheric before the piston starts on the exhaust stroke. During the exhaust stroke the pressure is slightly above atmospheric.

For full-throttle operation the pressure in the cylinder during the intake stroke is slightly below one atmosphere but the area of this negative loop is so small that it is usually ignored. For throttled operation the pressure during the suction stroke may drop as much as 10 psi to 12 psi below the atmospheric pressure. Although they approximate the polytropic curve $pv^n = c$, the compression and expansion curves of the actual diagrams are not truly adiabatic. This is due largely to the fact that there is a heat transfer between the gases and the cooled cylinder walls, but is further complicated by a variation in the specific heat as the temperatures of the gases change. The fuel vapors, which are compressed with the air, have specific-heat ratios which are lower for gases having many atoms per molecule. In general for monatomic gases at ordinary temperatures κ is 1.66; for diatomic gases, 1.40; for triatomic gases, 1.29; and for gasoline vapor about 1.08. For an engine using gasoline as fuel, the value of n is generally taken as 1.30 for both the compression and the expansion curves although values from 1.29 to 1.35 are sometimes used.

The compression pressures for engines operating on this cycle vary from 50 psi to 250 psi depending on the type of fuel for which the engine is designed. For automobile engines the pressures range from 100 psi to 160 psi. Some difficulty may be experienced with engines designed for the higher compression pressures when operating on ordinary gasoline. Even lower pressures are required with kerosene; blast-furnace gas may allow a compression to 150 psi, and alcohol as high as 250 psi.

In the Diesel cycle, combustion occurs at constant pressure and the heat is rejected at constant volume as shown in Fig. 293. The lines BC and DE are assumed to be adiabatics. All the heat must then be absorbed along the line CD and all rejected along EB. Let the heat received along CD be represented by H_1, and the absolute temperature at C by T_c and at D by T_d; and let the heat rejected along EB be represented by H_2, and the absolute temperature at E by T_e and at B by T_b. Assume the weight of the charge to be constant for the cycle and equal to w. This assumption is permissible, since the fuel is oil and the increase in weight of the charge is small.

Then

$$H_1 = wc_p(T_d - T_c)$$

and

$$H_2 = wc_v(T_e - T_b).$$

The work done $= H_1 - H_2 = wc_p(T_d - T_c) - wc_v(T_e - T_b)$.

$$\text{Efficiency} = \frac{H_1 - H_2}{H_1} = \frac{wc_p(T_d - T_c) - wc_v(T_e - T_b)}{wc_p(T_d - T_c)},$$

$$= \frac{c_p(T_d - T_c) - c_v(T_e - T_b)}{c_p(T_d - T_c)},$$

$$= 1 - \frac{T_e - T_b}{\kappa(T_d - T_c)}. \tag{7}$$

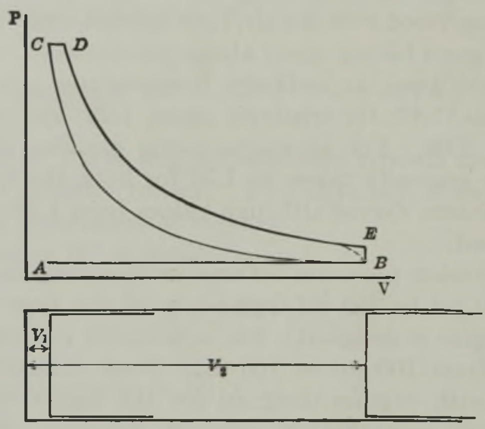


Fig. 293.—Diesel cycle.

Since CD is a constant-pressure line and EB a constant-volume line

$$T_d = T_c\frac{V_d}{V_c} \tag{8}$$

and

$$\frac{T_e}{T_b} = \frac{P_e}{P_b}. \tag{9}$$

But

$$P_e = P_d\left(\frac{V_d}{V_e}\right)^\kappa \tag{10}$$

and

$$P_b = P_c\left(\frac{V_c}{V_b}\right)^\kappa = P_d\left(\frac{V_c}{V_e}\right)^\kappa. \tag{11}$$

Substituting Eqs. (10) and (11) for P_e and P_b in Eq. (9),

$$\frac{T_e}{T_b} = \frac{P_d\left(\frac{V_d}{V_e}\right)^{\kappa}}{P_d\left(\frac{V_c}{V_e}\right)^{\kappa}} = \left(\frac{V_d}{V_c}\right)^{\kappa},$$

or

$$T_e = T_b\left(\frac{V_d}{V_c}\right)^{\kappa}. \tag{12}$$

Substituting Eqs. (8) and (12) for T_d and T_e in Eq. (7),

$$\text{Efficiency} = 1 - \frac{T_b\left(\frac{V_d}{V_c}\right)^{\kappa} - T_b}{\kappa\left(T_c\frac{V_d}{V_c} - T_c\right)}$$

$$= 1 - \frac{T_b\left[\left(\frac{V_d}{V_c}\right)^{\kappa} - 1\right]}{T_c\kappa\left(\frac{V_d}{V_c} - 1\right)}, \tag{13}$$

and since

$$\frac{T_b}{T_c} = \left(\frac{V_c}{V_b}\right)^{\kappa-1} = \frac{1}{r^{\kappa-1}},$$

$$\text{Efficiency} = 1 - \frac{1}{r^{\kappa-1}}\left[\frac{\left(\frac{V_d}{V_c}\right)^{\kappa} - 1}{\kappa\left(\frac{V_d}{V_c} - 1\right)}\right]. \tag{14}$$

From Eq. (14) it is seen that *the efficiency of the Diesel cycle depends not only upon the ratio of compression r, but also upon the ratio* $\frac{V_d}{V_c}$, *or the ratio of the volume at cutoff to the clearance volume.* It will be noted that the efficiency increases as this ratio decreases.

The dotted line in Fig. 293 shows the variation of the actual diagram from the theoretical. Here, again, the compression and expansion curves are practically adiabatics and in this case the value of κ is 1.4, since practically pure air is compressed in the cylinder. The fuel is injected at the end of the compression stroke and is burned at practically constant pressure. The value of κ for the expansion curve is generally assumed to be 1.4 also, although it is probably more nearly 1.35. With this cycle, no ignition device is used, the heat of compression of the air being sufficient to ignite spontaneously the injected fuel. Diesel-engine compression pressures are generally 450 psi to 500 psi, although, for some lower grade fuels, pressures as high as 600 psi are required.

Semi-Diesel engines may be designed to operate on cycles very closely approximating the Otto or the Diesel or may have one which is a combination of the two.

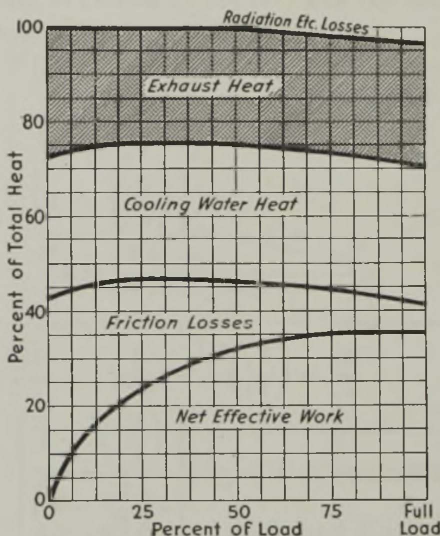


FIG. 294.—Heat balance of a typical Diesel engine.

In general, compression pressures of the semi-Diesel (fuel injection, spark ignition) are from 130 psi to 250 psi and peak pressures are well below that of a compression-ignition Diesel. The consequent lower stresses permit lighter parts to be used and the semi-Diesel may be made in many ways directly interchangeable with a gasoline Otto engine.

252. Efficiency and Losses.—Since of the heat energy that can be made available by combustion, only from 5 per cent to 35 per cent is delivered by the engine as useful mechanical energy, the remainder must be accounted for. This heat engine may be considered as a heat machine which in operation receives a constant inflow of potential energy transformed by combustion into heat energy. All of this energy flows out of the machine by various outlet channels. The most important outlet is the useful mechanical energy delivered by the engine. The heat energy leaving by way of the exhaust, owing to its high temperature and perhaps incomplete combustion, is usually the largest loss. Another loss of heat is that which is carried away by the cooling system and eventually delivered to the air. Heat is carried away also by direct radiation or conduction to the air moving past the hot engine parts.

Figure 294 shows how the heat leaving the engine by these channels varies with load conditions on a large Diesel engine. Because of the high expansion ratio of this engine, the exhaust temperature is relatively low; hence the proportion of heat loss through the exhaust is less than it would be in the case of a low-ratio Otto cycle engine.

253. Combustion and Flame Propagation.—One of the most complicated phenomena encountered in the study of the internal-combustion engine is the actual burning of the charge in an engine operating on the Otto cycle. The combustible mixture, which is usually heated before it enters the engine cylinder, absorbs more heat from the hot combustion-chamber walls, and then its temperature is further increased by the compression operation, until at the time of ignition this temperature is not far below the point at which combustion begins. Ignition is accomplished by raising the temperature of a small portion of the charge (that directly in the path of the electric spark) above the temperature at which the dissociation and subsequent oxidation of hydrocarbon fuel take place. The heat produced by the combustion of this small portion raises the temperature of the adjacent layer of mixture to the ignition point. Thus the flame is propagated in all directions away from the point of ignition until the whole charge is burned.

Although, in the Otto cycle, combustion is said to take place at constant volume, the temperature, pressure, and specific volume of each portion of the charge undergo a considerable change while the flame is sweeping across the combustion chamber. The changes in volume are illustrated by Fig. 295. At *A* is shown a combustion chamber filled with a charge divided into 10 equal portions. It is assumed that ignition occurs simultaneously along the whole wall at A. At B one-tenth of the charge has been burned and one-tenth of the heat energy liberated, raising the temperature and tending to raise the pressure of the combustion products. This one-tenth of the charge expands to equalize the pressure and occupies the space shown by the shaded portion, while the unburned nine-tenths are compressed somewhat into the remaining space. At C two-tenths have been burned and occupy practically one-half of the combustion chamber, while the remaining eight-tenths of the charge have been compressed into the other half. The flame, of course,

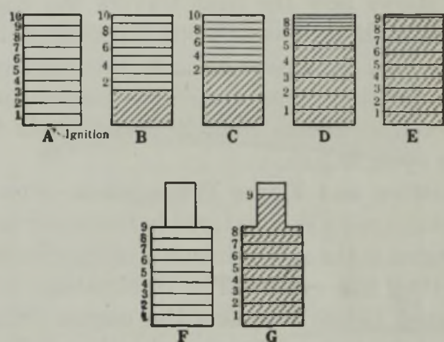


FIG. 295.—Diagram showing compression of unburned gases during combustion.

lies between the burned and the unburned sections of the charge. These diagrams thus show how each portion expands as it burns, compressing both the burned and the unburned sections of the charge until at E the last portion to burn has been tremendously compressed.

The rise in temperature due to this compression as well as that due to heat radiated from the flame may bring the unburned portion of the charge above its ignition point and virtually a spontaneous ignition of the unburned portion occurs. This causes the localized pressure of considerable magnitude in the region of the last portion to burn which is known as *detonation.* Characterized by a metallic "ping" due to high frequency vibration of piston and cylinder head, the detonation or "knocking" results in a loss of power, owing principally to the extremely high rate of heat radiation from the detonating gases. Figures 296 and 297 illustrate the differences in normal and detonating combustion. In Fig. 296 the flame travels across the combustion chamber to completion, with successive flame fronts being ignited in advancing flame. In the detonating sequence (Fig. 297) photograph number 13 marked $-0.2°$, shows the start of the spontaneous independent ignition. The following photograph, although taken only 2 deg of crank travel later, shows the whole unburned portion inflamed. The impact of the sudden pressure rise as well as the high localized temperature may eventually be destructive to the engine.

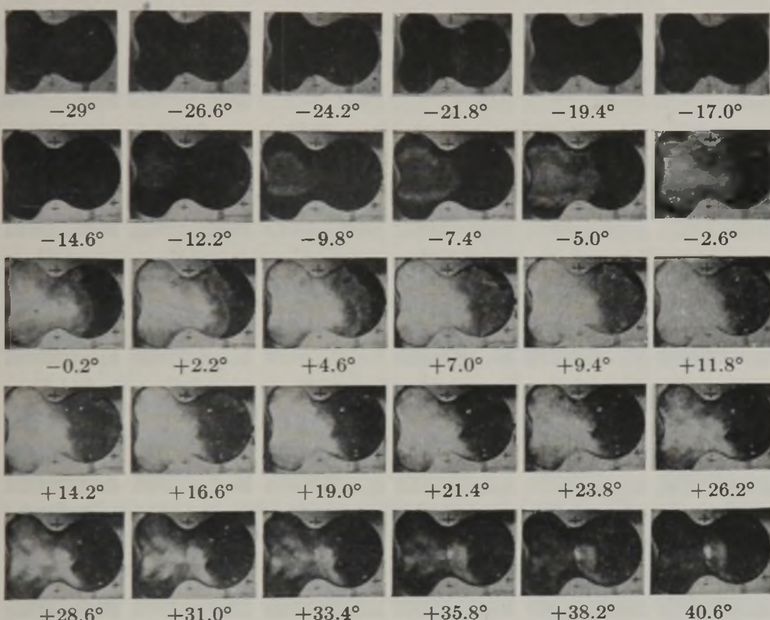


FIG. 296.—A nonknocking explosion in a gasoline engine running at 900 rpm.

As shown in Eq. (5), an increase in compression ratio will give increased engine efficiency; it also increases the temperature of the charge due to compression, and may well carry it to the point where the last portion of the charge to burn will detonate. Since the detonation is brought about by the last unburned portion exceeding its critical temperature, much can be done in engine design to reduce this temperature. The fundamental feature of combustion-chamber design to accomplish this is to provide the space in which the last portion of the charge is to burn with a high surface-volume relationship. The relatively cool piston head and combustion chamber walls may keep the temperature of the last portion below its detonating point. A chamber shaped as shown in Fig. 298*b* will permit a considerably higher compression ratio to be used without detonation of a given fuel than will one of the type of Fig. 298*a*. Obviously the characteristics of the fuel used are also major factors in controlling detonation. The arrangement of the atoms of carbon and hydrogen in a fuel molecule has been found to be of the greatest importance, and anticatalysts such as tetraethyl lead,

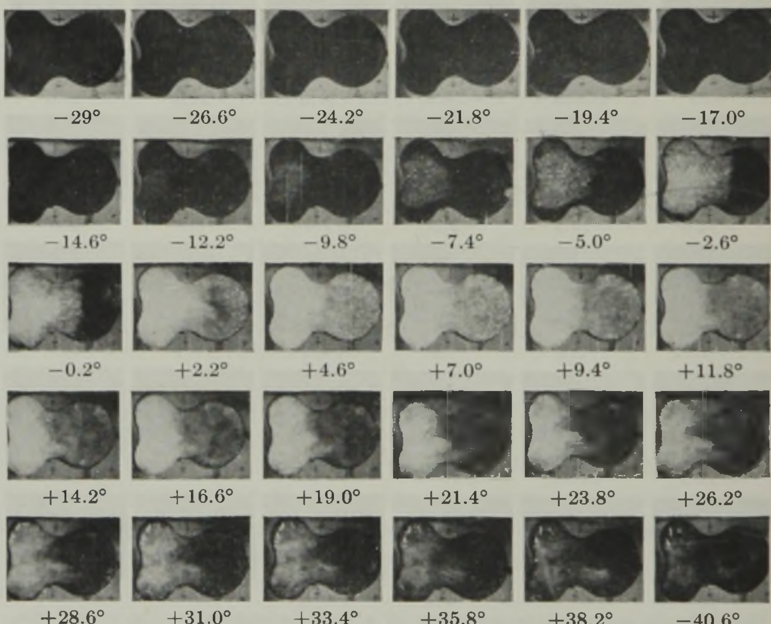
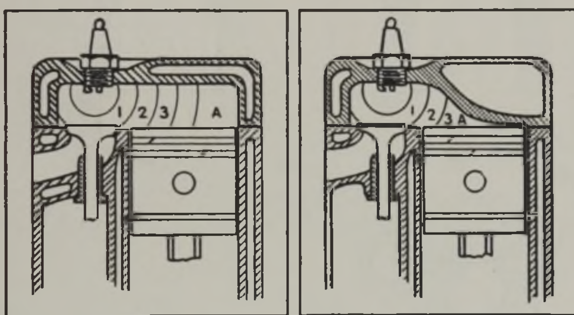


FIG. 297.—A knocking explosion in a gasoline engine running at 900 rpm.

iron carbonyl, iodine, and others markedly increase the anti-knock characteristics of the fuel.

Combustion in the Diesel cycle offers a somewhat different problem. Inasmuch as the fuel is not mixed with the air before compression, temperatures and pressures after compression may and, indeed, must be considerably higher than in Otto cycle engines. In the Diesel, liquid fuel is mechanically pulverized as it is sprayed into the combustion chamber filled with air at high temperature, but a finite time is required for the vaporization and ignition of the fuel. In high-speed Diesels this ignition lag becomes of major importance for, if a considerable quantity of fuel is injected before combustion of the first portion occurs, an extremely high rate of pressure rise will occur, resulting in a knocking or at least a rough-running engine.

It is also important that the fuel and air particles be thoroughly intermingled during an extremely short period of time; for this reason the shape of the combustion chamber of the Diesel engine has received a great deal of attention. The four drawings of Fig. 299 illustrate the principal types of chamber in use. Figure 299*a* is an example of the "open chamber" in which the nozzle spray pattern and combustion-chamber shape are matched in



(*a*) (*b*)

FIG. 298.—Combustion-chamber design for detonation control. (*a*) Poor; (*b*) good.

an effort to secure maximum dispersion of fuel in the air. Frequently the air is introduced into the cylinder tangentially and the whirling of the air is designed to facilitate this mixing. In *b* most of the air, upon compression, is forced into the turbulence chamber and given a high rotational velocity. Injection of fuel at the point shown produces the desired effect. An auxiliary air chamber may also be used as illustrated in *d*. Air is compressed into the auxiliary chamber and, after injection, when combustion has started and the piston moves downward, the rapid drop in cylinder pressure permits the air to flow out of the air chamber to mix and react with any remaining unburned fuel. A variation of this principle, known as the *energy cell*, is shown in *c*. Both fuel and air are present in the energy cell after injection has begun, but combustion starts in the open volume above the piston. The later combustion in the energy cell ejects the gases into the main chamber and forces a turbulence which results not only in a complete combustion but, because of the restriction at the exit of the energy cell, in a less rapid rate of pressure rise in the main combustion chamber.

The combustion characteristics of a high-speed Diesel engine can be illustrated by means of a pressure-time diagram such as

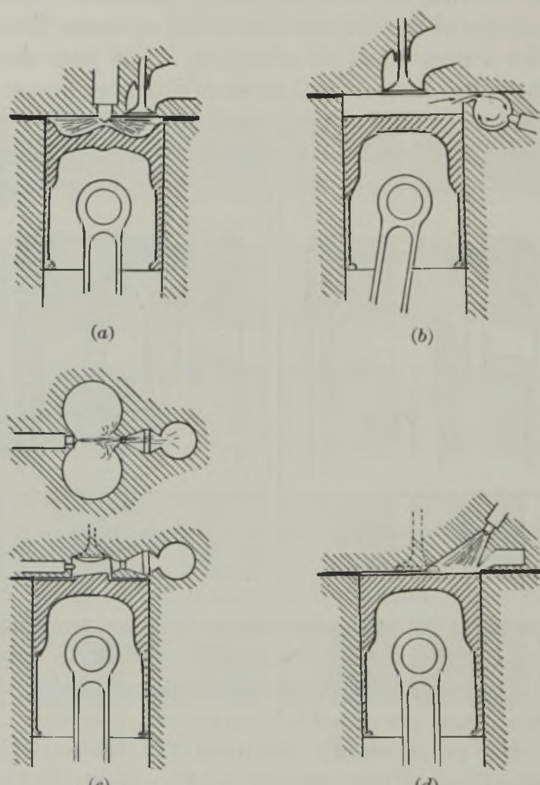


Fig. 299.—Types of Diesel combustion chambers. (a) Open chamber; (b) turbulence chamber; (c) energy cell; (d) air cell.

Fig. 300. A prolonged delay in the start of combustion after injection has begun will result in an excessively steep combustion line or high rate of pressure rise. It will be seen also that the modern Diesel engine deviates considerably from the theoretical Diesel cycle in the matter of constant-pressure burning. Only in the slower speed Diesel engines is the constant-pressure burning even approximated.

254. Arrangement of Cylinders and Timing.—The internal-combustion engine is peculiarly adaptable to multicylinder construction. The torque impulses transferred by a piston to the

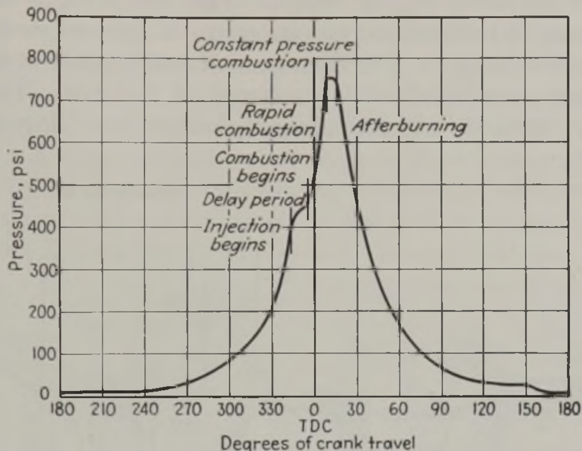


FIG. 300.—Pressure-time diagram of combustion in a high-speed Diesel engine.

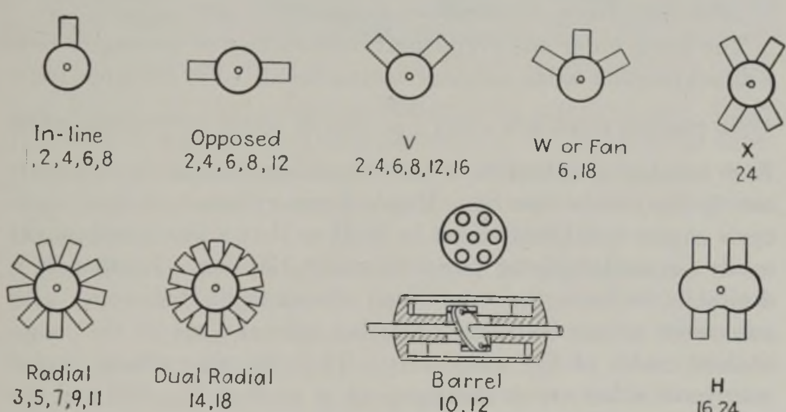


FIG. 301.—Methods of arranging cylinders in multicylinder four-cycle engines. The numbers below each type indicate the number of cylinders usually used in that arrangement.

crank in a combustion engine are necessarily more abrupt than in a steam engine; hence in order to smooth out the torque impulses during one revolution of the crank a number of cycles are overlapped, with the maximum impulse of successive cycles occurring at evenly spaced intervals of crank rotation. Since the power developed by any heat engine is a direct function of the quantity of medium it can utilize efficiently, combustion engines compromise on a relatively large number of small cylinders operating at high speed. This has brought attention to problems of balance and vibration which in slow-speed engines are of little moment. The arrangement and number of cylinders are closely associated with problems of balance. The more common methods of arranging cylinders are illustrated in Fig. 301.

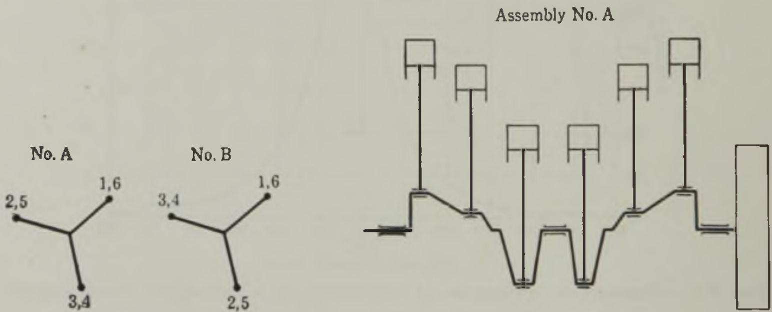


FIG. 302.—Figure of a six-cylinder crankshaft and piston assembly.

The firing order and crankshaft construction of the engine bear a direct relation to the cylinder arrangement. For example, four-cycle engines must fire every $\frac{720}{K}$ deg of crank revolution, when K = number of cylinders, if the torque impulses of the cylinders are to be evenly spaced. Hence a six-cylinder, in-line, four-cycle engine crankshaft must be built so that a piston is brought up to top center (firing position) every 120 deg. Further, it is desirable to have the crankshaft dynamically balanced, with successive torque impulses occurring fore and aft of the longitudinal center of the crankshaft. Thus the six-cylinder engine may have either crank arrangement *A* or *B* of Fig. 302 and the firing order will be 1-5-3-6-2-4 or 1-4-2-6-3-5, respectively. The alternate fore-and-aft firing reduces the torsional windup of the shaft which might otherwise be excessive. Even so, in every multicylinder engine crankshaft some torsional deflection occurs, and frequently the natural period of the shaft is in the range of engine power impulse rate and a severe torsional vibration is set up. Various dampeners have been devised to absorb this energy. Usually a light, friction-connected flywheel mounted on one or both ends of the shaft is used. A rigidly connected flywheel may change the period but will not dampen these torsional vibrations.

Radial four-cycle engines having a single throw crank of the type shown in Fig. 303 must always have an odd number of cylinders. Using the previous reasoning, a nine-cylinder radial engine must fire every 80 deg and, since there is an angle of 40 deg between cylinder axes, the firing order must be 1-3-5-7-9-2-4-6-8-1.

Torsional vibrations in radial engines have been very much reduced by hanging the crankshaft counterweight as a pendulum, the period of which corresponds to the rate of torque impulses when the engine is firing. Since both the rate of torque impulse

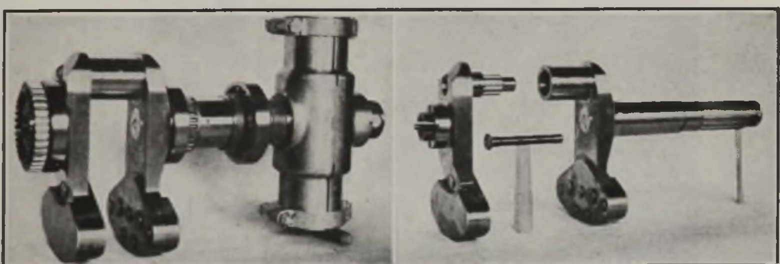


Fig. 303.—Crankshaft used in the Pratt and Whitney radial-type aircraft engine.

and period of the pendulum (neglecting gravity) are direct functions of engine revolutions per minute, the dampener is effective at all engine speeds. Part of the counterweight itself may be permitted to swing along a small radius and constitutes the dampener.

255. Valves.—Although many types of engine valves have been tried, the poppet type is now in almost universal use. It is closed by a spring and opened by means of a rotating cam and a cam follower, generally termed a *valve lifter*. The lifter is either of the mushroom type operated by a cam with curved flanks, or a roller-type lifter which may be used with a tangential cam having straight flanks. Each valve is opened once per cycle. If the engine has the cylinders built in line, it will have a long camshaft, revolving at one-half crankshaft speed, with a separate cam to operate each valve. If the engine is of the radial type, it will use a cam gear like that shown in Fig. 304. On the periphery of this gear are four lobes to operate the inlet, and four to operate the exhaust valves. The cam gear operates at one-eighth crankshaft speed; each lobe will operate a valve for each of the nine cylinders every revolution. As the lobe moves under a roller, it forces the valve lifter and the long push rod outward, moving the rocker arm on its pivot pin. The rocker arm pushes the valve inward and opens the passage into the cylinder.

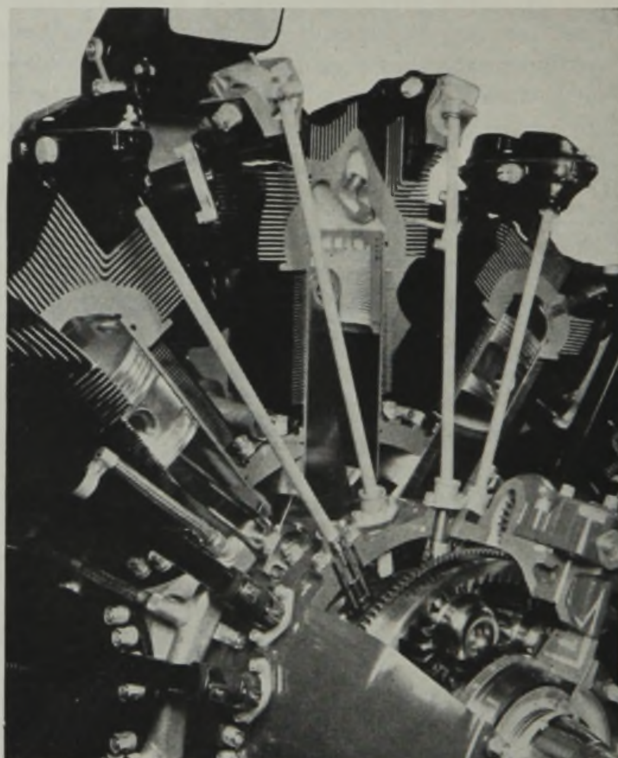


Fig. 304.—Radial engine with cam gear.

The cutaway section of the exhaust valve of Fig. 304 shows it to be hollow. This is quite general practice in high-performance engines because of the difficulty in cooling the exhaust valve. The hollow portion of the valve is filled with metallic sodium which melts at a temperature lower than the operating temperature of the valve and then acts as a liquid to transfer heat from the valve head to the stem, whence the heat may be more readily dissipated.

The actual passage through a poppet valve is a small annular opening between the valve head and its seat. At high speeds the exhaust gases are expelled and the charge drawn in through the port at velocities of several hundred feet per second. Acceleration and deceleration of even the short columns of gases in the intake and exhaust pipes become of considerable importance and the valves are timed accordingly.

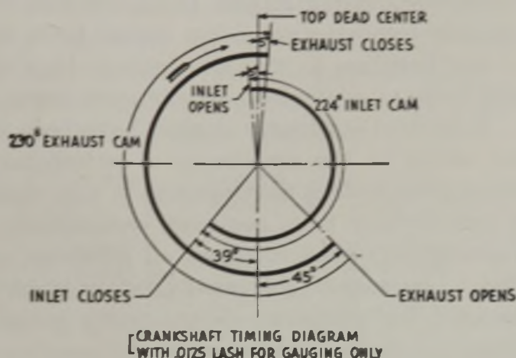


FIG. 305.—Typical valve-timing diagram for a high-speed four-cycle gasoline engine.

Figure 305 shows a timing diagram for a high-speed automobile engine. The inlet valve is opened slightly before top center, and the exhaust valve is still slightly open. The inertia of the receding gases in the exhaust pipe helps to scavenge the combustion chamber and to start the flow of inlet gas. The intake valve remains open 39 deg past the lower center to take advantage of the inertia of the incoming gases in charging the cylinder. The exhaust opening is timed 45 crank deg before the end of the expansion to allow the greater part of the products of combustion to escape owing to their own pressure, hence reducing the pressure on the piston as it rises on the exhaust stroke.

CHAPTER XIV

FUELS AND FUEL SYSTEMS

256. Internal-combustion Engine Fuels.—Solids, liquids, and gases are possible internal-combustion engine fuels; difficulty of preparation and handling as well as relatively high ash content of such solid fuels as coal virtually eliminate them from consideration. Coal-fired stationary engines have been operated in Europe, but owing to the abundance of petroleum there has been no stimulus for such a development in this country. Gas engines are considerably more common, economically using the gaseous by-products of blast furnaces, oil refineries, and similar plants as fuel. Gas engines are usually rather large, stationary installations used for pumping, electric-power generation, and similar service, but in recent years some automotive engines in Europe have been adapted to use the hydrocarbon gases obtained from destructive distillation of wood. Other vehicles carry a large balloon filled with illuminating gas as a fuel supply. Both developments are the result of restrictions on the use of petroleum. A relatively small number of gas-fueled vehicles are operated in this country on butane stored at high pressure in steel bottles. The expansion of the butane through a throttling valve lowers its temperature, so the fuel may first be used as a refrigerating medium to cool a bus or truck, then mixed with air and fed to the engine as fuel.

Liquid fuels, however, are more easily handled in vehicles than either solid or gaseous fuels, and other than the little-known atomic energy source, offer the lightest, most compact form of energy storage yet discovered. With existing engines, 1 gal of Diesel fuel oil, weighing but 7 lb may produce 30,000,000 ft-lb of shaft work. The liquid fuels most commonly used are obtained from petroleum, coal, or vegetable matter. Table XXXIII gives some of the more important physical and chemical properties of typical samples.

Crude petroleum is composed of a large number of different hydrocarbons having different boiling points. Most oil wells first produce natural gas, actually light hydrocarbons with boiling points below atmospheric temperature at atmospheric pressure. The heavier ends of crude petroleum have boiling points of 700 deg or higher so that they break down (crack) at atmospheric pressure before boiling. The gaseous light ends are frequently burned near the well, but may be compressed, liquefying a portion that is later blended with heavier fractions in gasoline. *Straight-run gasoline* is that portion which is distilled off from 85 F to 450 F. Kerosene is the 250 F to 600 F "cut"; the successively heavier fuel suitable for Diesel engines, the "cut" from 300 F up. The heavier oils, and those suitable for lubricants, are distilled under vacuum.

TABLE XXXIII.—LIQUID FUELS

Source and name	Specific gravity, H_2O= 1.00	Deg. Bé	Boiling range, deg F	Latent heat, Btu per lb	Higher heating value, Btu per lb	Lower heating value, Btu per lb	Per cent by weight			
							H	C	O_2	H_2O
Petroleum										
Naturalene	0.66	80	70 to 300	...	......	19,300				
Gasoline	0.75	59	120 to 400	135	20,450	19,150	14.84	84.65	0.51	0.57
Kerosene	0.80	45	350 to 550	110	19,900	18,650	14.37	85.25	0.38	0.43
Furnace oil	....	..	....	...	19,690	18,450	15.60	83.90	.....	0.50
Fuel oil	0.93	21	....	...	......	17,800				
Coal										
Benzol	0.87	30	176	160	......	17,400				
Vegetable Matter										
Ethyl alcohol	0.80	45	171 to 212	406	......	11,480	.....	.....	.....	1.50
Denatured alcohol	0.81	43	171 to 212	442	11,618	10,500	12.68	47.08	40.24	9.00
Methyl alcohol	0.83	39	151	500	......	9,630				

Because of the heavy demand for gasoline in relation to other petroleum products, the supply of straight-run gasoline is insufficient and other methods are used to produce it, notably cracking, polymerization, and hydrogenation. The first consists of heating the heavier hydrocarbons under pressure or in the presence of a catalyst which causes the molecule to break down forming two or more lighter molecules. Polymerization is the reverse and is applicable to the lighter hydrocarbons that would otherwise be too volatile for use as a combustion-engine liquid fuel. As the name implies, the hydrogenation process involves the addition of one or more hydrogen atoms to the fuel molecule. In general the detonating tendency of a straight-run gasoline is determined by the source of the crude oil from which it is made and the temperature range of cut, but the general practice of blending straight run with gasolines made by the other methods usually improves its antiknock quality. Kerosene, which was once the principal product of petroleum, is of little use as a combustion-engine fuel. Its low volatility and poor antiknock quality limit it to use in low-compression Otto engines. The heavier fractions of petroleum used as fuel for high-speed Diesel engines are as carefully refined as gasoline, and various qualities such as vaporization and ignition-delay characteristics must be controlled by the refining process.

Benzene, a coal derivative, is an excellent Otto cycle fuel from a detonation standpoint, but its high freezing point, 41 F, makes it undesirable for use in cold weather. Blended with gasoline, the mixture has a reasonably low freezing point and little tendency to detonate.

Alcohol, because of its high oxygen content, has a heating value much lower than that for gasoline. Nevertheless, because of its ability to withstand a very high compression and to recover a greater quantity of waste heat during vaporization, it can deliver almost as much power per pound of fuel as can gasoline. The high latent heat requires unusual heating devices to secure proper vaporization. Commercial denatured alcohol may have a water content from 5 to 10 per cent. Mixing gasoline with alcohol permits the use of a higher compression than with gasoline alone and also gives the necessary volatility for cold starting conditions. Although it has not as yet been economically possible to use alcohol in this country, it has been widely used in Europe and in tropical countries, and may be used in this country when the petroleum supply has been sufficiently diminished. When the petroleum reserves begin to fail, it will undoubtedly become profitable to make alcohol and to attack the shale-oil deposits which greatly exceed the known deposits of petroleum.

257. Heating Value of Fuels.—Usually it is best to determine the heating value of gas or liquid fuels by use of either the Junkers or some type of bomb calorimeter. In handling a very volatile fuel it is necessary to seal it into a glass bulb before weighing it. If the percentages of hydrogen and carbon are known, the heating value can be computed roughly by use of a formula similar to that used for coal (see page 104) giving the result in Btu per pound. Wherever carbon and hydrogen are in chemical union, a certain amount of heat is required for the dissociation of the elements before they can combine with the oxygen of the air. The amount of this heat varies with the constitution of the hydrocarbon.

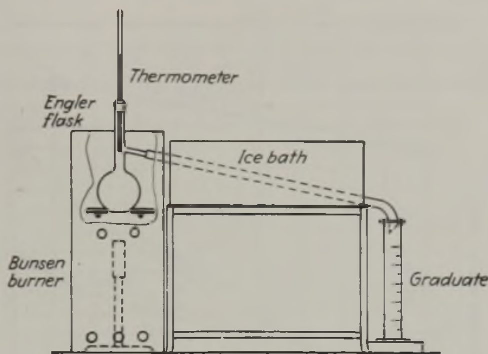


FIG. 306.—A.S.T.M. distillation apparatus.

The following equation has been developed for computing the higher heating value of most hydrocarbons of the general composition C_nH_m.

$$\text{Heating value per lb} = 14{,}600C + \frac{6n + 2}{4n + m} \times 52{,}200H. \quad (1)$$

Note that the heat of formation of the hydrocarbon, being related to the number of hydrogen atoms, is made a function of the hydrogen constituent and the calorific value assigned to the hydrogen is correspondingly less than that for free hydrogen.

Example.—Determine the higher heating value in Btu per pound of methane, CH_4.

Solution.—Methane is composed of 75 per cent carbon and 25 per cent hydrogen and in this case $n = 1$ and $m = 4$, hence, substituting in Eq. (1), the heating value of methane would be

$$14{,}600 \times 0.75 + \frac{6 + 2}{4 + 4} \times 52{,}200 \times 0.25 = 24{,}000 \text{ Btu per lb.}$$

This value agrees with experimental results.

258. Vaporization of Fuels.—In order to distinguish between the vaporization characteristics of various gasolines, kerosene, and fuel oil, the American Society for Testing Materials has set up a standardized distillation test. One hundred cubic centimeters of the fuel is distilled in a carefully specified manner in an apparatus, as illustrated in Fig. 306. The temperature at which the various portions are distilled is recorded and plotted. Figure 307 shows typical A.S.T.M. distillation characteristics of several fuels. This test does not indicate directly, however, the

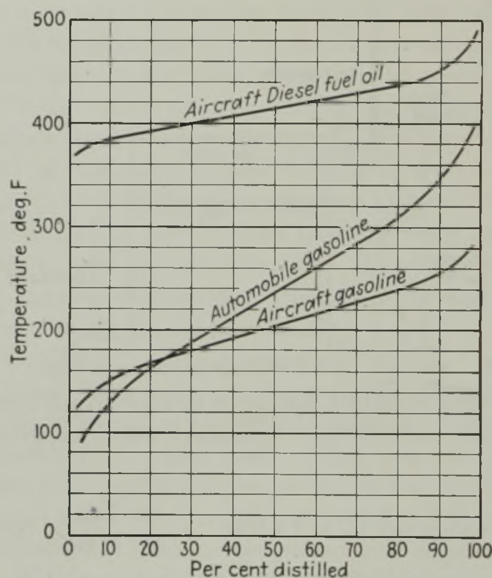


FIG. 307.—A.S.T.M. distillation characteristics of typical fuels.

degree of fuel vaporization in an engine. The latter is affected by

1. The temperature of the fuel.
2. The surface of the fuel exposed to heat.
3. The absolute pressure.
4. The time allowed for heat transfer and vaporization.
5. The degree of saturation of the air in contact with the fuel.

Because of a combination of these factors, the automobile engine gasoline shown in Fig. 308, for example, may be 98 per cent vaporized in a manifold at 13:1 air-fuel ratio, at atmospheric pressure and 120 F.

The heat necessary to vaporize the fuel is obtained from the incoming air, thus lowering the air temperature and possibly precipitating the moisture from the air. The combination of moisture and low temperature may sometimes form ice within the carburetor in sufficient quantity to choke off normal operation. To overcome this, heat from the exhaust gases is supplied to the mixture through a manifold hot spot. The latter is usually controlled either automatically or manually, because excessive heating of the intake would reduce the density of the charge and hence the power output of the engine.

In both the Otto and the Diesel engine the surface of fuel exposed is increased by spraying it with the air in finely divided

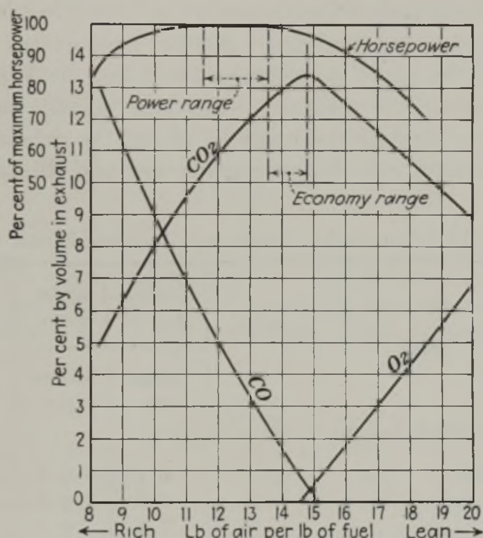


FIG. 308.—Effect of mixture ratio on engine power and exhaust gas composition for an average gasoline.

particles; but only in the Otto cycle engine is it possible to reduce the pressure by partially closing the throttle. Under light load the pressure in the intake manifold may be reduced to less than half an atmosphere.

The time in which vaporization must take place is very short. For a four- or eight-cylinder engine running at 2,000 rpm the time elapsed from the instant the fuel is sprayed from the carburetor nozzle until it has been completely burned is only about 0.06 sec. For a Diesel engine operating at the same speed, the time from the beginning of injection until combustion is complete should be only 0.005 sec. Exceedingly fine atomization by the fuel-spray nozzles together with very high temperatures is absolutely necessary in order to complete vaporization and combustion in such a short time.

It is evident that, although waste heat may be used to aid vaporization when the engine is running, it is not available for starting the cold engine. For starting, either heat must be supplied from some outside source, such as an electrical heater, or a more volatile fuel must be used until the engine has become warm.

Gasoline used for automobiles includes in its content a small portion of volatile fuels which vaporizes at a very low temperature and, for cold starting, perhaps 1,000 per cent excess fuel is provided so that the small percentage which can be vaporized will furnish a combustible mixture. The less volatile parts of the fuel merely run through the engine and out the exhaust or, passing by the pistons, run into the crankcase and dilute the lubricating oil.

TABLE XXXIV.—EXPLOSIVE LIMITS OF GASEOUS FUELS WITH AIR

Fuel	Ratio of air to gas by volume		Theoretical ratio of air to gas by volume
	Excess air	Excess gas	
Carbon monoxide	5.1	0.3	2.4
Hydrogen	9.6	0.5	2.4
Water gas	7.1	0.5	2.4
Acetylene	28.8	0.9	12.0
Coal gas	11.6	4.2	5.7
Ethylene	23.4	5.8	14.4
Alcohol, ethyl	24.3	6.3	14.4
Marsh gas	15.4	6.8	9.3
Ether	35.7	12.0	28.4
Benzene	36.7	14.4	36.0
Pentane	40.7	19.4	37.5

Heat applied to the fuel, after the air has been trapped in the cylinder by the closing of the intake valve, cannot reduce the weight of the charge drawn in or the power output of the engine. In many engines, fuel is wholly or partially vaporized by spraying it on a hot plate or metal surface in the combustion chamber. Obviously such a hot surface must be artificially heated for starting. Such engines, which use crude oil just as it comes from the wells, have been placed on the market and have given fair satisfaction. The difficulty in using crude oils is found in taking care of the heavier and less volatile portions such as paraffin and asphalt. As a general rule this difficult vaporization and the value of the by-products available in the heavier products have made it advisable to refine the oil burned in such engines.

259. Fuel Mixtures.—The mixture ratio of air and fuel used for internal-combustion engines is very important. For each fuel only a certain range of mixture ratio is explosive. For each fuel there is a mixture ratio giving the engine the maximum power, and still another ratio giving the highest efficiency. Table XXXIV gives the explosive limits for several gases. Experiments show that for gaseous fuels the best results are obtained when the air in the cylinder is slightly in excess of the theoretical mixture ratio.

Example.—A typical hydrocarbon of the paraffin series whose density and boiling point are about those of the average gasoline burns according to the equation $C_9H_{20} + 14O_2 = 9CO_2 + 10H_2O$. Find the theoretical air-fuel ratio.

Solution.

	$C_9\ H_{20}$	$+\ 14O_2$	$=\ 9CO_2$	$+\ 10H_2O$
Atomic weights	12 1	16	12 16	1 16
Molecular weights	128	32	44	18
Formula weights	128	448	396	180

Since the air contains 23 per cent oxygen by weight, the weight of air required to burn one pound of fuel is

$$\tfrac{448}{128} \times \tfrac{100}{23} = 15.2 \text{ lb.}$$

That is to say, the air-fuel ratio with excess of neither fuel nor air is 15.2.

If the composition of the hydrocarbon fuel is known in percentage by weight, the ratio may be computed as follows: The average gasoline mentioned above contains 84.4 per cent carbon and 15.6 per cent hydrogen. The weight of air required to burn 1 lb is then

$$\frac{84.4}{100} \times \frac{2.66}{0.23} + \frac{15.6}{100} \times \frac{8}{0.23} = 15.2 \text{ lb.} \quad \text{(See Art. 65.)}$$

It is to be noted that for every pound of fuel burned there is formed $\tfrac{180}{128} = 1.4$ lb of steam, which condenses to considerably more than 1 gal of water per gal of fuel. Airships use this fact to advantage by condensing water from the exhaust of their engines to replace the weight of fuel burned and to prevent the necessity of valving off any of their expensive helium to maintain the desired altitude. For open-throttle operation the ratio computed above usually gives the highest efficiency. For part loads, where more heat can be applied to the mixture, a greater ratio of about 18 will give the highest efficiency.

The effect of the air-fuel mixture on the power developed by the engine and the composition of the exhaust gas is illustrated in Fig. 308. It will be noted that if about 25 per cent excess fuel is used, the power output of an engine using liquid fuel will be considerably increased. To explain this we may consider the internal-combustion engine to be an air engine, as indeed it is when burning liquid fuel. If it uses the normal mixture ratio of 15 lb of air to 1 lb of fuel, it is really using about 10,000 volumes of air to 1 volume of liquid fuel. The power output is limited by what is often called the *breathing capacity* of this air engine. If 1 lb of air is used to burn carbon, about 1,260 Btu heat energy is developed; if that pound of air is used to burn hydrogen, about 1,800 Btu are obtained. It is therefore possible to use an excess of fuel, burning all of the hydrogen with its greater heating value and discarding some of the carbon, having a lesser heating value, as carbon monoxide or free carbon which appears as a black smoke. By this means a greater heat-energy input per working stroke is obtained, resulting of course in a greater power output. Thus a mixture of 12 lb of air and 1 lb of fuel will produce the maximum of power, burn more rapidly, and require least spark advance.

260. The fuel system includes a storage tank, shut-off valve, strainer or filter, settling chamber, a pump, and the carburetor or Diesel fuel pump, together with the necessary copper or steel tubing. To reduce fire risk the tank should be located at some distance from the hot engine. In an automobile it is usually at the rear; in an airplane in the wings or fuselage; in stationary-engine service it is usually buried in the ground. If it must be near the engine, it should be protected by a fire wall. It is possible to locate the supply tank so that fuel will flow to the engine by gravity; usually some type of pump is used which either furnishes fuel to the carburetor at a controlled low pressure, or supplies a much larger quantity than is needed with the excess fuel returning to the supply tank. For the latter arrangement a gear pump similar to that described in the lubrication system is used. The A. C. fuel pump shown in Fig. 309 is bolted to the engine crankcase with the rocker arm *A* extending inside and lying against an eccentric on the camshaft. Movement of the rocker arm which is pivoted at *B* will pull down the pull rod *C* together with the diaphragm *D*, which is held between metal disks, compressing the spring *E* and producing a vacuum in the pump chamber *F*. Fuel from the supply tank enters the glass sediment bowl *G* and then passes through the strainer *H* and the

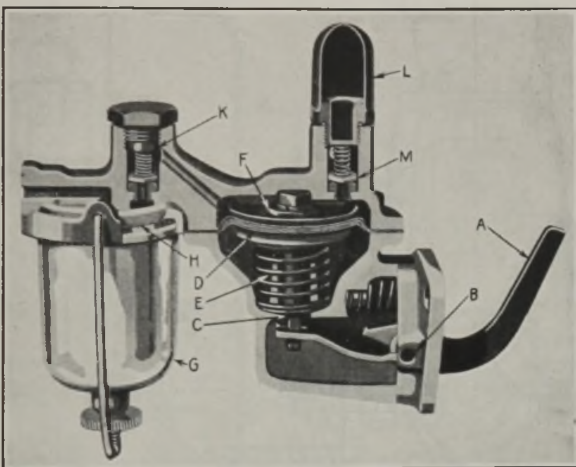


FIG. 309.—A.C. fuel pump, Type E.

inlet valve *K* into the pump chamber *F*. On the return stroke, the spring *E* pushes the diaphragm upward, forcing fuel from the pump chamber through the outlet valve *M* and out to the carburetor. When the carburetor float chamber is filled, the float will close the inlet needle valve (not shown), stopping the fuel flow and the upward motion of the diaphragm. Owing to the spring pressure, the diaphragm maintains a steady definite pressure on the fuel line until the carburetor requires further fuel and the needle valve opens. A vapor dome *L* reduces pulsation at the pump outlet. Such a pump supplies fuel to the engine at a predetermined pressure and operates only when the engine is running.

Figure 310 shows a diagram of the fuel system used by a typical Diesel engine. Fuel is pumped from the storage tank through a strainer into the metering-pump reservoir *A* by the fuel-oil service pump which is operated by an eccentric on the engine crankshaft. The service pump supplies more fuel than

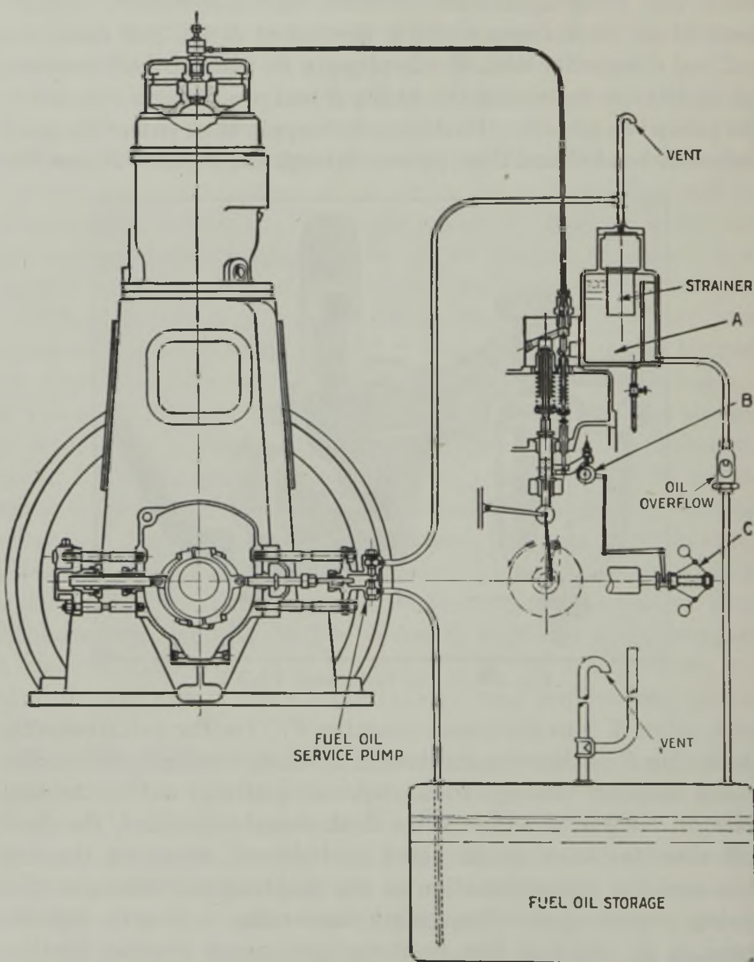


Fig. 310.—Diagram showing system of fuel distribution to a Worthington solid-injection Diesel engine.

the engine requires and a constant level of fuel is maintained in the metering-pump reservoir with the excess fuel being carried back to the fuel-oil storage tank through the fuel-oil overflow. Vents are provided for each tank or reservoir to allow the escape of any air that may become entrained in the fuel as it is pumped through the system.

261. Carburetors.—To ensure proper combustion in the engine cylinder, it is necessary to mix the fuel and air in the proper proportions. For gaseous fuel, a mixing valve, which obtains the proper mixture ratio by adjustment of the fuel and air orifices, is sufficient. For liquid fuels it is necessary to atomize or pulverize the fuel to obtain a homogeneous mixture and, by reason of the greater fuel surface exposed, to increase the rapidity of vaporization. Atomization of the fuel is accomplished by

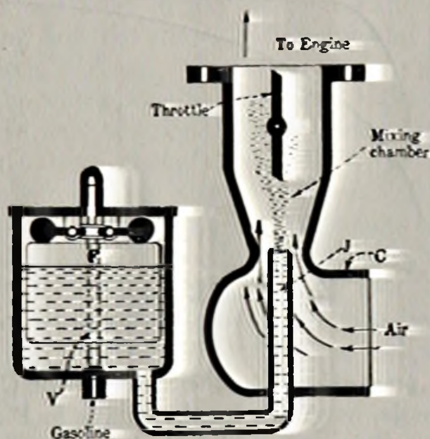


FIG. 311.—Diagram of simple carburetor.

spraying the fuel through a small nozzle into a swiftly moving air stream.

Figure 311 is a diagram of a simple carburetor having a single air orifice or Venturi C and a single fuel nozzle J. The float operates the inlet needle valve V to maintain a constant level of fuel in the float chamber. This level is far enough below the tip of the fuel nozzle so that the engine, when not running, may be tilted through a considerable angle without causing fuel to flow out of the nozzle. In operation the engine acts as a pump and draws air through the carburetor and intake manifold into the cylinders. It thus creates a partial vacuum in the carburetor mixing chamber. Air flow through the Venturi is then due to the difference between the pressure in the mixing chamber and the atmosphere. Since the float chamber has a vent connecting it with the atmosphere, the flow of fuel through the fuel passages is caused by the same pressure difference that causes air flow through the Venturi.

The general expression for the flow of fluids is

$$V = C\sqrt{2gh},$$

where V is the velocity of flow in feet per second, C a coefficient, and h the head, or difference in pressure producing the flow. To determine the fuel flow, h is expressed in feet of fuel, and for air flow, in feet of air. The variation of C for air flow through the

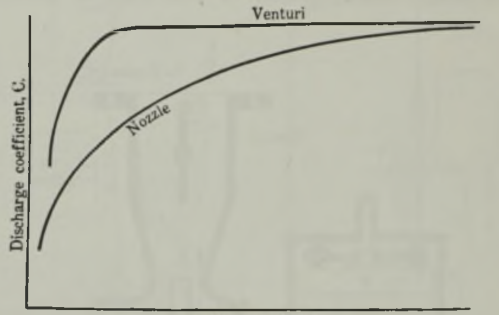


FIG. 312.—Relative variation of discharge coefficients for air in Venturi and fuel in nozzle.

Venturi is not proportional to the variation of C for fuel flow through the nozzle, as shown by the curves in Fig. 312. Also, some suction head, *i.e.*, some air velocity, is required to lift the fuel in the nozzle and to break the surface tension of the liquid. As a result, an increase in the flow of mixture from such a carburetor causes an increase in the proportion of fuel in the mixture, *i.e.*, the mixture delivered by the simple carburetor, consisting essentially of a single air orifice and a single fuel nozzle, becomes richer and richer as the engine speed or load is increased.

Although such a carburetor, being suitable only for an engine operating at constant speed and load, is practically never used, it is important to know its operating characteristics since most modern carburetors are made up of a simple carburetor plus one or more devices for correcting for the improper action of the simple carburetor.

These carburetors may be classified as follows according to the method of compensating for the major deficiencies of the simple carburetor:

1. Auxiliary air-valve carburetors which open up additional air passages as mixture flow increases.

2. Plain tube or air-bled jet carburetors which allow a controlled flow of air through the fuel passages in place of fuel as the mixture flow increases.

3. Expanding carburetors which open up or expand both the air and the fuel passages as the mixture flow increases.

It may be stated briefly then that carburetors of the first class correct mixture proportions by varying the air flow, in the second by varying the fuel flow, and in the third by varying both. An analysis of the operation of any of these carburetors may be made by finding first the simple carburetor and then the compensating devices.

Because of the wide operating range and flexibility required of a carburetor, various additional devices must be employed for satisfactory operation: choke valve, idle by-pass, accelerating device, and on aircraft, an altitude control. The use of each of these is dictated by the mixture requirements of the engine.

1. *Choke Valve.*—This is usually a butterfly valve near the air entrance to the carburetor which shuts off most of the air supplied and allows nearly full manifold vacuum to be applied to the fuel jet. The resulting mixture may be as rich as 1 lb of fuel per lb of air. Since in starting a cold engine, only a small portion of the fuel will vaporize, the resulting air-vapor ratio will be within the range of ignitibility. The same result is obtained in aircraft engines by means of a primer pump which injects fuel directly into the intake pipe before starting.

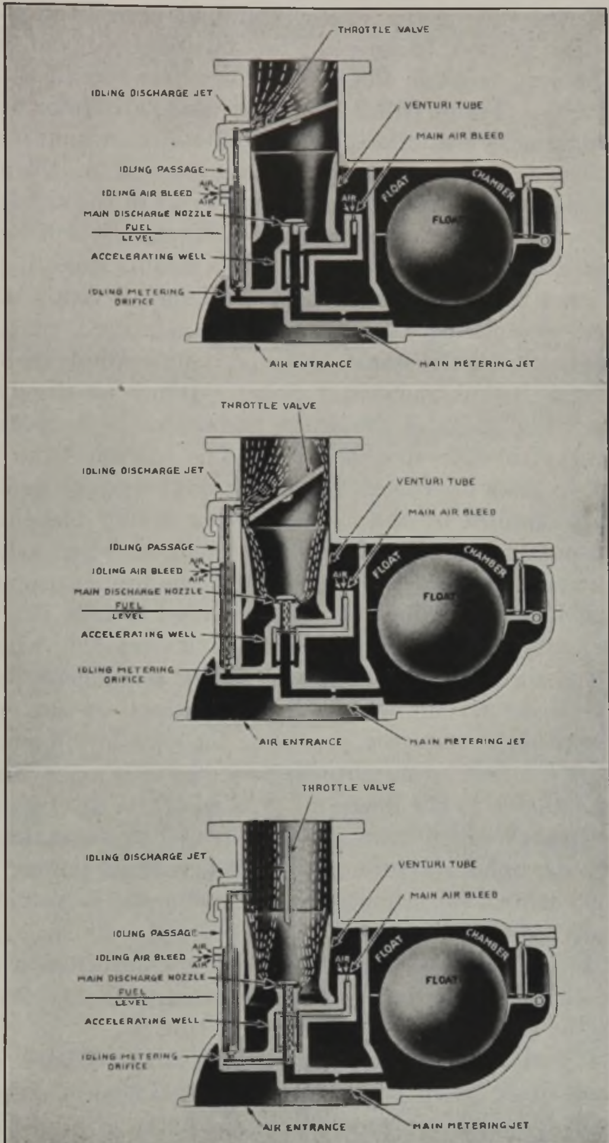
2. *Idle By-pass.*—In very low throttle operation the pressure drop of the air through the Venturi may be insufficient to draw fuel from either the main or the compensating nozzles and the high velocity, hence low static pressure, of the air past the edge of the throttle must be used to supply the engine with fuel. A direct passage of fuel from the float chamber is therefore provided, with a manually adjustable air blend. For best operation this is adjusted to give an air-fuel ratio of approximately 12.5:1, as the most easily ignitible, and releasing the most thermal energy per pound of air. Idle operation results in an excessive dilution of the fresh charge with residual burned gases in the cylinder, and maximum ignitibility is an essential requirement of the fresh charge.

3. *Accelerating Device.*—Obviously the mixture supplied for rapid acceleration should be of maximum ignitibility and thermal energy per pound of air. If the throttle is suddenly opened, the rate of air flow increases very rapidly, but the flow of fuel tends to lag because of its greater density and inertia. The resulting lean mixture may cause the engine to misfire and perhaps stall at a time when maximum power is demanded. In order to overcome this, most carburetors are provided with a positive displacement pump linked to the throttle, so that upon sudden opening of the throttle a small amount of fuel is forced into the air stream. This action not only provides time for the fuel in the nozzle passages to accelerate, but also provides the maximum power ratio during that time. Some carburetor designs omit the positive displacement pump and simply provide a small "well" or reserve of fuel which is dumped into the air stream upon acceleration. Auxiliary air-valve carburetors are usually provided with a dash pot on the air valve to restrict air flow temporarily.

4. *Altitude Control.*—Whenever carbureted engines are operated over a wide range of altitude, the diminishing density of air with higher altitude increases the weight ratio of fuel to air, *i.e.*, the mixture becomes richer and additional compensation is necessary. As higher altitudes are reached this may be accomplished either manually or automatically with an aneroid control by:

1. Opening additional air passages.
2. Restricting the flow of fuel by;
    a. reducing the air pressure on the fuel in the float chamber.
    b. restricting an orifice through which the fuel must pass.
    c. moving the entire Venturi so that the throat of the Venturi is drawn away from the fuel nozzle.

Figure 313-1 is a sectional view of the Stromberg plain tube carburetor used on aircraft engines. The throttle is shown in the very nearly closed position existing when the engine is running idle at a low speed. Fuel is being drawn from the float chamber through the main metering jet and the idling metering orifice. It is then mixed with air coming in through the idling air bleed, and is finally sprayed into the air stream where it is passing the edge of the throttle at a very high velocity due to



1.

2.

3.

Fig. 313.—Three views of Stromberg carburetor.

the high vacuum in the intake manifold above the throttle. Atomization is good, because the fuel has been sprayed into the air at the only point in the carburetor where the air attains a high velocity. Figure 313-2 shows the same carburetor with the engine running at medium speed. Fuel is discharging from the main nozzle as well as from the idling by-pass. As the suction on the main nozzle increases, the level of fuel in the accelerating well is lowered, uncovering more holes through which more air is bled into the fuel passage. Since this minute quantity of air goes through the fuel passage in place of fuel, it tends to make lean what would otherwise be too rich a mixture. Should the air bleeds be closed, the whole device becomes a simple carburetor and delivers the characteristic richer mixture as the demand increases. Figure 313-3 shows the carburetor as it operates at full power with wide-open throttle. The suction is no longer sufficient to draw fuel from the idling discharge jet and all of the fuel is supplied by the main jet. Air is now bleeding into the main-nozzle fuel passage from the idling air bleed as well as from the main air bleed and a mixture of the proper proportions is delivered to the engine. It will be noted that in this figure the accelerator well is empty whereas it was full when the engine was running idle. If the throttle is suddenly opened when the engine is running slowly, the increased suction empties the accelerating well at once, providing the necessary momentary rich mixture. Just as the throttle becomes fully open, a needle valve (not shown in the diagram) is lifted off its seat, opening a passage through which extra fuel is delivered to the main nozzle to supply the richer mixture needed for maximum power.

Altitude control (not shown) on this carburetor is provided by a manually adjusted valve which connects the air space above the fuel in the float chamber to the low-pressure region at the Venturi, thereby lowering the total pressure on the fuel and reducing the flow through the main metering jet.

Aircraft carburetor operation at varying angles of bank, climb, and glide is made possible by providing a dual float chamber, the mean level of which is always at the discharge level of the carburetor nozzle. Inverted flying during a rapid maneuver causes no particular carburetion difficulty because centrifugal force replaces the action of gravity in keeping the fuel level in proper relation to the discharge jet. For continuous inverted operation special provision must be made by either a manually adjustable fuel passage into the air stream or a complete carburetor unit inverted in relation to the normal unit.

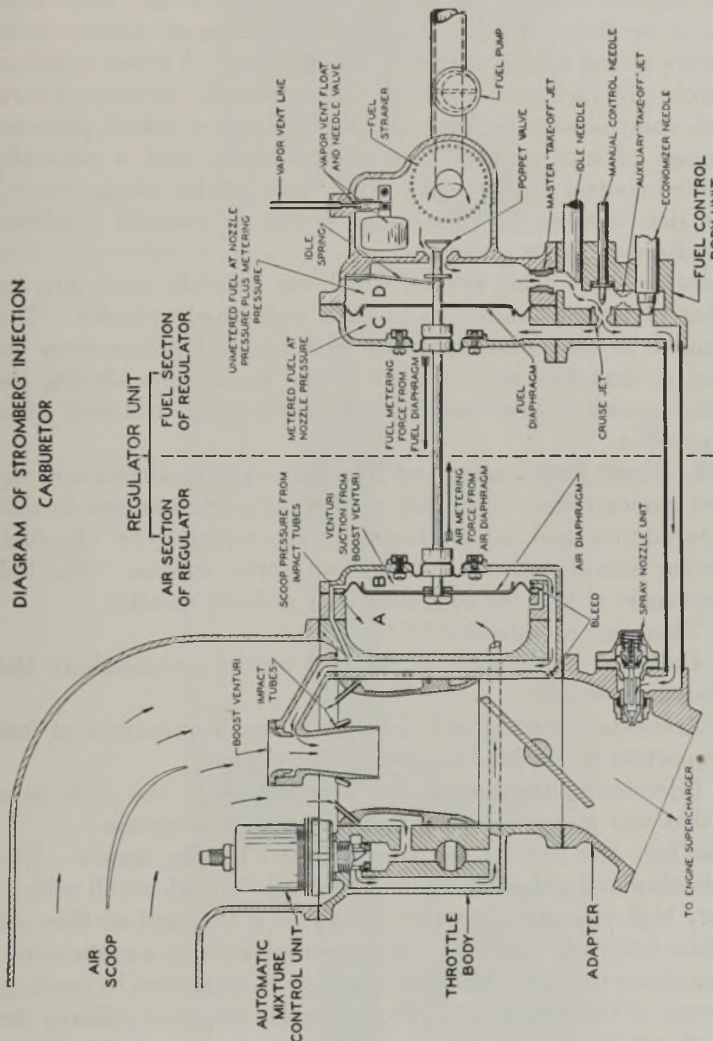


FIG. 314.—Diagram of Stromberg injection carburetor.

The special requirements of aircraft carburetion have led to the development of the injection carburetor, a diagram of which is shown in Fig. 314. This differs in principle from the flow carburetors previously discussed in that the difference in pressure between the Venturi throat and air scoop is utilized as a mechanical force to regulate the fuel flow. Thus the mass air flow at a given density controls the fuel flow to a fixed weight ratio at all rates of air flow. Fuel is discharged into the air stream under pressure giving a high degree of atomization. Various auxiliary controls are included, such as the automatic mixture control which compensates for altitude, an idle spring which prevents stoppage of fuel at very low air-flow rates, and a manually operated control needle which gives the operator an option of a fixed rich mixture for maximum power or take-off condition and a fixed ratio for cruising.

Another fuel-mixing system which has recently come into use is that of a timed injection to each individual cylinder. This is similar to the Diesel injection systems, but differs from the Diesel in that fuel is sprayed into the intake pipe during the suction stroke rather than into the combustion chamber after compression.

262. Diesel Fuel Pumps and Nozzles.—The fuel system of the Diesel engine differs from that of the carbureted engine in that the carburetor and intake manifold are replaced by the fuel-metering pump, delivery tubing, and spray nozzles. The last system must obtain the following very definite results:

1. Thorough atomization of the fuel.
2. Uniform distribution of the fuel particles throughout the combustion chamber.
3. Accurate metering and delivery of small quantities of fuel in proportion to the load carried by the engine.
4. Injection of the fuel at the proper time in the cycle at a rate that will promote steady and complete combustion.

The first two results are accomplished by the nozzle. The first is obtained either by driving the fuel through small orifices at very high velocity or by turbulent flow of fuel and air through irregular labyrinth passages. Since atomization is a preliminary to vaporizing the fuel, this operation may be improved by burning a portion of the charge in a precombustion chamber utilizing the heat of combustion to aid in vaporizing the remainder of the fuel. The second object is attained by choice of the number, form, and position of nozzles that will produce a spray to conform to the contour of the combustion chamber. The nozzle

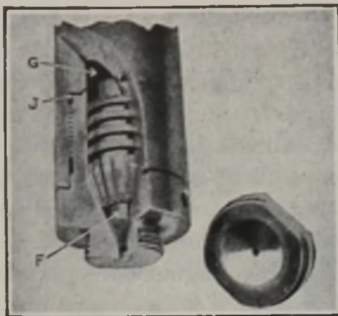


FIG. 315.—Section of lower part of spray valve on Busch-Sulzer Bros Diesel engine and disassembled parts of valve.

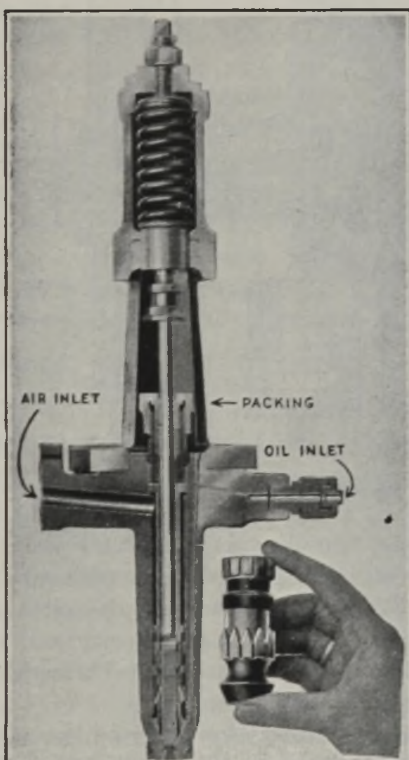


FIG. 316.—Cross section of spray valve (Hesselman system) on McIntosh and Seymour Diesel engine; at right, atomizer disassembled.

may have single or multiple spraying jets. It may have a circumferential orifice or a pin jet. The fuel may be sprayed into the cylinder with or without the aid of high-pressure injection air. If air is used, the method is called *air injection;* if fuel only is injected, it is called *solid injection.*

Figure 315 shows a form of nozzle used with air injection. The space G is always connected to the injection air supply. Just previous to each injection a metered quantity of fuel is forced through the opening J against the pressure of the injection

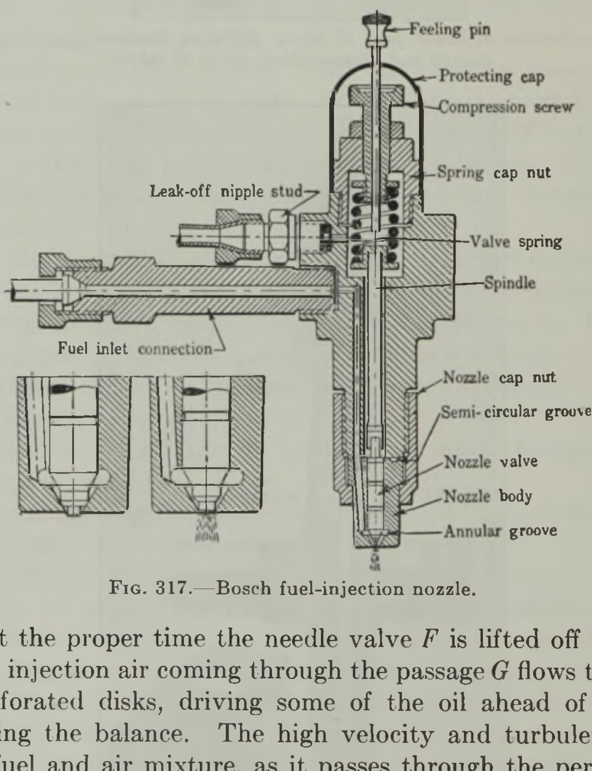


FIG. 317.—Bosch fuel-injection nozzle.

air. At the proper time the needle valve F is lifted off its seat and the injection air coming through the passage G flows through the perforated disks, driving some of the oil ahead of it and entraining the balance. The high velocity and turbulent flow of the fuel and air mixture, as it passes through the perforated disks, thoroughly atomize the fuel.

Figure 316 shows another type of atomizer which takes the place of the rings and cone (shown in Fig. 315) to produce the turbulent flow needed for proper atomization. In both of these nozzles, or spray valves, the fuel pump meters the fuel and forces it into the nozzle while the injection air under a pressure of 700 psi to 1,000 psi drives it through the atomizer into the cylinder. This injection air, which may be as much as 10 per cent of the total air charge of the engine cylinders, is usually supplied by a three-stage compressor built integral with the engine. The air is cooled after each stage of compression and stored in a small tank from which it is used as needed. When an engine has been stopped, it may be started again by delivering injection air, through starting valves which are operated by the camshaft, into each cylinder at the beginning of its working stroke.

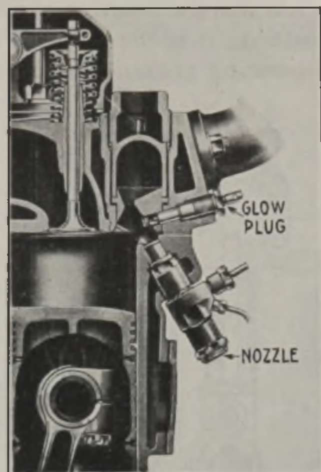
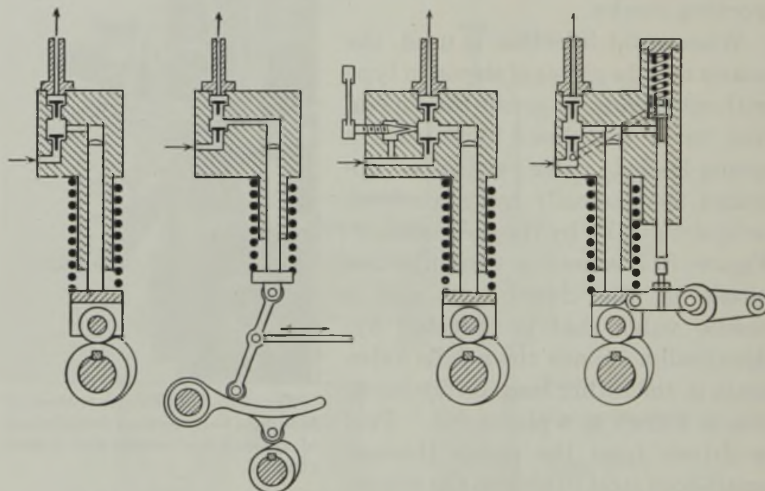


FIG. 318.—The Acro system of fuel injection showing installation of a Bosch fuel nozzle and a glow plug.

When solid injection is used, the nozzle may be either of the open type without means to arrest the flow of fuel, or of the closed type using a spring-loaded valve which is operated mechanically by cam action, or hydraulically by the fuel pressure. Figure 317 shows a fuel-injection nozzle of the closed type with a needle valve that is operated hydraulically. Since the needle valve seats in the rather large orifice itself, this is known as a pin nozzle. Fuel is driven from the pump through small-bore steel tubing to the nozzle. As the stroke of the fuel-pump plunger raises the pressure, the needle valve is lifted against the spring load and the fuel is sprayed out through the annular space, between the needle valve and its seat, into the combustion chamber.

Figure 318 shows the installation of this nozzle according to the Acro system. The combustion chamber is divided into two approximately equal volumes connected by a rather restricted passage. The fuel is sprayed into the lower part of the combustion chamber and is aimed directly at the throat of the connecting passage. At the moment of injection the air, owing to the shape of the combustion chamber, is in a state of great turbulence, which aids in proper distribution of the fuel spray. The temperature of the air in the upper half of the combustion chamber has been raised not only by compression but also by contact with the hot walls of the chamber which is only partially cooled. The additional heat increases the rapidity of vaporization and this engine is said to operate satisfactorily at 3,000 rpm. The glow plug used for starting is also clearly shown. It provides a coil of wire lying in the path of the spray which may be heated to incandescence by current from the battery to ensure ignition when starting in extremely cold weather.

As it is the function of the nozzle to provide the required spray pattern, it is the function of the injection pump to furnish the operating pressure at the proper time as well as to meter the



(a) (b) (c) (d)

Fig. 319.—Methods of regulating the delivery from a pump.

quantity of fuel injected. The fuel-quantity regulation is accomplished by either one of two methods: variable stroke or variable by-pass. As illustrated in Fig. 319*a*, the stroke of the plunger is varied by moving the variable contour cam along its axis of rotation, and in *b* it is changed by varying the position of the connecting link away from the center of oscillation of the oscillating link. The pumps, shown in Fig. 319*c* and *d*, allow oil to return to the inlet side through a by-pass—in *c* continuously during the whole stroke of the plunger, and in *d* by mechanically opening a by-pass valve at a given point in the delivery stroke of the plunger. The point at which the by-pass is opened is controlled by raising or lowering the center of oscillation of the actuating link. Figure 320 illustrates a variation of the by-pass method. The pump plunger may be turned on its own axis while reciprocating, to uncover the inlet port at any desired point in the delivery stroke. When the inlet port is thus uncovered, the fuel is discharged back through the groove into the inlet passage instead of through the spray nozzle. On the return stroke the fuel remaining in the chamber above the plunger is subjected to a partial vacuum which permits a fresh quantity of

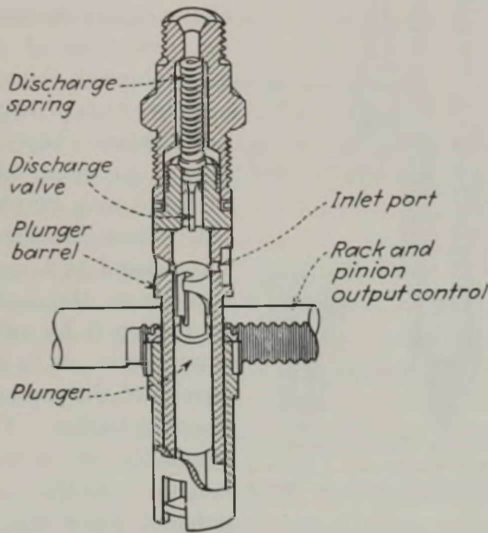


Fig. 320.—Bosch injection pump.

fuel to enter when the top of the plunger uncovers the inlet port.

Usually each cylinder of a multicylinder Diesel engine is provided with an individual pump and nozzle. This may be done by building the pump and nozzle as a unit as shown in Fig. 321 and operating each pump by a cam on the same shaft that operates the air valves. Another method is to build the pumps into a compact unit and provide pressure lines to each of the nozzles. Obviously it is desirable that each pump deliver to its cylinder exactly the same quantity as every other pump in the unit at all operating conditions, so with the multicylinder pump system it is sometimes necessary to provide all cylinders with the same length of fuel delivery line, regardless of the proximity of the pump to its cylinder. This is necessary because the compressibility of the oil in tubes of different lengths may cause uneven surging and distribution of the oil. A third method, the common rail system, provides a common pressure line for all cylinders with mechanically operated valves opening a passage to each nozzle in the proper sequence.

263. The exhaust system consists essentially of an exhaust manifold connecting the exhaust ports for each cylinder through an exhaust pipe to the muffler. A muffler is required because of the report caused by the initial discharge of the exhaust gases through the valve port at a rather high pressure. At the point where the exhaust valve begins to open, the cylinder pressure may be as high as 50 psi. Since the sharp report is propagated as a sound or pressure wave, the muffler is built to reduce it by reflection and interference, or to dissipate the energy of the pressure wave by suitable baffles. This must be accomplished without serious resistance to the passage of the exhaust gases into the atmosphere. Cooling the exhaust gases aids materially as it reduces their pressure.

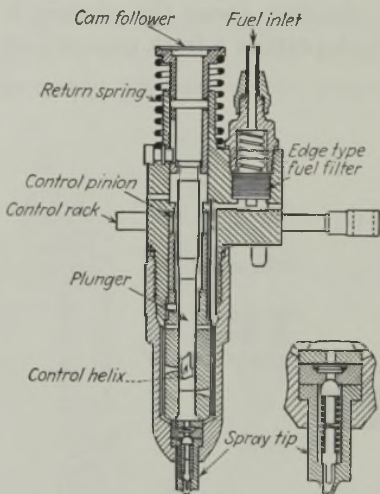


FIG. 321.—General Motors Diesel injection pump and nozzle unit. Helical by-pass control system with six-orifice spray tip.

264. Governing.—The aim of all governors is to adjust the supply of energy to the demand for work from the machine. Usually this is done in such a way as to maintain constant speed.

The following general methods of governing are used in gas engines:

1. The *hit-and-miss* system.
2. Variation in the *quantity of fuel and air charge* entering the cylinder, the air-fuel ratio remaining constant.
3. Variation in the *quantity of fuel only*, the quantity of air remaining constant. This implies a variation in the fuel-air ratio, or *quality* of the mixture.
4. Governing by changing the time of ignition.
5. Combinations of the above methods.

Method 1.—The most common of all these systems of governing is the *hit and miss.* In this system, when the speed exceeds the normal, the supply of fuel is cut off and the engine gets no explosion, causing the engine to "miss." The loss of the explosion causes the speed to slacken, the governor opens the inlet valve, and the engine again receives an impulse, or a "hit." The miss may be occasioned (*a*) by holding the exhaust valve open so that no suction may occur; or (*b*) by failing to open the intake valve. This is a simple and economical method used by small stationary engines where accurate regulation of speed is not necessary.

Method 2.—*Quantity governing* may be accomplished by varying the weight, or quantity, of *fuel and air* mixture entering the cylinders. This is usually accomplished by means of a throttle valve in the intake passage between the carburetor and the cylinder. Closing this valve, to reduce the quantity of charge, also reduces the compression pressure. Reducing the compression decreases the efficiency, hence this form of governing is not so economical at part-load operation as the hit-and-miss system. However, the engine operation is so smooth, and the speed regulation so accurate, that this type of governing is used for practically all carbureted engines operating in automotive equipment. In automobiles and aircraft the throttle valve is usually hand or foot operated.

Method 3.—*Quality governing* is accomplished by varying only the quantity of *fuel*, the charge of air being always the same. The compression pressure and temperature are always the same, but the temperature of the products of combustion depends on the quantity of fuel supplied. This system is not applicable to part-load operation of a carbureted engine since the mixture may become so lean that it will not ignite. It is, however, commonly used by Diesel engines, in fact by any engine that injects the fuel directly into the cylinder. For these, it is exceptionally economical at part loads and permits accurate regulation of the speed.

Method 4.—The speed may be controlled by changing the time of ignition. The power output may be decreased by retarding the spark, although the fuel consumption remains the same This method is so inefficient at part loads that it is used only on small engines where fuel economy is not essential.

Method 5.—A great many combinations of the above systems have been used. Often engines having *quantity* and *quality* governors for the heavy and medium loads change the governing system to *hit-and-miss* for light loads. A carbureted automotive engine, at or near full-throttle operation, is actually using a combination of Methods 2 and 3, since a fuel needle valve is opened by a link connecting it with the throttle, so that not only more mixture, but a richer mixture, is drawn into the cylinder as the throttle is fully opened. For part-load operation Method 2 is used; and for very slow idling, the driver may occasionally resort to Method 4.

A typical governor installation is shown in Fig. 310 which illustrates by a diagram the fuel system of a Diesel engine. The stroke of the metering-pump plunger is always the same, but the effective stroke and the quantity of fuel injected are controlled through by-passing the fuel through the inlet valve back into the supply reservoir as described under Diesel pumps. During the stroke of the metering-pump plunger, the time at which the inlet valve opens and injection ceases is determined by the position of the eccentric B. The position of the eccentric is in turn controlled, through a link and bell-crank, by the fly-ball governor which operates according to the engine speed.

CHAPTER XV

AUXILIARY SYSTEMS

265. Cooling.—Since the maximum temperature of the gases in the internal-combustion-engine cylinder may reach 3000 F, it becomes necessary to cool the cylinder. Cooling the cylinder walls prevents vaporization or burning of the necessary lubricant. Cooling the combustion-chamber walls prevents preignition and allows the use of a higher compression ratio without detonation. Included as a part of the combustion-chamber wall are the piston head, the valve heads, and the spark plugs or the fuel-injection nozzles. Cooling may be the most important factor to be considered in the design of these parts.

Cooling may be accomplished directly by means of exterior cooling fins made integral with the cylinder and cylinder head. The extended surface so presented to a blast of air is sufficient to dispose of the heat rapidly enough to keep cylinder temperatures down to a practical operating range. Head temperatures run from 450 F to 500 F as an upper limit for continuous operation, and wall temperatures must be kept about 100 deg lower. The flow of air is produced by forward motion of the engine, the action of a blower, or both, as in the case of the airplane engine. Stationary engines must be provided with a fan of sufficient capacity to cool them under maximum output conditions and are limited to the smaller sizes. Farm lighting plants, for example, are usually powered with engines of this type.

Cooling may be accomplished indirectly by surrounding the cylinder head and walls with a jacket through which water or some other liquid is circulated. This circulation is obtained by the use of a pump, or by thermosyphon action. As illustrated by Fig. 322, the water receives its charge of heat units from the metal of the engine cylinders, then travels into the upper part of a radiator. As the water trickles down through the many copper or brass tubes of the radiator, the heat is transferred first into the metal of the radiator and then into the air which is driven through and around the tubes by a fan or by the forward motion of the engine. The cooled water is then returned to the jacket by a centrifugal pump. This type of pump is preferred because circulation and the cooling operation will continue by thermosyphon action after rotation of the pump has ceased.

Heat transfer from the water in a radiator to the air is materially aided by the use of metal fins, thus presenting a larger metal

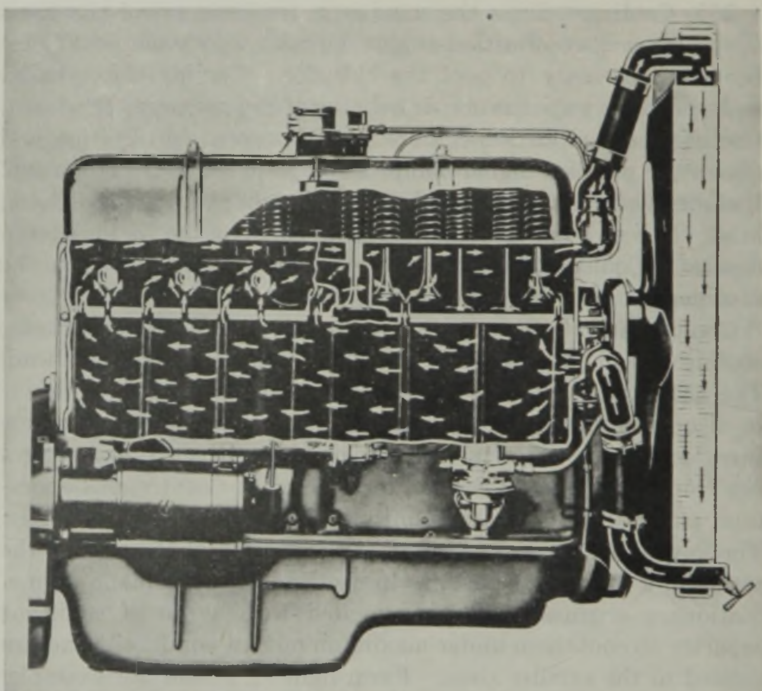


FIG. 322.—Water-cooling system. Water from the block is nozzle-directed under some pressure around the exhaust valve seats and spark-plug bosses.

surface to the air blast. It is also aided by the use of irregular passages which cause turbulent flow of both water and air. The turbulent flow wipes off the fluid film which tends to adhere to the metal surfaces, and to insulate them from the moving fluid. Turbulence thus facilitates heat transfer from the cylinder metal to the water, from the water to the radiator metal, and from the radiator metal to the air.

It is logical to use air cooling for aircraft engines and water cooling for engines in marine service. Even though the liability of leakage or freezing may present many difficulties, water cooling is generally preferred for automobiles, trucks, and stationary heavy-duty engines, because the boiling point places a definite limit on the operating temperature. The addition of glycerine, alcohol, etc., will lower the freezing temperature. The use of a liquid with a high boiling point, such as ethylene glycol, will increase the thermal efficiency of the engine and decrease the weight of the cooling system, and still provide a limit to the operating temperature of the engine.

266. Lubrication.—The reliability of the internal-combustion engine depends largely on its automatic lubrication system.

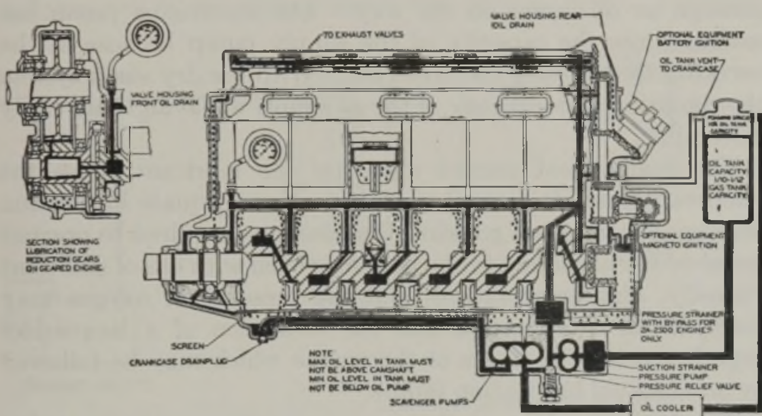


FIG. 323.—Lubricating chart of Packard aircraft engines, models 1500 and 2500.

Since from 10 per cent to 20 per cent of the power developed in the cylinder is used to overcome friction in the engine, a recirculating system is used, which not only lubricates the various bearings but also serves to carry away the heat produced by friction. The cylinder walls, the gears, and all plain bearings require continuous lubrication.

The oil reservoir may be within the engine itself in a sump which is kept filled to a certain minimum level so that the lower end of the connecting rod dips into the oil on each revolution and splashes lubricant to cylinder walls and wrist pin. More frequently a positive displacement pump, usually a gear pump, draws oil from the sump and distributes it under considerable pressure to the various bearing surfaces through tubing or drilled passages in the engine parts. This is the system commonly used on automobile engines. In many other installations, particularly of large engines, a dry sump lubrication system is used. Figure 323 illustrates this system applied to an aircraft engine. The oil supply is contained in a separate tank from which it is fed to a pressure pump and distributed to the bearings. Drilled passages in the crankshaft carry oil from the main bearings to each crankpin, and the throw-off from the main and connecting-rod bearings lubricates the cylinder walls and wrist pin. It returns by gravity to a small sump in the lower part of the engine, and is continuously withdrawn by a scavenging pump which returns it to the supply tank, usually forcing it through an oil cooler on the way. The scavenging pump has several times the capacity of the supply pump because of the necessity for handling entrained air. With the dry sump system the condition and quantity of the oil supply are at all times easily discernible.

The engine heat carried away by the lubricant raises the temperature and lowers oil viscosity. For adequate lubrication the viscosity must not go below the minimum required to prevent metal-to-metal contact in the bearings; even with oils of sufficient viscosity, high temperatures and the presence of oxygen may result in oil deterioration and the formation of a lacquerlike deposit coating the inside of the engine which may be followed by an eventual lubrication failure.

267. Ignition.—One of the most important events occurring in the internal-combustion-engine cycle is the ignition of the combustible charge. Probably the simplest form of ignition is that sometimes called *compression ignition*, used by the Diesel engine. In this engine, the charge of air is compressed to a very high pressure, with a temperature high enough to ignite the entering charge of gas or fuel oil as it is driven into the cylinder by a still higher pressure. This method requires no special apparatus and the time of ignition is accurately controlled by timing the admission of fuel into the cylinder.

Ignition for the Otto cycle engine is not so simple. The early engines made use of an open flame in much the same way a match was used to fire a muzzle-loading cannon. Flame ignition was uncertain and was soon replaced by the hot-tube type in which a closed tube, connected with the engine cylinder, was kept at a red heat by an external flame. The compression of some of the combustible gases into this hot tube ignited them at about the proper point in the cycle. The timing of the ignition could be regulated somewhat by the temperature of the hot tube, but was not accurate enough for the modern engine. A variation of the above method involves the use of a bulb-shaped chamber, with walls that are only slightly cooled, which is connected to the cylinder by a somewhat restricted passage. A considerable portion of the charge is forced into this chamber during the compression stroke, and heat absorbed from the hot walls added to the effect of compression serves to raise the temperature of the charge above the ignition point. A means

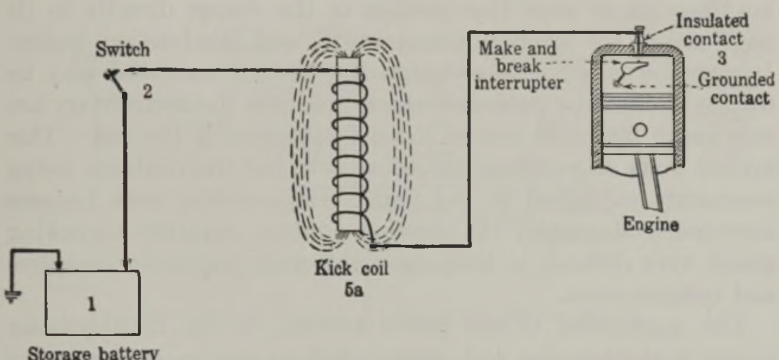


FIG. 324.—Wiring diagram of make-and-break ignition system.

of controlling the bulb temperature is necessary to control the time of ignition and an external heater is required for starting.

Accurate timing of the ignition of the charge is so essential to the smooth operation of the internal-combustion engine that the use of an electric arc was soon suggested. The first and simplest electric ignition device was known as the *make-and-break* system. It consisted of the following units (Fig. 324): (1) a source of electric energy such as a storage battery, dry cell, or magneto; (2) a switch which is used to stop operation of the system; (3) the breaker or igniter (between whose contacts the spark occurs); and (5*a*) the spark or induction coil which furnishes the necessary potential. The igniter has two contact points located with the combustion chamber. The stationary contact or electrode extends out through the cylinder wall but is insulated from it. The movable contact is attached to a shaft, having a bearing in the cylinder wall, which is operated automatically by engine mechanism outside the cylinder. The coil consists of a few turns of No. 14 or 16 insulated copper wire, wrapped around a soft-iron core.

The operation of the system was as follows: Just before the time of ignition, the igniter contacts are closed, thereby "making" the electric circuit which allows current to flow from the battery through the circuit, and to set up a strong magnetic field about the coil. At the instant ignition is desired, the contacts begin to separate and the stored energy of the magnetic field builds up a high potential in the winding of the coil, which forces some current across the gap and produces an arc. This arc supplies heat enough to raise that portion of the charge directly in its path above the ignition temperature, and combustion begins. It must be noted that although the battery potential may be 6 volts or less, the potential which produces the momentary arc may reach 200 volts due to the self-induction of the coil. This system gives an excellent arc when new, but the contacts, being constantly subjected to the flame temperatures, soon become corroded. Moreover, the moving contact requires a packing gland, very difficult to keep gas-tight under explosion pressures and temperatures.

The application of the above system, to the multicylinder engine is cumbersome and complicated, so that excepting for an occasional installation it has been superseded by the simpler and more dependable jump-spark system.

The *jump-spark* system is made up of seven major and indispensable units and two safety devices for protection of the induction-coil insulation. The following parts are to be found on the circuit diagram, Fig. 325.

6 to 8 volts:

1. A source of electrical energy
   - chemical
     - dry cell.
     - storage cell.
   - mechanical
     - generator.
     - magneto.
2. Switch (used to stop the engine).
3. Breaker.
4. Condenser.
5. Transformer coil:
   - (5*a*) primary winding; (5*x*) safety resistance;
   - (5*b*) secondary winding; (5*y*) safety gap.

4,000 to 5,000 volts:

6. Distributor.
7. Spark plug (one for each cylinder).

There are in this system two distinct electrical circuits: the primary circuit with a prevailing potential of 6 to 8 volts, and a secondary circuit which has a momentary potential of about 4,500 volts. The induction coil has a primary winding consisting of a few turns of insulated wire wound about a soft-iron core. Around the primary are wound a great number of turns of very fine wire which forms the secondary. The breaker which serves to interrupt the flow of current consists of a pair of contacts

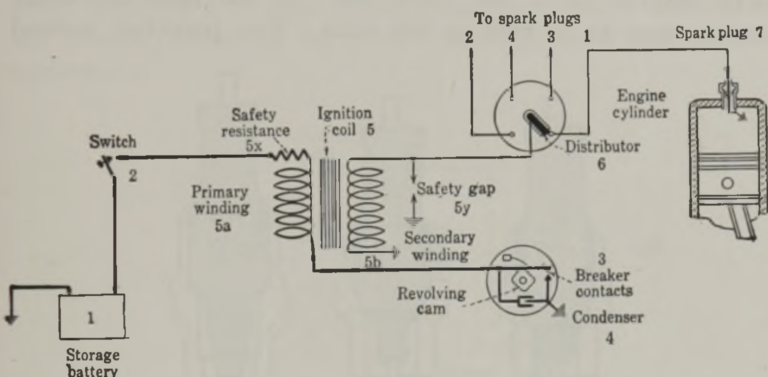


FIG. 325.—Wiring diagram of jump-spark ignition system.

which may be separated by the action of a rotating cam. The distributor is merely a rotating switch which leads the secondary current to the proper plug as the coil discharges. The breaker and distributor are generally built as a single unit called the *ignition head.* The distributor rotor is mounted on the upper end of the vertical shaft, just above the breaker cam. As the distributor must make one revolution per cycle, the breaker cam will then have a lobe for each cylinder. The condenser is nearly always mounted on or in the ignition head, close to the breaker contacts. In the base of the head is generally found a mechanism which utilizes the centrifugal force of some revolving weights to advance automatically the spark as the engine speed increases.

The primary circuit of this system is very similar to that of the make-and-break system. In fact the source, switch, breaker, and primary winding are almost identical in their manner of operation. Just before the time of ignition, the breaker contacts close and current begins to flow through the primary winding of the coil building up a magnetic field. The primary current is interrupted by the separation of the breaker contacts and the magnetic field collapses. The sudden collapse of the magnetic field induces a current in both primary and secondary windings of the coil. Because of the many thousand turns of fine wire, the voltage induced in the secondary is very high, yet without the use of a condenser is not sufficient to jump the spark-plug gap. The potential induced in the primary, in exactly the same manner as in the "kick coil" of the make-and-break system, may be as high as 300 volts. This potential, instead

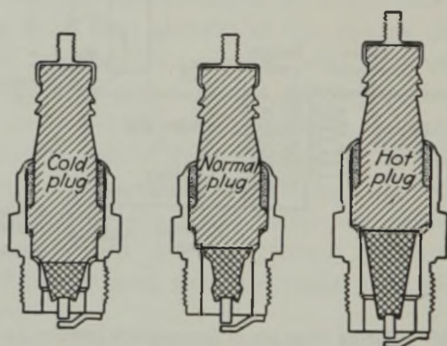


FIG. 326.—Spark-plug sections showing method of controlling the operating heat range.

of causing current to flow across the gap between the breaker contacts, charges the condenser. Then, as the induced potential subsides, the condenser discharges backward through the primary winding, magnetizing it with opposite polarity. Since the potential induced in the secondary winding of the coil is proportional to the rate of flux change, this use of the condenser raises the secondary voltage to about 4,500 which is sufficiently high to drive current across the spark-plug gap. The use of the condenser thus greatly increases the secondary voltage and prevents arcing and corrosion of the breaker contacts.

For aircraft engine ignition the high-tension magneto is used almost exclusively. A light compact system, it is made up of the same seven major units described in the battery system, differing only in the fact that the electric current of the primary coil is produced by mechanical means. Referring to Fig. 327, rotation of the four-pole magnet reverses the direction of magnetic current in the soft-iron core four times per revolution. This reversal of flux induces a voltage in both windings, but is not abrupt enough to induce the required high potential in the secondary. The voltage of the primary coil, which is part of a complete circuit, causes a primary current to flow. At or near the point of maximum induced current the breaker points separate and the action of the condenser and induction coil induces the high potential in the secondary necessary to jump the spark-plug gap. Operation of the magneto is stopped by closing the switch which provides a by-pass circuit from the primary winding to the ground, rendering the circuit interruption of the breaker inoperative.

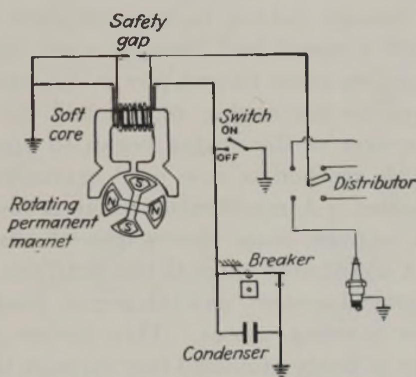


FIG. 327.—Inductor-type magneto circuit.

The spark plug consists of a central electrode imbedded in porcelain, or mica, insulation which is securely clamped in a metal spark-plug body. Attached to the lower end of the plug is the grounded electrode. The two electrodes are separated by the spark gap of 0.015 in. to 0.030 in.

The spark plug is a disturbing element in the heat flow away from the combustion-chamber walls, and spark plugs of the proper "heat range" must be used for a particular engine. In general, low-compression engines make use of a hot plug; high-compression and high-performance engines make use of a colder plug. As illustrated in Fig. 326, the length of insulator through which the heat must travel largely determines the heat range of the plug. In a given service too cold a plug will tend to foul with carbon and short out; too hot a plug may crack the insulator or cause preignition.

268. Starting the Engine.—Unlike the steam engine, which develops a high torque when standing still, the internal-combustion engine will not function properly or develop power until it is turning at a fair speed. Small engines may be turned by hand. Large engines require considerable power to start them. One of the early starter developments involved the use of compressed air which was introduced into the working cylinder just after the beginning of the working stroke, driving the piston very much as a normal explosion would. Air under pressure of 400 psi to 600 psi was led from a storage tank through a rotary distributor valve which timed the air injection and distributed it through tubing to the cylinders in the proper order, until such a speed had been attained that the engine continued to function under its own power. A small compressor, which was driven by the engine, replenished the supply in the storage tank as soon as the engine began to operate under its own power. This method is now used extensively in starting large Diesel engines. A modification of it is finding extensive use on aircraft engines, since the weight of the supply tank, compressor, and distributor valve is less than the corresponding parts in the electrical system, and the engine itself performs the function of the starting motor. This device also injects a definite quantity of finely atomized fuel through the distributing tubing directly into the cylinder, ensuring the presence of a combustible mixture in the engine cylinder during the starting operation.

To reduce the power required for the starting operation on some engines, the exhaust valves are held open during the compression stroke until starting speed has been attained, at which time they are released and the engine starts as usual.

For starting airplane engines, an inertia starter of very light weight is commonly used. It consists essentially of a small flywheel which may be driven to a very high speed through a train of gears and a hand crank. The starter shaft is then connected to the engine crankshaft through both a jaw clutch and a friction clutch. The energy stored in the flywheel, operating through the reduction gears, serves to turn the engine at a fair starting speed for a few revolutions. A small electric motor is sometimes used to spin the flywheel instead of operating it by hand.

Automobiles, trucks, tractors, and small marine engines are started by a small electric motor which drives the engine through a small pinion, engaging gear teeth on the engine flywheel From 1 hp to 2 hp is required to turn the engine at a speed varying from 100 rpm to 300 rpm. A motor with a series field is used in order to supply the high initial torque required to overcome the static friction of the engine. Power is supplied from a battery of the lead-plate type, with a potential of 6 volts

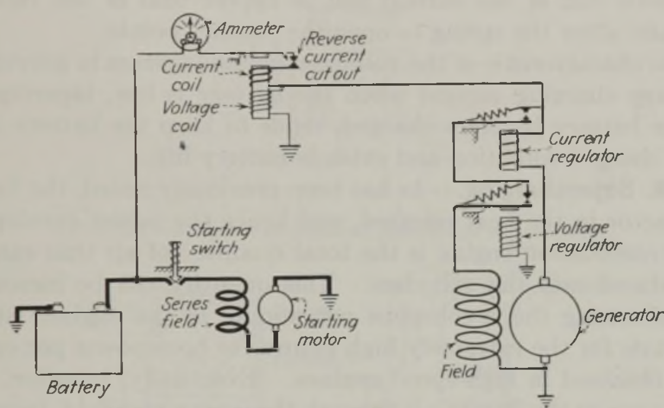


FIG. 328.—Wiring diagram for starting and generating system.

and a capacity from 80 amp-hr to 200 amp-hr. After the engine is started, the electric energy used from the storage battery is replaced by means of an electric generator driven by the engine.

Figure 328 is a diagram of the circuits for such a starting and generating system. An ordinary shunt generator producing voltage nearly as a direct function of speed would be unsatisfactory for use on a variable-speed engine, so the voltage- and current-limiting regulators are connected in the manner shown. Should the voltage of the generator exceed a predetermined maximum, the magnetism set up by the flow of current through the voltage coil opens the points of the voltage regulator, forcing all the field current to pass through a resistance. This reduces the field current, lowers the output voltage, and permits the spring to close the points, after which the action is repeated rapidly enough to give in effect a constant maximum voltage. A similar action occurs when the output current reaches a predetermined maximum. The reverse current cutout is a magnetic switch which closes the circuit to the battery when the generator voltage is high enough to magnetize the iron core. This is, of course, set slightly above the battery voltage. Once in operation the flux of the current coil of the reverse-current cutout aids the voltage coil in maintaining the closed contact points as long as current flows to the battery. Should the voltage of the generator fall below that of the battery, the reverse current will cause the magnetic flux of the current coil to oppose that of the voltage coil and allow the spring to open the contact points.

The characteristic of the voltage-regulated system in providing a heavy charging current when the battery is low, tapering off as the battery becomes charged, tends to keep the battery in a fully charged condition and extends battery life.

269. Supercharging.—As has been previously noted, the limiting factor in the heat released, and hence the power developed, by a combustion engine is the total quantity of air that can be introduced into the cylinders. This quantity can be increased by increasing the revolutions per minute of the engine, which accounts for the extremely high figures for horsepower per cubic inch obtained in high-speed engines. Eventually, however, the resistance to the flow of air through the intake manifold, increasing with the velocity of flow, reduces the density of the cylinder charge, and the increase of friction horsepower with speed results in a definite peak speed for maximum shaft horsepower. An auxiliary air pump or supercharger will, up to certain limits, permit an increase in revolutions per minute, hence horsepower. This is the principal use of superchargers in automobile and racing engines. For aircraft, however, the diminished density of air at high altitudes reduces engine output by approximately 2 per cent for each 1,000 ft of altitude, and the function of the aircraft supercharger is to maintain low-altitude output at high altitudes.

Two types of superchargers may be used: positive displacement and centrifugal. In general the positive displacement types, *viz.*, the piston displacement, Roots blower, and vane pump, are too bulky or heavy for aircraft use, and the centrifugal type is used almost exclusively. In the latter, the centrifugal blower may be driven either by an exhaust turbine or by a gear train from the crankshaft. The exhaust turbine presents problems of stresses at high temperature, but has the advantage of operating against a reduced back pressure at high altitudes, increasing the output when supercharging is most required. The geared centrifugal type, used on practically all radial aircraft

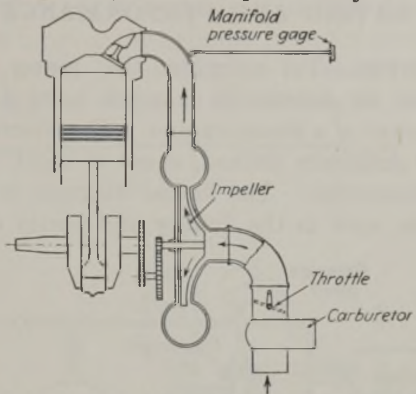


FIG. 329.—Diagram of a geared centrifugal supercharger.

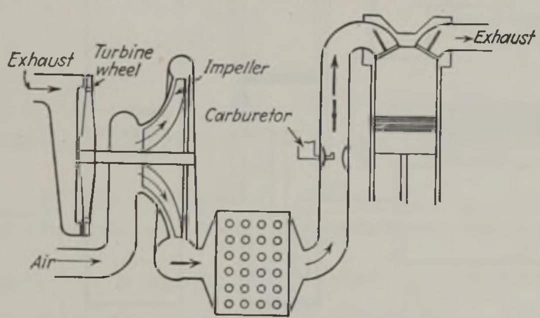


FIG. 330.—Diagram of a turbo-driven centrifugal supercharger.

engines, has the disadvantage of running as a direct function of engine speed, and an engine with supercharger capacity for high altitude must be throttled at low altitude. Figures 329 and 330 show how superchargers of each of these types are employed in the intake system. With the gear-driven impeller (Fig. 329) the gasoline-air mixture is compressed and incidentally well mixed by the impeller; with the exhaust turbo type (Fig. 330) only air is compressed and the entire carburetor operates under supercharge pressure.

CHAPTER XVI

RATING AND PERFORMANCE

270. Horsepower.—To estimate the power output of a gas engine from its dimensions is much more difficult than to estimate the power of a steam engine. The theoretical diagram, although quite definitely defined, is not of much value in determining the horsepower. The actual diagram is influenced by so many factors, such as the quality and purity of gas, temperature of the mixture, conditions of combustion, heat losses, location and kind of ignition, form of combustion chamber, and other items, that it is possible to obtain almost any result. The diagram factor as applied to the gas engine is of little value, as it shows variations under different conditions as high as 100 per cent. Added to this is the fact that an accurate indicator diagram is no such easy matter to secure as it is on a steam engine. The high speeds preclude the use of the type of indicator so useful on slower speed engines, because the inertia of the moving

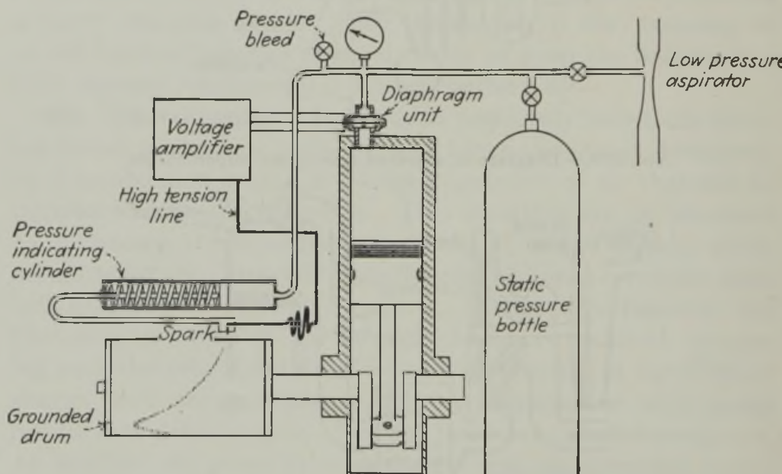


FIG. 331.—Schematic diagram of a point-by-point pressure indicator for high-speed engines.

parts of the indicator will cause the readings of pressure and volume to lag far behind the actual changes. High-speed engine indicators usually operate to give point-by-point pressure readings. A schematic diagram of such an indicator is shown in Fig. 331. It consists of a strong, light diaphragm exposed to cylinder pressure on one side and to a controlled static pressure on the other. Whenever the cylinder pressure exceeds the static pressure, or vice versa, the diaphragm breaks an electrical contact. This break in an electrical circuit may be used to cause a

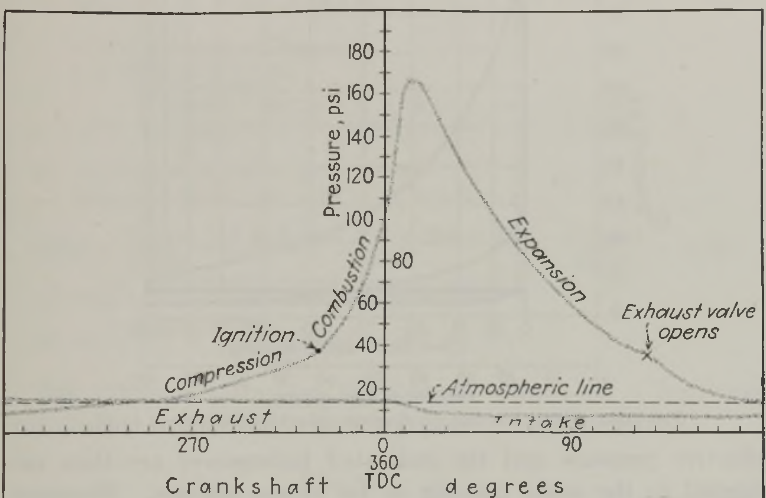


FIG. 332.—Pressure-time indicator diagram for a part-throttle cycle.

spark to jump from a pointer and perforate a paper on the rotating cylinder, which is driven by the crankshaft. The position of the pointer along the axis of the cylinder is determined by the static pressure applied to the diaphragm. In operation, the static pressure is brought above the highest pressure reached in the cylinder and slowly allowed to drop, while the engine is operating under constant speed and load. Thus the indicator diagram obtained will be the average of perhaps several thousand cycles, and the result will appear as shown in Fig. 332. It can be converted into a pressure-volume diagram by relating crank degrees to piston travel, but in many ways it is equally useful if studied on the pressure-time basis. For example, the slope of the combustion line is a direct indication of the rate of pressure rise, an important consideration in the design of combustion chambers and engine roughness analysis.

Transferred to a pressure-volume basis, the diagram will appear as Fig. 333 and the net work per cycle will be represented by the net area of the diagram. Note that the exhaust and intake strokes yield a negative work loop (shown crosshatched), particularly at part throttle, which must be deducted from the positive work area in determining the net work. The mean

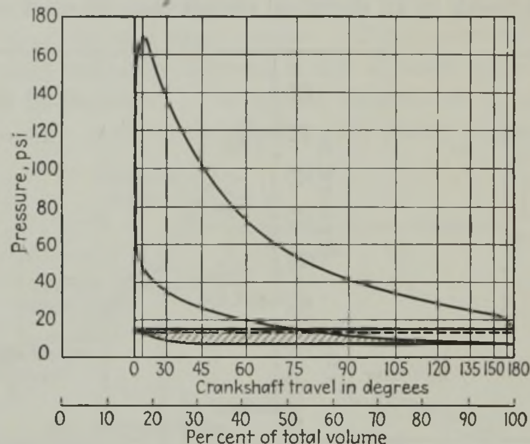


FIG. 333.—Pressure-volume diagram of a throttled Otto cycle.

effective pressure and the indicated horsepower are then calculated in the same manner as for steam engines. However, for internal-combustion engines the formula for indicated horsepower becomes

$$\text{Ihp} = \frac{plaNK}{33{,}000}, \tag{1}$$

where N = the number of explosions per minute;
K = the number of cylinders.

The brake horsepower (bhp) of a gas engine is found in exactly the same manner as the brake horsepower of a steam engine, the expression being

$$\text{Bhp} = \frac{2\pi Lnw}{33{,}000}, \tag{2}$$

where L = the length of the *brake arm* in feet;
w = the net weight on the brake;
n = the number of revolutions per minute.

It is thus seen that in making a test of a gas engine to obtain the indicated horsepower and brake horsepower both the explosions per minute and the revolutions per minute must be noted.

Example.—A 10-in. by 18-in. single-acting gas engine runs 200 rpm and makes 96 explosions per min. The gross weight on the brake was 140 lb, the tare 20 lb, and the length of the brake arm 60 in. The area of the indicator diagram was 1.18 sq in. and the length 3 in., and the scale of the spring used was 200 lb. Find (*a*) the indicated horsepower; (*b*) the brake horsepower; (*c*) the friction horsepower; and (*d*) the mechanical efficiency.

Solution.

(*a*) $\text{Mep} = \frac{1.18}{3} \times 200 = 78.7$ lb.

$a = \pi \times 5 \times 5 = 78.54$ sq in.

$l = 18 \div 12 = 1.5$ ft.

$$\text{Ihp} = \frac{plaN}{33{,}000} = \frac{78.7 \times 1.5 \times 78.54 \times 96}{33{,}000} = \frac{890{,}000}{33{,}000} = 27.$$

(*b*) Net weight $= 140 - 20 = 120$ lb.

Length of brake arm $= 60 \div 12 = 5$ ft.

$$\text{Bhp} = \frac{2\pi Lnw}{33{,}000} = \frac{2 \times 3.1416 \times 5 \times 200 \times 120}{33{,}000} = \frac{755{,}000}{33{,}000} = 22.85.$$

(*c*) $\text{Fhp} = \text{ihp} - \text{bhp} = 27 - 22.85 = 4.15.$

(*d*) $e_m = \frac{\text{bhp}}{\text{ihp}} = \frac{22.85}{27} = 0.846 = 84.6$ per cent.

For taxation purposes, the normal output of a four-cycle automobile engine has been determined by the formula

$$\text{Bhp} = \frac{d^2 \times K}{2.5}, \tag{3}$$

where d = the diameter of the cylinder in inches, and K the number of cylinders. This rule is based on a piston speed of 1,000 fpm and has, of course, an arbitrary and conventional value only.

The rated horsepower of an internal-combustion engine to drive an electric generator of a given size is quite different from that of a steam engine to drive the same machine. This is due to the fact that a gas engine as rated has very little overload capacity, while a steam engine can carry a 25 per cent overload continuously and a 50 per cent overload for a short period of time. In order to allow for the overload capacity of the generator, the gas engine must be sufficiently large to drive the generator under that condition.

As an example, to drive a 2,000-kw generator, a 4,500-hp gas engine is used; to drive the same generator with a steam engine, a 3,000-hp engine is used.

It should be noted that, at present, internal-combustion engines are rated on their output, or brake horsepower, and steam engines are rated on their indicated horsepower. As already stated, internal-combustion engines are rated at practically their maximum capacity; steam engines are rated at the indicated horsepower at which they give the best economy.

It is possible to overload most Diesel and semi-Diesel engines, as they are now rated, from 10 to 25 per cent. In general, the makers rate this type of engine at the power at which they give best operation, but it is possible at a sacrifice in efficiency to obtain a greater power output than the manufacturer's rating.

Governed engines are rated at the governed speed. Aircraft engines are rated at the speed at which the engine can drive the propeller. Tractors have two ratings, a belt horsepower and a drawbar horsepower. The former is that which may be delivered by a belt, and the latter that available to pull a vehicle or implement. The latter is usually 50 to 60 per cent of the former.

271. Engine Performance.—As in the case of the steam engine, the power output is measured by means of a dynamometer. "Dynamo-meter" means power-meter, but the machine actually measures torque, *i.e.*, a force at a definite lever arm. Before the power output is actually determined, the speed of the engine must be indicated by a tachometer, or measured by a revolution counter and a stop watch. Dynamometers may be classed as absorption, transmission, or motor dynamometers according to whether they absorb, transmit, or develop the power they measure.

The Prony brake, previously described, is the simplest type of absorption dynamometer. It does not actually absorb the power but converts the mechanical energy into heat energy by means of the mechanical friction between two rubbing surfaces. For slow-speed engines the Prony brake and the pressure indicator may work satisfactorily, but for high-speed engines the indicator is unable to trace accurately the pressure and volume changes, and these engines develop too much power to be absorbed by mechanical friction from a practical flywheel. The heat developed per square inch of brake area results in excessively high surface temperatures which change the coefficient of friction and cause erratic loads.

The water brake is an absorption dynamometer which utilizes viscosity or fluid friction, instead of mechanical friction between sliding surfaces, to change mechanical energy into heat energy which is carried away by circulating water. An impeller, driven by the engine, rotates in a casing mounted on trunnion bearings

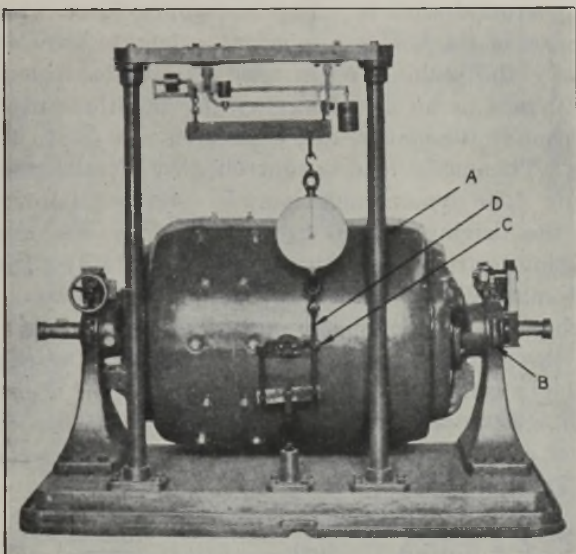


FIG. 334.—Sprague electric 300-hp dynamometer.

and restrained from turning only by a brake arm attached to the casing and a link connecting it to the scale beam. As the impeller rotates, moving the water with it, resistance to flow of the water reacts on the dynamometer casing and is registered on the scales. With no water in the casing the impeller runs free. The load is controlled by the quantity of water in the casing.

For testing aircraft engines a propeller or club is often used to absorb the power by the fluid-friction method, as well as to supply the air flow required to cool the engine. While an engine is developing the torque which turns a propeller in one direction, an equal and opposite torque is acting on the engine itself. By mounting the engine on trunnion bearings and providing it with a lever arm, this torque reaction may be easily measured.

For accurate testing of high-speed engines the electric dynamometer shown in Fig. 334 is used almost exclusively. This most versatile machine may be used as either a motor or an absorption dynamometer. It is a dynamo whose field casing A is supported on trunnion bearings B so that it is free to rotate, except that it is restrained at the knife-edge C by a link which connects it to the spring scales and the scale beam. The engine is coupled to the shaft of the armature whose rotation tends to drag the field casing around with it. The restraining force which prevents rotation of the field is applied at a definite lever arm and measured by the scales. When used as an absorption dynamometer, it acts as an electric generator and the current generated is usually dissipated, *i.e.*, converted into heat, in a grid resistance. The engine load is controlled by a resistance in the field circuit. The dynamometer may be used as a motor; it then measures the driving torque exerted to drive the armature. Compensation for the losses due to bearing and brush friction is made automatically. Whether operating as a motor or as a generator, the dynamometer measures exactly the torque delivered through the shaft either from motor to engine or from engine to generator. The speed of the motor is controlled by a field resistance. The dynamometer is usually supplied with a lever arm of such a length that the horsepower computation (Eq. 2) is simplified. For example, if

$$L = 1.3125, \qquad \text{then} \qquad bhp = \frac{nw}{4{,}000}; \tag{4}$$

or if

$$L = 1.75, \qquad \text{then} \qquad bhp = \frac{nw}{3{,}000}. \tag{5}$$

The fuel consumption of the engine may be measured by volume in gallons or liters, or by weight in pounds or kilograms. Usually a sensitive balance provided with electrical contacts is used to control the operation of a revolution counter and stop watch. The air consumption is measured by a gasometer or a calibrated orifice or Venturi. The efficiency of combustion is determined by analysis of the exhaust gases. In addition to the percentages of CO_2, O_2, and CO, the quantity of hydrogen and the unsaturated hydrocarbons should be determined.

Figure 335 is a curve sheet, on which are plotted the results of a standard-engine performance test made on a typical automobile engine.

The particular shape of the maximum torque curve on a given engine is determined in part by the carburetion and intake manifolding. The pulsating flow in the manifold gives rise to surging that may cause unequal mixture distribution, partly supercharging one cylinder and starving another. The maximum

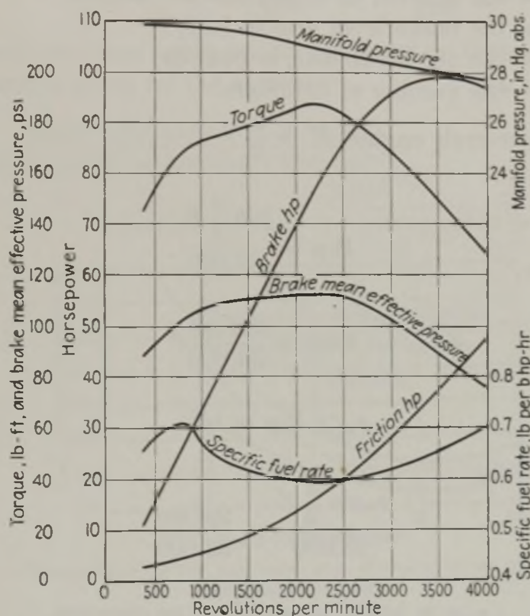


FIG. 335.—Performance curves for an automobile engine.

torque occurs at a speed where a balance is reached between friction loss and maximum average quantity of mixture introduced into the cylinders per cycle. Maximum brake horsepower occurs when a similar balance is reached between friction and the maximum quantity of mixture per unit time.

When engines of different size are to be compared, it is convenient to express the power output in terms of brake mean effective pressure. This is that part of the indicated mean effective pressure which remains after the pressure required to overcome engine friction has been subtracted. Stated in another way it is the indicated mean effective pressure multiplied by the mechanical efficiency.

Let e_m = the mechanical efficiency;
p = the indicated mean effective pressure in pounds per square inch;
l = the stroke in feet;
a = the piston area in square inches;
n = the number of revolutions per minute;
N = the number of explosions per minute;
L = the brake lever arm in feet;
w = the net brake load in pounds;
K = the number of cylinders.

For a four-cycle engine, $N = \frac{n}{2}$,

$$\text{Ihp} = \frac{pla\frac{n}{2}K}{33{,}000} \tag{6}$$

and

$$\text{Bhp} = \frac{2\pi Lwn}{33{,}000}; \tag{7}$$

but

$$e_m \times \text{ihp} = \text{bhp}. \tag{8}$$

Therefore,

$$\frac{e_m pla\frac{n}{2}K}{33{,}000} = \frac{2\pi Lwn}{33{,}000}.$$

Let

$$e_m p = \text{brake mean effective pressure,}$$

then

$$e_m p = \frac{4\pi Lw}{laK}. \tag{9}$$

If

Lw = torque, T, in foot-pounds, and $12laK$ = the engine displacement D in cubic inches,
then

$$e_m p = 150.8\frac{T}{D}. \tag{10}$$

The power output of an engine depends on the temperature and pressure of the atmosphere in which it operates. It is therefore desirable to correct the observed output so that it represents the power of the engine when operating in a standard atmosphere with a pressure of 29.92 in. Hg and a temperature of 60 F. The standard power output is obtained as follows:

$$\text{Bhp}_{\text{std.}} = \text{bhp}_{\text{abs.}} \times \frac{29.92}{(\text{barometer}_{\text{abs.}} - \text{in. Hg vapor pressure})} \sqrt{\frac{\text{absolute temperature F}_{\text{abs.}}}{520}}. \tag{11}$$

Thermal efficiency is almost always based on brake horsepower output and is given by the expression

$$e_t = \frac{\text{brake horsepower} \times 2{,}545}{\text{lb fuel per hr} \times \text{heating value per lb}} = \frac{2{,}545}{\text{specific fuel rate} \times \text{heating value per lb.}} \tag{12}$$

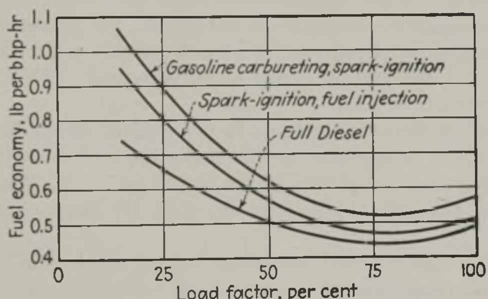


FIG. 336.—Comparison of the fuel economy characteristics of three types of combustion engines.

The specific fuel rate, in pounds of fuel per brake horsepower-hour is inversely proportional to the thermal efficiency for a given fuel and is frequently used when comparing the economy of operation of engines of different size. Figure 336 is an example of the use of the specific fuel rate as a basis of comparison.

272. Fields of Operation of Diesel and Otto Cycle Engines.—The innovation of Diesel-powered locomotives, particularly in light-weight high-speed trains, has spurred the development of steam locomotives, and numerous Diesel- and gasoline-powered rail car units are in operation throughout the country. Lack of flexibility and the apparently inescapable fuel-oil odor have as yet barred the Diesel from use in passenger cars, but its economy and reduced fire risk recommend it for use in buses, tractors, and trucks, although loss of compression after a time may reduce the new engine-economy figures. For this reason the oil-injection, spark-ignition engine has been developed to compete with the compression-ignition Diesel. Using a lower compression ratio, its "new engine" economy is not quite so good as a full Diesel, but it is contended that operation difficulties are reduced. A comparison of the relative economies of the three engines is shown in Fig. 336. The particular engine referred to can be used to burn gas, gasoline, or fuel oil by changing only the external fuel accessories, replacing the injection pump with a gas or gasoline carburetor as the case may be.

PROBLEMS

1. If the clearance volume of an Otto engine is 35 per cent, (*a*) what is the thermal efficiency when $\kappa = 1.4$; (*b*) when $\kappa = 1.3$?

2. Find the clearance volume in per cent of stroke for an Otto cycle engine that will give a thermal efficiency of 45 per cent. ($\kappa = 1.4$.)

3. If, in an engine operating in an Otto cycle, the pressure at the beginning of compression is 14 psia, (*a*) what compression will be required theoretically to give a thermal efficiency of 60 per cent? (*b*) What clearance will be required? ($\kappa = 1.4$.)

4. (*a*) Find the clearance volume in per cent of stroke for a four-cycle Diesel engine where the pressure at the beginning of the compression stroke is 14 psia and at the end of this stroke is 500 psia. (*b*) What will be the temperature at the end of compression, if at the beginning it is 100 F? ($\kappa = 1.4$.)

5. Determine the thermal efficiency for an eight-cylinder automobile engine whose dimensions are $3\frac{1}{16}$ in. by $4\frac{1}{4}$ in., compression ratio is 6.5, and piston displacement 250 cu in. ($\kappa = 1.4$.)

6. A 10-in. by 18-in. Otto engine has a volumetric clearance of 20 per cent. The pressure at the beginning of the compression stroke is 14 psia and the temperature 200 F. Find (*a*) the thermal efficiency; and (*b*) the pressure and temperature at the end of compression.

7. (*a*) If in a Diesel engine the fuel is injected for 6 per cent of the stroke, what will be the thermal efficiency if the clearance is 6 per cent? (*b*) What will it be if the fuel is injected for 12 per cent of the stroke?

8. In an internal-combustion engine developing 3,500 bhp the mechanical efficiency is 85 per cent, the jacket loss 33 per cent, and the thermal efficiency based on brake horsepower is 25 per cent. The jacket water enters at 85 F and leaves at 130 F. How many gallons of water must be circulated per minute?

9. The Society of Automotive Engineers formula for brake horsepower $= \dfrac{d^2 \times K}{2.5}$, where d = cylinder diameter in inches and K the number of cylinders. A certain eight-cylinder $3\frac{1}{4}$-in. by $3\frac{1}{2}$-in. engine develops 79 bhp at 4,000 rpm on a dynamometer test. Does the formula check? If not, state why.

10. An 8-in. by 15-in. single-acting gas engine runs 250 rpm and makes 100 explosions per min. The area of the indicator diagram is 1.05 sq in. and the length 3 in. The scale of spring used is 240 lb. A Prony brake was attached to the engine, the length of the brake arm being 63 in.; the gross weight, 75 lb; and the tare, 20 lb. Determine the mechanical efficiency of the engine.

11. A single-acting gas engine is 8 in. by 12 in. The indicator diagram from the engine has an area of 0.99 sq in. and a length of 3 in.; the scale of spring, 200 lb; rpm, 250; explosions per min, 200. The engine drives a generator developing 70 amp at 220 volts with an 80 per cent power factor. What is the combined mechanical and electrical efficiency of the plant?

12. An 8-in. by 12-in. single-acting, four-cycle gas engine, running 300 rpm, and having an average mean effective pressure of 70 psi, has a "hit-and-miss" type of governor and misses every fourth explosion. The length of the arm of the Prony brake is 48 in. and the tare on the scales 10 lb. Determine the gross weight on scales, if the mechanical efficiency is 85 per cent.

13. Given a single-cylinder, four-cycle gas engine whose bore is 5 in. operating with a piston speed of 1,200 fpm. Average area of indicator diagrams, 1.5 sq in.; average length of diagrams, 3 in.; scale of spring, 160 lb. Determine the indicated horsepower being developed.

14. A two-cycle gas engine is 24 in. by 30 in. and runs 120 rpm. It has a "hit-and-miss" type of governor and is running at less than full load, so that explosions occur every sixth stroke. An indicator diagram taken from the engine with a 100-lb spring has an area of 3.3 sq in. and is 3 in. long. (*a*) What indicated horsepower is the engine developing? (*b*) A Prony brake with an arm 6 ft long is attached to the engine, the gross weight on the brake is 1,000 lb and the tare is 150 lb. What is the mechanical efficiency of the engine?

15. An aircraft engine having a piston displacement of 1,000 cu in. per revolution is tested on a dynamometer having a lever arm 5 ft long. The engine runs at 2,000 rpm and produces a net load on the scale of 170 lb. It uses 180 lb of gasoline per hr. The heating value of the fuel is 20,000 Btu per lb. Find (*a*) the brake horsepower developed; (*b*) the thermal efficiency based on brake horsepower.

16. A six-cylinder, single-acting internal-combustion engine, 5 in. by 6 in., has a piston speed of 1,480 fpm. It develops 60 bhp and has a mechanical efficiency of 75 per cent. The mean effective pressure is 60.57 psi. The specific fuel consumption is 0.50 lb per bhp per hr. Fuel has a heating value of 19,500 Btu per lb. Determine (*a*) whether this is a two- or four-cycle engine; (*b*) the thermal efficiency based on brake horsepower.

17. A nine-cylinder, four-cycle, gasoline engine having a 6-in. bore and 7-in. stroke operates at 2,000 rpm. The indicator diagram area for each cylinder is 0.915 sq in. and length is 3 in. The spring constant is 300 lb; the mechanical efficiency is 85 per cent. Later a supercharger which requires 25 hp for its operation is geared to the shaft of the motor, and the area of the indicator diagram then becomes 1.13 sq in. Assuming that the mechanical efficiency of the engine proper remains unchanged, what is the net increase in the horsepower output of the unit?

18. A six-cylinder engine has a total piston displacement of 200 cu in. per revolution. The compression ratio of the engine is 6.7 to 1. When running with wide-open throttle at 3,600 rpm, the engine develops 80 hp. At this operating condition it uses 45 lb of gasoline per hr. The heating value of gasoline is 19,800 Btu per lb. Find (*a*) the clearance volume of each cylinder; (*b*) the torque developed in foot-pounds; (*c*) the specific fuel consumption in pounds per brake horsepower-hour; (*d*) the thermal efficiency, based on brake horsepower.

19. Given a two-cycle, four-cylinder marine gasoline engine, 4-in. bore, 6-in. stroke, running 500 rpm. Indicator diagrams from each cylinder show an average area of 1.2 sq in. and an average length of 4 in. The scale of the indicator spring is 250 lb. The engine is fitted with a Prony brake having an arm length of 15.75 in.; gross weight on scales, 180 lb; tare of brake, 20 lb. The engine consumes 0.8 lb of gasoline per ihp-hr. Gasoline has a heating value of 19,000 Btu per lb. Find (*a*) the indicated horsepower; (*b*) the brake horsepower; (*c*) the friction horsepower; (*d*) the mechanical efficiency; (*e*) the pounds of gasoline per brake horsepower per hour; (*f*) the actual thermal efficiency based on the indicated horsepower; (*g*) the actual thermal efficiency based on the brake horsepower.

20. The following data apply to a modern, high-speed, four-cycle single-acting Diesel marine engine: number of cylinders, 8; cylinder bore, 8 in.; piston stroke, 10 in.; speed, 750 rpm; developed brake horsepower, 325; heating value of fuel oil, 19,000 Btu per lb; engine consumes 0.42 lb of fuel oil per brake horsepower per hour; mechanical efficiency, 0.80. Determine (*a*) the mean effective pressure; (*b*) the indicated horsepower; (*c*) the friction horsepower; (*d*) the actual thermal efficiency in per cent based on brake horsepower; (*e*) the actual thermal efficiency in per cent based on indicated horsepower. (*f*) Assuming the engine to be fitted with a Prony brake having an arm length of 9 ft and a tare of 180 lb, what will be the total load in pounds on the scales?

21. (*a*) and (*b*) Determine the brake horsepower and the thermal efficiency based on the brake horsepower of a gasoline engine on which there are the following data: engine used 2 lb of gasoline, 19,800 Btu per lb in 4.67 min. The average speed during the run was 1,640 rpm and the net brake load 85.4 lb. The length of brake arm was 1.3125 ft. (*c*) Considering that the usual average amount of heat is lost to the cooling water, how many gallons per hour are required to cool the engine in the above run if the inlet temperature is 120 F, and the outlet temperature is 170 F?

22. An eight-cylinder aeroplane engine when tested at a constant speed and load requires 7 lb of gasoline, specific gravity 0.7, in 6.58 min. The engine turns 8,881 revolutions in this time. The net load on the brake is 280 lb with a brake arm of 21 in. Determine the fuel consumption in pounds per brake horsepower per hour.

23. A certain dirigible engine has a thermal efficiency of 25 per cent, develops 500 hp and uses a fuel having 20,000 Btu per lb and a specific gravity of 0.72. The dirigible can carry 1,000 gal of fuel, and fly 70 mph with this engine. How far and how long can the machine fly at this rate?

24. A gas producer produces $67\frac{1}{2}$ cu ft of gas containing 120 Btu per cu ft from each pound of lignite containing 11,100 Btu per lb. The engine supplied by this producer uses 100 cu ft of gas per bhp-hr when making 100 explosions per min. Find (*a*) the thermal efficiency of the producer; (*b*) that of the engine based on brake horsepower; and (*c*) the combined thermal efficiency of the producer and engine.

CHAPTER XVII

ECONOMY OF HEAT ENGINES

273. Relative Economy of Heat Engines.—Primarily, the efficiency and, in most cases, the economy, of heat engines depend upon the range of temperature of the working medium in the engine. As has been shown, the maximum thermal efficiency of any engine theoretically equals

$$\frac{T_1 - T_2}{T_1},$$

where T_1 is the initial absolute temperature of the working medium and T_2 is its final absolute temperature. In practice, it is found that the best heat engines are able to realize actually only about 60 per cent of the theoretical efficiency.

Table XXXV published by the Superheater Company shows how the steam consumption of various types of engines is decreased, or the economy increased, by the use of superheated steam and the consequent increase in the range of temperatures, $T_1 - T_2$.

TABLE XXXV.—RANGES IN STEAM CONSUMPTION OF PRIME MOVERS

Type engine	Steam consumption, lb per hp-hr		
	Saturated steam	100 degrees superheat	200 degrees superheat
Simple noncondensing	29 to 54	20 to 38	18 to 35
Simple noncondensing automatic	26 to 40	18 to 34	16 to 30
Simple noncondensing Corliss	26 to 35	18 to 30	
Compound noncondensing	19 to 28	15 to 25	13 to 22
Compound condensing	12 to 22	10 to 20	9 to 17
Simple duplex steam pumps	120 to 200	80 to 160	
Turbines noncondensing (kwhr)	28 to 60	24 to 54	21 to 48
Turbines condensing (kwhr)	12 to 42	10 to 38	9 to 34
Simple noncondensing locomotive	27	23	19.5
Steam shovels	42	34	29

An examination of the range of temperatures in the various forms of heat engines will give some clue to their probable actual efficiency. The following table gives a general idea of the possible efficiency of some of the more important prime movers:

Table XXXVI.—Thermal Efficiencies of Prime Movers

Types	Range of temperature in cylinders	Theoretical efficiency	Probable actual efficiency
Average noncondensing steam engine	116	14.5	8.7
Average condensing steam engine	226	27.8	16.7
High-pressure noncondensing steam engine	194	22.4	13.4
High-pressure condensing steam engine	279	32.2	19.3
High-pressure steam engine, superheated steam	381	39.6	23.8
Average condensing steam turbine, saturated steam	381	39.6	23.8
High-pressure condensing steam turbine, superheated steam	429	43.3	25.7
Small gas engine	900	39.5	19.5
Large gas engine	1,300	47.0	28.0
Large gas engine, high compression	1,400	52.2	31.6
Diesel motor, very high compression	1,900	60.0	36.0

This table gives some idea of the development and future possibilities of the various prime movers, considering them from a standpoint of heat efficiency. The internal-combustion engine is theoretically approximately twice as efficient as the steam engine.

274. Commercial Economy.—Heat efficiency, however, is not the only consideration. In actual operation, the important thing is the cost to produce a horsepower for a given period of time. A convenient unit of time is 1 year.

This cost of production involves a great many considerations. In determining this cost the following items should be considered:

1. Interest on the capital invested.
2. Depreciation of machinery and building structures.
3. Insurance.
4. Taxes.
5. Fuel cost.
6. Labor of attendants.
7. Maintenance and repairs.
8. Oil, waste, water, and other supplies.

The first four of these items are called the *fixed charges*, and remain the same no matter what the load on the plant may be. The last four items are the *operating expense*, and vary with the conditions of operation. The sum of the fixed charges and operating expense is the *total operating cost.*

In most plants the cost of coal is from 25 to 30 per cent of the total operating expense. A saving in the coal cost of operating is not always a saving in the total cost of operating. This saving may involve so much increased cost of installation that the additional fixed charges on the new capital invested will more than offset the saving in coal. This is well illustrated by the condition that exists in localities having very cheap coal.

A careful comparison of plant-operating costs for a condensing and a noncondensing plant often shows that the cost of operating the noncondensing is less than that of the condensing plant, owing to the fact that the increased cost of the condensing plant adds more to the interest and depreciation charges than is saved on the cost of coal used, which is less than in a noncondensing plant.

In central power stations, the cost of the boiler and piping system and the prime mover is approximately 40 per cent of the entire installation cost. In the larger stations it may be only 25 per cent to 30 per cent. With an average station cost of $100 per kw, 15 per cent should be allowed for fixed charges.

PROBLEMS

1. A 300-hp automatic engine uses 28 lb of steam per ihp-hr, and costs $4,500. A 300-hp Corliss engine uses 20 lb of steam per ihp-hr, and costs $6,000. Steam at the plant costs 20¢ per 1,000 lb. The plant runs 10 hr per day and 300 days per year. Allowing 6 per cent on the investment for interest, 10 per cent depreciation on the automatic engine, and 7 per cent on the Corliss, (*a*) which is the more economical engine to buy? (*b*) What is the saving per year of this engine over the other?

2. An engine developing 500 hp, 10 hr per day, 300 days per year, is guaranteed to use 14.5 lb of steam per ihp-hr, but by actual test it uses 15 lb. If it costs 17¢ to make 1,000 lb of steam, and interest is 6 per cent and depreciation 5 per cent, how much should be deducted from the purchase price of the engine to compensate the purchaser for the increased cost of operation above that required under the guarantee? (Suggestion: Find the present worth of an annuity equal to the cost per year of the excess steam consumption.)

3. It requires a 1,000-ihp engine to operate a mill running 24 hr per day and 300 days per year. The present engine uses 20 lb of dry steam per ihp-hr and has an average repair cost per year of $300. A new engine is to be installed costing $15 per ihp and using 15 lb steam per ihp-hr. The steam pressure is 150 psia; feed-water temperature, 200 F. The actual evaporation per pound of dry coal is 9 lb and the coal costs $3 per ton. The old engine was sold for $5 per hp. Interest is at 5 per cent; depreciation and repairs on new engine, 10 per cent. What is the net saving for the owner per year?

4. A power plant is to deliver 1,000 kw at the switchboard. A steam engine can be installed which will give an economy of $14\frac{1}{2}$ lb steam per ihp-hr. The steam pressure is 145.3 psi; feed-water temperature, 150 F; dry steam. The engine efficiency is 92 per cent; generator efficiency, 95 per cent. A steam turbine can be installed which will give a steam consumption of 20 lb per kwhr. Steam pressure, 145.3 psi; 150 deg superheat; feed-water temperature, 150 F. The cost of generating steam in each case is 20¢ per 1,000,000 heat units. (*a*) Which is the more economical installation? (*b*) How much is saved per hour by this one over the other?

5. A steam engine uses 20 lb of steam per ihp-hr and develops 200 hp. A 20-in. by 24-in. single-acting gas engine running 220 rpm is being considered for the place. It uses 10,000 Btu per ihp-hr when making 105 explosions per min and developing an average mean effective pressure of 100 lb. The efficiency of the boiler plant is 70 per cent; efficiency of gas producer, 80 per cent. The steam-engine plant costs $20,000; gas engine and gas-producer plant cost $30,000; the cost of labor is the same for both plants. Coal costs $3 a ton and contains 13,000 Btu per lb. The steam pressure in the boiler plant is 100 psi and the temperature of the feed water is 180 F. If the interest charges are 5 per cent, and the repairs and the depreciation 10 per cent, (*a*) which would be the cheaper plant, and (*b*) how much, to run 10 hr a day for 300 days a year?

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