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STEAM TURBINES

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STEAM TURBINES

A SHORT TREATISE ON THEORY DESIGN, AND FIELD OF OPERATION

BY

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PREFACE.

The purpose of this book is twofold, to provide, first, a textbook suited to a short course on steam turbines in engineering schools and, second, a book for the engineer on the principles and general design of turbines without going into refined treatment of the more difficult problems entering into their design.

At the end of each article is given a short list of references where those wishing to pursue the subject will find it treated most thoroughly. Other references are given among the problems to sources available to most students and it is suggested that these be made subjects for short reports, thus extending the reading of the student beyond the limits of the present text.

The writer wishes to thank Mr. Geo. A. Orrok of the New York Edison Company, the engineers of the prominent turbine manufacturers, for data and illustrations used, and especially Mr. C. C. Perry of the Sheffield Scientific School for his help in the preparation of the material.

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April, 1, 1911.

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CHAPTER I.

THE FREE EXPANSION OF STEAM.

Art. 1.—The Energy of a Jet.

The power developed in a steam turbine is derived from one or more jets of steam. The power available depends on the kinetic energy of the steam available under the given conditions of operation.

The total kinetic energy in a moving mass of weight W and velocity V is

$$K=\frac{WV^2}{2g} \qquad (1)$$

where the body is brought to rest. Where the velocity is reduced from V to V_1 the kinetic energy available in the change is

$$K=\frac{W(V^2-V_1^2)}{2g}. \qquad (2)$$

Or, from the standpoint of the force of the jet we have $Ft=mV$. Taking the time as unity the impulsive force acting for one second is

$$F=mV=\frac{WV}{g}. \qquad (3)$$

If A = area of the jet and w = the weight per cubic foot of the steam at the given pressure and dryness or superheat, the weight delivered per second is $W=wAV$. Substituting, Eq. (3) becomes

$$F=\frac{wAV^2}{g}. \qquad (4)$$

For any given pressure and dryness w is known. Both the impulsive force and the kinetic energy, K, available become determinate as soon as V is known. An important part of the theory of turbines has for its object the determination of this V for any given set of conditions.

Taking the weight of steam in Eq. (1) as unity we have, since the value of $2g$ is a known constant, a definite relation between K and V by which the velocity V may be calculated when the energy available per pound of steam is known.

The energy of a gas such as steam may exist in two forms, heat and kinetic energy. It may consist wholly of heat. In this condition the gas is confined and under pressure from the energy of its particles, due to heat. If allowed to do so it would expand into a medium at lower pressure, doing work; but since it is confined and at rest, this capacity for work is potential only and analogous to that of a weight suspended and doing no work, but capable of doing it if released.

Or, theoretically, having been allowed to expand under the influence of its heat tension or pressure until all the heat had been expended, the energy would be wholly kinetic. This condition, however, is unattainable, and is analogous to the energy of the weight if it had fallen freely from its point of suspension to the center of the earth. Practically, therefore, there is a third or intermediate condition where the energy exists in both forms, and the total energy of the steam is the sum of its heat energy and the kinetic energy. As the weight can fall only to the surface of the ground, so steam in expanding can not give up all its heat, but can part with it only to a point corresponding to the temperature of the medium into which it expands.

In the British system of units heat is measured in British Thermal Units and energy in foot-pounds, the two being interchangeable at the ratio of 1 B.T.U = 777.5 foot-pounds.[1] From the Law of Conservation of Energy the total energy in a pound of steam during expansion remains a constant. Expressing this in the form of an equation,

$$777.5\,H_1 + \frac{WV_1^2}{2g} = 777.5\,H_2 + \frac{WV_2^2}{2g}$$

[1] The heat values throughout this book are those given in Marks and Davis' "Steam Tables and Diagrams."

where H_1V_1 and H_2V_2 are the heat energies and steam velocity for any two stages of the expansion. Remembering that W is unity

$$777.5\ (H_1 - H_2) = \frac{V_1^2 - V_2^2}{2g}. \tag{5}$$

In the expansion of steam from a vessel $\frac{V_1^2}{2g}$ may be neglected, therefore Eq. (5) becomes

$$\frac{V^2}{2g} = 777.5(H_1 - H_2) \quad \text{or}$$
$$V = 223.8\sqrt{H_1 - H_2} \tag{6}$$

in which V is the theoretical velocity attained, and H_1 and H_2 the total heat content in B. T. U. per pound of steam before and after expansion.

Problems.

1. What is the impulsive force when five nozzles are each discharging .96 lb. of steam at a velocity of 3450 ft. per sec.?
2. What velocity must a pound of steam have to deliver 87,000 ft. lbs. of work per sec.?
3. What energy per lb. will steam give up when slowed down from 2150 ft. per sec. to 1210 ft. per sec.?
4. What energy per lb. of steam will be available for a decrease in velocity from 3090 ft. per sec. to 2150?
5. What weight of steam must be delivered per sec. to give 60,000 ft. lbs. of work per sec. when the velocities before and after expansion are 2500 and 1500 ft. per sec.?
6. What is the theoretical H. P. of a jet delivering 1.3 lbs. of steam per sec. at 3500 ft. per sec.?
7. What must be the initial velocity of steam which delivers in one jet 40 H. P. per lb. of steam delivered, the final velocity being 1600 ft. per sec.?
8. What is the impulsive force in .092 lb. of steam moving at 1330 ft. per sec., also at 3680 ft. per. sec.?
9. How many H. P. are available in a heat drop of 42 B. T. U. per sec.?
10. What heat drop is required to generate a theoretical velocity of 3380 ft. per sec.?

11. What is the H. P. per lb. of steam of a jet where the heat drop is 235 B. T. U.?
12. What is the heat equivalent in B. T. U. per lb. of the work done when steam is reduced in velocity from 3400 ft. per sec. to 1300 ft. per sec.?
13. How many B. T. U.'s are the equivalent of a H. P. hr.?

References.

THOMAS: "Steam Turbines."
JUDE: "The Theory of the Steam Turbine."
Standard works on thermodynamics.

Art. 2.—The Temperature-entropy Diagram.

The determination, then, of the theoretical steam velocities which are to govern the design, depends on the heat drop in the stage considered. This heat drop may be calculated analytically from steam tables, or determined more conveniently, and with sufficient accuracy, from the entropy diagrams published with steam tables or the works on turbine engineering.

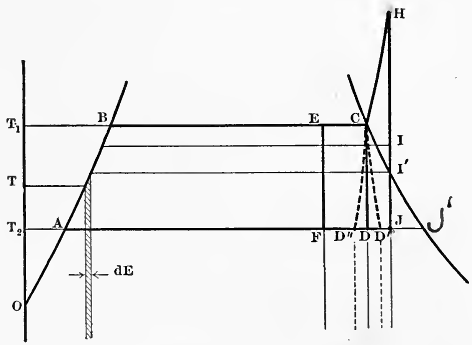


FIG. 1.—Temperature-entropy diagram.

In the temperature-entropy diagram for steam, the absolute temperature of the water or steam is shown by the ordinate of a point in a curve, and the total heat per pound by the projected area underneath the curve. The corresponding abscissa represents a quantity known as the entropy. As engine and turbine problems are concerned with certain changes in heat which never involve the total contents, that portion only of the diagram above the melting point of ice need be drawn.

The addition of a small amount of heat, dQ, at some temperature T would be represented by the shaded slice, Fig. 1, or

$$TdE = dQ. \tag{7}$$

If S be the mean specific heat during a change the heat added is $Q = S(T_1 - T_2)$, or for differentials, $dQ = SdT$. Substituting, we have

$$dE = S\frac{dT}{T}. \tag{7a}$$

Integrating, the change of entropy for a rise in temperature from T_2 to T_1, as from A to B, is

$$E = S\int_{T_2}^{T_1}\frac{dT}{T} = S\,\log_e\frac{T_1}{T_2}. \tag{8}$$

Equations (7) and (7*a*) represent the curve OB, the projected area beneath this line showing the heat contents of water. When heat is added as latent heat and goes wholly into changing the condition, as during vaporization, the temperature remains constant and Eq. (7) becomes $\frac{dQ}{dE} =$ a constant, *i.e.*, the entropy and the heat vary directly and the curve is a horizontal straight line represented by

$$E = \frac{Q}{T}.$$

As the latent heat varies for different temperatures the corresponding horizontal lines, such as BC, will vary in length and a curve, CI', drawn through their ends will be the locus of the saturation points. After the water has been wholly converted into steam, further addition of heat raises its temperature and the equation of the curve resumes the form (8), with S as the variable specific heat of superheated steam.

Expansion through an orifice or short nozzle is so rapid as practically to give the conditions of adiabatic expansion, in which there is no exchange of heat between the steam and the walls surrounding it. Such heat as is given up goes into external work only, which is done at the expense of the internal energy of the steam. For instance, let the steam be at the condition indicated at C, that is, at the temperature T_1, and just dry and saturated. If allowed to expand adiabatically to a lower temperature T_2 the vertical line CD would indicate adiabatic expansion, since any other line as CD' or CD'' would have an area under it indicating heat added to or subtracted from the steam, as heat, during the expansion. The external work done, equal to the heat energy represented by the area $ABCD$ is accounted for by the lowering of the quality of the steam, or the condensation of a portion. If the quality of the steam was originally below the saturation point, as at E, then the quality at the lower pressure will have some other value, as shown by the position of F. If the steam were originally superheated, as indicated at H, adiabatic expansion will reduce the amount of superheat as at I, or if continued bring it to saturation as at I', or if still further continued condense part of the steam as indicated by the position of J.

Problems.

(Use standard steam tables, such as Marks and Davis'.)

1. What is the increase in entropy from water at 65° Fahr. to dry, saturated steam at 320° Fahr.?
2. The entropy of evaporation at 360° Fahr. is 1.0514. What is the latent heat of evaporation for that point?
3. Referring to Fig. 1, under what conditions will steam expanding adiabatically from T_1 to T_2 have its quality raised, when will it remain about the same, when will it be lowered?
4. What will be the final quality of steam expanded adiabatically from the dry, saturated point, C, 140 lbs. gauge pressure to 28 inches of vacuum? 1#ab .87
5. Given steam at 170 lbs. absolute and superheated 100° Fahr., at what pressure will it be dry and saturated when expanded adiabatically, at what pressure will its quality be 90%? (Take the specific heat of superheated steam at 0.6.)
6. Given steam at 160 lbs. gauge pressure and 98 1/2% quality, what is

the quality or superheat at atmospheric pressure which corresponds to the same total heat?

7. Given 1 lb. of water at 70° Fahr., what will be its condition when 1148 B. T. U.'s have been added and its absolute pressure is 27 lbs.? 75° super

8. What is the B. T. U. drop per lb. for adiabatic expansion from 150 lbs. gauge pressure and 80° Fahr. superheat to a pressure corresponding to 28 1/2 inches of vacuum? 354.5

References.

THOMAS: "Steam Turbines."
JUDE: "The Theory of the Steam Turbine."
NEILSON: "The Steam Turbine."
MOYER: "Steam Turbines."
FRENCH: "Steam Turbines."

Art. 3—The Heat-entropy Diagram.

Prof. Mollier has developed a heat diagram based upon the temperature-entropy diagram, but more convenient to work with. Copies of this chart, for both British and Continental units, are published with a number of the standard works on steam turbines. An excellent one is published with Marks and Davis' Steam Tables, a part of which, on a reduced scale, is reproduced here.[1]

In this chart the heat content of the steam at any given condition of pressure and dryness or superheat, appears as the ordinate in a right-angled co-ordinate system, instead of an area as in the temperature-entropy diagram, and the entropy appears as the abscissa. Any condition of the steam may be expressed by the position of a point in this plane. The points of equal pressures being connected, there results the series of curves of equal pressures running upward from left to right. Similarly, by connecting points of constant dryness and constant superheat we have curves of constant quality and superheat. A vertical line from an initial condition to the pressure line of the final condition shows the heat drop due to adiabatic expansion where there is no change in entropy, and the heat change may be read off directly on the scale of the chart. Horizontal lines indicate changes at constant heat or changes of condition, and therefore give the change in quality for moist

[1] The Marks and Davis Diagrams I, Total Heat-entropy Diagram and II, Tota Heat-pressure Diagram, may be purchased at $0.40 net, from the publishers, Longmans, Green, and Co., 4th Ave. and 30th St., New York City.

steam, or in temperature or superheat for superheated steam. As the theoretical velocity developed in a nozzle is a function of certain constants and the heat drop only (Eq. 6), an additional scale may be plotted from which the velocity corresponding to the heat change may be read off at once. A third scale of the foot-pounds available per pound of steam is also added, in which the values are 777.5 times the B. T. U. drops.

Problems.

1. What is final condition of dry steam expanded adiabatically from 150 lbs. gauge to 1 lb. absolute? .774
2. What will be the quality of a lb. of steam at atmospheric pressure which has the same total heat as steam at 50 lbs. absolute and 95% quality? .973
3. Let steam be expanded adiabatically from 170 lbs. absolute and 220° superheat. What is its pressure when dry and saturated? 36 At what pressure will it have a quality of 88%? 3#
4. If dry steam is expanded adiabatically from 100 lbs. gauge pressure what would be its final condition if the velocity obtained is 3600 ft. per sec.? .815 at 2.5#
5. Let steam be expanded from 140 lbs. gauge and 100° superheat and attain a velocity of 3500 ft. per sec. How many ft. lbs. of work per lb. of steam are being carried away as lost heat in the exhaust?
6. Steam expands from 160 lbs. gauge and 120° superheat to 1.5 lbs. absolute and 89% quality. What would be the quality if it had expanded adiabatically to the same pressure? .826 What is the cause of the increase in quality?
7. A lb. of steam at 130 lbs. gauge pressure generates 350 H. P., what is the quality at 1 lb. absolute? What is the change in entropy?
8. How many ft. lbs. of energy are available per lb. in the adiabatic expansion of dry steam from 150 lbs. gauge to atmospheric pressure, and also from atmospheric pressure to 1 lb. absolute?
9. Let dry saturated steam be expanded adiabatically from 100 lbs. absolute to 1 lb. absolute. How many B. T. U.'s are available with an atmospheric exhaust? 138
10. With dry steam at 140 lbs. gauge pressure and expanding to 1 lb. absolute, how many ft. lbs. per lb. would be lost for a discharge at the saturation point over a discharge under pure adiabatic expansion?

References.

Marks and Davis: "Steam Tables and Diagrams."

Art. 4.—The Velocity of a Jet.

In calculating the heat drop, in order to get the velocity, H_1 is usually found directly from the initial conditions. H_2, however, is not known immediately, as only one of the conditions, usually the pressure, is directly given. The dryness or quality of the steam at the lower pressure must be determined before

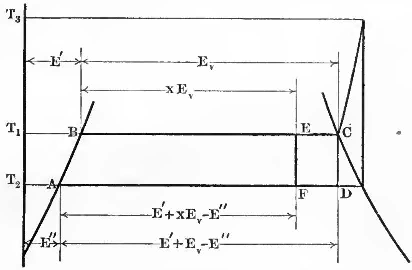


FIG. 2.—Temperature-entropy diagram.

H_2 is known. Assuming first an adiabatic drop in pressure, we will have the following equations, referring to the entropy diagram, Fig. 2:

$$V = 223.8\sqrt{q_1 + xL_1 - q_2 - T_2(E' + xE_v - E'')} \qquad (9)$$

for moist or dry saturated steam, x being the quality, and

$$V = 223.8\sqrt{q_1 + L_1 + S(T_3 - T_1) - q_2 - T_2\left(E' + E_v + S\log_e \frac{T_3}{T_1} - E''\right)} \qquad (9a)$$

for superheated steam, where

q_1 and q_2 = total of the liquid at the initial and final pressures, P_1 and P_2,
L_1 = latent heat of vaporization for P_1
T_1 and T_2 = the absolute temperatures at initial and final pressures,
T_3 = the absolute temperatures of the superheat, as indicated in Fig. 2,
S = the specific heat of superheated steam.

This last value has been the subject of extensive experiment, and is still open to discussion. A working value, in use by engineers engaged in handling superheated steam, is from .6 to .65. Variations in the specific heat are taken into account in the heat-entropy chart. As the heat drop in the temperature-entropy diagram, Fig. 2, is represented by the area $ABEF$, and as the side AB approximates a straight line, the figure may be assumed to be a trapezoid. The area then would be $(T_1 - T_2)\left(\frac{BE + AF}{2}\right)$.

With this assumption (9) reduces to

$$V = 158\sqrt{(E_1 + E_2)(T_1 - T_2)} \tag{10}$$

and (9a) to

$$V = 158\sqrt{(E_1 + E_2)(T_1 - T_2) + 2S\left[(T_3 - T_1) - T_2 \log_e \frac{T_3}{T_1}\right]}. \tag{10a}$$

E_1 in the case of initially moist steam is the entropy $BE = xE_v$, where x is the quality of the steam and E_v the entropy of evaporation. For dry saturated steam, E_1 becomes E_v, x being 1. E_2 is the entropy AF or AD according as initial steam is moist or dry. It may be scaled directly from the entropy diagrams or found from steam tables by adding to E_1 the difference between the entropy of the liquids at T_1 and T_2. The error introduced by using this trapezoidal approximation is only about 1%.

A nozzle will not, however, deliver steam at the full velocity represented by the heat drop between the inlet and outlet conditions. Whatever the contour of the nozzle, frictional resistances are offered to the flow, and the steam must give up some of its heat energy in overcoming them. The heat thus used raises the temperature of the surrounding walls and of the particles of the steam itself. As the temperature of the steam falls during further expansion, this heat re-evaporates part of the water of previous condensation and raises the quality of the steam. If the frictional work be sufficient to maintain its heat constant during the fall of temperature, there is of course no heat drop during the expansion, and consequently from Eq. (6)

no velocity. Such is the case where the work of friction is all spent in raising the internal energy of the steam as in a throttling calorimeter, where the final velocity is practically zero.

Adiabatic and constant heat expansion are shown by the entropy diagram, Fig. 3. With steam at condition E adiabatic expansion would be shown by the vertical drop EF, the heat drop causing increase of velocity being represented by the area $ABEFA$. An expansion at constant heat would occur along a line EF', such that the total heat, shown by the area beneath Amn would at all times be equal to that beneath ABE. At the

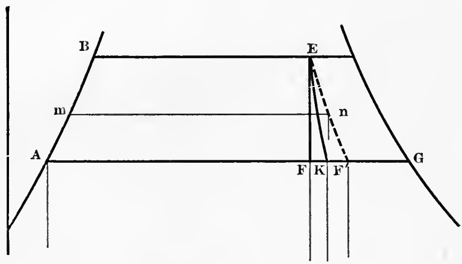


Fig. 3.—Temperature-entropy diagram.

final temperature the total heat area beneath AF_1 being equal to that below ABE, no heat drop would be shown and there would consequently be no velocity. Actual expansion in turbine nozzles occurs along some such curve as EK, between the adiabatic and this constant heat line. The heat drop producing kinetic energy is represented by the difference of total heat areas beneath ABE and AK. Under adiabatic conditions the area under AF subtracted from the total area under ABE, gives the area corresponding to the available heat energy. Under the actual conditions the area under AK must be subtracted. The excess area under FK represents, then, the heat which might have been applied to increasing the velocity of the steam, but is lost energy due to the frictional resistances. In dealing with the relative areas it should be borne in mind that the scale of the ordinates and the abscissas are widely different.

If y be the percentage of the heat drop H_1-H_2 which is so lost, $(H_1-H_2)\ (1-y)$ will be the heat energy converted into kinetic energy, and this expression may be substituted in equations (6), (10), and (10a) giving

$$V=223.8\sqrt{(H_1-H_2)(1-y)} \tag{11}$$

$$V=158\sqrt{(E_1+E_2)(T_1-T_2)(1-y)} \tag{12}$$

$$V=158\sqrt{\left[(E_1+E_2)(T_1-T_2)+2S\left\{(T_3-T_1)-T_2 log_e\frac{T_3}{T_1}\right\}\right](1-y)} \tag{12a}$$

where E_1 and E_2 are the entropy changes for adiabatic expansion as before.

The percentage of re-evaporation, or increase in the quality of the steam, FK, is readily obtained. The lost heat, $y(H_1-H_2)$, is equal to X_2L_2, where L_2 is the latent heat of evaporation or the area under AG, and X_2 is the percentage of FK to AG. Therefore

$$X_2=\frac{y(H_1-H_2)}{L_2}. \tag{13}$$

In turbine design y is usually determined by experiment for the type of nozzle to be used. An average of the results of many investigations gives about 10% as the value of y, which corresponds to a velocity of about 95% of the theoretical velocity.

Problems.

1. Let steam at 98 1/2% quality and 145 lbs. gauge expand adiabatically to 1.5 lbs. absolute, what is the velocity obtainable by Eq. (9)? Check this by the heat-entropy chart.
2. What velocity is obtainable for an adiabatic expansion from 180 lbs. absolute and 120° superheat to 80 lbs. absolute, and also from dry steam 3 lbs. absolute to 1 lb.?
3. What velocity is obtainable with an expansion of from 150 lbs. gauge and 60° superheat to atmospheric pressure, with an energy loss of 20%? What is the condition of the exhaust?
4. Let dry steam be expanded from 140 lbs. gauge to 2 lbs. absolute. What will be the quality of the steam when the energy loss during expansion is 14%?
5. What is the velocity of steam expanded from 155 lbs. gauge and 50° superheat to 1.5 lbs. absolute with 15% energy loss?
6. What is H. P. per lb. of steam represented by the conditions of Prob. 5?

7. What must be the initial conditions when steam expanded to 1 lb. absolute and 90% quality generates 370 theoretical H. P. per lb. of steam with an energy loss of 85 B. T. U.?
8. Construct a condition curve for the expansion of steam from 155 lbs. absolute and 80° superheat to 1.5 lbs. absolute, with a reheating loss of 15%. What is the condition of the steam at 100 lbs., 50 lbs., and atmospheric pressure?
9. What is the energy loss when the velocity loss is 6%?

References.

STODOLA: "The Steam Turbine."
THOMAS: "Steam Turbines."
JUDE: "Theory of the Steam Turbine."
MOYER: "Steam Turbines."
FRENCH: "Steam Turbines."

Art. 5.—Weight of Steam Delivered and Impulsive Force.

Both experiment and theory show that for a given upper pressure p_1 there exists a certain lower pressure p_2 called the critical pressure, for which the *quantity discharged* becomes a maximum, and that however far the final pressure be lowered the quantity delivered is not materially increased. The final velocity and weight discharged will depend upon the final pressure only when that pressure is higher than the "critical pressure" (about $.58p_1$). There exists in an expansion nozzle for final pressures lower than this critical pressure, a zone near the inner end where the velocity and weight of flow are the same whatever the lower pressure. Beyond this point further expansion to the final pressure may take place, with a marked increase of velocity, but the quantity delivered will not be increased. While the final velocity attained depends on the final conditions, the quantity discharged depends solely on the critical pressure, provided the final pressure is less than the critical pressure. This pressure varies somewhat with different gases, as shown below.

Gas	Critical Pressure
Dry air	$.526\ p_1$
Superheated steam	$.546\ p_1$
Dry saturated steam	$.577\ p_1$
Moist steam	$.582\ p_1$
The value for steam is usually taken at	$.58\ p_1$

It can also be shown that whatever the initial pressure, the critical velocity of the steam in the throat created by the drop in pressure to $.58\ p_1$ is about the same, and that for that and all lower final pressures the theoretical velocity in the throat is about 1450 feet per second.

To determine, then, the weight of flow, only the critical velocity and the area of the orifice, or, in the expanding nozzles, of the throat, can be considered. The weight delivered is

$$W=\frac{AV}{144v} \tag{14}$$

where $A=$ the area of the orifice, or least cross section, in square inches., and $v=$ the volume in cubic feet per pound at the critical pressure $.58\,p_1$. Substituting for V the value given in Eq. (10) we have

$$W=\frac{1.1A}{v}\sqrt{(E_1+E_2)(T_1-T_2)}, \tag{14a}$$

E_2 and T_2 being taken for the critical pressure, and not the final pressure.

The available energy per pound of steam multiplied by the pounds of steam per H. P. hour is equal to 33000×60. Therefore the pounds of steam per H. P. hour $=\dfrac{33000\times60}{777.5(H_1-H_2)}$ or the theoretical steam rate per H. P. hour $=\dfrac{2545}{H_1-H_2}$. (15)

This must be increased by the percentage of mechanical losses, such as windage, friction, etc., to obtain the commercial rating.

From Eq. (3) the impulsive force, F, may be determined by substituting the values of V and W found in Eq. (10) and (14a), remembering the E_2 and T_2 in Eq. (10) apply to the final pressure, while in Eq. (14a) they apply, together with v, to the critical pressure. A simple formula, developed by Mr. George Wilson, for determining the reaction of orifices flowing into atmospheric pressure, while largely empirical, agrees very closely with experimental results. The reaction $=1.23\ p_1-14.7$ pounds per square inch of orifice. This applies, of course, only to non-condensing turbines.

Problems.

1. What is the theoretical steam rate of a turbine having a heat drop of 235 B. T. U. per lb. of steam?
2. What heat drop must steam have to give a theoretical steam rate of 18 lbs. per H. P. hour?
3. A 200 H. P. turbine having eight nozzles has a heat drop of 250 B. T. U. per lb. of steam. What must be the throat area of nozzle to pass the required weight per sec. when the initial pressure = 150 lbs. gauge.
4. What is the reaction of an orifice .35 sq. inch in area discharging from 100 lbs. gauge pressure into the atmosphere?
5. What throat diameter is required in a nozzle to give a reaction of 12 lbs. when expanding from dry steam at 120 lbs. absolute to atmospheric pressure?
6. What area of throat is required in a nozzle to discharge .12 lbs. of steam per sec. when the specific volume is 5.1 cu. ft.?
7. Steam at 150 lbs. absolute is expanded with an energy loss of 5% at the throat, what area is required to discharge .1 lb. per sec.?
8. What is the impulsive force of the jet in the throat for conditions of Prob. 7?
9. What is the impulsive force of the jet if the expansion of Prob. 7 is continued down to 1.5 lbs. absolute with a total energy loss of 10%?

References.

THOMAS: "Steam Turbines."
MOYER: "Steam Turbines."
FRENCH: "Steam Turbines."

CHAPTER II.

UTILIZATION OF KINETIC ENERGY IN STEAM.

Art. 6.—Expanding Nozzles.

When expanding nozzles are used, the pressure in the throat, a, can not be less than $.58\ P_1$, but having passed this point, there is nothing to preclude further expansion down to the pressure of the surrounding medium. This further expansion increases the velocity of the steam, but unless the flow is concentrated in one direction the energy will be dissipated, and for any useful purposes lost. The function of the divergent nozzle used in turbine design is to direct the jet issuing from the throat in a definite direction so that the kinetic energy generated may be made available. There is no more energy in the flow from an expanding nozzle than from a simple orifice, but if

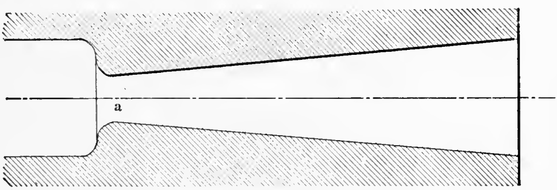


Fig. 4.—Expanding nozzle.

properly proportioned the nozzle allows the steam to expand from the critical pressure to the final pressure with a minimum of vibration, and renders the kinetic energy of the jet available along some definite line of action.

The proper proportions for expanding nozzles have been the subject of extensive experiments. The most notable results are those of Rosenhain, Stodola, Rateau, and Gutermuth, and the material brought out in these investigations is so great that little more can be done here than to point out some of the general results.

For short nozzles, whether rounded at the entrance, convergent, or straight, there is a sudden drop of pressure just beyond the throat or entrance, to a pressure below that of the discharge chamber. The steam returns to the pressure of the exhaust medium in a series of oscillations roughly proportionate to the depression, and decreasing, as a damped vibration, until at perhaps 1 or 1 1/2 times the length of the nozzle the pressure has settled down to that of the lower medium. It is the function of a well designed divergent nozzle to reduce these oscillations to a minimum. These vibrations are clearly shown in Fig. 5, taken from Stodola's experiments. Fig. 6 shows the effect of the divergent nozzle, where it is seen that there are practically no oscillations of pressure within the nozzle after the one created at the throat has died out. If the final pressure, beyond the nozzle, is either above or below that corresponding to the expansion within the nozzle, oscillations will be set up in the exhaust space. These are shown clearly from the experiments of Dr. C. E. Lucke, Fig. 7,[1] which were made with a De Laval nozzle, the best known for giving complete expansion. In this nozzle the divergent portion is conical. Parenty has deduced mathematically that the most efficient curve is that of an ellipse with the focus in the throat. Experiments seem also to indicate that the best form for the divergent part is to have it slightly concave, and some turbines have adopted this form. But for manufacturing reasons they are usually made conical, as the increase in efficiency is only slight.

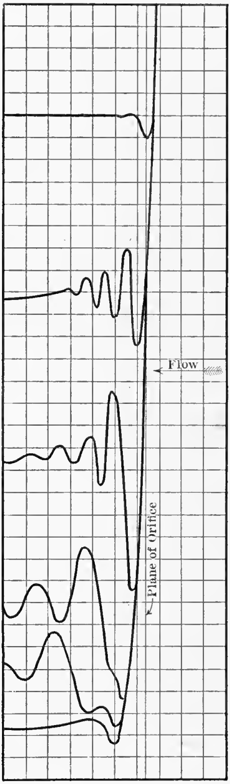


FIG. 5.—Expansion curves.

[1] A. S. M. E., Vol. XXVI., page 134.

The form recommended by Rosenhain had the inner corner but slightly rounded, and a straight divergent taper beyond, of about 1 in 10 or 1 in 12. A greater efficiency appears to be obtainable with a nozzle which under-expands rather than one which over-expands, as the loss of energy increases rapidly with over-expansion. With a well proportioned nozzle a velocity efficiency

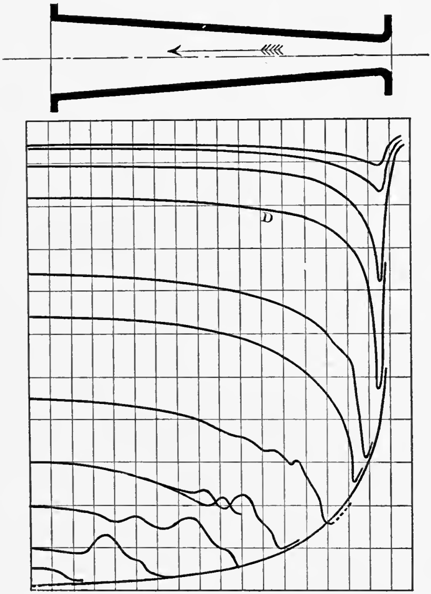


FIG. 6.—Pressure curves in an expansion nozzle.

of 95% may be relied on. Fig. 8 shows full size the contour of a De Laval nozzle for a non-condensing turbine.

Mr. Strickland Kneass, of Wm. Sellers and Co., has made an extensive investigation into the discharge of steam through nozzles and arrived at a form differing quite markedly from the De Laval nozzle, but which has shown an equally high if not higher efficiency. Fig. 9 shows a section of some of the nozzles tested and gives curves of the variations in pressure and velocity for steam at initial pressures of 20, 30, 60, 90, and 120 pounds.

Taking the 1 in 10 tube, No. 2, as a basis, various tapers were

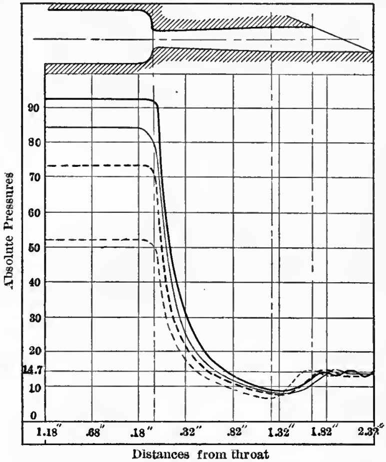


FIG. 7.—Pressure curves showing over expansion.

tried. The angle of divergence, as we have already been led to expect, had little or no effect on the inlet half of the tube. The tube was gradually shortened also from the outlet end with the

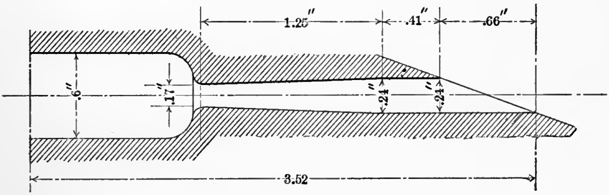


FIG. 8.—Section of a De Laval nozzle—non-condensing.

same result. The divergent curve of No. 5 was calculated to give uniform acceleration and to avoid the losses due to the changes of velocity, shown in the conical portion of Nos. 3 and 4, and which were characteristic of all the straight tapered forms. This form, No. 5, gave the best results. It is interesting to note that while Rosenhain and others advocate an only slightly rounded inlet (see also Fig. 8), Mr. Kneass obtained better results

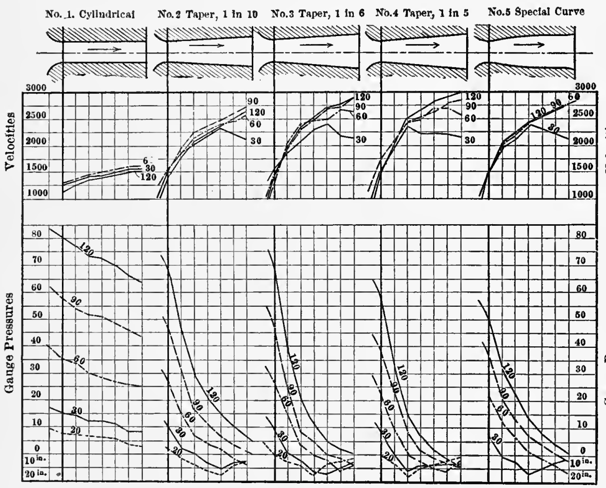


FIG. 9.—Kneass' experiments on nozzles.

with well rounded inlets and a taper of 1 to 6. The best efficiency obtained was from a nozzle proportioned for 30 pounds initial pressure and atmospheric exhaust. The results were as follows:

Gauge Pressure	Actual Velocity	Theoretical Velocity	Difference	Per Cent. of Loss
120	2690	2820	130	.046
90	2550	2650	100	.038
60	2290	2400	110	.045
30	1970	2000	30	.015
15	1400	1500	100	.066

The velocity loss, except at the lowest pressure, is within 0.5% and at the pressure for which the nozzle was designed, 30 pounds, there was but 1.5% loss. This test would indicate that an expanding nozzle should be designed for a pressure under, rather than over, the average to be encountered. The energy loss from the above, being proportional to the square of the velocity, varies from 3% to 13% with an average of about 10% as previously stated.

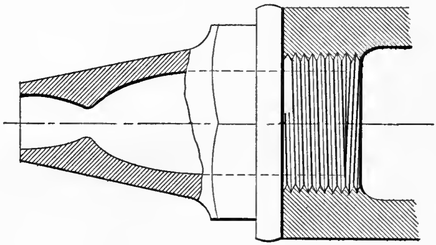


FIG. 10.—Nozzle on Kerr turbine.

A form of nozzle similar to No. 5 has been adopted in the Kerr turbine. In this turbine there is multi- stage expansion and consequently only a small pressure drop in any one set of nozzles. Fig. 10 shows a longitudinal section of this nozzle.

Problems.

1. Report on tests and conclusions of Sibley and Kemble in paper "Efficiency Tests of Steam-turbine" Nozzles. Trans. A. S. M. E., vol. xxxi, p. 617.
2. Report on the tests of Borsody and Cairncross. Reported in Transactions of A. S. M. E., vol. xxvi, p. 114.

References.

STODOLA: "The Steam Turbine."

THOMAS: "Steam Turbines."

RATEAU: "Flow of Steam through Nozzles."

MOYER: "Steam Turbines."

JUDE: "Theory of the Steam Turbine."

BORSODY: AND CAIRNCROSS: "Pressures and Temperatures in Free Expansion," Trans. A. S. M. E., vol. xxvi.

SIBLEY AND KEMBLE: "Efficiency Tests of Nozzles," A. S. M. E., vol. xxxi.

Art. 7.—Design of an Expanding Nozzle.

The following example in the design of an expanding nozzle, worked out first by the use of the steam tables and entropy diagram, will be used also to illustrate the use of the heat-entropy diagram.

Let the conditions be as follows:

Initial steam pressure,	=165 pounds absolute
Initial dryness of steam,	=98 1/2 %
Final steam pressure,	=1 pound absolute
Loss of energy during expansion, y	=15%
Weight of steam discharged per second	=.092 pound

Let it be required to find the diameter of a conical nozzle and the condition of the steam at the points where the pressures are 96, 75, 60, 45, 30, 15, and 1 pound absolute per square inch. The first of these pressures, 96 pounds, is .58 of the initial pressure, and is therefore the "critical pressure" in the throat. The first seven lines in the following table give the detailed operations in finding the adiabatic heat drop. The letters in the second column refer to Fig. 3. Lines 8 to 12 are the operations to determine the quality of the steam at the various pressures. With this known, the volume of steam to be delivered per second may be calculated. From this the areas and corresponding diameters required may easily be obtained.

Let it be given that the divergent portion of the nozzle is to be 4 inches long. From the results tabulated the nozzle may be drawn and the curves of pressures and velocities plotted as shown in Fig. 11. Values for 5 pounds and 2 1/2 pounds absolute are interpolated to assist in drawing the curves. The pressure and velocity for any point in the nozzle may be readily found from the curves. For instance for the section, *aa*, project downward cutting the two curves. The pressure curve will be cut at 15 pounds and the velocity curve at 2710 feet per second, as referred to their respective scales.

The operations from 1 to 12, requiring considerable time and care, may be arrived at in a few moments by the use of the heat-entropy diagram. Fig. 12 reproduces part of it for clearness. From the point A, giving the initial condition of the steam, 165

1	Pressures, p_2 (lbs. per sq. in. abs.)	96	75	60	45	30	15	1
2	H_1 (Total heat between initial conditions and water at pressures p_2 represented by area beneath ABE, Fig. 3.)	886.8	904.7	920	938.7	963.3	1001.1	1112
3	Change in entropy F to A, Fig. 3............	1.076	1.099	1.119	1.144	1.178	1.233	1.413
4	T_2 (Absolute temp. at pressures p_2)..........	785.5	767.2	753.3	735.1	709.9	672.6	561.4
5	H_2 (Item 3 × Item 4)......................	845.2	843.1	842.9	840.9	836.3	829.3	793.6
6	H_1-H_2......................(B. T. U.).	41.6	61.6	77.1	97.8	127.	172.	319.
7	$V=223.8\sqrt{(H_1-H_2)(1-y)}=206.6\sqrt{H_1-H_2}$	1330	1620.	1810.	2040.	2325.	2710.	3680.
8	Entropy of evap. at pressures p_2 (AG Fig. 3) .	1.1348	1.1778	1.216	1.2644	1.3311	1.4416	1.8427
9	L_2 (Heat of vaporization at pressures p_2)....	890.3	907.2	914.9	928.2	945.1	969.7	1034.6
10	Quality of steam for adiabatic exp. $\left(=\frac{\text{Item 3}}{\text{Item 8}}=\frac{AF}{AG}, \text{Fig. 3}\right)$	.949	.934	.921	.905	.884	.855	.767
11	Increase in Quality, $x=\frac{y(H_1-H_2)}{L_2}=\frac{.15\ (\text{Item } 6)}{(\text{Item } 9)}.$	.007	.0102	.0126	.0158	.0202	.0266	.0463
12	Actual quality at K = Item 10 + Item 11......	.956	.944	.933	.921	.904	.882	.813
13	Volume of 1 lb. dry steam at press. p_2.........	4.60	5.81	7.17	9.39	13.74	26.27	333
14	Volume of 1 lb. wet steam at press. p_2........ (Item 12 × Item 13.)	4.4	5.49	6.59	8.65	12.42	23.20	271
15	Volume to be delivered per sec. =........... (.092lb. × Item 14).	.405	.505	.61	.795	1.14	2.14	25.
16	Area required $=\frac{\text{Item } 15\times 144}{\text{Item } 7}$................	.044	.045	.048	.056	.0705	.114	.96
17	Diameter required........................	.236	.24	.248	.267	.300	.38	.99

pounds pressure and .985 dryness, drop a perpendicular to B on the line of the given final pressure, 1 pound. The distance AB, laid off on the heat scale gives 319 B. T. U., the heat drop for adiabatic expansion, corresponding with the calculated value in line 6 of the table. But 15% of the heat drop is lost in friction. Laying off $Bb' = .15$ of AB, we have Ab' as the actual heat drop. Projecting b' horizontally until it crosses the 1 pound pressure

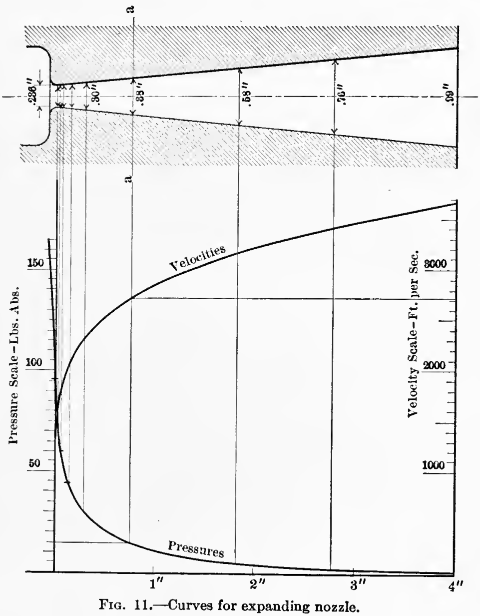


FIG. 11.—Curves for expanding nozzle.

line as at b'', we have, reading from the quality lines, .813 as the final dryness, which agrees with the calculated value line 12.

To ascertain the condition at any intermediate pressures, with B as a center and Bb' as a radius, draw the arc $b'c$. Draw a tangent to the arc $b'c$ through A. From the points d, e, etc., where AB cuts the pressure lines in question, draw arcs tangent to Ac. Project the points d', e', where the arcs cut AB, horizontally over to the points d'', e'', on their respective pressure lines. These points referred to the quality lines give the dryness. A "quality curve," Ab'', may be drawn through d'', e'', etc., from which the dryness for any intermediate pressure may be read off at once. The qualities thus found will agree with the values as calculated within limits of error justified by the nature of the problem. It may be noted here that this quality curve may be laid off almost as readily for a varying friction

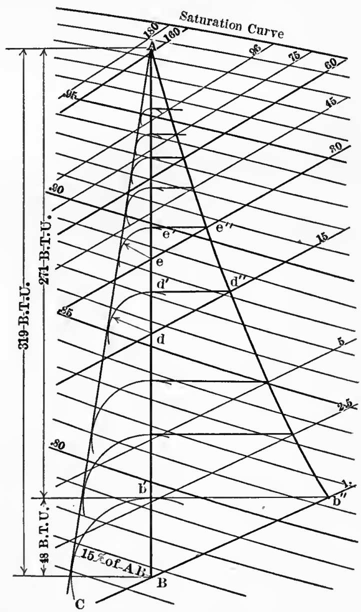


FIG. 12.—Condition curve for expanding nozzle.

loss as for a constant one. This is of value in calculations for multistage machines where the friction losses increase as the steam expands.

Problems.

1. Design an expanding nozzle for the following conditions: Initial steam pressure = 120 lbs. gauge; superheat = 40° Fahr.; atmospheric discharge pressure; energy loss, 16%; weight of steam delivered = .11 lb. per sec.; length of nozzle = 3 ins.

2. Design a nozzle for same conditions as Prob. 1, except that discharge pressure is 1.5 lbs. absolute and length = 4 ins.
3. Calculate the H. P. available in jet at throat of nozzle, Prob. 1, and also after expansion to atmospheric discharge pressure.
4. Calculate the H. P. available at throat and exit of nozzle in Prob. 2.
5. If the expanding portion of the nozzle in Prob. 2 is 4 inches long, what are the pressure and velocity at points 1/2 inch and 2 1/2 inches from the throat?

References.

STODOLA: "The Steam Turbine."
THOMAS: "Steam Turbines."
MOYER: "Steam Turbines."

Art. 8.—BLADE FORMS AND BLADE VELOCITIES.

The velocities which enter into the problem of turbine design are:

The absolute velocity of the jet as it enters the blades.

The velocity of the moving blades.

The relative velocity of the jet and the blades.

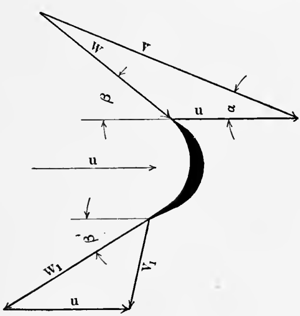


FIG. 13.—Velocity diagram.

The absolute velocity of the jet as it leaves the blades.

The ideal condition in all turbines, whether steam or water driven, is that the driving fluid "enter without shock and leave without velocity."

Let V, above, be the absolute velocity of the entering steam, and u the blade velocity. W, the relative velocity, is found, both in magnitude and direction, from the velocity triangle V, u, and W. The angle β determines the direction of the *back* of the blade on the entering side. The curved surface of the blade deflects the jet, discharging it backward with a velocity W_1, equal to W, less a certain loss due to friction. This friction can not be determined theoretically, but may be assumed from results of experiments and experience in turbine design. These indicate that W_1 varies from .8 to .95 of W, according to conditions, .92 being a fair average value for the single stage turbine. The angle

β' is ordinarily made equal to β for convenience in manufacture and to reduce end thrust. Having W_1 in magnitude and direction, the final absolute velocity V_1 is readily obtained by combining it with the bucket velocity u. From equation (2) the energy per pound of steam developed in buckets is $\frac{V^2 - V_1^2}{2g}$, and the theoretical efficiency is

$$E = \frac{V^2 - V_1^2}{V^2}. \tag{16}$$

Fig. 14 shows a convenient method of determining the form of the bucket, where entrance and exit angles are equal.

Lay off u, the bucket velocity, horizontally. Draw V to the same scale, making the angle α. In the De Laval wheel this angle is 20°. Closing the triangle we have the relative velocity

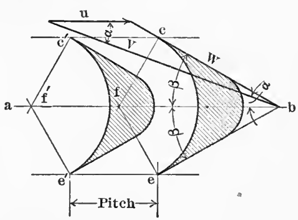


FIG. 14.—Determination of blade sections.

W in direction and magnitude. Draw ab parallel to u and lay off $\beta' = \beta$ on the exit side. Take c and e equidistant from ab and the width of the blade apart. Lay off cc' and ee' equal to the pitch of the blades. The face of the bucket may be drawn as the arc of a circle about the intersection, f, of the perpendiculars cf and ef and the center line ab.[1] The back of the blade is formed by two lines parallel to be and bc connected by an arc with f as the center.

The best width ce and pitch cc' are empirical and determined by experience, as no formulæ for them have been developed. Moyer says, "for turbines of less than 100 H. P., the width of the

[1] A well known German rule is to make the radius equal to twice the pitch times sine β, or $r = 2p$ sine β.

blades is often made about 1″, increasing to about 1.5″ in turbines of 1000 H. P." "The most efficient blade pitch appears to be between the limits of 1/2″ and 1″. Between these two values the efficiency of blades made according to conventional designs is practically constant. The usual blade pitches are 5/8, 3/4, and 7/8 inch."

It is more important that the backs of the vanes should be parallel to the entering steam than the faces, for steam striking the backs is deflected outward, retarding the wheel and causing loss. If it strikes the face the energy of deflection goes into useful work in the wheel. In the spacing of the blades there must be enough free area to provide for the unobstructed passage of the steam at the large volume corresponding to the exhaust side of the blade, and it should be as near uniform in section as conditions will permit. It must be borne in mind, too, that the opening of a round inclined nozzle is an ellipse, and the greater part of the steam is discharged through these openings near the middle of the nozzle.

Problems.

1. If steam issues from a nozzle with a velocity of 3400 ft. per sec. at an angle $\alpha = 20°$ and the blade velocity $u = 1200$ ft. per sec., what are the relative velocity W and blade angle β?
2. Assume blade angles β and β_1 as equal, what is the final absolute exit velocity neglecting steam friction in the blading?
3. What is the energy per lb. of steam available for the conditions of Probs. 1 and 2? What is the efficiency?
4. Given $V = 3400$ ft. per sec. $\alpha = 20°$, $u = 1200$ ft. per sec., and an energy loss of 20% in passage through the blading, what is the final velocity of exit V_1?
5. Determine the energy absorbed by the wheel per lb. of steam and the efficiency for conditions of Prob. 4.
6. Draw a cross section of the blading for conditions of Prob. 4. What length of blade could be used with a nozzle .8 inch exit diameter?
7. How many r. p. m. should a 42-inch diam. wheel run for a nozzle velocity of 2300 ft. and $\alpha = 22\ 1/2°$ and $\beta = 27°$.
8. With a blade velocity u = 400 ft. per sec., a nozzle velocity of 1720 ft. per sec., and $\alpha = 42°$, what will be the absolute angle of exit of the steam for a symmetrical blade?
9. Report on article on "Principles of Steam Turbine Buckets." Power, March 17, 1908, p. 391

References.

STODOLA: "The Steam Turbine."
THOMAS: "Steam Turbines."
MOYER: "Steam Turbines."
JUDE: "Theory of the Steam Turbine."
NEILSON: "The Steam Turbine."

Art. 9.—Conditions of Maximum Efficiency.

Let Fig. 15 be the velocity diagram for the entrance and exit velocities, where the relative velocity W is supposed to remain constant.

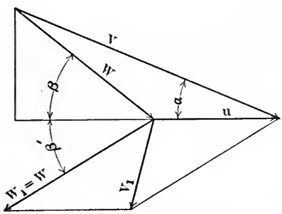


FIG. 15.—Velocity diagram.

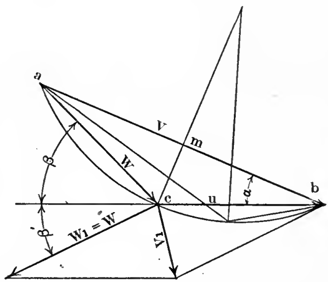


FIG. 16.—Velocity diagram.

From trigonometry

$$V^2 = W^2 + u^2 + 2uW \cos \beta$$

$$V_1{}^2 = W^2 + u^2 - 2u\, W \cos \beta_1 \qquad \text{subtracting,}$$

$$V^2 - V_1{}^2 = +2u\, W\,(\cos \beta + \cos \beta_1)$$

$$V_1{}^2 = V^2 - 2u\, W\,(\cos \beta + \cos \beta_1). \qquad (17)$$

The efficiency of the bucket, $\dfrac{V^2 - V_1{}^2}{V^2}$, Eq. (16), will be highest when $V_1{}^2$ is least. $V_1{}^2$ will be least, V^2 being constant, for the maximum value of the second term in Eq. (17). As in hydraulic problems the ideal condition is given by complete reversal where $\cos \beta$ and $\cos \beta_1$ are each positive and unity.

FIRST CASE.—*Where β, β_1 and V are fixed.*

Since the above quantities are fixed it is clear from Eq. (17) that V_1 is least where uW is greatest. In Fig. 16 the area of the triangle $abc = bc.\tfrac{1}{2}ac \sin \beta = \tfrac{1}{2}ab.cm$, therefore $\dfrac{ab.cm}{\sin \beta} =$

$ac.bc = W.u$. But $ab = V$, and $sin\ \beta$ are constant, therefore, comparing with Eq. (17), V_1^2 is least when cm is maximum. The length ab and the angle acb are fixed, therefore the locus of the possible positions of the intersection c with reference to ab is an arc acb and cm is maximum when $ac = cb$, or when $u = W$. We have then

$$u = \frac{V}{2\ cos \frac{\beta}{2}} \qquad (18)$$

and

$$\alpha = \tfrac{1}{2}\beta. \qquad (19)$$

Taking the relative velocities W and W_1 as equal, the efficiency, $E = \frac{V^2 - V_1^2}{V^2}$ becomes $\frac{2u.\ W(cos\ \beta + cos\ \beta_1)}{V^2}$.

If, in addition, β and β_1 are also equal

$$E = \frac{4u\ W\ cos\ \beta}{V^2}.$$

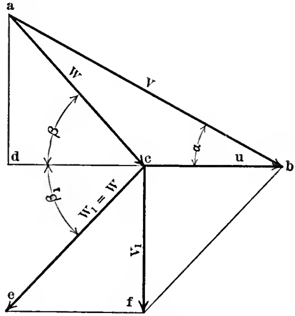


FIG. 17.—Velocity diagram.

Since $W\ cos\ \beta = V\ cos\ \alpha - u$, the efficiency in terms of α is

$$E = \frac{4u}{V^2}(V\ cos\ \alpha - u) = \frac{4u}{V}(cos\ \alpha - \frac{u}{V}). \qquad (20)$$

SECOND CASE.—*Where V and α are fixed, and β and β_1 are equal but not determined.*

Eq. (17) above becomes $V_1^2 = V^2 - 4uWcos\beta$, $V_1^2 = V^2 - 4u.cd$.

V_1 is least when $u.cd$ is maximum, bd, the sum of u and cd is constant and $= V \cos \alpha$. Therefore, $u.cd$ is maximum for $u = cd$, or

$$u = \frac{V \cos \alpha}{2} \tag{21}$$

and

$$\tan \beta = \tan \beta_1 = \frac{ad}{dc} = \frac{ad}{\frac{bd}{2}} = 2\frac{ad}{bd} = 2 \tan \alpha.$$

Also, since $W = W_1$, $ec = ac$ and the triangle cef can easily be shown to be similar and equal to the triangle acd. Therefore $cf = ad$ or $V_1 = V \sin \alpha$.

The efficiency $\frac{V^2 - V_1^2}{V^2}$ becomes then $= \frac{V^2 - V^2 \sin^2\alpha}{V^2} =$

$$\frac{V^2\,(1 - \sin^2 \alpha)}{V^2} \text{ or simply } = \cos^2\alpha \tag{22}$$

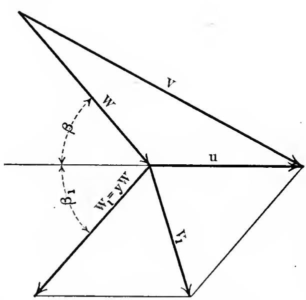


FIG. 18.—Velocity diagram.

THIRD CASE.—*When V, α, and β are fixed.* The triangle abc is fixed. W and u are therefore determinate. $Cos\ \beta_1$, then, is the only variable on the right hand side of Eq. (17), and V_1 is least for $\beta_1 = O$.

In the foregoing, losses in the relative velocity W, due to friction, have been neglected. If these be considered, $V^2 = W^2 + u^2 + 2uW \cos \beta$ as before, but V_1^2 becomes $V_1^2 = W_1^2 + u^2 - 2uW_1 \cos \beta_1$, where $W_1 = yW$.

The energy developed is

$$\frac{V^2 - V_1^2}{2g} = \frac{W^2 - W_1^2 + 2u\ (W\ cos\ \beta + W_1\ cos\ \beta_1)}{2g}. \tag{23}$$

The general expression for the efficiency

$$= \frac{V^2 - V_1^2}{V^2} = \frac{W^2 - W_1^2 + 2u\ (W\ cos\ \beta + W_1\ cos\ \beta_1)}{V^2}.$$

For Case II where $\beta = \beta_1$ it becomes

$$= \frac{W^2 - W_1^2 + 2u.\ cos\ \beta\ (W + W_1)}{V^2}.$$

It frequently happens that the best value of u must be sacrificed for safety or for cheapness of construction, or there may be offsetting gains in mechanical efficiency by adopting a

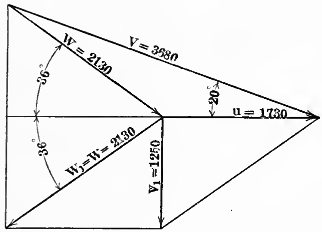


FIG. 19.—Velocity diagram.

slower speed. Take as an example, a De Laval nozzle, coming under Case II, delivering steam at 3680 feet per second at an angle, $\alpha = 20°$. Then $u = \frac{V\ cos\ \alpha}{2} = \frac{3680}{2}.94 = 1730$ feet per second. This would give a value for the relative velocity

$$W = \sqrt{3680^2 + 1730^2 - 2\ cos\ 20°\ 3680\ .1730} = 2130 \text{ feet},$$

and a blade angle, $\beta = tan^{-1}\ 2\ tan\ 20° = \tan^{-1} .728 = 36°$. Experience has shown that the blade velocity should not exceed 1400 feet per second. If the actual speed be 1250 feet per second the value for the relative velocity is

$$W=\sqrt{3680^2+1250^2-2\ cos\ 20°\ .3680\ .1250}=2540$$

and since $sin\ \beta=\frac{V\ sin\ \alpha}{W}$, the blade angle=

$$sin^{-1}\frac{3680\ sin\ 20°}{2540}=30°.$$

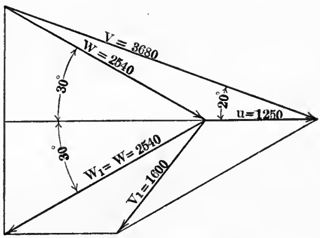


Fig. 20.—Velocity diagram.

In the De Laval turbine a nozzle angle of 20° has been established for all sizes. The angles β and β_1 are equal, varying from 30° or 32° up to 36°.

β and β_1 can not be equal to zero, unless the plane of the reversal of the steam makes an angle with the line of motion. This is done in some turbines, as the Stumpf and Terry machines.

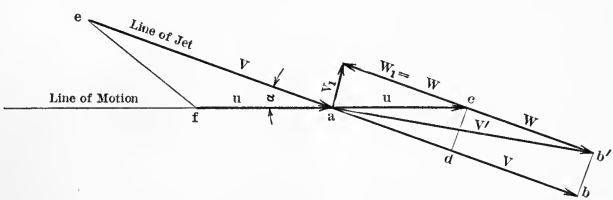


Fig. 21.—Velocity diagram—tangential type.

The introduction of this new angle partially offsets the advantage of complete reversal, as will be seen in considering Fig. 21. If V and u be represented by ab and ac the velocity, u, can be resolved into two components, $u\ cos\ \alpha=\ ad$, and $u\ sin\ \alpha=cd$. The component cd will be impressed on the entering jet giving it the direction ab' so that W is the relative velocity of entrance, not *cb* or *ef*. The complete reversal of the jet gives a relative exit velocity of W, equal to $-W$, and an absolute exit velocity of V_1, as shown.

Neilson shows that V_1 is minimum for $u=\frac{2.V.\cos\alpha}{3\cos^2\alpha+1}$. For this value $V_1^2=\frac{V^2(1-\cos^2\alpha.)}{3\cos^2\alpha+1}$.

The third of the principal forms of buckets used in single-expansion turbines is the Pelton type, which lends itself to use

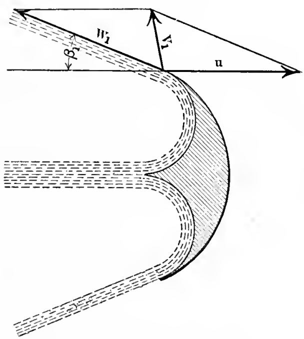


FIG. 22.—Pelton type of blade.

with steam as well as water. A complete reversal of the steam jet and a value of $u=\frac{V}{2}$ would give a theoretical efficiency of 100%. It is necessary, of course, to give the steam some exit velocity to clear the wheel. This velocity, V_1, can be determined graphically as in Fig. 22, or analytically by the equation

$$
\begin{aligned}
V_1^2 &= W^2+u^2-2\,W\,u\cos\beta_1. \quad \text{Since } W=V-u \\
&=(V-u)^2+u^2-2u\,(V-u)\cos\beta_1, \\
&=V_2-2u\,(V-u)\,(1+\cos\beta_1).
\end{aligned}
$$

The theoretical efficiency as before $=\frac{V^2-V_1^2}{V^2}$

$$
=\frac{2u\,W\,(1+\cos\beta)}{V^2}. \qquad (25)
$$

Problems.

1. Given $V=2420$ ft. per sec., $\beta.=45°$ and $\beta_1=40°$, what is the best value for the blade velocity and for the nozzle angle? Neglect steam friction.
2. What is the blade efficiency for the conditions of Prob. 1.
3. Given a steam velocity of 2250 ft. per sec. and a nozzle angle of 22°, what is the best blade velocity, neglecting steam friction? What is the efficiency of the blading?
4. With symmetrical blading what nozzle angle should be used when the steam velocity is 2700 ft. per sec. and the blade velocity 1430 ft. to get an efficiency of 87%?
5. What angles α and β should be used for a DeLaval nozzle and blade to give 88 1/2% efficiency with an entrance velocity of 3550 ft. per sec.? What blade velocity does this call for?
6. What is the power developed per lb. of steam for the conditions of Prob. 5?
7. Find the power developed in a DeLaval blade with $V=3550$ as in Prob. 5., u limited to 1240 ft. per sec., $\alpha=20°$.
8. Find the power developed under the conditions of Prob. 7 if in addition there is a velocity loss of 10% in passage through the blades; what is the blade efficiency under these conditions?
9. Show that for maximum efficiency, V and u are at right angles.
10. Prove that "dilution of the jet" or lowering the velocity of the steam by having mixed with a heavier fluid, as water, is a less efficient way of reducing speeds than by lowering the velocity.[1]

References.

Stodola: "The Steam Turbine."
Neilson: "The Steam Turbine."
Thomas: "Steam Turbines."
Jude: "Theory of the Steam Turbine."
Moyer: "Steam Turbines."
Rateau: "Method of Calculating Steam Turbines," London Engineering, Dec. 10, 1909, p. 804.
Briling: "Energy Losses in Turbine Buckets," Zeitschrift d. v. Deut. Ing., Feb. 12, 1910.

Art. 10—The Trajectory Of The Steam.

The absolute direction of the steam at any point may be determined by combining the velocity u of the vane with that of the steam as it moves relatively to the vane. Neglecting frictional losses, this would have a magnitude W, and a direction tangent

[1]See French, Steam Turbines, p. 63.

to the vane surface at the point in question. By taking a succession of points the absolute directions chould be found and the envelope of these would give an approximate trajectory. The error in this method is cumulative, however, and a better way is as follows: Let W be the relative velocity at entrance, and u the vane velocity as heretofore. If the inner vane surface be semicircular the time required to move from A to B will be $\frac{\pi r}{W}$. The distance forward covered in the time will be $u\frac{\pi r}{W}$. Laying this off along the path of B we have the end of the trajectory. For any intermediate point, as C, the time required will be $\frac{arc\ AC}{W}$ and the distance forward covered by C will be $CC_1 = \frac{arc\ AC}{W}u$. The path will depend on the size and shape of the vane and the relative velocities u and W. If, as is usual, the vane surface is only a portion of a semicircle, the method is not altered and the distance CC_1 for any point C is laid off along its path equal to $\frac{distance\ AC}{W}$ u, and so on for any number of points needed to determine the curve.

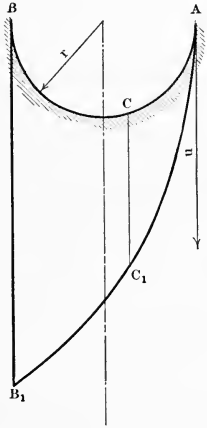


FIG. 23.—Trajectory of the steam.

If, however, the relative velocity W be reduced by friction to W_1 during the passage through the vane it will lengthen out the trajectory. Just at what rate the retardation occurs would be difficult to determine, but it may be assumed to be uniform. If W and W_1 be the relative velocities at entrance and exit the loss from A to $B = W - W_1$, and assuming this loss as uniform the velocity at $C = W - \frac{AC}{AB}(W - W_1)$ and the time required to move from A to $C = \frac{AC}{ave.\,rel.\,vel.} = \frac{AC}{W - \frac{AC}{2AB}(W - W_1)}$.

The point corresponding to C on the trajectory would then lie on its path a distance forward

$$= \frac{AC}{W - \frac{AC}{2AB}(W - W_1)} u.$$

A succession of points, as before, will determine the complete trajectory.

Below are given the trajectories for a blade of the De Laval form, for values of $\frac{u}{W} = .35, .4, .45, .5, .55$, and $.6$, and a frictional loss of $.08$.

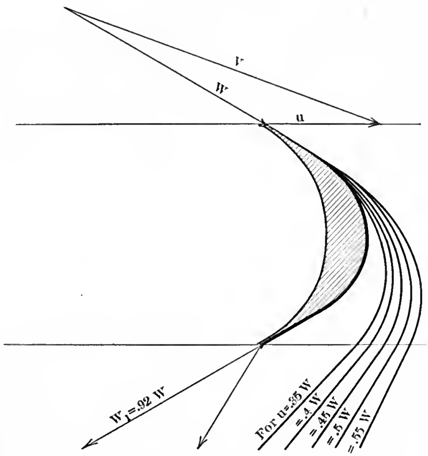


FIG. 24.—Trajectory curves.

The cross section of the passage between the vanes should be as nearly uniform as possible. Where the steam fills the passage and the backs of the blades differ in form from the faces, there may be a marked difference in the trajectories of steam particles, which means eddies and inevitable loss of energy.

Fig. 25 shows a difference in the paths of particles moving along the fronts and backs of the blades, which is to be guarded against as far as possible in designing turbine blades. In impulse turbines the expansion occurs in the nozzles or fixed vanes and there is no drop in pressure as the steam goes through the buckets. This renders the determination of the trajectory easy, but in reaction turbines the moving passages are themselves nozzles, and the steam velocity increases sharply in passing through them. The determination of the trajectory under these conditions becomes very involved. Approximations have been

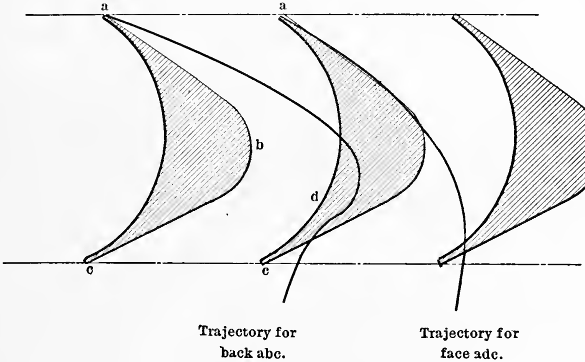


FIG. 25.—Trajectory at front and back of blade.

made by assuming that the relative velocity increases in some regular manner. Having made some such assumptions trajectories may be plotted by the method outlined, but as they are at best guess work and of necessity differ throughout the entire length of the rotor they are of little value. The methods outlined above of course apply to the Pelton type of bucket as well as to the side entrance type.

Problems.

1. With a steam velocity of 2200 and a blade velocity of 800 ft. per sec. and semicircular blade as in Fig. 23, how far will the blade travel during the passage of the steam, assuming no friction loss?
2. Given nozzle and blade velocities of 3300 ft. and 1200 ft. per sec., a nozzle angle of 20°, a blade pitch of 5/8 in. and a width of 1 in., how far has the blade moved when the steam crosses the center line, and how far at exit?

3. Assuming the W and blade face of Prob. 2, plot the trajectory for u = .3, .4, and .5 of W.
4. Given $V = 3400$ ft. and $u = 1220$ ft. per sec., $\alpha = 20°$ and a pitch of 3/4 in., draw a section of the blading and plot the trajectories of the steam moving along the face and back of the blade (see Fig. 25).

References.

JUDE: "Theory of the Steam Turbine."
STODOLA: "The Steam Turbine."

CHAPTER III.

CALCULATIONS OF TURBINE BLADING.

Art. 11.—Single-stage Impulse Turbines.

The simplest form of successful turbine is the single-stage impulse turbine. In this all the expansion occurs in one set of nozzles, where the heat energy available is converted into kinetic energy and directed against a single row of moving vanes.

As the steam velocity varies with the heat drop and the vane velocity with that of the incoming steam, the speed of the rotating vanes in the case of single expansion is very high, from 1,000 to 1,400 feet per second, or about half the velocity of a modern high powered rifle bullet. De Laval frankly accepted these speeds and developed a machine capable of withstanding the strains produced. Some idea of the problems involved may be had from the fact that in the 300-horse-power turbines the blades, weighing only 1/28 of a pound each, but running with a velocity of 1380 feet per second on a 15-inch radius, develop a centrifugal force of 3/4 of a ton. It called for engineering ability of a high order, and greatest care in the details of design and manufacture, for not only was a built up rotor required capable of running from 10,000 to 30,000 R.P.M., according to the size, but the velocities developed had to be reduced through gearing to ones which could be used. An efficient and practical machine has been developed, however, which runs with gear velocities of 100 feet per second with but little wear, no vibration, and high efficiency.

The relation between the pressures and velocities during the passage of the steam through an impulse turbine of the De Laval type is shown schematically in Fig. 26. The boiler pressure drops in a single expansion to the exhaust pressure before the steam enters the vanes. The wheel therefore has equal pressure on both sides. The velocity V of the steam is maximum as it enters the vanes and is nearly all absorbed in the passage through them. The steam enters the exhaust chamber with the low velocity v, sufficient only to clear the wheel.

As an example of calculation for the blading of a turbine of this type let the steam conditions be the same as for the example on page 22, namely,

Initial steam pressure,	=	150 pounds gauge
Initial dryness	=	98 1/2 %
Final steam pressure	=	1 pound absolute

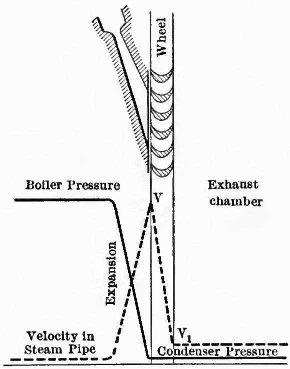


FIG. 26.—Scheme of single stage impulse turbine.

In addition let the

Output	=	110 H. P.
Revolutions per minute of wheel	=	13000
Blade velocity be limited to	=	1250 feet per second
Number of nozzles	=	6
Angle of nozzles	=	20°

Angle of entrance and exit of blades to be equal.

As already pointed out in Art. 8 the design of the blading, and in fact of the whole turbine, is materially affected by the frictional losses in the nozzles and blades. In determining the proportions for a given case assumptions must be made for these, based on experience. Stodola gives the following values for the various losses.[1]

[1] Steam Turbines, 2nd ed., p. 224.

Losses in the nozzle	$=15\%$ of the available energy
Losses in the blades	$=21\%$ of the available energy
Losses at exit	$=\ \ 4.6\%$ of the available energy
Total loss	$=40.6\%$ of the available energy

From the given conditions of steam and exhaust the theoretical heat drop available is 319 B. T. U. But assuming a nozzle loss of 15%, only $319\times .85=271$ B. T. U. will be transformed into kinetic energy. From Eq. (6) we have $V=223.8\sqrt{271}=$ 3680 feet per second. (See problem in Art. 7.)

From Eq. (21) $u=\dfrac{3680\times .94}{2}=1730$ feet per second for maximum efficiency. For mechanical reasons u will be limited to 1250 feet per second as given in the initial conditions. The

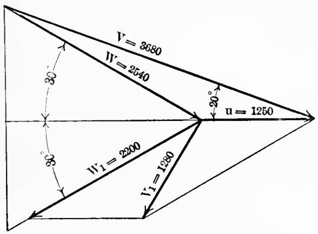


Fig. 27.—Velocity diagram.

energy loss in the blades is 21% of the total, or $\dfrac{.21}{.85}=24.7\%$ of that delivered to the blades by the nozzles. If 2540 feet per second be the relative velocity at entrance, as shown by the velocity diagram, Fig. 27, the relative velocity at exit will be such that the kinetic energy is 24.7 % less, or

$$\frac{2540^2}{2g}\times .753=\frac{W_1^{\,2}}{2g}.\quad \text{Hence}$$

$$W_1=\sqrt{2540^2\times .753}=2200 \text{ feet per second.}$$

The diagram shows that the blade angles are 30° and the final velocity $V_1=1280$ feet per second. Comparison of Fig. 27 with

Figs. 19 and 20 shows how friction and mechanical limitations modify the theoretical conditions.

Comparing reported tests of similar turbines, 18 pounds per B. H. P. hour may be assumed for the steam rate on which to base the nozzle areas. Then $\frac{18\times 110}{3600}=.55$ pounds of steam must be delivered per second by the six nozzles or .092 pounds for each nozzle. The proportions of a nozzle for these conditions have already been worked out in Art. 7 and are shown in Fig. 11. 13000 R. P. M. have been assumed for the rotor shaft, and 1250 feet per second for u, therefore $\frac{1250\times 60}{13000}=5.75$ feet is the circumference of the blade circle, which corresponds to a pitch diameter of 22 inches. From Art. 8 the pitch may be assumed as approximately 5/8 inch, which gives us 110 blades. Since the exit diameter of the nozzle, Fig. 11, is .99 inches we may make the blades, say, 1 1/8 inches high.

Problems.

1. Given the steam conditions of Prob. 1., Art. 7, namely:

Steam pressure	=120 lbs. gauge.
Superheat	= 40° F.
Exhaust at atmospheric pressure	
Output	=100 K. W.
R. P. M. of wheel	=12000.
Blade velocity	=1200 ft. per sec.
Number of nozzles	=8.
Angle of nozzles	=20°.
Blades symmetrical.	

   Design nozzles and blading for a single stage impulse turbine assuming a steam rate of 50 lbs. of steam per K. W. hr.
2. Design nozzles and blading for a condensing turbine, conditions as in Prob. 1, except exhaust pressure = 1.5 lbs. absolute. What changes of design are involved ? Assume a steam rate of 26 lbs. per K. W. hr.
3. Design nozzles and blading for a single stage impulse turbine. Steam pressure 15 lbs. absolute; exhaust pressure 1 lb. absolute; quality of steam at admission = 88%; blade velocity = 1200 ft. per sec. Take steam rate at 42 lbs. per K. W. hr.
4. Report on the construction and efficiency of the reduction gearing of the DeLaval steam turbine.

References.

STODOLA: "The Steam Turbine."
NEILSON: "The Steam Turbine."
THOMAS: "Steam Turbines."
LEA AND MEDEN: Paper, "DeLaval Steam Turbines," A. S. M. E., vol. xxv.

Art. 12.—Multi-stage Impulse Turbines.

As seen in the last article a single-stage impulse turbine must run at a very high velocity and be geared down.

There are two ways by which the velocity of an impulse turbine may be reduced, by subdividing the heat drop in a series of

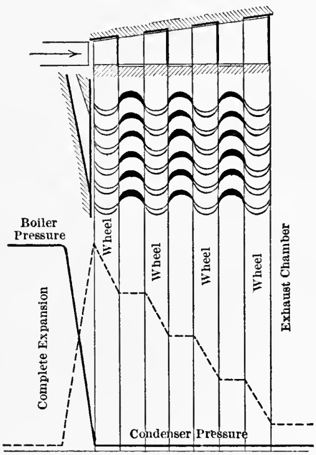


FIG. 28.—Scheme of impulse turbine with single pressure stage and several velocity stages.

expansions, and by having the steam impinge on two or more rows of buckets after expansion. The shaft velocity can thus be reduced to a point where the turbine can be directly connected to a generator, blower, or centrifugal pump. Such turbines are more complex and theoretically less efficient than the single-stage machine. This is offset, however, by the advantages of direct connection and lower speed. The De Laval type of single-stage turbine finds its limit at about 500 H. P. Beyond this the strains become too great and the problem of speed reduction too troublesome. In the multi-stage type, however, there seems to be almost no limitation of the power which can be developed by a single machine. Curtis turbines of this type are in use developing 20,000 K. W., and larger units might be used were they commercially desirable.

The field of the multi-stage turbine is much wider than that of the single-stage, and it constitutes the second of the main divisions into which turbines may be separated.

Fig. 28 illustrates the action of the original Curtis and the Riedler turbines. Complete expansion occurs as before in a

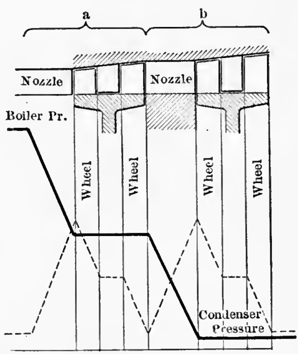


FIG. 29.—Scheme of Curtis turbine, multi-pressure, multi-velocity stage.

single set of nozzles, and the steam enters the first row of blades at high velocity as in the previous case. The steam leaves this row, however, with considerable velocity, is redirected by a set of stationary guide vanes and impinges on the next row, and so on to the exhaust chamber. The drops in steam velocity occur in the passage through the moving blades. The stationary vanes are merely guides, and the velocity during the passage through them is substantially constant, as shown. The increase of length in the successive rows of stationary blades is to allow for the decrease in velocity, not for increase in volume. Theoretically the exhaust pressure with its corresponding volume is reached at the nozzle, and all the rows of blades are driven by kinetic energy, revolve in a vacuum with the least possible resistance, and the wheels are balanced for pressures. This is a single-pressure, multi-velocity stage type and is still used on the smaller Curtis turbines up to about 35 K.W.

As now built the Curtis turbines have the pressure drop divided into two or more "pressure stages." Steam is partially expanded in a set of nozzles to some lower pressure and passes through a number of rows of moving blades as at *a*, Fig. 29, constituting one stage where its kinetic energy is absorbed. It is then expanded again and goes through another series of blades, or second stage, *b*. The number of stages and of rows in each stage varies with the

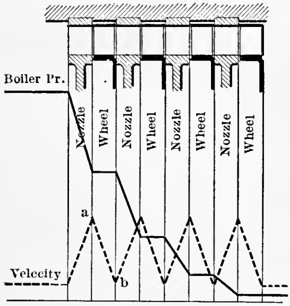


FIG. 30.—Scheme of multicellular turbine, multi-pressure, single-velocity stage.

conditions. As before, the wheels are balanced, and there is no end thrust. The pressure throughout each stage is constant, the drops occurring in the expanding nozzles at the beginning of each stage. This may be described as a multi-pressure, multi-velocity stage machine.

This idea of breaking up the pressure drop is carried to the limit in the "multi-cellular" turbines, Fig. 30, such as the Rateau, Zoelly, Kerr, and Wilkinson machines. Here a set of expansion nozzles after each row of blades replaces the fixed guide vanes, and accelerates the steam to a certain velocity, which is absorbed in the following running wheel. The multi-cellular turbine is therefore a multi-pressure, single-velocity tage machine. A turbine has been patented[1] which is similar to Fig. 30 except that, while the velocity drop is about as usual,

[1] U. S. Patent, J. F. M. Patitz, Oct. 4, 1910.

both velocities V and V_1 are high *i.e.*, the broken line in Fig. 30 is raised as a whole. The energy available in a single stage varies with $V^2 - V_1^2$, and is much greater for a given drop when both velocities are high. It is hoped by this means to reduce greatly the number of stages, but this will be partially offset by greater steam friction. See Probs. 3 and 4, p. 3.

In all of these forms there will be noticed the characteristic which defines impulse turbines, that there is no fall of pressure in the moving blades. Expansion occurs only in fixed passages or nozzles, and the moving blades are free from end thrust.

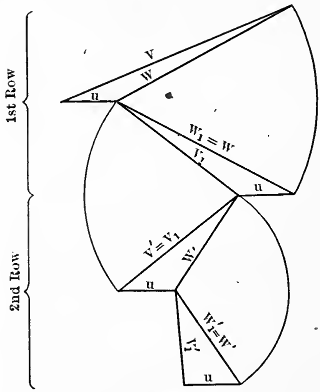


FIG. 31.—Velocity diagram for two stages.

The determination of the velocities in the multi-velocity stage turbine is an extension of the method already outlined. The final velocity, V_1, at exit from the first row, Fig. 31, is deflected and becomes the initial velocity for the second row of blades, and so on for the whole number of rows. When the rows of buckets have the same mean diameter, as is usual, u has the same value for each row. The diagram may be arranged conveniently and more compactly by extending the line of the vane velocity as in Fig. 32, and laying off a succession of lengths, each equal to u, and connecting each with the point a, as shown. With an initial velocity of 2430 feet per second, a blade velocity of 400 feet per second, and a nozzle angle of 22 1/2°, the exit velocity from the second row of moving blades is 1130 feet per second, and the blade angle 42°. Proper allowance for friction, however, materially affects the design and should, as before, be taken into account. Fig. 33 shows how this may be done, and gives the diagram where friction in the entering nozzle has reduced the velocity, V, to 2320 feet per second, and in each succeeding row a friction loss of 10% is assumed. Comparison of the blade section as determined by the two diagrams shows a marked difference. If the radii of the successive rows on the shaft are not

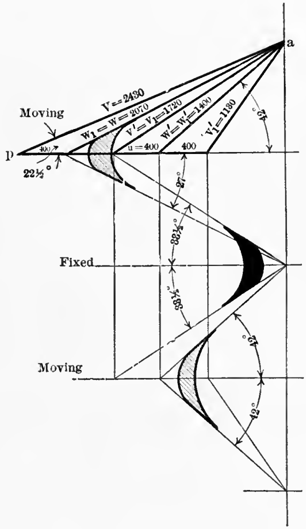


Fig. 32.—Velocity diagram for Curtis turbine, no allowance for steam friction.

equal, the length of u should be made to increase directly as the diameter. In general, the friction loss is not constant through the successive stages, but increases from 3% or 4% to perhaps 10% with increasing presence of water in the steam. Proper values can be assigned only from experience. The method of finding the velocities in Fig. 33 is applicable for an increasing friction loss as well as for a constant one.

As an illustration of calculations for blading in a multi-stage impulse turbine, let the following conditions be given:

Initial steam pressure	=175 pounds absolute
Initial superheat	=140° Fahr.
Final pressure	= 1 pound absolute
Assume a blade velocity u	=400 feet per second
Assume nozzle angles	= 22 1/2°

Assume 10% energy loss in the nozzles and 10% velocity loss in each row of vanes. Let the turbine be of the Curtis type with three stages and two rows of moving blades in each stage.

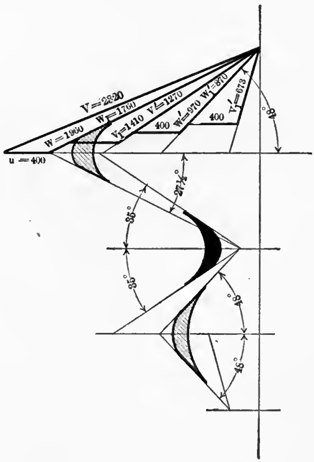


Fig. 33.—Velocity diagram for Curtis turbine with allowance for steam friction.

For convenience part of the heat chart is reproduced in Fig. 34. The initial condition of the steam is given by the intersection of the 175-pound pressure line and the 140° superheat line. A vertical line dropped to the 1 pound line gives 357 B. T. U. as the total heat drop for pure adiabatic expansion. Dividing this drop as nearly equally as possible at AC, CD, and DB, we have the pressure ranges for the first three stages.

First, 175 pounds to 45 pounds absolute.
Second, 45 pounds to 8 pounds absolute.
Third, 8 pounds to 1 pound absolute.

Deducting $CC' = 10\%$ of AC for the assumed heat loss in the expanding nozzle, we have AC', which, referred to the velocity scale, gives 2320 feet per second initial velocity. As u and α are given, we may draw the velocity diagram, allowing for the 10% velocity loss in each row. This is given in Fig. 33, where

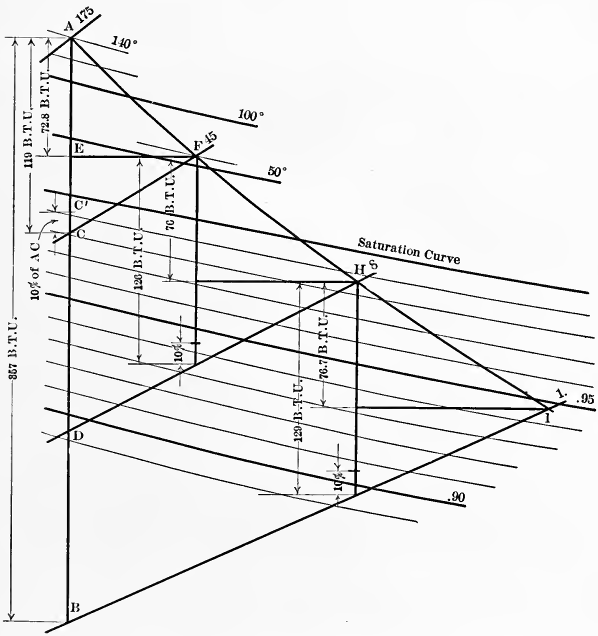


FIG. 34.—Condition curve, Curtis turbine.

the final velocity is found to be 673 feet per second. The blade sections thus determined are given in the same figure.

From Eq. (3.) the force per pound of steam acting in the line of motion, due to entrance, is $\frac{W}{g} V \cos \alpha$ and that due to the exit is $\frac{W_1}{g} V_1' \cos \beta'$. Remembering that W here is unity the total force acting on the first row per pound of steam is

$\frac{V \cos \alpha + V_1 \cos \beta'}{g}$ and the work done is this expression multiplied by the blade velocity, u, or the

Work done in the first row per second per pound of steam =

$$\frac{u}{g}(V \cos \alpha - V_1 \cos \beta'). \tag{26}$$

A similar expression gives the work done in the second row of moving blades. The velocities required may be scaled from the velocity diagrams.

The actual energy per pound of steam which has gone into work during passage through the stage is thus found to be 56540 foot-pounds or in its heat equivalent, 72.8 B.T.U. This is laid off on the heat drop at *AE* in Fig. 34 and the remainder *EC*, in this case about 40% of *AC*, is still in the steam as heat energy.

Since the steam is now at a pressure of 45 pounds absolute, and has a total heat corresponding to the ordinate of *E*, the condition of the steam at the beginning of the second stage is found by projecting *E* horizontally until it meets the 45 pound line as at *F*. From the position of *F* it is seen that the steam, instead of being 98% quality, is superheated about 60° Fahr. Dropping a vertical from *F* to the 8 pound line shows an adiabatic heat drop of 126 B. T. U., which, with 10% allowance for losses, gives a velocity of 2380 feet per second. Laying out the diagram as before, we have an exit velocity of 710 feet for the second stage. The work for this stage is found, by the previous method, to be 59,150 foot-pounds per pound of steam, and the corresponding actual heat drop, 76 B. T. U. Applying this to the heat chart as before, we find that the steam enters the third or last stage with a quality of .986. The adiabatic heat drop for the last stage is 129 B. T. U., and the velocity, allowing for losses, is 2400 feet per second. The energy developed in this stage is 59,600 foot-pounds per pound, and the corresponding heat drop, 76.7 B. T. U. From this last we can obtain the final condition of the steam at *I*, where it is found to have a dryness of .947%.

Adding the work of the three stages, we have

First stage	56,540
Second stage	59,150
Third stage	59,600
Total work per pound of steam =	175,290

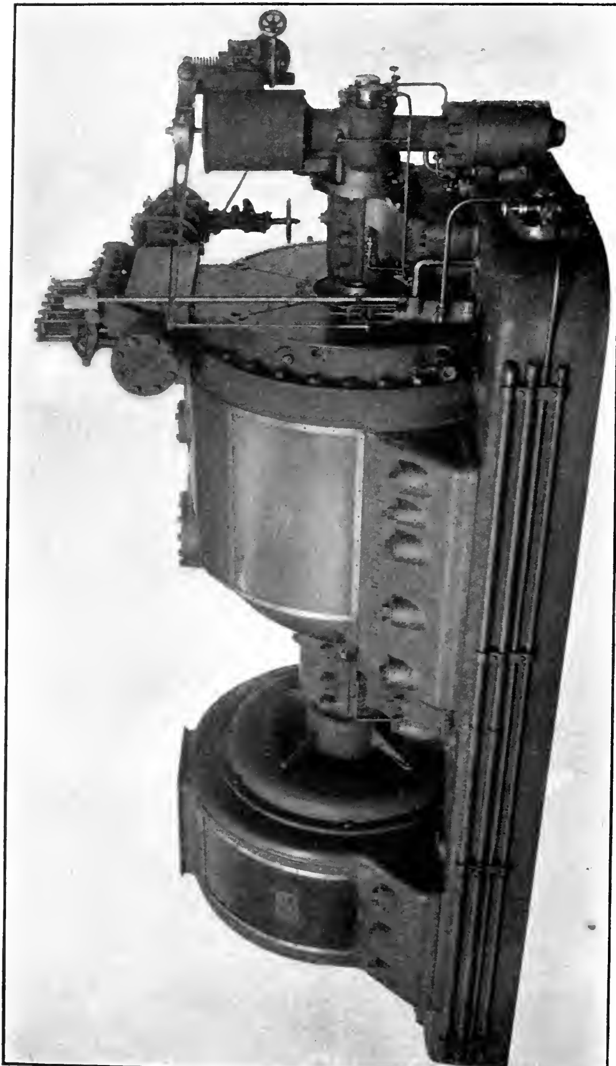


FIG. 35.—1000 K. W. horizontal Curtis turbine; five stages, 1800 R. P M.

The theoretical steam rate $= \frac{1{,}980{,}000}{175{,}290} = 11.3$ pounds per H. P. hour. Allowing 8% for mechanical friction, windage, etc., this becomes $\frac{11.3}{.92} = 12.3$ pounds.

If the generator efficiency be taken at 95% we have a steam rate of 13 pounds per E. H. P. hour, or 17.5 pounds per K. W. hour, on the basis of a power factor of unity.

As in Fig. 12, a quality curve may be drawn through the points *A*, *F*, *H*, and *I*, which will give the condition of the steam for any point during the expansion. With this determined, and with the H. P. to be developed, and the steam rate as found above, we have the basis for the calculation of the lengths of the blades.

Problems.

1. Make diagrams and calculations for a Curtiss turbine with 5 pressure stages, each of 2 velocity stages.

Initial steam pressure	=155 lbs. abs.
Superheat	=75° F.
Final pressure	=28 in. vac.
Assumed blade velocity	=425 ft. per. sec.
Nozzle angle	=22°.
Velocity loss in each nozzle	5%.
Velocity loss in each row of blades	10%.

Calculate the nozzle area and blading for a 500 K. W. turbine, the mechanical and generator friction being taken at the values given on p. 117.

2. Make diagrams and calculations for a L. P. Curtis turbine, the conditions being those of Prob. 1, except that the initial pressure is 15 lbs. absolute and there are two pressure stages, each with two velocity stages. The nozzle angles to be 25°.
3. Report on the Small Curtis Turbine. See paper of G. A. Orrok, Trans. A. S. M. E., vol. xxxi, p. 263.
4. Report on a Comparison of the Rateau or multicellular, and the Curtis Principles of Design. Power, Dec. 20, 1910, p. 2218, and Jan. 3 and 10, 1911, pp. 19 and 64.

References.

STODOLA: "The Steam Turbine."
THOMAS: "Steam Turbines."
EMMET: "The Steam Turbine in Modern Engineering," Trans. A. S. M. E., vol. xxv.
See also references in the problems.

Art. 13.—Single Flow Impulse-reaction Turbine.

An impulse turbine as indicated in Art. 12 is one in which expansion occurs only in the fixed passages or nozzles.

A reaction turbine is one in which the energy is derived from the reaction of steam issuing from openings or nozzles in the moving wheel. No pure reaction turbine is being built at this time. A number were built in this country many years ago by Avery, but they have long since been abandoned. Theoretically, there is no reason why a pure reaction turbine might not be built, but the velocities involved are so high as to render the type impracti-

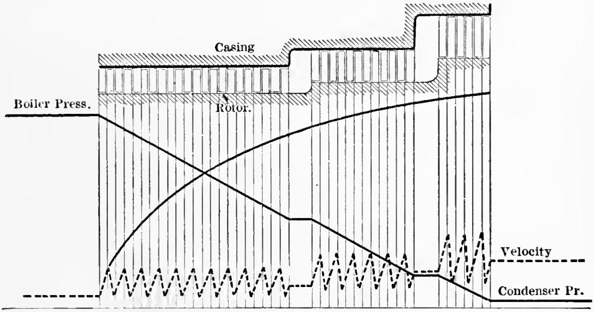


FIG. 36.—Scheme of impulse-reaction turbine.

cal. A combination of the impulse and reaction principles has, however, led to one of the largest and most important types of turbines yet developed.

Figs. 55 and 67 show sectional elevations of two Parsons or impulse-reaction turbines, both single-flow, *i.e.*, the steam enters at the end and flows axially in one direction. Fig. 36, corresponding to Figs. 23 to 26, illustrates their operation. The general form of the buckets is shown in Fig. 37, in which it is seen that the curve at the entering end of the moving vane acts as in impulse turbines, to absorb the kinetic energy discharged from the fixed nozzles. The long straight portion at the exit end of the moving blades forms a nozzle which restores the velocity just absorbed and adds the power of the reaction of exit to that of the impulse first received. In this type there is an almost uninterrupted lowering of the pressure. If there were no motion in the rotor there would be a corresponding rise in the steam velocity, as shown in the curved line, Fig. 36. In fact, the long passage between the rotor and the casing is essentially an enlarged nozzle, its sectional area increasing steadily, and conforming roughly to that which would be given an expanding nozzle.

The impulse-reaction turbine allows the lowest velocities which have been attained with high economy. Its construction is much more complicated than the simple De Laval wheel, but it has been successful in a very wide range of service.

In the impulse-reaction diagram, Fig. 37, the fixed vanes beyond the first row of moving blades receive the steam at a velocity, V_1, using the same notation as on page 29. The steam, in passing through the fixed vanes, expands, and the velocity increases to V. With respect to the following row of moving blades, the steam enters at the velocity W. In passage through the moving blades there is further expansion, and the steam leaves them at a high velocity, W_1. The total work done in the moving blades is: First, the energy $\frac{V^2 - V_1^2}{2g}$ per pound of steam produced in the guide blades and expended upon the moving vanes; second, the reaction accompanying the increase of steam velocity in the moving blade from W to W_1, which is equal to $\frac{W_1^2 - W^2}{2g}$ per pound of steam used. The guide blades and moving vanes usually have the same form and angles, consequently these two expressions are equal, and the total work is divided equally between impulse and reaction. This type of turbine is subject to end thrust which is balanced by pistons on the rotor as shown in Figs. 55 and 67.

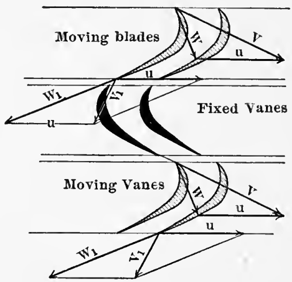


FIG. 37.—Velocity diagram, impulse-reaction turbine.

Before the vanes can be designed in this type, the heat drop in the various stages must be determined, and for this the following data and assumptions are required:

1. The initial steam pressure, with dryness or superheat.
2. The exhaust pressure.
3. The absolute velocity, V, of steam as it leaves guide blades. This varies progressively through the turbine.
4. Friction losses of the steam in passing through the turbine.
5. The exit angles of fixed and moving blades.
6. The peripheral velocity of the blades for the various cylinders or steps.

The exit angles are made the same for both guide vanes and rotating blades, ordinarily between 20° and 30°. The smaller angles than these, while apparently increasing the power, lead to contracted passages and increased friction. On the other hand, with too great angles the power for each stage decreases, requiring too many stages, and increasing the size of the turbine. The peripheral velocity u (item 6 above) at the H. P. stage varies from 100 to 135 feet per second and increases as the size of the rotor increases in the later stages. Table I, from a paper of E. M. Speakman, gives some basis for selecting the vane velocities.

Table I.

PARSONS TURBINE PRACTICE.

Electrical Work.

Rated Capacity	Peripheral Vane Speed		Number of Rows	Revolutions Per Minute
	First Expansion	Last Expansion		
5000 K. W.	135	330	70	750
3500 K. W.	138	280	75	1200
2500 K. W.	125	300	84	1360
1500 K. W.	125	360	72	1500
1000 K. W.	125	250	80	1800
750 K. W.	125	260	77	2000
500 K. W.	120	285	60	3000
250 K. W.	100	210	72	3000
75 K. W.	100	200	48	4000

Marine Work.

Type of Vessel	Peripheral Vane Speed		Mean Ratio of $\frac{u}{V}$	No. of Shafts
	In H. P.	In L. P.		
High speed mail steamer	70–80	110–130	.45–.5	4
Intermediate mail steamer	80–90	110–135	.47–.5	3 or 4
Channel steamers........	90–105	120–150	.37–.47	3
Battleships and cruisers.	85–100	115–135	.48–.52	4
Small cruisers...........	105–120	130–160	.47–.5	3 or 4
Torpedo craft...........	110–130	160–210	.47–.51	3 or 4

The steam velocity (item 3) varies from 2 to 3.5 times the blade velocities. When it would rise above this in the progressive expansion along the rotor, the diameter of the rotor is increased, beginning a new step or drum.

The heat drop in any stage (*i.e.*, one row of fixed vanes and one row of moving blades) may be found readily by the following

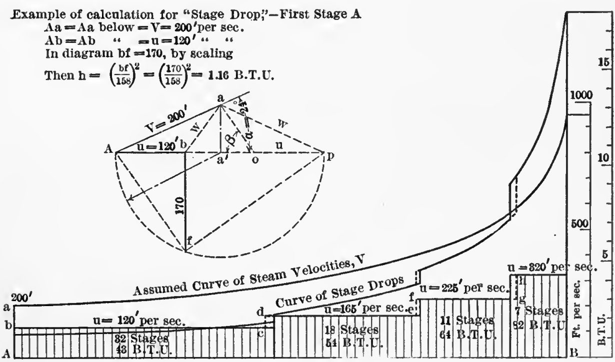


FIG. 38.—Velocity curves and heat drops, single flow impulse-reaction turbine.

simple graphical construction, Fig. 38. In the velocity triangle formed by V, W, and u, produce u and drop a perpendicular aa' upon it from a. With a', the base of this perpendicular, as a center, and Aa' as a radius, describe the arc afp. Drop a perpendicular from b, cutting the arc at f. The work done in foot-pounds per pound of steam in the stage is $W=\frac{(bf)^2}{g}$, where bf is measured to the same velocity scale as u and V_1.

The proof of this is as follows:

The separate velocity diagrams in Fig. 33 may be combined as shown, and where the work is divided equally between impulse and reaction the figures will be symmetrical with respect to aa'. The velocities of exit from the fixed and moving blades are W and V_1, and from Eq. (26) the work is done

$$=\frac{u(V\cos\alpha+V_1\cos\beta)}{g}=\frac{u(Aa'+a'o)}{g}.$$

But bf, being the perpendicular from f to the hypothenuse of the right-angled triangle a, f, p, inscribed in a semicircle, is the mean proportional between the segments ab and bp, therefore

$$\begin{aligned} bf^2 &= Ab\times bp \text{ and since } bp=ao,\\ &=Ab(Aa'+a'o)\\ &=u(V\cos\alpha+V_1\cos\beta)\end{aligned}$$

$$W=\frac{(bf)^2}{g}.$$

The heat drop producing the work will be

$$h=\frac{(bf)^2}{777.5g}=\left(\frac{bf}{158}\right)^2. \qquad (27)$$

If the heat drops in the various stages were equal, the number of stages could be determined at once by dividing the total heat drop from inlet to condenser by the stage drop assumed. The stage drops, however, vary, those nearest the condenser being much greater than the earlier ones. A method of determining these, outlined by Prof. Stodola, will be used in working out a definite case. The conditions will be taken as follows:

Normal capacity	=2000 K. W.
Overload capacity, without by-pass,	=2400 K. W.
Initial pressure	=175 pounds absolute
Superheat	=50°
Condenser pressure	=1 pound absolute

R. P. M.	$=3{,}600$	
Number of drums	$=4$	
Loss due to steam friction	$=$	28%
Loss due to leakage	$=$ 6%	
Loss due to windage, bearings, etc.,	$=$15%	
Energy in exhaust	$=$ 3%	
Total losses in addition to steam friction	$=$	24%.
Efficiency of generator	$=$	94%

We must first assume blade velocities. From reference to Table I on page 56 these may be taken as

$u=120$ feet for first stage, increasing to
$u=320$ feet at exhaust end.

Assume also exit angles for all blades and vanes $=24°$, and a clear area through the blades $=$ 1/3 of the cross section of the opening.

Lay off, as in the lower part of Fig. 38, a horizontal base AB to represent the length of the rotor. Let the steam velocity in the first stage be 200 feet per second, and in the final stage be 950 feet, these values being based on experience. The curve of steam velocities must be assumed. It follows approximately a hyperbolic curve except toward the exhaust end, where it rises rapidly. The curve here drawn is about as followed in practice.

The velocity $u=120$ feet per second, for the first cylinder, is laid off at the left, and the velocity 320 feet, for the last cylinder, is laid off at the right. The most advantageous ratio between the steam and blade velocities $\frac{v}{u}$ is from 2 to 3 and it should vary between about the same limits on each drum. It is customary also to make the ratio between the successive drum diameters substantially constant. The blade velocities, on this basis, come to 120, 165, 225, and 320 feet per second.

Moyer gives an empirical formula for determining the number of stages:

$$n=\frac{C}{u^2} \qquad (28)$$

where C is a constant which varies from 1,500,000 for marine turbines to 2,600,000 for electric generator service, and u is the mean blade velocity in feet per second of the cylinder considered. The n found is the number of rows required if the blade speed for the whole turbine were u. As the cylinder considered develops a certain fraction only of the total power, the number of rows required for that cylinder will be that fraction of the n found in the formula.

It can be seen from the wide range in the constant that reliance on this formula calls for experience on the part of the designer with the working conditions and their relation to the constant used. It may be used to great advantage by any one, however, as giving a good approximation for the number of blades on each drum or cylinder. Let us assume that the power developed by the various cylinders shall be approximately 1/6, 1/5, 1/4, and 3/8 of the total.

Then, taking C at 2,500,000, for

$$\text{1st cylinder,}\quad n_1=\frac{2500000}{120^2}.1/6=30 \text{ stages}$$

$$\text{2nd cylinder,}\quad n_2=\frac{2500000}{165^2}.1/5=18 \text{ stages}$$

$$\text{3rd cylinder,}\quad n_3=\frac{2500000}{225^2}.1/4=12 \text{ stages}$$

$$\text{4th cylinder,}\quad n_4=\frac{2500000}{320^2}.3/8=\ \ 6 \text{ stages}$$

$$\text{Total}\qquad\qquad 66 \text{ stages.}$$

The stages are spaced off uniformly on AB and the drum lengths located tentatively as at *cd*, *ef*, and *gh*.

The heat drop which would occur in a stage located at any point may be determined by the graphical method of Fig. 38. Having taken a sufficient number of these points, and plotted the corresponding heat drops, we get the broken curve marked "stage drops," the breaks being as shown in dotted lines. The average ordinate of this curve gives the average of the heat drops for all the stages, which, in this case, =3.58 B. T. U. This average drop, divided into the total heat drop from inlet to exhaust, will give the total number of stages. The total heat drop, determined from the steam-entropy tables, or from the heat-entropy chart (see Fig. 40) is found to be 338 B. T. U., for pure adiabatic expansion, but there is a loss of energy in the steam of $y=28\%$; therefore $338\times(1-28)=338\times.72=243$ B. T. U. Actual heat drop per pound of steam.

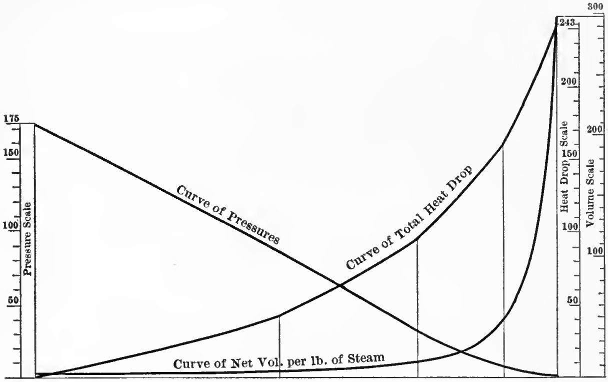


Fig. 39.—Pressure and volume curves, single-flow, impulse-reaction turbine.

The number of stages then will be the

$$\frac{\text{Total heat drop}}{\text{Average stage drop}}=\frac{243\ \text{B. T. U.}}{3.58\ \text{B. T. U.}}=68 \text{ instead of } 66.$$

The 68 stages may be laid off on the base AB and the cylinders, which have up to this point been divided only tentatively, may be readjusted, as shown in full lines. The division made here is as follows:

	Heat Drop, as Scaled from Curve
First cylinder........................32 stages	43
Second cylinder.....................18 stages	54
Third cylinder.......................11 stages	64
Fourth cylinder............. ...... 7 stages	82
Total............................68 stages	243

The progressive heat drop through the turbine may be shown in a curve (see Fig. 39) in which the ordinates represent the progressive sum of the heat drops from A. The last ordinate should check with the total net heat drop 243 B. T. U.

By the initial conditions there are further losses amounting to 24% of the actual heat drop. Therefore, $243 \times 76\% = 185$ B. T. U. per pound of steam, are available for power. Then

$$\frac{1980000}{185 \times 777.5} = 13.8 \text{ pounds of steam per B. H. P.}$$

Since the efficiency of the generator $= 94\%$,

$$\frac{13.8 \times 1.36}{.94} = 20 \text{ pounds} = \text{steam per K. W. hour.}$$

$$\frac{2000 \times 20}{3600} = 11.1 \text{ pounds steam per second for normal load.}$$

The steam rate may rise slightly for overloads, but hardly enough to change the rate inside of 20% overload; we have therefore $\frac{20 \times 2400}{3600} = 13.4$ pounds steam per second for overload conditions. Where a by-pass valve is used, as in Fig. 55, the steam rate should be further raised for the blading beyond it.

There remains the determination of the steam volume under the given conditions, and of the dimensions of the blades and vanes. The condition of the steam may be taken from Fig. 40, which is taken from the heat chart. A, as before, indicates the initial conditions. The vertical drop to B on the 1 pound line gives a quality of 79.2%. But the 28% energy loss raises the quality of the exhaust to 88.5%. The condition curve gives the quality of the steam for any intermediate pressure.

The heat drop for any point, read off from the curve of total drops, Fig. 39, and applied to Fig. 40 by measuring down from A and projecting over to the quality curve AD, gives the pressure and condition corresponding to the pressure. Curves of pressure and volume of Fig. 39 may thus be derived.

The blade lengths may be derived as follows: From Eq. (14) the area of the steam passage required, in square inches, =

$$\frac{\text{weight of steam per second} \times \text{vol. per pound} \times 144}{\text{steam velocity in feet per second.}}$$

$$=\frac{W\times v\times 144}{V}$$

and the length of blade required to give the necessary area (assuming the blades as occupying 1/3 of the clear passageway, and the blade angle $=24°$) is

$$L=\frac{\text{Area}\times 1.5}{\pi\times\text{mean blade dia. } sin\ 24°}=\frac{1.5A}{\pi\times d\times .4}.$$

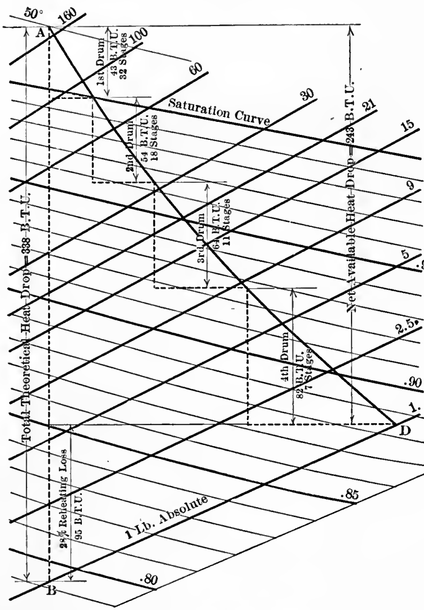


FIG. 40.—Condition curve, single-flow impulse-reaction turbine.

Combining these equations and reducing,

$$L=\frac{W\times v\times 144\times 1.5}{.4\pi Vd}=W\frac{171.5v}{Vd}.$$

For the normal load, 2000 K.W., $W=11.1$ pounds (p. 62).
For the overload, 2400 K.W., $W=13.4$ pounds (p. 62).

The mean blade diameter, d, for the various cylinders, is determined from the values of u, Fig. 38, and the initial condition of 3,600 R. P. M., or 60 revolutions per second.

$$u=60\times\pi\times d, \text{ from which}$$

For first cylinder $d=\frac{120}{189}=.635'=$ 7 5/8 ins. mean dia.

For second cylinder $d=\frac{165}{189}=.875'=$ 10 1/2 ins. mean dia.

For third cylinder $d=\frac{225}{189}=1.19'=$ 14 1/4 ins. mean dia.

For fourth cylinder $d=\frac{320}{189}=1.7'$ $=$ 20 1/2 ins. mean dia.

For a slower R. P. M. or higher blade velocities the mean blade diameters would of course be correspondingly larger.

Taking V from the curve V, Fig. 38, and volumes from Fig. 39, and with the values for w and d above, the lengths of the various blades may be determined. For manufacturing reasons the lengths, instead of increasing progressively along the rotor, are stepped into groups as shown in Fig. 44.

Problems.

1. Make diagrams and calculations for the blading of a single flow Parsons turbine, having four drums.

Capacity, rated,	1000 K. W.
Capacity at opening of by-pass valve,	1300 K. W.
Initial pressure,	150 gauge.
Superheat,	50° F.
Exhaust pressure,	28 1/2 in. vac.
R. P. M.,	3600.
Exit blade angles,	25°.
Steam velocities,	250 ft. to 1000 ft.
Blade velocities,	150 ft. to 350 ft.

Losses and generator efficiency, as given on p. 62.

2. Make diagrams and calculations for the blading of a single flow L. P. turbine of Parsons type.

Initial steam pressure,	16 lbs. absolute.
Quality,	95%.
Exhaust,	28 1/2 vacuum.
R. P. M.,	3600.
Exit blade angles,	28°.

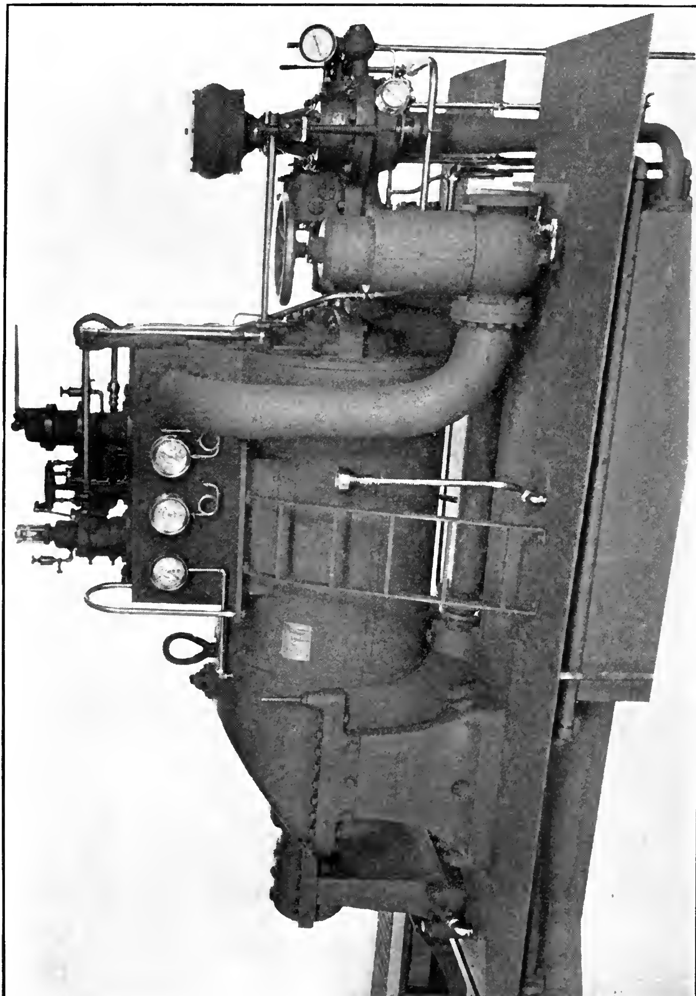


Fig. 41.—Westinghouse single-flow turbine

Assume same steam and blade velocities as for the low pressure stages of Prob. 1.

3. Report on the Avery turbine, interesting as being a pure reaction turbine and probably the first commercial one. See American Machinist, Nov. 9, 1905, p. 631.
4. Report on the article on Blading Calculation, Power, Aug. 9, 1910, p. 1412.
5. Report on article on Construction Details of a Reaction Turbine, Power, May 19, 1908, p. 761.
6. Report on article on the Internal Losses in a Parsons Turbine, by Prof. A. G. Christie. Power, Aug. 24, 1909, p. 299.

References.

Stodola: "The Steam Turbine."
Thomas: "Steam Turbines."
Moyer: "Steam Turbines."
Hodgkinson: Paper, A. S. M. E., vol. xxv.
Parsons: Paper, Inst. Naval Arch. (London), May, 1904.
See also references in the following problems.

Art. 14.—Double Flow Impulse-reaction Turbines.

On account of the drop in pressure through the moving blades in the Parsons turbine steam must be admitted around the entire circumference, which leads to very short blades in the high-pressure stages, and for the same reason the clearance over the ends of the blades must be very small. Leakage is further minimized by using only small pressure drops, which increases the number of H. P. stages, lengthens the rotor, and introduces trouble from expansion which in the larger sizes becomes serious. While the heat drop per stage is less at the H. P. end of the rotor than at the L. P. end, the pressure drop is greater. This, coupled with the fact that the end clearances over the short H. P. blades must inevitably give a greater percentage of leakage, makes the H. P. end of a Parsons turbine its least efficient portion. The pure reaction type, where the blades are pressure balanced and admission may be had through a portion only of the blade circle, is more efficient in the high-pressure ranges than the reaction type, but in the later stages where water is present in the steam it is less efficient.

These considerations have led to the combination of the two types in one machine, utilizing the advantages of both. Steam

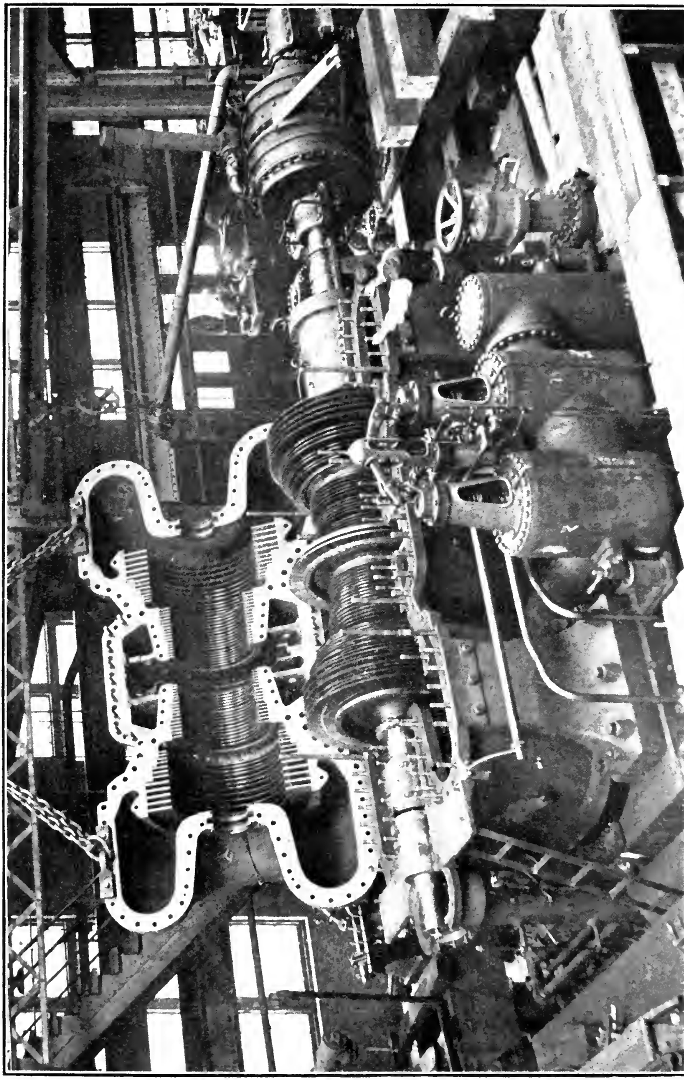


FIG. 42.—10,000-K. W. Westinghouse double-flow turbine.

is admitted through nozzles to a multi-stage impulse wheel and expanded to about atmospheric pressure. The rest of the expansion takes place in fixed and moving blades of impulse-reaction type. The reaction portion is divided in the latest form and steam flows axially in both directions from the impulse wheel. Fig. 42 shows a Westinghouse turbine of this design where the central impulse wheel is clearly seen, with the reaction portions on each side. Fig. 43 is a sectional elevation of the same turbine and shows clearly the course of the steam as it leaves the impulse wheel and is divided into two streams flowing in opposite directions. This arrangement automatically balances

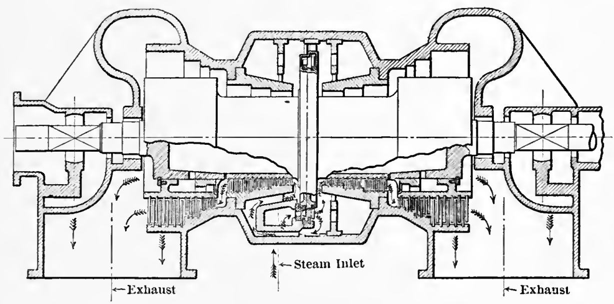


FIG. 43.—Section of 10,000-K. W. Westinghouse double-flow turbine.

the end thrust of the reaction blades and eliminates the balancing pistons necessary in the single-flow type. Besides a symmetrical casing, with smaller exhaust connections it materially shortens the distance between the bearings. Fig. 44 shows two rotors, both for 2000-K.W. Westinghouse machines, the upper one of the double-flow combined type and the lower the old single-flow type.

Double-flow turbines are found most desirable in the larger sizes, but for comparison with the previous problem of a single-flow turbine, let the same conditions be assumed for one of this type. Let the steam be expanded in a set of nozzles from inlet pressure to 25 pounds absolute and pass through two velocity stages of an impulse wheel of the Curtis type as shown in stage *a*, Fig. 29. Let the remainder of the expansion to 1 pound absolute take place in two sets of Parsons blading as in Fig. 42. Let the osses in the impulse element, due to disc and blade friction, leakage, etc., be assumed as 40%, the energy so lost reappearing in the increased quality of the steam. The friction losses in the low-pressure blading will be taken at 28% as before.

Fig. 45 gives the expansion diagram for these conditions. The net drop available for the impulse stage is 91.2 B. T. U. and for the low-pressure stages 142.8 B. T. U. It will be seen from last article that the number of stages in the low-pressure

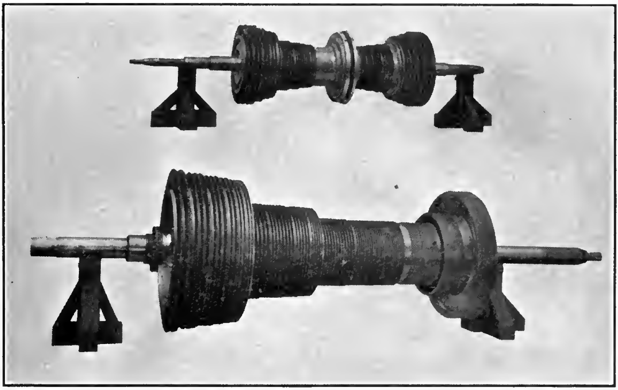


FIG. 44.—Comparison of single- and double-flow Westinghouse rotors—3600 and 1800 R. P. M.—same H. P.

blading depends on the available heat drop, the steam and blade velocities and the exit angles. Dividing the flow does not affect the number of stages on each side. It merely halves the length of the blades, an advantage in the last few stages where in the single-flow machines the length is excessive. Greater blade angles may be used to advantage in low-pressure stages than in high-pressure ones without unduly increasing the number of stages, and 30° will be used instead of 24° as before.

Let the blading be carried on two drums or cylinders running at 225 and 320 feet per second and the steam velocity increase from 400 feet per second to 950. This gives the same conditions as for the last two drums in Fig. 38.

If 3/8 of the total work be carried by the impulse wheel, 1/4 by the smaller drums, and 3/8 by the larger ones, the application of formula (27) gives 13 stages and 9 stages for the first and second drums, respectively. Laying these off tentatively and proceeding as in Fig. 35 we find the mean stage drop to be 6.8 B. T. U. (Fig. 46). Then

$$\frac{142.3}{6.8}=21=\text{number of stages.}$$

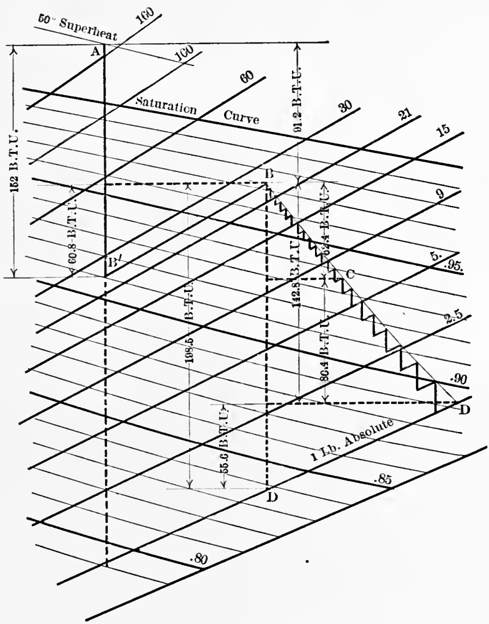


FIG. 45.—Condition curve, double-flow turbine.

Re-adjusting we may take 13 for the first cylinders and 8 for the second. The stage drops are laid off in Fig. 45, those from B to C being on the first cylinder, and those from C to D on the second. The pressures and conditions are found directly and the blade lengths determined as in the last article, which will be those for a single-flow turbine of 21 rows. In the present one there will be *two* complete sets of blades of 21 stages each, each set of blading one-half the length required for the single-flow. Curves of pressures and net volumes per pound of steam are shown in Fig. 46.

It must be remembered that in this and the foregoing articles certain constants, velocities and friction coefficients have been

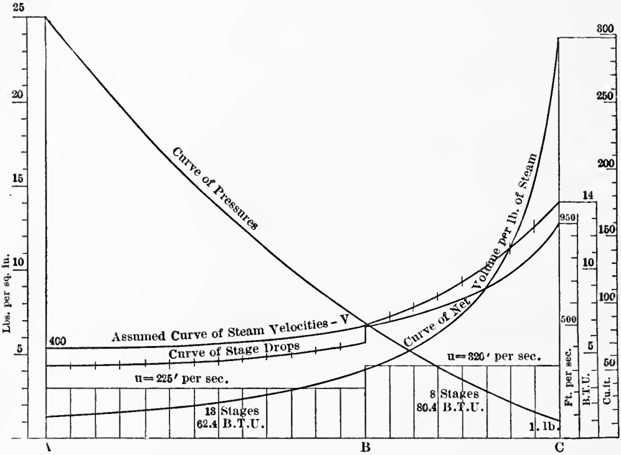


FIG. 46.—Velocity curves and heat drops, double-flow turbine.

more or less arbitrarily assumed. Accurate values for these can be assumed only after wide experience and from intimate knowledge of working conditions. Such information on the choice of values as is available may be found in the standard works on turbines referred to at the end of this article.

Problems.

1. Calculate the blading and draw diagrams for a double-flow turbine similar to Fig. 43.

Capacity,	2000 K. W.
Initial steam pressure,	160 lbs. gauge.
Superheat,	60°.
Exhaust pressure,	28 ins. vac.

Impulse wheel to have two velocity stages and to carry expansion down to 20 lbs. absolute, with reheating losses of 40%.

2. Make diagrams and blade calculations for a turbine for conditions similar to Prob. 1. The first pressure stage to be a two-stage Curtis wheel expanding to 25 lbs. absolute. The remainder of the expansion to take place in a series of wheels of Rateau or multicellular type (Fig. 30). Consult J. A. Moyer, Steam Turbines, p. 86.
3. Report on the Westinghouse double flow turbine described in Power, June 16, 1908, p. 931.
4. Report on the construction and test of a 10,000-K. W. double-flow, impulse-reaction turbine, described in paper by Naphtaly, Trans. A. S. M. E., vol. xxxii, Dec., 1910. (Shown in Fig. 43.)
5. Report on the Melms-Pfenninger turbine (Curtis-Parsons), described in Lond. Engineering, July 9, 1909, p. 39, and Power, Sept., 1907, p. 643.

References.

MOYER: "Steam Turbines."
STODOLA: "The Steam Turbine."
NAPHTALY: "Test of 10000-K. W. Turbine," Trans. A. S. M. E., Dec., 1910.

CHAPTER IV.

MECHANICAL PROBLEMS.

The problems already considered have dealt with the using of steam at the high velocities of turbine practice. These involve problems of a purely mechanical nature, among which are provision for centrifugal strains due to speeds of 3,000 to 30,000 R. P. M., the balancing of the shafts, bearings suitable for such speeds, and close and rapid regulation under varying loads.

Art. 15.—Centrifugal Strains.

A thorough consideration of this subject involves extended and complex analysis. Some only of its phases will be touched on here.

As the velocities in the De Laval turbine are much higher than in any other, it is in this turbine that the interesting behavior of a disc rotating at high velocities is most marked.

If a flexible shaft carry a round disc, one side of which is slightly heavier than the other, the center of gravity will lie at one side of the geometrical center, as in Fig. 47 "a". If they are rotated rapidly the unbalanced centrifugal force of the heavy side will deflect the shaft toward the heavy side, and cause the geometrical center of the disc to describe a circle as at "b". This action increases with the speed up to a certain point called the "critical speed," when the vibration becomes momentarily excessive. On further increase of speed they settle down and will run quietly, however much the speed be increased. At this critical speed the axis of rotation shifts from the center of the path of the geometric center, *o*, to the center of gravity of the wheel, as shown in Fig. 47 "c".

It can be shown that the deflection is

$$y=\frac{e}{1-\frac{V_c^2}{V^2}}$$

where e is the eccentricity of the center of gravity and V_c the critical velocity referred to. The greater the velocity V, the smaller y becomes, approaching the value $y=e$, until the wheel bursts. In the De Laval practice the critical speed is from 1/8 to 1/5 of the normal R. P. M.

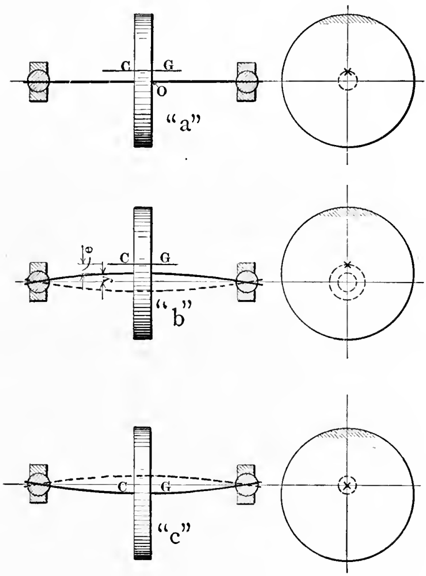


FIG. 47.—Action of the flexible shaft.

For cylindrical rings where the radial thickness is small, the tensional stress due to the centrifugal force is equal to

$$S=\frac{12\ w\ v^2}{g} \tag{29}$$

where S = stress in pounds per square inch.
w = weight per cubic inch of material.
v = linear velocity of ring in feet per second.

Calculation of the stresses in discs rotating at very high speeds is difficult. They vary with the weight of the material and the square of the speed. Fortunately the materials available have been so improved that a factor of safety can be used, sufficient

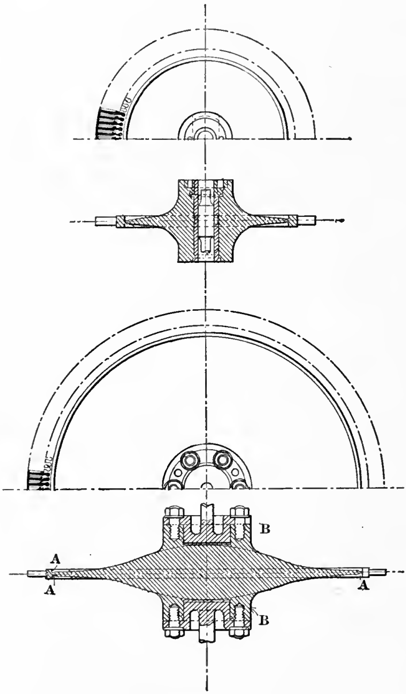


FIG. 48.—Large and small De Laval wheels.

to cover any uncertainty as to the centrifugal strains. Two facts are shown clearly by both theory and experiment.

First. The stresses are greater nearer the center than near the rim.

Second. The stress in the nave of a bored disc is greater than in a solid disc. A small hole may even double the strains.

These principles have been recognized in the design of the disc in the De Laval turbine shown in Fig. 48. The profile is a form of logarithmic curve asymptotic to the central axis of the disc. In the smaller turbines the flexible shaft runs through the disc, but in the larger ones the shafts are flanged to the sides, to avoid piercing the disc. The curved lines added near the center of the large wheel show the theoretical form, to which the attachments are tangent. Wheels of this form, when tested to destruction, would burst through the center into large pieces, and it is stated that these pieces have been driven through a steel wheel case 2 inches thick. This menace is obviated, however, by turning grooves, *A*, *A*, which reduce the thickness of the wheel close to the periphery, decreasing the strength of the wheel at this point so that the stresses here are about 50% higher than in the rest of the wheel. The factor of safety at this point being about 5, the rim of the wheel will tear off at a little over twice the normal speed, breaking up into small fragments, which are unable to damage the wheel case. When the rim lets go, the blades which are the impelling force of the wheel go with it, the centrifugal forces are at once much reduced, and the wheel is unbalanced. The heavy hub which projects into the casing at *B* with but small clearance comes into contact with the sides and acts as a brake, bringing the wheel to rest in a few revolutions. Discs of rolled plate are not used for a single-stage machine, as streaks or lines of weakness, due to "piping," difficult to detect, may become elements of danger.

In connection with the improvement in materials referred to, it may be noted that Krupp and Co. of Essen, Germany, have for this use a special nickel steel of 125,800 pounds tensile strength, 92,300 pounds elastic limit, and 12% elongation. They have recently produced nickel steel of even higher strength but of less elongation. The following figures are taken from their published tables:

Ult. Tensile Strength, Pounds per Sq. Inch	Elongation	Elastic Limit, Pounds per Sq. Inch	
255,600	7.	136,320	Measured on .472 inch diam. bar, 3.937 between points.
252,760	5.5	153,360	
251,340	6.	210,160	
258,440	4.1	227,200	
211,580	6.8	187,440	
310,980	(?)[1]	213,000	

[1] Broke in center punch mark.

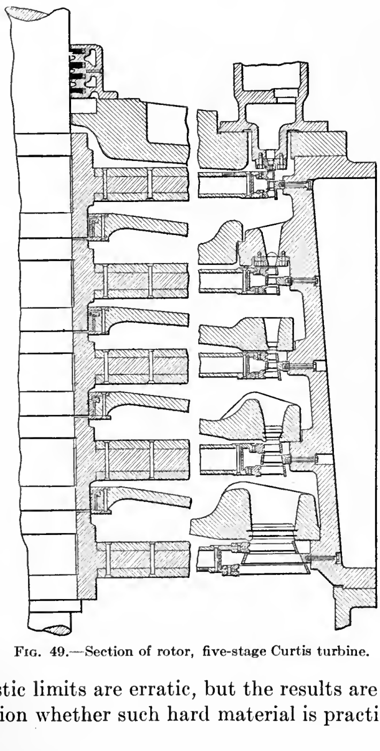


Fig. 49.—Section of rotor, five-stage Curtis turbine.

These elastic limits are erratic, but the results are remarkable. It is a question whether such hard material is practical but discs are used which show under test 134,900 pounds tensile strength, 14% elongation, and 106,630 pounds elastic limit.

In multi-stage impulse turbines where the velocity is about half that of the single-stage type, the question of centrifugal strains is not so difficult. In the Curtis turbine the wheels are built up of castings and plates. The blades are milled in a sectional blade ring bolted to the discs at the edge. Fig. 49 shows the detail of this construction.

In turbines of the Parsons type the velocities are still further reduced (see table, p. 56). The principle adopted by Mr. Parsons was to have the drum and shaft stiff, balanced as closely as possible, and carried in a bearing which would provide sufficient lateral motion to allow the shaft to find its own center.

Problems.

1. Report on the conclusions of paper on "Critical Speed Calculations." by S. H. Weaver, Jour. A. S. M. E., vol. xxxii, June, 1910.

References.

STODOLA: "The Steam Turbine" (especially 4th ed.).

JUDE: "Theory of the Steam Turbine."

MOYER: "Steam Turbines."

J. M. NEWTON: "Design and Construction of High Speed Turbine Rotors," London Engineering, July 8 and 15, 1910.

WEAVER: "Critical Speed Calculation," Jour. A. S. M. E., June, 1910.

Art. 16.—Bearings.

The earlier form of bearing used by Parsons is shown in Fig. 50, where the space between the shaft and the journal wall was taken up by two sets of rings. Those of one set fitted the shaft closely, and had play between their outer diameters and the journal wall. Those of the other set, which alternated with these, fitted the journal box tightly, and were loose on the shaft. They were held in contact by a side spring. A lateral movement of .005 inch or more was thus provided. This arrangement, while satisfactory, has been superseded by one (Fig. 51) in which there is a nest of bronze sleeves concentric with the shaft, each having about .002 inch play on the diameter. Holes are bored through them to permit the lubricating oil to reach the inner sleeves and shaft. Side play is thus provided as before, but with a much simpler construction. In the large machines, running below 1200 R. P. M., the flexible bearing is dispensed with and solid self-oiling bearings are used.

In the vertical Curtis turbine the problem has been to design

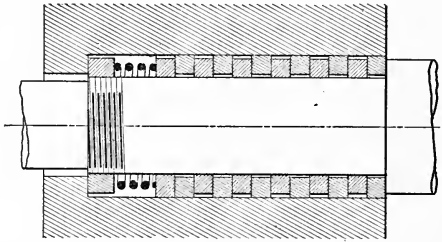


FIG. 50.—Old Parsons shaft packing.

a step bearing which would carry the entire weight of the shaft, wheels, and the field. The end of the shaft carries a cast iron block, which runs upon a similar block in the bearing. Oil or water is forced through the pipe *A*, Fig. 52, into the recessed portion between the bearing faces, and circulates out and

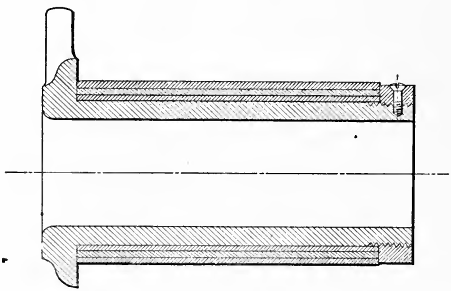


FIG. 51.—Later Parsons shaft packing.

upward, through the journal, leaving the bearing at the top *B*. The entire weight of the moving parts is thus carried on the film of lubricant between the blocks *a* and *b*. The pressure of the lubricant is from 150 pounds to 900 pounds per square inch, according to the size.

Problems.

1. Report on the principles of labyrinth packings, see Power, Sept. 8, 1908, p. 401.
2. Report on the general principles of the labyrinth packing, see Lond. Engineering, Jan. 10, 1908, p. 35.

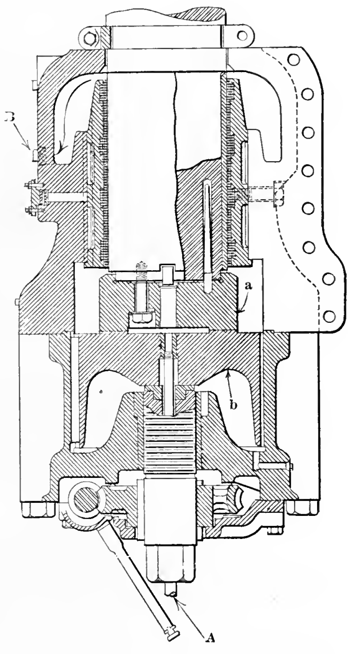


FIG. 52.—Step bearing of a Curtis vertical turbine.

References.

STODOLA: "The Steam Turbine" (4th ed.).
JUDE: "Theory of the Steam Turbine."
For theory of the labyrinth packing see London Engineering, Jan. 10, 1908, p. 35, also "Power," Sept. 8, 1908, p. 401.
Revue de Mechanique, Nov., 1910.

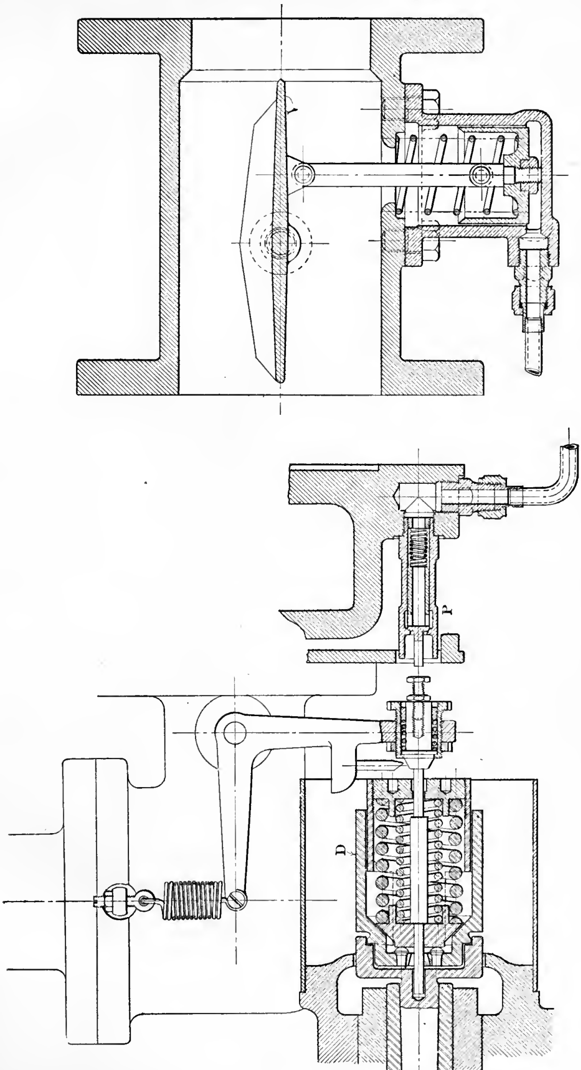


Fig. 53.—De Laval throttling governor.

Art. 17.—Governing.

There are three types of turbine governors in general use:

First, balanced throttling valves.

Second, multiple closing or relay valves.

Third, intermittent admission.

The well known turbines present examples of all three of these classes. A fourth method of control is used in conjunction with the third as supplementary to it. It consists of means for admitting H. P. steam directly to the later stages by means of a by-pass valve, in order to provide for heavy overload conditions.

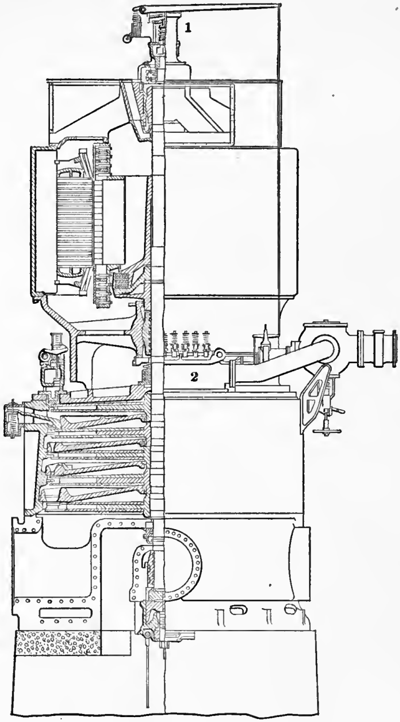
Throttling Type.

As applied on the DeLaval turbine, this governor has a spring-actuated double-beat valve, controlled by a spring and weight governor located at one end of one of the gear shafts where the speed has been reduced 10 to 1. Even here the speed is from 1,000 to 3,000 R. P. M. This high velocity alters the weight arms materially from the well known shape, and they take the form of semi-cylindrical leaves, *D*, turning about the knife edges, and closely enfold the controlling spring. The governing mechanism has a further control in addition to that of the throttle. If the balanced valve fails to govern and the speed continues to rise, the tappit, *P*, comes into contact with an independent valve, operating a butterfly valve, *V*, in the exhaust passage and raising the back pressure. This immediately puts a brake on the wheel and prevents any further acceleration of speed. The effectiveness of this action may be shown by the fact that a 150 H. P. turbine, with nozzles designed for 150 pounds gauge pressure, and 26 inches vacuum, if operated on a back pressure equivalent to an atmospheric exhaust, will slow down and can not be brought up to full speed even with the entire load removed.

The controlling mechanisms of the De Laval, Kerr, Zoelly, and a large number of other turbines follow these general lines.

Multiple or Relay Control.

In the smaller sizes of Curtis turbines governing is effected by throttling, as already described. In the larger sizes the speed is controlled by the opening and closing of small valves admit-



1. Govenor.
2. Steam Chest.

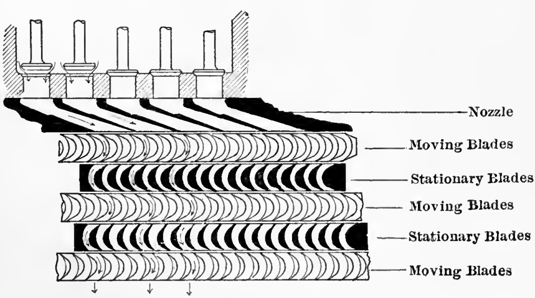


FIG. 54.—Relay control of Curtis turbine.

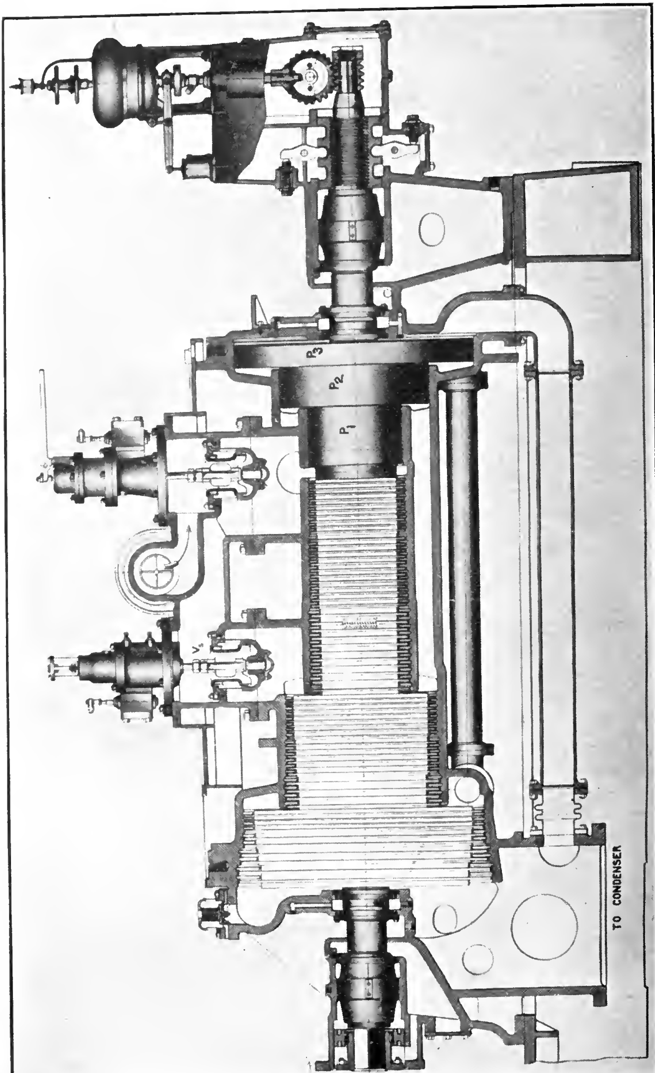


FIG. 55.—Section of Westinghouse turbine.

ting to the separate nozzle sections, as shown in the lower part of Fig. 54, which is a schematic section of the casing 2 above, the number in action depending upon the load. These nozzle valves are operated from the governor mechanism, 1, at the top of the unit, and are designed to be fully open or shut, like a pop-safety valve, in order to prevent cutting of the seats by wire drawing. The control of the nozzle valves by the governor has been effected by means of positive mechanical connections, electrical connections, and hydraulic cylinders. The last has been found most satisfactory, and is used on the later machines.

This method of control by multiple valves gives excellent economy, as the steam is used at an almost constant thermodynamic efficiency, whatever the load.

Intermittent Control.

The general arrangement of the controlling mechanism is shown on the general section of a Westinghouse turbine, Fig. 55.

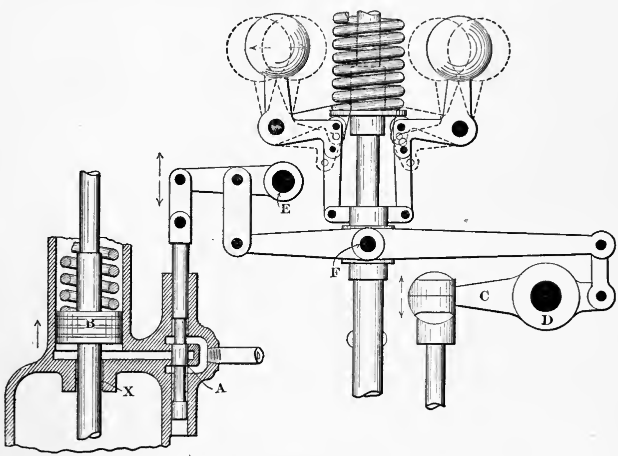


Fig. 56.—Governing mechanism of the Westinghouse turbine.

Admission is through a single double-beat valve controlled by a steam-operated piston above it, Fig. 56. A heavy spring above the piston tends to hold the valve to its seat. Steam is admitted to the piston B through the annular clearance X, around the valve steam and exhausts through a controlling valve A, which is actuated by the governor. The governor is of the bell-crank type, the horizontal arms bearing against an adjustable spiral spring. E and D are fixed fulcrums, F, a floating one, moving up and down with the governor sleeve. C is an oscillating lever driven by worm gearing and an eccentric, from the main turbine shaft. The motion of C is communicated to the pilot valve, A, and thence to the main valve, admitting steam to the turbine in gusts. The distance that the valve A opens as it moves up and down, and consequently the time it

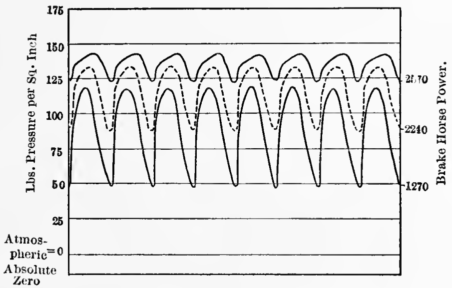


FIG. 57.—Pulsation diagram, Westinghouse turbine.

remains open, is controlled through F by the position of the governor. At light loads the main valve opens for only short periods and remains closed for the greater part of the stroke of A. As the load increases the valve remains open longer, until finally a practically continuous pressure is maintained in the H. P. end of the turbine. The effect of the governor on the initial pressures is shown in Fig. 57. The fourth method of control, that of providing for overloads by admission of steam to later stages is also shown in Fig. 55 where a secondary admission valve V_s is provided, to admit high pressure steam to the second drum of the turbine for overloads. It is not opened and closed regularly, as the main one is, but is controlled by another pilot valve connected directly with the governor sleeve. Another device for accomplishing the same thing, used by Brown, Bovari, and Co. in Europe, opens by-pass valves to the later stages as the pressure rises in the H. P. chamber. It is obvious that steam admitted directly to the later stages is not used to the greatest advantage, but as overload conditions are intermittent the loss in steam economy is admissible.

A comparison of tests of turbines using the three methods of governing[1] shows that from the standpoint of steam consumption under light loads the multiple valve control is most economical, intermittent admission next and throttling valve last. The throttling governor is, however, the simplest and is used almost universally on small units.

Problems.

1. Look up and report on the various methods of operating the multiple nozzle valve on the Curtis turbine, see Bulletins of the General Electric Co.
2. Report on the regulation of the Rateau low pressure turbine, described in Power, May 10, 1910, p. 845.

References.

Stodola: "The Steam Turbine" (4th ed.).
Jude: "Theory of the Steam Turbine."
Moyer: "Steam Turbines."
Revue de Mechanique, Oct., 1910.

[1] Mechanical Engineer, Jan. 20, 1906.

CHAPTER V.

COMPARISON OF TYPES.

Art. 18.—Vertical Turbines.

Up to the introduction of the Curtis turbines no one seems to have thought of arranging the shaft to run vertically, influenced, perhaps, by the accustomed position of the reciprocating engines. The builders of this turbine adopted this position and still adhere to it for their large units. Its use is limited to self contained generating sets. The following reasons are advanced in its favor.

1. Accurate control of the relative position of its parts by means of the adjustable step bearing. (Fig. 52.)
2. Symmetry of design.
3. All bearings relieved from side strain and lateral wear.
4. Friction practically eliminated.
5. A short shaft, free from deflection.
6. Small floor space.
7. Small foundations.

The arrangement, shown in Fig. 54, unquestionably has advantages. That it reduces friction to a minimum is shown by the fact that unless a brake is applied a 5,000-K.W. unit will run for four or five hours after steam has been shut off. The elaborate step bearing necessitated, with its system of high pressure, forced lubrication is one of the chief objections, and though little trouble seems to have developed from this source, it is an object of constant watching. The generator is located above the turbine, exposed to the heat arising from it, and to steam leaking past the glands. While the vertical position offers convenient access for inspection and minor repairs, it is inconvenient in case of overhauling. In the horizontal type the turbine and generator may be overhauled independently, and work can be done simultaneously on both ends. In the vertical, the machine has to be taken apart from the top down, and the discs stripped off one by one.

An interesting example of the vertical type was built in Berlin by the Allgemeine Electricitäts Gesellschaft or "A.E.G.," combining the features of the Curtis turbine with those of the Reidler-Stumpf. A section of this turbine is shown in Fig. 58. The weight of the moving parts was carried by a thrust bearing (*B*) between the generator and turbine, and a jet condenser was located at

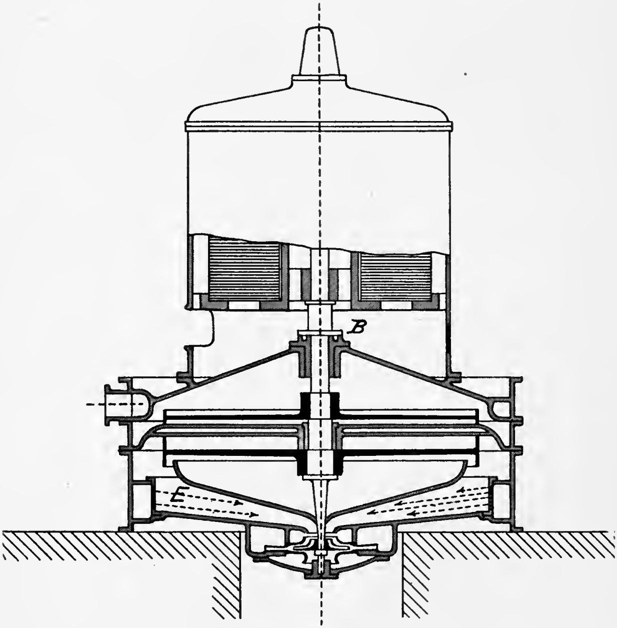


Fig. 58.—Section of old A. E. G. turbine.

the lower end of the shaft. The steam exhausted into the passage *E*, where it was condensed by water sprayed in from the side. The condensed water drained into a centrifugal pump on the end of the main turbine shaft. This arrangement is said to have given a high vacuum, and proven very satisfactory. The A.E.G. have, however, discontinued this vertical type and now

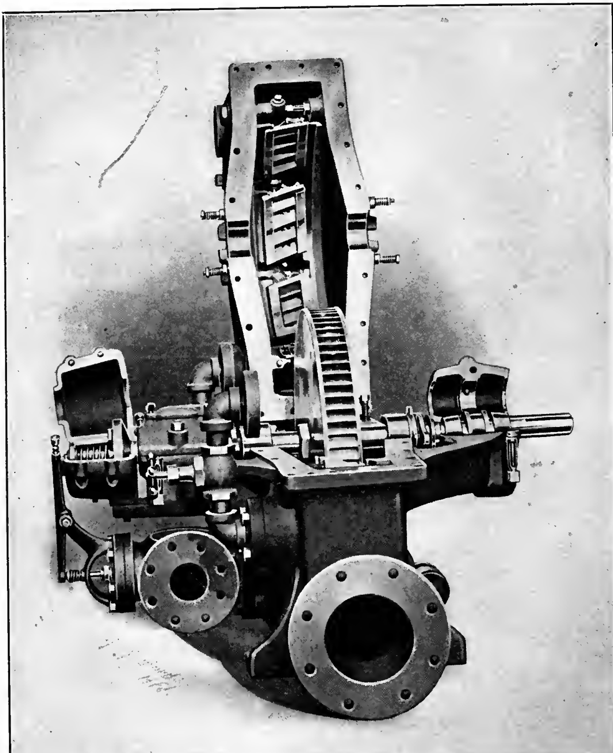


Fig. 59.—100 H. P. Terry steam turbine, opened.

build for their larger sizes a horizontal machine illustrated in Art. 20. This has a Curtis single-pressure, multi-velocity stage wheel and a low pressure end with a number of pressure stages, each having a single velocity stage. A section of this turbine is shown in Fig. 66.

Problems.

1. Report of paper, Emmet, "Steam Turbine in Modern Engineering," Trans. A. S. M. E., vol. xxv., p. 1041.

References.

STODOLA: "The Steam Turbine."
THOMAS: "Steam Turbines."
JUDE: "Theory of the Steam Turbine."
FRENCH: "Steam Turbines."
EMMET: Paper, "Steam Turbine in Modern Engineering," Trans. A. S. M. E., vol. xxv.

Art. 19.—Tangential Flow Turbines.

The Reidler turbine is one of the few turbines which use tangential jets. The buckets for tangential wheels may have two forms, those which curve in one plane only, as the Reidler

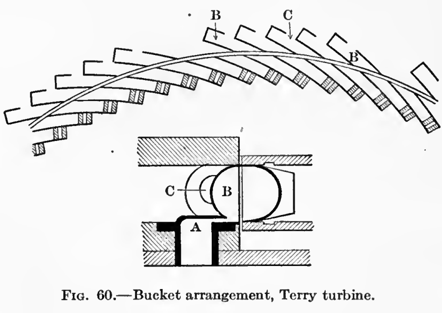


FIG. 60.—Bucket arrangement, Terry turbine.

and Terry turbines, with the jet divided or not, or they may be of the Pelton type, which curve in all directions. Prof. Rateau and Zoelly both experimented extensively with the divided bucket, but have abandoned it in favor of multicellular machines with axial flow. In the Reidler-Stumpf machines, however, it has been developed with success, and these have used it in units of 2,000 K. W. and over, having four or more stages.

The Terry turbine, Fig. 59, is manufactured in this country. In this, undivided tangential jets are used. The steam is expanded in one or more nozzles to exhaust pressure and enters at *A*, Fig. 60, is deflected to one side, and compounded on a single wheel by looping back into buckets *B*, in the casing, and returned from these into succeeding buckets in the single row on the wheel.

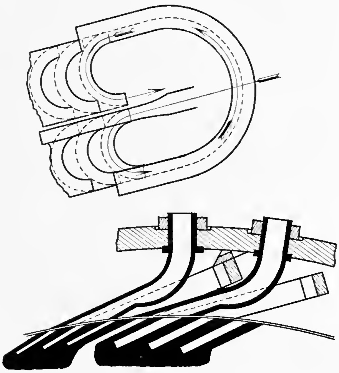


Fig. 61.—Nozzle arrangement of Reidler-Stumpf Turbine.

This reversal is accomplished four or five times, a portion of the steam escaping at each loop, through the exhaust openings, *C*.

Some of the dimensions of one of these turbines are:

Diameter of wheel	=2 feet.
Number of buckets	=70.
Width of buckets	=2 1/2 inches.
Pitch of buckets	=1 inch.
R. P. M.	=2600.
Peripheral velocity	=270 feet per second.

It developed 30 H. P. with 145 pounds steam pressure on 32 pounds steam per H. P. hour. Remarkable economy is not claimed for this type but it gives a compact and simple turbine with low cost and running at moderate speeds. In the Reidler turbines the return loops are made in separate sets of buckets, as shown in Fig. 61.

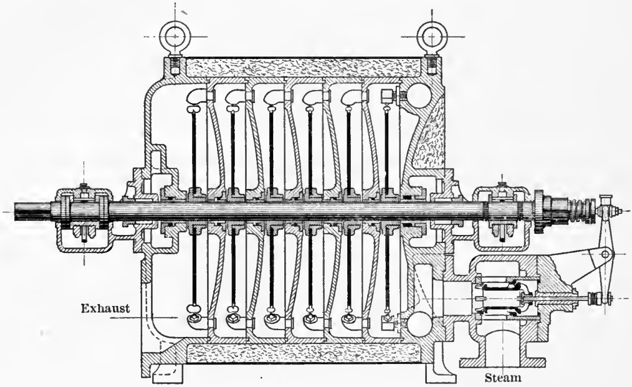


FIG. 62.—Section of Kerr turbine.

The Kerr turbine, developed by Mr. C. V. Kerr, uses buckets of the Pelton divided-flow type, mounted on discs. It is of the multicellular type shown in Fig. 30. The buckets are drop forgings, finished on the inside. The stages are repeating

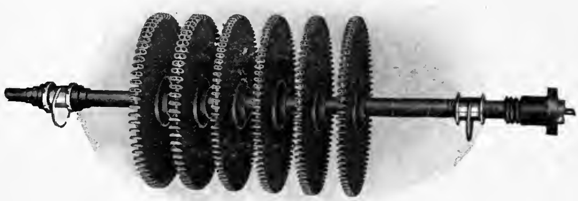


FIG. 63.—Rotor of Kerr steam turbine.

sections with the nozzles increasing in number, as rendered necessary by the expansion of the steam. The design shows careful consideration of the manufacturing problems involved, and adaptability for commercial production. The speed of the Terry and Kerr turbines is from 1200 to 3,000 R. P. M., and they are used direct connected for blowers, pumps, and generators, also for belt driving. The governors are of the throttling type, closely resembling the De Laval.

Problems.

1. Report on the Terry turbine, see Power, Nov. 1907, p. 801, and paper of Orrok, A. S. M. E., vol. xxxi, p. 263.
2. Report on the Kerr turbine as described in paper of G. A. Orrok, Trans. A. S. M. E., vol. xxxi, p. 263, also Am. Machinist, Mar. 21, 1907, p. 415.
3. Report on Sturtevant steam turbine, see paper of G. A. Orrok, A. S. M. E., vol. xxxi, p. 263.
4. Report on the Dake steam turbine. See Power, December, 1907, p. 886, and paper of Orrok, A. S. M. E., vol. xxxi, p. 263.
5. Report on the Bliss turbine as described in paper of G. A. Orrok, A. S. M. E., vol. xxxi, p. 263, and Power, August 10, 1909, p. 250.
6. Report on the Richards turbine described in the American Machinist, November 16, 1905, p. 660.

References.

Stodola: "The Steam Turbine."
Jude: "Theory of the Steam Turbine."
See also references in the following problems.

Art. 20.—Multicellular Turbines.

As mentioned before, Prof. Rateau's investigations led him into a multicellular type of machine having many pressure stages, each with a single velocity stage, which has been brought to a high degree of perfection in France and is one of the best known of the European turbines. It is also one of the few turbines which have been applied to Marine Service. Fig. 64 shows a section of a L. P. Rateau turbine as manufactured in the U. S. The H. P. machines are practically the same except for the number of stages.

The pressure and velocity changes are those shown in Fig. 30. The running joints are on the shaft, where for a given clearance there is a minimum of leakage. So many rotating discs would appear to offer large steam friction loss, but tests of turbines up to 1,000 and 2,000 H. P. show a friction of only from 2 to 4%, or about the same as drum turbines of the Parsons type. The expansion nozzles are inserted in openings in the fixed diaphragms. In the high-pressure stages they occupy but a small portion of the circumference, but extend farther around as the pressure falls and the volume increases. The nozzles or vanes in each successive diaphragm are set around in angular advance of the preceding one, corresponding to the trajectory of the steam, so that the steam as it leaves the moving blades is opposite the openings of the next expansion set. An advantage in this type is that the clearance around the blades may be very large,

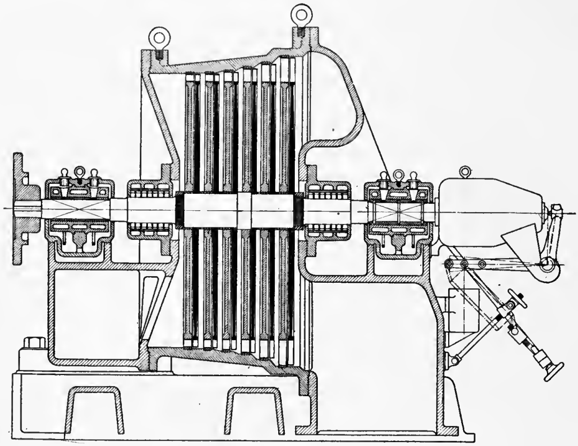


FIG. 64.—Section of Rateau low-pressure turbine.

running from 1/16 to 1/4 of an inch; in fact the builders do not trouble about giving it any precise value. The only small clearances are at the hubs, where friction is less to be feared. The diaphragm bushings are of soft material with little attempt at clearances, for the machine when started wears a sufficient play to turn without touching.

The Zoelly turbine has been mentioned as having originally had tangential flow instead of axial flow. The old form had a wheel with long spokes milled on their faces for two grooves, which gave them a Pelton form where the jet impinged upon them.

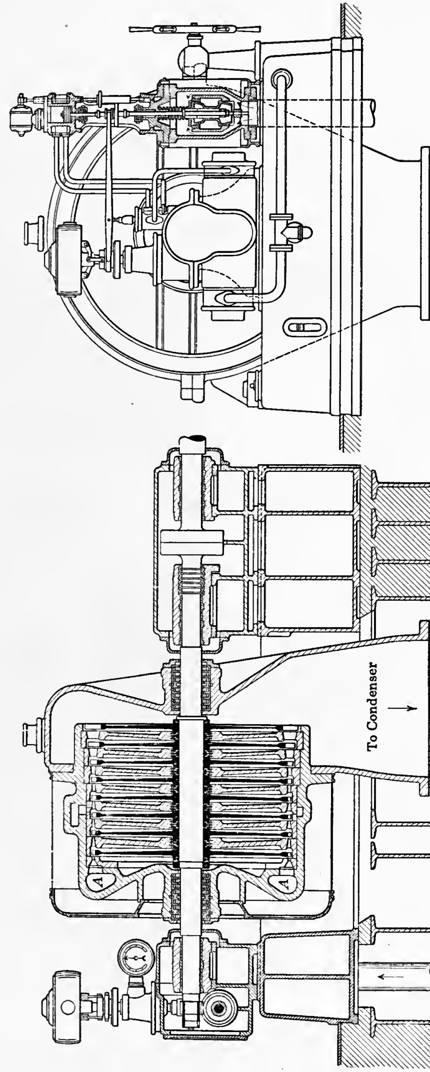


Fig. 65.—Zoelly steam turbine.

With the change to side admission the long spokes were abandoned, and as now built the Zoelly is a multicellular turbine differing from the Rateau only in its construction and details. It has been very successful and is widely known in Europe.

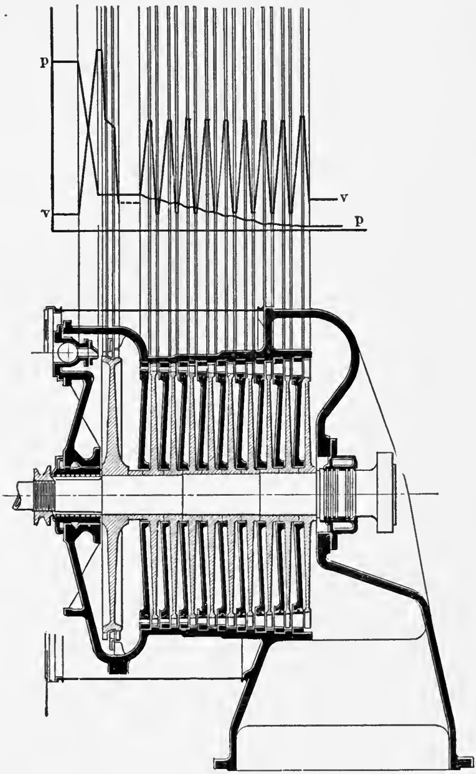


FIG. 66.—Section of A. E. G. horizontal turbine.

As stated in Art. 18, the Algemeine Electricitäts Gesellschaft are building a turbine combining a Curtis single-pressure, multi-velocity stage wheel, in which the steam is expanded to about atmospheric pressure, and a multicellular low-pressure end, making a Curtis-Rateau machine analogous to the Curtis-Parsons described in Art. 14. The Bergmann turbine is also of this type. Both are being built in large sizes.

Problems.

1. Report on the A. E. G. turbine described in Lond. Engineering, May 20, 1910, p. 639. Also Power, Feb. 7, 1911.
2. Report on the Rateau turbine as described and illustrated in Lond. Engineering, May 15, 1908, p. 639.
3. Report on the Zoelly turbine as described and illustrated in Lond. Engineering, July 3, 1908, p. 1.
4. Report on multicellular turbine described in U. S. Patent issued to J. F. M. Patitz, October 4, 1910. No. 971,555.

References.

Stodola: "The Steam Turbine" (4th ed.).
Rateau: Paper, "Different Applications of SteamTurbines," Trans. A. S. M. E., vol. xxv.
Jude: "Theory of the Steam Turbine."
Rey: Paper, "Rateau Steam Turbine and its Applications," Journal Am. Soc. Naval Eng., Nov., 1905.
See also references in the problems.

Art. 21.—Special Forms of Impulse-reaction Turbine.

The Allis-Chalmers turbine shown in Fig. 67 is a single-flow machine of the Parsons type differing from the Westinghouse only in the method of balancing, blading, and in various details.

In the typical single-flow Parsons turbine the end thrust is carried by the pistons P_1, P_2, and P_3 shown in Fig. 55. In the large sizes the large low-pressure piston, P_3, distorts under pressure and repeated heating and cooling, so that it has been necessary to give them clearances which permit leakage of steam. In the Allis turbine this large balancing piston is replaced by a smaller one, a, at the low-pressure end of the rotor where it is practically free from distortion, and nearly the whole area is effective. In the later forms of Westinghouse machines the balancing is accomplished by the double flow, as already described. In the blading, the ends are riveted to a channel-shaped shroud ring, B, Fig. 68, stiffening them against vibration. The openings

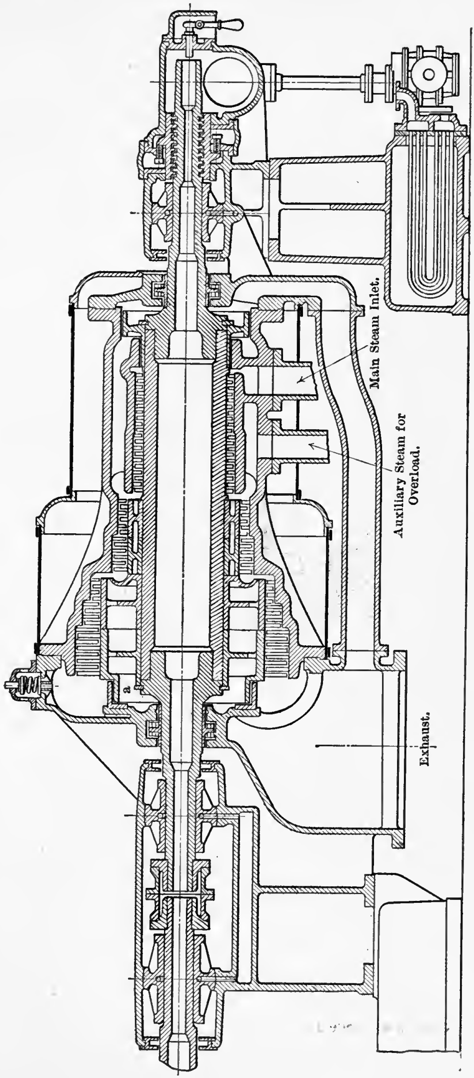


FIG. 67.—Section of Allis-Chalmers turbine.

for the blade tips are stamped in the ring, accurately spacing the blades. The shape of the ring forms a natural baffle, and the clearance may be smaller than in previous practice. Only the thin edges of the channel can touch the casing, and if they do, they wear free without any material heating. The

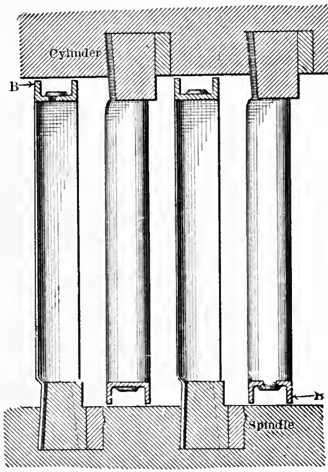


FIG. 68.—Blading of Allis-Chalmers turbine.

shroud ring itself is not new, as both the Rateau and Zoelly have used them, but the gain from its use is greater in reaction turbines than in these types.

Problems.

1. Look up and report on the early history of steam turbines, see Neilson, French or Gentsch.
2. Look up the Elektra turbine described in Power, April 6, 1909, p. 635. Compare with the new type of small Westinghouse turbine.
3. Report on the Wilkinson turbine and small type of Westinghouse turbine.
4. Report on the Eyermann turbine, a turbine having combine radial and axial flow. See Power, November 24, 1908, p. 855.

References.

STODOLA: "The Steam Turbine" (4th ed. illustrates and describes all the turbines of any prominence).
JUDE: "Theory of the Steam Turbine."
GENTZCH: "Steam Turbines." Translated by A. R. Liddell.
SOZNOWZKI: "Turbines a Vapeur," Paris, 1897.
(The last two are valuable for the history of steam turbines.)
See also references in the problems.

Art. 22.—Low-pressure Turbines and Regenerators.

Reference to the pressure-temperature curve, Fig. 69, or to the heat chart, shows that the heat drop on which the available energy depends is greatest at the low pressures. A reciprocating

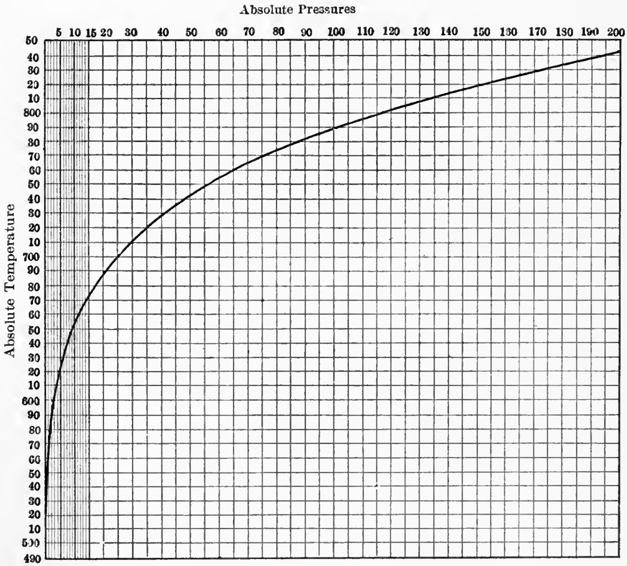


FIG. 69.—Pressure-temperature curve.

engine is most efficient at the higher pressures, down to about that of the atmosphere. The turbine is most effective in utilizing the lower vacuums, as excessive expansion in an engine entails prohibitive sizes of L. P. cylinders, and, with the wide range of power available below atmospheric pressure, it can be coupled with a reciprocating engine to great advantage. In fact, the economy of such a combined unit where the engine exhausts into a low-pressure turbine, is higher than when the entire pressure range from boiler to condenser is utilized in an engine or a turbine alone, and the installation of combined units of this kind is coming more and more into use. For marine service there is a further advantage of superior maneuvering power and economy at slow speeds over turbines alone, and the new White Star steamers Oceanic and Titanic are being equipped with engines and exhaust turbines. For power station work the *total* economy of a combined unit of new engines and new turbines, taking into account all running expenses, fixed charges, etc., is not so high as for a H. P. turbine plant alone. It furnishes, however, a valuable means of utilizing existing reciprocating plants and bringing them up to the highest efficiency. Where a good engine plant already exists its capacity may be increased more economically by the addition of exhaust turbines than by replacing the engines with H. P. turbines.

The conditions of operation for exhaust turbines fall under two classes:

First, where the steam supply is substantially constant or varies with the turbine load.

Second, where it is widely fluctuating or intermittent and is independent of the turbine load.

In the first class fall marine engines and those for power station service. A conspicuous example of the successful installation of exhaust turbines for the latter service is found in the Fifty-ninth Street Station of the Interborough Rapid Transit Co., N. Y. City.

This station was equipped with 7500-K. W. compound Corliss condensing engines direct connected to generators, each unit having two 42-inch H. P. and two 86-inch L. P. cylinders, all of 60-inch stroke, and an average steam rate of 17 to 18 pounds per K. W. hour. After consideration of various means of providing additional power, exhaust turbines were installed on three units to determine their desirability for the remaining sets. They were 3-stage vertical Curtis turbines, of 7500 K. W. maximum rating, each having 6 fixed nozzles, and 6 operated by hand so as to control the division of load between the engine and its exhaust turbine. An overspeed governor operating a 40-inch butterfly valve on the steam line connecting the separator and the turbine, and an 8-inch vacuum breaker on the condenser were the only forms of governor used.

Mr. Stott summarizes the results of the tests as follows:

"a. An increase of 100% in maximum capacity of plant.

"b. An increase of 146% in economic capacity of plant.

"c. A saving of approximately 85% of the condensed steam for return to the boilers.

"d. An average improvement in economy of 13% over the best H. P. turbine results.

"e. An average improvement in economy of 25% (between limits of 7500 K. W. and 15000 K. W.) over the results obtained by the engine units alone.

"f. An average unit thermal efficiency between the limits of 6500 K. W. and 15500 K. W. of 20.6%."

The steam rate of the engines alone was from 17 to 18 pounds per K. W. hour, of the exhaust turbines, 26 1/2 to 28 pounds, and of the combined unit, 13 to 14 pounds.

In an installation of this character where the load on the turbine is electrically connected with that of the engine, the turbine load rises and falls with the steam supply and no governor is required. When running at very light loads, where a turbine is inefficient, it is found economical to cut it out, and run the engine alone, condensing. Where the supply of steam from the engine is liable to long-continued interruptions, live steam can be admitted through a reducing valve from an independent source, the exhaust turbine thus drawing its supply from two sources.

The second class of installations, where there is intermittent steam supply, comprises rolling mills, forging plants, mine hoists, and other non-condensing engines. Such engines use great quantities of steam under widely varying loads, often with little or no expansion. Prof. Rateau has given much attention to the application of low-pressure turbines to such exhausts, and to equalize the supply from these constantly varying sources he has developed the steam regenerator or heat accumulator, shown in Figs. 70 and 71. The engine exhaust at about atmospheric pressure is condensed in the regenerator when it arrives in large quantities, and revaporized when the excess ceases. The condensation and re-evaporation of the steam follow the fluctuations of pressure as the supply exceeds or falls short of that required by the turbine.

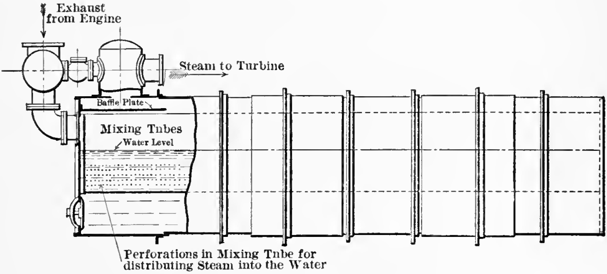


FIG. 70.—Side view of Rateau regenerator.

The regenerator takes a form like a tubeless horizontal boiler (Fig. 70) where the water is kept in forced circulation by the injection of steam. It is necessary to provide a relief valve to carry off any surplus steam, an automatic expansion valve, as in the previous case, to supply steam directly to

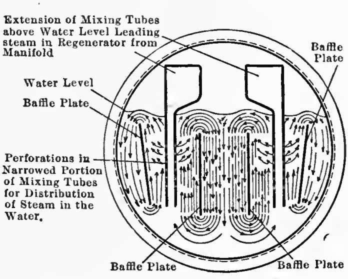


FIG. 71.—Cross-section of Rateau regenerator.

the turbine should the main engine be shut down for any length of time, and check valves to protect the regenerator and main engine when the turbine is running direct.

The steam flow from the regenerator to the turbine is constant compared to that from the engine to the regenerator and is, of course, substantially equal to the average rate of engine exhaust. Let T be the maximum time, in minutes, of stoppage of the engine, during which the regenerator must supply the turbines alone. Let R be the total steam consumption of the turbine in pounds per minute, L the mean latent heat at the regenerator pressures (usually 0 to 4 pounds gauge), t the allowed temperature drop of the water. Then

$$w=\frac{TRL}{t} \tag{30}$$

is the weight of water required to run the turbine T minutes with an allowed fall in temperature, and consequently pressure. This difference in pressure is not over 3 pounds, at the outside.

The capacity to absorb the engine exhaust, however, determines the amount of water required, rather than the question of the turbine supply. A regenerator may be called on to absorb fluxes of steam five times as great as the more uniform flow to the turbine and they may be thrown on it almost instantly and must be cared for. If S be the maximum pounds of steam the regenerator will be called on to absorb, L the mean latent heat of the engine exhaust, and 12.4 B. T. U. the difference in temperature between 0 and 4 pounds gauge pressure, then the water required is

$$w_1=\frac{SL}{12.4} \tag{31}$$

which will usually be found greatly in excess of the w above.

Proper values of T, S, and the mean engine exhaust are determined only by careful investigation for each installation. The form of regenerator shown in Fig. 71 has shown the greatest capacity for absorbing the steam. Two regenerators of an older type installed at the Bruay Mines in France to utilize the exhaust of a hoisting engine and drive a 300 H. P. Rateau generator set have been in successful service since 1902. With supply pressures ranging from 12 to 14.5 pounds absolute, and condenser pressures from 2.13 to 2.62 pounds absolute, the plant showed a steam consumption of from 37.4 to 45.2 pounds per E. H. P. hour. A great number of regenerators and L. P. turbines are now in successful use in Europe and in this country, some of them exceeding 3000 H. P. of turbine capacity.

Where there are long periods, as for instance over night, when the exhaust steam supply is not available, a high-pressure turbine has been coupled with the low-pressure one, the low-pressure turbine being so proportioned that it can carry the entire load, but if its steam supply fails regulating valves controlled by the speed and the pressure in the regenerator, cut in the high-pressure turbine, and kept up the output. The load may be carried

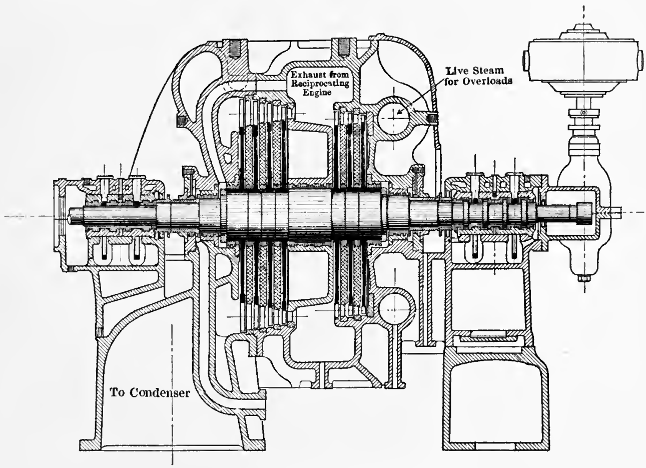


FIG. 72.—Rateau mixed-flow turbine.

entirely by either turbine, or distributed between them, as the momentary conditions require. Any exhaust steam used in the H. P. turbine passes through the L. P. one, and the group is thus working at the best economy possible. Running entirely on the H. P. machine, with steam at 110 pounds gauge pressure, one of the combined units has been found to give an economy of about 18 pounds of steam per E. H. P. hour, and running entirely on the L. P., with steam at atmospheric pressure, it gave about 36.

The two turbines are now combined in one to form a "mixed-flow" turbine as shown in Fig. 72, a much more compact arrangement.

Problems.

1. Report on the article on compounding piston engines with turbines, by Prof. Rateau, Power, October, 1907, p. 693.
2. Report on paper on Exhaust Turbines and Regenerators, by R. F. Halliwell, given in Lond. Engineering, February 5, 1909, p. 197.
3. Report on low pressure steam turbine as referred to in article by R. M. Neilson, Power, July 6, 1909, p. 1.
4. Report on article by C. H. Smoot, on Regenerators. Power, November 8, 1910., p. 1979.
5. Report on article, L. P. Turbines, by C. H. Smoot, Power, June 22, 1909, p. 1100.
6. Report on investigation into L. P. turbines for the Cambridge Elect. Light Co. by Prof. Hollis, see Power, November 10, 1908, p. 787.
7. Report on the Brush-Parsons double-flow exhaust turbine, described in Lond. Engineering, July 1, 1910, p. 2.
8. Report on English and German experience with L. P. turbines as described by Gradenwitz in Engineering Magazine, vol. xxxiv, p. 278.
9. Report on the test of engines and exhaust turbines at 59th street Interborough Station, see paper by Stott and Pigott, Trans. A. S. M. E., March, 1910, see also Power, December 14, 1909, p. 985.

References.

Stodola: "The Steam Turbine."

Moyer: "Steam Turbines."

Rateau: Paper, "Different Applications of Steam Turbines," Trans. A. S. M. E., vol. xxv.

Rateau: Article, Power, Oct., 1907, p. 693.

Burleigh: Paper on Exhaust Turbines, Power, Oct., 17, 1908, p. 828.

Halliwell: Paper on Exhaust Turbines, London Engineering, Feb. 5, 1909.

Stott and Pigott: Paper, "Test of a L. P. Turbine," Trans. A. S. M. E., 1910. See also Power, Dec. 14, 1909, p. 985.

Smoot: Articles, Power, June 22, 1909, p. 1100, and Nov. 8, 1910, p. 1979.

See also references in the problems.

CHAPTER VI.

CONDITIONS OF OPERATION.

Art. 23.—Effect of Superheat.

As with reciprocating engines, there is a gain in the economy of turbines by the use of superheated steam, but how much is gained can not be fully determined until the specific heat of superheated steam is definitely known.

While this has been the subject of repeated and extensive experiments from the time of Regnault to the present, the proper values have not been agreed upon. Grindley in England; Greissman, Lorenz and Knoblauch, Linde and Klebe in Germany, and Carpenter and Jones in America, have all done important work in this direction, but the results are contradictory and have not definitely settled how the specific heat varies with the pressure and temperature changes. Modern experiments show that the value .4805 of Regnault is too low. Apparently the specific heat increases with increase of pressure, but decreases with increase of superheat. This conclusion is indicated by Lorenz, and is sustained by Knoblauch, Linde, and Klebe. Mr. Emmett has deduced from a series of experiments made in a Curtis turbine at a pressure of 155 pounds that it varies from about .53 to .77 for temperatures of 360° to 620° Fahr. The usual practice is to take it as constant and at from .6 to .65. The bearing of the specific heat on the theoretical economy of the turbine using superheated steam is well shown by calculations made on a De Laval turbine. By taking first the water rate consumption as a basis for calculating, the gain was found to be 8.1 per cent. Then on the basis of the heat units in the steam the gain was found to be, for

Specific Heat	Gain
.48	4.8%
.6	4 %
.8	2.7%

In the diagram following are given the average of a number of steam consumptions for various degrees of superheat, up to 200°. These are based on tests of Parsons, Curtis, Zoelly, and Brown-Boveri turbines, and of such high grade engines as had records of tests under same conditions. The lines for the turbines are more reliable than for the engines, being based on a larger

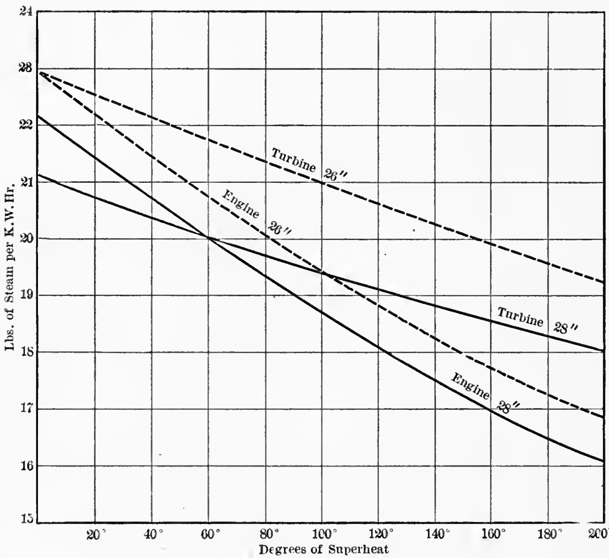


Fig. 73.—Effect of superheat.

number of tests. According to these curves the gain in economy is greater for engines than for turbines. The curves for the turbines would confirm the statement made by Mr. Parsons and by Messrs. Dean and Main that the gain in economy is about 1% for every 10° of superheat, up to about 150° or 180° Fahr. The gain from superheat in turbine practice comes chiefly from the reduced steam friction in the blades and passages. Reference to Fig. 40 shows that there may be 10% or 15% of water in the steam in the later stages. Superheating reduces the condensation and by lessening the water going through the blades, materially decreases the steam friction. It is claimed that a turbine can operate under much higher superheat than an engine. Doubtless this is true, but it remains to be shown that there is any advantage in the high superheats the turbine may be made to stand. The best practice indicates that a moderate superheat, of 50° to 100° is of advantage. Beyond this the results are not wholly favorable, for while the steam rate can be lowered it is done at the expense of extra fuel required for the high superheats.

References.

STODOLA: "The Steam Turbine."
JUDE: "Theory of the Steam Turbine."
MOYER: "Steam Turbines."
FRENCH: "Steam Turbines."
MARKS AND DAVIS: "Steam Tables and Diagrams."
Trans. of A. S. M. E., 1905 to 1910.

Art. 24.—Relative Effects of High Vacuums in Reciprocating Engines and Turbines.

The turbine gains relatively more than the steam engine as the vacuum is pushed toward its limit. In fact, this is one of the chief elements in its success in competition with the older form. The advantage in economy, under equal conditions, is usually in favor of the steam engine for vacuums lower than 26 inches or 2 pounds absolute. For higher vacuums the turbine gains rapidly. General practice has set the economical vacuum for reciprocating engines at 26 inches to 27 inches. Most of the turbine practice at the present time is based on 28 inches to 28 1/2 inches of vacuum.

The reasons why the turbine shows this greater benefit for the higher vacuums are:

First.—The low temperatures of the exhaust pressure can not reach back into the hotter parts of the machine, as in a L. P. engine cylinder, which is alternately exposed to wide differences of temperature.

Second.—If expansion is carried too far in the steam engine, the L. P. cylinder valves and passages become abnormally large, absorb power, and neutralize any thermodynamic gains. A compound engine with a cylinder ratio of 4 to 1, using steam at 150 pounds, might expand say 15 times. A turbine presents no difficulties to an expansion ratio of 100 to 110 or down to a pressure of 1 pound absolute. To carry this expansion in a compound engine calls for a cylinder ratio of 33 to 1, which is obviously prohibitive.

Since the expansion of the engine is limited to the L. P. terminal, *c*, Fig. 74, the only gain in a higher vacuum is the lowering of the back pressure, represented by the area *e*, *d*, *f*, *g*.

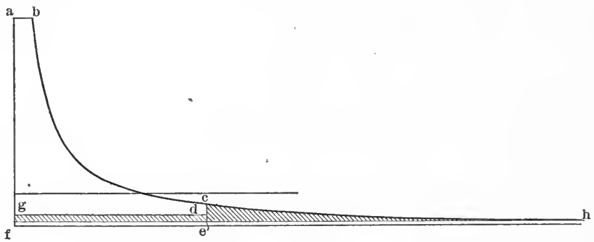


Fig. 74.—Pressure-volume curve for engines and turbines.

Since expansion can be carried down within a turbine to the lowest condenser pressures, the turbines can utilize the large triangular expansion area to *h*, in addition to all that the engine gains.

Theoretically the gain increases very rapidly for the rarer vacuums, but with the turbine, as with the engine, there is a practical limit, though not reached so soon, beyond which the cost, maintenance, and power required for the condensing apparatus offset the theoretical gains. As already stated, the limit is 28 to 28 1/2 inches, but its value for any given set of conditions is a question of engineering economics. Into it enter the following:

1. The steam consumption of
   (a) turbine,
   (b) auxiliaries.
2. The use-factor, or the ratio of the actual output to the rated output.
3. The cost of the coal, as fired.
4. Cost of injection water.

5. Fixed charges, on the condensing plant, covering
   (a) Interest on cost.
   (b) Interest on cost of masonry, space occupied, etc.
   (c) Depreciation and repairs.
   (d) Insurance, taxes, etc.

The method of determining the economical back pressure may be illustrated by an assumed case. Let

Capacity of turbine	= 1,500 K. W.
Use-factor (2)	= 15%.
Cost of coal (3)	= \$3.50 per ton, fired.
Suppose fixed charges (5)	= 10%.
Average boiler evaporation	= 7 pounds per pound of coal.

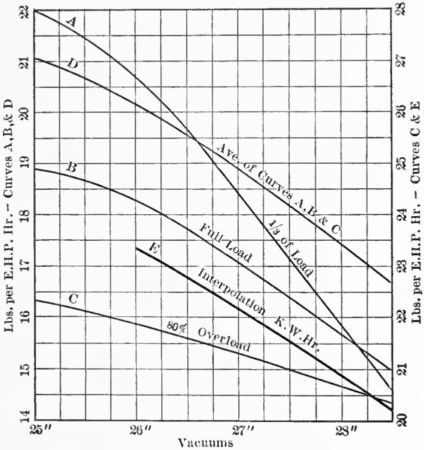


FIG. 75.—Approx. relation of water rate and vacuum.

Let the cost of condensing apparatus, based on injection water of 70° Fahr., obtained from estimates of builders for several different back pressures, be as follows: The plant for 28 1/2 inches and 29 inches would be the same as for 28 inches, except for an increase in the size of the dry vacuum pump. These vacuums are obtainable only with injection water of 60° Fahr. for 28 1/2 inches, and 45°–50° for 29 inches.

Inches of Vacuum	Cost of Condensing Plant	Increase in Cost	Increase in Annual Charge Based on 10%
26	$6000.		
26½	6900.	$900.	$90.
27	7500.	600.	60.
27½	8000.	500.	50.
28	9500.	1500.	150.
28½	10000.	500.	50.

Let the guaranteed steam consumption of the turbine be 21 pounds per K. W. hour, at 28 inches vacuum. The steam rates for varying vacuums may be interpolated from Fig. 75, which is based on some Westinghouse tests. The steam consumption of the auxiliaries may be based on their estimated H. P., and a steam consumption of 1 1/2 times that of the turbine.

The total steam consumption then would be

Vacuum Inches	Steam Consumption		
	Turbine	Auxiliaries	Total
1	2	3	4
26	23.4	3.4 %	24.2
26½	22.8	4. %	23.7
27	22.2	5.25%	23.3
27½	21.6	6.9 %	23.1
28	20.9	9. %	22.8
28½	20.2	12. %	22.6

The yearly running cost $= \frac{A\ B\ C\ D}{E} \times$ const. Where

A = K. W. capacity of the unit.
B = Total consumption (Coll. 4 last table).
C = Power factor.
D = Cost of coal, fired.
E = Average boiler evaporation per pound.

$$\text{Constant} = \frac{\text{hours per year}}{\text{pounds per ton}} = \frac{8760}{2240} = 3.92.$$

Calculating the yearly cost with the various consumptions, and tabulating, we have:

Inches of Vacuum	Cost of Steam per Year	Saving
26	440×24.2 =$10648	
26½	440×23.7 = 10428	$220.
27	440×23.3 = 10252	176.
27½	440×23.1 = 10164	88.
28	440×22.8 = 10032	132.
28½	440×22.6 = 9944	188.

Comparing these savings per year with the increase in the annual charge as the vacuum varies, we see that the gain changes to a loss at just under 28 inches. We see, too, that if an ample supply of cold injection water is available, it would pay to carry 28 1/2

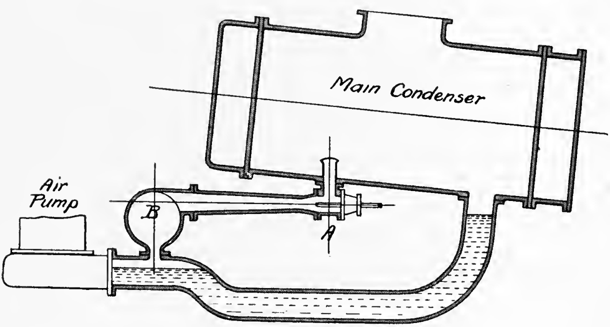


FIG. 76.—Parsons Augmentor.

inches where the temperatures will permit. This method would call for a vacuum somewhat too low, if anything, for the steam used in the auxiliaries is not all lost, as a large part of the heat in it is regained in heating the feed water.

Higher vacuums than 28 inches are carried in high-class power tations. Some condenser plants have even been installed under a guarantee to maintain a working vacuum within 3/4 inch of the barometer. Whether such a high vacuum is profitable remains to be shown by experience.

Mr. Parsons has invented a device, Fig. 76, which has been widely used in Europe, but for some reason has been little used in this country. He places the air pump, proportioned for about 26 inches vacuum, below the main condenser, and connects this by an inverted siphon, forming an air seal. Between the air pump and condenser, as a by-pass connection, is a steam ejector connection, *A*, delivering through a small surface condenser *B*, about 1/20 the size of the main one, into the air pump suction beyond the siphon. This ejector draws nearly all the residual air and vapor from the main condenser and increases the vacuum to 27 1/2 inches or 28 inches, without the increase in size of main condenser and air pump demanded by the ordinary arrangement. Mr. Parsons states that the live steam required by the ejector is about 1 1/2% of that used by the main turbine at full load, and is materially less than that required to drive the larger air pump, which would regularly be provided for the higher vacuum.

Problems.

1. Report on paper of G. I. Rockwood on "Condensers for Steam Turbines," Trans. A. S. M. E., vol. xxvi, p. 383.

References.

Stodola: "The Steam Turbine."

Jude: "Theory of the Steam Turbine."

Rockwood: Paper, "Condensers for Steam Turbines." Trans. A. S. M. E., vol. xxvi. p. 383.

CHAPTER VII.

POSITION AND FIELD OF THE STEAM TURBINE.

Art. 25.—Relative Economy of Engines and Turbines.

The heat equivalent of a H. P. hour is

$$\frac{33000+60}{777.5}=2545 \text{ B. T. U. per hour.}$$

If H_1 and H_2 are the total heats per pound of steam before and after adiabatic expansion to the exhaust pressure, and W the weight of steam per H. P. hour as tested, then

$$(H_1-H_2)\ P=\frac{2545}{W}$$

or the available heat drop $\times\%$ utilized = B. T. U. utilized per pound

$$P=\frac{2545}{W(H_1-H_2)}.$$

This "potential efficiency," P, which has been variously named by different authorities, gives a comparison between the heat actually utilized by the motor, and that available under its operating conditions. It applies equally to reciprocating engines, and to turbines, and can be used as a basis of comparison, provided the water rate can be brought to the same basis. The almost universal basis used in determining the economy of the reciprocating engine is the steam rate per indicated H. P. hour. No indicator has been devised for turbines, and it would be difficult to show the internal power developed. While some device might show the energy of the jet of steam from the nozzle, it would be practically impossible to register definitely the energy given up to the blades in a multi-stage turbine, where there are losses from eddies, friction, etc. Engineers, however, are so familiar with economies based on indicated H. P. that comparisons are often made by reducing the steam consumption of the turbine to the basis of the I. H. P. of a reciprocating engine having the same electrical output or brake H. P., as the case may be. Where it is possible there is a growing practice of basing the steam rate on the K. W. hours or electrical H. P., which requires no assumptions, especially as the bulk of turbine work outside of the marine service is in direct connected electrical units.

Comparisons based on I. H. P. must take into account the efficiency of the generators, turbines, and engines. The efficiencies of engine-driven generators for alternating currents on sizes from 500 to 5,000 K. W. will range from 87 1/2% at 1/4 load up to 96% at 1 1/4 load. For direct current generators, 100 K. W. to 3,000 K. W., it will range from 89% at 1/4 load to 94% at full load. The generator efficiency with the De Laval units is somewhat lower, as the electrical work is divided between twin generators. Their efficiency may be taken at from 88% at 1/2 load to 91% at full load, for direct current, and at 86% at 1/2 load to 92% at full load.

The following generator efficiencies are reported in various tests.

Name	Capacity K. W.	Load				
		$\frac{1}{4}$	$\frac{1}{2}$	$\frac{3}{4}$	Full	$1\frac{1}{4}$
Westinghouse	1,250	86	93	......	96	
Westinghouse	400	......	94.6	95.7	96.6	
Allis-Chalmers	5,500	......	97	97.9	98.3	98.5
Curtis	......	......	96	......	97.5	

In comparing with engines a value of 95% may fairly be assumed. The mechanical efficiency of reciprocating engines has been exhaustively investigated, and the friction losses appear to be nearly constant for all loads. The efficiency consequently falls off rapidly for light loads. A summary of various modern engines as given by French is as follows:

Mechanical Efficiency of Engines at or near Rating.

Engine	Efficiency, %	
	Engine alone	Engine and Generator
Large vertical Corliss, compound...........	96.5 to 97.5	92 to 94
Large horizontal Corliss, compound.........	94.5	90
Large horizontal Corliss, triple.............	91	
High speed, simple.........................	94	
Small and medium, hor. compound.........	87 to 93	
Large pumping engine......................	90	

The mechanical efficiency for turbines, as it involves the comparison of the brake H. P. with the indeterminate internal power developed by the steam, can only be assumed. This factor is taken from 90% to 95%. For large units the higher figure is probably nearer correct. These values give a combined efficiency of 85% to 93% for a turbine generating unit, which may be used in reducing K. W. output to an I. H. P. basis, for comparison with reciprocating plants. The following tables, from a paper by Mr. C. V. Kerr before the A. S. M. E.,[1] give means of comparing reciprocating engines and turbines on the basis of their potential efficiencies.

[1] Trans. A. S. M. E., vol. xxv., 1904.

Potential Efficiency of Steam Engines.

Type	Steam: Gauge Pressure	Steam: Temperature of Saturated Steam	Steam: Superheat at Engine	Vacuum, inches of Mercury	Steam per I.H.P. Hour	Available Heat	Potential Efficiency	Notes
	Lbs.	*F.*	*F.*		*Lbs.*	*B. T. U.*	*P. C.*	
Double vertical single acting	100	338	0	0	26.19	147.8	65.8	20 × 16 Westinghouse Standard, 257 I. H. P. Shop test.
Vertical single acting compound	120	350	0	0	21.9	159.6	72.8	14 & 24 × 14 Westinghouse Compound, 170 I. H. P. Shop test.
				25	18.1	265.5	53.0	
Vertical double acting compound	150	366	0	0	20.0	173.2	73.5	17 & 24 × 24 Westinghouse Compound. Rated 600 I. H. P. Shaft governor. Shop test.
Vertical three cylinder compound	184.6	381.5	0	25.19	12.24	286.9	72.5	43.5 & 2–73.5 × 60 Cylinders. I. H. P 5,310. Economic rating 4,900–5,500 I. H. P. New York Edison-Waterside.
	185.6	382.0	0	27.25	11.93	315.3	67.6	
Horizontal double single acting	126	353	288	0	17.7	211.3	68.0	Trials "A" and "C," by Prof. Ewing, at Sheffield, England. Schmidt System.
	110	344	0	0	29.7	168.1	51.0	
Single acting tandem compound	158	369	219	26.14	11.75	361.0	60.0	Trial by Prof. Ewing, at Knocklong, Ireland. 65 I. H. P.
Double acting compound	140	360	0	27.5	17.2	313.6	47.0	Trial by Prof. Ewing, at Middlepolder, Am. 125–184 I. H. P.
			197	27	10.4	344.1	71.0	
Single acting twin tandem compound	157.3	369	295	26.87	8.97	370.1	76.7	Trial by Prof. Lewicki on Thale Engine. 257.6 I. H. P.

Potential Efficiency of Steam Engines.—*Continued.*

Type	Steam			Vacuum, inches of Mercury	Steam per I.H.P. Hour	Available Heat	Potential Efficiency	Notes
	Gauge Pressure	Temperature of Saturated Steam	Superheat at Engine					
	Lbs.	*F.*	*F.*		*Lbs.*	*B. T. U.*	*P. C.*	
Double acting twin tandem compound	140	360	331	25.4	8.96	352.9	80.5	Official Trial. 1,045 I. H. P. Schmidt Engine, Pabianice, Poland.
Four cylinder triple expansion Sulzer engine........	184	381	0	26.25	11.57	305.7	72.0	3,000 I. H. P. Rated. See *Eng. News*, Oct. 2, 1902.
	188	382.7	223	28	8.97	392.1	72.3	
Horizontal tandem compound	132.2	356.6	0	27.9	12.06	315.7	67.0	Van den Engine. Tested by Prof. Schröter. 220 I. H. P.
	128.9	354.8	98	27.8	11.00	330.6	70.0	
	131.6	356.2	310	27.8	8.86	384.7	74.7	
Vertical triple expansion pumping engine, 780 I. H. P.	155	368	0	26	11.38	279.0	80.2	20 Mill. Gal. Snow Pump. Test by Prof. Goss, at Indianapolis Water-works.
Cross compound Corliss ...	145.1	363.2	0	25.24	13.84	279.2	65.8	Rice & Sargent Engine. Tested by Prof. Jacobus, at Millbourne Mills, Phila.
	142.4	361.8	374.5	26.79	9.56	387.1	68.8	
Cross compound Corliss......	151.3	366.2	0	28.63	12.1	338.0	62.2	Rice & Sargent Engine. Tested by Prof. Jacobus for Amer. Sugar Ref. Co., Brooklyn.

Potential Efficiency of Steam Turbines.

Type	Steam: Gauge Pres.	Steam: Temp. Sat.	Steam: Super-heat at Turbine	Vacuum, inches of Mercury	Water per H. P. hour: Indicated	Water per H. P. hour: * Brake	Water per H. P. hour: * Electrical	Available heat	Potential efficiency	Notes (Gen. Eff. =95%, Mech. Eff. =90%)
	Lbs.	*F.*	*F.*		*Lbs.*	*Lbs.*	*Lbs.*	*B. T. U.*	*P. C.*	
De La Val	206.2	390	0	26.6	13.26	* 14.73	15.5	318.0	60.4	Tests by Dean & Main on 200 K. W. Unit. Full Load.
	208.3	391	81	27.2	12.2	13.55	14.3	341.4	61.2	
Westinghouse 200 K. W. Unit	150	366	0	27	12.92	.....	* 15.2	306.9	64.2	Full Load. Shop Test.
Westinghouse 400 K. W. Unit	153	367	0	28	12.27	13.63	....	325.0	63.9	Tests by Dean & Main.
	151	366	182	28	10.13	11.25	....	360.3	69.7	Full Load.
Westinghouse 1,000 K. W. Unit	149	365	0	28	12.6	....	14.73	312.0	64.7	Full Load.
	136	358	56	28	11.2	....	13.1	321.0	70.8	Shop Test.
	154	368	140	28	10.8	....	12.66	341.0	69.2	
Westinghouse 1,250 K. W. Unit	147.1	364.3	0	27.11	12.4	....	14.52	304.1	67.5	Tests 6 & 11. Full Load. A. M. Mattice, Power. March, '04.
	146.0	363.8	78.25	28.1	11.25	....	13.17	340.3	66.5	
Westinghouse 1,500 K. W. Unit.............	148	364.8	0	27	12.65	....	14.8	301.7	66.8	Full Load.
	146	363.7	28	27.5	11.68	....	13.67	317.0	68.8	Shop Test.
Brown-Boveri Turbo-Alternator. 2,600 K. W. Unit	173	376	196	27.75	9.84	....	11.5	372.8	69.4 (+)	Frankfort. Corporation Test at Full Load. Exciter (+), or (−).
					9.5	....	11.1		71.8 (−)	
Rateau Multicellular.......	136.7	359	0	26.7	13.42	....	15.7	293.4	64.7	525 E. H. P. Full Load. A. Rateau. Eng. Mag., Oct., '03.
Curtis 600 K. W..........	140	360.7	0	28.5	12.2	....	14.25	332.9	62.7	Full Load. W. L. R. Emmet. St. Ry., Rev. 4-20-03.
			150		10.68	....	12.5	365.4	65.3	
Curtis 2,000 K. W	156.0	368.6	212	28.5	9.76	....	11.42	389.4	67.0	Eng'rs Club, Phila. 3-19-04.

* Results of tests.

Looking at the economy on a basis of steam per H. P. hour, the following table gives the results obtained from some reciprocating engines of exceptionally high economy.

Engine	Horse Power	Steam Pressure Gauge	Vacuum—Ins.	R. P. M.	Superheat—Deg. Fahr.	Steam per H. P. Hr.—Lbs.	Authority
Westinghouse Vertical Brooklyn, N. Y.	5400	185	27.3	76	.....	11.93	Eng. Record May 28, 1904.
Rockwood-Wheelock Natick, R. I.	595	159	25.4	76.4	.....	13.	F. W. Dean A. S. M. E., 1895.
McIntosh and Seymour Webster, Mass.	1076	123	27.1	99.6	20	12.76	F. W. Dean A. S. M. E., 1898.
Rice and Sargent Brooklyn, N. Y.	627	151	28.6	121	.....	12.1	D. W. Jacobus A. S. M. E., 1903.
Rice and Sargent Phila., Pa.	420	142	25.8	102	297	9.56	D. W. Jacobus A. S. M. E., 1904.
Horizontal Four-Valve	658	150.4	26.4	80	16.4	12.03	Barrus' Eng. Tests.
Leavitt Pumping Engine Chestnut Hill, Mass.	575	175.7	27.25	50.6	.....	11.2	E. F. Miller Tech. Quart'ly, vol. ix.

The following is a table of tests of 23 engines in commercial operation, and may be taken as indicating average results.

Indicated Horse Power	Steam per I. H. P. Hr.	Indicated Horse Power	Steam per I. H. P. Hr.
659.	11.89	725.	13.27
658.	12.03	1714.	13.27
670.	12.29	1030.	13.21
798.	13.28	843.	13.53
1017.	13.26	382.	14.05
689.	12.69	873.	14.18
708.	12.45	676.	14.6
636.	13.28	1540.	14.1
280.	13.37	300.	15.78
719.	13.09	606.	16.28
741.	13.23	716.	19.36
739.	13.01		

In comparison with these results are the following.[1] In reducing

[1] French. "Steam Turbines."

the given steam rates to I. H. P. the combined efficiency factors used are:

For units 300.... 400 K. W. 88%
For units 500....1500 K. W. 90%
For units 2000....3000 K. W. 92%

Where 93% is used the test was made on brake H. P. basis.

Engine efficiency alone without generator, for units from 400 to 500 H. P., or 300 to 400 K. W., taken at 93%.

Engine efficiency corresponding to the 300 H. P. De LaVal turbine, taken at 92%.

Typical Steam Consumptions of Turbines.

Turbine	Nominal Power	Steam Rate			Estimated Equiv. Consumption per I. H. P.	% Efficiency Assumed to get I. H. P. Result.
		Per Brake H. P.	Per Elect. H. P.	Per K. W.		
Saturated Steam						
De Laval	300 H. P.	15.17	.....	.....	13.96	92.
Rateau	500 H. P.	.....	14.9	.....	13.11	88.
Zoelly	500 H. P.	.....	16.05	21.5	14.12	88.
Curtis, American	500 H. P.	.....	14.76	19.78	13.28	90.
Westinghouse-Parsons	400 H. P.	13.63	.....	.....	12.68	93.
Westinghouse-Parsons	1250 H. P.	.....	14.13	18.95	12.72	90.
Moderate Superheat Up to 150°						
De Laval	300 H. P.	13.94	.....	.....	12.82	92.
Zoelly	500 H. P.	.....	14.05	18.82	12.36	88.
Curtis, English	500 K. W.	.....	15.29	20.50	13.76	90.
Curtis, American	500 K. W.	.....	13.28	17.79	11.95	90.
Parsons	300 K. W.	.....	14.96	20.06	13.16	88.
Parsons	1500 K. W.	.....	13.44	18.	12.10	90.
Parsons	3000 K. W.	.....	11.79	15.8	10.85	92.
Westinghouse-Parsons	400 K. W.	12.07	.....	.....	11.23	93.
Westinghouse-Parsons	1250 K. W.	.....	13.78	18.48	12.40	93.

Typical Steam Consumptions of Turbines.—*Continued.*

High Superheat
180° to 290°

Curtis, American......	500 K. W.	.....	11.26	15.1	10.14	90.
Curtis, American......	2000 K. W.	.....	11.27	15.12	10.36	92.
Parsons..............	3000 K. W.	.....	11.	14.74	10.12	92.
Westinghouse-Parsons .	400 K. W.	11.17	.....	.....	10.39	93.

In comparison with the last test in the table, with the high super heat, may be taken the tests of a Van der Kerchove engine, made by Prof. Schroeter.

I. H. P.	Superheat in Deg. Fahr.	Steam per I. H. P. Hr. Lbs.	Equivalent Pounds of Saturated Stm.	Heat Units per I. H. P. Hr.
222.	0.	12.06	12.08	250.
226.	43.7	11.58	11.77	244.
227.	97.7	11.	11.44	237.
223.	151.7	10.67	11.33	234.
223.	221.2	9.81	10.69	221.
218.	310.9	8.86	10.01	207.

The Rice and Sargent engine, in table, p. 122, is perhaps a still better comparison.

The following list of tests of engines and turbines may be taken as representing the comparative economy obtained by each type to January, 1911. The B. T. U. per K. W. minute are obtained by multiplying the steam rate per K. W. minute by the difference between the B. T. U. per pound of steam at the initial conditions of pressure and superheat and the B. T. U. in 1 pound of water at the vacuum pressure, col. 4. The last column gives the best basis for comparison.

	1	2	3	4	5	6
	Load K. W.	Steam Press. Abs.	Super-heat or Mois-ture	Vacu-um	Lbs Steam per K. W. Hour	B.T.U. per K. W. Min.
Turbines						
N. Y. Edison, Westinghouse......	9870	192	97°F	27.31	15	294
City Electric, Westinghouse......	9173	182	59°	27.90	14.57	282.4
Bklyn Rpd. Transit, Westing-house.	11466	192	106°	28.15	14.45	287.5
Chgo. Edison, Curtis.............	10186	191	147°	29.47	12.9	270
Carville, Parsons................	5059.4	209	95°	29.27	13.35	271.7
Frankfort, Brown-Bovari.........	3521.6	157	130°	28.90	13.70	278
Turin, Escher-Wyss.............	3540	187	90°	28.35	15	297.4
Berlin Moabit., A. E. G..........	3169	185	215°	29	12.7	268.4
Rummelsburg, A. E. G.,.........	4239	188	285°	29	11.9	258.5
Engines						
N. Y. Edison, Westinghouse......	3872	200	dry	27	16.78	312
Interborough, 59th St., Allis[1] ..	5496	190	dry	25.02	17.82	225
Boston Edison, McIntosh.......	1500	176.5	92°	25.4	16.47	315
Berlin Moabit....................	1920	202.7	223	28	13.35	276
Berlin Luisenstrasse.............	1950	202.7	264	27	14	295

[1] Without L. P. Exhaust Turbines, see Art. 22.

The results brought out in the foregoing indicate

1. That with high vacuums the larger turbines show economies somewhat better than reciprocating engines.

2. That with moderate vacuum, in the medium sizes, the advantage in economy still lies with the refined types of reciprocating engine.

Problems.

1. Look up and report on methods of testing steam turbines. Paper by Dickinson and Robinson, Am. Inst. of Elect. Eng'rs, December, 1910.
2. Report on "Test of a 9000 K. W. Turbo-generator Set." Paper by F. H. Varney, Trans. A. S. M. E., vol. xxxii, December, 1910.
3. Report on chapters on testing of turbines in Moyer's "Steam Turbines."

References.

Stodola: "The Steam Turbine."
Moyer: "Steam Turbines."
Jude: "Theory of the Steam Turbine."
French: "Steam Turbines."

NEILSON: "The Steam Turbine."
THOMAS: "Steam Turbines."
KERR: Paper, Trans. A. S. M. E., vol. xxv.

Art. 26.—General Comparison of Engines and Turbines.

Other considerations, however, enter such as steadiness and economy under widely fluctuating working loads, the room required, the foundations, cost of installation and operation.

The governors of reciprocating engines regulate the effort from stroke to stroke to meet varying load conditions, but fluctuations due to the effect of the connecting rod and the weight of the reciprocating parts can be cared for only by the flywheel. The flywheel can at best only reduce these fluctuations to within prescribed limits. In direct connected generating units, especially when running in parallel, the turbine has the advantage of a total absence of reciprocating motion and a uniform turning effort, the high speed of the rotor and heavy armature acting as a powerful regulating force. In fact it has been found possible with turbines to run railway, power, and lighting circuits from the same machine, and where a turbine has been installed to run in parallel with piston engines it has always been found to have a steadying effect on the whole system.

But the turbine is a one-speed machine. In this respect it is distinctly inferior to the reciprocating engine, for it cannot be run at speeds lower than that for which it is designed without serious effect on its economy. This is one of the limitations to its use in marine work, and has proved one of the difficult problems in that connection. Even turbine advocates do not recommend them for intermittent services and varying speeds. The field available for the marine turbine seems to be deep sea service, where the work consists of long runs at uniform speed. Tests made on a Curtis turbine for variation in steam consumption in relation to the speed resulted as follows:

R. P. M.	Steam per K. W. hr.
1900	19.75
1600	20.74
1300	22.7
1000	27.

In the torpedo-boat destroyers built by the British Navy for comparing turbines and reciprocating engines, the turbine boats were superior in economy to the engine-driven boats at normal speeds, but invariably inferior when running at slow cruising speeds.

In good practice reciprocating engines are not installed to run under frequent overloads greater than 50% of the normal capacity. Corliss engines can be made to carry overloads of 100%, but not with good economy. The tendency is to provide an engine large enough to handle the maximum loads easily, and let it operate at an average load considerably below its rated capacity. Prof. Carpenter has reported tests of thirty-five street railway power plants. In these this tendency is clearly marked. The following table gives a summary of seven of these tests made on compound condensing engines of the Corliss or similar types. The remaining ones not given show the same tendency.

H. P. of Engine	Mean observed H. P.	% of Capacity	Steam per I. H. P. Hour.
825	482	58.2	22.7
1,000	277	27.7	21.9
1,000	314	31.4	20.0
350	182	52.2	16.64
500	290	58.	16.90
2,000	814	40.7	14.50
200	145	72.	17.30
Average	....	48.6	18.6

The average load on these engines was less than half their rated capacity, and they were operated at an average steam rate of over 18 pounds instead of 14 and 15 pounds, which the type of engine is capable of.

As turbines have been rated they have been capable of operating under heavy overloads and at very fair steam rates. Guarantees were regularly made on the rates for overloads of 25%, 50%, and 100%.

As a result turbines have been installed more with reference to the average load to be carried, and less with reference to the maximum load than a steam engine, the turbine operating at nearly its best economy most of the time, whereas an engine, rated nearer the maximum load, would be operating at less than its most efficient rating.

The following table shows the variations in consumption reported for a Westinghouse turbine on loads varying from 1/2 load to 100% overload.

Load	Steam Rate Saturated	Steam Rate 100° Superheat
1/2 load	15.86	14.34
3/4 load	15.05	13.45
Full load	13.89	12.48
1 1/4 load	13.85	12.41
1 1/2 load	....	12.79
100% overload	15.12	13.55

Here an overload of 100% was carried with an increase of only 10% in the steam rate. Such a load could not be carried indefinitely with safety to the generator, but overloads of 50% are often met.

It should be added however, that turbine manufacturers are rating their machines higher now and this difference is becoming less marked.

In respect to the room occupied, the advantage in favor of the turbine alone is very marked. If the condensing apparatus be included in the comparison the difference is not so great, but even then it is usually in favor of the turbine. This is brought out clearly in Fig. 77, which shows a 3,500 K. W. engine-driven unit,[1] and two turbine-driven units, one 5000 K. W. and behind one of 7500 K. W. in the Waterside Station of the N. Y. Edison Co. Four of the engine-driven units similar to that shown in the picture are being replaced by three turbine units, each of 20,000 K. W. capacity.

A comparison of these turbine units with the reciprocating engine shows the equally marked difference in the foundations.

The selling price of turbines to-day is governed largely by that

[1] The engine of this unit is really capable of turning out 6000 H. P., although the generator is rated at 3500 K. W.

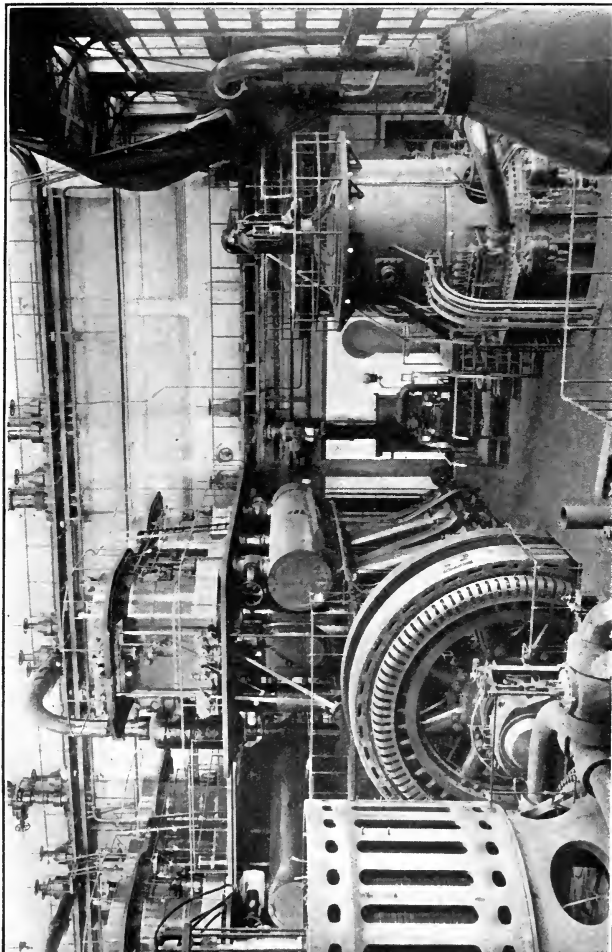


Fig. 77.—3500 K. W. Engine driven unit and two turbine units of 5000 K. W. and 7500 K. W. capacity. N. Y. Edison Co., Waterside Station.

of the competing engines. The manufacturing cost is probably less, but as there are few turbines in the market and comparatively little competition as yet between them, the competition has been rather with the old type engines. Any estimate of cost, however, must include the land, buildings, and foundations. These elements vary so with different conditions that no general results can be given. The general adoption of the turbine shows that for large units at least it is the cheaper, and it has practically driven out its rival for large power station work.

Mr. Stott, Supt. of Motive Power of the Interborough R. T. Co., New York City, in a paper before the A. I. E. E., makes the following comparison of the operating cost of large engine-driven and turbine-driven units. He says: "The turbine shows more uniform steam rate for varying loads, practically as good economy with saturated steam, and a thermal economy of 6.6% better for superheated steam." The various items are derived from actual costs.

Comparison of Charges per K. W. Hr.

Maintenance	Comp. Cond. Engine 7500 H. P.	Turbine 5000 H. P.
Engine room, mechanical	2.57	.51
Boiler room	4.61	4.30
Coal and ash handling apparatus	.58	.54
Electrical apparatus	1.12	1.12
Operation		
Coal and ash handling labor	2.26	2.11
Removal of ashes	1.06	.94
Dock rental	.74	.74
Boiler room, labor	7.15	6.68
Boiler room, oil, waste, etc.	.17	.17
Coal	61.30	57.30
Water	7.14	.71
Engine room, mechanical labor	6.71	1.35
Lubrication	1.77	.35
Waste, etc	.30	.30
Electrical labor	2.52	2.52
Relative cost of maintenance and operation	100.	79.64
Relative investment in per cent	100.	82.52

The general statement that the turbine requires much less oil is borne out in this table, where the turbine requires but 20% of that for the engine generator. In a turbine oil is used only in the bearings, free from contact with the steam, so that the steam is returnable to the boilers as soon as condensed, a property of great value in places where feed water is expensive.

Problems.

1. Report on the power of plant of Public Service Corp'n. of N. J., described in Power, November 23, 1909, p. 853.

References.

Stevens and Hobart: "Steam Turbine Engineering."
Moyer: "Steam Turbines."
French: "Steam Turbines."
Murray: "Electric Power Plants."
Waldron: Paper, "The Steam Turbine from an Operating Standpoint." Trans. A. S. M. E., vol. xxiv.

Art. 27.—Relative Fields of Engines and Turbines.

The reciprocating engine is superior to the turbine in starting power, in reversing, in operating at varying speeds on intermittent and irregular service. It requires less condensing apparatus and gives a higher economy for saturated steam at average vacuums. It can run at moderate speeds and will therefore have a monopoly of many services where the high speeds of the turbine are prohibitive. These considerations indicate that the steam engine will hold its place in locomotive work, hoisting, rolling mills, lighter marine work, and belt and rope driving. Turbines are being applied to centrifugal air compressers and to pumps, with considerable success, but it will be some time before they will be a serious rival in this field or in blowing engines. If the steam engine encounters serious competition in the smaller sizes for miscellaneous work it seems much more likely that it will come from the gas engine than from the turbine.

The turbine, on the other hand, has demonstrated that it, too, has its field. For direct connected services such as for generators and marine work, especially in the larger units, it has proven superior to the reciprocating engine. In central power station service it has practically superseded the old type of engine.

Up to February, 1911, the sales of the three foremost manufacturers of large turbines in the United States were as follows:

General Electric Co.	2,150,000 K. W.
Westinghouse Machine Co.	1,600,000 K. W.
Allis-Chalmers Co.	350,000 K. W.
Total	4,100,000 K. W.

When it is realized that scarcely a dozen of these machines are 9 years old, and the vast majority not more than 5, one can see the hold the turbine has obtained on this field. About 70 or 75% of the turbines built are for electric light, power, and traction companies.

Here the turbine is at its best. It is practically immune from danger from priming. Its uniform velocity and close regulation, its high economy under widely varying loads, its compactness and inexpensive foundations, and its exhaust free from oil, together with the great saving in the generators due to the high speeds, have made it distinctively the central station machine for the generation of alternating current power.

In marine service the turbine has also been successful, although its ascendancy here is not so marked as in power station work. Mr. C. A. Parsons has lead the way into this field, and it is largely to his indomitable will that the successful application of the turbine to marine service is due. When he first began his experiments, propeller shafts ran at from 80 to 120 R. P. M., and the slowest turbines at 3,000 to 5,000 R. P. M. The "Turbinia," the first experimental vessel, was equipped with 2,000 H. P. turbines designed for 3,000 R. P. M. It was found that at the designed speed of rotation only 18 knots could be obtained. Experimental investigation developed the existence of "cavitation," a cylindrical vacuum formed around the propeller, causing great loss of power.

After clever and extensive experiments propellers were developed on new lines, the power was divided up between 3 turbines driving 3 shafts, the revolutions reduced about 1/2, and a combination of turbine and propeller obtained which gave remarkable results. A speed of over 34 knots, never before reached by any vessel, was obtained.

From that time, 1894, the development has been steady. From the Turbinia, with its 100 feet of length, 9 feet beam and 44 1/2 tons displacement, the turbine has been successfully applied to destroyers, small passenger steamers on the Clyde, the Channel, and Irish Sea services, to the Allan liners, the largest Cunard liners and to the heaviest battleships. The Mauritania and Lusitania have maintained average speeds of over 25 knots, and developed over 70,000 H. P., 70% more power than has ever before been concentrated in one set of engines.

The main advantages of the turbine in the marine service are:

1. Absence of engine vibration.
2. Lower center of gravity with consequent greater stability.
3. Less weight of machinery with consequent greater carrying power.
4. Higher average economy.
5. Smaller propellers—and therefore, from this and item three, less draft.
6. Lower cost of operation—*i.e.*, attendance, supplies, etc.

Mr. Parsons, in a paper before the Institute of Marine Engineers, speaking of the future of the marine turbine, says:

"I think we are safe in predicting that it will soon supersede entirely the reciprocating engine in vessels of 16 knots sea-speed and upward, and of over 5,000 I. H. P., and probably also including vessels of speed down to 13 knots, of 20,000 tons and upward, and possibly still slower vessels in course of time. At present it may, I think, be said that the above most suitable field comprises about 1/5 of the total steam tonnage of the world; but it must be remembered that the speed of ships tends to increase, and the turbine to improve, and so the class of ships suitable for the turbine will increase.

Much interest and attention are being centered on the combination of reciprocating engines and turbines for marine work as in the Olympic. The advantages to this combination have been considered in Art. 22, but it seems especially adapted for ships, giving high economy and a greater flexibility than turbines alone.[1]

[1] "Combination System of Reciprocating Engines and Steam Turbines," paper by C. A. Parsons, Institution of Naval Architects., Lond., April 9, 1908. Reprinted in London Engineering, April 17, 1908.

See also articles on the Engine of the White Star SS "Olympic," London Engineering, Oct. 11, 1910.