STEAM TURBINES:
THEIR DESIGN AND CONSTRUCTION.

This work gives the most recent results in practice, and a succinct account of the latest types of Steam Turbines. It contains exhaustive comparisons between steam turbine and piston economics, and from the results rational conclusions are formulated as to the respective cases in which these two types of prime movers should be employed. Weights, costs, speeds, number of wheels and blades and other leading data of the various types are contrasted in tabular form. More complete details than have been published before of many of the most recent steam turbine Generating Stations on both sides of the Atlantic are included, with a unique series of comparative tables. Marine Steam Turbines are similarly dealt with, and fuller information than has before appeared is tabulated in form for the readiest reference.


"Is one of the latest and best descriptive books on the subject. . . . A work which will remain of permanent value because it records the position and prospects of steam turbine design and manufacture more thoroughly than any book yet available."—Sir William White in the Times Engineering Supplement.

"Surpasses by a long way all other books on this subject in its utility to engineers. . . . It is practically a complete compendium of the facts of existing turbine practice. Its production has been guided by a thorough scientific knowledge of the most recent developments in other countries as well as our own. Every kind of information will be found in it."—Engineer.

"The authors have succeeded at infinite labour in collecting data and in placing before engineers what the steam turbine has accomplished up to the present. . . . No test of value has escaped the scrutiny of the authors."—Electrician.

WHITTAKER & CO., LONDON, E.C.
STEAM TURBINES:
THEIR DESIGN AND CONSTRUCTION.

BY

RANKIN KENNEDY,

AUTHOR OF
"MODERN ENGINES AND PRIME MOVERS," "ELECTRICAL INSTALLATIONS,"
"FLYING MACHINES," "ELECTRICAL DISTRIBUTION," ETC., ETC.

WITH 62 ILLUSTRATIONS.

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PREFACE.

This small work is designed for the use of Engineers and Students who desire to obtain an insight into the methods whereby the principal dimensions of Steam Turbines are calculated, referring to first principles. Formulae are given for calculating stages and numbers of wheels and dimensions with worked examples.

The leading features of Turbine construction are shown and fully illustrated, the series arrangement of wheels and its effect upon the velocities of the steam and the Turbine blades and economy of steam are fully considered. Labyrinthine steam packing and high vacua condensers are shown as essential to high economy.

The endeavour of the author has been to provide a simple system of introduction to the design and construction of the Steam Turbine, leaving out the many more intricate problems, which are fully treated in more advanced and larger works.

RANKIN KENNEDY.

Glasgow, 1910.
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Turbine Principles of Calculation

The principles of steam turbines are similar to those of the water turbine in many respects. The principal effects in the conversion of the energy in the steam are due to the velocity given to the steam by the heat energy multiplied by its weight, momentum being the form of the energy or kinetic energy. Some turbines may be called pressure turbines, but no really pure pressure turbine is in use in practice. A pressure turbine should be actuated by pressure only, the same as a piston is acted upon by steam pressure.

Most turbines are actuated by the momentum of the fluid giving a pressure on the turbine blade equal to \( \frac{WV}{g} \), where \( W \) is the weight of fluid passing per second, \( V \) its velocity in feet per second, and \( g = 32 \). And the effect of the momentum upon a moving blade is called the impulse. And we have many examples of purely impulse, or, as some call them, action turbines. Other turbines actuated by pressure are called reaction turbines. The largest class of turbines are actuated by pressure and impulse together.

A turbine acting purely by pressure alone could be designed without reference to the fluid velocities. For instance, the Barker's Mill or Hero type of turbine, if it were actuated by the pressure of the fluid alone, would give a thrust driving the turbine, equal to the pressure multiplied by the area of the nozzles, and that only, and that pressure would be independent of the velocity of the
orifice. But as a matter of fact, the pressure is just the same as if it were an impulse turbine, and, if the wheel is fixed, the thrust is equal to twice the pressure multiplied by the area of the nozzle, just the same as that for an impulse turbine, and yet this turbine is the nearest approach to a purely pressure turbine. The proper division of turbines is into three classes: action turbines, reaction turbines, and combined action and reaction turbines.

The De Laval turbine is a good example of the first class; the Barker's Mill or Hero turbine, a good example of the second; and the Vortex water turbine and the Parsons' steam turbine are good examples of the third class.

As we deal with the weight per lb., the volume and temperatures of steam, and the relative volumes of steam and water very frequently in this work, a table is here given for reference. If the figures in the last column are multiplied by 2.3, a factor $R_v$ will be found for use in velocity formulæ employed by the author.

**TABLE I**

**Properties of Saturated Steam**

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<th>Absolute pressure in lbs. per sq. in.</th>
<th>Temperature in degrees, Fahrenheit</th>
<th>Total heat in B.T. Units from steam at 32° F.</th>
<th>Latent heat in B.T. Units.</th>
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### THEORETICAL, MECHANICAL AND PHYSICAL PROBLEMS.

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It may be as well to go over the first principles of turbines and their primary construction. Sometimes turbines are called impulse and sometimes pressure turbines, or action turbines and reaction turbines. There is a distinction between them, but it is not fundamental, and great importance need not be attached to their difference. The earliest and simplest and the only purely pressure or reaction turbine known, is the Barker's Mill or Hero type of turbine.

In describing the action of fluid streams on blades, or

Fig. 1.—Fluid pressure jet on plate.
wheel blades, but nowhere should the stream be broken up. All its movements must be smooth and gradual, hence the form of plate Fig. 2. Properly it should, for a cylindrical jet, be a figure of revolution, shown by the conical plate revolved on axis $a-b$. If the plate is fixed, the fluid spreads gradually down the cone and is deflected aside at right angles to its first path, and it exerts a pressure $P$ on the plate, which has for its value

$$P = \frac{W V a}{g} \quad \ldots \quad (1)$$

velocity of jet $V a$=feet per second. Wherever we have mutual action and resulting motion at a velocity, $V$, the thrust or pressure produced by the action or reaction is proportional to $W$, the weight of the body acting, and $V$ the velocity of its motion, and by dividing the product of $W \times V$ by $g$, the thrust in lbs., when $W$ is in lbs. and $V$ in feet per second, is found, and the energy of the reaction is equal to

$$\frac{W V^2}{2g} \quad \ldots \quad (2)$$

The student of the steam turbine must first of all know what is $W$ and $V$ in turbine problems. The simplest element of a steam turbine is the stator, in the shape of a nozzle delivering steam, under a pressure inside the nozzle of $P$, and a pressure outside of $P_1$, $dP$, the pressure

![Fig. 2.—Jet on conical plate.](image-url)
difference being \( P - P_1 \); \( V_s \) is the velocity of the steam, and \( V_t \) the velocity of the blade or bucket upon which the steam acts or reacts. And there is another velocity of the steam which is of importance, and that is the velocity with which it leaves the blade or turbine bucket, \( V_e \), the velocity of exit.

When a jet strikes a flat fixed plate like Fig. 1, it is broken up, and most of the force wasted in turbulent effects, but when it is thrown against a fixed blade or bucket having a figure of revolution round axis \( a, b \) (Fig. 2), that is a plate with a rising cone, the fluid is gradually divided, and it slides smoothly down the cone until it glances off at right angles in all directions.

![Fig. 3.—Hero reaction wheel.](image)

In such a case the thrust \( T \) of the jet against the plate \( T = \frac{WV}{g} \). If the weight of the water or steam equals 6.4 lbs. and the velocity (\( V = \) velocity of water or steam) is 100 feet per second, then \( T = \frac{6.4 \times 100}{32} = 20 \) lbs. If the fluid issues from the jet and the jet is free to move round a centre from which it gets its steam, as in Fig. 3 (shown with two jets to make the balance true, each jet being half the area of the one shown in Fig. 2), we may calculate as if there were only one of same area as in Fig. 2. The law of action and reaction being opposite and equal, the jet arms will be forced back by a reaction equal to \( \frac{WV}{g} = \frac{6.4 \times 100}{32} = 20 \) lbs. when the arms are fixed, so that by employing a movable jet or jets we could get a turbine without blades, and some are so made.

In both these cases (Figs. 2 and 3) the action and reaction are obviously opposite and equal, but in this first instance the action is on the blade on which there is no reaction, tending to drive it in the same direction as the jet (observe this point carefully). Reaction is due to the fluid leaving
the blade or bucket, hence we have purely reaction when
the jet is made the movable member, and pure action when
the plate is made the movable or driven member. In these
cases it is usual to call the fixed member the stator, and
the driven one the rotor.

The reaction which takes place on the conical bucket is
at right angles to the motion of the bucket blade and jet,
and is equal and opposite in all directions from the centre,
and hence has no pressure effect on the blade, it is lost.
And the pressure is equal to a column of the fluid whose
base is the area of the jet, and its height double the height
due to the velocity, hence if, in the movable jet case, the
pressure inside was 100 lbs. above that outside per
square inch, and the jet area at its smallest part was
0.1 square inch in sectional area, the thrust would be
\[ T = 2 \times 0.1 \times 100 = 20 \text{ lbs.} \]

The whole energy of the jet being equal to \( \frac{WV^2}{2g} \) its
whole thrust possible is four times that due to the height
or head to which \( V \) is due, hence we get only half the
possible thrust from a jet or blade in which the fluid jet
is deflected through only one right angle, and acts only by
action alone.

Hence it is desirable to form blades in which we can
get all the thrust possible, and this can be easily done by
making a blade in which the fluid is compelled to action
and reaction. Fig. 4 shows this blade single-sided, and
Fig. 5 double-sided. Now it will be seen that for one half
of its course after entry into the bucket or blade, it is
acting purely by action as it did in the first case (Fig. 2),
turning through one right angle, but during the other half
of its course it is acting by reaction purely, as it did in the
nozzle when it was made movable, therefore we get
one half the energy exerted as action and the other half
as reaction in one blade, and twice the pressure which was
obtained in the Figs. 2 and 4, and

\[ T = \frac{2WV}{g} = \frac{2 \times 6.4 \times 100}{32} = 40 \text{ lbs.} \]

And even when the angle is not exactly two right angles,
the pressure is nearly equal to that given above, even
when the two angles are 45°, that is, 45° to the line of
motion, as in Fig. 6, it is only 17 per cent. less than when
at two angles of 90°. It must be noted that it is quite
a different thing to deflect the fluid through one angle of 90, as in the Fig. 4, and to deflect it through \textit{two} angles of 45° to the line of motion of the blade (see Fig. 5). In most steam turbines the actual effect of the blades employed as guide blades and wheel blades give in effect pressures practically equal to that as if the steam were turned through two right angles, and as such they may be calculated in the designs.

**MOVING BLADES**

Hitherto the blades have been considered at rest, in which condition there is a maximum of thrust, either by action or reaction, or both, but as there is no motion there is no power produced. Again, if a cup, or blade, or plate, were placed in front of the flowing stream and allowed
THEORETICAL, MECHANICAL AND PHYSICAL PROBLEMS.

to move with the stream, unresisted, there would be a maximum of motion, \( V_s \) and \( V_t \) being equal, the pressure or thrust would be a minimum, and no power obtained. Between these two extremes there is a velocity \( = V_t = \frac{1}{2} V_s \), at which the pressure multiplied by velocity is a maximum.

In order to calculate the thrust on the blade in motion,

\[
T = \frac{d \cdot a \cdot V_s (V_s - V_t)}{32}
\]  

for a blade which turns the fluid through one right angle.
(Figs. 2 and 4). In proof of above formula, \( V_s - V_t = V \), the actual velocity with which the fluid enters the blade. The blade and the fluid are moving in the same direction; if both moved at same velocity \( V \) would equal nothing. If the turbine blade were fixed, the steam would enter at full velocity \( V_s \), but the work done would be nothing, as there is no velocity of the blade. Generally the best practical result is obtained when \( V_t = \frac{1}{2} V_s \).

The student is supposed to have an elementary knowledge of theoretical mechanics, from which he will know that the reaction thrust of a fluid is \( T = \frac{W V}{g} \), hence, as \( d a V_s \) equals \( W \), and \( V_s - V_t = V \),

\[
T = \frac{d a V_s(V_s - V_t)}{g}.
\]

Hence, suppose a case in which \( V_s = 150, V_t = 50 \), and \( d a_s = 0.5 \). Then by (3),

\[
T = \frac{d a V_s(V_s - V_t)}{g} = \frac{0.5 \times 150 \times 100}{32} = 235.
\]

For blades or buckets of type shown in Fig. 6, wherein the fluid is deflected through two right angles or nearly so, the formula above is the same, but multiplied by 2.

\[
T = \frac{2 \cdot d a_s V_s(V_s - V_t)}{g} . \tag{4}
\]

In the Parsons' turbine the guide blades and moving blades are almost similar. They are shown in diagram (Fig. 7).
The steam is thrown at an acute angle against the curved entrance to the moving blades, where it acts by impulse or action; it is then turned through a large angle, and reacts on the blades at \( a, a, a \), as it leaves the moving wheel. It may be safely calculated that the thrust \( T \) is equal to \( \frac{2WV}{g} \) in this turbine.

In the Rateau, Zoelly, and other turbines, the steam acts upon the blades by momentum without reaction, and each ring of moving blades is in a separate compartment, and the guide rings are partial flows, at the high-pressure end; the angle of entry is about 20° and leaving about the same, so that the fluid is turned twice through 90—20 or 70°, or 140° altogether.

The formulae is then—

\[
\text{Thrust} = T = \frac{daV_s(V_s - V_t) \times 1 + \cos \theta}{g} \quad (6)
\]

\( \theta \) being the angle made by the entering plus the angle made by the leaving fluid stream. Referring to Fig. 6, the whole angle is \( a,d=90 \); the entering angle = \( abc=45 \), and the leaving angle \( =cbd=45 \), hence \( \theta=90 \).

The term kinetic energy, \( E \), in mechanics means capacity for performing work, in a mass moving with a definite velocity \( V \). A jet of fluid flowing at a rate of \( W \) lbs. per second with a velocity, \( V \), exerts a thrust, \( T \), against a turbine blade which moves the blade against a resistance, \( R \), with a velocity \( V_t \). The kinetic energy delivered is equal to \( T \times V_t \). \( R \) is equal to \( T \), hence \( E \) is also equal to \( R \times V_t \); hence, if either \( R \) or \( T \) is known and \( V_t \) measured, \( E \) can be found. And when \( R \) or \( T \) is in lbs. and \( V_t \) in feet per second, the result is in ft.-lbs. per second. And as there are 550 ft.-lbs. per second to the horsepower (H.P.),

\[
\text{H.P.} = \frac{TV_t}{550}.
\]

The kinetic energy, \( E \), delivered to the moving blade, is found by multiplying the thrust \( T \) by \( V_t \) the blade velocity.

\[
E = T \times V_t = \text{ft.-lbs. per second} \quad (7)
\]

If \( T = 235 \) and \( V_t = 50 \)

\[
E = 235 \times 50 = 11750 \text{ ft.-lbs.}
\]
STEAM TURBINES.

It is also found by taking the fundamental formula for energy, \( E = \frac{WV^2}{2g} \), which when used for a moving blade, whose motion is due to a moving jet of fluid, is,

\[
E = \frac{W(V_s^2 - V_e^2)}{2g} \quad (8)
\]

\( V_s \) being the fluid velocity just before it enters the blade, and \( V_e \) its velocity when it leaves the blade.

\( E \) also equals

\[
E = \frac{W \times V_t(V_s - V_t)}{32}
\]

wherein \( W = d\alpha V_s \), and hence, if \( d\alpha = 0.5 \),

\[
V_s = 150, \quad V_t = 50.
\]

\[
E = \frac{0.5 \times 150 \times 50 \times 100}{32} = 11750 \text{ ft.-lbs.}
\]

This for a conical or other blade which deflects the fluid through one right angle.

For blades which deflect through two right angles, or nearly so, the value is double the above (see Fig. 6).

Turbines in which the blades are set with two different angles, a wide angle to the line of motion at entry and an acute angle at exit, are most common, because the wide angle at entry is best suited for action or impulse, and the acute angle at exit best for reaction or pressure. This may be noted in Parsons' turbines, also in Prof. James Thomson's Vortex turbine, and others.

Turbines constructed strictly according to theory, \( i.e., \) that the fluid should be deflected twice through an angle of 90, are not common among steam turbines, the Pelton wheel being the nearest approach to theory in this respect (see Fig. 8).
The blades of a turbine should have a velocity $V_t$ equal to one-half that of the steam velocity $V_s$; this, however, is seldom reached in practice. The device of putting turbine wheels in series is resorted to in order to reduce the velocity $V_s$ by stages, wherein the pressure is let down by small differences.

The maximum velocity of steam is reached when it is expanded in a divergent conical nozzle from a high pressure to a very low pressure, say 0.5 lbs. absolute. This is an exceptionally low pressure, but is taken to show the immense velocities attainable. In falling from 185 lbs. pressure to 0.5, the steam will give up heat energy equal to 382 British Thermal Units equal to kinetic energy $382 \times 778$ ft.-lbs., hence $V_s = V_s$, then $V_s = \sqrt{382 \times 778}$. per lb.

\[ V = \sqrt{382 \times 778 \times 2g} = 4370 \text{ ft. per second}, \]

a velocity far beyond any practicable turbine blade velocity. Such a velocity can be obtained only by allowing the steam to expand in a conical nozzle from a pressure $P$ to a lower pressure $P_1$ nearly zero.

Steam flowing from an orifice under a pressure $P$ into a pressure $P_1$ reaches a maximum weight flow, when $P_1$ is less than $0.6 \times P$; but when $P_1$ is greater than $0.6 \times P$, the velocity is found as follows:

If we find the quantity of heat energy available, $E$, in the steam falling from pressure $P$ to $P_1$, then,

\[ V_s = 8 \sqrt{E} \]

hence,

\[ V_s = 8 \sqrt{\text{B.Th.U.} \times 778} \]

And as we may once for all reduce $8 \sqrt{778}$ to $= 223$, the velocity is found by

\[ V_s = 223 \sqrt{\text{B.Th.Unit}} \]

(that is the B.Th.U. in the fall of pressure). And if the velocity is known, then,

\[ \text{B.Th.U.} = \frac{V_s}{223} \]

A thermal velocity curve is given in Fig. 9 to show the values at a glance for any ordinary velocities.

Now this holds good for small differences of pressure with ordinary converging nozzles or guide blades.
The curve (Fig. 9), it will be seen, tends to become flat, indicating that a maximum velocity is soon reached, beyond which no additional pressure or thermal units can increase the velocity through the narrowest part of the nozzle. But if an expanding nozzle is employed to allow the stream to expand after it passes the throat, then a further increase of velocity is given to the steam in proportion to its remaining thermal energy. This is true whatever the difference of pressure, for steam will expand in passing from a higher to a lower pressure however small the difference; but at small differences the increase in velocity due to an expanding nozzle is not enough to warrant the expense in construction entailed by their use. As a matter of fact, the expansion and increased velocity due to it occurs in the clearance spaces between the rings of guide blades and rings of wheel blades in the Parsons' type of turbine, and hence it is not necessary to have the space between the pairs so small as it was at one time considered necessary.

When, however, \( P_1 \) is less than \( 0.6 \times P \), an expanding
nozzle is absolutely necessary at common working pressures, say 180 lbs. initial \( P \), then \( P_1 \) would equal 108 lbs. and the difference would be 72 lbs.

\[
P \times 0.6 = P_1 = 180 \times 0.6 = 108.
\]

Now it will be necessary to find the velocity possible due to this difference in pressure, and the most correct method is to find the ft.-lbs. of work, \( E \), due to the total heat in \( P - P_1 \) of a drop in temperature, and from that find the B.Th.U. in the range of temperature. The thermal units, given by a drop in pressure from 180 to 108 lbs. per square inch, must be found, and the following symbols are used in the calculation. It is assumed that the steam is allowed to acquire its maximum velocity in an expanding nozzle—

\[
\begin{align*}
T_1 &= t_1 + 460. \\
T &= t + 460. \\
t_1 &= \text{temperature of the steam at initial pressure } P \text{ absolute.} \\
t &= \text{“ “ lower pressure } P_1. \\
H &= \text{total heat of the steam at initial pressure } P. \\
t_1 &= \text{the temperature at } 180 = 373. \\
t &= 108 = 333. \\
dt &= \text{difference of temperature} = 40. \\
H &= 1195 \text{ at initial pressure.}
\end{align*}
\]

\[
\begin{align*}
\text{B.Th.} &= H \left( \frac{T_1 - T}{T_1} \right) = 1195 \left( \frac{40}{833} \right) = 58. \\
\end{align*}
\]

And \( V = 223 \times \sqrt{\text{B.Th.}} \cdot \text{U} = 223 \times \sqrt{58} = 1630 \text{ feet per second.} \)

**Table of the Velocity of Outflow and the Working Capacity of Dry Saturated Steam.**

*With Expanding Nozzle.*

<table>
<thead>
<tr>
<th>Initial Steam Pressure, Pounds per Square Inch.</th>
<th>Counter Pressure, 1 Atmosphere.</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Per Pound of Steam per Hour.</td>
</tr>
<tr>
<td>100</td>
<td>2717</td>
</tr>
<tr>
<td>120</td>
<td>2822</td>
</tr>
<tr>
<td>140</td>
<td>2913</td>
</tr>
<tr>
<td>160</td>
<td>2992</td>
</tr>
<tr>
<td>180</td>
<td>3058</td>
</tr>
<tr>
<td>200</td>
<td>3115</td>
</tr>
</tbody>
</table>
Without the expanding nozzle the velocity would be 1600 or thereabout. The actual H.P. in a good turbine would be probably one-fourth of the value given in table.

In most cases, however, we have given Vs, for Vt is limited for practical reasons and Vs should be twice Vt.

It is therefore necessary when Vt is given and the initial pressure, to be able to find, \( t_1 \) the temperature of the lower pressure, and from that obtain the corresponding pressure, \( P_1 \).

The general formula to find the temperature of steam at the lower pressure side of the wheels or stages in a turbine is

\[
T = \left( \frac{V_s^2}{JH} \times T_1 \right)
\]

and \( t = T - 460 \).

wherein \( T \) is the lower temperature; \( T_1 \), the initial temperature; \( Vs \), the velocity of the steam; \( J \), Joule's equivalent; \( H \), total heat.

Given \( P = 180 \), and \( Vs = 947 \) required the lower pressure \( P_1 \) necessary to give a difference of pressure to produce \( Vs = 947 \).

\[
T_1 = 373 + 460 = 833. \quad t_1 = 373.
\]

Then, \( \left( \frac{947 \times 947}{778 \times 1195} \times 833 \right) = 800 \), the temperature \( T \).

Hence, \( t = 800 - 460 = 340 \).

And from table I, we find this temperature belongs to a pressure of 120 lbs., a difference per square inch = 180 - 120 = 60 lbs.

The enormously high velocities due to large difference of pressure entail the use of very specially designed wheels to withstand the high centrifugal forces.

On the other hand, the reduction of the pressure differences entails the use of many wheels in series, for each wheel can only receive a small fraction of the total energy.

For marine purposes, speeds of propellers limit the turbine velocity \( Vt \). And in order to obtain a high \( Vt \) with as low a steam velocity \( Vs \) as possible, the diameter of the wheels is necessarily large, as \( Vt \) is the peripheral velocity of the blades at their mid length.

The number of wheels required in the series is found by dividing the total thermal units in the steam between
P and \( P_1 \), by the total thermal units found as above required to give the necessary \( V_s \) to the steam equal to twice \( V_t \).

For example, the steam falling from 180 to 3 lbs. absolute, will by (12) equal 332 B.Th.U. And the B.Th.U. required to produce a \( V_s = 947 \) will by (11) equal 18.

\[
\text{hence } n = \frac{332}{18} = 18 \text{ wheels in series.}
\]

\( V_s \) being equal to 947, \( V_t \) should be \( \frac{947}{2} = 473 \) feet per second, a very high velocity, which can be reduced only by a greater number of wheels and a corresponding smaller \( V_s \).

\( V \) the velocity of efflux of steam with a smaller difference of pressure than \( P_1 \times 0.6 \) \( P \) can be calculated by taking the two pressures \( \frac{P_1 + P}{2} \), that is their mean pressure, and the value in the last column of table I., represented by symbols \( R_v \).

\( V \) then equals

\[
V = 8 \sqrt{R_v \times dp \times 2.31} \quad \ldots \quad (14)
\]

\( dp \) being difference of pressure, and 2.31 the height of a column of water equal to one lb. pressure per square inch.

Although the velocity of the steam may be increased in a nozzle, the weight of steam flowing per second is strictly limited by the throat of the nozzle, its area in section; and when \( P_1 \) is equal to \( P \times 0.6 \), it reaches a maximum value, and no increase of pressure difference will increase the weight of the flow if the nozzle is of best form, then.

When \( P_1 \) is greater than \( P \times 0.6 \), the formulae below may be used for quantity or weight of flow \( q \)

\[
q = 0.03a \sqrt{(P-P_1) \frac{P_1}{P}} \quad \ldots \quad (15)
\]

\( q \) being lbs. per second, \( a \) the area in square inches, \( P \) and \( P_1 \) the two pressures in lbs. per square inch.

With properly shaped nozzles \( a \) should be multiplied by a constant between 0.92 and 0.95.

By another formula the quantity in lbs. per second can be calculated.

\[
q = 6.6 \times A.D. \quad \ldots \quad (16)
\]

\( D \) being the density of the steam at \( P \) as given in table I., this formulae also requires a constant for \( A \) about 0.92.
Conversely, if we have \( q \) given, we could find \( A \) from

\[
A = \frac{6.6 \times D}{q}
\]  

(17)

Or, if the quantity and velocity are given, \( A \) can be found. The velocity is generally given in feet, the area will be in square feet, and the quantity in cubic feet; for instance, 22 lbs. of steam, at 160 lbs. pressure, can be reduced to cubic feet by taking the volume from table I. per lb. at pressure of 160 = 2'8 cubic feet, hence \( 22 \times 2'8 = 61'6 \) cubic feet. If the steam velocity were 500 feet per second the area of nozzle would be

\[
A = \frac{\text{cubic feet}}{Vs} = \frac{61.6}{500} = 0.123
\]

square feet, or 17'7 square inches.

The diameter \( D \) of turbine wheels when in series depends, of course, upon \( Vt \), and the pressure or thrust on the blades, and in many cases in marine practice available space will have to be considered, the draught of the vessel and so on, suppose revolutions to be 440.

An empirical rule, deduced from experience with turbines already built, is \( D \) in inches:

\[
D = \frac{Vt \times 228}{\text{Revs. per minute}}
\]  

(18)

hence a turbine with \( Vt = 150 \) per second would have wheels equal to \( D = \frac{150 \times 223}{440} = 6 \) feet 3 inches nearly.

This is the mean diameter, the diameter of the wheels measured from middle of blade lengths.

Already formulae have been given for the steam velocity derived from thermal and also from gravitational units. The formula given for Parsons' turbine by Mr G. Storey, according to which the velocity of the steam depends on the drop in pressure between one row of blades and the next, it can be shown that

\[
v^2 = 2g \frac{dp}{p}
\]  

where \( g \) = gravity, \( H \) is the homogeneous head of steam which is about 63,000 feet for high pressure steam and about 45,000 feet for steam at atmospheric pressure, \( p \) is the absolute pressure at any row of blades, and \( dp \) is the drop of pressure between one row of blades and the next, and therefore \( \frac{dp}{p} \) is the amount of expansion that takes place in any row of blades.
Take, as an example, a Parsons' turbine, as shown in Fig. 7, the blading diagram for a 500 kilowatts turbine for 3000 revolutions per minute, and 150 lbs. pressure, with a vacuum of 28 inches. If we assume that the turbine is of the same diameter throughout, it is easily seen that since the speed of the blades is constant, in order to make the velocity ratio between the steam and the blades constant, the velocity of the steam through the blades must be constant, and as the steam at each row expands by the amount $\frac{dp}{p}$, it is evident that each row must have larger openings than the one before by the amount $\frac{dp}{p}$, and also that this ratio $\frac{dp}{p}$ is a constant throughout the turbine on the assumption that $H$ is constant, which is approximately the case, and for a preliminary consideration may be assumed.

From the equation $v^2 = 2gh\frac{dp}{p}$ the velocity of the steam can be calculated at any point along the turbine.

This method is called the $H$ method, since it works from the head of steam $H$.

It would require a table of "Heads" corresponding to pressures to make it useful, whereas in the formula given on page 17.

$$v = 8\sqrt{Rv \times dp \times 2.31} \quad . \quad . \quad . \quad (19)$$

$Rv$ can be found in most steam tables, such as table I, where it is given in the last column.

In the actual turbine each row of blades is $\frac{dp}{p}$ higher than the preceding one.

Now since, in an ordinary turbine working from, say, 150 lbs. steam pressure down to a pressure of one pound absolute, or a vacuum of 28 inches, there are about 150 expansions by pressure—or, say, 100 expansions by volume—we require the blades at the low pressure end 100 times as long as the high pressure end, so that it will easily be seen that the blades at the low pressure end would be so long as to be impossible to put on the spindle. The device, therefore, is adopted of increasing the diameter towards the low pressure end, and, since an increase of diameter also increases not only the circumference but also the velocity of the blades, it is easily seen that
the height of blade will vary inversely as the square of the diameter: that is, double the diameter, double the velocity of the blades, therefore double the velocity of the steam, so that we require only half the area through the blades, but since the circumference is doubled we only require blades of one quarter the height. The usual custom in land turbines is to have three drums each $\sqrt{2}$ or 1.4 times the diameter of the other, and thus the blades on the second drum are half the height of what they would be if they had been on the diameter of the first drum, and the blades on the last drum one-quarter the height of what they would have been if they had been on the diameter of the first drum. This, in the case mentioned above, where there are 100 expansions by volume, reduces the ratio of blade heights from 100 to 25.

The following remarks by Mr Speakman on the general subject of turbine design for marine purposes will be useful, and instructive:

"Expanding through a definite range of temperature and pressure, steam exerts the same energy, whether it issues from a suitable orifice or expands against a receding piston. Two transformations of energy take place in the steam turbine—first, from thermal to kinetic energy; secondly, from kinetic energy to useful work. The latter alone presents an analogy to the hydraulic turbine, the radical difference between the two lying in the low density of steam compared with water, and the wide variation of its volume under different temperatures and pressures.

Fig. 7 gives a sectional elevation of a marine turbine blading arrangement, and though this is only for an H.P. cylinder the principle is exactly the same throughout. The expansion, which is approximately adiabatic, is carried out in this annular chamber from A to B, which essentially resembles a simple divergent steam nozzle, but with this difference, that whereas in a nozzle the heat energy of the working steam is expended upon itself in producing high velocities, in Parsons' turbine the total expansion is sub-divided into a number of steps, in each of which a certain dynamic relationship between jet and vane is maintained. The expansion of steam at any one stage is typical of its working throughout the turbine. Each stage consists of a ring of stationary blades which give direction and velocity to the steam, and a ring of moving blades that immediately convert the energy of velocity into useful torque. The
total torque on the shaft is due to the impulse of steam entering the moving blades and to reaction as it leaves them, this process being repeated throughout the turbine.

Parsons' turbines, however, have been built with \( V_t \) (Fig. 7), varying from 0.25 to 0.85 of \( V_s \), where \( V_t \) represents blade velocity at mean diameter, and \( V_s \) the steam speed due to expansion across the row in question. A very usual ratio in electrical work for large units has been \( \frac{V_t}{V_s} = 0.6 \), but this involves a greater number of rows than is possible in marine work, and the ratio must be reduced. These ratios need very careful calculation. The steam consumption must be accurately known in order to proportion them correctly throughout the turbine, and the necessity (which is inevitable with the present form of caulking piece) of having the same area of openings in so many rows while the steam volume increases so rapidly, adds to the difficulty of close calculation. The potential energy of the steam, corresponding to the “head” in water turbines, can easily be calculated for given pressure differences.

\[
B.\text{Th.\,U.} \times 778 = \text{Energy in ft.-lbs. per lb. of steam} = E.
\]

\[
V_s = 8 \sqrt{\frac{E}{223 \sqrt{B.\text{Th.\,U.}}}}
\]

For a given blade velocity, it is obvious, then, that the speed ratio between jet and vane must affect the number of stages, and the greater the ratio of \( V_t \) to \( V_s \) the greater will be the required number of rows, that is, to obtain the required \( V_s \) at each stage a smaller pressure drop per row is necessary, or vice versa.”

The best blading arrangement, scientifically and commercially, is the result of much theory and practice. The mean diameter is an arbitrary dimension capable of wide variation without affecting the efficiency, provided that the number of rows is correct; it is found by assuming, from experience, a blade velocity, whence—

\[
\text{Mean diameter (Blade velocity in feet per sec.} \times 228 \text{ in inches} = \frac{R.P.M.}{V_t}. \]

To arrive at the corresponding number of rows, the revolutions being given, the ratio of \( V_t \) to \( V_s \) must be settled, from which the steam speed can be obtained; it is a convenient assumption at the beginning of any design to consider the turbine as parallel throughout and of constant efficiency, and to design on this basis.
In early marine turbines the turbine drums were all made of the same diameter, and the higher speed necessary on the L.P.'s was got by running at considerably higher revolutions than on the H.P. shaft; but, following up the increase in propeller efficiency found to be due to the use of larger screws the speed for each shaft is now more nearly equal, while the wing drums are made larger in diameter. The vagaries of the following wake, however, necessitate slightly different propeller dimensions on each shaft, or else slightly different revolutions with the same screws; and it is noticeable that in a triple-screw arrangement, the centre screw being right-handed and the wing screws revolving outwards, that the starboard propeller is influenced by the centre one, and almost invariably revolves at a lower speed. In a four-shaft design, due to the varying wake values at different speeds, and possibly, also, to some unequal distribution of power, the outer screws run slower at low speeds, and faster at high speeds than the two inner shafts, but exact data as to this, and the possibility of allowing for it in the design, are still wanting.

In all types of turbines—Parsons', Rateau's, Curtiss', etc.—a certain ratio must be maintained between the blade velocity and steam velocity, and as steam acquires very high velocities by expansion, the blade velocity must be maintained either by the revolutions or by large diameters, or both. As the weight increases very rapidly with the diameter, and extraordinarily so with the reduction in rotative speed, it is preferable to increase, if possible, the revolutions or the number of stages rather than the diameter, and especially should this be done in cases where, as in the Rateau or Zoelly types, the weight increases more rapidly in inverse proportion to the R.P.M. and the diameter than it does with other types. To increase the revolutions, it may be necessary to increase the number of shafts and propellers, thus reducing the power per shaft and the effective thrust through each screw. Increasing the diameter of the turbine adds largely to the constructional difficulties, especially of the cylinder.

Having obtained the number of rows and the diameter, the blading arrangement can be worked out in detail. The height of blade depends on the volume of the steam and the speed at which it is to flow, and also on the ratio of the area of exit openings between the blades to that of the
annulus between spindle and cylinder, which is about one-third in normal blades. The necessary clear area to pass the steam being equal to volume \( \div \) velocity, and knowing this annular factor, say 3, for a ratio of one-third (or 2 for \( \frac{1}{2} \), etc.), then

\[
\text{Height of blade} = \left( \frac{\text{Clear area in square inches} \times 3}{\text{Mean circumference in inches}} \right)
\]

The ratio of blade height to mean diameter should not be less than 3 per cent. or more than 15 per cent., because in the former the leakage will be excessive, and in the latter the bending moment on the blade becomes too great, and the radial divergence of the blades too much. The width of blade, the shape of section adopted, and the circumferential pitch, are standard considerations, and affect the factor 3 given above. It is not proposed to enlarge upon them in this work. It may, however, be remarked that for \( \frac{V}{t} \) greater than 0.6 the usual shape of Parsons' section, as shown in Fig. 7, should be modified to a somewhat different form of blade, with a sharper entrance edge. This section is not to be recommended, as, owing to the necessity of strengthening the blade sufficiently, the metal must be placed nearer the exit edge, thus increasing the angle between the face and the back of the exit edge of the blades, and giving, in fact, an inferior shape of opening compared with that obtainable with a blade section adapted to ratios under 0.6.

Due to the action of the steam, an end thrust occurs in the direction of the propeller, which is advantageously used in partially balancing the propeller thrust, thereby reducing the size of thrust block necessary. A margin must be allowed here, and the propeller thrust is not entirely balanced by the pressure on the annulus between the dummy-ring and the spindle plus the end pressure on the blades. For the diameter D to give the required annulus, as well as that of the propeller, the effective thrust must be carefully calculated; and experience shows that there is a drop in steam pressure varying from 10 to 15 lbs. per square inch between the pipe inlet to the H.P. receiver and the first row of blades, which should be considered in designing this balancing area. The number of rows of dummy packing used varies according to the designer's judgment very
largely, and may be modified according to the pressure and the clearance allowed—say a 7-1000th to a 15-1000th of an inch in electrical work, and rather more in marine work.

It may be remembered that in a marine turbine the spindle is in compression and the cylinder in tension when working. In electrical turbines where the end thrust must be eliminated by the use of balancing pistons, the spindle is in tension and the cylinder is balanced. The shafts between the turbine bearings and the drum must be made amply stiff enough, as well as strong enough, for any sag in the spindle will destroy the clearance. The stresses due to centrifugal force are very low in the Parsons' turbine, and except in occasional L.P. barrels do not exceed about 7500 lbs. per square inch, while at the H.P. end they are usually under 2000.

The vane speeds adopted in practice vary considerably; for some time 100 feet per second was regarded as a standard for the first row, and the Westinghouse Co. at Pittsburg was first to make a radical departure in this and adopt far higher speeds. The maximum vane speed used for Parsons' blading is about 375 feet per second in the low pressure blades, and 170 in the H.P. blades of electrical turbines; the lowest speeds used are in marine work, and are only about one-third of these. To some extent blade speed is governed by blade height; the speed should be so modified so that this may be at least 3 per cent. of the mean diameter to reduce the proportion of clearance losses. Leakage over the tips of the blades is perhaps not so detrimental on account of actual leakage loss as in its superheating effect on steam between the row past which it leaks and the last row, because this reheating effect upsets calculations regarding openings by increasing the steam volume, and thereby affects the fluid efficiency. This leakage over the tips must be taken into account in designing reaction turbines. Temperature and diameter influence the clearance, and the stiffer the cylinder is to resist distortion due to heat the less the clearance may be made.
CHAPTER II

ELEMENTARY TURBINES

Different Types of Turbines

The early Hero type of turbine is worthy of notice, although it is apparently not to be of immediate practical use. Its principles are, however, of value and instructive.

![Fig. 10. — Hero wheel converging nozzle.](image)

It is shown in the simplest and commonest form in Fig. 10. That design, however, is faulty, for this type of wheel will always have a large pressure drop, and therefore

![Fig. 11. — Hero wheel diverging nozzle.](image)

an expanding nozzle is necessary, in order to obtain the benefit of the further expansion of the steam. The form of wheel should be approximately that shown in Fig. 11, where the steam is allowed to expand in a conical passage before passing round the deflecting bend, and it has acquired the full velocity by the time it reaches $x y$. 

25
An early form of this turbine is shown in Fig. 12, built in Glasgow in 1854. The nozzles are formed by a cut-out ring between two steel discs, but they are too heavy for the velocity which would be required of a single wheel. A better construction is that shown in Fig. 13, where the nozzle is connected to an expanding conduit, so that the full velocity of the steam is obtained before it turns the bend and exerts its momentum. The ring is cut out to form the nozzles, and then bound by two discs of steel; the steam enters by the hollow shaft. There is, however, always a difficulty with this class of turbine, especially with wheels in series. Nevertheless, it formed the subject of one of Mr Parsons' patents. The illustration (Fig. 14) shows an end view and a longitudinal section of this turbine, which has four arms in four separate compartments, the steam passing from one pair of arms to the next through the hollow shaft. A sectional view (Fig. 3) shows one of the pairs enlarged, in which it will be seen that the orifice is merely a hole in the side of the arm.
In this construction the initial pressure must have been intended to be not much greater than the pressure outside the arms, \( P_1 \), the outside pressure in the casing being at least 0.6 times \( P \) the pressure inside the arms, so that \( V_s \) would be about 1400 feet per second.

The idea was to expand the steam in these wheels so that its volume would be enlarged before entering the usual rings of guide and wheel blades in the ordinary Parsons' turbine. That idea is an excellent one in itself, but this method of putting it into practice was not the best.

The same notion has been put into practice in the Westinghouse turbines by means of wheels of De Laval type, shown in elementary form (Fig. 15.)

Here, in Fig. 15, we have two wheels of large diameter in order to obtain a high-wheel blade velocity. The steam expands through the nozzles of the first wheel and a short range of guide blades, GB, so that it reaches the ordinary series of rings of blades, P, in considerable volume, and hence the first rows of blades are not so short as they would require to be to take the initial pressure direct.
Fig. 14.—Parsons’ series of Hero wheels.

Fig. 15.—Diagram, Westinghouse turbine.
A similar design is due to Curtiss', shown in Figs. 16 and 17. Here the nozzles, E, E, E, are only in a partial flow wheel; the wheels have complete rings of blades; the steam expands through all the passages, wheels and nozzles.

This type of turbine has much to recommend it, and will yet become almost universal on the plan of the Westinghouse, with a series of De Laval wheels of large diameter and a series of rings of guide and wheel blades to follow the expansion down to vacuum pressure. Wheels are sometimes grouped as impulse or action wheels and reaction or pressure wheels, but the distinction is not very clear in steam turbines. The Pelton wheel is referred to as an impulse turbine (see Fig. 8), which shows one of the buckets and the jet. It is purely an impulse wheel, the fluid being
deviated through two right angles, so that its thrust is
\[ T = \frac{2WV}{g}. \]

Fig. 18 shows the combined action and reaction wheel of Prof. James Thomson, known as the Vortex wheel, wherein the fluid is guided tangentially at considerable velocity upon the wheel tips. Here it acts by impulse, and is turned through a large angle, and being under pressure, reacts at the outlet. Such a wheel gives a thrust proportional to \( \frac{2WV}{g} \), one-half being due to impulse, and the other half due to reaction.

The smaller turbines are more generally employed in electrical generator driving, the best known being the De Laval, with the single wheel. The design of this machine is simplicity itself, but it requires the highest class of constructional work, the leading features being the flexible shaft
to allow the wheel, at the very high velocities necessary, to centre itself. The wheel itself is specially formed to withstand the great centrifugal forces, and the double helical gearing to reduce the speed to a suitable one for driving. All these points have been admirably attended to by the designer.

Fewer revolutions would be necessary with wheels of larger diameter, but the stress caused by centrifugal forces in a wheel is greater in proportion to its diameter, the greater the diameter at any given peripheral speed.

If the stress were the same at the same peripheral speed, whatever the diameter, then the wheel of large diameter at slow revolutions would be used, but that is not the case. Without going into any far-fetched proofs, it will readily be seen that the total weight of a wheel of large diameter for a given peripheral speed must be much greater than the total weight of a smaller wheel for the same peripheral speed.

Hence in the De Laval wheel we find small diameters from 5 to 18 inches.

These wheels take from 22 to 24 lbs. of steam per horse-power hour, and as it is for present purposes simpler to obtain the principal dimensions beginning with the steam consumed per horse-power hour than otherwise, we
can start with this figure, say for a 25 horse-power turbine, taking the higher figure 24 lbs. per hour, we get a total of $24 \times 25 = 600$ lbs. of steam per hour, or $0.166$ per second. And as the steam flow is always, in this turbine, the maximum weight flow at the throat, we can now find the area of the throat of the nozzles, or throats, if we have more than one. The velocity $V_s$ of the steam at the throat is given by De Laval as 1500 feet per second, a figure we may use.

It will be necessary to reduce the lbs. of steam per second to cubic feet per second, and for that purpose we take the initial pressure at 160 lbs. per square inch, with a volume of 2.8 cubic feet per lb., hence $2.8 \times 0.166 = 0.47$ cubic feet. If now we divide $\frac{0.47}{1500} = 0.0031$ square feet as area of throat of nozzle, or 0.045 square inches. This is very nearly $\frac{1}{4}$ of an inch diameter for one nozzle throat, but when the nozzle diverges beyond the throat, it becomes of wide bore at the mouth—about 1 1/4 inches in this case—and that entails wheel buckets about 1 1/2 inches long, a length which for various reasons is too great for a small-power wheel. Probably four nozzles of throat diameter, equal to 1/8 inch each, would be employed. The nozzles have regulating needles on hand wheels (Fig. 19), whereby their entrance can be restricted or shut, hence the throat may be slightly larger than that calculated, to allow a margin for difference in boiler pressure.

$V_t$, the velocity of the wheel is, in the small and

![Fig. 19.—De Laval nozzle regulator.](image)
medium sized wheels, about \( \frac{1}{4} \) of \( V_s \), that is about \( \frac{1}{4} \) of 4000 feet per second, that being the velocity of the steam expanded in the nozzle, giving 1000 feet per second for \( V_t \).

The highest number of revolutions per second possible is chosen for such wheels, and, \( R \) the revolutions, is limited to 510 per second for smaller sizes. From this and the peripheral speed, we get the circumference of the wheel at the middle of the blades. The circumference would be \( \frac{1000}{500} = 2 \) feet, diameter = \( 7\frac{1}{8} \) inches.

The diameter may, however, be somewhat larger, as the 500 revolutions is for smallest size, and the wheels decrease in revolutions as the size mounts up, while the peripheral speed is increased in the larger sizes.

The buckets are only about \( \frac{1}{4} \)-inch broad, the steam entering at an angle of 20° to the plane of the wheel or line of motion, and leaving at same angle.

**The De Laval Nozzle.**

This nozzle is shown in Fig. 19, also in Fig. 20, with an expanding curve in connection with it, as calculated and plotted by Prof. Andrew Jamieson.

With the wide end as the inlet and the narrow end the outlet, we get the fluid gradually increasing in velocity up to the narrow exit, where it issues, slightly converging at the full speed due to the height or pressure.

If we reverse the nozzle we get a very different result.

Suppose we consider the nozzle connected to a steam boiler at 200 lbs. pressure about atmospheric, or 214.7. According to Prof. Zeuner, in a nozzle in which steam is adiabatically expanded the potential energy (heat) is converted into kinetic energy. The kinetic energy of
1 lb. of steam at velocity \( V \) in feet per second is of course equal to
\[
\frac{V^2}{2g} \text{ ft.-lbs.}
\]

The black line curve represents the expansion curve of dry saturated steam to \( \rho v^\frac{2}{3} \) = constant.

The dotted line curve represents the adiabatic expansion of steam as per the above data in the De Laval turbine steam nozzle to \( \rho v^\frac{2}{5} \) = constant.

Suppose steam is let down from pressure \( p_1 \) to pressure \( p_2 \) adiabatically in an expanding nozzle. The internal heat of the steam at \( p_1 \) = (the internal heat of the steam at \( p_2 \)) + (kinetic energy at \( p_2 \) in heat units); or, in other
words, the kinetic energy is proportional to \( H - h \), where \( H \) is the internal heat at \( p_1 \) and \( h \) that at \( p_2 \). And if \( J \) is the mechanical equivalent of heat, the velocity of the steam acquired by the time it had fallen to \( p_2 \) would be—

\[
V = \sqrt{2gJ(H-h)}
\]  

(20)

The internal heats depend on the percentage of moisture in the steam, calculated on the assumption of constant entropy during the expansion.

Theory and experiment agree that at a certain ratio between \( p_1 \) and \( p_2 \) a maximum amount of steam flows through a converging nozzle. This limiting ratio is—

\[
\frac{p_2}{p_1} = 0.6; \text{ and hence } p_2 = 0.6 \times p_1
\]  

(21)

In the case cited of a pressure \( p_1 \) equal to 214.7 lbs., \( p_2 \) at section B of the nozzle would be equal to \( 0.6 \times 214.7 = 128.8 \) lbs.

The full-line curve represents the natural loss of pressure in dry saturated steam as it is expanded in accordance with Prof. Rankine's well-known formula \( pv^{1.5} = a \) constant; where \( p \) is the pressure in lbs. per square inch absolute, and \( v \) the corresponding volume in cubic feet per lb. of steam. This curve is drawn from 475 lbs. per square inch—at which pressure 1 lb. of the steam occupied 1 cubic foot—down to 1 lb. absolute, at which it occupied 330 cubic feet. In the Fig. 20 is included the range of pressures. The dotted line represents an adiabatic expansion curve to \( pv^{1.9} = a \) constant, from 215 lbs. absolute at Section A, before entering the nozzle, down to 0.93 lb. at Section C, where it occupied 256.8 cubic feet, and left the nozzle with a velocity of 4127 feet per second with 24 per cent. of moisture. This curve passed through the point B, where the steam occupied 3.5 cubic feet, and has 4 per cent. of moisture, with a velocity of 1500 feet per second. Now, it was evident from this curve that if the potential energy of each lb. of static steam at A had been so far converted into kinetic energy at B, that it had there a velocity of 1500 feet per second, and contained 4 per cent. of moisture, with an increase of volume from 2.11 cubic feet at A to 3.5 cubic feet at B, it must of necessity have fallen in pressure from 215 to 125 lbs. absolute pressure in doing so. This is in strict accordance with the natural law for the adiabatic expansion of steam. The temperature of the steam must also have fallen from 382° Fahr. to at least 340° Fahr. in this short passage.
It might be that the steam in passing from A to B, and from the very small throat of \( \frac{1}{4} \)-inch diameter, at the rate of 8 lbs. weight per second, naturally formed a *vena contracta* due to "throttling." In any case, it must there lose potential energy due to friction and increased velocity.
Taking the three sections of the cones:—

Section A—
Pressure, 200 lbs. above atmosphere.
Moisture percentage = 0.
Unit quantity of steam, 1 lb.

Section B (smallest section)—
Pressure, 110 lbs.
Quantity of steam, 0.96. Moisture 0.4 lbs.
Velocity \( V = \sqrt{2gJ(H - h)} = 1500 \) feet per second.
Volume of steam, 3.5 cubic feet.
Section C (the wider end)—
Pressure, 2 inches of mercury (absolute pressure).
Percentage of moisture in the steam, 24 per cent.
Specific quantity of steam, 0.76.
Velocity of the steam, 4127 feet per second.
Specific volume of the steam, 256.8 cubic feet per lb.

The proportion between the areas of the large and small section of this nozzle should be as $27.2345$ to 1, or the proportion between the diameters of these two sections as 5.2187 to 1. If, for instance, the diameter of the small section is 6 millimetres, or very nearly $\frac{1}{4}$ of an inch, the diameter of the large section should be 31.31 millimetres, or nearly 1.31 inch. Through such a nozzle there passes a certain constant weight of dry saturated steam of 200 lbs. pressure per hour, neither more nor less. This fact of the nozzle passing only a certain amount of steam per hour is used as a measure of steam.

The quantity of steam passed by the nozzle is measured by the section B, and is equal to $Q = 370 AD$ so long as the difference of pressure is greater than $\frac{4}{5}$th the initial pressure.

### TABLE OF VELOCITIES OF STEAM. BY KONRAD ANDERSEN.

**THE VELOCITY OF OUTFLOW AND THE WORKING CAPACITY OF DRY SATURATED STEAM.**

<table>
<thead>
<tr>
<th>Initial Steam pressure, lbs. per square inch.</th>
<th>Counter-pressure 1 atm.</th>
<th>Counter-pressure 2.4 lbs. per sq. in. absolute corresponding to 29 in. vacuum.</th>
<th>Counter-pressure 0.93 lbs. per sq. in. absolute corresponding to 28 in. vacuum.</th>
</tr>
</thead>
<tbody>
<tr>
<td>---------------------------------------------</td>
<td>-------------------------</td>
<td>-------------------------------------------------</td>
<td>--------------------------------------------------------------------------------</td>
</tr>
<tr>
<td>60</td>
<td>2421</td>
<td>25.29</td>
<td>0.046</td>
</tr>
<tr>
<td>80</td>
<td>2595</td>
<td>29.06</td>
<td>0.053</td>
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<td>100</td>
<td>2717</td>
<td>31.86</td>
<td>0.058</td>
</tr>
<tr>
<td>120</td>
<td>2822</td>
<td>34.37</td>
<td>0.062</td>
</tr>
<tr>
<td>140</td>
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<td>36.62</td>
<td>0.066</td>
</tr>
<tr>
<td>160</td>
<td>2992</td>
<td>38.63</td>
<td>0.070</td>
</tr>
<tr>
<td>180</td>
<td>3058</td>
<td>40.35</td>
<td>0.073</td>
</tr>
<tr>
<td>200</td>
<td>3115</td>
<td>41.87</td>
<td>0.076</td>
</tr>
<tr>
<td>220</td>
<td>3166</td>
<td>43.26</td>
<td>0.079</td>
</tr>
<tr>
<td>280</td>
<td>3294</td>
<td>46.83</td>
<td>0.085</td>
</tr>
</tbody>
</table>
As will be seen from the foregoing table, the velocity of outflow of steam expanded in suitable nozzles to the proper ratio is very high. Steam expanded in a nozzle from 280 lbs. pressure above the atmosphere down to 28 inches vacuum leaves the nozzle with a velocity of 4229 feet per second, or over 48 miles per minute. This steam jet would pass round the earth in 8 hours 37 minutes.

The proportion of the wide to the narrow end of the divergent nozzle is given above as 27 to 1 nearly.

When \( V \) the velocity at the throat is known and its sectional area, and \( V_1 \) is also known, that is the velocity at the wide end, the proportion of the nozzle can be found.

The volume of the steam at the throat pressure \( P \) is taken as cubic feet per lbs. = \( C \), and the volume at \( P_1 \) the lower pressure \( C_1 \), then—

\[
\frac{C_1}{C} = \frac{256}{3.5} = \frac{73}{27} = 27.
\]

That is to say, in this case, the wide end should have twenty-seven times the area of the throat.

Fig. 19 shows the arrangement of the nozzles in the turbine casing with the needle valve and hand-wheel regulator.

Fig. 21 shows the various parts of the complete turbine dissected, and Fig. 22 is a sectional view of the whole machine, and Fig. 23 a complete view of a machine for belt-driving from a pulley.

The arrangement of multiple nozzles is shown in Fig. 24. This arrangement is adopted in order to reduce the length of the blades. The blades must, of course, be a little longer than the diameter of the wide end of the nozzle; four nozzles, therefore, require much shorter blades than one or two nozzles of same combined total throat area.

The De Laval turbine has met with considerable success, and is the only one of high efficiency of small size. It is easy enough to make a large power turbine of high efficiency, but difficult to do so in small powers.

In order to reduce the high velocity necessary with a single wheel, turbines have been designed for lower speeds by placing De Laval wheels and nozzles in series order.

In the Hero or Barker's Mill type of turbine (Fig. 3), the principles are different from the De Laval turbine, which is
purely impulse in operation. In the Hero type, without expanding steam passages to the nozzle, the thrust is simply equal to $2P.a$, when the wheel is at rest, and when in motion,

$$ (2P.a) \times 1 - \frac{V_t}{V_s} \quad \ldots \ldots \quad (23) $$

But if expanding passages are employed, then the thrust is proportional to formula (3), the steam being turned through one right angle only.

The number of revolutions would, in case of expanding passages, be very high, or the peripheral speed very high. If this type of wheel is put in series to reduce $Vt$, then the expanding passage is not of much advantage, and its thrust becomes simply proportional to (22).

$$ (2P.a) \times 1 - \frac{V_t}{V_s}, $$

wherein $P$ is pressure of steam in lbs. per square inch and an area of nozzles per square inch.

If $P=160$ lbs., and $a=0.1$ square inch, and $Vt=500$, and $Vs$ 1000,

then, \[ T = (2 \times 160 \times 0.1) \times 1 - \frac{500}{1000} = \]

\[ T = 32 \times 1 - \frac{1}{2} = 32 \times \frac{1}{2} = 16 \text{ lbs}, \]

and the energy $E=16 \times 500 = T \times Vt=8000 \text{ ft.-lbs.}$

As a turbine it presents many mechanical difficulties, and has not been a practical success.

The combined impulse and reaction wheel is best illustrated by the Parsons' wheel (see Fig. 7). Here the steam strikes the entrance to the blades with a velocity $Vs$, and is turned through approximately, one right angle giving a thrust proportional to formula (3). And as there
is a difference of pressure, $P$, between the entering side of the wheel and the exit, this pressure produces a thrust exactly in the same way as the pressure in the Hero wheel, proportional to $2 P a \times 1 - \frac{V_t}{V_s}$, so that we have both impulse and reaction at work on the one wheel in the Parsons' type of wheel.

The Rateau and Zoelly turbines are simply multiples of De Laval turbines, and purely impulse motors.

An impulse turbine for the high-pressure end, combined with a Parsons' turbine for the low pressure, has much to recommend it.

A large diameter of the impulse wheel can be used at the high-pressure end, with only one or two delivery nozzles, so that $V_t$ is high, although the revolutions are low.

Four impulse wheels for the high pressure may thus take the place of a long series of Parsons' wheels.

There is the Curtiss' turbine, also a combined form of turbine, partly impulse wheels and partly reaction wheels.

These are the present practical forms of turbines; in all probability, before long, one-standard type will alone prevail.

In dealing with the mechanical construction of turbines, especially those of this single-wheel type, or with only a few wheels in series, centrifugal forces must of course be taken into account in designing the wheels, the blades and the shafts. The scope of this work does not include the designing or dimensioning of the parts in relation to the stresses or forces tending to bend or break them. Those calculations belong rather to the domain of structural engineering, but a brief reference to centrifugal forces may not be out of place here.

In elementary books on the principles of mechanics, it is proved that if $\frac{W}{g}$ be the mass of a body of weight $W$, Fig. 24.—De Laval multiple wheels.
which describes a circle of radius, \( r \), with a uniform velocity, \( V \), under a force, \( F \),

\[
F = \frac{W \times V^2}{gr} \times \frac{V}{r}.
\]

An unbalanced wheel, at high rates of revolutions, produces vibrations, even when the balance is very nearly correct, a wheel 4 feet in diameter revolving at 160 feet per second peripheral speed, and with an unbalanced weight on the periphery of only 0.1 lbs., would produce a side-pull causing vibrations equal to

\[
F = \frac{WV^2}{gr} = \frac{0.1 \times 160 \times 160}{32 \times 2} = 40 \text{ lbs.}
\]

A bucket on a De Laval wheel on a 1-foot radius, and weighing only \( \frac{1}{10} \)th of a lb., would at a peripheral velocity of 500 feet per second, be pulled outwards by a force,

\[
F = \frac{WV^2}{gr} = \frac{0.1 \times 500 \times 500}{32 \times 1} = 800 \text{ lbs. nearly.}
\]

Turbine wheels are, however, rarely run at any speeds above 200 feet per second, as they are now nearly all constructed with wheels in series, in order to reduce peripheral speeds.

Balancing is of greatest importance; strength to resist the centrifugal forces when balance is perfect is easily provided.
CHAPTER III

TURBINE WHEELS IN SERIES

Inventors early conceived the idea of placing wheels in series in order to reduce the velocity \( Vt \) and still obtain efficiency. Pilbrow was a pioneer in this line, and his patent of fifty years ago shows he clearly understood the principles.

One of his propositions is shown in Fig. 25, showing two wheels and one nozzle, the wheels running in opposite directions.

Another early arrangement is that shown in Fig. 27, which was designed by a Greenock engineer, Wilson. The shaded blades are the fixed guide blades and the white blades the guide blades; the expansion is clearly shown from the high pressure to the low-pressure end. Another of Wilson's designs is that shown in Fig. 26, in which the steam expands radially through alternately fixed and movable blades.

Another of Wilson's designs is shown in Fig. 28, in which the steam is directed repeatedly through one wheel, the guide blades increasing in number. All these primitive
designs have been adopted for series working in modern steam turbines.

Wilson's patents date away back to 1848 and Pilbrow's Patent 9658 is dated 1843.

Pilbrow also found that he got a better thrust when he allowed the steam jet to expand before striking the wheel blades, and he says the best distance of the orifice of
the jet from the blades is \( \frac{3}{4} \) inch, with steam at 60 lbs. pressure, and with a \( \frac{3}{4} \)-inch nozzle he got 14 lbs. thrust.

The high velocities were, however, too much for these early inventors, the constructional work being quite beyond the then existing tools and materials.

For stationary power purposes, such as dynamo driving, the wheels may have a very high rate of revolutions, and consequently the steam velocity \( V_s \) and blade velocity \( V_t \) may be high, and therefore a few wheels in series may drop the whole pressure difference.

But in marine propulsion the revolutions are reduced to accommodate the screw propeller, and hence the marine steam turbine has a long series of expansions, the pressure difference between wheel and wheel being small.

It has already been shown that an enormously high ratio of revolutions, and peripheral speeds, are entailed by the use of only one wheel, and now it may be shown how the speeds may be lowered by wheels in series. Fig. 28.—Wilson's turbine, 1848.

Taking first the case of one wheel at 500 velocity = \( V_t \), and 1500 the velocity = \( V_s \), \( V_s \) is here about the velocity of flow when \( P_i \) is \( 0.6 \times P \).

The turbine to be of De Laval type, with a nozzle throat area of \( \cdot007 \) square feet or \( 1 \) square inch section, and the initial steam pressure equal to 180 lbs. absolute, it may be assumed that the angles of entry and exit are small and the full thrust obtained by (4).

The density of the steam at 180 lbs. pressure is \( 0.4 \).

Then by (4) the thrust is,

\[
T = \frac{2daVs(V_s - Vt)}{32} = \frac{2 \times \cdot007 \times 0.4 \times 1500(1500 - 500)}{32} = \frac{8.4 \times 1000}{32} = 262 \text{ lbs.}
\]

wherein as before—

\( d \) is the density of the steam at the given pressure, table I. col. v.
a the area of the fluid stream or jet in square feet, 
Vs, velocity of fluid,
Vt, blade velocity of turbine,
and 262 by Vt will equal the ft.-lbs. by (7) 
\[ E = \frac{131,000}{550} = 238. \]
The weight of steam per second is 
\[ 2dV_s = 4.2 \text{ lbs.}, \]
hence the steam consumed per horse-power hour equals 
\[ \frac{4.2 \times 3600}{238} = 64 \text{ lbs.}, \]
and that without taking any losses into account.

Increasing Vt would reduce this steam consumption, but it is desirable rather to decrease Vt. This can be done if we add wheels in series. And if we add wheels in series, the steam consumption per horse-power hour diminishes, and this is now to be shown.

First, let more wheels similar to that already calculated be added to diminish the steam consumed per horse-power hour. We have seen that so long as \( P_1 \) is less than \( 0.6 \times P \), the velocity remains about 1500 feet per second in a jet delivery from \( P \) to \( P_1 \).

In the wheel just calculated, the initial pressure was 180 lbs. absolute pressure per square inch, and in order to get the full velocity in the jet, the pressure into which it delivers, in a chamber in which the wheel is confined, would be \( 180 \times 0.6 = 108 \) lbs., so that if we provided another chamber with another wheel, 108 lbs. would be the delivery pressure \( P \) from the first to the second wheel.

\( p \), the thrust on this second wheel, would be found as before, but \( d \) and \( a \) would have to be calculated from table I., corresponding to the lower initial pressure, 108 lbs. At that pressure the weight per cubic foot = 0.25, and the volume is 4 instead of 2.5 at 100, hence the jet area will be increased in the ratio of 4 to 2.5 \( \frac{4}{2.5} = 1.6 \) square inches = 0.011 square feet. \( V_s \) and \( V_t \) remain the same as before.

\( 2d \alpha \) will be 0.0056, hence the second wheel will give as much horse power as the first, and the steam consumption will be reduced to 32 lbs. per horse-power hour.

Now how far this can be carried may be seen by running out the pressures until 3 lbs. or less per square inch is reached.
This is done as follows, so long as $P_1$ is equal to or less than $P \times 0.6$, the same velocity of steam is given at each wheel and the same weight of steam passes.

$$
\begin{array}{cc}
P & P_1 \\
First \text{ wheel} & 180 \times 0.6 = 108 \\
Second & 108 \times 0.6 = 64.8 \\
Third & 64.8 \times 0.6 = 39.7 \\
Fourth & 39.7 \times 0.6 = 23.8 \\
Fifth & 23.8 \times 0.6 = 14.3 \\
Sixth & 14.3 \times 0.6 = 8.4 \\
Seventh & 8.4 \times 0.6 = 5 \\
Eighth & 5 \times 0.6 = 3 \\
\end{array}
$$

This is a theoretical arrangement, not allowing for frictional or condensation or other losses.

Eight wheels at 238 horse power each would give a total of 1904 horse power, and the steam consumed per horse-power hour would be equal to

$$
\frac{3600 \times 4.2}{1904} = 8 \text{ lbs.}
$$

In practice, however, the consumption would be double this amount, or thereabout, but with careful design, 12 to 13 lbs. may be reached per horse-power hour.

The foregoing shows the advantage of wheels in series. It will be understood that as the steam expands in the successive nozzles, the wheel blades and nozzles increase in area in proportion to the increase in volume of steam at each stage.

500 feet per second for $VT$ is a fairly high velocity, and if the revolutions were limited to, say, 2400 per minute or 40 per second, the circumference of the wheels at mid length of the wheel blades would be

$$
\frac{500}{40} = 12.5 \text{ feet, or about 4 feet diameter.}
$$

Recently some turbines have been introduced with only four wheels in series for dynamo and other high-speed driving.

The pressure drop is from 160 to 60 in first wheel, 60 to 30 in the second, 30 to 15 in the third, and 15 to vacuum in the fourth.

Using expanding nozzles, the speed is necessarily
high, but sufficiently low to dispense with reducing gear wheels.

The Rateau turbine is another one in which each wheel must be in a separate chamber, separate from the next, and no attempt is made to pack the shaft.

This type of steam turbine is somewhat like the Parsons parallel flow-motor, but differs from the latter in this respect, that each rotating ring of blades revolves, as it were, in a compartment by itself. If we can imagine a number of De Laval turbines placed side by side with the wheels in parallel planes, and if we imagine a large number of nozzles extending from the exhaust side of one wheel, through the casings to the steam side of the next wheel, we have in principle a Rateau turbine (Fig. 29).

The shaft where it passes through the casing is clear of it by about \( \frac{1}{4} \) th of an inch at the high-pressure end, and greater of course at the low-pressure end; the hole it passes through is fairly long, and has its inside face grooved as in the Deluel pump (Fig. 30), to retard the escape of steam.
The steam enters at the high-pressure end through guide blades, of which one is shown at A. These direct the steam on to a revolving wheel, carrying a single row of buckets. The steam having passed through these and lost its velocity there, expands again through a second set of guide blades, mounted on a fixed diaphragm B, which discharge it into the buckets, carried by a second revolving wheel, on leaving which the steam expands through a third set of guide blades, and so on. In fact, the Rateau turbine consists essentially of a number of simple turbines, through which the steam flows in series. In the turbine illustrated there are fourteen of these, each of which constitutes a stage of the turbine. In the first of these the pressure will fall, say, from 155 lbs. absolute to about 130 lbs. absolute,

in passing through the guide-blades at A, and by this expansion the steam acquires a velocity of about 750 feet per second. This velocity is taken out of the steam as it passes through the moving bucket, but the pressure of the steam does not alter in this passage. The steam now expands through the guide blades fixed in the stationary diaphragm, B, falling in pressure from 130 lbs. per square inch to about 108 lbs. absolute, and in this again acquires a velocity of about 750 feet per second, which is again abstracted by a moving wheel. Thus the steam flows through each turbine in succession, acquiring a velocity by expansion through the fixed guide blades, and losing it again by passing through the revolving buckets. In the high-pressure end of the turbine the steam velocity is, as stated, about 750 feet per second.

At the high-pressure end the Rateau turbine is designed to work with partial admission, so that the guide blades cover, say, only one-sixth of the complete periphery of the
diaphragm. The fraction covered, of course, actually varies from diaphragm to diaphragm, and at the low-pressure end the guide-blades extend over practically the whole circumference.

These types of turbines, while preferable for the high-pressure end of a series and for high speeds, are inferior to the Parsons' type for low speeds and low pressures.

In marine steam turbines, where \( V_t \) must be low, \( V_s \) is correspondingly low, and a long range of expansion necessary. \( V_t \) is kept up as high as possible by large diameters of wheels, but upon this follows small lengths of blades at the high-pressure end, with consequent large leakage over the blade tops, but in the multiple wheel of

![Fig. 31.—Parson's early turbine.](image)

De Laval type, the wheel can be made large in diameter with large blades and small guide nozzles, hence it seems that the combination of the two types would have advantages.

The original Parsons' turbines were double-ended; the steam entered between two sets of turbine wheels and expanded towards both ends, where it exhausted. A sectional view of this turbine is shown in Fig. 31. It was perfectly balanced and required no driving pistons nor elaborate packing, and may have some advantages for low-pressure steam even now. It, however, presents difficulties otherwise, for it reduces the quantity of steam entering at the high pressure, thus making the first group of blades very small in height.

The introduction of the dummy pistons for balancing the pressures improved the turbine considerably, as also did the labyrinthine steam packing of the shaft, both designed on the same principles.
The principle is old and well known, and was used in an old air-pump, known as Deluel's, in which the piston moved free of the cylinder without packing. The piston was long and had grooves cut at intervals along its length (see Fig. 30); the air or gas which escaped past the first ring expanded into the first groove, losing in doing so some of its pressure. In passing from one groove to the next it again lost pressure, until it reached a stage at which it had no pressure, and ceased to flow any further, so that the gas formed its own packing, and there was no metallic friction between the piston and the cylinder.

Mr Parsons adopted this principle in his turbines, and it
STEAM TURBINES.
has since been adopted in all others. Any fluid flowing in a channel which has sudden changes of sectional area falls in pressure, and if the changes of section are numerous and sudden, the whole flow is checked. Fig. 32 shows a dummy piston packed in this way at P, against the high-pressure steam entering the high-pressure end at A. Brass rings are let into slots in the casing, and undercut grooves on the wheel fit these rings loosely, with faces close to the rings. The steam has not only to expand suddenly many times in its flow, but also to flow in a zigzag course, both of these devices retarding its flow (see Fig. 33).

The construction is shown more clearly in the enlarged Fig. 33. The turbine then became of the well-known construction shown in Fig. 34, where the dummy pistons, D.E.F., balancing the three groups of wheels are shown. The packing of the shaft where it passes through the casing is now carried out on the same principles. A section of a water-cooled packing is shown in Fig. 35. It consists of metallic rings; one set loosely fitting the shaft are split into segments and held in place by spiral springs; the other set fit into the bore and between the shaft rings, thus forming a labyrinthine packing which effectually closes the passage against steam.

Fig. 36 represents a section of a Parsons’ turbine with one diameter of rotor drum and three steps of blades, each step calculated to provide passages of correct sectional area for the steam as it expands in volume along the rotor. This construction is suitable only where there are several turbines in series, each with its own casing, forming large powers, such as in marine turbines. In other cases, where the whole power is in one casing, the rotor drums are stepped up in diameter, so as to increase the area of the passages without unduly increasing the length of the blades.

Referring to Figs. 37 and 38, the steam expands as indicated by the curves in which the blade lengths should
fit at their tips along the turbine stator. This cannot be done at a reasonable cost, but by stepping up the rotor as shown by the shaded parts, it is closely approached and the ratio of the height of the blades between the high and low-pressure ends is about 12 to 1.

In order to more clearly understand the design, it is good practice to take the case of a turbine of large power, and calculate back from the known steam consumption per horse-power hour of similar turbines already constructed and working, a course not practicable with a new type, but quite safe with old types well tried. The quantity of steam flowing in a turbine can be calculated from the area of the blade passages and its velocity.

In taking the area of the blades it is necessary, not only to take the actual area through the blades, but also that of the clearance space above them, and further to the quantity thus calculated there has to be added the leakage through the dummies, and also, if steam-packed glands are used, the quantity of steam required to pack these glands.

Both these items, especially the latter, are small ones, but when all these allowances are made, it is found that within errors of observation the quantity of steam used by a turbine is the calculated one. It is thus easy to calculate the quantity of steam used by a turbine, but the horse-power it will produce is another question.
Comparison of H.M. cruisers, Amethyst and Topaze, fitted with turbines and reciprocating engines. Vessels 360 feet x 40 feet x 14'6 feet, 3000 tons displacement. Designed speed, 21'75 knots, with 9800 I.H.P.

<table>
<thead>
<tr>
<th></th>
<th>Amethyst</th>
<th>Topaze</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>24 Hours' Trial at 10 Knots.</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>I.H.P.</td>
<td>897</td>
<td>897</td>
</tr>
<tr>
<td>Speed</td>
<td>10</td>
<td>10:058</td>
</tr>
<tr>
<td>Total water per hour</td>
<td>26,260</td>
<td>21,294</td>
</tr>
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<td>Water per I.H.P. per hour</td>
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<td>23.74</td>
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<td></td>
</tr>
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<td>I.H.P.</td>
<td>2250</td>
<td>2251</td>
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<td>Speed</td>
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<td>14.08</td>
</tr>
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<td>Total water per hour</td>
<td>44,090</td>
<td>42,260</td>
</tr>
<tr>
<td>Water per I.H.P. per hour</td>
<td>19.6</td>
<td>18.77</td>
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<td><strong>30 Hours' Trial at 18 Knots.</strong></td>
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<td>I.H.P.</td>
<td>4770</td>
<td>4776</td>
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<td>Speed</td>
<td>18.186</td>
<td>18.069</td>
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<tr>
<td>Total water per hour</td>
<td>76,493</td>
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<td>Water per I.H.P. per hour</td>
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<td>18.95</td>
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</thead>
<tbody>
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<td></td>
<td></td>
</tr>
<tr>
<td>I.H.P.</td>
<td>7280</td>
<td>6689</td>
</tr>
<tr>
<td>Speed</td>
<td>20.6</td>
<td>20.06</td>
</tr>
<tr>
<td>Total water per hour</td>
<td>100,606</td>
<td>134,248</td>
</tr>
<tr>
<td>Water per I.H.P. per hour</td>
<td>13.8</td>
<td>20.07</td>
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<table>
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<tr>
<th></th>
<th>Amethyst</th>
<th>Topaze</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>4 Hours' Trial at Full Power.</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>I.H.P.</td>
<td>{13,000}</td>
<td>9573</td>
</tr>
<tr>
<td>Speed</td>
<td>{23.06}</td>
<td>21.826</td>
</tr>
<tr>
<td>Water per hour</td>
<td>(176,845)</td>
<td>209,950</td>
</tr>
<tr>
<td>Water per I.H.P. per hour</td>
<td>13.6</td>
<td>21.93</td>
</tr>
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</table>
The vessels were each 350 feet long and of 3000 tons displacement. The turbine engines were similar to those in the case of the *King Edward*, only cruising turbines were added to give better economy at low speeds, while the reciprocating engines were of the usual triple expansion, Admiralty type. The result was that at all speeds above
14 knots the turbine was the more economical, being 15 per cent. higher than in the reciprocating engines.

A comparative test between H.M.S. *Amethyst*, turbine-driven, and a sister ship, the *Topaze*, with reciprocating engines, is here given, which shows the steam consumption in practice of a Parsons’ set of marine turbines.

For the purpose of calculation we may assume a turbine set of, say, 5000 horse-power, taking at full load on the turbine, 15 lbs. of steam per horse-power hour at initial pressure of 160 lbs. absolute, equal to a total mass of steam of $5000 \times 15 = 75,000$ lbs. per hour, or $\frac{75,000}{3,600} = 20$ lbs. per second. This is the first figure in the calculation.

And suppose the revolutions of the shafts to be 420 per minute, that is 7 per second, the diameter of the first ring of blades must be determined, and this entails fixing
upon the velocity of the wheel blades $V_t$, which in a marine turbine may be 140 feet per second. The revolutions being 7, the circumference of the ring of blades at its mean diameter would be $\frac{140}{7} = 20$ feet, say, 6 feet 6 inches diameter.

![Diagram](image)

**Fig. 39.**—Steam flow in Parsons' turbine.

If $V_t$ is 140 feet, $V_s$ being twice $V_t$ would equal 280 feet per second.

The quantity of steam per second must be reduced to cubic feet. The volume per lb. at 160 lbs. pressure is 2.8 cubic feet, hence $2.8 \times 20 = 56$ cubic feet. At a velocity $V_s$ of 280 feet per second, the guide passage area required would be $\frac{56}{280}$ equal to 0.2 square feet, the area of the passages between the first row of blades.

This area depends to a great extent upon the angle of the blades, as shown in Fig. 40, and varies between .37 and .86 of the actual clear area of the annular space between the rotor and stator. If taken at 0.5, the area would be $\frac{0.2}{0.5} = 0.4$ feet, which upon a 20-foot circumference would give $\frac{0.4}{20} = .02$ feet depth of space, or 0.24 inches.

This short length of blade would be difficult to deal with and have a large proportionate leakage at this high-pressure end. The only way to increase the length of blade is to decrease the diameter, giving a lower $V_t$. 


If $V_t$ were made 105, then $\frac{105}{7} = 15$ feet circumference, $V_s$ may remain the same 280 feet.

The depth of space would now be $\frac{0.4}{15} = 0.027$ feet, or 0.324 inches.

At the large end the volume would be about 120 cubic feet per lb., and the blade area correspondingly larger—42 times larger. The height of the blades, however, are rarely more than a fifth of the drum diameter, otherwise they fantail too much, and it is better to increase the drum diameter to get the increased area. This gives a higher $V_t$ and $V_s$ and a larger area with shorter blades.

![Fig. 40.—Influence of blade angle on area.](image)

The steam to be passed at low-pressure end is $120 \times 20 = 2400$ cubic feet per second. If $V_s$ is increased to 300, the area required will be 8 square feet, and if the blades occupy half the area, 16 square feet will be required in the clear area, and if $V_t$ is half $V_s$ at this end, it will be 150 feet per second, giving a circumference of $\frac{150}{7} = 21.4$ feet, say, 7 feet at the mean diameter and $\frac{16}{21.11} = 0.74$ feet the annular space between rotor and stator in depth, and 0.7 feet the length of the longest blades.

It might be an advantage to increase $V_t$ at the large end still more, if no objections arise against increase of diameter, by increasing $V_s$ to 400 and $V_t$ to 200, the blades then come out shorter at the low-pressure end.

There is compromise at every stage in the design. At the small end we have blades too short, and at the
large end too long, and these must be arranged with due regard to the velocities; but these velocities can be varied to meet other requirements within reasonable limits.

There does not seem to be any direct method of designing the turbine, except that of taking the steam consumption, the velocities, and initial and terminal pressures, and calculating the stages and number of wheels in series, from the thermal or pressure differences.

Small steam turbines are handicapped for land power installations where there are no condensing water available. The efficiency of a turbine is bound up with the condenser, which enables it to far more efficiently utilise the heat energy of the steam than a reciprocating engine; still there are many purposes in which power from a simple engine like a turbine, even although not highly economical, may be preferable to a reciprocating engine. Small turbines are better designed on the action principle, with large drops in pressure at each stage, and few wheels, and with velocities of from 30 to 60 feet per second of the wheel blades.

Wherever condensing water is available, the turbine is by far the most efficient steam motor, in fact, in many cases it pays to provide an artificial water supply, cooled by large cooling ponds or evaporative towers, to enable turbines to be employed for land power purposes.

At sea there is no water difficulty, and Mr Parsons has therefore given special attention to improving the condenser. From Mr Gerald Storey's Cantor papers the improved condenser's description here given is taken (Fig. 41). The chief improvement is the vacuum augmentor, whereby the last remainders of steam and air are effectually removed. It is combined with a special wet and dry air pump. The steam taken to work the augmentor is about 6 per cent. of the whole, but the gain in power is much more.

The design of condensers has been especially influenced by the introduction of steam turbines. As has been shown, in the old days of reciprocating engines, the condenser giving 25 inches vacuum was quite good enough, but now-a-days, on account of the great improvement in economy of steam turbines, with higher vacua, it is common to have between 28 and 29 inches. As a rule in the case of a condensing plant the temperature of the cooling water is fixed, and therefore, in order to obtain as low a temperature
Fig. 41.—High vacua condenser with augmentor.
of the outlet water from the condenser as possible, as large a quantity of cooling water as is practicable should be used. This again is limited by the power required to pump the water, and also by other considerations, especially where cooling towers are used, but as a fair average it is generally found that somewhere between 50 and 70 times the steam condensed can be obtained. This means a rise in temperature of the cooling water of about 17° Fahr., as it takes on an average about 1,000 B.T.U. to condense one pound of steam. This figure is rather lower than the one for dry steam, but it must be remembered that exhaust steam is nearly always wet, and therefore takes rather less B.T.U., and we have found in practice that 1,000 B.T.U. per pound of steam is a very fair figure to take.

The maximum vacuum which can be obtained from a condenser is the vacuum due to the temperature of the outlet water, and the closer to this we can get the vacuum actually obtained the better. There are two ways of expressing this difference: one is in inches of mercury, and the other is in temperature, and for condenser work the latter is the more convenient. When it is remembered that from about 24 to 27 inches, each inch of vacuum makes 4 per cent. difference in the steam consumption of a turbine, between 27 and 28 inches about 5 per cent., and from 28 to 29 inches 6 or 7 per cent., or that approximately 3° Fahr. difference in the temperature of the exhaust means an increase or decrease of about 1 per cent., it is easily understood how important it is to keep the difference of temperature between the outlet water from the condenser, and the temperature due to the vacuum as small as possible. This difference in good modern condensers, when condensing, say 12 lbs. per square foot per hour, can be kept as low as 5° or 6° Fahr.

Another way of looking at the efficiency of the condenser is the B.T.U. transmitted per square foot of cooling surface per hour per 1° Fahr. difference of temperature, and this figure can in well-constructed condensers be as high as 1,000 to 1,200 B.T.U. The resistance to the heat passing from the steam to the water may be considered in three stages. There is first the heat transmitted from the steam to the tubes of the condenser, and this resistance is affected by the quantity of air in the condenser, and the efficiency of the air pump. With suitable arrangements,
however, this resistance can be reduced to a very small figure, especially if appliances are used, such as dry-air pumps, or, still better, Mr Parsons' vacuum augmentor, to withdraw the air completely from the condenser. If air is present it not only vitiates the vacuum, but also reduces the rate of condensation of the steam by causing a blanket of air to form round the tubes, and thus preventing fresh steam getting to them. The second resistance to the transmission of heat is in the metal of the tube itself, but this, with metal tubes, such as are always used, is an exceedingly small figure, and may be neglected. The third and last is the resistance to the passage of heat between the metal of the tubes and the cooling water, and this is apparently one of the principal losses, and varies enormously with the cleanliness of the tube. If there is any slime or dirt or deposit on the inside of the tube, it is found that conduction of heat very rapidly goes down, and therefore to get the best results the tubes must be kept clean. In this connection also it is necessary to have sufficient velocity of flow of the water to make turbulent flow in the tubes and not streamline flow, that is, a velocity sufficient to make the water mix up as it is travelling along the tubes and not to have a cold core of water with a hot envelope outside it next the tubes.

It is in connection with the extracting of air thoroughly from the condenser that the greatest improvements have been made of late years, and amongst these dry-air pumps, and the vacuum augmentor mentioned above and shown in Fig. 41, are especially prominent. This latter consists simply of a jet of steam drawing the air and vapour from the condenser and delivering it through a small auxiliary condenser to the air pump, and thus, although the air pump may only produce a vacuum of, say, 27 or 28 inches, there may be a vacuum of 28 to 29 inches in the condenser, and in practice this appliance has been found most satisfactory.

On these same points the remarks of Mr Speakman at the reading of his paper on Parsons' steam turbines are of interest:—

Considerable modifications have been found desirable in the proportions of condensers and air pumps for working in conjunction with turbines, due to the greatly increased volume and reduced temperature of the steam. With well-designed apparatus, a vacuum of from 27 to 28
inches can generally be maintained with about 1 square foot of surface per I.H.P., and a circulation at 70° Fahr. of 30 times the feed water. A vacuum, however, of from 27 to 28 inches will affect a saving of about from 5 to 6 per cent. in the steam consumption of the main unit, but to obtain this, a higher ratio of cooling surface to H.P.—possibly 1.4 square feet—must be allowed, as well as a considerable increase in the quantity of circulating water—never a difficult matter in marine work, where, however, the surface is rather less. A proportionally larger air pump should be employed as well, but the additional power required for this and the larger circulating pump will not exceed from 1 to 1.5 per cent. on the total power, leaving a net gain in economy of about 4.5 to 5 per cent due to the high vacuum.

The action of the augmentor is to draw off the residual air to a much greater extent than is possible with ordinary pumps alone, and the condensation takes place with much greater rapidity and allows a great reduction in the cooling surface, provided that the circulating water is not diminished in volume, and that the velocity through the tubes is kept about 5 feet per second, the total quantity being equal to at least 50 times the feed water.

The following table illustrates the gain of energy due to these higher vacua, assuming adiabatic expansion from 150 to 1.5 lbs. absolute:

<table>
<thead>
<tr>
<th>VACUUM.</th>
<th>26 in.</th>
<th>27 in.</th>
<th>28 in.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Temperature, degrees F.</td>
<td>126.5</td>
<td>116</td>
<td>102</td>
</tr>
<tr>
<td>Volume per lb. cubic feet</td>
<td>137.5</td>
<td>177.5</td>
<td>256</td>
</tr>
<tr>
<td>B.Th.U.'s available per lb.</td>
<td>277</td>
<td>293</td>
<td>312</td>
</tr>
<tr>
<td>Increase in B.Th.U.'s</td>
<td>—</td>
<td>16</td>
<td>19</td>
</tr>
<tr>
<td>Increase per cent.</td>
<td>—</td>
<td>5.77</td>
<td>6.48</td>
</tr>
</tbody>
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To obtain a similar increase in available energy by merely increasing the pressure above 150 lbs., compared with reducing it from 27 to 28 inches, it would be necessary to go up to over 200 lbs. absolute, while in the case of the 26-inch vacuum it would be necessary to go to over 250 lbs., and either pressure would greatly increase the weight and cost of turbines and boilers.

The vacuum augmentor or intensifier is almost a
necessity with exhaust steam turbines, and several other types of augmentors have been proposed, so that the continuous steam jet may not be required. It is less expensive to produce a jet effect mechanically. This is done in effect in the exhaust steam jet ejector condenser,

where a jet of cold water, or several jets of cold water, are employed to produce a high vacuum.

Jet ejector condensers cannot, however, be employed as main condensers, for the condensed steam which forms good feed water for the boilers is lost, being mixed with the condensing water.

But a jet ejector can be used as an intensifier, for the small quantity of steam it removes with the remaining air is negligible. The best of these ejectors at present is the

Fig. 42.—Leblanc high vacua jet vacuum augmentor.
Leblanc, used by the Westinghouse Company on high vacuum work. It is described by M. Maurice Leblanc in a paper to the Ass. Technique Maritime, Paris, 1908. Fig. 42 shows this apparatus in section. It will be seen to be an ejector condenser, in which the water jet is given the necessary velocity by an inverted turbine wheel; the wheel is driven by a belt or other gearing and water enters from the inner ends of the blades; it is thrown out and downwards in sheets or flakes with a velocity high enough to overpower the outer pressure in the ejector expanding cone. The velocity is calculated to operate against atmospheric pressure 15 lbs. per square inch or 34 ft. head. The velocity will be \( V = 8 \sqrt{h} = V = 8 \sqrt{34} \), about 48 feet per second. The sheets of water flowing down at this rate form water pistons, gradually narrowing in area and forcing all air and steam before them through the narrow ejector throat, through which there is no return. The vacuum attainable is very high, but a point is soon reached at which the cost of producing the high vacuum is greater than the value of the additional power obtained, and this fact must not be forgotten in considering vacuum intensifiers.

It has been found that in the reciprocating engine there is little use, considering the temperature at which the feed water is returned to the boiler, in going to a better vacuum than 25 inches. With the steam turbine, vacua of 28\( \frac{1}{2} \) or 29 inches, or absolute pressures of from \( 3 \) to \( 1 \) lb. per square inch, can be utilised, as the difficulty of dealing with large volumes of steam does not occur in the case of the steam turbine as in the case of the reciprocating engine, and it has been found that with the steam turbine the gain due to vacuum goes steadily on up to the highest attainable vacua. Between 25 and 26 inches, or 26 and 27 inches, there is a gain of about 4 per cent.; a further gain of 5 per cent. is made with the vacuum increased to 28 inches, and a still further gain of 6 to 7 per cent. when it is increased to 29 inches.

This is more easily understood if we consider that the theoretical power to be derived from the steam is almost proportional to the logarithm of the expansions, and thus practically the same power can be obtained working from 400 lbs. to 1 lb. absolute, or 28 inches vacuum, as from 200 lbs. to \( \frac{1}{2} \) lb., or 29 inches vacuum. In each case there are 400 expansions by pressure, and in each case the
theoretical consumption of steam by Clausius' cycle would be about 9·3 lbs. per kilowatt hour. With 150°F. superheat this would come down to 8·7 lbs., and under the conditions of 200 lbs. pressure and 29 inches vacuum with 150°F. superheat, 13·2 lbs. per kilowatt hour has actually been obtained with an overall efficiency, including the alternator, of about 66 per cent., or 71½ per cent. on the turbine shaft, allowing for the electrical losses. Professor Ewing, in his book on *The Steam Engine*, gives a list of principal results obtained from condensing reciprocating engines, and in no case does the ratio of the consumption of steam by Clausius' cycle, compared with that used per indicated horse-power, exceed 64 per cent. As the ratio of brake horse-power to indicated horse-power is never more than 90 per cent., this means an efficiency at the engine shaft of not more than 58 per cent.

Considering the steam turbine makes, the best use of the low-pressure steam and the reciprocating engine has some advantages for the high-pressure steam. It was proposed early to use the two combined on vessels of smaller sizes, but whether the gain in power will compensate for the extra cost of multiple shafts remains to be seen. A more promising direction in which the turbine
has already achieved success is the exhaust steam turbine utilising the steam exhausted from ordinary engines.

Where good condensing is possible, much power may be thus obtained.

The utilisation of the exhaust steam from reciprocating engines is made advantageously possible by means of specially designed turbines, particularly in those cases where the intermittent but frequent working of the main engines gives rise to difficulties in the application of condensing plants, and consequently the majority of such engines for winding purposes or rolling-mills, or steam-hammers, discharge directly into the atmosphere, the quantity of steam lost being considerable.

Mr Parsons has patented the combination of a reciprocating engine and a low-pressure turbine arranged in series, so that when the high-pressure steam has been duly expanded in the engine it passes on to the turbine and thence to the condenser. This system was adopted for cruising purposes in H.M.S. Velox, the engines being placed on the main shafts and disconnected at the higher speeds, while numerous sets of independent low-pressure turbines have recently been installed in steel works and breweries.

If a pound of steam, at 150 lbs. absolute pressure, be adiabatically expanded to atmospheric pressure, the available B.Th.U.'s per lb. are about 165, while a further expansion of 27 inches of vacuum yields another 130, an increase of 80 per cent., or 90 per cent. if expanded to 28 inches of vacuum. These high vacua being economically and easily obtainable under the modified conditions that turbine engineering has evolved, admit of a proportionate gain in power, with a very slight increase of capital cost in the case of an exhaust turbine, while the proportions of reciprocating engine cylinders and valves at the very low pressures forbid the use of ordinary engines for the purpose.

A recently installed Parsons' exhaust turbine of 250 kilowatts, working with steam between 2 and 2½ lbs. absolute pressure, consumed only 34½ lbs. per kilowatt hour, giving an efficiency of 57 per cent. compared with the Rankine cycle. Such turbines are of the ordinary type, the proportions of the blading being suitably modified to allow for the altered conditions.

Prof. Rateau has also done a considerable amount of work on this subject, employing a suitable turbine of
his own type in conjunction with a steam regenerator accumulator that forms an essential feature of his installations. This accumulator is intended to regulate the intermittent flow of steam from the piston engines before it passes to the turbines, its function being to absorb the heat units in the steam while the main engines are exhausting, and to re-evaporate the water condensed therein when the engines are at rest.

Several of these turbines are at work in England and on the Continent, one of the largest being at the works of the Steel Company of Scotland. This turbine, of about 450 k.w., during a test made in January 1906, consumed about 36'0 lbs. per k.w. hour, with steam of 11'4 lbs. absolute and a vacuum of 27'9 inches, the revolutions being 1,500, and the generator of the direct current type.

When working with live steam, wiredrawn in pressure from that of the boilers—perhaps 140 lbs.—down to atmospheric pressure, as must occur during long periods of rest of the main engines, a considerable degree of superheat is imparted to the steam, increasing the efficiency about 7½ per cent. over that obtained when working with ordinary exhaust steam.

The simplicity of such installations and the great gain in power on the same steam consumption effected by their use will certainly result in a much wider application in the near future.
The number of revolutions is necessarily high, even in large turbines. This cannot well be overcome; but several systems of power transmission have been suggested, whereby the high-speed turbine may be geared to slow-speed shafts. De Laval’s wheel gearing is applicable only to small powers.

Electrical transmission has been tried, but it is costly, bulky, and on a large scale, where it would be most required, is far beyond any electric power we have experience of. Hydraulic and compressed air transmissions have also been proposed.

But, after all, marine engineers have preferred to face the high speeds without any intermediate gearing. Direct coupling may have some difficulties, but they seem to be less than with any form of gearing hitherto proposed.

Parsons, at a very early date, met the necessity for reversing the ship’s motion by placing a reverse turbine on the same shaft and in the same casing as the low-pressure turbine, and employing three propellers, the middle one on the high-pressure turbine and the two side shafts on two low-pressure turbines, as shown in Figs. 43 and 44. The reversing turbines are, therefore, only applied to the two side propellers. A sectional diagram of the arrangement of the low-pressure turbine, with the inside reversing turbine, is shown at Fig. 45.

The stationary rings of vanes of the main turbine are fixed, as is usual, to the casing c, the moving rings being attached to the drum o, which is fixed to the shaft f.
The reversing turbine has its fixed rings of blades attached to the exterior of the cylinder \( p \), which is fixed to the casing \( d \), while its moving rings are carried by the casing \( 7 \), which is rigid with the drum \( o \). The steam enters the main turbine at \( n \), while it gains access to the reversing turbine by the pipe \( r \). The exhaust ends, \( g \) and \( h \), of both turbines open directly into the condenser passage \( k \).

Many have attempted the design of a reversing turbine, but none have succeeded. A turbine employing flat blades would reverse by simply reversing the direction of the steam flow, but such turbines are very inefficient, and use more than twice the steam per horse-power.

The reversing question and the speed-reduction questions are still open for settlement, and many schemes have been advanced for a solution of the problems.
CHAPTER IV

DESCRIPTION OF SOME STEAM TURBINES

CALCULATING THE PRINCIPAL DIMENSIONS

In Chapter III. the method for calculating out the principal dimensions of the rotor and stator elements of the turbine has been already explained and illustrated by worked examples. In this work it is not intended to enter upon the discussion of the dimensions of the metallic wheels, strength and rigidity, nor of the proper dimensions of castings, casings, pipes, bolts and nuts, nor of the lubricating system, foundations and couplings. All these mechanical details are of importance, and form the subject-matter of another work for the use of engineering draughtsmen, whose duty it is to proportion the parts to the work they have to perform or stresses to bear.

The principal dimensions of the stator and rotor of the turbines are those with which we have to deal here. These being determined, the mechanical design follows; but different draughtsmen would no doubt make out different designs, while all adhered to the principal dimensions.

In the Parsons' type of turbine we have a total drop of heat energy between the pressure of steam at the first row of blades and that at the last row.

The velocity of the turbine blades is generally a given quantity, \( V_t \). \( V_s \), the steam velocity, is generally about \( V_t \times 2 \), hence, if \( V_t \) were chosen or given as equal to 100 feet per second, \( V_s \) might be taken as 200 feet per second.

Now, by (12) \( \text{B.Th.U.} = H \left( \frac{T_1 - T}{T_1} \right) \), from which we can
find the B.Th.U. in the total drop from initial to condenser pressures.

From (10) and (11) a table can be calculated and a curve made, showing the B.Th.U. required to produce a steam speed required; this is shown in Fig. 46. Suppose \( V_s \) is to be 200 feet per second, then referring to the curve we find \( V_s \) corresponds about 0.8 B.Th.U. per row of blades. Then, if for instance we by (12) found B.Th.U. for the whole range of temperature equal to, say, 240 units, then the number of rows of blades or wheels in series would be \( \frac{240}{0.8} = 300 \), if the drop in heat energy was made equal in each. At a steam velocity of 500 feet per second—not an uncommon velocity—the B.Th.U. per row is shown in the curve as equal to 5, hence in that case \( \frac{240}{5} = 48 \) rows of blades or wheels in series.

In the Parsons' and some other turbines, it is the practice to increase the diameter of the wheels as they approach the exhaust end, and also to increase \( V_s \). In these cases the method of calculation is to divide the
steam turbines. turbine into stages, and to take the energy drop in each stage, with its particular \( V_s \), and calculate each separately.

In order to determine the rotor area of blade passages, the first row is calculated out, and also that of the first row of guide blades, then the last row area should be found.

As we have the steam velocity, \( V_s \), the area can be found from the volume of steam to be dealt with, if \( a \) is the area in feet \( V_s a = C \), the volume in cubic feet.

\[ a = \frac{C}{V_s} \]

\( C \) is the actual volume of the steam flow.

If the initial steam pressure is 160 lbs. and the final pressure 3 lbs. absolute, the volumes will be, by table I., 2.79 and 117.5 respectively, so that the blade passages would be approximately as 2.79 at the high-pressure end to 117.5 at the low-pressure end in clear area.

Hence, if the turbine is to be 1000 H.P., and is one known to take, say, 15 lbs. per horse-power per hour, the steam required will be \( \frac{15 \times 1000}{3600} = 4 \) lbs. per second approximately.

\( C \), the volume at the high-pressure end would be equal to \( 4 \times 2.79 = 11.16 \) cubic feet per second.

Now, suppose \( V_s = 300 \) feet per second, the area of the blade passages would be \( \frac{11.16}{300} = 0.037 \) square feet = \( a \).

Now, from Fig. 40, it will be seen that the obliquity of the blades has a great influence on the area of blade passages. The two extremes of obliquity to the axis is shown 68°, making \( a \) the area of blades in the annular space \( A \) between the base rotor and stator, drum and casing equal to \( a = A \times 0.38 \); and 30°, making \( a = A \times 0.86 \).

The greater angle would be used at the high-pressure end in order to lengthen the blades as much as possible; while the smaller angle would be used at the low-pressure end where the passages have to be wide and the blades long.

As the space \( A \) comes out very short at the high-pressure end if \( Vt \) is the same there as it is at the low-pressure end, the value of \( Vt \) is made less at the high-pressure end, \( V_s \) we will assume to be the same at all the
turbine stages = 300 ft. per second. Then at the high-pressure end \( V_t = \frac{300}{3} = 100 \), and at the intermediate stage, \( V_t = \frac{300}{2.5} = 120 \), and in the low pressure, \( V_t = \frac{300}{2} = 150 \).

These values are chosen only as examples; both \( V_t \) and \( V_s \) may vary in the different turbine stages, and turbines have been made with all the blades on the same diameter of drum. It is the method of calculation we are dealing with, not the actual values, of every possible modification. The higher \( V_t \) at the low-pressure end enables shorter blades to be used there, and the lower \( V_t \) at the high-pressure end enables the use of longer blades there.

The circumference of the rotor at the middle of the blades is of importance, it is equal to \( \frac{V_t}{n} \) where \( n \) is the revolution per second. In a turbine of 1000 H.P. for dynamo driving, probably \( n \) would be chosen at 20 per second, hence, circumference equals \( \frac{100}{20} = 5 \) feet at the high-pressure end, and \( \frac{120}{20} = 6 \) feet at the intermediate stage, and \( \frac{150}{20} = 7.5 \) at the low-pressure end, with diameters 1.6 feet, 2 feet, and 2.4 feet respectively. The area of the first stage blade passages we have found to be 0.09, but allowing for blade-edge thickness, it might be 0.1 square feet, to pass 4 lbs. of steam per second at 160 lbs. pressure initial. At the low pressure and intermediate stages the area must be found later, when the initial pressures at these stages have been found.

It is necessary now to find the heat energy in the drop of temperature at 160 lbs. pressure = 363 degrees to that at 3 lbs. pressure = 141 degrees temperature.

By (12) \( B \text{.Th.}\text{U.} = H\left(\frac{T_1 - T}{T_1}\right) \) and
\[
T_1 = t_1 + 460 \text{ and } T = t + 460, \text{ from table I.}
\]
\( t_1 \) for steam at 160 lbs. absolute pressure = 363.4,
\( t \) for steam at 3 lbs. absolute pressure = 141.6,
hence, \( t_1 + 460 = 363.4 + 460 = 823.4 = T_1 \),
and, \( t + 460 = 141.6 + 460 = 601.6 = T \),
\( H \) for steam at initial pressure of 160 = 1192.2,
\[
\ldots B \text{.Th.}\text{U.} = 1192.2 \left(\frac{823.4 - 601.6}{823.4}\right) = 321
\]
the total energy. Now here again the judgment of the designer may be exercised, for he may divide the energy equally between the three stages of the turbine, giving it out in each \( \frac{321}{3} = 107 \) units, which we will here do, but the energy may be divided unequally if necessary.

To find the pressure at the end of the first stage we must find the temperature of the steam there by (13), or by

\[
t = t_1 - \left( \frac{V_s^2}{JH} \times T_1 \right)
\]

\( V_s \) here equals 300,
\( J = 778, \)
\( H = 1192, \)
\( t_1 = 363, \)
\( T_1 = 823 = t_1 + 460, \)

hence

\[
t = 363 - \left( \frac{300 \times 300}{778 \times 1192} \times 823 \right) = 280,
\]

which from table I. corresponds to a steam pressure of 50 lbs. per square inch.

Similarly for the end of the intermediate stage the pressure will be,

\[
t = 280 - \left( \frac{300 \times 300}{778 \times 1188} \times 740 \right) = 206^\circ = 13 \text{ lbs.}
\]

Similarly for end of third stage,

\[
t = 206 - \left( \frac{300 \times 300}{778 \times 1145} \times 666 \right) = 140 = 3 \text{ lbs.,}
\]

we can now calculate the area of the blade passages in the high, the intermediate, and low-pressure turbines.

The volume of the steam to be passed in the intermediate turbine = \( 4 \times 8.3 = 33.2 \) cubic feet, and in low-pressure turbine = \( 4 \times 30 = 120 \) cubic feet.

Now in practice the steam should expand gradually through all the stages, and therefore each row of blade should be a little longer than the preceding one (see curves, Figs. 37 and 38), but that is too costly a construction, it is, therefore, usual to make the stages all one diameter. The volume at the first row is 11 cubic feet. Referring back to page 95, the ratio of the area of the exit from the blades to that of the annulus is about \( \frac{1}{3} \), hence the length of blades is equal to—
Height of blades = \left( \frac{\text{Clear area of annulus in square inches} \times 3}{\text{Mean circumference in inches}} \right).

11 cubic feet is the volume at the high-pressure end, \( V_s = 300, \frac{11}{300} = 0.037 \) square feet clear, and with blades = \( 0.037 \times 3 = 0.111 \) square foot, mean circumference = 5 feet,

Height of blades, \( \frac{0.111}{5} = 0.022 \) feet,

\( 0.022 \times 12 = 0.264 \) inches.

In the intermediate stage the volume is 33; hence, \( \frac{33}{300} = 0.11 \) square feet clear area, and with blades \( 0.11 \times 3 = 0.33 \), and the mean circumference is 6 feet; hence, height of blades, \( \frac{0.33}{6} = 0.055 \) feet equal to \( 0.055 \times 12 = 0.660 \) inches.

In the low-pressure end the volume is 120, \( \frac{120}{300} = 0.4 \times 3 = 1.2 \), divided by 7.5 = 1.6; \( 0.16 \times 12 = 1.92 \), nearly 2 inches.

From these results it is easy to calculate the diameter and bore of the rotors and stators.

As the drop in heat energy is the same in the three divisions of the turbine, the number of wheels in series will be the same in each stage.

Theoretically, the turbine wheels should gradually increase in length from the first wheel to the last; and in some of the Westinghouse turbines the steam is first let down in pressure through two impulse wheels and then expanded through a series of wheels, each one of the series being longer than the preceding one, so that their tips touch the curve (Fig. 37), and the steam expands as it should do in an expanding nozzle.

In the turbine under consideration, we have three groups of wheels; the wheels in series will number according to the drop in heat energy from the first wheel in the series to the last; the drop we have found to be 107 B.Th.U., and from the curve (Fig. 46) we find the B.Th.U. corresponding to the velocity \( V_n = 300 = 1.85 \), hence the wheels in series are equal to \( \frac{107}{1.85} = 56 \) nearly.
The number of blades on each wheel is found by taking the circumference and dividing by the space from centre to centre of each blade.

Blading material is now made of standard shape and sizes; drawn rods of long length can be bought both for the blades and the caulking pieces. They run from $\frac{1}{16}$th in breadth up to $\frac{3}{8}$th inch. When narrow blades are employed, they must be close together; and hence numerous. A good breadth of blade is $\frac{3}{8}$th inch, these space out about $\frac{1}{8}$th inch apart, making a wheel with a moderate number of blades.

On a wheel of mean circumference, equal to 5 feet or 60 inches, the number of blades would be $60 \times 8 = 480$.

It has now been shown by actual calculations how to find the principal dimensions of a 1000 horse-power steam turbine, with a speed of 1200 per minute; initial pressure, 160 lbs.; final pressure, 3 lbs.

The design might, of course, be considerably varied; high-steam velocities and blade velocities may be taken at the low-pressure end.

A rule, found from experience, has already been given, page 21, in inches.

Mean diameter or wheels = \(\frac{Vt \times 228}{\text{revolutions per minute}}\) or \(\frac{Vt \times 3.8}{\text{revolutions per second}}\).

This would give for our example at the low-pressure end, \(\frac{150 \times 3.8}{20} = 2\) feet 4 inches.

Then it has also been found that the blade heights should not be less than 3 per cent. of the mean diameter, nor more than 15 per cent.; with smaller heights the percentage of leakage is high, and with longer blades the vibration is excessive.

It has already been indicated on pages 46 and 47, how to calculate out the principal dimensions of the impulse wheels, like the Rateau and Zoelly turbine wheels.

In the Parsons' type, the thrust on the blades is partly due to reaction at exit and partly to impulse at inlet. At inlet the pressure is equal to \(\frac{W(V_8 - V_t)}{32}\); at the outlet of the blades it is due to \(2Pa\), where \(p\) is the pressure of the
FIG. 47. — Impulse turbine design.
steam in the wheel, and \( a \), the area of the outlets in square inches, the same as in a Vortex turbine wheel.

But in the purely impulse wheel, like the Rateau and Zoelly, the thrust is due to the impulse of the steam jets deflected through nearly two right angles in the wheel blades, and the thrust is proportional to \( \frac{2W}{(V_s - V_t)} \)

very nearly, and the fall in energy is nearly twice as great in each wheel as it is in a reaction wheel.

Fig. 47 represents this type of design, each wheel being in a separate compartment. The calculations are much the same as that already given. The total heat energy in the steam between initial and final temperatures is, however, divided up into fractions, each representing one wheel; the best ratio of \( V_s \) to \( V_t \) is 2 to 1, but as this type of turbine may be a partial-flow turbine, we can avoid the use of small blades in the wheels and use only two or three jets at the high-pressure end, and the full blast at the low-pressure end, and the ratio of \( V_s \) to \( V_t \) may be the same throughout.

If from curve we find \( V_s \) corresponds to, say, \( 89 \) B.Th.U. and the total heat is \( 321 \), then the number of stages would be \( \frac{321}{9} = 36 \) wheels,

\[
V_s \text{ would be } = 223 \sqrt{\text{B.Th.U.}} = 223 \sqrt{9} = 669,
\]

\[
V_t \text{ may be } \frac{1}{4} \text{ of } V_s = \frac{669}{4} = 167 \text{ per second.}
\]

As an example of the design and dimensions of the Zoelly impulse turbine, a 6000 k.w. steam turbine may be taken, and by analysing the dimensions, we can arrive at the principal dimensions, and see the reasoning in the design.

6000 k.w. = 8140 H.P. nearly.

This table, unfortunately, gives only the steam consumption for the power given out at the dynamo terminals. From column VIII. at full load we find the dynamic efficiency \( 71.5 \) per cent. The efficiency of the alternator is \( 96.2 \) per cent. or a loss of only \( 3.8 \) per cent., so that the loss in the turbine is found from the total loss \( 100 - 71.5 = 28.5 - 3.8 = 24.7 \) per cent. The thermal dynamic efficiency on the turbine shaft would, therefore, be \( 100 - 24.7 = 75.3 \).
TABLE I.—Summary of Steam Consumptions, etc., in lbs. per kw-hr.
(Measured at Alternator Terminals, excluding Auxiliaries.)

<table>
<thead>
<tr>
<th>Col.</th>
<th>I. Howden's Guarantee with 27&quot; Vacuum, Total Steam Temperature 523° F. (140° F. Superheat), Barometer 30 in.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Col. II.</td>
<td>&quot;&quot; corrected to 28&quot; Vacuum &quot;&quot; 523° &quot;&quot; (140°) &quot;&quot;</td>
</tr>
<tr>
<td>Col. III.</td>
<td>Actual Test Results with 27·56/28·75&quot; Vacuum &quot;&quot; 514/532° &quot;&quot;</td>
</tr>
<tr>
<td>Col. IV.</td>
<td>&quot;&quot; corrected to 28&quot; Vacuum &quot;&quot; 514/532° &quot;&quot;</td>
</tr>
<tr>
<td>Col. V.</td>
<td>Results of Column IV. corrected for a &quot;&quot; 523° &quot;&quot; (140°) &quot;&quot;</td>
</tr>
<tr>
<td>Col. VI.</td>
<td>Column III. &quot;&quot; &quot;&quot; &quot;&quot; 483° &quot;&quot; (100°) &quot;&quot;</td>
</tr>
<tr>
<td>Col. VII.</td>
<td>Alternator Efficiencies.</td>
</tr>
<tr>
<td>Col. VIII.</td>
<td>Thermo-dynamic Efficiencies.</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th></th>
<th>Col. I.</th>
<th>Col. II.</th>
<th>Col. III.</th>
<th>Col. IV.</th>
<th>Col. V.</th>
<th>Col. VI.</th>
<th>Col. VII.</th>
<th>Col. VIII.</th>
</tr>
</thead>
<tbody>
<tr>
<td>25 per cent. Overload</td>
<td>15·1</td>
<td>14·4</td>
<td>14·9</td>
<td>14·2</td>
<td>14·1</td>
<td>15·3</td>
<td>96·4 per cent.</td>
<td>70 per cent.</td>
</tr>
<tr>
<td>Full Load</td>
<td>-</td>
<td>-</td>
<td>14·4</td>
<td>13·7</td>
<td>14·6</td>
<td>13·9</td>
<td>13·8</td>
<td>15·0</td>
</tr>
<tr>
<td>½-Load</td>
<td>-</td>
<td>-</td>
<td>15·4</td>
<td>14·6</td>
<td>15·0</td>
<td>14·3</td>
<td>14·4</td>
<td>15·6</td>
</tr>
<tr>
<td>¾-Load</td>
<td>-</td>
<td>-</td>
<td>16·7</td>
<td>15·9</td>
<td>15·9</td>
<td>15·1</td>
<td>15·2</td>
<td>16·4</td>
</tr>
<tr>
<td>¾-Load</td>
<td>-</td>
<td>-</td>
<td>19·8</td>
<td>18·8</td>
<td>17·5</td>
<td>16·7</td>
<td>16·7</td>
<td>18·1</td>
</tr>
</tbody>
</table>
From column III. we find the actual steam consumption =14.6 per K.w. hour, 1 K.w. = 1000 watts, 746 watts = 1 H.P., therefore the consumption is equal to—

\[
\frac{1000 \times 14.6}{746} = 20 \text{ lbs. per H.P. hour.}
\]

Twenty lbs. per horse-power hour at alternator terminals, and 3.8 per cent. less at the turbine shaft = 19.25 lbs. per horse-power hour.

This superheating very much improves the economy, and as it is likely to be much more extensively applied, this turbine affords an example showing how the superheating is taken into account in the calculation. H, the total heat in the steam at any saturated pressure we readily get in tables like table I.

For superheated steam, H, the total heat is found by the general formula given by Prof. Peabody—

\[
H = 0.48(T - 10.38(\sqrt{P}) + 857.
\]

From the particulars given of this turbine the temperature of the steam was 523° with 140° superheat, and therefore 523 - 140 = 383°, which corresponds very nearly to 200 lbs. pressure per square inch, or 28,800 lbs. per square foot.

In above formula, \( T = 460 + t = 460 + 523 = 983° \), and \( P \) is lbs. per square foot, hence

\[
H = 0.48(983 - 10.38 \sqrt{28,800}) + 857 = 1264.
\]

The low temperature of the vacuum is 126°, hence from formulae

\[
\text{B.Th.U.} = H\left(\frac{T}{T_1}\right) \text{and } T_1 = 983 \text{ and } T = 586,
\]

hence

\[
\text{B.Th.U.} = 1264\left(\frac{983 - 586}{983}\right) = 506.
\]

Total thermal units per lb. of steam.

There are twenty wheels, hence the drop in each equals \( \frac{506}{20} = 25.1 \) B.Th.U., from which we can find—

\[
V_s = 223 \sqrt{\text{B.Th.U.}} = 223\sqrt{25.1} = 1115
\]

feet per second.

In the impulse type of turbine, \( V_t \) may be \( \frac{1}{4} V_s \), hence,

\[
V_t \text{ may be } \frac{1115}{4} = 278 \text{ feet per second,}
\]

\( R \) the revolutions = 1000 per minute or 16.6 per second,
Mean circumference of ring of blades \(\frac{Vt}{\text{revols. per second}}\), hence, Circumference, \(\frac{278}{16.6}\) = 17 feet, corresponding to a diameter of 5.4 feet at the mean.

The outside diameter given is seen from following maker’s statement:

- Diameter of high-pressure turbine over blading - 67\(\frac{3}{4}\) inches.
- Diameter of low-pressure turbine over blading - 90\(\frac{1}{8}\) "
- Diameter of turbine shaft - - - - 11 "
- Revolutions per minute - - - - 1000 "
- Overall length of turbine and alternator - - - 40 feet.

The low-pressure turbine is increased in diameter in order to reduce the length of the blades. The steam velocity \(V_s\) and the blade velocities \(V_t\) are both increased proportionately, so that we can calculate, as has been done here, as if the mean diameters of all wheels were equal. Or two calculations can be made, one for the eleven wheels in high-pressure end, finding the low pressure there as the initial pressure for the low-pressure end.

The student can work this example out from the other starting-point, given \(V_s, V_t,\) steam pressure, and superheat, the horse-power, and vacuum, and the consumption per horse-power or kilowatt, and revolution per second.

The usual bye-pass arrangement is provided, by which high-pressure steam can be fed into the lower stages if an emergency load is required, and with this it is possible to increase the output of the machine to even 10,000 K.w. if necessary. Without the pass-valve being opened the turbine will comfortably develop 8000 K.w.

The makers claim the following advantages for their turbine:

The turbine has only from one-third to one-fifth the number of blades of the ordinary multistage reaction turbine for the same power and speed, and the average velocity of the steam being low, the wear on the blades is practically negligible.

There is ample clearance between all the moving and stationary parts, and no adjustment is necessary either in the thrust or the main bearings.

As the principle of the turbine entirely eliminates the axial thrust, balancing pistons are not required, and the leakage incidental to such pistons is avoided.
The greatest advantage can be taken of superheat, as, owing to the form of casing adopted, there is no danger of warping. The turbine is not affected by great variation of temperature, and will work quite satisfactorily with steam at a temperature as high as 700° F.

The lubrication is entirely automatic, and the cost of oil is very little on account of its being used over and over again.

The governing is exact and instantaneous, and the speed practically remains constant under widely varying loads; as steam is not admitted in gusts, as in the case of most reaction turbines, the paralleling of alternators is very much simplified.

As no part of the turbine except the bearings is lubricated, the exhaust steam is pure, and may be condensed and used as feed-water.

Each running wheel is separately balanced, and after erection the whole of the rotating system is balanced, the result being that the turbine runs dead true, no vibration whatever being perceptible.

The steam consumption does not increase after running some time, as is the case with other types of turbines, and is independent of the attention and skill of the engine-room attendants.

The turbine having been adopted for the largest steam- ships, and for large generating electricity units, has received much attention from the best of engineers, and is now in a high state of perfection. Small steam engines are rapidly being displaced by the gas, the oil, and the electric motor, so that small turbines are not of much importance. They present difficulties for small powers due to their high velocities, and gearing is necessary for most drives by small turbines.

One line of investigation seems to the author to be capable of development, and that is in the direction of studying the effect of different guide blades with the different wheel blades.

In the Parsons' turbine the guide blades and wheel blades are similar in shape and form, although their functions in the motor are quite different.

In some turbines the guide blades and wheel blades are different, but no exhaustive investigation has been published on the subject. It is mentioned here only to attract attention to it.
In Fig. 48 is shown a section of guide blades and wheel blades of different shapes; the shaded wheels are the guide wheels. It is an impulse turbine, in which the object is to impinge the steam upon the wheel blades at the highest possible velocity, and at the smallest possible angle to the face of the wheels. This, to the author, seems to be quite in accordance with the theory, and is consonant with the best practice in hydraulic turbines, a section of a good design of which is shown in Fig. 49. Water, of course, does not expand, hence it is possible to make the

![Impulse turbine blades](image)

entry to the wheel wider than the jet, and the water does not entirely fill the space between the wheel blades. But with steam the space is filled by expansion.

In the impulse turbine the fluid arrested in its motion and redirected by the guide blades, has its momentum first reduced to pressure and then to velocity in the guide blades; in the wheel the velocity is converted into pressure, doing the work, and it issues at a lower pressure into the next guide blades. Hence the guide blades should be designed to convert pressure into velocity with the least possible loss, and the wheel blades conversely should convert the velocity into pressure with the least possible loss. The form of blades shown in Fig. 48 seem to meet these requirements, so far as the guide blades are concerned, but it is doubtful if the wheel blades in an impulse turbine should have a wider entry than the outlet, when steam is the fluid employed, for it produces a pressure on the back of the blade, which is undesirable.

In the pressure or reaction turbine the function of the
guide blades is to arrest the steam after it leaves the wheels, and convert its velocity into pressure. This can be done by semicircular guide blades, and they may be spaced wider apart than the wheel blades, less in number, so that the construction is cheaper. And clearance between the tips of the guide blades and the rotor need not be so fine as that between the tips of the wheel blades and stator. And the space between guide blades and wheel blades need not be small, except when length is of importance.

In one way of regarding this question it would seem that, if in Fig. 49 the upper blades are the wheel blades, and the lower the guide blades, it would be a pressure turbine, but if *vice versa*, it would be an impulse turbine. In the first case, owing to the contraction of the outlet, there is an axial pressure, which must be balanced by dummy pistons, for the reaction of the steam has an axial component, and the pressure is greater on the entry side than it is on the outlet side.

But in the second case the pressure is equal on both sides of the wheel and has no axial thrust.

In the first case, there is partitions between wheel and wheel, hence for large and slow speed turbines, where the wheels must be very numerous and the partitions numerous, in an impulse turbine, the pressure or reaction turbine is preferable.

A turbine for small powers is made with four partial-flow wheels, the first two are in a compartment with a guide block of blades between them; the second two are in another compartment also with a guide block of blades between them where the steam is applied to the wheel. In this way a cheap turbine is made which, owing to the large diameter of the wheels, runs at a moderate number of revolutions per second, and is as economical as a small reciprocating engine.

It is interesting to note that in the early days of the screw propeller the great difficulty was to make the engines run fast enough for the screw, and spur gearing was adopted in many cases in the first half of the last century. Gearing has been entirely dropped for the last fifty or sixty years, but now the difficulty in many cases is to make the turbine run slow enough for the screw, and once more gearing is being considered so as to make the turbine adaptable for use in slow speed steamers which, after all, constitute by far the greater part of the shipping of the world.
The combination system described above does this, but gearing a high-speed turbine to a slow-speed screw would also accomplish what is needed.

Sixty years ago, there was nothing but primitive spur gearing, with generally wooden teeth in one member, but now we have steel gears accurately cut by modern machinery, often with helical teeth, and running in oil baths. Chain gears of some form are also promising, as well as various forms of electrical and hydraulic gear. The electrical gearing is especially promising, one or more turbines with dynamos or alternators, practically identical with those used on land, being employed to drive one or more motors on the screw shafts at reduced speeds of revolution. Again, in the combination system of reciprocating engine and turbine, the exhaust turbine may be a high-speed one, driving a motor on the shaft of the reciprocating engine instead of a screw on its own shaft.

Such questions are now receiving much attention, and may result in all the steamships of the world being wholly or partly turbine-driven instead of only those vessels of comparatively high speed.

There is no doubt of the promising advantages of a speed-reducing gear, if it can be realised; the great difficulty is the large powers to be dealt with. On the whole, it seems that a hydraulic system would be much preferable to an electric system, and such a system is now under trials in Germany by Dr Föttinger for the German Admiralty.

It is essentially a combined pump and turbine, and, on the smaller scale tested, answers very well in reversing and in controlling the speeds.

Whether it will deal with the power on large vessels remains to be seen, but at present there has been no difficulties met with.

In America experiments are being made with cut-steel helical gearing like the De Laval.

In this country the solid shaft connection is preferred, and endeavours made to accommodate the turbine and screws by good design.

The nozzles and wheel buckets are calculated, as already explained, beginning with the steam consumption.

Small impulse turbines can be cheaply made of four wheels, two in each of two compartments. In the De Laval wheel we have $Vt$ about $\frac{1}{4}Vs$, and $Vs$ extremely high by expanding in a conical nozzle. By putting four wheels in
series, the $V_s$ is reduced to $\frac{1}{4}$th, and hence the $V_t$ can be reduced to $\frac{1}{4}$th also, and so dispense with the wheel gearing for dynamo driving.

In the matter of velocities, the steam turbine presents some difficulties. The best speed of screw propellers is comparatively low, and this, as may be seen from the foregoing calculations, entails large turbines in order to bring down the velocities. Many proposals have been made to interpolate some gearing between the turbine and propeller, so that the turbine might run at high velocity and the screw at a lower velocity. None of the proposals have as yet proved of any practical value, but they may be here mentioned.

They naturally divide into two classes—First, simple reduction gearing, such as wheel-gearing, toothed or by belt. Helical toothed gearing has been successful only on small turbines like the De Laval, but it is now being tried on a larger scale on American naval vessels.

The second class constitutes a conversion of energy system, such as electricity, hydraulic and pneumatic conversion.

The electric system presents no engineering difficulties, but it is a frightfully heavy, expensive and dangerous system on board ship. Consider only a small power, say, 10,000 H.P., in, say, three turbines and three screws. With the direct drive we have only three turbines each of 3,333 horse-power to install, but if in order to secure more efficient screws and a complete control of the vessel's speed and reversing, electrical machinery were added, there would be—in addition to the three 3,333 H.P. turbines—three electrical generators and three electrical motors, each of 3,333 H.P., amounting in all to nine units, each of 3,333 H.P., to do the work of 10,000 H.P.

Fairly high-pressure electricity would be necessary with such powers, and this in accordance with electrical engineering practice would require an enormous amount of precautionary care and trouble and a highly complicated and elaborate switchboard.

The steam engineering staff would still be required as before, and, in addition, some electrical engineering attendants.

Hydraulic or pneumatic systems are not so bad, they are safe, and perfectly well understood by the ordinary marine engineering staff.

All of them, of course, are additional sources of loss in
transmission of the power. The electricity promoters claim, however, most extraordinary high efficiency, but their claims are based on ideal conditions which never occur in practice. Most of these proposals are made by men totally unacquainted with marine engineering and the prevailing conditions in the engine-room of a steamship.

Another type of turbine propeller is that known as the jet propeller, which produces a thrust exactly as the thrust is produced in a turbine, that is, by the reaction of a fluid having momentum.

It has been tried on several vessels but without success. However, an examination of the conditions of the tests reveals the fact that the failures were due to very inefficient machinery; the water jets were produced by centrifugal pumps of less than 50 per cent. efficiency, and the velocity of the entering water was largely destroyed at the inlet of the pump.

The nett efficiency of an engine and screw propeller, reckoned from I.H.P., is about 0.5.

A jet may have an efficiency of 0.75, and a modern pump an efficiency of 0.75. Total, 56.25.

It may yet be possible, by some injector or ejector appliances, to produce jet propellers by direct steam pressure and velocity, so doing away with all moving machinery on board ships.

While we can calculate the horse-power of a turbine with fairly close results, it is desirable that some means of ascertaining the actual brake horse-power should be available, or if not the brake horse-power some approximation to indicated horse-power.

It will be admitted that if the torque on the shaft could be ascertained that the horse-power could then be measured under various loads. A power meter on this
principle has been applied by Messrs Denny Brothers, Dumbarton (Fig. 50). Two large discs are attached to the main shaft transmitting the turbine power, at a distance, L, from each other, and a line drawn parallel to the shaft coincides with a line drawn on the edge of each disc. If now a torque is applied to the shaft, the shaft twists and the lines no longer coincide, showing by their divergence from the parallel line an angle of twist proportional to the torque. Torque can be applied by known weights and levers and the values found for the angles of twist in pounds. By direct experiment it has been found for ordinary homogeneous steel shafts that \( T \), the torque in ft.-tons, is equal to

\[
T = \frac{140d^4\theta}{L}
\]

wherein \( d \) is shaft diameter in inches, \( \theta \) the angle, and \( L \) the length between the discs, in feet.

Mr G. S. Goudie has deduced this also from first principles, using a coefficient of rigidity modulus = 5250 per square inch, from the following calculations.

The formula was given by Mr Ward of Messrs Denny Brothers.

It could easily be shown if a portion of a shaft AB, of diameter \( d \) inches and length \( l \) inches (Fig. 51) had the plane B twisted relatively to the plane A by a torque \( T \), so that any radial in the plane B was deflected from \( Oa \) to \( Ob \) through an angle of \( \varphi \) radians, that,

\[
\varphi = \frac{32Tl}{\pi Cd^4} \quad \text{and the torque } \quad T = \frac{\pi Cd^4\varphi}{32l} \quad \text{inch-lbs., } C \text{ being the rigidity modulus in lbs. per square inch.} \]

In the notation given by Mr Ward, the length \( L \) was in feet, and the angular deflection \( \theta \) in degrees, so that, in the above expression, \( l = 12L \), and \( \varphi = \frac{\theta}{57.3} \). When these were sub-
STITUTED, AND THE LEFT-HAND SIDE FURTHER DIVIDED BY 12, TO BRING THE TORQUE TO FT.-LBS., IT BECAME—

\[ T = \frac{3.1416 \times C d^4 \theta}{12 \times 32 \times 12L \times 57.3} = \frac{C d^4 \theta}{84,046L} \text{ ft.-lbs.} \]

AND WHEN 11,760,000 LBS. PER SQUARE INCH WAS SUBSTITUTED FOR C, THE EXPRESSION TOOK THE FINAL FORM,

\[ T = \frac{140d^4 \theta}{L} \]

AS GIVEN BY MR. WARD.


VARIOUS MODIFICATIONS OF THIS POWER METER ARE NOW KNOWN. IT DOES NOT MEASURE SMALL LOADS, BUT THAT IS NOT OF MUCH IMPORTANCE. WHAT IT IS DESIRABLE TO FIND OUT IS THE POWER GIVEN AT THE MAXIMUM, AND AT THE MAXIMUM POWER THE TWIST GIVEN TO THE SHAFT IS OF COURSE A MAXIMUM.

INSTEAD OF ELECTRICAL INDICATORS, A TUBE IS FIXED SOMETIMES TO ONE OF THE DISCS, EXTENDING ALONG THE SHAFT CLOSE UP TO THE OTHER DISC, OR TO ONE OF THE FIXED BEARINGS OF THE SHAFT, AND THEN A MECHANICAL MULTIPLYING GEARING IS USED TO SHOW THE AMOUNT OF TWIST BETWEEN THE TWO POINTS. THE ARC OF TORSION ON THE CIRCUMFERENCE OF THE SHAFT IS BUT SMALL, BUT CARRIED OUT TO THE CIRCUMFERENCE OF THE DISCS IS MAGNIFIED IN PROPORTION. MECHANISM IS SO ARRANGED BETWEEN THE TWO DISCS THAT THEIR RELATIVE MOVEMENT IS REPRODUCED AND MAGNIFIED BY THE SAME KIND OF PENCIL ARRANGEMENT THAT IS FOUND IN ORDINARY STEAM INDICATORS. THE PAPER CARRYING THE DRUM IS ALSO CONCENTRIC WITH THE SHAFT, BUT IS PLACED ON THE OTHER SIDE OF THE FIXED DISC TO THAT OF THE TUBE. THE DIAGRAM IS TAKEN IN THE USUAL WAY.

FIG. 52 SHOWS THE FÖTTINGER DESIGN OF THIS TYPE OF TORSION POWER INDICATOR.
CHAPTER V

THE CONSTRUCTION OF TURBINE WHEELS

The De Laval single-stage turbine construction has already been described. Turbines consisting of a series of impulse turbines with velocity stages in each series are more commonly in use, the speed being low enough for direct coupling without gearing.

We have seen that steam speeds may reach as high as 4000 feet per second, and to get economy of the highest
order the turbine blade speed should be half this = 2000 feet per second, but in practice, at the very most, only one-fourth is safely attained, or 1000 feet per second as blade velocity. Wheels for this enormous speed must be carefully designed and constructed. The danger from bursting is most at the centre, hence these wheels are enlarged into a massive hub (see left-hand figure in Fig. 53), which shows the construction of a small wheel, in which the shaft passes through the hub. In the larger sizes, the wheel must be solid throughout—to bore out the centre would reduce its strength below the safe limit—the shaft is therefore bolted on by two flanges (Fig. 53, right hand).

If we consider a wheel with a circumference of 3 feet, intended to run at 1000 feet per second or 900 feet per second we get 300 revolutions per second or 10,000 per minute. For dynamo driving this is reduced by a ten to one gearing to 1800.

In the case of water turbines, when the water velocity is very high, the practice is to use a partial-flow wheel of large diameter, so that although the angular velocity is very high, the revolutions are moderate. Fig. 54 shows a section of this kind of turbine. This practice cannot be followed in the steam turbine with a single wheel, owing to the difficulties increasing with increased diameter and weight.

The A. E. G. Company make a compound wheeled turbine with two wheels, and each wheel has two rows of blades, the steam passing from the first row in the first wheel through guide blades into the second row. A diaphragm with a labyrinthine packing is placed between the first and second wheel. The steam then passes through the two rows of the second wheel. Fig. 55 shows the construction in section.

Fig. 55 shows the blades and guide blades as constructed for a speed of 3000 revs. per minute, having only two wheels running at the fairly high peripheral speed of about 600 feet per second. The discs have the profile proper to a disc of uniform strength, and being
perforated have very massive naves. In the case illustrated, the discs are not forced on the shaft, but are gripped by split cones screwed at the end pulled tightly between nave and shaft by the nut shown. This construction is adopted to make it easy to remove the discs from the shaft.

There is only one diaphragm where a low clearance is essential, but the leakage per square inch of clearance area is not only more serious in absolute amount than at any diaphragm of Fig. 47, but in this case the leakage steam by-passes half the turbine—in the former case only about one-twentieth of the turbine at any diaphragm. On the other hand, the small number of discs leads to a short shaft, and this in turn allows a small diameter, which is favourable to a small leakage area.
The blades are drawn to the correct section. Dovetail grooves of the special section shown (Fig. 56) are turned in the heavy rim of the wheel, and the blades having their roots cut to the correct profile are then inserted through suitable slots in the groove with distance pieces of ductile material between each blade to ensure correct spacing and position of the blades. When all the blades in a ring are in position, the distance pieces are caulked down hard. The tops of the blades are then riveted to the circumferential shroud strip.

With a turbine speed of 3000 revolutions and a blade speed of 600, we get \( \frac{3000}{600} = 5 \) feet, as the circumference at the middle of the blades or a diameter of 19.5 inches.

This turbine for dynamo driving is cheap and efficient for small powers.

These double wheels are sometimes made up of two steel rings on which the blades are milled out of the solid; the rings are then bolted to a steel disc, as shown in Fig. 57. This, however, makes a heavy wheel, and is expensive to make.

Reference has already been made to the combination of a partial-flow impulse turbine with a reaction turbine. The impulse wheels letting down the pressure by large drops, is better calculated to operate with high pressure and small volume of steam, while the reaction turbine deals best with the steam after it has been reduced in pressure and increased in volume.

Fig. 58 shows this design in diagram; the impulse wheel has two rows of blades, as in Fig. 56, and the reaction blades are carried on a drum of the same diameter throughout.

Turbine blades are made now with great accuracy, by drawing, like wire, and may be had in lengths up to 8 feet ready for cutting.

The blades are drawn in copper or brass, and are spaced either by suitable distance pieces of soft brass,
also drawn to the required section, or are let into slots cut in a continuous brass foundation ring. In the former case the distance pieces are caulked directly, in the latter case a separate caulkling strip is used, and the caulkling becomes a somewhat quicker and easier operation. On the other hand the use of a foundation ring involves the root of the blade being stamped flat or pressed to a sharp angular section, and this is an undesirable feature, since it weakens the blade against bending stresses.
Where separate distance pieces are used, the blade root is stamped with two small rounded corrugations, as shown in Fig. 59. The walls of the groove are also corrugated, and the metal of the distance piece is caulked into all the crevices so formed, thus completely locking all the blades.

A wire threaded through the blades near their tips serves to maintain them in equal distance apart and prevents oscillations; this wire is silver soldered at each blade.

In some turbines a ring is placed like a tyre over the blade tips, as shown in Fig. 60. And due to its weight the blade speed is lowered to about 400 feet per second, so that three or four wheels are necessary.

A typical reaction turbine rotor is shown in Fig. 61, the rotor in question being built by the Brush Company for an output of 2000 k.w. at 1500 revs. per minute. Different from the disc rotors of the preceding turbine types, it consists of a hollow steel drum to which are attached the end shafts which form the journals. According to the design the drum is stepped in two, three, or four diameters—this construction being dictated by the effort to reduce to a minimum the leakage of steam from stage to stage without cutting too fine for safety the radial running clearance.

A large axial clearance being kept between successive blade rows, partial admission is impossible in reaction turbines, and the condition for low steam leakage is that the radial blade clearance is small in comparison with the effective length of the blades. The ratio of radial clearance to blade length is kept below 0.06 or 0.07, unless economy be deliberately sacrificed to secure some other advantage. Frequently the ratio is 0.05 at the high-pressure end, diminishing to 0.01 at the exhaust end.

The diameter at the exhaust end is limited by the stress caused by the centrifugal forces. Passing from the exhaust towards the steam end of the turbine, the steam pressure rises, and the total volume of steam flowing decreases, the blades being shortened approximately in proportion.

When the blades become so short as to give a disproportionately large clearance, the drum diameter is reduced, and keeping a constant ratio of steam speed to blade speed, and a constant radial clearance, the ratio of radial
clearance to blade length decreases approximately inversely as the square of the mean blade diameter, while the number of blade rows for a given pressure drop increases inversely as the square of the mean blade diameter.

The labyrinthine packing finally adopted is interesting, and is shown in Fig. 62.

Collars (A) turned on the rotor run in close proximity to but not in contact with the collars (B) projecting from the cylinder. By setting the thrust block, the axial clearance, and therefore the clearance area, can be finally adjusted and brought to a low figure. The whole forms a series of constrictions separated by relatively large chambers. Since a definite clearance exists, steam can leak past the piston, and the flowing steam is subjected to a series of throttlings, the velocity of exit of the steam from one constriction being destroyed by eddies in the chamber following.

By reducing the clearance or by increasing the number of throttlings, the weight of steam leaking past the dummy may be reduced as far as desired. The relation between these quantities is approximately shown by equation first given by Mr H. M. Martin.*

\[ W = 68 \Omega \sqrt[3]{\frac{p_1 (1 - \frac{1}{\rho^2})}{V(N + \log_{10} \rho)}} \]

Where \( W \) = steam leakage, pounds per second.
\( \Omega \) = clearance area at each constriction—square feet.
\( p_1 \) = initial steam pressure, pounds per square inch absolute.
\( V \) = initial steam specific volume, cubic feet per pound.
\( N \) = number of throttlings.
\( \rho \) = ratio of expansion of the steam flowing from one to the other side of the piston.

The doubtful factor in this formula \( \rho \) is difficult to value correctly.

The total leakage through the steam packings in the

* See Engineering, 10th January 1908, p. 35,
Parsons’ type of turbine from beginning to end is about 8 per cent. of the steam used.

On the whole subject it may be said that the present status of the steam turbine entitles it to a leading position hardly approached by any other motor for the driving of dynamo electric machines. In this connection, the speeds of both machines can be chosen as the best for highest economy.

It is different, however, with marine propulsion and ordinary manufacturing drives, wherein speeds are much lower. In the marine cases the turbines have to be large in diameter to bring the revolutions down to a suitable value for the screw propellers, or gearing of some kind must be used, or perhaps another type of propeller better suited for high speeds. An improved type of hydraulic jet propeller has been proposed by the author, and ranks among other proposals already referred to.

The steam turbine is a steam-engine, and goes always with the boiler. The tendency of later-day engineering in prime movers is to adopt the more direct method of converting heat into power afforded by the internal combustion engine. The boiler and its appendages, the funnel or brick chimney, etc., are in many ways expensive, troublesome, and objectionable, and no improved steam-engine can remedy the boiler troubles. The steam turbine, however, has come to stay, and will go on extending its sphere of usefulness for many years.
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