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# Steam Turbines

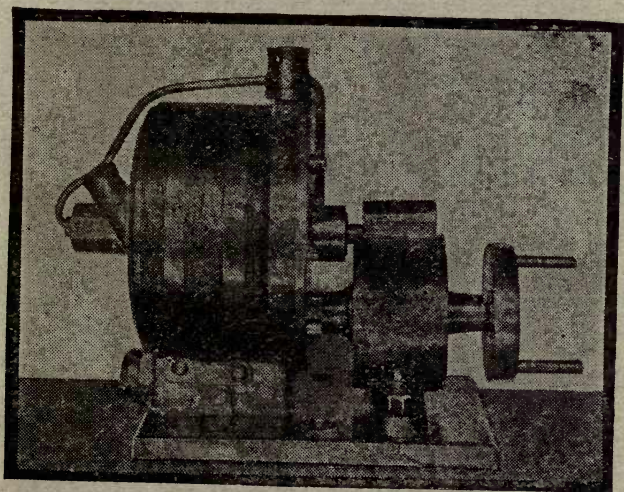
HOW TO DESIGN AND BUILD THEM

*A Practical Handbook for Model-Makers*

BY

H. H. HARRISON

*FULLY ILLUSTRATED*



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The "Model Engineer" Series. No. 23

# MODEL STEAM TURBINES

HOW TO DESIGN AND BUILD THEM

BY

H. H. HARRISON



*FULLY ILLUSTRATED*

THIRD EDITION—REVISED

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## PREFACE

THE steam turbine has rapidly come to the front during the last few years on account of its many advantages as compared with its older rival, the reciprocating steam engine.

From a model point of view also, several advantages may be claimed for it; thus for model steamer work its light weight for a given output, and the fact that the centre of gravity is low, are two conditions which make it specially applicable for high-speed craft.

In this small book it is proposed to lay down the principles on which model steam turbines may be designed; and some examples, either built or projected, are given, being for the most part taken from the columns of *The Model Engineer*.

H. H. HARRISON.

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## PREFACE TO THIRD EDITION

THE need for a third edition of this book having arisen, advantage has been taken of the opportunity to partially re-write it and to add further matter.

The reader is earnestly recommended to thoroughly master Chapters I. and II., as he will then have a very fair knowledge of the theory underlying the design and construction of steam turbines in general.

H. H. HARRISON.





# MODEL STEAM TURBINES

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## CHAPTER I.

### GENERAL CONSIDERATIONS.

STEAM turbines may be divided into two classes: the *impulse* or action type, in which a jet of steam or fluid at high velocity impinges on a number of vanes or paddles fixed around the rim of a wheel free to rotate, and the reaction type, more generally known as Hero's Engine or Barker's mill.

Both these types are pretty familiar to us from our childhood's days, in the windows of the toyshop, and no doubt in many cases (the writer's also) formed our first steam model. They are illustrated in figs. 1 and 2, and frequently appear in patent specifications, though, of course, with considerable modification.

In order that the principles on which these machines are designed and for which they depend on their action may be thoroughly understood, the following elementary mechanical considerations are necessary.

A body has both *mass* and *weight*. The weight of

a body is the force of gravity acting on it; whilst mass is the quantity of matter which the body contains.

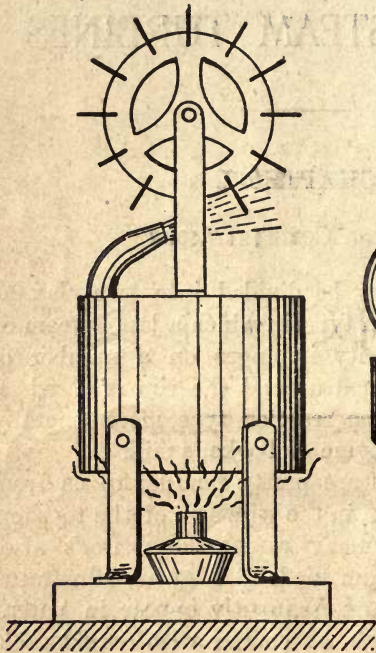


Fig. 1.—Simple Impulse Wheel.



Fig. 2.—Hero's Reaction Engine.

Unit weight is the standard pound, and the unit of mass is the quantity of matter in a body weighing 32.2 pounds.

*Force*, which is defined as so many pounds or tons

(according to its magnitude), may be measured by the velocity generated in one second on a body free to move in any direction whatever. Unit force is that force which, acting on a free body of one pound weight, generates in it a velocity of one foot per second. In our latitude, any body free to fall acquires in doing so (neglecting the resistance of the air) a velocity of 32.2 feet per second in each second during the interval between its release and arrest by the earth or any other body. This fact enables us to compare the force acting on a moving body when its weight in lbs. and its initial and final velocities in feet per second are known. An example will serve to make this clear.

Assume that a body weighs say 8 lbs.; the pull of the earth or the attraction of gravity would generate a velocity, as we have seen, of 32.2 feet per second after it is allowed to fall, and at the end of the second time interval another 32.2 feet would be added, and so on. The force acting in this case is of course 8 lbs. These successive increases in velocity are termed the *acceleration* of the body, and the figure 32.2, the acceleration due to the force of gravity, is symbolised by the letter *g*.

Now, suppose the body to be at rest but free to move in any direction horizontally or vertically, and that a force is suddenly applied in such a way that the velocity of the body in its direction of motion is found, after a time interval of one second, to be 40 feet per second. Since the body was at rest at the beginning, and found to have the above velocity at









and the increase or change of momentum

$$= \frac{W}{g}(v_2 - v_1) . . . (11)$$

The change of momentum in (11) as the result of the blow is termed an impulse, and the action of a constant force may be regarded as an indefinitely large number of impulses gradually changing the momentum of the body.

If the force which we will call  $F$  acts for  $t$  seconds, then

$$Ft = \frac{W}{g}(v_2 - v_1),$$

or  $F = \frac{\text{whole change of momentum}}{\text{time}} . . . (12)$

Where the initial velocity is zero, the whole change of velocity is  $V$  feet per second, and (12) =

$Ft = \frac{W}{g}V$ , and if  $t$ , the time of application of the force, is unity,

$$F = \frac{W}{g}V . . . . . (13)$$





## CHAPTER II.

### PRESSURE DEVELOPED ON SURFACES BY AN IMPINGING JET. VELOCITY AND FLOW OF STEAM THROUGH ORIFICES.

It has been seen from Chapter I. that if a body of mass  $\frac{W}{g}$  travelling at velocity of  $v$  feet per second, receives a push or impulse  $P$  during time  $t$  seconds, the velocity, and consequently the energy, of the body is increased.

In fig. 3 a piston is shown mounted in a cylinder, the back end of which terminates in a nozzle. The cylinder is filled with water which, the piston being moved to the right, leaves the nozzle with a certain velocity  $V$ , and impinges on the flat stationary surface. On striking the plate its velocity in the direction of flow is nil, and the whole change of velocity is  $V$ . Hence

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

where  $P$  is pressure in lbs. per square inch,  $W$  weight of fluid or gas discharged per second, and  $V$  the original velocity of the jet.

If the flat surface in fig. 3 forms one of a series of vanes round the rim of a wheel, it will move away from the jet with a certain velocity which is some fraction of the velocity of the jet. Calling this fraction

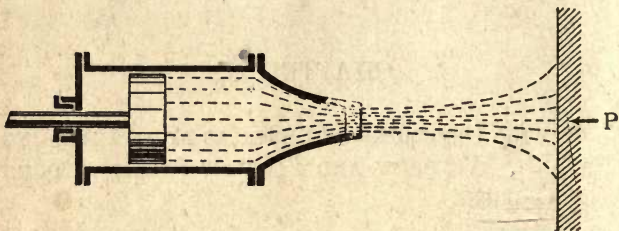


Fig. 3.

$\frac{1}{x}$ , the vane velocity is the jet velocity divided by  $x$ ; thus—

$$V \text{ (the jet velocity)} \times \frac{1}{x} = \frac{V}{x} \text{ (the vane velocity).}$$

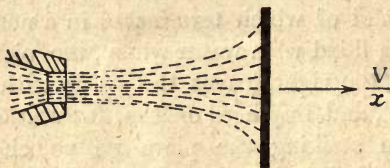


Fig. 4.

It is evident that in this case the velocity of the jet will not be completely destroyed, as it is still able to follow the vane up (see fig. 4). The change of velocity of the jet is therefore only  $V - \frac{V}{x}$ , and the

pressure  $P$  becomes

$$P = \frac{W}{g} \left( V - \frac{V}{x} \right) \dots \dots \dots (2)$$

The work done per second is, obviously, the pressure in lbs. on the vane multiplied by the distance in feet through which the pressure is exerted; and the distance in feet traversed by the vanes in a second is the velocity of these in feet per second, or  $\frac{V}{x}$ .

Therefore the work is equal to

$$P \times \frac{V}{x} \text{ foot pounds.}$$

inserting the value of  $P$  from (2);

$$\text{the work in foot pounds} = \frac{W}{g} \times \left( V - \frac{V}{x} \right) \times \frac{V}{x} \dots (3)$$

The efficiency is given by the ratio of the work done on the vanes to the energy possessed by the jet.

From Chapter I. the jet energy is  $\frac{Wv^2}{2g}$ , and the efficiency is therefore

$$\text{Equation (3)} \div \frac{Wv^2}{2g} \dots \dots \dots (4)$$

For example :—

Supposing the velocity of the jet is 300 feet per second, and the size of the nozzle is such that 2 lbs. of water or gas are discharged in one second, the energy of the jet =  $\frac{Wv^2}{2g} = \frac{2 \times 300 \times 300}{2 \times 32.2} = 2795$  foot pounds, and assuming that the vane velocity is

one-third that of the jet, or  $\frac{1}{x} = \frac{1}{3}$ ,  $\frac{V}{x} = \frac{300}{3} = 100$  feet per second.

Work done on the vanes is from (3)

$$\frac{2}{32.2} \times (300 - 100) \times 100 = 1240 \text{ foot pounds,}$$

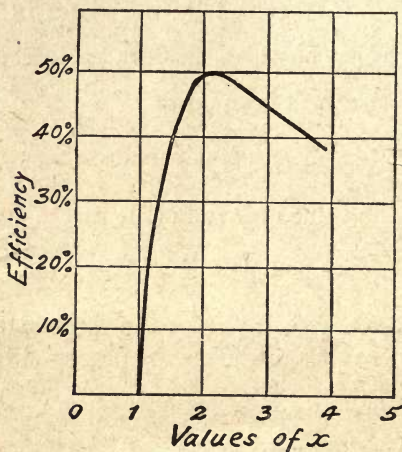


Fig. 5.—Variations of Efficiency with Different Speeds.

and the efficiency which is equal to the ratio  $\frac{\text{Work got out}}{\text{work put in}}$  is by (4) and the preceding calculations

$$\frac{1240}{2790} = 44.5 \text{ per cent.}$$

Taking the same figures as to velocity of jet and weight of fluid discharges, but varying the vane velocities, the curve, fig. 5, is obtained, showing

that the efficiency is a maximum when the vane speed is half that of the jet. It will also be seen that the maximum efficiency is only 50 per cent., which is all that can be attained with a flat plate.

This loss of efficiency is due to shock, and to the fact that the jet moves on with the wheel, and the energy which it still contains is not utilised. The

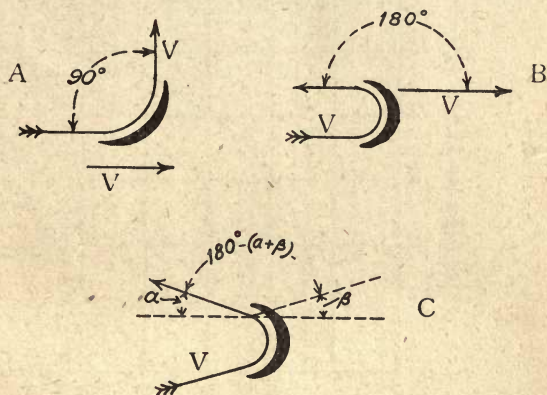


Fig. 6.—Action of Curved Blades.

first loss can be avoided by curving the vane as shown at A, fig. 6, which gradually changes the direction of flow of the jet. If matters are so arranged that the direction of flow of the jet is completely reversed as at B, fig. 6, it is clear that the energy abstracted will be the maximum possible; and if the vane speed is, as before, half that of the jet, the efficiency becomes 100 per cent. In practice, however, owing to the necessity of arranging for

the entrance and exit of the jet, the angle between the initial and final directions is somewhat less than  $180^\circ$ . If a Pelton type of wheel is employed, as in the Rateau single wheel turbine and that of Messrs Riedler & Stumpf, this angle becomes  $180^\circ$ , but the steam nozzle has to be inclined, so that the

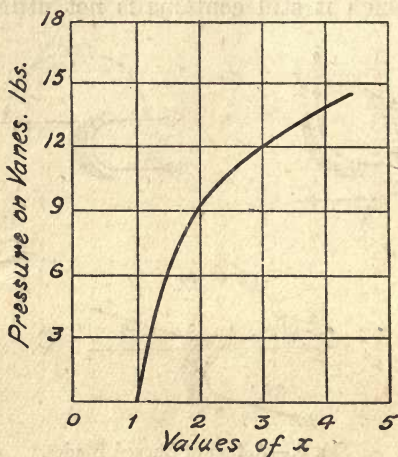


Fig. 7.—Variations of Pressure on Blades with Different Speeds.

effective velocity of the jet is less than its absolute velocity, *i.e.* the velocity with which it leaves the nozzle. Fig. 7 shows the increase of pressure on the vanes as the peripheral speed falls below half the jet velocity.

From the foregoing it is clear that to design a turbine to do a certain amount of work all that is required to be known is the weight discharged in

unit time and the velocity attained by the working agent, whether it be a gas, as steam or compressed air, or a fluid such as water.

The extended development of the turbine in recent years has directed the attention of experimenters to

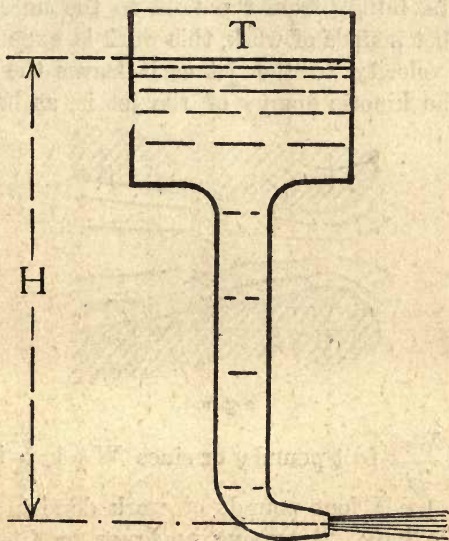


Fig. 8.

the at one time obscure subject of the velocity and outflow of steam through nozzles, and much valuable information is now available in the various technical journals. The following approximate method of treatment will, however, be sufficient for the purpose of designing model steam turbines.

Let T (fig. 8) be a tank in which the water is kept

at a constant level, and let it be provided with a discharge pipe terminating in a nozzle as shown. Suppose the relation between the height  $H$  and the area of the nozzle is such that one pound of water per second is discharged; each pound of water in falling from the tank to the nozzle does  $1 \times H$  foot pounds of work, this work is expended in giving velocity to the jet as it leaves the nozzle. Now the kinetic energy of the jet is, as has been

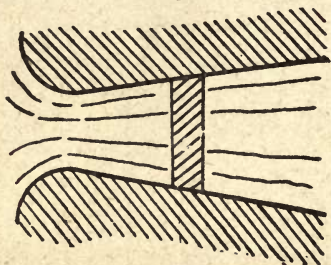


Fig. 9.

proved,  $\frac{Wv^2}{2g}$  foot pounds, or since  $W = 1$ ,  $\frac{v^2}{2g}$  is equal to the  $1 \times H$  foot pounds of work done in falling from the tank. Assuming no losses by friction in piping or nozzle, then

$1 \times H$  or  $H$  (foot pounds) =  $\frac{v^2}{2g}$  (foot pounds), and therefore  $v^2 = 2g$  ( $H$  foot pounds), or the velocity acquired  $v = \sqrt{2g \times \text{the work done in falling through } H \text{ feet}}$  (5)

In exactly the same way we may calculate the velocity of steam in falling from an initial pressure  $p$  to final pressure  $p_2$ . The steam does work on



itself, as each layer may be considered as acting as a perfectly free piston to the steam behind it, as in fig. 9. Similarly a projectile in a gun acquires velocity by the expansion of the gases generated by explosion. If, therefore, the work done on itself by steam in falling through a certain difference of pressure (the 'H' of the hydraulic analogy) is calculated, the velocity can be obtained as indicated by equation (5).

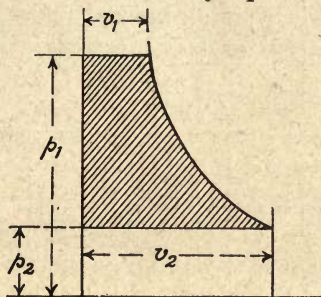


Fig. 10.

For example:

The work done on itself by steam expanding from 50 lbs. absolute (gauge pressure 35 lbs.) to  $\frac{1}{2}$  lb. absolute (a vacuum of  $29\frac{1}{2}$ " mercury) is 276,300 foot pounds approximately, and (5) becomes

$$v = \sqrt{2 + 32 \cdot 2 \times 276,300};$$

or since the square root of  $2 \times 32 \cdot 2$  is 8 nearly,

$$v = 8 \sqrt{276,300} = 4200 \text{ feet per second.}$$

There are two methods of evaluating the work done by steam in expanding from one pressure to another. One is to construct an 'indicator diagram' for one lb. of steam, as in fig. 10; or, alternatively, to

calculate it. Both are tedious, and the latter is both tedious and difficult to those not accustomed to such calculations.

For model turbines it will not be possible, as a rule, to apply a condenser, and the expansion will not therefore be anything like so high as in the foregoing example. The following table, calculated on the assumption of atmospheric exhaust, gives the velocities attained by steam jets under most conditions as to initial or boiler pressures likely to be met with.

TABLE I.

Gauge pressure, lbs. per sq. in.,	15	20	40	50	60	80
Velocity of steam at throat of nozzle, . . . . .	1430	1450	1460	1470	1475	1480
Terminal velocity, . . . . .	1600	2050	2256	2368	2512	2720

In fig. 11 a steam nozzle of the De Laval type is shown, and by reference to this it will be seen that the nozzle consists of two parts, the inlet part to the left of the figure being convergent up to the minimum area or 'throat'  $A_m$ , after which the nozzle diverges.

On reference to Table I. it will be noticed that the velocity of the steam at the throat of the nozzle is almost constant, whilst the terminal velocities increase as the initial pressure is raised. At 15 lbs. pressure (30 lbs. absolute) the ratio of expansion is  $\frac{30}{15}=2$ , and the final velocity when leaving the

nozzle 1600 feet per second; whilst at 80 lbs. with a ratio of expansion of  $\frac{(80+15)}{15} = 6\frac{1}{3}$ , the velocity is 2720, or 1.7 times as great. From the very close agreement of the throat velocities we should expect that up to the throat the ratio of expansion is the same whatever the initial pressure may be; and this is so, it being a well-known fact that steam discharging through a converging nozzle expands to 58 per cent. of the initial pressure. This peculiarity

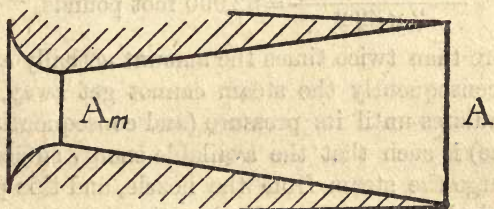


Fig. 11.—De Laval Nozzle.

(which will be useful for calculating the size of the orifice for passing a given weight of steam) may be explained shortly as follows:—

In expanding to any extent, the volume of steam increases directly as the ratio of expansion. Thus at 20 lbs. absolute the volume of a pound of steam is 19.73 cubic feet, and at 10 lbs. absolute (ratio of expansion 2) 37.83 cubic feet. With a fixed discharge orifice, if the volume of steam is doubled, say, by increasing the ratio of expansion, the velocity required to discharge it in the same time must be

doubled also. Assume the velocity is 1000 feet per second, and that one lb. of steam passes per second. The energy in foot pounds

$$= \frac{Wv^2}{2g} = \frac{1 \times 1000 \times 1000}{2 \times 32.2} = 15,500 \text{ foot pounds.}$$

If the ratio of expansion is twice that in the first instance, the energy developed will be proportionately increased, but the energy required to double the velocity is equal to

$$\frac{1 \times 2000 \times 2000}{2 \times 32.2} = 62,000 \text{ foot pounds,}$$

or more than twice times the amount actually available, consequently the steam cannot get away, and accumulates until its pressure (and consequently its volume) is such that the available energy suffices to discharge the steam from the nozzle, and this point is reached, as before stated, when the throat pressure is 58 per cent. of the initial. This explains why a boiler does not instantaneously discharge the whole of the steam when the safety-valve lifts. We are now in a position to understand the function of the diverging part of the nozzle. By progressive enlargement of the nozzle area as the expansion of the steam is extended, the jet is able to expand laterally as well as in the direction of flow. The increase of area of the jet along the nozzle as the volume of steam grows, makes the velocity required to discharge it at any given point the same as the velocity actually produced by the degree of expansion at that point. If the nozzle were convergent, only, the steam would

spread laterally as at A, fig. 12, and the velocity would be about 1400 feet only, and would not be increased even if the nozzle were discharging into a perfect vacuum; but by providing a taper extension piece, the discharge is as sketch B, and the energy developed is much greater. The following table

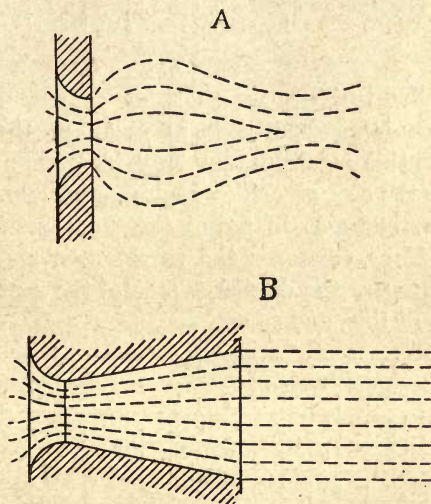


Fig. 12.—Difference between Simple Orifice and Taper Nozzle.

gives the weights of steam passed through a De Laval nozzle having a throat area of 1 square inch, and also shows the increased area required at the discharge end, atmospheric pressure in the turbine casing being assumed. The taper of the flaring end should be 1 in 12, which, with the throat and discharge dimensions, fixes the length of the nozzle.

TABLE II.

Gauge pressure, lbs. persq.in.	15	30	40	50	60	80
Weight of steam passed per second per sq. in., lbs.	0.41	0.53	0.66	0.757	0.88	1.09
Area of discharge end of nozzle, . . . sq. in.	1.015	1.02	1.12	1.2	1.4	1.53

In model turbines we shall, of course, not deal with such large quantities as given in the above table, but this is useful, not only in enabling us to calculate the size of orifice required for any weight of steam, but also in fixing the final area of the nozzle. Thus, whatever the throat area may be, at 15 lbs. pressure the discharge area of the nozzle must be 1.015 of this, and so on.

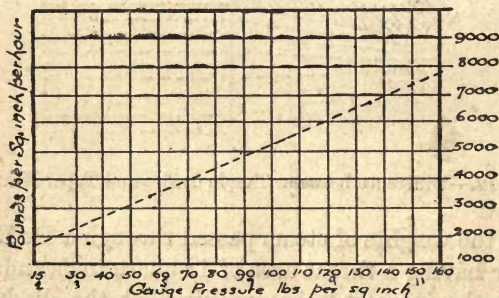


Fig. 12A.—Steam passed per hour per square inch for different Boiler Pressures.

The above curve, the result of actual experiments, will be found useful for reference.

### CHAPTER III.

#### METHOD OF DESIGNING A DE LAVAL STEAM TURBINE.

THE size of a model turbine is determined by the maximum evaporation of the boiler which is to furnish the steam. A standard rule has been to reckon on one cubic inch of water per minute per 100 square inches of heating surface. Five cubic inches evaporation from this surface has been obtained, and designs promising as much as ten cubic inches have been published. We will take one cubic inch, and if the evaporation is more or less than this quantity areas will have to be increased or reduced in exact proportion. One cubic inch of water weighs 0.036 lb., and the table overleaf gives the corresponding volumes of steam both in cubic inches and feet at the respective pressures considered.

Assuming that the boiler pressure is 30 lbs. (gauge), the volume of steam delivered to the turbine per minute is 0.33 cubic feet, weighing, of course, 0.036 lbs.; hence the volume and weight

passing *per second* will be one-sixtieth of these figures, that is, .0006 lb. and .056 cubic feet respectively. From Table II, 0.53 lb. of steam is given as being passed per second per square inch at 30 lbs. pressure, therefore the throat area of our nozzle becomes  $1 \times \frac{0.0006}{0.53} = 0.00113$  square inches, and the value of the final area is, from the same table, 1.02 times this =  $0.00113 \times 1.02 = 0.00115$ . The *diameters* can be calculated as follows:—

$$\text{Area} = \pi r^2, \text{ where } \pi = 3.1416. \quad \therefore \frac{\text{Area}}{\pi} = r^2,$$

from which  $r = \sqrt{\frac{\text{Area}}{\pi}}$ ; and since  $\delta$ , the diameter,

$$= 2r \quad \therefore \delta = 2r = 2 \sqrt{\frac{\text{Area}}{\pi}}.$$

TABLE III.

Gauge Pressure. Lbs. per sq. in.	Volume of Steam from 1 cubic inch of Water.	
	Cubic inches.	Cubic feet.
15	830	0.48
30	571	0.33
40	467	0.27
50	398	0.23
60	346	0.20
80	276	0.16



Applying this to the above figures, the throat diameter becomes

$$\delta = 2 \sqrt{\frac{0.00113}{\pi}} = 2 \sqrt{\frac{0.00113}{1.77}} = 0.038 \text{ ins. } (1.77 = \sqrt{\pi})$$

The details of this calculation given below will make the method of operation quite plain.

$$0.00113 = \frac{113}{100,000} \therefore \sqrt{\frac{113}{100,000}} = \frac{\sqrt{113}}{\sqrt{100,000}} = \frac{10.6}{316} = 0.0336; \text{ and this multiplied}$$

by 2 and divided by 1.77,

$$\frac{0.0336 \times 2}{1.77} = 0.038, \text{ or } \frac{38}{1000} \text{ths of an inch.}$$

In exactly the same way the diameter at the end of the nozzle is found to be 0.0384 inches. The conically expanding portion of the nozzle should

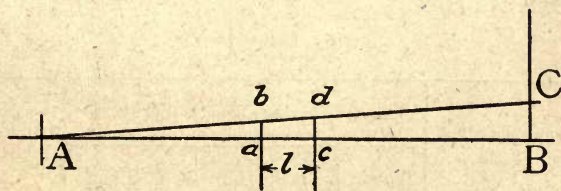


Fig. 13.

have a taper of 1 in 12, and this will fix its length, which can be obtained graphically as shown in fig. 13 in the following manner. Draw a straight line AB, and at B erect a perpendicular  $BC = \frac{1}{12}$ th

the length of A. Now find by trial at what point  $a$  along A.B the vertical  $ab$  is equal to the throat diameter of the nozzle, and also the position  $c$  at which  $cd$  equals the final diameter, then  $ac=l$ , the length of the nozzle.

Using the diagram for the case in question, the length works out to  $\frac{6}{1000}$  of an inch, so that the nozzle amounts to little more than a simple orifice in a thin plate. Even if the throat area of the nozzle were somewhat nearer that of the nozzles used in actual work, it is doubtful if a properly designed De Laval nozzle would give better results than a simple orifice with the low pressures we are considering.

In the following table particulars as to areas, diameters, and lengths are given, and a reference to this will show how extremely minute the dimensions are as a result of the small quantity of steam available.

TABLE IV.

Gauge pressure. Lbs.	Diameters, inches.		Areas, square inches.		Length. Inches.
	Throat.	Final.	Throat.	Final.	
15	0.042	0.0426	0.0014	0.00142	0.008
30	0.038	0.0384	0.0013	0.00115	0.006
40	0.034	0.0356	0.0009	0.001	0.018
50	0.032	0.0348	0.00079	0.000948	0.033
60	0.0292	0.0337	0.00067	0.000938	0.054
80	0.0264	0.0328	0.00055	0.00084	0.077

The nozzle may be made as follows :—

A piece of brass rod, A, fig. 14, is screwed to take a standard union and bored to  $\frac{1}{16}$  of an inch nearly the whole way down. A drill somewhat smaller than the required area according to Table IV. is used to complete the bore, and the throat orifice is then made the exact size by using a needle as a rimer and working from the discharge end of the nozzle. The throat diameter should be slightly under rather than

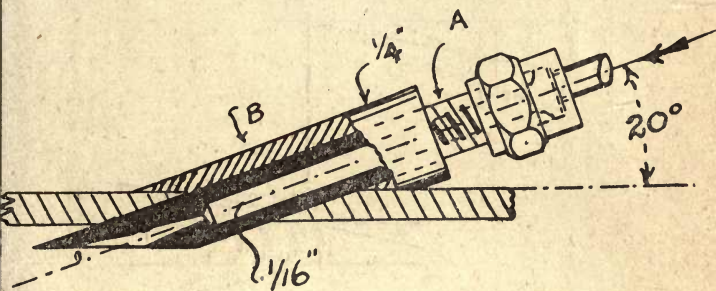


Fig. 14.

over the exact dimension, since the area of the nozzle increases as the *square* of the diameter, and not as the diameter simply.

The next points to be decided are, the diameter of the wheel and the speed at which it is to be run. Fig. 15 illustrates in plan the nozzle, and a portion of the turbine wheel and the diagram of velocity is set out in the upper portion of the drawing. O C is the velocity of the steam jet as it leaves the nozzle, or the 'absolute velocity'; O B is the peripheral velocity of the wheel at the point where

the steam enters, and  $O D$  is the velocity of the steam jet relative to the wheel. The peripheral velocity of the wheel is the number of feet per second at which a point on the rim moves in space and is equal to  $2\pi rN$  feet, where  $r$  is the radius of the wheel in feet and  $N$  the number of revolutions per second. In Chapter I, it was shown that for maximum efficiency the peripheral speed

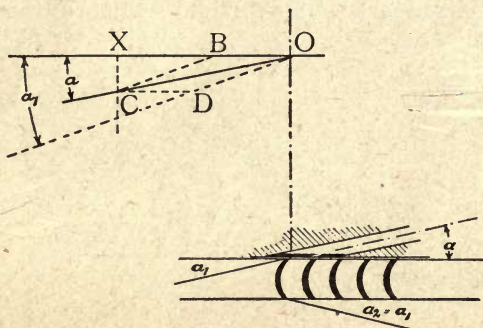


Fig. 15.—Diagram of Velocities.

of the wheel should be half that of the jet, and as the nozzle is inclined at an angle of  $20^\circ$  to the wheel, the component velocity parallel to the wheel must be obtained. This can be done graphically by extending the line  $O B$  and dropping a perpendicular from  $C$  to  $X$  as shown in dotted lines in fig. 15, when  $O X$  to the same scale as  $O C$  will give the parallel component in feet per second. From Table II.  $O C$  equals 2050, and  $O X$  is therefore about 1930 feet per second. The velocity  $O B$  of the

wheel is then  $\frac{1930}{2} = 965$  feet, and we can now fix the diameter of the wheel. The diameter does not mean the extreme diameter, but twice the radius from the centre of the wheel to the centre of the steam nozzle; and as it is advisable to fix the latter some way down the vanes, say half an inch, the actual diameter of the wheel from the tips of the vanes will be an inch greater than the one on which our calculations are to be based. Taking a wheel whose extreme diameter is 4 inches, the working diameter will be 3 inches and the circumference 0.78 feet. The number of revolutions necessary in order that the wheel speed is half the steam velocity, *i.e.* 965 feet, is  $\frac{956}{0.78} = 1230$  revolutions per second, or 73,800 per minute; and if 2000 revolutions is the maximum which the driving shaft may make, reduction gear of  $\frac{73,800}{2000} = 37$  to 1, will be necessary. As in most cases probably old clock wheels will be utilised for gearing, it will not be possible to accomplish this reduction in one stage, and a compound arrangement as indicated in fig. 16 will be necessary. A simpler form of reduction gear is to employ a worm wheel on the turbine shaft and a single wheel on the propeller or driven shaft. We have now to consider the shape of the vanes. In fig. 17 the velocity diagram of fig. 15 is transferred to the base line  $X_1 O X$ , the lettering

remaining the same. The angle  $a_1$ , which the relative velocity of the jet makes with the horizontal, is the entrance angle to the vanes in order that the

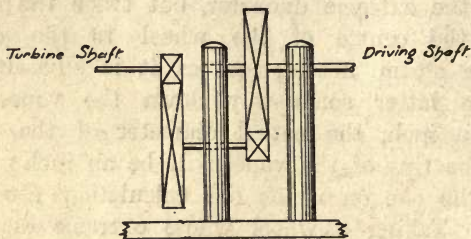


Fig. 16.—Compound Reduction Gear.

steam may glide on to these without shock, and in the diagram this angle is reproduced to the left of  $O$  both above and below the horizontal  $X^1 OX$ , and the discharge angle thus has the same value, although in actual practice this is not so. From

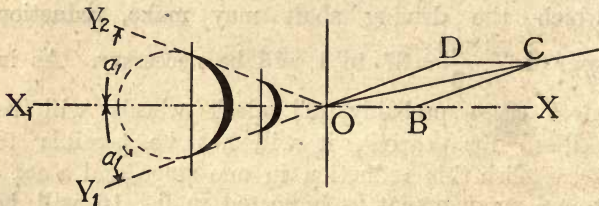


Fig. 17.—Formation of Blades.

different points in the line  $O X^1$  arcs can be struck which will fulfil the necessary conditions; that is, that their direction at the extremities bounded by the lines  $O Y_1$   $O Y_2$  coincide with the latter. It is evident, therefore, that we must have some rule to

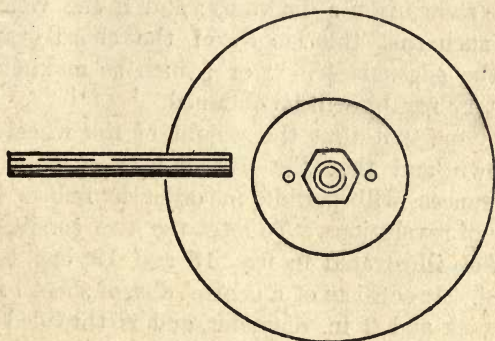


Fig. 19.—Elevation.

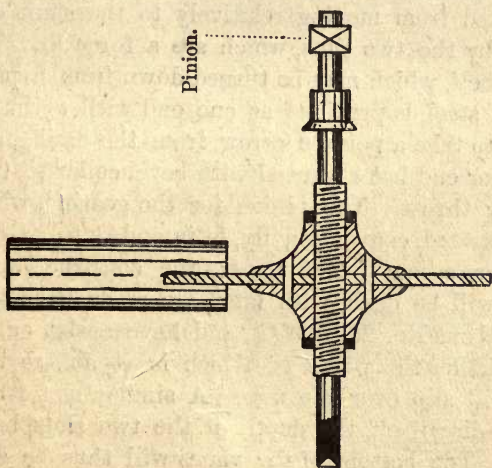


Fig. 18.—Section.

Design for De Laval Wheel.

decide which of the series of vanes should be adopted (two are shown in the diagram); and if the vane is chosen such that the length of the chord drawn across the edges is  $\frac{1}{4}$  inch or  $\frac{3}{8}$  inch as maximum, satisfactory results will be obtained.

It is important that the weight of the wheel be kept down and that the diameter be as large as circumstances will permit, in order to reduce the number of revolutions. To meet the two conditions the wheel illustrated in figs. 18 and 19 has been designed. It consists of a central disc of sheet brass  $\frac{1}{16}$  in. thick and 2 in. diameter, and is threaded on the shaft, being held in a vertical position by the two bosses as shown. These are in turn locked by two back-nuts, shaded black in section, and the wheel is prevented from moving relatively to the shaft and bosses by the two pins, which are a force fit. The shaft itself, which may be turned down from a piece of  $\frac{3}{8}$  in. steel, is provided at one end with a conical centre to take a pointed screw from the casing, and the other end has a journal with bevel collar to take up any thrust. The pinion for the gearing, which is illustrated conventionally, is intended to be outside the casing. The vanes, the construction of which will be dealt with later, are made from thin brass, about No. 28 B.W.G., and have a slot cut in the middle, the width of which is  $\frac{1}{16}$  in., so that they will slip over the disc. A similar slot is cut in the disc itself, the depth of the two slots being  $\frac{3}{16}$  in. The bottom of the vanes will thus be  $\frac{3}{8}$  in. below the edge of the disc, and the provision of the



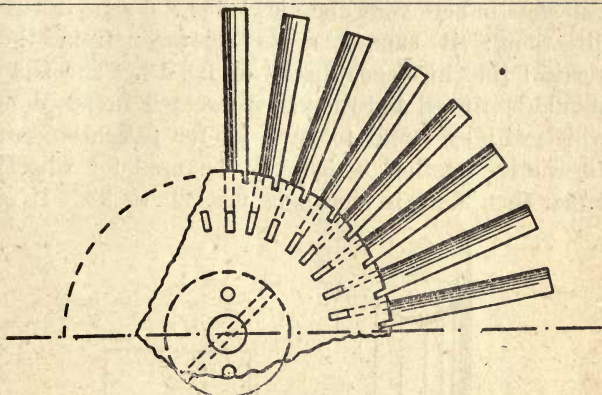
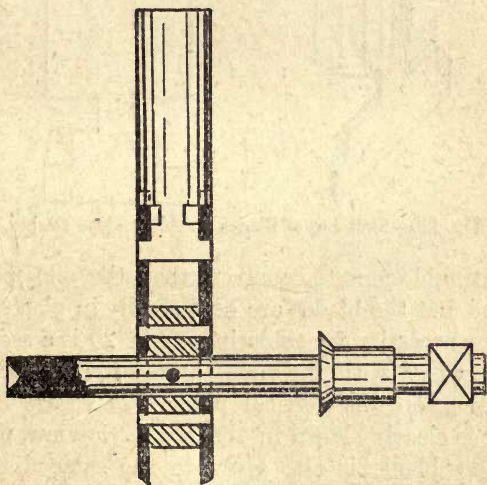


Fig. 20.—Elevation.

Fig. 21.—Section.  
Alternative Design for De Laval Wheel.

two slots in both vane and disc locks the vane in two directions; it cannot move sideways from the vertical line in either fig. 18 or fig. 19. The slots should be tinned and the vanes sweated in position, which will be found to be a job for patience. An alternative method which can be used for wheels larger than 3 in. is shown in figs. 20 to 23. It is

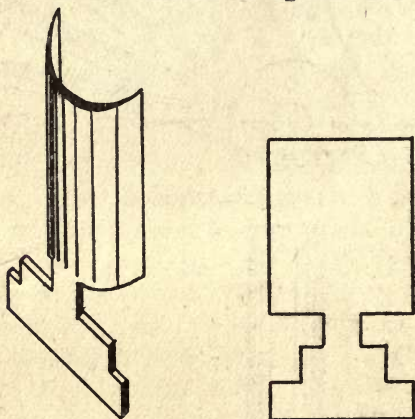


Fig. 22.—Sketches of Blades for Alternative Design.

more troublesome to construct than the one just described, but the blades are held rigidly in place without the necessity for soldering. Fig. 20 is a segment of the wheel with the vanes attached, and fig. 21 is a section on the vertical plane. The form of the vanes is clearly shown in the isometric view, fig. 23, and the 'blank' is also given in the same drawing. The wheel consists of a central boss and two outer discs of  $\frac{1}{16}$  in. sheet brass. The boss is attached to

the shaft by means of a pin of a length exactly equal to the diameter of the boss, so that when the pin is driven home it does not project either way and put the wheel out of balance. The two discs are fastened to the boss by means of pins of uniform

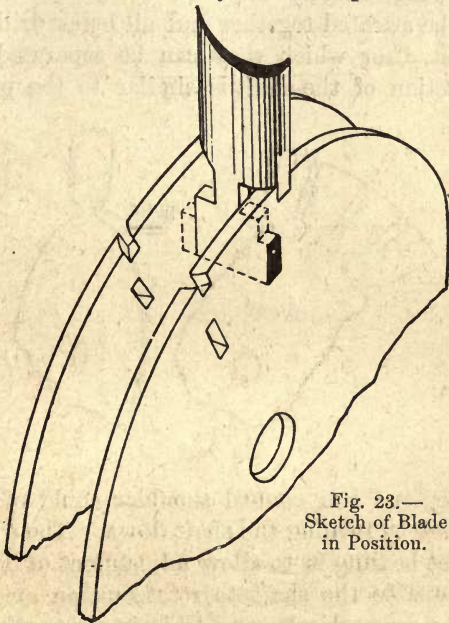


Fig. 23.—  
Sketch of Blade  
in Position.

diameter, which may be allowed to project each side slightly, and then riveted over by spreading the ends with a centre punch. Only one disc should be fixed first, as the vanes have to be inserted before the second disc can be finally fastened. The vanes are held in place by their bottom lugs entering small

slots some way down the discs, and an additional steadying is given by the saw-cuts in the edges of the discs, which allow the bottom of the vanes to enter, as clearly shown in fig. 23. To ensure the discs being exactly alike in every respect, they should be sweated together and all holes drilled and slots cut, after which they can be separated. The construction of the shaft is similar to the previous

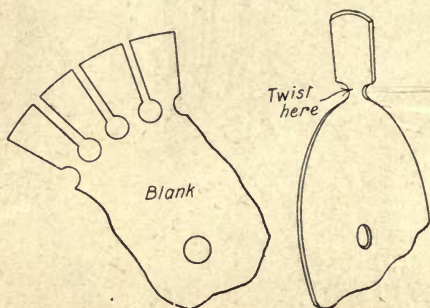


Fig. 24.

example, and the conical shoulder could of course be formed by turning the shaft down. The object of the pivot bearing is to allow adjustment of end-play and to enable the shaft to rotate on an axis other than its geometrical one. This is accomplished in the De Laval machine by using a long thin flexible shaft, as is well known, but such a method is impracticable in a model. An ingenious method of forming the vanes and wheel out of *one* disc of brass is due to a firm of German toy-makers, and is illustrated in fig. 24. A series of holes are drilled round

the disc and radial cuts are made outwards to the edge. The flat pieces between the saw-cuts are then curved, and afterwards twisted so as to be at right angles to the disc.

A development of this idea due to the writer is illustrated in fig. 25, which allows the pitch to be

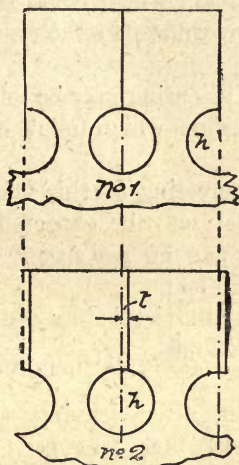


Fig. 25.

reduced while keeping the blades the same width. Two identical discs (Nos. 1 and 2) are first marked out and the rings of holes  $h$  in each are drilled; No. 1 is then finished off in the ordinary way. In No. 2 the radial saw-cuts are made off the centre by a distance  $t$  equal to the thickness of the metal. The two discs are now riveted together, the vanes of one being between the vanes of the other, and the result

is a wheel with a pitch one-half that of a single disc wheel. The object of displacing the radial cuts in No. 2 is to keep the tips of all the vanes in line. Were it not for this arrangement, the vanes would be "staggered." By using thin brass, reducing the width of the blades, and using two or even three discs, a very efficient wheel could be made, and the pitch would approximate to the cross-section of the steam jet.

The question of casing may be left to the reader, the numerous designs which follow offering sufficient choice.

It will be interesting at this stage to see what power the model may be expected to exert. The weight of steam passing per second is 0.0006 lbs. at a velocity of 2050 feet.

The energy in the jet before entering the wheel is  $\frac{.0006 \times 2050 \times 2050}{2 \times 32.2} = 39$  foot pounds, and the

energy remaining after leaving the wheel at a velocity of about 900 feet per second will be  $\frac{.0006 \times 900 \times 900}{2 \times 32.2} = 7.7$ , or 8 foot pounds nearly,

therefore the energy given up to the wheel is equal to  $(39 - 8) = 31$ , and putting the turbine efficiency at 40 per cent. only to cover losses in nozzle and due to friction of disc and of bearings, the work done in foot pounds is equal to  $12\frac{1}{2}$  per second, or 750 per minute. If the driving shaft is running at 2000 revolutions per minute, the work done by the turbine per revolution multiplied by the

revolutions should equal 750, or work done per revolution =  $\frac{750}{2000} = 0.375$  foot pounds. This will show that shafting and propeller gear, if the turbine is to be applied to boat propulsion, must be of the lightest description and friction must be kept as low as possible, also the pitch and area of the propeller will require to be reduced considerably below that for boats driven by slower running reciprocating engines.

## CHAPTER IV.

### COMPLETE DESIGNS FOR DE LAVAL TURBINES. METHOD OF MAKING VANES. SHROUDING.

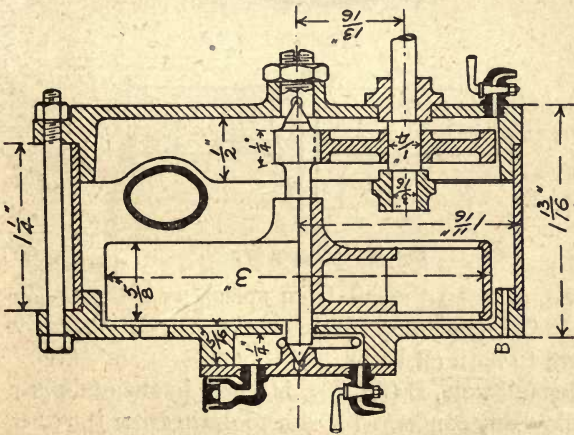
FIGS. 26 and 27 are sections of the prize design in the competition instituted by *The Model Engineer*.

Referring to fig. 26, it will be seen that the left-hand bearing for the shaft consists of a steel plate with a countersunk boss in the middle, which is attached to the end casting by four  $\frac{1}{8}$  in. studs, shown in fig. 31. The boss must be deeply case-hardened for the end of the shaft to run in. The space between the steel plate and the end-plate forms an oil chamber, the oil being conveyed to the bearing by a ring as in dynamos.

The right-hand bearing is a  $\frac{1}{4}$  in. countersunk steel stud, screwed through a boss in the end-plate, and held in position by a lock nut. This stud is also hardened. The pinion on the shaft gears into a gun-metal wheel  $1\frac{1}{2}$  inches diameter, mounted on a  $\frac{3}{16}$  in. shaft. This is supported on the left by a gun-metal bearing (fig. 34), screwed to bosses on the end-plate.

A thin brass plate is soldered on the inside of





the right-hand end-plate, as shown by dotted lines in fig. 31. This forms an oil bath in which the gear wheel runs, thus carrying oil to the pinion and bearing, and a thin brass cover should be put, as in

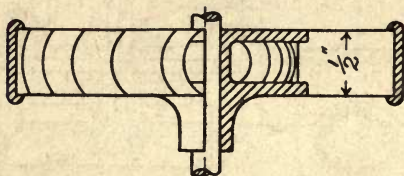
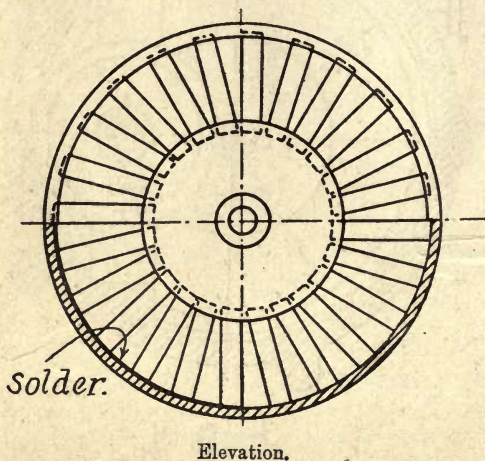


Fig. 28.—Plan of Wheel.

fig. 33, to keep the oil from splashing all over the case. Oil cups and drain cocks can be fitted as shown to both oil baths.

A small hole, B (fig. 26), is bored in the end-plate to allow any condensed steam to drain from the case,

and another in the end-plate of the turbine case, the latter being rimmed and filed until the steam nozzle fits in it at an angle of  $20^{\circ}$  to the plate. The nozzle is held in place by a brass bracket sweated to it, which is fastened by two screws to the end-plate. The joint between the nozzle and the end-plate should be sweated with soft solder to make it steam-tight. The exhaust steam passes through a  $\frac{1}{2}$  in. diameter tube soldered into the side of the casing.

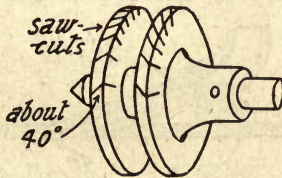


Fig. 29.—Boss of Wheel.

The vane wheel of the turbine consists of a boss, round which a number of suitably shaped vanes are fixed. Outside these vanes is a brass band to keep the steam from escaping through centrifugal force. The dimensions and appearance of the wheel can be seen in figs. 26 and 28. The vanes are made from No. 28 B.W.G. brass  $\frac{3}{4}$  in. long, and having the proper curve.

The boss (fig. 29) is made of either brass or gun-metal, and can be built up or turned from a casting, the latter course being preferable. Its dimensions are:—Diameter of side plates,  $1\frac{1}{2}$  in.; diameter of

centre,  $\frac{3}{8}$  in.; thickness of plates,  $\frac{3}{32}$  in.; distance apart of plates,  $\frac{5}{16}$  in.

The centre of the boss extends on one side of the wheel, so that it can be fixed to the shaft by a pin driven right through. In the edges of the plates twenty-four equidistant cuts are made with a fret-saw, into which the bottom edges of the vanes are sweated. These cuts are at an angle of about  $40^\circ$  with the side of the plate.

The shaft is turned from a piece of steel  $\frac{7}{16}$  in. diameter, and has a disc left at one end in which

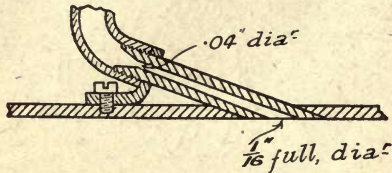


Fig. 30.—Detail of Nozzle.

teeth are cut for the necessary gearing. Its dimensions are:—Length, 2 in.; diameter,  $\frac{3}{16}$  in.; diameter of boss,  $\frac{3}{8}$  in.; width of boss,  $\frac{1}{4}$  in. The ends of the shaft are pointed and hardened, and run in two hollow centres. The boss of the wheel is bored a good fit on the shaft, and is secured by a small pin driven right through it.

The steam nozzle is made of a piece of brass rod outside diameter  $\frac{7}{32}$  in. (bare). A hole 0.04 in. diameter is bored through the middle of it, and is tapered out as in sketch, fig. 30, which clearly shows how the nozzle is attached to the casing. The nozzle

is designed to suit a boiler evaporating 1·2 cubic inches of water per minute, and a reference to Table IV., Chapter II., will show that it is much too long.

The various details of the casing are shown in figs. 26, 27, 31, and 32. The case of the turbine consists of two castings, fig. 31, fitting into the

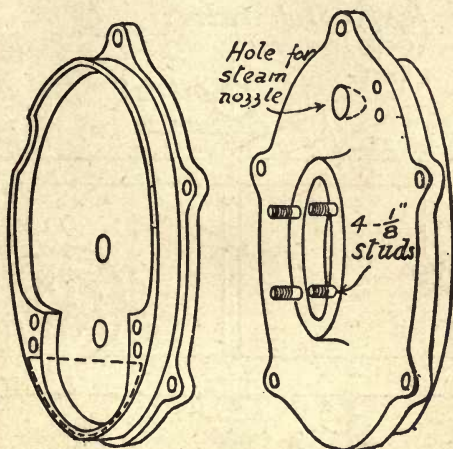


Fig. 31.—Ends of Turbine Casing.

ends of a piece of  $3\frac{1}{2}$  in. diameter solid drawn brass or copper tube, the whole being held together by five bolts, as shown. These castings can be made in either brass or aluminium, preferably the latter.

The vanes are twenty-four in number, and the method of their construction is explained at the end of this chapter. The method advised by the designer for fixing these into the boss is as follows:—

The boss, mounted on a mandrel, is firmly fixed to a piece of wood with a hole  $3\frac{1}{4}$  in. diameter by  $\frac{5}{8}$  in. deep turned in it. A circle of ordinary pins must now be driven into the bottom of the hole at equal distance apart. The diameter of this circle

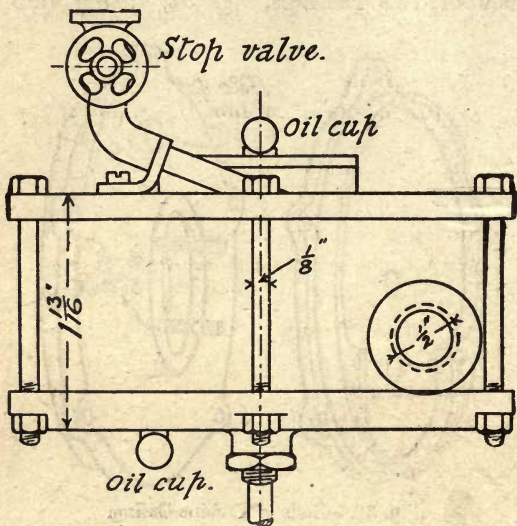


Fig. 32.—Plan of Turbine.

should be about  $2\frac{3}{4}$  in. (fig. 35). The vanes are now pushed into the saw-cuts in the boss, the backs of their outer ends being against the pins. In this way the vanes are held in their places radially.

When all the vanes are in position, a thin mixture of plaster-of-Paris must be poured into the hole in the wood. When the plaster has thoroughly set,

the boss and vanes, firmly imbedded in the plaster, can be taken from the board, the pins having been previously removed. It must then be mounted



Fig. 33.—Oil Bath Casing.

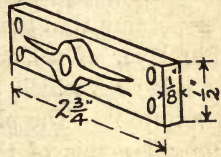


Fig. 34.—Bearing for Countershaft.

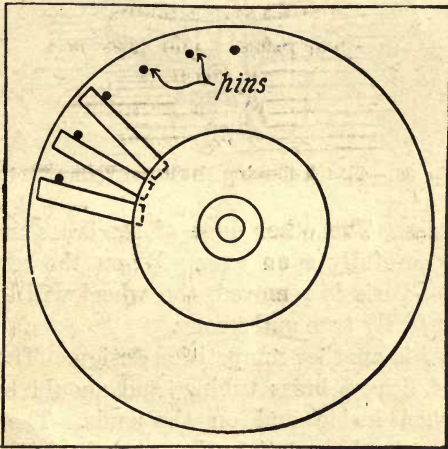


Fig. 35.—Fixing Blades (Plan).

between centres, and carefully turned down to  $2\frac{7}{8}$  in. diameter. The outside ring for the wheel is made from a piece of solid drawn brass tubing,

3 in. diameter and  $\frac{11}{16}$  in. wide; this must be thoroughly annealed, and one edge spun over, as in figs. 28 and 73.

The plaster-of-Paris must now be carefully chipped away for a depth of about  $\frac{1}{4}$  in. from the ends of the vanes, and the brass rings slipped into place. A thin layer of solder is then sweated into the inside of this ring, which fixes the outer ends of the vanes firmly to it. The plaster is then chipped away from the other ends of the vanes, which are then sweated

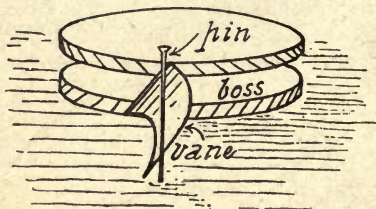


Fig. 36.—Sketch showing Method of Fixing Blades.

to the boss. The other edge of the brass ring must now be carefully spun over. When the remaining plaster-of-Paris is removed, the wheel will be found to be perfectly true and strong.

Fig. 37 is another competition design. The casing *cc* is of drawn brass tubing, and should be carefully trued inside and on the ends. The covers *ee* are preferably castings, but may be built up or turned from the solid, as may also the wheel. These are held together by three bolts *d*, a slight fillet being turned to fit inside casing. The bosses are drilled  $\frac{3}{16}$  in. to take the bearing sleeves *ss*;



these must be of good gun-metal and a sliding fit. The oil-cups act as dowels to prevent them turning. The wheel *a* requires to be carefully turned and drilled centrally for shaft; a piece of knitting-needle makes a good shaft if a driving fit for wheel, so that the original temper and polish be preserved.

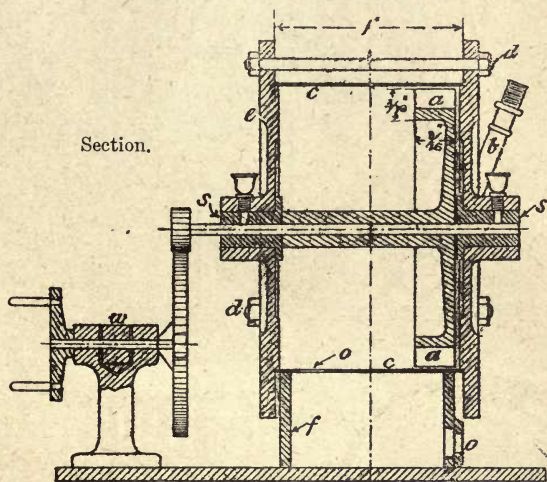


Fig. 37.—Design for Model De Laval by the Rev. W. Bredin Naylor.

The vanes are cut out of the solid as detailed further on, and their dimensions, together with those of the steam nozzle and the oil cup, are given in figs. 40 to 42. The nozzle is turned from  $\frac{3}{16}$  inch brass rod, and its exact size is given in fig. 42. The dotted lines show the taper discharge end. It is sweated into the casing as seen in figs. 37 and 38, at

an angle of  $60^\circ$ . The designer does not state why this angle is chosen; but, presumably, the

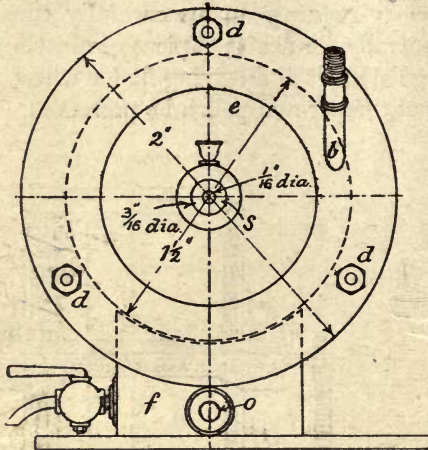


Fig. 38.—End View.

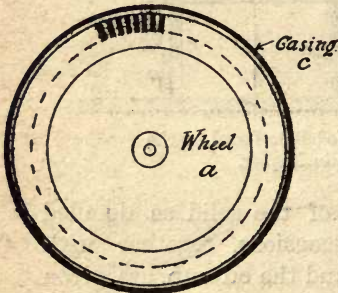


Fig. 39.



Fig. 40.



Fig. 41.—Oil Cup.



Fig. 42.—  
Nozzle.

object in view was to reduce the velocity at which the wheel would have to run. While this would

certainly be accomplished, the energy in the steam is not so fully utilised as it might be.

The saddle *f* (fig. 38) on which the casing rests was made from 1 in. brass tube hammered to a

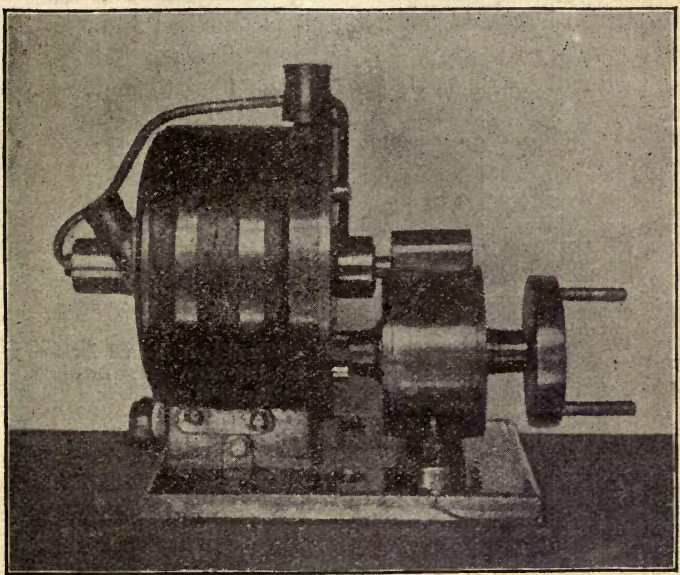
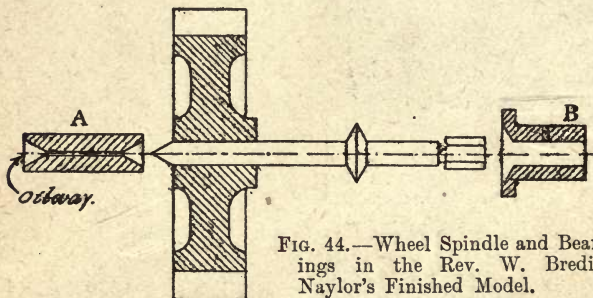


Fig. 43.—Photograph of the Rev. W. Bredin Naylor's Model Steam Turbine.

rectangular shape on a former, and serves as a chamber to collect the condensed steam from the outlet *o* in under part of casing. The turbine is geared down to 1 in 10 *at least*. The designer subsequently completed a model turbine of which

fig. 43 is a photograph, and fig. 44 gives details of the wheel and bearings.

The wheel is  $1\frac{1}{2}$  in. in diameter by  $\frac{3}{8}$  in. thick, and the blades are cut  $\frac{3}{16}$  in. deep, with a space between of  $\frac{1}{4}$  in. The oiling arrangement is explained by fig. 44, and the oil is supplied from the cup seen on the top of the casing. The nozzle has a throat diameter of  $\frac{1}{25}$  in., and the final diameter is



$\frac{1}{16}$  in. The gear runs in oil, and the reduction is 10 to 1. With steam at 30 lbs. pressure, the number of revolutions is said to be about 10,000 per minute; but it is not stated whether this figure refers to the turbine or the propeller shaft.

Figs. 45 to 49 are drawings for a turbine to develop "anything up to 5 H.P.," and a glance at the sectional drawing of the steam nozzle will show that the turbine is too large for the ordinary model boiler. It consists of a disc mounted on the centre of a flexible steel shaft, the supports of which are placed at a considerable distance from the disc. As it is impossible to accurately balance the wheel, this

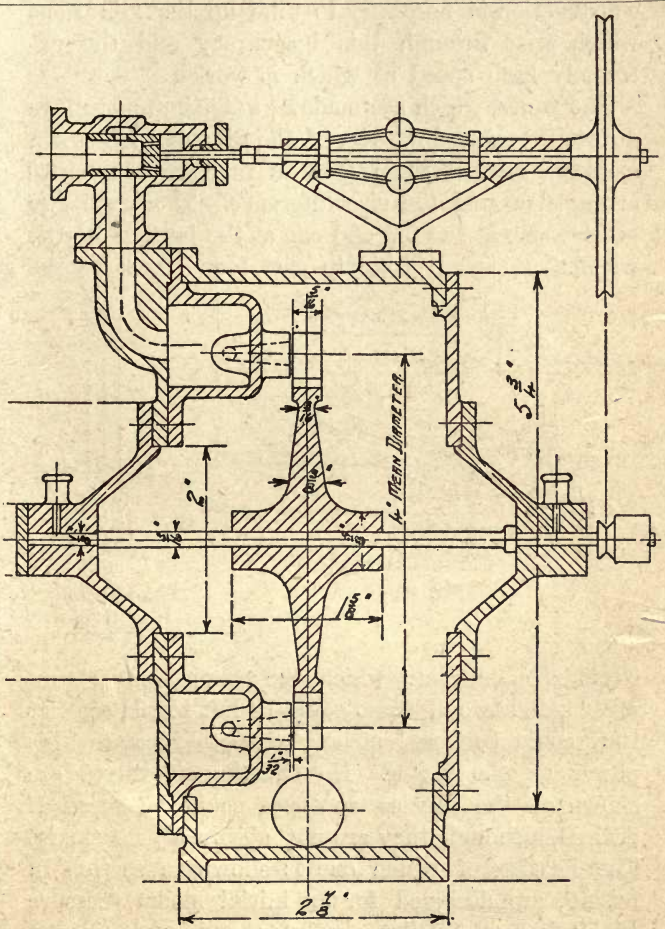


Fig. 45.—Design for Larger De Laval Type Turbine. Longitudinal Section. (Half full size.)

construction is necessary to take up the vibrations which arise through this inaccuracy and the extremely high speed at which it works.

The vanes, which are made of tough gun-metal, are dovetailed around the rim of the disc, and the steam is blown against them from a number of nozzles arranged around the circumference. There are two nozzles, one at the top and one at the bottom of the wheel. It is advisable to cast four bosses to the

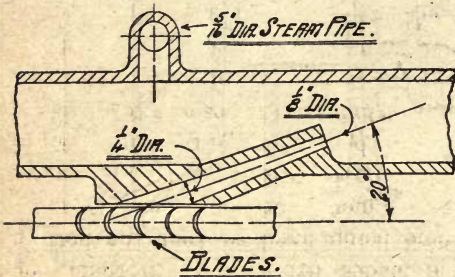


Fig. 46.

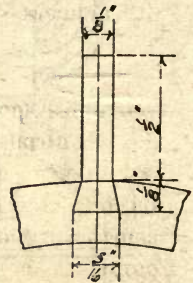


Fig. 48.—Enlarged Drawing of Blades (Elevation).

circular chamber to which the steam-pipe is connected, and two of these could be left blank, so that they could be bored at any time to increase this power of the motor. No shutting-off valves are shown on the nozzles, as these could be fitted if desired, although they are not absolutely necessary. The nozzles, of which one is shown in section in fig. 46, are designed for an initial steam pressure of 50 lbs. or 65 lbs. absolute; and as it is not intended to apply a condenser, the steam expands



Fig. 48a.—Enlarged Drawing of Blades. (Plan.)

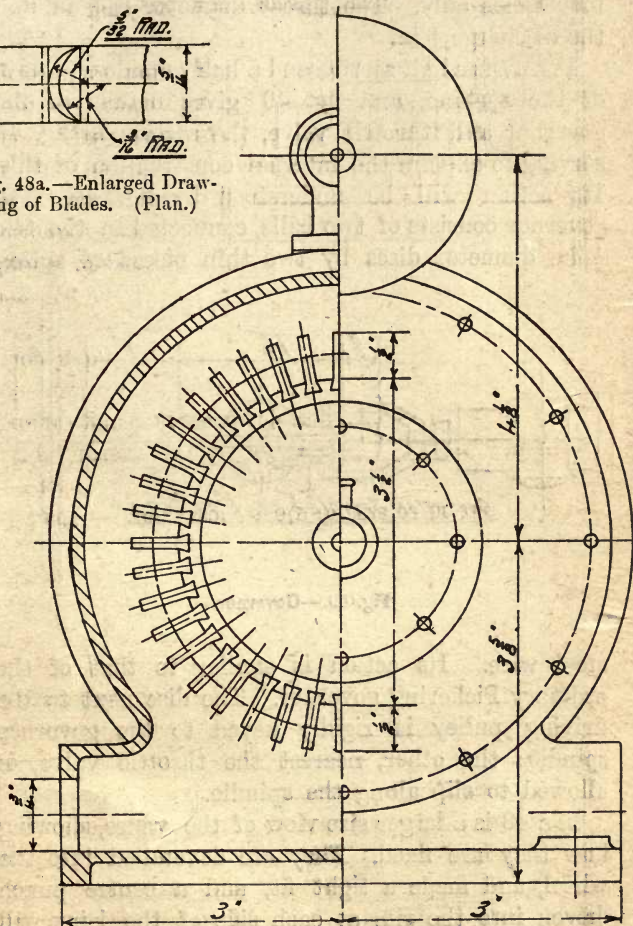


Fig. 47.—End View and Section of Larger De Laval Turbine.

four times only. The throat diameter is  $\frac{1}{8}$  in. and the discharge  $\frac{1}{4}$  in.

Fig. 47 is a half section and a half outside elevation of the turbine, and fig. 49 gives details of the governor and throttle valve, the section in fig. 45 serving to explain the internal construction of this. Its action will be understood by fig. 49. The governor consists of two balls, connected to the two  $\frac{1}{2}$  in. diameter discs by two thin pieces of spring

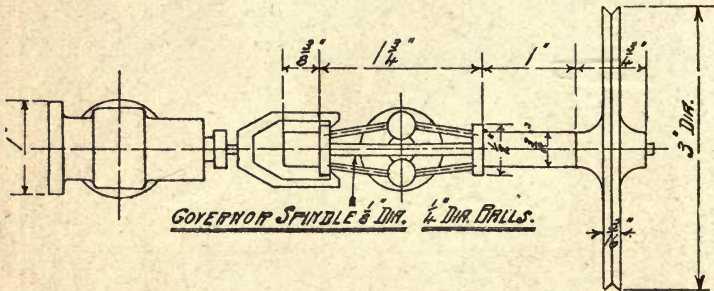


Fig. 49.—Governor.

steel wire. Its action is similar to that of the ordinary Pickering governor. The disc next to the driving pulley is rigidly keyed to the governor spindle: the other, nearest the throttle valve, is allowed to slip along the spindle.

Fig. 48 is a larger size view of the vanes, showing how they are fixed. They are dovetailed into the wheel, and made a tight fit, and a centre punch driven into the rim at each side of the joint will prevent any lateral movement.



To those to whom the gearing may prove a stumbling-block, the friction gear illustrated in the two following drawings may be of interest.

The gear is formed of plain bevel wheels, of which one element of the pair is of metal and the other of some such substance as pressed leather or silicated cardboard. The turbine shaft is supported in bearings  $B_2 B_3$ , and the end thrust in the direction  $B_3 B_2$  keeps the bevel wheel tight against the two opposing wheels. This thrust can be increased by making the exit angle of the vanes slightly less than the inlet

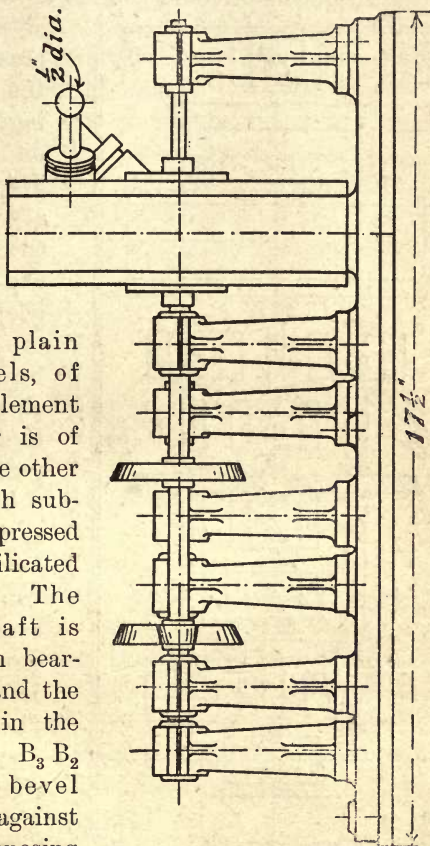


Fig. 50.—Elevation of a Model Steam Turbine, showing the Reduction Gear.

angle—that is to say, angle  $A_2$  less than  $A_1$  (fig. 15), or the turbine shaft could be kept up to its work by

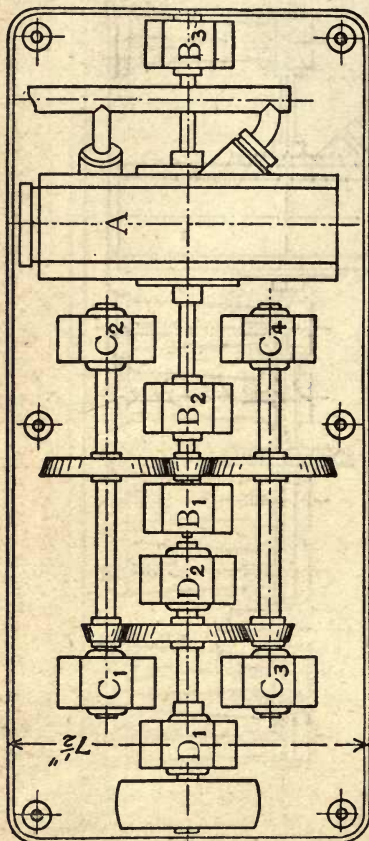


Fig. 51.—Plan of Turbine and Gear.

a spiral spring pressing against the turbine casing and a collar on this shaft. This form of gear allows of a high-speed reduction, and even if not adopted in its entirety, the bevel wheels offer an alternative to toothed gearing.

There are several methods for forming the vanes, and the simplest, and certainly the least tedious, is the following, due to the designer of the turbine described on p. 43:—

Referring to fig. 52, two pieces of iron are clamped together and a hole, of a diameter depending on the

curvature of the blades, drilled through them, the centre of the hole being fixed so that the bottom piece of iron forms a mould for the vanes. A piece of rod should then be turned down to a diameter less than that of the hole by twice the thickness of the brass sheet of which the vanes are to be made. The rod is now filed down till it is flush with the top of the mould, when a piece of the sheet brass is interposed. The vanes are formed by taking a flat piece of brass well annealed and about the

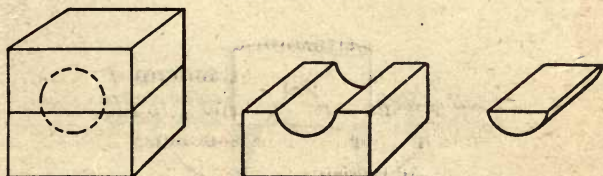


Fig. 52.—Method of Curving the Blades.

size of the mould, laying this on the top and forcing the segment of the rod, or 'core,' on top with a hammer. This bends the brass to the correct curvature, and the projecting edges can be afterwards removed. The writer has had some experience with this process, and has found it very easy and satisfactory, and it can always be employed where the vanes are symmetrical, that is to say, form a segment of a circle.

Mr Bredin Naylor cuts his vanes out of the solid rim of the wheel in the following ingenious manner:—

The tool for cutting the curved vanes is shown in

fig. 53, and it is made from a  $\frac{1}{2}$ -in. steel rod having a hole  $\frac{3}{8}$  in. diameter drilled centrally at  $h$ , and turned down on the outside till the thickness of the blades is  $\frac{1}{32}$  in. The edge is nicked with a chisel to make a saw edge, and is then hardened. The cutter is held in the lathechuck, and the hand-rest is set

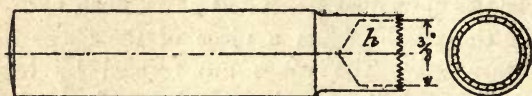


Fig. 53.—Tool for Forming Blades.

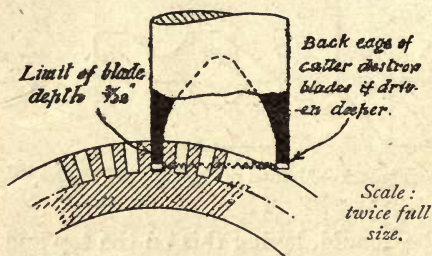


Fig. 54.—Sketch showing how Annular Cutting Tool Limits the Depth of the Blade.

$\frac{1}{16}$  in. below the lathe centres and almost touching the teeth of the cutter. The work is held in the fingers and thus fed up to the cutter. The sketch (fig. 54) shows very clearly how the depth of the blades is limited for a given size of blade and cutter, the back of the latter destroying the other blades, and to obviate this Mr Bredin Naylor designed the improved cutter (fig. 55).

The cutter is made from tool steel rod about  $2\frac{1}{2}$  in. long, the diameter being a quarter greater than the thickness of the washer to be cut. The rod is chucked and trued for  $\frac{1}{2}$  in. from the end; a hole is drilled in the end of such diameter as to leave the sides about  $\frac{1}{16}$  in. thick; the sides are filed away, leaving the part *a* projecting; this is shaped

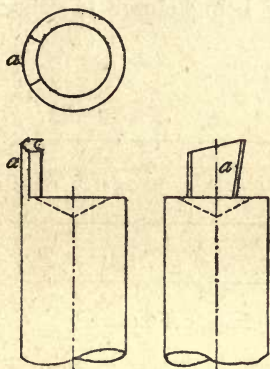


Fig. 55.—Improved Cutter.

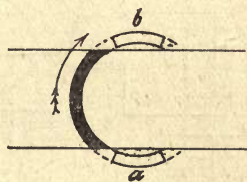


Fig. 56.—Diagram showing how the Improved Cutter Forms the Blades.

for clearance, and, of course, hardened and tempered. Fig. 57 shows the method of using it; *b* and *c* are 'steadies' for the wheel *w*, and are held in the slide-rest and fed up to the cutter in the chuck, which is worked by hand in semicircular sweeps. There should be a very slight clearance to allow the wheel to be turned round on the pin *p* without loosening the slide-rest nut. A hand vice may be used to hold the wheel steady while it is being cut.

The sketch (fig. 56) shows the action of the cutter *a* at the beginning of the cut; *b*, after cutting; from *b* the cutter is brought back to *a*, and the washer advanced for another cut. Spaces of any dimensions can be left between the blades.

This method has the advantage over the built-up wheels with separately formed vanes, that for a given diameter of wheel the number of vanes can be much greater owing to their being closer together.

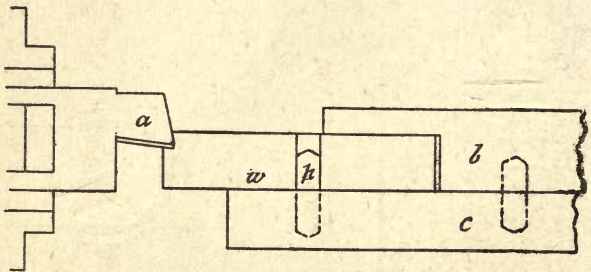


Fig. 57.—Method of Using Improved Cutter.

Where the distance apart is greater than the cross-section of the steam jet, it is obvious that for constantly recurring infinitesimal spaces of time the jet is blowing through the wheel without doing any useful work. On the other hand, if the vanes are *too* close together, they continually meet the jet, and eddying is set up, which means, of course, loss of energy. Only experiment can settle the best width to adopt. It has been pointed out that Mr Naylor's cutter could be used for cutting grooves in the rim of the wheel boss into which vanes made of

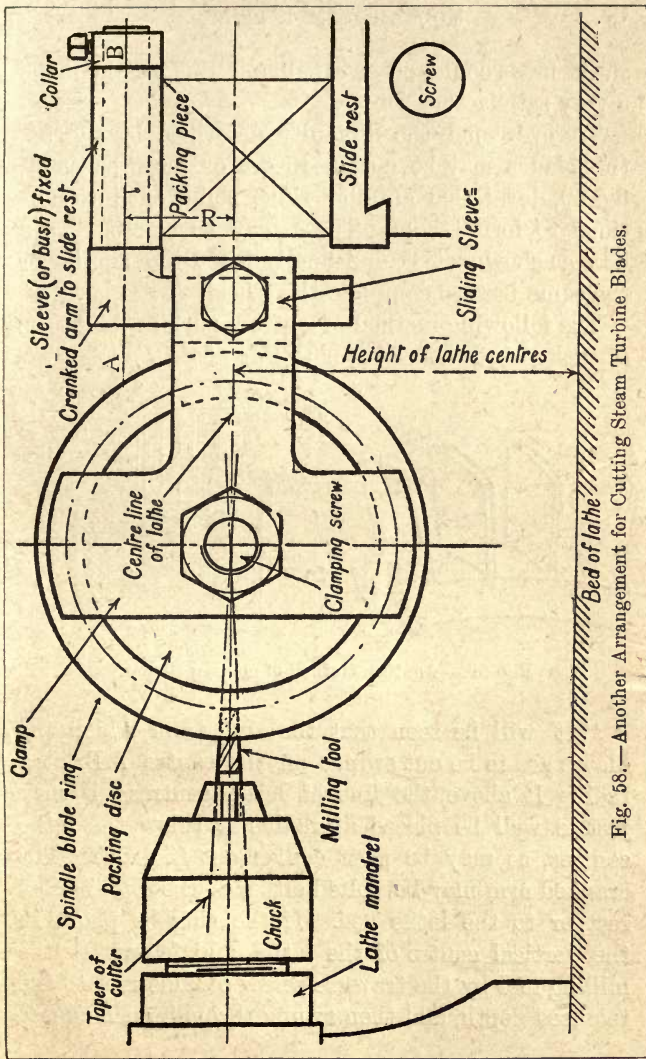


Fig. 58.—Another Arrangement for Cutting Steam Turbine Blades.

sheet brass could be sweated, and this would make a very satisfactory job.

To overcome the difficulty described by fig. 54, the blades may be cut in threes or fours as at A, fig. 59, dovetailed annular rings being cut into portions to form them. These may be clamped to a boss as shown at B, and finally fitted to a similar or the same boss to complete the wheel.

The following method of cutting turbine blades is suggested by a contributor to *The Model Engineer* :—

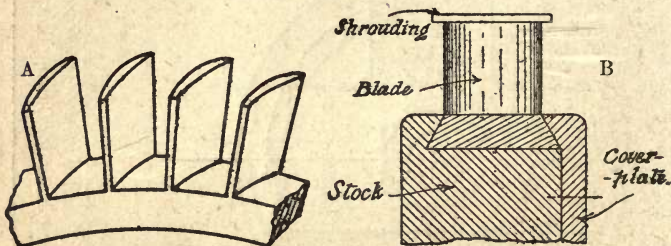


Fig. 59.—Another Method of making Blades.

“ It will be seen that the ring from which the blades are to be cut swings on the centre A B, at a radius R above the line of lathe centres; it may just as well be placed at radius R below the lathe centres, as may be most convenient (fig. 58). The cranked arm may be bolted either to the top of slide-rest, or to the lathe bed. If the ring be placed in the vertical centre of the lathe, and fed up to the milling tool by the traverse screw of slide-rest to the required depth, and then swung through an angle at



the radius  $R$  sufficient to clear the milling tool, a groove will be cut, as shown in fig. 60.

“Those not possessing a slide-rest may accomplish this by allowing an amount of play between the collar and the bush (fig. 58) equal to the length of blade, and a small amount for clearance between point of milling tool and ring, and feed up by hand.

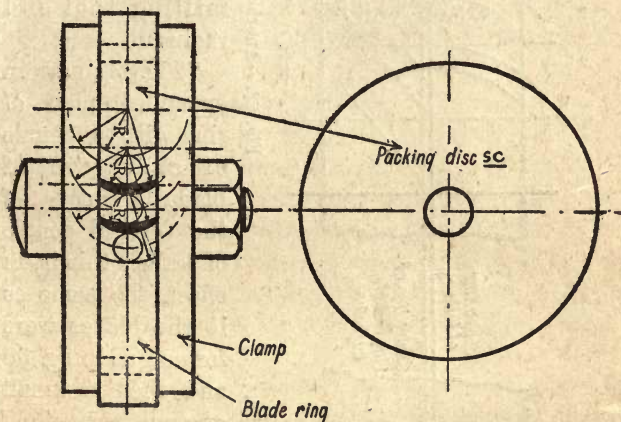


Fig. 60.—End View.

Fig. 61.—Packing Disc.

“The next operation is to slacken clamping screw (fig. 60), and turn the ring round on the packing disc (fig. 61) to where the next groove has to be cut, leaving a space in between which is the blade. It will be seen that with the ring mounted on the vertical centre of the lathe, the pitch of the blades equals the diameter of the cutter *plus* the thickness of the blade at the centre of the blade

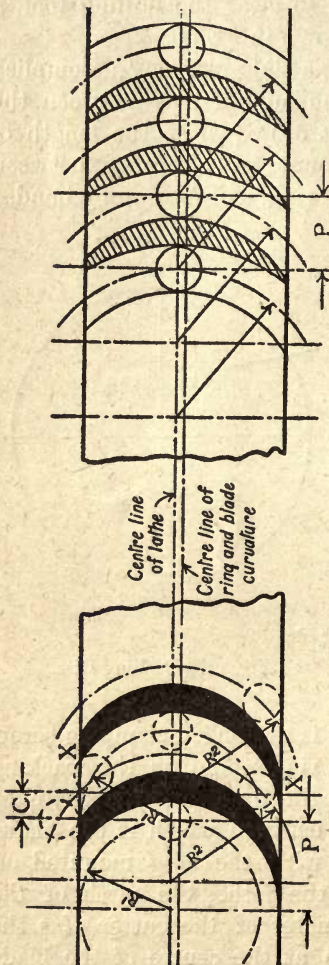


Fig. 63.—Another Form of Blade.

curvature, which in this case is the vertical centre of lathe. The greatest width of blade (or thickness of ring) is fixed by the diameter of milling tool and radius  $R$ .

“Fig. 63 shows an expanded plan of ring, with the centre of curvature of blade slightly off the vertical centre of lathe; this is, in effect, the same as if the blades were formed as in fig. 58 and then a small amount turned off the one side of ring.

“Fig. 62 shows a blade of somewhat more difficult construction; here the pitch of the blade is not determined by the diameter of cutter and the thickness of blade.

Fig. 62.—Expanded Plan of Blades.

“Taking  $R^1$  (fig. 62) it will be noticed that the positions X and X' of the milling tool limit the greatest angle at which the ring may be swung through; for, if continued, the milling tool would be cut into the back of the next blade.

“To form the back of the same blade it is necessary to turn the ring round on the packing disc through the distance C, and with a new radius ( $R^2$ ) swing the ring through an angle sufficient to cut to the side of ring.

“It is advisable to cut all the fronts of the blades first with radius  $R^2$ , turning the ring round through the distance P (which is the pitch) for every blade, then to set the sliding sleeve (fig. 58) to the radius  $R^2$ , and cut the backs of the blades. The pitch in this case is equal to the diameter of milling tool + distance C + thickness of blade. To obtain a blade parallel through its length, it is necessary to taper milling tool as shown in fig. 58.”

The writer who makes the above suggestions further adds that a Parsons turbine might be constructed by cutting so many rings and putting them on a spindle. The guide vanes which fit into the turbine casing would be made by cutting rings into halves and forming the blades on the *inside* curvature of the halved rings.

Whatever method for making the blades be adopted, the edges should be filed quite sharp, except at the roots where the blades enter the boss or wheel disc. Otherwise the steam will meet the flat edge of the vane and rebound (see fig. 64).

With regard to shrouding, it is an open question as to whether this is necessary in a model turbine.

Mr Naylor's wheel is not so provided, and a small turbine that the writer has experimented with was also unshrouded. The function of the shrouding is to prevent the steam being thrown out of the wheel radially by centrifugal action, but in addition it prevents the steam jet expanding laterally during its passage through the wheel. A piece of tubing of an

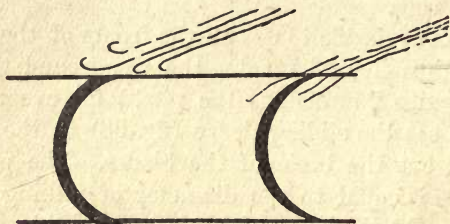


Fig. 64.—Diagram showing best form of Blade.

internal diameter equal to that of the wheel may be used as in fig. 26, but if shrouding is decided on, it is better to use very thin sheet brass. The ring may be made by taking a circular disc of brass and turning a piece of hard wood the diameter of the wheel and of thickness equal to the width of the shrouding. The brass disc should be larger than the wood, and being placed on this with the centres coinciding, the projecting metal edge should be gently and carefully hammered down or spun so as to present the appearance of a canister lid or boiler end. From the centre of the disc a circle should be

struck of such radius that it approaches the flange, and this circle cut out. The ring will then appear as at A, fig. 65, which is a section on the centre line. This ring can now be slipped over the vanes, the wheel mounted in the lathe, and the other edge spun over as at B. The advantage of this method is that the shrouding is much lighter than it would be if

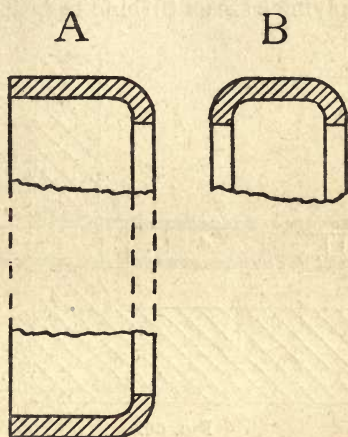


Fig. 65.—Method of making Shrouding.

made from tubing; and further, it does not limit the diameter of the wheel to any particular size to suit the tubing. The vanes will require supporting during the operation, and if they are mechanically attached to the wheel, white metal may be run in between them temporarily. If they are soldered in, plaster-of-Paris should be used. The shrouding should be sweated to the vanes to prevent its rotation relative

to the wheel, but all excess of solder should be carefully removed.

The wheel should be spun between centres and adjusted so that it is in balance everywhere, and any corrections which can be made by filing the discharge sides of the vanes should be done at the *extreme* diameter.

When applying a steam turbine to boat propulsion,

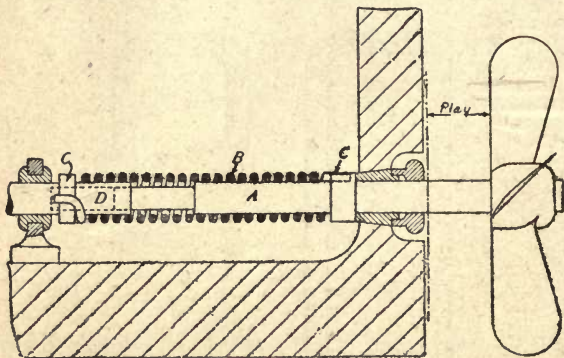


FIG. 66.

it has to be remembered that the torque they give out is small, and it is therefore advisable to employ a "buffer" between the turbine and the propeller shaft. A suitable device is shown in fig. 66. •

A is the propeller shaft and B a coiled spring fitting it exactly, and fixed at its ends in the flanges CC. The tail shaft is free to move in the circular hole D in the turbine shaft, the two pieces being flexibly coupled together by the spring. When the

propeller revolves clear of the water the position is as shown; when it enters the water, the resistance causes it to move forward, compressing the spring and gradually transmitting the load to the turbine without shock. For a five-foot boat with fine lines a steel spring 4 in. of  $\frac{1}{16}$  in. diameter wire with the pitch of the coils  $\frac{1}{32}$  inch apart and with a play of  $\frac{1}{2}$  in. is recommended.

## CHAPTER V.

### THE THEORY OF MULTIPLE STAGE TURBINES.

IN the second chapter it has been shown that for maximum efficiency the peripheral speed of the turbine wheel must be equal to half that of the parallel component of the steam jet velocity. As a result of this, either the revolutions of the driving shaft have to be high, or gearing must be adopted. The use of gearing in practical work would seem to limit the size of the impulse or action turbine, which is the class to which the De Laval machine belongs; but its only objection, from a model-maker's standpoint, is the difficulty or the extra cost of its construction. Gearing is obviated in practice by 'staging' the work done by the steam. This will be made clear by reference to fig. 67. Here the 'indicator diagram' is divided into several strips, the area of each of which represents a fraction of the total work of the steam. If a De Laval turbine were supplied with steam at an initial pressure  $p_1$ , and the exhaust pipe had a throttle valve so adjusted that the fall of pressure during the passage of steam through the turbine was from  $p_1$  to  $p_2$  only, it is



evident that the velocity of the steam jet could be made as low as we please, since this is proportional to the square root of the work done by the steam in expanding between the two pressures. In this way a single wheel turbine could be made to run at a low rate of speed without gearing; but it is evident from inspection of fig. 67 that this would be an inefficient

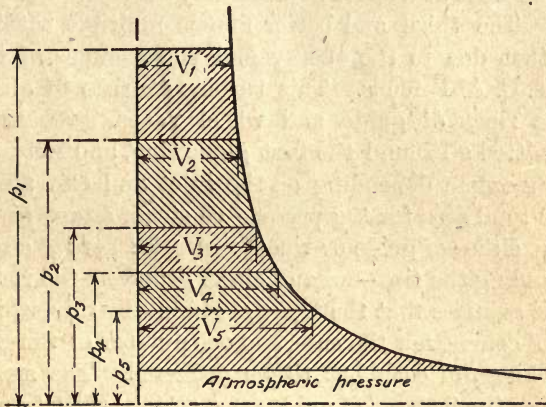


Fig. 67.—Diagram illustrating Work done by "Staging" in Steam Turbines.

arrangement, as the remaining available energy in the steam simply passes uselessly to the exhaust. If a second turbine were provided in which the steam was expanded from  $p_2$  to  $p_3$ , doing work on the second wheel, a further fraction of the energy of the steam would be utilised, and by adding still more wheels in series with one another and mounted on a common shaft, we get the multicellular turbine of Rateau.

An alternative method of securing a low rate of revolution is to expand the steam completely in a nozzle, and then pass it through a series of alternate guide and running wheels. This is known as 'velocity' staging, and is the method adopted in the well-known Curtis turbine. Each wheel abstracts a fraction of the velocity energy of the jet, and can thus rotate at a speed considerably below that of the jet. The third and better known multiple turbine is that due to the energy and perseverance of the Hon. C. A. Parsons. This turbine consists of alternate rings of guide and wheel vanes. Steam is admitted all round the first guide ring, and expands to an extent depending on the inlet and exit areas. The total area for the passage of steam at any given ring, either of guide or wheel vanes, is: area of circle bounding the tips — area of circle bounding the roots. This assumes that the vanes are infinitely thin, which is, of course, not the case, and the number of vanes  $\times$  thickness of each has to be deducted. The steam leaves the first set of guides at a certain velocity and enters the first wheel. Here it gives up its velocity energy by doing work on the turbine shaft, but at the same time it expands further, thus generating kinetic energy for utilisation in the next stage. By choosing a suitable number of stages the steam velocities are kept low, and the turbine runs efficiently at a rate of revolution such that gearing is not required (a Parsons turbine is stated to have been run satisfactorily at the same number of revolutions as a high-speed engine, *i.e.* about 450).

There are other types of multi-stage turbine on the market, but the three above described hold the lead so far, and it now remains to be seen whether the construction of one or the other in model sizes is practicable.

The simplest to both design and make appears to be the Rateau type. There are no fine clearances required, and each stage necessitates only one set of vanes—an important consideration.

The points requiring investigation are: (1) The best diameter of wheel; (2) the most suitable steam pressure; (3) the number of stages to adopt; (4) the number of revolutions. The latter cannot well be under 2000 per minute without either using wheels of too large a diameter for model boat work, or having a large number of stages, which for models is impracticable.

Table V. gives the peripheral speeds at 2000 and 3000 revolutions per minute respectively, of wheels of from 2 in. to 5 in. diameter.

TABLE V.

Diameter of Wheel.	Peripheral Speed (feet per second).	
	2000 revs.	3000 revs.
2 in.	19·44	26·16
3 ,,	29·16	39·24
4 ,,	38·9	52·32
5 ,,	48·6	65·4

In Table VI. the steam velocities for varying ratios of expansion are given. For instance, the figure 1·65

means that the steam is expanded to  $\frac{1}{1.65}$  of its initial pressure (absolute pressures considered).

TABLE VI.

Ratio of Expansion.	Velocity (feet per second).
1.65	1370
1.5	1230
1.25	910
1.1	220
1.05	134
1.02	86
1.01	61

The above values are those obtained by theoretical calculation, and must not be considered as rigidly correct. Steam expands in a nozzle according to the law  $PV^k = \text{constant}$ , and some uncertainty attaches to the value of  $K$ , which alters with the steam pressure and with the initial dryness of the steam. They are, however, near enough for the purpose in view.

Assuming a steam pressure of 60 lbs. (75 absolute) expanding to the atmosphere, the number of expansions will be  $\frac{75}{15} = 5$ .

Referring to Tables V. and VI. we must rule out all but the last three ratios of expansion, as the velocities are too high. Further, the diameter of the wheels should not—for boat work, at all events—go beyond 4 in.

Taking a 4-in. wheel and an expansion ratio of 1.01, let us see what number of stages will be re-

quired for a rate of revolution of 2000 per minute. Calling  $x$  the number of stages, it is evident that  $r$  (the ratio of expansion) multiplied by itself  $x$  times ( $r \times r \times r \dots$  to  $x$  factors)  $= r^x$  will give the number of times the steam has expanded. Supposing, for instance, that the number of stages  $x$  is three,  $r$  being 1.01; then,  $r^x = 1.01 \times 1.01 \times 1.01 = 1.03$ , or the steam has increased its volume by  $\frac{3}{100}$  only, whereas it is required to expand five times to bring it to atmospheric pressure. The number of stages required to attain this expansion will therefore be given by the equation:

$$r^x \text{ or } 1.01^x = 5.$$

The simplest way to solve this will be to take logarithms of both sides thus:—

$$x \times \log r = \log 5.$$

and 
$$x = \frac{\log 5}{\log 1.01} = \frac{0.6990}{0.0043} = 162.5.$$

or 163 stages, an impossible number.

The construction of a multiple turbine of model proportions is thus out of the question, and, as in the case of the model locomotive, a compromise has to be effected.

One difficulty with very small turbines is the enormous speed of the rotor. This means that the least inaccuracy in balance or in the construction and alignment of the bearings will cause considerable loss of power; and it therefore seems desirable to adopt some means for enabling the rotor to be run at reasonable efficiency while its peripheral speed is

considerably below that fixed on for theoretical reasons. One way of doing this is to fully expand the steam in a nozzle and then send it through the wheel several times before it reaches the exhaust.

The photo, fig. 68, illustrates a turbine constructed by the Rev. Bredin Naylor in which this principle is carried into effect. The vanes are mounted on the

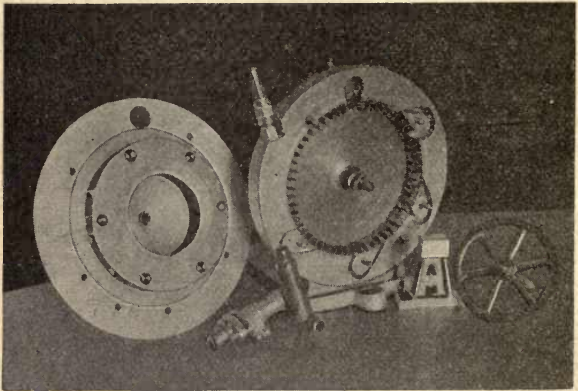


Fig. 68.

side of the rotor disc, and passages are formed in the covers so that the steam passes through the disc nine times. The effect of this is that the disc may run with equal efficiency at one-ninth its theoretical speed. Thus, with a comparatively small ratio of reduction, the propeller shaft revolutions may be kept down to reasonable limits. Mr Naylor has tested this turbine under actual working conditions, coupling it up to a propeller completely immersed

in water contained in a tank. With steam at 30–35 lbs. per square inch, a steam nozzle of 0.032 inches diameter, and a 2-inch propeller, a speed of 600 to 700 revolutions per minute was attained.

After some considerable thought on the matter, the writer has come to the conclusion that the problem of the model marine steam turbine will only be solved in an efficient manner by the use of two or three turbines of the type just described, put in series with one another, thus combining velocity and pressure staging. As it would not be practicable to drive the feed-pump from these high-speed turbines, the following arrangement might be adopted. The turbines should be arranged to exhaust at a pressure of 60 lbs. per square inch into the feed-pump, so that the fall of pressure in the turbines would only be the difference between the boiler pressure and 60 lbs. To fix our ideas, let us suppose that the boiler pressure is 90 lbs., and that three turbines, of the class just considered, are arranged in series with one another, and exhaust into the feed-pump at 60 lbs. The fall of pressure from the boiler to the last turbine exhaust pipe is  $90 - 60 = 30$ , or 10 lbs. fall in each turbine. The steam velocities will therefore be considerably reduced, and for the case considered are about 220 feet per second. For a single-inlet turbine the peripheral speed for maximum efficiency would have to be 110 feet per second, which, with a wheel of 3 inches mean diameter, works out at 8500 revolutions per minute. Sending the steam through the wheel four times, would bring the theoretical

speed down to 2100 revolutions without the necessity for gearing of any kind. The piping from boiler to turbines and pump is indicated diagrammatically in fig. 69. A is a two-way cock which in one position

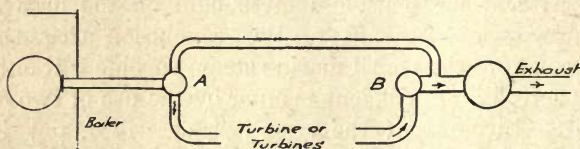


Fig. 69.

admits steam from the boiler to the turbines. In the other position the boiler steam passes into the feed-pump direct. When this takes place, it is necessary to make some provision for preventing the steam going backwards through the turbines, and a

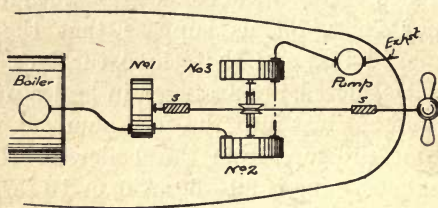


Fig. 70.

non-return valve B—a simple ball check valve—is fitted on the turbine exhaust. In the ordinary way, steam reaches the feed-pump after doing work in the turbines, as explained.

Fig. 70 shows the arrangement of the three turbines. No. 1 is coupled direct to the propeller



shaft through a coiled spring S. Nos. 2 and 3 are placed longitudinally in the boat, and drive on to the propeller shaft by means of pulleys and a coiled spring belt. The steam distribution from the boiler is clearly shown in the sketch. The turbine arrangement sketched out here gives promise of success, and does not require a great deal of work. All pipes should be well lagged, the steam should be superheated, and independent drain-cocks fitted to the turbine casings. A small reservoir might be formed in the bottom of each turbine case, so that the condensed steam which accumulates on a run does not waterlog the turbine wheels.

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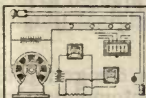
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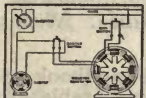


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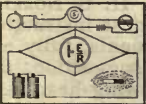


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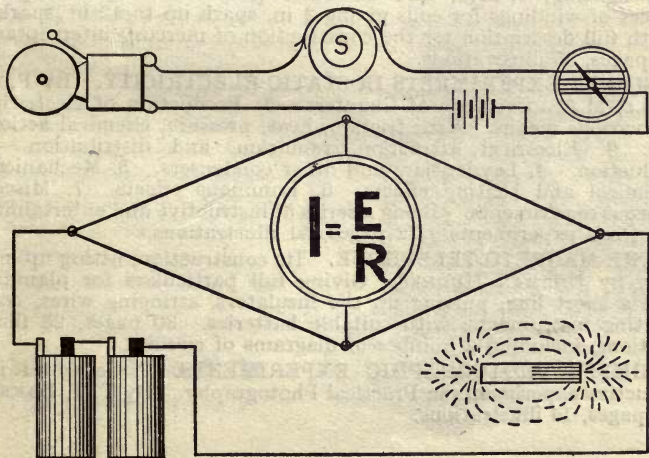
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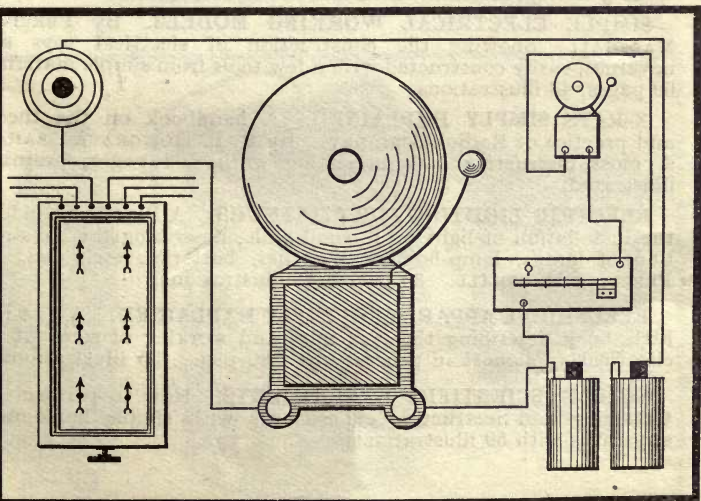
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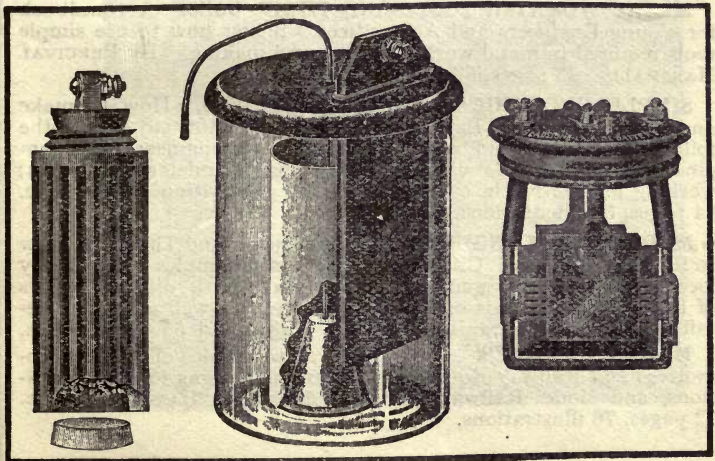
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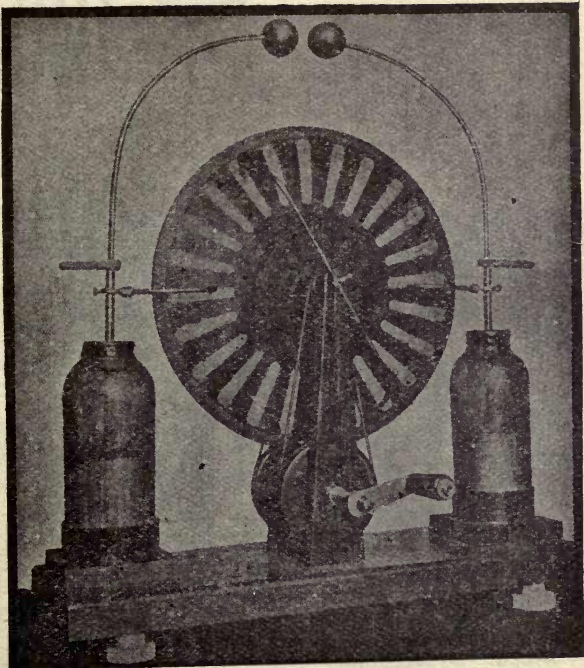
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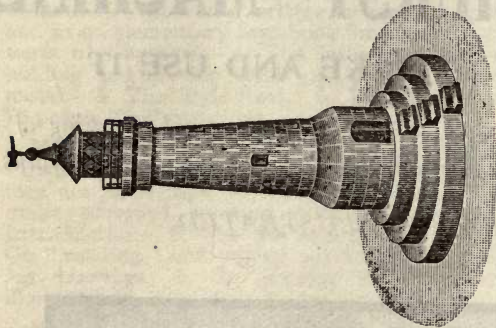
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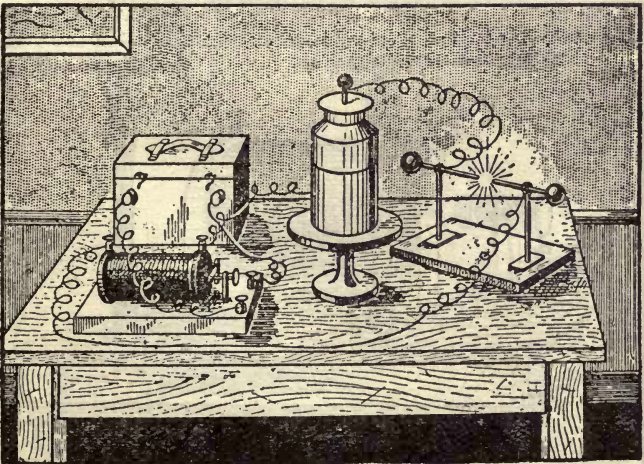
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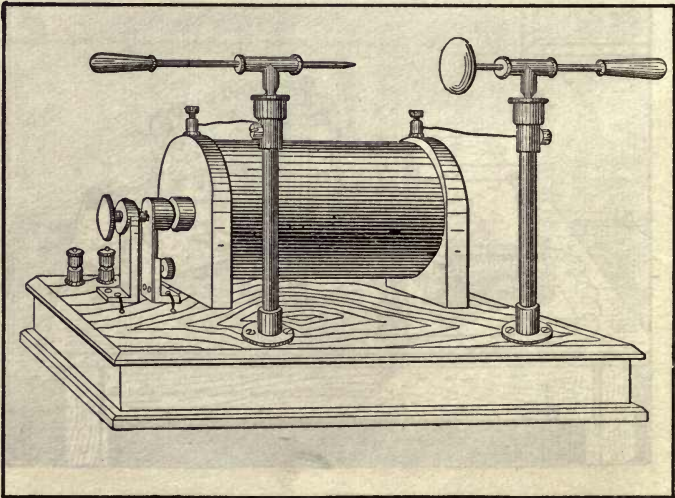
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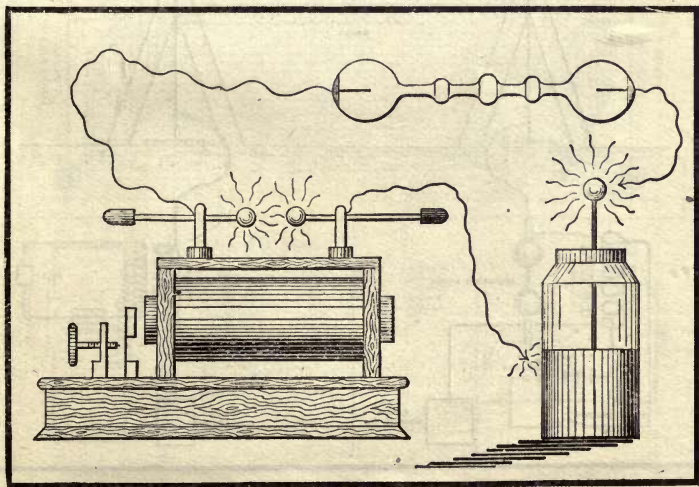
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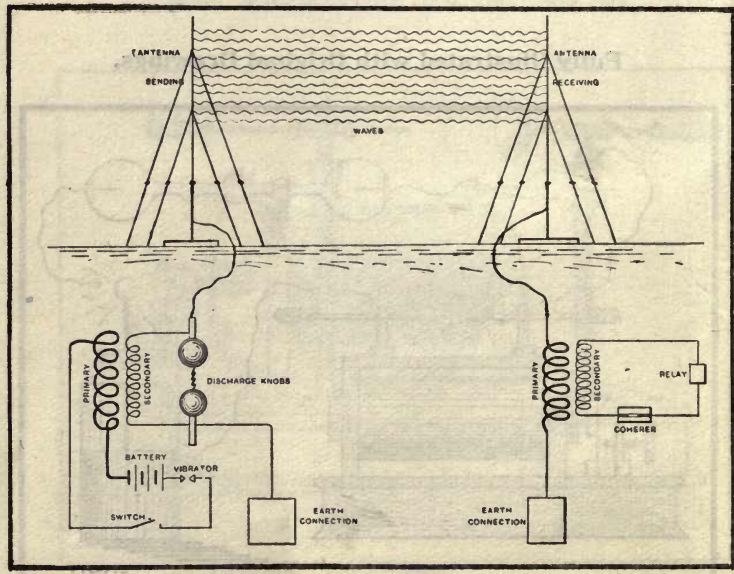
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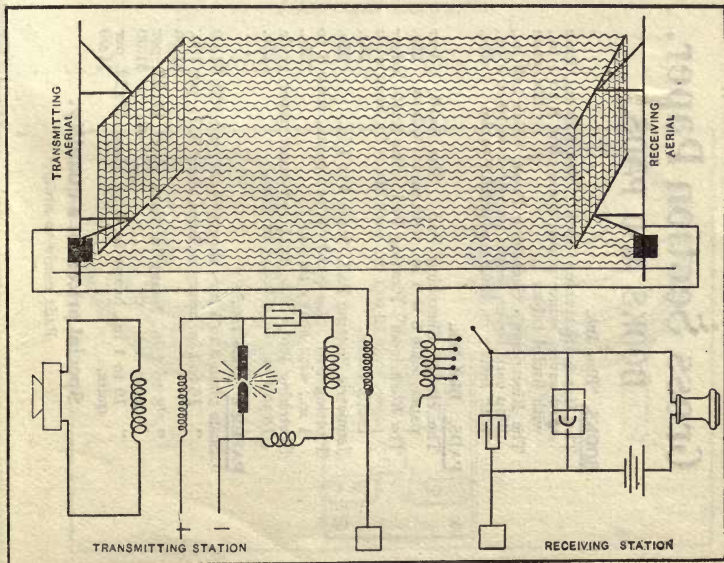
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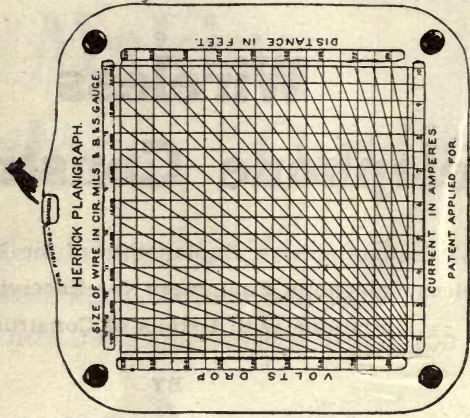
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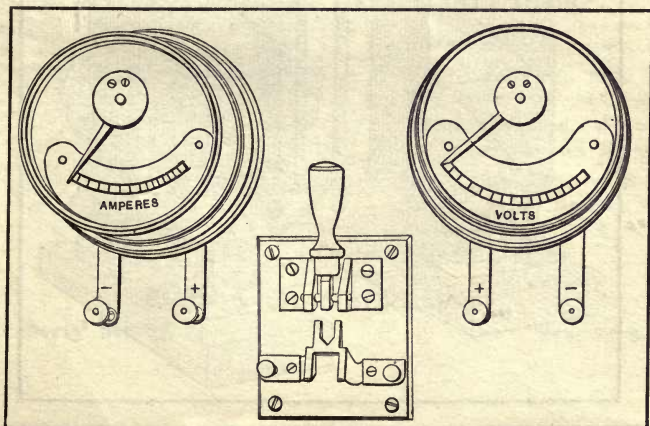
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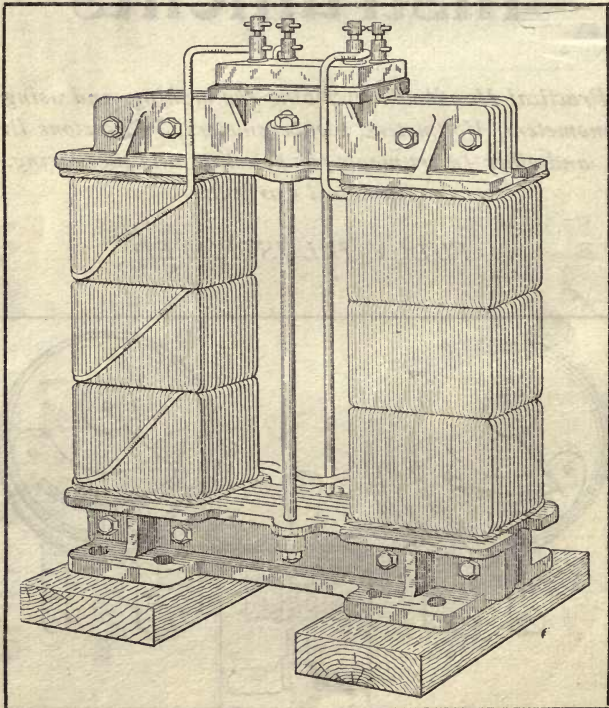
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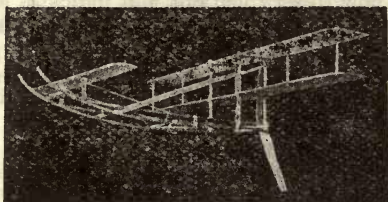
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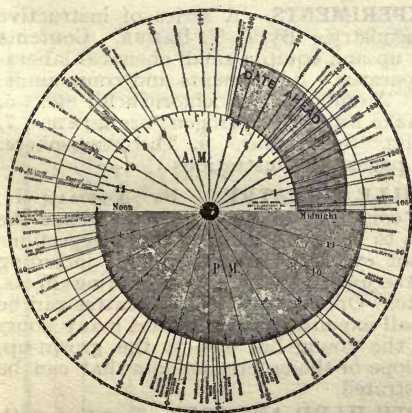
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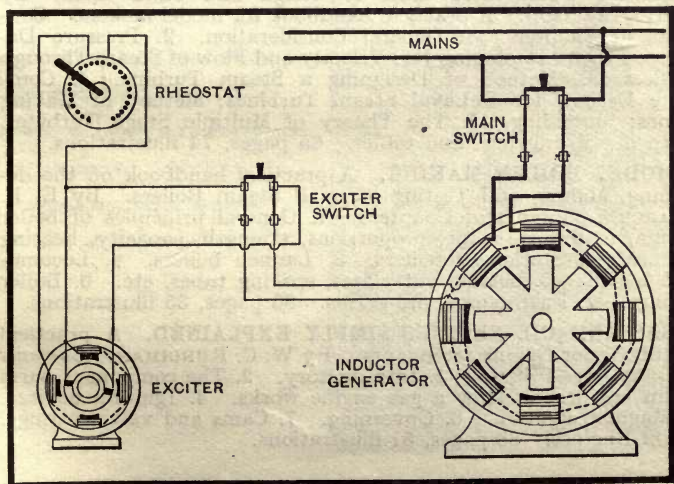
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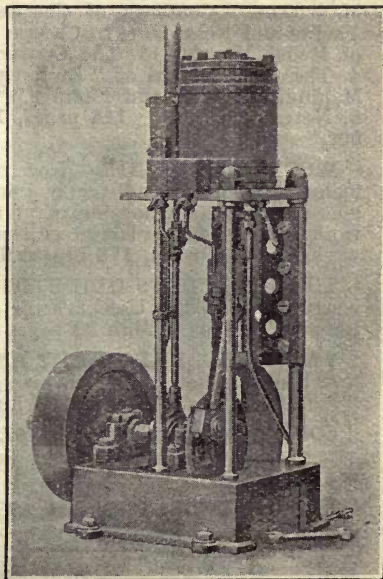
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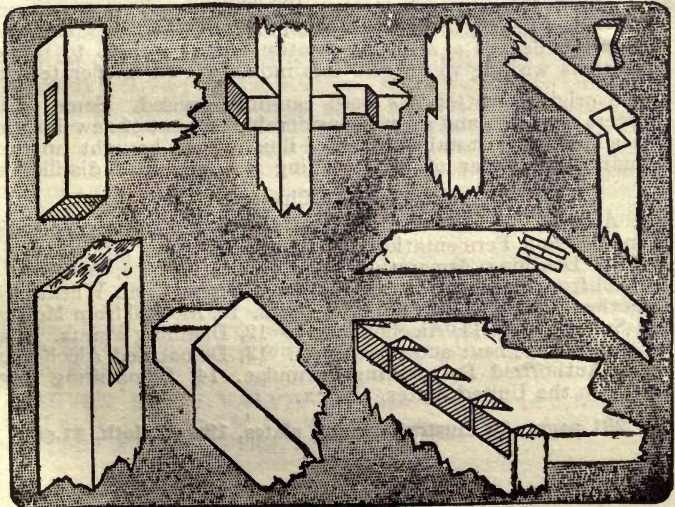
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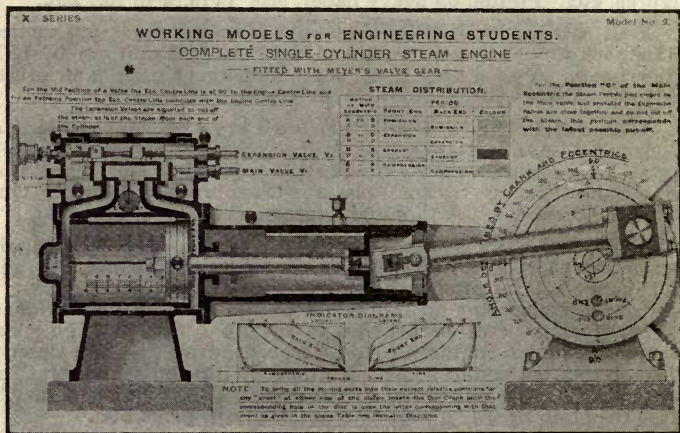
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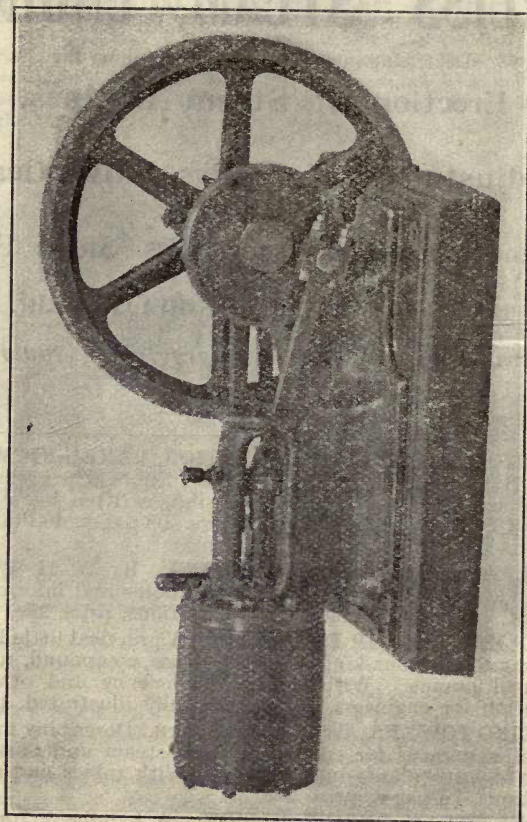
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