ECONOMY OF TURBINES UNDER CRUISING CONDITIONS.

In the early days of the introduction of steam turbines into H.M. Navy, the performance of these units was sensibly independent of the method of operation, and, although satisfactory economy was obtainable at full power, the steam consumption at cruising speeds was undesirably high, while there appeared to be little hope of any immediate advance in this respect. The attention of the engineer and of the designer was thus diverted to the performance of the auxiliary machinery, with the result that during the past few years a marked improvement has been realised in the overall economy at low powers.

The modern requirements of warships, including, as they do, both higher maximum speeds and greatly reduced fuel consumption under cruising conditions, have of necessity stimulated the production of turbines capable of meeting the foregoing needs which could not otherwise be realised. These modern turbines are, however, by reason of their somewhat complicated construction, more susceptible to the effects of improper operation. A correct understanding of the principles involved in their design thus becomes increasingly necessary to the sea-going engineer, and this article has been written with a view to providing information on this subject.

It is of course well known that in every turbine, whatever the type, the maximum efficiency can only be realised at one speed of revolution, and that at all other speeds the efficiency will be less than this maximum. This effect is shown in Fig. 1 where the curve *OAB* represents the variation in efficiency with speed of revolution for a particular turbine wheel operating under constant steam conditions.

The variation in turbine efficiency with respect to that of the speed of the ship is even more marked, and, since the power required to drive a vessel varies approximately as the cube of the speed, the efficiency curve in this case is somewhat as *OCB* in Fig. 1, where the abscissæ refer to the speed of the ship.

The marine turbine designer is faced at the outset with the difficulty of deciding the particular speed of the ship at which the main engines shall realise their maximum efficiency. In general, if the speed selected is too low, the falling off in efficiency at the higher powers will be excessive, due to overspeeding—that is, to the steam striking on the backs of the blades and thus retarding their motion. The losses due to the opposite effect of underspeeding are not so marked, and it therefore becomes desirable to design for maximum efficiency at some power exceeding 80 per cent. of the full output. Even in this case, unless special arrangements are adopted, the total efficiency will decrease at a rapid rate as the power is reduced.

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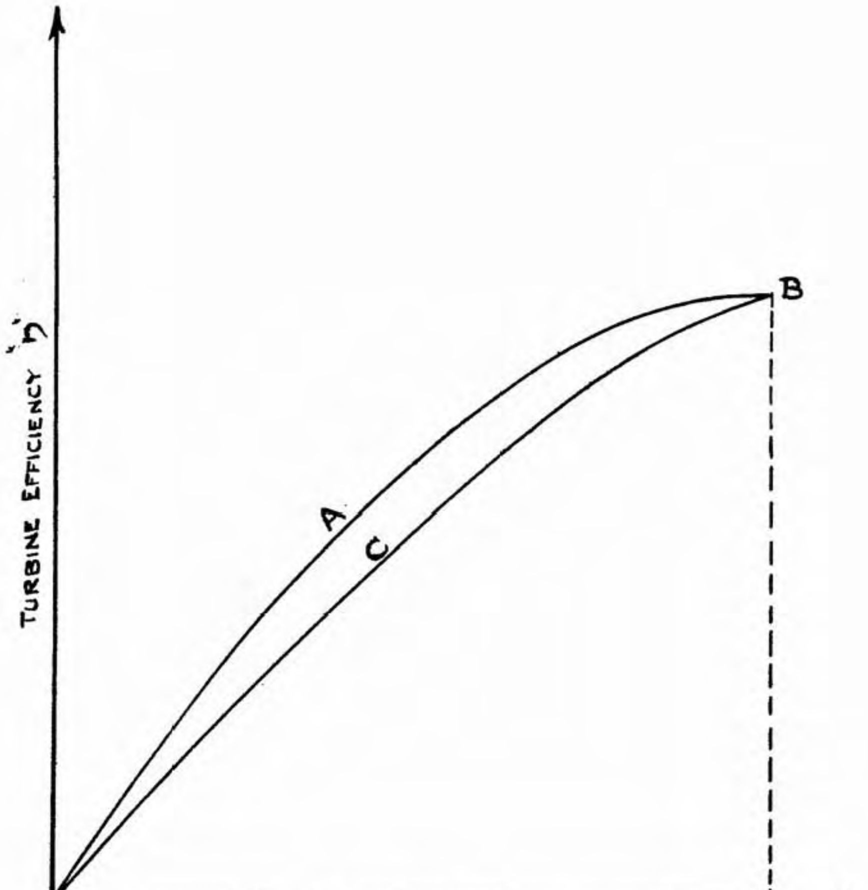

VARIATION OF TURBINE EFF^{Y} WITH SPEED.

FIGURE 1.

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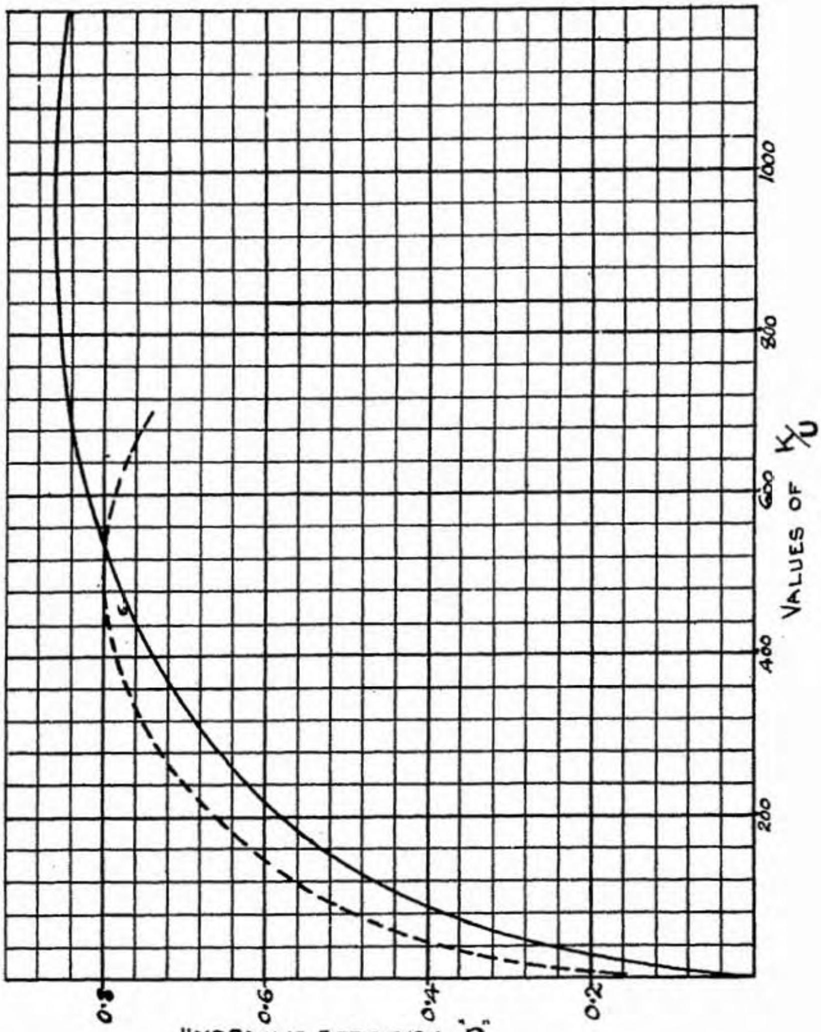

FULL LINE :- REACTION TURBINES
DOTTED " :- IMPULSE —"—

U = TOTAL THERMODYNAMIC HEAD, IN B.T.U's, CORRECTED FOR REHEAT & LOSSES ARISING FROM SUPERSATURATION.

$$K = \Sigma\left(\frac{d}{10}\right)^2 \cdot \left(\frac{R.P.M}{100}\right)^2$$

WHERE d = MEAN BLADE DIA^R IN INCHES & THE SUMMATION INCLUDES MOVING BLADES ONLY.

P.1419. FIGURE 2.

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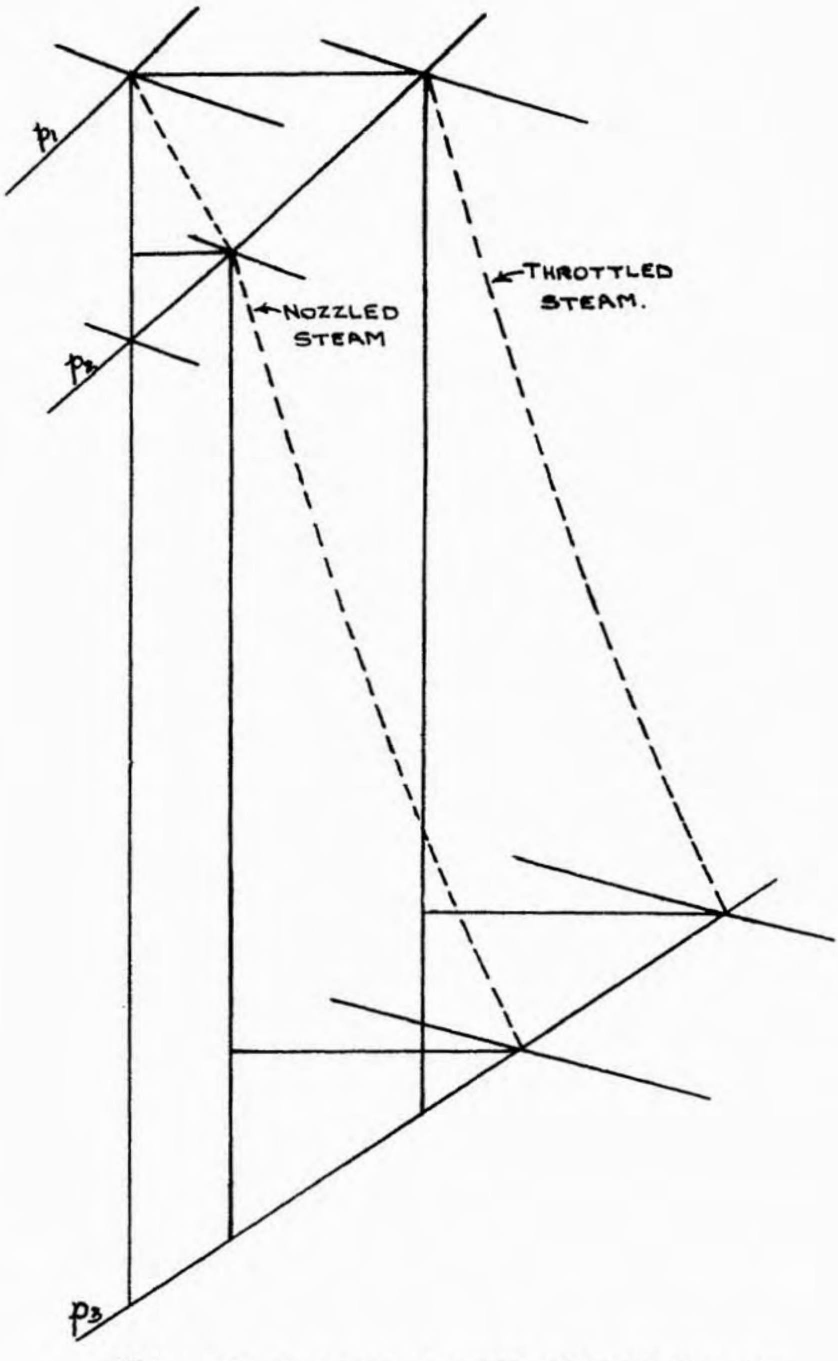

MOLLIER DIAGRAM SHOWING THE EFFECT OF NOZZLING & THROTTLING.

FIGURE 3.

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Efficiency of Turbine Blading.—Before discussing the broad principles of design it is necessary to consider those factors which affect the efficiency of a group of turbine blades. There are, broadly speaking, two such factors, viz. :—

1. The speed ratio ρ between the blade and steam speeds.
2. The condition of the steam, which influences the frictional losses in the system.

Speed Ratio $\frac{(\textit{Blade speed})}{(\textit{Steam speed})}$ clearly depends upon the diameter and revolutions of the rotor and also upon the " head " or " heat drop " available for giving the steam the required velocity. The head, in turn, depends chiefly upon the condition of the steam at the turbine.

The curve of Fig. 2 has been plotted to show how the efficiency of turbine blading varies with variation of a factor $\frac{K}{U}$, the meaning of the symbols being indicated on the figure. It will be observed that $\frac{K}{U}$ is clearly proportional to ρ^2, and that the curve in effect shows how the efficiency of a complete turbine is influenced by the speed ratio. In passing, it is of interest to note that these curves have been derived from practical observations and may readily be used for the solution of problems involving the steam consumption at reduced speeds, under different steam conditions or with rows of blading removed from a given turbine.

The problem of maintaining a constant efficiency under all conditions involves therefore the requirement that the speed ratio also is maintained approximately constant, observing however, that, as shown by the curve in Fig. 2, considerable latitude is allowable in this respect for turbines designed for the values of $\frac{K}{U}$ adopted in current Naval practice, viz., 500 and 900 in impulse and in reaction turbines respectively.

The value of this design coefficient in a given turbine will clearly vary as the revolutions and the steam conditions are altered; the former of these two variables can be predicted from a knowledge of the propeller and hull characteristics; it remains to be seen how the steam conditions will vary as the power is reduced.

Operation of turbines at reduced power.—The power of a set of turbines may be varied by one of two general methods, namely (1) by throttling the steam before it enters the turbine receiver, or (2) by reducing the number, and with it the effective area, of the initial nozzles.

The latter method is the more economical since the effect of throttling is to cause the steam to be exhausted in a higher state, more heat being rejected to the condenser than in the case when pure nozzle control is employed. The Mollier diagram of Fig. 3 clearly illustrates the foregoing statement.

Pressure distribution at reduced powers.—It is an experimental fact, of which considerable use is made by designers, that at any stage in a turbine, a linear relation exists between the pressure of the steam and the fraction of the total full power developed at a given instant, provided that the reduction in power is made by throttling alone. This law also applies approximately to the case of a turbine where the power variations are effected by altering the number of nozzles in use and throttling between the changes of nozzle area. In this case the pressure in the initial stage may be recorded by a series of straight lines, each corresponding to a particular arrangement of nozzles; in the remaining stages the agreement with the true straight line law becomes increasingly exact as the exhaust end of the turbine is approached (*vide* Fig. 4).

Heat drop in initial stages at reduced powers.—It will be seen on reference to Fig. 4 that as the power is reduced so the pressure in the second stage is lowered, with the result that the heat drop available between the maximum possible pressure at the turbines and that in the stage becomes increasingly greater: the curve *D* in Fig. 5 has been drawn to indicate the heat drop available in the first stage under these conditions, expressed as a percentage of the total heat drop for the installation.

It is thus evident that as the power is reduced so the potentialities of the initial stage or stages increase while the efficiency of these stages becomes more and more important.

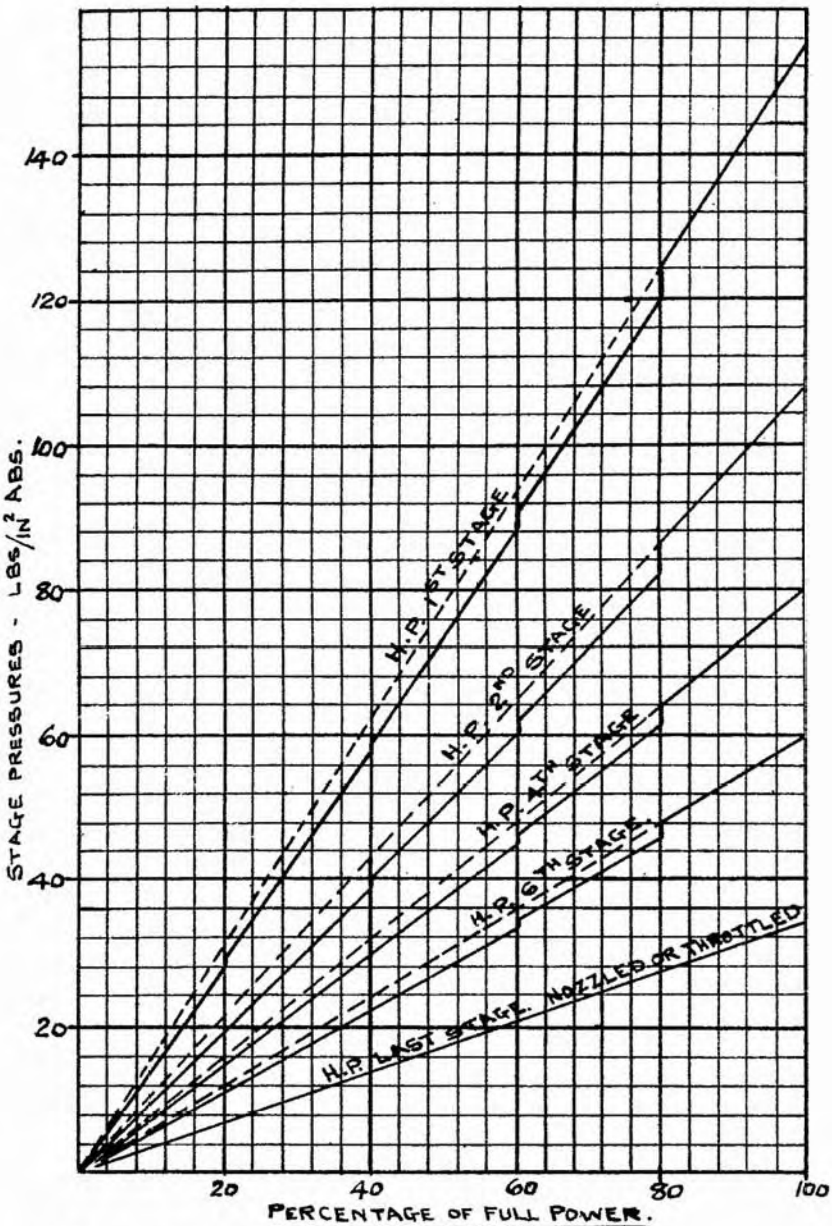
Reduction in power is thus attended by two important effects, namely, a reduction in blade speed, and, as far as the initial stages are concerned, by an increase in the available heat drop with an accompanying addition to the steam speed. The speed ratio in the initial stages is therefore reduced at an increasingly rapid rate, and the efficiency of the initial stages suffers in a like manner if the full steam pressure is maintained at inlet to the turbine.

Efficiency under cruising conditions.—The problem set to the designer thus becomes one of dealing with the increased heat drop in the initial stages in such a manner as to preserve a reasonable value of the speed ratio in these stages under all conditions.

This problem can clearly be attacked in two ways, viz. :—

> (*a*) by increasing the blade speed in the initial stages as the power is reduced, or (*b*) by adding more stages in order to sub-divide the increasing heat drop in sensible agreement with the decreasing blade speed.

Method (*a*) has only one practical application, namely that in which a cruising turbine is introduced into the steam circuit at reduced powers; the use of gearing between the cruising unit and the main turbine enables the blade speed of the former to be adjusted to the desired condition without the necessity for a rotor of large diameter.



DOTTED LINES - THROTTLED.
FULL LINES - NOZZLED.

FIGURE 4.

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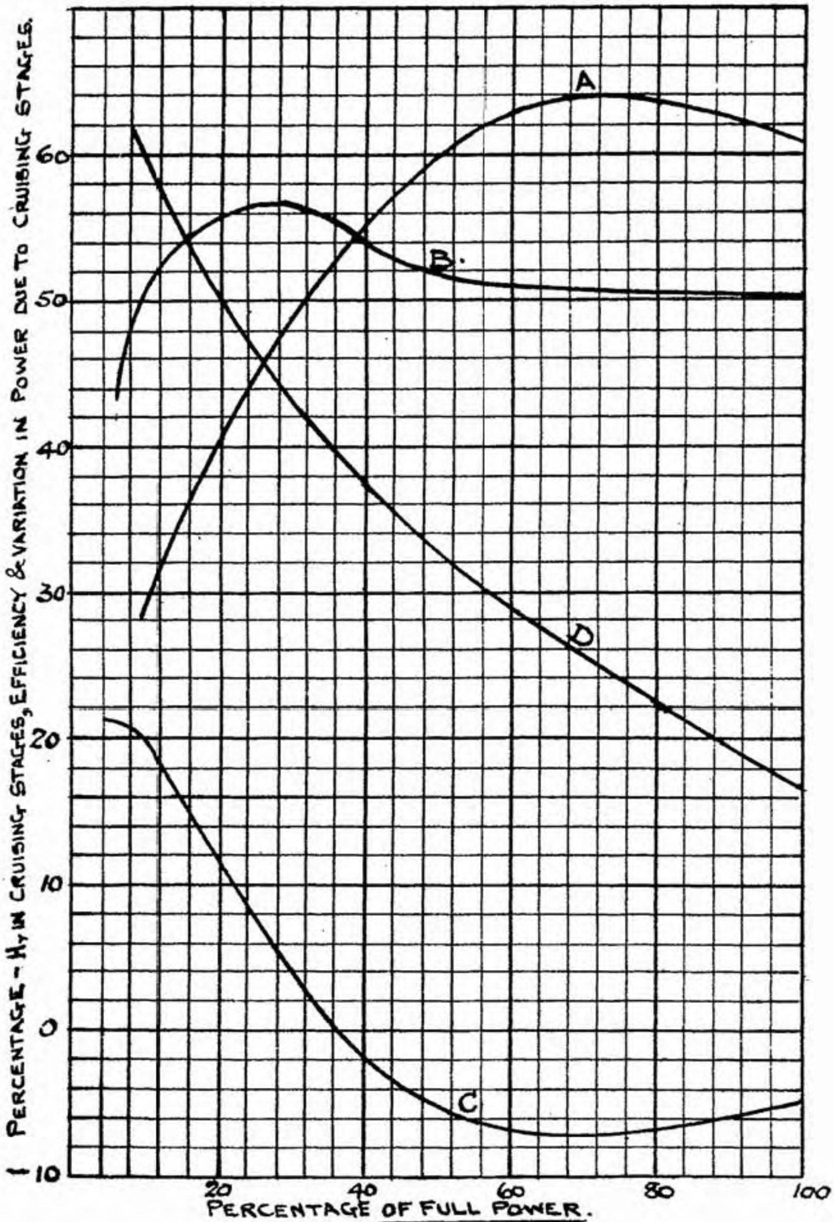

A - EFFICIENCY FOR SINGLE VELOCITY STAGE.

B - —"— " THREE —"— STAGES.

C - VARIATION IN POWER DUE TO CRUISING STAGES.

D - PERCENTAGE H_T IN CRUISING STAGES.

FIGURE 5.

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Method (*b*) is that most largely employed, and is achieved in practice by the adoption of separate cruising turbines (geared or ungeared) or by the use of special cruising stages arranged on the H.P. turbine shaft at the inlet end.

Separate cruising turbines, however, demand extra weight and complication, while it is usually necessary to arrange for their disconnection at comparatively low speeds of the ship as considerations of weight and space do not permit of these units being designed to rotate when the main plant is developing more than (say) 30 per cent. of its full power. The use of geared cruising turbines involves still further complication, but an appreciable reduction of the space requirements is achieved thereby, without any substantial alteration in weight, as compared with direct coupled units.

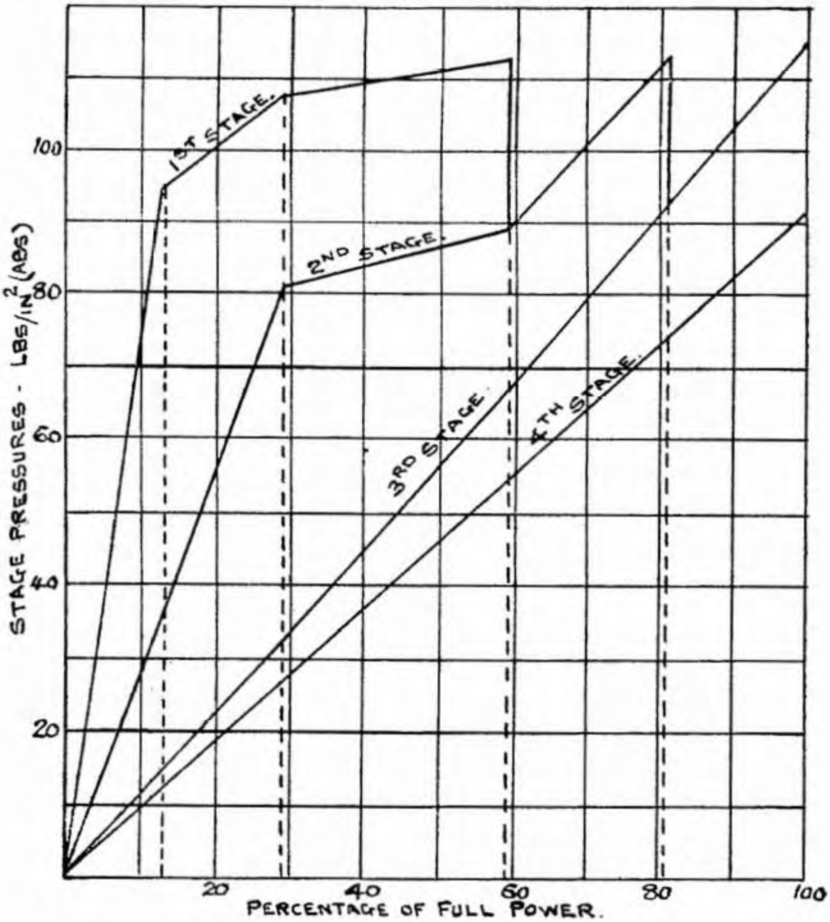
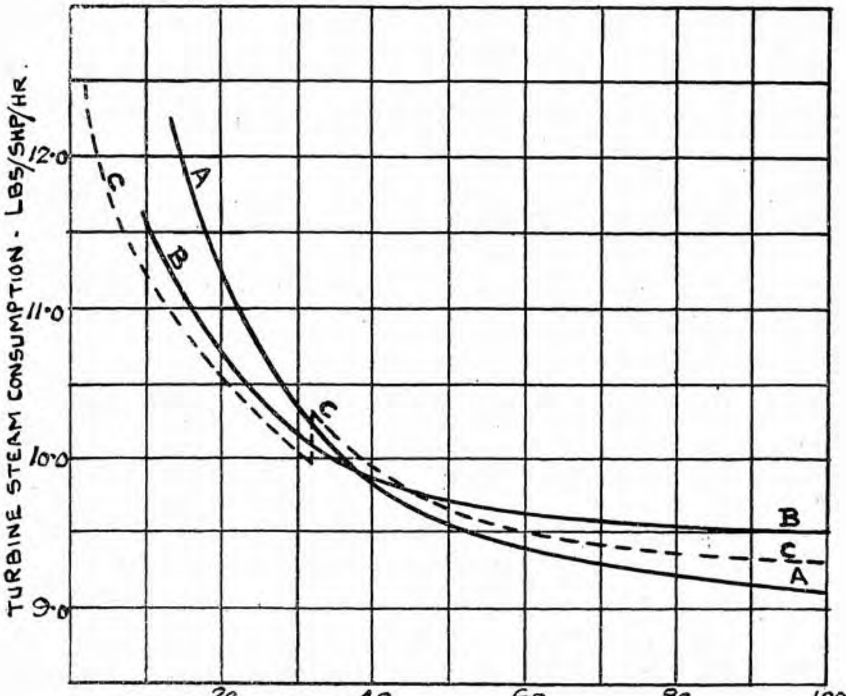


Fig. 6.

Cruising stages are arranged on two different systems, one being suitable for use in Impulse turbines only, and the other being applicable to any type of turbine, but finding most general use in those of the reaction type.



AA - SINGLE VELOCITY STAGE. BB - THREE —"— STAGES. CCC - SEPARATE CRUISING TURBINES.

FIG. 7.

In the former method, the stages are arranged to be worked in parallel at the highest powers and all in series at low outputs, combinations of series and parallel working being employed under intermediate conditions. The alternative system provides for the admission of the steam supply at stages progressively more remote from the inlet end of the H.P. as the power is increased, thus shortening the total path of the steam under such conditions; at full power, therefore, all the cruising stages are running idly in steam at the pressure obtaining in the first main stage.

Whatever the method employed, unless the cruising arrangements can be disconnected when no longer required, a loss of efficiency at high powers must necessarily follow the adoption of such devices, since not only is the efficiency of the latter poor under such conditions but also they add appreciably to the rotational losses of the turbine. This effect may clearly be seen by examination of the curves in Fig. 7, where comparisons are made between the steam consumptions of a particular design of turbine fitted with different cruising arrangements. It will be seen that the shape of these curves may be altered as the designer wishes, subject to the limitations imposed by any particular design. Turbines with characteristics such as curve A are evidently suitable for vessels which run at one particular speed, *e.g.*, liners and other merchant craft, while curves such as C represent the type desirable for warships which are required to make prolonged ocean voyages at moderate speeds and yet be capable of very high speeds for shorter periods.

The curves illustrate also that by the use of separate cruising turbines very high efficiencies can be attained at extremely low proportions of full output, such arrangements being essential for particular requirements where an extreme radius of action is demanded at very low proportions of full power.

Cruising Turbines.—Special cruising turbines have been included in a large number of Naval designs, varying from the directly connected units fitted to the early turbine battleships to the geared arrangements in the later vessels. The curves C and C^1 indicate the steam consumption of a set of turbines fitted with a geared cruising turbine, C being that realised without the use of this adjunct. It will be observed that a discontinuity exists between these two curves in the neighbourhood of 30 per cent. full power, corresponding to a speed of 21 knots in the particular vessel where this installation was fitted. The break in the curve is due to the change from the cruising to the main turbines, an operation which is usually performed immediately that the maximum available steam pressure is attained at the inlet belt of the former units. Improved economy may, however, be obtained by delaying the change-over until the safe limit of peripheral speed has been reached in the cruising turbine, which may be worked in parallel with the main set during the intervening period.

Cruising turbines have generally been arranged to drive the same propeller shaft as the H.P. turbine into which they exhaust. In a few cases of four-shaft installations, however, cruising turbines have been provided in connection with (say) the two wing shafts, being arranged to exhaust into the H.P. turbines on the centre shafts. This arrangement may be somewhat more economical at particular speeds than the more usual one, since the output of each cruising turbine is thus approximately doubled, for a given speed of ship : the principal advantage of this arrangement lies in the fact that the condensing plant in connection with the outer main turbines need not be run, provided that the cruising turbine drains are led to the condensers on the inner shaft.

Cruising Stages, Reaction Turbines.—Fig. 8 illustrates an arrangement of cruising stages fitted to reaction turbines, consisting of an impulse stage, compounded for velocity, followed by reaction stages, all situated at the inlet end of the H.P. turbine. Suitable valves are fitted to enable steam to be admitted at C

between the cruising stages, while the full power inlet is arranged at D, that is at the inlet to the main portion of the turbine : the nozzles to the velocity wheel are fitted with control valves, thus enabling further economy in operation to be obtained.

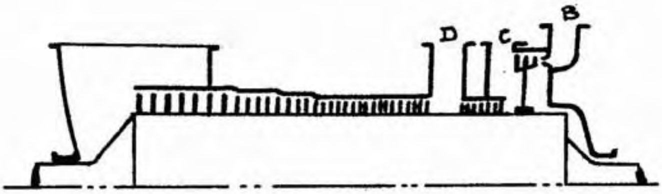
At the lowest powers steam is admitted to the velocity stage and passes through all the cruising stages in series. As the output is increased, steam is permitted to enter at C till about 60 per cent. of the full output is reached, when the full power inlet D is opened.

The blading is so proportioned that at full power the pressure at D is less than that in the cruising nozzle box and thus a slight flow of steam may be maintained through the cruising stages in order to avoid overheating of the blades. It is to be noted, however, that the H.P. dummy piston is arranged in connection with the velocity stage and thus even if no steam is admitted at B and C under high power conditions, there will still be a flow of steam through the cruising stages, but in the reverse direction. Experience alone can decide which is the more economical condition, but it appears preferable to maintain a slight flow of steam in the normal direction through B in view of the greater security and more ready operation so obtained.

Cruising Stages in Impulse Turbines.—A typical arrangement of cruising stages as fitted to a set of impulse turbines is shown in Fig. 9. The particular installation illustrated consists of three sets of velocity compounded wheels; nozzle control valves and bye-pass valves are arranged so that the stages may be worked in series at the lowest powers and all in parallel at full power, while combinations of series and parallel working may be employed under intermediate conditions.

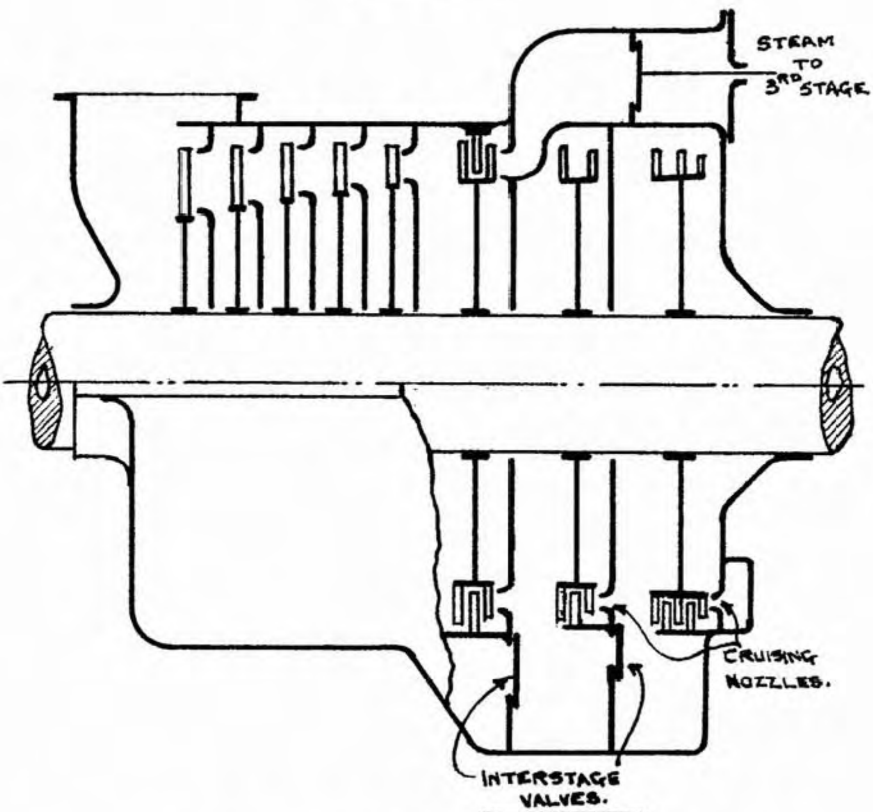
Each of these wheels is designed to develop its maximum efficiency at a different turbine speed, and the efficiency of the whole turbine is determined by the judgment of the designer in this respect. In order to maintain a reasonable economy at full power, it is usual to design one of these wheels (conveniently the last one) to deal with the majority of the steam flow under these conditions, when also it must be arranged to develop its maximum efficiency. The initial and intermediate wheels in such a design are usually arranged for smaller proportions of the total steam flow, being designed to realise their maximum efficiencies at speeds corresponding to, say, 30 per cent. and 60 per cent. full power respectively. Thus at high powers the third stage plays the predominant part, while, as the output is reduced, the first and second stages provide a very satisfactory degree of economy, despite the low performance of the third stage under such conditions.

The effect is well illustrated by Fig. 10 where the efficiency of each cruising stage is separately indicated over the whole range of power. The nett efficiency of three such stages may be



SECTION THRO' H.P. REACTION TURBINE WITH IMPULE WHEEL

FIGURE. 8.



SECTION TROUGH IMPULSE TURBINE WITH 3 VELOCITY COMPOUNDED STAGES.

FIGURE:- 9.

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CURVES OF EFFICIENCY OF A TURBINE WITH THREE VELOCITY STAGES.

I. 1ST STAGE
II 2ND —"—
III 3RD —"—

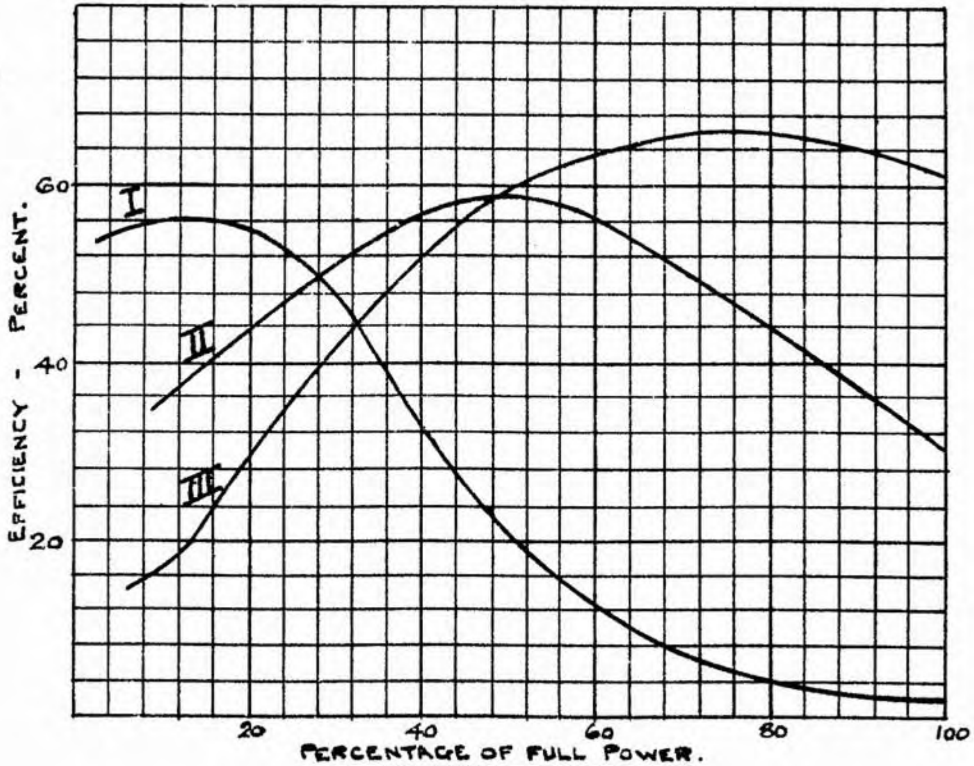

FIGURE. 10.

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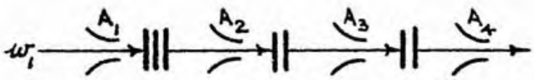
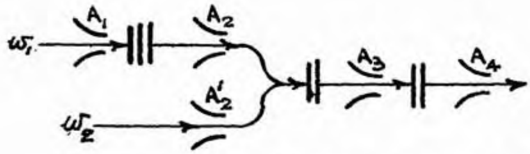
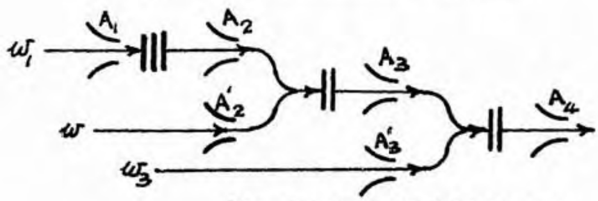


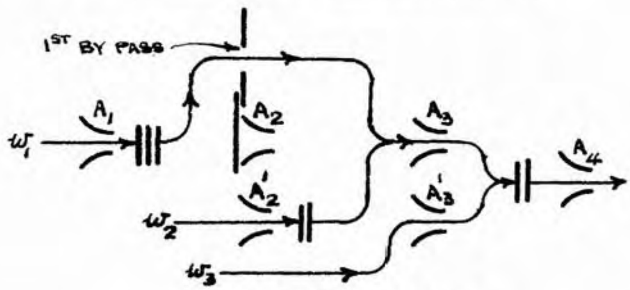
ALL STAGES IN SERIES.



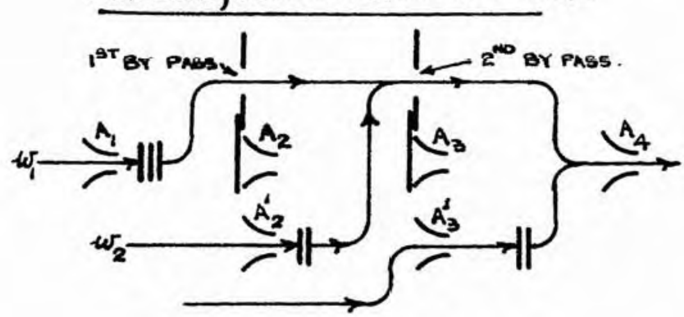
ALL STAGES IN SERIES BUT
LIVE STEAM TO 2ND STAGE.



ALL STAGES IN SERIES BUT
LIVE STEAM TO 2ND & 3RD STAGES.



1ST & 2ND STAGES IN
PARALLEL; INCREASED STEAM TO 3RD STAGE.



ALL STAGES IN PARALLEL.

FIGURE:- II.

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compared with that of a single initial stages by examination o: curves A and B in Fig. 5: the gain or loss of power due to the cruising stages is shown by the curve C in the same figure.

Arrangements for controlling the steam to cruising stages.—The initial nozzles for the first stage are arranged in the centre of the lower half of the turbine as are also the "series" nozzles for the second and third stages: the term "series" is here used to indicate those nozzles through which the steam passes from one cruising stage to the next. Control valves for the admission of steam at the initial pressure to the nozzles of second stage are arranged on the forward end ot the turbine on either side of the first stage nozzles, suitable channels being cast in the turbine casing to convey the steam to the second stage: such nozzles are hereafter referred to as "primary" nozzles.

The primary nozzle control valves for the third stage are situated at the forward end of the turbine but in the top half, being connected to their appropriate nozzles in the same way as arranged for the second stage.

Bye-pass ports are provided in the first and second diaphragms, and slide valves, arranged to be operated from convenient positions, are fitted in such a manner as to cover the bye-pass ports when in one position while closing off the series nozzles when at the opposite end of their travel.

The action of the bye-pass valves may best be understood from a consideration of Figs. 6 and 11, on which are recorded the steam pressures in the cruising stages at various percentages of full power, together with diagrammatic sketches of the method of steam distribution.

Let us follow the process of working up to full power from slow speed. Initially all the cruising stages are in series, the slide valves therefore being set to cover the bye-pass ports and steam being admitted through the control valves to the first stage. As the power is increased, by admitting more steam to the first stage, the pressure in all stages gradually rises till a point is reached when the first stage nozzles are passing their full quantity of steam. The heat available for this stage when the point of maximum flow is first reached, is, however, more than sufficient to produce this steam flow, the surplus heat being partially used as reaction in the blading. Additional supplies of steam are now provided by opening the nozzle control valves to stage two, thus causing a further progressive rise in the stage pressures throughout the turbine.

The rise of pressure in the first stage does not at first entail a decreased steam flow, merely resulting in reduced reaction, but eventually when the pressure in the stage is approximately equal to the critical pressure, it becomes necessary either to accept a reduced steam flow or to lower the stage pressure by bye-passing. When the pressure in the first stage has risen to the point B (Fig. 6) the bye-pass valve in the first diaphragm may be raised to cover the series nozzles, thus lowering the first-stage pressure to that obtaining in the second stage and so permitting maximum flow conditions to continue in the first stage nozzles.

Alternatively the pressure in the first stage may be permitted to rise, the accompanying reduction in steam flow in this stage being made good by the admission of further supplies through the second or third stage primary nozzles.

Theory indicates that this latter arrangement should result in a somewhat reduced efficiency since, not only will the rotational losses in the first stage be increased on account of the greater density of the medium in which the wheel is revolving, but also the speed ratio may be unfavourably affected by the resulting reduction in the steam speed. On the other hand the admission of further quantities of steam to the second or third stages may tend to improve the speed ratio for these wheels and thus to increase their efficiency; trial alone can determine which alternative is the more economical under given conditions.

After the maximum steam flow in the second stage primary nozzles has been reached, further supplies of live steam are admitted to the third stage. The third stage series nozzles are bye-passed when the second stage pressure has reached the point C, and thereafter the unit is worked with all three stages in parallel till the Full Power is attained.

In some designs the bye-pass valves are arranged merely to uncover the ports without shutting off the series nozzles—this may result in a somewhat different efficiency to that of the alternative arrangement described, but here again trial alone can satisfactorily decide which is the more economical method, although it appears probable that the difference between the two designs will be inappreciable in practice. This latter method leads to simplification in design and to a small saving in weight.

It will be apparent that such arrangements of cruising stages in impulse turbines offer some scope for experiment in order to establish the most economical method of operation, and it is hoped that the foregoing brief description will serve to point out the possibilities of the system. The requirements for the maximum economy are :—

(1) The maintenance of the maximum possible pressure at entrance to all primary nozzles.

(2) Distribution of the steam among the stages in order that the speed ratio corresponding to maximum efficiency may be realised in each case.

(3) The maintenance of the minimum stage pressures in order to reduce windage losses and leakage through the diaphragm glands.

As regards (2) it may be noted that the value of the speed ratio ρ corresponding to maximum efficiency is about $0 \cdot 225$ in

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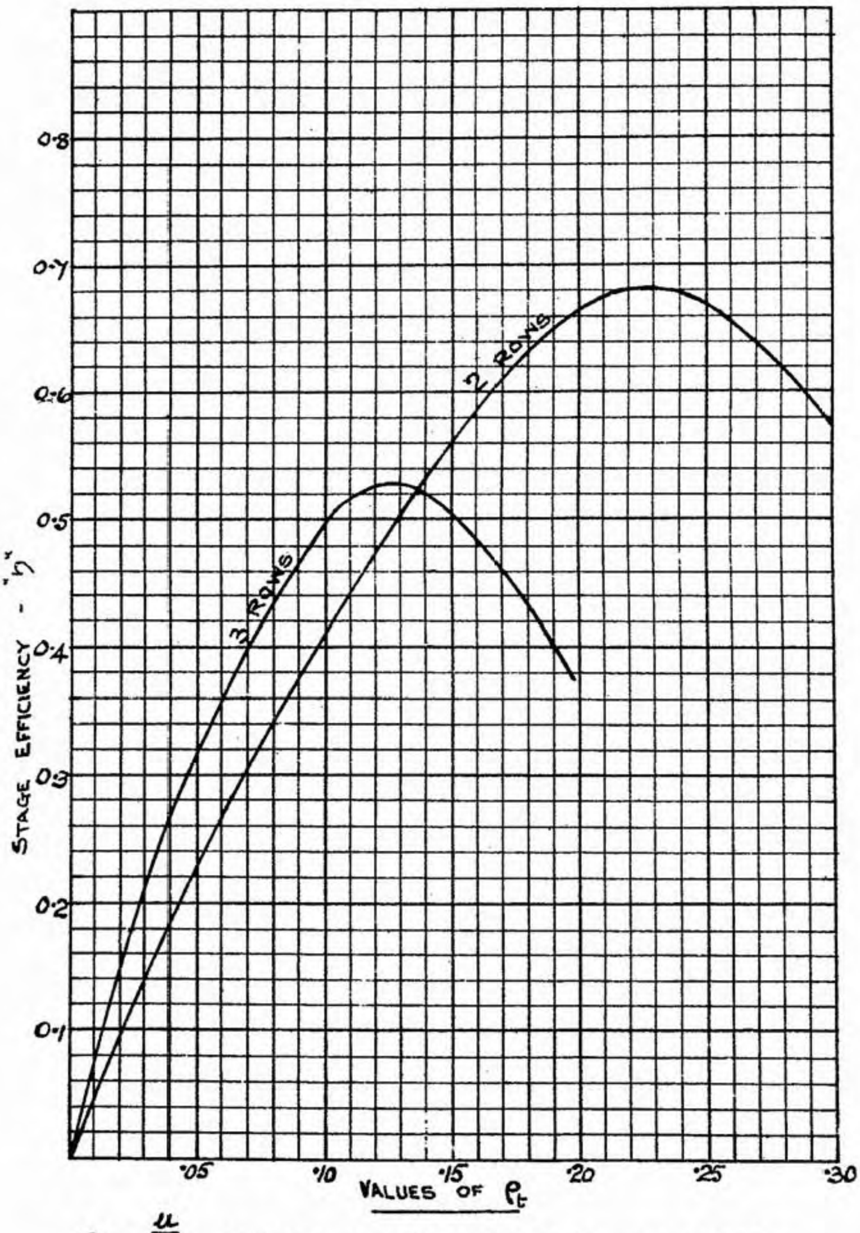

$\rho_t = \frac{u}{V_t}$ WHERE:- u = MEAN BLADES VELOCITY IN FT/SEC.

$V_t = 223{\cdot}7\sqrt{h_r}$

h_r = RANKINE HEAT DROP IN NOZZLES & BLADES.

FIGURE:- 12.

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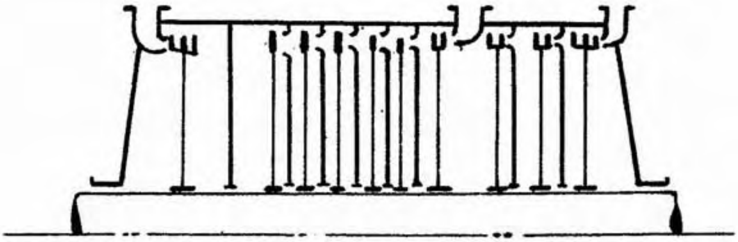

FIGURE:- 13.

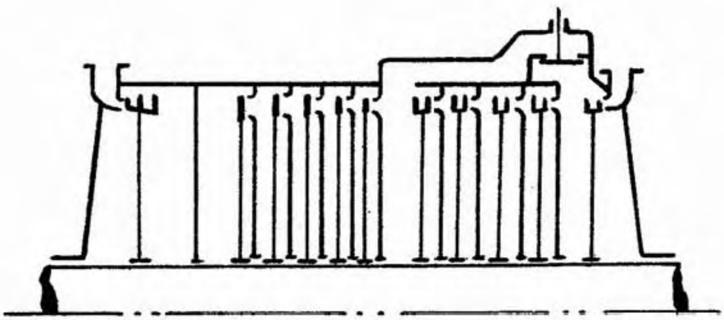

FIGURE:- 14.

TYPICAL ARRANGEMENT OF BELLUZZO TURBINES.

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two-row stages and 0·125 in three-row stages. The values quoted are for the theoretical speed ratio

$$\text{ratio } \rho_t, \text{ given by } \frac{\pi DN}{60 \times 223\cdot 7\sqrt{Hr}}$$

Where D = pitch circle diameter in feet.
N = revolutions per minute.
Hr = Rankine Heat drop from the state at inlet to the nozzles to that at exit from the last moving blade, that is the whole heat drop allotted to the stage.

Curves showing the variation in efficiency with change of ρ in typical designs of velocity compounded wheels are given in Fig. 12.

Methods not in use in H.M. Navy.—It has been pointed out earlier in this article that efficiency at cruising speeds can only be obtained by the use of devices whereby the increased heat drop which obtains in the initial stages under these conditions can be economically dealt with. A simple method of partially achieving this end entails the provision of special cruising nozzles at inlet to the H.P. turbine; these nozzles are designed to deal with the large heat drop at low power, their angles being adjusted to improve the efficiency of the first stage under such condition. This method does not, however, result in any very marked improvement, while in many designs its incorporation becomes impracticable.

The principal disadvantage of arrangements of cruising stages such as those described is that the rotational losses of the stages designed for low power working become of some importance at full power, as these stages then revolve in steam of considerable density; the arrangement of Fig. 9 is clearly better in this respect than that shown in Fig. 8.

Some doubt exists, however, regarding the extent of the rotational losses in turbine wheels, and designers are not agreed as to whether any real advantage is to be gained at Full Power by admitting steam to the purely cruising stages. An Italian designer, Dr. Beluzzo, holds the view that the low power wheels should be permitted to revolve idly at high outputs of the set, claiming that the design and operation of the unit is much simplified thereby. Typical designs on this principle are illustrated by Figs. 13 and 14, of which the latter is the more simple to operate since the whole of the steam enters through A at all times, the bye-pass B being opened as the full output of the turbine is approached.

Conclusion.—The design of turbines suitable for modern naval requirements is by no means an exact science, the complexity of the problem preventing it from attaining that dignity; it is thus important that all possible information should be rendered regarding the working of the arrangements of turbines that are introduced into the Naval service. When one method of operation appears to give superior results to another it is, however, very necessary to make certain that the improvement is in fact due to the cause to which it is ascribed, and that it has not been made apparent by unnoticed factors, *e.g.*, variations in steam pressure or vacuum may completely mask the effects of different methods of operation.

Information regarding new features in machinery installations may be of the utmost value, but only if it is complete ; redundancy of material is of far greater service than any insufficiency, however strongly supported by opinions based upon superficial observations. Accurate and complete analysis is usually impossible under ship conditions, but the value of any data so obtained is greatly enhanced if accompanied by statements regarding the probable accuracy of any included figures and by estimates of those factors which could not be observed.