SOME ASPECTS OF NAVAL TURBINE DESIGN

PART I

by

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STEAM CONDITIONS

The mechanical work which a suitably designed turbine can extract from each pound of steam supplied to it increases, in general, as both the pressure and temperature of the steam increase.

These increases are very marked at moderate temperatures and pressures, but fall in magnitude as the higher values are reached.

The increases in work obtained from higher quality (*i.e.*, higher temperature and pressure) steam are gross, not nett, increases for the cycle, because to obtain the higher quality steam extra heat has to be added in the boiler and superheater.

The overall efficiency of the cycle does, however, increase with rising steam conditions.

To show graphically the theoretical increases is not easy, for there are many variables and a given increase in steam temperature does not have the same effect in cycle efficiency at one pressure as it does at another. Nor is the effect of a given variation of pressure constant for varying temperatures.

In general, for any total steam temperature which may be decided upon, there is a range of suitable pressures. Outside the range, factors which tend to unbalance the design become significant. For example, if the pressure is raised unduly without corresponding increases in its total temperature, the steam in the later stages of the turbine will become very wet, with a consequent loss of efficiency and increased incidence of blade erosion. A combination of conditions is usually chosen which will give a final wetness of, say, 5% to 8%. In some turbines where special arrangements are made to separate water from the steam in the later stages a higher percentage wetness is allowed ; but such special arrangements can only be incorporated at some cost in other directions and have not so far been used in naval designs.

Materials in general lose strength with increasing temperature ; and in modern naval designs turbine rotating parts are stressed as highly as prudence permits as a result of the search for high efficiency. Up to 700°F the loss in strength of carbon steel is not very significant. At temperatures above 700°F either the parts must be designed for lower stresses or less common materials employed. Above about 800°F another factor—creep—must be taken into account, and not only must the stresses to be imposed be considered in relation to the U.T.S. or proof stress of the material at the working temperature, but it must also be decided whether the dimensional changes that these stresses will produce (due to creep) during the stipulated working life are acceptable.

The range of suitable materials of which there is working experience at 800°F or so, and which can be forged in the sizes required for the rotors of a large set of naval machinery is not very great, and is almost entirely composed of the relatively low alloy steels, heat treated to give the optimum mechanical properties.

At 800°F, creep in rotors made of such steels is only just becoming a significant factor at normal working stresses ; at 875° it may well be a controlling one in many cases. Above about 1,000°F, at the rotor the low alloy steels become unsuitable for most designs and it would be necessary to turn to the austenitic steels—going out of the present range of steam experience and into a field of many potential difficulties, and reduced potential gains.

The austenitic steels are costly and have seldom been produced in large forgings ; they will be dealt with a little more fully under the heading " Rotors ". If, however, they are to be employed in spite of their many disadvantages there is probably very little reason as far as they are concerned why the temperature should not be advanced further—that is, if the cost and other drawbacks are to be accepted it would be reasonable to advance the temperature say to 1,150°F at the rotor—in order to reap to the full the diminishing benefits available.

The above remarks have been largely confined to the revolving parts of the turbine ; the problems of differential expansions, absorption of clearances, casing distortions and the like can be expected to increase with temperature ; and can also be expected to be superable by development and design.

The result of increases of temperature upon the cycle as a whole and upon each item of the installation affected must be similarly examined. Really high pressures lead to boiler tube plates of great thickness—for a 1,500 p.s.i. boiler drum 50 in diameter the tube plate might be 6 in thick—with consequent increases in weight and manufacturing difficulty ; and steam temperatures above 950°F probably demand superheater tubes of material which could at present be expected to involve difficulties in manufacture and hazards in service.

Auxiliary turbines are, at higher temperatures and pressures, likely to detract so much from the efficiency of the cycle as a whole, particularly at low total powers, that alternative methods of drive might become imperative. Steam pipes at such advanced conditions, although they become relatively extremely thick and probably inconveniently stiff, would not be expected to present problems of comparable magnitude.

These points are set down in order to emphasise that the steam conditions can only be decided after a close examination of the whole cycle and installation.

Partial load characteristics

Until a few years ago the propelling machinery of British warships was designed to give its greatest efficiency at or near full power. As the efficiency cannot be kept constant throughout the power range this necessitated reduced efficiencies at reduced powers. The hull resistance curve of a ship falls much faster than does the efficiency of the machinery as speeds decrease, so that the economical speed was always low ; but the fuel consumption at the economical speed was never so low as it would have been if the efficiency of the engines could have been maintained at the reduced powers. In many instances cruising turbines were added in an effort to improve the partial load efficiency of the plant ; but many of these did not completely solve the problem, and apart from increasing weight, space requirements and cost, had operational disadvantages.

War experience showed that the endurance at full power was seldom of such operational importance as endurance at cruising powers, and the design aim was changed accordingly. Maximum efficiency in the 20% to 25% power range is now the design target.

The design point. A definition of this term is " that percentage of full power at which the adiabatic efficiency of the turbine is at its maximum, while all the steam supplied to the turbine is passing all the turbine stages," *i.e.*, the highest efficiency with no by-passing taking place.

In order to achieve the highest economy at 20-25% full load it would be desirable to bring the design point down to this region ; but so far in naval designs it has not been found possible to get the design point below about 60%. Whilst stress is no longer laid upon obtaining the highest efficiency at full power, too steep a fall in the turbine efficiency curve cannot be tolerated : not only for reasons of reduced full power endurance, but also because as the turbine full power efficiency falls more steam is required to obtain full power ; and this increase is not only reflected in increased sizes of boilers and steam pipes, feed pumps and other auxiliaries, but also in the turbine itself ; the last stages have still to be designed to pass the full quantity of steam.

There are various ways of lowering the design point, and these will be dealt with later in these notes ; meanwhile before proceeding there are two other matters to be dealt with which intimately affect the turbine design.

FULL-POWER EXHAUST PRESSURE

One of the heaviest and bulkiest items of the whole machinery plant is the condenser, size and weight and other things being equal, the higher the designed full power vacuum, the larger and heavier the condenser.

The effect on the turbine of changes in designed full-power vacuum are considerable ; the volume of saturated steam at 29 inches vacuum is about 3·7 times that of steam at 26 inches vacuum.

The effect of a change from former to the latter upon the dimensions of the later L.P. stages is very considerable even allowing for the greater quantity of steam to be passed for the required power at the lower vacuum.

It is by no means a simple matter to decide what is the optimum vacuum for any particular service ; some of the points to be taken into consideration are :—

(*a*) Reduced vacuum means a lower efficiency, and therefore, more steam (at given initial conditions) for a given power. Consequently, low vacuum means larger boilers, steam pipes and auxiliaries, but smaller turbines and condensers (the latter in spite of the increased heat content of the exhaust steam).

(*b*) Lower vacuum allows a smaller annulus area for the last L.P. stages, consequently shorter blades. Blade loading in the last L.P. rows is frequently the limiting factor upon L.P. turbine blade speed. In present working ranges, the higher the blade speed, the better will efficiency be maintained at reduced powers.

Alternatively, the reduced loading and stage diameter may allow the addition of a further stage, or stages, adding to the efficiency particularly at partial loads.

(*c*) A lower vacuum will enable the final stages to be designed so that the leaving velocity of the steam can be reduced, thereby lowering the leaving loss. In a modern destroyer turbine designed for 27·5 in vacuum at full power the leaving loss may amount to perhaps 16 B.T.U.s/lb. about 5% of the useful work obtained from the steam by the turbine. In the same set with the vacuum lowered to 26 in, and the same power output, the above loss will be about halved.

(*d*) The higher the initial steam conditions, the higher the overall heat drop, and the lower the percentage effect upon the adiabatic heat drop of a reduced vacuum.

(*e*) The reductions in rotor diameter and length mentioned above allow a given turbine to be designed with (i) greater resistance to shock loading and (ii) a more compact casing with less risk of distortion and consequently permitting smaller clearances and reduced losses.

(*f*) In a set designed for high vacuum the full capacity of the condenser is employed only at full power—at lower powers a portion of its area is not used. If, however, a lower vacuum is accepted at full power, high vacuum will still be available at reduced powers, and the smaller quantity of steam passing at these powers will effectively fill the blade passages.

(*g*) A set designed for high vacuum can only achieve this vacuum if sea-water temperature is low enough to permit it. Thus a set designed for 29 ins vacuum cannot attain it if the sea-water temperature exceeds about 75°F ; but there is no theoretical reason why a set designed for 26 ins vacuum should not attain this with any sea-water temperature likely to be encountered.

(*h*) The lower the vacuum the higher should be the condensate temperature, and therefore the less the heat which must be added to attain a given feed temperature (29 in vacuum corresponds to 79°F, 26 in to 125°F). This gain is only real, however, if the whole installation is skillfully designed so that additional sources of feed heating (*e.g.*, closed exhaust) are not redundant.

(*i*) The higher the vacuum the greater the quantity of unwanted air which will enter the system and have to be extracted ; and also the lower the vacuum the less the volume of a unit quantity of the air to be extracted and the more economic is its extraction in terms of ejector steam.

(*j*) For given initial steam conditions, the lower the vacuum the better will be the final quality (*i.e.*, dryness) of the steam ; and consequently the higher the mean efficiency of the whole turbine.

All these and other factors must be examined and assessed before a final decision is made.

It will be noted that nearly all the above arguments seem to be against the employment of the highest full-power vacuums. The fact remains, however, that for a turbine in which maximum efficiency at full power is the predominating requirement the highest possible vacuum must be used.

GEARING

For a given hull and performance the highest speed at which a propeller may be run is fairly clearly defined, and if this speed is exceeded with present propeller design a sharp drop in propulsive efficiency is to be expected. The engine builder always wishes for the highest shaft revolutions, but if the optimum revolutions of the propeller are far exceeded the price paid is very heavy ; the shaft revolutions are thus readily decided.

With " modern " steam conditions—say 650° p.s.i. and 850°F—naval requirements for turbine designs lead inevitably to rotor revolutions of such an order that double-reduction gears are essential. Once committed to double reduction, the weight and space of the gear units is affected only slightly by small changes in reduction ratio. For a given power the higher the ratio, in general, the greater the weight and bulk of the gearing ; but within the probable limits of compromise the variations are unlikely to be significant and consequently the designer of a turbine set with D.R. gearing can choose his turbine revolutions with negligible limitations or necessity to compromise as far as the gearing or propeller are concerned.

It will be understood that these remarks apply to what may now be considered "conventional" designs ; the case requires and receives re-examination for novel layouts.

The loading of hobbed, shaved and lapped gears is rising and in consequence the weight and the bulk of gearing are being reduced. The expected introduction of types of gearing with a considerable step-up in loading will further reduce these factors, and this in consequence will render the present small differences associated with variations in turbine-designed revolutions smaller still. In the future, therefore, even more than at present, the turbine designer can choose his revolutions unhampered by gearing considerations.

In general the best way the turbine designer can help the gear designer is by a suitable choice of the distribution of power between the cylinders.

Secondary effects only are involved and the issue may be influenced by the desirability of having certain parts identical for both H.P. and L.P. trains ; but in general a well-balanced set of double reduction gearing will be secured if the power distribution between the H.P. and L.P. turbine is inversely proportional to the square root of the full-power revolutions.

Such a distribution is based upon minimum departures from selected P/D loading values throughout the set of gearing, and is usually attainable by the turbine designer without any attendant disadvantages.

A turbine design which to allow suitable spacing between the cylinders or for any other reason involves the introduction of idler wheels is clearly a disadvantage and would require sound reasons to justify it.

IMPULSE OR REACTION ?

Earlier teachings were sometimes to the effect that pure reaction turbines were only exemplified by the Lungström design : the implication was that other reaction turbines had some degree of impulse effect, and this is correct. But it was seldom pointed out that impulse turbines normally have also a degree of reaction ; and in modern sets this is often deliberately designed into the turbine. An "all-impulse" (H.P. and L.P.) naval set is likely to carry an increasing degree of reaction through the L.P. stages, until the last rows may in effect be reaction stages, although the mechanical construction will probably follow the recognised impulse technique of the earlier stages of the turbine.

The division between impulse and reaction turbines is therefore not always quite so hard and fast as is often thought, particularly in L.P. turbines. It is, however, quite sufficiently definite to maintain fundamental differences between the two types in methods of construction.

Fundamental considerations indicate that the maximum efficiency of an impulse stage is achieved when the velocity ratio is about 0·45 : for a reaction stage the corresponding value is about 0·8.

If stages of the two types are compared, each having the same blade speed (it being supposed that this is the maximum the design permits) it is clear that for maximum efficiency the steam inlet speed in the impulse stage may be nearly double that in the reaction stage, and the heat drop in the fixed blades or nozzles will therefore be about four times that in each reaction blade. Because little expansion normally takes place in a moving impulse blade, while it does in both moving and fixed reaction blades, the net result is that an impulse stage can take about twice the heat drop that a reaction stage can ; and for this reason alone the impulse turbine can be designed with about half the number of stages that would be needed for the reaction type.

The reduction in the number of stages allows the rotor to be made shorter, with a consequent rise in the whirling speed. A higher blade speed may therefore be employed and a still greater heat drop taken per stage while maintaining the desired velocity ratio ; this again helps to reduce the number of stages.

In a specific instance two naval turbines have been designed to fulfil precisely the same functions and operate under the same conditions with the same efficiency ; they have stages as follows :—

Reaction ... ... Curtis wheel + 22 reaction pairs.

Impulse ... ... Curtis wheel + 8 stages.

The axial length of a Rateau stage is, of course, greater than that of a reaction pair ; nevertheless there is a considerable saving in length of the impulse rotor.

The shortening of the rotor allows an improved resistance to shock and thermal distortion, and the values of the differential expansions between the rotor and casing are also somewhat less, allowing smaller clearances to be used, with consequential reduction of losses.

The rise in steam condition has brought into prominence the evils of casing distortions, and efforts have been made to understand where these distortions arise, and how they may be minimised. This is a matter to be dealt with more fully in these notes when casings are considered in detail ; here it is sufficient to say that in naval turbines distortion will always take place, and there are four ways to avoid serious damage :—

(1) To design the casings in such a way that the distortions are a minimum.

(2) Reduce the number of fine clearances as far as possible.

(3) To arrange the position of the fine clearances so that they are reduced as little as possible by distortions that are likely to occur, and

(4) Open out fine clearances as necessary to avoid fouling—accepting the losses of efficiency entailed.

Once the turbine is built (4) is the only method open to us, and it is unfortunate that it is one to which resort has sometimes had to be made.

Regarding (1), the casing of an impulse turbine is likely to have less tendency to distort than that of the relatively longer reaction one ; but there is often not very much in this point.

Concerning (2), the reduction of the number of stages to a half or a third clearly gives the impulse turbine an advantage, particularly as for this type there is only one set of clearances per stage as opposed to two sets for an end-tightened reaction pair. The clearances are also radial and unaffected by axial differential expansion of rotor and casing.

Most important for the impulse case, however, is (3). The fine clearances are at the centres of the diaphragms, on comparatively small diameters, where not only may the relative movements be expected to be smaller, but also, if contact occurs, the rubbing speeds are lower, and contact areas smaller ; whilst the smaller diameters permit more generous clearances without large losses.

The most dangerous place for a rub to occur is almost certainly in a dummy cylinder. Unfortunately the location of a dummy cylinder at the hot high pressure end of the turbine is such that casing distortions are almost unavoidably communicated to it ; while the changing temperatures to which it is subject during changes of power, together with its high peripheral speed, and limited capacity for dissipating heat all make it a device to be viewed with foreboding in advanced naval designs, and a standing argument against reaction turbines.

In the past impulse turbines have not proved satisfactory in naval service ; they have suffered from various defects, some of which are discussed below ; the majority of the points will also be discussed more fully under various other headings.

Loosening of shrunk-on discs

This has been a recurrent trouble due in large part to lack of knowledge regarding allowable disc stresses and thermal effects not only at full power but also when warming through and changing powers (with the consequent opening of by-passes, etc.). In modern naval designs the discs are solid with the rotors so that loosening cannot occur. It is necessary, however, to guard against over-stressing the material due to thermal effects, particularly at the point where the discs join the hub or spindle ; sudden changes in steam temperature may be closely followed by the disc which has a large area for its mass ; the reverse, of course, holds for the spindle. Much research on this matter is in hand.

Blade Vibration

The existence of this ill as a real source of turbine breakdown was quite unappreciated until long after the completion of the majority of the impulse turbines in the Navy designed prior to World War II.

Although our knowledge on this matter is far from complete, it became sufficient during World War II to enable every failure from this cause to be corrected in such a way that it did not recur, and reasonable confidence can now be felt that no widespread troubles from this source should again be experienced.

Wheel Vibration

In the days of the older impulse turbines this matter was not understood. Since then methods of determining the natural frequency of wheels, both practically and by calculation, have been evolved, and here again confidence that this trouble can be avoided should not be misplaced.

The other main causes of turbine unreliability apply either only to reaction types (*e.g.*, dummy fouling) or to both impulse and reaction ; whilst such trouble as blade fouling due to differential expansions* or casing distortion can occur in the impulse type, it is clear that the reaction type is far more subject to them for the reasons given elsewhere.

Although practically all ships of the British Fleet have at present either all-reaction or impulse-reaction turbines, the introduction of more advanced steam conditions have caused a review of policy and a swing to an all-impulse H.P. turbine in association with either a reaction or impulse L.P. The lower temperatures in the L.P. turbine appear at first sight to permit a free choice between the two types, but the greater length and diameter, the proximity of the condenser, and the incorporation of astern turbines all tend to cause large distortions in the L.P. so that in reality much the same arguments against the reaction type apply as for the H.P. turbine.

The two types may be expected to have about the same overall efficiency, the higher theoretical stage efficiency of the reaction type being offset to some

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* This term is normally used in these notes to mean axial differential expansion between casings and rotor.

extent by its greater inherent parasitic losses ; a constant speed turbine could be designed in the knowledge that it would always be " nursed," would have a better chance of proving the reaction the more efficient type ; but for a naval turbine where reliability under difficult conditions is of paramount importance the larger clearances necessary to achieve robustness in a reaction turbine would almost certainly destroy its apparent advantage.

In spite then of the overall satisfactory performance in the service of reaction turbines at the lower steam conditions, the weight of opinion now tends towards the impulse type, especially for the high-pressure cylinder.

THE CURTIS WHEEL

It is at first sight surprising that so many designs of naval turbines embody a Curtis wheel* as the first stage. Close examination of the alternatives, however, show good reasons for this choice. Research is in hand to examine these alternatives ; but at present it remains to be proved that the incorporation of a Curtis wheel is not the best way to meet naval requirements in a straight-forward two- or three-cylinder design.

Various fundamental facts contribute to the choice of the Curtis wheel for a first stage. In older turbines by-passes were sometimes fitted in such a way that the number and types of nozzles which were used in stages after the first could be controlled, *e.g.*, the " series-parallel " arrangement in Brown Curtis turbines ; these were not always satisfactory either from the point of view of giving good flow conditions or mechanical reliability ; and with advanced steam conditions the reliability would decrease while the interference with the symmetry and compactness of the casing (very necessary to give freedom from distortion) would be tolerated by few designers.

The typical modern naval turbine does not therefore have any way of controlling live steam after it has passed into the first stage ; the subsequent stages must be designed with sufficient areas to meet the maximum flow which has to pass them under any conditions ; at reduced powers these full areas will be available for the reduced steam flow, and expansion will consequently take place to a greater extent in the earlier stages of the turbine. That this is so is shown by the fact that the wheel-case (*i.e.*, the first stage) pressure is approximately proportional to the power output of the turbine.

At low power, therefore, a large proportion of the pressure drop (and consequently a large proportion of the available heat drop) unavoidably takes place across the first-stage nozzles, resulting in very high steam speeds. This, combined with the reduced blade speed produces a most unfavourable velocity ratio. The peak of the stage efficiency curve for a Curtis wheel occurs at a velocity ratio of 0·22 ; that for a Rateau† stage at about 0·45, and under the lower partial load conditions the velocity ratio is likely to be even less than the former value. With a velocity ratio of 0·2 in each case the stage efficiency of a Curtis wheel is perhaps 10% to 12% better than that of a Rateau wheel ; and on a basis of equal blade speeds the former will take about four times the heat drop that the latter can.

This is a very significant effect ; in a modern design, at 10% power one half of the available heat drop takes place in the first-stage nozzles ; at 20% power one-third. It is clear that the efficiency of this stage under these conditions

* Curtis wheel—A 2 row velocity compounded impulse stage.

† Single row impulse stage.

is of great importance. Inefficiency will give rise to re-heating and although some of the lost energy may be regained later in the turbine, the inefficiency of the first stage will have given rise to an increase in entropy and only a fraction of the loss will be recoverable even under the best conditions.

Throttling the boiler steam must be avoided as much as possible to prevent increases in entropy and the inefficiency attendant upon the consequent reduced availability of the energy ; and at partial loads nozzling is therefore necessary, the nozzles being kept supplied with steam at high pressure.

Although under reduced power conditions, the velocity ratio of a first stage is very low (say, perhaps 0·2), it will be considerably raised at higher powers, the reasons for this being the converse of those given above to explain the low velocity ratio at low powers, *i.e.*, firstly the blade speeds will be higher ; and secondly the stage pressure will rise (in part due to the building up of pressure throughout the turbine as the flow increases ; and in part because in most naval designs by-passes are fitted which by admitting steam to subsequent stages " back-up " the pressure in the earlier ones).

This low pressure drop in the first-stage nozzles permits only a low heat drop and lower steam velocities. The velocity ratio under such conditions will be so high that the stage efficiency of the Curtis wheel will be well past its peak ; and indeed little if any useful work will be expected from it at full power. In this condition it is said to be " over-speeded ". The major portion of the steam will be by-passed round it, and the backing-up of the first-stage will reduce the flow through it until it becomes a very low proportion of the whole. Under these conditions the design must ensure that the steam passing is sufficient to prevent overheating, as damage will occur if the flow becomes too small or ceases ; but in a highly over-speeded condition the first-stage exhaust steam may well be hotter than at entry. Such a condition must give rise to inefficiency ; but in an external by-pass turbine only a small part of the total steam is effected, and as at full power the maximum efficiency is of less significance than at cruising powers, this is relatively unimportant.

What happens to the steam thermo-dynamically in a highly over-speeded Curtis wheel is at present a matter for conjecture and research.

Practically, the Curtis wheel has also much to recommend it ; again on account of the large heat drop it can employ it replaces four Rateau stages, or eight reaction pairs ; the saving in rotor length thus made will permit either of adding one or more extra stages, or of raising the speed of rotation whilst maintaining the same margin below the whirling speed.

The Curtis wheel is also not infrequently fitted in designs in which the partial load characteristics are not important ; and also in land turbines where the constant speed maintained at all loads weakens the arguments based on matters of velocity ratio. Here the reason may be that of saving rotor length ; but another strong reason exists ; that is, that the large heat drop causes a considerable fall in both the temperature and pressure in the first stage. In a modern design a limiting factor may well be the lowering of the physical properties of the rotor material under high temperature ; and the drop associated with the Curtis wheel nozzles will in effect allow the cycle to employ a much higher temperature than could be the case if steam at the full temperature reached the rotor. This may well be a vital factor in determining the upper limit of temperature of the cycle.

The lower temperature on the rotor also reduces difficulties with glands ; but not so much as does the reduced pressure, which enables fewer restrictions to be employed, shortening the gland and the rotor as a whole ; these facts apply equally to naval and land designs.

Unfortunately, in naval designs with heavy by-passing, as shown above, the wheel-case temperature and pressure will rise at full power, and the advantages just described will be lost under this condition. The type of by-passing affects this issue, allowing more advantage to be taken of the initial fall in temperature in some cases than in others.

In any naval example where creep of the rotor is a limiting factor, this aspect of the matter again becomes of importance as the highest temperature on the rotor will be confined to the time the by-passes are open, which may be a very small fraction of the total life of the machinery.