SOME CONSIDERATIONS AFFECTING DESIGN, CONSTRUCTION AND WORKING OF LARGE STEAM TURBINES.

In a paper dealing mainly with large turbines for land power stations read recently at the Institution of Electrical Engineers,* instances of serious breakdowns in some of these units were enumerated, and a general consideration of the failures serves to bring into prominence some of the important difficulties which have to be surmounted in the design and construction of large turbines. Although in principle the design of turbines for land and marine use follows the same general lines, there are certain essential features which in the respective cases have a modifying effect and call for special consideration. These may be understood if the main requirements for such turbines are briefly discussed.

In the land turbine the questions of weight and space arise only so far as they may be concerned in initial cost, manufacturing, and transport arrangements, and full scope may usually be given to the attainment of the maximum possible efficiency. In the general case the economic consideration, with which is always associated reliability, is of paramount importance in shore installations. On the other hand, weight and space have an important bearing on the design of turbines for marine and particularly for naval use, and the economic consideration may have to give way somewhat to these other factors. Prior to the introduction of gearing or other forms of transmission, the necessary requirements of the propeller, essentially a low-speed unit, also exerted a most important effect on the marine designs. The corresponding effect is not experienced to any great extent in land turbines, which are constructed almost entirely to drive dynamos, essentially high-speed units. With the adoption, however, of an intermediate form of transmission between turbines and propeller shaft, the turbines are to a large extent designed independently of the propeller requirements, and with such designs, except as affected by the weight and space features, land practice is more nearly approached.

As a general principle it may be stated that the main difficulties arise with land turbines in the attempts to attain the high efficiencies necessary. It must be remembered that in some cases enormous powers are required and the machinery for a considerable portion of its time is working at high output, and in any case individual machines are generally worked at a high-load factor. On the other hand, in the marine, and more particularly the naval designs, the weight and space features, seeing also that cruising and astern arrangements are necessary, are liable to lead to the greatest difficulties. How these features affect the design will be seen in the following, which it must be pointed

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* Some recent developments in large steam turbine practice.—K. Baumann.

out is not intended to describe how large turbines are designed, but only to indicate in their very broad outlines some leading features involved, where the main difficulties arise, and the methods of overcoming these difficulties.

In any steam turbine a very high thermodynamic efficiency can only be obtained by working between as wide a temperature range as practical considerations permit, *i.e.*, by the use of superheated steam and a high degree of vacuum. Working between such limits will give a large heat drop per pound of steam, and for a definite number of stages in the turbine this will result in a higher velocity of steam at the different stages than would obtain with a turbine using saturated steam and a lower vacuum. As the blade speed for best efficiency should bear some relation to the steam velocity (the exact relation, which in general varies from ·5 to unity, depending on the type of turbine) high-blade speeds are necessary in such designs unless a very large number of stages are fitted. Fitting a large number of stages, however, has attendant disadvantages. The length of the turbine is increased involving the construction of stiffer and heavier rotors to resist deflection and whirling and entailing additional leakage losses at blade tips or diaphragm glands, and the gland mouths. In addition there is a loss of efficiency owing to the additional shock and friction losses, involved in the longer steam path to be traversed and increased windage losses. The same difficulties regarding leakage, stiffness and weight of spindle also arise if the high-blade speeds desired are obtained by an increase in mean diameter with the number of stages kept down.

It will be appreciated therefore that some compromises in this respect based on experience have to be effected.

One important compromise is to fit a compound system, *i.e.*, divide the turbines into two units H.P. and L.P., but the same general considerations, although considerably reduced in complexity, arise in dealing with these units separately.

The division of power into two units is necessary in any case in high-powered geared installations in order to obtain a practicable design of gearing, but the division of power is also desirable for the sake of the efficiency and reliability of the turbines, the simplification of construction and, possibly, a better utilisation of the available space.

A stage in the design is therefore reached when a certain number of stages and blade speed are provisionally decided upon.

In naval practice with twin H.P. and L.P. units fitted in connection with gearing, the use of saturated steam has enabled a sufficient number of stages to be fitted without involving the necessity of exceeding a maximum blade speed, at the mean diameter, of 500 f.s. in large designs, whereas with superheat land practice mean blade speeds exceeding 750 f.s. have been reached in special cases. This question of blade speed has a determining influence on the design in the last stages of the L.P. portion of the turbine. If the high vacuum is to be effectively utilised the steam in the last nozzles and last row of blading must fall to a pressure only slightly above the condenser pressure. With a 28 inch vacuum in this last row, the specific volume of dry steam stands at the large figure of 333 cu. ft. per lb. and with the use of a high degree of initial superheat, coupled with the reheating effect on the steam due to leakage, shock and friction losses in its passage through the turbine, it will be about 90 per cent. dry in this position, *i.e.*, possess a volume of 280-290 cu. ft. per lb. A very considerable area of flow will therefore be necessary in a large output machine if the steam velocity is still to bear its correct relation to the blade velocity at this stage.

This steam velocity has been provisionally fixed by the number of stages and it is inadvisable to increase it at this last stage by allowing a greater pressure drop through the final nozzles, as, apart from a loss of efficiency due to its incorrect relation with the blade speed, the high resulting leaving velocity will entail additional energy being carried over to the exhaust. The requisite area of steam flow is governed in the main by the angling of the nozzles and blading and by the length of the blades. Wider spacing of the individual blades has a detrimental effect on the efficiency due to the possibility of a direct flow through of the steam and a loss of impulsive or reaction effect. The blade length cannot be increased inordinately, as it will give too great a circumferential spacing at the tips, and it is therefore unusual, except in some very special cases, to make the length of blades more than 20 per cent. of the mean diameter at which they are set. The important factor to be considered, however, as regards long lengths is the question of centrifugal stress on the blades. This stress on a blade of uniform section will vary directly as the length of the blade, directly as the square of the blade velocity and inversely as the mean diameter of the blade annulus; or as it is sometimes stated, directly as the square of the blade velocity and the blade "ratio" this latter figure being the ratio of the height of the blade to the mean diameter. If the blade has a reduced root section, as for example where it is dove-tailed into the rim of the wheel, the stress will be proportionately increased at this section, as the support of the packing or distance pieces is indefinite and is usually ignored in stress calculations. The bending stress on the blades due to the action of the steam must also be taken into consideration, and it is on the satisfactory compromises effected in these last stage particulars that the correct relation between mean diameter and revolutions of the turbine or of the L.P. unit of a compound system, is mainly determined.

With the speeds reached in naval practice the necessary compromises have been possible and suitable stresses obtained as permit of the use of phosphor bronze blading, which material possesses a high resisting power to erosion and has practically no tendency to corrode.

With superheat, however, and with the high stresses pertaining in the large land units, special alloy steels containing nickel or nickel chromium are used to a large extent, and these steels, although resisting erosion if the steam is comparatively dry, are generally affected by corrosion, especially if CO_2 and oxygen are present in the steam. The use of stainless steel for turbine blades is referred to in an article in No. 3. of these Papers.

For very large powers, it is found impossible always to co-relate successfully the various factors outlined above, and the last stage difficulty has to be overcome in other ways, and it will be interesting to describe briefly some of these methods. Even with these modifications, the last stage considerations still have the most important effect on the main features of the design for the large output machines.

(1) The L.P. Unit is constructed as a double-flow turbine the steam entering at the centre and flowing through what amounts to two turbines mounted on the same spindle, the exhaust at each end being into a common chamber connected to the condenser. The blade heights through the L.P. Turbine are therefore one-half those that would obtain in a single-flow turbine of the same number of stages. In re-action machines where this arrangement is frequently adopted this modification has the advantage of eliminating axial thrust, and the necessities of a dummy piston with its attendant leakage. The tip leakage losses are, however, increased as there are now two sets of clearances instead of the one, and these clearances depend not on the blade heights but on the diameter over the blades. It leads also to a longer and heavier turbine involving additional consideration from the point of view of stiffness of the rotor to keep it within its critical speed of revolution. This modification is met with in naval practice in the Parsons all-geared arrangements.

(2) A single flow is fitted for all stages except the last, in which two equal rows of blades are fitted, but with steam flowing in opposite directions through these two rows respectively. Certain difficulties in construction and the arrangement of a free exhaust are involved with this method.

(3) In some very large units of 60,000 K.W. (*i.e.* 80,000 H.P.) which have been constructed in America, the arrangement is a triple-cross-compound, the H.P. turbine exhausting to each of the two L.P. turbines, each of which is of the double-flow type, thus giving in effect four final stages.

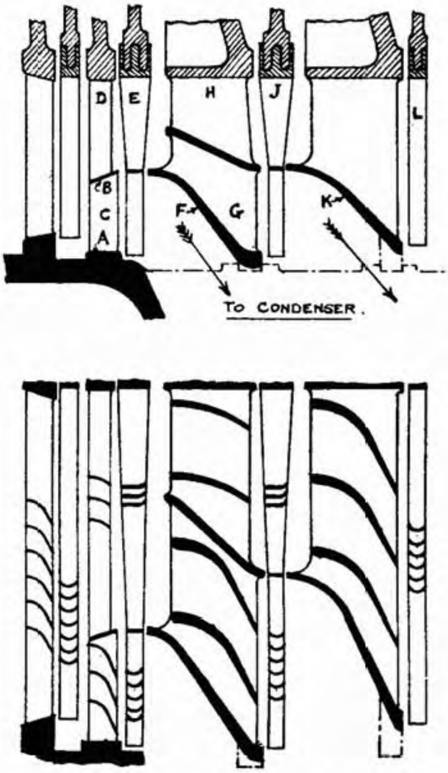
(4) A British Company has introduced what is known as a multi-exhaust turbine for overcoming the difficulty, and this, although not fitted in the naval service, is worthy of detailed description. The steam passes through the H.P. end of the turbine in the usual way until it reaches the special diaphragm A Fig. 1. At this point the steam is divided into two parts by the annular division ring B in the diaphragm. Passing through the outer annulus C, the steam is expanded to the condenser pressure,

the shape of the diaphragm blades and that of the co-operating row of moving blades at this part being suitably proportioned. In the inner annulus D, the steam passes without appreciable expansion through suitably formed guide blades. Opposite D is the inner portion of the row of moving blades E shaped in such a way that the stresses due to the centrifugal force are within the limits fixed by the material, the passages left between the blades being such as to pass the maximum quantity of steam with the smallest drop in pressure. The steam then reaches the diaphragm F which also has two distinct annular parts, the outer part G being suitably shaped to expand the steam to condenser pressure, and the inner part H to pass the steam on without appreciable expansion, through suitably shaped moving blades J to another diaphragm K, where the steam is expanded finally to condenser pressure over the full lengths of the blades L. Due to the difference in pressure existing in the inner and outer annular passages, a small leakage occurs through the clearance spaces between the guide blades and the corresponding moving blades. This leakage is of a minor order having regard to the large volume of steam dealt with at this part of the turbine. With this construction it is claimed that it is possible to increase the leaving area of the final stage in the ratio of 1 to 1·6 with one additional row of exhaust blades, in the ratio of 1 to 2·2 with two additional rows and in the ratio of 1 to 2·7 with three additional rows. The output of the machine can therefore be increased over a considerable range, and without the use of the two-cylinder construction. It is also claimed with justification that with this method of construction it is possible to construct different multi-exhaust turbines for relative exhaust capacities of 1, 1·6, 2·2, and 2·7 on the same mean diameter and with the same length and type of blades, thus utilising the same patterns for a considerable range of output, with a special distance pattern for the additional stage and length required. The smaller powered turbines will work with partial admission at the earlier stages, and the larger turbines with full peripheral admission.

The objections to the foregoing are the additional complexities surrounding the special diaphragms and also the non-uniform blades and nozzles required in the final stages.

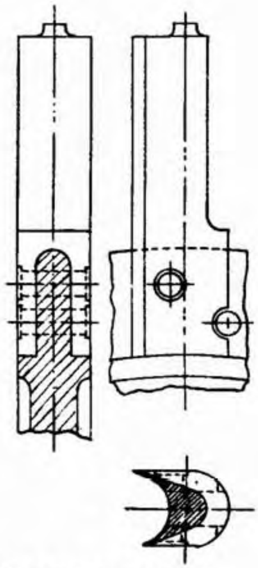
It might also be pointed out in connection with the last stage problem that it is to a certain extent simplified in some large stations by arranging to tap off a portion of the steam at one or more stages towards the L.P. end and utilise the steam so tapped off for feed heating, a modification that secures a slight gain in overall thermal efficiency besides reducing the amount of steam to be passed through the last stages and exhausted to the condenser.

Returning again to the question of blade stresses. Such stresses are a minimum at the tip of the blades and in a uniform blade gradually increase to a maximum at the end where the blades adjoin the wheel or drum. With dovetailed blading, this



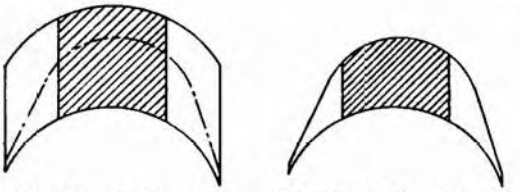
MULTIPLE EXHAUST.

FIGURE 1.



RATEAU STRADDLED BLADE.

FIGURE 3.



COMBINED SECTION, 48C.
AREA AT ROOT = ·2475 □"

ORDINARY SECTION, 48.
AREA OF BLADE = ·24 □"
AREA AT ROOT = ·162 □"

BROWN CURTIS BLADE SECTIONS.

FIGURE 2.

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stress is proportionately increased at the root section, due mainly to the reduction of area and possible existence of sharp corners in the dovetailing, which latter, correct design and machining avoids to some extent.

In the highly stressed cases such reduction is obviously undesirable. By combining the blade with the packing or distance piece, it is possible with dove-tailed blading to maintain a section at the reduced position at least as large as the section of the blade itself (Fig. 2). This is a usual practice in the last rows of blading in large Brown-Curtis turbines and in view of the advantages generally of such stronger blading, also allowing for the fact that it is less liable to be strained during the initial operation of blading the turbine, is now specified for all future naval work. Another method of attachment is to rivet the blading to the rim of the wheel, the design being such that the area at the minimum section is at least equal to that at the root end of the blade (Fig. 3). Apart from such methods it is arranged in extreme cases of land practice to increase the section of the blades gradually from a point on the blades to the root either by making them thicker in the circumferential direction or wider in the axial direction. Such methods have some effect on the steam flow but any such effects must be accepted at the expense of obtaining the necessary strength and therefore the main consideration, reliability, for without this latter feature all striving for efficiency is valueless as the gain in efficiency may be easily discounted by the repair bill and economic losses involved in the breakdown and the time such a unit is out of operation.

There is now sufficient knowledge and experience available to enable the working stresses on the various parts of a turbine to be at least approximately determined although certain details, particularly the diaphragms, drums and wheels present great difficulties in this respect. The greatest analytical difficulties, however, do not arise directly in connection with the ordinary working stresses, but in the tendency of the various parts to vibrate and a number of important failures that have occurred in power-plant practice have been attributed to failures of details under the effect of vibratory stresses that have been superimposed on the ordinary design load stresses. Every turbine, as a whole, has a series of critical speeds of revolution at which speeds and for a range either side of these speeds, dangerous vibrations may be set up in the rotor. These speeds synchronise with the frequencies of vibration of the rotor considered as a loaded beam. If the time of one revolution corresponds with a time of a complete oscillation of the rotor considered as a beam, then an error in the balance, which it is never possible to eliminate completely, may cause these dangerous vibrations to occur. This point can be fairly easily dealt with in the design stage, and in all large units it is usual to arrange for the rotor to be of sufficient stiffness to possess a lowest critical speed which is at least 30 per cent. above the maximum running speed.

In small high speed turbines of the De Laval type this is not the case, the shafting being purposely made very flexible, and the running speed is arranged to be considerably higher than the lowest (which is the most important) critical speed and which is passed through rapidly on speeding up or shutting down. Such a procedure is not permissible with large turbines owing to the risks involved in passing though the critical range, the speeding up or shutting down being necessarily slower and thereby involving the turbine running at a dangerous speed for an appreciable time. Obviously the procedure is also quite inadmissible for any variable speed turbines as are necessary for marine work and which may have to run for periods at any speed up to full speed.

The greatest differences between land and marine practice arise from the necessity in the latter cases of providing for astern running and, in the case of warships, for working economically at reduced powers. In vessels fitted with electric or hydraulic gearing between turbines and propeller shafts, the provisions for astern working are incorporated in the motors or transformers respectively, the turbines working in one direction only.

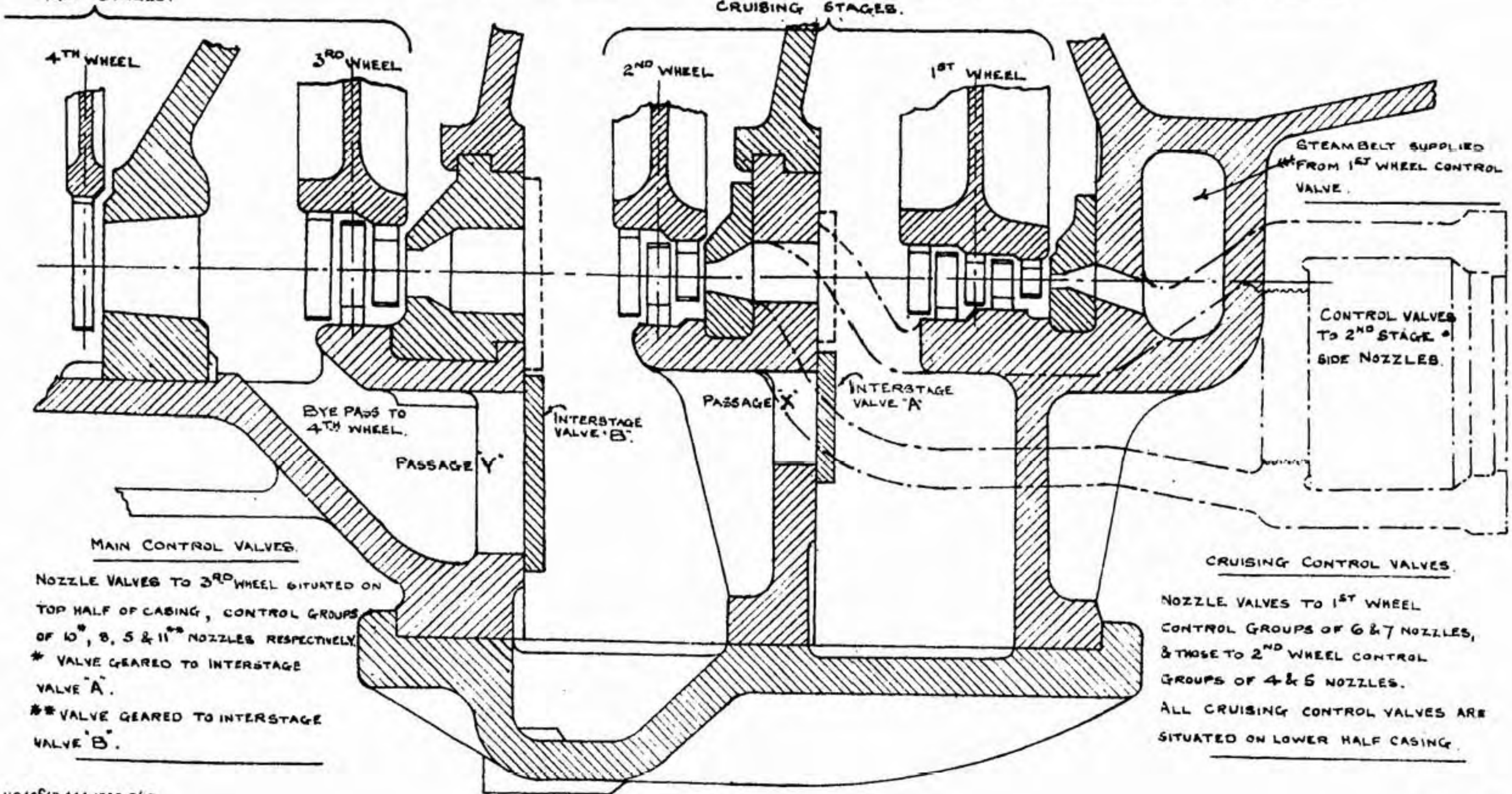
In the case of mechanically geared or direct-drive installations an astern turbine is, however, necessary and confining our attention to recent British all-geared practice it is usual to arrange in capital ships for a maximum astern power of about one-half and in T.B.D.'s of about one-third of the full power ahead. This power is actually much larger than that normally required but it ensures that the efficiency of the astern turbines is not unduly low and thus provides for a reasonable stopping power and astern speed being available when working with a proportion only of boilers in use. The maximum power required in current designs can be obtained on a single pressure stage velocity-compounded wheel.

It should be borne in mind that provision for astern working entails a loss equivalent to the work of driving the idle turbine. The astern turbine is now always incorporated in the same casing as the L.P. turbine, an advantage from space and other considerations, but this involves an increase in weight and a larger and stiffer rotor spindle is required which entails a possible increase in the leakage losses at the gland mouths and in the diaphragm glands of the ahead portion.

In warships, efficiency at low powers is of considerable importance. The falling off in economy resulting from the reduced velocity ratio of the turbines at low powers is compensated for, in a measure, either by the use of special cruising turbines or by cruising stages incorporated in the main units. The additional stages which are brought into use at low powers ensure a reduction in the heat drop per stage and consequently a lower stage steam velocity. This enables a higher velocity ratio between blade and steam being maintained with consequently a better efficiency than would otherwise be the case.

TWIN SCREW GEARED T. B. D. — 27000 S.H.P.

BROWN CURTIS H.P. TURBINE WITH TWO CRUISING STAGES. FIGURE 4.



MAIN CONTROL VALVES.

NOZZLE VALVES TO 3RD WHEEL SITUATED ON TOP HALF OF CASING, CONTROL GROUPS OF 10*, 8, 5 & 11** NOZZLES RESPECTIVELY.

* VALVE GEARED TO INTERSTAGE VALVE "A".

** VALVE GEARED TO INTERSTAGE VALVE "B".

CRUISING CONTROL VALVES.

NOZZLE VALVES TO 1ST WHEEL CONTROL GROUPS OF 6 & 7 NOZZLES, & THOSE TO 2ND WHEEL CONTROL GROUPS OF 4 & 5 NOZZLES.

ALL CRUISING CONTROL VALVES ARE SITUATED ON LOWER HALF CASING.

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The incorporation in the main turbines is now generally preferred although the additional wheels necessary must then always rotate at the high powers in comparatively dense steam and thus entail additional frictional losses. The provision of a stiffer rotor is also necessary and this has its effect in increasing diaphragm and gland losses. At the same time there is a net saving in space, weight and complication as compared with the arrangement of separate cruising turbines. If full advantage is to be taken of the latter arrangement, de-clutching fittings are necessary between the cruising and main turbines to eliminate the loss entailed in driving the cruising turbines at high powers. On returning to low power conditions the main engines must be stopped to re-engage the cruising turbines, and this disadvantage, under war and fleet steaming conditions, does not enable full use to be made of the cruising arrangements.

With the present design of Brown-Curtis turbines the cruising wheels incorporated in the main turbines are not bye-passed to be inoperative at the high powers but are, at all powers above those obtainable under the various cruising conditions, worked at a high capacity. The general arrangement of the early stages of such turbines with two cruising wheels, as fitted in the later T.B.D's is shown in Fig. 4.

The conditions of working are as follows :—

(1) At very low powers the control valves to the first stage on bottom casing of the turbine are open, the steam supplied passing through, in series, all nozzles of the first stage, bottom nozzles of the second and third stages, and those nozzles of the fourth stage *not* supplied by the bye-pass.

(2) At low powers but exceeding those obtained in (1) the steam supplied to the first stage nozzles pursues the same course but is now supplemented with additional steam supplied by control valves on the bottom casing direct into the second stage chamber and pass together through the bottom nozzles of the third stage and those nozzles of the fourth stage *not* supplied by the bye-pass.

(1) and (2) supply steam for all powers up to about 25 per cent. of the full power and approximately 15 per cent. can be obtained with (1).

For powers above 25 per cent. additional steam is obtained by opening the control valves to the third wheel on the top casing and, with the opening out of the first of these top half valves, the slide valve " A " is moved upwards, from the position covering the bye-pass " X," to a position covering the second stage bottom nozzles and thereby opening the bye-pass " X " from the first stage into the third stage.

The following now holds—

(*a*) Steam supplied to first stage after doing work thereon is bye-passed into the third stage by passage " X."

(*b*) Steam supplied to the second stage continues to pass through the third and fourth stages as before.

Steam supplied by the top casing control valve (which valve has, through gearing, effected the movement of the interstage valve " A ") passes to the third stage and so through the turbine. With this setting, further top casing control valves may also be opened until, at about two-thirds the power, the opening of a further control valve causes, through gearing, the slide valve " B " to move upward from the position covering bye-pass " Y " to a position covering the nozzles into the third stage. The covering of these third stage nozzles causes the steam, which has done work on the second stage, to bye-pass by the passage " Y " to the nozzles of the fourth stage. With this setting corresponding to conditions at higher powers the passage of steam is as follows :—

(*a*) Steam supplied to the first stage bye-passes from the first stage casing to the fourth stage nozzles through passages " X " and " Y."

(*b*) Steam supplied to the second stage bye-passes from the second stage casing to the fourth stage nozzles through passage " Y."

(*c*) Steam to the third stage is now supplied entirely by top casing nozzles and passes from the third stage through the fourth stage nozzles. In the fourth stage casing all steam becomes common to the subsequent stages.

By means of this arrangement and opening only the nozzle valves required to obtain the desired speed with a high pressure in the control belts, an economy of the turbines can be maintained, approximating to that at the high powers, down to a very low proportion of the full power.

It is now proposed to note some important running difficulties that have occurred in large turbines under various classifications of failures.

(1) *Distortion of Turbine Rotor and Stator.*—This is a consideration that has always occupied the careful attention both of turbine designers and users and generally increasing care is required both as the size of the unit and the pressure and temperature range to which the various details are exposed, are increased. It is this liability to distortion under the temperature and pressure conditions that governs the various clearances permitted and therefore has an effect both on the reliability and efficiency. The design stage involves the consideration of careful proportioning of the important details to fair off any changes of thickness as much as possible, such as are involved in the provision of facings for mountings, flanges and any necessary supports, and also in the ribbing. When in use, the warming-up requires careful application, and the working-up of the turbine should be gradual, any sudden changes of speed being avoided as far as possible. These points are now well understood, and generally appreciated. Another feature that has to be carefully attended to is to ensure

that stresses, resulting from the expansions of the steam pipes supplying the steam to and connecting the various units, are not transmitted to the turbine casings. The provision of expansion joints in correct positions will overcome any such effects.

Two partial failures of large 30,000 K.W. machines were attributed to this feature. In one the labyrinth packing on the L.P. element failed due to the buckling of the cylinder caused by rigid piping between the two surface condensers which were rigidly bolted to the turbine, and when the repair was made good flexible piping was installed between these condensers. In the second case the labyrinth packing on the H.P. element failed three times, the failure being attributed to heavy distortional stresses caused by rigid bracing of the steam pipe near the turbine, although excessive clearance in the thrust block was cited as a contributory cause of these failures. As might be expected, such failures are more likely to occur in the reaction machines where the clearances are generally less than in the impulse machines, although in the case of a comparatively small impulse machine (3,000 K.W.) failure was attributed to the distortion of a last stage diaphragm, which fouled the last wheel, owing to a sudden change in temperature as a result of changing over from condensing to non-condensing condition. Under such a condition the last diaphragms will heat up much more quickly than the more massive turbine casing. The diaphragms will consequently expand at a greater rate than the casing and if the clearances are not sufficient, the diaphragms may distort in an axial direction under the pressure exerted by the casing in the radial direction, and if excessive, will cause rubbing between the diaphragm and rotor. The difficulties in this connection are obviously more acute in connection with diaphragms of large diameter and a certain amount of radial clearance where the diaphragm is held in the cylinder is a necessity.

Apart from possible distortion due to temperature differences, the diaphragms are also exposed to the distortion due to the pressure difference between the two sides, and it is difficult, without introducing corresponding disadvantages, to arrange for a very stiff support of these fittings where attached to the cylinder casing. The complete wrecking of a large 35,000 K.W. machine was attributed to the deflection of a diaphragm, which amounted to more than the axial clearance and so caused fouling of the adjacent wheel. The accident took place when, due to a breakdown of another machine on the system, the large unit was called upon to supply overload. The rubbing between the diaphragm and disc was so intense that it caused the diaphragm to revolve with the rotor until it fractured and burst the turbine casing.

It is not possible to calculate with any degree of accuracy the deflection of a diaphragm owing to the peculiar nature of its construction and support, and especially too if the diaphragm is in halves. It is now considered advisable in this connection to test these large diaphragms for deflection before they are actually used in the turbine to ascertain whether this deflection is well within the limits of the axial clearance, allowances being necessary for distortion due to temperature changes.

(2) *Cracking and Breaking of Turbine Discs.*—Considerable difficulties have been experienced in land practice due to the cracking of turbine discs and this feature is one of the greatest difficulties in large high speed turbine design. The cause of failure may be due to various reasons.

(*a*) Excessive stresses due to the centrifugal forces.
(*b*) Vibration of the discs.
(*c*) Excessive stresses due to uneven distribution of temperature.
(*d*) Faulty material.
(*e*) Faulty design, manufacture or treatment during manufacture.

In the general case the turbine discs bursts with practically no warning, completely wrecking the turbine and adjoining machinery. With the exception of cases where the turbines have run away due, say, to a governor not operating, the failure of the disc has been due to a fatigue fracture which may have taken considerable time to develop. This fatigue fracture is caused by the vibration of the turbine disc giving rise to considerable vibratory stresses being superimposed on the static stresses due to centrifugal force, and uneven distribution of temperature. These static stresses (using the qualification " static " in the sense of " steady " although the stresses may be caused by dynamic effects) are susceptible to calculation although presenting considerable analytical difficulties, and are in any case somewhat uncertain in proximity to steam balance holes, keyways with sharp corners, sharp edges, or any rough machining work on the disc. Such local stressing of the material must therefore be avoided if possible. The vibratory stresses when set up result in local stresses exceeding the elastic limit of the material, and this will ultimately result in a fatigue fracture.

It is very important, therefore, that the turbine discs should be designed in such a way that synchronous vibration due to small periodic disturbing forces of any kind is not possible. It appears from experience that as a general proposition difficulty in this connection arises if the discs are run at speeds above about 650 f.s. at the mean blade diameter. If this speed be exceeded the discs have to be made much heavier than would be necessary from the consideration of centrifugal stresses only.

A turbine disc is liable to vibrate in various ways. The vibration may be of the umbrella-shape type, or it may be of the segmental type with two or more nodal diameters. The former vibration is likely to be set up by some axial disturbances of the shaft such as may arise, say, in a flexible coupling. The latter type may be caused by the uneven distribution of the admission of steam over the periphery of the turbine wheel, as may occur with partial admission or with small inaccuracies in the nozzles such as often exist near the horizontal joint of the diaphragm. In such a case the nodal diameters are stationary and the disc rotates in a deformed shape which shape is stationary in space. There is evidence, however, that dangerous vibrations also occur where the nodal diameters rotate with the disc. Such vibrations may be set up by a slight vibration of the machine resulting from a rotor out of balance.

The frequency of natural vibrations of a disc depend not only on the form and the physical properties of the material, but also on the internal forging and machining stresses, and the distribution of temperature in the disc. The last-named factors greatly complicate the problem of determining the natural frequency of vibration by calculation, and it therefore behoves the designer to keep well within safe limits rather than attempt to find it by trial and error or by actual experiment, as the experiment to be satisfactory can only be carried out on the actual machine under working conditions.

Some of these failures may now be mentioned :—

15,000 K.W. Machine: wrecked through bursting of the last disc, the fracture starting from the edge of a balancing hole.

35,000 K.W. Machine; 19th wheel burst and wrecked machine. Previous trouble due to vibration and shrouding coming off had been experienced. The disc cracked from a balance hole close to the rim; these balance holes were roughly machined and had sharp corners.

15,000 K.W. Machine; failure of last wheel causing wrecking of L.P. cylinder. Wheel cracked between two balance holes.

5,000 K.W. Machine; wrecked through bursting of third disc. Fatigue crack developed from balance hole in third wheel causing fracture.

20,000 K.W. Machine; severe vibration; all blades were stripped off seventh wheel and fifth wheel showed a radial crack. The wheels were made of cast steel, and cracks have developed in other discs of similar machines. The discs were replaced by others of forged steel.

5,000 K.W. Machine; wrecked through bursting of first disc. Attributed to vibration on four nodal diameters. Stiffer disc fitted in the replace design.

6,250 K.W. Machine; wrecked through bursting of first disc. Cause rather obscure but possibility of vibration on four nodal diameters is supposed to have caused fracture to commence from a local weakness due to faulty material. A duplicate disc in a similar machine has run satisfactorily for years. A replace disc failed in 10 days. Stiffer disc fitted subsequently.

30,000 K.W. Machine; wrecked through bursting of last disc. Failure appeared to originate from a deep circumferential tool mark made in machining the disc and situated about midway between hub and rim. The material did not appear entirely satisfactory and stresses due to running and possible vibratory causes would appear to have further accentuated stresses which would be already above those anticipated in the neighbourhood of the deep tool mark.

An interesting case of vibration has been reported from a Zoelly machine. It was the 21st of a series of machines all of the same power and revolutions, and no trouble had been experienced with the first 20. In the 21st machine, disc vibration occurred about three nodal diameter, 120° apart. The only difference between this machine and the remainder was that it was connected to an alternating current generator giving a number of cycles different from the others, and this arrangement led to the same number of pole-changes in this machine in a given time to be equal to the revolutions, and synchronous vibration was believed to have been set up. The defective wheel was replaced by one of stiffer material.

Although lightness is a special feature in some classes of naval designs, there have been no failures similar to those above, and the generally lower peripheral speeds combined with correct design, careful machining, balancing and fitting appear to have kept such designs well within safe limits.

Blading Failures.—The blading has at all times been looked upon as the most vital feature in a steam turbine and the majority of turbine failures have originated in these details. Generally speaking, however, with such failures it is most unusual for the machine to be absolutely wrecked, as although in cases the damage is extensive, it is confined to blades and nozzle plates, the main structural details remaining serviceable without machining work being necessary. In a number of instances the cause has arisen outside the cylinder by a failure of the lubricating supply, leading to overheating and wear of the rotor bearings or adjusting block, with consequent destruction of the blade clearances, whilst in other cases a foreign body or a loosening of an internal fitting has initiated the stripping. These are apart from those cases where contact has occurred through distortion due to insufficient or uneven heating.

There are, however, a large number of cases where blade failure, other than due to the above causes, has taken place, and it has been observed in some of the naval instances that the failure has occurred at positions in which the normal working stresses have been well within safe limits. In such cases it can only be assumed that the blade or blades were defective, either initially from the material point of view, or suffered mal-treatment during the process of blading, or were subjected to a shock effect by a sudden admission of water to the turbine, either with the main steam or auxiliary exhaust supply. Investigation of such cases always presents a certain amount of difficulty, as, due to extension of the defect, it is generally not possible to locate the initial source of failure. In view, however, of the possible causes of failure indicated the remedy is to eliminate all such causes from the design to the running stage. The correct design ensures that the proportions are sufficient to enable the stresses incurred to be easily borne by the material adopted, and all sharp corners, such as may arise in dove-tailing, must be eliminated to avoid an excessive local stress, which may easily lead to fracture and failure in course of time. In the manufacturing stage the material of each individual blade must be up to sample and must be free from any cracks even of a microscopic nature. Very careful examination of the whole of the finished blades is indicated as a necessity. Coming to the blading operation the blades must at no time be subjected to mal-treatment such as being driven heavily round the groove in the wheel, as apart from the driving strains introduced it indicates that the blade itself is also being locally stressed owing to its tight fit. There is reason to believe that under the stress of war conditions failures of blades can be attributed to treatment of this nature. Further stresses may be introduced in setting the blades when fairing them up to accommodate the shrouding or binding strips and such work should be carried out with discretion. The shrouding itself is a potential cause of failure which must not be overlooked. The holes in the shrouding should have rounded corners to obviate introduction of local stresses, and should be evenly and centrally spaced, otherwise the blades and shrouding are strained during the fitting.

Then finally due attention and care on service must be given to obviate any shocks being brought on the blading.

There still remains the question of vibratory stresses which may arise especially with the long blades. The vibration may be set up by vibration of the disc, by excessive vibration of the machine due to an unsatisfactory rotor balance, or it may be due to vibration of the blades themselves. If in the latter case an alteration of the blade shape is not practicable, a more rigid element can be obtained by staying intermediate points of the blades.

There have been no failures in the naval service that can be attributed to vibratory causes pure and simple, but in two very large land units it would appear that blades failed on the L.P. wheel due to an accentuation of the normal stresses by vibration. The damage in such cases was local and in one case a loss of efficiency was accepted by running with the last row of blades absent. In the case under consideration the blades were 34 inches long on a mean diameter of 11 ft. 7 in. and running at a speed at the mean circumference of 720 f.s., the design being of an extreme nature.

Intermittent vibration is sometimes noticed in turbines especially on first starting after warming up. This may be due to distortion destroying the clearance of the diaphragm glands which consequently make contact with the rotor spindle. After standing or running easily for a short period this distortion is generally overcome. In the earlier Brown-Curtis designs these glands were fitted to touch the spindle, the comb-rings in segments being held by a garter spring. Temporary seizing sometimes took place while standing and vibration was experienced till the combs wore or the seizing was overcome. In all the later designs, however, there is a definite clearance between combs and spindle, the gland rings being fitted with a shoulder which ensures the clearance being always maintained.