Paper read at the Institute on Tuesday, 24 October, 1972

MARINE STEAM TURBINES—SOME POINTS OF DESIGN AND OPERATION

K. M. B. Donald, B.Sc., C.Eng., M.I.Mar.E., M.I.Mech.E.*

Mr. Donald

The author believes that the time is ripe for a discussion of some of the problems affecting reliability in currently operating modern marine steam turbines.

The paper begins with an examination of the types of vessel for which steam turbine main machinery has been selected, and the trend of requirements in the immediate future. It is shown that steam turbine-propelled vessels today represent large capital investments, and thus even more vitally than hitherto, are expected to maintain a high standard of reliability. Whether reliability has been appreciably forfeited in the quest for better fuel rates is left open for further discussion. The author then goes on to enumerate some of the problems which have adversely affected the satisfactory operation of main steam turbines in the past, treating each point in some detail, starting with an examination of the machinery when running, followed by an inspection with the casing top half removed. There is a section on the important subject of blade vibration, which points out some of the difficulties which face the designer after he has satisfied the thermodynamic and aerodynamic requirements in blade design.

There is also a section devoted to possible causes of rough running of steam turbine machinery, both during trials and after a period of satisfactory operation. This leads into an introduction to the balancing of rotors, including critical speeds, modal balancing (multi-plane balancing), and oil film whirl.

The paper is concluded with a short section on limits and measurement of vibration in turbine machinery.

INTRODUCTION—THE IMMEDIATE FUTURE (SHIPPING STATISTICS UP TO JULY 1971)

It is worth examining the market for steam turbines in the immediate future so that some idea of the numbers and powers required can be seen. It is not intended to present arguments in favour of, or against, the motor or turbine for main propulsion of vessels but merely to illustrate the present trends and the likely numbers of steam turbines required.

Leaving aside vessels like passenger ships and general cargo ships on the grounds that there are insufficient of the former built to make much impact on turbine manufacturers, and that both size of the latter and tradition make the use of steam turbines unlikely, four other types of vessel were examined:

a) the oil tanker;
b) the container ship;
c) the ore, bulk and ore/oil carrier;
d) the liquefied gas carrier.

Figs. 1, 2, 3 and 4 illustrate the total number of steam turbine-powered vessels built and on order each year with accompanying curves of the average power/screw for each vessel.

Tankers

The service speed of tankers ranges between 27·8 km/h to 31·5 km/h (15 to 17 knots), and this average speed is carried over to the vessels on order up to 1975.

The cost per ton of carrying a cargo of oil decreases with increasing size of vessel in theory but the present trend shows that the cost of freighting levels off at a deadweight size for a VL CC of about 305 000 tonnes (300 000 tons).[1] With these limitations of the major parameters it is not surprising that the average shp requirements for tankers are rising slowly.

On the other hand, the numbers of vessels greater than 81 300 tonnes (80 000 tons dwt) on order for the next three years or so is at least maintained at a high level of 60 to 70 vessels per year. This market can therefore be said to be steady both in numbers and in power requirements.

Container Ships

The average shp requirements per screw for container ships are steadily rising as both deadweights and service speeds increase. Some power units will be required of up to 42 MW (60 000 shp)/screw. These are 44 100 tonnes (43 400 tons deadweight) vessels with twin screws capable of 61·2 km/h (33 knots).

Of the total number of container ships on order, about 50 per cent are twin screw. One 21 640 tonnes (21 300 tons deadweight) vessel is to be powered by a steam turbine delivering 36·8 MW (50 000 shp) on a single screw with a service speed of 46·35 km/h (25 knots). The number of container ships built since 1968 is comparatively few, however, and although the number on order rises to 31 at the end of 1972, 1973 does not show a continued increase in the numbers.

Surprisingly, there were 227 container ships in operation at the time of this survey. Some 60 or more converted by the Americans from oil tankers, cargo vessels and naval craft operate at 24·1 km/h (13 knots) to 29·65 km/h (16 knots) and are steam turbine-powered. In fact, the number of container ships now in service may be too many for the market, hence the decline in numbers on order for 1973. Nevertheless, this is a market for the highest power steam turbines afloat.

Ore, Bulk, Ore/Oil Carriers, Liquefied Gas Carriers

Ore, bulk, ore/oil carriers (OBO) and liquefied gas carriers have not presented a large market for steam turbines, but average power requirements show an upward trend as sizes of vessels increase each year. Therefore, it may be expected that the numbers

* Senior Engineer Surveyor, Lloyd's Register of Shipping

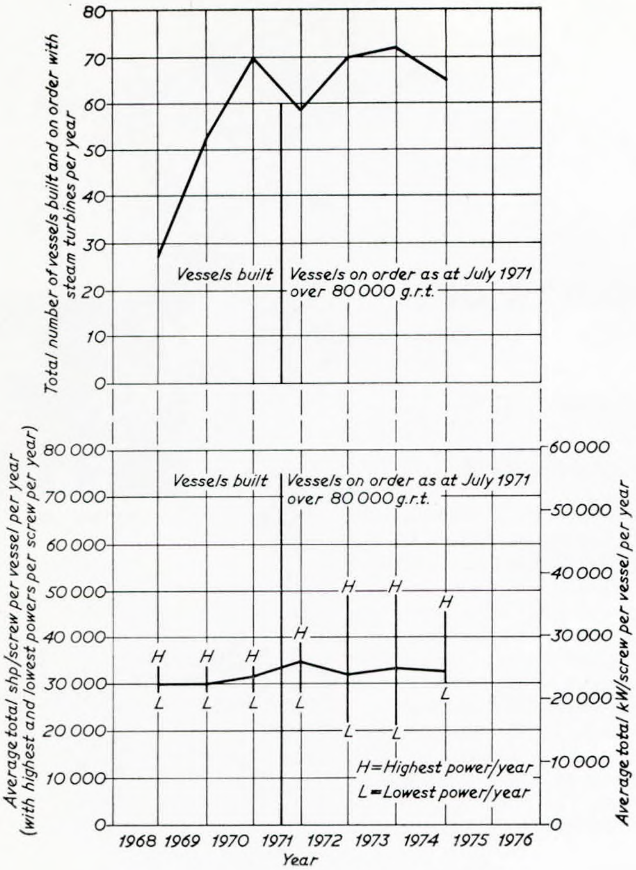

FIG. 1—*Tankers*

powered with steam turbines will rise as deadweights increase and may be considered to be a developing market.

Like oil tankers, average speeds are 27·8 km/h (15 knots) to 31·5 km/h (17 knots), so the power requirements will be roughly in line with increasing size, and steam turbines could be preferred for propulsion as deadweights increase.

There is some evidence that liquefied gas carriers may require larger powers because service speeds are tending to increase.

Marine Steam Turbine Reliability

It can be seen from the foregoing that steam turbine-propelled vessels today represent large capital investments, and thus, even more vitally than hitherto, are expected to maintain a high standard of reliability.

Unfortunately, in the quest for better fuel rates, designers have had to pay the inevitable penalty of more complex steam cycles, including in some cases the use of reheat, and higher steam pressures, all of which tend to act against the reliability associated with the simple cycle of earlier years.

The biggest single improvement in fuel rates is achieved by the adoption of the "reheat" cycle at the higher steam operating conditions of about 98 bar (1420 lb/in^2) and 520°C (968°F) re-heated to 520°C (968°F), whilst the next single biggest improvement is to employ the higher inlet conditions in a "straight" cycle. It is doubtful whether reheat is justified, however, at powers of less than about 22·1 MW (30 000 shp).

Other ways of improving the fuel rate are to incorporate

a) a main turbine-driven generator;
b) regenerative feed heating, with up to as many as five bled steam points;
c) condensate-cooled lubricating oil coolers;
d) vacuum pumps instead of "steam" air ejectors;
e) scoop circulation of the condenser cooling water.

Only service experience will determine whether such improvements have shown the desired benefits without impairing the reliability of the plant compared with the conventional simple cycle.

A main turbine-driven generator, for example, could be difficult to operate at constant cycle frequency over a long period of time as the propeller revolutions fall with increase in hull fouling etc., and furthermore, heavy weather conditions may make it necessary to declutch the generator from the main turbine due to variations in rotational speed. Separate turbine-driven generators of the back-pressure type would probably be preferable in most instances.

One other possibility, namely, a boiler feed pump driven by a separate back-pressure turbo-generator, is probably also impractical in view of the constant speed requirement of the turbo-generator.

With regard to re-heat, apart from the added complexity of control and the careful feed-water conditioning necessary at the higher operating conditions, the main supply and reheat piping would be somewhat complicated, especially if twin boilers were installed.

Automatic control is becoming far more widely adopted for steam installations but, in general, the simpler the system, the simpler to automate.

CONCLUSION

The prospects for steam turbines are going to be determined to a large extent by the satisfactory operation of current turbine machinery, and all associated equipment such as boilers, gears etc.

Only by experience in operation and constructive discussion of the problems which arise can progress be made. It is with this objective that the author has written this paper.

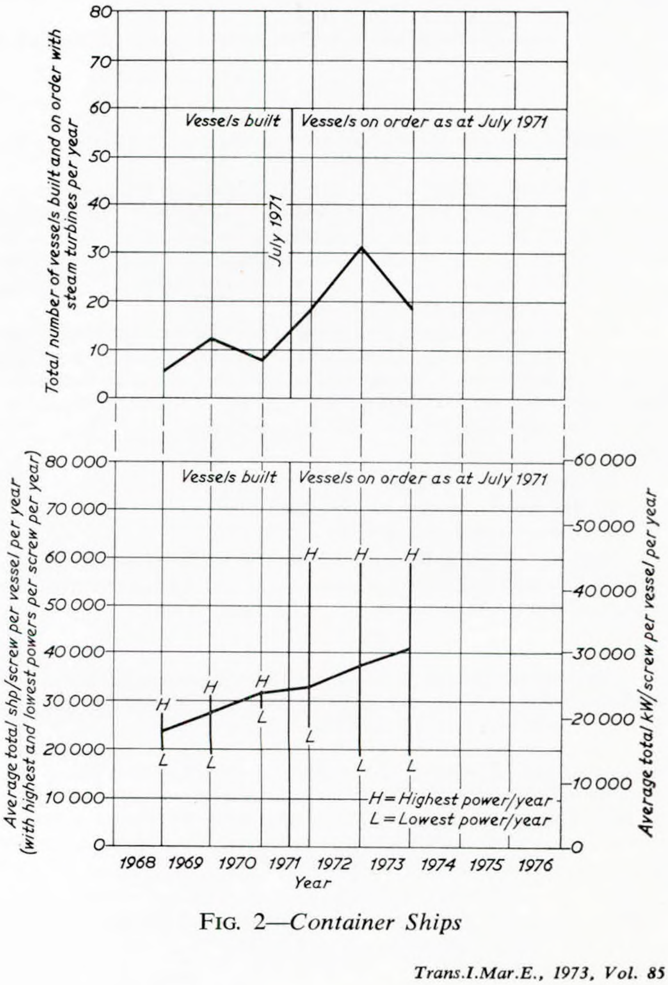

FIG. 2—*Container Ships*

SOME PROBLEMS WHICH AFFECT THE SATISFACTORY OPERATION OF MARINE STEAM TURBINES

Overall reliability is of course determined by the component parts of the main engine installation such as those of boilers, gearing and associated equipment, but this paper deals almost exclusively with the main turbines, and in some detail.

There are a number of different designs of turbine still operating from earlier days, but basically, these do not differ greatly from modern turbines, and many of the service problems are common to all.

External Examination of the Turbine When Running

Steam or Lubricating Oil Leaks

Apart from being wasteful, both are potential hazards in an engine room space.

Lagging

Apart from thermal efficiency considerations, reduction of heat losses with properly lagged casings, particularly the H.P. turbine, is important to reduce the amount of casing distortion, and to maintain a cooler engine space. This also applies to steam pipes, of course.

Steam Pipe Forces and Restraints

To keep external forces on the turbine casings to a minimum, piping such as main steam pipes should not be fixed rigidly, or restrained rigidly at positions near the turbine flanges, so as to minimize forces and moments acting. Bled steam pipes, and lubricating oil pipes should not be run so as unduly to restrain the natural thermal expansion of a turbine casing.

Internal Examination with the Top Half Turbine Casing Removed

Checks should include the following:

a) erosion of blading—(i) ahead blading; (ii) astern blading;
b) fouling of blades with deposits;

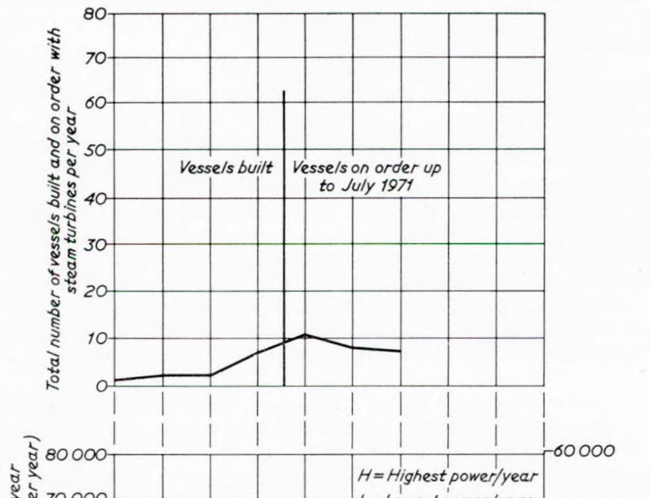

FIG. 3—*OBO*

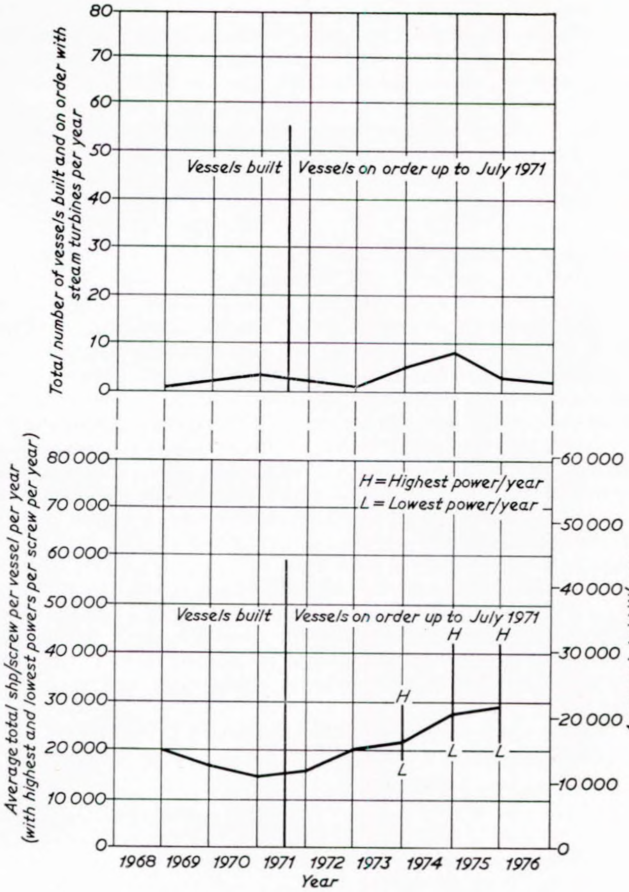

FIG. 4—*Liquefied gas carriers*

c) rubbing of labyrinth glands;
d) rubbing of moving blade tip seals—(i) axial rubbing; (ii) blade tip rubbing;
e) blockage of bled steam drains or water extraction drains;
f) lacing wire brazing failure;
g) failure of stator blading or nozzles;
h) warp of horizontal joints.

Erosion of Blading—Ahead Blading

The last two or three stages in a condensing turbine could be subject to erosion damage because to achieve the best possible thermal efficiency, expansion of the steam into the saturated region is necessary.

It is usual in present-day designs to provide water extraction channels in the annular space between fixed blades in the final stages so that water centrifuged off the moving blades can be collected and drained away. Some manufacturers also fit stellite shields on the leading edges of the last two or three rows of moving blades in the L.P. turbine to minimize the rate of erosion.

Turbine designers are well aware of the dangers of erosion, and one of the main design parameters of the L.P. turbine is the limitation of the last row tip speed to minimize erosion damage.

The incidence of erosion in modern marine turbines is very small, and cases where it has become significant are usually due to operation at low superheat temperatures, high vacuum, lack of, or inadequate, water extraction channels, blocked extraction drains, or combinations of all these factors over a period of time.

Erosion of Blading—Astern Blading

The astern turbine is usually located on the L.P. rotor, so if there has been extensive erosion of ahead blading resulting from high moisture content, it may be found that astern blading is also eroded due to water droplets forming on the outer casing and dripping into the exhaust end of the astern casing.

When the astern turbine is turning ahead, it tends to act like a compressor, drawing in the drips of water falling past the exhaust, which erode the astern wheel blades in much the same manner as in the ahead stages, except that the water drops strike the trailing edges of the astern blades. This effect may occur more often when both ahead and astern exhausts are adjacent and the flow is downward into the condenser than in the case of the single-plane layout, where both ahead and astern casings exhaust axially in the same direction downstream.

Fouling of Blades with Deposits

Boiler salts can enter the turbine in three ways. Firstly by priming as when the concentration of boiler solids is high, there is a tendency for the water to foam. During ebullition, bubbles explode on the surface carrying over tiny droplets of water into the superheater, and on into the turbine. Secondly by taking over a quantity of water as a result of heavy rolling, or due to faulty water-level control and thirdly, in a closed system consisting of water containing dissolved matter, in equilibrium with the vapour, the dissolved matter distributes itself quantitatively between the vapour and the water, in a ratio which depends on the nature of the dissolved matter and the temperature and pressure of the system(2). (This ratio is called the distribution ratio, and it increases as the temperature is raised). At boiler pressures of 58·6 bar (850 lb/in^2) or above, the effect becomes significant with regard to silica. In effect the silica becomes volatile in steam. The presence of caustic soda reduces the ratio for silica, but if the silica content is not kept low, it will be carried over in quantity and precipitated out at some stage in the turbine, and deposited on the blades.

During some experiments on land sets(3) it was found that salts had precipitated in areas of the turbine roughly as follows:

sulphates—in early H.P. turbine zone
sodium—around the 315·5°C (600°F) area (melting point 317·6°C [604°F])
chlorides—a little lower down the turbine
iron—all through the machine
silica—lower H.P. zone and downstream

The nature of the sulphates and chlorides was not stated.

Rubbing of Labyrinth Glands

There are many designs of shaft glands, but probably the most common type in use today incorporate replaceable spring-back knife-edge segments fitted in steel housings.

Up to temperatures of 399°C (750°F) leaded-nickel-bronze glands have been frequently used by most manufacturers, and on occasions this material has been used up to 482·5°C (900°F) plus, but at these higher temperatures rapid de-zincification and resulting cracking may occur which could lead to severe shaft damage if rubbing took place.

13 per cent chromium stainless iron gland material is generally used above operating temperatures of 427°C (800°F).

Causes of gland rubbing include casing thermal distortion, a thermal bend in the rotor and displacement of the casing relative to the bearings, due, for example, to pipe thrusts or moments.

Theoretically, a bent shaft should be the only circumstance under which rubbing aggravates the bend by heating the outward bowed perimeter of the shaft, thus increasing the bend in that direction.

If a permanent bend results, it is interesting to note that when cool, the shaft will be bowed in the opposite direction to the sector of the rotor which has rubbed. This is because the outer skin of the rotor has yielded in compression when heated locally by the rub and pulls the rotor into a bent condition with the rub on the concave side when cool.

Rubbing of Moving Blade Tip Seals

Because the moving blades in the H.P. cylinder are sometimes designed to have a small degree of reaction, the shrouding incorporates a radial, or axial, seal, and in some instances both are employed. Clearances of 0·25 mm or less are typical in the initial stages because the specific volume of the steam is small.

Axial Rubbing: The thrust bearing is usually located at the steam inlet end of a turbine, so that when working up to service power, the rotor which is heated more rapidly than the casing, expands at a faster rate, and the rotor discs and blades move away from the diaphragms, increasing the axial clearance.

When power is being reduced, the opposite effect takes place. Closing in the manoeuvring valve or throttle valves, the steam is throttled to a lower temperature and pressure and the rotor is cooled more rapidly than the casing, and axial clearances are accordingly reduced.

It is important to emphasize, therefore, particularly in the case of H.P. rotors, which have a relatively small mass compared with the casing and its insulation, that differential expansion is the important parameter (i.e. the expansion or contraction of the rotor relative to the casing). L.P. turbines may well have a reverse relationship of masses. Thus an indicator which shows only casing expansion is of little value, other than to indicate whether or not there is a restraint to free expansion.

In general, therefore, if an axial rub has occurred, the cause will be reduced clearance from a rapid drop in the steam temperature which is also affected by the different rates of temperature change of casing and rotor. It could also be due to incorrect adjustment of clearances when first installed, or after later overhaul.

The checking of clearances will only be correct when both rotor and casing are at the same temperature, and furthermore, various angular positions of the rotor should be tried to see if clearances are the same all round.

Axial rubbing could overheat shrouding, diaphragms, or blade roots (if there is a blade root seal as well as a tip seal) and may result in localized yield.

Blade Tip Rubbing: The last few stages of an L.P. turbine usually embody moving blades with a relatively high degree of reaction at the mean radius, and although shrouding would also be an advantage to prevent flow spillage over the tips, the length of the blades employed and their shape do not always enable shrouding to be readily fitted.

In such cases tip leakage is minimized by designing for small clearances between casing (or diaphragm rim extension) and moving blade tips.

The tips of the moving blades are machined down to fine edges so that if radial tip rubbing should occur, the blade tip will be worn away without serious damage to the blade.

Some designs of L.P. casing are integral with an underslung downward flow condenser, and the rotor incorporates the astern turbine. During manoeuvring operations, going ahead and astern, the L.P. casing is usually subject to thermal distortion so that sometimes the blade tips touch the casing. If lacing wires are incorporated, the rubbing could put a strain on the brazing between wire and blade, and cause failure of the brazing.

Blockage of Bled Steam Drains or Water Extraction Drains

It is most important that bleed belt drains, and water extraction belt drains are kept open and free of scale and other debris. They are usually orifices led by a pipe to an atmospheric drain tank, or directly through the bottom of the inner casing into the condenser.

Modern L.P. turbines almost always have water collector channels between the stationary blades of the last stages

The wet steam droplets centrifuged to these channels coalesce and flow to the bottom of the casing where they pass through a drain orifice. If these holes are blocked, the channels fill with water, until they spill over into the next stage, causing erosion damage to the moving blades, particularly at the tips.

Bleed belts can also fill with water unless drains are clear. This can cause not only a local thermal shock on the casing and blades but could also distort the turbine cylinder sufficiently to cause interstage gland rubbing.

Drains led to a drain tank can be readily tested for blockage, but the type which drains directly into the condenser below are more difficult to test without lifting out the rotor. The simplest test in these conditions is to pour water into the drain channels to see if the water drains out into the condenser.

Lacing Wire Brazing Failure

The principal object of lacing wires is to inhibit certain modes of moving blade vibration which may otherwise cause fatigue failures. A number of blades are secured to one another by brazing the wire to each blade, a process which should be closely controlled in the building stage to ensure that the brazing material runs well into the hole and fills the spaces between the wire and the hole by capilliary action, leaving neat fillets on both sides of the blade. If, for any reason, the holes are not adequately filled, the brazing may fail after a period of running and one or more blades is then no longer inhibited and could vibrate as a single blade. Damping wires do not prevent blade vibration, but limit the amplitude at resonance by providing dry friction damping.

These latter wires are not brazed to the blades and may be single-pitch lengths, or cover a number of blades, depending upon the manufacturer's design. One of the major problems in the use of damping wires is that they tend to wear away due to frettage.

Failure of Stator Blading or Nozzles

If stator blades, such as are found in the "Parson's" 50 per cent reaction turbines, break off, they will cause damage to moving and fixed blades downstream.

They are unlikely to fail due to vibration, however, because they are less highly loaded than the corresponding moving blades.

Diaphragm nozzles sometimes become eroded at the trailing edges, and small pieces break away, causing damage to moving blades.

Thermal shock stresses resulting from "slugs" of water quenching nozzles, blading, and rotor, are much more serious than any deposition of salts and there is also a possibility of thrust bearing failures, due to the sudden axial additional forces exerted on the rotor.

On occasions, it may be necessary to remove a row of moving blades as a temporary measure, following moving blade damage. In these circumstances, if the corresponding row of stator blades or nozzles is not damaged, they should be left in place to reduce the steam pressure before the following set of fixed blades, otherwise excessive pressures could be set up at the maximum operating condition which would cause damage to diaphragms downstream.

Warp of Horizontal Joints

It is sometimes found that when the top and bottom half H.P. casings are parted, the horizontal joint is not flat. A straight edge placed diametrically across the flanges may show the outer edge of the flange to be higher than the inner, and in some instances the joint has been made good again by filing and scraping the flange faces to meet flat.

In fact, this is not necessary because the cause of the warp is high temperature creep of the inner fibres of the casing in compression and making good the joints will merely allow further creep to take place later. The joints should be left alone and the horizontal joint bolts pulled up hard to close the gap on the inside. Although the cold pull-up in the bolts will produce high stresses, when warming through the turbine and running up again, the stress in the bolts will be relieved immediately. Remaking of the flanges will only be necessary where it is found that there has been leakage which has wire-drawn the flange face, and under those circumstances the manufacturer should be consulted. The leakage may have been caused by creep in the joint bolts, relieving the stress on the inside edge of the flange, and allowing a gap to appear when reducing steam flow and temperature.

Hull Deflexions and Machinery Alignment

The effects of hull deflexions, particularly with regard to VLCC at the extremes of loaded and ballast draughts, are already acknowledged, if not readily computable to sufficient accuracy. As VLCC become larger, the problem of maintaining turbine to gear, and gear to gear, alignment will worsen, particularly if higher tensile steel is used in ship construction, thus tending to increase structural deflexions. Further work is needed in this direction, so that engine-builders can design machinery to tolerate given types and magnitudes of movement of engine and line shaft seatings.

Gearing

With the exception of certain epicyclic primary reduction gearing (which has been well reported elsewhere[(4)]), there have not been many major problems.

Some Aspects of Blade and Wheel Vibration

Blade vibration is an important feature of turbine design, for if one considers that there may be upwards of 3000 blades in a compound turbine, and the failure of a single one could temporarily put half the unit out of action, it is perhaps not surprising that most turbine manufacturers devote special attention to this subject. Regrettably, however, blade failures do still occur.

Geared marine steam turbines are inherently variable-speed machines, and must therefore be considered in that light when blade resonant frequencies and stresses are being calculated.

A few of the blade vibration problems facing the turbine designer are posed in the following, after a brief explanation of some of the basic concepts of blade and wheel vibration.

Blades

Most modern turbines are fitted with "impulse-reaction" moving blades of radially constant cross-section, with the exception of the last three to four rows in the L.P. turbine which are usually twisted and tapered along their length.

Each row of moving blades is preceded by a diaphragm containing a full circle of nozzles, except for the first stage "control" nozzles which occupy only a small arc of the circumference, termed "partial admission".

Blade Shrouding, Lacing Wires, Damping Wires

"Impulse-reaction" blading is usually shrouded with a cover-band, fixed to the blade tips, which in addition to preventing steam leakage over the tips, serves in particular to reduce the amplitude of vibration of the blades, by careful choice of the number of blades covered by a continuous length of shrouding. Blades connected in this manner are referred to as a "batch" or "packet" of blades.

Lacing wire, on the other hand, which is "threaded" through a small hole in each blade, and fastened to each blade, is intended to inhibit a particular mode of vibration of the "batch".

Damping wires may sometimes look very similar, but are not fastened to each blade, therefore they limit the amplitude of vibration by dry friction damping, as previously mentioned.

Rotor Wheels

The wheels onto which the moving blades are fitted can vibrate like discs (the blades acting as an extension of the wheel), and each disc has definite modes of vibration, such as nodal diameters, and nodal circles, illustrated in Figs. 5(a) and 5(b).

If nodal diameter or nodal circle frequencies coincide, then a complex pattern of both modes occurs, illustrated in Fig. 5(c). If blade and wheel frequencies occur at or near the same frequency, they tend to interact, each modifying the other somewhat.[(5)]

Blade Roots

There are many methods by which blades are fixed to the wheels. Fig. 6 illustrates some of the more common forms of root fixing in use today. Blade vibration characteristics depend upon the type of root employed, so far as natural frequencies, damping and vibration stresses are concerned. At the same time since blades are subjected to high centrifugal force when the wheels rotate, the roots and wheel rims must be capable of withstanding high centrifugal (steady) stresses.

Some manufacturers have a standard size root for various blade lengths of the same profile. In this way a rotor can be fitted with blade lengths appropriate to each stage, depending on the cycle requirements of individual owners. One manufacturer, for example, uses 24 different blade lengths from 12·6 mm to 67 mm, employing an identical root size and blade profile for each.

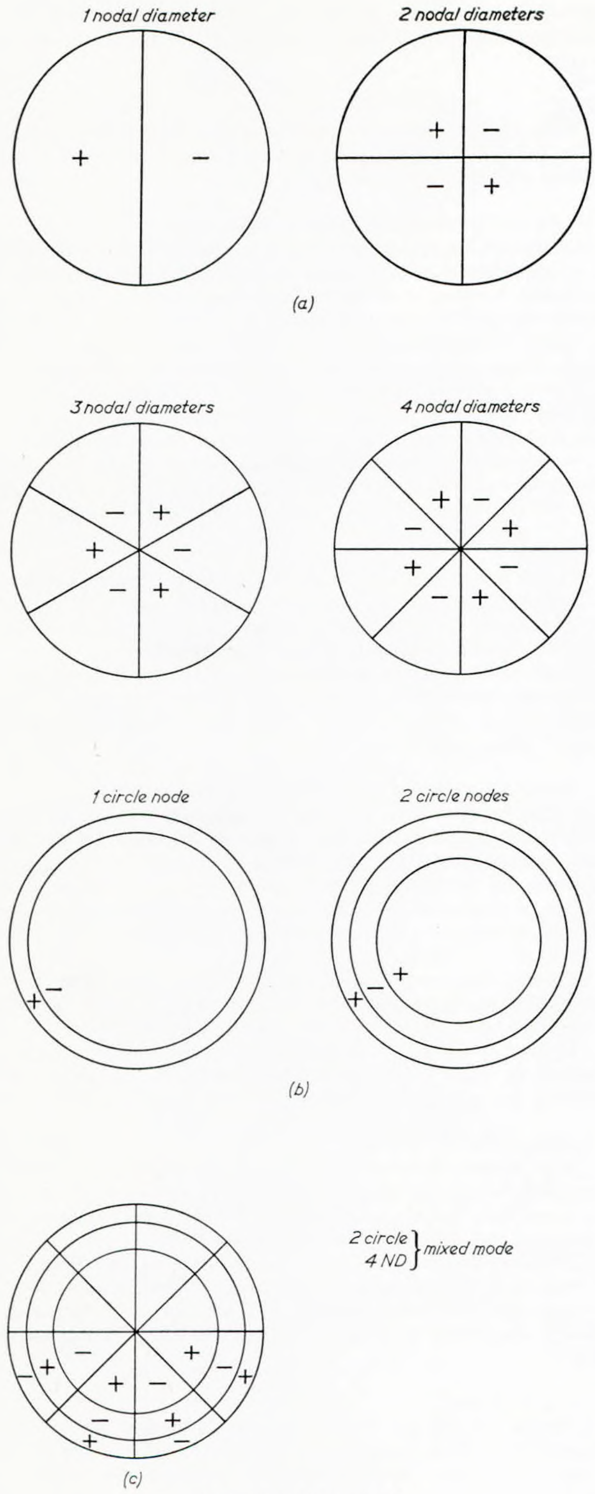

FIG. 5 (a, b and c)

Modes of Blade Vibration

Strictly, blade vibration should always be considered in terms of the total dynamic system of wheel and rotor, since resonant frequencies, damping, and energy distribution are all affected by the total system. For convenience, however, and since the degree of coupling may be small, it is acceptable in most instances to consider a single blade, or a "batch" of blades (whichever is applicable) when reference is made to the various modes of vibration.

For blades arranged in batches, connected by shrouding, there are modes of vibration of the batch, which are unique, called the "clamped-pinned" modes of vibration and these may occur in the tangential, axial or torsional directions.

The name "clamped-pinned" is something of a misnomer because the tips are not truly pinned nor the roots truly clamped. The shrouding introduces a small bending moment, dependent upon the shroud and fixing stiffness.

Whilst other modes of vibration are just as important as the "C-P" modes, it is these modes of vibration of batches of blades which have received increasing attention in recent years.

Excitation

The most common form of excitation is the steam impulse which a moving blade experiences as it passes the stationary blades, or nozzles. With the exception, perhaps, of the first stage, where groups of nozzles are concentrated over a small arc of the circumference, and therefore exert considerable steam force on the moving blades, this is probably the strongest form of excitation the blade will be subjected to, particularly in the H.P. turbine. Other forms of excitation are those from mechanical sources transmitted through the rotor from claw couplings (not used in modern turbines) and gearing. Not many cases exist, however, with proven mechanical excitation as the exclusive cause of moving blade failure.

Steam excitation can also result from variations in flow, or pressure, or both, around the circumference of a diaphragm, due to obstructions in the flow path, bled-steam extraction holes, or localized eddies in the flow path.[6] If this variation is known, harmonic analysis of the distribution may reveal a component which could excite blades.

Damping

There are three sources of damping when the blades are working under operating conditions:

a) internal damping of the blading material;
b) inherent dry friction damping of the blade assembly at root and tip, or by means of damping wires;
c) fluid or viscous damping of the steam environment.

It is usual to consider the dynamic magnification factor Q; however, since this is the quantity which is often measured in vibration testing of blades.

Vibration tests to determine damping are not carried out during running conditions, so the fluid damping is an unknown component as also is the effect of root fixing when subjected to centrifugal force.

Values of Q used in calculations vary, but a somewhat empirical value of 100 is sometimes quoted, which is intended to include all three sources of damping mentioned above.

Measurement of Blade Resonant Frequencies

It is usual for a manufacturer to carry out blade vibration tests on new or improved blading. This may be done by mounting a packet of blades on a solid metal segment and exciting the resonant frequencies in various modes. Resonant frequency corrections are then made to account for centrifugal "stiffening" effects, temperature effects, and so on.

One further effect of centrifugal force due to rotation under operating conditions which has to be considered is that of root fixture and how it may change with operation at, or near, full power.

Referring back to Fig. 6 for example, if the "aerofoil" blade length of the inverted T-root is considered as the distance from the tip to the line x, the actual frequency under operating conditions will in fact be lower than that measured statically, because, under the action of centrifugal force when rotating, the blade root jams up on the "lands" of the T-root at z; so in effect the actual length of blade now vibrating is no longer the tip to x but tip to z. Thus, if a manufacturer has a number of blade lengths with the same root dimensions, then the shorter the blade, the greater the percentage error in calculated natural frequency if the root length is neglected as being an integral part of the vibrating blade.

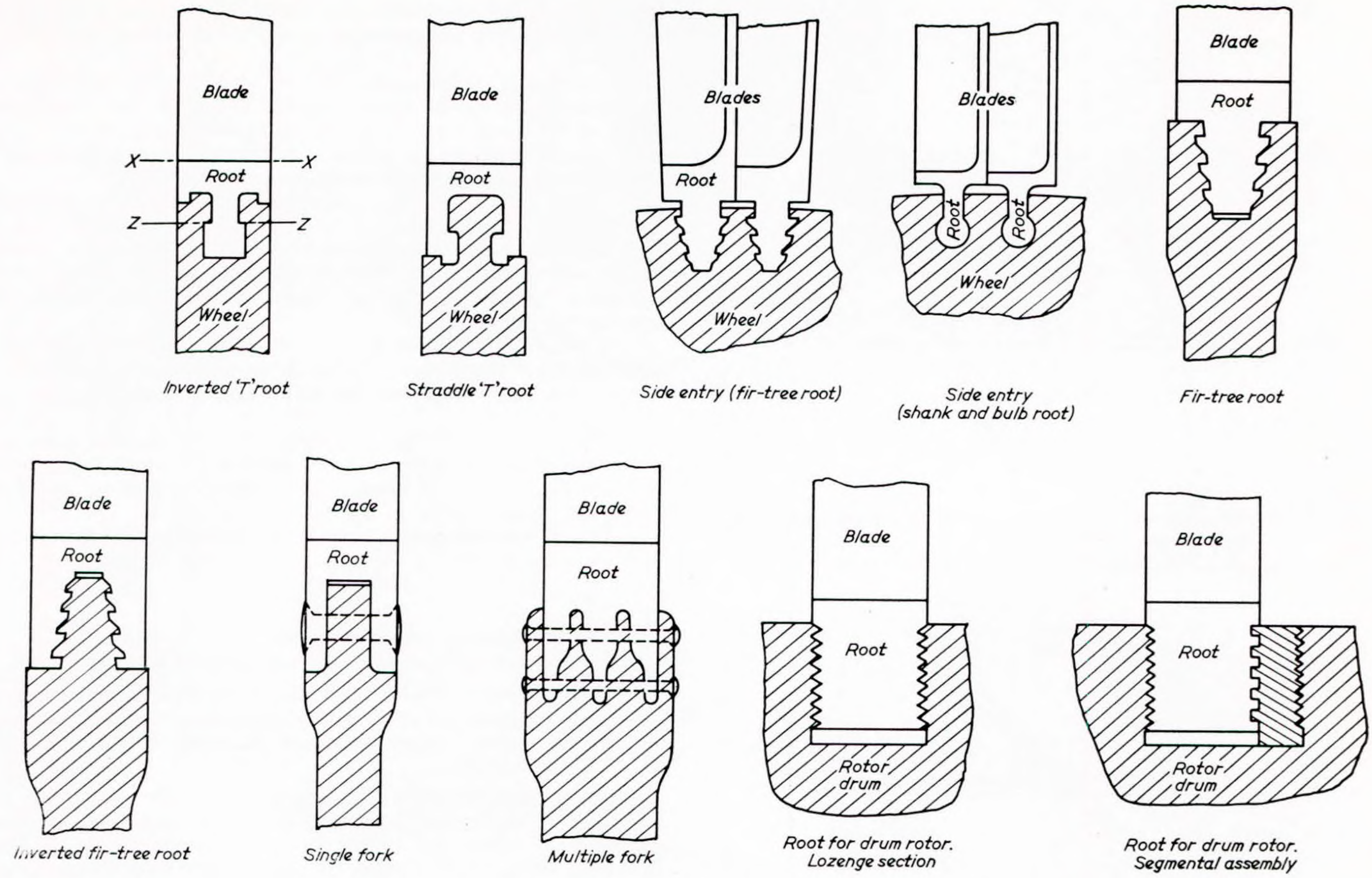

FIG. 6

Static blade tests which attempt to allow for this condition are sometimes misleading, for to jam a wedge under the root end to simulate the hard contact on the T-root "lands", introduces some damping on the bottom of the root, whereas in practice that lower end of the root would probably be free.

Manufacturers are constantly researching into these testing problems, aiming at approaching as nearly as possible to actual operating conditions.

Tests or research "in situ", particularly with main steam turbines, are rarely practicable on account of the availability requirements of large tankers and fast container ships.

The Campbell Diagram

Before either the steady, or the vibratory stresses in a row of blades can be calculated, it is essential to know the angular velocity of the rotor, at which the blade resonant frequencies (for all modes of vibration), correspond with each of the possible exciting frequencies. One of the most convenient ways of portraying the relationship between the resonant and the exciting frequencies is to construct a Campbell diagram, as illustrated in Fig. 7. The resonant frequencies for each mode of vibration of a batch of blades (for example) are plotted up the Y axis (making due allowance for the centrifugal stiffening effect), and rotor speed along the X axis.

All the known exciting frequencies are then drawn as a direct function of rotor speed, passing through the origin. In addition, it is usual to consider harmonics of the exciting frequencies.

As will be seen from Fig. 7, there are many points of coincidence between blade resonant frequencies and exciting frequencies, not all of which would necessarily imply blade failure under normal operating conditions.

It is clear that the more information with which the designer is provided, such as the "dynamic magnifier", the magnitude of the various exciting forces, stress concentration factors, and the metal fatigue properties, etc., the better he will be able to calculate the factors of safety applicable to the row of blades considered.

Blade Vibration Under Operating Conditions

In addition to sophisticated aero-dynamic requirements in the design of turbine blades, they must also be strong enough to withstand the various forces acting upon them, in a particular type of environment.

The forces acting upon them are basically:

a) steam-driving force;
b) centrifugal force;
c) vibratory forces.

At the same time they will be affected by temperature, and possibly also by humidity and corrosive chemicals.

(*a*) *The steam-driving force:* This is effectively the force on each blade turning the rotor and causes a steady bending moment on the blades. If the blades are connected by shrouding, then the position of maximum stress will depend upon the restraining moment at the blade tips, root, binding wire and so on. In general if the force distribution up the length of the blade is known, together with the tip and root restraints, then it is possible to construct a bending moment diagram and to find the position of maximum bending stress. This stress is usually referred to as the "steady steam bending stress'.

(*b*) *The centrifugal force:* The position of the largest stress will depend upon the blade shape, its radius of rotation, and angular velocity.

In addition, the centrifugal force can produce a bending stress in the root of the blade if the geometrical centroid of the

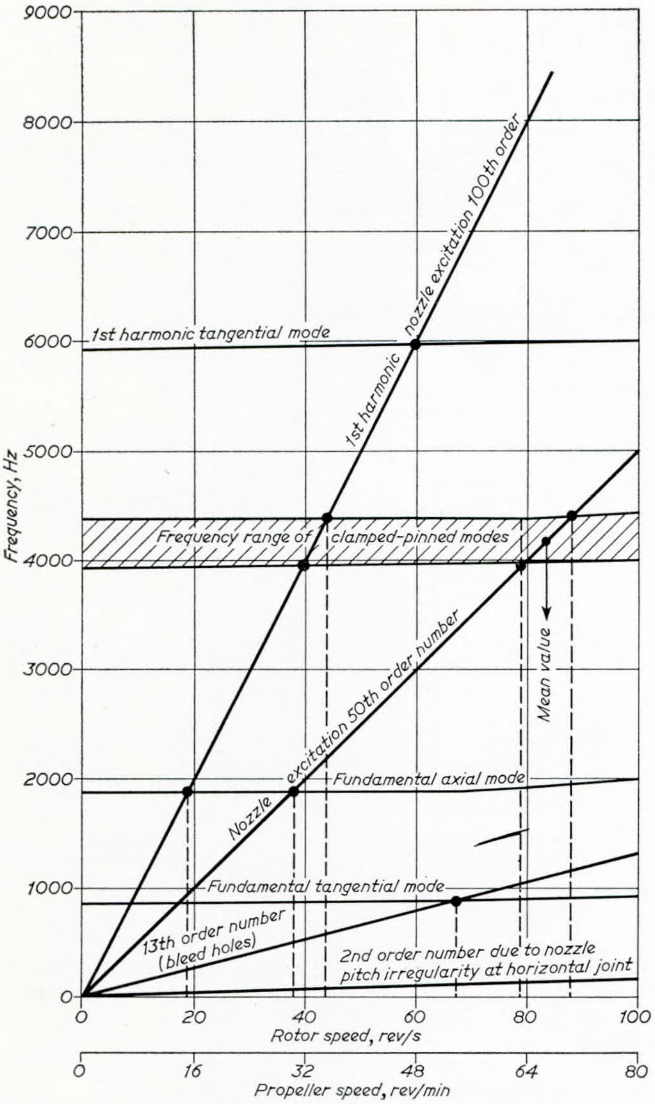

FIG. 7—*Campbell diagram (fictitious case)*

aerofoil section is offset from the root geometrical centroid, in either the tangential or axial direction, and this can be additive to or subtractive from the steady steam bending stress, depending upon which way the moment acts.

(*c*) *Vibratory force:* The partition, or wall, between each nozzle reduces the velocity of the steam flowing past it due to friction, whereas in the core of the nozzle jet the velocity is a maximum. Thus, a blade moving past the nozzles alternately experiences changes in steam velocity in addition to the steady steam-driving velocity component mentioned in (*a*).

A moving blade will *always* be subject to the alternating force in addition to the centrifugal steady steam driving force, but when the frequency of the alternating force is equal to a natural frequency of the moving blades, resonance occurs, and the amplitude of vibration of the blades is then dependent upon the amount of damping due to the three separate sources mentioned previously, and the magnitude of the vibratory force component.

The vibratory force is sometimes expressed as a fraction of the steady force, and called the "stimulus" or "excitation", *e*.

The vibratory stress at the base of the aerofoil section of the blade, is related to the amplitude of vibration of the blade, for a particular mode of vibration, and the corresponding swinging form of the blade, or blades, when vibrating at the resonant frequency. It is this vibratory stress (excluding stress concentration factors) which is sometimes quoted when dealing with blade vibration.

One way of defining the vibratory stress is to express it as a function of the steady steam binding stress, at the rotational speed of the rotor corresponding to blade resonance, i.e. $\sigma_v = \pm D.\ \sigma_{sb}$ MN/m^2 (lb/in^2) where D is the product of e the excitation fraction, and Q, the dynamic magnification factor.

Values which have been assigned[6] (in some designs) are $e = 0{\cdot}10$, $Q = 100$, whence $\sigma_v = \pm 10{\cdot}\sigma_{sb}$ MN/m^2 (lb/in^2).

The stimulus, or excitation fraction, is dependent upon many factors such as the nozzle trailing edge thickness, axial clearance between nozzles and blades, surface finish, degree of reaction and so on.

It will be appreciated that the above is a very approximate, semi-empirical expression of a much more complex relationship, which most manufacturers are well aware of and fully able to apply in practice.

Another form of the above relationship[7] includes a resonant-response factor, (K), in the value for D which takes into account the number of blades, the number of nozzles and the harmonic order number of the exciting force.

The expressions quoted are not applicable to the H.P. turbine first stage blades, nor to the long, twisted, and tapered blades of the L.P. turbine final moving rows.

The Working Environment of Blading

Temperature effects upon the vibration of a blade are:

a) the value of Young's modulus E is reduced with increase in temperature so the natural frequency will be lowered;
b) the metal fatigue limits are reduced with increase in temperature.

Some manufacturers are having to consider the effect of humidity in activating corrosive chemicals deposited on the blades as possibly one of the most significant factors in the failure of certain L.P. turbine blades in recent years, suggesting that failure may be by corrosion fatigue. These blades were calculated to have high factors of safety against failure by conventional standards (F.S. > 10). Fretting fatigue has also been known as the cause of failure of some machinery parts, including turbine blade root fixings. As a general rule the conditions for fatigue failure due to fretting in a blade root may be satisfied if the fretting occurs at an area of high stress concentration, such as in a small or sharp radius at a change of section. Like corrosion fatigue, the normal standards of factors of safety against fatigue failure no longer apply.

A further aspect of blade fatigue loading is the effect of hull fouling and roughening, and heavy weather conditions on the shaft torque developed. As the hull resistance increases, the engine revolutions at full power tend to decrease, by 3 per cent to 5 per cent in some cases, whereas, the turbine being a roughly constant power machine, the shaft torque developed increases. This rise in torque is achieved by an augment in steady steam bending force on the blades.

If there is any doubt about the safety of blades against failure, this factor should be written into the calculations, but, additionally, in all cases a direct-reading torque meter should preferably be fitted ensuring a maximum limiting torque is not exceeded during operation.

Elimination or Detuning of Resonant Frequencies

In some instances where the cause of fatigue failure has been determined in a row of moving blades, certain standard remedial measures can be considered, as listed below:

a) changing the number of nozzles;
b) skewing the nozzle trailing edges to the radial line of the blades;
c) eliminating nozzle pitch errors at the horizontal joint or round the circumference elsewhere;
d) increasing the axial clearance between nozzle trailing edges and moving blade inlet edges;
e) removing or detuning the effect of stringers or radial spokes downstream of the last stage blades;

f) increasing or decreasing the number of bled steam holes downstream or upstream of a row of moving blades, or any such arrangement which causes irregularities in steam flow around the circumference;
g) detuning the moving blades by reinforcing the blade near the root;
h) thickening the whole profile of the blades and thus having fewer moving blades;
i) installing lacing wires to inhibit clamped-pinned tangential modes;
j) providing snubbers or damping wires to limit the peak amplitudes at mid-span or at the tips.

Manufacture and Fitting

Blades which are fitted to the wheel rim by means of circumferential slots may need some hand-dressing to ensure good butting of adjacent blade roots. Fir-tree roots and variants of the fir-tree type, where a number of "lands" share the centrifugal load are carefully machined to ensure equal distribution of the load over each "land" to avoid high crushing stresses, and the possibility of fretting on the least loaded "land" surface.

Whenever hand-dressing is necessary, a careful inspection should be made to ensure that no file marks or scratches remain in way of the highly stressed areas near the root.(6) To avoid the effect of notching, leading and trailing edges should not be sharp-edged.

Shrouding should be fitted to suit any inequalities in the pitch of the tenons when the blades are in place. Bending or twisting the blades to fit the tenons in the shroud-holes should be avoided. Particular care should be taken to provide adequate fillet radii on the tenons and shroud holes.

Rough Running of Turbine Machinery

When a marine steam turbine is first run a number of tests are carried out in the manufacturer's test bay. Generally these spin tests cannot establish much more than that the machine will run satisfactorily up to its maximum speed plus about 10 per cent, coupled to the gearing train. The reason for limited tests is that normally, for large horsepowers, manufacturers do not have the steam capacity in their test bays to run the main propulsion machinery at full load.

The opportunity is taken to set the overspeed trip gear and ensure that this is operational. Checks are made of rotor balance and bearing oil film stability by measuring vibration levels, but short of ensuring that the turbine will turn smoothly when steam is applied, the tests are of little other use. The main sea trials will establish the matching of the propulsion system and propeller and whether full load can be achieved according to the design data. It is only when the full steam flow and inlet steam temperature and pressure are achieved for an adequate period that the turbine machinery can be said to have been properly tested.

During Trials

During trials, rough running will probably be associated with the effects of the high rate of heat flow into the turbine, not apparent during spin testing such as:

a) casing distortion;
b) pipe forces;
c) rotor/gearing misalignment.

(*a*) *Casing Distortion:* A turbine casing is far from being a continuous homogeneous cylinder. There are horizontal joints requiring thick flanges, a shell of varying diameter, penetrations in the shell, for steam inlet and outlet, and ends which are re-entrant and penetrated for the rotor shaft. The different masses of metal heat at different rates and distortion becomes extremely complicated to visualize. During transient conditions, such as starting up and working up to full power, casing distortion can be quite significant depending upon the rate of increase of power, especially when one considers the fine tolerances between fixed and moving parts. If a gland rub occurs, vibration of the rotor may ensue. To continue to run the turbine up to speed could be dangerous, therefore it is usual to run the set at reduced speed until the vibration is within accepted limits before running up to full power. Persistent vibration from rubbing means that some excessive thermal distortion may be taking place. It may be the presence of water in the bottom half casing possibly in an inadequately drained bled steam belt or it may be that the proper expansion of the casing in an axial or transverse direction is inhibited by sticking palm support keys, or by coming hard up against pipes and other structures, giving rise to displacement of the journal bearings relative to the casing.

(*b*) *Pipe Forces:* While H.P. casings are becoming smaller and more compact, steam pipes are increasing in size and pipe thrusts and moments becoming greater. Torque reaction is also increasing. All these factors make it difficult to prevent the cylinder from lifting off supporting palms, and to provide keying arrangements adequate to allow the casing to expand freely and yet withstand thrusts in various directions. Careful attention should always be paid to the flexibility of the pipework to the turbine, to the position of hangers, clearance of holes through bulkheads for pipes and to pipe expansion movements generally. It may be necessary for powers in excess of, say 29·4 MW (40 000 shp) to copy land practice with separate steam chests and symmetrically arranged inlet steam loop-pipes to the H.P. casing, for instance. The possible use of pressure-balanced cross-over pipes may also need to be considered in some cases.

Pipe forces which push the turbine casing upwards or sideways may adversely affect the gear shaft/rotor shaft alignment, quite apart from possibly initiating a gland rub.

(*c*) *Rotor/Gearing Alignment:* Gearing pinions and turbine rotors should ideally be in a straight line when running at the maximum design power, yet in most cases the only means to check alignment is when stationary, and cold. To run the propulsion machinery at full power for some time then shut down and check the alignment could be misleading. If the total relative movement in the vertical and horizontal direction of both the pinion bearings and the turbine bearings can be calculated, then an offset alignment can be set cold. However, the problem is one not only of parallel offset but also of angle between the axes of the shafts and this is extremely difficult to predict since one needs to know precisely the direction of movement from cold to hot of both pinion bearing centres and both rotor bearing centres.

The side of a gearbox near the turbine could reach temperatures higher than those at the aft side because of the proximity of turbine casings, gland seal pipes, cross-over pipes, etc. and thus displace the pinion bearing centres by different amounts. Similarly, turbine pedestals and seatings may not be equally heated. Typical examples of this occur when bleed pipes are located nearer to one bearing pedestal than another.

Fine tooth couplings will accept an amount of misalignment but when transmitting higher torques it is possible that couplings could lock hard and instead of being truly flexible, become a possible source of unbalance, or even exert bending effects on the pinion shaft.

After a period of satisfactory running a turbine rotor can develop vibration, and under such conditions one can state the apparently obvious—something has changed. It may be:

a) loss of blades, shrouding, lacing wire, closing pieces, etc.;
b) rubbing;
c) coupling unbalance;
d) erosion damage.

The (*a*) type of vibration is usually evident by the sudden development of vibration as would be expected. However, the loss of a piece of lacing wire or shrouding may not always result in large unbalance. Unless there is a continuous chart recording of vibration level, the sudden slight change may go unnoticed.

If vibration monitoring equipment is fitted, it may be set to alarm, or to trip, at predetermined limits, depending upon requirements.

Rubbing, as already mentioned, can be initiated for a number of reasons, but after a period of satisfactory running at or near the service speed, when all conditions are stable, gland rubbing is unlikely unless a slug of water has passed over from the boiler, and bows the shaft, or water enters the gland steam and bends the rotor, or water enters the turbine cylinder from a bled steam pipe and distorts the casing or rotor. Very often, even during stable conditions, such events can occur, but are more likely under transient conditions.

Coupling unbalance over the period of time between inspections can start from an accumulation of oil sludge in the coupling tube, or wear taking place in the coupling teeth due to inadequate lubrication, or both. Whatever the cause, the nature of the build-up of unbalance will usually be gradual.

Erosion, as already mentioned, is a gradual process and is generally fairly evenly spread over all blades, so that unbalance due to loss of metal from the moving blades may not be noticeable on the slower running L.P. rotor in terms of increased vibration.

If, however, blades have been poorly fitted for any reason so that the axial distance from the fixed blades is not the same for all moving blades then those which are nearest the fixed blades will erode more than the rest and could result in slight unbalance and a gradual build-up of vibration levels.

One detrimental effect of erosion on balance of a rotor is on the trimming weights (if fitted) placed on the disc of the last stage L.P. ahead blades. If water drips from the top half casing into the condenser (at low superheat temperatures) the balance weights will be struck by water droplets and eroded away at the exhaust end. Depending upon the distance of weights from the axis of rotation, and the magnitude of the weight, there will be a slow change in the balance condition.

The Balancing of Flexible Rotors

Critical Speeds—Historical Introduction

Since marine steam turbines are inherently variable speed machines, the question of critical speeds is an important consideration.

It was once common to specify rotor critical speeds as occuring not less than 125 per cent[8] of the design speed. It was thought that in this way rotor criticals could be dismissed as an item of concern in design.

However, it became apparent that the as-built efficiencies were poorer than could be obtained with more moderate shaft sizes and that the internal efficiency of the turbine fell away rapidly because of leakage losses resulting from a high rate of wear of the steam sealing arrangement under abnormal operating conditions. The heavier shafts were more liable to thermal distortion such as bowing, and differential expansion. More important, perhaps, there came the realization that the 125 per cent rule did not ensure that the first critical speed was above the design speed. As better methods of measurement became available, it was shown that bearing supports were far from rigid as had been assumed in calculations, and that the measured critical speed was always below the calculated value, as seen in Fig. 8 which is a plot of many such measurements. As can be seen, it is impracticable to place an actual critical speed above the running range, unless turbine speeds are greatly reduced from those of long-standing practice.

For example, high pressure turbine rotors are commonly designed to run at about 6500 rev/min at service speed. The 125 per cent rule would call for a calculated rigid bearing critical speed of 8125 rev/min. From Fig. 8 it would actually be between 4000 to 5000 rev/min or 75 per cent of the service speed. Again, efforts to stiffen the shaft would only result in moving the critical speed nearer to the service speed, but never beyond it.

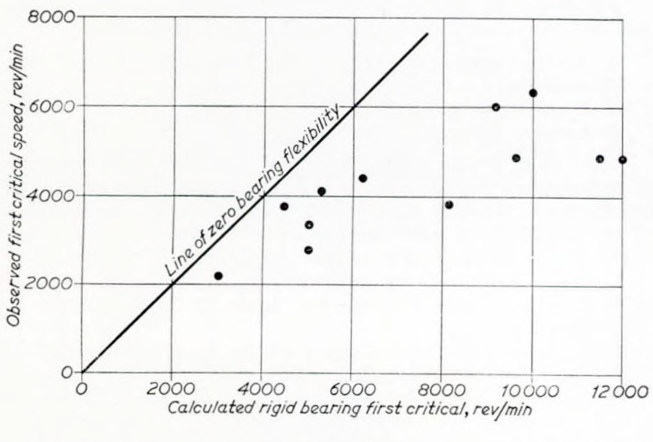

FIG. 8—*Observed first critical speeds of turbine rotors*

The reason for the depression of the observed critical speed is, of course, the flexibility of the supporting structure of the bearings and that of the oil film. Contrary to the early methods of calculation, the shaft bearings cannot be considered as nodal points, since they do in fact deflect and the nodes will be outside the bearing span.

Another important characteristic of bearing performance is the strong damping effect of the oil film. It is this damping which has in most cases permitted trouble free operation at the critical speed. Thus the 125 per cent critical speed rule was for many years thought to be achieving its objective, since no obvious critical speeds were observed during operation. It was only by deliberately fitting a large unbalance to the stiff rotors of those days that the presence of critical within the operating range was found.

Balancing of Flexible Rotors

Unlike the solid, or drum-type rotors of earlier years, rotors today are more flexible, with wheels (or discs) integral on small diameter shafts (sometimes called gashed rotors) and are termed over-critical (or super-critical) which means that the operating speed is higher than the first critical speed. When the operating speed is below the first critical speed they are termed under-critical or sub-critical.

The results[9] of passing through critical speeds are:

a) The vibration amplitude increases and starts to change in angle relative to some datum point on the rotor. As the critical speed is reached, the vibration peaks are about 90° out of phase with the force on the bearings. As the critical speed is passed, the phase angle increases to 180°.

   In effect, the heavy side of an unbalanced rotor swings inwards towards the axis of rotation as the shaft comes near and passes the first critical speed.

   The light side of the rotor will then be on the outside of the deflexion curve until the second critical speed is passed when it will once again turn through 180° so that the heavy side is on the outside.

b) Although, in the absence of damping, theory shows that the vibration amplitudes increase to infinity at the critical speed, bearing oil film damping forces prevent this from happening.

c) Between the first and second critical the rotor tends to rotate about its mass centre, so that vibrations will tend to decrease.

d) Because of the rapid change of vibration phase and amplitude at a critical speed, a rotor running at the critical speed is extremely sensitive to small changes in conditions such as speed and temperature. Consequently, it is difficult to balance a rotor at the critical speed. Clearly, critical speeds should always be designed away from manoeuvring speeds, wherever possible.

*One Method of Achieving Balance**

One method used for the balancing of flexible rotors which gives satisfactory results is briefly as follows. The rotor, Fig. 9(a), is balanced as a rigid rotor in a slow speed balancing machine with force and couple balance weights added, Figs 9(b) and 9(c). The rotor is run near to its first flexural mode critical speed and weights are added as illustrated in Fig. 9(d). The effect of these weights is to compensate for the modal unbalance, but not to disturb the rigid rotor force balance of Figs. 9(b) and 9(c). If necessary the rotor is then run near its second critical speed and weights added as illustrated in Fig. 9(e). The effect of these weights is to compensate for the modal unbalance, but will not disturb either the couple balancing of Figs. 9(b) and 9(c), since the resulting moments about the centre line are equal to zero, or the balancing of Fig. 9(d).

* Federn's Method.

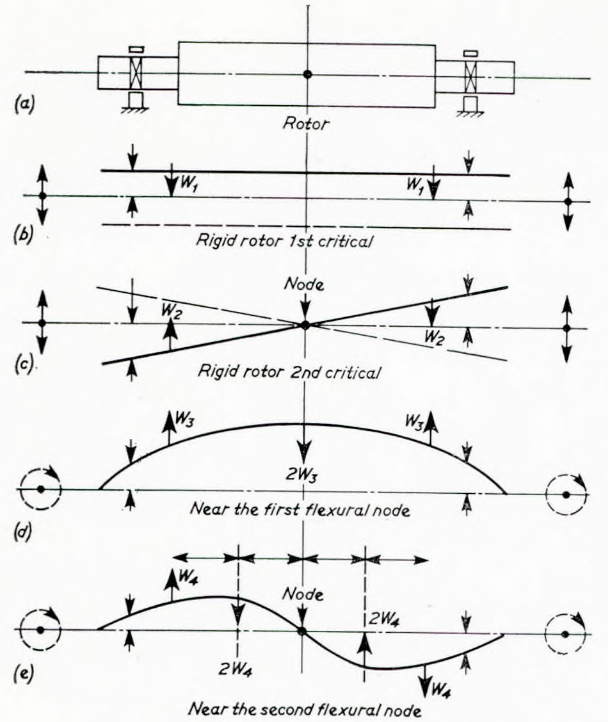

FIG. 9

The object of balancing the flexural modes is to reduce the deflexion of the shaft at the critical speeds by balancing out the forces which produce the deflexion, or in effect to reduce the corresponding bending moment diagram to a negligible amount.

Most manufacturers would probably agree that modal balancing should be carried out when the rotor is in the turbine on its true supports, and when the rotor has attained a steady temperature. This is difficult to achieve, however, because of the need to lift the top-half casing to be able to put weights in some other balancing plane.

Another problem is that of running the rotor at high rotational speeds, due to windage problems, unless a partial-vacuum chamber is constructed.

Oil Whirl

Rotor vibration can also be caused by what is termed "oil whirl" or "oil whip" (though modern turbines usually have short, highly loaded bearings to avoid such phenomena).

Oil film whirl is a self-excited vibration which is characterized by a circular or elliptical motion of the shaft journal within an oil film lubricated journal bearing, generally occurring at the shaft half-order frequency (one complete cycle for every two revolutions of the shaft).

The problem of oil whirl is not a clear-cut phenomenon, and although many papers have been written on the subject, not all authors agree on the actual mechanism. It is quite possible that a number of mysterious cases of rotor vibration in recent years may have been caused by oil whirl, despite the provision of short, highly loaded bearings.

Some manufacturers fit anti-whirl bearings as standard equipment in modern turbines, though the type fitted varies from one manufacturer to another.

Vibration Limits in Marine Steam Turbines

Most manufacturers of marine turbines have adopted a limit of vibration for the "installed" set, though there is no apparent conformity of limits between different manufacturers.

The "V.D.I. Recommendations"(10) (2056, Group T) appear to have been the basis of some of the limits employed.

There are some differences of opinion as to whether amplitude of vibration of the rotor journals relative to the bearing, or vibration velocity (r.m.s.) measured on the bearing keeps should be the criterion.

Again, there are suggestions that "acceleration" should be the criterion for measurement of vibration levels. However, H. G. Yates(11) in 1949 showed clearly that velocity is the most suitable unit of vibration measurement, which "will result in similar values for all machinery of the same type, and constructed with the same degree of precision".

It is perfectly valid to specify an amplitude of vibration, provided it is applied at a particular rotor speed, such that the equivalent velocity of vibration is within acceptable limits.

The scope of the paper does not permit a lengthy discourse on the subject, but there is clearly a need for research into acceptable levels of vibration, to safeguard main steam turbines on the one hand without being overcautious on the other.

A maximum allowable vibration velocity of about 16 mm/s (r.m.s.) is probably a reasonable limit, preceded by a warning signal at about 10 mm/s (r.m.s.).

It is debatable whether a maximum allowable level of vibration should automatically trip the main turbine, or reduce speed under automatic control, since this is largely dependent upon the limit chosen for the maximum allowable vibration.

A suitable period of operational experience of modern marine steam turbines should provide a satisfactory solution to the problem in due course.

Measurements and Limits of Vibrations

Terms and Definitions(10): Vibration velocity V in mm/s has been adopted as the standard form of measurement for vibratory levels in steam turbine machinery.

In the case of sinusoidal vibrations where $V = \hat{V} \cos \omega t$ and also in the case of vibrations consisting of several components at different frequencies, the root-mean-square value (Veff) of the vibration velocity V is taken as the characteristic value for the vibration strength.

The value Veff can be measured directly by electrical instruments provided with a square law characteristic of velocity indication.

If the non-harmonic, periodic or non-periodic vibrations are in recorded form, the effective value of the velocity must be formed as a root-mean-square value of the instantaneous value $V(t)$ according to the following formula:

$$\text{Veff} = \sqrt{\frac{1}{T}\int_{0}^{T} V^2(t)\,dt}$$

If the vibration trace has been analysed into components of amplitudes $\hat{S}_1$, $\hat{S}_2$, etc., with corresponding angular frequencies ω_1, ω_2, etc., then

$$\text{Veff} = \sqrt{\tfrac{1}{2}\,(\hat{S}_1{}^2\omega_1{}^2 + \hat{S}_2{}^2\omega_2{}^2 + \text{etc.}}$$

$$\text{or} \quad \text{Veff} = \sqrt{\tfrac{1}{2}\,(V_1{}^2 + V_2{}^2 + \text{etc.}}$$

In practice a recorded vibratory trace may show a predominantly single sinusoidal wave form and a fair assessment of the velocity of vibration may be obtained by measuring this predominant wave form amplitude and frequency from the trace.

$$\text{then Veff} = \frac{1}{\sqrt{2}} \cdot \hat{S} \cdot \frac{(2\pi N)}{(60)}$$

where N = frequency in cycles/min

$\hat{S} = \frac{1}{2} \times$ peak-to-peak trace width in mm (the single amplitude in mm)

Measuring Instruments: The vibration measured on the bearing keeps can be indicated or recorded with either mechanically or electrically operated instruments.

Instruments with linear characteristics such as linearly operating surface-contact rectifiers can produce inaccurate readings in the case of mixed frequency vibrations. These should be

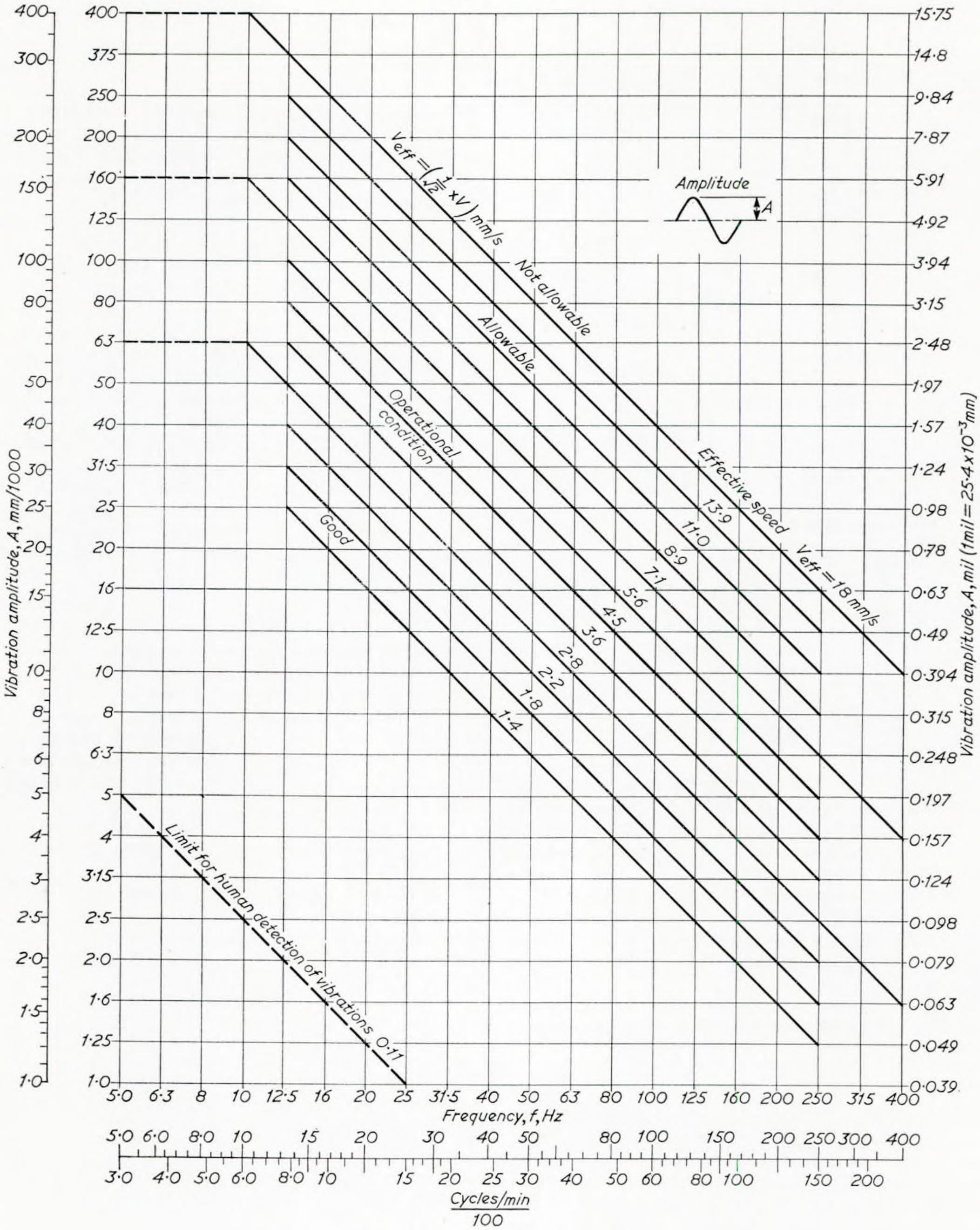

FIG. 10—*Main turbine vibration levels*

used only if there is a predominantly sinusoidal wave form having small mixed components.

The linear working frequency range of measuring instruments must be wide enough for all the frequency components which are needed for the evaluation of the vibration level to be indicated without distortion of amplitude. Its range of modulation and environmental stability must correspond to the measuring conditions in question.

To exclude the possibility that vibrations normal to the direction of measurement will distort the indication of the vibrations in the direction of measurement, the vibration pick-up must have low cross-sensitivity.

The error limits of each individual measurement are in addition affected by the coupling between the pick-up and the object being measured, and also by the reaction of the pick-up on the object. A mass ratio of 1/10 should in no circumstances be exceeded.

In the case of steam turbines the points of measurement should be on the bearing housing (keeps) in the horizontal direction at a height corresponding to the shaft centreline, vertically above the centre of the shaft centreline in the middle of the housing, and axially parallel to the shaft axis at shaft height.

The graph Fig. 10 is taken from the V.D.I. Directive 2056 of October 1964, machinery group T, and is generally considered to give the most suitable limiting vibration levels in marine steam turbines to date.

Bands of vibration levels are denoted good, operational condition, allowable and not allowable, which are terms which take into account the three quantities, frequency, amplitude, and velocity of vibration.

REFERENCES

1) PLATT, E. H. W. 1970. "Operation, Maintenance and Manning Problems in Large Tankers". *Trans. I. Mar. E.*, Vol. 82, p. 425.
2) C.E.G.B. "Modern Power Station Practice." *Chemistry and Metallurgy*, Vol. 5.
3) KEOGH, F. G. 1970-71. "On-load Cleaning of Steam Turbine Blading". *Proc. I. Mech. E.*, Vol. 185, Steam Plant Group, Paper 15/71.
4) JONES, T. P. 1972. "Design, Operating Experience and Development of Main Propulsion Epicyclic Gears". *Trans. I. Mar. E.*, Vol. 84, Part 15.
5) KANTOROWICZ, O. P. T. 1962-63. "Of Steam Turbine Wheel, Batch and Blade Vibrations". *Trans N.E.C.I.E.S.*, Vol. 79, p. 51.
6) FLEETING, R., and COATS, R. 1970. "Blade Failures in the H.P. Turbines of R.M.S. *Queen Elizabeth* 2 and Their Rectification". *Trans. I. Mar. E.*, Vol. 82, p. 49.
7) WEAVER, F. L., and PROHL, M. A. "High Frequency Vibration of Steam Turbine Buckets". *Trans. ASME.*, Paper No. 56-A.119.
8) ROHDE, E. C. 1960. "Design Aspects of Modern Marine Propulsion Turbines". SNAME, Paper presented at the annual meeting, New York, 17-18 Nov.
9) LAST, B. P. 1964. "The Balancing of Flexible Turbine and Generator Rotors". I.Mech.E. Student/Graduate Paper, Viscount Weir First Prize.
10) *Verein Deutscher Ingenieure*. Oct. 1964. Recommendation 2056, Machinery Group T.
11) YATES, H. G. 1948-49. "Vibration Diagnosis in Marine Geared Turbines". *Trans. N.E.C.I.E.S.*, Vol. 65, p. 225.

ACKNOWLEDGEMENTS

The author would like to thank the Committee of Lloyd's Register of Shipping, and Mr. B. Hildrew, M.Sc., C.Eng., the Technical Director, for their kind permission to publish the paper.

He was indebted to Dr. S. Archer, C.G.I.A. (h.c.), C.Eng., for his advice in preparing the paper and for the many hours of his own time spent in editing it.

Most of the contents were taken from a larger paper presented to the Lloyd's Register Technical Association in February, 1972, therefore he should like to thank all the members of that Association, especially the President, Mr. A. R. Hinson, C.Eng., Principal Surveyor, and also to Mr. C. Chartan, B.Sc., C.Eng., Principal Surveyor, whose persistent encouragement had led him to present the paper to the Institute.

Finally, he should like to thank all his colleagues within the Society, and his many friends outside connected with the marine industry, who had contributed to his knowledge over many years.

Discussion

MR. R. E. BURN, M.I.Mar.E., opening the discussion, said that as a member of the only British company designing and manufacturing marine steam turbines he was grateful for this opportunity to comment on Mr. Donald's paper. A lot of useful data was provided but also the paper showed the steam turbine to be coping very well with the requirements of the industry. Mr. Donald, with the experience gained from his long service with Lloyd's, was providing a great service to the steam turbine protagonists by pointing out areas where problems had arisen. Mr. Burn thought he was saying—"For goodness sake don't lose out to the Diesel brigade because of faults which should be easily avoided".

The introduction illustrated the current trends for the higher powered ships and the proportion engined with steam turbines. Some exception might be taken to his conclusion in that whilst agreeing that the future of steam turbines would depend on their satisfactory operation, it was necessary to compare the overall performance of these engines with that of the main competitors in the form of direct coupled and geared Diesel engines and, to some extent, with gas turbines. Although problems did arise with steam turbines he had noticed a re-awakening of interest, for some applications, in steam turbine propulsion for lower powers, currently wholly supplied by geared Diesel engines. This interest appeared to be associated with owners who operated ships over the greater part of their useful life and who were interested in the long term economics, as compared to the purchasers of ships which were put out on charter, in which first cost and short term economy appeared to be the main consideration.

Mr. Burn said that there were several design aspects mentioned in the paper on which he would like to comment. Lagging tended to be a neglected area and this was inexcusable with the excellent modern materials available which were both durable and highly effective as insulation.

Erosion of blading should be fully considered during the design stage and extensive research and investigation had led to the development of methods of forecasting when this phenomenon would be of significant proportions. The factors considered included tip speeds, water droplet size, blade, diaphragm and casing geometry.

In addition EBW of protective stellite shields was replacing the less reliable brazing methods previously used. He wondered whether, in cases of erosion of astern blading, this had resulted due to a lack of properly designed steam deflectors in the common exhaust chamber of underslung condenser designs.

Labyrinth gland rubs could affect the reliability and efficiency of the turbine and whilst the use of spring-backed leaded nickel bronze and stainless iron glands appeared to be standard practice and caused very little trouble he felt that there was still room for development in this area, for example, the use of modern designs of carbon glands which were far more amenable to radial rubs without causing either vibration or loss of efficiency. As a result of these beneficial qualities they could be designed to run at very high rubbing speeds and with very much smaller clearances than with traditional materials.

The author had stated that axial clearances of 0·25 mm or less were typical in the initial stages of marine steam turbines. Mr. Burn suggested that to operate a steam turbine with axial clearances as tight as 0·25 mm placed an unnecessary hazard on the reliability of the machine. Where a low degree of reaction was designed into the moving blade, axial clearances of 0·75 mm could be adopted as minimum and would result in a small, and hence acceptable, increase of tip leakage loss. His company used this approach on all standard frame marine turbines.

Vibration characteristics could dictate that a lacing wire was required to inhibit low order, in-phase tangential vibrations and that the wire formed no more than three or four "packets" of blades. In this case brazing the wire to the runner blades was not practical and could lead to broken joints or cracked lacing wires. For many years Mr. Burn's company had used a lacing wire tube crimped, by means of a controlled pneumatic process, between the blade forms. This successful solution gave the necessary damping without the hazards introduced by the brazing process which in this application was very difficult to control, and could result in a reduction in properties for the lacing wire and blade due to too high a temperature being reached.

The warping of horizontal joints could only be the result of very high thermal stresses. Modern photo-elastic stress analysis techniques had led to the development of design methods that had improved the geometry of horizontal joints, particularly on the HP cylinder, which gave much improved thermal symmetry with the remainder of the casing (Fig. 11). Two factors commonly overlooked by maintenance staffs when dealing with high pressure high temperature joints were: the capability of using existing mechanical aids to control tighten all high temperature bolts and the flagrant misuse of jointing compound that was mistakenly thought to provide a steamtight seal.

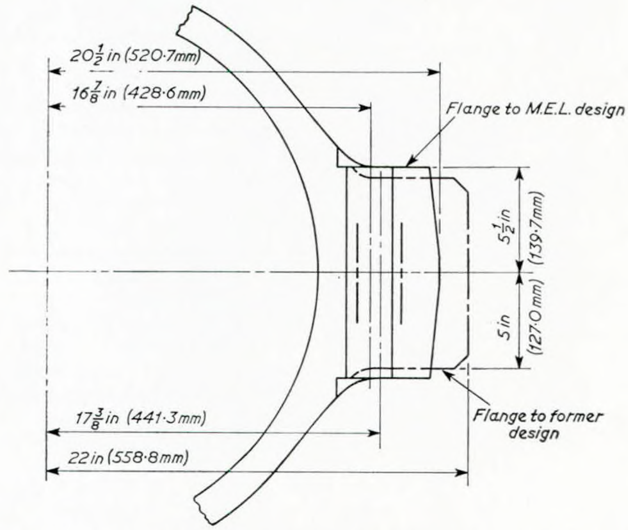

FIG. 11—*Flange example, comparison between M.E.L. and former design, for same duty*

In following a policy of producing robust turbines it was advantageous to design the rotor with discs having sufficient gyroscopic stiffness to nullify completely the effect of wheel vibration modes upon the basic characteristics of the moving blades. This design process could be carried out with complete accuracy and ensured that the frequency of all nodal patterns were well above blade modes in the operating speed range. This simplified the blade vibration analysis with the consequent improvement in reliability factor.

It was not entirely true to say that the effect of root fixing was unknown when calculating resonant frequencies. Whilst this unhappy state of affairs did exist when some of the fixings illustrated by Mr. Donald were used, other roots enabled a very close correlation to be obtained between calculated and actual frequency of the blade when installed in the turbine. Mr. Burn's company considered it an essential design requirement that only the latter type should be used for marine turbines. Full scale wheel case test chambers had existed for many years whereby the dynamic frequencies of various types of blading and root fixing had been accurately measured and compared against theoretically computed frequency levels. This type of research had led to the exclusive adoption of the two or three land-straddle and multi-fork root fixings for marine turbines and the company could thus confidently predict all the major resonant frequencies to within five per cent of the actual operating levels (Fig. 12).

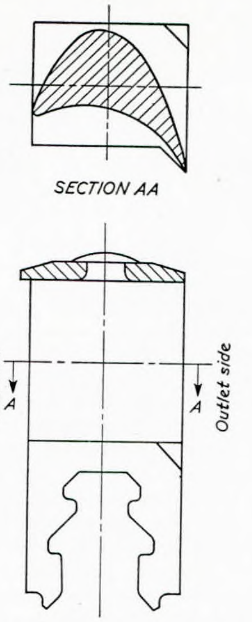

FIG. 12—*Typical teardrop section blade and fir tree root*

Problems associated with turbine to gearing shaft alignment had largely been eliminated in recent years by careful design of the support and keying arrangements within the turbine. Calculations taking into account all known external forces were carried out to ensure that no key was excessively loaded and that, at all conditions, at least 50 per cent of the natural casing CG force remained on the turbine support palms. Keying arrangements ensured the maintenance of axial alignment under all operating conditions. Membrane type couplings between turbines and gearing, capable of transmitting large powers and accepting mal-alignment without detriment to flexibility, had also been adopted.

Within the last few years the subject of rotor critical speeds had received considerable attention and the introduction of large computers and new computation methods had allowed an analysis in detail of rotor dynamic behaviour—taking into account the effect of the individual bearing design and its inherent oil film flexibility. It was possible to produce rotor response curves showing peak to peak amplitudes at given speeds, and also to introduce an out of balance component into any part of the rotor and plot the resulting resonance curve. Thus the relative merits of different critical speeds could be analysed and this technique was particularly useful in showing effective balance planes for the reduction of vibration amplitudes, as shown in Figs. 13 and 14. An *in situ* check balance of a flexible rotor at running speed and service temperature was now possible, and any necessary correction could be applied without removing the turbine cylinder top half.

To improve the accuracy of such critical speed computations it was advantageous to directly support bearing pedestals from the ship's seatings. This policy could represent an important factor in the overall layout of, particularly, the LP turbine in a cross-compound machine.

A. R. HINSON, A.M.I.Mar.E., referred to the use of steam turbine machinery in container ships of high power, and to the author's statement: "This is a market for the highest power steam turbines afloat".

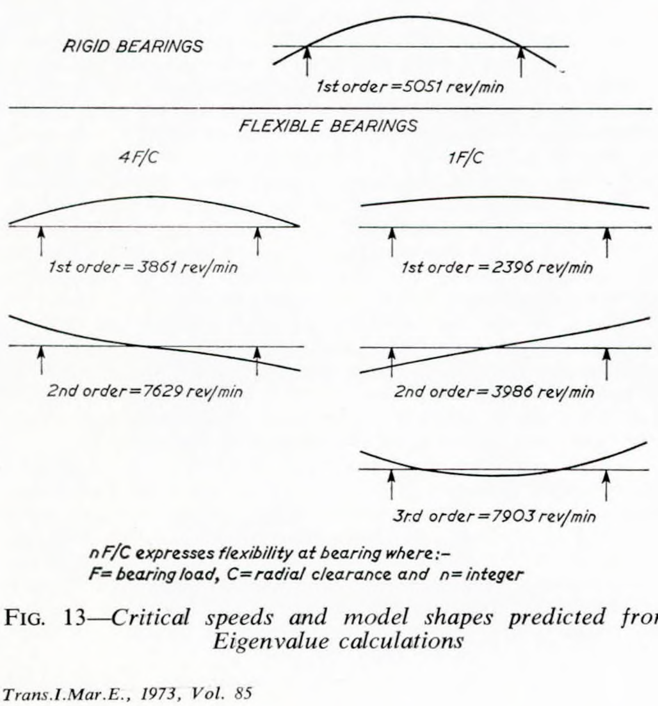

FIG. 13—*Critical speeds and model shapes predicted from Eigenvalue calculations*

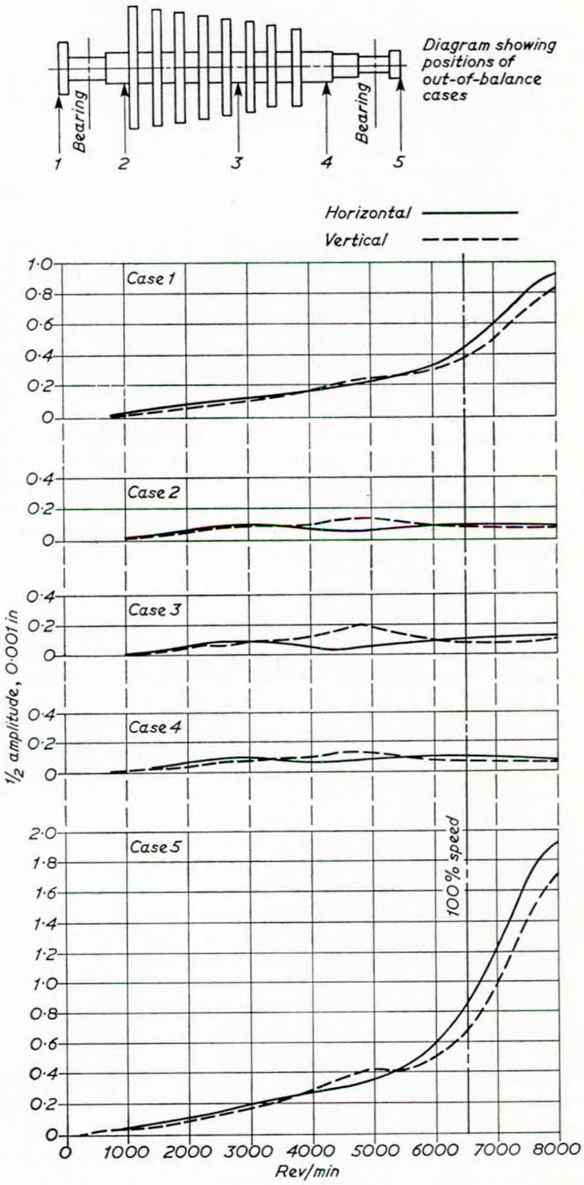

FIG. 14

A fortnight ago in this Institute we were privileged to hear Messrs. O'Hare and Holburn present a paper*, on the opening page of which the authors wrote: "in the total system economics, the highest return on investment would not necessarily be obtained from the installation which offered the best propulsion efficiency, lowest maintenance cost or lowest fuel rate".

The paper then went on to discuss service experience with twin screw gas turbine container ships and mentioned that a complete power plant—gas generator and free turbine could be replaced in 18 hours.

The price for that flexibility would be paid in fuel.

Steam turbine designers had always been aware that reliability could be increased at the expense of fuel economy and at least one article in the technical press had made a case for a simple steam plant with moderate steam conditions. Increased reliability in steam turbine plants would be obtained when turbines were free of the problems the author had mentioned. Mr. Hinson then discussed one of these problems, namely, blade failure caused by vibration.

The standard method of dealing with vibration problems was to reduce the exciting force and move the resonant frequency away from the operating speed.

Various methods were given to calculate natural frequencies of a blade or groups of blades, but none seemed to be very accurate. There were too many factors which could only be approximately estimated. There were several reasons for this uncertainty, the most important of which were that the fitting of the blades in the rotor, the brazing of the lacing wires and the fitting of the shrouding were manual operations.

When vibration occurred in shrouded blades, failure of the shrouding was probably the first sign of trouble. When the blade had a thin cross-section the tenon could fail in bending fatigue. With thin blades the tenons usually had a rectangular cross-section in order to get sufficient cross-sectional area. The holes in the shrouding were, therefore, rectangular and the stress-raisers at the corners of the holes could be the origin of fractures. This was likely when the shrouding was crushed by excessive force when the tenon was peened over.

Would the author give his opinion as to the accuracy with which it was possible to determine resonant frequencies of package blade vibration?

With regard to the measurement of turbine vibration on the bearing keeps mentioned in the paper, it appeared that the sensitivity of this method depended to a certain extent on the stiffness of the bearing seatings. A more sensitive method employed a proximity pickup at or near an antinode on the shaft between the turbine and gearing. The author's comments on this would be appreciated.

MR. J. M. CRUIKSHANK said his remarks were based mainly on hindsight, and observations arising from the results of investigations into failures. Having jotted them down, there appeared to be a common thread running through the whole business, i.e. the flexible rotor, and it now seemed to be creating one of the major niggers in the woodpile. Simple things such as increased gland clearance could throw turbine alignments out very greatly, and it illustrated the situation shown by Mr. Burn of what could happen as a result of out-of-balance couplings, which was usually where malalignment appeared—there would be a number of people who would not sleep easily if they knew they had badly worn flexible couplings; so attention to glands and leakage in that area was important. Perhaps some designers should look at prevention of any leakage which occurred in that area because of the adverse effects it could have.

Again on the question of glands, when there was a nickel-iron content in the gland packing material, it could have unfortunate results in that it acted rather like a machining device on the rotor when it touched the glands.

With regard to casing distortion, Mr. Cruikshank had found there was a tendency for the casings to hog. It could be a transient one, or one which was induced during the manufacturing process. Perhaps the author could say whether the method of boring bolt holes in the casing flanges had been standardised, in that they were rough bored before the casings were permanently stress relieved.

Mr. Hinson had referred to the question of tenons, and pointed out that the state of the hand fitting was an imponderable. Too often, in the mass production of stamping out of shrouding strips, one found a fitter on the job using a file, when nobody was looking, on the root of the tenon to achieve a fit. That was unfortunate.

Mr. Burn's remarks about the effect of out-of-balance forces at the end of the rotor were just as important in that connection. The importance of proper bearing design, including some method of damping, was paramount in order to achieve minimum dynamic magnification of the rotor amplitude.

With reference to vibration transducers, the type of transducer should be carefully selected. If it were not of the positional type measuring the distance between the bearing and the probe, and was one measuring vibration only, it must cut off the critical frequencies coming from the hull vibration. Therefore any equipment must be insensitive to hull frequencies of below 10 Hz.

On the question of soundproof control rooms, most of the large turbine ships would be sailing UMS where there was nobody asleep in the control room. It was essential with flexible rotors that in the bridge control equipment there should be some dead band, or an area through which speed was taken up or down, so that the turbine could not run continuously within the rotor critical. Previously everybody concerned had been used to the fact that there was a normal barred range due to the gearing criticals, and one did not run the equipment in those revolutions, but now it was very important that the criticals in the rotors themselves, especially the H.P., should be recognised in the control equipment design.

The author remarked that steam turbines were acting as compressors. Another possible dangerous factor was leakage from the manoeuvring valves into the astern casing, which had a detrimental effect on blade life—even small quantities and safeguards should be installed.

DR. A. W. DAVIS, M.I.Mar.E., said the author had made an extremely useful collection of the ills that could beset steam turbines, and there would be a healthy widening of education on the subject as a consequence. Insofar as it added to the effectiveness of the operating personnel and surveyors it was all very admirable but, where it reflected on shortcomings of design, it showed a rather sorry state of affairs. There should not be so broad a scope for such a presentation two-thirds of a century after the steam turbine first made its dramatic and successful appearance.

Allowing that powers continued to rise, particularly relative to propeller speeds, the characteristic remained that a large proportion of the power range at the lower end was vulnerable to competition—larger in Europe than America. Now, however, it was also to be challenged at the upper end of the scale by the gas turbine, aided by the dexterity with which the latter could be given a low profile for its capital charges on spare units. This situation had been largely aggravated by enthusiasm for economy, involving complications of unproven worth, often in equipment ancillary to the main turbines, and in the boilers sometimes by an addiction to size reduction.

Much interest was aroused by the proposals referred to regarding the adoption of a simplified steam cycle embracing, among other features, a steam temperature of 482°C (900°F), the reduction being to promote better boiler reliability on the basis of a study of American operating records which had shown a sharp adverse turn when that temperature was exceeded.

For a long time the reliability of the turbine itself was its greatest asset. However, for the reasons the author had given, that could no longer be accepted without question. Apart from the factors introduced by the application of ancillary features and the continuing ailments to which boilers were prone, lightness of construction, aimed at minimizing capital cost, had contributed significantly to failures arising from misalignment. In the last third of the twentieth century one did not expect to find turbines

---

* O'Hare T. L. R. and Holburn J. G., 1972, "Operating Experience with Gas Turbine Container Ships." *Trans.I.Mar.E., Vol. 85, p.1.*

lifted from their seatings by known forces of uncalculated magnitude which was, of course, the kind of situation that could be achieved by the designer if he was given sufficient encouragement to reduce the cost of the product at the expense of quality.

Some failures arising from blade vibrations were difficult to predict with certainty but, in able hands, they could be forecast to the extent of knowing if some zone of uncertainty was, or was not, being encroached upon. In higher powers an element of uncertainty could sometimes only be avoided by the adoption of double-flow L.P. turbines, but at a cost that made competition with blythe spirits difficult.

There had been reference to lacing wires and their difficulties. The real way to overcome that problem was not to fit any lacing wires. Side entry blades had a great deal to recommend them, but when one looked at the sectional views of side entry blades given in the paper the question could be raised as to how they were prevented from sliding fore and aft. Some methods of stopping such movement were completely satisfactory and others were less so but cheaper to provide.

The author had dealt with the subject of choked drains, and in the course of his remarks he perhaps put the designer in a more favourable light than was sometimes deserved. The generous provision of drains had become a less desirable subject of discussion in these days of remote control of machinery. The satisfactory remote control of drainage systems was costly and provided more sources of possible trouble. A paper devoted wholly to this subject could be more erudite than its title might imply.

DR. J. COWLEY, M.I.Mar.E., said he welcomed the paper, especially as at one stage he had thought that the Institute would not have another paper presented on steam turbines. That was ten years ago when motor ships appeared to have the field all to themselves. Even so, at the present time turbines were not being built for installations of less than about 14·7 MW (20 000 hp). To compete with motor ships, steam turbine designers had had to resort to complications such as reheat, as mentioned by the author.

From practical considerations marine installations were restricted to single stage reheating. The gain in efficiency was therefore limited as it depended both on the amount of heat which could be added during the reheating process and the average temperature at which that heat was added. For optimum efficiency it was found that the reheat pressure must be relatively low, and calculations showed that it was about 20–25 per cent of the boiler pressure. Unfortunately this meant that the specific volume of the reheated steam was about five times as great, and the large volume of steam in the reheater and steam lines was distinctly embarrassing when unexpected manoeuvres were required. Ships operated in the channel with reheaters in use and it would be interesting to learn from the author whether he had any figures for the amount of time it took, not for the vessel to go astern, but for the engineers to be in a position to put the ship astern. It seemed that a very long time would be required.

The second point related to periodically unmanned machinery spaces, i.e., ships with the UMS notation. The author had mentioned that on one ship, turbine vibration had wakened off duty engineers but had passed unnoticed by the watchkeeper in the control room. It had been suggested that a ship was safer if the engineers were asleep in bed rather than asleep in the control room! However, the figures available did not indicate that shipping companies shared that view as, of 131 UMS ships, 122 were propelled by Diesel engines, three by gas turbines and six by steam turbines. It would seem that steam turbine operators were either being very timid or there must be some other reason for their reluctance to operate steam ships in the UMS condition. It might be because of the long distances involved in these large ships or it might be timidity; in view of the fact that steam ships had been automated for many years and as they would not operate without automatic equipment, one wondered why owners continued to use traditional watchkeeping in preference to UMS operation. Perhaps the author would comment on this practice.

MR. A. N. S. BURNETT, M.I.Mar.E., suggested that the author was a very brave man to make the statement that there was going to be a large market for the steam turbine so far as OBO ships were concerned, and to infer later that the power plant in liquified gas carriers would increase in power, therefore providing a market for the steam turbine. Mr. Burnett questioned these assertions which were, in his view, dangerous assumptions.

The outstanding fact emanating from detailed market research and other survey by his company, was that the answer was never what one thought it might be when the exercise commenced. There were a number of new ideas now available which had not been mentioned in the paper and which would affect the situation in the market place. The curve for these ships shown by the author might well take a steep dive if a number of these items were taken into serious consideration. Mr. Burnett advised anybody who was interested in marketing new or existing equipment to carry out detailed studies before launching their equipment, with any certainty, on to the marine market.

With reference to the question of reliability, it might be preferable to call it availability. In the case of some VLCCs, the cost of downtime—or out of service costs per day—might well be between £30 000 and £40 000. That was the cost to the owner when such a ship was unavailable. The ship owner should be interested in establishing whether the downtime was caused by a boiler or turbine failure or from any other cause. There were systems on the market, some computerized and some not, which could make valuable use of that information only if detailed and accurate records were kept. If such records were kept in proper detail and the fault could be attributed, for example, to the turbine, then the owner had the basis of a cost assessment of what cost effective action could be taken, even if the answer was that no action would be worthwhile for that particular ship.

These realistic ideas were used in the aircraft industry every day. This was how such aircraft as the VC10 could turn round now in half an hour instead of three or four hours as previously. When an aeroplane failed to leave on time somebody had to write down exactly why. The information went to the technical department who fed it into computers or an assessment system. It was then compared with existing data, and the appropriate equipment designer was called in at the very earliest moment. The designer welcomed this because he became aware of the kind of cost factor which owners were looking for. He could then use the information to rectify the defect where appropriate. This method was not used to a great extent in the marine industry.

Mr. Burnett appreciated that large international oil company tanker owners already operated Kalamazoo or other methods of assessment of defects. This action did not apply to all VLCC owners, two-thirds of whom were independent operators, and the Kalamazoo system was not fully cost effective in his opinion.

Regarding automation and UMS, there were quite a number of turbine driven VLCCs now under construction which would be operating UMS, and numbers would increase quite considerably over those referred to in the paper. In this connection, reference had been made to complication and simplicity, which led to the further point that if ship owners could assess accurately which equipment or system had caused the failure, then certain parts of many marine automation systems might disappear overnight as the cost effective value could make certain parts of the system unattractive. It would be of interest to know whether the author had any information on the average number of days turbine driven ships were out of service where the turbine was the prime cause, as opposed to the boiler or condenser or any other part of the machinery plant.

With reference to the bending of rotors, this often occurred in the manoeuvring part of the overall ship operation. A number of automatic control systems were in service which took care of this cycle, and it would be interesting to hear the author's view whether such equipment did in fact prevent the possibility of bending rotors during manoeuvring; or did one have to face the fact that, in the future, there would probably be many more bent rotors?

Correspondence

MR. T. E. NOBLE, A.M.I.Mar.E., in a written contribution, referred to the section of the paper dealing with steam nozzle impulse excitation of blade vibration and asked if this form of excitation could be applicable to the second moving row of blades in a velocity compounded stage—as the steam must pass the first moving row blades and stationary blades before reaching the second moving row.

He asked the author's opinion concerning the major modes of response of a shrouded "packet" of blades to the shock effect of partial admission.

MR. R. LOVELL, A.M.I.Mar.E., in a written contribution, stated that during the discussion a speaker had asked for some figures on the reliability distribution of the component parts of a steam system. This could be answered by referring to a question given in the engineering knowledge paper of the extra first class exam of Jan. 1972, which stated: " . . . an analysis of the areas of unreliability in marine steam turbine systems gives the following distribution:

boilers	49 per cent
condensers and circulating pumps	12 per cent
boiler feed pumps	11 per cent
main turbines	7 per cent
all others	21 per cent . . . "

MR. C. H. VERITY, B.Eng., M.I.Mar.E., wrote that the importance of maintaining shaft alignment and the correct orientation or meshing of the rotating elements in the gearcase was of particular interest to him at the present time. Thus, it was gratifying to note that the author had not only made special mention of hull deflexion and its effect on machinery alignment, but had also drawn attention to the need for further work to be undertaken, directed towards the construction of propulsion plants capable of "tolerating given types and magnitude of movement of engine and line shaft seatings".

Mr. Verity's company had been interested in these aspects for some time, and was currently undertaking a close study of what might be achieved with high powered single screw machinery, utilizing the potential offered by the employment of high torque capacity flexible couplings in the main shafting. Experience with such couplings mounted directly in main propulsion shafting systems carrying high powers at large torque ratings had proved highly satisfactory in service for some years and, when interposed between main gearing and thrust bearing, had reflected their value in the smooth running of the main machinery as demonstrated by measured low noise levels and vibration amplitudes.

Two designs of coupling had been generated, both using rubber, or similar, as the flexible medium and both designed to place the rubber in compression, as distinct from shear, when transmitting torque. This not only enhanced coupling reliability, but also permitted a more compact coupling to be manufactured. The first design used radial rubber cylinders associated with radial meshing vanes, thus achieving a coupling having a rubber volume/total volume ratio higher than had ever been reached previously. Consequently, with the rubbers always substantially in compression, high loadings were possible. A second design, offering more flexibility with reduced cost of manufacture and simpler construction also used rubber in compression combined with a small amount of shear. Centred around the use of rubber elements, each comprising a series of rubber laminations bonded to thin steel plates, this offered a much reduced deflexion in compression, under direct loading, than would have resulted from a rubber block of equal volume having the same lateral dimensions. This coupling utilized a series of such elements arranged circumferentially with the laminations disposed radially between the driving and the driven members (Fig. 15).

Shafting misalignment manifested itself in two basic ways which were usually present together in various proportions. These comprised angular misalignment—where the two shafts met but were not parallel—and radial misalignment, where the two shafts were parallel but offset. Either of the couplings now

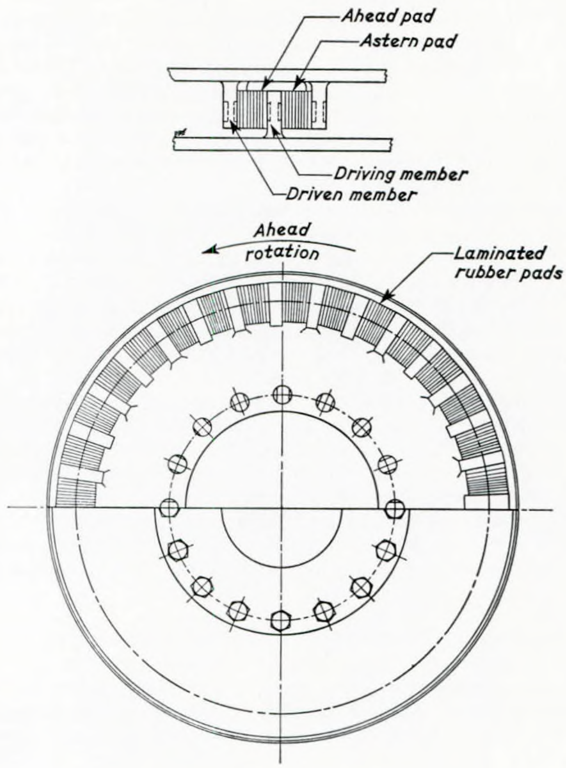

FIG. 15—*High torque capacity couplings*

described would readily accommodate angular displacement and a limited amount of lateral displacement. Where large radial or lateral displacements were contemplated, the coupling elements were usually associated with a cardan shaft whose length had been adjusted to provide the desired excursions without over-stressing the rubber.

For problems arising from structure deflexions in very large tankers with relatively short stiff propeller shafting, a solution could lie in the elimination of bending moments in the shafting, arising from curvature of the surfaces on which the machinery was mounted, by incorporating high torque capacity flexible couplings of the design described.

Following publication in Bunyan's Paper* a design of flexible couplings using laminated rubber pads was investigated for transmitting 29·4 MW (40 000 shp) at 80 rev/min. With such a coupling introduced between the main gearing and the main thrust bearing, in a tanker experiencing main tank top hogging of the order described in the paper, calculations showed that the bending moment so induced in the main shafting would be reduced to negligible proportions, thus making the gearing entirely free from the main shaft influences.

In recent years Mr. Verity's company had introduced a novel type of co-axial gear which, in his opinion, was particularly adaptable for inclusion with single screw geared turbine machinery installations for large ships with engines aft. The author would, therefore, be pleased to learn that further studies were currently in hand to examine what might be achieved with such machinery

* Bunyan, T.W., Paper presented at the Europort 71 Congress, British Day, 12 November 1971, organised by BMEC (see *Marine Engineers Review*, Dec. 1971, p.25).

utilizing the potential offered by high torque capacity flexible couplings, which, in single or cardan shafting configurations, might be combined conveniently with co-axial gearing and main thrust bearings in idealized positions.

MR. R. C. MCLEOD wrote that, under the heading "Warp of Horizontal Joints", the observation was made that, when a turbine had been in service and the top half casing was lifted, the joint face was often found to be no longer flat. He had also found this to be the case and agreed that it was better to re-assemble the turbine without re-bedding the joint. However, he did not agree with the statement: "Although the cold pull-up in the bolts will produce high stresses, when warming through the turbine and running-up again, the stress in the bolts will be relieved immediately".

With conditions as stated in the paper, the joint faces would appear as in Fig. 16, before tightening the bolts. After tightening the bolts enough, the gap at A would be closed. Further tightening of the bolts would stretch them elastically and compress the flanges elastically.

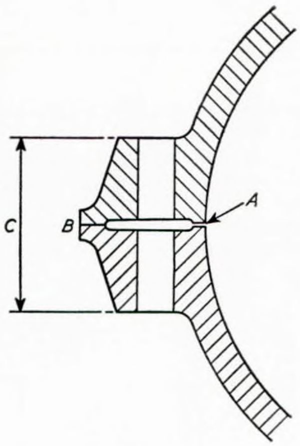

FIG. 16

Dimension C would remain nearly constant, but would decrease slightly as the tension in the bolts was increased.

The coefficient of expansion of the casing and bolt materials was assumed to be virtually similar. If the casing and bolts heated up at the same rate, there could be no change in the bolt stress. After starting and reaching normal conditions of running, the distortion due to creep, resulting from service conditions, would tend to vanish, making the joint faces flat when hot, and the flange nip would be reduced at B and increased at A, but dimension C would only have changed due to expansion, which applied to both casing and bolt material.

Unless the turbine was designed with bolt and flange heating, the chances were that the temperature rise of the bolts would lag behind that of the casing and the bolt stress, in his view, would not be "relieved immediately", but would increase to some extent.

MR. B. WOOD, in a written contribution, stated that many of the points raised by the author relating to marine turbines applied equally to turbines driving compressors in chemical and oil industries. These also ran at high and variable speeds—up to 25·7 MW (35 000 hp) at 10 000 rev/min.

He complemented the author on his valuable survey with which he was broadly in agreement, but there were points where he believed correction was desirable; thus "impulse-reaction" was to be deprecated as a nomenclature. What the author had described was impulse construction. This was unaffected by the fact that for 50 years the brighter impulse designers had been applying some reaction. An "impulse reaction" turbine he understood to be one in which the main reaction stages were preceded by one or more impulse stages either to get rid of the initial pressure and temperature quickly, or for regulation.

Possibly more important was the issue of disturbing frequencies. The author seemed to pass over low multiples of revolution frequency as if they did not matter, but most blade failures Mr. Wood had encountered were attributable to this sort of excitation. Multiples of rev/s up to the 5th had been recognized as lethal for many years (Fig. 17). With the tendency

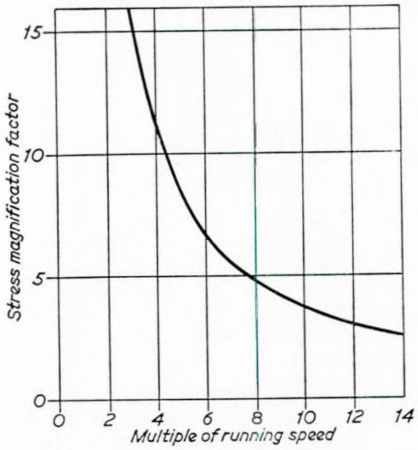

FIG. 17—*Stress magnification factor curve*

to adopt higher stressing the higher multiples 6th, 7th and 8th had had to be included in the danger zone. Failures firmly attributable to resonance at 10 and 13 per revolution were also known, although in these cases some other ingredient, such as a stress raiser at the critical place, apparently had to be present. The author did not mention one of the most useful methods of avoiding such resonances, namely, batching, primarily in the shroud. Thus five batches would eliminate the 5 per revolution impulse within the packet (though not out of phase vibrations).

The author mentioned two of the makers' old wives' tales. Mr. Wood had rarely met a turbine breakdown where the maker had not suggested: slug of water, pipe reactions, and, more recently, corrosion fatigue. He had never found the first two to be relevant but the third contained an element of truth in that blade failures sometimes occurred after, say, 4000 hours. If the lowest natural frequency was 1000 Hz the implication was that the blade had undergone some $1{\cdot}4 \times 10^{10}$ reversals. The classic Wöhler test in air would show failure within 5×10^{6} reversals or not at all. There were various possible explanations and, without strain gauging in service, it was difficult to say which one fitted. Thus, it was possible that resonance only occurred during some off design condition when running up or at overspeed. The alternative was that in wet conditions, especially in water containing salts or dissolved gases, the fatigue resistance was grossly lowered and there might be no fatigue limit (Fig. 18).

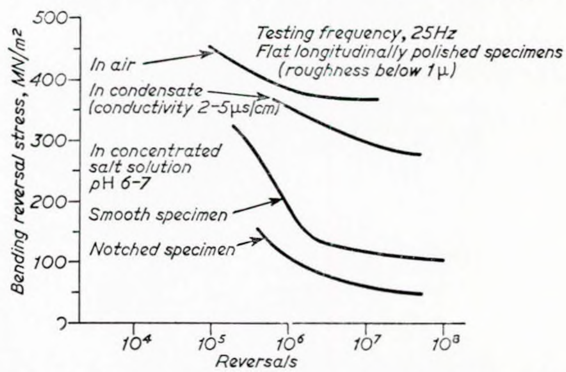

FIG. 18—*Fatigue and corrosion of A 0·1C, 13CR blade steel (after Pohl)*

Despite this, Mr. Wood had not yet encountered blade fatigue failures that had not been overcome by mechanical means. In some cases his company could not wait to see whether mechanical alteration would suffice so had also adopted a lower grade alloy. Such alloys characterised by lower UTS were commonly less sensitive to stress corrosion. In this connection the steam load stress which most designers employed was a false criterion. No practical blade had ever failed by steam load; blades failed by resonance (fatigue). Robust (low stressed) blades not only had higher margins and could tolerate some pitting but they were more likely to be out of tune.

He would like to throw in another inference from experience although this was not fully proved, i.e. if the foot failed it indicated a wheel resonance; failure in the blade proper indicated blade resonance.

All the mystery could not yet be eliminated from blade technology. Thus it was easier to explain a blade failure after the event—since it was known where the fracture had occurred—than it was to foresee the risk and eliminate it at the design stage.

Author's Reply

The author was pleased that a turbine designer as experienced and competent as Mr. Burn should open the discussion, and thanked him for his valuable contribution to the paper.

Whilst the author agreed that the future prospects for steam turbines should, for completeness, be compared with other types of main engine machinery, it was not within the scope of his paper to do so. There were many aspects to consider when comparing different types of prime mover, certainly sufficient for another paper.

Blade erosion rates were more readily predicted when considering land based, constant speed, steam turbine prime movers, but much less so their marine counterparts, where steam inlet pressure, temperature, condenser vacuum and propeller speeds, were all subject to considerable variations during the life of the vessel.

The electron beam welding method of shielding the leading edge of a blade was probably a spin-off from developments in the land based turbine field. If this manufacturing process prevented the "shedding" of shields, as had occurred in some cases in the past where brazed stellite shields were fitted, it would indeed be a step forward.

Regarding the erosion of astern blading, the author had included some of the more interesting details (see Figs. 19, 20 and 21) of the case he investigated in 1966, which was referred to during the discussion. As could be seen from the L.P. turbine layout, the ahead L.P. turbine was a double-flow type, with an astern turbine on the after end of the rotor.

The aft-flow ahead turbine outlet and the astern turbine outlet both faced aftwards. In the sketch it would be seen that the astern blades were eroded in way of the trailing edges, on the top half of the blade length, and on the leading edges of the bottom half of the same blades. It was clear that drips which fell past the astern turbine intake, from the inner and outer casings, and from the deflector plate downstream of the astern turbine, were being drawn into the astern turbine at the outer periphery, and being forced out at the inner periphery. The trimming weight on the astern wheel was also heavily eroded, on the face corresponding to ahead rotation as indicated.

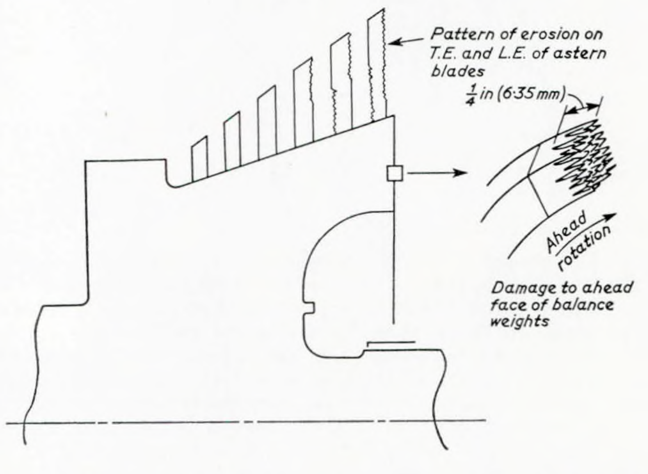

FIG. 20—*Erosion damage on astern turbine blades and balance weights due to excessive moisture when running ahead*

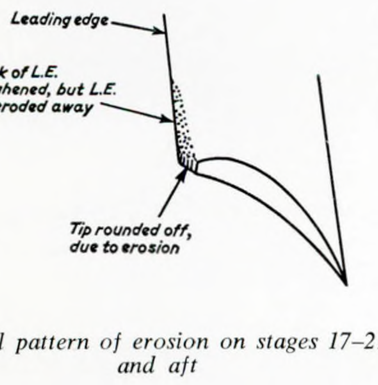

FIG. 19—*Typical pattern of erosion on stages 17–21, forward and aft*

The disposition of the erosion on the astern blading was interesting, because it clearly illustrated the circulatory path of the steam within the astern turbine when it was turning ahead.

Regarding gland sealing arrangements, if carbon glands were less liable to promote, or aggravate, a thermal bend, they would be a worthwhile development.

Crimped tubing had certainly proved to be a success in the last row blades of *Canberra's* turbines, but whether this method would be suitable for the closely pitched blading of a high speed H.P. rotor, was another question.

The M.E.L. design of casing horizontal flange "looked right", compared with the former design, by bringing the bolts closer to the inner surface, and reducing the radial width of the flange, also the area of the mating surfaces etc.

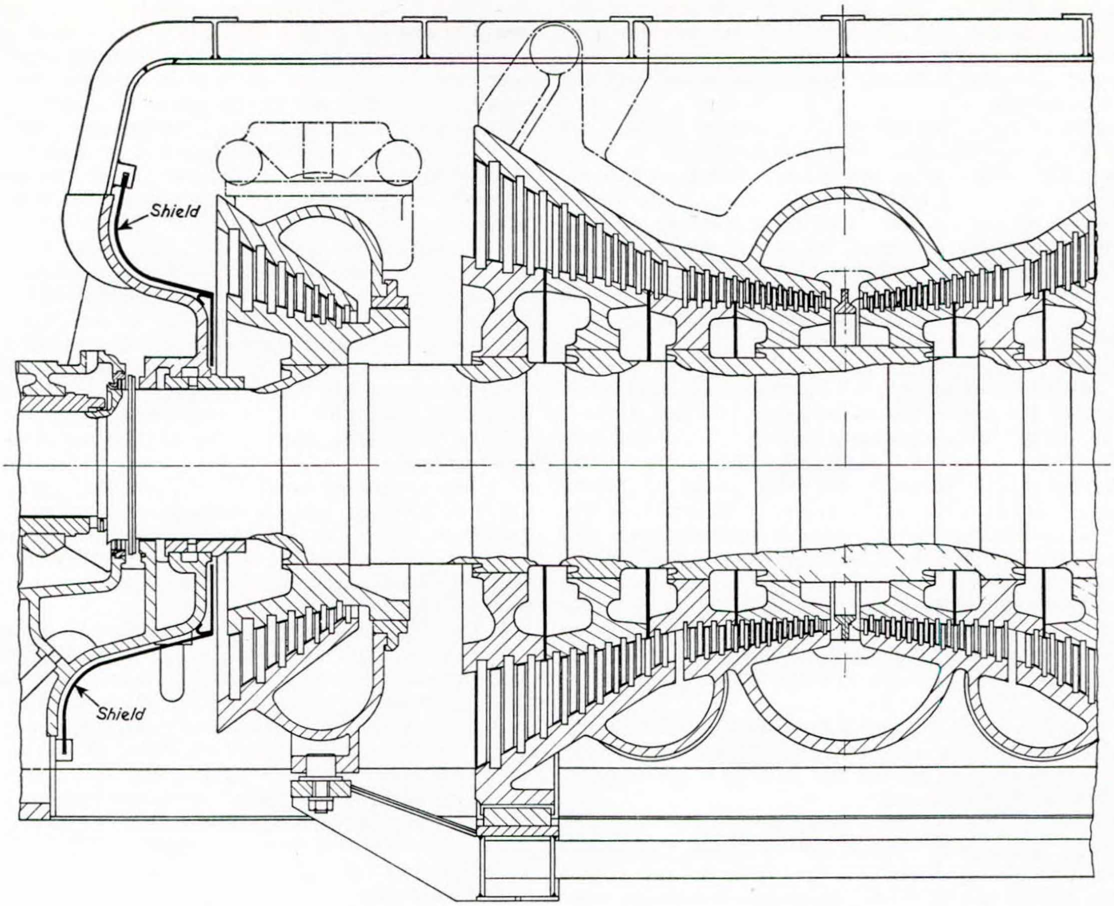

FIG. 21—*Cross-section through double flow L.P. turbine*

The inverted fir tree type root, of the blade illustrated by Mr. Burn, embodied an overhanging trailing edge, but the diagram did not make the geometry of the overhang clear. One would assume that there would be a carefully machined and generous radius at the base of the overhanging trailing edge, with a corresponding cut-away on the corner of the next root platform, to avoid the abutment of a sharp edge in an area of stress concentration. It was, no doubt, a well tried and tested design, but possibly there was an unnecessary importance placed upon accuracy of frequency prediction, if, due to blade manufacture and hand fitting in the wheel, the spread of "as-built" frequencies of different batches of blades was equal to, or greater than the five per cent quoted by Mr. Burn. One wondered, in fact, whether the quoted accuracy was not so much dependent upon the degree of sophistication of the calculations, as on the measured results with which they were being compared.

It was probably as important, therefore, to measure periodically, the "range" of as-built frequencies on a number of rotors, to ensure constancy of skill in manufacture and hand fitting.

There were few, if any, "open windows" in the rotor speed range of most marine turbines, where coincidence of blade resonant frequencies and likely excitation frequencies would not occur. It was therefore essential, where coincidence was inevitable, to be able to predict accurately the levels of steady and vibratory stresses in blades and shrouding.

The accuracy of stress prediction depended, among other things, upon correct assessment of stress concentration factors, yet this was one factor which could be underestimated, due to poor manufacture and fitting of blades and shrouding.

With regard to axial tip sealing clearance, it was presumed that Mr. Burn was allowing for the amount of axial "float" of the rotor, between the ahead and astern thrust pads, when he mentioned a minimum figure of 0·75 mm axial tip clearance. Certainly it was better to err on the high side when setting axial clearance, to avoid damage to blading, diaphragms, and possibly, thrust bearings.

Radial tip seals in the initial stages would probably be preferable for "reheat" H.P. turbines, of the type common to most manufacturers, where inlet steam flowed forward, and reheated steam aftward, in the same H.P. turbine casing.

In reply to Mr. Hinson, Mr. Burn had probably answered his question about the accuracy to which blade and packet resonant frequencies could be calculated, though one would imagine that a lot of research and development experience lay behind Mr. Burn's, almost facile, claim of accuracy to within five per cent, particularly with regard to the longer tapered and twisted blades fitted on the last few stages of L.P. turbines.

The author would favour the proximity pick-up, which measured the amplitude of vibration of the shaft at some convenient position near the bearing, if this were to be installed during building of the turbine (though one normally used a "seismic-mass" type pick-up, when carrying out vibration measurements on a turbine where vibration-monitoring equipment had not been installed). The reason for his preference was that, at low rev/min, one would have a more positive indication of a possible bent rotor and be in a better position to correct the situation, before reaching the rotor critical speed (which in the case of some H.P. rotors could be as low as 50 per cent of service rev/min). Of course the amplitude would need to be related to the frequency, in order to apply the vibration limits, and this might tend to complicate both the vibration-monitoring equipment installation and, possibly, the information read-out as well, if, for example, one dial indicated mils displacement and the other vibration velocity.

With regard to shrouding failures, it should be remembered that when a packet (or batch) of blades was vibrated in some specific mode, the blade tenons and shroud, being part of the structure, were subject to bending stresses just as the blades were. If that part of the structure were comparatively weak, it would fail first. In many instances, he agreed that failure of tenons or shrouding was due to faulty workmanship.

Mr. Cruickshank had provided a useful contribution to the paper from the operator's viewpoint. His observations were well founded and needed no further comment.

In reply to his question, the author could not be sure whether all turbine manufacturers bored flange bolt holes prior to stress relieving the casing, but the tendency for H.P. turbine casings to hog was probably a transient condition, more common among double-casing designs (an inner and outer casing) during warm-up, or shut-down. Although the rotor would be turning and circulating the hot air/steam mixture in the inner casing, there was no such circulation in the space between the inner and outer casings, since there was no steam flow. A pocket of hot gas tended to collect in the top half between inner and outer casing, increasing the temperature there and causing the outer casing to hog. Depending, of course, upon the design of support for the inner casing, it could be forced upwards by a hogged outer casing, and start a rub between the rotor and the bottom half of the inner casing.

In view of the fact that the author agreed with the substance of Dr. Cowley's remarks, he was somewhat embarrassed by the fact that he was unable to answer Dr. Cowley's questions as comprehensively as they deserved.

The author was not particularly surprised to learn that ships operated in the Channel with "reheat on" to achieve full power and speed, but considering the fact that propeller rev/min would probably only fall by about ten to twelve per cent in the non-reheat mode, it seemed to be taking an unnecessary risk.

The case the author quoted, during the presentation of the paper, of an H.P. turbine which had hammered out its bearings during the run-up to speed, and which had gone unnoticed by the watchkeeper in the sound-proofed control room, was an example of what could happen if H.P. and L.P. turbine vibration-monitoring equipment were not installed, when the control room was remote from the engine flat, particularly during manoeuvring operations, or working up to full speed. If a proximity probe had been installed in the turbine, the operator would have had warning of the thermal bend at low revolutions, and could have "heat-soaked" the rotor on steam at low rev/min for some time until the "run-out" was reduced.

Regarding Dr. Cowley's second point about periodically unmanned machinery spaces, the author had looked at the figures given in the 1970, 1971 and 1972 Annual Reports of Lloyd's Register of Shipping, which gave the numbers of automated ships built to class with Lloyd's Register in 1969, 1970 and 1971 respectively (Tables I, II and III).

It was clear that in each of the years considered there were more Diesel powered ships than steam turbine powered ships which had been assigned the UMS notation, but considering that there were 14 to 19 times more Diesel engined ships built every year, the larger number of such ships with UMS notation was not so surprising. With the exception of the latest figures for 1971 a greater percentage of steam turbine powered ships had the UMS notation than Diesel powered ships.

Whilst the 1971 figures (Table III) showed a dramatic fall to only one steam turbine powered ship with the UMS notation out of a total of 23, evidence of a steady decline in numbers of steam powered ships with UMS notation could only be judged in the light of the figures for 1972 and the following years.

Dr. Davis had mentioned the challenge to the steam turbine at the upper end of the power scale by the gas turbine. This was certainly true, particularly of the large fast containerships, but even the gas turbine could suffer eventually from the fate described by Dr. Davis as the "enthusiasm for economy". There could be an even greater incentive to improve the fuel rate of the gas turbine than there had been for the steam turbine, beginning with the introduction of heat exchangers etc. Dr. Davis's advocacy for a return to the simplified steam cycle for marine propulsion* was becoming increasingly recognized†. His proposition had been based upon the premise that an extensive simplification of today's marine steam power plant would enhance the reliability of the plants and make them more economical to build and install.

It should be pointed out, of course, that Dr. Davis's economic analyses had been based on a high speed containership system operating on a four to five day North Atlantic trade route, though he had also claimed that the analyses were generally applicable to other containership systems, and to tankers. At first glance, therefore, it would appear that Dr. Davis's proposal for a return to the simplified steam cycle to achieve a higher degree of reliability was aimed primarily at that area of the marine propulsion market presently being challenged by the gas turbine.

The advantage which Dr. Davis could claim for the steam turbine, however, was the extensive background of experience, based on American maintenance and repair records, which enabled the designer to make a better evaluation of the most suitable steam conditions and plant thermodynamic cycle for the various applications considered.

Dr. Davis's admonition of designers for allowing situations to develop in which turbines could be lifted from their seatings by known forces of uncalculated magnitude was no doubt prompted by the section on "pipe forces", in the paper. In fact pipe forces were calculable, but the installed pipework rarely conformed to the theoretical assumptions made in the calculation. The situation could become even more complicated where reheat turbines of the combined H.P./I.P. type were fitted with not less than three main steam pipes running between turbine casing and boiler, in addition to the usual cross-over pipework. The latter also needed careful design and fitting to avoid applying large moments and forces to the turbine casings.

The author agreed completely with Dr. Davis on the subject of turbine blade vibrations. It was far more difficult to predict the vibration characteristics and sources of excitation of turbine blades with certainty at the design stage than it was to account for the most likely cause of failure after the event. It was not asking too much, however, that a manufacturer should have sufficient expertise and data on his own blades to know if "a zone of uncertainty" was being encroached upon.

The fitting of lacing wires was sometimes a necessary safeguard against failure when the "zone of uncertainty" had been determined, but could not otherwise be avoided.

In that respect side-entry blades were at a disadvantage, because the pitch of the blades could not be changed at a later date, to permit blades of a more robust section to be fitted.

The "bulb and shank" root, side-entry blades were indented in way of the bottom edge of the bulb on both sides of the wheel, which "spread" the bulb ends and prevented fore and aft movement. The spread of the bulb raised other problems, however, notably the tendency to impose localized loading on the corners of the bulb root.

As Dr. Davis had intimated, the provision of sufficient numbers and properly located drains was an important feature of turbine plant design which deserved more attention than it was often given.

The author had made reference in the paper to one of the problems associated with bled steam, and water extraction drains, particularly the possible blockage of orifices of the

---

* Davis, A. W. March 1971. "A Simplified Steam Plant for Marine Propulsion." *Westinghouse Engineer,* Vol. 31, No. 2, pp. 34–41 incl.

† Kasschau, K. "A Proposed Simplified Steam Plant for Marine Propulsion." *Marine Technology,* Pub. by SNAME (quarterly), Vol. 9, No. 2, pp 161–172 incl.

Author's Reply

TABLE I—NUMBERS OF AUTOMATED SHIPS BUILT TO CLASS IN 1969

Type of ship	Oil engine			Steam turbine		
	A	B	C	A	B	C
Cargo	71 (34)	*203		10 (2)	11	
Tanker	38 (9)	64		13 (4)	20	
Trawler	34	105				
Tug	22 (1)	34				
Ferry	9	12				
Passenger				1	1	
Others	*27 (2)	**43			1	
Total	201 (46)	461	44 (10)	24 (6)	33	73 (18)

A = Number of ships automated
B = Number of ships built to class
C = Percentage of ships automated

*Includes 1 Diesel electric ship
**Includes 3 Diesel electric ships. The figures in brackets refer to ships which have been assigned the UMS notation.

TABLE II—NUMBERS OF AUTOMATED SHIPS BUILT TO CLASS IN 1970

Type of ship	Oil engine			Steam turbine		
	A	B	C	A	B	C
Cargo	117 (69)	196	60	4 (1)	4	100
Tanker	37 (15)	51	72	15 (6)	16	93
Trawler	10	31	32			
Tug	25	39	64			
Ferry	10 (1)	15	66			
Passenger	1	1	100			
Others	*34 (3)	41	83			
All types	234 (88)	374	62	19 (7)	20	95

A = Number of ships automated
B = Number of ships built to class
C = Percentage of ships automated

*Includes 1 Diesel electric ship. The figures in brackets refer to ships which have been assigned the UMS notation.

TABLE III—NUMBERS OF AUTOMATED SHIPS BUILT TO CLASS IN 1971

Type of ship	Oil engine			Steam turbine		
	A	B	C	A	B	C
Cargo	165 (52)	217	76	4 (0)	4	100
Tanker	50 (8)	58	86	18 (1)	19	95
Trawler	43 (7)	50	86			
Tug	24 (9)	33	73			
Ferry	12 (2)	14	86			
Others	28 (11)	39	72			
All types	322 (89)	411	78	22 (1)	23	96

A = Number of ships automated
B = Number of ships built to class
C = Percentage of ships automated

The figures in brackets refer to ships which have been assigned the UMS notation.

"continuous drain" type with dirt and débris, which were not readily accessible for inspection or cleaning.

Some manufacturers claimed that a 4 mm to 5 mm diameter continuous drain orifice would be large enough to avoid blockage, yet small enough to minimize steam loss. However, the author believed it was essential that an efficient dirt trap should precede each drain orifice. Where a thermal drain valve was fitted it should be preceded by a suitable strainer. Both types should be accessible for cleaning.

The avoidance of dirt and débris in the steam system was dependent to a large extent on the care taken in cleaning the system before installation in the vessel and during subsequent maintenance. In many instances the critical period was during the sea trials, followed by the first few months of operation, during which time drains and strainers could become blocked, valves jammed, or the turbine blades, gland seals etc, damaged due to the passage through the system of swarf, scale, grit, weld spatter, pieces of gasket etc.

In reply to Mr. Burnett, the author would like to correct Mr. Burnett's opening paragraph, by pointing out that a large market for steam turbines in OBO and LNG ships was never suggested. Indeed, a glance at the upper curves in Figs. 3 and 4, made it clear that no such statement could be made for the immediate future.

The author assumed that Mr. Burnett was not implying that reliability and availability were interchangeable terms, but that when referring to VLCC, it was more convenient to consider availability rather than reliability.

The point about LNG ships was that, unlike other cargoes, some portion of the liquefied gas would evaporate at a rate dependent upon the heat-flow through the container into the cargo. The rate of evaporation could be reduced, by increasing the container insulation, or by the use of refrigeration plant, both of which added to the already high cost of building and operating such vessels.

Alternatively, the time elapsed between loading and discharging the liquefied gas cargo could be cut by increasing the service speed of the vessel, thus reducing evaporation losses. A balance between these alternatives could be arrived at, for a particular size of vessel, but because the initial cost of building such vessels was so great, an increase of main engine power would make only a marginal difference to the final cost of the ship and, therefore, a higher speed could probably be the more attractive proposition.

With regard to Mr. Burnett's question about time lost due to turbine main machinery failures, compared with the rest of the plant, he would of course realize that such information was only readily available to owners and managers of shipping companies.

However, one fleet operator had recently shown the author a chart of some 16 ships, all oil tankers, built since 1960, and the author had deduced the following facts.

The eight built between the beginning of 1960 and the end of 1964, had an average total time-loss of 2·2 days/year. The average ratio of time lost due to boiler failures, to time lost due to main turbine failures, was 1·9 to 1·0.

For the remaining eight ships built after 1965 and up to 1969, the average total time-loss was 6·3 days/year, and here the ratio of boiler time-loss, to turbine time-loss, averaged 1·0 to 1·8, an almost complete reversal of the previous ratio.

Admittedly, this was not a representative sample of world shipping, nor of different makes of turbine, but it must surely suggest the need for further research, to determine whether a less conflicting trend was reflected elsewhere in marine fleets.

The author would like to thank Mr. Lovell, who sent the following contribution. "From a question given in the engineering knowledge paper, of the extra first class exam of January, 1972, quote,—an analysis of the areas of unreliability in marine steam turbine systems gave the following distribution:

Boilers	49%
Condensers and circulating pumps	12%
Boiler feed pumps	11%
Main turbines	7%
All others	21%

The source of the information was not known, but showed the boiler as the least reliable component, by a large margin.

At the Marine Steam Power Plant Seminar, 1970, a contribution by the Babcock and Wilcox Co, based on an extensive analysis of boiler casualties from service reports, and U.S. Government and private repair costs, showed that on a maintenance cost basis, boilers accounted for 29·5 per cent of the total engine room maintenance cost, and steam generation (including boilers) was 43·9 per cent of the total engine room maintenance cost.

Taken in conjunction with the previous analysis of unreliability, it would appear, at first glance, that although boilers were the least reliable components, the cost of maintaining them was only a third of the cost of all other components. However, one was always wary of the presentation of statistical evidence of this sort in support of an argument, since it could, even with the best of intentions, be omitting more contrary evidence than that stated in favour. In that context the use of a computer was not necessarily a safeguard, since the results obtained would depend, both on what was fed into the computer and on the questions it was asked.

With regard to Mr. Burnett's question about the bending of rotors during manoeuvring, the author could see no reason why a properly designed automatic system could not be devised to prevent the initiation or augmentation of a thermal bend. Perhaps, more important, it was essential to know if the shaft was bent before commencing to increase rotor speed, by installing suitable instrumentation to monitor and warn of such a condition. The bend itself was not necessarily dangerous or crippling, as long as the rotors could be kept turning at low speed, by admitting a small quantity of steam to equalize the temperature distribution.

The author thanked Mr. Verity who had written a most interesting and informative contribution concerning the possible use of heavy duty flexible couplings in the main line shafting of VLCC and other similar vessels.

The possible returns from improving the after-end designs of VLCC in order to minimize hull deflexions seemed at present to be very limited when taken alone, but taken in conjunction with the installation of flexible couplings in the main line shafting could no doubt bring about a decided improvement in the maintenance of good propulsion machinery alignment. This would apply equally to geared turbine and Diesel installations.

Mr. Verity had not quoted any figures regarding the angular and offset displacements which could be accommodated by the flexible coupling (and there might also be an axial displacement to accommodate), but it seemed to the author that care would still have to be taken to ensure correct positioning of the coupling, and shaft alignment, during installation of the line shafting and propulsion machinery, even if a flexible coupling were fitted. There might be a tendency for some to believe, for example, that less care would be necessary during installation of the line shafting if a flexible coupling were fitted which could tolerate large offset and angular deflexions. It was just possible, however, that an alignment error during installation augmented by hull deflexions in service might exceed the total acceptable displacement of the coupling.

With regard to the Cardan shaft arrangement mentioned by Mr. Verity, the author wondered about the weight of the shaft which would have to be supported by the flexible couplings at either end, unless some centring devices at each end could be arranged to carry the load.

Alternatively, if only a single coupling were employed, the weight of half the coupling might be supported on the stub shaft overhanging the after bearing of the main gearwheel, imposing a bending moment on that end of the gearwheel shaft and an additional load on the main wheel after bearing.

In accommodating a particular displacement, all flexible couplings transmitted a corresponding misalignment force, the magnitude of which was dependent upon the displacement accommodated and the design of the coupling used. Therefore, the effect of such forces on the line shafting would have to be carefully considered at the design stage of both hull and machinery to avoid excessive forces and moments being applied to the shafting.

There was at least one main turbine/gearing design, to the author's knowledge, in which the main line shaft thrust bearing was forward of the main reduction gearwheel, but in general the provision of a thrust bearing aft of a flexible coupling should not present an insurmountable problem.

It would be necessary, of course, to examine the design in more detail before commenting further on the concept, and then of carrying out the practical application and reporting the results. The author looked forward to hearing more about this development in due course.

In reply to Mr. Noble, there was not to the author's knowledge, any published information about the effects of nozzle excitation on the second moving row of a two-row, velocity-compounded Curtis wheel. However, Heymann* had described the "pulse strength", associated with the wake from nozzles, as being represented by the product of the "velocity defect" and the "wake width", both of which, he says, "will decrease with increasing distance from the nozzles". He had thus described the effect of nozzle excitation on adjacent moving blades and the effect of axial spacing from the nozzles. In the author's opinion, it was unlikely that nozzle excitation would carry through to the second row of moving blades unchanged, either in frequency, or magnitude. It was probable that some velocity variation, due to the nozzle wakes, would be manifest round the circumference, at inlet to the first row of fixed blades, and that this would be further modified, by its passage through the row of fixed blades. Fixed blade passing frequency was likely to be the most important source of excitation, acting on the second row of moving blades.

Mr. Noble was quite correct in drawing attention to the shock effects of partial admission however, and the author believed that this effect would most certainly carry through to the second moving row.

Many designs of marine turbine incorporated nozzle group control. Anything from three to seven separate nozzle groups were used, in the control of some modern marine steam turbines, and it seemed clear that careful attention had to be paid to the sequence of opening of each group, with due regard to the resonant frequencies of the corresponding moving blades.

A paper by Kroon† described clearly the effect produced on moving blades by partial admission. With regard to Mr. Noble's second question, the author's own views were, that if one were considering the first (control) stage of a multi-stage turbine, where blade lengths were generally very short and robust, the mode of vibration most likely to be excited by partial admission would be the fundamental, in-phase, packet mode in the tangential and axial directions.

In fact, Kroon had stated that, "the designer should strive for short blades and light steam loads, as long as this did not result in poor efficiency. Shorter blades also meant higher natural frequencies, so that vibratory motion could die out more rapidly between successive loading periods".

The author was pleased to have received a written communication from Mr. McLeod, a former designer of turbines who had built up a wealth of experience over a number of years.

Mr. McLeod had expressed his disagreement with the concept of flange bolts being relieved of stress when the turbine was started up and had reached normal running conditions, and believed that the bolt stress would probably increase to some extent. However, the author had meant his reference to immediate relief of stress to apply only to that component of stress produced in the bolts from closing the gap on the inside of the casing, and not to the total bolt stress. He agreed with Mr. McLeod's analysis of the thermal stress conditions, but of course there was an additional stress due to pressurization of the cylinder to be considered.

With regard to steam heating of flanges and bolts, it was to be hoped that marine steam turbine designers would not have to resort to such a system at any time in the future, because, apart from being contrary to the concept of simplification, it could lead to operational difficulties due to unexpected casing distortion, unless it were a carefully designed and operated system.

In reply to Mr. Wood, the author would like to point out that it was "blading" which was referred to in the paper, as being of the "impulse-reaction" type, not the turbine.

The author was familiar with the "impulse-reaction" type turbine, described by Mr. Wood, but by "main" reaction stages, he doubtless meant 50 per cent reaction stages, whereas most modern marine propulsion steam turbines were of the "wheel and diaphragm" type, the percentage of reaction at the mean height increasing progressively, from the first stages in the H.P. turbine, to the final stage in the L.P. turbine. For example, one manufacturer had described his L.P. turbine as having 20 per cent reaction, at the mean height of the first moving row, with an increasing percentage in each stage, to the eighth, and final row, which was designed for 45 per cent reaction at the mean height.

On the question of nomenclature, it might be more logical, and more readily understood by the uninitiated, if all staging were considered as either "pure" impulse, "impulse-reaction" or "pure" reaction, treating the 50 per cent reaction stage as a special case of impulse-reaction, which allowed a particular type of construction to be used.

Mr. Wood would find that the section on "shrouding", in the paper, made reference to "batching" of blades to limit vibration. This served to limit only the amplitude of vibration of the batch fundamental tangential mode, at the resonant frequency.

Again, under the heading "Excitation", there was the reference to flow variations round the circumference, which implied low multiples of revolution frequency. In Fig. 7, multiples of two and thirteen were illustrated in the Campbell Diagram.

The author was not surprised that Mr. Wood discounted both water carryover and the effect of pipe forces on turbine alignment, if he was only concerned with break-downs in turbine machinery, since neither of these items necessarily resulted in failure of the turbine, but they could certainly lead to operational problems. In the marine industry it was sometimes just as expensive as a break-down if reduction in operating speed of a vessel was necessary over an extended period.

With regard to corrosion fatigue, as Mr. Wood had correctly argued, in wet steam conditions where salts such as chlorides might be present and the blade material subjected to corrosion attack, not only was the limiting fatigue strength reduced, but it was no longer independent of the number of load cycles. It should be emphasized, of course, that due to nozzle wakes there was always an exciting force present which caused a non-resonant stress reversal on the blades.

If foaming took place in the boiler drum, there would be the danger of salts being carried along with the steam, so proper boiler treatment control was a vital requisite. In the final stages of an L.P. turbine, the concentration of water droplets would probably be sufficient to flush away any salt deposition before it had time to start an attack. One would expect an attack to occur at a particular stage in the L.P. expansion, where the steam condition coincided with the dew point on the condition curve. If the steam inlet conditions, the vacuum, load etc, on a marine main steam turbine remained constant for a sufficiently long period, so that the dew point occurred at one particular stage, it might be expected that corrosive attack could begin there.

The author could not agree with Mr. Wood's remark that the steam load stress was a false criterion, though it would be preferable to describe it as an important factor in the combined steady and vibratory stress components. The lower the level of steady stress, generally the lower also the vibratory stress and the greater the margin of safety against fatigue failure. This was the condition achieved by making the blade more robust. It would be dangerous to conclude, however, that the more robust

* Heyman, F. J. 1969. "Turbine Blade Vibration Due to Nozzle Wakes." *Trans. ASME,* Journal of Engineering for Power. October, pp 223–238.

† Kroon, R. P. 1940. "Turbine Blade Vibration Due to Partial Admission." *Trans. ASME,* Journal of Applied Mechanics, Vol. 62, December, pp A161–A165.

blade having higher natural frequencies would thus be unconditionally safe from excitation and, possibly, failure, at some higher frequency.

Finally, the author could not agree with Mr. Wood's observation that a failure in the root of a blade automatically suggested a wheel resonance as being responsible for the failure. He had investigated a number of cases of blade root fatigue failure and had shown that the cause of the root failure was due to a particular mode of blade vibration in which the root took part. Reference to this phenomenon was made under the heading, "Measurement of Blade Resonant Frequencies", in the paper.

Related Abstracts

The performance of a throttle controlled propulsion steam turbine

The methane tanker *Hassi R'Mel* is propelled by the first marine propulsion turbine with throttle control and completely symmetrically built turbine casings for ahead and astern operation, designed to secure high reliability and manoeuvrability, and low fuel consumption; the output at the propeller shaft is 16 250 hp, at 120 rev/min. This article assesses the trials and the first six months of operation. Particular attention was given to the testing of the astern turbine as long operation at full speed astern is usually considered a particularly critical stage. The regulations of the classification society require capacity for operating at full speed astern for 30 minutes. As the ahead valve is closed during the astern operation and the ahead turbine turns under vacuum in the opposite direction, very complex turbulence and temperature conditions occur owing to ventilation in the blading. So as to be able to judge better the thermal behaviour of the H.P. and L.P. turbines during astern operation, the steam temperatures at the H.P. inlet, in the cross-connexion between H.P. and L.P. turbines, at the extraction from the L.P. turbine and in the exhaust connexion behind the outlet of the astern turbine, were taken. Whereas the steam temperature at the H.P. turbine inlet drops gradually from 505°C to 430°C because the ahead valve is closed, the temperature in the cross-connexion rises slightly from 210°C to about 240°C. This is presumably because subsequent to the stuffing box steam a weak flow from the H.P. to the L.P. turbine occurs, which transports heat from the hot H.P. casing towards the condenser. The curve indicates that steam temperature in the connecting line will not, even during astern operation at full speed for several hours, rise much above 250°C. The considerable drop from 150°C to about 60°C at the extraction point of the L.P. turbine proves that the L.P. turbine blading is not warmed up by the exhaust steam of the astern turbine. For this reason the same flow direction was chosen, as the propulsion unit was being designed, for the ahead and astern turbines, so that the exhaust steam from the astern turbine could be conducted directly to the condenser. The exhaust steam temperature behind the astern turbine is particularly important for the thermal load of the condenser. It reached its highest value of 165°C after approximately 36 minutes at full speed astern and then remained almost constant. Thus unlimited astern operation can be carried out with this unit. The test was completed after about 43 minutes duration because no alterations in the operational behaviour were to be expected. No difficulties arose from the usual reverse manoeuvres and crash stop tests. The thermal behaviour of the H.P. turbine is of decisive importance for rapid starting-up, large changes in load and reverse manoeuvres. Much information about these thermally unstable processes for large power plant and industrial turbines is available, but little attention has been given to them on marine turbines. Yet with marine propulsion units, more frequent and faster changes in speed, load and temperature must be reckoned with, and flange heating of the lower casing half, usual in power plant turbines, are dispensed with. The maintenance of the radial clearances in the blading and at the shaft seal under thermally unstable operating conditions can only be reached by a heat elastical design of the turbine casing. For this reason the turbine casings are designed completely symmetrically about the joint. In addition the throttle control system guarantees a symmetrical temperature distribution within the H.P. turbine casing at all load points and in all cross-sections. Thus distortion of the turbine casing is almost totally excluded, so that the radial clearances are maintained. Leakages at the joint flange can, for example, be caused by too high a temperature difference between the flange inner and outer sides. In addition, perfect insulation of the turbine casing is also essential for even heating of the casing. Experience shows spray insulation to be most favourable. Casing distortion and flange tightness can be checked quite easily by measuring corresponding temperature differences. With simple casing forms it is in general sufficient to measure the wall temperatures above and below in the middle cross-section, as the differences between these values represents a measure for the distortion of the casing. The wall temperatures of the inner and outer flange sides can be used for checking the joint tightness. On board, these measurements were limited to the H.P. turbine as the test stand trials indicated that no thermal distortion or leakages in the L.P. turbine are to be expected owing to the low temperature. It was shown that the thermo-elasticity of the H.P. turbine had been attained by means of the symmetrical design and the throttle control. This unit is insensitive to changes in thermal conditions. Six months does not suffice for final judgement of the unit, but experience to date shows very satisfactory results, in particular:

1) the efficiency in the upper load range is higher than with nozzle group control;
2) the preheating which is normally necessary for marine steam turbines, is dispensed with;
3) the unit can be run up from the cold condition to full speed within a few minutes; the manoeuvrability of the unit is similar to that of Diesel engines and gas turbines;
4) astern operation at full speed is unlimited.—*Geisler, O. et al: Shipping World and Shipbuilder, July 1972, Vol. 165, No. 3871, pp. 877–879.*

Steam turbines and gearing

Approximately ten per cent of the total number of ships classed with Lloyd's Register are propelled by steam turbines, but this represents about 25 per cent of the total horsepower. Since there are fewer turbines than oil engines, it is more difficult to generalize about their defects. Also, there are several types of turbine so that the experience gained with any one type is not so great as that with some of the slow running oil engines. Blading defects predominate and these are followed by defects of the rotor, its thrust and journal bearings, and defects of the gearing wheels, pinions, their bearings and flexible couplings. A steam turbine is more compact than an oil engine and a defect in one component is more likely to lead to defects in others. A blading defect can easily lead to turbine unbalance which may affect the bearings, conversely, a journal or thrust bearing failure can affect the blading. The causes of thermal stress and distortion are identified as:

a) incorrect warming through and cooling down; keeping a hot turbine stopped too long when manoeuvring;
b) water entering the turbine owing to the boiler priming, or through gland steam or bled steam connexions;
c) steam leaks from the astern manoeuvring valve when running ahead;
d) prolonged running astern.

These conditions are examined, and methods adopted to counter them are outlined. Excessive vibration is often the first sign of trouble. Vibration detectors have been used which sense movement of the turbine casing or bearings, and also proximity pick-ups which sense shaft eccentricity. The latter appear to be the most suitable since the rotor and shaft always run eccentrically when there is turbine vibration. If the bearing supports are stiff there may be insufficient vibratory force to generate significant amplitude at the turbine casing. Attention is also paid to bearing temperature alarms and turbine shut-down devices, and a check-list of automatic controls and alarms is given.—*Hinson, A. R.: Paper on "Defects in Marine Machinery and Associated Automatic Controls", given to a*

Conference on "Factors in the selection of marine machinery and plant with particular reference to reliability, maintenance and cost" held at the Institute of Marine Engineers, June, 1971; Proceedings, pp. 67–79.

Under water propulsion using cryogenic fuel

As compared with "conventional" hydrogen peroxide/kerosene submarine drives, a methane/oxygen fuel has a volumetric advantage of up to 20 per cent and a weight advantage of up to 50 per cent. One of the main advantages of cryogenic fuels, however, is the possibility of complete condensation of the exhaust components. For H_2O this process is natural, but disposal of CO_2 is not so simple. With methane/oxygen cryogenic fuel, the heat required for evaporation from the liquid stage can now also be used to condense the CO_2. The condensed exhaust gases can then either be pumped out, or stored for the highly important weight equilibrium in the case of submarines. A proposed underwater system is designed for an effective output of approximately 400 hp. Liquid methane and liquid oxygen would be stored in tanks of either glass or carbon reinforced plastics, noted for their tensile strength and stability, with much reduced weight, in the lower temperature range. To prevent diffusion of the fuels through the stressed fibre, the tanks would be internally coated, glass fibre with plastic, carbon fibre with thin metal. Pressure delivery is suggested, the pressure gas being produced inside the tank by an external supply of heat. An internal combustion working process is envisaged. After stoichiometric combustion the gases are mixed directly with and impart their heat to pure water which is converted to superheated steam and conveyed to a turbine. The supply of cooling water is regulated so that a gas/steam mixture at approximately 1000°C is fed to the turbine; approximately ten per cent combustion gas and approximately 90 per cent make-up water in closed circulation. The combustion gases are extracted from the condenser and conveyed to a storage tank and, therefore, go through an open process. The performance data are:

turbine output (effective)	410 hp
turbine speed	36 000 rev/min
turbine throughput	200 g/s
fuel throughput (methane/oxygen)	300 g/s
specific fuel consumption	0·25 kg/bhp h
steam generator temperature	1200°C
number of turbine stages	5
turbine outlet temperature	450°C
heat rate to sea water	510 kW=122 kcal/s
isotropic efficiency	45 per cent
overall efficiency	34 per cent

Walter, H.: Marine Engineer and Naval Architect, Jan. 1971, Vol. 94, p. 27.

Shore testing of steam turbine machinery

The state of steam plant reliability is reviewed and the factors contributing to operational unreliability are identified. The influence of unique environmental effects and the prevalence of interface and system integration problems suggest a broader approach than is offered by shore testing alone, although, in a development sense, it can form part of the process. The wealth of shipboard experience available offers a valuable data bank of reliability information. From this, a broad spectrum of corrective programmes can be derived, involving changes not only in technical matters, but in management practices as well. It is concluded that good reliability is available for modern steamships, but analysis of reliability data suggests many opportunities for improvement, and more stringent demands upon ships require such improvements. Clearly shore testing is not a panacea although, as a development tool, it can form part of an overall process. In generalizing upon experience, a number of factors stand out; for example the instances of unreliability are well distributed throughout the plant and unique shipboard environmental factors are strong influences.—*Rohde, E. C.: Paper on "Shore Testing of Steam Turbine Machinery, with Particular Reference to Reliability, Maintenance and Cost", given to a Conference on "Factors in the selection of marine machinery and plant with particular reference to reliability, maintenance and cost" held at the Institute of Marine Engineers, June, 1971. Proceedings, pp. 35–43.*