LLOYD'S REGISTER OF SHIPPING
№ 71, Fenchurch Street E.C.3
LONDON

# Lecture Delivered before the 1957 Conference of The Ship Machinery Manufacturers' Association in Japan

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M.I.Mech.E., M.I.Mar.E.

Held in Tokyo on 17th January, 1957

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T. W. Bunyan, B.Sc.(Eng.), M.I.Mech.E., M.I.Mar.E.,

Principal Engineer Surveyor for Research to Lloyd's Register of Shipping.

Mr. Chairman, Gentlemen,

I am deeply grateful that you have allotted me the lion's share of the time of this Conference, in order that I may speak about matters relating to the main propelling machinery of ships. The honour you do me is greatly appreciated, and I sincerely hope that you will feel at the end of the lecture that the time has been profitably spent.

By way of introducing myself and my work to you, I would say that I hold the privileged position of being head of the Engineering Research Department of Lloyd's Register of Shipping, with headquarters in London and a Research Laboratory at Crawley, in Sussex, England. I have a well-tried and experienced team of keen engineers, stress men, mathematicians, vibration and electronic experts on my staff, and some of us have the exciting jobs which take us all over the world, often at very short notice, to tackle engineering problems mostly connected with ships. The remainder have the less spectacular role of being more or less static, either in London or engaged in fundamental research work at the laboratory. The department deals with some 300 investigational jobs annually, and as many as five independent teams have been out on field work at the one time. Last year the department made 87 foreign flights ranging from Indonesia and South Africa to the western seaboard of Canada. It has been the policy of Lloyd's Register of Shipping not to withhold the services of the Research Department from anyone, whether the ship be classed with Lloyd's or not. It is a wise policy which has paid dividends. Fees charged for any work are usually based purely on actual cost. The activities of the department are not entirely confined to shipping but also embrace a very wide non-marine field. For instance, last year the department carried out complex stress analyses on the atomic reactor vessels at Calder Hall and on the very large supersonic wind tunnel at Bedford, to mention only two non-marine activities.

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I have dared to take as my subject "Main Machinery of Ships", which embraces a very wide field indeed. For reasons of time and in order not to weary you with the dull and monotonous sound of my voice for hours and hours (an affliction I would not wish on to my enemies, let alone my friends, which I hope you will all remain after I have dealt with you!), I shall try to highlight very briefly the tender spots in ships' main machinery as these have made themselves evident from our own experience. I shall be very happy to attempt to answer your questions, which may or may not deal with the particular problems I have selected. And I suggest that we might discuss each main group of problems as it arises.

## GROUP I

### Turbines

With the advent of the super tanker of 60,000 tons and more, in increasing numbers using machinery capable of delivering 20,000 s.h.p. and more continuously, the double reduction geared turbine installation is often the only practical choice. For smaller powers the supercharged oil engine is a very serious competitor. Given a good geared turbine installation, the reliability and low maintenance costs make it the obvious choice, particularly so with tankers, where the quick turn round and short time in port can be an embarrassment for the routine overhauls required by diesel main machinery.

I do not wish to be thought conservative in outlook, but as the number of turbine troubles in service appears to bear a certain ratio to the superheat temperature, it might be a wise precaution to adopt a conservative attitude towards high temperatures until more and experienced personnel are available to run these installations. By far the most common steam conditions met with in new installations today are 600 lb./sq.in. pressure and 850°F. superheat, giving a specific consumption of about ·55 lb./b.h.p./hour.

(a) *Bent turbine rotors*

Each year brings its crop of bent turbine rotors and, while priming of the boilers is at once suggested, followed by the alternative suggestion of careless warming through or cooling down, gland steam arrangements—particularly those supplied from bled steam when full away—can often be criticised for inefficient draining of the steam collector (which should preferably be automatic). It is a simple matter to straighten rotors by the thermal shock method, but experience has shown that in a few cases this has not been without hazard. Alloy steel h.p. rotors are preferably renewed rather than straightened.

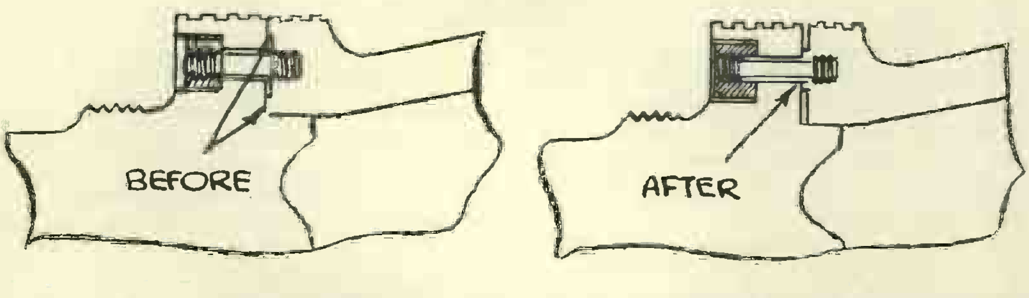
Much has been achieved towards resistance to thermal bending by the "gashed" rotor design coupled with resiliently mounted labyrinth boxes, which appear to be able to withstand without distress or distortion thermal shocks which would produce severe bending in "barrel" type rotors. Some designs employ a built-up construction where a large rotor is made up of two or more forgings shrunk, welded or bolted together. It may be interesting to describe a strange condition of apparent thermal instability which occurred with the design of a European manufacturer, which unfortunately involved several large tankers, two of which were out of commission for months. As the symptoms suggested thermal instability, both rotors were given repeated thermal stabilisation heat treatments, but on re-installation on board after careful rebalancing, were found to develop very serious vibration rendering them unserviceable. The owners finally called in Lloyd's Research Department in spite of the fact that the ships were not classed with Lloyd's. An examination was made of one of the rotors removed for the third time from the ship. At the joint, which was made in way of the dummy piston, an arc of fine rust was observed which gave the clue to the trouble.

It was estimated that the stretching length of the studs was such that an elongation of only one-thousandth of an inch was all that was necessary to produce complete loss of pre-stress in the studs. It was deduced that it was possible to create this straining by the thermal differential expansion of the rotor barrel at the point of entry of the steam, which would produce the pivoting of the barrel relative to the journal piece at the outer fitting strip of the joint, as shown in Fig. 1. By replacing the two strips with one in way of the pitch circle of the studs, this action was removed. The studs were also increased in length.

(b) *Turbine blading failures*

Since Stodola's classic work on blading, there has been a tremendous advancement in Cascade testing and instrumentation which enables a fairly precise fore-knowledge of the frequency response, critical excitation and damping of blading vibration under service conditions to be obtained. So it is not to be wondered at that blading failures in modern turbines are seldom encountered. They do, however, occur and priming of boilers is quite naturally the first reason suggested by the turbine designer as the cause of the failure. It is, of course, often true that a heavy slug of water will induce high mechanical stresses and even higher thermal stresses in the first impulse stages of an h.p. turbine. Indeed, the waviness sometimes seen on the trailing edges of blading is characteristic of this combined effect. Portions of heavy shrouding have been thrown when this happens and, while the turbine designer cannot be held solely to blame, certain design precautions can be taken, such as the elongation of rivet holes and a reduction in the length of continuous shrouding which would greatly reduce the hazard. It does, however, still happen that rows of reaction blading are critically excited, either by steam excitation, produced by the discontinuities in the throat area between the fixed blades on either side of the casing joint, or by claw coupling harmonics induced by malalignment between turbine rotor and pinion shaft.

Recently, in the case of five large passenger vessels, the first two expansions were wrecked after two to four years' service ; here again the turbine manufacturers were first in the field with a serious criticism of the boilers, which were blamed for heavy priming. I could not subscribe to this, as the condition of the turbine I examined did not suggest priming. An interesting series of tests was made on the boilers, in which a continuous 24-hour record was made during a week's voyage, of the wetness of the steam before entering the superheaters, also the temperature of the steam at the manœuvring valve, and in the h.p. turbine at the commencement of the reaction stages. The boilers were, in fact, unintentionally mal-operated during the tests, but at no time was there any suggestion of priming and the maximum wetness recorded before the superheaters, on one of the boilers, was only 1·5 per cent. This ruled out the boilers. During the voyage a record was made of the fore and aft movements of the turbine rotors, and it was found that where the trailing action of the propeller drove the turbine during manœuvring or quick throttling, the rotor would move forward smartly as was to be expected. On checking blade tip clearances and turbine thrust clearances, and making allowances for the differential expansion of the housing relative to the rotor, it was found that this simple fact was responsible and that very serious interference was produced between rotor and stator blading when the rotor moved against the forward thrust pads.




SHOWING MODIFICATION TO SPIGOT JOINT - L.P. ROTOR

FIG. 1

It is the fashion generally to chase that last ounce of fuel consumption and, with some designers, blading clearances and labyrinth clearances appear to be a happy hunting ground; in their burning zeal for economy they introduce a hazard that sometimes produces disastrous results, and in one fell swoop offset what precious little saving might have been achieved in the life of the ship by heavy repair bills, demurrage and loss of earnings, which are never enthusiastically accepted by either owners or underwriters. As in so many other marine applications, reliability must never be jeopardised for the sake of the odd 1 per cent gain in efficiency. Two severe rotor failures which occurred last year resulting from bending were directly attributed to too fine clearances.

(c) *Whirling*

It occasionally happens that an incorrect assessment of the rigidity of bearing supports produces a serious lowering of the whirling critical of turbine rotors. Indeed, some designers, aware of this uncertainty, deliberately lower the whirling to occur at 50 to 60 per cent of the running speed. This appears to be quite rational in that with a true and balanced rotor, the presence of the whirling critical will be undiscernible and after a period of service when some small distortion of the rotor occurs, the whirling will produce a slight roughness at this speed, which can be run through and easily avoided for continuous running, whereas in the case of the whirling occurring too close to the service speed, the same rotor distortion could produce a very large force with possibly disastrous results to the bearings. A very simple method of closely estimating the whirling critical on board ship is to excite the rotor into its transverse bending mode of vibration by gently tapping at mid-length with a rawhide mallet. The vibration is picked up by a small vibration pick-up, the output of which, suitably amplified electronically, is fed into an electronic analyser which will accurately indicate the frequency, which as a first order will bear a very close relationship to the true whirling speed.

An offshoot of the whirling problem is oil whip, which is not infrequently found in small turbines such as turbo generators. The symptoms are rough running, and oil whip can be diagnosed by a study of the vibration at the bearings. A sub-multiple of $\frac{1}{3}$ or $\frac{1}{2}$ the running speed indicates oil whip, which can often be cured by increasing the bearing pressure. If this is not possible, the loading can be obtained by suitably grooving the top half bearing. A useful paper has been read in the "Transactions of the American Society of Mechanical Engineers", and I shall be pleased to supply the reference to anyone interested.

(d) *Turbine vibration*

With large tankers or other large vessels with machinery aft, the turbines and condensers may be situated at an antinode for most of the transverse propeller-excited modes of hull vibration occurring either in deep or ballast draught. While such vibration can be easily tuned away from the service speed by judicious alteration of ballast, it may be a considerable embarrassment if such conditions occur in the loaded draught, as in this case no ballast tuning is possible.

The realistic step in the early stages is for the machinery contractor to have some small say in the design of the after body, aperture of the ship and also the propeller. Were he allowed to do this, then he could almost guarantee beforehand that no problem of machinery vibration due to propeller-excited modes of hull vibration would arise. Aperture clearances should be as large as practicable. The stream lines of the stern frame should be sharply tapered and the rudder should be as slim as practicable with a sharpish nose. Fig. 3 gives proportions of the significant aperture clearances which should be regarded as minima.

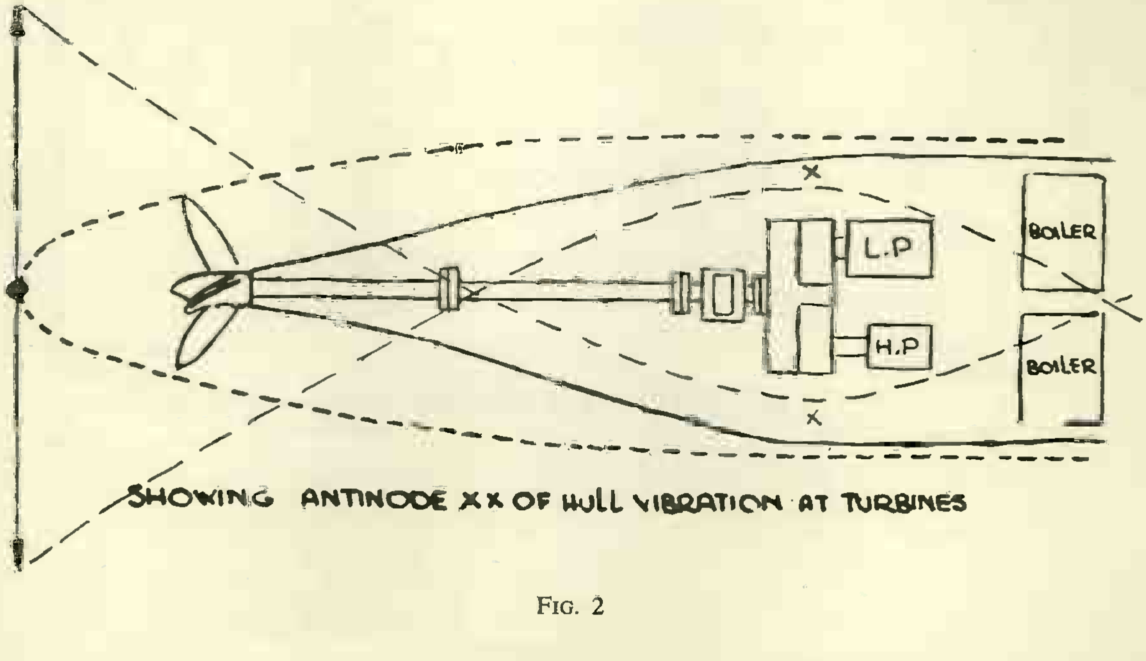



FIG. 2

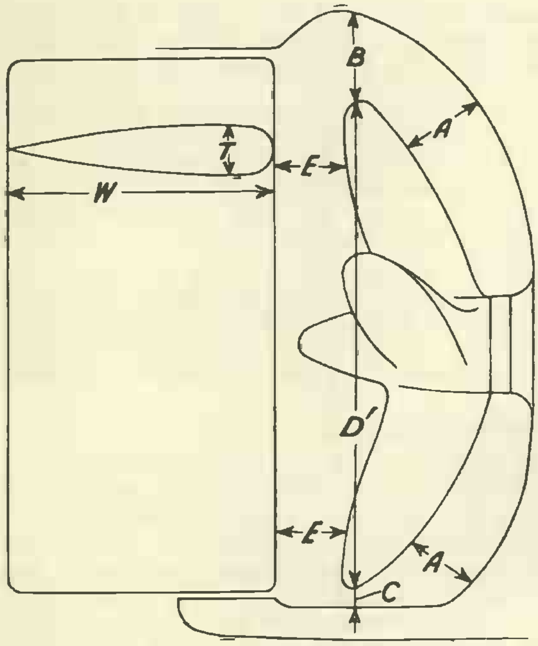



FIG. 3.—SINGLE SCREW APERTURE CLEARANCES

| Position | Clearances |
|---|---|
| $A=$ | $0{\cdot}15 \times D$ feet |
| $B=$ | $0{\cdot}10 \times D$ feet |
| $C=$ | $0{\cdot}03 \times D$ feet |
| $E=$ | $0{\cdot}08 \times D$ feet or $T$ feet, whichever is the greater |

Some of the serious troubles which have resulted from excessive vibration in way of turbines and condensers have been:—

(i) Failure of main steam pipe supplying astern turbine. It may be advantageous to have a flexible connection to main steam pipes adjacent to h.p. and astern turbines, as this will considerably reduce the dynamic bending stress in the pipes produced by vibrating turbines. The indiscriminate staying and securing of vibrating steam pipes can apply dangerous bending stresses to the pipes and flanges.

(ii) Progressive damage to pipe lagging.

(iii) Damage to tube plates and slackening of condenser tubes.

(iv) Straining of l.p. turbine casing producing heavy rub of shrouding. In this case the transverse vibratory movements of the condenser relative to the ship's structure were about $\pm$ 1 mm. The condenser was finally strutted at the spring supported feet to the ship's side, permitting flexibility in the vertical plane but rigidity transversely. A number of other ships have been similarly dealt with because of excessive condenser vibration.

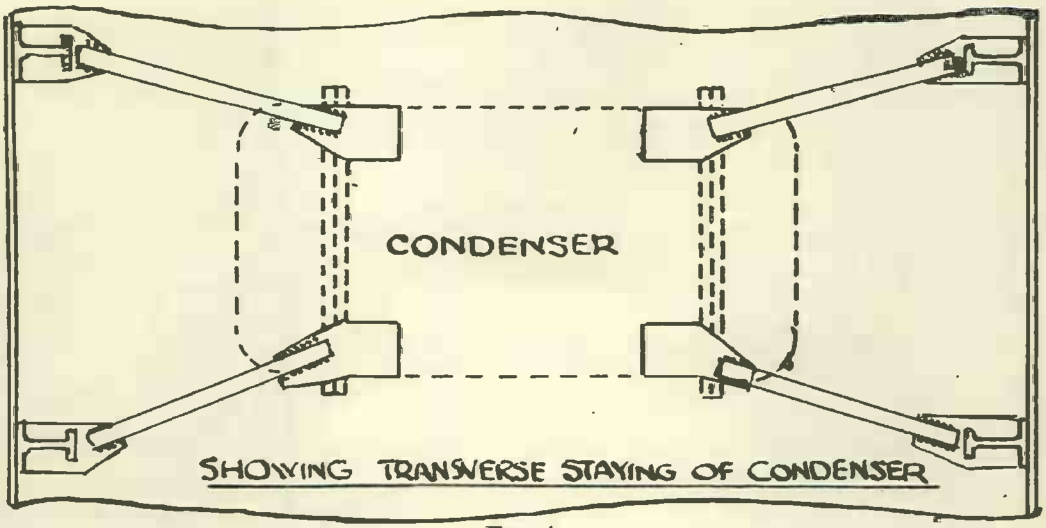



FIG. 4

# GROUP II

## Gearing

I now come to what could be the greatest anxiety of all—main reduction gearing. Gearing failures which occur from time to time are often disastrous, laying a ship up for months. Tankers mostly appear to have been involved.

I am not fully conversant with the overall picture of Japanese gearing, but have the impression that while there has been a certain amount of anxiety on account of excessive noise and pitting, the actual failures which have occurred in service have been very few indeed.

Commencing with design, there is much that can be done to guarantee optimum conditions for reliability in service. The articulated drive, where primary and secondary trains are coupled through flexible quill shafting, is to be preferred, and is in fact now most commonly adopted, as against the nested type where primary wheels and secondary pinions are rigidly coupled. The articulated design is more costly and of greater overall dimensions and weight, but has the considerable advantage that the dynamic stresses applied to the gear teeth resulting from unavoidable gear cutting inaccuracies are greatly reduced by the quill shaft flexibility.

The choice of materials and tooth loadings is another problem which has caused a lot of grief. A very good combination which has been widely used for years is a $3\frac{1}{2}$ per cent nickel steel pinion working with a ·35–·45 carbon steel wheel. The permissible tooth loadings are naturally limited by the use of the soft wheel material. Usual "K" factors with this material and the articulated design are 80 for primaries and secondaries. The materials are completely compatible and have the considerable advantage that slight undulations and poor surface finish will be bedded down fairly early in the life of the gear without producing serious pitting or scuffing. Further, large forgings such as gear rims present no difficulty as to obtaining metallurgical consistency and soundness. There is naturally a trend towards harder materials, for which the classification societies permit increased "K" factors in direct ratio to the hardness; and where high precision gear cutting and grinding machinery are available, the ultimate may be reached with hardened and ground gears. Designs employing an all-welded construction in which the wheel rim is welded to the diaphragms are preferable to the shrunk on rim. Such designs, however, call for advanced welding techniques where alloy or ·45 carbon steels are used for the rim, and the stress relieving heat treatment must be good in order to avoid rim distortion due to hobbing.

The use of fine tooth claw couplings is now common practice. Here again the choice of materials is important and the claw teeth should have a hardness approaching 300 and 350 Brinell before these couplings will work trouble free with malalignment. It is, of course, appreciated that there should be no malalignment between turbine rotors and primary pinions, but even assuming that by careful measurements and tests it can be established what the cold alignment should be, to give true alignment when the running temperature is reached—quite appreciable variations in alignment occur with different loading conditions of the ship.

The efficient lubrication of claw couplings is important. Utilising the oil spill over from a bearing is not sufficient; a separate nozzle pipe—often supplied from the bearing well—should be provided in all cases.

Naturally, the accuracy of the gear cutting machine largely determines whether an excellent design of gear will be satisfactory in service or not, and a good quality machine of proved and certified accuracy is a very sound investment. The author's personal preference is for a solid table type of hobbing machine having two independent drives to the master wheel. The one has a robust worm and worm wheel of coarser pitch which will be used for the roughing cuts, and the second a worm and worm wheel having as large a number of teeth as practicable (in the region of one thousand for a 120 inch machine) fully corrected and of high accuracy, which will be used for the finishing cut only. The machine should be installed in a thermally insulated house without windows and air conditioned with a temperature control that actually controls the internal ambient temperature to $\pm 1\frac{1}{2}$°F. The hobbing machine should be mounted on a concrete raft of several hundreds of tons in weight and isolated from the rest of the floor structure. The site of the building should be well away from any service generating earth tremors such as heavy forges, drop hammers and the like, or railway lines over which heavy locomotives and rolling stock are moved.

By this means two very important hobbing errors will be effectively dealt with; the first being hobber or "worm wheel cyclic" errors and the second "diurnal errors", both of which are magnified both in amplitude and wave-length by the larger diameter of the cut gear. By choosing a large number of teeth in the master wheel of the hobbing machine, a very important reduction in the undulation wave-length is achieved: this produces three important advantages. Firstly, the undulation amplitude is automatically reduced, secondly, the noise resulting from hobber errors in the gear will be approaching supersonic frequencies, and last but by no means least, post hobbing processes such as shaving or lapping can be completely effective in removing these undulations, which is not the case where the undulation wave-length is large, i.e. more than 3 in. produced by a master wheel with fewer teeth. The diurnal error, that is the error caused by day to night changes of temperature, is, of course, independent of the number of teeth in the master wheel and of the quality or condition of the gear hobbing machine, and will be of the most serious proportions for machines housed in the open shop when cutting large gears. On a large bull gear a temperature variation of $\pm 2\frac{1}{2}$°F. will be clearly evident as a noisy gear-frequency about octave "C" at full power, and will most probably cause pitting at the crest of the diurnal undulations.

Fig. 5.—Main wheel showing diurnal undulations ("daily bands")

Fig. 6.—Showing pitting below pitch line on main wheel (forward end, after helix)

While pitting is in itself largely a self corrective process normally and may take several months to stabilise, in cases where certain hard materials are necessary because of high "K" factors, once pitting commences due to any reason, such as undulations or malalignment, then there is a good chance that no stability will be reached and the gear will have to be replaced.

Scuffing of gear teeth, which may show up in the first few running hours on load, is always a cause for anxiety; the process is often non-stabilising and will progressively damage the gear teeth. On the assumption that a normal tooth profile has been used with normal slide roll ratios and normal pitch line speeds, then corrective measures are usually straightforward. Lack of tip relief is most commonly the reason—usually on pinions where it is a simple matter by smooth filing and stoning to re-apply the necessary relief, the greatest care being taken that the relief is not more than about ·002 in. at the tip being washed away to zero at about 1/5th of the tooth height from the tip. Excessive tip relief will considerably increase the specific load on the gear teeth. At the same time the lubricating oil should be changed to an E.P. oil and the load on the gears limited to about 70 per cent for a period of weeks for bedding in purposes. It is, of course, concluded that magnetic filters are installed in the lubricating oil system, this being a "must" anyway.

Incidentally the lapping process can remove tip relief quite effectively and Fig. 8 shows the serious scuffing produced entirely through this means.

Fig. 7.—Scuffed second reduction pinion (7/10-in. deep tooth): astern

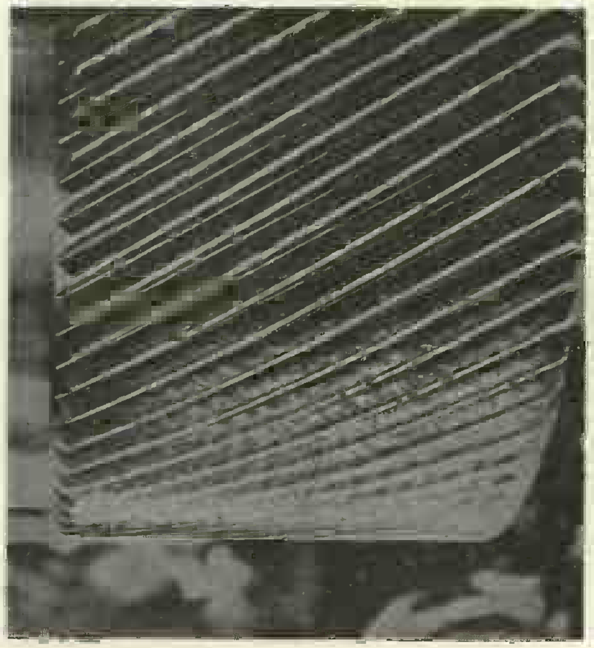


FIG. 8.—SCUFFED A.A. L.P. FIRST REDUCTION PINION SHOWING UNDULATIONS.

There is no need to underline the vital necessity for good bedding either in alignment or matching of helix angles. With indifferent bedding the specific load can be increased enormously resulting in collapse of the tooth profile through surface fatigue and producing a highly dangerous bending fatigue condition in way of the roots of the teeth. Where localised pitting is observed at the ends of the helices, a very careful magnetic crack test should be applied, as the chances are quite good that one or more teeth will be found cracked at the loaded ends of the helix. When this happens, all is not necessarily lost, in so far that it may be possible to keep the ship in service, depending on the extent of the damage, by removing the affected zone circumferentially or by realignment of the journals as the case may be, which would enable the ship to remain in service at a power which must be reduced in direct ratio to the extent of the damage, until such time as a replacement gear is available.

One classic case occurred last year where, due to tooth breakage and cracking of one of the main wheel rims of the bull gear on a large tanker, the ship was maintained in service at 60 r.p.m., about ten knots for a round voyage Continent/Persian Gulf, by turning off the damaged helices of both secondary pinions and attaching a temporary thrust arrangement secured between the free ends of the pinion journals and the bearing brasses. In this case the rim material was at fault metallurgically in that pronounced sulphide inclusions resulting from the hardening process were found to be present in way of the roots of the teeth. The steel used was silicon-manganese steel (C 0·46, Si 0·9, Mn 1·0–70–85 Kg/mm.[2]). The vessel was not classed with Lloyd's Register. The temporary repair was carried out in 14 days with little or no local facilities and reflects great credit on the engineers of a Continental builder, who were flown out to the vessel. This case is referred to in a recent paper by my colleague, Mr. S. Archer[2].

A direct result of hobber errors and incorrect tooth profiles is noisy operation. While it would be difficult for classification societies to specify maximum permissible gearing noise levels, it can be said that an accurate gear correctly aligned in a welded steel gear case should not generate more than about 102 decibels of noise at a distance of 3 ft. anywhere around the gear case. With a cast iron gear case this figure should be more easily achieved. Unfortunately, noise levels normally exceed this value by 3 or 4 decibels (an increase of $2\frac{1}{2}$ times in the noise intensity) and there have been cases where the level has been as much as 118 decibels (an increase of 40 times). This is quite unjustifiable and means that either the owner will be seriously embarrassed in keeping engineers, or he will be faced with serious compensation claims because of resulting deafness of engine room personnel. As the bulk of the noise will be either hobber or diurnal error in primary or secondary trains or tooth contact frequency, the noise will be piercing and psychologically most unpleasant because of the frequency range, 800 to 3500 cycles per second, to which the human ear is most sensitive. Incidentally, I shall probably be called to book by some of you for quoting decibels and not "phons" when talking about noise levels, but I have a reason for thinking in terms of decibels in that when dealing with gearing errors one disregards in the first analysis the physiological factors and concentrates on the cause of the noise. That being the case, "weighted" networks must not be used with the microphone amplifier—a flat characteristic is essential for obtaining relative sound intensities in order that one may be able to discover the predominating cause of the trouble. It is most remarkable with what accuracy a good quality noise analyser will break down a noise spectrum from a gear. Each component will be exactly recognisable from a knowledge of the kinematics of the gear cutting machine and the gear itself, including the claw couplings.

I have purposely refrained from making reference to hobbing machines with "creep" drive to the master wheel because I believe the only contribution—an important one—a "creep" machine makes, is in producing a "scrambled" noise spectrum as against the purer and more penetrating tones of the solid table machine. This is accomplished by dispersing the worm cyclic error in spirals around the gear instead of axially as is the case with the solid table machine, but in doing this a further error is introduced resulting from the worm wheel to creep ring cumulative pitch error, which is often a worse error than the worm cyclic error of the solid table machine and certainly very much more difficult to eliminate. Another bad feature is that the crests of these long undulations are substantially in line circumferentially, which increases the possibilities of local overheating and breakdown of oil films with resulting scuffing.

As the geared turbine is becoming increasingly installed as larger and larger powers are demanded, and will come into even greater prominence when atomic fuels are available, it would be a very sound investment to achieve a high standard of gear cutting accuracy, and every encouragement, governmental or by collective effort of the industry itself, should be given towards this end.

Before leaving the subject of gearing, a few remarks on torsional and axial vibration may be of interest. It can be said with almost complete assurance that the torsional and axial vibration problem in geared turbine installations is reduced to normal proportions by good propeller and aperture design. There was a very spectacular demonstration of this when we were called to investigate a serious hull and gearing vibration problem on two large tankers. Both problems were severe. The hammering in the gearing at the 1-node critical was impressive and covered a speed range three times what would be normal. The after part of the ship was complete bedlam with propeller-excited vibration. The ship was drydocked and on our recommendation the rudder nose was cut back as shown in Fig. 9, in order to increase the excessively small trailing edge clearance. The results were most satisfactory.

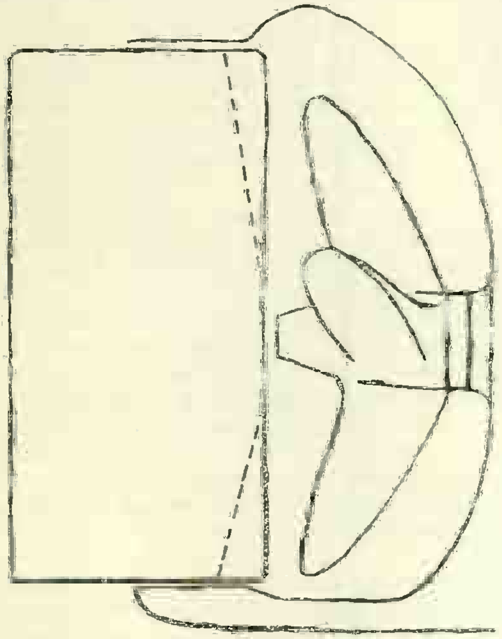


Fig. 9.—Rudder nose cut back to increase propeller trailing edge clearance

One final matter before we start asking one another questions: In any study of gearing and the control of the quality of gearing, it is necessary to have apparatus designed specifically to do these jobs. The first essential, in my opinion, is a good undulation recorder. I have brought along Dr. Tomlinson's N.P.L. undulation recorder, which is straightforward to use. Fig. 10 shows the instrument in question, and I would again refer you to Mr. Archer's paper, which details the simple formulæ for setting the instrument and assessing the magnification factors.

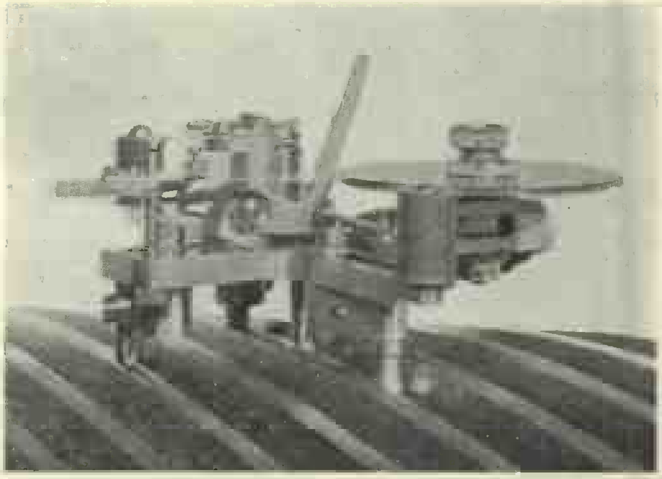


Fig. 10.—Tomlinson undulation recorder (N.P.L. type)

There are sundry other gauges which should be part of the equipment of the gauge room:—

(a) Cumulative pitch gauge.

(b) Pitch to pitch gauge.

(c) Axial pitch gauge.

The reason I have given the undulation recorder prominence is that, in my experience, it is the undulation error which largely determines a dangerous or noisy gear, taking for granted of course that helix angles are matched, tooth profiles are accurate and bedding is good. With modern hob production and the post hobbing method of shaving, very high accuracies are obtainable both in tooth profile and pitch to pitch accuracy.

Complementary with good gauge room practice is the ability to assess the quality of the cut gear when operating under load. For this purpose I would recommend a noise level meter and sound analyser. There is a wide range of these electronic instruments of varying qualities and cost commercially available, and I would be happy to supply this data to anyone interested.

The above would be a good start towards producing good gearing. For the further study of dynamic loads and stresses applied to gearing as a result of gear cutting inaccuracies, there is quite a selection of audio frequency amplifiers, recorders and the necessary torsional and axial vibration pick-ups. A lot of our own work in this field was published about ten years ago before the North East Coast Institution[8].

# GROUP III

## Diesel Engines

(a) The diesel engine has perhaps taken the place of the old reliable triple expansion engine and is more often used for propelling ships than any other type of prime mover, which in itself explains why there are generally available more engineers with diesel tickets than steam.

The modern trend with advanced supercharging has extended the range of the diesel engine, which will soon become a serious competitor with its superior economy—up to 15,000 s.h.p. and more. But, like all machinery, the diesel engine has its tender spots, although I must admit that with the art and artifice of the diesel designer, such damage as may be caused is usually closely confined and even a crack in a crankshaft does not cause the frightful havoc that one single broken tooth can do with a geared drive. The latter often entails a towing job from some remote part of the world, followed by a lengthy stay in port while there is a general scramble around for forgings, followed by a further scramble to find a suitable gear cutting machine that is not solidly committed day and night for the next three years. Think how simple it would be if there were standard designs for gearing throughout the world, as there are for crankshafting of proprietary types of diesel engine. I beg your pardon for this diversion.

With large oil engines the trend is for fully fabricated designs, and one hears surprisingly often the downright criticism that this is a retrograde step, particularly with bedplates. I personally do not subscribe to this, but can appreciate the cause of such criticism. Cracking in fabricated bedplates is fairly common but is often much more alarming in appearance than of any consequence practically. It is generally agreed that where the various bits and pieces which go together to form the bearing girder are assembled by a skilled fitter using a ·002 in. feeler as a criterion of fit, following by an intelligent welding procedure followed by effective stress relieving heat treatment, then the end result would be as reliable as a heavy casting, cheaper to produce and maintain in service and lighter in weight. One of the tenderest spots in a fabricated bedplate is in the attachment of the bearing pocket, particularly where this is a heavy element flame cut from a slab, and must be attached to fairly light diaphragm plates and ribs. As often as not the welding is cracked as soon as completed. The stresses applied to the structure due to gas load are usually quite nominal as designed, but where crankshaft journals are not true but have considerable helical and conical errors (often introduced unwittingly during the turning operations in the lathe), then the eccentric dynamic loads which can be applied to the bearing pocket may multiply the nominal designed value by several times. It is very common indeed to find the bearing white metal cracked and for the bearing itself to be barrelled where damage is found to the structure in way of the pockets. Malalignment of the crankshaft due to original sin or to unequal wear down or hammering of the chocks is another cause.

Cracking of the landings and pockets for through bolts is often a matter of poor design initially, but here again careful set up welding and stress relieving would pay dividends. It is now quite common practice to use cast steel girders welded or bolted into the bedplate structure.

(b) *Crankshaft failures*

In this day and age, with the careful check automatically carried out by leading classification societies of the torsional vibratory characteristics of the dynamics of the shafting installation, there should be no failures directly due to torsional vibration, because such vibratory stresses are predictable accurately, can be checked by actual measurement during trials, and are such as do not alter with time—unless some alteration is made for any reason to the inertias or stiffness of the shafting, such as changing a propeller. But here again, such action should be taken only with the approval of the classification society concerned. However, failures do occur, infrequently, and are one of those "gremlins" that diesel designers worry about—assuming they are the worrying type. In the number of failures of large crankshafts investigated by us, there has most often been some extraneous factor involved, such as corrosion pitting, malalignment, knifing of fillets or oil holes. Corrosion pitting of fillets sometimes occurs with engines burning boiler fuels. The increase in the stress concentration effect under these conditions of corrosion pitting may be large indeed. An interesting feature about such corrosion is that very often it is selective in character and confined to the underside of the pin or journal in forged shafts. This is due to two factors; one is that the combined stress condition here is several times that in any other part of the fillet, and the other is that the internal locked-up stresses remaining in the forging as a result of the heat treatment are also concentrated in this position.

Possibly the predominating cause of trouble with crankshafting is unequal weardown of bearings, which normal maintenance should automatically rectify. Sometimes severe malalignment is induced by the failure of a main bearing through wiping or hammering out, and it can often be proved that the initial bearing failure has been brought about by excessive load caused by malalignment or by crankshaft errors, or by hammering in of the bedplate chocks, sometimes found deeply fretted into the tank top or a temporary breakdown of the lubrication. In tanker installations it is often necessary, for torsional vibration considerations, to instal massive intermediate shafting in order to raise the major critical of 4 and 5 cylinder two-stroke engines above the running speed. A number of crankshaft failures have occurred in way of the aftermost crank due to the serious malalignment between line shafting and the heavy intermediate shaft. Crankweb deflection readings are a reliable indication of the alignment of crank shafting in almost any engine, with the exception of the Doxford type engine, which is a very popular engine in Europe. Here, due to the complex nature of the crankshaft web, deflection readings are difficult to interpret.

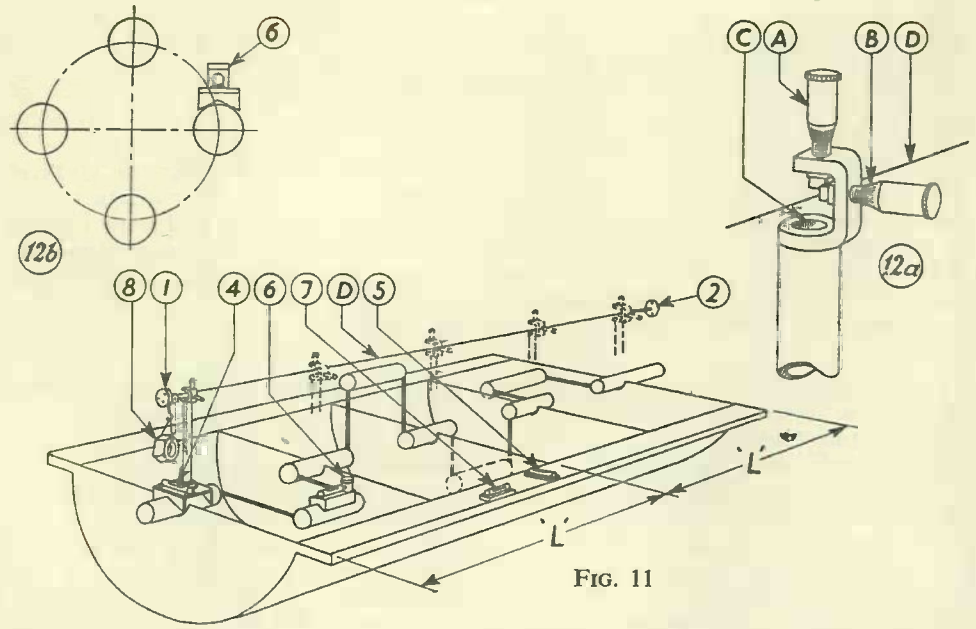



FIG. 11

*Alignment of Journals*

Piano wire 0·018 in. (*D*) anchored to endplates of engine entablature over pulley at one end (*1*) and fixed block (*2*) at other; height of wire set to be approximately the same at each end and in approximately the same vertical plane as crankshaft centre line by micro readings. Micro staff set approximately vertical by spirit level (*4*) to be read only when master level (*5*) reads "zero". Master level left undisturbed throughout tests—note micro readings for only just making contact with wire when bulb (*C*) lights up. Repeat for $X_2$ $X_3$ $X_4$ $X_5$, etc.

*Checking Crankpin Errors*

Take clinometer readings (*6*) when mounted on vee block on pin—the clinometer to be read only when master level (*7*) reads zero—take readings on the four quarters as shown in Fig. 12(b).

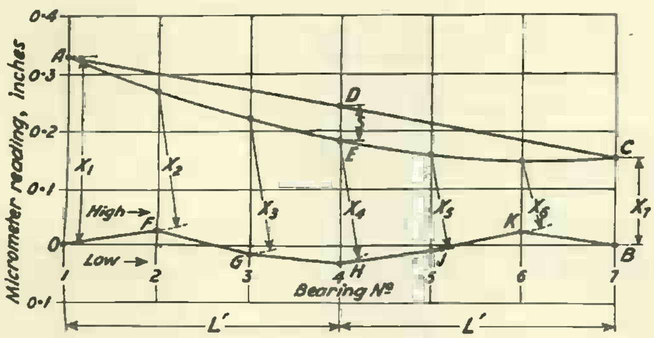
*Important*

(1) Weight (*8*)=42lb.

(2) Spirit levels (*4*) (*5*) (*6*) (*7*) sensitivity=0·001 per ft.

(3) Piano wire is 0·018 in. diameter.

(4) Sag of wire to be allowed for= $\frac{L^2}{8}$ thousandths of an inch where $L$=half length of engine (feet).




VERTICAL ALIGNMENT OF CRANKSHAFT

NOTE:

1) Position of bearing centre line plotted out to scale—1, 2, 3, 4, 5. 6, 7, etc.

2) Actual micrometer readings $X_1$ and $X_7$, i.e. first and last bearing, plotted *OA* and *BC*.

(3) Join *A* and *C* (set wire so that *OA*=*BC* approximately).

(4) At point corresponding to mid length of crankshaft *D*, drop down *DE*=*S* inches=the sag of the wire. $S=\frac{L^2}{8}$ thousandths of an inch where *L*=half length of engine in feet.

(5) Draw in smooth curve *AEC* showing sagging wire.

(6) From intersections of curve *AEC* with bearing centre lines, drop $X_2$ $X_3$, etc., actual micrometer readings.

(7) Then *O*, *F*, *G*, *H*, *J*, *K*, *B* is actual bearing alignment; those above *OB* are high, below *OB* are low.

(8) *Important*. Wire is 0·018 in. diameter piano wire, tension 42lb.

A simple and accurate shaft or bedplate alignment indicator was developed by the department which is now widely in use in Europe on all types of engines, but is of particular attraction in the case of the Doxford, where a set of alignment readings can be obtained in two hours, using the access to the journals provided by the oil supply holes in the bearing keep. The figures show the arrangement.

(c) The top end bearing, as many of you may know, can, without any apparent reason, develop the most alarming symptoms. Overheating or hammering out of the white metal is a common feature, and renewing the bearing, dressing up and re-surfacing the pin will not produce the desired results. This trouble can develop in new engines or in engines that have given trouble-free service for years. The cause of the trouble, however, more often than not does not lie in the crosshead, but is brought about by eccentric loading or uneven loading of the crosshead pins; in some double acting engines this eccentric loading does not affect the crosshead bearings but fractures the piston rod. Such eccentric loading is caused by "weaving" of the bottom end bearing—by that I mean the backwards and forwards movement of the bottom end bearing from web to web. There will be evidence of bright marking on both webs which will confirm this action. There are three factors, all associated with the crankshaft itself, which can cause this condition. In a new engine, helical or conical errors in the crankpin resulting from inaccurate machining are one factor. Fig. 12 will demonstrate what I mean.

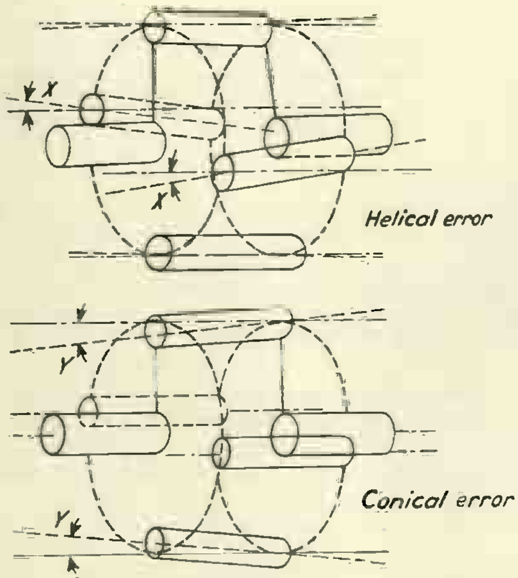



FIG. 12.—SCHEMATIC DIAGRAM OF CRANKPIN ERRORS

While the engine is in the shop, these errors are checked by sensitive spirit level and a vee block. With engines that have run satisfactorily in service there are two factors to look for, the first is malalignment of the crankshaft due to unequal wear down, particularly so if No. 1 top end is involved as No. 1 main bearing rate of wear down is less than the others and so is that of the last bearing adjacent to the thrust. The last factor to look for is the barrelling wear pattern of the crank pin. Once the cause is established it is a straightforward matter to proceed with the rectification.

Main bearing failures may involve alignment and this is the first thing to check. In a new engine hot main bearings may be indicative of crankshaft journal errors, and an excellent way to gauge these is to proceed as indicated in Fig. 13.

While the quality of white metal is one of the first things criticised when a bearing failure occurs, it is seldom in fact responsible. There are, however, occasions when the metal is at fault. One recent case of top end bearing failure, which I will refer to again later, was in fact associated with the quality of bearing metal, which was found covered with fine fatigue cracks in way of the bearing area. An analysis showed serious lead contamination, which greatly lowered the fatigue resistance of the metal, which was also not adhering properly to the bearing brass. Thick white metal linings are less reliable than thin ones.

(d) Each year brings its crop of fractured or cracked pistons and liners, and Lloyd's statistics show that Japanese production is not free from these troubles. The most favourable conditions for producing such failures are at the end of a voyage; when approaching port the speed is reduced to dead slow or stop, and super-cooling of a hot engine occurs unless precautions are taken to ease off the piston or jacket cooling water. While much can be done to reduce thermal or physical stress concentrations in way of necessary discontinuities, such as strengthening ribs, valve bosses and openings, the personal element in operation is a big factor where thermal cracking is concerned, as has been demonstrated so often by entirely satisfactory and trouble-free performance on some ships, but continual trouble from this cause on others with identical installations.

(e) The failure of bolts holding together the running gear of diesel machinery is a matter that is accepted with philosophic calm, yet a critical examination of the bolted assembly will indicate that the factors of safety can be very small indeed. It can be demonstrated that the dynamic load to be carried by a bolt is dependent on the elasticity ratio of the bolt to that of the bolted assembly. A good design criterion is that the bolt should be about four times more elastic than the bits and pieces it is holding together. With top ends and bottom ends, for reasons of space and lightness the scantlings of the bolted assembly are kept down to a minimum with the resulting reduction

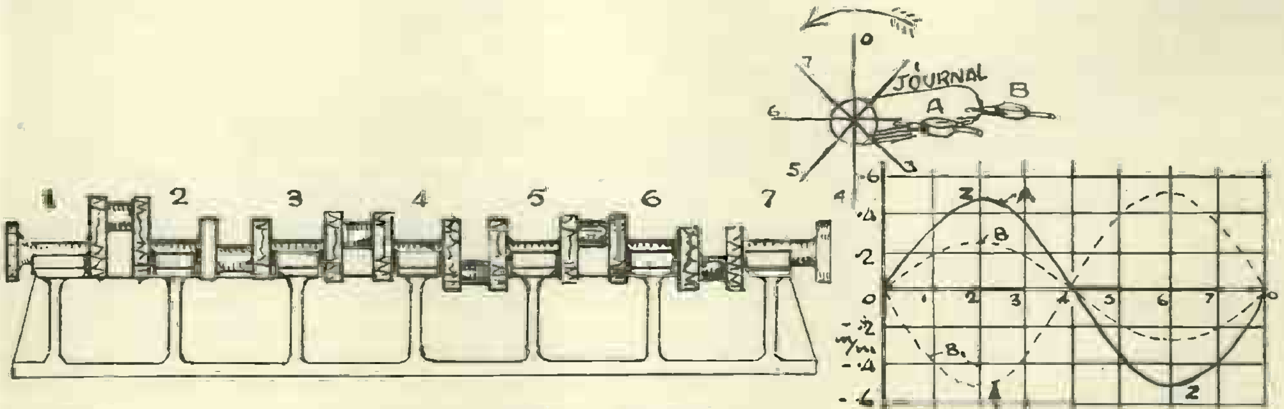



FIG. 13

*Procedure:* Starting with No. 2, remove bottom half main bearing, fix clock gauges "A" & "B" horizontally against No. 2 journal as shown, plot readings from "A" & "B" for 45° rotation of crankshaft as shown—difference between readings from "A" & "B" should be less than $\frac{D}{5000}$ m/m, where D=journal diam. m/m. total throw "Z-Z" to be less than $\frac{D}{2000}$ m/m, curve $B_1$ denotes serious wobble of journal—repeat for journals 3, 4, 5, 6, excessive errors to be filed and lapped out progressively commencing at No. 2 journal in the rigidity of the assembly. It must be remembered that the fatigue strength of a well designed nut and bolt in reversed tension is about 3½ tons per square inch, and to obtain this value the greatest care is taken in the mounting of the bolt in the testing machine to avoid eccentric loading. In practice such a bolt will be used to transmit 5 tons per square inch gas load, which is quite safe if the elasticity ratio of bolt to assembly is 4 to 1, which will give a working load in the bolt of only one ton per square inch. With slight eccentricity of loading the fatigue strength of the bolt may be halved, and with slight undertightening the initial elasticity ratio is seriously reduced, so it is not to be wondered at that each year a large number of dynamically stressed bolts will fail.

With large bolts, particularly where access is cramped, it is very difficult to get sufficient pre-stress by the use of the flogging hammer. A simple fitment shown in the next figure enables the correct pre-stress to be obtained.

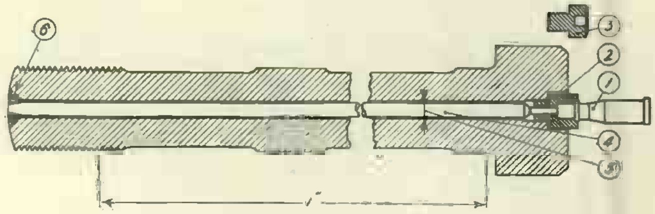


FIG. 14.—MICROMETER GAUGE FOR INDICATING BOLT STRAIN ($=0{\cdot}0006 \times L$ INCH FOR M.S. BOLTS)

(1) Standard 0-½ inch micrometer head; (2) brass bush screwed ½-in. Whit. into bolt head; (3) steel ½-in. Whit. socket screw for plugging hole; (4) ¼-in. diameter ground silver steel rod; (5) $\frac{9}{32}$-in. diameter hole drilled the length of the bolt; (6) rod inserted and welded or brazed there.

There is one other matter I would finally raise in regard to bolted connections of bearing halves. The top end bearing trouble I mentioned previously was considerably aggravated by distortion of the bearing by tightening the bolts

# GROUP IV

**Propellers and Shafting**

I have already stressed the vital importance of having generous propeller aperture clearances with single screw ships, and think it would be valuable in this final section of my talk to catalogue the most undesirable effects which result from inadequate propeller aperture clearances.

1. Increased cyclical torque at propeller bladed frequency. In a well designed aperture and propeller arrangement, the torque variation may be less than 4 per cent, whereas with tight clearances this value may be trebled. The effects of this will be heavy hammering of gearing in the vicinity of the propeller-excited torsional vibration critical speeds. Transverse rocking of the turbines, gearing and condensers, caused either directly by torque variation or by propeller-excited transverse hull vibration, may sometimes reach severe proportions and may be a considerable embarrassment where such vibration is critically excited in the vicinity of the service revolutions. In diesel driven ships—tankers in particular—propeller torque variation has been known to produce severe transverse rocking modes of vibration, and indeed it is quite common practice to secure large engines of 8 cylinders and upwards to the ship's side framing at the level of the top of the cylinders or entablature.

2. Small aperture clearances seriously increase the propeller dynamic thrust forces. In one case recently investigated the propeller-excited axial vibration was so violent that serious damage was being done to the main reduction gearing. In this case the design of the propeller was also involved, and I am pleased to say that not only was a complete cure obtained with a re-designed propeller, but a useful increase in the ship's speed was thrown in for good measure. The original and the new propeller were both four-bladed designs, but the new propeller was designed specially to reduce the dynamic axial thrust component. Incidentally, the five-bladed propeller was introduced a few years ago specifically to reduce axial vibration, which was then being experienced in some super tankers in which the main thrust bearing was built into the forward part of the main gear case. The lowering of the thrust impedance was, I understand, sufficient to cause serious axial vibration at the service speed. The five-bladed propeller achieved two advantages over the four-bladed propeller. Firstly, the speed at which axial resonance would occur was reduced as 4 to 5, also the dynamic thrust component at propeller blade frequency causing the vibration was approximately half of that with the original four-bladed propeller.

3. Reference has already been made to the heavy propeller-excited hull vibration which can be produced by a propeller working in an aperture having small clearances. Some work published about 18 months[3] ago has indicated that these forces may reach values of $\pm$ 16 tons applied to the most effective part of a ship, i.e. the stern, for causing hull vibration.

4. Small aperture clearances, as would be expected, can greatly increase the applied bending stress in a propeller tailshaft assembly, which can reach dangerous proportions at the driving face of the keyway at the large end of the cone. It has been established[4] by strain gauge measurement that these stresses can reach dangerously high values, particularly when a single screw vessel is being driven hard in bad weather. This is a particular danger with turbine driven ships where there is no apparent racing of the engines due to the very large flywheel effect of the l.p. turbine, yet the shafting and gearing may be subjected to very high overloads. The five-bladed propeller is a worse offender than the four-bladed propeller from a consideration of tailshaft bending stresses. In some large tankers which have suffered rapid tailshaft failures and heavy erosion damage to tailshaft liners (*see* Fig. 15), it has been found that the transverse bending mode of vibration is either coincident or in close proximity to the service revolutions.

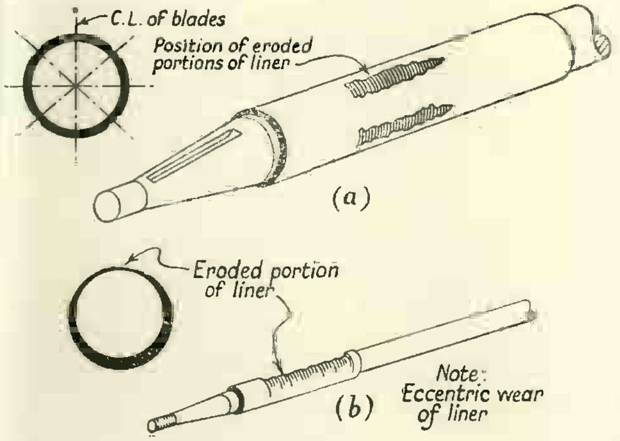



Fig. 15

Recommendations have been made for replacing five-bladed propellers with four-bladed in these cases. Incidentally, erosion grooving in distinct patterns corresponding to the number of blades is an indication that such resonance is occurring near the service revolutions. It is interesting that such vibration seldom shows up at the trials, i.e. when the stern wood is new and not bedded in to the running alignment of the tailshaft. It sometimes takes six months or more before this transverse tailshaft vibration shows up. It is easily distinguishable, as its symptoms are heavy pounding in the stern bearing, particularly when the ship is pitching slightly. The tailshaft coupling or sometimes the last plummer block bearing will be observed to be moving at propeller blade frequency (i.e. number of blades × r.p.m.).

5. Insufficient aperture clearances will enable a propeller to cavitate more easily. Erosion pitting of the rudder nose is also aggravated.

## CONCLUDING REMARKS

Mr. Chairman and Gentlemen:

I hope that I have not left a feeling of deep gloom and uncertainty. I do not want to give the impression that Japanese ships are heir to all of these depressing weaknesses. They may be particularly free from most. But as and when these troubles arise, it is comforting to know that there is no mystery about them. With the use of modern instruments and the experience of engineers well tried in the field of trouble—in which grade I class myself and my team—a solution, often a surprisingly simple and practical one, is quickly found.

I have to thank you, Mr. Rinoiye and your Committee, for inviting me to speak, and I have to thank you gentlemen for being one of the best and most enthusiastic audiences I have ever addressed. Your questions have been important ones, and I hope my answers have been adequate. I apologise that there has not been time to have a Japanese translation made of this lecture before the meeting, but hope I shall again have the honour and pleasure of taking part in another of your excellent conferences.

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Printed by Lloyd's Register of Shipping
at Garrett House
Manor Royal, Crawley, Sussex, England