STERNGEAR DESIGN FOR MAXIMUM RELIABILITY THE GLACIER-HERBERT SYSTEM

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In recent years the growth in size and power of ships has revealed a number of weaknesses in conventional sterngear. At the same time, the emergence of the mammoth tanker and expensive special purpose ships has emphasized the need for utter reliability to avoid delays, diversions and loss of hire. Against this background, the paper reviews the sterngear function as a whole, and describes the authors' philosophy as marine engineer and bearing specialist respectively, in developing the concept of the Glacier-Herbert Sterngear System. The design and installation of three prototype sets and their subsequent performance in service is described in detail. The paper concludes with illustrations of the application of the system to different classes of tonnage and an analysis of the economic benefits.

INTRODUCTION

Before enlarging on proposals for improved sterngear, it is worth pausing briefly to trace the evolution in stern bearing practice and outline some of the problems experienced.

Bearings

For over a century the conventional sterngear was water lubricated and consisted of a long aft bush and a short forward bush fitted concentrically into the stern tube. The bushes were lined with staves of lignum vitae or, more recently, of resin bonded asbestos or other fibre.

That this arrangement survived for so long was not due to its intrinsic merits, but rather because its behaviour was predictable and so long as shaft powers were low, the wear was not too serious a problem. But with the rapid escalation in size and power of ships, higher bearing pressures were inevitable following the effect of the square/cube law. This resulted in wear increasing to intolerable levels.

An extensive survey[1] disclosed wear rates of 0·5 to 3·0 mm (average 1·0 mm) per year for dry cargo ships, but increasing to 1·0 to 7·5 mm+ (average 3·0 mm) per year for tankers and ore carriers with engines aft. So, whilst dry cargo ships could expect three years or more between "re-wooding", the tankers needed attention in less than three years and the worst vessels required it every year. There is little doubt that the main cause of reduced life was the increase in power and heavier bearing loads, although the positioning of engines aft and the influence of high block coefficients on wake distribution were contributory factors.

The excessive wear combined with short, stiff shafting gave rise to trouble in the gearbox and engine bearings.

Resulting from these experiences and following the development of suitable seals, the oil lubricated whitemetal lined stern tube bearings became popular, and this type is probably fitted to the majority of ships built today.

The wear of such bearings, when correctly fitted, is virtually negligible, being usually less than 0·25 mm in four years. They are, however, more sensitive to misalignment.

The lignum vitae bearings survived by the contact areas rubbing down until the load was supported over the full length of the bearing which by then had adapted itself to the deflexion curve of the shaft. Whitemetal bearings, on the other hand, if subjected to severe edge loading due to misalignment are likely to wipe which may lead to failure in extreme cases. Hence with whitemetal bearings good alignment is more important.

The conventional bearing arrangement does not provide good alignment if the two bores are concentric and in line, since they will not match the shaft curve. As a result of unfavourable experience, the practice of slope boring the aft bush was introduced, although some operators avoided the issue by increasing the clearance of the forward bush until it ceased to be a bearing at all and functioned only as a guide whilst shipping the shaft.

One severe drawback of this system was the inability to check alignment after assembly.

In the early 1960s the single bush system began to be adopted, the forward bush being replaced by the aftermost shafting bearing moved further aft[2]. This allowed the shafting to be aligned much more accurately than hitherto, and it became possible to check the alignment. However, the alignment could not be adjusted once the shafting and engines were installed. except at great expense.

If there is lack of rigidity in the aft structure of the hull misalignment can arise with changes in load or weather conditions. The major cause of alignment variation is that produced by thrust eccentricity, the more so as this can vary significantly between full load and ballast condition. Superimposed is a smaller cyclic fluctuation due to the number of propeller blades. An alternating variation of similar magnitude can occur due to ship movement in a sea-way[3,4].

The Influence of Length and Alignment on Bearing Performance

The stern bearing must operate under widely varying conditions in addition to the misalignment which can be imposed by the factors described. The load varies with the shaft bending and the speed may vary between turning gear and full power. The most arduous condition probably occurs after a long period at full power when revolutions are reduced for manoeuvring, perhaps concluding with a lengthy period on turning gear. This is when the oil film will be thinnest and, if edge loading is present, abrasion may occur.

The bearing length is fixed by classification society rules and

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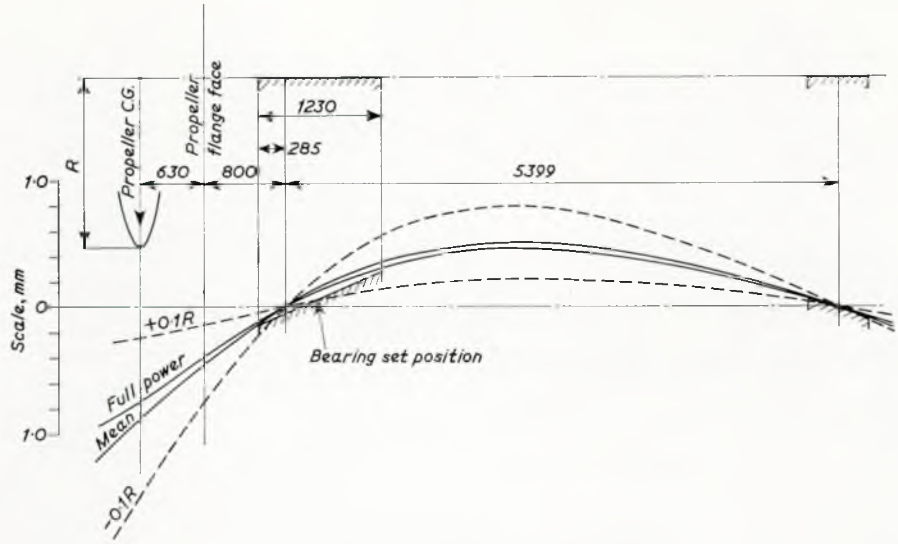


FIG. 1—*Shaft lines m.v. "Laurita"*

has been successively reduced from the L = 4D for water lubrication to the present L = 2D for oil lubrication with the proviso that the mean pressure is not to exceed 0·62 MN/m^2 (90 lbf/in^2).

In view of the shaft bending and alignment it is very doubtful if bearings of length greater than 2·5D are any safer, and indeed may have had a lower margin. But caution should be exercised before any further massive reductions in length are made. One sometimes hears statements that short bearings are better able to withstand shaft bending and misalignment and that L/D = 1 is an "optimum" proportion. The authors disagree.

There is no fixed optimum proportion for a bearing, the optimum length varies according to all the parameters imposed on the bearing.

Although the variation in film thickness caused by shaft bending over the length of the bearing will be less in a shorter bearing, the actual film thicknesses may be so much less that the safety margin is still reduced. In cases of misalignment the short bearing has no advantage over the long, except in such extreme cases, as where the shaft slope is so great that it contacts the opposing surface of the bearing at the opposite end—a condition so gross that it must be avoided in any event.

A short bearing will not have such good damping action as a longer bearing.

It is the author's view that for large vessels there is little to be gained by further reduction in length. However there is a need to give more attention to design on an individual basis where proportions could be optimized.

As an example, Fig. 1 shows the computed shaft lines for m.v. *Laurita*. The mean line is for the "warm, buoyant" condition without thrust (or with concentric thrust) and the full power line allows for the thrust eccentricity anticipated for this vessel. Superimposed are the shaft lines computed with thrust eccentricity of ±0·10 R (R = propeller disc radius) such as may be reached with a vessel of high block coefficient under conditions of full power loaded and ballast respectively. The position at which the bearing was set is shown. Table I summarizes the minimum oil film thicknesses which result. These have been computed allowing for the shaft curvature and slope. The ratio L/D = 2·14 arose through the use of a standard size with a shaft diameter below the maximum in the range.

As a comparison similar figures are given for a bearing with L/D = 1·0 and it will be seen that in all cases the film thicknesses are considerably less.

Only the vertical plane has been shown here. In practice there is also a significant horizontal component of thrust eccentricity and, therefore, of shaft deflexion which complicates the oil film geometry.

TABLE I—MINIMUM OIL FILM THICKNESSES

Rev/min 135	L/D = 2·14		L/D = 1·0	
Shaft Line	Aft End	Ford. End	Aft End	Ford. End
Mean	0·32 (·013)	0·39 (·015)	0·12 (·0047)	0·15 (·0059)
Full Power	0·35 (·014)	0·35 (·014)	0·13 (·0051)	0·13 (·0051)
+0·10 R	0·54 (·021)	0·30 (·012)	0·23 (·0091)	0·08 (·0031)
−0·10 R	0·19 (·0075)	0·56 (·022)	0·04 (·0016)	0·26 (·010)

Film thicknesses in mm, inch conversions in brackets.

Also, to simplify the computation the position of bearing reaction has been assumed fixed, but in practice the position of the reaction will follow the position of resultant film pressure as it changes according to shaft slope; thereby modifying both the bearing load and the shaft line.

To illustrate the effect of misalignment on film thickness the curves shown in Fig. 2 have been plotted. These show the relation

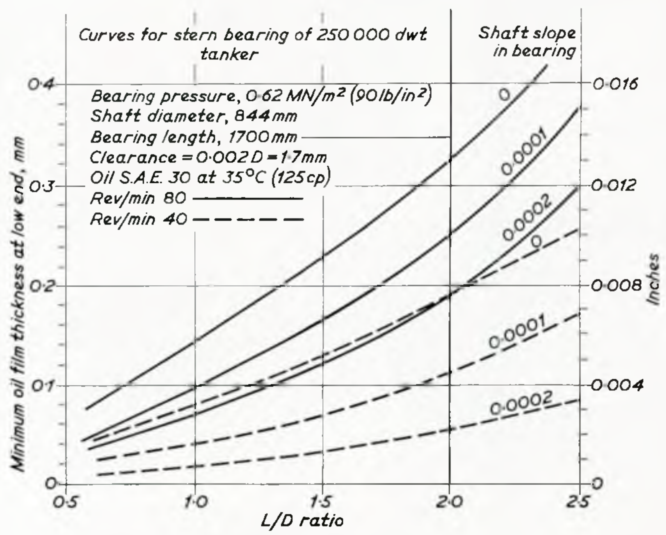


FIG. 2—*Effect of misalignment on minimum film thickness*

between minimum film thickness and shaft slope in bearing. They are based on a typical bearing for a 250 000 dwt tanker at full and half speed.

It will be observed that the minimum film thickness decreases quite rapidly with decrease in length, and corresponding increase in specific load. Moreover, the effect of misalignment becomes more marked with the shorter bearings, especially at lower revolutions.

This is an area in which further work is being done at the present time(13).

Seals

The original stuffing box and soft packing used for water lubricated bearings, with its tendency to groove the shaft, has been superseded by improved forms of seal, although slight leakage of water is not of major importance in this case. The oil lubricated bearing however depends for its operation on clean oil of correct viscosity and, hence, it is necessary to fit seals to retain the oil and to completely exclude sea water.

In the authors' experience the seals currently in use, though continuously being improved(12), are more prone to failure than any other component in the sterngear, In fact it is likely that a large proportion of bearing failures have been caused by prior failure of the seal.

The frequency of seal failures increased with the growth in size in ships and in 1968 a committee investigating seal failures in a major shipbuilding country reported that out of 29 ships studied, 12 had to be dry docked within the first year through stern seal failure(5).

There seemed little doubt that the cause lay in the increase in pressure range resulting from the deeper draught. As palliatives there has been recourse to fitting a second, lower level, header tank for operation in ballast, pressure balance devices and so forth. Separate oil circulation has been provided to cool the seals. All these features add complexity and increase the dependence upon the human element.

Seal failures when they occur are usually without warning and necessitate the vessel putting into port and unloading cargo. If the circumstances are such that the fault can be attended to very quickly it can sometimes be done outside of dry dock by tilting the ship. If however there is any suspicion of weardown or obvious contamination of the lubricating oil system, examination of the shaft and bearing will be required. This means diverting the ship to a suitable dock, and the larger the vessel, the greater is the diversion and delay likely to be. At the time of writing, there exist only 21 repair dry docks in the world able to take a vessel over 200 000 dwt and of these seven are in Japan and four in Northern Europe.

Shafting

In the light of statistics produced by leading classification societies even the conventional propeller shaft cannot be considered to be thoroughly reliable. Reports indicate that of all shafts surveyed casualty rates due to cracks in the cone or keyway varied from 0·7 per cent for geared turbines amidships to 4·9 per cent for oil engines fitted aft(6, 7).

It is worth noting that the cracks do not occur at the point of maximum bending moment and stress in the shaft—i.e. in the bearing—but at a place where the stresses would be expected to be lower. The statistics suggest that the adoption of "keyless" propeller fittings, will eliminate that proportion of failures caused by keyway cracking but this is not yet shown to be a complete solution to the problem.

In contrast the authors have found no instance of failure of a shaft with a flanged connexion to a c.p. propeller, in spite of the general increase in weight and overhang relative to shaft diameter.

Many of the problems referred to are intensified due to the uneven wake distribution. This is the province of the naval architect, but the authors would make the observation that improvements carried to the point where propeller excited hull vibrations are no longer objectionable do not necessarily resolve the engineering problems.

It is not proposed to spend more time enlarging on the problems. Detailed information will be found in the references already quoted and in Refs. 8–12.

Guide Lines for Development

By relating the foregoing fundamental reasoning to a practical study of conventional sterngear elements and an assessment of the known consequences of their failure it is possible to establish principles for future design. These form the basis of the "Glacier-Herbert" concept and may be summarized as follows.

1) The propeller and shaft should be supported in a two bearing system in which the set angle of the sterntube bearing can be chosen to suit the mean inclination of the shaft at the point of support and in which the aftermost pedestal bearing is freely positionable in space.
2) To maintain effective load distribution within the sterntube bearing in spite of varying shaft inclinations (which occur due to changes in loading, weather conditions and cyclic influences from the wake) the support system should be so arranged that the bearing can be set relative to the chosen inclination line with absolute precision. It should be possible to make the final adjustment with the ship complete and afloat.
3) To minimize repair costs and loss of hire due to docking or diversion, the equipment should permit full inspection and replacement of wearing parts in the seal and bearing assembly by the ship's personnel with the ship afloat in any condition of loading.
4) The design should enable full survey inspection and propeller replacement to be carried out with minimum time and cost. This is substantially realized if both survey and propeller changing can be carried out without removing the shaft from the vessel.
5) It follows from (4) that a shaft which can be relied upon to last the life of the vessel is necessary. The design of the propeller connexion must therefore be free from stress concentrations of the type which have led to trouble with conventional cone connexions.
6) Seal failure is by no means uncommon in spite of the improvements made by various specialist manufacturers. The design of the stern gear system should therefore be suitable for the accommodation of a wide range of alternative types of seal.
7) The risk of bearing damage is greatest when it becomes necessary to continue a voyage with a defective seal. It should therefore be possible to detect seal leakages at onset. The bearing design and lubrication should be arranged so that the risk of bearing damage following seal failure is minimized.
8) In assessing the real capital cost of any equipment it is necessary to include its cost of installation and its effect on the rational construction of the vessel. The risks of trouble arising from dirt or defects in installation work must also be minimized.

   It is particularly desirable to eliminate the "critical path" operation of boring out on the building berth and the assembly itself should be capable of being fitted as a complete sub assembly with minimum cost. An additional advantage is obtained if a common stern-frame design can be adopted to serve both fixed pitch and c.p. applications.
9) In designing to satisfy the above requirements, the need to achieve utter reliability must always be borne in mind. A practical means to this end is to utilize existing proven elements where possible.
10) To enable any bearing system to function as designed the rigidity of its supporting structure must be assured. It must be shown how such a structure might be arranged consistent with the provision of ample space for dismantling the bearing. To achieve ample lateral strength is particularly important.

    In making such provisions it is essential to avoid unduly large boss diameters or characteristics in the hull form in way of the propeller which may be judged to cause losses in propulsive efficiency.
11) The outboard seal must be well protected from damage and the design should thus include a really effective rope-guard.

There is often evidence of accelerated corrosion in the enclosed water space under the rope-guard and the resultant debris can be detrimental to seal performance and life. All sea water exposed parts should therefore be of corrosion resistant material. Similarly marine growth during lay up has been known to produce expensive trouble following re-entry into service. It should therefore be possible to seal off the boss and conserve the boss cavity during prolonged periods of enforced idleness.

12) The lubrication of the sterntube bearing is particularly important. There should be positive circulation of oil through the bearing and through the end cavities between bearing and seal. Ample cooling must also be ensured.

In addition it is desirable to be able to detect and separate water which may enter the lubrication system in the event of seal failure.

THE GLACIER - HERBERT DESIGN

Fig. 3 shows a "Glacier-Herbert" bearing assembly suitable for a propeller shaft diameter of 590 mm.

The bearing housing is a flanged S.G. iron tube in two halves lined with whitemetal. They are firmly bolted together and supported at the aft end by a spherical carrier ring to which is bolted the outboard seal housing, and at the forward end by a circular diaphragm to which is bolted the inboard seal housing.

The bearing halves are completely symmetrical and thus reversible and invertible within the sub assembly.

A full assembly of the complete sterngear is shown in the axometric drawing (Fig. 4).

The bearing assembly, consisting of the bearing (1), the spherical carrier ring (2) and the diaphragm (5) with outboard and inboard seal housings respectively attached, is housed entirely within the sternframe.

The sternframe casting is shaped to accommodate a short outer bore within which the spherical seating ring (3) is fitted and secured with substantial bolts. This ring in turn receives the spherical carrier ring (2), the whole forming an all round supporting annulus of extreme rigidity around the aft end of the bearing. At its inboard end the casting is provided with a bulkhead which stiffens the frame itself and in which is formed a short stepped bore backed by two partial flanges.

The diaphragm is received loosely within the smaller diameter of the stepped bore, the larger diameter of which forms a seating ring for the blocks (7) and (8) which locate and support the forward end of the bearing. The whole assembly is secured in position by the axial bolts (9) which are anchored to the frame casting.

"O" ring seals are provided on the periphery of the spherical carrier ring and the forward diaphragm to form an oiltight space surrounding the bearing which is filled with lubricating oil. The "O" rings used at the forward diaphragm are of large diameter (25–30 mm). This allows a working clearance between the diaphragm and its mating bore within which the diaphragm may be eccentrically displaced to achieve final precision alignment of the bearing relative to the shaft line.

The chosen location of the diaphragm is determined by adjustment of the distance pieces (7) fitted in way of the lower supporting chocks (6). The upper wedge chocks (8) hold the diaphragm firmly in this position with radial pressure determined by the axial bolt loading.

This arrangement allows for the fact that the bearing will expand when warming up at a greater rate than its housing and permits this to take place without detracting from the rigidity of support at the forward end of the bearing. The Belleville washer packs (10) fitted to each axial bolt ensure virtually constant loading of these bolts and those securing the spherical seating ring. The initial controlled load is applied by hydraulic nuts.

The spigot arranged on the after side of the spherical seating ring (3) runs concentric with the flange carrier ring (4) with a nominal radial clearance of 3 to 4 mm. This latter ring and the rotating element of the outboard seal, is bolted to the forward face of the propeller shaft flange. Alternatively, the ring may be formed by appropriate shaping of the forward end of the propeller boss. Two inflatable seals (15) each with independent air supply, are housed within recesses in the spherical seating ring spigot and seal against the inner wall of the shaft carrier ring when brought into action for under-water withdrawal. The outer diameter of the two rings are machined parallel to receive an external bandage which can be fitted by a skin-diver as alternative to using the inflatable seals or to seal off the gland space from the sea if the ship is laid up.

Most careful attention was given to the choice of materials in the enclosed sea-water filled spaces between propeller boss and

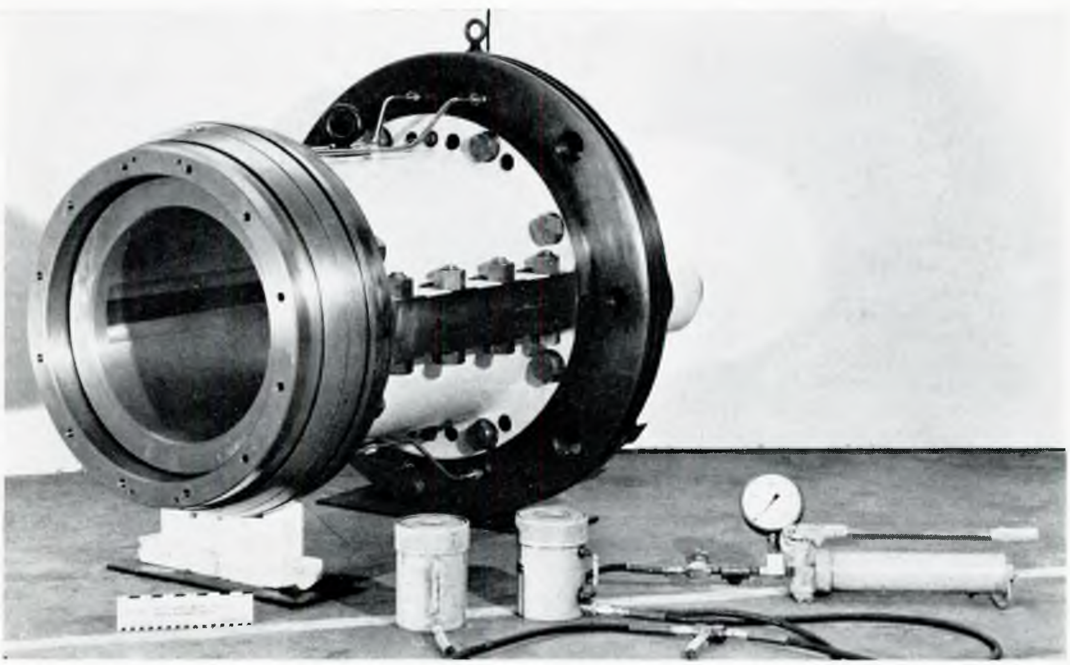


FIG. 3—*Stern bearing assembly in works looking on aft end—the two jacks used for installation and withdrawal are shown*

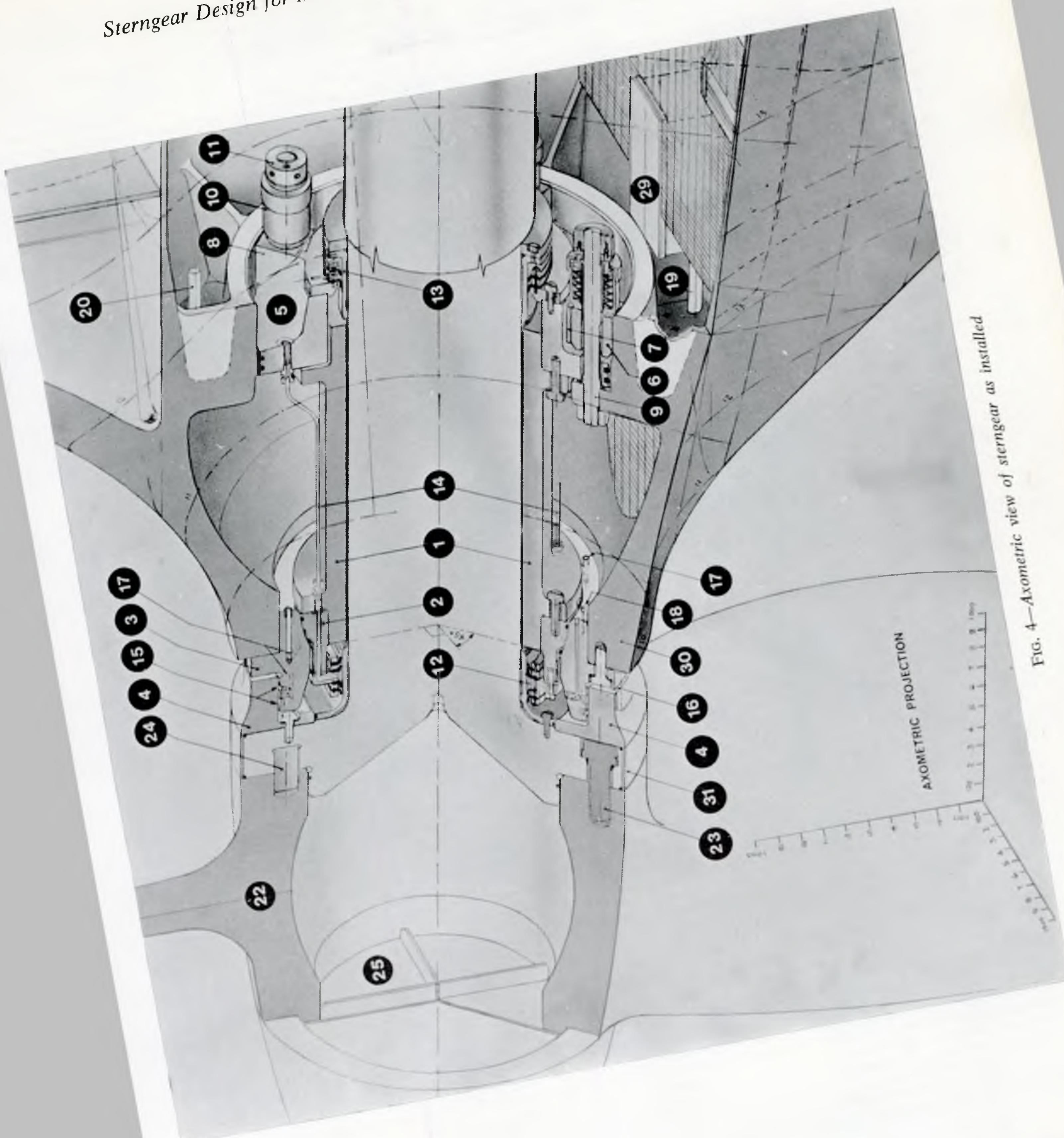


FIG. 4—*Axometric view of sterngear as installed*

frame to avoid corrosion and thus the accumulation of circulating abrasive debris in the space surrounding the seal. Additionally the labyrinth form of the assembly and rotation of the enclosed water by entrainment discourages the penetration of suspended abrasive in shoal conditions.

There are arranged top and bottom pipes connecting the space between working and inflatable seals to within the vessel. The prime function of these is to test the effectiveness of the inflatable seals and to drain the enclosed space before underwater withdrawal of the bearing and seal assembly. These lines terminate as drilled passages in the spherical seating ring. By inclining the radial drilling in the direction of ahead rotation and connecting the internal pipes to a suitable service line during normal operation, the natural rotational flow within the enclosed area is converted into a positive cleansing action and the water is constantly changed.

The high velocities of entrainment produced in the outer regions of restricted clearance discourage marine growth. A further thought is that in ships which produce a surplus of fresh water, a nominal flow of this would produce virtually fresh water conditions in the interior.

In applications with stainless steel propellers the spherical seating ring, spherical carrier ring and gland housing are made of "Ni-Resist" iron and the shaft carrier ring of stainless steel to the propeller specification. With bronze alloy propellers, compatible bronze alloys are used for these parts.

The influence of these choices on capital cost is significant and there are cheaper alternatives, but by far the most important initial objective is to demonstrate utter reliability in the long term direct from the design stage. In such circumstances, the authors believe this to be the only responsible approach.

Seal Arrangement

In the illustration a development of the well known Simplex seal has been used.

In all units so far manufactured, the static part of the seal assembly has been attached to the bearing assembly to be withdrawn with it. This has proved a convenient way of moving the seal assembly to a place where ample space for dismantling is available. When lip and radial seal assemblies are dismantled and reassembled in this way the bearing sliding along the shaft maintains the seal elements in perfect alignment with the mating sleeve. Repeated full scale rig tests were made to prove the effectiveness of the method.

After intensive study of the conventional Simplex seal design in relation to modern bearing systems, it was concluded that the floating bush and associated bellows were of questionable value. There was in fact, some evidence that they could be a source of trouble. These were therefore dispensed with and an equivalent arrangement of simple lip rings adopted.

The outboard gland cavity, conventionally filled in dock, is instead filled via the internal pipe connections (14) after assembly. This has three main advantages. First, the supply oil can be changed as desired and by temporarily raising the pressure the seals can be flushed. During this latter process oil is introduced to the space forward of the aftermost seal lip. Secondly, when drained, the cavity can be used to check the effectiveness of oil and water seals. Thirdly, if seal damage is apparent, it is possible to run with the cavity drained to form an open coffer-dam and minimize the risk of water contamination of the lubricating oil or of sea pollution.

Thus in the outboard seal there is improved lubrication and cleanliness and a major source of heat is removed. Again the pressure differential outboard is not high and there is ample external cooling. The rate at which grooving of the bushes is experienced is much lower than inboard, particularly on deep draught ships and it is reasonable to expect a marked increase in the life of the outboard bushes due to the improvement described. In supplement to this, the life of the bush is doubled by the fitting of a spacer disc in the original assembly which can be removed when necessary to present the lips to an unworn part of the bush.

At the inboard end where grooving is a more serious problem due to the higher pressure differential and limited cooling, the design incorporates a recess at the end of the bearing which allows the use of a much longer bush which can be repositioned several times.

For large ships, the inboard bush can be provided with three lips inboard to reduce pressure differentials, the space between the first and second lips being permanently connected to the ballast header tank or to a small intermediate header tank fitted at an appropriate level. It should also be noted that the inboard bush is not difficult to change and if a spare is carried mounted on the shaft, forward, the original can be cut off and a replacement fitted afloat.

It is submitted that such measures, coupled with current developments by lip seal specialists minimize the limitations of this type of seal. Alternatively there is now available a radial seal alternative which is claimed to be capable of running on fully split bushes.

Equally the design will accommodate the known types of face seal now marketed.

Bearing Lubrication

As already described, the bearing runs fully submerged in a cooled oil bath, positive circulation of the oil through the bearing being achieved without the complication of forced lubrication.

The bearing is provided with side grooves connected by holes to the surrounding oil. A system of grooves in the bearing end spigots likewise connect the ends of the bearing to the oil bath. With this arrangement a pumping action, stimulated by shaft rotation, draws oil into the bearing through the side ports along its length and discharges it from the ends of the bearing to return to the oil bath via the end channels referred to.

This system ensures a balanced oil distribution and thermocouples embedded in the lower half at each end of the bearing measure the temperature of the whitemetal in the loaded area. The indication given is thus both meaningful and sensitive and if an alarm is fitted, warning of an undesirable condition developing is immediate. The disposition of these thermocouples relative to the flow pattern provides an indication of load distribution on the bearing length and thus verification of the correctitude of alignment.

Additionally, the arrangement is self cleaning since entrained dirt or debris has ample opportunity to settle out rather than be recirculated. Moreover, water which may pass the outboard seal in event of chronic seal leakage, gravitates directly to the undisturbed area at the bottom of the oil bath from which it can be drained.

Experience, which will be described, has shown that water can accumulate in the lower part of the oil bath in significant quantity without being drawn into the circulating system. Simple monitoring devices are available to give alarm before the presence of water introduces a hazard. The system thus permits the use of normal lubricating oil with safety.

The Propeller and Shaft Connexions

With controllable pitch propellers the shafting design is entirely conventional except that a short spigot is required on the forward side of the shaft flange to receive the shaft carrier ring. The normal protecting plate covering the propeller connecting bolts and the conventional rope guard are dispensed with.

Statistics available from several hundred c.p. propeller installations show the flanged shaft connexion to have a most enviable record of freedom from failure compared with the conventional cone connexion in which the rate of failure is significant and well known. Why, therefore, should fixed pitch propellers not be attached in the same way? Considered in conjunction with the characteristics of the Glacier-Herbert bearing and seal arrangement the advantages are numerous:

1) the life expectancy of the shaft will equal that of the ship itself;
2) the propeller itself can be made some 5 per cent lighter with corresponding reduction in cost;
3) the propeller can be quickly changed without removing the shaft from the vessel.

Combining the significance of (1) and (3) with the fact that the sterngear design permits special survey inspection, seal and bearing maintenance without shaft removal, there can no longer be valid objection to shafts fitted from outboard on the grounds that this requires dismantling of the rudder to effect removal.

With this obstacle overcome, there are in fact several additional reasons for flange mounting of the propeller and for fitting the shaft from outboard:

i) the aftermost position of the engine and hence the position of the engine-room forward bulkhead is determined in some cases by the length of intermediate shaft required to permit inboard withdrawal of the propeller shaft—in some cases examined, it has been possible to move the aft end of the engine two frames aft; the thrust block need no longer be placed forward of the removable shaft section;

ii) initial fitting is easier and there is no need to provide heavy lifting tackle and withdrawal gear inboard or to reserve space for rollover of the intermediate shaft;

iii) the statistics of failure of shafts with flange mounted propellers compared with cone mounted fairly indicate that a spare propeller shaft, and stowage arrangement for same, can be safely dispensed with—the cost saving of this can be fairly included in relative cost calculations.

The S.K.F. coupling used inboard is in fact what is used outboard in some "keyless" arrangements of propeller attachment but in this location it is not subject to excessive bending or to water penetration, and it does save considerable time and cost at survey. The shaft for the flange mounted propeller is akin to a standard shaft reversed but without its attendant cone, screwed extremity, nut and mounting tools. It is also shorter than the conventional by the length of the propeller boss and there is a reduction in forging weight. Similarly the intermediate shaft has only one flange which reduces its cost. When these facts are fairly considered and the convenience of the S.K.F. coupling taken into account, the total cost of shafting and coupling is quite comparable with the conventional.

The Influence of Overhang

In the original design a propeller boss of the type shown in Fig. 5 was conceived, primarily with the idea of reducing overhang. It was subsequently found that the configuration severely restricted the p.c.d. of the bolt circle and hence the strength of the flange connexion if the propeller boss diameter is not to exceed that essentially required at the sternframe boss to accommodate the bearing and seal assembly. It was also found that the reduced propeller weight was not fully matched by a reduction in

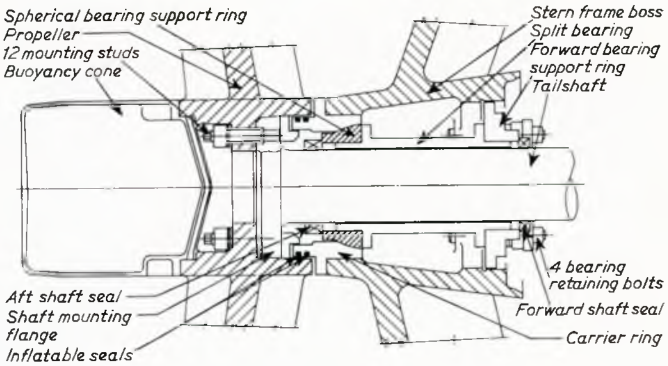


FIG. 5—*Solid propeller with hollow cylindrical boss and internal flanged mounting*

casting weight and cost on account of casting limitations.

In the matter of overhang, there is no evidence that the shaft itself is too weak at its point of maximum bending moment. There is also no fundamental reason to become over-awed by sheer size in this context since as sizes increase the shaft modulus increases much more quickly than the propeller weight and the overhang itself increases at a lower rate than does the shaft diameter.

It is not to be implied that it is not desirable to minimize overhang as, apart from the above consideration, it does affect the bearing reaction, but in this respect we are working within known parameters and not at the point where reduction must be sought at the expense of boss diameter and complication of manufacturing or mounting.

What is very much more necessary is to ensure effective load distribution and therefore to align the bearing with unquestionable precision relative to the immersed static bending line to give the bearing maximum freedom to accommodate the superimposed and variable flexures produced by thrust eccentricity.

It follows from this last statement that the ultimate capability of any sterngear design tends to be determined by the skill of the naval architect in producing a reasonable wake pattern.

Ship Design Considerations

The mechanical assembly thus far described allows the achievement of sternframe boss propeller diameter ratios ranging from

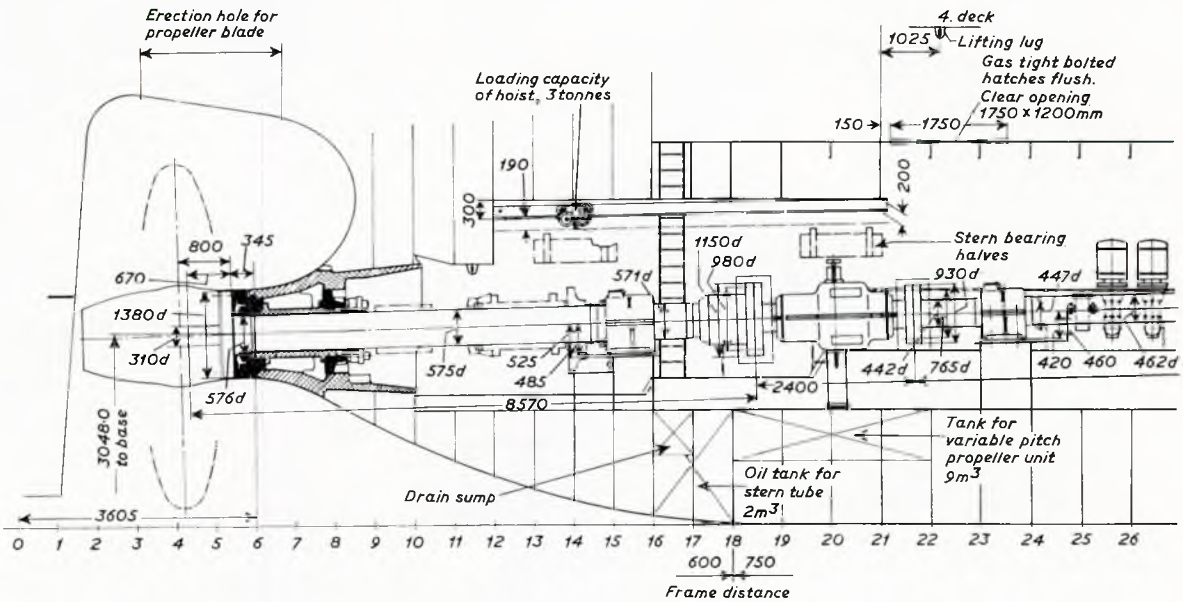


FIG. 6—*Shafting installation in m.v. "Laurita"*

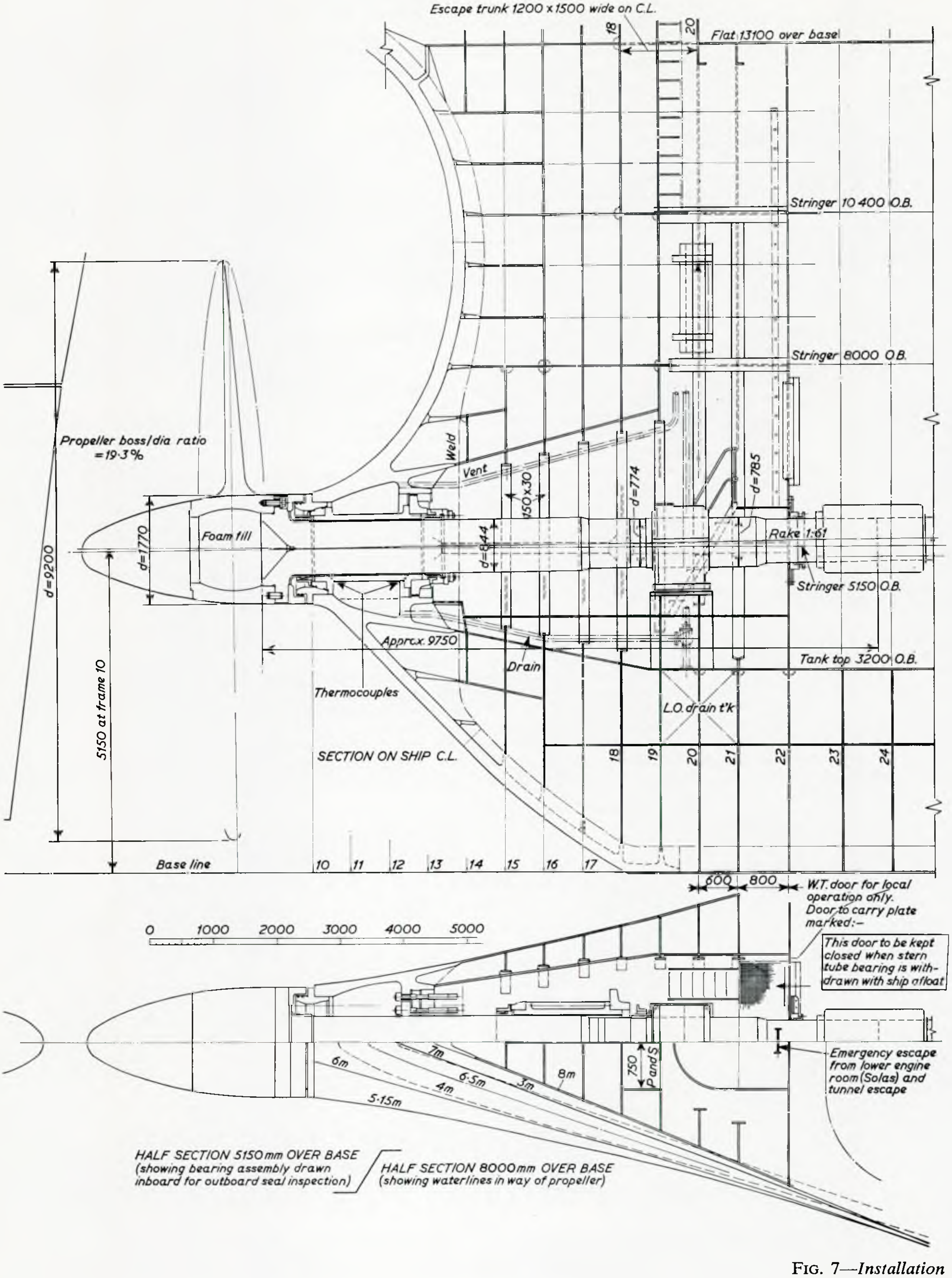
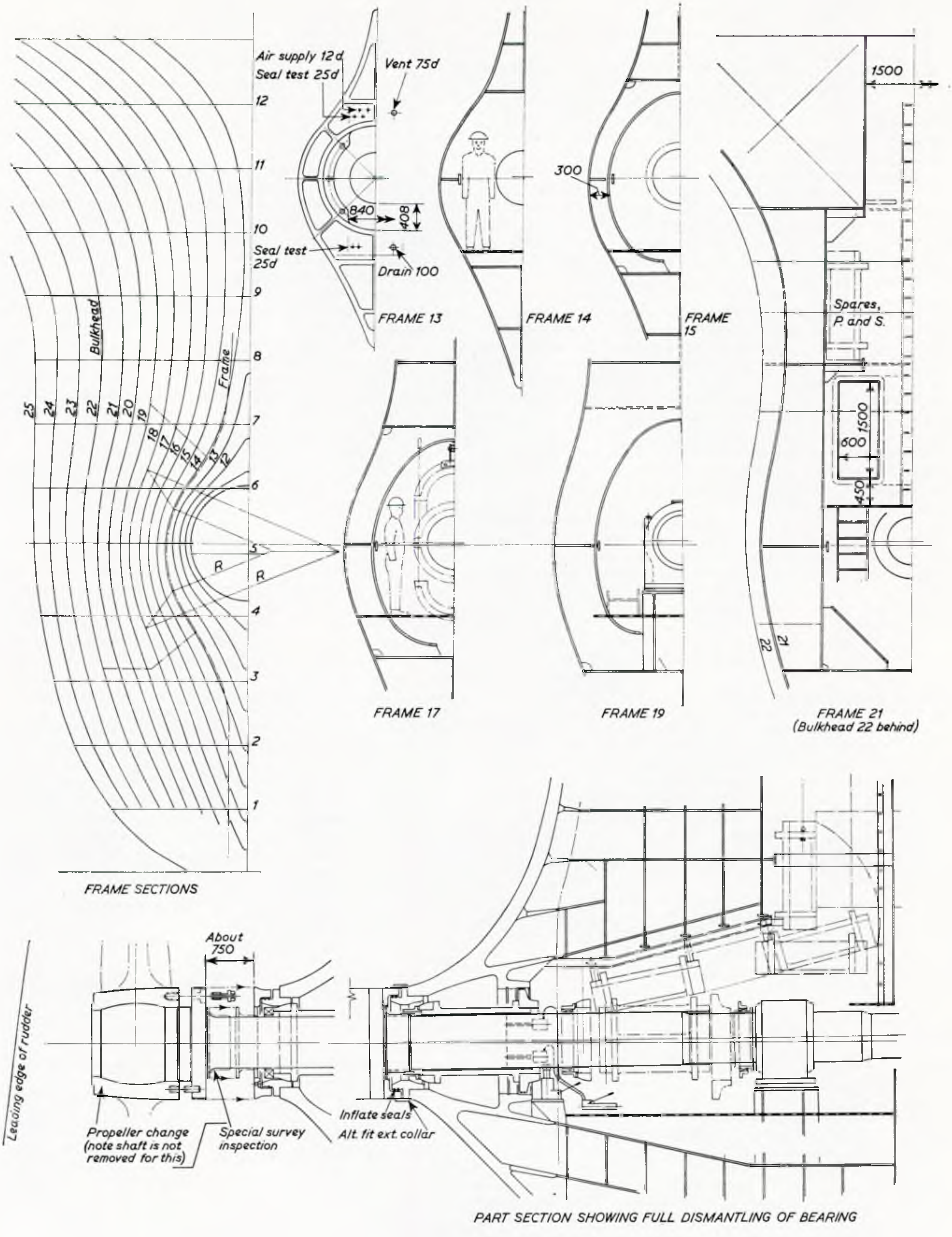


FIG. 7—*Installation design stu*



250 000 dwt tanker

24 per cent in the smaller sizes with shafts 500–600 mm in diameter to 20 per cent with shafts above 800 mm diameter. It is submitted that these are not excessive figures which penalize the quasi propulsive coefficient.

Most importantly perhaps, in this context, it has been found possible to develop the fairing of lines to the boss in a manner which ensures ample local strength and access for overhauling, coupled with the avoidance of sudden increases in section immediately forward of the boss, or added impedance of flow to the low wake regions behind the sternframe.

Our initial installations Fig. 6 were made in 22 knot ships with a block coefficient of only 0·54. The original model had a bulbous afterbody which needed very little modification to accommodate our sterngear design and subsequent model tests confirmed that this had in no way impaired quasi propulsive coefficient or wake distribution. The published photographs show that this was not achieved at the expense of restricted access.

In the larger full ships the provision of ample access and working space combined with structural strength and rigidity is not at all difficult but the structure must be developed with due regard to the engineer's functional requirements for withdrawal and dismantling of the bearing assembly. It is of course of no value to achieve precision alignment of the bearing system if the rigidity of the structure supporting the system is questionable. Lateral rigidity in particular is most difficult to achieve and deserving of particular attention.

Fig. 7 shows an installation design study prepared for a typical supertanker of 250 000 dwt with machinery of 36 000 shp at 80 rev/min. The aft sections shown are developed from the original lines of the model at frame 25 and the propeller boss ratio is less than 20 per cent. The profile of the sternframe casting is identical with that of the standard ship and the waterlines below 3·5 metres and above seven metres are unaltered.

In the immediate vicinity of the propeller boss the waterlines are faired in the manner shown on the plan producing very good flow lines with waterplane sections conducive to the development of superior strength and rigidity.

It will be seen that the waterline at the shaft centreline is perfectly straight for some three metres forward of the frame casting and the sections 14 to 18 are based on circular arcs of constantly increasing radius. The heavy aftermost plates in the open area can then be developed from flat plates rolled to a true cone between the constant limits of arc indicated. The adjoining reverse curvature is not excessive and the associated waterlines are virtually straight so that the rolled plates can be finished and faired without unusual difficulty.

Certain statutory bodies require the stern bearing dismantling area to be isolated from the engine room when the bearing is withdrawn underwater and in some ships this necessitates an additional bulkhead. In the case under consideration this bulkhead was formed by moving the forward face of the aft peak, originally on Frame 20, to Frame 22. There is thus only a marginal increase in steelweight but a marked improvement in the stiffness of the whole structure enclosing the two bearing system. The trunk abaft the bulkhead serves as a channel for removing components direct to deck, and in some cases can be used as the escape route from lower engine-room now required by SOLAS.

Dismantling and Reassembly

For inspection and maintenance the bearing and seal assemblies are drawn inboard to a location where dismantling space can be provided. When this is done the shaft must of course be supported but when the operation is to be carried out underwater the obvious method of slinging the shaft, or providing a temporary support fitted from outboard, are not acceptable.

One of the proprietary features of the design which makes underwater withdrawal possible is the provision of a permanent support in proximity to the shaft outboard of both bearing and seal onto which the shaft can be lowered. One alternative envisaged was to permanently fit an encapsulated jack in similar position, this being operable from inboard to engage the shaft when required.

Before commencing withdrawal with ship afloat the safe operation of the water seals must be demonstrated and trapped water drained from the enclosed cavity.

The two pipes already referred to which connect the top and bottom of the cavity to within the vessel are provided inboard with linked cocks so that both open and close together. The cocks are first opened and water discharged into the vessel through both pipes. This demonstrates freedom from obstruction. The aftermost water seal is then inflated and the drainage of the enclosed space commences. This is complete when the lower pipe ceases to discharge.

A duplicate arrangement of test pipes is connected to the space between the two water seals and the above process is repeated. There are then two independent seals preventing water ingress and withdrawal can commence.

It is a principle of safety that both seals must be shown to be in working order. In the event of failure to achieve this the alternative external bandage must be fitted by skin divers and its effectiveness similarly checked. This neoprene bandage is of substantial proportions and provided with a simple stretching arrangement which can easily be operated by the diver.

The bearing lubricating oil is next run down to the drain tank and the inboard seal assembly disconnected and slid forward along the shaft. Obviously, if it is the inboard seal only which requires attention, no further preparation is required.

To withdraw the bearing assembly, two horizontal jacks hydraulically coupled are mounted on the diaphragm to apply pressure against the two half flanges located behind the diaphragm housing. A further lifting jack is mounted under the shaft about 150 mm forward of the inboard end of the assembly. These jacks, though supplied with the sterngear, are useful general purpose tools.

The hydraulic nuts and Belleville packs, together with the upper wedge chocks, are first removed. Following this, load is applied to the vertical jack and the shaft assembly lifted through the bearing clearance. The forward end of the bearing assembly is thus fully unloaded and the load on the aft spherical seating partially relieved.

The lower supporting chocks are then removed and pressure applied to the horizontal jacks to disengage the spherical seat and move the assembly about 125 mm forward, in which position the bearing assembly hangs fully supported on the shaft and the diaphragm is clear of its immediate housing.

The vertical jack is next released allowing the shaft and propeller assembly to settle until supported by contact between the external spigot of the spherical seating ring and the internal surface of the shaft carrier ring. The jack is then removed.

A "Tirfor" winch or tackle is now applied to haul the assembly forward along the shaft into the dismantling position.

During assembly the procedure is reversed, the only difference being that the horizontal jacks are reversed and tension rods fitted through their hollow spindles. These are screwed into tapped holes in the sternframe casting.

Figs. 8 to 13 illustrate the above operations in sequence. These were taken onboard the prototype installation and show that even in an extremely fine ship there need be no shortage of space.

Essential survey work in dock is limited to examination of the propeller connexion. For this, and for propeller changing, the inboard coupling is disconnected and the propeller/shaft assembly hauled outboard some 600 mm by sliding in the bearing. In this position the propeller can be removed, the inflatable seals examined or replaced and magnaflux testing of the shaft carried out.

These operations are shown in diagrammatic form in Fig. 7.

A further Glacier-Herbert innovation is illustrated in Fig. 14. This is a trunk fitted above the propeller blade through which the blade can be drawn into the vessel. In this case it enables c.p. propeller blades to be changed without shore facilities by deballasting and tipping the ship. The idea has been further developed for very large ships wherein the use of fixed pitch propellers with detachable blades is becoming of interest. Practical means for cleaning the interface underwater, and for securing the blade underwater, have been worked out.

FIG. 10—*The stern frame bearing recess—this shows the ample access space to the shaft surface and the outboard seal bush*

FIG. 8 *(Left)—Commencement of the withdrawal operation* FIG. 9 *(Right)—Withdrawal of the bearing: the inboard seal has been removed along the shaft. The bearing has been jacked clear of its seating and is now being winched along the shaft.*

FIG. 11—*Bearing withdrawn: inspector is in recess examining surface of outboard bush and shaft (If necessary the bush can be withdrawn along shaft to permit inspection of flange fillet).*

FIG. 12—*Replacement of bearing*

FIG. 13—*Replacement of bearing—jack raising shaft from aft support to centralize and slide bearing home*

FIG. 14—*Fitting propeller: hatch through which individual blades may be lifted to deck can be seen*

Installation Onboard

The sternframe casting is best fully machined in the machine shop but the design does allow for a minimum amount of final machining of the principal bores on the berth when the builder prefers this.

The three prototype installations were made at the Hamburg Shipyard of Blohm and Voss who chose the former method. The ships concerned were high speed car transporters built for A/S Uglands Rederi of Grimstad, Norway. These were single screw ships with machinery developing 16 000 bhp at 135 rev/min and equipped with Kamewa propellers. The propeller shaft diameter was 575 mm.

The machined sternframe casting was welded into the largest handleable sub assembly which was then aligned to the main hull structure on the berth using targets erected in the fore and aft machined bores. The sub assembly was welded in position using a controlled sequence of symmetrical welding directed by constant observation of the alignment.

In all three ships final concentric mounting of the assembly was achieved without undue difficulty within the limits of bedplate chock adjustment permissible.

The outer diameter of the spherical carrier ring was then machined to suit the aft bore of the frame. It was considered permissible to machine this ring slightly eccentric to the inner spherical seat if necessary to correct any alignment shift during welding, but in no case was it necessary to make use of this facility.

The spherical seating ring was fitted into the after bore and bolted in position. Internal pipes for seal test and air supply to the seals were fitted and pressure tested, lubricating oil drain and vent pipes were connected and tested and the whole system and interior spaces carefully cleaned.

Meanwhile the propeller hub and shaft complete with outboard gland bush and shaft carrier ring were completely assembled in the shop and the complete bearing assembly with seal elements attached suspended by blocks in its normal withdrawn position. The shaft assembly was then introduced through the open frame, the suspended bearing assembly and the aft pedestal bearing until its aft end was supported on the spigot of the spherical carrier ring. This proved a straightforward and rapid procedure.

The shaft line was next connected and the bearing assembly fitted using the procedure already described in the previous section, the only difference being that at this stage the shims under the mounting chocks have to be machined after positioning the bearing. At this stage there is a calculated clearance under the forward end of the shaft which is diminished when the ship is floated. A final check of alignment was made in each case with the ship fully completed immediately before running the machinery. For this adjustment the vertical jack is positioned directly under the projecting face of the diaphragm to move the forward end of the bearing.

Trials

On all the ships excellent results were obtained, the bearing temperature rising slowly and steadily to stabilize at 38–44°C according to sea temperature after some hours on load. The two thermocouples showed an average difference of 2°–3°C and their accuracy was confirmed by reversal of leads.

On completion of each trial an underwater withdrawal was made for training purposes and to test the equipment. Each such examination was carried out within a normal working day including the actual inspection, and the work itself involved two to three men only.

In each case the condition of shaft surface, seals and supporting contact surfaces was found to be excellent with bearing clearances and alignment exactly as checked on assembly.

Service Performance

Each installation was similarly examined at guarantee after 4000–5000 hours of operation with equally satisfactory results.

The first ship completed trials in December 1969 and at the time of preparing this paper had recorded over 13 000 hours in service.

The only instance of trouble was on the second ship, *Torinita* in April '71, after 3400 hours of operation

At morning inspection on passage Panama-Los Angeles the Norwegian Chief Engineer, Mr. R. Viksand, noted the bearing temperature had risen to 100°C. Power was immediately reduced to idling and the shaft eventually stopped.

The pipes connecting the outboard gland cavities to within the vessel were disconnected and water found to be dribbling from one of these. The bearing oil chamber was run to bilge and found, in the words of the chief engineer, to be "more water than oil".

Suspecting seal failure, the central gland cavity was kept drained, the bearing oil chamber was filled with clean oil and the oil side seals found to be tight.

The engines were then restarted and the shaft turned slowly with bearing temperatures under constant observation. The temperature did not rise abnormally and there was no apparent overheating of the aft pedestal bearing as the load was increased gradually to full power over 15 hours. Passage to Los Angeles was completed in 100 hours at full speed with bearing temperatures stable and no vibration or other ill effects apparent.

One of the authors boarded the vessel during an eight hour call at Los Angeles and studied the condition on passage to San Francisco.

On arrival the inflatable seals were brought into use and the bearing assembly withdrawn and dismantled. This was carried out alongside the unloading berth with discharging operations proceeding normally.

The bearing lower half was found fully wiped along its length with weardown about 0·5 mm. The top half was completely unworn and unmarked except for deep scores in one or two isolated places caused by rotating debris. The shaft was found in good condition and free from significant ridging. Less than one hour of hand honing was necessary to satisfy all concerned. The bearing was landed for remetalling.

The seals themselves were in fact found in very good order and required no attention beyond cleaning. Location of the leak path was by no means difficult. It was seen that several plugs used to seal the right angle drillings in the outboard seal housing were heavily corroded and disintegrating whilst one plug had corroded away completely.

This allowed water to pass virtually unchecked from the sea to the innermost lip of the outboard seal. The fact that this allowed water to leak in rather than oil to leak and was not explained until experiments were made on the subsequent passage.

A translucent hose was rigged from a convenient shipside connexion aft to measure the actual seawater head at rest and at speed and a difference of one metre head was measured, due to quarter wave effect. This, and the fact that the draught at the time of the incident was greater than that advised when the header tank was positioned, combined to reduce the net pressure difference between sea and oil to a calculated 0·6 metres in calm sea conditions. Thus, with only moderate pitching, contamination became inevitable.

The plugs in question were, as suspected, found to be mild steel material fitted in error into the stainless steel housing.

After replacing the plugs in proper material the remetalled bearing was reassembled and refitted and the ship resumed trading.

Remetalling and remachining was carried out by Glacier's U.S. representatives, Pioneer Motor Bearing Co., in 36 hours. Apart from this the entire work was carried out by the ship's engine room staff with the assistance of one of the authors. The total working time involved was 48½ hours which included 17 hours occupied in removing three hatches in the cargo decks and handling the bearing to and from shore transport. Of the crew involved none had prior experience of the work except the chief engineer who had witnessed the examination made on completion of trials.

The foregoing incident provided a sharp lesson in the need for meticulous attention to detail, a lesson which is unlikely to be lost on those concerned and which, in fact, should be kept in mind by all engineers.

Considered in retrospect the experience is regarded as most valuable as it provided a demonstration that the basic design is sound and fully capable of doing what it was designed for. More specifically:

1) the bearing assembly has withstood the shock of a severe wiping at full power without damage to the mountings;
2) the form of shell housing adopted does not distort when remetalled;
3) considerable water penetration can occur before this is drawn into the bearing to cause failure;
4) the top half of the bearing was virtually intact and as new—if necessary the bearing halves could have been reversed and refitted after dressing the lower half;
5) the complete task of bearing removal and replacement can be carried out afloat by the normal ship's personnel provided the senior engineer responsible is familiar with the procedure.

Following the diagnosis of the trouble the bearings of the two sister ships were withdrawn during normal turn-round and the corresponding plugs removed and replaced with stainless material.

All shaft and bearing surfaces, seal lips and bushes were found to be in first class condition without measurable wear. When these inspections were made during May '71 the first vessel *Laurita* had logged over 10 000 hours of operation.

The Economic Case

To test the logic of our reasoning in financial terms detailed work lists with estimated times and costs were prepared for survey work and failures of the type known to be experienced, and the comment of representative owners invited. Some confirmed that the figures were substantially correct or conservative and none said they were exaggerated.

Table II summarises the results of these studies for installations ranging from 10 000 to 36 000 shp.

Capital costs were computed from experience so far gained. The installed cost differential depends largely on the design and construction of the conventional arrangement, and, at this stage of development, on the outlook of the builders concerned. It can thus vary considerably.

The guidance figures shown are based on comparison with conventional single bush systems in which the cast iron whitemetal lined bush is pressed directly into the sternframe.

When the original design includes in addition a cast iron sterntube with nut, as is usual with two bush conventional systems, the above differential cost is reduced.

Again the comparison is based on a standard specification i.e. without spare propeller shaft. If the actual specification includes a spare conventional shaft and this is deleted when the safer design with flange propeller mount is used, following established c.p. propeller practice, the capital difference is considerably reduced.

The "gross daily income" used in the calculations is that advised from an authoritative source to be the average daily income required to justify building at 1971 costs with the expectation of writing off investment in the vessel in 12 years.

All other figures are calculated on 1971 costs and will therefore increase with inflation.

Estimated Savings include the effect of "loss of hire" at the daily average income rate when the time saving can be fairly predicted. In the case of full survey inspection with fixed pitch propellers, the calculated saving is based on a reduction in total survey time of two days. With c.p. propellers the saving at full survey when the shaft is drawn is calculated to reduce the total survey time by six days.

Fields of Application

Vessels for which installations are now being manufactured include 265 000 dwt tankers and other ships of over 100 000 tons. Among these is a tanker of 130 000 tons for Uglands Rederi in whose ships the prototype installations were fitted. Designs have been completed in full detail for shaft sizes up to 940 mm dia. which in general terms means that power demands up to 50 000 shp/shaft can be accommodated.

Contrary to what may be supposed on superficial examination, it does not follow that increase in size must impose greater problems. On the contrary the majority of special design features just described are easier to arrange as size increases. Such aspects of bearing or seal design which might be felt to pose limitations

TABLE II—ECONOMIC STUDIES

Type classification	20 000 ton product tanker or ship of equivalent cost		130 000 ton tanker or O.B.O. with single screw		260 000 ton tanker with single screw
Power	10 000–14 000 shp		20 000–25 000 shp		36 000 shp
Size of propeller shaft	560 to 590 mm		710 to 750 mm		840 mm
Type of Propeller	c.p.	solid flange mtd.	c.p.	solid flange mtd.	solid flange mtd.
Estimated nett extra cost of vessel with sterngear fitted	£13 000	£12 500	£23 000	£21 000	£25 000
Estimated saving at *each* Survey during which shaft withdrawal is required when using conventional sterngear	£20 050	£6800	£51 150	£19 750	£27 080
Estimated saving at *each* incidence of service defect					
(a) Bearing replacement at survey	£6600	£6600	£15 800	£15 800	£23 960
(b) Propeller change at survey	—	£6400	—	£16 200	£24 060
(c) Seal failure requiring simple rings change (ship tipped)	£4275	£4275	£11 395	£11 395	£16 370
(d) Seal failure requiring inspection of bearing and shaft (ship docked)	£29 200	£16 418	£89 445	£46 550	£73 730
(e) Bearing failure or bearing replacement following (d)	£35 800	£22 918	£105 345	£62 350	£97 820
(f) Emergency propeller change (erosion or impact damage)	—	£20 100	—	£53 350	£88 120
Average gross daily income expected from ship in service	£2150	£2150	£6750	£6750	£10 000

as sizes increase are common to all sterngear designs. Where it is felt that such limits are being approached then there is increased incentive to use an arrangement in which the seal and bearing components are readily accessible.

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CONVERSION TABLE

Imperial Units	Recommended Unit Equivalent
1 in	25·4 mm
1 ft	0·3048 m
1 ton	1·016 tonne
1 ton/in^2	15·44 MN/m^2
1 hp	0·7457 kW
1 UK knot	1·8532 km/h
5/9 (°F−32)	°C

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Discussion

DR. S. ARCHER, Vice-President, I.Mar.E., said that these two papers represented two parallel approaches to the vital problem of providing utter reliability for the most important bearing in the ship with its associated sealing arrangements. Both pairs of authors had reminded them of the present high and escalating docking and off-hire charges, especially for the mammoth tankers and costly special purpose ships. The number of drydocks currently available for the very large ship was probably already inadequate for the needs of the industry and their rate of building would undoubtedly not match the growth in the numbers of V.L.C.C. and other large vessels. It could thus fairly be claimed that the crying need, generated mainly by the sheer economic pressures of the oil transportation business, had been father to the two solutions advocated today. It was also a matter of no little pride to see British marine engineers—all of them members of this Institute—leading the world in this important field today. Other recent British examples of notable improvements to stern gear which sprang to mind were the Stone Manganese Marine/Milton low bending moment tailshaft arrangement and Mr. Bunyan's Pilgrim keyless bore propeller. These few examples should confound those prophets of doom who bewailed the passing of British inventive genius in marine engineering design matters.

It was true that, up until about the early 1960s, no really important research effort had been applied to improving the design of stern bearing arrangements and little enough on seals for very large shafts. These two papers went far to redress this glaring omission, not so much from the viewpoint of the hydrodynamics of the oil-lubricated, babbited bearing, but mainly with respect to its arrangement in the ship, ready access for survey and replacement afloat and also for checking shaft alignment.

It might be asked how serious a problem was the reliability of the stern bearing/shaft seal today. In attempting to answer this crucial question, the authors had pointed out that it was likely that a large proportion of bearing failures had been caused by prior failure of the seal. This was very probably true and was supported by the report by Messrs. Herbert and Hill that, out of 29 ships studied, 12 had to be drydocked within the first year through stern seal failures. As a further contribution to this question, it might be interesting to reproduce some recent statistics prepared by Lloyd's Register for ships of various types of 15 000 grt and above, built in the four-year period 1966–69 inclusive, and in service during the four-year period 1967–70 inclusive.

Table III showed a breakdown of the composition of this group of ships by type of service, and whether oil gland or continuous liner, for each year and for the total period. The tanker had very definitely swung over to the oil gland, and in 1969 there were only two out of 26 without this type; for the bulk carrier there was less of a swing, the figures being 28 to seven. The final figures were 117:9 for tankers and 141:40 for bulk carriers.

Table IV gave the failure statistics for oil-lubricated stern bearings and stern seals, respectively, for this same group of ships over the same period, as reported by the society's surveyors, whether during normal surveys or in damage cases. The aggregate years of service were given and the incidence of failure was expressed as "per 100 years' service". For tankers the incidence of damage was 3·1 for stern bearings and 5·0 for stern seals. For bulk carriers a much lower failure rate—0·5—was shown for stern bearings, but nearly the same rate—4·8—for stern seals. So it looked as if the seal failure incidence was approximately independent of the type of ship, whereas the bearing failure was present more in the tanker, for reasons which might appear later.

Unfortunately, in these statistics it had not been possible to tell from the reports in each case whether the bearing failed as a result of seal damage, or *vice versa*, or independently. However, the authors were probably right in blaming the seal as the prime cause of many bearing failures.

In view of the major importance of seal efficiency and durability, it was encouraging to note that considerable research was currently in progress in this country aimed at improving shaft seals. Large scale test rigs were now in use at the British Hydromechanics Research Association laboratory at Cranfield, and a well known oil company was also active in this experimental field. As they had heard that afternoon, from Mr. Wilkinson, considerable research and development work had also been done by his company on large face type seals. Similar work was proceeding in Japan. From these failure statistics it was concluded that the stern bearing/shaft seal combination was of major importance today as a potential source of financial loss to owners, shippers and underwriters. The incidence rates, of course, related to existing conventional designs of stern bearing arrangements and thus, given more reliable shaft seals, it might be reasonable to expect a better overall performance with the two improved bearing designs described in these papers.

It was clear, from the vulnerability of these two key elements of a ship's stern gear, that any design which could successfully and expeditiously, at a reasonable cost, permit their survey, adjustment and replacement without drydocking or tipping the ship, could not fail to be justified on economic grounds. This Messrs. Herbert and Hill appeared to have done in their "Economic Studies" given in Table II and the later Table III now presented, from which it was clear that the economic advantages rose steeply with size of ship, as might be expected.

He was specially interested in the sections of the second paper dealing with bearing length/diameter ratios and with shaft alignment. The former had been a controversial subject for many years and the authors' investigations and conclusions were indeed very timely, in particular, that a reduction of L/D ratio below about two, might not improve the bearing reliability in terms of oil film thickness and that misalignment effects were more important with short bearings, especially at lower revolutions. This work tended to justify the reluctance of certain classification societies—in particular Lloyd's Register—to approve L/D ratios of less than two with specific bearing pressures of more than 90 lb/in^2.

The ingenious method adopted by the authors for adjusting bearing attitude to secure optimum alignment, in his view, justified the extra complication involved. Experience to date with the very large ships, such as V.L.C.C., strongly suggested that, although the naval architect had rightly concentrated on limiting structural stresses, in many cases, due to the sheer scale effect, unexpectedly large deflexions occurred at the after ends of these vessels in way of the machinery. As an example, a recent test carried out in collaboration with a Danish shipyard, using a laser beam in the double bottom as a reference, showed a total vertical deflexion of no less than $1\frac{1}{2}$–2 mm over the length of the engine room between light and fully loaded 280 000 dwt ship conditions. Lloyd's Register's own survey records also showed supporting evidence for the importance of adequate aft end stiffness, both longitudinal and transverse, in that a high proportion of the larger ships built in recent years had reported damage to stern gear of one kind or another. This problem was currently under active investigation at Lloyd's Register.

Finally, on behalf of Lloyd's Register, he thanked the authors for their kind acknowledgment of the society's part in helping to ensure the success of their enterprise.

TABLE III—INVESTIGATION INTO STERN BEARING AND OIL GLAND FAILURE

Numbers of tailshaft installations fitted in ships built to Lloyd's Register class in each of the years 1966 to 1969, inclusive, of 15 000 tons gross and over, sub-divided by the ship type and by oil or water-lubricated stern bearings (oil gland or continuous liner).

Year	1966		1967		1968		1969		66–69 incl.	
Ship type	o.g.	c.l.	o.g.	c.l.	o.g.	c.l.	o.g.	c.l.	o.g.	c.l.
Cargo	1	—	1	—	5	—	11	—	18	—
Tanker	45	5	24	—	22	2	26	2	117	9
Bulk carrier	34	10	42	9	37	14	28	7	141	40
Passenger	2	2	—	—	—	—	—	2	2	4

TABLE IV

Numbers of failures reported by Lloyd's Register surveyors in four-year period, 1967 to 1970, with oil-lubricated bearings and outer oil glands in the installations under consideration.

A = number of failures
B = aggregate years' service
C = incidence of failure/100 years' service

Ship type	Stern bearings A	Stern bearings B	Stern bearings C	Outer oil glands A	Outer oil glands B	Outer oil glands C
Cargo	—	36·5	—	1	36·5	2·7
Tanker	11	358	3·1	18	358	5·0
Bulk carrier	2	417·5	0·5	21	417·5	4·8
Passenger	—	8	—	—	8	—

MR. R. LOFTUM, B.Sc., said that, as a classification engineer, having studied the Glacier-Herbert design at earlier stages, he had found it most interesting to see that a practical solution for this type of stern gear was attained in the end.

If now a real breakthrough took place, after a certain time, would it be easy for shipbuilders and suppliers of stern gear to handle such a construction correctly—or perhaps produce it—assuming that the present makers might not always be able to meet the demand for assistance?

The assembly and adjustment work would probably require more special attention from the classification societies than had been usual up to now.

Would the authors comment on one particular detail question, the four radial supporting chocks at the front end were understood to need grease lubrication of their bearing surfaces—was this found necessary from experience?

Should serious consequences be expected if the lubrication was neglected, this must in part be due to the strong clamping forces on the chocks' surfaces, combined with thermal expansion movements.

Conventional stern tubes, which were sometimes arranged centrally in a rigid hull tube, were in such cases often supported free to move at the forward end, only sealed against the bore in the bulkhead by O-rings. To his knowledge, no routine lubrication was prescribed for such mating surfaces.

It would be interesting to learn if a system of lubrication without frequent attention would be introduced.

In the paper it was described how it was possible to run with partly defective gland seals without sea pollution. Had this been demonstrated? This was most valuable and so was the possibility of checking the tightness.

Returning to the main point of the paper, that supervessels should not need docking because of stern gland trouble, it was natural to ask: how many large ships were already in existence with conventional stern tubes and how many would be completed like that, before repair-afloat type equipment became a standard specification?

Perhaps owners of conventionally built superships some day would wish that their vessels could have the stern gear converted to the modern type. It seemed to be justifiable that an attempt was made to find a solution to that, but would the authors consider such an idea to be within the bounds of possibility?

MR. C. G. VOLCY, M.Sc., M.I.Mar.E., in a contribution read by Mr. I. J. Day, M.I.Mar.E., agreed with most of the views put forward. In his opinion, the fitting of the old fashioned stern tube with two bushes was not at all logical and might be considered to be the cause of many of the problems which occurred on these vessels.

He agreed with the warning not to decrease the length of the aft bush of the stern tube to less than two diameters—which value was introduced into Bureau Veritas rules several years ago. In his experience it was necessary for the aft support of the tailshaft to have extra length for emergency cases. He knew of examples where, due to broken propeller blades, the white metal of the aft bush had been damaged (the length of this bush being two diameters) where the damage was spread from the aft extremity over a length of 2D, which meant that, in the case of damage to white metal, the surface of which was spread over $2\frac{1}{2}$D, sufficient smooth surface was provided on the lower part of the bush to enable the creation of the oil film. Thus, the tailshaft could run without the further overheating which could lead to the white metal running out and subsequent cracking of the tailshaft. That could endanger not only the security of the stern gear assembly, but also the security of the ship itself.

He was rather surprised at some of the values of minimum oil thickness in Table I, particularly for bushes having an L/D 2·14. Were these values calculated or measured? In his experience, the oil film thickness did not exceed 0·08 to 0·12 mm, irrespective of the L/D ratio, or even for L/D = 1. In fact, the case quoted by the authors of L/D = 2·14 could not actually exist. The curved tailshaft was practically never married over the whole length of a bush having L/D = 2. He would be satisfied if, for such a bush, the actual contact was spread over a length L/D = 1, and this could be obtained, either by scraping the aft extremity of the bush, or by arranging for double slope boring.

Another factor which influenced the actual behaviour of the tailshaft and its aft support was the bending moment applied to the propeller, due to the eccentricity of the thrust. During alignment, only static forces were present and thus, the alignment could not take the bending moment into account. In the running condition, this bending moment was variable. Thus, the curvature of the tailshaft was also variable and the contact conditions between the tailshaft and its aft support were changeable. From his experience, the values of this bending moment and the eccentricity of the thrust were seldom obtained, and then only with difficulty. These considerations showed that, in reality, there was no contact, or that contact was not achieved over the whole length of the aft bush, which led to the conclusion that the minimum oil film thickness would be somewhat closer to those indicated in column L/D = 1 of Table I, but not exceeding 0·12 to 0·15 mm. In connexion with what had been said previously, there arose the problem of the creation of this minimum oil film thickness in the shafting of large powerful ships when their turning gear was in use. Mr. Volcy's opinion was that this was the most severe condition to be met during the life of the aft support of the tailshaft and its corresponding journal. Bureau Veritas were studying this problem deeply and hoped to come to some conclusions in the near future.

MR. J. H. MILTON, M.I.Mar.E., said that, although ships had to be dry docked at periods not exceeding 2 to $2\frac{1}{2}$ years and at every 4 to 5 years for special survey, the fact that the flange mounted propeller and tailshaft could be examined *in situ* and afloat if need be, must, until oil glands were made trouble free (especially in the case of large tankers), be a big advantage.

Had a classification tailshaft survey on one of these split bearing installations been done afloat? In view of the importance of the bolts securing the propeller to the tailshaft flange, could these be withdrawn individually for examination at such times?

Both designs allowed the bottom half bearing to be removed

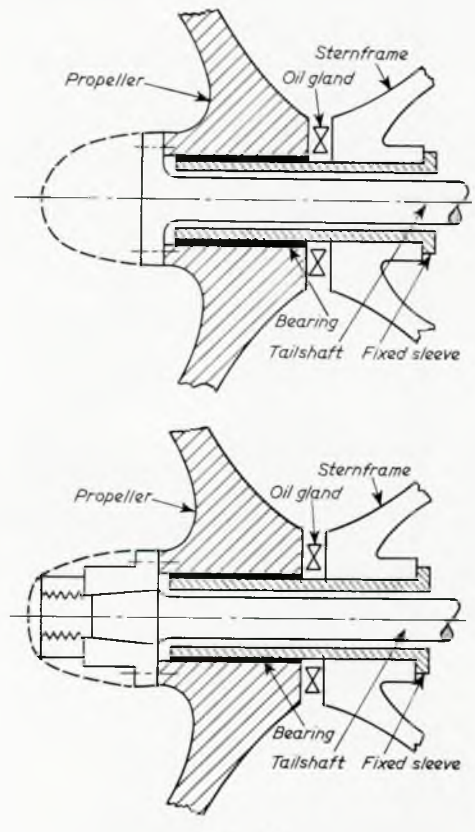

FIG. 15

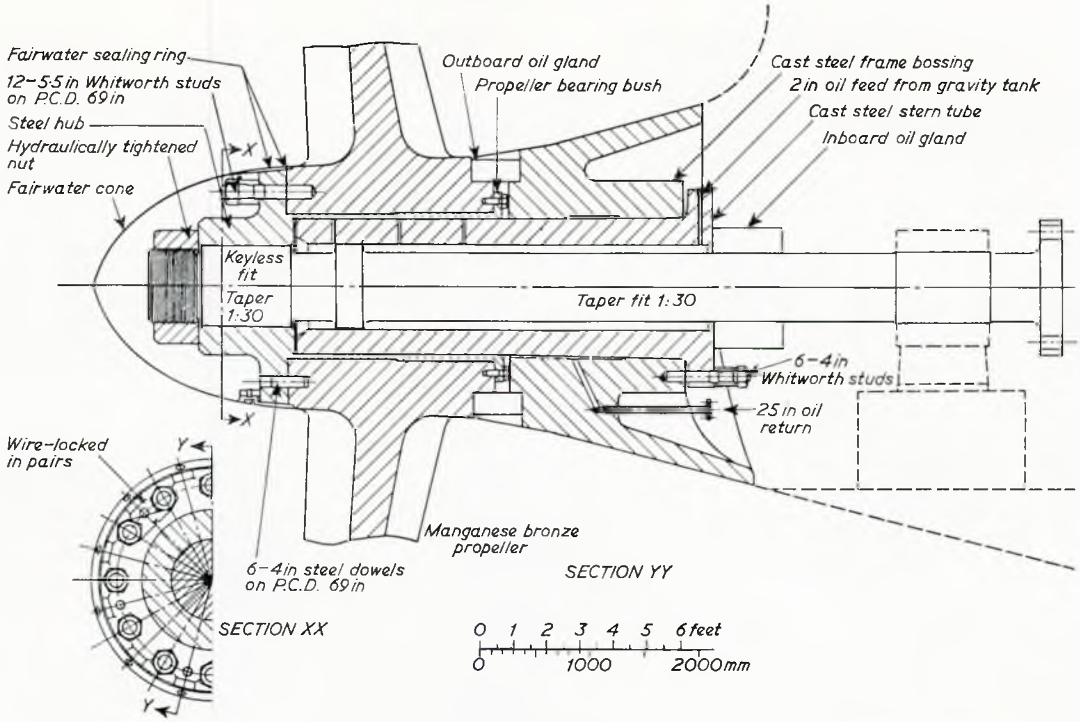

FIG. 16

for examination, while the shaft and propeller appareutly dropped on to an external support. Were these supports not liable to corrosion and did the spherical carrier ring have to lift the weight of the propeller and shaft when the bearing was being re-assembled? If so, was it not possible that scoring of the surfaces would take place? Furthermore, were any means provided for checking that the spherical seating ring joint had become solid before the inboard flange landed on the forward frame ring?

The title of the second paper mentioned "sterngear design" and it was contended that a new design of stern gear, not only the stern bearing, was long overdue. Was it good engineering practice to secure a heavy propeller to the free unsupported end of a tailshaft and then, as a result, to subject the tailshaft and shafting to bending moments and deflexions through the overhanging weight and out-of-balance thrust of the propeller?

It was felt that the automobile industry could perhaps teach us something in this field. Some years ago it was quite a common sight to see cars and lorries canted over to one side with one rear wheel off, through a half shaft breaking. Those shafts, like our tailshafts, were subjected to torsion and bending.

The automobile engineers found the answer by letting the wheel revolve on the outside of the axle casing, taking the bending off the shaft and letting it transmit torque only.

A stern gear on these lines was shown in Fig. 15 and one roller bearing manufacturer maintained that he could provide bearings, for a 40-ton propeller, which would last the life of the vessel, but again, lubrication was one of the major difficulties.

There was a fixed sleeve in the stern frame and the propeller revolved on the outside of it. The tailshaft transmitted torque only, and thus could be of smaller diameter. The tailshaft passed through the sleeve and drove the propeller either by a bolted flange, as shown in the top sketch, or by a core and nut, as shown in the bottom one. The oil gland came between the propeller and the stern frame.

The advantages claimed for such a stern gear were:

1) bending stresses are largely eliminated from the tailshaft;
2) increased flexibility of the tailshaft reduces the possibility of misalignment troubles;
3) the keyless connexion eliminates the usual stress concentration at the forward end of the keyway;
4) the steel to steel taper connexion eliminates variation in grip at different temperatures;
5) the shafting system is torsionally more flexible and this is often advantageous when the engines are aft;
6) the tailshaft can be examined without removal of the propeller;
7) running heat will be dissipated much more readily than in a conventional arrangement.

Fig. 16 showed the proposed taper connexion for the propeller, for a 30 000 hp installation. This was the Stone-Milton arrangement.

MR. A. N. S. BURNETT, M.I.Mar.E., said that here two separate, independent marine companies had succeeded in stripping off the cloak of convention too prevalent in the marine industry and had tackled a problem from its grass roots.

These two papers represented what marine engineering was about: researching the market to determine what the customer would like if he could get it, carrying out value engineering and feasibility studies, with more research, and even development, and then finally arriving at a solution. All the facade of convention had to be torn away and a system, which the customer would buy in the market place, put in its place.

Many ingenious devices had been witnessed over the last few years, e.g. new ways of mounting a conventional propeller on a conventional propeller shaft, but one was now faced with a new concept, achieved by system and value engineering techniques. It was a pity there had not been such a gathering at the earlier annual session, when value engineering was explained in detail*. This development was, even so, meritorious.

Two personal experiences emphasized the highlights of the second paper. Firstly, in Australia, Mr. Burnett's ship had had to drydock at Williamstown because some trouble with the propeller shaft was suspected. It was found that one of the shaft nuts was loose and the thread corroded. The only solution was to remove the tailshaft and to build up and remachine. The nearest lathe which could take the tailshaft was 100 miles away, so it had to be taken out, transported there and back by lorry, and replaced, which added up to a considerable cost and loss of time. With the equipment described in the paper, this would not have occurred.

Secondly, at one stage, he was responsible for vessels classed

* Matossian, B. G. 1968. "Developing and Organizing an Effective Value Engineering Programme." *Trans.I.Mar.E.*, Vol. 80, p. 417.

under a major classification society, with controllable pitch propellers, on two shafts. It was explained to the classification society that the aft stern bearing could be removed for examination in dry dock without the need to remove the flanged pod. They insisted that the tailshaft should be withdrawn according to standard convention. Therefore the c.p. propeller pod and tailshaft all had to be withdrawn at considerable expense: with the present arrangement, this would not have happened.

There was, however, another important factor not referred to in the paper. In certain types of c.p. propeller, the control rod ran down inside the tailshaft to the pod, and there were some further problems regarding dismantling and reassembly of the c.p. propeller control gear. No such gear need now be touched at tailshaft surveys. Had the authors any comments?

In the eternal triangle of the marine industry—the ship operator, the building yard, and the supplier—none could-exist profitably without the help and co-operation of the other two, In the paper, under "Guidelines for Development", items 3, 4, 5, 8 and 9 involved all three sides of the triangle. It looked easy when put down on paper, but in practice one's own managing and financial directors had to be convinced of the need to examine these problems and to cater for them—it was not easy when money for R and D was not in plentiful supply.

There were two items to add to this list of guidelines. The first was to ensure, from the outset, the co-operation of the classification societies, and that must be long before any metal was cut. One must try to persuade them that shaft survey intervals could be further relaxed, giving further benefits to the user. This was beginning to happen, e.g. docking intervals had been relaxed when impressed current systems and other arrangements were utilized.

Secondly, in all developments there must be control cut-off points, if it was evident that the development was unsuccessful, and there must always be built-in marketing plans from a very early stage, if there was to be any reasonable success. In the air and other transport industries the world over, members from each side of the triangle were willing to sit down together and work out developments needed by all parties, so that each party did not have to work in isolation.

In the paper the question "Why?" had been asked. This question must be asked continuously and the answers would often be surprising.

The paper was packed with research data. These were facts: sound innovation and development could only be based on sound market research.

Before looking at the crux of the affair and asking whether it would pay to adopt this system, there was first another question to ask: did the marine industry realize that the cost of its maintenance was approximately double that for any machinery, plant or process in existence elsewhere? Mr. Burnett thought that the authors had underplayed the economic case, which was very real. They did not appear to have taken into account savings from the reduced scale of labour needed for dismantling and reassembling the propeller and bearing. Also the reduced manufacturing costs as more systems got on to the production line were not taken into account. No claim was made for cost saved by the shorter engine room mentioned.

We could now look confidently forward to performance checks at shaft surveys by probe, ultrasonic and other techniques, without the need to dismantle the bearing at all, unless it were found to be defective.

Mr. G. Eller said that his company had installed the first three Glacier-Herbert stern gears on the three Ugland ships mentioned.

The casting of the stern gear was fully machined in the machine shop, and thus it was not necessary to machine it on the slipway. After machining, the casing was welded into the ship's body, and there were no problems during the process if the casing was always welded on two side points at the same time. One point to watch was that the aft boring (No. 13 in Fig. 4), where the spherical seating was fitted, must be absolutely round—the difference between the vertical and the horizontal diameter must not be more than 0·05 mm, otherwise the spherical seat would be deformed and would not fit tightly. It had to carry the whole weight of the shaft and the propeller and had to be watertight.

A good point in this design was the ability to align the bearing after launching. On the conventional ship the two bearings in the stern tube were aligned by optical equipment, or other means, before launching, after which it was impossible to adjust them to the curvature of the shaft, if necessary. In the case of the authors' installation, after aligning the shafting, the inflatable seal was employed, and the alignment of the aft bearing could be corrected and there was the ability to follow the bending line of the shaft. There was also the possibility, following damage to the ship, e.g. to the hull, of adjusting the aft bearing and it would not be necessary to re-align the main engine. One could work on the bearing and on the seal all the time the ship was afloat. It was not necessary to drydock the ship to inspect the bearing or the seal.

The inflatable seal had to be extra tight, as its proper functioning was very important when working on the bearing, otherwise water would enter the tunnel. In this case, his company used one of the starting air bottles for this purpose only.

Mr. H. E. Assen, in a contribution read by Mr. J. W. E. Mansfield, said that he had been concerned with the installation of the prototype units in the Ugland ships.

In 1968 when the proposals for the sterngear were submitted to Lloyd's Register of Shipping, they were very carefully assessed, at both the society's Hamburg and London offices. The comments then made stressed the necessity for accuracy of construction and alignment and, by virtue of the good co-operation between all the parties concerned, this essential requirement had been well met.

One aspect concerning the welding of the sub-assembly, referred to under the sub-heading "Installation Onboard", was the timing of the welding so that the stern frame was welded into the sub-assembly, and then the sub-assembly was welded-in at a relatively late stage in the construction of the aft end, using reference points and targets. This sequence avoided distortion due to a large weight of fabrication being put on the aft end after the stern gear had been welded in place.

Again, from a practical aspect, great care was taken with such items as locking devices, seals and bearings, as well as clearances. It was worthy of note that the Ugland ships were fitted with KaMeWa variable pitch propeller units and the running temperatures of these units were found to be slightly higher than would be expected for a conventional solid shaft. This necessitated the aft bearing clearances being somewhat larger than for a conventional shaft.

The references to designs up to 50 000 shp/shaft inferred that multiple screw projects had been designed. These would be of interest, particularly in view of the latest constructions for fast container ships having fast lines and relatively flexible shaft bossings, or at the other end of the scale, for large single screw tankers having a shaft section modulus approaching that of the aft end structure of the ships about the vertical axis at the after sections.

The paper was a justification of a sound design which could be confirmed as being very satisfactory from the practical point of view.

Although several similarly purposed designs had been publicized, the Glacier-Herbert system was so far the only one to have been installed in larger vessels in Germany. It should not, however, be taken from this that the local designers were inactive in this field.

Mr. J. W. E. Mansfield noted from Fig. 7, a design study for a 250 000 dwt tanker, that a watertight door was shown in the bulkhead at frame 22, giving direct access to the stern bearing compartment from the engine room.

Lloyd's Register, however, would expect, in this type of installation, that the bulkhead at frame 22 be retained intact, i.e. without a watertight door, and sole access to the stern bearing compartment should be by a trunk, situated aft of the bulkhead at frame 22 and extending to a position above the bulkhead deck, or the first deck above the L.W.L., as the case might be, with a suitable closing appliance at its upper end.

In the installation shown in Fig. 6, where the normal shaft tunnel space existed, a watertight door fitted on the after engine room bulkhead, giving access to the tunnel and with a notice plate on the door to the effect that "this door is to be kept shut when the stern tube bearing is withdrawn with ship afloat" had been accepted, the usual escape trunk being provided from the tunnel space.

The view taken was that a tunnel space offered a larger floodable space in the event of external seal failure, even if the tunnel door at the engine room was inadvertently open at the time. This allowed some time for closing the watertight door without risk of flooding the engine room. Such a margin would not exist in the aft end machinery installation with the watertight door as proposed.

Correspondence

MR. G. C. VOLCY, M.Sc., M.I.Mar.E., wrote that, in his own paper*, he had illustrated white metal bearings which had deteriorated due to the non-rational alignment leading to negative reactions being present in way of the forward bush and, consequently, to loss of contact between the lower part of the bush and the corresponding tailshaft journal. This led to a considerable decrease of the natural frequencies of the considered elastic system and the appearance of a forced vibration resonator, amplifying dynamically the excitations from propeller forces. Such a resonator and the corresponding vibrations of the tailshaft and after part of the hull had also been illustrated in his paper. By detuning this resonator and by rational misalignment of the shafting, the dynamic amplification of the excitation forces was cancelled out and the shafting behaved normally, whilst in the case of the non-rational alignment phenomenon described, the white metal bushes were destroyed. However, it was not only a question of calculation, but also of being sure that the results of the calculation were actually realized. For a conventional two-bush stern tube, where double sloping in way of the two bushes was adopted, one was never sure what had really been done during the machining of these parts. These conclusions from his paper led to the solution to which everybody must come and which had been adopted and realized so well by the authors' company. Once somebody had the courage to break with all out of date principles, the advantages thereof multiplied. He fully agreed with the authors that alignment operations, in the one-bush stern tube, were greatly simplified, in fact they became child's play. This he could say from experience, a number of his society's clients having adopted the one-bush stern tube solution, several years ago.

The authors deserved praise for further continuing their research and resolving the troubles which could always eventually occur, e.g. leakage past watertight devices; also the design concept of their one-bush stern tube was important, in that it could be removed afloat.

* Volcy, G. C. 1967. "Forced Vibrations of the Hull and Rational Alignment of the Tailshaft" *Nouveautés Techniques Maritimes.*

Authors' Reply

In reply to the discussion, the authors said that Dr. Archer's statistics of failure confirmed that current sterngear performance was poor in relation to that of many other machinery elements, which were themselves recognized as troublesome.

Considering, in particular, the tanker group in which the higher power installations were probably concentrated, the statistics revealed an incidence rate of failure involving the outboard seal or the bearing itself of 8·1/100 shaft years in vessels built between 1967 and 1970. This showed that every vessel in the most modern sector of the world tanker fleet must expect such an incident during its normal working life.

The resulting time lost would, of course, depend on the circumstances of the incident, but with "Glacier-Herbert" facilities available, the saving in such lost time would be expected to be:

One to two days for a simple seal failure detected and dealt with immediately (i.e. usually those occurring whilst in ballast and conventionally dealt with by tipping the ship).

Four to six days in the case where a ship had to run on after seal failure to the extent that a weardown check and bearing inspection in dock was called for (i.e. usually those occurring during the early part of a loaded voyage).

Six to ten days in the event of a bearing failure (this might follow a seal failure or be the result of a bearing problem).

The average time to be saved was thus five days at each incident and this must be anticipated on the basis of the statistics. To this must be added a possible greater saving due to the elimination of diversion, waiting for dock, docking and repair bills.

It could be argued, and reasonably expected, that improvement in component design would improve the statistics. On the other hand, the average age of the ships to which the figures related was only four years. The inclusion of a reasonable "waiting or diversion" factor adequately protected the validity of the "five day loss expectation" even if there were to be a significant improvement in the statistics themselves.

Dr. Archer's reference to vertical deflexion over the length of the engine room was interesting. A recent report coming to the authors' notice showed a vertical deflexion over the length of machinery of over 2 mm between light ballast and full load conditions in a large tanker.

The effect was to produce a *change* in shaft alignment of about 0·0003 radians. Comparing this with the curves in Fig. 2, if the bearing alignment could be set so accurately that this change became $\pm$0·00015 then the bearing could probably tolerate it. If, however, the change produced a slope range of 0 to 0·0003 (or perhaps $+$0·0001 to $+$0·0004) then edge wear might occur. The authors submitted that this accuracy of alignment could only be achieved in practice by a gear which allowed final adjustment with the ship afloat. With such a gear, the shaft alignment in the bearing could be measured for both extremes of loading and a desired setting made.

A further influence to be considered in this context was that of increasing service demands in relation to sterngear capability. As power outputs, ship speed and size increased, many aspects of sterngear became more critical and these the authors sought to anticipate when defining the "guide lines for development" summarized in the paper. The achievement of a significant improvement in reliability under determinate conditions and in ultimate capability under limiting conditions had been a fundamental objective in developing the design.

Mr. Loftum had asked if shipbuilders and suppliers of sterngear could handle the execution of the installation correctly if present suppliers could not meet eventual demand for technical assistance.

In the matter of installation design, the authors and their colleagues had made a number of design studies for various ship types with shafts from 450 mm to 950 mm diameter to show how the form of the bossing and structural arrangement of the stern could best be related to the functional requirements of the sterngear system. The gear could be installed in the majority of ships without recourse to difficult steelwork arrangements and there were already indications that builders with a "will" to make this type of installation experienced no difficulties.

With regard to the installation itself, there were no aspects which were particularly demanding in relation to normal marine engineering practice and in fact, in some aspects, the work was simplified. Mr. Eller had confirmed this in his paper(14) which dealt with the prototype installations for which he was responsible. It was significant that the builders felt sufficiently confident to make the third installation without requesting technical assistance from Glacier.

In respect of the actual manufacture of the sterngear, it contained relatively simple and robust components and there were no features which would be beyond the competence of the average bearing manufacturer acquainted with heavy engineering practice. Arrangements were in fact being made for manufacture of the sterngear in areas where there was special incentive to do so.

The radial chocks which located and secured the forward end of the bearing within the forward bore of the stern frame were provided with grease connexions as a precautionary measure, since they were arranged to accommodate small expansion differentials between the bearing assembly and stern frame. The object was primarily to guard against the possibility of corrosion at the interface rather than to lubricate. Frequent greasing had not been found necessary in service and examination after a year's service showed no trace of fretting. It would probably be quite satisfactory to coat the surfaces with water displacement grease on assembly.

Mr. Loftum had also asked about the ability of the design to prevent sea pollution in the event of partial failure of the lip seals. Protection against this and against pollution of oil with sea water was provided by disconnecting the pipes fitted for supplying oil to the central cavity of the outboard gland. The cavity then formed an open cofferdam. The effectiveness of the arrangement was amply demonstrated during the *Torinita* incident, when disintegration of the mild steel plug fitted in error in the gland housing made a 10 mm diameter connexion between the sea and the seal cavity. With this cavity drained and the bearing system refilled with clean oil, the vessel ran for 2000 miles at full power without ill effect and there was no sign of water contamination when the bearing was subsequently withdrawn. With two 15 mm pipes draining to within the vessel, leakage would have to be very large indeed to cause a build up of level in the cofferdam sufficient to introduce the hazard of contamination. An added advantage of this arrangement was that oil leakage could be recovered.

The two items 17, Fig. 4, were seal test and drain connexions the function of which was to verify the effectiveness of the pneumatic seals, or external bandage, and to drain off the trapped water in the boss cavity before the bearing assembly was withdrawn.

These connexions are shown more clearly in Fig. 17. Prior to underwater withdrawal of the bearing and seal assembly, the linked cocks (1) are opened and water discharged to verify that the pipes are not choked. The aft pneumatic seal is then inflated and when water flow ceases, the seal is proved effective and drainage complete. A duplicate system (2) is fitted to the annulus between the two pneumatic seals. By repeating the above described procedure, the performance of the forward pneumatic seal is also checked.

A secondary function of the aft pneumatic seal test system is to inject filtered water into the outboard cavity in a circumferential manner in order to prevent the penetration of abrasive media into the space surrounding the outboard working seal.

Referring again to Fig. 17, there are two "O" rings (3) at the ends of the spherical interface with a circumferential groove between them. A tapping from this groove with small bore tail pipe to the face of the forward diaphragm forms an open drain from which a discharge will be seen in the event of leakage past the water or oil side "O" ring. This connexion also prevents entrappment of air or fluid during assembly, and provides a means of pressure testing the tightness of the joint after the bearing is fitted and secured.

Mr. Loftum had also referred to the possibility of converting existing ships. The authors' view was that, although this was feasible, it was unlikely to be economic.

With regard to the prospect of the real breakthrough to which Mr. Loftum had referred, the authors felt that marine engineers in operating companies, who had experienced the serious consequential losses which attended a simple fault, and those in building yards who were alive to the very real problem of ensuring precision alignment, had not been slow to appreciate the logic of the solution offered. So far, the principal difficulty had been to convince shipbuilders that these problems did exist and hence to induce them to think progressively when calculating the cost of the relatively simple structural changes involved. There was initially some resistance from naval architects due to increased boss diameters, but current Glacier-Herbert designs with boss diameter ratios down to 20 per cent had overcome this objection.

The latest available designs also showed significant improve-

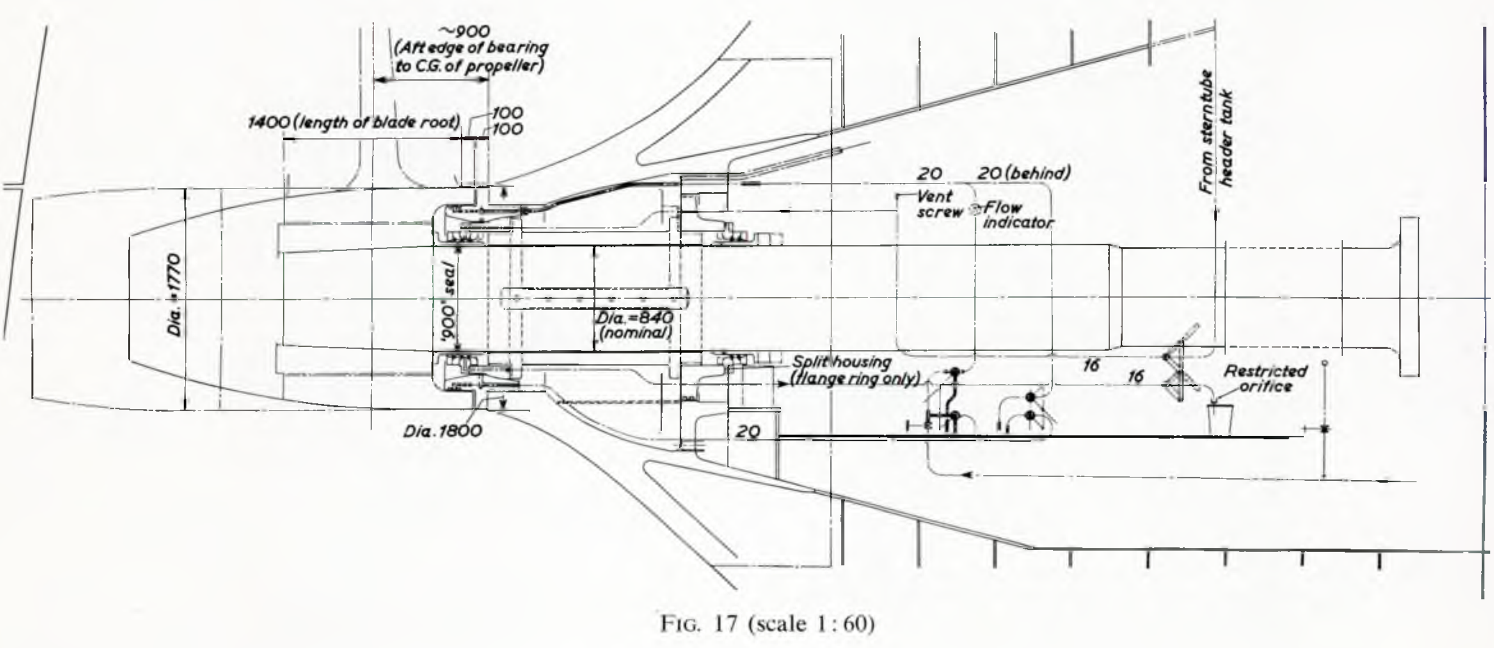

FIG. 17 (scale 1:60)

ment in cost effectiveness. Fig. 17 showed a stern gear assembly for 36 000 shp at 86 rev/min using the Pilgrim Mk III propeller referred to in the paper. It would be noted that the shaft and propeller were perfectly conventional and induced no extra cost. The net extra cost was, therefore, the difference between cost of the Glacier-Herbert assembly and that of a standard bush with seals, plus the cost of steelwork changes. The latter was not of major significance on a steelweight basis and those builders who had studied the design with a will to adopt it had confirmed accordingly.

It could further be noted from Fig. 17 that propeller overhang was minimized and the arrangement was such that all inspection and survey operations, including propeller changes, could be effected without removing the shaft or breaking the coupling.

Lip seals were shown in the illustration and both inboard and outboard bushes could be replaced from outboard when the propeller was removed and the bearing dismantled for examination at survey. In the interim there was provision for axial repositioning of the seal lips.

The arrangement shown was also offered with face seals of fully split type and all proprietary seal designs known to the authors could in fact be accommodated.

The authors were grateful to Mr. Volcy for his stimulating comments and for the work he had published on shafting alignment and vibration.

The figures in Table I were calculated values, but allowance was made for the curve of the shaft over the length of the bearing, the bearing bore being straight. For the $L/D = 2{\cdot}14$ case, the "mean line" curvature was estimated to be 0·05 mm rise at the centre of the bearing. The oil viscosity used was consistent with the temperatures measured during trials of the vessel.

The effect of operating temperature and consequent viscosity was very significant. For example, the film thicknesses shown in Fig. 2 would reduce to approximately one half if the oil temperature rose to 50°C.

With a conventional stern bearing arrangement cooling was likely to be less effective and the temperature would, therefore, be higher. This might account for the lower figures obtained by Mr. Volcy.

The film thicknesses were calculated by a method which was in close accord with the experimental results on misaligned bearings obtained by Mr. Asanabe (ref. 13, since published). These included trial results on a stern bearing in a 200 000 dwt tanker.

However, the authors' main purpose was to expose the problem and thereby highlight the need for good alignment if wear and damage were to be avoided. On Mr. Volcy's figures, the problem was shown to be even more acute.

One of the problems in stern bearings was the change in shaft slope and curvature in the bearing due to changes in the

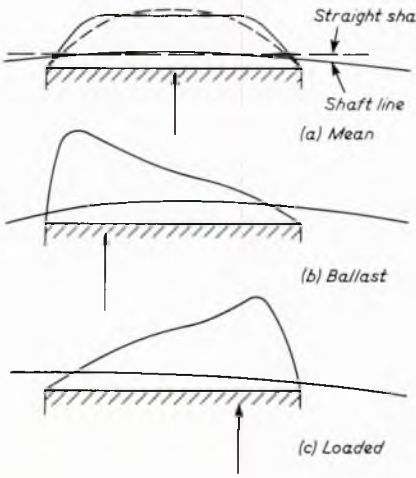

FIG. 18—*Indicating effect of shaft slope in bearing upon film pressure distribution and position of resultant action*

loading of the vessel and the power being transmitted. Not only did these react on the loads carried by the bearings, but the position of resultant bearing reaction also shifted.

Fig. 18 had been prepared to illustrate this. The condition of correct alignment, zero net slope, is shown at (a) for both straight and curved shafts. The distribution of oil film pressure is indicated. The resultant reaction is, of course, central. Shaft slope results in an asymmetrical film pressure, as shown at (b) and (c), with a corresponding shift in the position of the resultant reaction. This may well affect the whirling characteristics.

Had Mr. Volcy taken this into account in any of his work?

His suggestion for a deliberately shaped bore was interesting. On several occasions the authors' company had suggested the use of a curved bore to match more closely the shaft curvature, but to date no builder had ventured to apply this in practice. It was, however, a logical solution.

The authors agreed that the most severe conditions occurred during operation of turning gear, or very slow speed running, particularly after a period at full power when everything was warm and the oil viscosity at its lowest. Again, this condition required the best alignment over the whole length of the bearing.

It was felt that with further increase in size and power, considerations of safety and avoidance of wear would dictate the use of jacking oil. This would be very beneficial at slow speed, but involved the added complication of a high pressure system. However, this had long been standard practice in land installations and there was no reason why it should not be adopted for stern bearings where conditions were even more onerous.

In reply to Mr. Milton, it was not the authors' original intention to provide for surveys afloat, since these could be programmed ahead and would coincide with a drydocking. But there was nothing to prevent a survey being carried out afloat. With the tailshaft moved aft by about 500 mm, the propeller bolts may be withdrawn in rotation for inspection without removing the propeller.

It was agreed that the propeller support surfaces were liable to corrosion and it was for this reason that, in the Glacier-Herbert system, all surfaces in contact with sea water had been made of non-corroding materials.

During dismantling and reassembly of the bearing, the weight of the shaft was taken by a jack placed immediately for'd of the bearing—see Figs. 7 and 13. This avoided risk of scoring the spherical surfaces. No scoring had been experienced in practice, even after several withdrawals. The analogy to automotive practice was interesting, but few, if any, tailshafts failed at the point of maximum bending stress.

Mr. Milton's proposed sterngear was very ingenious, but the authors felt impelled to comment that it did not meet all the desiderata for modern vessels. Had the effect of thrust eccentricity been fully considered?

Mr. Burnett's remarks were typical of many which had sustained the authors in their endeavours. On his reference to co-operation with classification societies, this had been maintained throughout development and their helpful guidance and constructive criticism were gratefully acknowledged. The contribution to the discussion by representatives of three major classification societies was ample evidence of their continuing interest.

Present requirements concerning survey intervals were by no means onerous with this type of sterngear and the authors did not feel the need to campaign on that issue.

The excellent film shown earlier by Messrs. Crombie and Clay brought one to realize that there had been little change in sterngear from Brunel's *Great Britain*. Was it a reflection on marine engineering that the design had been virtually unchanged to the present day?

Mr. Burnett had mentioned two personal experiences. There were, of course, many cases of failures and delays—one heard time and again of ships being delayed in out-of-the-way ports for 10, 20, 30 days and even longer. Yet even after such experiences, many of the executives concerned still hesitated to depart from the conventional system. One could only hope that the example of the more enterprising owners would overcome this cautious attitude.

However, apart from incidents of failure, the authors

TABLE V

Type of ship	20 000 dwt product tanker		35 000 dwt bulk carrier		130 000 dwt tanker		260 000 dwt tanker
Horsepower	10–14 000		10–14 000		20–25 000		36 000
Size of propeller shaft	560–590 mm		560–590 mm		710–750 mm		840 mm
Type of propeller	c.p.	solid	c.p.	solid	c.p.	solid	solid
Estimated net extra cost fitted	£13 000	£12 500	£13 000	£12 500	£23 000	£21 000	£25 000
Cost saving at 4 year survey (assuming constant values)	—	£ 2 500	—	£ 2 500	—	£ 6 250	£ 7 000
Net debit after 4 years (assuming no incident)	£13 000	£10 000	£13 000	£10 000	£23 000	£14 750	£18 000
Cost saving at 8 year survey (assuming constant values)	£ 7 000	£ 2 500	£ 7 000	£ 2 500	£10 250	£ 6 250	£ 7 000
Net debit after 8 years (assuming no incident)	£ 6 000	£ 7 500	£ 6 000	£ 7 500	£12 750	£ 8 500	£11 000
Average daily income (minimum economic)	£ 2 150	£ 2 150	£ 2 750	£ 2 750	£ 6 750	£ 6 750	£10 000
Additional days 'on hire' in 8 years required to offset debt	2·79	3·49	2·18	2·73	1·89	1·18	1·10
Essential survey time for tailend work (a) conventional (b) Glacier-Herbert	8/9 2	2 × 4/5 2 × 1·5	8/9 1·5	2 × 4/5 2 × 1·5	8/9 2	2 × 5 2 × 2	2 × 5 2 × 2
Possible days saving by Glacier-Herbert	6–7	5–7	6·5–7·5	5–7	6–7	6	6

believed that the economics were favourable on a basis of the savings which could be made in normal survey time and costs.

Some owners had said that with modern cleaning and painting techniques, and with ships on continuous survey, the off hire times could be reduced as assumed in Table II. Others were sceptical, but nevertheless agreed that with the time for tail end survey so much reduced, it was reasonable to expect some reduction in total survey time.

The figures in Table V were, therefore, produced to show more clearly what must be achieved in additional hire time to justify the extra cost involved. The table showed the savings in direct costs of surveys up to eight years, and the additional days "on hire" required to offset the initial cost of the installation. The extra days on hire which it was possible to obtain by savings in survey times were shown. Comparing these, it could be seen that the initial cost should be recovered in eight years. As in Table II, all figures were at 1971 values. No allowance had been made for the cost of servicing the additional capital, nor for the effect of inflation. The latter would significantly increase the savings under present day conditions.

Mr. Eller's contribution and that of Mr. Assen were particularly welcome, in view of their practical experience with the prototype installations. Mr. Eller's simple adaptation of a standard air driven grinder which trued out the after bore in three hours was most effective in controlling the dimension he referred to.

With reference to Mr. Assen's comments on future applications, those he referred to had already been studied in detail and with suitable treatment of the afterbody design, the gear could be accommodated in vessels with extremely fine lines.

Replying to Mr. Mansfield and his remarks on flooding, the authors had been primarily concerned to design a system with adequate safeguards against this possibility and the consequences of total water seal failure had indeed been considered.

In the event of a simultaneous total collapse of both pneumatic seals, water would enter through the annulus area, which was equivalent to a 110 mm hole in a small ship with 500 mm shaft. In a large vessel with 840 mm shaft, the equivalent hole diameter would be 160 mm. Assuming a coefficient of discharge of 0·5 and depth of immersion of six metres for the small ship and 15·25 metres for the large one, the maximum ingress rate would be 3 tons/min and 8 tons/min respectively.

Referring to the tanker installation shown in Fig. 7, the time lapse to fill the floodable space to the sill of the watertight door would be five minutes, which the authors felt was ample time for evacuation of the space and closing of the watertight door.

In the authors' view, it should be quite reasonable to treat all installations as for conventional mid-ship designs in which the tunnel escape served as the SOLAS escape route from the lower engine room. Alternatively, the watertight door could be dispensed with and the stern compartment entered via a separate trunk from above the load line. This would be quite reasonable in modern vessels with full instrumentation to monitor the system.

REFERENCE

14) ELLER, G., 1971. "The Glacier-Herbert Sterngear System—A New Sterntube Design". Verein Deutscher Ingenieure, Hamburg.