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STERN BEARINGS AND SEAL FAILURES

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There are many papers and documents in circulation touching in depth on the merits of specific designed bearing seal systems employed on modern naval and commercial vessels. Each speaks of the successes and the failures of the system being promoted. In fact, the advances made in stern tube bearings and seal designs cover but a short span of time since the adoption of the screw propeller over 100 years ago.

The early naval architect and designer was confronted with the problem of supporting the shaft and bringing it through the hull with the minimum of bearing losses and ingress of sea water. The materials and techniques then available, were reviewed with the hostile environment of the sea always in mind. Early selections were wanting in both the ability to do the job efficiently and avoid costly maintenance. Eventually a fibrous bearing material was found which proved to be successful, although wear still persisted and a limited amount of maintenance was still required. Further refinements were definitely needed.

Many years passed before this problem was again taken up with enthusiasm and the determination to do that which had not been done before; progress was about to be made and bearings using rubber and plastics were introduced. Both the wood fibre and the elastomer bearing were water lubricated. Sea water was excluded from the ship only by an internal packing gland system. Little design attention had been paid to this gland since its maintenance was nominal and required no particular skill. Meanwhile, the overall problem of bearings and seals persisted, since a full solution had not been resolved. For years stern tube frames and tail shafts were refitted on a routine basis with new bearings and liners, with the knowledge that limited performance and eventual replacement were to be expected.

The adoption of the oil lubricated stern bearing was delayed because the conventional packing gland could not be used for both the exclusion of sea water and the retention of oil in the stern tube bearing chamber.

Progress has again been made through the development of several types of sealing systems which now assure the good performance of oil lubricated stern bearings.

This paper is an attempt to appraise bearings and seal systems and report on failures thus far encountered.

ANALYSIS OF BEARING FAILURES

This paper will attempt to bring the record up to date on the failures which have been known to occur during the past ten to fifteen years and includes analyses of three types of stern bearing designs and several seal systems.

It is obvious that there is a close physical relation between the stern bearing and the seal system which prevents the ingress of water around the shaft. In some cases, the failure of one can cause the failure of the other.

For many years the accepted material for stern bearings in propeller driven vessels has been fibrous lignum vitae material. It was selected because of its ability to maintain a propeller shaft relatively true, and to support it as it passes through the stern frame. To prevent the ingress of water, a waste packing gland is used inboard of the bearing. This type of bearing depends largely upon sea water for its lubrication and, while in an unworn condition, is quite effective. On vessels which ply the deep oceans and seas, reasonable life can be expected from this bearing material but when ships are required to manoeuvre in shallow rivers and estuaries, the water is highly polluted with suspended abrasive materials which pose a threat to the life of the bearing material. Some vessels require rewooding and others bearing and liner maintenance on short overhaul schedules. Wear-down of the bearing may be from 0·020 to 0·300 in, as noted for tankers with machinery aft. The inboard seal, however, can be maintained with less difficulty than the bearing, since it is more readily accessible.

As wear continues on this type of bearing material, the shaft is further displaced from its alignment and vibration and pounding often occur. This is the sign that something must be done or severe failures of the shafting etc., may occur. A review of the engine room logs of many ships will attest to this operating condition and it is generally accepted as one of the prices which must be paid for sea propulsion.

The problem of wear and shock has been evaluated by competent engineers in the hope of at least reducing the wear rate of a wood, water lubricated, stern bearing. Studies into various grades and cuts of lignum vitae have given much promise of improved wear resistance and longer life. Materials other than lignum vitae, e.g. specially compounded elastomers and plastics, have been introduced for this application with considerable success. With these bearing materials, a better quality can be maintained and improved wear resistant qualities incorporated. Some are employed as standard for naval and commercial ships operating in all water conditions. In these cases, internal gland seals are employed in much the same manner as with the lignum vitae bearing material.

The history of sea water lubricated bearings is clouded

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by the bearings' consistent demand for periodic attention, which becomes routine with time and can be given only the minimum of attention. It is, however, an area which is vital to the ship and the overall efficiency of the power plant.

The use of oil lubricated stern bearings has been considered seriously by naval architects for many years. They have, however, been engrossed in other pressing problems of ship design deemed more important than the perfection of an oil bearing for this application. Twenty years ago, this problem was given priority and consideration was accorded to one continually troublesome problem. While it was relatively easy to develop an oil lubricated bearing design which could be employed in many cases within the same stern tube and frame configuration as wood, the companion problem of seals was not so readily solved. Of course, this could not be ignored and it was obvious that the water lubricated seal gland could not fulfil the requirement. A search was made for acceptable materials and designs which would perform a dual function of retaining both sea water and lubricating oil. Many seal designs have been produced to meet these requirements. However, the requirements of the application have been much beyond previous experiences and the use of the oil lubricated stern bearing awaited the development of a practical retaining seal system.

Two types of seal have been developed for this marine application, a lip seal type and a face seal type; and both have their inherent advantages and disadvantages. Many of both types are employed aboard ship for sealing other vital systems; however, none are required to seal such a large diameter shaft system. Until a successful large seal was introduced, the oil lubricated stern bearing could not be employed to meet the requirements of progressive ship owners and operators.

With the advent of a practical seal system, the first oil lubricated stern bearings became a reality. The first such systems were designed and built in European yards with much success and were developed for shaft journal diameters up to 500mm which at that time was considered large for merchantmen. The largest passenger vessels continued to use lignum vitae since their architects were less venturesome in the field of advanced bearing and seal technology.

With the advent of larger and faster tankers and cargo vessels, and now, the supertankers, the bearing and seal engineer is again being challenged to develop systems for shafting up to 1500mm. We truly do not know what will be expected tomorrow —the problem is to meet today's demand with the assurance that a practical and reliable system of stern bearings and seals can be guaranteed.

Modern oil lubricated bearings are dependent upon hydrodynamic lubrication. The moving surfaces of the journal and the stationary bearing are separated by a lubricant film somewhat thicker than the roughness of the surfaces. Friction is at a low level and little or no wear will occur. This condition would be the ideal; however, in practical applications, it is nearly impossible to obtain. Misalignment between the journal and the bearing in most applications makes it difficult to build up a thick film even with high viscosity lubricants. This condition is further exaggerated when the speeds are low and the loads on the bearing are high. Thus, the bearing approaches a boundary lubrication condition and the lubrication is largely dependent upon specific lubricant and surface conditions, rather than upon the viscosity of the lubricant. This condition is frequently observed during starting and stopping a system under heavy load. A good lubricant under these conditions is one which will exhibit good wetting possibilities. Since there is a likelihood of journal to bearing contact at the high points of each surface, adhesion will be prevented. With continued boundary conditions, friction heat raises the local temperatures to a point of near seizure between the bearing and the journal. The high points of such an inter-action frequently shear off of the bearing in the load zone and are deposited upon the journal or are spun off into the lubricant as minute amounts of bearing material This can accumulate outside the load zone and adhere to the bearing. This effect is frequently observed in bearings which under load have experienced a condition allied to boundary lubrication. Most oil lubricated stern bearings fail in this manner (See Fig. 1).

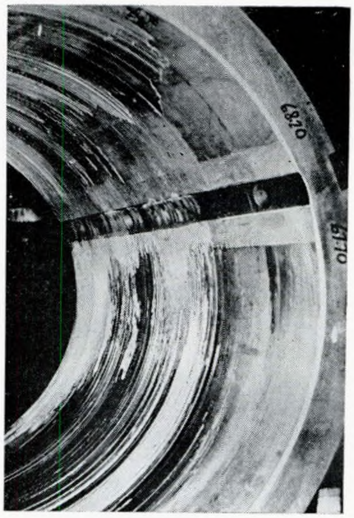

FIG. 1—*Failed oil lubricated white metal bearing*

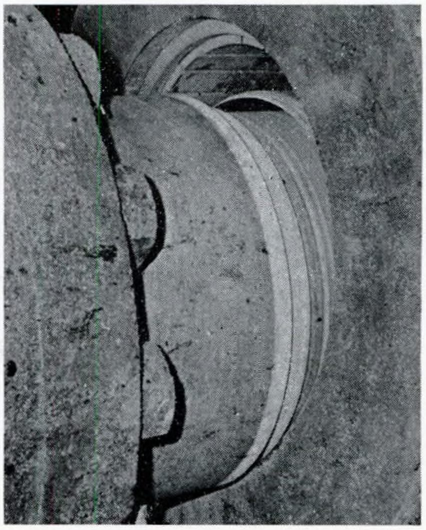

FIG. 2—*Worn lignum vitae water lubricated bearing*

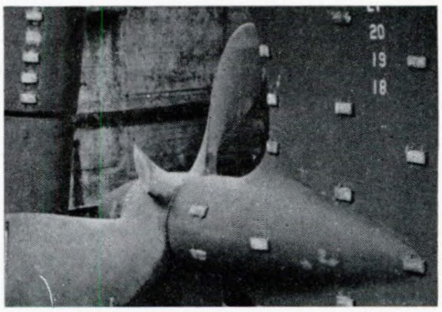

FIG. 3—*Cause of failed oil lubricated bearing*

It is difficult to define and catalogue the reasons for stern bearing failures. A review of performance records and ship histories reveal many reasons for failure. Some failed bearings exhibit characteristics of a slow failure rate, such as lignum vitae which fails primarily due to excessive wear, and rubber and plastic bearings which fail in a similar but more rapid manner. The oil lubricated bearing can fail quite rapidly under adverse conditions and is thus more sensitive to the elements which cause failure (See Figs 2, 3, 4, 5 and 6).

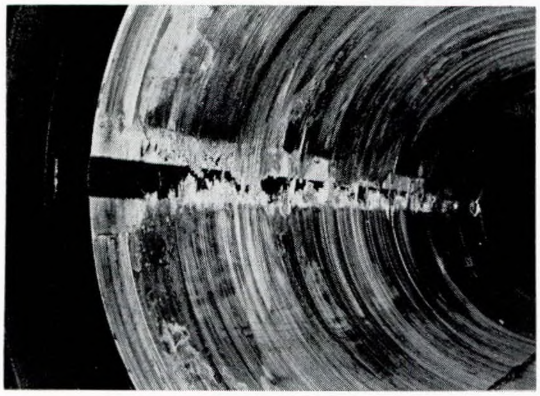

FIG. 4—*Failed oil lubricated bearing by overload*

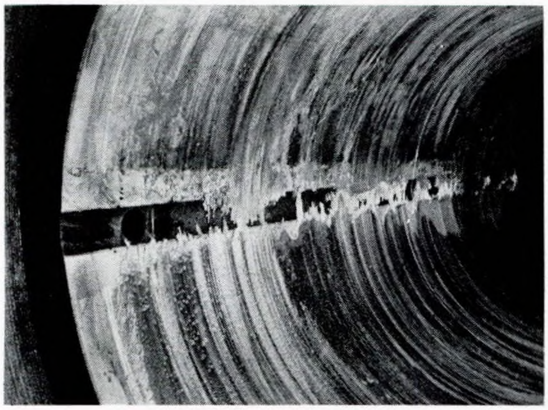

FIG. 5—*Failed oil lubricated bearing by overload*

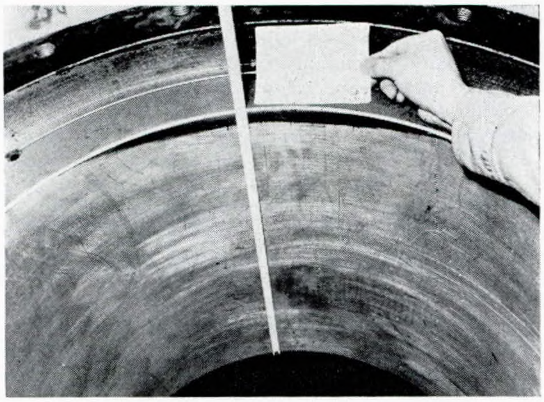

FIG. 6—*Forward end of forward bearing system as Figs 4 and 5*

The principal cause for stern bearing failure, in oil lubricated babbitt lined bearings, is a misalignment between the bearing and the journal. When this occurs, the system goes quickly from a hydrodynamic lubricated bearing, through boundary conditions into failure, largely due to non-uniform loading. Here, loads are concentrated in a small load zone and the lubricant does not have the ability to keep the sliding surfaces apart.

Failure under these conditions can occur either during initial line up of the journal, or by subsequent changes in the alignment after a period of what appears to be satisfactory operation under hydrodynamic conditions. It is possible that this operating condition may have been very marginal; however, upon a slight further misalignment of the system the bearing may fail.

Properly aligned bearing systems operating under hydrodynamic lubrication will function for indefinitely long periods. When these conditions are modified beyond the capability of the film to accommodate the displacements, failure will occur at an early stage.

It is obvious that the lack of the proper lubricant or the contamination of the lubricant with foreign materials can adversely affect the bearing and lead to early failures.

Seal systems which leak and allow water to enter the lubrication system contaminate the lubricants and reduce the load carrying capability of the oil film. Such contamination has led to failure in a relatively short running period. As pointed out earlier, it is important that the system be under hydrodynamic lubrication at all times. When this system goes through the boundary lubrication stage, failure is most likely to occur.

Babbitt metal fatigue which generally occurs in the highly loaded areas, at the aft and forward ends of the stern bearing has been the cause of some failures. Some white metal alloys are more susceptible to this type of failure than others, but even though an alloy may have been selected for its fatigue resistance, its physical properties can be measureably changed during casting. Complete and knowledgeable control of all cleaning and casting techniques is essential to produce bearings with the maximum of fatigue resistance—loose cracked babbitt metal has been a major cause of bearing failure. Good adhesion between the white metal and the shell must be ensured and the shell materials which show resistance to bonding unless the shells are especially cleaned and surface treated should be given particular attention. Loose babbitt metal can occur as flakes, the smaller flakes becoming suspended in the lubricating oil, flushing into the oil wedge and disrupting the oil film. These will finally restrict the oil grooves, thus reducing the lubrication, with eventual bearing heating and metal failure.

White metal looseness is attributed to lack of bonding to the shell; poor methods and facilities, and faulty quality control during the shell cleaning process are largely responsible. The strength of the bond is dependent upon good tinning of the clean shell and temperature control of the tinned shell and white metal when cast. Proper cooling and stress relieving is necessary to achieve satisfactory bond strength values; and methods and equipment are available to permit physical testing of the white metal casting to determine the bond strength.

The most commonly used device is a tension instrument which can actually measure the bond strength between the white metal and the backing shell. The system normally employed is known as the Chalmers Bond Strength Test (1). The Chalmers Test is conducted after preparing the bearing by isolating a plug of white metal by means of a trepanning tool and then drilling, with a flat ended drill, a co-axial hole through the shell to the junction of the shell and white metal. The plug is then pushed away from the shell by means of a force applied by a plunger through this hole. Typical values of bond strength of a high tin babbitt alloy are represented by Hedges (2) in Table 1. It should be noted that the cooling rate of the casting drastically affects the bond strength. Here the cooling rate is indicated and the temperature of the test versus the bond strength plotted.

TABLE I—THE EFFECT OF DURATION AND TEMPERATURE OF TINNING ON THE BOND STRENGTH BETWEEN 7 PER CENT SB, 3·5 PER CENT CU ALLOY AND STEEL

Time of tinning	Temperature, °C	Bond strength, tons/in²	Average bond strength tons/in²
15 s	280°	5·31, 5·41, 5·02, 5·57, 5·44, 5·21, 5·25, 5·46	5·3
15 s	380°	5·01, 4·68, 5·72, 5·35, 5·12, 5·08, 5·29, 5·31	5·2
5 min	280°	5·07, 5·28, 5·44, 5·50, 5·27, 5·21, 5·35, 5·25	5·3
5 min	380°	5·65, 4·88, 4·80, 5·60, 5·31, 5·17, 5·24, 5·20	5·2
60 min	380°	4·42, 5·05, 5·15, 4·61, 4·55, 5·15, 5·16, 4·62	4·8
60 min	450°	0·69, 0·70, 0·68, 0·55, 0·68, 0·74, 0·72, 0·66	0·7

The strength of the bond is dependent upon the duration and temperatures held during the tinning operation. Representative values are shown as reported by Hedges in Fig. 7.

During the tinning cycle of a ferrous shell, an iron/tin compound will be formed at the contact line of the two metals. It is the thickness of this compound which largely affects the strength of the bond. Prolonged exposure to tin, affects the thickness of the compound. It is quite obvious that acceptable

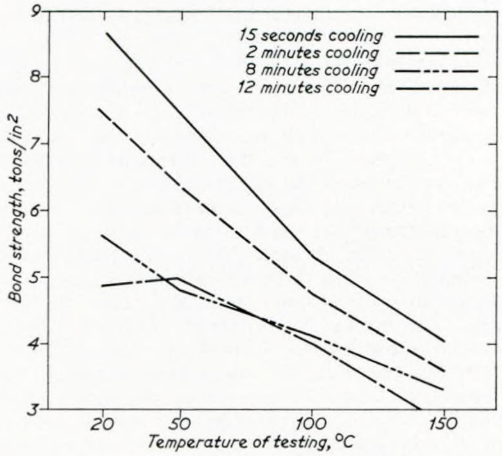

FIG. 7—*Influence of rate of cooling on bond strength*

results can be obtained with relatively short periods allowed for tinning.

The bond strength of high leaded babbitt alloys will approach those of high tin babbitt and the criterion for acceptable bonds are similar.

Recent methods in ultrasonic inspections have been developed which will identify areas of non-adhesion, which in some cases is deemed satisfactory. Normally a satisfactorily bonded bearing is one in which there is adhesion at the edges, and at least 95 per cent soundness elsewhere. The strength of the bond, however, cannot be measured by non-destructive means. Satisfactory bonds can be consistantly produced and quality monitored by proven methods (See Figs. 8, 9 and 10).

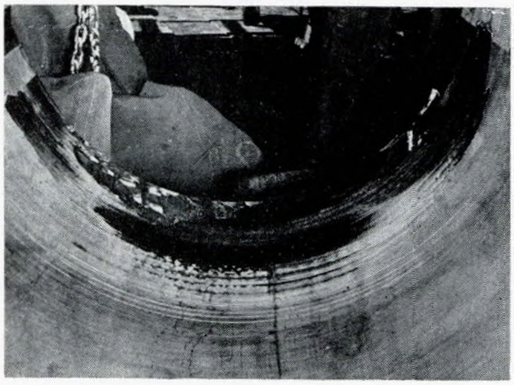

FIG. 8—*Failed oil lubricated bearing-bond and fatigue*

FIG. 9—*Distressed area of bearing in Fig. 8*

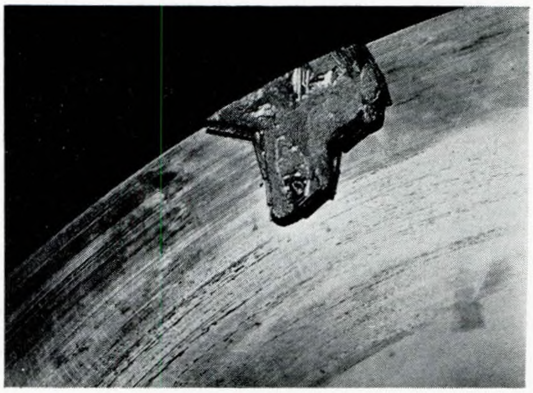

FIG. 10—*Distressed area of bearing in Fig. 8*

Proper design and materials selection in themselves, do not guarantee a successful oil stern bearing application. Proper installation technique at the shipyard is essential. Every yard encounters real difficulties on the first installation and some yards have carried these difficulties over into the second and even the third installation. But the problem need not be of such magnitude. The greatest difficulties have been encountered in the proper use of the boring bar; generally shipyards use the wrong sized bar. Boring bars which are long and flexible do not produce a true bore in which to press the bearing and shipyards with six inch bars have had to weld up stern tubes and rebore as many as two or three times. Some ships have been retrofitted for oil bearings without boring the tube, simply by the facing and drilling of the stern tube to accept the necessary bolting for the seal housings. In these cases the outside diameter of the bearings were turned to fit the existing tube bore. This can only be done if the old tube is round and is in true alignment.

Another problem in some shipyards is the failure of their outside yard people to recognize that this is a precision bearing and sealing assembly and that it must be treated with the same degree of respect shown to comparable bearings in precise turbines and gear equipment. Quite frequently grit blasting operations have been conducted near the assembly of the bearing and seals. Many stern tubes were found to be loaded with grit even though extensive flushing and cleaning was carried out. The volume of the oil in a stern tube is of such a magnitude (200 to 250 U.S. gallons) that flushing is quite ineffective in removing the bulk of the grit. Large volumes must be removed by hand before flushing. The work of assembling the bearings and seals to the ship should be scheduled in such a manner that the parts can be kept clean up to the point of closing the lubricating oil system (Figs. 11 and 12).

FIG. 11—*Dirt and residue in oil slot*

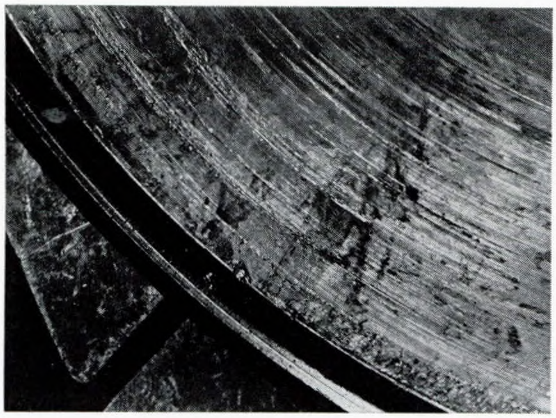

FIG. 12—*Effects of dirt and residue in bearing*

Current stern tube bearing practices dictate a design employing a shell wall thickness of roughly eight to ten per cent of the journal diameter. White metal, uniformly applied with thickness of approximately $\frac{5}{32}$ in or 4mm and finished to values of at least 32 r.m.s. are specified. The length of the aftermost bearing is generally an L/D ratio of 2·5:1. Projected loadings of 70 lb/in^2 are not uncommon and loads as high as 100 lb/in^2 are practical. It is assumed that there is proper alignment of the journal and the bearing, and that the structure is rigid enough to maintain this precise alignment. With these restrictions on line-up, it may be possible to reduce the L/D ratio and to increase the allowable loadings.

It is recognized that permanent alignment and uniform loading throughout the length of the bearing is quite difficult if not impossible on conventional vessels. To assist in relieving highly loaded zones, some naval architects advocate a relatively short aft bearing i.e. L/D of 1·5:1 and no forward bearing in the stern tube. A spring or plummer block bearing is positioned on the tailshaft just forward of the stern tube and aft of the tailshaft flange. This allows for adjustment of the spring bearing beyond the stern tube and relieves the loading of the forward bearing when mounted in the stern tube. Many ships are so equipped and are giving good service.

Further consideration of the problem of bearing and shaft alignment leads to a system where a self-centring bearing can be employed. Such a design has been developed and put into service on relatively small shaft systems where the journal diameter is 18 in. In this design, the bearing shell is fitted with a spherical or knuckle fit which allows the bearing to take its reference from the displaced shaft and afford more uniform loading throughout its length.

This design can be further refined by the use of segmental pads or tilting pads so located within a bearing housing that they align themselves to the displaced shaft. This design has definite advantages over the self-aligning bushing since the pads can be loaded to higher values with improved lubrication. On very large shaft systems where loads exceed 250 lb/in^2, it is possible to design a tilting pad system which incorporates means for shaft suspension through hydrostatic forced oil lubrication. No vessels have been equipped with such advanced stern tube systems although line shaft bearings are in service which employ this design concept.

In summary, stern tube bearing failures can be attributed to one or more of the following causes:

1) misalignment of journal and bearing;
2) contaminated lubricating oil systems;
3) excessive bearing or sleeve wear;
4) faulty bearing engineering and design;
5) bearing material failure;
6) faulty white metal lining technique.

This report touches only on the failure aspects of stern tube bearings. There are, of course, many satisfactory bearing systems which continue to perform well over long periods.

The merits of various bearing designs are quite well known. The capability of bearing materials are as well known and from these facts, the proper materials and design can be selected. It should be noted that nearly all newly constructed large tankers, as well as fast cargo liners and bulk carriers are fitted with oil lubricated stern tube bearings. It is the belief of naval architects and shipowners that they will thus have the minimum of difficulties and the maximum of performance.

ANALYSIS OF SEAL FAILURES

Any sealing system selected must be compatible with the type of bearing to be employed.

Fundamentally, the sealing system must be thoroughly reliable and operate with a minimum of attention and maintenance. Any seal system must perform at least one of the two basic functions—to keep water out of the ship, and to retain the lubricant within the bearing system.

A brief history of the seal evolution, however, will plot the course of their developments.

With the advent of the screw propeller more than a hundred years ago, the designer of the power system was confronted with a problem largely of supporting the shaft and the propeller, and bringing the shafting through the hull and bearing with the minimum of shaft friction and ingress of sea water. A wood stave type, water lubricated bearing made from lignum vitae was selected since it possessed a natural built-in oiliness which has a lubristic character when run in clear water and was readily available the world over. There remained the problem of affording a seal which would encircle the shaft and employ a resiliant packing material to prevent the ingress of sea water into the ship. This was solved by the development of an inboard gland system which has shown little improvement to date except in the quality of packing material used. Current glands employ modern flexible packings which have reasonably good wear life characteristics and successfully resist the ingress of sea water This type of sealing system requires a considerable amount of maintenance and periodic replacement of the resiliant members. It is in itself a low cost item and has been deemed successful for the average ship built up to the 1950's. With the advent of larger ships with a deeper draught as well as a larger shaft system, the gland with its resiliant member becomes more questionable and a considerable amount of discretion is necessary when it is considered for large diameter shafts and deep draught vessels.

With the design of larger ships employing propeller shafts to carry much more horsepower, it was necessary to develop a sealing system of such design that oil lubricated stern tube bearings could be used. The first work in this direction was conducted in Europe employing the principles of the lip seal design. The first sizes were developed for shafting up to 500 mm diameter, used with oil lubricated white metal bearings and were quite successful. In the years from 1950 to 1960 many European ships were refitted with the new improved oil lubricated bearings and lip type seals to the ultimate satisfaction of the shipowner and operator. The fundamental requirement of the seal had been met. The sealing system prevented the ingress of sea water into the ship and effectively retained the oil within the bearing system. Little maintenance was required as the seals performed well for extended periods of time.

A parallel development was under way to accomplish the sealing means through the use of face contact type seals. This type of sealing was not as readily accepted as the lip type seal since its operation was not as obvious as that employed in the lip type seal. Some large installation of the face type seal were employed as a sealing means to prevent the ingress of sea water into the ship and with considerable success but their popularity has not been as great as those designs employing the lip seal since ship experiences are somewhat limited.

Designs have been developed which employ both the principles of the lip seal and the face seal for application aboard ship. The experiences gathered to date on such an application are limited since relatively few ships have been equipped with this type of sealing system.

The most popular and successful type of seal system is one which employs resiliant lip seal rings which contact a revolving shaft liner system. The seal rings are made from a variety of elastomers which have been especially compounded for this specific type of seal and are giving satisfactory service. The majority of these ships are equipped with seal systems for medium to small size shafting. With the advent of the supertanker with large diameter shafting for high horsepower, some problems have arisen, since the periphial speed of the shaft, the internal and external pressures of the sea water and the lubricating oil, are considerably greater than on smaller and more lightly draughted vessels. The face type seals, likewise, have had this same problem; however, their applications to these large vessels have been relatively few. Histories of difficulties, therefore, are not available for review. It is known that this type of seal is sensitive to foreign matter in the lubricating oil system which works its way into the contact area and parts of the faces. Gyration or excursions of the shaft and rotating seal member also affect the rate of oil loss through this type of seal.

There are many reasons for shaft seal failures. Some of the failures do not stem from the seal itself but are caused by the bearing and lubricating oil—dirt and foreign material in the lubricating oil system have been particularly troublesome. Many problems arise because of the failure of the shipyard's personnel to clean the bearing and seal system thoroughly before the lubricating oil is applied. Likewise, the various lubricating oil tanks and piping system have been found to be contaminated with scrap material such as grit, sand, rust, weld spatter, gasket trimmings, paper, rags etc. The consequence of such inclusions in the system is generally detected shortly after the system goes into service and becomes evident by leakage of hot running oil from the seal, or even the failure of the bearings and seals.

Foreign material becomes to some degree, suspended in the lubricating oil system and is circulated throughout the seals and bearing which causes them to malfunction. It is extremely important that the complete lubricating oil system be purged of any foreign material by first carefully inspecting the system and then flushing with approved hydraulic techniques. Foreign matter in the lubrication system is particularly troublesome to face type seals and to a lesser degree, the lip type seal, since, in either case, the sealing surface is disrupted and leakage can occur. With the entrance of suspended dirt and foreign material into the bearing itself more critical damage is done to the bearing. With the continual flow of a contaminated lube system excessive wear and bearing failure can occur (Fig. 12).

The specific purpose of the seal system is to exclude the sea water and retain lubricating oils within the bearing structure. Experience has shown that oil leakage is more likely to occur across the forward seal system than it is across the aft system. When the balance of pressure within a bearing seal system is considered it is noted that there is a much greater pressure drop across the forward seal system than the aft where internal pressures are counterbalanced to a degree by the sea water pressures. The forward seal system, because of the greater pressure drop, is exposed to higher contact frictional loads which are converted into heat energy for dissipation within the shaft alley. A similar situation exists on the aft seal. Here, however, less pressure and lower frictional loads are encountered and the seal is more readily cooled by the sea water. The effect of the frictional heat in the forward seal system is a deteriorating effect on the sealing material which can change the characteristics of the material to a point where fractures can occur causing a possible oil leakage into the shaft alley.

Various manufacturers of lip type seal systems have developed methods to reduce frictional heat through circulating oil systems, thus affording a means to dissipate the heat before it can adversely affect the sealing mediums. These have been particularly effective on forward seal systems. Various high temperature sealing elastomers have been developed with the ability to withstand higher temperatures before deterioration. These materials show much promise for use on shaft systems of high peripheral speeds as currently being encountered on large supertankers. Since the pressure of the lubricating oil at the bearing is considerable, particularly at a deep draughted load line, the pressure differential frequently causes deformations in the elastomer, thus exposing more of the lip to contact with the liner. Mechanical supports are frequently used and prevent undue distortions to the seal and limit this adverse factor.

Some manufacturers of lip type seals are offering sealing elastomers with better thermal and mechanical characteristics. To date there are about 30 ships equipped with high temperature elastomers and the results of this experience have been quite encouraging. These seals have been in service now for approximately two years with few or no problems being encountered.

Some lip seals show evidence of cutting or grooving in the metal at the riding contacts. It has been noted that some early U.S. built seal systems have caused a considerable amount of grooving at the point of seal ring contact, caused by a combination of the forces on the seal ring at temperatures which change the characteristic of the elastomer. This cutting or grooving has been eliminated in the last four years by the selection of a more compatible metal alloy.

The use of a hard surface coating material applied to the liner riding surface appears to have definite advantages on large seal systems in the reduction of friction and prevention of grooving. Coatings have been applied to both the forward and aft systems with promising results. Seal systems employing this coating on the aft seal liner have not proven to be satisfactory, as failures have occurred due to sea water corrosion. On the other hand, coatings applied to the forward seal liner have been quite successful. The first such coating has been in service now for nearly four years and when inspected two years after application was found to be in perfect condition, with no deterioration or wear on the coating and little wear on the seal rings. Since the system is mounted inboard, there is no corrosion attack like that experienced on the aft system liner.

It has been the practice of shipowners and operators to carefully examine and replace any of the expendable elements of seal systems whenever the vessel is drydocked and the shaft withdrawn for inspection. This is a good practice to follow. The continued success of the seal systems can be assured when the vessel goes back into service. The period of time between drydocking for tailshaft inspection is dependant on the regulatory bodies; owners are anxious to extend periods of time between inspections whenever possible. Sealing systems with the ability to withstand extended periods in service are available and performing satisfactorily.

It is at the shaft inspection and seal wear element replacement fitting that much knowledge can be gained about the performance of the seal system. There have been instances where the wear on the elements has progressed to a point when immediate attention is necessary. Some inspections have shown that serious defects in both materials and labour would have caused failures had they not been detected. A variety of problems have been presented to design engineers in this way and corrective steps have been taken to improve the integrity of the product. Some of the problems remain for further study, particularly as they pertain to very large shaft systems. The findings for this type of application most certainly will be incorporated into the smaller sizes, thereby improving the whole spectrum of the product.

Failures of sealing systems invariably cause ingress of sea water or loss of oil. In some cases, the ingress of sea water is into the ship or it may be into the oil lubricating system. Loss of oil may be into the ship or into the sea. Most seal failures are mechanical in nature, though some show failures through chemical and corrosive means. In general, failure by fouling is quite rare, even though damage occurs to the propeller and rudder. Low temperature operation has little or no effect on the seals.

The principal elements of the seal are the members which actually form the barriers. The various supporting members, though important, are seldom involved in the failure of the seal. Wear, in the form of scores and grooves caused by inclusions of abrasive materials at the pressure contact, are a principal cause of leakage.

The excursions of the shaft operating within the confines of the bearing can promote wear. In some seals these excursions provide a means for the entrance of abrasive materials while in others, this displacement is absorbed within the resiliances of the seal elements. Most scoring and grooving occurs on the revolving elements of the seal. Materials are carefully selected to be wear and abrasion resistant as well as to be compatible with the mating stationary element. Hardness alone is not the criterion for materials selection. The sliding faces must withstand the normal operating environment including to some degree, suspended foreign materials. Deterioration of the material must not occur with normal operating service.

The sealing pressures which maintain the contact between the rotating and stationary elements must be of such magnitude that the elements never part. Thermal expansion of shaft systems affect the integrity of some seals and may be of sufficient magnitude to allow leakage to occur. Machining and assembly adjustment inaccuracies may also be sufficient to cause leakage.

Seal elements employing elastomers are prone to deterioration with age. Failures can occur through abnormally high operating temperatures at the sliding contacts. This reduces their resiliant qualities by hardening and cracking, thus causing eventual leakage. Some elastomers deteriorate in stages and release abrasive residue into the contact area, which further intensifies the rate of deterioration by rubbing friction, causing scoring of the elements and the generation of additional friction heat. Careful compounding of the elastomer can largely overcome this difficult problem.

Some seals are sensitive to the pressures exerted by the sea water and the counterbalancing effect produced by the lubricating oil reservoir pressures. The imbalance of the outboard and inboard pressures may cause distortion in the shape of the stationary element and the area of sliding contact. The additional contact area increases the frictional heat which is detrimental to the sealing material.

Some seal elastomers show lack of strength and excessive distortion under pressure and mechanical supports are used as a restraint to prevent the effect of excessive distortion. Reinforcement of the seal element is accomplished by the inclusion of fabrics or resilient metals in the elastomer during moulding. Improved elastomers are now available which have more resistance to distorting pressures.

Newly developed lubricating oil systems are available with pressure controlling devices which are regulated by the sea water pressure as the vessel's draught varies. Other systems are employed to control pressure by the use of multiple head tanks, located at optimum heights, which can be operated either automatically or manually to keep the pressure within good operating range.

Heat deterioration of the seal elements is not the only cause of seal failures. Some are caused by faulty assembly practices such as shown in Fig. 13. Here is evidence of elastomer failure due

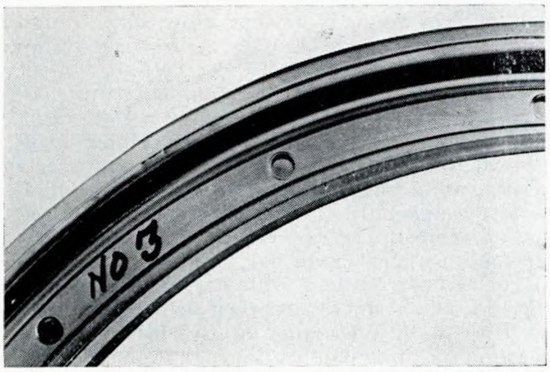

FIG. 13—*Failed seal ring—excessive tightening*

to excessive compression of the clamping medium. When the elastomer was flexed over some period of time, cracks and fractures have occurred; excessive tightening in other areas have produced a similar condition.

Heat deterioration of the elastomer is evident in Fig. 14. Here the elastomer has failed by wear and disintegration and pieces of the elastomer have broken out across the contact surface of the element. Fig. 15 shows a similar condition of failure.

Still other failures have occurred (in rare cases), of elastomer "delamination" effect, largely caused by a poorly moulded part. This appears to be a layer separation of the elastomer at the rupture.

Surface blisters have occurred on the elastomer and where probed were found to contain either air or fresh water. Fig. 16 gives the general appearance of such blisters. Sometimes these do not become evident until the element has been in actual service for some time. Proper compounding and moulding of the elastomer can largely control this condition.

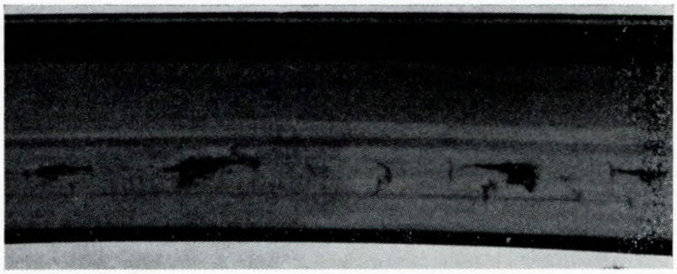

FIG. 14—*Failed seal ring—riding contact area*

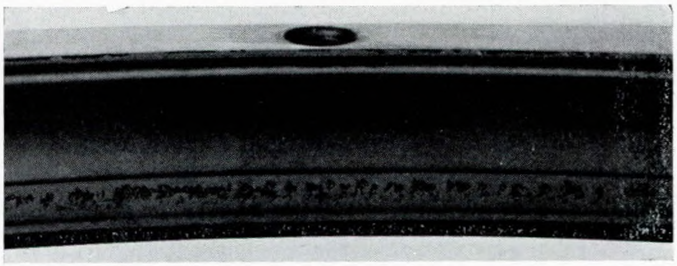

FIG. 15—*Failed seal ring—riding contact area*

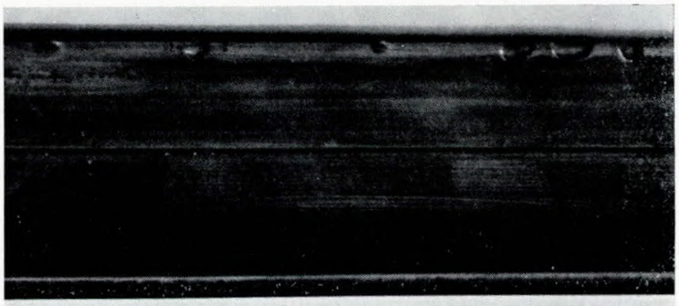

FIG. 16—*Seal ring imperfections—blistering*

Surface blemishes have been noted which lead to the question of the integrity of the element. They are generally small areas of surface imperfections and can be determined by visual inspection but no known failures have occurred through the use of elements which have shown these imperfections, though they appear to be suspicious. Diligent quality control is essential during the manufacture of all sealing elements.

In summary, stern tube bearing seal failures can be attributed to one or more of the following fundamental causes:

i) dirt and foreign material in the lubricating system;
ii) pressure differentials across the sealing medium;
iii) wear of the seal parts;
iv) deterioration of the seal at higher temperatures.

An ideal seal system would be one which prevents the ingress of sea water into the vessel, and/or oil loss from the bearing system. It would be a device which needs no maintenance or repair and has infinite life. Further, it would be a system which could be easily installed and thereafter forgotten. It is unfortunate however, that such a system cannot be built with the present materials and known techniques. It is, therefore, the responsibility of the scientist and engineer to develop systems which are as free of failures as possible and perform well under a wide and varying set of operating conditions. With the widening of marine horizons, the demand for equipment which will function as intended is becoming a challenge to all involved in the design and construction of modern vessels. The development of new techniques and designs to employ oil lubricated bearings and seal systems is evidence of progress being made to meet present day demands.

Without a doubt, there will continue to be failures in stern tube bearings and seal systems. Many will stem from causes discussed in this paper. Some can be prevented by advanced technology; some cannot because of the human element in the installation and operation of the systems. It is the hope of all concerned that these failures can be minimized by the use of modern materials, advanced technology and competent ship operators.

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Discussion

MR. A. ROSE, B.Sc., A.M.I.Mar.E., like the author, felt that manufacturing technique had considerable influence on the failure of the stern bush under the severe pounding and vibratory loads which it experienced. It was well established that unbonded babbitt would rapidly break up due to fatigue and it was easy to discover whether this was due to the original sin of the manufacturer in providing poor bond, or whether, in spite of a good bond, the vibratory loads had been too high. That was to say it was easy, provided the bearing could be inspected before complete failure had occurred, as unfortunately after that point most bearings looked alike. If the bearing could be examined in the state shown in Figs 8 and 9, then failure due to lack of bond would generally be shown by lack of tin in the fatigued areas, whereas if the original bond were good, the shearing of the babbitt from the backing would take place above the tinned layer, leaving the tinned surfaces. Sometimes, after fatigue failure, erosion of the cavities would take place, which could remove some of the tin, giving an impression of lack of bond. Generally, however, close examination of the smooth surfaces at the bottom damaged zone would usually reveal whether the tin had been originally present.

Metal manufacturers were producing fatigue-resistant tin-based babbitts, whose structures were modified and refined by the addition of alloying elements such as cadmium and silver to the basic tin, antimony and copper. Mr. Rose's own view was that whereas these elements would successfully increase the fatigue strength of a test sample of babbitt, their effect upon the finished article, the bearing, was small. Rather than add alloying elements to increase the fatigue resistance of the babbitt, the addition of wetting agents would help to increase the overall fatigue resistance of the bearing by ensuring better bonds.

These remarks applied to conventional babbitt material. So far the advantages of bearing materials used in other fields, notably the automobile industry, had not been made use of by marine engineers. It seemed to Mr. Rose that the stern bearing could be improved in this way, and Mr. Rose would be glad to hear the author's views regarding other materials.

The section on cooling rates and stress relieving was most interesting. Often, too little emphasis was placed upon the stress relieving of large bearings. One large Diesel manufacturer insisted that bearings should not be artificially cooled, thus giving them the chance to get some stress relieving. But most people insisted on high cooling rates. If very high rates were used, then the babbitt cooled very rapidly to a point well below its solidus and, with the difference between coefficients of thermal expansion of babbitt and steel, high interface stresses were set up. Mr. Rose had had experience of actual shearing of the interface during subsequent machining. Had the author's company experienced this problem, apparently due to too high a cooling rate? Mr. Rose agreed that theoretically the higher the cooling rate, the stronger the bond, but thought that very high cooling rates should be used with caution.

In the case of cast iron stern bushes, too high a cooling rate could result in the cracking of the casting, although this was a very minor problem in well controlled workshops. Most large stern bushes were at present cast on spinning machines and the main danger associated with slow cooling appeared to be segregation of the white metal rather than lack of bond. Whatever the cooling times it was surely the cleaning prior to tinning which had most effect on bond strength. It would be of considerable interest if the author could enlarge a little on his own company's methods of babbitting these large bushes.

So far Mr. Rose said he had assumed that the babbitt would be cast in a molten state. There seemed to be many advantages in using a metal spraying technique to line large stern bushes. This allowed the babbitt to be deposited, using the same machines as for rough boring. Hot shells did not have to be handled, and since only a small machining allowance had to be left, machining time and wastage were reduced. The bond strength of a sprayed babbitt was probably not as high as the best obtainable for cast babbitt but, on the other hand, the bond was uniform all over the bearing and there were no local areas of very poor bond to initiate failure. In the past, lamination of sprayed babbitt had presented some problems, but these had been overcome. Mr. Rose's own experience of fatigue-testing sprayed bearings confirmed his remarks about white metal in that those which tinned most easily gave the best results in this rig, and the sprayed bearings had been comparable with the very best of the cast ones. Although the tests on the bonds had shown them to be slightly weaker, they were uniform all over the bearing. He asked for the author's views on this point.

In a paper recently presented at the Institute, Emerson, Sinclair and Milne asked for a reduction in L/D ratio. In the discussion to that paper, Mr. Rose had presented some graphs which showed that the small L/D ratio normally acceptable in most systems would be unacceptable for stern gear. This was because at the low duty parameters at which the stern operated, the oil temperature rise was such as to reduce the viscosity beyond a critical point for a very short bearing. However, he agreed very much with the ratio 1·5/1 proposed, but would be very wary of dropping below it.

Mr. Rose asked the author to give further details about his suggestion of tilting pad bearings for stern gear. Mr. Rose's work so far had led him to the conclusion that the tilting pad was most unsuitable for stern bearings. When A. G. M. Michell invented the tilting pad journal bearing, he had assumed that, like his thrust bearing, it would have had a larger minimum film thickness than its plain counterparts. This did not prove to be the case. In the range of loads and speeds encountered on stern gear, the minimum film thickness was probably inadequate to accommodate the deflected shape of the shaft. Spherical pivots would allow some degree of self-alignment but there could still be sufficient shaft deflexion from end to end, or bending over a length of 45 to 60 in, to cause rubbing at one end, particularly at slow speed. The particular advantage of tilting pads, if any, would be in the ease of removal of the segments. In this case Mr. Rose suggested that more than five were needed to use this advantage fully. In one of the slides Mr. Koons had shown there seemed to be hardened pivots set into the pad. Mr. Rose asked the author to comment on their stresses and the degree of hammering they experienced. They seemed much too small and would hammer out.

Mr. Rose had outlined these disadvantages, he said, in the hope of provoking the author into giving more information than was in the paper—he pointed out that the author's company had spent considerable time investigating tilting pad stern gear.

To see the advantages of tilting pad bearings two such bearings should be used fairly close together. The after one should be the stern bearing of tilting pad design to hold the shaft steady, while forward of it a plain bearing should be mounted to take a considerable measure of load from the actual stern bearing. Alignment would be critical for two bearings so close to one another, and Mr. Rose suggested a hydraulic jacking system, followed by cast-resin chocking, for the forward bearing, and that a hydraulic jack be fitted between the lower-most tilting pads to facilitate both alignment and pad removal.

Were there not dangers of instability when using hydrostatic oil with a tilting pad? Rather than using a part hydrostatic part hydrodynamic action on a tilting pad, would it not have been preferable to have used an entirely hydrostatic system in a plain bush?

If hydrostatic bearings were used, the viscosity effect would be negligible and water could again be used as a lubricant, as there was a plentiful supply, and sealing problems would be minimal. Water-lubricated hydrostatic stern gear was a practical suggestion and sea water could be the lubricant, filtered to about 15 micron before admission. Pressures would depend upon bearing load, but 250 to 300 lb/in^2 should normally suffice. A simple forward seal and an aft restriction would be necessary, as the outward flow of filtered water would keep out the bulk of foreign matter.

Mr. T. W. Bunyan, B.Sc., M.I.Mar.E., said he had been particularly interested in the slide of the hydrostatic bearing shown at the meeting. From the aspect of physical dimensions, it had looked as if this would mean using a much larger stern frame boss to accommodate the tilting pad segments of the bearing, which appeared to have about the same radial thickness as the stern tube itself.

The author had underlined many of the factors which had led to trouble, few instances of which could be laid at the door of the designer; and Mr. Bunyan largely agreed with his comments. Given the same care and cleanliness one would give to setting up journal bearings in main propelling machinery, a modern stern bearing with Viton rubber oil-retaining seals could be expected to provide many years of trouble-free service. As for the problem of wear in the stainless steel liner this was a matter which was dealt with at the Special Surveys, and there was no great problem in resurfacing these. Under reasonably good conditions, it was possible to get out of a liner a service life corresponding to three Special Surveys.

Such troubles that did occur involved the loss of oil, but in Mr. Bunyan's experience this had very rarely led to grooving of the tailshaft. A temporary remedy for a heavy loss of oil was to top up the header tank with oil of say three times the normal viscosity, which would substantially reduce the leakage rate until repairs could be undertaken. Disastrous wiping of stern bearings was very rare indeed with degraphitized cast iron white metal lined bushes which did not exhibit the sinister side effects of the now obsolete white metal lined bronze stern bushes.

In the early days great changes in draught of very large tankers and bulk carriers produced problems with lip seals and called for cooling of the forward seal, and the use of Viton rubber labyrinths. Fortunately, with these large ships which employed powers over 30 000 hp, the rev/min was reduced to about 80 in order to achieve good propeller efficiencies, utilizing larger propeller diameters. He had not known of vibration problems either with the hammering of the stern bearings or in the hull itself at around 80 rev/min. This could not be said where propeller speeds exceeded 100 rev/min when serious hammering of the tail shaft in the stern bearing was often experienced producing the high frequency excitation of the after structure of the ship together with all the usual attendant problems. Quite heavy marking of the top of the stern bearing had confirmed that the problem was one of hammering. Mr. Bunyan was interested to know the author's own experiences of this problem. Maintenance costs and dry dockings for large ships were becoming astronomical. The availability of large dry docks at short notice was a problem which made the fear of sudden disasters a serious anxiety. Therefore a great deal of ingenuity was being exercised in achieving the maximum possible reliability of propelling machinery, coupled with the logical consideration of increasing the periods between dry dockings by making use of the latest advancements in underwater coatings of the hull to maintain ship performance.

The bogey of fatigue failure in tailshafts could well have been removed by eliminating the shaft key and keyway by resorting to the keyless bore propeller. Fatigue failure made a four-year inspection of the shaft a necessity. With modern underwater coatings, it was now quite common on large ships to operate for two years between dry dockings. After the guarantee docking of a new ship two two-year dockings were contemplated for large ships with a tailshaft survey at the end of the third dry docking i.e. after the first five years' service, and again at the end of a further three two-yearly dry dockings (after a further six years).

This could well be the pattern of the future. It would mean that trouble free service from large stern bearings and seals could be expected for periods up to six years.

Mr. S. C. W. Wilkinson said that his company first considered the development of a marine stern shaft seal range early in the 1950s and the decision to concentrate on radial face seals resulted from a considerable amount of work carried out on inflatable seals considered for emergency application at full power, and also as main seals on oil lubricated installations. He felt that the results of this investigation were very relevant to lip seal potential and felt that they might be of interest to the discussion because of the apparent absence of any theoretical research work on lip seal design.

Basically the inflatable seal comprised a 'U' shape inverted split tyre which surrounded the shaft and was loaded into contact with it by an internal pressure (Fig. 17). This enabled the sealing force 'F' to be controlled and the resultant contact stress 'S' whilst at the same time observing the leakage rate 'L'.

The first major facts that had come out of this work were those showing the effects of the loss of the reaction pressure due to leakage at the moment when complete sealing occurred. Thus for any given pressure the pattern of leakage relative to variations of 'F' could be illustrated by the graph (Fig. 18). From these diagrams it could be seen that to achieve satisfactory performance, one had to attempt to operate in the area indicated by the cross at the peak of each curve. This meant, however, that there was a serious risk that any fluctuation of 'F' could result in a complete loss of the reactionary pressure and immediate heavy overloading of the contact area, followed by high friction heat and possible damage.

Mr. Wilkinson went on to say that bearing in mind that

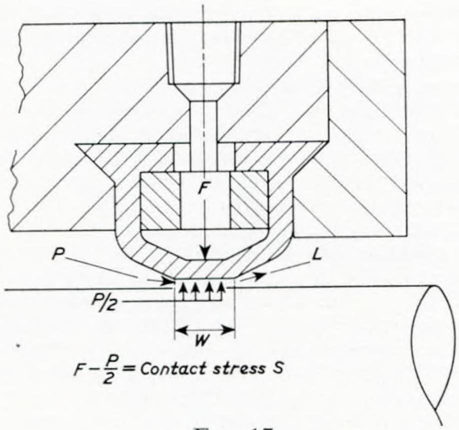


FIG. 17

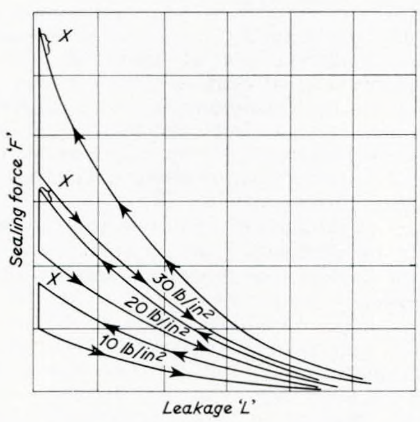


FIG. 18

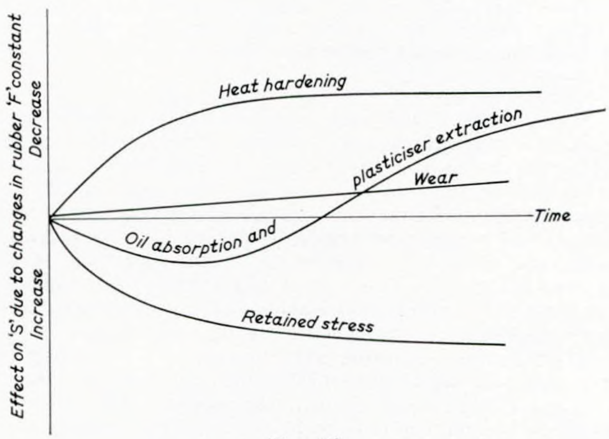


FIG. 19

in most conventional lip seal applications a constant spring load provided the force 'F', the problems of the design became very apparent. It was interesting to look at the factors which provided varying reactions to 'F' assuming it to be a constant. Fig. 19 illustrated the five major variants that had to be considered with elastic materials such as synthetic rubber and plastics:

a) Retained stress. The retained stress fell off fairly rapidly with all these materials—in the case of those with a poor characteristic of this sort, such as Viton, in a matter of minutes, and those with the best characteristics such as natural rubber in several days.

b) Wear. In a sealing application one had to consider wear, which was removing the material, making necessary further compression of the rubber hoop at the lip. This would be a very small factor having an effect over a very long period.

c) Heat hardening. In the case of some of the best materials for lip seals such as Nitrile rubber, the effects of heat were to bring about a massive increase in the hardness of the material. In the case of a seal with a constant 'F', a higher temperature was liable to be generated in the early stages of running, resulting in rapid hardening giving a fairly substantial increase in resistance to 'F' which eventually resulted in a drop of temperature and a settled-down running condition (unless hardening had occurred to such an extent that mechanical damage resulted).

d) Heat softening. Because of the mechanical damage resulting from excessive hardening due to heat, materials had been employed which softened due to heat and these would give an inverted curve as opposed to heat hardening, i.e. a fairly rapid fall off in resistance to 'F' in the early stages of running. But this in fact was an undesirable situation because it did not produce a self-compensating condition. This resulted in the rather disappointing performance of materials such as Viton.

e) Oil absorption and plasticizer extraction. These two effects could be considered together because one was a function of the other. They had the short term influence of softening in the early stages, perhaps for two or three weeks, and then a progressive hardening over several months. Associated with hardening would also be some degree of shrinkage, and in many cases it was this factor which brought about lip cracking in the longer term as opposed to simple heat hardening. Fig. 19 gave a general impression of the relationship of these curves and made it obvious that in the higher pressure applications it was extremely difficult to establish a set of operating conditions putting 'X' in the desirable position on the loading relative to leakage curve.

Viewed all together, these experimental results clearly explained why under low pressure conditions where 'F' could be low and therefore the reaction low and its loss of little consequence, satisfactory operation would be achieved in spite of the varying factors. When attempts were made to use this form of seal at higher pressures, experiments indicated that above 30 lb/in², under virtually no leakage conditions, there was a high incidence of failures in the experiments due in most cases to uncontrolled overheating occurring during the early stages of operation. But having survived this by carefully controlling 'F', it was also found that even after several months, a factor such as a change of ambient temperature could initiate the onset of a rapid breakdown condition. If this did not occur, there was the further problem that plasticizer extraction and resultant cracking would eventually lead to failure after a period usually in excess of a year.

Fig. 20 showed the results of another experiment carried out to establish the effects of the minimum possible oil film which it was hoped would give improved conditions over dry running. In an experiment set up to give conditions where rubber would run dry on metal at a settle-down temperature of about 50°C Mr. Wilkinson said the team had been surprised

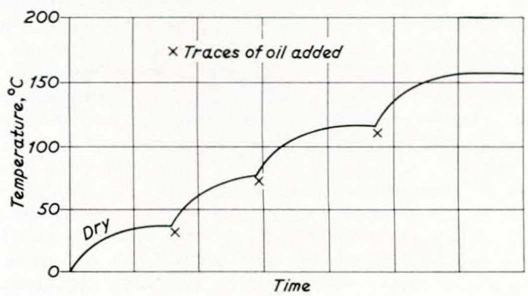


FIG. 20

to observe that the addition of traces of oil had resulted in an increase of running temperature in the order of 40 or 50°C which would not recover, and a subsequent addition of further small quantities of oil gave the surprising effect of a stepped build-up in temperature which would not recover until an appreciable degree of hardening in the case of Nitrile rubber had occurred. Undoubtedly this phenomena presented a further risk of unpredictable failure when attempting to use rubber as a seal under a virtually zero leakage condition.

This experience explained, Mr. Wilkinson, helped him and his colleagues to conclude that as ships increased in size a pressure barrier would be reached above which lip seals would prove to be unsatisfactory, and this justified the development of a range of very large face seals including 41 in diameter units on oil, units which had up to the present time been at sea for two years. Also designs were at the moment being prepared for 58 in diameter units for oil lubricated bearings operating at 60 lb/in^2. Experience in this field up to the present had established that a 20 in seal at 100 rev/min could be designed to operate satisfactorily at pressures ten times higher than those normally encountered in such applications, thus indicating the greatly improved safety factor made available in the case of surface ships as opposed to the lip seal with a maximum pressure limit not exceeding 50 lb/in^2 under similar conditions.

The author had referred to the problems of dirt penetration between the sealing members and Mr. Wilkinson emphasised that when correctly designed the face seal with its square contact edge was not nearly so susceptible to dirt penetration and damage as the lip seal with its angled entry to the position of contact. He and his colleagues had been able to prove this point in the case of seals at sea on both oil and water.

The radial design also offered the possibility of consistent known and predictable material characteristics which in turn allowed concise design criteria to be established once the problems of the operating requirement had been completely analysed. A full investigation of the operating conditions over a broad range of ships had taken them over ten years and it was only in the last two years that certain major aspects of the problem of applying face seals to these conditions had been unravelled; but he regretted that there was understandably insufficient time at this meeting to go into this work in depth.

In closing, he said that inevitably fully split face seals would be considerably more expensive than lip seals, but there was little doubt that their costs were more than justified by their very much greater margins of safety, predictable performance and the ease with which they could be serviced without moving the shaft.

MR. W. N. McILVEAN, A.M.I.Mar.E., said that the problems and limitations of white metal lined stern tubes provoked the question: "Why have white metal?" About three years ago an investigation had been started to find an alternative bearing material with the following properties in mind:

1) it should be capable of accepting high bearing loads, and so allowing a reduction in bearing length;
2) it should be compatable with water lubrication to reduce seal failure problems;
3) the frictional properties should be better than white metal to reduce wear caused by friction and low revolutions;
4) it should be no more expensive and if possible cheaper than white metal;
5) it should be easy to replace in the event of bearing trouble or complete failure.

This investigation had led to the choice of a resin impregnated asbestos composition produced in cylindrical form and subsequently pressed into the stern tube, oil lubrication being supplied in the conventional manner. So far the results had been most encouraging; in fact one ship converted to this type of stern tube some 2½ years ago had run for several months following a seal failure, the stern tube lubricant during the period being predominantly water. On inspection, no damage had been apparent to either shaft or tube. The L/D ratios of the plastic line stern tubes had been reduced from 2·5 to 1·5 in more recent conversions producing an increase in nominal bearing pressure to 8 kg/cm^2.

Referring to stern tube seals Mr. McIlvean said the objective was to keep the water out and the oil in. Unfortunately, a seal did not yet appear to exist which could do this reliably for say, five years, and the stern seal still had to be overhauled after two years. However, the techniques of lip seal bonding developed by the seal manufacturers had helped enormously in the overhaul of stern seals. The main problems affecting the reliability of the seal appeared to be those of material selection for the seal elements and liner. The factors affecting the seal elements could be assessed as:

1) the temperature at which the seal operated, due to the heat generated by friction—on the test rig which was run in conjunction with a shipyard, thermocouples embedded in the seal lip indicated temperatures of up to 140°C;
2) contaminants in the lubricating oil, which could be responsible for some of the liner grooving problems which occurred;
3) the differential pressure at which the seal operated caused deflexion on the seal lip—as the differential pressure was increased the seal lip could be flattened out against the liner thus increasing the area of contact, resulting in variations of sealing efficiency.

The effects of high temperature and oil on the various elastomers, in particular Nitrile were well known. Their tests exposing Nitrile rubber to various oils at different temperatures for varying periods of time had revealed a wealth of information. However the inevitable result had been that Nitrile rubber tended to harden and crack. It had been suggested earlier in the meeting that Viton rubber which was a high temperature elastomer, would solve some of these problems. This could be so, but could the author indicate any problems which he had had with the phenomena of "stick slip" when running Viton against a chrome steel surface?

MR. P. J. ADOLPH, M.I.Mar.E., said he was particularly interested in this paper insomuch as his company had a class of very deep draught tankers fitted with stern tube seals manufactured by the author's company. These seals were of the floating lip type.

Initially a number of problems had arisen with the stern tube seals on these vessels but these problems did not reflect on the seals themselves. These particular vessels originally had only had one header tank, and the problems arose both from sea water contamination of the lubricating oil at loaded draught and, in one case, leakage of lubricating oil into the sea at light draught. The difference in draught between loaded and ballast condition for these vessels was normally approximately 44 ft. During cargo discharge operations, ballasting was carried out concurrently with the discharge of the cargo, but even so it was sometimes possible that this 44 ft difference would be exceeded for a short period. Nevertheless this amply illustrated the problem of seal manufacturers and the solution adopted on the vessels in question had been to fit two header tanks at suitable levels.

The author's comments regarding cleanliness had been extremely pertinent and deserved strong support.

Mr. Adolph said he would like to read the comments on a lubricating oil report on a sample of oil from the stern tube on one of the tankers just referred to: "The oil as represented by the sample from the port stern tube contains 15·7 per cent salt water and considerable amounts of rust flakes, sand and other abrasive material." This had been on a nearly new ship. It was not surprising that this vessel had to be dry docked shortly afterwards. This had been prior to the fitting of two separate header tanks, and hopefully the problem of sea water contamination was now completely cured. However, this case amply illustrated the serious consequences of inadequate cleaning of stern tube lubrication systems, and the author's recommendation to treat these systems as one would a hydraulic system was very pertinent.

One point which the author had not touched on was the use of forced lubrication for stern tubes. It would appear that the stage had been reached on a number of vessels presently in service where existing conventional methods for lubricating stern tubes had possibly reached their limit. It would therefore appear that the emergence of forced lubricated stern tube bearings was not far off.

MR. A. HILL, B.Sc., M.I.Mar.E., said that it was not easy to overload a bearing, examples of which were shown in Figs 4 and 5, because usually the total load was known fairly accurately. Had misalignment caused overload in a local area? It was European practice to make stern tube bearings in one piece, the length being usually 2 to $2\frac{1}{2}$ times the diameter. Mr. Hill thought it was the practice of the author's company to make them in two sections and to thread these into the stern tube. In doing so, it was almost impossible to get the bore in line. What limit was allowed in putting the two sections together? Or had this practice now been abandoned?

The author had said that naval architects were advocating a single bush. Mr. Hill agreed with this choice because it offered practically the only hope of getting correct alignment of the aft bearing, virtually impossible with the conventional arrangement. The only value of the forward bearing was to assist in entering the tail shaft; as a bearing it was redundant. Indeed it was not unusual to increase the clearance to prevent it acting as a bearing.

Mr. Hill went on to say that it was better still to have a system which allowed alignment of the bearing after the ship was in the water and completed. It was possible to do this in something less than two hours. Service experience on this equipment gave temperature readings of 38°C aft and 34°C forward. When the loads were changed round, as a check the readings were 37°C and 35°C respectively.

Mr. Hill was interested in what was said regarding tilting pads. Mr. Koons had mentioned the use of hydrostatic forced oil lubrication. Was this for continuous load carrying, or was it a device used during starting, stopping or turning gear operations? In any event, it introduced more complication and cost, and risk of failure, and was better avoided for as long as possible. He did not know of any bearing equipment which needed this at the present time. The pumping losses would be very high, and this had to be considered.

A risk attendant on pivoted pads was that of pivot fatigue, or fretting under conditions of vibration. He wondered if the author's experience had shown any sign of this. It would not occur with normal stern conditions, but it could under conditions of heavy vibrations. About fifteen years ago there were several cases of severe fretting of the pivots on aft plummer bearings. The pivoted pads were removed and replaced by plain bearings after which there was no more trouble.

During the last five years it had seemed to Mr. Hill that seals were far more prone to failure than bearings. Did the author agree? The problems of aft and forward seals, and the pressures they had to carry, had been mentioned. The aft seal was generally in balance, apart from changes in ship loading conditions, and these could be met by having two header tanks. The forward seal carried full oil pressure, but probably had better conditions for cleanliness. Which seal had the greater incidence of failure?

The author mentioned the use of hard surface coatings and their value on forward seal liners. Would he state what type of coating was used and how it was applied? Was it metallic or ceramic?

MR. C. W. HERBERT, M.I.Mar.E., said that the main problem appeared to be that many conventional arrangements of stern gear made it difficult, if not impossible, to predict with certainty that all the essential ingredients for success would be present in the final installation.

Regarding the bearing, the dynamic bending moments to which a shaft was subjected could increase or decrease the static bending moment according to the particular service conditions applied. So, to get the best alignment of the bearing it was vital to ensure that the main line of deflexion was chosen at the beginning with absolute precision. If this was done in design, the shorter the stern bearing, the greater its capacity to accommodate angular deflexions of the shaft and still retain good load distribution. It seemed sensible to use the maximum diameter tail shaft that could be conveniently accommodated in the stern bushing in conjunction with a short bearing both to reduce shaft deflexion along the bearing length and to optimize the bearing proportions.

Perhaps one inhibiting factor was that the diameter was chosen by the classification rules for strength, and it was very difficult to persuade a shipbuilder to add half an inch to this in the interests of reliability, considering it from the bearing point of view.

As for seals, the author's company, as well as others, had appreciated the shortcomings of what had been standard for many years, particularly in the rate of seal failure. In the three ships of Glacier Herbert design after 10 000 hours running there did not appear to be any disadvantage in removing the floating bush. This was done to eliminate it as a heat source, which he believed had been one cause of problems on outboard seals. At the same time, arrangements had been made for full oil lubrication within the seal space, and it had been possible to check that there had been no undue temperature rise in the area.

Marine engineers had to strive for perfection. The thought being applied to seal and bearing development at the present time should take the profession some considerable way along this road. But one had also to appreciate the enormous consequential losses which were associated with failure, so it was logical to try to design a system which allowed the best elements available at any given time to be combined in such a manner as to promote the full servicing and inspection of these with the ship afloat.

Mr. Herbert's firm had done this on four ships with complete success making it possible to check the lay of the shaft with precision before trials and to make full service inspections with the ship afloat and loaded, with the same facility as one inspected bottom end bearings.

MR. G. CROMBIE, A.M.I.Mar.E., said that in his opinion a basic mistake had been made in designing oil lubricated white metal stern bearings on the same principles as sea water lubricated lignum vitae bearings, i.e. using a stern tube. The problems of the inaccessibility of bearing and tail shaft remained, and their inspection or survey required the removal of the propeller, tail shaft and some intermediate shafting and, additionally for controllable pitch propeller installations, the rudder and the muff coupling. Furthermore, it was difficult, if not impossible, to achieve contact between the tail shaft and the bearings over the complete length of the stern tube, and thus overloading of some parts of the stern tube bearings was unavoidable.

Split stern bearings said Mr. Crombie, of the type designed by his company enabled the bearing and tail shaft to be examined and the outboard seal wear to be measured with the vessel afloat and at loaded draught without removing either the propeller or the shafting. This work could be carried out in six hours. The stern tube was eliminated, and the stern bearing had a L/D of 2·0 and there was no forward stern tube bearing, a plummer block being fitted aft of the tail shaft forward flange. The author himself had advocated a similar design in the paper. Three such bearings said Mr. Crombie, had completed more than six years total service time and given trouble-free operation.

With regard to alignment problems mentioned by the author, it was considered that the single stern bearing would enable better alignment to be achieved than was possible with a stern tube with two bearings. However, an improved design had been produced, in which the complete stern bearing assembly could be angled to achieve full contact with the tail shaft over the length of the bearing. This operation could be carried out with the vessel afloat and at a draught representative of the most frequent operating condition. The highly loaded areas at the forward and aft ends of the stern tube were thus eliminated.

Some other advantages of split stern bearings said Mr. Crombie were:

a) tail shafts could be flanged at both ends—therefore both fixed bladed and c.p. propellers could be flange mounted, and the troublesome keyway eliminated;

b) the forward muff coupling was eliminated;

c) flange mounted, fixed bladed propellers were lighter and cheaper than similar taper mounted propellers;

d) flange mounted propellers could be replaced without removing the tail shaft;

e) tail shaft bending moments were reduced by virtue of the lighter propeller.

The three split stern bearings in operation were fitted with radial face type seals, as mentioned earlier by Mr. Wilkinson, the outboard seal being 1070 mm (42 in) diameter, with a sliding speed of 7·8 m/sec at service speed. The inboard seal was of smaller diameter and the seal face temperatures during operation were about 47°C for the outboard seal and 110°C for the inboard seal. It would be interesting to compare this diameter, speed and operating temperature with similar figures for the high temperature elastomers for the lip type seals mentioned by the author.

Could the author give some details about the metal alloy for eliminating grooving of the seal liner when using lip seals?

The statement that the use of a hard surface coating had not proved satisfactory on the outboard seal liner was most important. The grooving of the liner, which would occur when using lip seals, would necessitate dry docking the vessel and, unless a fully split seal was fitted, the propeller and tail shaft had to be removed to replace the seal liner. The split stern bearings already mentioned, although not subject to grooving of the liner, since radial face seals were used, were fitted with seals in which every component was split. The outboard seal wear could be measured from inside the vessel while afloat and it was possible therefore to minimize the risk of failure, since this measurement could be carried out regularly.

The author also mentioned that an extension of the time between surveys was desirable to owners besides which it was also important to reduce the time and labour required at each survey. When a split stern bearing was fitted, there was no need to remove either the propeller or the shafting at time of survey, since the bearing and tail shaft could be examined by removing the bearing cap, and the propeller mounting bolts could be withdrawn one at a time for examination. There was no propeller keyway or tail shaft taper and therefore there was no need to remove the propeller.

Mr. J. C. Hodge, A.M.I.Mar.E., supported Mr. McIlvean's views on finding a new type of material which would alleviate many of the problems encountered with conventional white metal stern bearings.

Mr. Hodge's company had been producing and developing water lubricated stern bearings for the past fourteen years for both merchant and naval ships. The naval side of the business was still very strong but asbestos phenolic compounds were too expensive and insufficiently superior technically to compete with lignum vitae. Rapid wear-down often occurred due to suspended sand and silt in estuarial waters.

The question was asked by an earlier speaker: Why not use this material in an oil filled stern tube? Theoretical calculations showed that above about 20 to 30 rev/min a typical merchant ship stern bearing would operate hydrodynamically on a mineral oil S.A.E. 30. The problem it was thought might lie in the inability of the reinforced plastic material to transfer the frictional heat generated under boundary lubrication conditions, resulting in serious overheating of the bearing.

Small-scale rig tests were made but they were of little use for a comparision with a large tanker stern bearing.

Fortunately, said Mr. Hodge, in 1966 an American operator asked his company to design and produce a reinforced plastic stern bearing for a 50 000 ton oil tanker.

This had been done and the sea trials were most successful. The highest oil temperature recorded under any condition had been no more than 15°C above seawater temperature. No additional oil cooling had been necessary other than the usual precaution of sailing with the aft peak tank flooded.

There had been certain problems during the first trip from Africa to the U.S.A. when the aft seal failed seriously and the chief engineer pumped seawater through the bearing after loosing all his oil.

Inspection had revealed that the bearings and shaft were still in perfect condition and after adjustment to the seal the system was refilled with oil and the ship sailed on again.

There were now three small tankers owned by Mr. McIlvean's company fitted with this type of bearing, two of which had had somewhat chequered careers. One had suffered from early seal failure when once again the bearing had proved its ability to run safely on seawater for about seven to ten days without the slightest sign of wear or damage to the shaft or bearing material.

On another occasion a thin steel wire had cut through the aft seal and wrapped itself around the end of the tailshaft like a clock spring. Again the bearing and shaft were found to be in excellent condition. This same ship was also mined by the Vietcong in the Far East and had undergone serious hull repairs in Singapore.

The second tanker had suffered serious damage to one of her four propeller blades which necessitated half cropping the opposite blade for balance. There was no spare propeller available so the ship had to sail in this condition for over three months. Vibration was quite severe, but again the resilient nature of the reinforced plastic bearing material absorbed the shocks and no damage was reported.

Would a white metal bearing have withstood such adverse conditions without showing signs of stress?

In the near future this type of bearing would be fitted to the largest class of vessel under construction today. The advantages of such a bearing were:

i) it was capable of operating in oil or seawater under emergency conditions;

ii) a choice of dockyard was allowed because under such an emergency the ship could proceed normally for a sufficient period of time without damaging the shaft or bearing;

iii) the bearing material was porous to oil and water and would therefore operate much better under start, stop and boundary conditions than would a white metal bearing;

iv) it was resilient and would not flow under shock loading conditions;

v) misalignment should be less of a problem than with a metallic bearing;

vi) the material was relatively soft so that foreign particles were absorbed and sank below the surface;

vii) the material couldn't seize onto the shaft in the same manner as a metal to metal system.

The material permitted smaller L/D ratios to be used e.g. 1·5/1 was employed on two ships now in service. Safety steps forward, although not normally employed, brought the L/D ratio up to 2·5/1.

Mr. D. W. R. Price said that he was reminded of a remark by a public speaker when asked to give a definition of the word "statistics". He said: "Statistics are a bit like a bikini. What is revealed is most interesting; what is concealed is vital." The paper stated that some manufacturers of lip type seals were offering elastomers with better thermal and mechanical characteristics, and that to date there were about 30 ships equipped with high temperature elastomers, and that the results of this experience had been quite encouraging. Mr. Price asked the author which elastomers gave this performance, and which ships were being referred to.

Correspondence

MR. C. A. SINCLAIR, M.I.Mar.E., said that development of large diameter shaft seals had contributed dramatically to the industry's ability to transmit larger powers in machinery aft installations. However, it was not surprising that some failures resulted in any major development and it was noted that this paper dealt in particular with failures. In machinery aft installations with water lubricated lignum vitae bearings it was frequently found that when once the wear exceeded $3\frac{1}{2}$ mm dramatic wear rates were encountered and frequently vessels were detained shortly after drydockings and refittings in order to effect necessary repairs.

There were many discussions and arguments in Japan in connexion with the clearances necessary in white metal bearings and as far he remembered the maker's recommendation was $1\frac{1}{4}$ thou. per inch of diameter where the local surveyors were of the opinion that $1\frac{3}{4}$ thou. was preferable. It would seem that some of the early failures could have been avoided had it been possible to reduce temperature rises in the early period of the bearings' operation and it seemed that trials with sister vessels indicated at least 5° lower temperature at the after bushes where larger clearances were used. It would be interesting to know what the maker's present recommendation was and what the comparative service results have been with different clearances. Mr. Sinclair understood that the maker's early objections to the larger clearances were that they would upset the seal. In this respect face seals seemed to have some advantage.

With regard the local bonding defects it had been noted that on occasions the loose metal had been removed and the edge washed away. Could the author testify to the subsequent satisfactory working of such bushes or was this always regarded only as the temporary expedient. It would be interesting to know whether any *in situ* patching of white metal had been found possible.

With regard to the fitting of the bushes into the stern tube Mr. Sinclair recalled that the use of molysulphide grease had been tried but the contamination in the system caused by this was found to be detrimental. Could the author state what was their present practice?

Turning to the seals, he said there was little mention in the paper of the radial cracking frequently encountered in seal failures and he understood that each seal was now made of two grades of rubber, the contact section being of specially developed high temperature rubber. Was this in fact what had been done and had there been any cases of radial cracks since this development?

The author had stated that development of suitable metals had made it possible to avoid ridging on the sleeves in way of rubbing area; however, it seemed to Mr. Sinclair that ridging still frequently occurred. Would the author agree that any ridging was highly undesirable because of possible axial movement in the shafting with consequent leakage? It had been said that eccentricity of the sleeve by even as much as 0·5 mm was sufficient to cause considerable difficulty with these shaft seals. It would be interesting therefore to know how accurate centring was accomplished.

The paper stated that a circulating pump was used for cooling the forward seal. Sometimes a special tank was arranged for maintaining the cavity between the second and third seal rings at the after end full of oil. Two tanks were being used—one for ballast and another for loaded condition or at times an automatic pressure arrangement was installed. All this added up to a degree of complication and in view of the high cost of drydocking for emergency repairs it was felt that simplicity should be the main objective. In any event it was clear that the crew should be given full operating instructions written in their mother tongue if accidents were to be avoided.

In connexion with the need for drydocking it was understood that other forms of seal had been developed which could be split, thus avoiding the need for propeller removal and possible drydocking. Had the author's company any new developments in this connexion? Finally would the author state it as absolutely necessary to renew the seals at the time of shaft inspection (usually four years)?

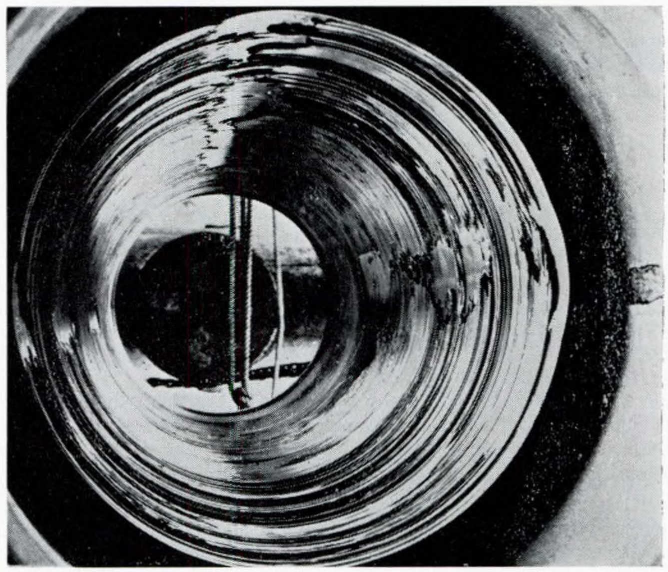


FIG. 21—*Seized white metal bush of stern tube*

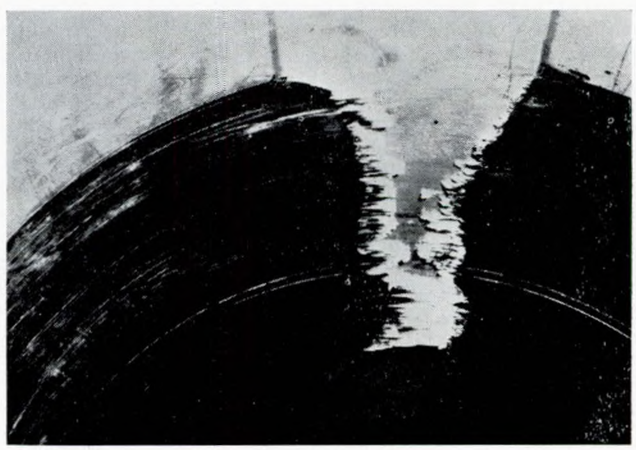


FIG. 22—*Upper part of the aft (white metal) bush found seized after sea trials*

MR. G. C. VOLCY, M.I.Mar.E., wrote that the failure of tail shaft bearings was causing great problems for shipowners, often due to the impossibility of being able to drydock at short notice. For this reason the reliability of stern gear was of the utmost importance.

Numerous examples had been shown in the figures of the paper, of failed stern bushes. Mr. Volcy agreed with the author that these failures were attributable not only to dirt in the oil, but also to problems of alignment—in particular to straight alignment and unpredictable misalignment of the shafting. From his association with Bureau Veritas Mr. Volcy had encountered analogous cases and two examples were shown in Figs. 21 and 22.

These failures, which were very similar to those shown in the paper, were due to the non-rational alignment of the shafting, which had been lined up in the normal way. Figs. 23 and 24 showed the results of shaft alignment calculations, and the values of reactions of shaft bearings. There was a negative reaction in way of the forward stern tube bush, and hence there would be a loss of contact between the lower half of the bush and the tailshaft. This loss of contact induced a lowering of the natural frequency of whirling vibration as shown in Fig. 25.

Due to the dynamic increase of the excitation forces attributable to the propeller, a whipping action occurred causing damage to the aft white metal bush as shown in Figs. 21 and 22. Such damage could be avoided in a very simple manner by adopting a logical curved alignment of the shafting.

Mr. Volcy agreed with the author on the question of the length of the aft bush of the stern tube. In his experience, too great a length of this bush was not only expensive and difficult to mount and draw, but it detracted from the satisfactory operation of the propulsion system. Experience had led Bureau Veritas to reduce the length of the aft bush to L/D=2 in the case of a shaft system which was aligned according to the curved alignment method. He also agreed that this length could be reduced a little further, but it would

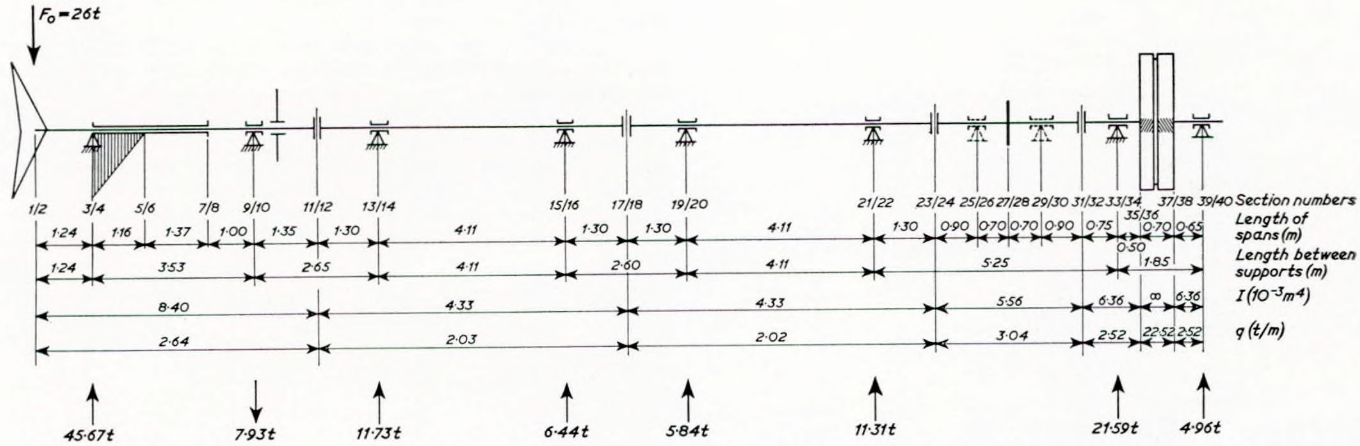
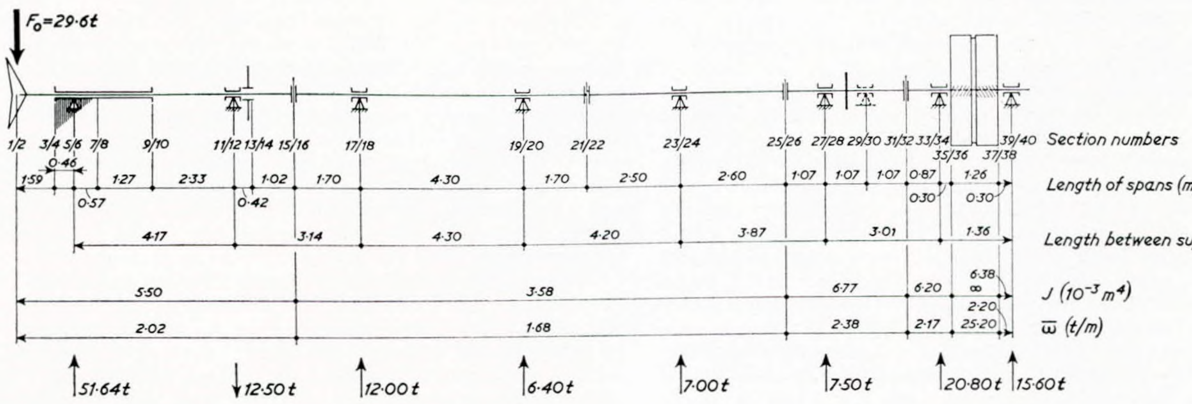


FIG. 23—*Distribution of static reactions of the line shaft supports of a 48 000 dwt tanker*



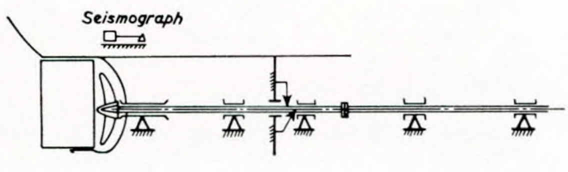


FIG. 24—*Distribution of static reactions of the line shaft supports of a 35 000 dwt tanker*

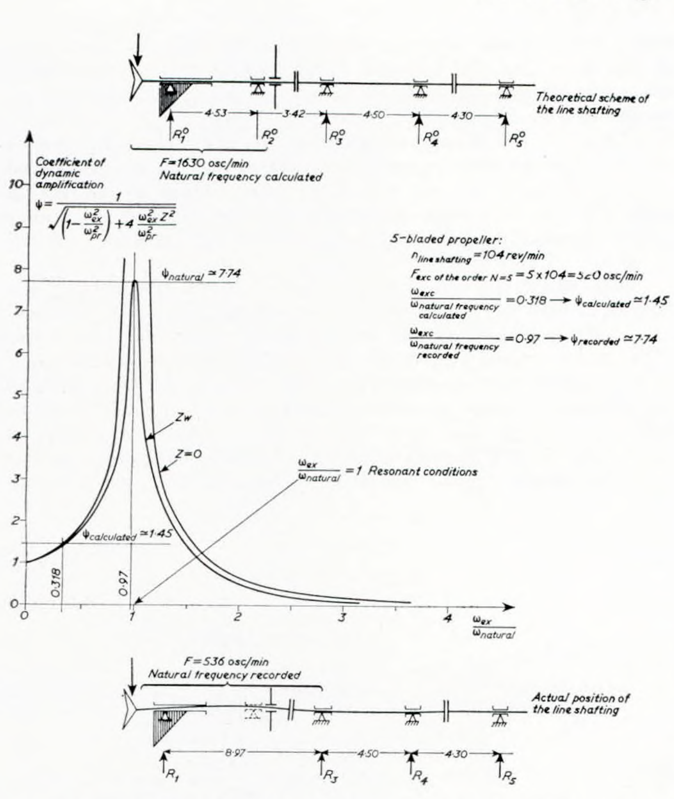


FIG. 25—*Effect of conditions of liasion between the tail shaft and its forward bush on the dynamic comportment of the line shafting*

be prudent to bear in mind the following precautions.

First, as illustrated in Fig. 3 the correct dynamic lubrication had been disturbed, the oil film broken and the white metal destroyed. In such a condition the remainder of the bearing length had to be sufficient to ensure that the tail shaft journal would still be supported by an efficient bearing surface despite the loss of the disturbed metal. Of course the unit pressure on the white metal was higher and heating was provoked, and in the case of a bush which was too short the white metal could melt out and endanger the tail shaft itself.

Secondly, prudence was required because of the couples and forces generated by the propeller in its working in the wake field. During lining-up these forces did not appear; on the other hand at full power they were producing forces on the tail shaft which were bending it in such a way as to force it into contact with the aft bush so that a longer length was in contact than during the alignment. For this reason, the bush should be of sufficient length to ensure that the correct hydro-dynamic lubrication conditions were not disturbed.

More details concerning the question of rational alignment of the tail shaft could be found in the paper, "Forced Vibrations of the Hull and Rational Alignment of the Tail Shaft".[1]

He came to the conclusion that the presence of the stern tube did not present any advantage; on the contrary it led to difficulties and he would go as far as to say that the presence of the stern tube could be considered as a source of additional troubles for the stern gear. It was advisable to omit it or shorten its length to the length of the aft bush. This shortening was a function of the accessiblity of the shaft sealing devices. Omission of the stern tube was mostly recommended for big tankers, bulk carriers or container ships.

1. Volcy, G. C., 1967, *Nouveautés Techniques Maritime.*

Author's Reply

Mr. Koons said he was interested to learn of new bearing materials. His company had concentrated their efforts on white metals, since they believed these had proven to be quite successful for stern tube bearing applications. They had carefully studied the effects of additional alloying elements and agreed with Mr. Rose that their advantages were small and their use involved greater risks during the manufacturing process.

The stress relieving operation of the babbitting process was important for producing good quality bearings. Slow cooling (in air) would lead to weak bonds and fast cooling without stress relieving to internal stresses and possible bond failure. Some fast cooling temperature limits had been established and when these were reached fast cooling was discontinued and the bearing allowed to cool slowly to ambient. These limits would vary with the white metal alloy, since they were largely dependent on the white metal reaching a solid state. Since fast cooling always progressed from the outer diameter inwards towards the molten white metal, it had been found that precooling was required before white metal pouring on thick walled bearings. This technique would help to reduce segregation of the alloying elements and present a more uniform white metal casting. Shells exceeding four inches in wall thickness would require precooling.

It was obvious that a good bond could only be produced with a well cleaned and tinned shell. Some shell contaminants were difficult to remove however and special processing was necessary to secure good tinning.

To secure a good white metal casting, the temperatures of the tinned shell and molten white metal were important as well as fast cooling and stress relieving. The proper combinations of temperatures and cooling cycles would produce optimum results.

Mr. Koon's company had had limited experiences with white metal spraying. The author himself did not believe that this method was as good as the molten metal technique because a dense homogenious metal could not be produced and a metal with a lower unit bearing capacity would result.

Replying to Mr. Bunyan, bearing to journal clearances of 1·5 mm on large tail shafts were not uncommon or excessive. In fact, some classification societies preferred larger clearances. In general, with well-balanced wheel and shaft systems and stiff tailshafts, excessive pounding vibration had rarely been seen. Severe pounding of the tail shaft in the bearing at speeds lower than 100 rev/min had been experienced when the system was dynamically unbalanced, and the magnitude of the vibration was amplified by the resonance reaction of the hull at certain speeds. Of course, said Mr. Koons, this severe pounding was abnormal and called for prompt corrective action.

The author had not experienced marking at the top of the stern bearing referred to by Mr. Bunyan but he suspected that this pattern was traceable to the number of blades in the wheel and the pressure points of the rudder horn while the shaft was turning at normal cruising speed. It could also be more severe while operating in a ballast condition and solid water was not available to all blades.

The parameters used by Mr. McIlvean's company, said the author, were most admirable and utopian. The company were to be congratulated for their advanced thinking and their willingness to pioneer.

Although a seal with an extended overhaul cycle of five years did not appear to be currently available, there were reports of results approaching this, involving little if any operational maintenance. And with the use of improved elastomers and wear part coatings, such a goal for all applications would become a reality in the near future. The replacement of seal elements on a shorter term basis had not to be confused with true seal failures. Many replacements were made as a convenience during periods of drydocking for other hull work and were considered to assure the integrity of the seals and an extended life. Such practice undoubtedly would be continued regardless of satisfactory performance of improved elements.

Fluroelastomers applied to lip seals promised extended periods of performance. The characteristic "stick-slip" frequency was somewhat lower than that found in conventional Nitrile lip seals. This dynamic response did not appear to be harmful until operating temperatures well below the freezing point were encountered. At these low temperatures, the response was so slow that leakage could possibly occur. This effect would not alter much by a change of the riding surface material to one with the same degree of smoothness.

Replying to Mr. Adolph the author said it was obvious that the total application of seals and bearings had to be carefully studied and developed so that the limits of satisfactory performance were not exceeded. Designs were available, which, when applied to the largest of vessels now in service or contemplated, would give satisfactory performances.

Force feed or hydrostatic bearings had a definite place in stern tube manufacture. With ever-increasing shaft loadings and shaft diameters, and with attendant misalignment caused by hull displacements, compensating means were required. Hydrostatic lift was desirable for these high loads and might become mandatory for very low speeds particularly under turning gear operation.

In reply to Mr. Hill, Mr. Koons said that in his own company the aft bushes were currently being made in two pieces and carefully babbitted and dowled together to form one full length aft bearing as Mr. Hill had described. The axial centrelines of the two aft bushes coincided with a maximum allowable variance of 0·003 in. There was no difficulty in installing the two aft bushes since the stern tube was finish bored for both bushes at one machine setting, thus producing a true centreline for the bushes. Each of the two bushes were machined on the o.d. and referenced from the finished bore by the installing yard. There had been no difficulties encountered with this system in erection or operation; in fact, yards frequently commented on the greater ease of installation of the two short bushes over the long one.

Lip seal failures were more likely to occur on the inboard or forward seal than on the outboard seal, in general for the reasons cited by Mr. Hill. Of course, inspection and replacement of the former was much simpler than of the latter. In the case of the ceramic coatings on the inboard seal, a marked reduction in frictional heat generation was recorded since the coefficient of friction was lower. This gave a cooler seal and lengthened the life of the elastomers.

Mr. Crombie's description of a bearing system employing face seals was a departure from the widely accepted design developed for rigid full bearings, all the more striking since his design could only be incorporated on new construction with reasonable costs. The advantages listed by Mr. Crombie of such a system were accurate provided all other factors were optimized and there was a freedom of design not normally accepted.

The author said he was interested to hear about performance temperatures on a 1070 mm aft face seal system but comparable data was not available for lip seal installations of this size. It was obvious that temperatures on the smaller inboard seal would materially reduce the life expectancy of a Nitrile lip seal.

The alloy employed for aft liners had to be selected to be compatible with the specific elastomer compounded and with the operational environment. Considerable research was required to balance the metal alloy with the elastomer in order to reduce the tendency of grooving to a minimum.

Mr. Price's inquiry referred to the use of elastomers with better thermal and mechanical characteristics than those found in conventional Nitriles. At least two manufacturers of lip type stern tube seals were offering fluoroelastomers. Their principal advantage was better heat stability, a factor which would substantially reduce deterioration at the lip contact. In most cases, this elastomer was used on seal sizes of 800 mm and larger, or on systems where the propeller shaft speed caused excessive frictional heat.

Regarding the journal to bearing bore clearances referred to by Mr. Sinclair, Mr. Koons said it had been the general practice to decrease the unit allowance as the journal diameter was increased. Clearances of 0·0018 in per inch of diameter for small journals from 14 in to 24 in, and 0·0013 in per inch for diameters from 26 in to 40 in were not uncommon. There had been no other reports on temperature reductions with larger internal clearances.

The removal of loose babbitt and washing away at the edge had been permitted only as a temporary measure to reduce the possibilities of lube oil contamination with loose babbitt. Patching of the white metal could be done provided the bushing was in the stern tube. In some cases, patching was impossible due to the location of the defective area, and the bush had to be removed for this repair. It was always better if possible to replace the defective bush with one with sound babbitt.

Molydisulphide grease had been used to help press the bushes into the stern tube and no unfavorable or questionable results had been noted. Any possible contamination after physical cleaning of the stern tube and bushing would be minimal and of little consequence.

Most lip seal rings were made from a single elastomer throughout. Some producers of lip seals, however, did offer seal rings of two elastomers—one for its ability to withstand frictional heat, and the other for its superior mechanical properties. The two cured elastomers were physically bonded together and treated thereafter as a single seal ring. There has been no radial cracking noted in the two elastomer seal rings as has been found in some of the single elastomer Nitrile seal rings.

Any ridging or grooving of the rubbing area was undesirable. Although the seal rings eventually seated themselves radially in the wear grooves, axial movement of the shafting system would tend to displace the riding track and could cause some degree of leakage. The use of sleeve alloys and coatings selected to minimize this condition was highly desirable, wrote the author.

It was important that the riding sleeves be mounted as concentrically as possible to the tailshaft and propeller. This could be controlled by accurate fitting into the propeller of the aft sleeve, and by careful positioning and indicating of the forward sleeve. Eccentricity of 0·5 mm would lead to leakage from both the aft and forward seal systems.

Simplicity in any lubricating system was desirable. But a system which was automatic and reliable was preferable to a manually operated system. In both systems the operators had to be fully conversant with operating principles and techniques.

There was a tendency among some engineers to shorten the aft bushes beyond conventional practice. Mr. Volcy had explained the limits of such developments and warned of the dangers of exceeding current minimum lengths, and his paper "Forced Vibrations of the Hull and Rational Alignment of Tailshaft", had emphasized the need for careful consideration of alignment to promote an optimum operating system.

Related Abstracts

Wear of Lignum Vitae Bearings of Propeller Shafts

This paper treats a practical example of previous studies concerned with the deflexions of ships' propeller shafts. The author gives an explanation of the rapid wear of lignum vitae bearings and shows how wear can be avoided by appropriate positioning of shaft supports and by an alignment which allows sufficient initial loading of the outermost tailshaft bearing.—*Paper by J. Lachassagne presented at the 1971 meeting of the Association Technique Maritime et Aéronautique (in French).*

Propulsion Machinery for Contra-rotating Propeller Systems

Contra-rotating (c.r.) propellers offer one attractive solution to meeting the needs for higher power and more efficient ship propulsion. A planetary second-reduction gear system is proposed to provide the c.r. energy to a pair of propellers from a conventional turbine power source. The concentric line shafts, bearings, and lubrication system are discussed as vital components to a c.r. system. Some favourable full-scale operating experience has been obtained on all components except a high-power second-reduction planetary gear operating in the c.r. mode. The use of a c.r. propeller system should have a significant economic impact on the powering of ships.—*Steele, T. W., Marine Technology, October 1970, Vol. 7, pp. 425–432.*

Working Life of a Seal

The life of an oil seal is shown to be directly proportional to the rate at which it can absorb the effects of friction horsepower and inversely proportional to the rate at which it develops friction horsepower.

The life equation,

$$t = BD/M\omega^2 \qquad \text{Eq. [3]}$$

can be used to compute seal life in varying shaft sizes and speeds if the seal life constant B is known for a specific fluid, fluid temperature, eccentricity and shaft finish.

The oil seal life factor B is shown to be equal to the total work done in the life-time of the seal times the angular shaft speed at which the work was done per inch of shaft diameter.

The self-limiting physical laws of work and energy apply to all friction devices. Thus, it is probable that the life equation can be employed in other fields involving rotating mechanisms.—*Brink, R. V., Lubrication Engineers, Journal of the American Society of Lubrication Engineers, October 1970, Vol. 26, pp. 375–380.*

Vibrations in Complex Shaft Systems

The increasing use of geared medium-speed engines in marine propulsion installations, together with the trend towards driving important auxiliaries (e.g. generators, cargo pumps etc.) via power take-offs from the gearing, leads to quite complicated multi-branched shaft systems, incorporating numerous elastic couplings, clutches, etc. The author discusses, for such systems, the three main types of shaft vibration, viz: whirling (rotating bending); axial; and torsional. The first two are considered only briefly, as the vibration modes which occur in practice are analogous to those encountered in low-speed Diesel or in turbine machinery. Whirling can result in rapid wear of water-lubricated stern-tube bearings or even shaft fracture. Axial vibration (usually propeller-excited can overload the gears if the thrust block (commonly built into the gearbox) is not sufficiently rigid. The computer programs used by Det norske Veritas for calculating natural frequencies, modes, amplitudes, etc. for whirling and axial vibrations are noted.—*Larsen, O., Veritas, October 1970, Vol. 16, pp. 8–11; (in Norwegian); Journal of Abstracts, BSRA, April 1971, Vol. 26, Abstract No. 30,827.*

Maximum Temperature for Bearings Under Steady Load

The limiting metal temperature in large babbitt bearings was observed to range from 268° to 390°F. Surface creep was encountered at this failure temperature with the local pressure in the oil film matching the yield stress for the bearing material. Expressions are developed for calculating the maximum temperature in either journal or tapered land bearings. These adiabatic solutions permit evaluating the corresponding maximum operating loads and speeds possible with various bearing geometries in either the laminar or turbulent regimes.—*Booser, E. R., Ryan, F. D. and Linkinhoker, C. L., Lubrication Engineering, July 1970, Vol. 26, pp. 226–235.*

Unified Regulations for the Design of Propeller Shafting

A thorough analysis of the propeller shafting of large ships (both Soviet and non-Soviet) has shown the reliability of this shafting to be affected by a large number of physically dissimilar factors, which can be divided provisionally into two groups: those associated with the design and production of propeller shafting, and those associated with the characteristics of the propeller and the dynamic characteristics of the shaft. These dynamic characteristics were not considered in previous U.S.S.R. Register Regulations, only static loading being allowed for.

Having become operative as of January 1, 1969, the new Regulations contain recommendations on the composition of propeller shafting; the general arrangement of the propeller shaft and its elements; the design of stern tubes, stern-bearings, stern glands, shafts, and their parts; shaft-turning arrangements; the choice of plummer and thrust blocks, bulkhead glands, brakes, etc.; and the calculation of propeller shafting. They apply to two provisional groups of propeller shafting: with a propeller-shaft diameter of up to 400 mm (group 1) and over 400 mm (group 2).

The Regulations place appreciable emphasis on the improvement of the watertight integrity of propeller-shafting elements. The sealing of the propeller cone on its forward and after ends is discussed in some detail from the design of the sealing assembly to the method of locking fastenings. —*Zelikson, I. L. and Sverdlov, N. L. Sudostroenie, December 1970, pp. 20–22, (in Russian); Journal of Abstracts, BSRA, May 1971, Vol. 26, Abstract No. 30,891.*

Experimental Determination of the Static and Dynamic Properties of Journal Bearings for High-speed Shafts

The investigation concerns the influence of the spring and damping properties of the oil-film of journal bearings on the vibration and stability behaviour of high-speed shafts. Both the experimentally determined static characteristic curves and the spring and damping coefficients are set up in a suitable form for the numerical use. Furthermore, the differences between experimentally and theoretically determined dynamic properties of bearings and the influences of the length to diameter ratio are discussed. A comparison is made of the vibration and stability behaviour using the example of a symmetrical one-mass rotor.—*Glienicke, J., UDI-Z, Vol. 112 (1970), No. 22, p. 1496 (in German).*

Contra-rotating Marine Propeller Drive (Patent Specification)

This invention concerns improvements relating to contra-rotating marine propeller drives. It seeks in particular to overcome the problem, which arises with contra-rotating marine propeller drives, of dealing with bearing loads on two concentric shafts of large diameter.

Fig. 1 shows, simply in outline, the aft portion of a ship with outboard openings (1 and 2) for the rudder (3) and forward and aft propellers (4 and 5) respectively, the rudder post being located at (6). The propellers (4 and 5) are secured by nuts (7) on respective cones (8, 9) on outer and inner shafts (10, 11) which extend forward to a gear box assembly (12) located in an inboard space (13) of the ship.

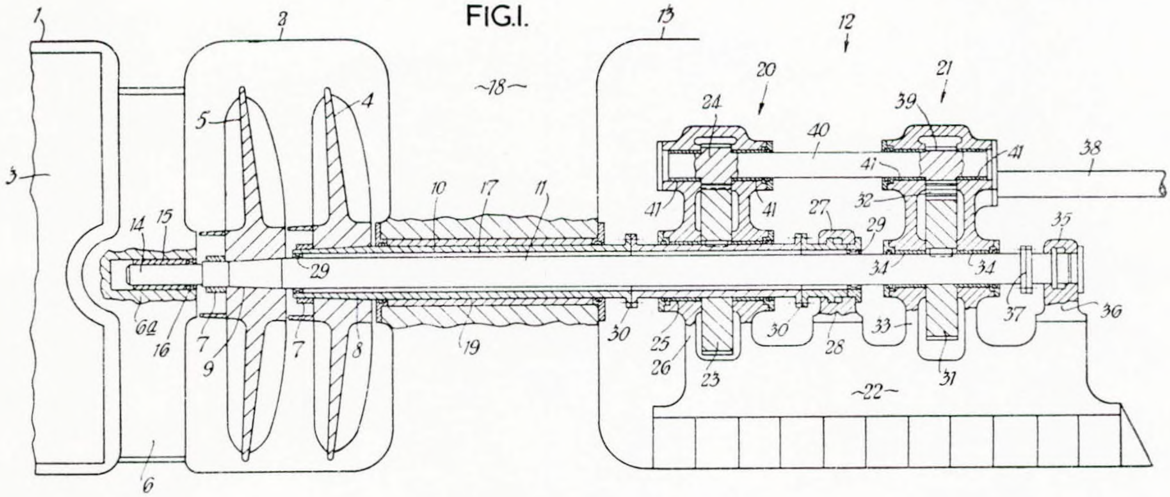
The outboard end (14) of the inner, solid, shaft (11) is supported in a bearing (15) carried in the rudder-post at (6). The bearing (15) is housed in an annular enlargement (6a) formed on or provided directly in a straight vertical rudder post or in an approximately V-shaped rudder post. Such enlargement may be faired at the aft end. Alternatively the bearing (15) may be supported from the stern-frame structure adjacent to the rudder, for example by a bracket or brackets. Suitably, the bearing (15) is an "underwater" white-metal bearing, but it could be a bearing lubricated through passages in the post. A gland (16) is provided at the forward end of the bearing. The tail end of the shaft (11) is enclosed.

Closely forward of the forward propeller (4), the outer, hollow, shaft (10), through which the inner shaft (11) passes with clearance (17), is supported from the stern-frame structure (18) in a stern-tube or tail-shaft bearing (19), completely independently of the inner shaft. Suitably the bearing (19) is a close-tolerance white-metal bearing having a length of the order of four times the external diameter of the shaft (10), but it could be a lubricated bearing.

Immediately forward of the bearing (19), the concentric shafts enter the gear-box assembly (12), which comprises two sections (20, 21) on a common foundation (22). The outer shaft (10) extends only to the forward side of the aft section (20) which encloses a gear-wheel (23) fast on the said shaft and a pinion (24) located above the said wheel and meshing therewith. There is no bearing between the shafts (10 and 11), but only a journal bearing or, as shown, bearings (25) between the outer shaft (10) and a pedestal (26) of the assembly (12) and a thrust-block bearing (27) supported by a pedestal (28) of the said assembly. The clearance (17) between the shafts (10, 11) is sealed at its forward and aft ends by glands (29). For ease of assembly and disassembly, the part of the outer shaft within the assembly (12) is subdivided into sections with flanged couplings at (30).

The inner shaft (11) extends into the forward section (21) which encloses a wheel (31) fast on the shaft and a pinion (32) located above and meshing with the said wheel. The shaft (11) is supported from a pedestal (33) of the assembly (12) by a journal bearing forward of the wheel (31) or, as shown, by bearings (34) forward and aft of the wheel. A thrust-block

FIG.I.



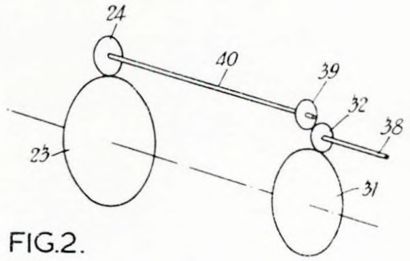


FIG.2.

bearing (35) for the shaft (11) is supported by pedestal (36) on the forward side of the section (21). The shaft (11) is also subdivided, with a coupling at (37).

The wheels (23 and 31) on the shafts (10 and 11) are driven from a single common power input shaft (38) extending aft from the power unit (not shown) of the ship and terminating at the assembly section (21). The pinion (32) is fast on the shaft (38) and directly drives the wheel (31) on the shaft (11). It also drives a pinion (39) fast on a transfer shaft (40) which extends aft to the assembly section (20) and on which the pinion (24), meshing with the wheel (23) on the shaft (10), is also fast. The shaft (40) is supported by bearings (41) in the pedestals (26, 33), as is also the aft of the shaft (38). If required, the shaft (40) may be sub-divided by a coupling at its mid-length. The shafts (10 and 11), with their respective propellers (4 and 5) are thus driven with contra-rotation. The relative speeds at which the propellers (4 and 5) are driven is determined by the ratios selected in the several parts of the gear train. With the arrangement illustrated, the propellers will rotate at the same speed.

Pinions driving the wheels (23, 31) could alternatively be driven through separate shafts extending aft from the power unit or units.

By way of example, with a turbine power unit whose high-pressure and low-pressure sides drive a single output shaft through an epicyclic or spur gearing giving a 12 to 1 reduction from, say, 6000 to 500 rev/min, the above-described gearing in the gear-box assembly (12) may provide further equal reductions of 5 to 1, say to 120 rev/min of the contra-rotating propellers (4 and 5).—*British Patent No. 1 221 201 issued to Stone Manganese Marine Ltd. Complete Specification published 3 February, 1971.*