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STERN BEARINGS—IN SERVICE MEASUREMENTS OF TAILSHAFT ATTITUDE, BEARING CLEARANCE, AND OTHER PARAMETERS

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Stern bearings and oil seals have been a source of considerable problems to shipowners during the past decade. Many of these problems are believed to be due to the inherent disadvantages of the conventional stern tube. It is also believed that not enough work has been done to establish the operating conditions present in a stern bearing, perhaps because of the difficulty of carrying out tests on a conventional stern tube. Tests carried out on the split stern bearing of a 280 000 dwt VLCC are described and discussed in this paper. The purpose of the tests was to establish the tailshaft attitude within the bearing, the minimum bearing clearance, the axial movement of the shafting, the lubricating oil pressure variation at the outboard oil seal and various stresses in the bearing assembly and associated hull plating. The measurements cover both ballast and full load operation under a variety of main engine speeds and weather conditions and were obtained during normal commercial operation without any special manoeuvres being carried out for test purposes. The main findings of the tests were the unexpected effects of hull deflexions, the likelihood of achieving better shaft alignment by carrying out the alignment procedure at ballast draught and the fact that the tailshaft frequently runs in the oil ways normally cut into stern bearings.

INTRODUCTION

Stern bearings and oil seals have been a source of considerable operating problems during the last ten years. Table I indicates the frequency of failures experienced by outboard oil seals and oil lubricated, whitemetal stern bearings. The table has been compiled from data recently obtained from a leading classification society and refers to vessels built to class between 1965 and 1973 inclusive. They are, therefore, all modern vessels. It can be noted that each type of vessel has a significant number of reported defects, although bulk carriers and large tankers show a higher incidence of failure than other types of vessels. It can also be seen that oil seal defects are more frequent than stern bearing defects for all types of vessels. Since the table shows only the reported defects it is possible that the actual number is considerably higher.

A recent paper [1] tabulates the items affecting docking surveys for 101 ships during the period January 1965 to November 1972. Examination of this data reveals the fact that 28 out of a total of 110 items, i.e. 26 per cent, were due to stern bearing and oil seal defects. In fact, the number of stern bearing and oil seal defects was nearly equal to the total number of hull defects.

It is clear, therefore, that there is a need for improved reliability of stern bearings and oil seals.

TABLE I—INCIDENCE OF FAILURE OF STERN BEARINGS AND OIL SEALS

Ship Type	No. of Ships	Average Operating Time-Years Per Ship	Number of Reported Defects: Outboard Oil seals	Number of Reported Defects: Oil Lubricated Stern Bearings
Dry Cargo	800	5·2	118	58
Bulk Carrier	329	5·2	86	24
Tanker (1)	219	5·4	25	14
Tanker (2)	100	7·4	47	27
Tanker (3)	128	4·3	30	25

Tanker (1)—Below 30 000 dwt
Tanker (2)—30 000 to 100 000 dwt
Tanker (3)—Over 100 000 dwt

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The increase in the frequency of stern bearing and oil seal problems has coincided with the advent of large and/or high powered vessels. Various theories have been proposed to account for this increase and it seems probable that one of the main reasons is the reduction in the ratio of hull stiffness to shaft stiffness. For a given size of ship, shaft stiffness has increased very rapidly due to the much higher engine powers now common while hull stiffness has remained constant or even reduced. The position is further aggravated by the fact that most vessels are now built with engines aft and there is very little flexibility in such short shafting lengths. The propellers and shafting are also much heavier because of the higher engine powers and high, local bearing loading is thus more likely if any mis-alignment is present.

One of the prime reasons inhibiting proper analysis of stern bearing problems is the lack of knowledge of the actual operating conditions present within the bearing. The main reason for this lack of knowledge may be the difficulty and expense of fitting the necessary instrumentation on a conventional stern tube bearing. With the split, withdrawable stern bearing, however, the problem is eased since it is a simple matter to fit the instrumentation without even drydocking the vessel.

It is considered that accurate shaft alignment is one of the most important requirements for increased stern bearing and oil seal reliability, especially on large and/or high powered vessels. It has been stated by some research work that even a properly aligned whitemetal lined stern bearing will, under certain loading conditions, be operating under boundary lubrication conditions which could lead to premature failure. The main purpose of the test programme, therefore, was to establish exactly the tailshaft position relative to the stern bearing for various operating conditions.

Another item which can be a source of problems for some types of stern bearings and oil seals is the axial movement of the tailshaft. This has two components, one relatively slow due to thermal expansion and contraction and a second, transient, vibratory movement due to variations in propeller thrust. Axial movement is of little consequence to whitemetal or other types of journal bearings since the shaft merely slides through the bearing. For roller type bearings, however, this movement may cause problems if it cannot be accommodated within the bearing itself.

Axial shaft movement may also affect the efficiency of oil seals. Lip type seals run on a liner attached to the shaft and tend to bed themselves in. Any axial movement will cause the seal to run in a different position on the liner and

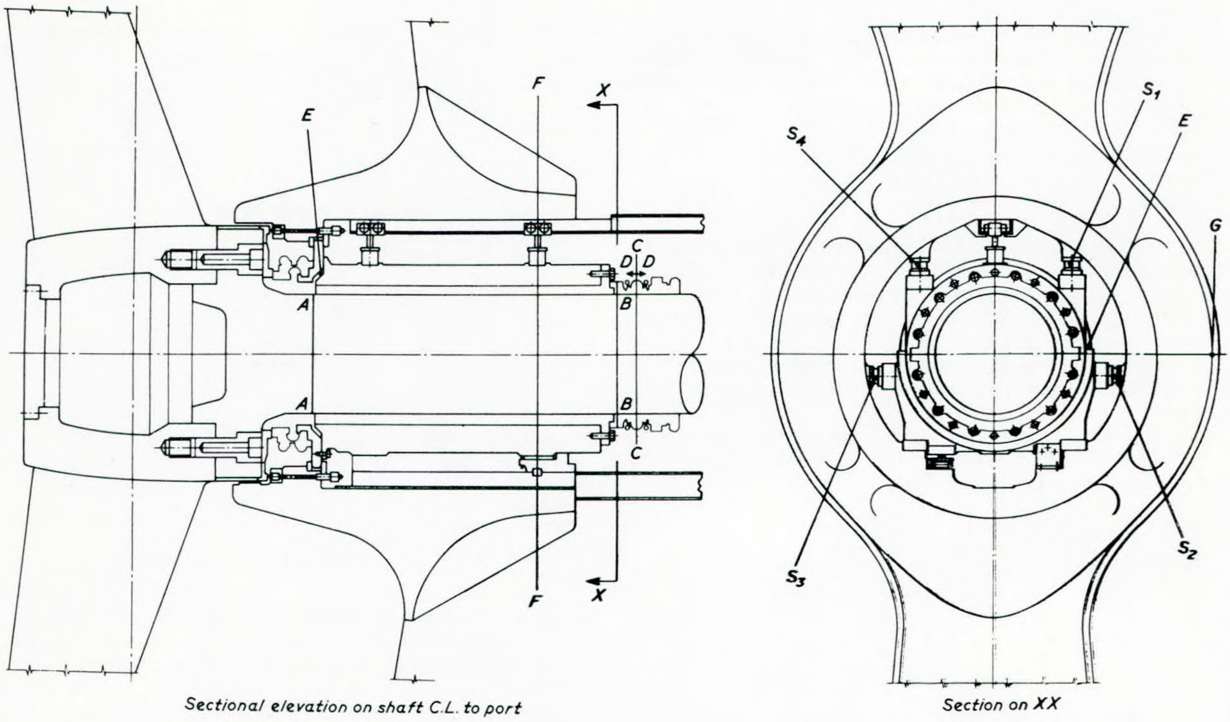


FIG. 1—*General arrangement of test bearing*

might lead to oil leakage or sea water ingress. Axial shaft movement will cause face type seals to be subjected to continuous compression and expansion which might possibly lead to eventual fatigue of the components. It has also been suspected for some time that the axial vibration can lead to compression and expansion of the lubricating oil in the space between the outboard seal and the tailshaft thus causing pressure increases or reductions compared to the pressure in the inlet pipeline.

For these reasons axial shaft movement and lubricating oil pressure were measured in this series of tests.

It was also decided to take the opportunity of measuring the loads in the bearing holding down jacks and the stress in the hull plating adjacent to the welded joint between the stern bearing and the hull. Similar tests had been carried out some years ago on one of the early Turnbull Mark I bearings fitted to a 12 000 dwt vessel and, although the results showed that the various jack loads and hull stresses were low and of no consequence, it was felt desirable to obtain this information from a large vessel. Also, no information of this type was available for the improved Mark IV design fitted on the test vessel.

The Turnbull Mark IV split stern bearing has been designed to overcome the disadvantages of the conventional stern tube by offering accessibility and the ability to examine all the component parts with the ship afloat. In addition there is the added advantage of being able to adjust the stern bearing position (afloat and at any desired draught condition) to enable the best possible shaft alignment to be achieved.

The test bearing is illustrated in Fig. 1 and a general arrangement of the shafting on the test vessel is shown in Fig. 2.

The Turnbull bearing concept has already been described in a previous paper to the Institute [2]. Briefly the bearing consists of two halves, each of which can be withdrawn separately into the vessel (afloat and loaded if necessary) to enable bearing and tailshaft examination to be carried out without disturbing the propeller or tailshaft. Chocks under the lower half bearing allow re-alignment to be carried out at any time during the ship's life.

The vessel on which the tests were carried out was a very large crude carrier of 280 000 dwt.

Length:	324 m
Breadth:	53·5 m
Depth:	28 m
Power:	26 845 kW at 90 rev/min
Propeller diameter:	8·6 m

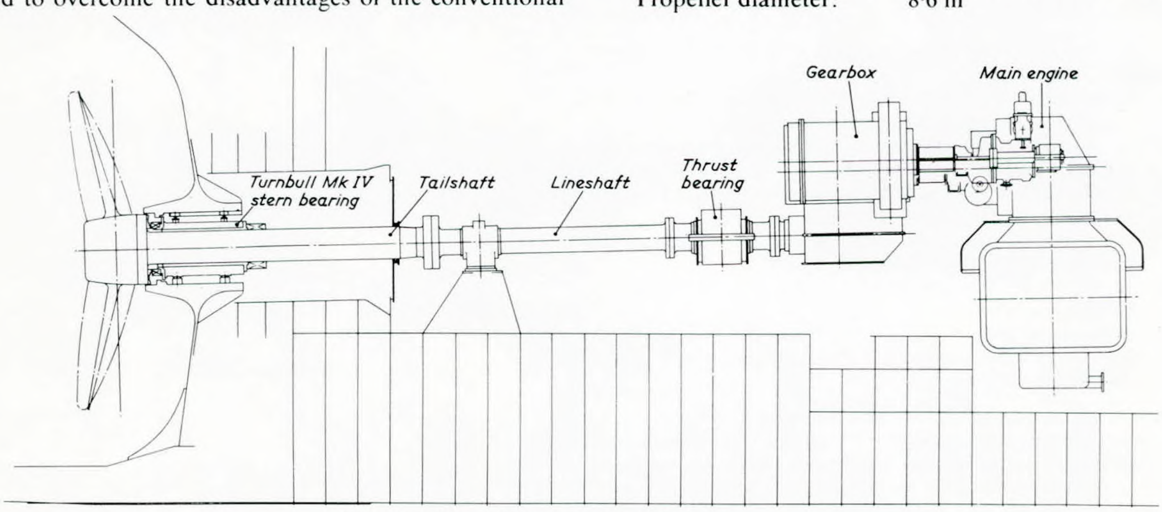


FIG. 2—*General arrangement of shafting*

Test measurements were obtained at a variety of draught conditions ranging from light ship to fully loaded. However, extended running only occurred at ballast and loaded draughts. Draught conditions were as follows:

Light ship:	5·3 m
Ballast:	12·5 m
Loaded:	21 m

The bearing alignment chocks had been fitted before the vessel first went into service in 1973 and there was no need to change this alignment at the time the test instrumentation was fitted. However, the actual alignment was checked at this time using bridge and feeler gauges and was found to be correct. The draught at the time of these checks was 5·3 m which was similar to that at which the original alignment was carried out.

The tests were carried out during the period 20th July to 30th August 1975, on a voyage from Japan to the Arabian Gulf in ballast and from the Arabian Gulf to Cape Town fully loaded.

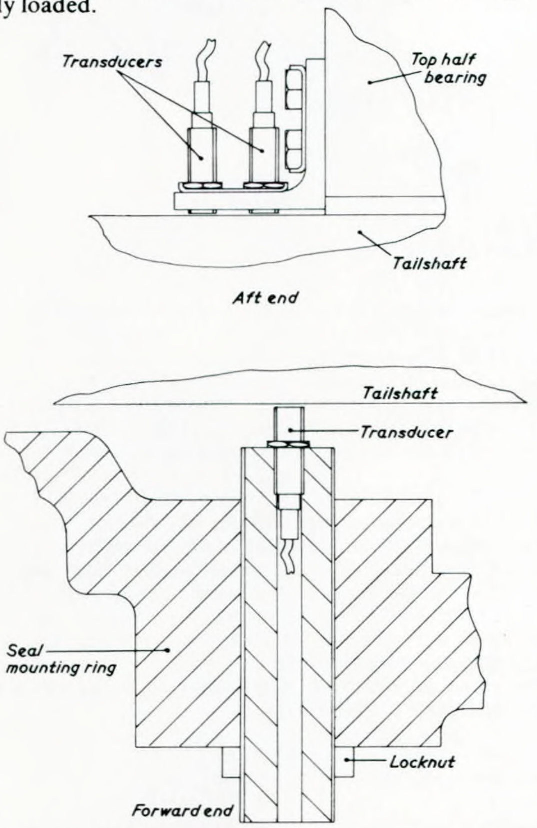


FIG. 3—*Radial proximity transducer mounting details*

TEST PROGRAMME

Objectives

i) to establish the variation in the radial position of the tailshaft relative to the stern bearing;
ii) to establish the minimum clearance between the tailshaft and the stern bearing;
iii) to establish the variation in the axial position of the tailshaft relative to the stern bearing;
iv) to measure the variation in the lubricating oil pressure in the outboard seal oil space;
v) to measure the variation in load in the bearing holding down jacks;
vi) to measure the variation in stress in the hull adjacent to the welded connexion between the stern frame and the hull.

Description and Location of Test Equipment

Non-contacting, inductive displacement transducers were used to measure the radial and axial position of the tailshaft. The loads in the hydraulic jacks and the hull stress were measured by means of strain gauges. The lubricating oil pressure was measured by means of a pressure transducer.

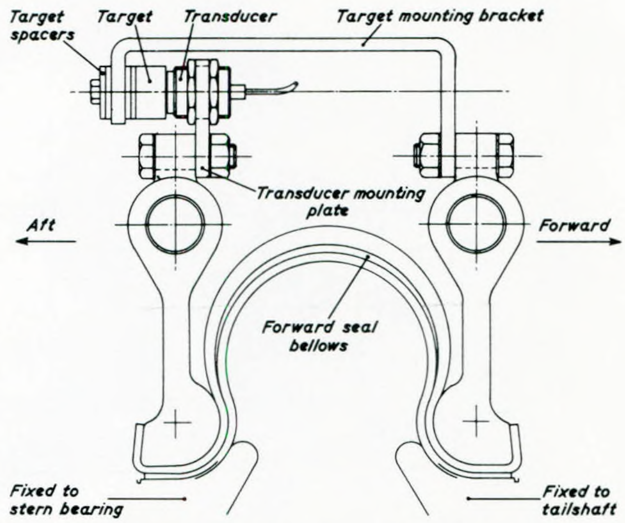


FIG. 4—*Axial proximity transducer mounting details*

A multi-channel, modular carrier amplifier system was used for energizing the transducers and strain gauges and for output signal conditioning. Read-out was also taken from the same unit.

The shafting arrangement is shown in Fig. 2 and a general arrangement of the bearing is shown in Fig. 1. The radial proximity transducers were fitted at both ends of the bearing in planes *A-A* and *B-B* as shown in Fig. 1. The methods of mounting the individual transducers are shown in Fig. 3.

The axial proximity transducer was fitted in plane *C-C* (Fig. 1) in the position shown as *D-D*. Fig. 4 gives details of the mounting arrangement. The calibrated spacers were required to increase or decrease the datum gap since the expected shaft movement was greater than the working range of the transducer.

The pressure transducer was fitted at position *E* (Fig. 1) and a detail of the arrangement is shown in Fig. 5.

Strain gauges were attached to the bearing holding down jacks in plane *F-F* (Fig. 1) in the positions shown as S_1, S_2, S_3 and S_4. Fig. 6 shows details of the positions of the strain gauges.

Strain gauges were also attached to the hull plating at location *G* (Fig. 1) just forward of the welded joint.

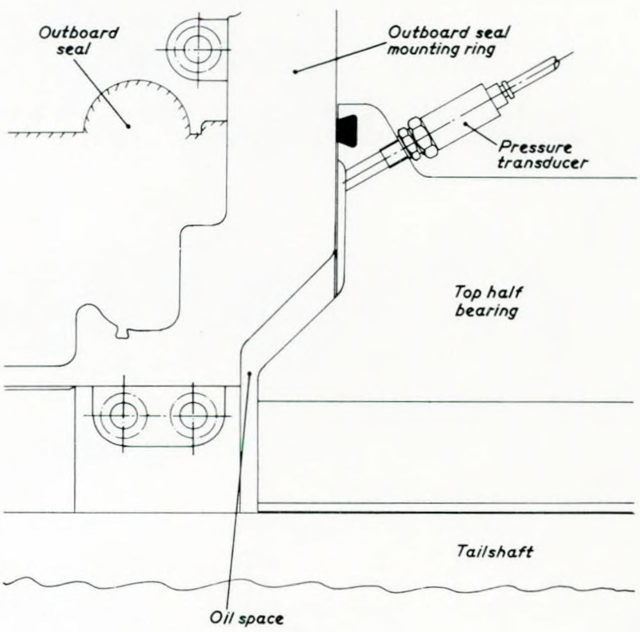


FIG. 5—*Location of pressure transducer*

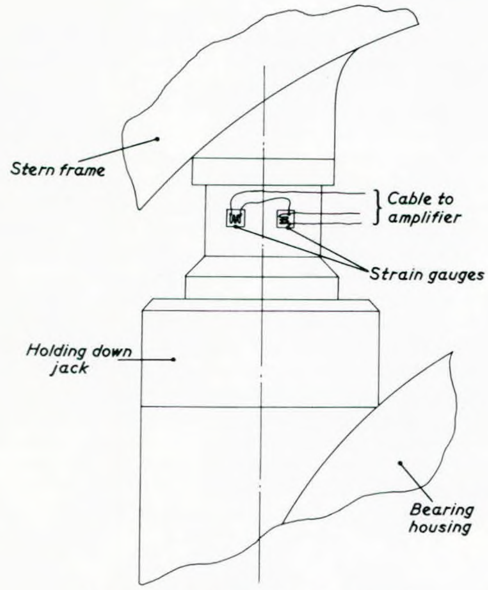


FIG. 6—*Position of strain gauges*

Discussion of Results

Measurements were recorded at regular intervals throughout the day while the vessel was at sea, at anchor, or on the loading berths and continuously while manoeuvring. The results are shown in graphical form in Figs. 7 to 14.

For convenience each of the graphs has been plotted to a base of rev/min, although the separate readings are not necessarily variable with respect to rev/min. This is especially true for the loads in the jacks and the stress in the hull shell plating and explains why there is such a wide scatter on some readings.

In the curves for the loaded condition, Figs. 9 and 11 to 14 (inclusive), there is no significance in the fact that, as rev/min increases, the graphs become asymptotic. The reason for this is that there was no extended running in the loaded condition at speeds below 80 rev/min and, in fact, the results for the range 50 rev/min astern to 80 rev/min ahead were all obtained in 1·5 hours after leaving the loading berth. Since the vessel had been at anchor for several days previously the results would obviously be quite different as the plant warmed through and expanded.

It should be noted that, although several different sea conditions occurred during the voyage, no exceptionally heavy weather was encountered at any time.

The test programme produced some rather unexpected results (especially with respect to hull deflexion and shafting alignment) which may help to explain why the reliability of stern bearings and oil seals is lower than might be expected.

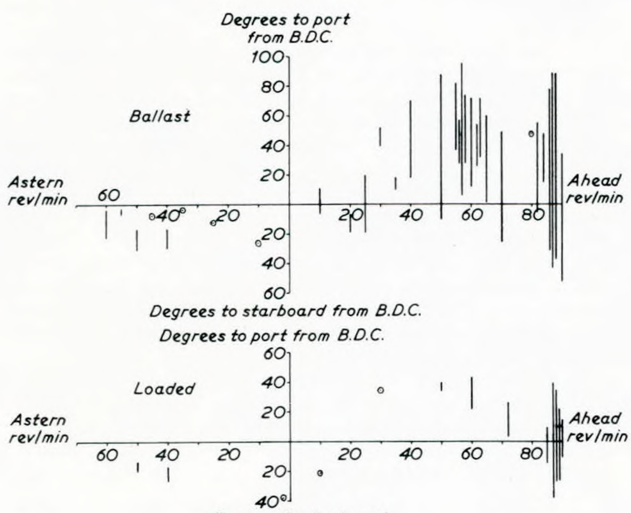


FIG. 7—*Tailshaft attitude*

(i) *Tailshaft Attitude*

Fig. 7 shows how the tailshaft attitude varies within the stern bearing clearance and attitude angle θ is defined in Fig. 15. The information from Fig. 7 together with the corresponding bearing clearance measurements discussed in section (ii) below means that the actual position of the tailshaft within the bearing is known for all conditions throughout the test period. It is to be understood that attitude refers to the angular position of the tailshaft within the bearing as defined in Fig. 15. It does not refer to the possibility that a tailshaft may be angled in the axial direction relative to the bearing.

In the ballast condition it can be noted that between 60 rev/min astern and 70 rev/min ahead the movement of the tailshaft within the bearing follows the expected movement for a clockwise rotating shaft (i.e. clockwise when viewed from aft), i.e., movement to port when going ahead and *vice versa* going astern. There is a certain amount of scatter which is probably attributable to two main reasons. Firstly, the movement of the vessel in the seaway causes the direction of the load applied to the bearing to be varied and hence the attitude angle changes. Secondly, the propeller operates in a non-uniform wake due to the disturbances caused by the ship's hull. This non-uniformity leads to a varying torsion being transmitted to the tailshaft and this in turn will effectively alter the position of the shaft within the bearing. Up to 70 rev/min ahead the tailshaft frequently rises up to 70° from the bottom centre position. In the full speed range of 80 to 90 rev/min, however, there is a considerably greater shaft movement ranging between 90° to port and 50° to starboard from the bottom centre position.

It should be noted that this range (and also subsequent ranges) was recorded during several weeks at sea and does not mean that the tailshaft moves through 140° at any one instant of time. At this speed range the general level of vibration throughout the vessel was noticeably greater and there was also very much more propeller noise noticeable to the ears. It is therefore likely that the wake effects are also more pronounced and this, together with the movement due to the seaway, explains why the results are more erratic than those at lower speeds.

In the loaded condition it can be seen that, between 50 rev/min astern and 70 rev/min ahead, the tailshaft again follows the expected movement for clockwise rotation although the angular variation is not as great as that found in ballast. The variation in the 80 to 90 rev/min range is again greater than for the lower speed ranges as was also evident in ballast. However, the range at all speeds is considerably less than that in the ballast condition varying from 40° to port to 30° to starboard. The reduction in the range of movement measured at full load compared to ballast is likely to be due to the greater immersion of the propeller and the subsequently more uniform wake condition which will reduce the torsional variation on the tailshaft. At ballast draught the propeller tips were 1·35 m below the surface while at full load draught this was increased to 9·6 m. Although the propeller could not be observed from on board the vessel it is believed that it did not break surface in the ballast condition even while the vessel was pitching since there was no evidence to suggest this. However the non-uniformity of the wake would be further aggravated in ballast by the disturbances due to sea swell. It is also probable that the loaded vessel will be less prone to flexural movement in the seaway.

Since the tests were carried out during normal commercial operation of the vessel it was not possible to operate for extended periods at selected speed ranges. The only speed range at which extended running occurred in both ballast and loaded conditions was 80 to 90 rev/min. Table II has been compiled from the test results for that speed range.

It can be deduced from the table that the range of attitude angle is influenced by both weather and draught conditions. The range in ballast, regardless of weather, is always greater than when fully loaded. For both ballast and loaded conditions the range in rough weather is generally greater than in calm weather if the average values are considered. However, the maximum reading in the calm, ballast condition is slightly greater than in the rough, ballast

TABLE II—ATTITUDE ANGLE VARIATION AT 80 TO 90 REV/MIN

		Ballast	Loaded
Calm	Average	$-20°$ to $+51°$	$-11°$ to $+11°$
	Max.	$-52°$ to $+89°$	$-24°$ to $+29°$
Rough	Average	$-21°$ to $+70°$	$-18°$ to $+15°$
	Max.	$-44°$ to $+89°$	$-38°$ to $+40°$

+ Denotes Movement to Port
– Denotes Movement to Starboard

condition. This observation, together with the fact that there is a considerable range of movement present in all conditions, varying from an average of 22° in the calm, loaded condition to 91° in the rough, ballast condition, leads to the conclusion that this continuously changing shaft attitude is the normal mode of operation.

If one considers the only other speed range at which extended running occurred, that is 55 to 65 rev/min in ballast in calm weather, it is found that the average range of attitude variation is between 30° and 57° to port with a maximum recorded range between 0° and 95° to port. These results back up the conclusion in the previous paragraph that it is quite normal for the shaft attitude to vary continuously and over such a large range.

This conclusion leads one to suppose that it is quite possible that operating problems might occur on the types of stern bearings which have large axial oil ways situated along the horizontal centre line on both sides of the bearing. If these designs are studied it will be found that the bottom edges of the oil grooves usually begin between 60° and 75° from the bottom centre of the bearing. It can be seen from Table II and Fig. 7 that, in ballast, the tailshaft will frequently run into the oil way. It is possible that this will cause the hydrodynamic oil film to break down due to the discontinuity and change in direction of the bearing surface with the result that boundary lubrication and metal-to-metal contact may occur.

The tailshaft attitude is of little importance in the Turnbull system, however, since plain journal bearings without oil ways are normally used and the results discussed in section (ii) below proved that metal-to-metal contact did not occur at any time on the test vessel.

(ii) *Bearing Clearance*

The variation in the minimum radial clearance is shown in Fig. 8.

It can be seen that there is very little difference in the minimum bearing clearance between ballast and full load conditions. The test results indicate that the minimum clearance occurs at either full ahead or full astern regardless of the loading condition of the vessel. The bearing clearance is adequate at all times with a minimum value of 0·025 mm when going full astern in the ballast condition. The minimum clearance when going full ahead is 0·051 mm and occurs in both the ballast and loaded conditions. The general value of the clearances at speeds other than 80 to 90 rev/min is higher than theory would suggest for both ballast and full load operation. This is very pronounced at very low speeds where the test results show that the clearance is around 0·5 mm. This indicates that the vessel has deflected relative to the tailshaft from the datum light ship condition since it would be expected that the shaft would be in contact with the bearing at very low speeds.

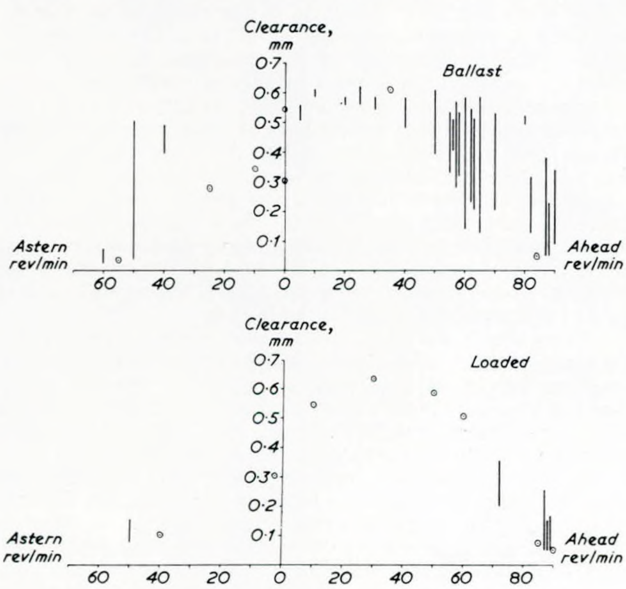


FIG. 8—*Minimum radial clearance*

If this is generally true then it is of major importance for shaft alignment purposes. The measuring instruments were set up with the vessel at light draught and at that time the tailshaft was lying in the bottom of the bearing at zero rev/min, i.e., zero clearance. Shaft alignment is normally also carried out at this same draught condition and the measured variation in clearance would seem to indicate that shafting alignment set at light draught could well be incorrect at ballast and loaded draughts. Because of this it would seem to be better if alignment was completed at ballast draught at least.

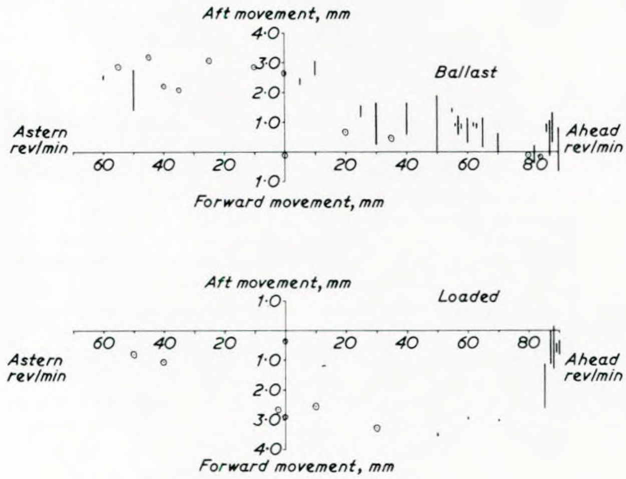


FIG. 9—*Axial shaft movement*

(iii) *Axial Shaft Movement*

Axial shaft movement variation is shown in Fig. 9. The total measured variation was 6·73 mm between full astern in ballast and full ahead in the fully loaded condition. It is interesting to note that in the loaded condition the shaft appears to be compressed relative to the datum position.

Although it could be expected that there would be more compression of the tailshaft at full load than in ballast due to the greater propeller thrust the recorded difference was greater than that due to shaft compression under propeller thrust. At 80 to 90 rev/min in ballast the shaft position was about 0·76 mm aft of the datum whereas for the same speed range in the loaded condition the shaft position was 0·51 mm forward, i.e., a difference of 1·27 mm. The calculated shaft compression due to the total end thrust was only 0·53 mm and would be considerably less for the difference in thrust (at 80 to 90 rev/min) between ballast and full load. The shafting was pushed forward onto the ahead thrust pads when setting up the instrumentation and thrust clearance can, therefore, be ignored.

It may be useful to consider the sequence of events leading up to this finding.

The vessel loaded a part-cargo of about 110 000 tonnes at one port. During the time the vessel was at anchor waiting to go on to the loading berth and during the loading period the shaft contracted until it returned to the datum position present when the transducer was set-up and calibrated.

The vessel then proceeded to another port to load the remaining 170 000 tonnes of cargo. During the whole time the vessel was at anchor or on the loading berth the shaft continued to contract beyond the datum point and eventually a compression of 2·92 mm was recorded. After the vessel got under way and proceeded on voyage the shaft compression gradually reduced until it settled out at around 0·51 to 0·76 mm. This took about 2·5 days and coincided with the vessel leaving the approaches to the Arabian Gulf. The reduction in compression began immediately the vessel was under way, however, and a 0·76 mm reduction was recorded only one hour after leaving the loading berth.

Since the effect was first recorded while the vessel was at anchor and with the main engines just turning over the compression is not due to propeller thrust. Instrument error can also be discounted because the transducer was visible at all times since it was fitted to the outside of the bearing and the change in the gap could be observed by eye. It would seem that when the vessel was loaded beyond a certain point there was some hull deflexion which caused the hull to lengthen relative to the shafting with the result that the shaft appeared to be compressed. It is also possible that air and sea water temperatures might have been responsible for some hull deflexion and this is discussed more fully in section (v) later.

There does not seem to be an immediate answer to why the compression reduced once the vessel was under way unless there was some kind of stress relaxation in the hull due to vibration and movement in the seaway.

Fig. 16 sketches out a possible (exaggerated) deflexion of the aft end of the vessel showing how the hull could be assumed to lengthen relative to the shafting.

In the ballast condition the recorded axial movement due to thermal expansion and steady propeller thrust indicated that the tailshaft expanded aft up to 3·18 mm from the datum position. In the loaded condition the movement was recorded as a compression of up to 3·55 mm from the datum position.

There was a vibratory axial shaft movement superimposed on the thermal expansion and steady propeller thrust movement. This vibratory movement is due to the variable propeller thrust resulting from the non-uniform wake conditions. The amplitude of vibration did not change between ballast and loaded condition being ±0·038 mm in calm weather and increasing to ±0·114 mm when the vessel was rolling. The frequency appeared to be more related to the rolling motion of the vessel than to any shaft speed/propeller blade combination.

It should be noted that the general vibration level (which was not measured) seemed to be low for this type of vessel and it is possible, therefore, that higher amplitudes than those given above might be present on some vessels.

(iv) *Lubricating Oil Pressure*

The lubricating oil pressure in the outboard seal oil space is shown graphically in Fig. 10. It can be seen that this pressure varies considerably from the lubricating oil inlet line pressure of 0·21 N/mm²(g). In general, the seal space pressure is higher than the inlet line pressure when going ahead and lower when going astern. Zero (gauge) oil pressure was recorded on more than one occasion when going astern but generally the pressure was between 0·07 and 0·14 N/mm²(g)

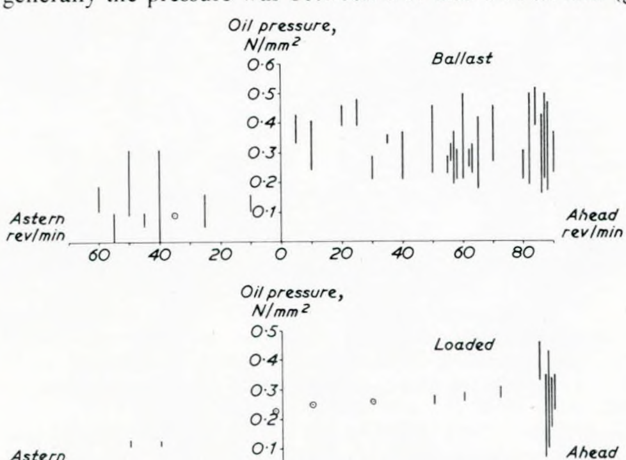


FIG. 10—*Oil pressure variation in outboard seal oil space*

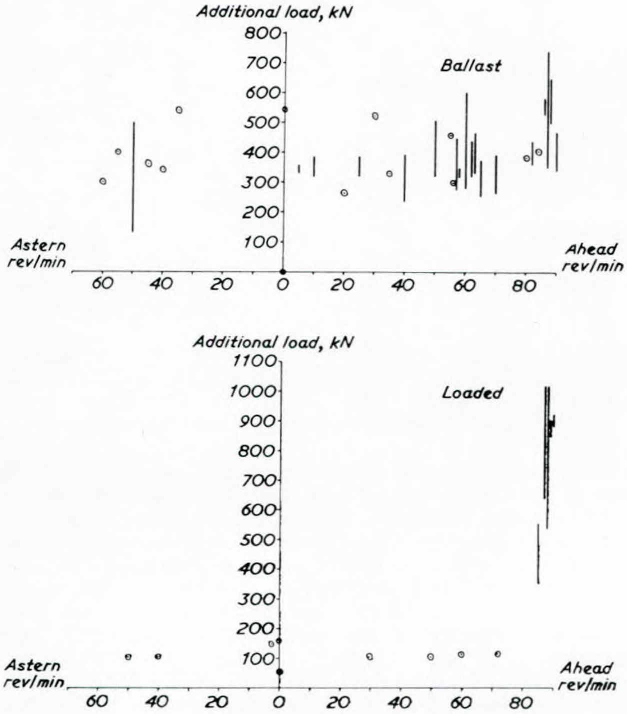


FIG. 11—*Load in port forward top jack*

for astern operation. The maximum recorded pressure was 0·52 N/mm²(g) and pressures up to 0·41 N/mm²(g) were common at most ahead rev/min in both the ballast and loaded conditions. However, at 80 to 90 rev/min ahead in the full load condition the pressure distribution did not follow the general pattern and the pressure frequently dropped below the inlet line pressure. It was observed, however, that these low oil pressures coincided with the vessel being in colder waters than at any previous time during the test programme. There was a considerable instantaneous fluctuation in oil pressure at all rev/min and all loading conditions when the vessel was rolling.

On the test vessel the sea water pressure at the outboard oil seal at full load draught was 0·16 N/mm²(g). The seal was, therefore, working with differential pressures ranging from +0·36 N/mm², (i.e., lubricating oil pressure greater than sea water pressure) to −0·16 N/mm², (i.e., sea water pressure greater than lubricating oil pressure). These differentials are considerably greater than those recommended by the various seal manufacturers. The variations have virtually no detrimental effects on the radial face type of oil seal fitted on the test vessel but lip type seals could very well be subjected to oil leakage or sea water ingress due to the considerable pressure differentials across the sealing lips.

(v) *Jack Loads*

Figs. 11 and 12 show the variation of load in the forward top jacks. This load is additional to the stressing load induced in the jacks when the bearing is assembled. The additional load is due to the thermal expansion of the bearing and also to the effects of hull deflexion.

It can be seen that both jacks exhibit a similar loading pattern although the actual loads are different. At 80 to 90 rev/min ahead in the fully loaded condition the jack loads are considerably greater than for similar rev/min in ballast, being about twice as much in the starboard jack and about 50 per cent greater in the port jack. The general level of the load in the starboard jack was, however, only about half that of the port jack.

This inequality of loading in the jacks was rather unexpected since the arrangement is symmetrical and it was previously believed that the loads would be similar. However, it has already been stated that the tailshaft generally ran in the port side of the bearing during ahead operation and, although the bearing metal temperature was not measured, there was a noticeable temperature difference between the port and starboard sides of the bearing (the port side being

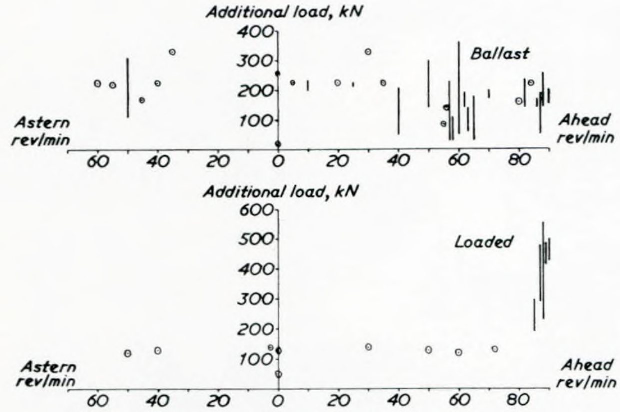


FIG. 12—*Load in starboard forward top jack*

hotter) readily detectable to the hand. The jack temperatures were also different due to heat conduction and the subsequent different expansion rates could be expected to give different loads in the jacks. There is also the possibility that hull deflexion may have played a part in producing different jack loads and this point is discussed in more detail below.

It should be noted that the additional loads are well within the capacity of the jacks and do not cause any problems.

Fig. 13 gives details of the loads in the forward side jacks. There was a considerable scatter of results for the port jack in the ballast condition. During the loading period in the Arabian Gulf the additional load increased to 2192 kN which was very much higher than expected and it was believed that the strain gauges had been damaged. However, once the vessel was under way again and out of the Arabian Gulf, the readings became comparable with those previously obtained thus indicating that the strain gauges were not damaged (see Table III). The very high loads were only recorded in the

TABLE III—PORT SIDE JACK LOAD

	Load (kN) Average	Max.
Week before entering Arabian Gulf	373	700
In Arabian Gulf	1184	2192
Week after leaving Arabian Gulf	421	690

Arabian Gulf and only when the vessel was at anchor with the main engines just rotating. However, above average loads were also recorded in Japan when the vessel moved from the shipyard to deep-water anchorage and also on the day of sea trials prior to sailing from Japan. The maximum value

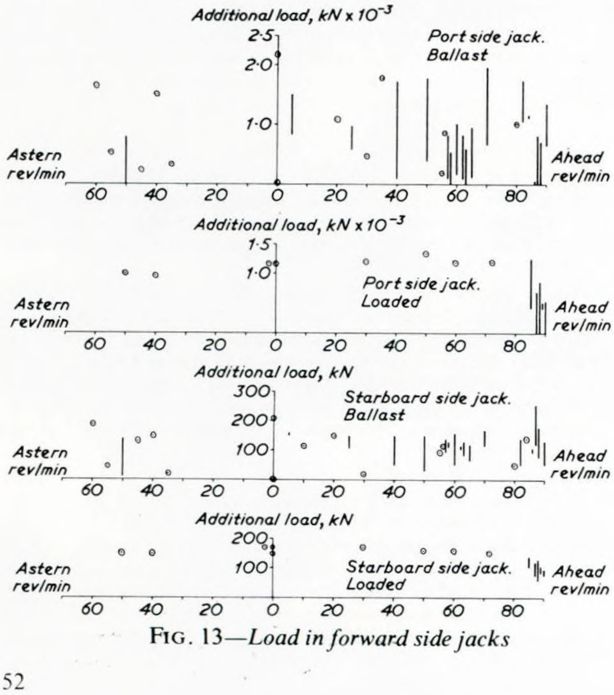


FIG. 13—*Load in forward side jacks*

recorded was 1355 kN which was much less than the maximum value in the Arabian Gulf but well above the average value of 524 kN recorded between Japan and the entrance to the Arabian Gulf.

It was also noted that the first reading of the day for the port side jack load (at 0830 hours) was invariably very much lower than at any other time. It was thought initially that this was due to the position of the sun but this was later proved incorrect since, even with the vessel on different headings and with the sun in a different position relative to the ship, the morning reading remained low.

A possible explanation might be that the high loads are due to built in stresses in the hull introduced during construction and which become magnified by hull deflexion. The geographical position of the building berth on which the test vessel was constructed is not known but it is a well-known fact that temperature differentials can have significant effects on a ship under construction.

Since the early morning reading was low compared with readings later in the day it may be assumed that, as a result of the expansion of the hull due to direct heating by the sun or to variations in sea water temperature, these built-in stresses lead to hull deflexions which affect the side jack load.

In the Arabian Gulf the much higher temperatures have evidently led to greater hull deflexions and even higher loads. The air temperature was over 54°C and Table IV shows the average sea water temperatures before, in and after the Arabian Gulf.

TABLE IV—AVERAGE SEA WATER TEMPERATURE

	Average Sea-Water Temperature
Week before entering Arabian Gulf	27·6°C
In Arabian Gulf	32·9°C
Week after leaving Arabian Gulf	24·6°C

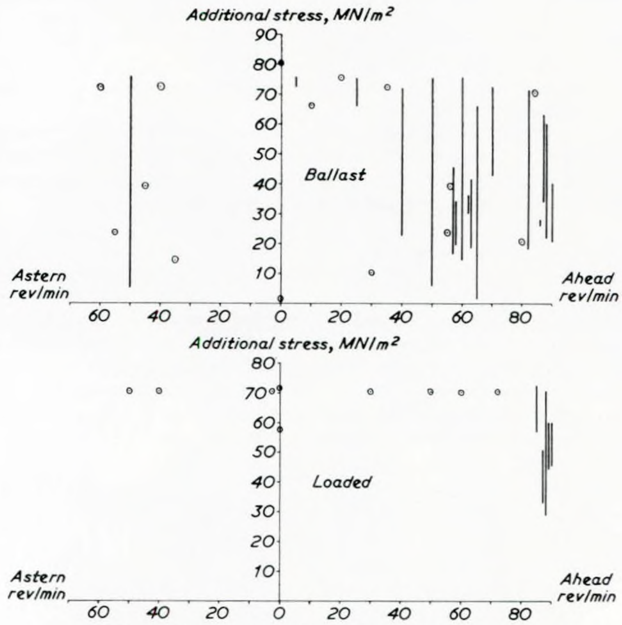


FIG. 14—*Stress in hull shell plating*

Since there is no apparent correlation between these abnormal readings and the position of the sun it seems more likely that it is the general temperature rise which acts on the locked in stresses rather than differential temperatures between the ship's sides. The reason for this conclusion is that the same magnitude of loading was recorded when the vessel was fixed between buoys as when at a single point mooring and free to swing through 360°.

During normal running conditions, for both ballast and loaded conditions, the additional load in the port jack was between 299 and 847 kN.

The starboard jack load was reasonably steady during normal running conditions with a general value of between

100 and 199 kN which was considerably less than that in the port jack. However, the starboard jack was affected in a similar way to the port jack during the loading period in the Arabian Gulf. The trend to increased loads was noticeable but was very much less marked than for the port jack as can be seen from Table V.

TABLE V—STARBOARD SIDE JACK LOAD

	Load (kN) Average	Max.
Week before entering Arabian Gulf	126	180
In Arabian Gulf	154	212
Week after leaving Arabian Gulf	92	128

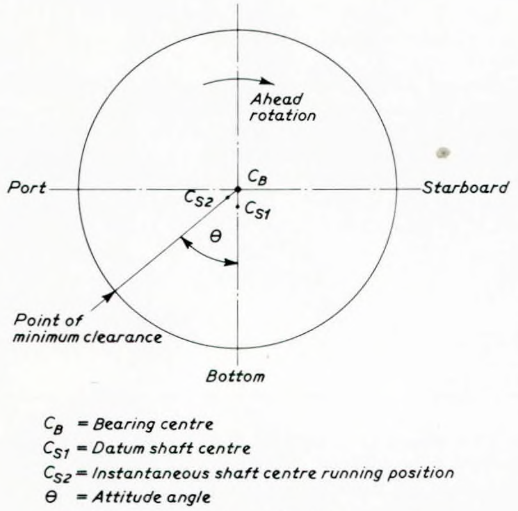


FIG. 15—*Definition of attitude angle*

It must be emphasized that even the very high jack loads were not transmitted to the tailshaft since there was a clearance between the tailshaft and the bearing at all times. There was, therefore, no additional force on the tailshaft.

(vi) *Stress in Hull*

Fig. 14 shows the variation in the hull shell plating stress just forward of the welded joint between the hull and the stern bearing casting.

It is significant that all the values greater than 62 MN/m^2 were recorded while the vessel was in the Arabian Gulf. It should also be noted that the measuring point was on the port side of the vessel and the high stresses in the hull coincide with the high loads in the port side and top jacks. This seems to support the hypothesis in section (v) that built-in hull stresses are responsible for hull deflexion.

During normal operation at sea, (i.e., out of the Arabian Gulf) the value of the hull stress in ballast ranged between 17·2 and 44·8 MN/m^2 rising to between 34·5 and 58·6 MN/m^2 for the loaded condition. It should again be noted that these values were recorded over the duration of the voyage. They do not denote the stress fluctuation at an instant in time.

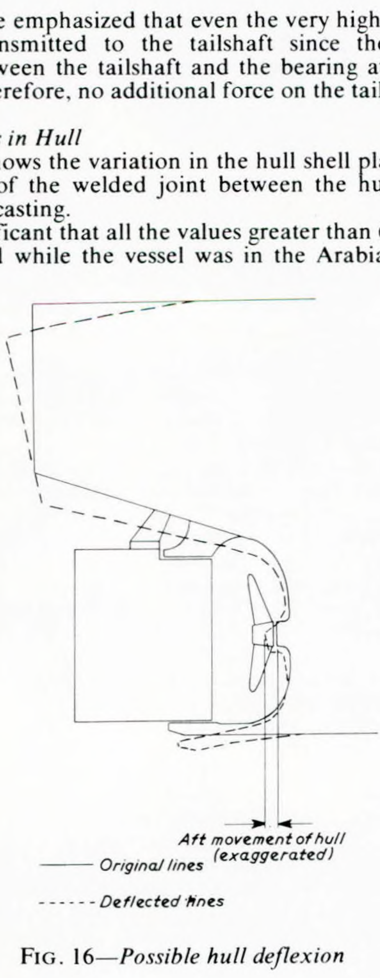


FIG. 16—*Possible hull deflexion*

CONCLUSIONS

A number of conclusions may be deduced from the results of the test programme. Before describing them, however, it should be noted that there is no reason to believe that any of the results were influenced by the fact that the measurements were obtained from a split stern bearing. The split bearing simplifies the problems (and reduces the cost) of fitting the necessary instrumentation to a stern tube. It is firmly believed that similar results would be obtained from measurements on a conventional stern tube.

One of the most important findings was that hull deflexion has a significant effect on most of the results and is responsible for greater variations in readings than had been previously supposed. Bearing clearance, axial shaft movement, jack loads and hull stresses all appear to have been influenced by hull deflexion. There was nothing about the construction of the test vessel which was different to normal and it must therefore be concluded that similar deflexions and subsequent effects may be present on all large vessels.

The angular position of the tailshaft within the stern bearing varied considerably but the tailshaft was always in the lower half bearing. This contrasts with earlier findings on a 12 000 dwt container ship fitted with a Turnbull Mark I stern bearing. On that vessel the tailshaft was found to run in the top half of the bearing practically all the time.

The tailshaft generally takes up its expected attitude within the bearing except at full ahead operation. This is true for both ballast and full load operation although the range of angular movement at full load is noticeably less than in ballast. At full ahead operation the tailshaft position is much more unstable and it frequently runs with an attitude which is unexpected, i.e., in the starboard side of the bearing. It is believed that this instability is due to the more pronounced propeller wake variation present at full ahead speeds. The variation in shaft movement was found to be dependent on both the draught and the weather conditions.

The range of movement is such that the tailshaft frequently encroaches into the oil ways which are normally positioned on the horizontal centre line of most types of stern bearings. This could lead to bearing damage and perhaps eventual bearing failure since very high local loading is likely. The problem does not arise on the test bearing, however, since it is usual for plain bushes without any oil ways to be fitted on this type of stern gear.

The minimum values of the minimum bearing clearance were found to be at full ahead and full astern and were practically independent of draught. The minimum measured clearance was adequate at all times and metal-to-metal contact did not occur at any time. It was clear from the manner in which the clearance varied from light ship to ballast that final alignment of the shafting could be more accurately carried out if completed at ballast draught and not light ship condition as is usual. The reason for this is that there was a definite, measurable hull deflexion when the draught was increased from light ship to ballast condition and thus any alignment carried out at light ship condition is likely to be in error at ballast draught. Since most vessels will be operating between ballast and full load draught there is little point in aligning the shafting at any lesser draught condition. The change from ballast to full load draught, however, did not significantly affect the alignment.

Of the total measured shaft axial movement of 6·73 mm over half was directly attributable to hull deflexion. This movement, plus the superimposed axial vibration, is of little significance for journal bearings but might cause design and operating problems for roller type bearings since it is unlikely that such large variations could be accommodated within the bearing itself.

The variation in lubricating oil pressure due to the axial shaft movement is greater than expected and might lead to oil leakage or sea water ingress on some types of seals due to the high pressure differentials present. The cyclical nature of the pressure pulsations might also eventually lead to fatigue problems. The recorded oil pressure was sensitive to sea water temperature being generally lower than average in cold waters and higher than average in warm waters.

The level of jack loads and hull stress was quite acceptable and even the very high loads which were recorded in the port forward side jack at one stage of the tests were well within the design capacity of the jack. However, it was significant that these loads were quite definitely influenced by the effects of hull deflexion.

Further investigation of the following items appears to be desirable: influence of temperature and loading conditions on hull deflexion, and the effect of hull deflexion on alignment and component stresses.

ACKNOWLEDGEMENTS

The author wishes to thank Mobil Shipping Company Limited for allowing the tests to be carried out on board one of their vessels. His thanks are also due to the ship's company for their assistance. The work was carried out with part support from the Ship and Marine Technology Requirements Board of the Department of Industry and their assistance, and permission to present this paper, is gratefully acknowledged. The author also wishes to thank the members of the staff of Ross Turnbull Ltd. who contributed to the production of the manuscript.

REFERENCES

1) Boylan, F. N., and Atkinson, F. H. 1975. "The Hull Survey of VLCC Including the Use of In-Water Techniques." Joint Conference on In-Water Maintenance on Ships. *Trans.I.Mar.E.*, Series B, pp. 10–26.
2) Crombie, G. and Clay, G. F. 1972. "An Improved Stern Bearing System — Design Features of, and Operating Experience with, Turnbull Split Stern Bearings." *Trans.I.Mar.E.*, Vol. 84, pp 349–366.

Discussion

Mr. D. McKinley was grateful to Mr. Crombie for such a detailed description of the measurements carried out to establish the operational characteristics of the Ross Turnbull design of stern bearing.

Mr. McKinley's organization were interested in the results of all shipboard measurements and of course as to how the results could be applied to the problems of ensuring that all parts of the machinery operated in a safe condition. Shipboard measurements of the type described in the paper always made one aware of the necessity to install equipment which would operate on the flexible platform offered by a ship.

The organization had examined in detail the effects of hull movements upon shafting, gearing and engine crankshafts, and had developed calculation techniques to enable predictions to be made. The shipboard measurements possible were becoming more sophisticated as electronic instrumentation improved and as more effective transducers were produced. The increased sophistication enabled Mr. McKinley's organization to establish more clearly what was actually happening to the machinery.

The problem of effects of hull movement on gear teeth had now been examined upon numerous occasions, by means of strain gauges in the roots of the gear teeth. Modern telemetry systems had enabled signals to be transmitted from the rotating components. The use of this technique had now become almost commonplace and required little modification to the machinery, except to bring wires through the casing and the supporting of instrumentation on the rotating elements.

Calculations had been refined to such an extent that the effects of hull movement on the crankshaft of a large engine could be calculated and the calculations backed up by measurement if required.

In the case of the sterntube bearing, measurements were made periodically to establish operational conditions. Mr. McKinley could not recollect his organization ever having been in such a fortunate position as Mr. Crombie, who had even been able to see the transducer. When the organization had worked in this area, it had always seemed to be in the unfortunate position of fitting proximity probes outside the hull with its obvious environmental problems, or drilling into sterntubes etc.

Mr. McKinley had noted that radial transducers were fitted to each end of the bearing and yet Fig. 8 appeared to refer to the measurements at one end of the bearing, presumably the forward end. It would have been useful to have seen the results at both ends and indeed, the relationship of one end to the other, as it was postulated that the hull movement affected the angle of the tailshaft to the bearing. Wide ranges of movement had been noted in the shaft attitude, but it was not clear to Mr. McKinley what was affecting these ranges. Were the movements related to blade order, pitch and roll motions of vessel, or rudder wake?

To discuss shaft attitudes further: Was it so surprising that shaft attitude appeared on the starboard side at full power with a right hand turning propeller? Mr. McKinley believed that it would suggest that forces due to eccentric thrust modified the attitude described as normal in the paper. His organization had noted groove damage likely to have been caused by the attitude movement described in the paper, and it was not at all unusual for it to have appeared on the starboard side.

The axial movements of the shaft were most interesting, but he was somewhat puzzled that there was no reference to the thrust movement under load. He would have expected that this movement would have been apparent with the method of measurement used.

The conclusion of the paper suggested that there was little point in aligning the shafting with the vessel in the light condition. Unfortunately it was a fact that many vessels had to be aligned in such a condition. Therefore, a method of checking with a shafting system design which allowed adjustments, would seem to be the way these alignment variations would have to be handled.

Mr. McKinley concluded by saying that information based upon actual measurement could be so difficult to obtain in the shipboard environment and the author was to be congratulated on what had been achieved in his excellently presented paper.

Mr. M. Joyce said he had been involved during the last thirty years in matters related to sterntubes and sterngear, and had found the paper fascinating. His company had specialized since the turn of the century in the production of sterntube lubricants, and had designed a stern seal in 1901, which was fitted to ships of all types until it was withdrawn a few years ago. He was, therefore, well aware of the many problems which had been experienced with oil-lubricated sterntubes. Many had simple, mechanical or human explanations, but others remained unexplained due to lack of knowledge or practical research into that section of a ship.

The work covered in the paper had helped to throw new light on the subject, but there were a number of points upon which clarification, or opinion, would have been of value.

No reference had been made to the diameter of the tailshaft. As the vessel concerned in the tests was a 280 000 tonne VLCC, it could be assumed that the diameter would have been in the region of 1000 mm or more. The importance of the diameter was in its relationship to the peripheral speed of the shaft, which could have materially affected the behaviour of the lubricating oil in terms of flow in the bearing, and oil pressures. Had consideration been given to this aspect and if so, what conclusions were drawn?

The conclusions expressed regarding the effect of oil ways in a sterntube bearing were interesting in that they tended to support the view that axial oil grooves cut into the top half of the bearings could create problems which would contribute to outboard oil losses. Until recent years, Mr. Joyce's experience had shown that there had been little standardization of theory or practice. Axial grooves cut in a variety of radial positions could still be found; some extended the whole length of the bearing, others stopped short. Some might be linked with circumferential grooves, and even helical grooves running the length of a bearing had been noted. Width, depth and section had varied widely over the whole spectrum. The use of plain bearing for such a large tailshaft appeared to be unusual, and he asked whether the author and his colleagues advocated such a design for all types and size of installation?

No reference had been made to the oil supply system installed in the test ship. The type of supply system, its installation and operation, was recognized as a potential source of pressure fluctuation, and could contribute to outboard oil losses, or inadequate lubrication. Was the oil supply system a simple gravity-feed or a pump-operated, circulatory system?

The observation that oil pressure in the oil seal space dropped below that of the inlet line pressure in colder waters was interesting but somewhat puzzling. One would have expected a lower temperature to increase the viscosity of the lubricating oil and, if anything, tend to thus increase pressure.

Mr. Joyce inquired whether any explanation of the observed phenomena had been advanced, and whether it would have been realistic to assume that it would occur in all types of ships and sterngear installations.

There was no reference in the paper to temperatures — either in the bearing or at the inboard and outboard seals. It would have been useful to know whether these were recorded during the test period.

Some concern had been expressed regarding the running temperatures recorded in the sterntubes of VLCC from time to time. The effect of temperature on the sterntube lubricant was important. If high temperatures were experienced the basic characteristics of the oil could be seriously affected, and its efficiency reduced. This could result in more rapid bearing wear and, in the extreme, to failure or seizure. Any information that would provide guide lines as to what could be expected as "normal" running temperatures relative to shaft speed, sea temperature and loaded condition of the vessel, would be of extreme value.

Dr. R. A. Goodman, F.I.Mar.E., stated that the paper had provided a great deal of useful information about, and given a deeper insight into, the real operating conditions in a stern tube bearing. It was of particular interest to note the comments and conclusions on the effects of hull deflexions. The Society had been aware, for some time, of the relatively flexible nature of aft end structures in larger ships, and of the consequent implications on the machinery operating conditions. It was, therefore, encouraging to see attention focussed on this potential problem from the commercial viewpoint of ship operation economics.

Over the past few years, developments in computer technology and finite element methods had made the theoretical prediction of some of the major effects mentioned in the paper a realistic proposition. For example, the prediction of relative hull deflexion between different loaded conditions had been carried out in several instances by the Society, as a bi-product of the more complex analysis of the dynamic characteristics of aft end structures. In such analyses, it was possible to predict the influence of draught on shaft alignment, bearing reactions, etc., thus enabling the most favourable alignment for all operational loaded conditions to be determined in relation to gearbox, stern tube bearing or line shaft bearing requirements. The change in relative axial position between shaft and stern tube was also readily shown in these analyses. Correlation of the theoretically predicted hull deflexion with full scale measurement was encouraging, agreement having been obtained in the few cases investigated to within the limits of experimental accuracy.

The cost of such theoretical analyses was not insignificant, although still small in comparison with the potential expense involved should gearing or stern tube bearing problems be encountered, after the ship had entered service. For the typical aft end structure of large ships, the cost of the basic analysis into relative hull deflexions and changes in shaft alignment under different loaded conditions should be of the order of £25 000. The major part of this cost was taken up with the preparation of data for the finite element model and the formulation and decomposition of the stiffness matrices. For relatively little additional effort, the finite element model could be utilized to provide further useful information, for example:

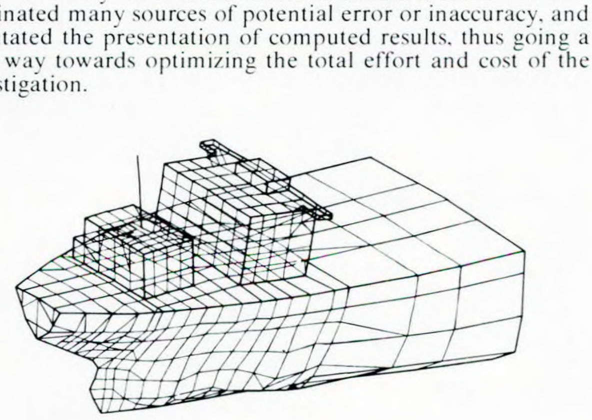
Relative hull deflexions between the afloat and dry dock or slip-way condition, thus enabling machinery alignment to be set up in any of these conditions;

Thermal hull deflexions, provided realistic temperature distributions could be defined;

Total flexibilities of bearing/hull structure arrangement for stern tube, line shaft or thust bearings, thus enabling a more accurate determination of shaft lateral and axial vibration characteristics.

Further extension of the model to include the superstructure, a simple representation of the midship and fore-body sections and the added mass, enabled a comprehensive analysis of the vibration and dynamic response characteristics to be carried out, including coupled shaft/hull modes.

An example of an aft end structure modelled for the latter type of analysis was given in Fig. 17, where, for clarity, all the internal structure had been omitted. The data preparation time for such a model had been reduced to just a few weeks by the development of interactive graphics capabilities, which not only minimized model simulation time, but also eliminated many sources of potential error or inaccuracy, and facilitated the presentation of computed results, thus going a long way towards optimizing the total effort and cost of the investigation.



MODE 6 FREQ = 7.742 Hz
MODE SHAPES FOR FULL LOAD CONDITION

Fig. 17—*Mode shapes for full load condition (mode 6 frequency = 742 Hz).*

Intelligently used, the finite element method, together with the considerable experience gained in its practical use, presented a valuable basic design method. As such, it deserved greater consideration than it had been given in recent years. When used in conjunction with carefully prepared and executed fullscale investigations, designed to identify undesirable modes of behaviour, as for example, that described by the author, it became a powerful tool for prediction and elimination of many current problems experienced in marine engineering and naval architectural fields.

Mr. S. C. W. Wilkinson commented that from the seal maker's point of view, the absence of accurate information on shaft movements in various classes of ships presented a major obstacle to a well based development programme. The information provided by Mr. Crombie and his colleagues, therefore, fulfilled a much needed requirement for the improvement of seals on such installations.

In 1961, his company had endeavoured to establish criteria for shaft movements which had to be met, this being essential for the design study that we were carrying out on the feasibility of adapting radial face seals for large marine propeller shafts. Suggested specifications varied from ±25·4 mm (±1″) at one extreme to ±0·254 mm (±0·010″) at the other whilst, typical of customers' ideas, was one case in which 12·7 mm ($\frac{1}{2}$″) movement was predicted on a long shafted ship when an investigation on board showed only 3·175 mm ($\frac{1}{8}$″) clearance on a trailing collar.

During the subsequent ten years up to 1974, the company were recording actual measurements taken from a wide range of ships, but these were questionable in accuracy because of the difficulty in obtaining information under all conditions of service. Mr. Crombie's paper related to a supertanker giving a total shaft outboard movement of between 6 mm and 7 mm. By comparison, readings taken on a small refrigerated ship gave a total movement of 14 mm, a further indication of unexpectedly large variations in seal length. Mr. Crombie's conclusion that reference to axial shaft movement was misleading was very pertinent because the conclusion that his company had come to was that most of the movement was related to the hull, and that, for example, the combination of an amidships engine room and frozen cargo space could have led to movements which were of very great consequence to seal and bearing performance.

The accommodation of these movements did not necessarily imply a change or modification in the seal design. Their greatest relevance was in establishing the as-fitted length. To illustrate this point: in a ship in which all the known predicted movements were shaft outboard, some of which had only occurred at launching, it was desirable to initially instal the seal in a shorter space, thus achieving the correct working length for the average working condition in the ship. For this reason, the only modification made to the seal in the case of the ship with the 14 mm movement was to allow for 7 mm greater compression.

The seals which had been developed were now designed to accommodate these degrees of movement and the pressure pulses which could result from them. The pressure pulses mentioned in the paper compared well with some recorded in smaller ships by a special instrument that the company had developed to monitor oil pressure within the outboard seal. These recorders, based on a pressure transducer, simply registered the number of times a preset pressure was exceeded and, in the case of seals designed for a maximum of 0·2 N/mm² (30 lb/in²), a monitoring pressure of 0·4 N/mm² (60 lb/in²) was commonly exceeded which in a typical case was more than fifteen times in a year.

In parallel with the test ship referred to in this paper, two other 230 000 tonne vessels were fitted with 1473 mm (58″) diameter seals, i.e. 254 mm (10″) larger than those in the paper. All these units had given trouble-free service, exemplified by the first of the 58″ units to be examined after two and a half years in which, during the six months prior to docking, total loss from the system was only 20 gallons. The seals were found to be in such good condition that they were reassembled untouched after which their performance had continued to equal that up to the time of refit.

The greater understanding of the complicated patterns of movement which this paper described should inevitably lead to recognition of the fact that the radial face principle for stern shaft sealing had a great deal to offer in the case of large ships. It had taken more than ten years of experience to justify this and to reach the stage at which conversions to face seals were being considered, the first example of which was at present being manufactured for installation.

The much more accurate information now provided by Mr. Crombie in this paper would certainly make it possible to ensure that these improved seals were fitted in such a way that they could give optimum performance, accommodate all the variations in shaft position, and retain and exclude pressure without detriment.

MR. R. C. BECKETT, F.I.Mar.E., said that the statistics and supporting technical information highlighted the current problems being experienced in ships stern bearings and sealing devices for the protection and lubrication of the tailshaft. It was considered that at least once in the life-time of a ship (20 years) she would suffer sterntube troubles.

There were many possible causes which could lead to trouble and possibly to failure of either the stern bearings or seals, or both. A number of such causes were indicated from the results of the test programme which had been carried out and ultimately would be of benefit in terms of "feed back" to manufacturers of sterntube appliances.

Low temperature operational conditions had not been reported as a cause of sterntube appliance failure, provided a suitable lubricant was employed.

The lubricant, whether a "conventional" i.e a straight mineral oil, or one of the "specialized" i.e. emulsifying oil — (with its ability to protect metal parts in the presence of sea water, thus avoiding the possibility of corrosion) was one of the important components of the stern sealing device, the other one being the sterntube lubrication system.

During the test programme were lubricant "samples" taken at any stage for analysis, considering the possibility of the presence of sea water in the sterntube?

Attention was drawn to some instances of VLCC which had sterntubes pressurized to much higher figures than would be considered appropriate for LIP type rubber seals and which could be regarded as pressure sensitive. Naturally, this high pressure prevented ingress of sea water entering the sterntube. But unfortunately this very often resulted in excessive oil loss through the seals.

These two important components of the "trinity" namely the sterntube lubrication system and the lubricant, when under operational conditions were also influenced by shaft attitudes, rotational speeds, and oil viscosity.

DR. J. M. CONWAY-JONES congratulated Mr. Crombie for the great quantity of data on a subject about which there had been much theorizing, but little hard information. Dr. Conway-Jones hoped that the author would clarify two points. Firstly, would the author state the diameter, length, and diametrical clearance of the bearing, and secondly were the probes, for which oil film clearance data was presented, positioned aft or forward? If the data was from the aft probes, would the author comment upon the reason for an oil film clearance of 0·3–0·5 mm at zero rev/min as in Fig. 8?

Did this indicate that the journal rested upon the forward end of the stern tube bearing, and hence that there was considerable misalignment in both the loaded and ballast condition? If this was so, it clearly indicated the desirability of being able to align the bearing to the shaft, when the ship was afloat and in the ballast condition.

MR. W. MARSH, F.I.Mar.E., in a contribution read by Mr. R. G. Boddie, congratulated Mr. Crombie for his well thought out series of tests to establish by measurement the relation between tail shaft and stern bearing during actual operation under various ship loads and operating conditions.

This was most important since it was unlikely that a shore based test rig could reproduce the complex conditions which occur at sea.

Table I, listing the incidence of failure of stern bearings and oil seals certainly indicated the need to improve the reliability of this part of the ship.

Reliability was usually measured by its reciprocal, the rate of failure, but in this case it was easier to comprehend the facts if one tabulated the mean time to failure, i.e. the number of ships times the average operating time divided by the number of failures.

Using the information on Table I of the paper the failure rate was given as the number of ship-years between failures. This showed that the mean time between failures, i.e. the reliability, varied remarkably between the various ship types, so far as the stern bearings were concerned. Much could be gained by concentrating future work on the bulk carriers and the larger tankers.

Correspondence

MR. C. H. SHAW, F.I.Mar.E., noted with interest Mr. Crombie's remarks on lubricating oil pressures and consequent pressure loadings on stern gland seals. The author's figures of differential pressure were indeed in excess of maximum recommendations for the old form of bellows type lip sealing rings.

At a pressure of approximately 0·3 N/mm^2, this sealing ring "trailing edge" was pressed entirely flat on to the shaft liner, friction losses peaked and thermal deterioration of the ring was considerable. Beyond this pressure, the "leading edge" actually tended to lift away from the liner, and permitted an oil wedge to establish a leakage route under the ring.

Consequently the bulb sealing ring was developed with a view to tolerating the increasing pressures. Sealing ring deformation under increasing pressure was then markedly reduced and, as pointed out by Mr. Beckett, the ring was self compensating since higher pressures would have sealed the ring tighter against the shaft liner. Due to the reduced landing area of the ring, frictional losses were approximately halved, and therefore, heat deterioration of the ring was much reduced. Indeed the bulb sealing ring showed no likelihood of inadequacy to current marine requirements.

It would have been of interest to know the author's opinion on whether the high pressures recorded might have been traced back to the inherent pumping action of the bellows piece during transitory axial movement of the shaft. This coupled with the policy of omitting the oil grooving in the bearing, thus restricting the free flow of oil along the bearing, might have caused a pressure build up which would not have occurred in the conventional lip seal/bearing arrangement.

Author's Reply

In reply to Mr. McKinlay, the author firstly emphasised that, although the test programme was carried out on a particular design of split stern bearing, there was no reason to believe that similar results would not have been obtained from other stern bearing and conventional stern tube designs.

Whilst agreeing with Mr. McKinlay that enormous advances had been made in calculation techniques it was nevertheless true to say that theoretical calculations were usually based on the assumption that ships were built to design dimensions which was unfortunately not true. Apart from the fact that it was almost impossible to achieve such accurate tolerances on large steel fabrications, there was also the problem of thermal expansion due to the weather conditions prevailing at the time of construction. Because of these factors it seemed unlikely that design dimensions could be adhered to, and there must therefore be some degree of error in calculations based on design figures.

He pointed out that only the axial proximity transducer, as shown in Fig. 4, was visible from within the vessel. However, it was one of the bonuses of a withdrawable stern gear that fitting of instrumentation outside the bearing could be carried out from within the vessel, afloat, without extensive drilling of holes and at very little cost.

Mr. McKinlay had correctly assumed that Fig. 8 referred to measurements at one end of the bearing only, but they were for the aft end, and not the forward end, as suggested. Two radial proximity transducers were fitted at each end of the bearing since the end to end variation was indeed very important. Unfortunately, one of the transducers at the forward end did not record at all, and it was not possible to fit another because this would have interfered with the ship's schedule. It was not possible to calculate the shaft position, from the output from one transducer. However, the output from the one working transducer was enough to be able to say that the shaft was not resting on the bottom of the forward end of the bearing.

Mr. McKinlay asked for clarification about what caused such large variations in shaft attitude. Unfortunately, the author could not answer this but it was probable that all the items referred to by Mr. McKinlay made some contribution. The test programme was carried out to establish the nature and magnitude of some of the more important items influencing stern bearing design and reliability. It was believed that this objective was achieved. However, the test programme was not intended to explain what caused the variations to occur. Further research and development was undoubtedly required to answer this question.

The author was grateful to Mr. McKinlay for his explanation and comments regarding the shaft running in the starboard side of the bearing at full power. It was interesting to note that groove damage had, in fact, been noted by his organization.

It might have been possible to estimate thrust movement under load from the test results although this was not done since, as previously mentioned, the test programme was concerned with the conditions prevailing at the stern bearing. The reasons for the shaft movement variations were not considered in detail, since it was thought likely that similar effects would be present on the stern bearings of similar vessels. However, an approximation to the thrust movement might be made as follows. The variation in shaft axial movement from ahead to astern in ballast was about 4 mm. (see Fig. 9). The calculated shaft compression due to ahead thrust was 0·53 mm, as mentioned in the paper. The calculated shaft expansion due to astern thrust was 0·26 mm. and the design thrust clearance was 1·0 to 1·5 mm. The theoretical variation in shaft axial position from ahead to astern was, therefore, between 1·79 and 2·29 mm, i.e. about 2 mm less than the measured variation. This might be a near approximation to the thrust movement.

Although Mr. McKinlay was correct in saying that many vessels had to be aligned in the light ship condition, it was important to remember that this was usually because conventional stern tubes were fitted. With the conventional stern tube it was virtually essential to line out shafting from the stern tube forward, finishing up with the main engine. Since chocking a main engine was a lengthy job it was, therefore, necessary to complete alignment at an early stage of fitting out, i.e. at light ship. However, if an adjustable stern gear was fitted it was possible to complete main engine chocking prior to final shaft alignment which could then be left until much later in the schedule. Final alignment could be completed in 24 hours, and could easily be carried out at ballast draught. With a shafting system which was fixed at both ends, i.e. a stern tube at one end and a main engine at the other, only a limited amount of re-alignment was possible by adjusting the plummer block bearings and it might not be possible to correct misalignment, except by the very expensive methods of reboring the stern tube, or moving the main engine.

Replying to Mr. Joyce the author advised that the tailshaft diameter was 890 mm and the maximum shaft speed was 90 rev/min. He agreed that peripheral speed was relevant when comparing different bearings, but it was not thought to be of prime importance in this instance since the peripheral speeds of stern bearings fell within fairly narrow bands, due mainly to the fact that shaft diameter was fixed by Classification requirements. In addition, since stern bearing lubricating oil temperatures were either uncontrolled, or manually controlled, fairly wide variations in operating temperatures, and hence oil viscosity, could occur thus reducing the effects of small changes in peripheral speeds. Also, since stern bearing oil pressures were governed by sea water pressures they were, therefore, reasonably constant for similar sized vessels. In vessels in service fitted with the author's company's stern bearings peripheral speeds varied between 3·9 and 4·9 m/sec for vessels varying in size from 4000 to 280 000 dwt.

The author's company had in service plain, split stern bearings of 940 mm diameter, transmitting 37·28 kW (50 000 hp) and they had been completely trouble free during five ship-years operation. Since this bearing system was surrounded by air a pumped, circulatory lubricating oil system was used so that the oil could be cooled if necessary. This forced lubrication might, of course, assist in lubricating a bearing without oil grooves. If a system with only a head pressure and no oil flow was fitted then the bearing operation might not be trouble-free. The author's company would certainly offer plain bearings for any application of their split stern bearing. A further important factor for trouble-free operation could well be the alignment capability of adjustable stern gears since this allowed more precise shafting alignment to be achieved.

Regarding the oil pressure in the outboard seal oil space, Mr. Joyce's assumptions of lower temperature leading to increased viscosity were correct. Why this had not led to increased pressures might have been due to the finite time lag between the pressure peak being reached and the transducer sensing it through the hole drilled in the top half bearing (see Fig. 5). Since the pressure pulsations built up over a very short time period (perhaps less than 0·1 seconds), and it was likely that the time lag increased as oil viscosity increased, it might be that the pressure started to dissipate before the transducer "saw" the peak pressure thus giving the anomaly of lower pressure with thicker oil. When the test programme was being researched attempts were made to find pressure transducers which could be fitted directly into the oil space but, unfortunately, these just did not exist. The author's company were aware that similar pressures had been recorded in a completely different type of vessel — on an 8000 dwt reefer ship (as mentioned in Mr. Wilkinson's contribution). The pressure generated in the oil space was proportional to the change in oil volume (dv) divided by the original oil volume (v) and any vessel with a similar dv/v could be expected to show similar characteristics.

Oil temperatures were not measured by specially installed test equipment but were noted on the standard ship's equipment as a check that the vessel was operating normally. The lubricating oil temperatures at the bearing outlet varied between 39°C and 48°C during the test period while the bearing metal temperature varied between about 38 and 42°C with the aft end being generally 1–2°C higher than the forward end of the bearing. The temperature rise through the bearing varied between about 10 and 14°C for shaft speeds varying between 60 and 90 rev/min. No difference was noted between the temperatures for ballast and full load operation. These were quite normal operating conditions in the author's experience. The sea water temperature varied between 18 and 35°C during the test period but, since the bearing inlet oil temperature was controlled manually, there was, therefore, no relationship between sea and bearing oil temperatures. The accuracy of these temperature readings would be that expected from normal ship's equipment.

Dr. Goodman's comments and the slides he had shown were extremely interesting although the author could not claim that he fully understood them. It was certainly encouraging to learn that theory and practice were so close in those cases investigated.

The author would agree with Dr. Goodman that sophisticated techniques such as finite element methods seemed to have been somewhat neglected but, while this was perhaps unjustified, it was always difficult to justify expenditure such as the £25 000 quoted by Dr. Goodman when, in fact, the benefits might not be immediately apparent (if they were present at all). The author had been faced with exactly the same problems when justifying the use of adjustable sterngear which, although acknowledged to have several advantages over conventional stern tube bearings, normally cost more and the potential benefits were not easily quantified.

Additionally, as mentioned in the reply to Mr. McKinlay, the dimensional variations inherent in shipbuilding could lead to errors in theoretical calculations. Shafting problems might also arise due to the physical difficulties of achieving a theoretical shaft alignment in practice. Another problem was that vessels were known to "shake-down" after a period in service, effectively altering the shaft alignment. Even if this effect could be predicted by theory it was clearly impossible to position the shafting so that it was correctly aligned in both the original and subsequent condition. The use of adjustable stern gears could alleviate, if not eliminate, this problem for which, in the not unknown cases of very bad settling down, on a stern tube installation the alternatives would be rechocking the main engine, or re-boring the stern frame.

The wide range of guesses of axial shaft movement given to Mr. Wilkinson some years ago were of great interest and were typical of the lack of knowledge present in this area of ship design, hence the necessity for further test programmes similar to that in the paper.

It was interesting to note that Mr. Wilkinson's company also found that hull deflexion was significant.

The author agreed that it was more important to establish an "as-fitted" seal length, rather than a design length, so that the seal compression was correct when the vessel was in its working condition. The problem was that the seal manufacturer, and also the stern bearing manufacturer were in the hands of the shipbuilder, and depended on him for details of estimated variations in the ship/shafting combination. They, therefore, had little control over a situation which had significant effects on their equipment, and if the correct information was not forthcoming, for any reason, there was little they could do.

The lubricating oil pressures of 0·4 N/mm^2, recorded in the small vessels, indicated that the pressure pulsations measured on the test vessel were not unusual and were not, apparently, related to the size or type of vessel.

As Mr. Beckett had stated, it was generally accepted that the approximate frequency of failure for stern tubes was about once in 20 years. However, this failure rate might be artificially depressed to a lower value than it would otherwise be because oil seals (the most vulnerable component) were often replaced at more frequent intervals than might be absolutely necessary, to coincide with a scheduled docking for hull cleaning and painting. The shipowner thus incurred higher overall maintenance costs, but gained an insurance against unscheduled maintenance and docking due to failure at a later date. This method might be the most acceptable way of working with the conventional stern tube, which was inaccessible, thereby eliminating any possibility of carrying out routine inspections. The author believed that an equally acceptable method was the use of split, withdrawable stern gears which allowed routine inspections to be carried out during loading and discharging of cargo. This might allow the life of stern bearing components to be extended, thus leading to overall economies.

The author endorsed Mr. Beckett's views on the importance of the lubricating oil. He believed that stern bearings were best served by emulsifying oils, which offered the kind of protection when sea water ingress occurred (which was probably in all stern bearings regardless of type) not possible with turbine or crankcase oils which were commonly specified for that duty. The author was, of course, referring to those emulsifying oils of similar viscosity to the turbine and crankcase oils, and not to the "old fashioned" type of tar-like oils still available. Unfortunately, low viscosity emulsifying oil was usually only available from specialist manufacturers, since the oil majors seemed to be either unwilling or unable to supply them. This could cause some problems with regard to worldwide availability. A further relevant point was that shipowners preferred to reduce to a minimum the different types of oils carried on board, and usually used the main engine lubricant in the stern bearing system, even though it might not be the most suitable.

Some details of the lubricating oil system had already been given in the reply to Mr. Joyce.

Lubricant samples were not taken as part of the test programme. The vessel was fitted with a coalescer filter in the stern bearing L.O. system, and it was normal ship's practice to run off and measure the amount of oily water in the drain tank twice a day. This gave a reasonably accurate estimate of the amount of sea water ingress and the author had checked this daily in the ship's log, to ensure that the vessel was operating normally. The sea water ingress was found to have been dependent upon the loading condition of the vessel, varying from about 0·6 litres/day in ballast to about 3 litres/day in the loaded condition, disregarding any variations due to sea condition and temperature.

Mr. Beckett's comments regarding the relationship between lubricating oil pressure and sea water pressure emphasized one of the problems of VLCC operation, i.e. the vast difference between ballast and loaded draughts which dramatically altered the pressure differential between the stern bearing lubricating oil pressure and the sea water pressure. This made the use of either more than one header tank or an automatic pressure control almost essential when lip-type oil seals were fitted. As mentioned in the paper, these lubricating oil pressure variations did not have much effect on face type seals except that the face load varied. It was thus possible to eliminate multi-header tanks. The test vessel was fitted with a circulatory lubricating oil system which operated at constant pump outlet pressure, regardless of draught variations.

Replying to Dr. Conway-Jones, the author stated that the diameter, length and diametral clearance of the bearing was 890 mm, 2225 mm and 1·4 mm, respectively.

Dr. Conway-Jones had mentioned oil film clearance and the author would like to clear up any possible misunderstanding about this. The clearance measured, and shown in Fig. 8, was that between the tailshaft and bearing surfaces. No measurement of oil film thickness as such was carried out, although it was quite likely that the oil film thickness might have corresponded with many of the clearances shown in Fig. 8.

Details of the probe positions had been given in the reply to Mr. McKinlay. Although it was possible to ascertain that the shaft was not lying in the bottom of the forward end of the stern bearing it was not unlikely that there might have been contact at some intermediate point up the side of the bearing.

It seemed likely that the 0·3–0·5 mm clearance recorded at zero rev/min was due to hull distortion probably caused by the different hull support conditions present at light, as compared with ballast draught. This did mean that, at ballast and loaded draughts, there was some degree of misalignment which was not present at light draught when the shafting was aligned. It was for precisely this reason that the paper advocated that alignment should be carried out at ballast draught, at least.

The author agreed with Mr. Marsh that, although shore-based test facilities were invaluable tools, only measurements taken at sea could give the true results.

Mr. Marsh's method of tabulating the failure rate was appreciated, since it was possible to present statistics in so many different ways. As Mr. Marsh had pointed out, however, it was clear that, no matter which method was used, bulk carriers and large tankers stood out as being the most vulnerable.

The author was grateful to Mr. Shaw for his description of the dynamics of lip seal ring operation. It was encouraging to learn that the latest designs had been developed specifically to handle the higher pressures more common today, although Mr. Shaw had not stated what these pressures were. It would have been interesting to learn whether these new designs would operate satisfactorily with the pressure differentials measured in the test programme (see *Lubricating oil pressure*, section (iv)).

As mentioned in the reply to Mr. Joyce, the change in pressure in the outboard seal oil space was proportional to the ratio dv/v. Although the transitory change in volume (dv) of a bellows type seal was clearly greater than that of a lip-type seal, it was also true that the original volume (v) of a lip seal was less than a bellows seal and, therefore, the ratio dv/v could conceivably remain constant, thus indicating that the peak pressures could be similar for both types of seals.

It was not possible to say whether the omission of the oil grooves could have been a factor which affected the value of the peak pressure. However, the author's opinion was that oil grooves (or the lack of them) were not of major importance since the pressure peaks occurred so quickly that, regardless of the clearance space in the bearing, they would not have been dissipated since the oil would not have been able to flow away fast enough.

As far as the bearing design itself was concerned, i.e. providing sufficient oil flow for both lubrication and cooling purposes, it was clear that there was no lack of oil flow since the bearing temperatures were low, and the bearing surface which was examined after almost two years in service showed no evidence of inadequate lubrication. Several similar designs had been in service for up to eight years, and they were also in good condition when inspected.

The advantages of smooth, circular bores devoid of oil grooving were well known from the point of view of building up an hydrodynamic oil film. They were also preferable for bearings where the direction of load varied continuously, such as ship's stern bearings.