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PERFORMANCE OF MARINE REDUCTION GEARS WITH MODERN DESIGN FEATURES

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A new type of marine reduction gear for two or more medium speed diesel engines has been developed, and was thus far applied to total ratings between 14 and 34 MW (20 000 and 46 000 hp). For these extreme ratings, modern design features have been introduced, which are: absolute symmetry of all essential structural elements, four point support, and cardan-shaft characteristics of input shafts. By these measures, certain problems were expected to be solved which have, from earlier experience, turned out to be of increasing significance with larger dimensions of ships and reduction gears. These problems are unequal load distribution over the face width and radial displacements of shafts. During trials, the performance of these modern gears was excellent. This was proved by special investigations, which confirmed that the above mentioned problems were indeed satisfactorily solved, and that marginal problems, which should always be anticipated with new equipment, did not appear.

INTRODUCTION

Among the essential aspects of present trends in marine propulsion, the advance of the medium speed diesel engine appears to be one of the most spectacular. Advantages are obvious, progress is continuing, and earlier drawbacks are being eliminated effectively. As one of the disadvantages, the need of a "vulnerable gearbox" was emphasized some years ago by an experienced author(1). It appears that the application of a reduction gear, which is inevitable for medium speed diesel drives, is considered to be a hazard, particularly from the point of view of the operator. Suspicion has been confirmed by the occurrence of breakdowns which have kept ships out of operation for several months.

The all over reliability of a single propeller plant with respect to failure of engines can be increased by the application of two or more engines. If one engine is out of operation, service can be continued with the others. However, a similar statement would not be true for the reduction gear. Although there will be two or more pinions, only one main wheel exists. Thus should only one of the pinions fail in the beginning, the main wheel and the other pinions will usually suffer consequential damage. It follows that the demand for utmost reliability of the reduction gear is greater than that of the diesel engines. Therefore, it is a challenge to marine gear manufacturers to supply and put into service reliable equipment.

To bring this task to a successful conclusion to the benefit and confidence of the ship owner, the following rules should be obeyed and consequently acted upon in proper sequence:

1) previous designs and experience to be carefully studied in general;
2) reasons for malfunctions and breakdowns to be investigated thoroughly;
3) principal solutions to problems to be established;
4) new designs to be created in accordance with these principles;
5) the efficiency of expected improvements to be verified by workshop and ship board tests;
6) potential marginal problems to be anticipated and eliminated, if necessary;
7) unexpected marginal problems to be detected in due course and eliminated readily;
8) originally satisfactory status during trials to be kept under control in future operation.

DESIGN FEATURES

As a solution to problems which will be explained later, a reduction gear with new features has been designed and successfully put into service. During this development, the above mentioned rules have been followed as closely as possible.

Detailed information on the design and its background has been given in an earlier paper(2). Therefore it may suffice to summarize only the major points.

The design is concerned with a multi-engine reduction gear for medium speed diesel engines. Fig. 1 shows the one which was manufactured first. It is a double-engine reduction gear for an Australian tanker. Its total rating is 14·7 MW (20 000 hp), and the centre distance between the engines or pinion shafts amounts to 4250 mm. The main distinction from conventional designs is that the casing is supported at four points only, which are located on the axes of symmetry. Also, all essential structural elements are absolutely symmetrical, particularly with respect to the lateral axis. This statement is true for the casing, the gear wheels and shafts, and for the arrangement of bearings. In order not to disturb the symmetry of the casing, a thrust bearing is not incorporated. It will be arranged separately further aft. For the same reason, double helical teeth have been applied, which will produce symmetrical forces.

Another remarkable feature is the arrangement of input shafts which are separate from the hollow pinion shafts and are not supported in bearings. At the forward end, each input shaft will be connected to the engine by means of a coupling which is flexible in torsional, axial and angular, but not in radial direction. The aft end, which carries a multi-disc clutch, is connected to the pinion shaft by a gear type coupling, which allows for axial and angular movement only. By virtue of the angular flexibility at either end, the input shaft assumes the characteristics of a cardan shaft. Therefore, the input shaft is capable of accommodating radial displace-

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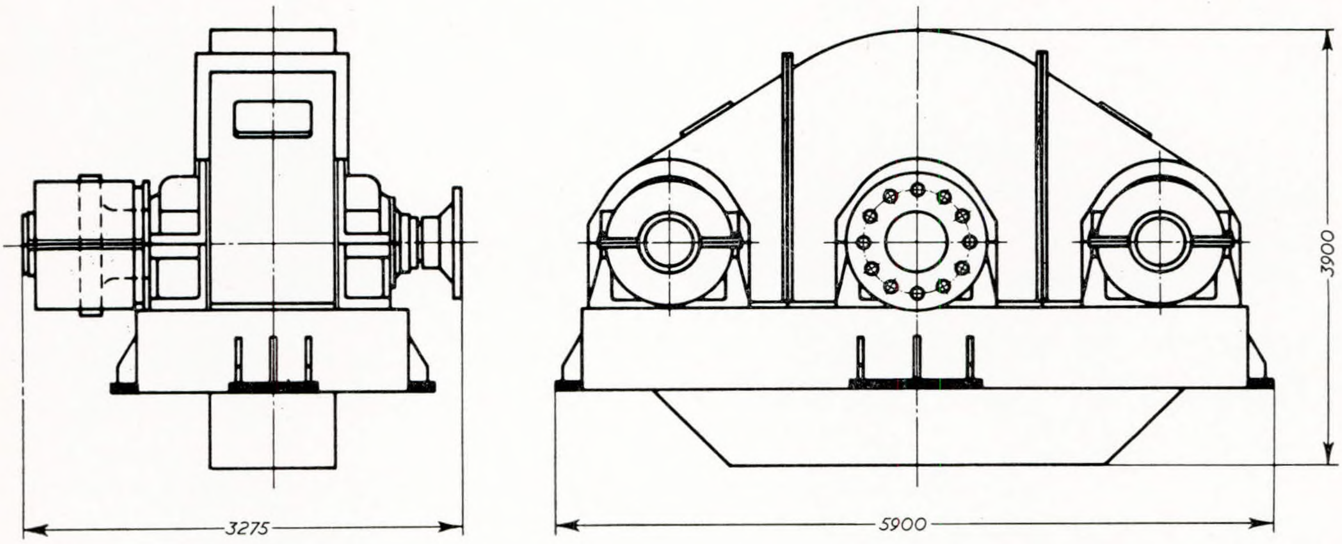


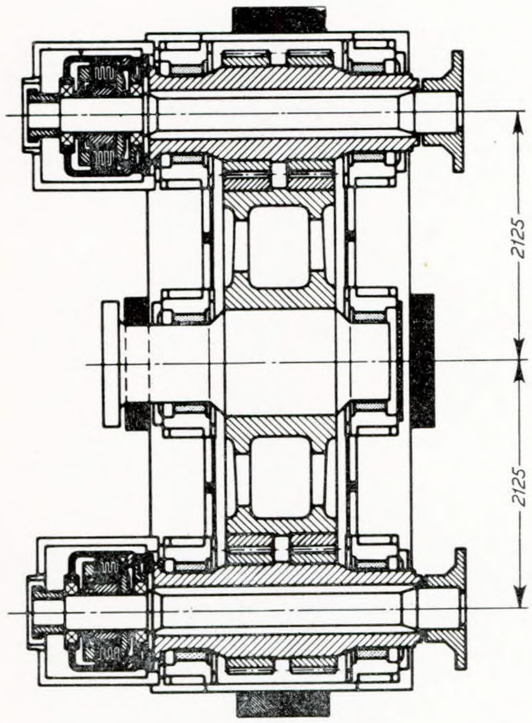


FIG. 1—*14·7 MW (20 000 hp) double-engine marine reduction gear of modern design*

ments up to 2 mm between engine and pinion shaft. This results in input shaft inclinations of about 1 mrad with no appreciable reaction forces. Fig. 2 shows the reduction gear close to completion in the workshop.

INSTALLATION ON BOARD

The arrangement of a similar double-engine reduction gear on board a refrigerating vessel is shown in Fig. 3. With only slight differences in size, the total rating is even larger, namely 17·2 MW (23 400 hp). Two medium speed diesel engines a, a′ are connected to the main gear c by two highly flexible couplings b, b′. The other units shown, namely the thrust bearing d, two generator speed-up gears e, e′, two other flexible couplings f, f′, and two 1920 kW generators g, g′ are all mounted apart from the main gear. Fig. 4 shows a side view including the propeller and the intermediate shaft, and also a section of the double bottom. As demonstrated more clearly by Fig. 5, the whole propulsion plant is located far to the aft end of the vessel, the narrow space available necessitating a compact arrangement of the individual units as described.

The supports of the machinery are arranged on the ship's structure as shown in Fig. 6. Particular attention should be given to the location and design of the four isolated supports 1 to 4 of the main reduction gear. As an extension of the principle of symmetry as applied to the design of the gear, the axes of symmetry coincide with the girders of the hull. As seen from the sectional views, each support is based on a separate structure. As a result, any support may be deflected individually without disturbing the deflexion of another, to be compatible with the deflexions of the gear. Also, the support b for the thrust bearing is designed as a separate structure so that deflexions due to propeller thrust will rarely be transmitted to the reduction gear.

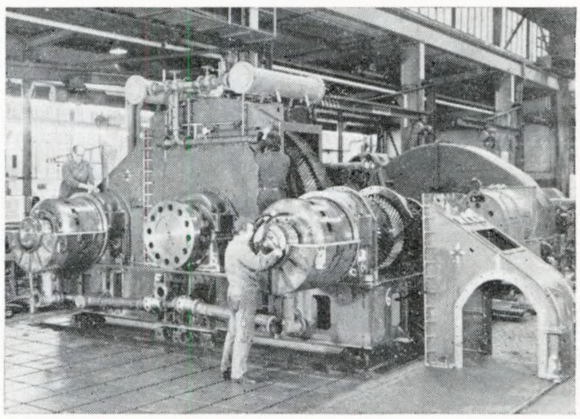


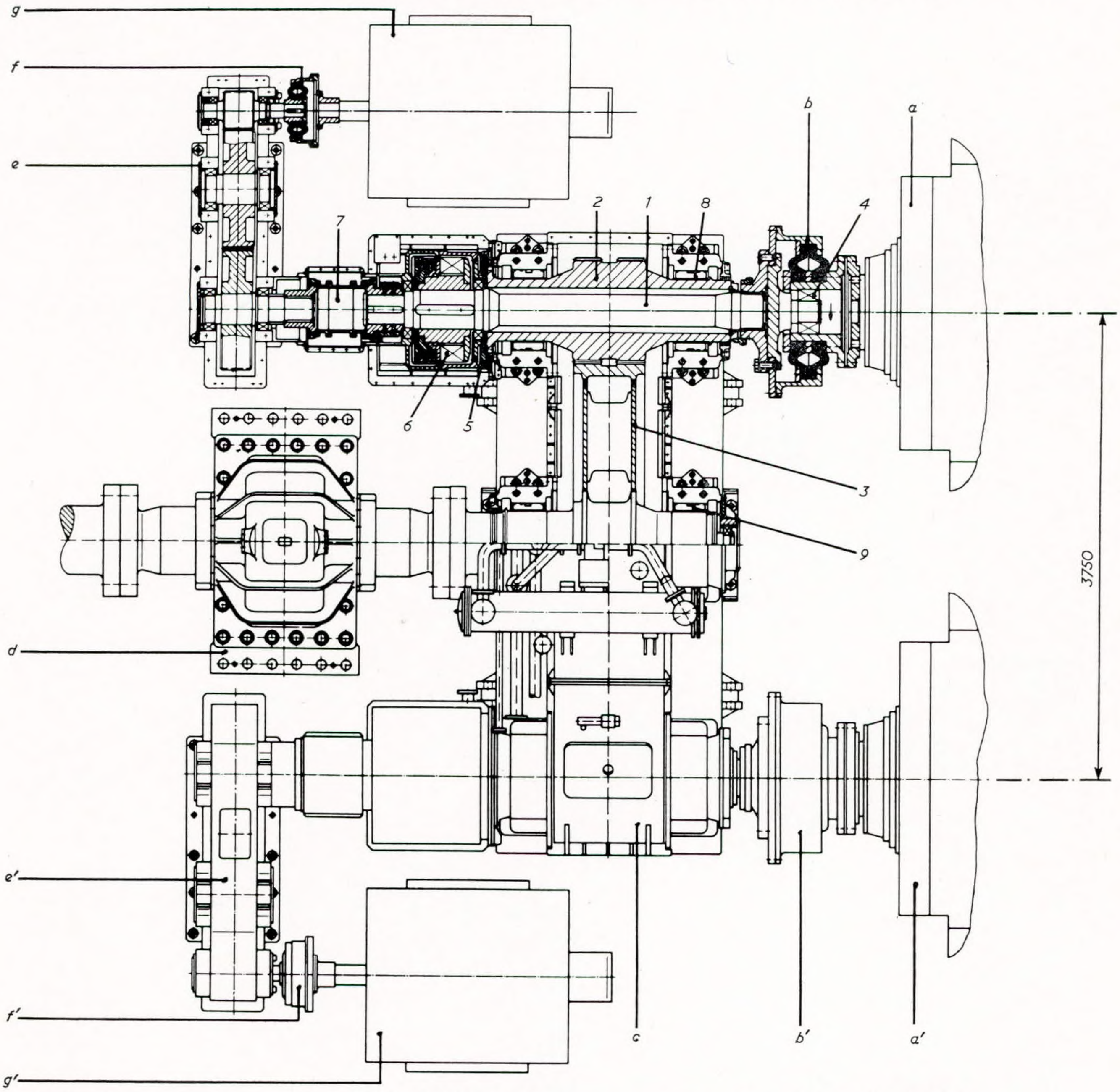
FIG. 2—*14·7 MW (20 000 hp) marine reduction gear close to completion*

APPLICATION TO A THREE-ENGINE DRIVE

The principles described for the design of a double-engine reduction gear and for its installation on board were also applied to a three-engine propeller drive, Fig. 7. With a total output of 34 MW (46 000 hp), this reduction gear is at the present time considered to be the largest ever to come into service on a merchant vessel with diesel engine drive.

SOLUTIONS TO PROBLEMS OF THE PAST

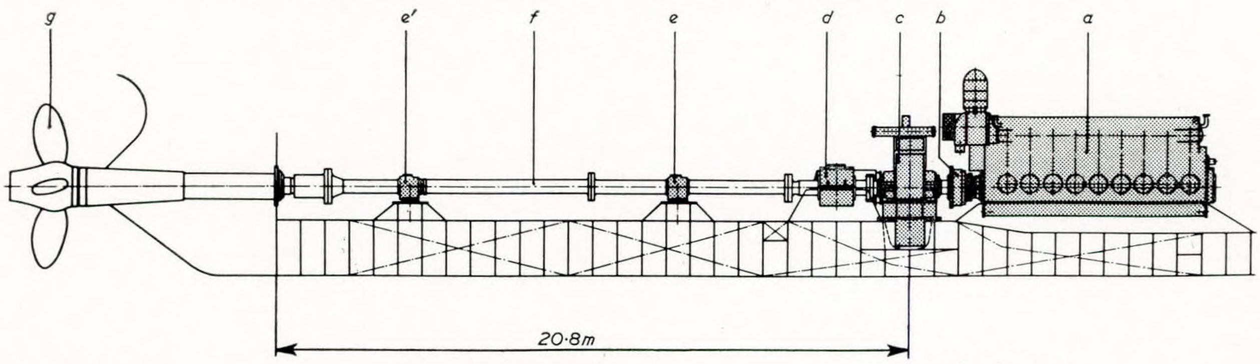
As mentioned before, the new gear design was conceived as a solution to problems which have caused some troubles in the past. Most of these problems were related to the fact that ratings and dimensions of marine reduction gears have increased rapidly during the last decade. Although basic considerations of various problems may be applicable



a, a′	diesel engines	b, b′	flexible couplings	c	main reduction gear
d	thrust bearing	e, e′	generator gears	f, f′	flexible couplings
g, g′	generators				

1 input shaft, 2 pinion shaft, 3 main wheel, 4 supporting bearing, 5 gear type coupling.

FIG. 3—*17·2 MW (23 400 hp) marine reduction gear installed with other machinery*



a	diesel engine	b	flexible coupling	c	reduction gear		
d	thrust bearing	e, e′	intermediate shaft bearings	f	intermediate shaft	g	V.P. propeller

FIG. 4—*Arrangement of machinery on the double bottom*

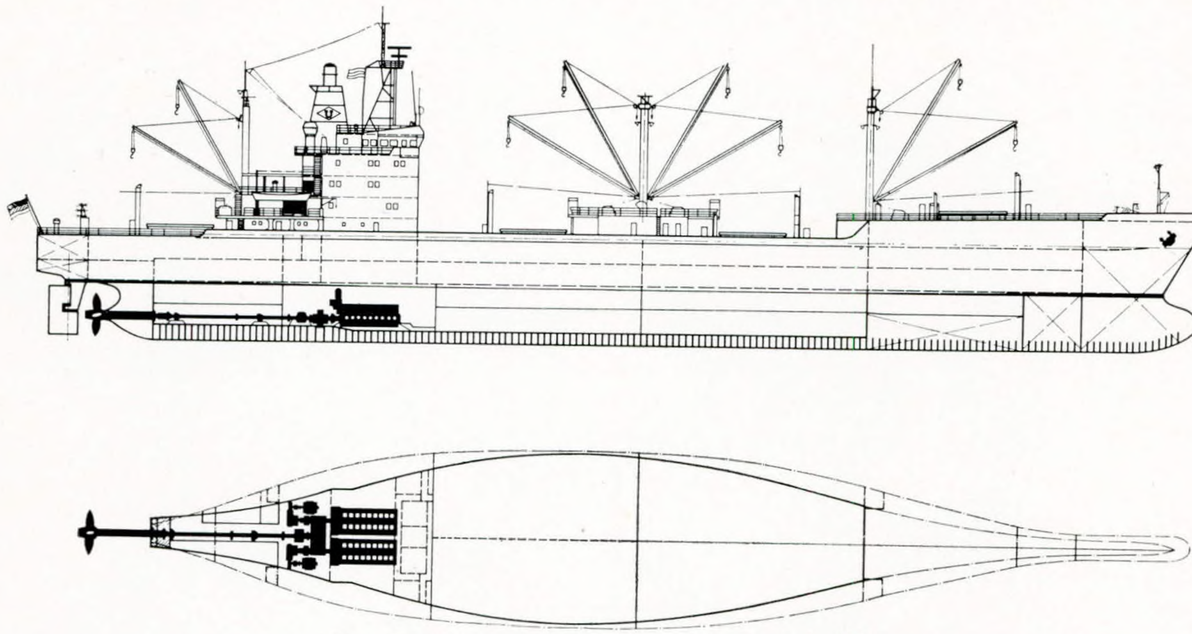
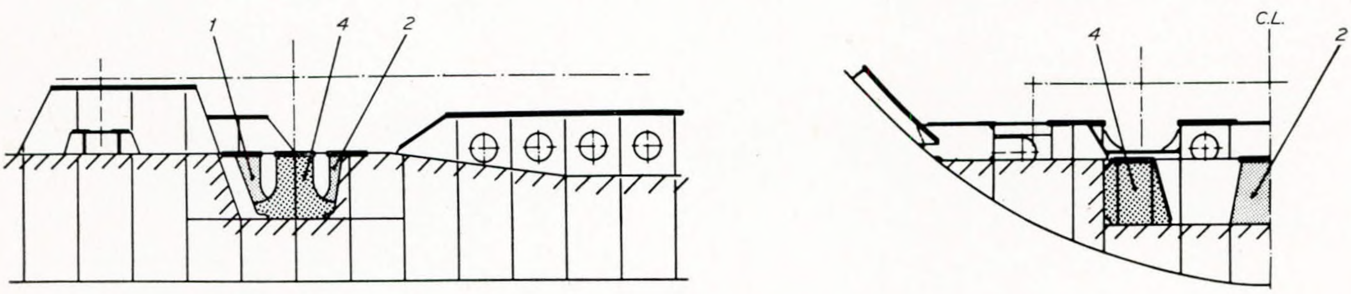
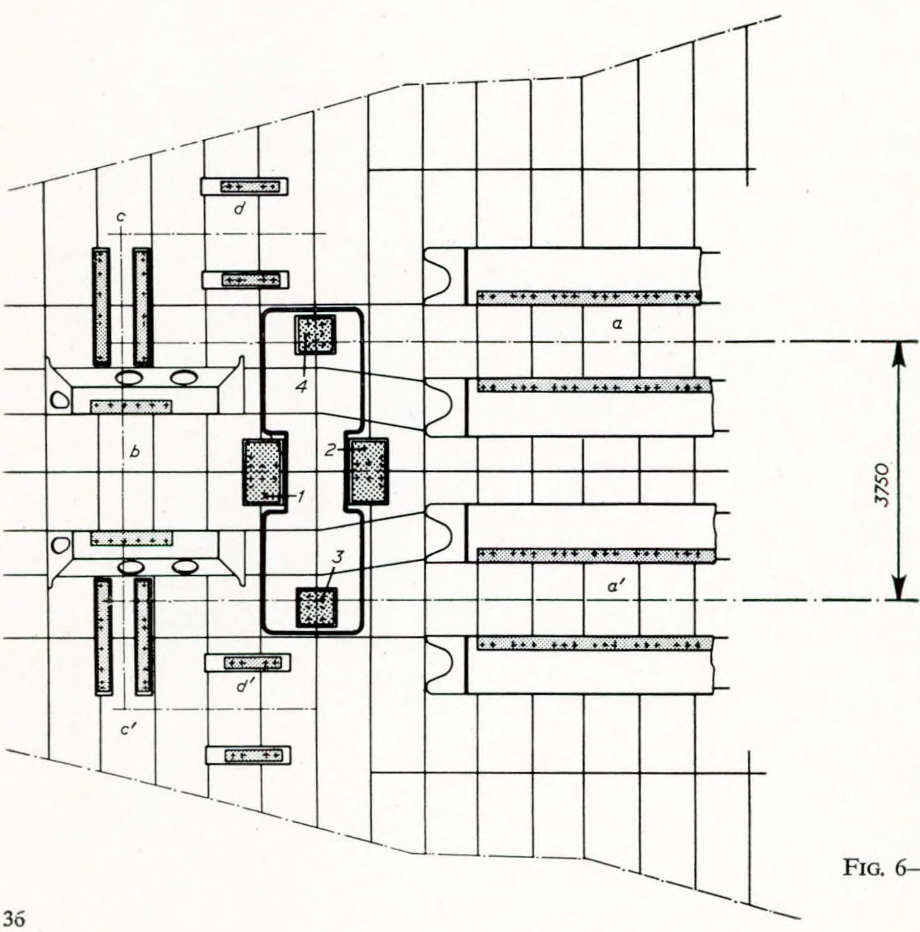


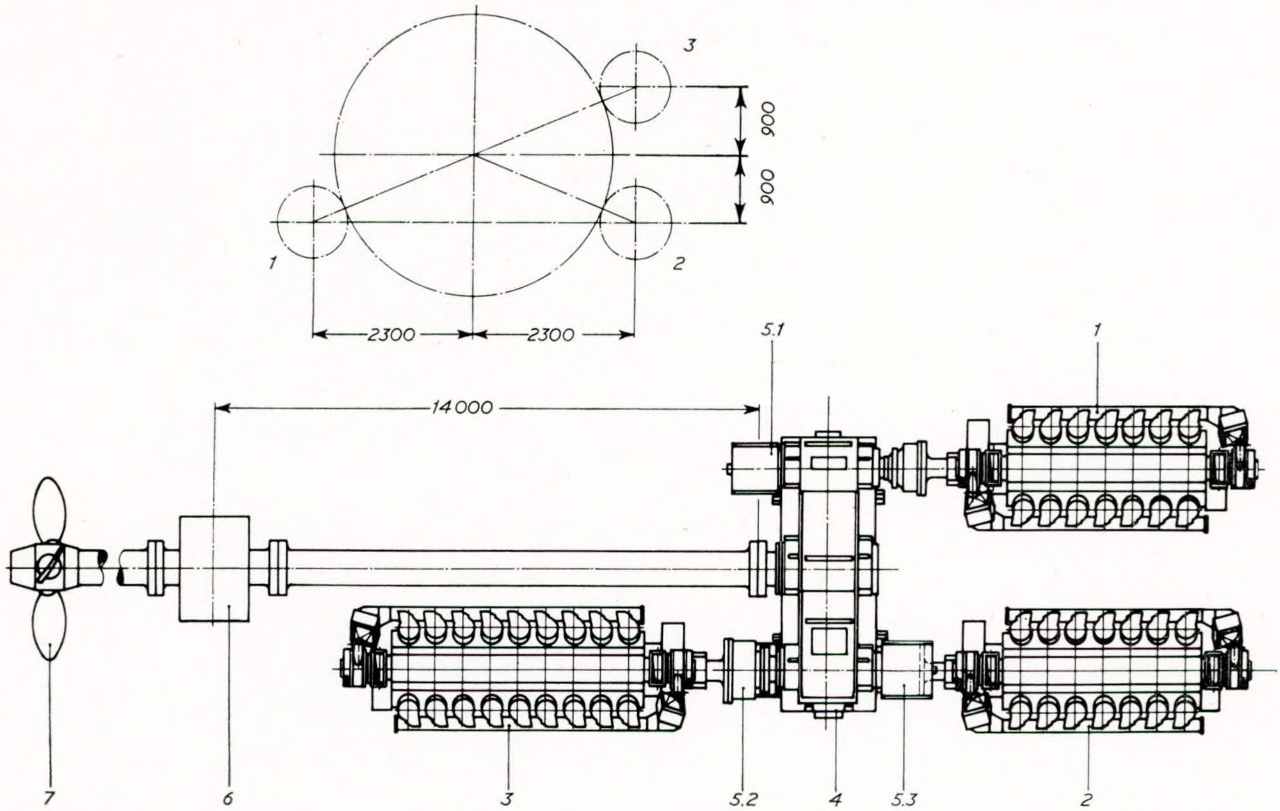
FIG. 5—*Refrigerating vessel m.s* Blumenthal, *1974*





a, a′ diesel engines
b thrust bearing
c, c′ generator gears
d, d′ generators
1 to 4 main reduction gear

FIG. 6—*Arrangement of machinery (top view)*



1, 2, 3 diesel engines 4 reduction gear 5 multi-disc clutches
6 thrust bearing 7 V.P. propeller

FIG. 7—*Propulsion plant with three diesel engines and 34 MW (46 000 hp) reduction gear*

to smaller units as well, they will assume practical significance only when dimensions are increased above a critical value.

By general experience with a great number of marine reduction gears, the following problems turned out to be of increasing importance:

1) unequal load distribution over the face width of the teeth;
2) radial displacements between the shafts of engines and reduction gear;
3) reactions between the reduction gear and the intermediate propeller shaft.

Before taking successful action against these phenomena, the problems had to be investigated in detail.

Unequal Load Distribution

Unequal load distribution over the face width has, in severe cases, caused overload at one end of the gear teeth and eventually led to fatigue failure by cracking. In the ideal case, tooth load per unit length may be expected to be the same over the whole face width. In practice, deviations from the ideal situation are quite normal and acceptable within reasonable limits. In an extreme case, the stresses could be doubled at one end and reduced to zero at the other, which would no longer be permissible. To establish a fair tooth contact, it has always been good practice to check the load distribution by painting several teeth with oil resistant lacquer and observing its abrasion by the action of torque transmission under service conditions. However, this method has its limitations. Better information is obtained by measuring the tooth rot stresses with strain gauges[3].

By the application of the strain gauge method it was found that the load distribution over the face width changed when the transmitted torque was increased. This can be explained by unsymmetric deflexions of the complex structural unit which consists of the gear casing and the ship's hull supporting the gear. As demonstrated in Fig. 8 for a pinion shaft, vertical froces F_1 and F_2 are acting on the bearings and the structure below. Asymmetric geometry of these elements will result in two different spring constants c_1 and c_2, which are effective at the locations of the bearings. Correspondingly, the deflexions f_1 and f_2 are different, and the pinion shaft will suffer a non-parallel displacement at increasing torque. The distribution of tooth root stresses changes accordingly. This is shown on the right hand side of Fig. 8, which was derived from strain gauge measurements on a particular vessel. Differences of bearing deflexions in the order of 0·1 mm are sufficient to explain the unequal distribution as shown by the upper curve which describes the full load condition. When compared to an ideal uniform distribution, the additional stresses at one end of the tooth amount to 60 per cent of the mean stress. At quarter load (lower curve) the stress distribution is quite uniform, which indicates that the original non-loaded condition of the reduction gear was almost perfect.

Apparently, displacements of the hull due to other sources (water and cargo load, propeller thrust, heat expansion, plastic deformations), again in the order of only 0·1 mm in difference, will have the same adverse effect.

Radial Displacements at the Engines

The second problem mentioned is the radial displacement between the crank shaft of the engine and the input shaft of the reduction gear. Fig. 9 shows the results of measurements obtained at 23 different occasions on one vessel[4]. Readings were taken in the customary way by dial gauges and slowly rotating shafts. The following parameters, which describe the various attitudes of the vessel, were variable:

1) cargo load (ranging from empty to fully loaded);
2) quantity of bunker oil (15 to 240 m^3);
3) sea water temperature (7 to 28°C);
4) engine room temperature (23 to 44°C);
5) motor oil temperature (20 to 52°C);
6) reduction gear oil temperature (22 to 48°C).

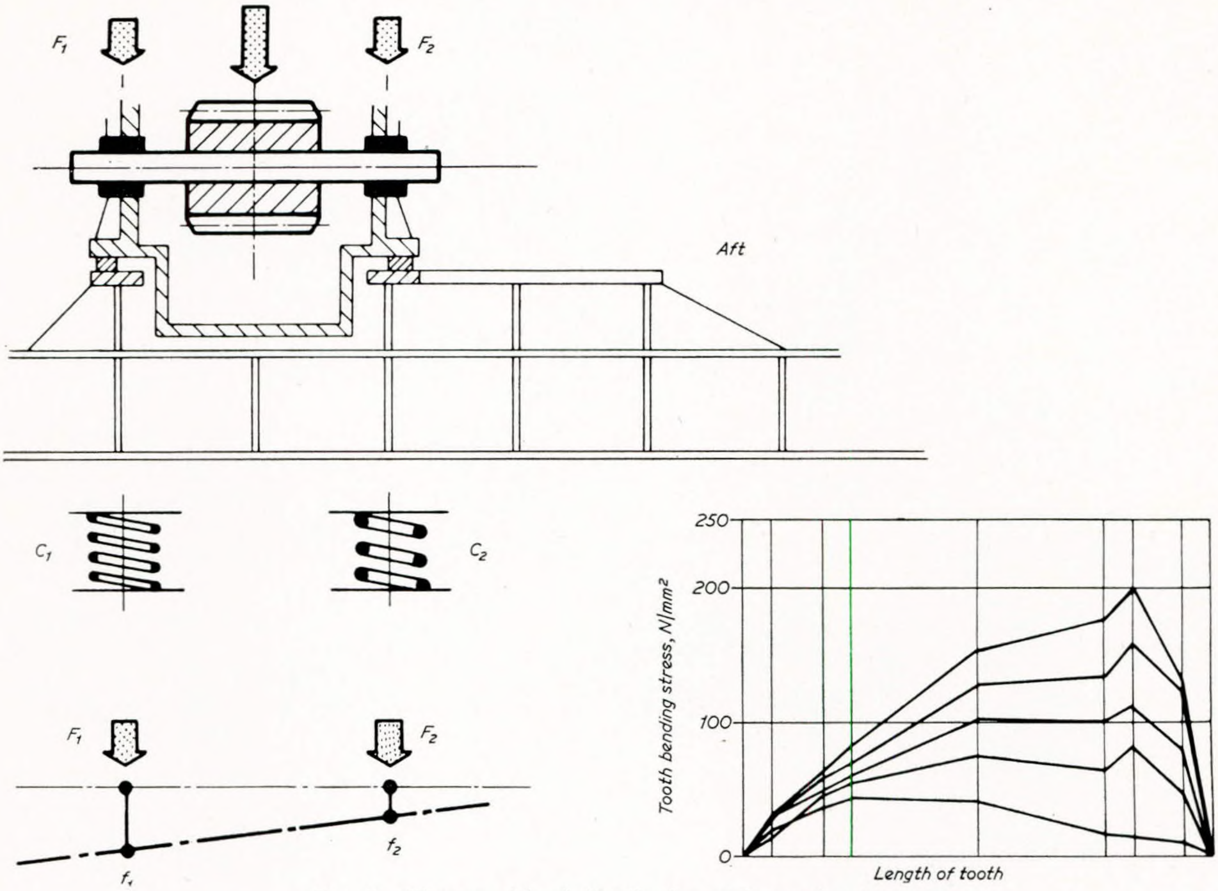


FIG. 8—*Change of load distribution at increasing torque*

As a consequence, the radial displacement turned out to be variable also. The points in Fig. 9 describing magnitude and direction of the radial displacement are scattered within a circular area of about 1 mm in diameter.

Points A correspond to the cold and unloaded condition. Points O describe the centres of the pinion shafts taken as reference points. On the basis of this analysis it is true that alignment could be improved by changing the position of the engine centres so that the reference points move from O to B, i.e. to the centre of the 1 mm circle. But the scatter within this circle would still remain.

Further radial displacements occur in the running condition, when bearing clearances allow additional shaft movements. Another investigation, where a rotating strain gauge pick-up was used in place of dial gauges, revealed that the additional radial displacement between the engine and the input shaft amounted to as much as 0·6 mm.

It is customary to provide a flexible coupling between the engines and the reduction gear. Some designs allow radial displacements up to 2 mm, which would be sufficient, others allow less. However, even if displacements are permissible for the couplings, reaction forces are inevitable. But

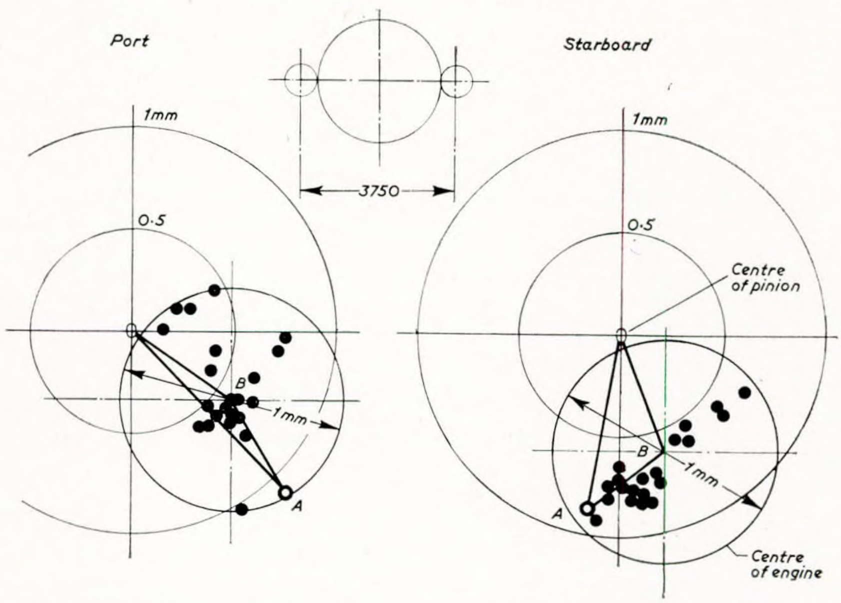


FIG. 9—*Change of radial displacement between engines and pinion shafts*

these reaction forces may not only cause overloads on the bearings, but also deflexions of pinion shafts which, in turn, will contribute to the development of unequal loads distribution over the face width.

Reactions at the Intermediate Shaft

The third problem, the reactions between the reduction gear and the intermediate propeller shaft, can also be described in terms of potential radial displacement at the point of connexion. With due consideration to the weight of the main wheel, the shaft, and the propeller, and to elastic deformations and heat expansions as well, the problem is usually solved by the aid of computer calculations and appropriate alignment procedures. One of the objectives is to arrive at bearing forces at the two main wheel bearings which do not differ too much from each other. Deviations will normally occur due to false estimates particularly for heat expansions. Therefore, it is not advisable to locate the first intermediate shaft bearing too close to the reduction gear.

forces acting on a symmetrical structure will produce identical displacements, and parallelity of shafts will be maintained. Also, if any one of the four supports is lowered or raised, the shafts remain parallel. These displacements of supports may be due to forces which originate from the reduction gear, or to movements of the hull itself. As a result, because of the undisturbed parallelity of shafts, tooth contact will not be altered by any of these influences. If a perfect load distribution over the face width is established in the beginning, it will stay like this.

The problem of radial displacement between engines and reduction gear is most effectively solved by the application of separate input shafts which act like cardan shafts.

Reactions between the reduction gear and the intermediate propeller shaft should be kept within allowable limits by arranging the first intermediate shaft bearing at a reasonable distance. Since the separate thrust bearing is normally installed closer to the reduction gear, the thrust bearing does not incorporate radial bearings, which would be in contradiction to the above mentioned principle.

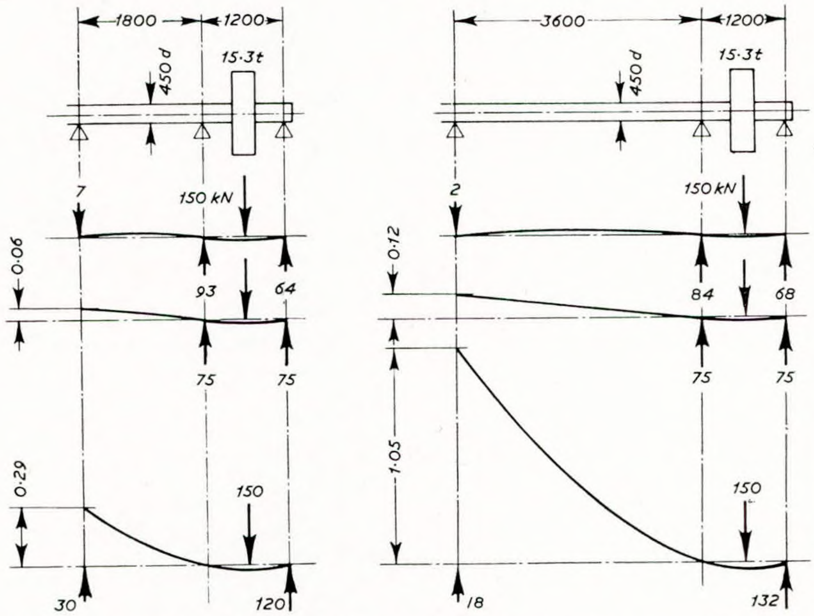


FIG. 10—*Reactions between main wheel and intermediate propeller shaft*

In the illustration on the left of Fig. 10, the bearing distance of 1800 mm is relatively small. With simplified assumptions, the weight of the main wheel will cause bearing reactions of 93 kN and 64 kN at the reduction gear, and of 7 kN at the first intermediate shaft bearing, provided that all three bearings are in line. This condition would be permissible, but it could be improved by raising the intermediate shaft bearing by 0·06 mm. However, should it be raised by 0·29 mm, which might easily be true as a result of false estimates or inaccurate performance of alignment work, one of the bearing forces would vanish. This would no longer be permissible. In the illustration on the right, the bearing distance is doubled and now amounts to 3600 mm. In this case, the bearing force at the reduction gear will vanish only after the intermediate shaft bearing has been raised by 1·05 mm. Thus a much larger margin of safety is provided by this arrangement.

Solution by the New Design of Reduction Gear

It was the objective of the design of the new reduction gear to bring the above mentioned problems to a satisfactory solution. As can be made plausible by theoretical considerations[5], the absolutely symmetrical design of the gear in conjunction with a four point support on the axes of symmetry ensures that the shafts of the gear always remain parallel to each other. First, this statement is true when deflexions by bearing forces are considered. Symmetrical

ANTICIPATION OF MARGINAL PROBLEMS

Whenever a new design of machinery is developed, particular attention is paid to well-known problems which are to be solved. Special features are introduced in view of this objective, as it was in the case of the multi-engine reduction gear described. One is even prepared to expect that the success of the measures taken will undoubtedly be verified in practical performance. However, any new design feature may turn out to be the source of new problems. In case these marginal problems should make more trouble than the old problems did before, no real progress would be established. Therefore, it is essential to anticipate as many of these marginal problems as possible.

Double Helical Teeth

In the case of the new reduction gear, double helical teeth had to be applied rather than single helical, for the sake of symmetry of forces. Although double helical gears are well proven with marine gears, particularly with turbine gears, it is sometimes claimed that for diesel engine applications a single helical gear would be better, because satisfactory load sharing between two helices might become a problem. Also, helical angles at the two helices are never exactly the same. Therefore, whenever one helix should have equal load distribution, the other helix would not. On the other hand, when shaft displacements have to be expected, double helical teeth will react less sensitively because the effective face width in this respect is that of one helix only, which is only half the value of a comparable single helical gear. To arrive at a good judgement, it is necessary to determine the practical relevance of the individual components mentioned.

Fig. 11 explains the situation schematically. Suppose that the tooth load is measured by strain gauges at four locations of a double helical gear as indicated to the lower left. Four values of tooth root stresses will then be obtained, σ'_1, σ'_3, σ''_1 and σ''_3. In the ideal case of equal load distribution, the stresses would be the same:

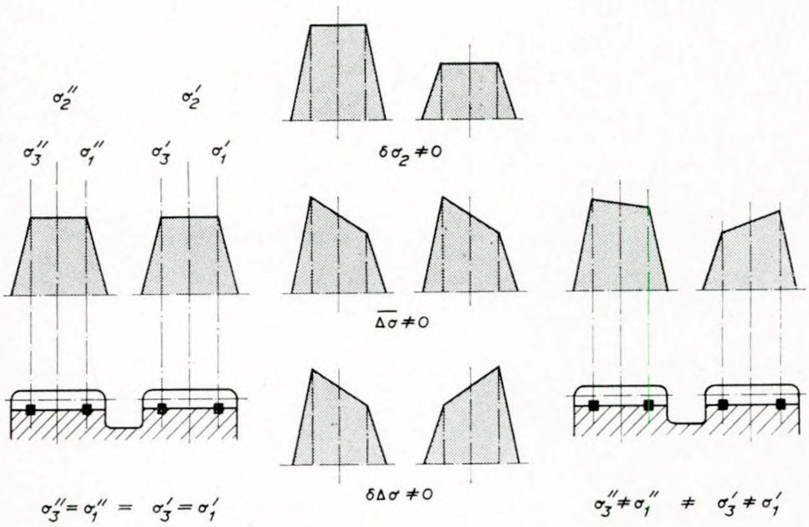


FIG. 11—*Additional stresses of various nature at a double helical gear*

$$\sigma'_1=\sigma'_3=\sigma''_1=\sigma''_3=\bar{\sigma}_2 \tag{1}$$

$\bar{\sigma}_2$ being the mean stress. Additional stresses of different nature are demonstrated in the centre of the illustration. The additional stress $\delta\sigma_2$ is due to unequal load sharing between the two helices, $\Delta\bar{\sigma}$ is due to deviations of shaft parallelity, and $\delta\Delta\sigma$ relates to differences of helical angles between the two helices. The summation of mean stress and additional stresses leads to four different stresses at the points of measurement, as shown in the diagram on the right, namely:

$$\begin{aligned}
\sigma'_1&=\bar{\sigma}_2-\delta\sigma_2-\Delta\bar{\sigma}+\delta\Delta\sigma\\
\sigma'_3&=\bar{\sigma}_2-\delta\sigma_2+\Delta\bar{\sigma}-\delta\Delta\sigma\\
\sigma''_1&=\bar{\sigma}_2+\delta\sigma_2-\Delta\bar{\sigma}-\delta\Delta\sigma\\
\sigma''_3&=\bar{\sigma}_2+\delta\sigma_2+\Delta\bar{\sigma}+\delta\Delta\sigma
\end{aligned} \tag{2}$$

When σ'_1, σ'_3, σ''_1 and σ''_3 are known by measurement, Eq. (2) can be solved for the components $\bar{\sigma}_2$, $\delta\sigma_2$, $\Delta\bar{\sigma}$ and $\delta\Delta\sigma$.

Axial Movements of Shafts

A double helical gear requires free axial movements of shafts. For this and other reasons, all the shafts of the new reduction gear do not have axial bearings. The axial position of the main wheel is determined by the axial position of the intermediate shaft which is supported at the thrust bearing. Thus the main wheel can freely move when propeller thrust or heat expansion causes it to do so. The pinion shafts will then follow the movement of the main wheel because they are guided by the double helical teeth. The axial position of the input shafts, however, will be different. It will be determined by the crank shafts of the engines and the flexible couplings to which the input shafts are connected. The gear type couplings which connect the input shafts to the pinion shafts are expected to compensate the differences between the axial movements of these shafts. However, the movements may also include vibratory components. In addition, at the gear type couplings, axial friction forces may be induced. As a whole, it is not easy to exactly predict the axial movements which will really occur.

Other Anticipations

Other problems to be anticipated are related to torsional vibrations, to vibrations of the gear casing, or to noise. In particular, vibrations of the gear casing might be expected because a structure supported in four points only may be lacking the required stability.

PERFORMANCE DURING TRIALS

A definite statement on the success of a new development can never be made before the first unit has been manufactured, tested, and its superiority proved under service conditions. Adverse effects, which a certain design claims to eliminate, may still be present to some extent in practice. It is just as important to make sure whether marginal problems will still exist. Therefore it is necessary to discover this as soon as possible.

Preliminary Tests

The first reduction gear of the new type to be manufactured was carefully tested in the workshop, with particular attention to the symmetry of deflexions and the parallelity of shaft displacements. Fig. 12 shows the displacements f_1 to f_6 of the six bearings when the casing was lifted by $f_7=1$ mm at one of the lateral supports, or by $f_9=1$ mm at one of the central supports. This was done by applying vertical forces F_7 or F_9, respectively. As the measured values indicate, the principle of maintaining parallelity of shafts is verified to a considerable degree of accuracy. Nevertheless, slight differences are apparent. In a similar way, deflexions were measured when the bearings were loaded by a vertical test force.

Load Distribution

During the first sea trials, the 17·4 MW (23 400 hp) unit described in Fig. 3 was tested for load distribution over the face width, axial movements of shafts, torsional vibrations, mechanical vibrations of the casing, and noise emission.

Fig. 13 shows the position of four strain gauges, which are called B1, B3, C1 and C3, to measure the tooth root stresses σ'_1, σ'_3, σ''_1 and σ''_3 at four points which were well selected over the face width of the double helical teeth of the main wheel. At every revolution, each strain gauge is stressed in accordance with the local forces transmitted to the respective tooth by the port or the starboard pinion. In the course of the time, different teeth of each pinion come into contact with a measuring tooth of the main wheel. Fifty two consecutive stress peaks were recorded. When plotted against the tooth number j_1 of the pinion by distinguishing between odd and even numbers of pinion revolutions, stress diagrams like in Fig. 14 are obtained. The illustration shows the tooth root stresses σ''_1 and σ''_3 as measured at the forward and the aft end of the aft helix, when only the port engine was operated at full load. Stresses σ'_1 and σ'_3 at the forward helix follow a similar pattern.

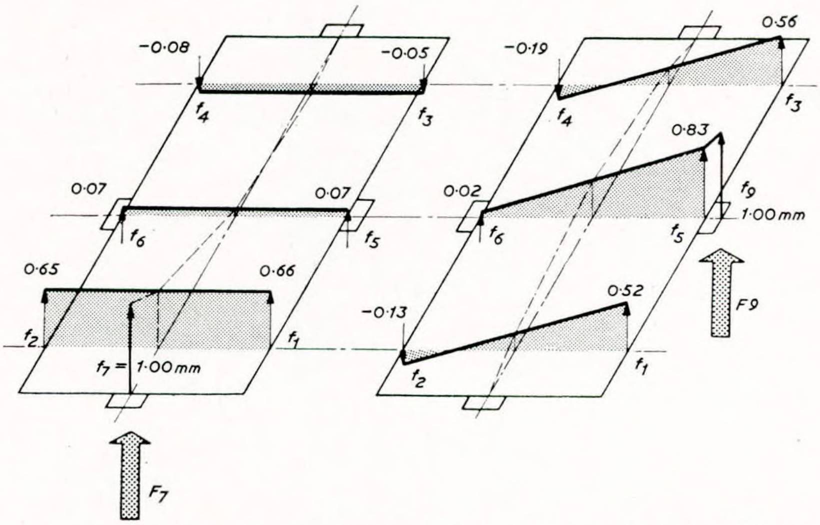


FIG. 12—*Experimental verification of the principle of maintaining parallelity of shafts*

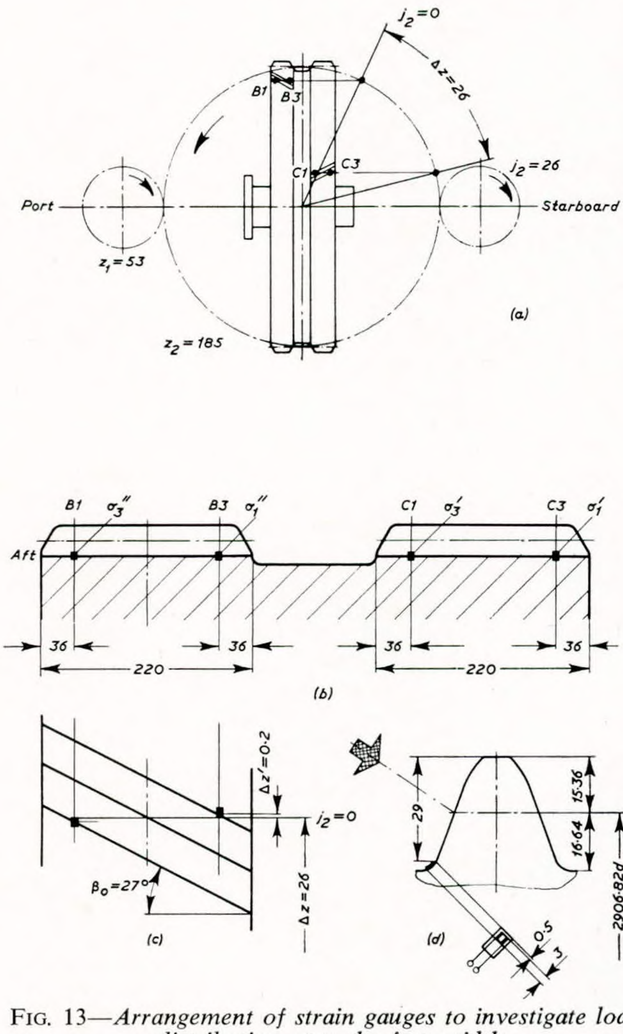


FIG. 13—*Arrangement of strain gauges to investigate load distribution over the face width*

Both diagrams show a sinusoidal variation which roughly follows curves b. A 1.5 order variation is predominant. It can be explained by a 1.5 order torsional vibration transmitted by the diesel engine. About curves b, the measured values are scattered at random with a standard deviation s_R. Mean values are indicated by the straight lines a. The mean values of σ''_1 and σ''_3 differ only slightly. This proves that the load distribution over the face width of the aft helix is almost perfect.

Fig. 15 (upper diagram) shows the distribution of the mean values of σ''_1 and σ''_3, and also of σ'_1 and σ'_3 at the forward helix, when plotted over the face width. Similarly, mean stresses obtained from measurements when both engines were operating, are plotted in the lower diagrams of Fig. 15. Lines a represent the full load, lines b the quarter load condition.

A great amount of information can be derived from Fig. 15. In all running conditions, the load distribution on either helix is most satisfactory. Also, the load sharing between the two helices is almost perfect, and differences of helical angles are negligible.

Numerical values for the mean stresses and their components as defined by Eq. (2) are, in the case of only the port engine operating at full load:

$$\begin{aligned} \sigma'_1 &= 97\ \mathrm{N/mm^2} \\ \sigma'_3 &= 79\ \mathrm{N/mm^2} \\ \sigma''_1 &= 77\ \mathrm{N/mm^2} \\ \sigma''_3 &= 83\ \mathrm{N/mm^2} \\ \bar{\sigma}_2 &= 84\ \mathrm{N/mm^2} \\ \delta\bar{\sigma}_2 &= -\ 4\ \mathrm{N/mm^2} \\ \Delta\sigma &= -\ 3\ \mathrm{N/mm^2} \\ \delta\Delta\sigma &= -\ 6\ \mathrm{N/mm^2} \end{aligned} \qquad (3)$$

The maximum stress, $\sigma'_1 = 97\ \mathrm{N/mm^2}$, when related to the mean stress, $\bar{\sigma}_2 = 84\ \mathrm{N/mm^2}$, reveals a total load distribution factor of not more than:

$$K_{F\beta} = 97/84 = 1,\ 15$$

i.e. the summation of all additional stresses amounts to only 15 per cent of the mean stress. This can be considered as an excellent practical result.

Axial Movements of Shafts

To investigate axial movements of shafts, inductive transducers were installed at the forward end of the hollow pinion shaft and the input shaft. The most vivid movements were recorded during a crash stop manoeuvre, Fig. 16. With respect to arbitrary zero points, x_1 represents the axial displacement of the input shaft, and x_2 that of the pinion shaft, x_2 also corresponds to the displacement of the main wheel. The difference of both, x_1-x_2, is the relative axial displacement at the gear type coupling between the two shafts. During the crash stop manoeuvre, the propeller pitch was changed from $+10$ to -10 within approximately 70 s. When decreasing the pitch from $+10$ to zero, the value of x_1 changes by 1·5 mm, the input shaft moving this much to the aft. At negative pitch, the input shaft moves forward again. The pinion shaft, however, moves to the aft steadily, altogether as much as 2 mm. Both shafts move independently.

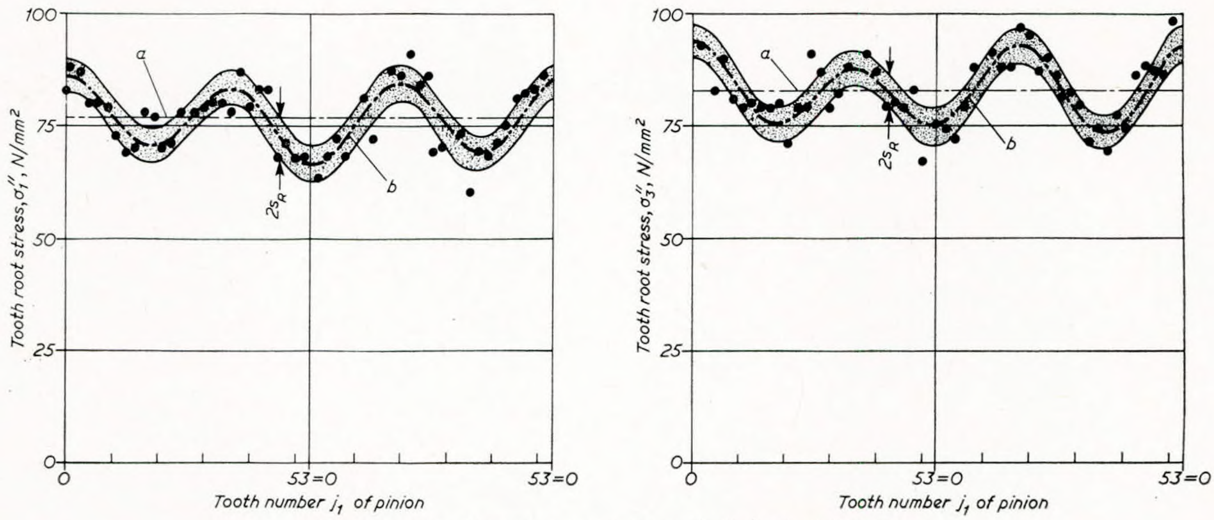


FIG. 14—*Tooth root stresses* σ''_1 *and* σ''_3 *at the aft helix*

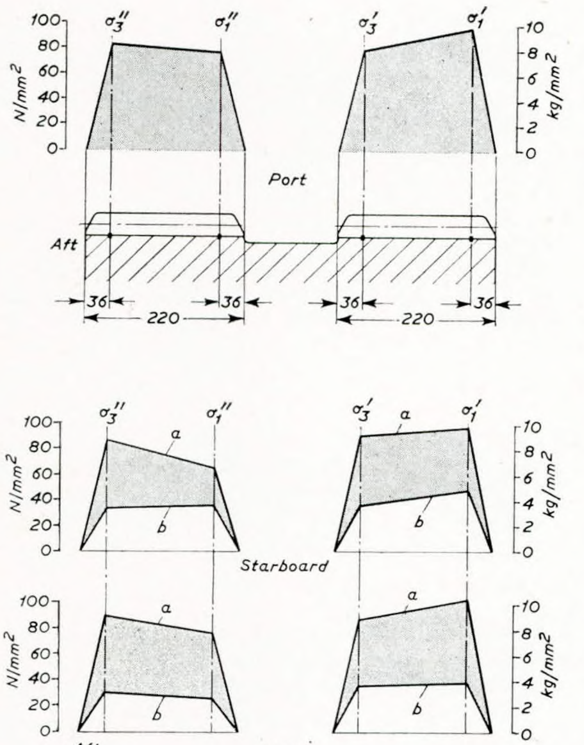


FIG. 15—*Stress distribution over face width–up: port engine at full load, down: both engines at full load* (*a*) *and quarter load* (*b*)

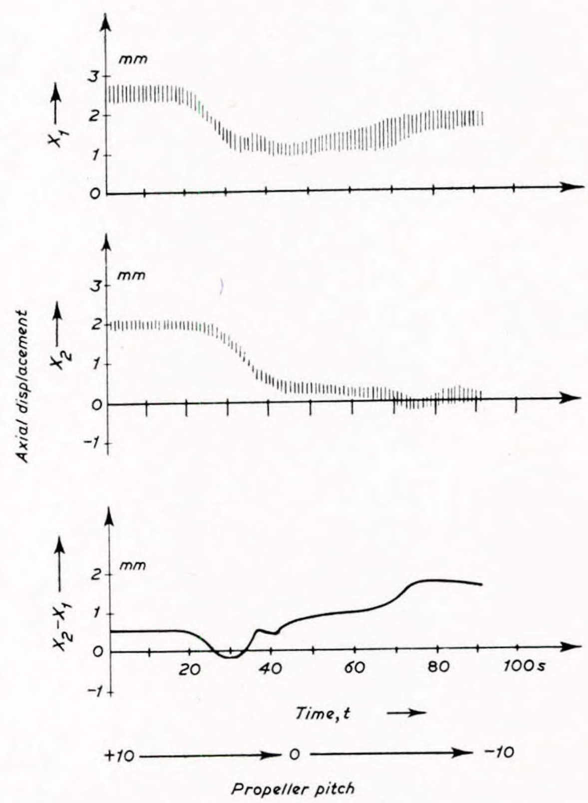


FIG. 16—*Axial displacement of shafts during crash stop manoeuvre* x_1 *input shaft,* x_2 *pinion shaft,* $x_1 - x_2$ *gear type couplings*

The movement of the pinion shaft is explained by the deformation of the thrust bearing and its support at changing propeller thrust, and that of the input shaft by a reaction of the flexible coupling when the engine torque is changed. As a result, the gear type coupling has to accommodate differential axial displacements of about 2 mm, which is evidently accomplished without difficulty.

The shaded lines in the diagrams for x_1 and x_2 describe the amount of axial vibrations of the shafts. At the maximum, these vibrations have amplitudes of $\pm 0{\cdot}3$ mm at the input shaft, which is not excessive.

Other Results

By further investigations, it could be verified that excessive mechanical vibrations of the gear casing did not occur. Also, the noise level remained within limits which are normal for a marine reduction gear of that size. Torsional vibrations, as revealed by Fig. 14, were also in permissible limits.

PROGRESS OF STANDARDS

The statement that, with the new design of reduction gear, additional stresses due to unequal load distribution over the face width are reduced to a negligible minimum, should be supported by comparison with the results of similar investigations on marine reduction gears of conventional design. Fig. 17 shows this comparison. The additional stress $\Delta\bar{\sigma}$ (or $\Delta\sigma$ with single helical gears) is plotted against the mean stress $\bar{\sigma}_2$ (or σ_2), thus indicating how $\Delta\bar{\sigma}$ or $\Delta\sigma$ changes when the transmitted torque is increased. Curves a to e correspond to vessels with reduction gears of conventional design, and curve

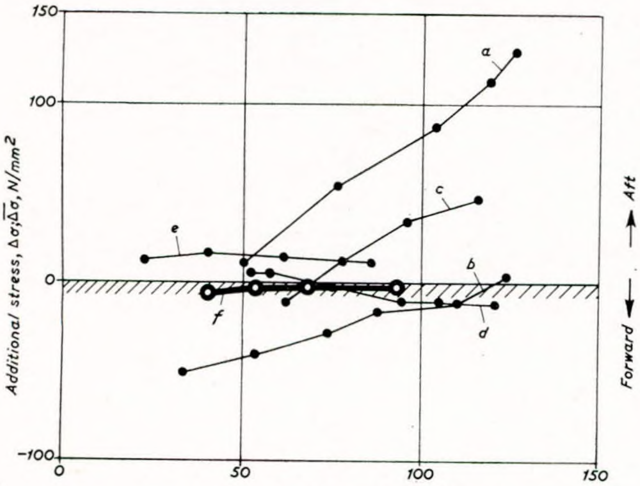


FIG. 17—*Additional stresses $\Delta\bar{\sigma}$, $\Delta\sigma$ due to shaft displacements measured at various reduction gears—a to e conventional design, f new design*

f corresponds to the new design. With the shaded zero line being perfection, it follows quite clearly from the diagram that considerable progress has been made.

CONCLUSION

Strain gauge measurements of tooth root stresses during sea trials have proved that the problem of unequal load distribution over the face width has been satisfactorily solved by the new design of reduction gear. At the same time, it turned out that the anticipation of poor load sharing between the two helices was immaterial. Also, no difficulties were involved with axial movements of shafts, vibrations, or noise. The performance was altogether excellent. Thus the new gear can be considered as a successful contribution to the progress of engineering standards in the field of marine propulsion.

ACKNOWLEDGEMENTS

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REFERENCES

1) SHORT, K. I., 1972: "Operational experience with medium speed diesel engines". *Trans. I. Mar. E., 1972, Vol. 84, p. 37.*

2) PINNEKAMP, W. and POLLAK-BANDA, E., 1973. "Marine reduction gears for medium speed diesel engines". The International Symposium on Marine Engineering, ISME Tokyo '73, 2-6-11.

3) PINNEKAMP, W., 1972: "Zur Lastverteilung über die Zahnbreite bei grossen Schiffsgetrieben". *Konstruktion* 24 (1972) 11, p. 428.

4) PINNEKAMP, W., 1973: "Untersuchungen über den Ausrichtfehler bei Schiffsmaschinen". *Schiff und Hafen* 25 (1973) 3, p. 234.

5) PINNEKAMP, W., 1974: "Symmetrieeigenschaften grosser Schiffsgetriebe mit Vierpunktabstützung". *Konstruktion* 26 (1974) 3, p. 85.

Discussion

MR. J. F. BUTLER, F.I.Mar.E., began by saying that he was not a gearing designer. His company's expertise came from the Hawthorn side and not the Doxford side. However, they experienced similar problems with engines. To maintain even pressure along two surfaces in close contact meant that it was necessary not only to maintain symmetrical loading but also symmetrical reaction from their supports. Such problems arose in the case of engine bearings, and had solutions similar to those described by the author.

With very heavy loads, it was essential not only to keep the load the same at the two ends but to design the surfaces so that the deflexions under load matched, so that if one part bent to a convex surface the other bent to a concave surface, and there was even loading from middle to the ends. That had been done by one of their competitors at the top end of a two-stroke engine, and had been successful. His own company had another way of achieving the same result.

It appeared that when there was a double pinion on a shaft of same length it would bend to some extent so that the load in the inner edges of the two helices would become less than at the outer edges. He wondered if it would not be better to put the drive into the mid-point of the hollow shaft so that the extra load in the middle, due to torsional effect, could match the additional load at the outer ends due to the bearings. But perhaps these were two different orders of deflexion.

In designing geared engine arrangements his firm laid out their gear box as Dr. Pinnekamp had done, with four mounting points on the axis of symmetry but with two main differences. In the first place they had felt that the four feet should not be fixed, but should have provision for sliding between the gear case and double bottom to allow for temperature expansion. The same principle was used for turbines, otherwise one got hog because the double bottom was cold and the turbine was hot. Therefore they had fixed one of the feet and allowed the others to slide towards it to cater for free expansion. The other difference was that at that time they designed the gear box to have an integral thrust block.

He agreed, however, that this was absolutely wrong and one should not make gear boxes suffer thrust loads. He would have preferred to put the thrust block as far aft as possible as the shape of a ship—tapering towards the stern—was such that near the aft end the hull was much better adapted to accepting thrust load. The thrust block should be supported all round as well as underneath, otherwise it would tend to tip and to put uneven loading on the thrust pads and bending stress on the ship. Mr. Butler made one final point. Dr. Pinnekamp's gear box had a main wheel located axially to even up the load on the two helices. He wondered if the author had met any problems of vibrational "shuffling" of the pinion, when it moved relative to the main gear. There were all the elements there to produce such vibration. He asked if anything had been done in the way of damping in the couplings, to avoid this difficulty.

MR. C. J. CHARLES, M.I.Mar.E., said that the philosophy of gear design described in the paper was an impressive solution to a thorny problem, and the excellent presentation matched the balanced approach in the written paper. He would like to comment on a few aspects.

His first point referred to the tests done in the workshop to demonstrate the principle of the four-point support. Was the method of support used there identical to the arrangement in the ship? It was apparent that the extent to which the four supports behaved as pivots was a very important factor. He would be interested to know how flexible these supports were in the ship and whether the support arranged in the workshop reproduced the ship condition.

Secondly, the influence of the line shafting, on the loading on the main wheel bearings, was a very important point. Certainly reference had been made to this in the paper. Were there any figures which showed quantitatively how this effect could influence the distribution of load across the gear face? For example, could it be shown, by reference to measurements, that the introduction of a main shaft flexible coupling could be justified?

Moving on to the proof of the pudding—the results of the strain gauge tests—he would ask about the position in which the gauges had been placed. From the diagrams in the paper it could be seen that the gauges were placed 36 mm from the end of the tooth. The tooth size appeared to be about 12 module, so taking into account end relief which he assumed was present on the gears, the gauge was placed very near the end of the effective tooth face.

If one then considered where a typical line of contact lay on the tooth flank, as illustrated in Fig. 18, it was apparent that the stress measured in the root of the tooth was a function of the load/unit of the line of contact, and the effective length of the moment arm.

Since, for a helical gear, the line of contact was inclined to the gear axis, different effective moment arms were generated at the two extremities of the tooth face. Hence, it was necessary to interpret the measured stress before drawing conclusions about the degree of uniformity of loading across the gear face. Roughly speaking, he thought that for a double helical gear of the type being considered, the stress measured near to the apex of the gear would be 1·5 of 2 times the stress measured at the base of the helix for the same level of unit loading, or surface stress. He asked whether this effect had been taken into account in drawing up the diagrams in the paper?

Finally, he had been rather disappointed to find that there was a toothed coupling in the transmission train. Possibly this was introduced because of the uncertainty about the amount of axial displacement that a flexible coupling in that position would have to accommodate. Now that these displacements had been measured, would the author be inclined to introduce a coupling with a more defined axial characteristic, possibly a membrane or link coupling which would not suffer from the potential hazard of locking up and creating very high axial loads such as one could get with a toothed coupling?

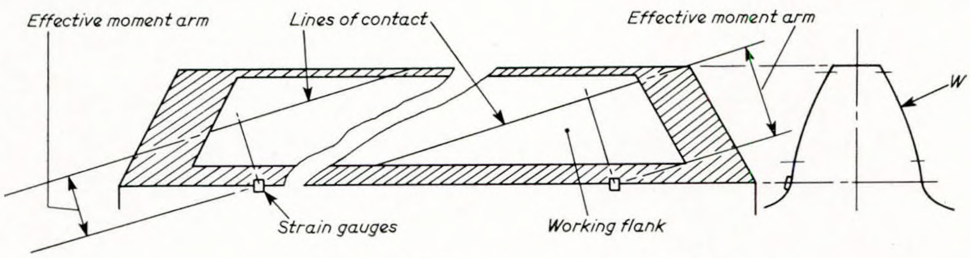

FIG. 18—*A typical line of contact on the tooth*

MR. A. E. TOMS F.I.Mar.E said that the paper had concentrated upon the design criteria and the results obtained during trials. The experience of the classification societies indicated that more emphasis needed to be given to the realistic capabilities of manufacture. He was sure that the author would agree that the ideal design could run into difficulties due to manufacturing limitations; he would have liked this to have been specifically mentioned in the paper, rather than being left with the impression that any troubles arising would be due to inadequate design.

The Rules of Lloyd's Register contained a factor for "high elastic" couplings in diesel engine gearing and a descriptive note had been added regarding the coupling flexibilities required to minimize the misalignment effect on the teeth. Generally the gear manufacturer would have to take care of the radial movement. However, all aspects had to be considered to be eligible for the application factor within these Rules. Couplings were generally introduced for torsional considerations and it had been requested for the Rules to be simplified in order to accept the gearing solely on the torsional characteristics of the couplings. He welcomed the author's belief in the need to cater for all movements in this respect.

If any difficulty were encountered, it would be likely to be concerned with the axial movement of the gear type coupling at the aft end of the quill shaft. Attempts with such a device had proved troublesome in the past. The author had said that it was not easy to predict these movements. Nevertheless, no doubt he had anticipated this and could do something about it, if necessary.

The important aspect of all installations was connected with the alignment of the shafting. One could calculate the deflexions of the hull by means of finite element methods, but with all the different types of ship loading, thermal conditions etc, there was a multiplicity of deflexions at the shafting supports from which one had to determine the best alignment for the gear in service.

Forced to provide two gear bearings in close proximity, he thought it advisable to have a seperate thrust without shaft support bearings. He would suggest that there should be an even greater span to diameter ratio between the aft gear bearing and the intermediate shaft bearing than that shown in Fig. 10. On paper, 1·05 mm may appear to be large, but with calculation errors, slight errors in setting up, and the tooth load involved, this was not so large a margin in operating conditions where, if there were more than one engine, operators liked to ring the changes.

The initial alignment would be carried out in the cold, static condition and he believed that Fig. 8, for the loaded condition, merely showed that this had been done incorrectly. The meshing contact markings probably showed the area to be acceptable, but this method was very subjective and it needed a very experienced person to determine the quality of the marking, which was the important factor. Such marking showed the combined contact at all speeds during the trials, whereas what one wanted to know was the condition at each speed. This marking was a simple classification requirement, but in the contributor's Society, they were recommending that telemetry be used more extensively to safeguard the gear manufacturer in that respect. It was good to see the author using this method for his judgment of good contact.

He had noticed what he believed to be a minor error in the calculations, equation 3, where the effect of helical angle error should be $+6 N/mm^2$. and not $-6 N/mm^2$. The value of $K_F\beta$, given in the paper for a face width/diameter ratio of 0·68, appeared to be high since most authorities would give a value of 1·15 for a ratio of approximately 1·5 in uncorrected gears. Without the mismatch of helical angles, the factor would have been 1·08 which was much nearer the I.S.O. figures. This, again, showed the effect of manufacturing capabilities. If the errors had been measured, perhaps the author could give these values, which would be invaluable in assessing the effect more objectively and would add to the general understanding on this point.

The fact that the effect of misalignment alone was so small—one obtained a value of $K_F\beta=1{\cdot}025$—confirmed the view that, from the design aspect, the symmetrical supports and the connexion between the pinion and the prime mover was a successful contribution to progress within this field.

MR. C. GAY M.I.Mar.E stated that he had a fairly brief point about the position of the thrust bearing. Dr. Pinnekamp had mentioned the importance of taking charge of the flexibility of the hull. One other speaker had pointed out that if the thrust bearing support was flexible, it could tip and impose axial forces on the gearing. He suggested that the thrust bearing should be positioned well aft of the gear shaft. However, 95 per cent of the mass of most ships was forward of the gear shaft. Traditionally, because of the practical constraints of the large cathedral diesel engines and the steam turbine seated over a large condenser, it had been customary to position the thrust bearing aft of the gear shaft, but this resulted in a tremendous reduction in the stiffness of the hull immediately forward of the thrust bear in order to accommodate the gearwheel—a reduction in hull stiffness where it could least be afforded. Would it not be preferable in all cases where practicable to put the thrust bearing forward of the gear box?

MR. A. E. WOLSTENCROFT expressed his gratitude, as a visitor, in being allowed to participate in the discussion. He wished to take up the point the author had made about marginal problems. In viewing the gear units described, it seemed to him that certain problems were likely to occur. Some had already been mentioned, such as the use of the gear type coupling to support the quill shaft.

A further one was the use of rolling contact bearings to support the clutch mechanism. When the clutch was engaged, there would in fact be no relative movement in those bearings. Was it possible that the author had encountered brinnelling in those conditions? Had the bearings been damaged?

Secondly, he would emphasize the point concerning lock-up of the gear type coupling which Mr. Charles had made. Had the author met with any form of fretting which might result in lock-up?

Thirdly, he would ask whether he considered that the combination of what appeared to be a tyre type coupling at the engine and quill shaft was so effective in damping torsional vibrations as something of the nature of a Geislinger coupling which was damped by the passage of oil past leaf spring?

MR. H. A. CLEMENTS congratulated the author on his very interesting paper.

He would like to comment on the author's emphasis on the symmetry of the gear box design to avoid relative distortions taking place as would affect the gear performance. There was a general acceptance of the quill shaft drive between a marine turbine, or diesel engine, and the pinion, but a reluctance to accept such a quill drive connexion between the propeller shaft and the main gearing. Such a quill drive would reduce the value of the offset forces which take place with hull distortion, such forces acting on the gear wheel with the comparatively small axial distance between its support bearings, had a considerable effect on the attitude of the gear wheel. The use of an output quill drive would certainly improve the load distribution on the gears maintained in the symmetrically shaped gear casing and overcome many problems caused by hull distortion. Was the reluctance to accept such a solution due to the need to reduce initial costs?

In the double helical gear design described by the author, any axial movements of the gear wheel would be transmitted to the pinion. Assuming the quill shaft was located axially from the engine by the bearing within the rubber coupling, the pinion axial movements must be accommodated by sliding of the gear type coupling interconnecting the friction clutch assembly with the pinion. When the gear was transmitting high power, a high force was required to slide the teeth of the gear type coupling, and the forces must react on the bearing within the rubber coupling. It would be interesting to learn whether this bearing was entirely satisfactory, considering that it had no relative rotational movement (except the small amount due to any torsional movements across the rubber coupling), also was subjected to angular movements when there was any misalignment between the engine and the pinion shaft.

The bearings supporting the friction clutch also had to transmit axial forces and were not subjected to relative rotation when the clutch was engaged. Such a condition could result in "brinnelling" of the bearings and it would be interesting to learn if this had been a problem.

MR. C. GAY raised one further point. In multi-diesel propulsion drives it was often necessary to operate with one or more engines going. Under these circumstances, the position of the bull gearwheel shaft in its bearing would be different in one mode of operation than in others. Mr. Clements had mentioned the reluctance in the marine industry to connect flexibly the bull gear to the propeller shaft, and with these various modes of operation, it was clear that substantial misalignment was involved.

Whereas in the vertical plane there seemed to be quite a lot of technology employed in the alignment of the propeller shaft to the bull gear, sophisticated techniques for satisfactorily aligning in the horizontal plane seemed less well known. Could the author say what provisions he would make for catering for these various modes of operation?

MR. H. J. MILLER, F.I.Mar.E., stated that he would like to take up five points with regard to the paper.

The author had shown the structure in way of the thrust block, gear box, and main engine. Mr. Miller believed that this was a very bad example, as in his view it was essential to get rather more continuity than was shown in Fig. 6. He suggested that this particular drawing should be changed at some time so as to show the continuity and integration of the structure. The paper would have been further improved had the author included the actual position of the L.O. sumps—whether forward or aft.

Secondly, he would ask the author to explain what precautions would be taken to ensure that the gear box was correctly installed, and more specifically would it be installed to equate the hot (working) condition of the propulsive unit? Then there was the question of how it could be proved to the owners and to the classification society, that it was correctly installed, as the traditional way of examining the blue markings on a set of gears immediately after the normal acceptance trial of the vessel was clearly not satisfactory: in his experience the only sure way was by the use of strain gauges continually recording during the acceptance trials—would the author endorse that view?

Thirdly, to deal with Fig. 14, he had been rather surprised to hear Dr. Pinnekamp say that although torsional vibration was apparent, the effect was small. Reading the figures from Fig. 14 (left hand graph) it would appear that the maximum and minimum tooth root stress was about 80 and 60 respectively; this amounted to a torsional variation of about 33 per cent, whereas he understood that L.R.S. allowed a torsional design margin of 30 per cent. He had seen this sort of thing during acceptance trials he would have been rather worried, as during acceptance trials the main engines were usually well balanced, and this degree of balance could not be maintained in normal service. Each manufacturer of gear boxes produced gear boxes in different power steps and he now adopted the principle that if the gear box suggested by the manufacturer indicated a working stress near the design limit then the next step up should be ordered—there had to be a sufficent reserve to cover the exigencies of the service of the vessel, and any shortcomings in the basis of design.

Fourthly, he would like to know the author's views about the different types of materials to be used in gears. People talked about "soft", about "case hardened" and about "nitrided" gears, and differing views were held by different manufacturers. In Germany manufacturers seemed to be tending to go for the case hardened gears, and this seemed to him to be the proper approach.

Finally, he was surprised to note that the paper seemed to deal only with double helicoidal gears; he had been dealing with gears up to about 6700kW (9000 hp) and had formed the opinion that single helicoidal gears offered many advantages. He wished to underline the "shuttling" aspect of double helicoidal gears (an earlier contributor to the discussion, Mr. Butler, had referred to it as "shuffling")—this was the axial oscillation of such gears which resulted in unequal load sharing by the forward and after helices and prompted the question, were the gear teeth of a double helicoidal gear designed sufficiently strong to take the whole load on either the forward or the after helix?

Correspondence

DR. J. F. SHANNON, wrote that the author had presented a very clear statement of the problems and their solution for a multi-engine single-reduction, single propeller gear box, placing great emphasis on symmetry and the positioning of the four supports on the girders of the hull.

It was generally agreed that it was best to place the thrust block as far aft as possible, making use of the flexibility of the intermediate shaft and its setting to ensure near-equal loads on the main wheel bearings. The weight of the main wheel and the shaft flexibility permits this to be achieved in the case quoted.

With larger powers and lower speeds this might not be possible, and some additional flexibility would be required. Flexible couplings placed fore and aft of the main wheel with a cardan shaft through the bore of the main wheel was one solution.

Similar flexible transmission was required at the input shaft, as pointed out by the author, who had the quill shaft through the hollow pinion and flexible couplings at either end. This was good, and most suitable where space between the engine and the gear was limited. Such was the case with the flexible drive to the high speed pinion of a steam turbine locomotive, built by Metropolitan Vickers Co. some 40 years ago and where the couplings were of the flexible disc type.

The fitting of the multi-disc clutch on the input shaft with a gear-tooth coupling and the use of the "rubber-type" flexible coupling associated with the diesel engine was neat. It would be helpful to have the torsional vibration characteristics of the system and vibration records so that the damping by the "rubber-coupling" could be appreciated, and to see why the 1·5 order vibration from the diesel engine got through to the teeth and the propeller torque variation and vibration does not.

The positioning of the four strain gauges on the two helices and the skilful analysis produced simple and clear results, showing a good gear. No doubt there were more than one set of gauges similarly placed on the wheel both as an assurance of gauge assembly life and a check on other pairs of teeth.

The unbalanced load sharing effect would be related to the axial vibrations which were seen to be present before and after the crash stops—Fig. 16. Could these be expressed quantitatively with the values obtained from the strain gauges on the teeth?

Author's Reply

The author, in reply, thanked the speakers for their contributions.

He was indebted to Mr. Butler who had raised several points. The principle of compensating deflexions of matching components was frequently used in gear design as well as, for example, the connexion of the driving shaft to the centre of the pinion and not to is end. It was a helpful means to overcome difficulties with slender pinion shafts, where torsional and bending deflexions would be of essential magnitude and, by the application of the principle as mentioned, would cancel out each other to some extent. However, a hollow pinion shaft as described in the paper, was stiff enough in bending and torsion, so that the problem would appear to be negligible. The central connexion would only involve unjustified additional cost. The correctness of this statement was verified by the strain gauge measurements, as described in Fig. 15. Further analysis had shown that the combined effect of bending and torsional flexibility had led to an increase of tooth root stresses of only $4N/mm^2$.

Provision of sliding between the gear case and certain supports on the double bottom would certainly be an excellent precaution to distortion of the gear case because of temperature expansion. However, Dr. Pinnekamp believed that although fitted bolts were only applied at one support while the others would move to some extent, this movement was not really needed because of sufficient flexibility of the hull.

It was helpful that Mr. Butler supported the idea of a separate thrust block as far aft as possible by making reference to his own comparative experience. Also, he was to be agreed with on the request to support the thrust block all around to prevent tipping. This, Dr. Pinnekamp thought, could also be accomplished by a support in the horizontal plane on propeller shaft level, which he believed to be more practical for hull designers.

It was true that axial "shuffling" of the pinion had caused problems. For this reason, axial vibrations of the pinion shaft had been measured. The amplitude under full load had been found not to exeed $\pm 0{\cdot}1$ mm. The vibrations had been of 1·5 order, thus definitely caused by engine excitation and related to a corresponding 1·5 order torsional vibration. Consequently, 1·5 order irregularities were indicated by the strain gauge measurements of tooth root stresses, not only as far as the mean value of both helices was concerned, but also with respect to differences between the two helices. These stress differences had shown an amplitude of ± 3 N/mm^2, which was not much. Thus the "shuffling" effect could be verified, but had likewise turned out to be of no practical significance. It should be mentioned that also in this respect the benefit of a separate input shaft (quill shaft) had clearly been demonstrated, because the 1·5 order axial vibrations had an amplitude of $\pm 0{\cdot}3$ mm there. Only about one third of this value had been transmitted to the pinion shaft, by virtue of axial movement of the tooth coupling between the two shafts. There were doubts whether damping forces of the rubber coupling might reduce axial vibrations effectively, but it was not believed that these damping forces were required.

The first question raised by Mr. Charles concerned the differences between the flexibility of supports during workshop tests and on the ship. In the workshop, an attempt had been made to make the supports as stiff a possible, for consideration of test procedure. The effect of flexible supports was simulated by tilting the support. On the ship, it was anticipated that the supports would have considerable flexibility, so that practical conditions would be favourable. This was proved by temporary tilting of supports by shims, which had almost no effect on the tooth root stresses, because the tilting was compensated by the flexibility.

Changes of the loading on the main wheel bearings had not been studied in detail during the investigations mentioned in the paper. However, no load variations across the face width had been observed which might have indicated such an effect. Therefore, a main shaft flexible coupling would not be justified.

The position of strain gauges one sixth of the face width away from the ends had been found to be a proper location to show load differences most clearly and, at the same time, avoid disturbing influences by end relief, which was present.

Mr. Charles's considerations on inclined contact lines and different effective moment arm were absolutely correct, Dr. Pinnekamp said. But the complications derived from this theory would only be encountered if conclusions about the uniformity of Hertzian pressure should be drawn from root stress measurements. When uniform load distribution was interpreted as a uniform distribution of tooth root stresses, as it should be, no further conversions would be required. For practical considerations, this interpretation was justified, when circumstances were as such that the hazard of fatigue cracks would be predominant to the hazard of pitting. An attempt to interpret the results of measurement in terms of Hertzian pressure had not been made.

The replacement of the tooth coupling by a membrane or link coupling might be considered. However, other provision would have to be made to accommodate axial displacement which could, according to measurements and other experience, well amount to several millimetres. So far, no locking up of tooth couplings could be observed with the reduction gears in question.

Dr. Pinnekamp was ready to agree that manufacturing limitations, as pointed out by Mr. Toms, were decisive for the practical success of a theoretically well conceived design. This aspect was more or less included in the "efficiency" mentioned in point 4 of the introduction of the paper. But it should be worthwhile to emphasize this more. It had been part of the objective of the tests to make sure that such limitations of manufacturing would not overshadow the benefits of the design principles, in which case the realistic progress would have been minimized. But, for instance, the effect of inaccurately manufactured helical angles of the double helical gear was included in the results of the stress measurements. Also, deviations from symmetry due to welding tolerances in the casing were believed to be the main reason for the still prevailing small changes of stress distribution under load. Thus, the final conclusion of the paper, that realistic progress was achieved, did include the unavoidable drawbacks due to manufacturing imperfections.

As to span to diameter ratio between the aft gear bearing and the intermediate shaft bearing, it was certainly a correct statement that the span should be as large as possible. Fig. 10 should demonstrate an example only, the value 1·05 mm being of no further numerical significance. However, it was quite interesting to learn that Mr. Toms considered this value to be rather too small than too large.

It was a good recommendation that telemetry should be used more extensively during trails. However, once the behaviour of a certain type of vessel and a certain type of gear had been studied in detail, the conventional method of marking teeth should still be suitable for subsequent applications.

Certainly, the stress component $\delta\Delta\sigma$ in equation 3 should be $+6$ N/mm^2 instead of -6 N/mm^2. Dr. Pinnekamp was thankful that Mr. Toms had emphasized that the factor $K_{F}\beta$ due to misalignment alone was not more than 1·025. However, he was doubtful that a value of 1·15 could always be realized in marine applications, according to the authorities mentioned.

As was argued by Mr. Gay, the arrangement of the thrust bearing forward of the bearing should be recommended, if space would permit. However, in most cases particularly with multi-engine applications, this engine particularly with multi-engine applications, this space would not be available without sacrifice of valuable cargo space.

In the author's experience, different positions of the bull gear shaft at various operational conditions had not caused too serious misalignment effects. Again, strain gauge measurements had not shown appreciable changes of load distribution, even with three-engine reduction gears, which allowed more different positions of the shaft than in the case of a double-engine installation. So far, no special provisions had been made to cater for movements in the horizontal plane. This was justified because weight components would not act in this direction and would not disturb equal tooth forces so much.

Mr. Wolstencroft had mentioned the problem of brinnelling of clutch bearings without relative movement. Brinnelling had been observed in exceptional cases only, where extraordinary vibrations had occured. Otherwise, the bearings had not suffered, because the geometrical situation required overdimensioning of these bearings anyhow.

Also, fretting of the tooth coupling was not observed during a check-up. It had to be pointed out that also these tooth couplings had to be overdimensioned for geometrical reasons.

In cases, where large torsional vibrations had to be anticipated, Geislinger couplings had been considered and applied. However, this aspect would require special treatment in individual cases. Rubber type Vulkan couplings had been successful as well. Although their damping might be smaller, their flexibility was larger, which might solve vibration problems even more effectively.

Mr. Clements had drawn attention to the reluctance to apply a quill shaft connexion between the main gear and the propeller shaft. Such an alternative would definitely be an improvement, although—in the author's opinion—it would not solve too many problems. This, in connexion with the reduction of initial cost, might be the explanation why a quill shaft had not been adopted too frequently in this place.

To answer the question about the properties of the special bearing inside the rubber coupling, it should be mentioned that this bearing was allowed to move freely axially by ± 12 mm. Angular movement would not affect the rollers, but would be accomplished by a special rubber bush surrounding the bearing.

Mr. Miller had made some remarks on the hull structure in the vicinity of the gear box. The author would agree that improvements were possible from the point of view of hull designers. The scope of the paper had been to show the points of view of the gear designer, which were considered to be realized effectively. So far, no drawbacks had been experienced. The lubrication oil sumps of the engines were forward of the gear, that of the gear was incorporated in the gear casing.

Once strain gauges had been applied to a marine gear, the author would agree that records should be made during the whole trial time. However, he never noticed any appreciable changes during such a small period. Larger changes might well occur later. Therefore, it had been recommended to monitor strain gauge readings continually during regular operation. A special electronic monitoring unit—known as the RENK-CHECKER—had been developed and successfully applied for this purpose.

The author was still of the opinion that the torsional vibration shown on Fig. 14 was small. With 80 and 60 N/mm^2 as the maximum respective minimum values, a percentage amplitude (not double amplitude) of $(80-60)/(80+60)\cdot 100$ per cent $=$ 14 per cent would follow, i.e. much less than the permissible 30 per cent. Thus, a margin was left for unbalance to occur later.

Case hardened gears would have the best durability and were preferred. In the case described, the main wheel was "soft", however, i.e. with natural hardness. Nitrided gears would be more economical whenever the dimensions of the gear had to be larger than required by strength, e.g. when the arrangement of engines determined the dimensions.

To the "shuttling" (or "shuffling") effect, comments had been given in reply to an earlier contribution. Of course, one helix would eventually not suffice to take the whole load.

Dr. Shannon made a written contribution in which he mentioned the requirement of a thrust bearing as far aft as possible. Also, he mentioned the possible application of a quill shaft at the main wheel. The author had given his comment to these questions before. Vibrations coming from the propeller were not appreciable in calculation. In any case, they would not appear in the strain gauge records, because the strain gauges were fitted to the main wheel, the teeth of which would come into mesh with the pinion always at the same phase angle of a propeller excited vibration, which should be assumed to be periodical with propeller or main wheel revolutions.

It was, in fact, possible to find a relationship between the axial vibration of the input shaft which generates an axial force variation due to axial spring constants of the rubber coupling, and the vibratory component of the unbalanced load sharing of the two helices. Quantitative analysis had revealed matching results.