Progress and Development in Naval Propulsion Gears 1946-1962

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The formation of the Admiralty-Vickers Gearing Research Association in 1946, its objectives and some of its work have already been described by other authors. Against this background, the present paper gives details of some post-war naval propulsion gears and accounts of their performance on trials and in service.

In the second part of the paper, some opinions on lubricating oil filtration, cleanliness and flushing are expressed and experience with extreme pressure lubricating oils is described.

Prefinished steel shell bearings are fitted in the latest naval gear sets and the use of these bearings is discussed.

An account is given of recent full scale tests of nitrided and induction hardened gears and of recent developments in the technique of induction hardening. The results are encouraging and suggest the increasing use of such gears.

In the concluding remarks a comparison is made of the sizes and weights of some of the gears described and it is seen how improvements in accuracy of manufacture, use of surface hardened materials and of more highly loaded bearings have all contributed to reductions in the sizes and weights of naval propulsion gears.

Finally some thoughts on the future are expressed.

INTRODUCTION

It was in 1910 that Sir Charles Parsons installed reduction gearing in s.s. *Vespasian* and this was followed by the large scale introduction of geared turbine machinery into the Fleet of the First World War period. Tostevin[1] has described the experience with the gears of these ships:

> ". . . of the 596 sets of all-gear installations on service in the Navy, some extending up to nearly six years, it has only been necessary to remove three for refit . . . and here it must be emphasized that no actual breakdown occurred and the gears, after dressing up and, in one case, new pinions being supplied, were subsequently re-utilized".

This was a remarkable record of achievement and a striking tribute to the gear engineers of those days.

The experience with turbine reduction gears during the Second World War was less praiseworthy and Joughin[2] has described some of the failures which occurred. His paper invites the conclusion that the comparatively small increases in gear loading since the early days had brought with them a disproportionate sacrifice of reliability. Furthermore, the ships of the Royal Navy were fitted with single-reduction gears whereas, in the U.S.A., double-reduction gears had been successfully developed before the war and were fitted in a very large number of U.S.N. warships.

In 1946, with recent war experience in mind, the Admiralty-Vickers Gearing Research Association (A.V.G.R.A.) was formed as an association between the Admiralty and certain industrial firms, representing the user, gear manufacturer and gear-cutting machine maker. In due course, B.S.R.A. (ex Pametrada) became a member of the Association which, since its formation, has been greatly assisted by the staff and facilities of the National Engineering Laboratory.

A.V.G.R.A.'s objectives were, briefly, improvement in gear cutting accuracy, the development of post-hobbing processes, evaluation of alternative materials and the development of surface hardened and ground gears. Much of the work carried out has already been described by Braddyll[3], Chamberlain[4], Chesters[5], Newman[6] and Page[7] and this work forms the background to Part I of the present paper in which a number of post-war naval gear designs are described and an account is given of their performance on trials and in service.

These designs are presented in chronological order and it will be seen how each has been influenced by the advances in knowledge and manufacturing techniques which have resulted from A.V.G.R.A.'s work.

E.P. lubricating oils are used in the majority of ships designed and built since the war and in Part II of the paper, experience with these oils is described and the cleaning and flushing of gearing is discussed. An account of experience with prefinished steel backed bearings is also given. Although carburized gears have been used extensively in recent designs, the distortion which occurs during hardening necessitates excessive grinding and thus increases in time and cost of production. Both induction hardening and nitriding result in very much less distortion and the load carrying capacity of gears hardened by these processes is being investigated by A.V.G.R.A. Although this work is not yet complete, the results of some recent full scale gear tests are given and recent advances in the technique of induction hardening are described.

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PART I

POST-WAR NAVAL PROPULSION GEARS—DESIGN, TRIALS AND SERVICE EXPERIENCE

Daring *Class, Marks I, II and III*

The first ships designed and built after the war were the *Daring* Class destroyers and these were also the first major R.N. ships to be fitted with British double-reduction gears. Three gear designs were fitted:

Mark I:

A dual tandem articulated design, with hobbed and shaved double helical gears and integral main thrust block forward of the main gear wheel. The gear loadings were 90-100 K (Lloyd's K factor). Some details of this design were given by Page[7].

Mark II:

This design was similar to Mark I but the gear loadings were higher (up to 130 K) and a separate main thrust block was fitted close to the aft end of the gearcase.

Mark III (H.M.S. *Diana*):

This design had single helical, carburized, hardened and ground pinions and primary wheels and an air hardened main wheel in a dual tandem, articulated arrangement. The gears were designed and manufactured in Switzerland. The gear tooth loadings were up to 260 K.

Although pitting of the main wheel teeth was experienced during the shore trials of the first design, all these gears have given satisfactory service. They will not be discussed further here but are referred to again later in the paper when comparative sizes and weights are considered.

The Y.100 Gears

These gears were fitted in the first post-war anti-submarine frigates, the twin-screw *Whitby* Class and the single-screw *Blackwood* Class vessels. In both classes it was necessary to make very substantial reductions in the weights and sizes of all items of machinery and this requirement had a very considerable effect on the gear design.

Lack of manufacturing capacity and experience of the production of hardened and ground marine propulsion gears in the U.K. made it necessary to use through hardened, hobbed and shaved materials. At the same time, for a similar duty, the Royal Canadian Navy decided to use carburized, hardened and ground gears and to set up the necessary manufacturing facilities in Canada. Their very successful experience with these gears has been reported by Nicholson[8].

Design Details

The arrangement of the Y.100 Mark I gearing is shown in Figs. 1 and 2 and general design data are given in Table I. The gears transmit 15,000 s.h.p. with a reduction ratio of 5,750:225 (cruising gears 8,400:154).

The drive from the main turbine was transmitted through double helical, double-reduction, dual tandem, articulated gears and that from the cruising turbine through an automatic clutch and an additional set of gears to the outboard primary wheel. The clutch was situated between the cruising turbine intermediate shaft and primary pinion (see Fig. 1). Its purpose was to disengage, automatically, the cruising turbine at approximately 30 per cent full power when power was increased and to re-engage automatically on reduction of power to this value[9].

To meet the requirements of minimum size and weight the tooth loadings were raised to 270 K for the secondary gears and 230 K for the primaries, i.e. between two and three times higher than those previously used in R.N. propulsion gears of comparable power.

To withstand these loads the materials chosen by the Admiralty were En 26 ($2\frac{1}{2}$ per cent nickel-chromium molybdenum steel) oil quenched and tempered to 70-75 tons for the pinions and En 30b ($4\frac{1}{4}$ per cent nickel-chromium molybdenum steel) air hardened and tempered to 59-67 tons for the wheels, these being the hardest materials that could be satisfactorily hobbed and shaved. At the time this choice was made there was no direct evidence in favour of this combina-

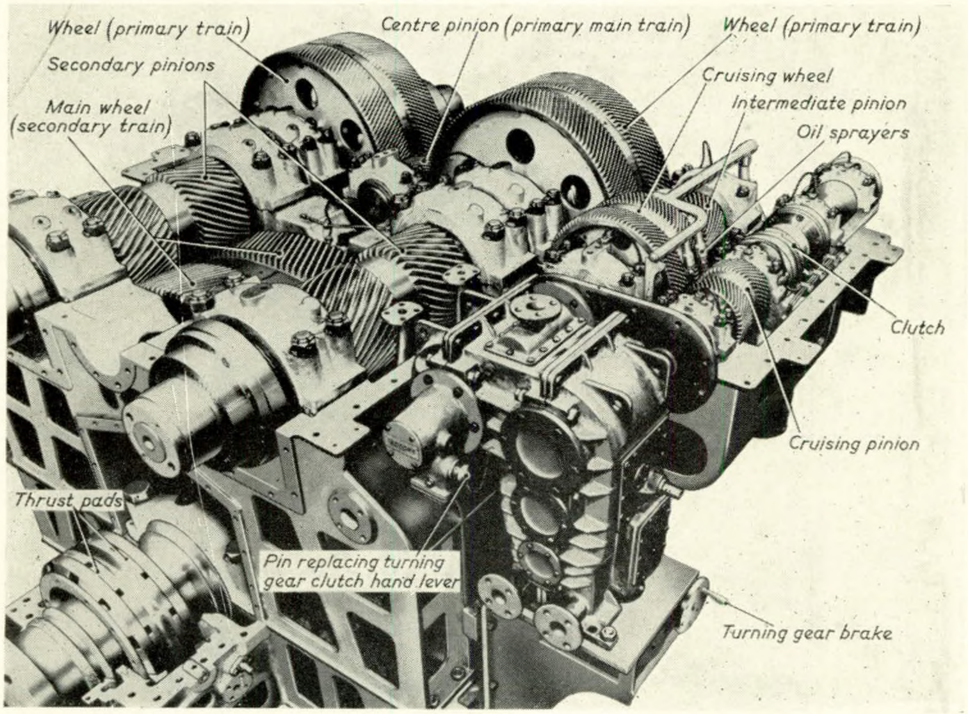


FIG. 1—*Y.100 gears*

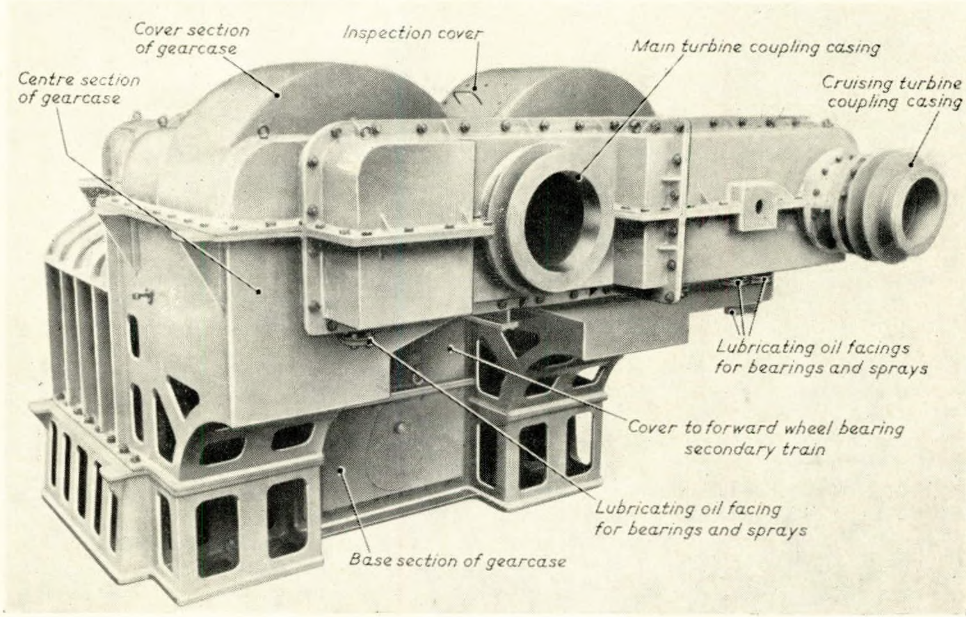


FIG. 2—*Y.100 gearcase—View from forward end*

tion of materials, for although full scale gear tests had been planned they had not yet been carried out. However, disc tests[5] of other materials indicated that surface load carrying capacity increased with the square of the tensile strength and the choice of materials was made on this basis.

The first three series of S.G.B. (steam gunboat type) gear tests, carried out at Pametrada and subsequently reported by Newman[6] had shown that helix correction designed to compensate for the full load distortion could provide a considerable increase in load carrying capacity. Helix corrections were therefore adopted for the secondary pinions of the Y.100 gears and details of the original corrections are given in Fig. 3. It was appreciated that the rather large normal pitch (1in.) of the secondary gears might cause a tendency to scuff but E.P. additive lubricating oils were then becoming available and could be used if required.

The gearcase was a fabricated steel structure with steel covers and cast steel bearing housings. White metal bearings in thick steel shells, equipped with thermocouples and mercury-in-steel thermometers, were fitted. The drive from each turbine was transmitted through nitrided gear type, fine tooth couplings. Gear type fine tooth couplings were also fitted at the aft end of each quill shaft connecting the primary wheels and secondary pinions. The axial position of the main wheel was determined by the main thrust bearing, the secondary pinions being free to position themselves axially. The position of the

TABLE I. Y.100 GEARS. GENERAL DESIGN DATA.

	Main turbine				Cruising turbine			
	Primary		Secondary		Primary		Secondary	
	Pinion	Wheel	Pinion	Wheel	Pinion	Wheel	Pinion	Wheel
Number of gear elements	1	2	2	1	1	1	1	1
Number of teeth	37	150	34	211	47	102	37	150
Pitch circle diameter, in.	8·508	34·492	12·604	78·22	8·832	19·168	8·508	34·492
Face width at pitch line, in.	$12\cdot3+2\frac{3}{4}$ gap Double helical		$18\cdot1+4$ gap Double helical		$6\cdot56+2\frac{3}{4}$ gap Double helical		$12\cdot3+2\frac{3}{4}$ gap Double helical	
Tangential load/Unit width:								
Full power, lb./in.	1,582		2,935					
Full cruising power, lb./in.					869		1,044	
Reduction ratio	4·05		6·21		2·17		4·05	
Total ratio	25·16				54·60			
K Factor:								
Full power	232		270					
Full cruising power					144		153	
Helix angle, deg.	30		30		32		30	
Normal pitch	0·625		1		0·5		0·625	
Normal pressure angle, deg.	16		16		16		16	
Addendum ratio, pinion/wheel	60/40		60/40		60/40		60/40	
Material Pinion	En.26 70–75 tons u.t.s./sq. in.		En.26 70–75 tons u.t.s./sq. in.		En.26 70–75 tons u.t.s./sq. in.		En.26 70–75 tons u.t.s./sq. in.	
Wheels	En.30 60–65 tons u.t.s./sq. in.		En.30 60–65 tons u.t.s./sq. in.		En.30 60–65 tons u.t.s./sq. in.		En.30 60–65 tons u.t.s./sq. in.	
Method of manufacture	Hobbed and shaved							

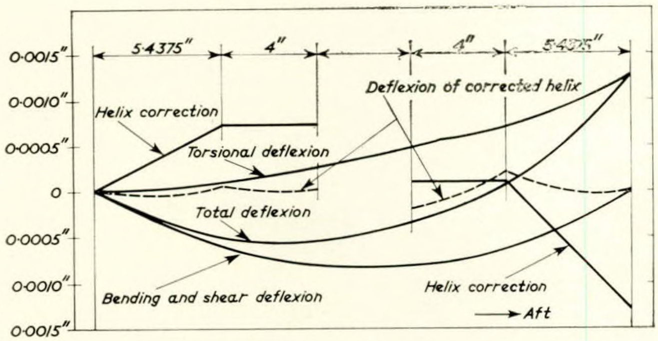


FIG. 3—*Y.100 gears—Original helix correction as applied to second reduction pinion*

outboard primary wheel was located by thrust faces at the ends of the primary wheel forward journal, the primary pinion and inboard primary wheel being free to position themselves axially relative to this located wheel. The weight of the gear set, complete in all respects, was 17 tons. The overall dimensions are shown later in Fig. 33.

Y.100 Mark I Gears—Shore Trials Experience

The port gear set for the first ship, H.M.S. *Whitby*, was tested at Pametrada in conjunction with the Y.100 machinery and was subsequently installed in the ship. Throughout these trials the lubricating oil used was OM100, which did not contain an E.P. additive.

a) *Gear Teeth*: Preliminary running of approximately 20 hours at up to 50 per cent torque and 7 hours at 50-100 per cent torque was completed satisfactorily but after a total of 7½ hours full power running, deterioration of the secondary gear teeth had begun. Light scuffing, over a length of 1½in. was found at the forward ends of the secondary pinions and main wheel teeth. On the main wheel, the scuffing was towards the roots and on the pinions, it was at the tips of the teeth for about $\frac{3}{16}$in. down the flanks. Slight pitting had occurred (i) on the main wheel near the pitch line above the scuffed area and at the aft end of the aft helix; (ii) on the outboard secondary pinion at the forward end of the forward helix and near the pitch line of the aft helix at a position 4½in. from the gap; (iii) on the inboard secondary pinion at the aft end of the aft helix. This is shown in Fig. 4. The cruising and primary gears were in a satisfactory condition. Further examination and measurement revealed:

i) that shaving of the secondary pinions had produced a profile which was proud at the pitch line and tooth tips with a hollow between, of 0·0003-0·0006in. in depth. The errors were greatest at the extreme ends of the pinions where most shaving had taken place in applying the helix corrections. It was considered that this had contributed to the pitting and scuffing observed.

ii) that the axial position of the pitting on the aft helix of the main wheel coincided with the junction of the corrected and true helices on the secondary pinions. Its position down the tooth corresponded to the pitch line.

Attempts to re-shave the pinions failed, the material having developed a high surface hardness during running. After hand stoning the scuffed areas, the pinions were replaced without further correction, to permit the continuation of main machinery trials. The difference in the pitting and contact marking on the secondary pinions had suggested slight inequality of load sharing and the gears were re-torqued accordingly.

The gearing continued in use until it had completed 166 hours of which about 30 hours were run at between 70 per cent and 100 per cent power. The scuffing practically polished itself out during this running but the pitting continued to spread until the gears were removed, although at no stage could the teeth be described as bad. The primary and cruising gears remained in good condition.

The complete gear set was returned to the makers for replacement of the secondary train by the ground pinions and shaved wheel originally made for the starboard unit. Grinding of the secondary pinions was resorted to as an expedient, pending the production of suitable shaving cutters.

For the original pinions, modifications to the shaving cutter profiles had been specified on the assumption that they would produce a conjugate shape on the gear being shaved and thus the desired gear tooth profile was only indirectly specified. The original shaving cutters were designed with 0·001-0·0012in. tip and root relief for both wheels and pinions and it was subsequently found that they left very little, if any, unmodified involute profile. For the new ground pinions, 0·001in. tip relief, running out tangentially 0·17in. down from the tip and a root relief of 0·0005in. running out tangentially 0·10in. from the end of contact was specified. Profile records of these pinion teeth are shown in Fig. 6.

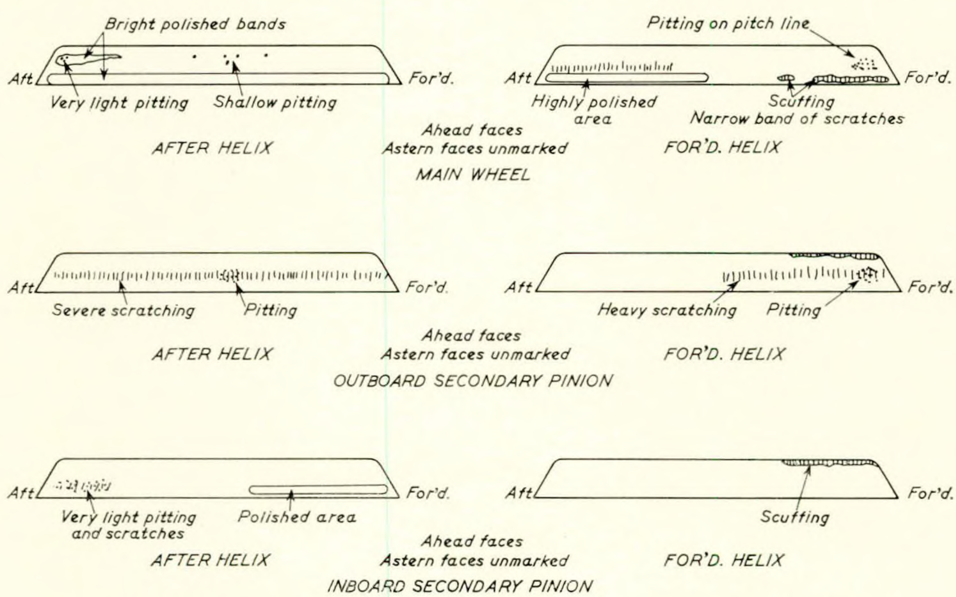


FIG. 4—*Y.100 gears—Condition of secondary gears after 7½ hours at full power*

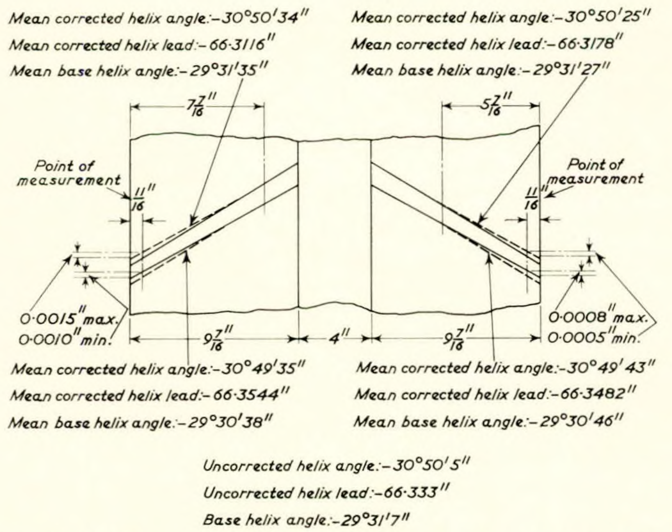


FIG. 5—*Y.100 gears—Modified helix corrections as applied to inboard final reduction*

In addition, both the magnitude and disposition of the helix correction were modified to reduce the tendency to pit at the junction of the true and corrected helices. Details of the revised corrections are shown in Figs. 5 and 6.

Examination of the original gears had shown evidence of fretting corrosion between the quill shafts and bushes in the bores of the primary wheels and secondary pinions, these bushes being provided only to limit the amount of sagging of the primary wheel and secondary pinion assembly during fitting. It was thought that interference at these positions and hence restriction of the gears within their bearing clearances, might have contributed to the pitting at the forward ends of the forward helices of the pinions and the clearance was therefore increased from 0·008in. to 0·030in.

Trials were resumed and the new secondary train gave a satisfactory performance, successfully completing 50 hours at full power and short periods at up to 130 per cent torque. Helix correction was considered justified and the tooth marking tended to corroborate the theoretical deflexion curves. Measurements of the gearcase distortion under load indicated that the forward secondary pinion bearings were slightly more rigidly supported than the aft bearings. This would tend to concentrate the loading towards the forward ends of both helices and could have been contributory to the scuffing and heavier pitting observed in the initial trials and also to the slightly heavier loading at the forward ends of the forward helices which appeared to persist during the final trials with the ground pinions. The effects of distortion in this design have been analysed and reported in detail by Waterworth[10].

b) *Bearings*: No troubles were experienced with the original journal bearings but after the first series of trials, bearings of reduced length were fitted. Comparative dimensions and loadings are given in Table II. The reduction in length was achieved by machining back the white metal at each end of the original bearing, leaving a running strip in the centre. These bearings performed excellently at all loads and speeds.

At the conclusion of the trials the gear set was installed in H.M.S. *Whitby* and its subsequent performance is mentioned later in this paper.

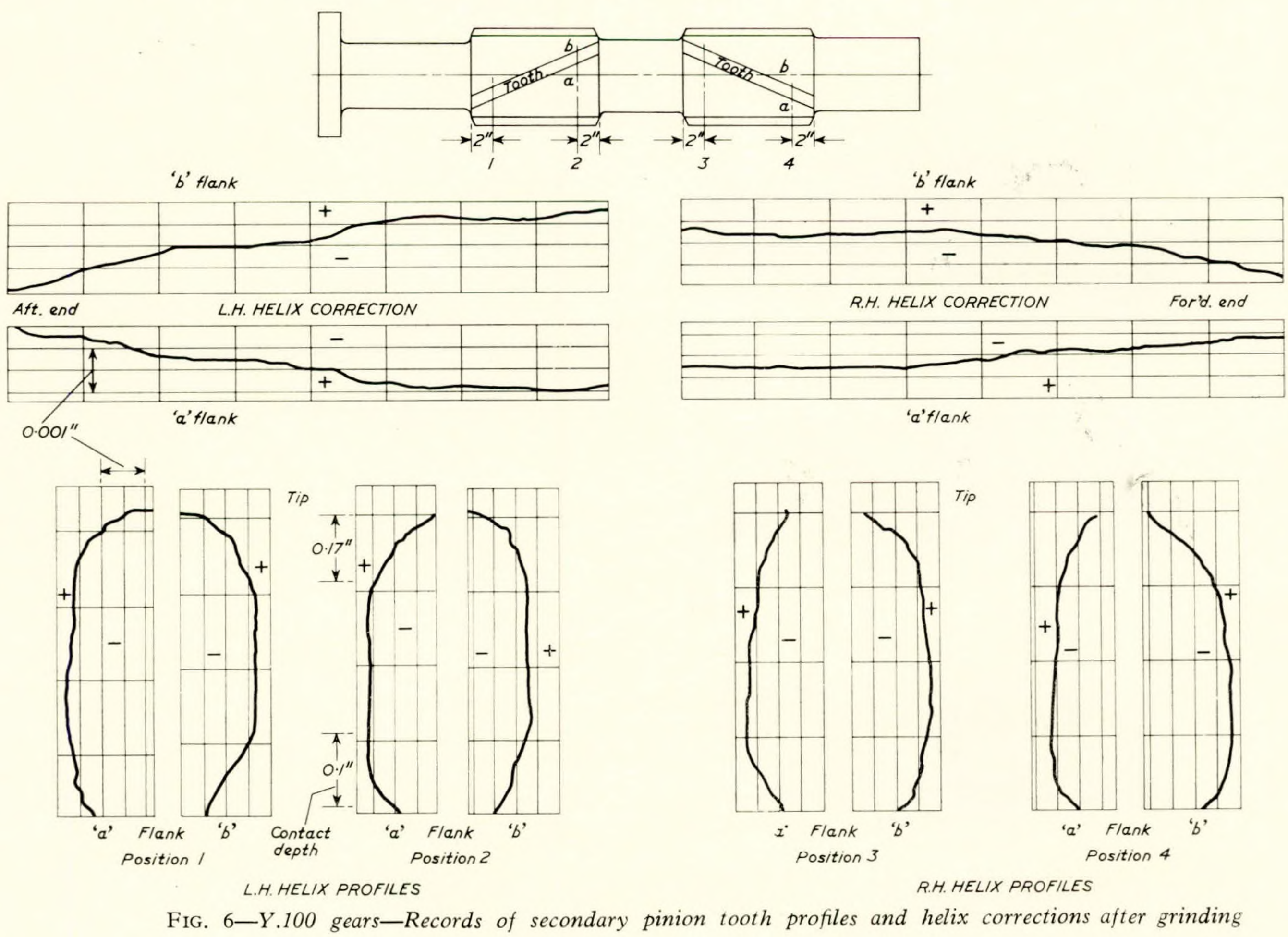


FIG. 6—*Y.100 gears—Records of secondary pinion tooth profiles and helix corrections after grinding*

TABLE II.—Y.100 GEARS. COMPARATIVE DIMENSIONS AND LOADINGS OF ORIGINAL AND REDUCED LENGTH BEARINGS.

Bearing	Journal diameter, in.	Bearing length		L/D ratio		Loading modified, lb./sq. in.
		Original, in.	Modified, in.	Original	Modified	
Cruising primary pinion	4	4	$1\frac{1}{2}$	$\frac{1}{1}$	$\frac{1}{2\cdot66}$	500
Main primary pinion	6	6	$3\frac{3}{4}$	$\frac{1}{1}$	$\frac{1}{1\cdot66}$	—
Cruising primary wheel	6	6	$3\frac{3}{4}$	$\frac{1}{1}$	$\frac{1}{1\cdot66}$	460
Cruising second pinion	6	6	$3\frac{3}{4}$	$\frac{1}{1}$	$\frac{1}{1\cdot66}$	460
Main primary wheel	$8\frac{1}{2}$	$7\frac{1}{2}$	3	$\frac{1}{1\cdot14}$	$\frac{1}{2\cdot8}$	430
Main second pinion	10	8	7	$\frac{1}{1\cdot25}$	$\frac{1}{1\cdot4}$	415
Main wheel	15	$11\frac{3}{4}$	$11\frac{3}{4}$	$\frac{1}{1\cdot28}$	$\frac{1}{1\cdot28}$	220

Y.100 Mark I Gears—Service Experience

Eighteen starboard and six port gear sets were required for the R.N. ships (six twin-shaft *Whitby* Class and twelve single-shaft *Blackwood* Class frigates) and manufacture was entrusted to a number of firms. Gear-cutting was undertaken by six firms, five of whom produced complete gear sets. In addition, another five firms manufactured gearcases but not gears.

The performance of these gears in the ships has been disappointing. In several sets, quite severe pitting of the secondary pinions occurred during the first few hours at full power. Usually, but not always, this was near the gap, i.e. on the uncorrected portions of the helices and a typical example is shown in Fig. 7. Minor scuffing was also experienced in some sets and it was therefore decided to use E.P. lubricating oils.

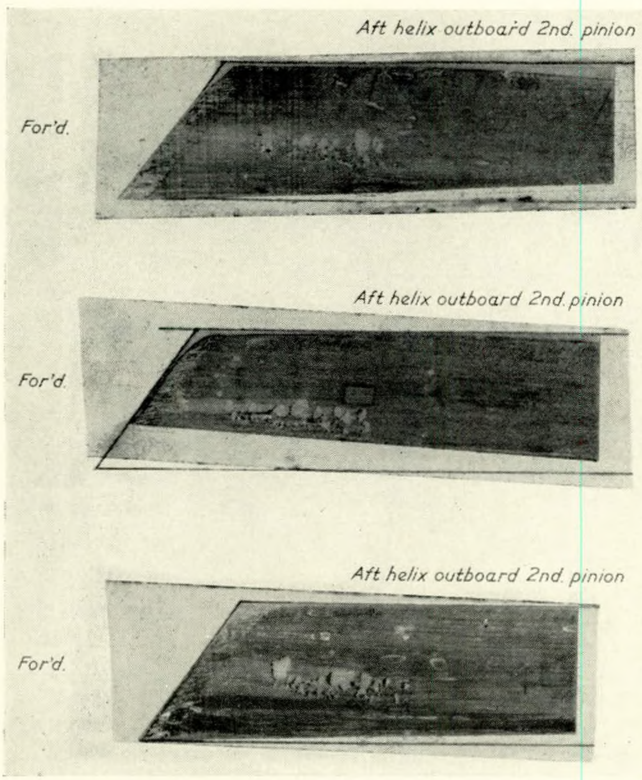


FIG. 7—*H.M.S.* Torquay *(Y.100 gears)—Selotape records showing pitting of secondary gears after 1 hour at full power*

The condition of the eighteen starboard and six port sets after contractors' sea trials was as follows:

Starboard Sets (18)	*Port Sets (6)*
10 pitted	1 pitted
4 minor scuffing	3 minor scuffing*
7 satisfactory	3 satisfactory

* Including the port gear set of H.M.S. *Whitby* which had been satisfactory during shore trials.

In all cases the condition of the primary gears was satisfactory. Since sea trials there has been further deterioration in the condition of most of these gears. The port and starboard gears of three of the twin screw *Whitby* Class vessels are still in excellent condition but the remainder (15 starboard and 3 port sets) have all pitted to varying extents. Also, several sets of primary gears now show fine pitting across the full face width in the vicinity of the pitch line. In addition, the secondary gears in a few sets have recently started to pit after being in good condition for a number of years.

In all sets, scuffing has been successfully eliminated by the continued use of E.P. oils.

A number of factors have been suggested to account for these variations in performance:

i) Gear-cutting errors, particularly in the application of helix correction. The problems encountered in shaving the first set have already been mentioned but it is thought that shaving errors cannot be held entirely responsible, since pitting has also occurred in ground pinions. In some cases there is evidence that excessive helix corrections were applied, resulting in overloading of the uncorrected portions and there is also evidence of excessive tip relief. It is perhaps significant that the successful gears of two of the three *Whitby* Class ships mentioned above were manufactured by the same firm and, in addition, the main gear wheels were lapped and not shaved. In cases where a firm manufactured several gear sets the performance of the later sets has invariably been better than that of the earliest sets made by that firm.

ii) Defects occurred mainly in the starboard sets whereas the port sets were comparatively trouble free. The gears are identical but the directions of rotation are different, the starboard sets running with the apices of the helices leading. With trailing apices, the helix corrections have an effect similar to that of end relief, in reducing the impact on entering mesh and it is possible that this is related to the better performance of the port gear sets. Recently, Boron and Welch[11] described experience with large primary gears, running with trailing apices, which suffered from heavy loading at the gap and they proposed a theory to explain the phenomena. This effect was not experienced in the Y.100 secondary gears and it is possible that other factors were more important.

iii) Gearcase distortion, measured during the shore trials at Pametrada, tended to increase the loading at the forward ends of both helices of the secondary pinions. As in the shore trials, all round chocking of the gearcase was employed in the ships but the greater flexibility of the seatings in the ships may have resulted in greater distortion and resultant maldistribution of load.

iv) The Y.100 gear materials were eventually tested in the S.G.B. gear test rig at Pametrada and, as described by Newman[6] the results were disappointing. It appeared that this combination of materials was prone to scuffing and showed a pitting resistance very much lower than was expected from considerations of hardness and ultimate tensile strength. In fact, later tests showed that the combination En 26/En 9 was more satisfactory in both respects.

v) Andersen and Zrodowski[12] and others have shown how the methods of aligning propeller shafting and main gear wheel can affect internal gear alignment on load and result in local overloading. During the trials of the first two *Whitby* Class ships there was evidence that the starboard forward main wheel journal was lifting into the top half of its bearing during high speed turns to starboard. The cause of this has never been established. Neither has the thermal rise of these gear sets nor the alignment actually achieved in a particular case been checked.

Regrettably, it has not been possible to investigate fully and explain the different behaviour of the gears in different ships. However, it seems reasonable to conclude that for the materials chosen and the standards of accuracy achieved during manufacture, the designed tooth loadings (270 K secondaries, 230 K primaries) were too high. Experience with these ships and the results of full scale shore trials suggest certain conclusions regarding through hardened, hobbed and shaved gears:

a) with the best available material combination (En 26 pinions and En 9 wheels are now favoured) the maximum permissible loads should not exceed 200 K for primary gears and 160 K for secondaries. Even at these loadings the margins of safety from pitting will not be great and first class manufacture (Grade A1 B.S. 1807:1952) and installation is necessary.

b) for warship propulsion gears where further reduction in size and weight are required, it becomes necessary to use surface hardened gears. Factors other than tooth strength may then influence the size and geometry of the gears, e.g. spread of turbines, and at present it seems likely that there will be no advantage in using loadings in excess of about 500 K. At such loadings surface hardened gears can have very substantial margins of safety from failure by pitting or tooth breakage, as indicated by the full scale gear tests carried out by A.V.G.R.A. and reported by Newman[6] and the excellent performance of the carburized and ground gears fitted in the *St. Laurent* Class frigates of the Royal Canadian Navy and reported by Nicholson[8].

Y.100 Mark II Gears

For the Y.100 Mark II gears fitted in the *Rothesay* and *Leander* Classes and the ships of a number of Commonwealth Navies some major changes were made, although the gear tooth geometry and loading were not altered:

i) Carburized, hardened and ground secondary pinions in En 36 steel were fitted.

ii) Port and starboard gears were handed so that both ran with the apices trailing.

iii) Prefinished medium wall bearings, with intermediate sleeves were fitted and, where necessary, these sleeves could be ground eccentrically to obtain correct alignment.

When tested in the A.V.G.R.A. second reduction test rig[6] this En 36 and En 30b material combination showed only a small improvement in load carrying capacity, for minor pitting of the wheel teeth began at 340 K.

However, more than thirty of these gear sets are in service and so far the only damage experienced has been caused by dirt and swarf left in bearing housings and oilways during manufacture. In addition, one of these ships completed two years' service, including approximately 50 hours at 90-100 per cent full power and throughout this period the inboard secondary pinion of the starboard gears had been running in a grossly misaligned condition. So far there is no evidence of damage to the pinion or main wheel. The misalignment has, of course, been corrected.

Nitrided Primary Wheels

One of these gear sets has been fitted, for trial purposes, with a carburized and ground En 36 primary pinion and nitrided and ground En 40c primary wheels. The face width of these gears has been reduced to raise the tooth loading to 450 K. This ship will go into service shortly.

Y.100 Mark III Gears

This design is a further development of the Y.100 Mark II and is being fitted in the latest *Leander* Class frigates. In view of the recent occurrence of pitting in the primary gears of some of the original Mark I gear sets, a carburized hardened and ground En 36 primary pinion has been specified. Other modifications have been adopted to permit more thorough flushing of internal oilways to remove dirt and swarf left in during manufacture and, because of difficulty experienced by some manufacturers in achieving satisfactory alignment, adjustable bearings will be fitted. In addition, the cruising gears will be omitted since these ships are not fitted with cruising turbines and the gearcase has been redesigned accordingly.

The Y.E.A.D. 1 Gears

The 30,000 h.p. Y.E.A.D. 1 gears (Fig. 8) represented the first British attempt to design and manufacture a carburized and ground marine propulsion gear set of large power. These gears and their associated turbines and boiler were extensively tested at Pametrada but were not fitted in any ships. The

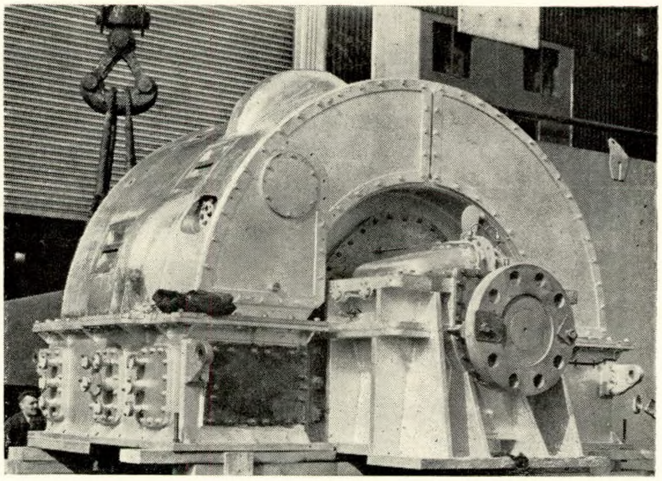


FIG. 8—*Y.E.A.D.1—External view of gearbox*

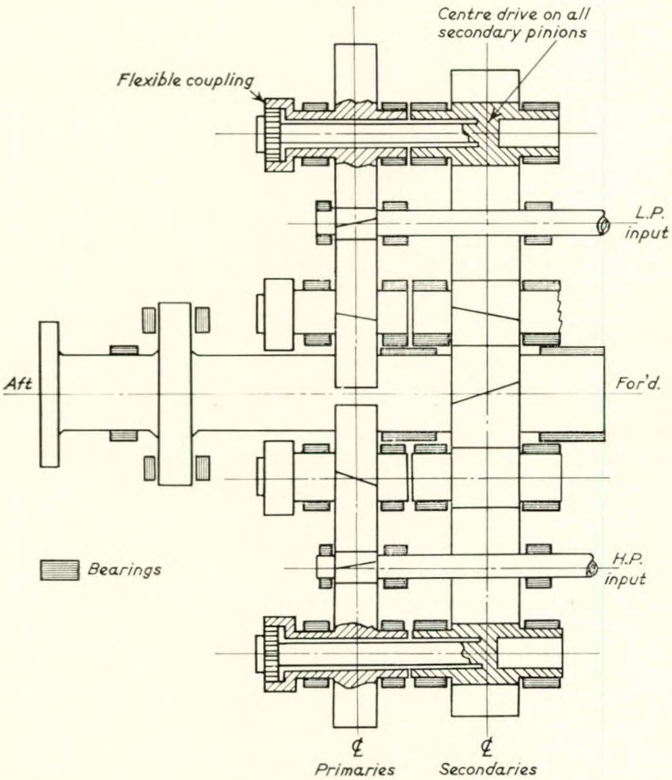


FIG. 9—*Y.E.A.D.1 gears—Arrangement of gears*

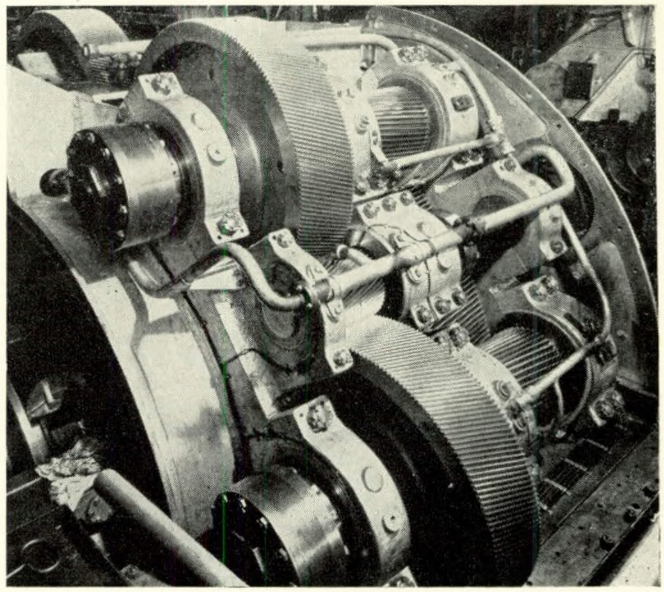


FIG. 10—*Y.E.A.D.1—L.P. gear train*

design was of the double-reduction, dual tandem, articulated type, with single helical, En 36a carburized, hardened and ground gears and ran at the following speeds at full power:

H.P. pinion	7,767 r.p.m.
L.P. pinion	6,023 r.p.m.
Main shaft	199·5 r.p.m.

The gears were required to run satisfactorily on OM100 lubricating oil which does not contain E.P. additives, and in view of the scuffing troubles experienced with the Royal Canadian Navy's carburized and ground Y.100 gears[(8)], it was decided to use somewhat larger pressure angles and smaller tooth pitches. The details are shown in Table III.

The layout of the gearing is shown in Figs. 9 and 10. It will be seen that the primary gears are situated at the aft end of the gearcase and the Michell thrust block, which was a separate unit, was bolted to the aft end. The reason for this arrangement was associated with the use of "three area support". The major forces to be resisted are those from the secondary gear tooth reactions and it is desirable that these should be transmitted to two areas of support abreast the main gear wheel via transverse strength members. It is convenient for the third area to incorporate the seating for the integral main thrust block, situated in the arch formed below the first reduction gears. If the first reduction gears are situated at the forward end the thrust block becomes inaccessible. A disadvantage of the Y.E.A.D. 1 type of arrangement is the need for a long high speed shaft to the primary pinions and the use of an additional bearing.

Transmission to the four secondary pinions was made via fine tooth couplings and quill shafts (Fig. 11) and an important feature of the design was the centre drive to the secondary pinions, intended to counterbalance the torsional deflexion by the bending deflexion and achieve even loading across the face width at full power. The fabricated steel gearcase was constructed in two parts, the lower part being a heavily ribbed box structure which carried the main wheel bearings (Figs. 12 and 16). The upper half was built in the form of three bridges over the main wheel shaft and the forward end of the thrust block, with the H.P. bearings on the outboard side of the bridges and the L.P. bearings on the inboard side. Each set

TABLE III.—Y.E.A.D. 1 GEARING. GENERAL DESIGN DATA FOR GEARS AND BEARINGS.

Particulars	H.P. primary pinion	L.P. primary pinion	H.P. and L.P. primary wheels	H.P. and L.P. secondary pinions	Main wheels
Number of teeth	38	49	210	46	324
Pitch diameter, in.	6·426	8·287	35·515	11·805	83·159
Diameter of addendum circle	6·776	8·636	35·801	12·366	83·610
Diameter of root circle	5·977	7·837	35·002	11·086	82·330
Normal pressure angle	25 deg.	25 deg.	25 deg.	22 deg. 30 min.	22 deg. 30 min.
Centre distances	20·971	21·900	—	47·479	47·479
Helix angle	19 deg. 46 min.	19 deg. 46 min.	19 deg. 46 min.	7 deg. 9 min.	7 deg. 9 min.
Normal pitch, in.	0·5	0·5	0·5	0·8	0·8
Circular pitch, in.	0·5313	0·5313	0·5313	0·8063	0·8063
Face width, in.	8	8	8	$13\frac{1}{8}$	$13\frac{1}{8}$
Load/inch of gear face, lb.	2,370	2,370	2,370	4,340	4,340
K factors	436	353	—	422	—
Bearing size, diameter and length, in.	5·25×3·1	5·25×3·1	9×4·5	10·5×6·375	14·5×12
Load/sq. in. projected area, lb.	448·5	454·5	427	495	493
Load per bearing, lb.	7,300	7,400	17,300	32,600	85,800

of steel bearing housings was cast in one piece and recessed and welded into the bridges (Fig. 13). Aluminium covers were fitted.

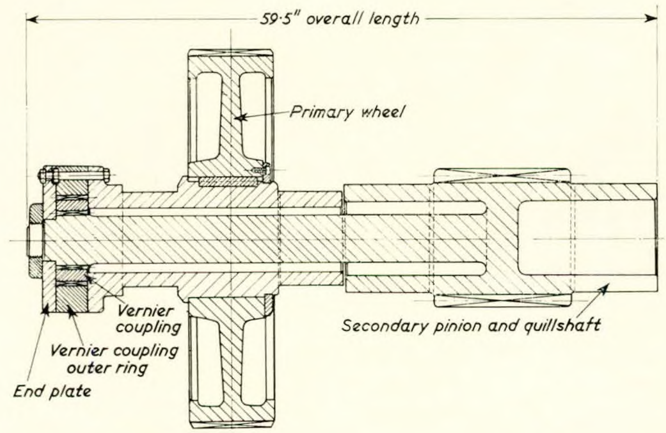


FIG. 11—*Y.E.A.D.1 gearing—Assembly of primary wheel and secondary pinion*

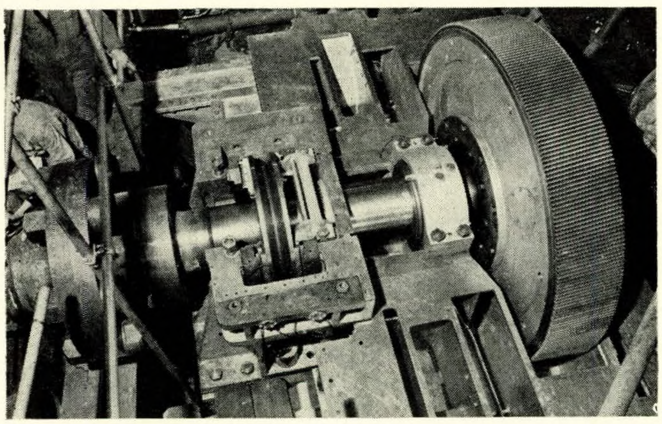


FIG. 12—*Y.E.A.D.1—Gearcase with top half removed*

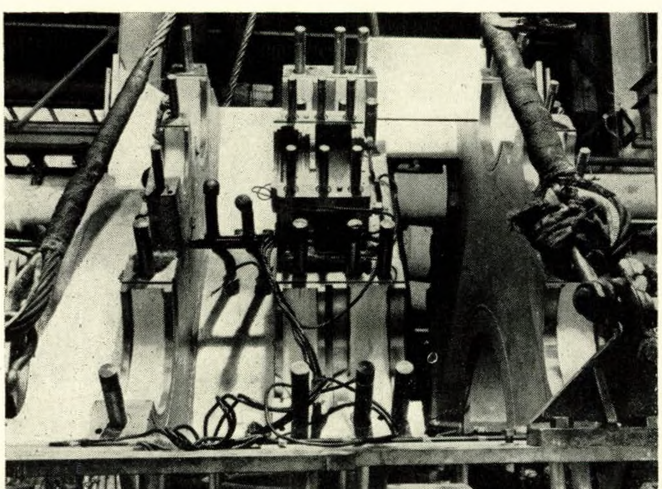


FIG. 13—*Y.E.A.D.1—Bearing housing and bridges*

The weight of the gearbox, complete, was 27½ tons; the overall dimensions are shown later in Fig. 33.

Gear Proving Trials

Gear proving trials were carried out over a range of powers from "light load" to full power, with instrumentation arranged for measuring bearing temperatures, conditions in the lubricating oil system and gearcase distortion. The lubricating oil used throughout the trials was OM100 and did not contain E.P. additives.

The gears performed very satisfactorily throughout the trials and the gear tooth surfaces remained in excellent condition with indications of full face contact at full power. Originally some of the high speed bearings were unreliable and ran hot but no further trouble was experienced after their diametral clearances had been increased from 0·007in. to 0·012in. Typical records of lubricating oil pressures, temperatures and flows are given in Table IV.

TABLE IV.—Y.E.A.D. 1. GEARING LUBRICATING OIL FLOWS, ETC. AT 100 PER CENT. POWER.

Bearings	Flow gal./min.	Pressure, lb./sq. in.
H.P. intershaft	52	11
L.P. intershaft	43	9
Primary pinion and thrust	69	7
Main wheel	4·6	9
Main thrust	22·3	4·5
Secondary gear sprayers	15·7	9·5
Primary gear sprayers	16·0	9·0
Total oil to gearcase	222·6	—
Distribution manifold pressure	—	13
Lubricating oil pump discharge	—	22·5
Lubricating oil cooler pressure drop	—	2
Cooler inlet temperature	153 deg. F.	
Cooler outlet temperature	120 deg. F.	

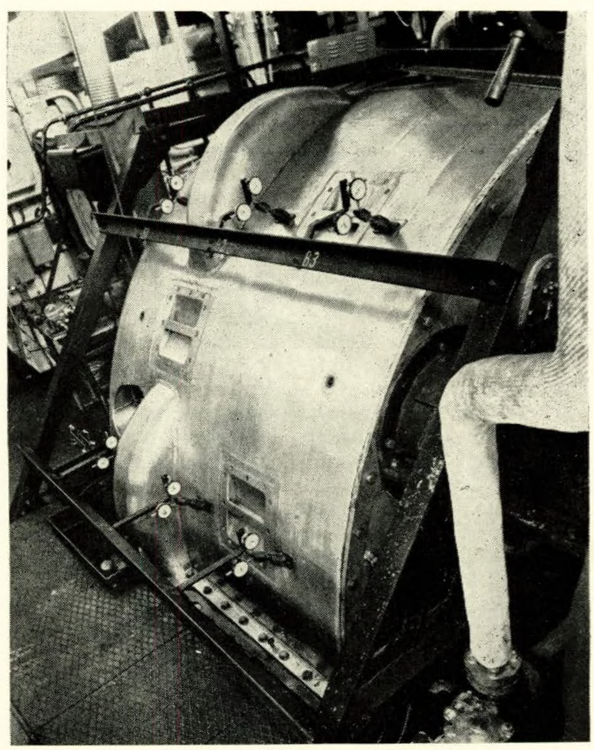


FIG. 14—*Y.E.A.D.1—L.P. side showing distortion frame and dial gauges*

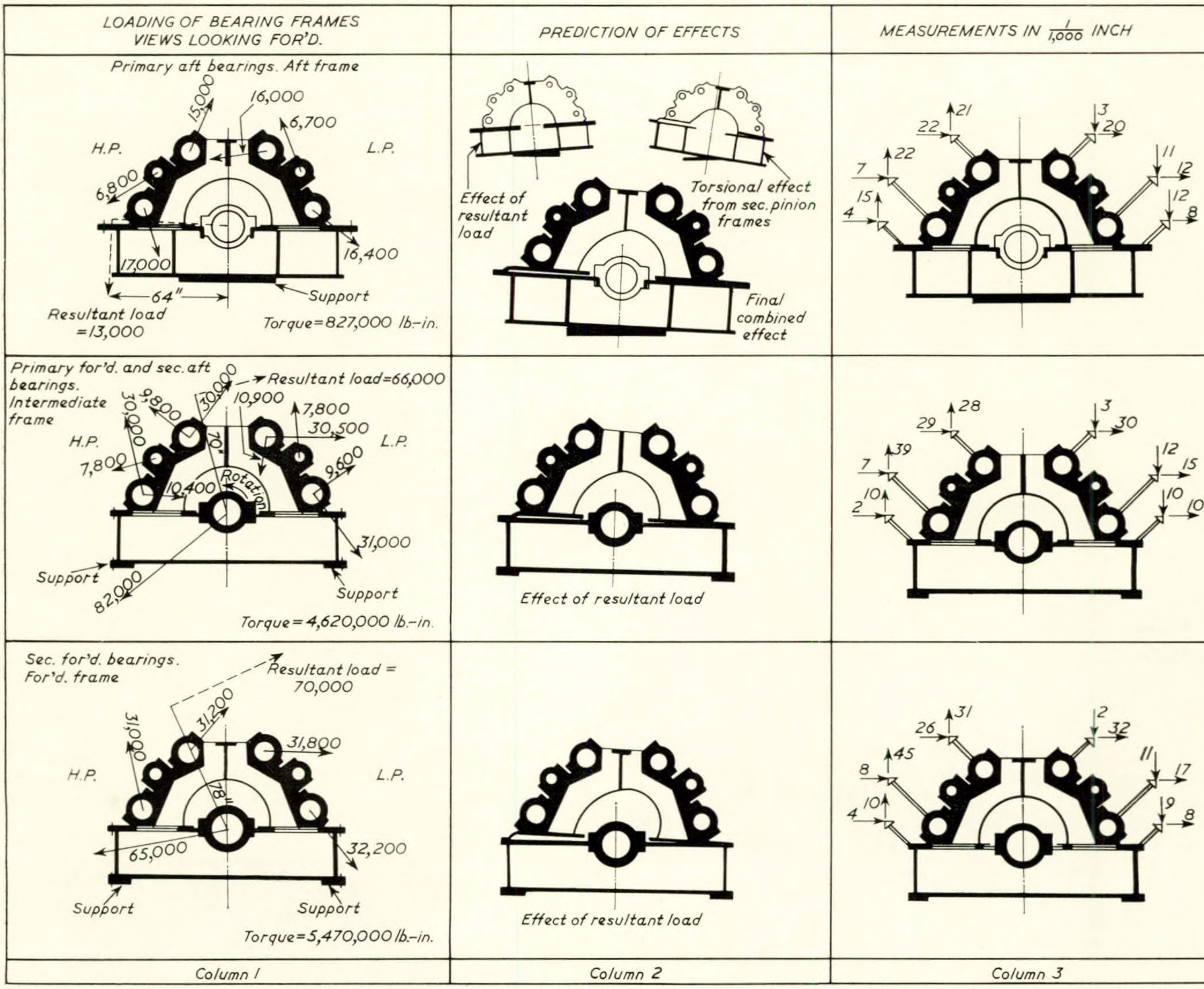


FIG. 15—*Y.E.A.D.I—Analysis of deflexion tests*

To measure distortion under load a rigid frame was built up around the gearcase from the test house floor. It was entirely independent of the gearcase and machinery seatings and carried dial gauges by means of which the absolute movement of several points on the gearcase could be determined.

Fig. 14 shows a portion of the distortion frame, the dial gauges and the measuring stalks which were screwed into tapped holes on the bearing keeps and passed out through clear holes in the gearcase covers sealed by brass bellows (0·005in. thick). The end of each stalk had two flat faces at right angles to each other against which the dial gauge spindles bore, these faces being machined in each case so that they were respectively horizontal and vertical.

Just before the end of each run, all the dial gauges were set to zero and the engines were stopped in about one minute. Immediately the turbines had come to rest the dial gauges were read again. The engines were then taken up to power once more in about one minute, dial gauge readings were taken and the procedure was repeated to obtain check readings. In this way the distortion of the gearcase caused by bearing loads alone was obtained while the thermal distortion effect was practically eliminated. In all, the vertical and horizontal movements of twenty points around the gearcase were measured and are shown, for the full power condition, in column 3 of Fig. 15. Column 1 of this figure shows the bearing loadings at each frame, the resultant of the loads in the top half of the gearcase being shown with a radius from the main gear wheel axis to define its position. An analysis of the effects of these forces on the complex structures of the gearcase (Fig. 16) would be lengthy and uncertain and will not be attempted here. However, visual inspection of the various loads and their directions suggests the effects shown in column 2 of Fig. 15 and these are confirmed by the recorded measurements in column 3.

The Y.102A and Y.111A Gears

The Y.102A and Y.111A gears are fitted in the new *County* Class Guided Missile ships and the *Tribal* Class General Purpose Frigates respectively. The main propulsion machinery of these ships consists of steam and gas turbines and the gear designs permit:

i) ahead and astern operation with the ahead and astern steam turbines;
ii) use of gas turbines to boost the ahead steam turbine power output;
iii) use of gas turbines alone for ahead power;
iv) use of gas turbines alone for ahead and astern operation, i.e. for manœuvring the ship.

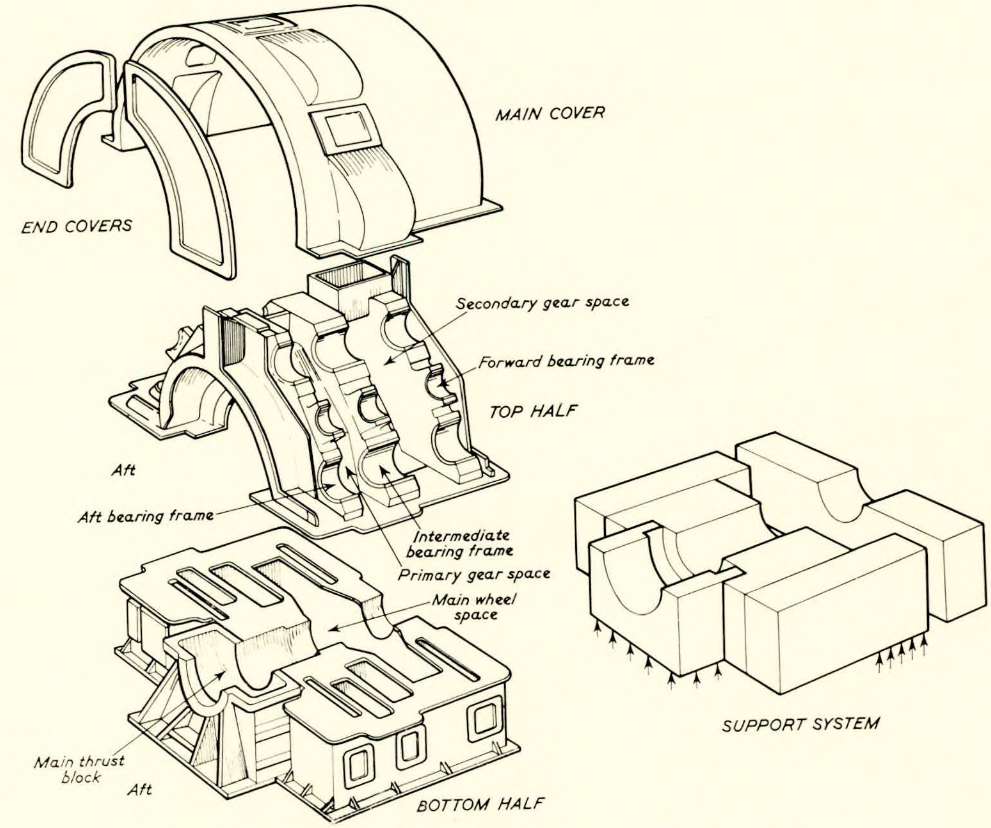


FIG. 16—*Y.E.A.D.1—Main structural components of gearcase*

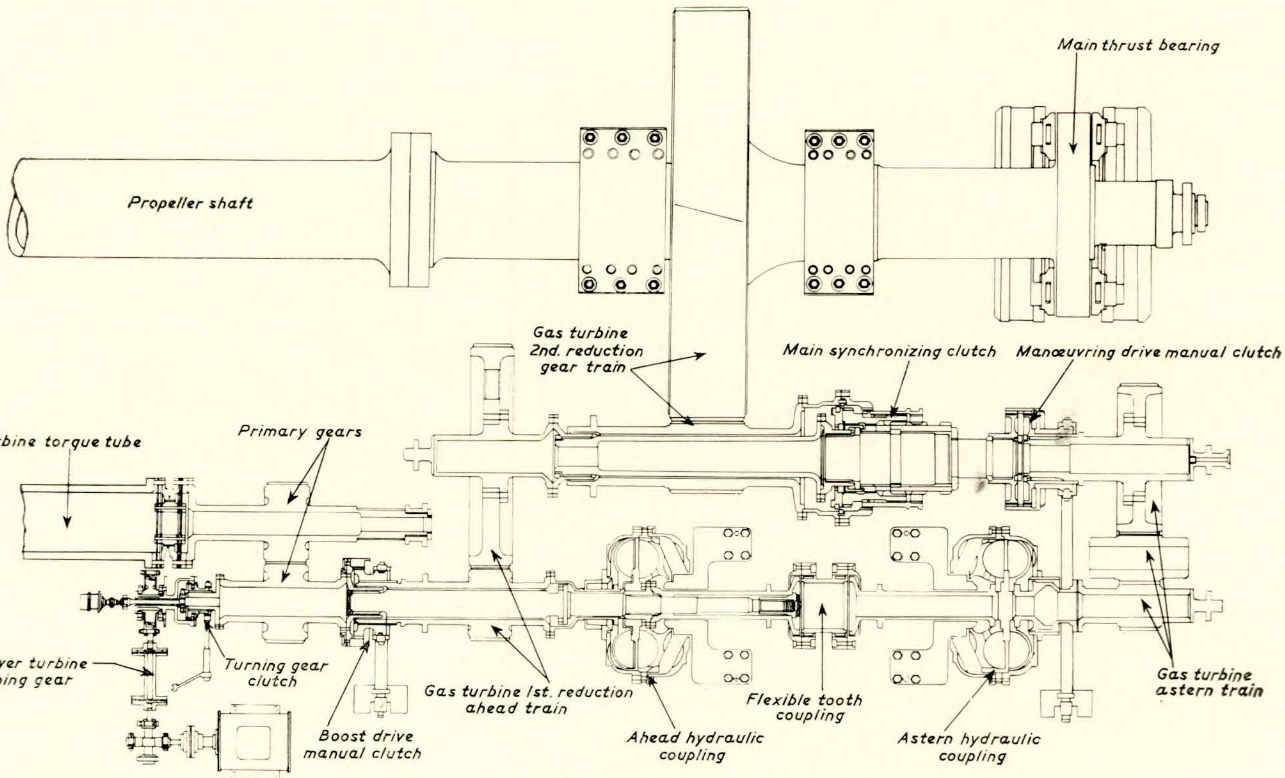
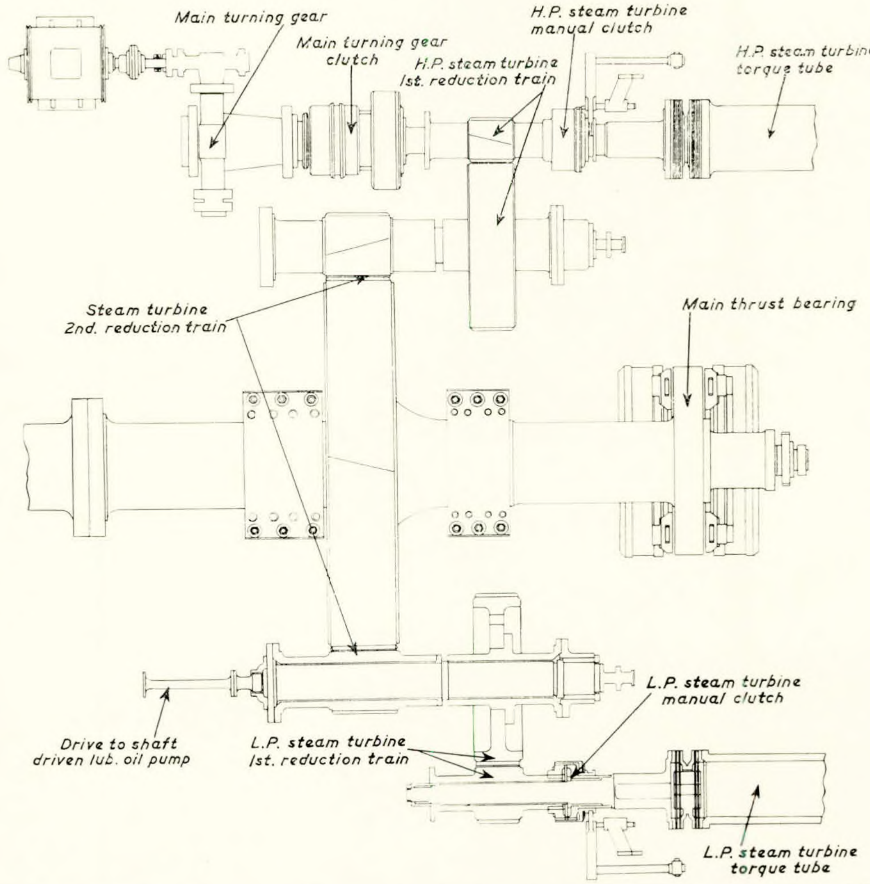
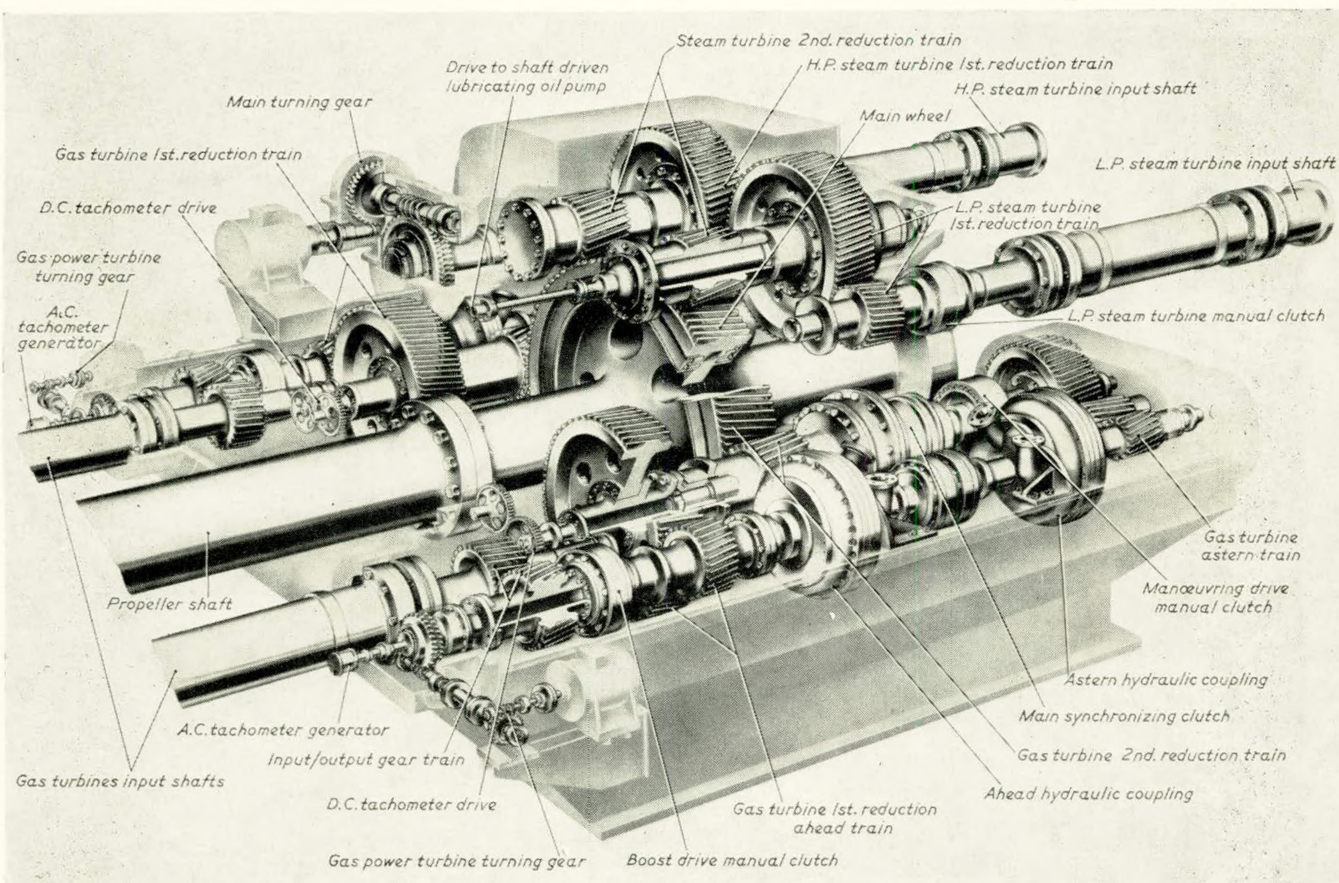


FIG. 17—*Y.102A—Port gearcase, inboard gas turbine gear train*



Right: FIG. 18—*Y.102A—Port gearcase steam turbine trains*

Below: FIG. 19—*Section view of Y.102A gearbox—Port set*



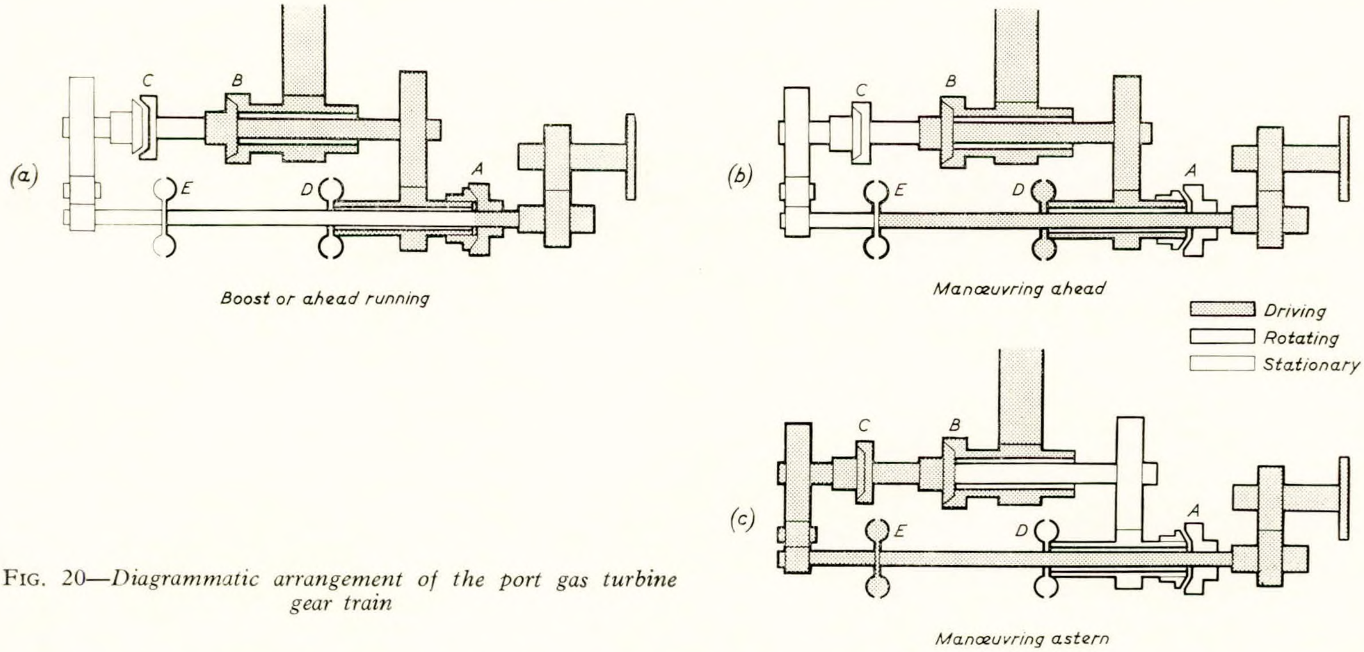


FIG. 20—*Diagrammatic arrangement of the port gas turbine gear train*

The machinery installations of these ships have already been described by Good and Dunlop(13) who also gave an account of the extensive shore trials carried out on the prototype gas turbines and Y.102 gearbox and of sea trials experience in the first ships of each Class. In order to present a complete picture, some of the information given in their paper will be repeated below but, as far as possible, the authors will confine their remarks to the more detailed aspects of these gears which have not yet been published.

General Description

The general layout of the Y.102A gears is shown diagrammatically in Figs. 17 and 18 and in a section view in Fig. 19. The steam turbine gear trains and gas turbine boost gear trains

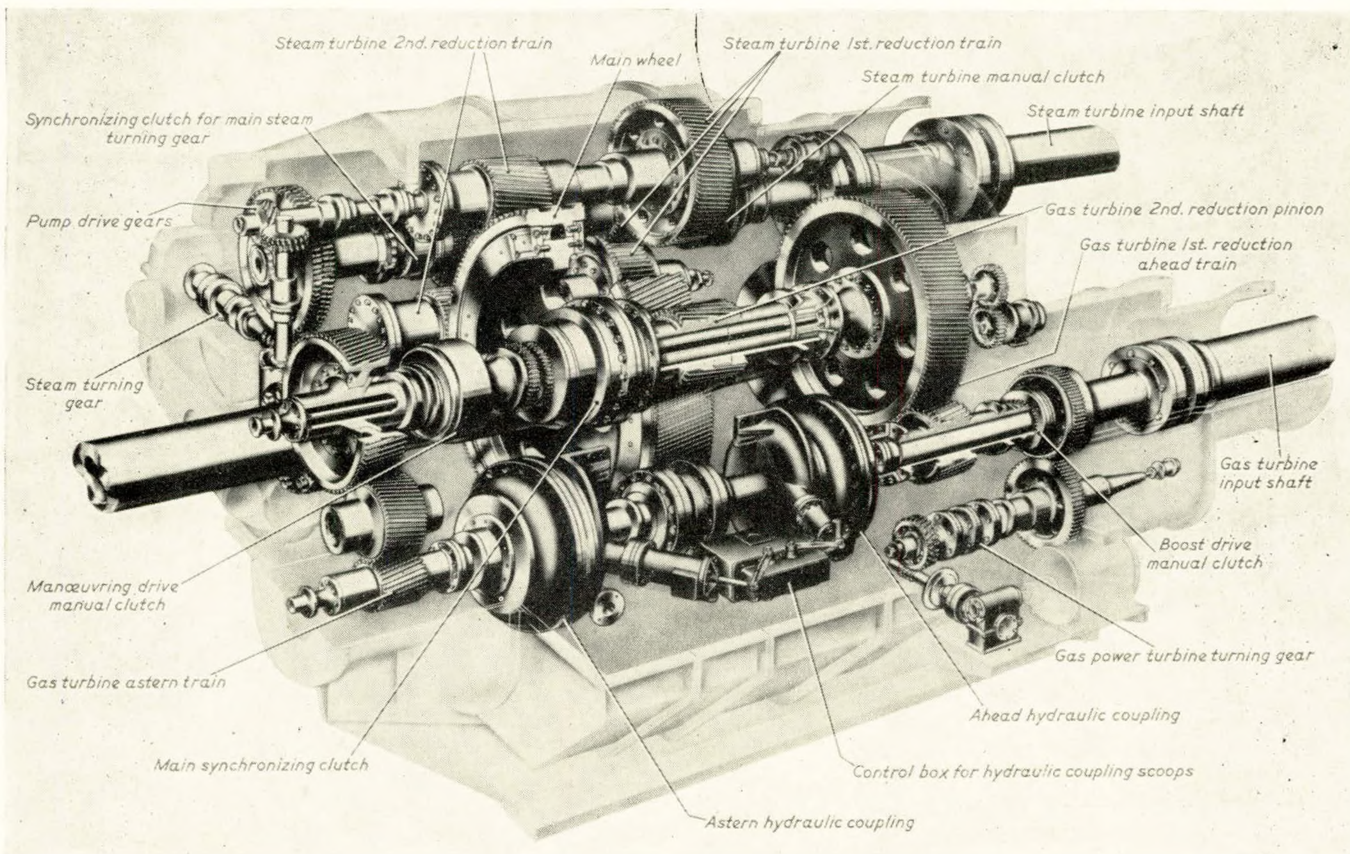


FIG. 21—*Section view of Y.111A gearbox*

are of the double-reduction, tandem articulated type and the manœuvring drive from each gas turbine comprises ahead and astern gear trains with associated hydraulic couplings, manually operated clutches and main synchronizing clutches. The main synchronizing clutches permit the gas turbines to be connected to the propeller shaft when it is rotating in the ahead direction. They engage automatically at synchronism and are then locked into engagement so that they are capable of transmitting ahead or astern torques. A diagrammatic representation of one of the gas turbine gear trains is shown in Fig. 20, from which the various operating functions of the gas turbines can be visualized. Thus, if a gas turbine is required for boost or ahead running only, the drive is taken through the boost drive manual clutch (A) which is engaged, through the first reduction ahead gears to the main synchronizing clutch (B) and thence through the second reduction pinion to the main gearwheel. The manœuvring drive manual clutch (C) remains disenaged.

When required for manœuvring, the gas turbine drive is connected to the hydraulic couplings (D) and (E) with the manœuvring drive manual clutch (C) engaged and the boost drive manual clutch (A) disengaged. Oil is supplied to either the ahead or astern hydraulic coupling as required.

Fig. 21 illustrates the Y.111A gearbox, which is similar in principle but simpler, since only one gas turbine and one steam turbine are involved. The steam turbine gear trains are, however, of the dual tandem, articulated type.

Hydraulic Couplings

A diagrammatic arrangement of a pair of hydraulic couplings with their associated control gear is shown in Fig. 22. A sectional arrangement of an ahead coupling is shown in Fig. 23. The couplings are designed so that oil supplied from the main lubricating oil system is continually flowing through the working circuit via ports in the impellers into the rotating scoop chambers where it is picked up by the scoop tubes and returned to the sump. Thus the amount of oil in the working circuit is determined by the radial position of the tip of the scoop tube. An hydraulic cylinder connected directly to the top of the scoop tube determines its radial position. The valve gear in the control unit controls the admission of oil to the two hydraulic cylinders (ahead and astern). A single hydraulically-operated double acting cylinder controls, via links and levers, the scoop positioning valves and also the oil diverter valve which diverts oil from one coupling to the other. When running in the boost condition or on steam turbines alone the main flow of oil to the couplings is shut off but a bypass orifice provides a reduced oil flow for cooling, which escapes through leak off orifices in the peripheries of the couplings.

The hydraulically-operated double acting cylinder has three positions: ahead, neutral and astern which give, respectively:

a) Ahead scoop withdrawn and ahead coupling filled with oil. Astern scoop extended, coupling empty and rotates at 200 per cent slip less the slip in the ahead coupling.
b) Both scoops partially withdrawn and the working circuits of both couplings partially filled. Both couplings rotate at 100 per cent slip. Diverter valve admits reduced flow of oil to both couplings.
c) Astern scoop withdrawn and astern coupling filled

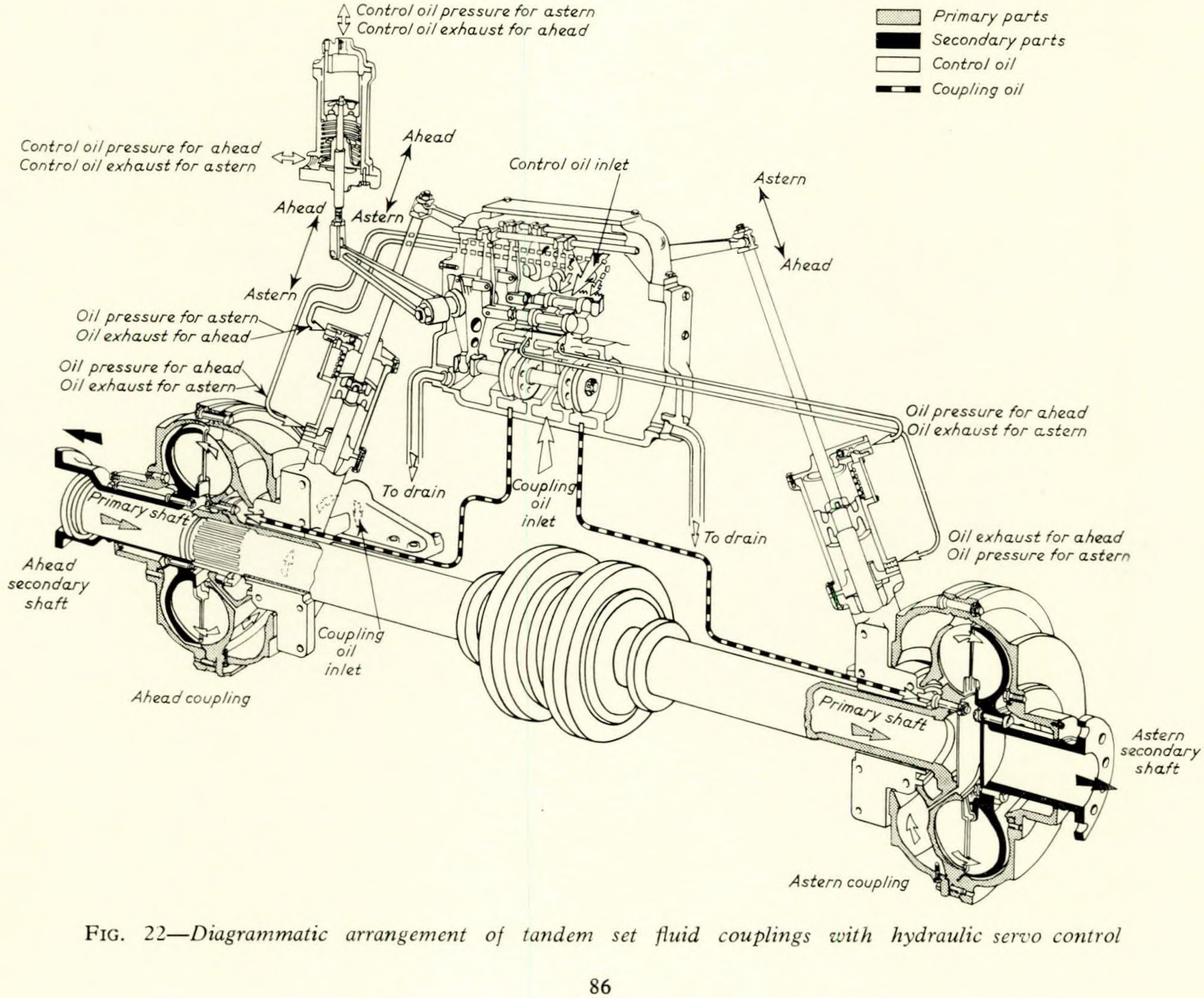


FIG. 22—*Diagrammatic arrangement of tandem set fluid couplings with hydraulic servo control*

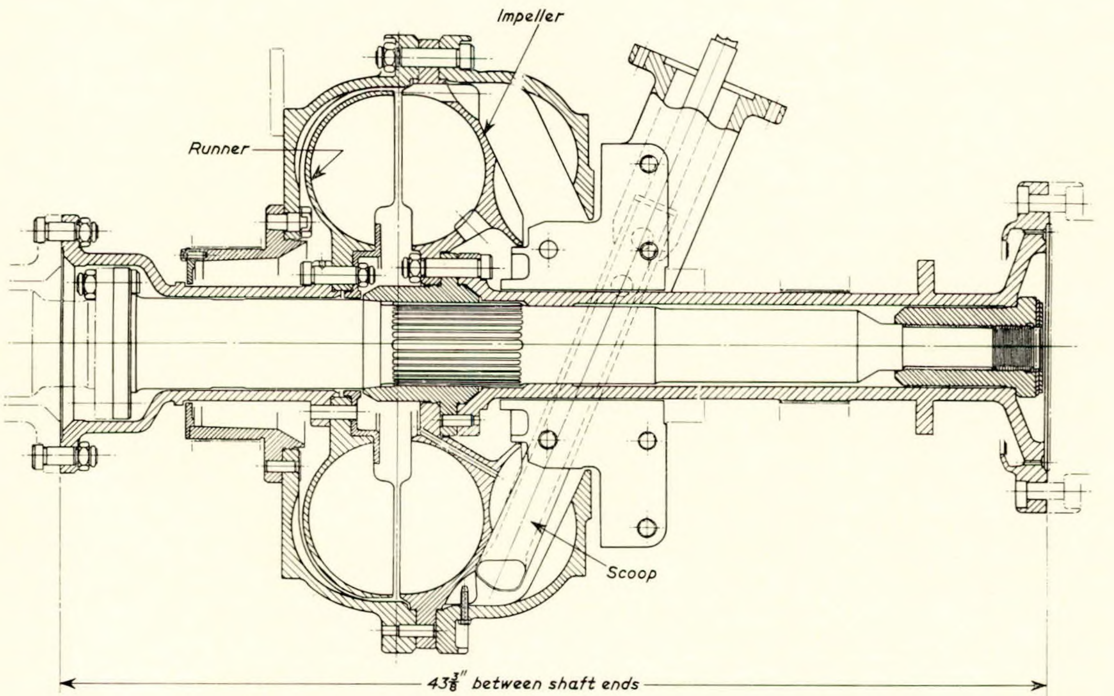


FIG. 23—*Ahead hydraulic coupling—Sectional arrangement*

with oil. Ahead scoop extended, coupling empty and rotates at 200 per cent slip less the slip in the astern coupling.

When manœuvring, the output from the gas turbine is restricted to 3,500 h.p. both ahead and astern, in order not to exceed the permitted maximum operating speed of the hydraulic couplings. In both classes of ship the corresponding propeller shaft r.p.m. are in excess of 50 per cent full power r.p.m. so that the ships' speeds and manœuvring capabilities are adequate. This limitation is imposed mainly by centrifugal stresses in the hydraulic couplings.

The Main Synchronizing Clutches

The main synchronizing clutches are vital components of these boost installations and, although this paper is concerned primarily with gearing, it is believed that a short account of these clutches will be of interest.

It has already been mentioned that an automatic clutch was fitted in the Y.100 machinery installations, to connect and disconnect the cruising turbine when reducing and increasing power. This clutch was actuated by a frictional speed sensing device and could not be locked in engagement. Although it performed satisfactorily under the limited conditions of shore trials, it was unsatisfactory in service at sea.

A new design, of the synchro-self-shifting type, with positive pawl actuating means, was developed and gave satisfactory service during prolonged trials in H.M.S. *Scarborough* and *Keppel*. However, changes in the operational roles of

TABLE V.—Y.102A GEARING. LEADING DESIGN DATA.

	Gear	Pitch circle diameter, in.	No. of teeth	Normal pitch, in.	Face width, in.	Helix angle	Normal pressure angle	K factor (maximum)
Steam	H.P. first reduction pinion H.P. first reduction wheel L.P. first reduction pinion L.P. first reduction wheel	7·96 33·54 11·65 33·54	28 118 41 118	0·8	8·5	26 deg. 23 min.	20 deg.	424 312
	H.P. second reduction pinion L.P. second reduction pinion Main gear wheel	12·83 12·83 71·92	33 33 185	1·2	13	10 deg. 43 min.	20 deg.	419 464
Gas	Primary input wheel Primary output wheel	13·61 13·61	52 52	0·8	7	13 deg. 24 min.	20 deg.	379
	First reduction pinion First reduction wheel	12·04 38·48	46 147	0·8	7	13 deg. 24 min.	20 deg.	314
	Second reduction pinion	12·83	33	1·2	13	10 deg. 43 min.	20 deg.	424
	Astern pinion Astern idler Astern wheel	8·12 8·38 25·65	31 32 98	0·8	7	13 deg. 24 min.	20 deg.	819* 530

***Note:** Full power astern on one gas turbine (emergency condition)

TABLE VI.—Y.111A GEARING. LEADING DESIGN DATA.

	Gear	Pitch circle diameter, in.	No. of teeth	Normal pitch, in.	Face width, in.	Helix angle	Normal pressure angle	K factor (maximum)
Steam	First reduction pinion First reduction wheel	8·81 28·91	32 105	0·8	7	22 deg. 20 min.	20 deg.	455
	Second reduction pinion Main gear wheel	12·83 71·92	33 185	1·2	13	10 deg. 43 min.	20 deg.	336
Gas	First reduction pinion First reduction wheel	12·04 38·48	46 147	0·8	7	13 deg. 24 min.	20 deg.	331
	Second reduction pinion	12·83	33	1·2	13	10 deg. 43 min.	20 deg.	441
	Astern pinion Astern idler Astern wheel	8·12 11·78 25·65	31 45 98	0·8	7	13 deg. 24 min.	20 deg.	602 356

these frigates now require the maximum endurance at higher speeds and the cruising turbines have therefore been removed from the majority of the ships.

The main synchronizing clutches fitted in the *Ashanti* (Y.111A) and *Devonshire* (Y.102A) Classes are also of the synchro-self-shifting type and the selection of the design was influenced by experience of the operating requirements obtained with the earlier Y.100 cruising turbine clutch. A description of the clutch and an account of the clutch trials carried out ashore will be found in the Appendix to this paper. Although, at the time of writing, seagoing experience is limited to approximately one year's service in the first ship and sea trials of three later ships, the performance of this clutch has been entirely satisfactory.

Gearing Design Details

The gears of the Y.102 shore trials set and in the majority of ships' sets are in En 36 steel and are carburized, hardened and ground. In some of the ships' sets, a number of induction hardened and ground gears in En 24 steel and nitrided and ground gears in En 40c have been installed. All gears are single helical. Leading design data of the gears are given in Tables V and VI. It will be seen that, under normal ahead operating conditions the tooth loading does not exceed 465 K but that very much higher loads can be realized when manœuvring on gas turbines. In emergency, when manœuvring at full power astern on one gas turbine alone, the associated astern gear train in the Y.102A design is loaded to 819 K.

The need for helix correction was examined in the initial design stage and, in addition to bending and twisting, slew of the pinion and wheel due to side thrust of the single helix was also considered. Calculations indicated that the slew effects might be corrective and, in addition, earlier experience with single helical gears had not revealed maldistribution of load from these causes. In view of this and the lack of data on the attitudes of journals when slewed in their bearings and of the fact that adjustable bearings were to be fitted, it was decided not to apply helix corrections.

Gear Proving Trials

Gear proving trials were carried out during the Y.102 shore trials, at no load, half load and full load, to prove the bedding of the teeth under various loads and to determine whether or not helix corrections were necessary. The bedding obtained extended fully across the face widths of the teeth and it was concluded that the stiffness of the gearcase was adequate, that helix corrections were unnecessary and that the methods of manufacture, tolerances, alignment and inspection had been satisfactory. A point of special interest is that during these trials the outboard gas astern pinion and idler ran for one hour at 819 K. The bedding was good and the teeth remained in excellent condition. A photograph of these gears is shown in Fig. 24.

To date, the Y.102 shore trials gears have completed over 1,000 hours running including 110 hours at full power and 33½ hours continuously at 130 per cent full torque. The gears are in excellent condition. E.P. lubricating oils have been used throughout the trials.

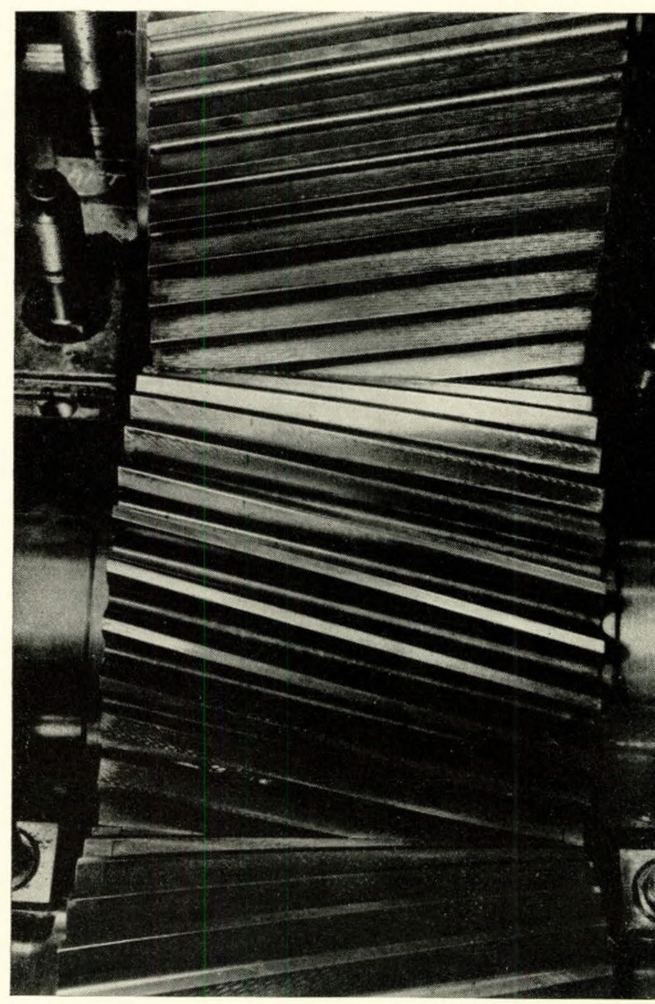


FIG. 24—*Y.102 outboard astern manœuvring gear train after completion of bedding trials*

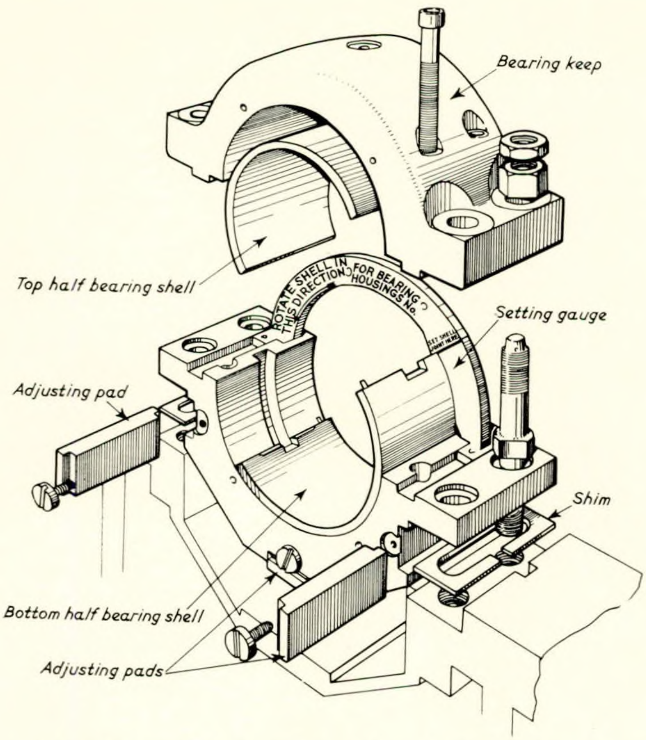


FIG. 25—*Typical bearing housing showing shell and setting gauge*

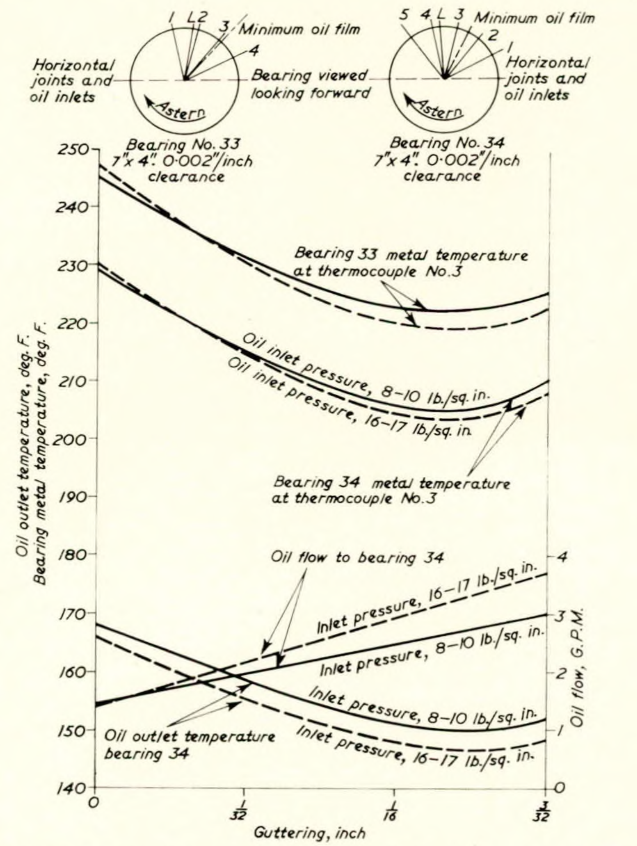


FIG. 26—*Effect of guttering and oil inlet pressure on metal temperature and oil flow at 3,500 h.p. and 171 r.p.m.*

Journal Bearings

The majority of the journal bearings comprise adjustable steel bearing housings into which are fitted pre-finished medium wall white metal lined steel shells. An interference fit is provided between shells and housings, the two parts of the latter being clamped together by socket headed bolts and studs as shown in Fig. 25. The correct radial position of the shell joint and oil inlets is achieved with a setting plate and maintained by a dowel in the bearing keep registering with a counterbore in the shell. Adjusting pads are fitted and permit true alignment during manufacture. Thermocouples located as near as possible to the positions of minimum oil film thickness are mounted radially in the bearing keeps and are embedded in small white metal plugs, held in contact with the bearing shells by means of screwed sleeves and springs.

With the exception of the main gear wheel bearings, all journal bearings are provided with two diametrically opposed oil inlet holes at the shell joints. During the early shore trials a number of high speed bearings wiped or ran at high temperatures. After repositioning these bearings to provide a greater arc between oil inlet and load line and, in some cases, increasing the diametral clearances from 0·0015in. to 0·00225in. per inch diameter, further troubles were avoided.

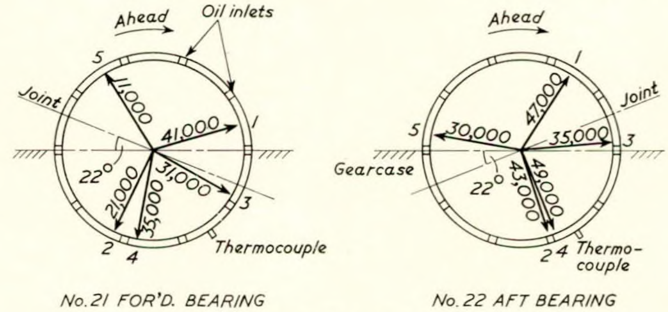
A useful expedient was resorted to on a number of occasions, in order to reduce high bearing temperatures, when time did not permit bearings with larger clearances to be obtained. This involved guttering the joints at 45 deg., to a depth of approximately $\frac{1}{16}$in., the gutters running axially from the oil inlets to the ends of the bearing. The effect is not only to increase the oil flow, as would be expected, but also to reduce the white metal temperature. To evaluate this practice, trials were carried out on the highly loaded, high speed, gas astern idler gear bearings without guttering and with gutters $\frac{1}{16}$in. × 45 deg. and $\frac{3}{32}$in. × 45 deg. The results are shown in Fig. 26 and it is seen that the optimum amount of guttering in this case was $\frac{1}{16}$in. and, with an oil inlet pressure of 8-10lb./sq. in. the white metal temperature was reduced 20 deg. F. approximately, whilst the oil flow was increased from 1·5 to 2·5 gal./min.

Main Gear Wheel Bearings

In a boost installation of this type the various possible running conditions both ahead and astern, result in a multiplicity of load lines (Fig. 27) in the main gear wheel journal bearings, such that it is difficult to obtain a safe location for a single oil inlet.

Trials of a conventional bearing (19in. diameter × 12$\frac{3}{4}$in. long, 0·02in. diametral clearance and single oil inlet 2in. × 1$\frac{1}{2}$in. with spreader) were carried out at 124 r.p.m. and oil inlet pressure 10lb./sq. in., the bearing being loaded to 232lb./sq. in. The direction of loading was varied and it was found that the bearing wiped when the load line approached within 40 deg. (approximately) of the oil inlet, in both the leading and trailing conditions.

It was therefore decided to test a bearing provided with



1) Steam ahead
2) Steam astern
3) Steam and gas ahead
4) Gas ahead
5) Gas astern

FIG. 27—*Y.111A—Main wheel bearing showing load lines*

a central circumferential groove, 1in. wide × ¼in. deep, with twelve equally spaced oil inlet holes each 1in. in diameter. The bearing was loaded to 345lb./sq. in. with oil inlet pressure 10lb./sq. in. and at this loading performed satisfactorily throughout the speed range 20-184 r.p.m. A further test involved four hours running at 1½ r.p.m. at 50lb./sq. in. loading. Evidence of slight rubbing was found afterwards but the bearing was undamaged. This type of bearing, with a circumferential oil groove, was subsequently fitted in the ships' sets. Experience so far has been entirely satisfactory.

Sea Trials Experience in H.M. Ships Ashanti, Nubian *and* Devonshire

Good and Dunlop[13] have given an account of the extensive sea trials of the first three ships and have described the techniques of operating the gas turbines, clutches and hydraulic coupling controls when boosting and when manœuvring on gas turbines. In this paper, no attempt will be made to describe the manœuvring trials in greater detail or to analyse the large number of records obtained. A separate paper would be required to cover this subject and it is thought that such a paper, given by the designers of this machinery, would be of considerable interest.

In all ships the performance of the gears themselves was very satisfactory, the tooth flanks generally showing good bedding and being in excellent condition after the trials. The starboard gear set in H.M.S. *Devonshire* contained a number of induction hardened gears, including the main gear wheel and these also were in excellent condition.

In the authors' opinion the sea trials of these ships have shown that the gears, including the hydraulic couplings and main synchronizing clutches, fulfil the design requirements and, notwithstanding their complexity, will prove to be robust and durable. In stating this opinion, it is, of course, assumed that the specified standards of manufacture will, in all cases, be achieved and that the standards of operation and maintenance will be suited to the precision of the machinery.

Brave *Class—Epicyclic Gears*

Although many epicyclic reduction gears are giving satisfactory service in turbo-generators and other auxiliary machinery in the post-war Fleet, they have so far been used for main propulsion gears only in a limited number of smaller vessels. Of these, the most interesting application has been in the *Brave* Class fast patrol boats, in which use is made of epicyclic gearing not only to give a primary reduction between the marine Proteus gas turbines and flexible cardan shafts, but also in the reverse/reduction gearboxes which incorporate carburized, hardened and ground spiral bevel vee drive gears. In this type of warship the use of small, lightweight, high powered machinery is of paramount importance and the epicyclic gear has proved to be extremely suitable for this application.

Fig. 28 shows a sectional arrangement of the reverse/reduction gear, which comprises three basic units; the vee drive bevel gear unit, the reverse/reduction gear consisting of two epicyclic gears in series and a parallel shaft gear unit. The construction of the epicyclic units follows the well known Stoeckicht principles and has already been described by Allen and Jones[15]. Nitrided sun wheel and planet wheels are fitted, the latter revolving on white metal faced spindles. The gear is capable of transmitting 3,850 h.p. through a speed reduction of 4,988-1,707 r.p.m. and is designed to transmit full power ahead or astern, thus permitting handed propellers. The total weight of the gearbox is less than 2 tons and the overall length is 6ft.

Ahead or astern operation is achieved by holding one or other of the two brakes, which comprise hydraulically actuated steel and sintered bronze shoes bearing on spheroidal graphite cast iron drums. In the neutral position both brakes are freed. Before installation in the first boat, the gears were run up to full speed, in both directions, ashore but unfortunately it was not possible to carry out any running under load. It is not, perhaps, surprising that the early sea trials at full power resulted in two failures of the planet spindle bearings in the first train of the port gearbox. In each case the damage was confined to the gear train in which the failure occurred. The trouble was cured by the addition of a separate, motor driven lubricating oil pump operated by a pressure switch to supplement the supply of lubricating oil at low speeds and by the provision of slightly greater oil clearance and an additional oil supply hole in the planet spindle bearings.

Further sea trials were entirely satisfactory and fully demonstrated, not only the flexibility of the machinery instal-

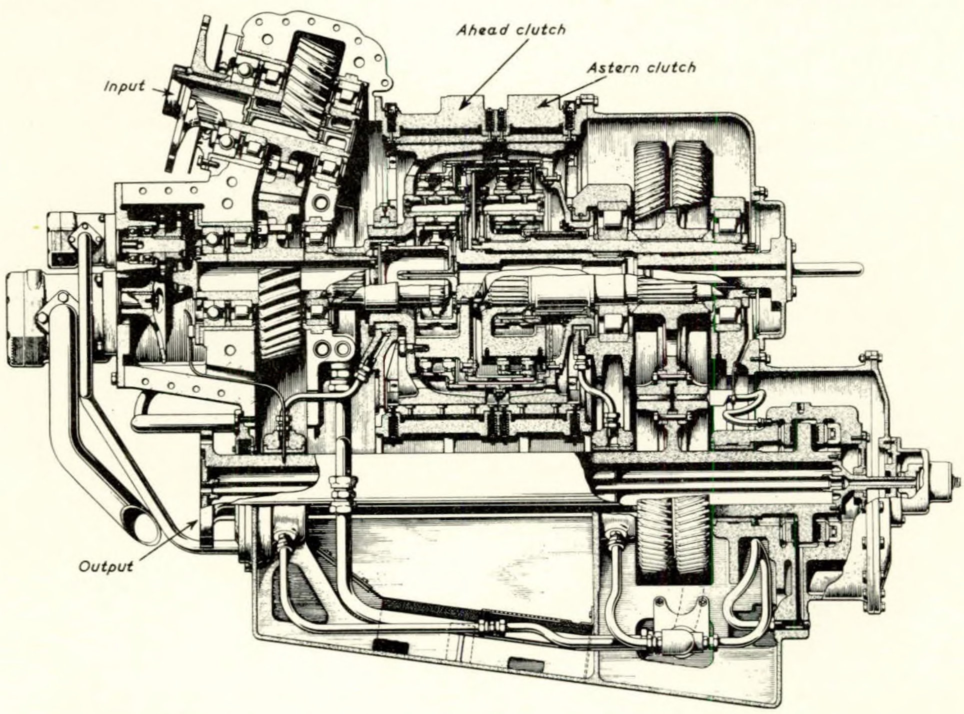


FIG. 28—Brave *Class fast patrol boat's reverse reduction gear*

lation as a whole but also the robustness of the gears. For example, the boats have been accelerated from rest to full power in 30 seconds, then decelerated to "stopped in the water" in the same interval. On one occasion when proceeding on three engines at more than 30 knots, one of the gear levers was inadvertently moved from the ahead to the astern position. The gearing and boat were heard to protest and the mistake was rectified. No troubles were experienced with this gearbox, which remained in service. During the course of a subsequent routine examination the only evidence that this incident had occurred was the torn appearance of the astern brake shoes, and particles of sintered bronze which had become welded to the brake drums.

In service the performance of the epicyclic units in these gear sets has been excellent and that of the gear sets as a whole has been good, particularly so when it is recalled that these were advanced, highly loaded designs and no testing or development was possible before installation in the boats. Minor troubles which have been experienced include:

i) Porosity of some of the spheroidal graphite cast iron brake drums. Although no failures have occurred a number of drums have been replaced in forged steel.

ii) Wear of lubricating oil pump drive gears. Gears with increased face width have been fitted.

One major breakdown has occurred, caused, it is believed, by failure of a light alloy cage in a vee drive roller bearing, permitting gross misalignment of the vee drive bevel gears and resulting, nine months later, in fracture of the heavily pitted teeth. Steel cages have now been fitted to these bearings.

PART II

GENERAL INFORMATION

Trials of E.P. Lubricating Oils

It has already been mentioned that minor scuffing of the secondary gears occurred during the sea trials of some of the *Whitby* and *Blackwood* Class vessels and because of this it was decided that E.P. lubricating oils should be used in these ships.

At this time the Admiralty had no ship experience of such oils and it was not known whether or not the oils supplied by the various manufacturers would be compatible when mixed together. In addition to the need for such oils in the ships mentioned, it was believed that they would also be required in future ships, to enable full advantage to be taken of the higher loadings permitted by hardened and ground gears.

A large scale experiment was therefore started, in which ships of the *Whitby* and *Blackwood* Class with Y.100 gears were run on nine formulations of E.P. oils supplied by seven manufacturers. Notwithstanding the logistic problems, the various supplies of oil were not mixed, i.e. each ship used oil of one formulation only, supplied by one manufacturer.

Initially it was hoped that the use of E.P. oils would permit these gears to "run in" and would then prove unnecessary. This hope has not been realized, for during the course of the experiment certain ships were unable to obtain the necessary supplies and were obliged to recharge their lubricating oil systems with the conventional turbine gear oil (OM100). In some cases but not all, this resulted in further scuffing, which was once again arrested by the use of E.P. oils as soon as supplies were received.

The trials were continued over a period of five years and although only one case of scuffing occurred with an E.P. oil, various other troubles were experienced, resulting in the withdrawal from service of certain oils, and revision of the Admiralty specification which has, in some respects, been made more severe. Some of these troubles were probably not related to the presence of extreme pressure additives and may well have occurred had the conventional turbine gear oil (OM100) been used in these ships. Experience so far suggests the following conclusions:

i) The use of copper, zinc or alloys containing these metals in lubricating oil systems is undesirable, particularly where high temperatures exist, since these materials can react with the E.P. agent in the oil and the reaction products may give rise to corrosion and/or sludge formation. Other reactive metals, such as cadmium may also be undesirable and should be avoided. The use of steel pipes, valves and bearing shells is therefore recommended but the retention of non-ferrous materials for lubricating oil coolers is permissible. In particular, the use of zinc in galvanized lubricating oil filter cages and of copper in lubricating oil heater tubes should be avoided.

ii) The design of lubricating oil heaters should ensure that local overheating of the oil does not occur, even when flow is restricted by partial fouling. Operating personnel must be made aware of the possibilities of overheating which can result from leaking steam valves and mal-operation.

iii) In machinery installations using steam at high temperatures it is most desirable that a supply of lubricating oil should be maintained, for cooling purposes, to auxiliary machines after they have been stopped. The steam temperature in the Y.100 frigates is only 850 deg. F. but apparent white metal temperatures of 400 deg. F. have been recorded, due to heat soakage after shutting down, in the bearings of some of the larger auxiliaries, not fitted with independently driven lubricating oil pumps. Such temperatures are suspected to have resulted in breakdown of the lubricating oil, blackening and corrosion of bearings.

iv) In some steam driven auxiliaries the ingress of water to the lubricating oil was excessive and provision for removing water was inadequate. This is believed to have caused a number of bearing troubles and cases of excessive wear of gears. In one ship, hydrolysis of the E.P. lubricating oil occurred, resulting in the formation of strong acids and corrosion fatigue failure of two turbo-generator gear pinions. The oil concerned was withdrawn. For future ships, consideration is being given to the provision of an auxiliary lubricating oil renovating system.

v) It was found that the lubricating oil "make-up" rates in these ships was sufficiently high to maintain adequate load carrying capacity of the oils and only slight reduction in the latter, due to depletion of the additives, was experienced.

vi) The use of certain preservatives in gearcases and other parts which are subsequently in contact with E.P. lubricating oils can result in the lowering of the demulsification properties of the oils. It is necessary therefore to ensure by efficient flushing and recharging that the preservative is removed before operating the machinery.

Throughout these trials the Admiralty received valuable advice and assistance from members of the Admiralty Fuels and Lubricants Advisory Committee, on which are represented the major oil companies, leading marine gear manufacturers and the Admiralty.

As a result of the trials, satisfactory formulations of E.P. turbine gear oils are now available and are in service in a large number of the post-war ships of the Fleet.

Lubricating Oil Filtration

Prior to the *Daring* Class, lubricating oil filtration was provided by wire mesh gauze, muslin or the self-cleaning disc and scraper type filter. In addition, centrifugal separators were fitted, operating in bypass circuits and although intended primarily for the removal of water, they also removed quantities of solid matter. Nevertheless, in ships so fitted it was considered necessary to take bridge gauge readings periodically and the renewal of worn bearings was a laborious and expensive procedure.

To improve bearing life, particularly in double-reduction gears with their increased numbers of bearings and to permit higher bearing loadings, a finer degree of filtration was desirable. After comprehensive tests at the Admiralty Engineering Laboratory, West Drayton, it was decided to fit felt filters in the *Daring* Class ships, capable of filtering out solids larger than about 30 μ and similar filters have been used in all subsequent designs up to the present day.

After the lubricating oil system has been flushed out and the first service change of filters has been made, it has been found that, under normal conditions, a life of approximately two years can be expected from the felt elements. Usually, differential pressure gauges are fitted across these filters, to give an indication that the felt element is becoming choked. Automatic bypass arrangements are now being fitted in the latest classes.

Service experience in the *Daring* Class and subsequent ships has been satisfactory and, after the initial dirt has been removed from the lubricating oil system, bearing wear has been negligible. So far, there has been little justification for the use of filters giving a finer degree of filtration and the additional size and weight of such filters has prevented their use.

Since felt filter elements cannot be cleaned, the performance of fine gauze filters and the possibility of cleaning them has been investigated. Such gauzes showed a similar efficiency of filtration but the extended life obtained by reflux cleaning has not warranted the added complication and weight of pipes and change-over valves involved. In addition, compared with felt, gauze elements are more expensive and, when choked with dirt, have shown a tendency to disintegrate.

In addition to felt filters, it is still customary to fit centrifugal purifiers and, resulting from tests carried out at the Admiralty Engineering Laboratory, it has been found possible to increase the throughput of oil from three to four times that originally permitted without significant decrease in performance.

Cleaning and Flushing

A problem that is still receiving serious attention is the matter of cleanliness. Scoring of teeth has occurred in a number of Y.100 gearboxes and far too many bearing shells have had to be renewed after shop trials in Y.100, Y.102A and Y.111A gearboxes for the same reason.

The scoring has been partly caused by mill scale and rust from the fabricated ferrous material used in the gearcase and lubricating oil system and partly from machining and other debris left in the gearcase and system. Pickling followed by phosphating is now specified for the removal of mill scale and rust on fabricated ferrous material but the removal of the debris left in oilways, housings, pipes, etc., is a much more difficult matter, particularly when the gearbox has been assembled. It must therefore be tackled during the final erection and with this in view revised instructions are being issued. The basis of these instructions is that all machining and fitting work must be completed before final erection and that all components must be clean, scale free and suitably protected to prevent contamination or corrosion during assembly. Personnel should be impressed with the need to maintain scrupulous cleanliness and, among the psychological aids to this, is the provision of transparent plastic sheeting to cover opened machinery, clean overalls, rubber soled shoes, vacuum cleaning facilities and adequate lighting. The erection site should be clean and where possible, covered in by a plastic tent.

After final assembly and before any running in is commenced, the lubricating oil systems must be flushed with a copious supply of clean, filtered, hot oil (120-170 deg. F.) preferably section by section in order to obtain the maximum flow of oil. Flushing may take anything up to 48 hours.

When machinery is first run on shop trials it is to be run in the astern direction to minimize the danger of damage to the ahead bearing surfaces. Similar precautions will be followed before machinery is run in the ship.

Journal Bearings

Newman[14] has given an account of bearing tests carried out at Pametrada for the Admiralty and this work resulted in the use of steel backed bearings with thin white metal linings in the Y.100 Mark I gears and in a considerable increase in bearing loadings. As a general rule, maximum loadings of 500lb./sq. in. are permitted for pinion and primary wheel journal bearings and 350lb./sq. in. for main wheel bearings. Experience at these loadings has been entirely satisfactory and the resulting reductions in bearing length ($^L/_D$ ratios of $^1/_3$-$^2/_3$ are permitted for all high speed bearings) have helped to reduce gearing losses. After successful experience in Diesel engines and a limited but successful trial in the Y.E.A.D. 1 gears prefinished white metal lined steel shell bearings were fitted in the Y.100 Mark II gear sets and all subsequent designs. Prefinished bearings have the following advantages over the conventional type of thick shell bearing:

a) Interchangeability of shells of the same nominal bore size.
b) Spares are lighter and more easily stored.
c) Cheaper. The cost can be as low as 10 per cent of the conventional type.
d) No re-metalling problems.

Above all, the use of this type of bearing provides an attractive solution to the problem of obtaining and maintaining gear alignment, a matter of particular importance with higher gear loadings and where silent running is required. With conventional bearings the use of boring jigs has been successful in some cases but the high cost of producing and maintaining such jigs is unattractive for small, peacetime building programmes. The alternative of using large, super accurate jig type boring machines[16] also permits the desired standards to be achieved but involves high capital costs for such machines and seems unlikely to be adopted by many firms.

With prefinished shell bearings manufactured to close tolerances on bore and outside diameter, it is only necessary to provide a means of adjustment between the bearing and the supporting structure. Subsequent bearing renewal can then be effected within the alignment tolerances without any fitting work. This has been of particular advantage in several cases, where bearings have been damaged, during shop trials, by dirt and metal swarf.

Two types of prefinished bearing shell are now available:

a) medium walled and
b) thin walled

comparative figures for a typical bearing of 7in. bore being:

	Thin	*Medium*
Wall thickness	0·1425in.	0·2188in.
White metal thickness	0·02in.	0·02in. nominal
Weight	2·8lb.	4·3lb.
"Nip"	0·005in.	0·0122in.
Clamping load	10,000lb.	13,000lb.

In the Y.100 Mark II design, the original housing bore sizes were retained in case it became necessary to change back to the conventional bearings fitted in the Mark I design. Split liners or sleeves were used to make up the difference in thickness and to clamp the medium wall bearing shells. The sleeves had a slight interference fit in the housings and, where neces-

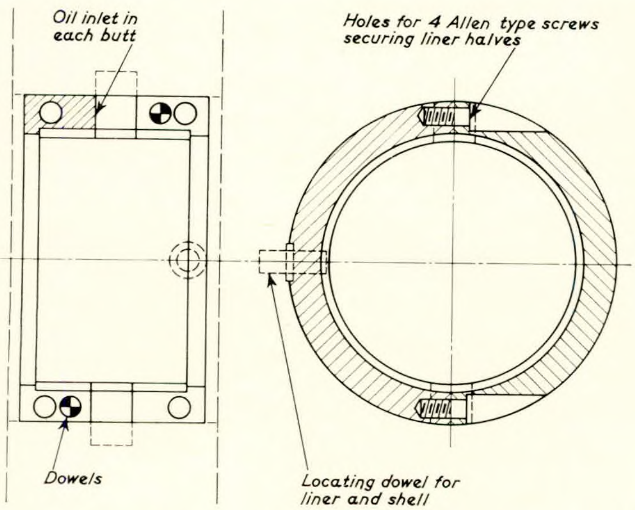


FIG. 29—*Y.100 Mark II—Arrangement of journal bearing*

sary, they were bored eccentric to correct any misalignment of the housing bores. This design has been satisfactory but it would have been preferable to use thicker sleeves to provide the necessary interference or "nip" of the medium walled shells. The general arrangement of this design is shown in Fig. 29. Lubricating oil is supplied to the bearing via a circumferential groove in the sleeves and oil inlets in the butts of the shells. The sleeve type of bearing has been used in a later design with thin walled bearings. With these it is not possible to provide an oil groove in the bore of the sleeve, as the shells require support over their full length and the groove is therefore machined around the outer diameter of the sleeve. The adjustable type of bearing used in the Y.102A and Y.111A designs has already been described (Fig. 25). A further advantage of this type of bearing has been the ability it confers to adjust the gear alignment, if required, after first running on load. In a number of ships it has been found that although the gear meshing was apparently satisfactory during shop trials at light load, examination after sea trials has shown that adjustment was desirable. A problem arising with prefinished bearings concerns the question of undersize journals which have been machined in error or to remove damage. With the conventional bearing it is possible to re-metal and bore undersize to suit the journal but with prefinished bearings it is necessary to adopt a standard range of undersizes. It is usually desirable to limit the white metal thickness to avoid too much reduction in fatigue strength and, at present, the Admiralty has made provision for standard undersizes of 0·02in. and 0·04in. on diameter. It is also possible to increase the thickness of the steel backing of the shell but care is necessary to ensure that the clamping bolts are capable of providing the increased "nip" which is required. Many cases of small errors in journal size have been rectified by nickel plating, provided the finished thickness does not exceed 0·012in.

Recent Full Scale Gear Tests

Newman[6] has described some of the full scale gear tests carried out by A.V.G.R.A. and which have played an essential part in the development of the surface hardened and ground gears now in service. Further tests have been carried out in recent years and are briefly described below. Since descriptions of the two test rigs have already been published[6] they will not be repeated here. The particulars of the test gears are, as follows:

	First Reduction Test Rig		*Second Reduction Test Rig*	
	Pinion	*Wheel*	*Pinion*	*Wheel*
Pitch circle diameter, in.	8·186	35·814	13·256	70·208
No. of teeth	24	105	27	143
Normal pitch, in.	1·047 (3D.P.)		1·496 (2D.P.)	
Normal pressure angle	20 deg.		16 deg. 35 min.	
Helix angle	12 deg. 14 min.		14 deg. 07 min.	
Face width, in.	8		10	
Speed (r.p.m.)	6,000	1,370	1,500	283

First Reduction Gear Tests—Nitrided Gears

Chamberlain[4] and Newman[6] described the manufacture and initial tests of an En 40c nitrided wheel and further testing was carried out after the publication of their papers. Details are given in Table VII from which it will be seen that after

TABLE VII.—A.V.G.R.A. FIRST REDUCTION GEAR TESTS.—NITRIDED EN 40C WHEEL.

All testing carried out at 6,000 r.p.m. pinion speed.

Duration, hrs.	Horsepower	Loading		Cycles × 10^6	
		K value	lb./in. face tangential	Pinion	Wheel
36 hrs. approximately	Running up to 8,250	0–398	0–2,652		
30	9,000	434	2,890	10·8	2·47
30	9,750	470	3,130	10·8	2·47
30	10,500	506	3,370	10·8	2·47
30	11,250	542	3,610	10·8	2·47
30	12,000	578	3,850	10·8	2·47
32	12,750	614	4,090	11·5	2·63
29¾	13,500	651	4,340	10·7	2·44
28½	14,250	687	4,580	10·3	2·35
30	15,000	723	4,810	10·8	2·47
30	15,750	759	5,050	10·8	2·47
29½	16,500	795	5,300	10·6	2·42
30	17,250	832	5,545	10·8	2·47
25¼	18,000	868	5,780	9·1	2·08
Torque reversed to load other wheel flank and a 10^8 wheel cycle run commenced.					
4	—	289-1,060	—	1·44	0·33
259	22,000	1,060	7,060	93·3	21·35
Pinion tooth cracked. Wheel and direction of loading reversed so that damaged astern pinion flank used to load the original ahead wheel flank and test continued.					
¼	22,000	1,060	7,060	0·1	0·02
Cracked tooth broken off pinion. Wheel undamaged. New pinion installed.					
613¾	18,000	867	5,780	220·8	50·5
288¼	22,000	1,060	7,060	104	23·75
Testing stopped for replacement of loading wheel. Test gears undamaged.					
59¼	22,000	1,060	7,060	21·3	4·87
30	24,000	1,156	7,700	10·8	2·47
30	26,000	1,254	8,360	10·8	2·47
30	28,000	1,350	9,000	10·8	2·47
Test gears undamaged at end of these tests.					

21 × 10^6 wheel cycles at 1,060 K a pinion tooth was found cracked. In an attempt to prolong the test, the nitrided wheel rim and direction of loading were reversed but the cracked tooth failed immediately the tests were recommenced.

A new carburized and ground pinion was installed, 50 × 10^6 wheel cycles at 867 K were successfully completed and the load had been raised to 1,060 K when the cast steel centre of the loading wheel collapsed and caused fracture of the loading wheel rim. The nitrided test wheel remained in excellent condition.

After replacement of the loading wheel, tests were resumed and the nitrided wheel successfully withstood over 2×10^6 cycles at each of the loads corresponding to 1,156 K, 1,254 K and 1,350 K.

The carburized and ground pinion was then replaced by an En 40c nitrided and ground pinion. Testing was continued on the Astern Wheel Flanks II and $3{\cdot}6 \times 10^6$ pinion cycles were completed at each of the following loads: 578 K, 675 K, 770 K, 868 K, 963 K, 1,060 K, both pinion and wheel remaining in satisfactory condition. Testing was then resumed on the Ahead Wheel Flanks I and the following programme was satisfactorily completed: $3{\cdot}6 \times 10^6$ pinion cycles at each load, equivalent to 578 K, 723 K, 868 K, 5×10^6 cycles at 1,060 K and $1{\cdot}08 \times 10^7$ cycles at 1,156 K. After $0{\cdot}96 \times 10^7$ cycles at 1,254 K, a pinion tooth broke and, in passing through the mesh, caused severe damage to the remaining teeth and to the wheel. Metallurgical examinations are not yet complete but it is known that the depth of nitrided case of this pinion did not exceed 0·015in.

Second Reduction Gear Tests

i) *En 24 Induction Hardened and Ground Wheel*: Testing of a 2 D.P. En 24 induction hardened wheel was also described by Newman[6] and further tests on this wheel were carried out after the publication of his paper. The back-tempered wheel flanks, running against a carburized hardened and ground En 36A pinion, were run in, before increasing the load to 909 K. After 5×10^6 wheel cycles at this loading, pitting again began towards one end of the wheel teeth and increased somewhat throughout 50×10^6 wheel cycles. The load was then increased to 1,360 K and, after $1{\cdot}3 \times 10^6$ wheel cycles the teeth scuffed over the full face of the wheel and the test was abandoned. Subsequent tests of the lubricating oil (OEP.90) showed that it had suffered an appreciable reduction in load carrying capacity. Metallurgical examination showed that the position of the pits on the wheel teeth corresponded to soft areas at the ends of the teeth where hardening had been imperfectly carried out. Both the equipment and operating technique have been improved since this wheel was induction hardened and today it is possible to achieve satisfactory hardening throughout the length of the teeth.

ii) *En 9 Induction Hardened and Ground Wheel*: Testing of this 2 D.P. wheel was commenced with an En 36A carburized and ground pinion. After short runs at lower powers the load was worked up to 904 K (10,146lb./in. face) and, after $21{\cdot}7 \times 10^6$ wheel cycles tooth breakage of the pinion occurred. The wheel was undamaged and testing was resumed using an En 35 carburized and ground pinion, which had already undergone earlier tests with other wheels. After another $8{\cdot}3 \times 10^6$ wheel cycles at 904 K this pinion also failed, the wheel fortunately remaining in excellent condition.

iii) *En 40c, Nitrided Pinion*: Testing was continued against an En 40c nitrided pinion. This pinion was ground after hobbing, only to improve the surface finish and obtain the desired helix and profile corrections. After nitriding, the pinion was checked for alignment and meshing and testing was begun. Some heavy bedding was observed during the early stages of testing and the roots of the pinion teeth were therefore honed. Subsequent bedding was excellent and this pinion had completed 10^8 load cycles ($18{\cdot}9 \times 10^6$ wheel cycles) at 904 K when repair of the loading wheel became necessary and testing had to be halted.

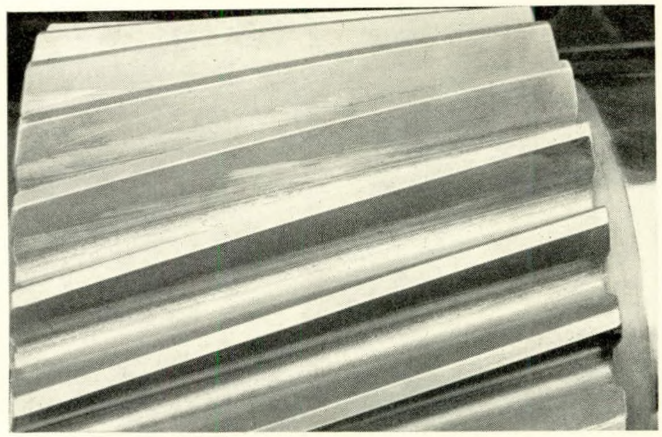


FIG. 30—*AVGRA—2nd reduction EN40C nitrided pinion*

The wheel had then completed $46{\cdot}9 \times 10^6$ wheel cycles at 904 K on the same flank and was in excellent condition, as was the unground nitrided pinion. Figs. 30 and 31 illustrate the appearance of the pinion and wheel after 58×10^6 and 39×10^6 cycles, respectively, at 904 K. It will be noticed that the matt finish produced by the nitriding process is still evident,

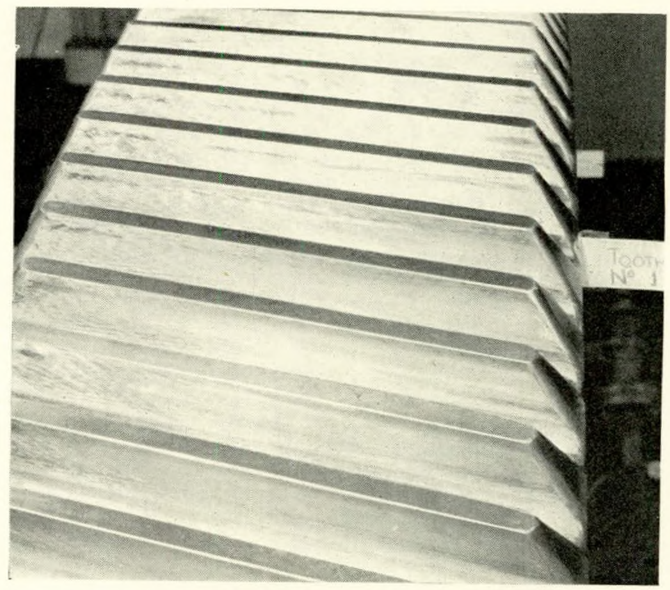


FIG. 31—*AVGRA—2nd reduction EN9 wheel*

except in the roots, where honing was carried out. Testing was resumed after repairs to the load wheel and, at the time of writing, 10^6 wheel cycles at 1,023 K have been successfully completed. Further tests are planned, to include 10^6 wheel cycles at 1,137, 1,250 and 1,364 K.

Induction Hardening

Chamberlain[4] has described A.V.G.R.A.'s earlier work in this field and mentioned the installation of a new gear induction hardening machine, in which the gear is immersed in oil. Since then, further development has been carried out and this machine was used to harden, at 8·3 Kc/s, the En 9 second reduction test wheel described above and a number of production gears, including two main gear wheels and ten primary wheels, for the G.M. destroyers and G.P. frigates. Etched hardness contours for three of these gears are shown in Fig. 32 and these are typical.

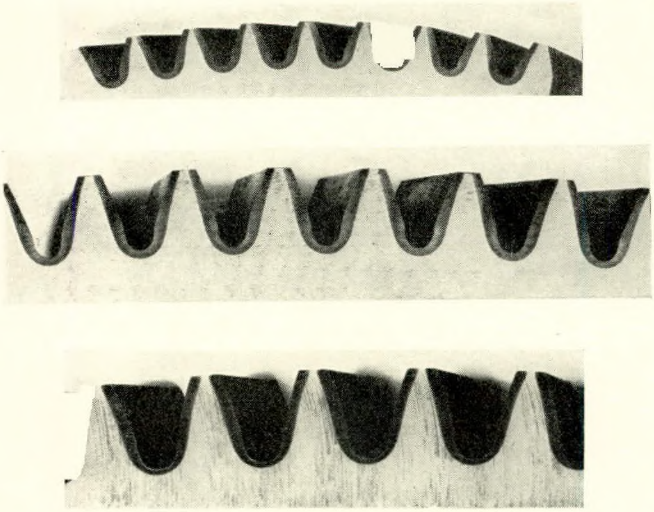


FIG. 32—*Induction hardened gears—Hardness contours after etching*

So far, it has not proved possible to dispense with the "before" and "after" test arcs, which are used to establish and check suitable hardening conditions. A satisfactory method of measuring, non-destructively, the depth of hardening is still being sought.

It has, however, proved possible to obtain satisfactory hardness contours at the ends of the tooth faces and, by the use of quenching jets, suitably directed to reduce considerably the "back tempering" effects. Typical hardness figures measured on the production gears mentioned above were in the range 550-650 V.P.N. on the ahead flanks and 540-640 V.P.N. on the astern flanks. The ahead flank of the first tooth, which is back tempered during hardening of the last tooth was some 20-50 V.P.N. softer than the other teeth.

Further work remains to be done to investigate the effects of variations in inductor size and clearances, alternative quenching media and to establish the minimum size of teeth which can be satisfactorily hardened at 8·3 Kc/s.

The response of various steels to induction hardening and their load carrying capacities as gears, when hardened, are also being investigated. Fifty-eight 8in. P.C.D., 4 D.P. power circulator gears, in ten different steels, have been induction hardened and will be tested.

With fine pitch gears, shallower hardened contours are required and induction hardening at higher frequencies is necessary. Preliminary trials have been completed and a 3ft. diameter 6 D.P. first reduction test wheel is to be induction hardened, at 250 Kc/s, and tested.

CONCLUDING REMARKS

Only a few gear designs have been required for the Royal Navy's post-war shipbuilding programme and in each of these, very considerable departures from past and proved practice have been made.

Thus in the *Daring* Class destroyers, double-reduction gears were fitted for the first time in British warships and although this step forward appears to have been taken with considerable misgivings at the time, it was, nevertheless, successfully accomplished.

In the Y.100 gears, harder alloy steels were utilized at tooth loadings more than twice as high as those in the *Daring* I and II designs and substantial reductions in size and weight were achieved. Although some troubles were experienced, only one of the original twenty-four Mark I sets has been renewed and it is probable that this set would have continued to run satisfactorily for a considerable period. These troubles were entirely eliminated in the Mark II design by the use of carburized, hardened and ground secondary pinions.

Although the excellent performance of the *Daring* III (*Diana*) and R.C.N. *St. Laurent* Class gears had already demonstrated the advantages of carburized, hardened and ground gears, the Y.E.A.D. 1 design was the first British attempt to utilize such gears and its successful performance on shore trials was encouraging and gratifying.

In all these gear designs, the desire to reduce size and weight, without sacrifice of reliability, was of primary importance. Comparisons of gear sizes and weights can be misleading, unless the gears compared are of similar type, power and reduction ratio but the relevant details of the designs mentioned above are shown in Table VIII and Fig. 33 and

TABLE VIII.—COMPARISON OF WEIGHTS OF POST-WAR GEAR DESIGNS

Design	S.h.p.	Reduction ratio	Lloyd's K value		Weight tons
			Primary	Secondary	
Daring I	27,000	17·7	83 94	100	44
Daring II	27,000	22·9	130	125	37½
Diana	27,000	23·2	236 260	200	32
Y.E.A.D. 1	30,000	H.P. 38·9 L.P. 30·2	436 353	422	27½

Note:—Weight of thrust block included in all cases.

it will be seen that the three *Daring* Class and the Y.E.A.D. 1 designs are of comparable power, although the reduction ratio of the latter is considerably greater. Notwithstanding this, the use of hardened and ground gears and shorter, more highly loaded bearings has resulted in substantial reductions in size and weight without sacrifice of reliability.

The Y.102 and Y.111 gear designs are so unlike their predecessors that comparisons of weights and sizes are meaningless—but had it been necessary to use through hardened, hobbed and shaved gears in these designs, the authors believe that this would have necessitated larger machinery spaces in the ships. The experience obtained during manufacture and in the shore trials and sea trials of these gears has already pointed the way to more compact and efficient gears of similar type. Should a requirement for further boost machinery installations, or perhaps, an all-gas-turbine ship arise, it is thought that the design and manufacture of the gears could now be undertaken with confidence.

Thus, in the period under review, the transition has been made from the comparatively large but simple, single-reduction, hobbed and filed gears fitted in wartime construction to the complex, double-reduction, highly loaded, hardened and ground gears fitted in the latest warships. It can be said that A.V.G.R.A.'s original objectives have, in large measure, been attained and although much work remains to be done, this seems likely to result more in consolidation of the present gains than in further substantial reductions in weight and size of naval gears.

The development and testing of induction hardened and nitrided gears will continue and experience so far suggests that they will replace carburized gears in future designs, with substantial reductions in manufacturing costs. Highly loaded gears of both types are now going into service and tests of an induction hardened primary wheel (6 D.P.) and nitrided secondary wheels (4 D.P. and 6 D.P.) will be commenced shortly. Notwithstanding the big improvements in gear cutting accuracy which have been made during this period, it is believed that even greater accuracy will be required in future warship gears. At present, B.S. 1807: 1952 Grade A1 standards are specified and can be attained by the majority of British warship gear manufacturers. It is doubtful whether further improvements in gear accuracy can be made with the gear measuring equipment in general use at present. The use of better equipment,

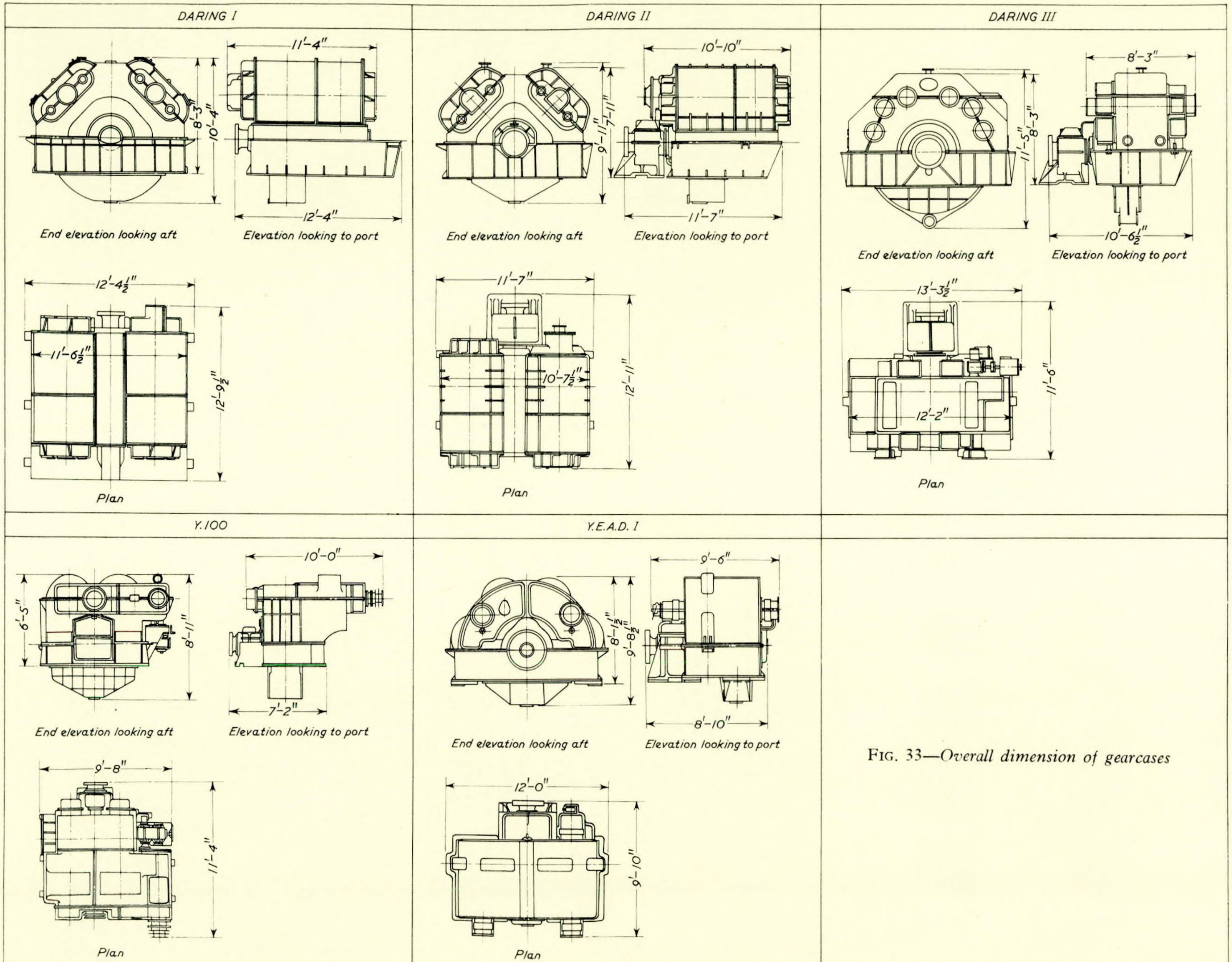


FIG. 33—*Overall dimension of gearcases*

now becoming available, is most desirable and should permit the present standards of accuracy to be assessed with more certainty and, it is hoped, to be surpassed in the future.

Although the use of adjustable journal bearings in recent designs permits correction of misalignment which may be apparent after full power trials, the authors believe that the problem of maintaining good alignment, throughout the wide range of powers at which naval vessels operate, requires further study. Compared with the gears themselves, the design and construction of gearcases seems to have received little attention and it may be that further work in this field, including, perhaps the use of model structures, would be rewarding. In addition to the maintenance of gear alignment there is a need for improvements in design to permit thorough cleaning during assembly. The troubles experienced from dirt and lack of care during manufacture have been referred to earlier in this paper but the resultant delays and expense are so great that no apology is made for again stressing the need for improvement.

ACKNOWLEDGEMENTS

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The authors are also much indebted to the many firms and organizations which have contributed, directly and indirectly, to this paper and in particular to the members and staff of the Admiralty-Vickers Gearing Research Association, which has been the greatest single factor in naval gear progress since 1946.

It is also most appropriate to acknowledge here the work of the successive A.V.G.R.A. Research Officers: Mr. A. Fisher, Mr. J. R. G. Braddyll, Mr. A. Chamberlain and Mr. W. G. Smith.

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Appendix

THE MAIN SYNCHRONIZING CLUTCH (Y.102A AND Y.111A)

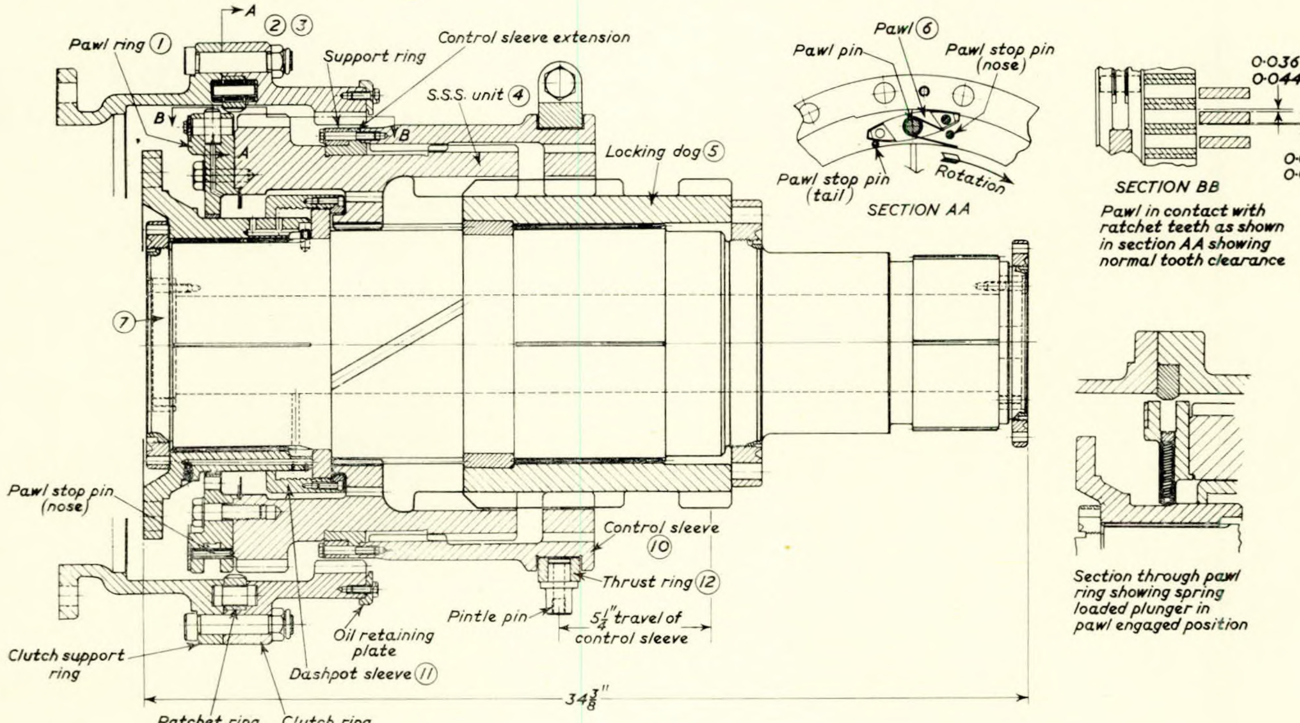
Description

A sectional arrangement of the clutch is shown in Fig. 34. The gas turbine is connected, via the intermediate gearing, to a helically splined shaft (7) on which slides the S.S.S. Unit (4). The pawl ring (1) is bolted to the S.S.S. Unit and carries four pairs of case hardened, spring loaded pawls (6) which can engage with the ratchet ring (8) mounted on Metalastick bushes (2, 3) to the clutch ring (9). A control sleeve (10) is connected to the S.S.S. Unit through straight splines. The positions of the various parts of the clutch during an engagement are shown diagrammatically in Fig. 35.

When the associated gas turbine is out of use, the clutch is locked in the disengaged "pawls free" position permitting free rotation of the clutch ring in either direction, as when manœuvring on the steam turbine. Gas turbines are only started with the propeller shaft stationary or rotating ahead and movement of the gas turbine starting lever actuates relays to unlock the clutch and put it into the pawls engaged position. To do this the control sleeve is moved through a spring link mechanism by means of a double acting hydraulic cylinder and piston. Thus when oil is transmitted to the cylinder the piston moves its complete travel (5¼in.) but the control sleeve only moves 3¼in. since its teeth abut axially against the locking dog teeth (5) and the spring link is compressed 2in. to take up the difference in movement. This axial movement causes the pawls to line up in ratcheting relationship with the ratchet teeth.

When the clutch has moved to the pawls engaged position, the gas turbine is automatically started and immediately the clutch primary components tend to pass through synchronism with the clutch ring, the pawl tips meet the ratchet teeth and precisely align the teeth of the S.S.S. Unit with the clutch ring tooth spaces. The pawls then transmit the small force required to initiate movement of the S.S.S. Unit along the helical splines to engage the clutch driving teeth. During this travel the pawls pass axially out of engagement with the ratchet teeth so they do not transmit any driving torque. As the S.S.S. Unit travels towards engagement, it abuts against the dashpot sleeve (11) for the last 1¼in. of its travel, thus cushioning the action, as the clutch takes up the drive.

The S.S.S. Unit completes its helical travel when its internal driving teeth contact the flanks of the teeth on the locking dog (5). The rotational movement of the control sleeve (10) during the helical travel of the S.S.S. Unit precisely aligns the locking teeth whereupon the control sleeve com-



1) Pawl ring	5) Locking dog teeth	9) Clutch ring
2) Metalastik bushes	6) Spring loaded pawls	10) Control sleeve
3) Metalastik bushes	7) Helically splined shaft	11) Dashpot sleeve
4) S.S.S. Unit	8) Ratchet ring	12) Thrust ring

FIG. 34—*Assembly of main synchronizing clutch—Port*

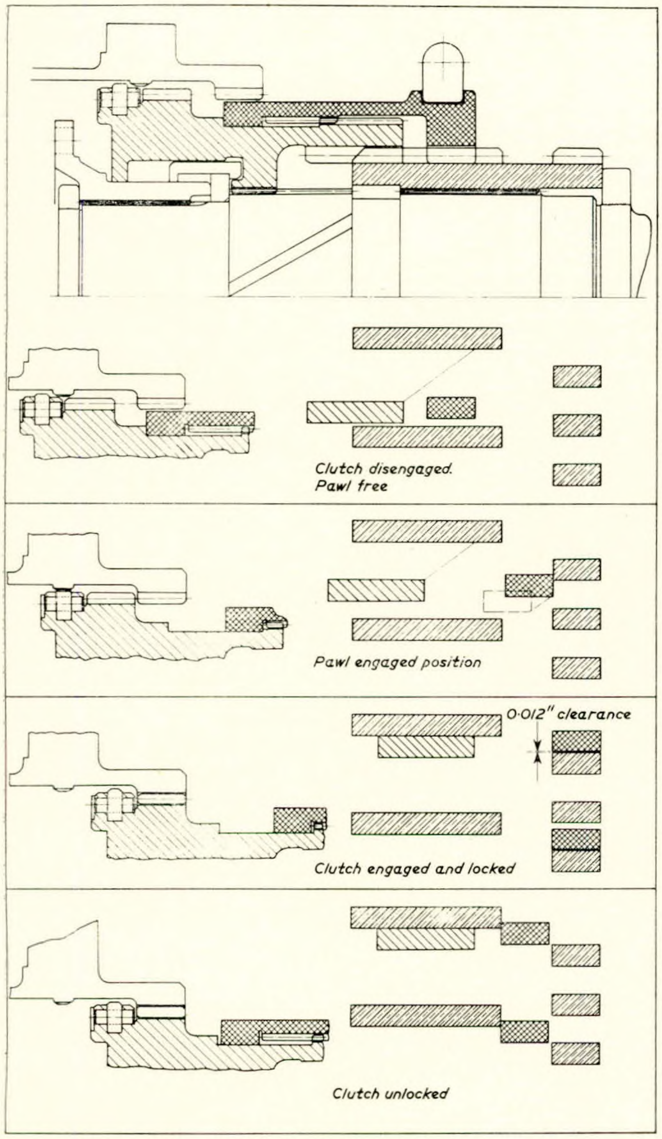


FIG. 35—*Y.102A. Assembly of main synchronizing clutch—Port. Positions of various parts during engagement and disengagement*

pletes its travel to lock the clutch. After locking, the clutch is then able to transmit the drive in both the ahead and astern directions.

Preliminary Clutch Trials

Preliminary tests were made in a special rig in order to prove the pawl mechanism and establish that it could withstand long periods of continuous ratcheting without wear. A special rig was also used to permit repeated engagements and disengagements of the clutch at operational speeds but with differential accelerations considerably higher than those expected in service. Early running experience showed the need for minor modifications and subsequently some 1,500 clutch engagements were satisfactorily carried out and the clutch was stripped and found to be in excellent condition.

Shore Trials in Y.102 Installation

During trials, in moderate seas, of the original type of Y.100 cruising turbine clutch in H.M.S. *Whitby,* cruising pinion accelerations and decelerations of the order of 300 r.p.m./sec. were measured. These arose solely from the movement of the ship in a seaway and it was apparent that with the ship under helm in rough seas this figure would have been exceeded.

In the Y.100 installation the gear reduction ratio between clutch and propeller shaft is 54·6 : 1 and thus speed variations in the latter, due to sea effects, were very considerably magnified at the clutch. In the Y.102 and Y.111 installations the corresponding ratio is 5·609, or approximately one tenth that in the Y.100. Furthermore, since the clutch runs at intermediate shaft speed, the acceleration of the incoming gas turbine is also reduced by the appropriate gear ratio (3·196) at the clutch input shaft. The estimated maximum differential acceleration across the clutch is 60 r.p.m./sec. or approximately one-ninth that of the Y.100 clutch. The Y.102 clutches were designed to engage satisfactorily under maximum differential accelerations of 100 r.p.m./sec. and were in fact, tested in the shore trials installation, at 108 r.p.m./sec. To achieve this condition, a technique was evolved in which the incoming gas turbine was accelerated rapidly and at the same time the other gas turbine was quickly reduced to idling from maximum speed. By varying the rates of acceleration and deceleration and by using different settings of the Froude brake on the propeller shaft it was possible to test the clutches over the complete range of speeds and at differential accelerations up to approximately twice as high as those expected in service. A further important advantage of installing the clutch in the intermediate shaft system, as in Y.102 and Y.111, is that when the associated gas turbine is at rest the high speed gearing is stationary.

Soon after the commencement of clutch trials and after successfully carrying out a number of engagements at various speeds and small differential accelerations, it was observed that the inboard clutch failed to engage at synchronism. This clutch was stripped and it was found that only the pawls, pawl carrier and ratchet ring were severely damaged and the remaining parts of the clutch were in satisfactory condition. The circumstances and cause of the failure were not conclusively established but investigation showed that:

a) due to the experimental nature of the trials, with interlocks out of action, the clutch could have been moved to the "pawl engaged" position at a moment when a negative speed differential existed between the ratchet ring and the pawls, e.g. when the input was already rotating faster than the output shaft, or possibly

b) that the control spring was not strong enough to ensure at all times the complete travel of the clutch to the "pawls free" position. In addition, an error in installation had reduced the distance between the side faces of the pawls and the ratchet ring. It thus seemed possible that an engagement could have been made with the pawls in edge contact, against control spring and sleeve forces, thus resulting in excessive pawl loading.

Notwithstanding the disappointment which accompanied this early failure, its value is now appreciated, for it resulted in a critical re-appraisal of the clutch and clutch controls and of the various operating conditions. Modifications were made, including provision of the device to lock the clutch in the "pawls free" position. This lock is remotely actuated and is removed automatically by the initial movement of the gas turbine starting lever.

When trials were resumed, approximately 60 engagements of each clutch were satisfactorily made under different conditions of speed and differential acceleration and finally, 100 engagements were made under the most severe conditions. Records of strain, also differential and absolute speeds, were taken at each engagement and Fig. 36 shows graphs of differential clutch revolutions, speed, acceleration and torque plotted against time, under severe conditions. During an engagement there are two actions, upon passing through synchronism which are completed before the control sleeve can move into the locked position:

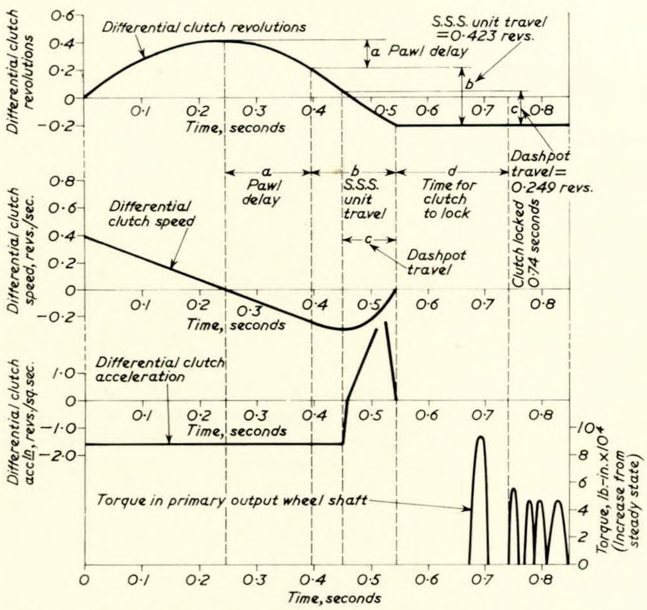


FIG. 36—*Y.102—Main synchronizing clutch—Conditions during engagement*

i) The pawls must "pick up" the ratchet teeth. Hence if synchronism occurs at the instant, just after one pair of pawls has passed a pair of ratchet teeth, the clutch members will rotate relative to each other until the next pair of pawls "pick up". This "pawls delay" is arbitrary and is between zero and a maximum of 0·0179 revolutions.

ii) The S.S.S. Unit moves on the helical splines until the driving teeth engage; the last part of this movement being against the resistance of the dashpot.

At the end of the engaging travel of the S.S.S. Unit, the control sleeve snaps axially to lock the clutch. These positions are shown in Fig. 36 as "a, b, c and d".

From the curve of differential clutch speed against time it is seen that, during this engagement, the speed increased linearly through synchronism until the speed difference between the input and output members was about 0·3 r.p.s. and was then decreased by the action of the dashpot. The action of the dashpot is powerful enough to accelerate the slower member and decelerate the faster member, thus bringing into driving contact the teeth of the final reduction gear train just prior to the moment of full engagement of the clutch. If impact should occur at the end of the travel of the S.S.S. Unit, the two clutch members will attempt to rebound but this is prevented by the control sleeve which, having a lead on the edge of its locking teeth, as well as backlash, engages and locks the clutch just before the full engagement point is reached. The control sleeve then completes its travel, completely locking the clutch and while this occurs the driving torque increases to its full value.

As two elastic shaft systems are connected, a torsional vibration is excited and it will be seen in Fig. 36 that after the torque stress reached its maximum value following clutch engagement, the driving torque dropped momentarily to zero and rose again: this cycle being repeated with decreasing amplitude, dying away in about six oscillations. Prior to an engagement, the accelerating torque of a gas turbine measured at the strain gauge was found to be negligible and hence these fluctuations in torque can be taken as absolute values. During the return swings of the brief oscillation, gear tooth separation occurs because the oscillating torque is greater than the accelerating torque from the gas turbine at the start of the engagement. The maximum amplitude of the oscillating torque shown in Fig. 36 is $9 \cdot 4 \times 10^4$lb./in. and this is considerably less than the maximum steady torque with one engine driving.

On completion of these trials the clutch was found to be in completely satisfactory condition. Subsequent experience in the ships has been equally satisfactory.

Discussion

DR. A. W. DAVIS (Member), in opening the discussion, said that the paper presented in a very artistic way, and very comprehensively, what had been done in the design and manufacture of naval gears during the relevant period. It was a record of great achievement. Not only was the paper an artistic one but the subject was an artistic one. The difficulties had been immense and the mistakes that had been made were relatively few, most of the units having operated in the way that was intended.

He did not wish the sincerity of his opening remarks to be lost in the criticisms that would follow, but the authors really were inviting criticism when in a paragraph of four and a half lines they attempted to sweep away (rather than describe) all that happened between 1943 and 1948.

There were considerable misgivings in relation to double reduction gears and these misgivings were not misplaced. When, in November 1943, it was first decided that double reduction gears should be adopted, there was not a machine in the country suitable for cutting them to the necessary accuracy and it was only by a very considerable effort in the following five years that the *Daring* Class gears came successfully into being. The authors referred to the filing of gears in 1946. Some people may have been filing them then and some people might still be doing so, but certainly the success achieved with the *Daring* Class gears was not through the means of filing. Possibly the authors should have selected a different period of years in their title; in fact, in his opening remarks, Commander Weaving had spoken of 1948 to 1962, which would have been a fairer description in print.

For the first two years of the organization that was represented by A.V.G.R.A. there was a great effort to try to broaden the accuracy that was already being achieved, and at first rather to over-emphasize the significance of the developments afoot. When the design came out for the Y.100 gears an unsuccessful representation was made to the Admiralty imploring them to adopt more conservative loadings for the secondary gears, so that while the ultimate operation was, as the authors have said, disappointing, it was not a surprise.

It might have been fitting had it been possible to anticipate events that may occur shortly after 1962, because the developments being made by A.V.G.R.A. were likely, in his view, to lead to quite a revolutionary change not only in naval gears but in merchant gears when nitrided elements would come into more general use. There was a description in the paper of very interesting developments in that direction, and with the possible lack of need for profile grinding after hardening these gears could become really a workable commercial proposition. He found it difficult to grow enthusiastic about induction hardened gears. He appreciated that the development was striking but his fear was that, each tooth being treated individually, consistency could never really be ensured. He would hate to be under the ice at the North Pole when an inconsistency was revealed.

He was not in favour of helix correction and thought he sensed something of the same thought in the authors' minds. He had noted the point that gears had successively improved as manufacturers had gained more experience, and this was just a demonstration of the fact that they were dealing with an art and not a science when it came to helical correction. Surely it is basically unsatisfactory to have a gear mesh that cannot at any time be properly checked in a gearbox and where reliance must be placed on records of gymnastics, carried out on a meshing frame at great cost during final tooth grinding.

The authors had stressed how necessary it was that more thought should be given to the construction of gearboxes, but for this to be well directed the policy of extreme weight saving may require reconsideration.

If a specialist section were given the task of re-designing a gearbox and in so doing they saved half its weight this would in itself be an achievement, but if looked at as a whole the weight reduction would correspond to something like the fuel that the machinery would consume in an hour at full power. This should be examined in relation to the sacrifices made in design and it is feared that these have gone so far as to prejudice the shape retaining strength of the gearboxes themselves. It seems a cumbersome and sometimes ineffective palliative to provide adjustable bearings, cumbersome in construction and in the dirt catching pockets introduced, and sometimes ineffective because adjustment cannot be phased to coincide with the onset of loss of gearbox shape. It is of course recognized that the design does provide means of making good for lack of proper production equipment where large jig boring machines are lacking.

MR. W. G. SMITH said that in presenting this review of sixteen years of progress and development in naval propulsion gearing the authors had found it possible to include a tremendous amount of detailed information which was of absorbing interest. Having been associated with a great deal of the work under review his comments were offered in the hope that they would add a little to the value of the paper.

Although it was recorded that the performance of the Mark I Y.100 gears was disappointing, it was fair to say that the technical design was exceptionally accurate. These gears were, in fact, a border-line case: they did not conceal, as so many conservative designs had done in the past, excessive factors of safety and, for the very reason that troubles were experienced with these gears, much valuable information for the future had been learned from them.

It was interesting, also, to remark on the adoption of two nickel/chromium/molybdenum steels, namely En26 and En30b, for the pinions and wheels in these gearboxes. All past experience had indicated that the use of similar alloy steels for pinion and wheel might lead to difficulty and indeed, after giving the matter most detailed consideration, the A.V.G.R.A. Materials Committee recommended, in January 1949, that the pinions should be made of En27, heat treated to 60-65 tons/sq. in. and the wheels of En9, heat treated to 45-50 tons/sq. in. It was believed that this combination would have given more successful results. The Admiralty, however, decided on the En26/En30b combination, hoping not only to achieve an added factor of safety on root strength, but also some margin of safety on surface stress, leading to the possibility of even higher loadings in the future. This hope was based on separate results of disc tests in which En30b and En26 were separately tested against case-hardened steel rollers and it was not till later that both gear tests and disc tests emphasized how important was the compatibility of the two materials.

Having been associated with the development of the lapping process it was very gratifying to note that the authors regarded it as significant that the successful gears in two of the three *Whitby* Class ships had main wheels, which were lapped and not shaved. Put another way, it would appear that out of a total of four ships having shaved gears, one was successful, whereas the lapped gears achieved a 100 per cent performance record. A.V.G.R.A. had always claimed that lapping was in no way inferior to shaving, and even now they would not wish to claim any more than that. It seemed necessary, however, to emphasize the success of these lapped gears again, because the most irresponsible statements were made from time to time concerning the ill effects of lapping.

It was no surprise to read that defects occurred mainly in the starboard sets, whereas the port sets were comparatively trouble free. This, indeed, had been their experience of twin-screw vessels, particularly in the merchant shipping field, for which they had records extending over a period of 40 years. Each case needed treating on its merits, as exemplified by Welch and Boron (reference (11) of the paper), who recommended running with apex leading for U.S. Navy gears. In the merchant ship and tanker field, however, they had come to the conclusion that the predominant effect was due to the fact that pinions ran approximately 5-6 deg. C. (10 deg. F.) hotter than their mating wheels. This lengthened the axial pitch of the pinion in relation to the axial pitch of the wheel. The pinion also suffered distortion in relation to the wheel due to bending and twisting, and an analysis would show that whereas, if the gears ran with apex leading (i.e. conventional starboard set), then the distortions were cumulative, whereas if the gears ran apex trailing (i.e. conventional port set), then the distortions effectively tended to cancel each other out. It was for this reason that during the past seven years it had been customary to cut tanker gears so that the apex trailed, rather than cutting them as starboard sets, as was the practice up to about 1955. It would be interesting to see the results of an analysis on this basis applied to the Mark I Y.100 gears.

With regard to the Y.E.A.D.1 gears, it was felt that there were two points worth emphasizing: firstly, that this prototype was 100 per cent successful, although it was the first attempt to design and manufacture a carburized and ground marine propulsion gear set for large powers. It incorporated gear tooth forms, which were unconventional, in relation to any known practice in this field, and these achieved the desired result of not requiring an E.P. oil. The central drive to the secondary pinion was also of interest and was entirely successful in achieving its designed objective, namely, of compensating bending deflexion by torsional deflexion.

In considering the whole field of development for the future it was felt that the stage had now been reached where unground nitrided secondary pinions could be considered. The point was of importance because, firstly, the grinding of these Y.100 double helical secondary pinions was a lengthy and costly operation and a further important point was that the contact conditions between two hobbed gears were likely to be superior to the contact conditions between a hobbed gear and a ground gear. This statement might sound surprising, but the accuracy of hobbing had reached the stage where tooth spacing, measured as circular pitch, was little inferior to that of ground gears, and had the compensating advantage that the tooth spacing, measured as base pitch, would be superior. Furthermore, the axial pitch around the gear would be more constant if the gear had been generated by hobbing than if it were ground, because in the latter case the spacing of the teeth and the uniformity of axial pitch were dependent upon the accuracy of the index plate.

Finally, it was felt that also the advantages, or indeed the absolute necessity, of having adequate inspection equipment should be emphasized also. Mention had been made of the factors accounting for the variable performance in the different sets of gears, including gear cutting errors, which referred more precisely, it was suggested, to the difficulty of providing the specified helix corrections by the shaving process. There was no doubt that generally in this country, at least, large marine gears could be produced to a higher degree of accuracy than they could be measured. The provision of adequate U.K.-manufactured inspection equipment was well behind the progress made in the improvement in accuracy of gear producing machines. Autographic recording of errors was the only satisfactory method and was being covered to some extent by instruments manufactured by Continental firms, but gear producing firms in this country possessing this equipment were few and far between. Actual experience had been obtained of the production in the past three years of about 700 carburized and case-hardened commercial gears of marine gear proportions, the accuracy and quality of which had been greatly enhanced since the inception of the product, simply by the acquisition of adequate inspection equipment providing autographic records.

DR. J. F. SHANNON said that the authors presented in chronological order the progress in design and development in naval propulsion gearing over the last 16 years. They showed the advance due to changes in pinion and wheel material from soft on soft, to hard on soft, and finally to hard on hard. These designs were parallel developments with back to back testing at A.V.G.R.A. and they culminated in the Y.E.A.D.I. design.

When Y.102 and its variants came to be designed the company which he represented was privileged in undertaking the task although at that time they were not members of A.V.G.R.A. Their experience in hardened and ground gears stemmed from their own developments for the G.2 and G.4 gas turbine naval marine propulsion machinery. The G.2 gas turbine was one of his company's aero-engines slightly derated, and was fitted in H.M. *Bold* boats as gas turbine boost with Diesel cruising. The gears were suitably matched and derated from aero-engine gear loading according to life factors on a one-fifth power basis to a nominal 600 K at full power on case-hardened and ground material to En39 specification. The final rating of the engines however gave a loading in service of 580 K. The G.2 gears were fitted with roller bearings and the G.4 with plain bearings, this latter being a parallel development with the engines. The gears and highly loaded bearings were successful in every respect. The power/weight ratio of these single reduction gears was 3·5 and 4·3 for the G.2 and G.4 respectively, expressed as h.p./r.p.m./ton.

These experiences were carried forward by A.E.I. to the design and development of the Y.102A and Y.111A as described by the authors. It was to be noted that the gear tooth loading factors specified were appropriate to the life factors associated with the steam and gas turbine machinery.

In the early initial scheming, epicyclic gears were considered for the gas turbine reversing drive which in Fig. 17 could have been co-axial with the secondary pinion. However, reversing epicyclics and the necessary brakes for the power required were not developed at that time. Other brake reversing schemes were considered but were also laid aside because of the lack of a suitable brake. The hydraulic coupling scheme selected gave a simple, sure and successful arrangement.

There were penalties to pay with reversing gearboxes in weight and efficiency. The power/weight ratio for the Y.102A and Y.111A was about 2·12 expressed as h.p./r.p.m./ton, whereas the value for the non-reversing Y.E.A.D.1 gearing was about 5·5. On the same basis, the power/weight ratio for the full power reversing epicyclic box shown in Fig. 28 was 1·13. This should be compared with the G.2 and G.4 for comparable gear ratios which gave 3·5 and 4·3 respectively.

The losses in Y.102 calculated and measured as far as possible by various methods, including the difference between input and output torque meters, gave an overall efficiency at full power with gas turbine boost of 94·5 per cent. This included losses due to journal and thrust bearings, fluid couplings, oil pump and auxiliaries. With steam only the maximum efficiency was 96·5 per cent (see Fig. 37).

As the Y.102A was the most advanced combination mach-

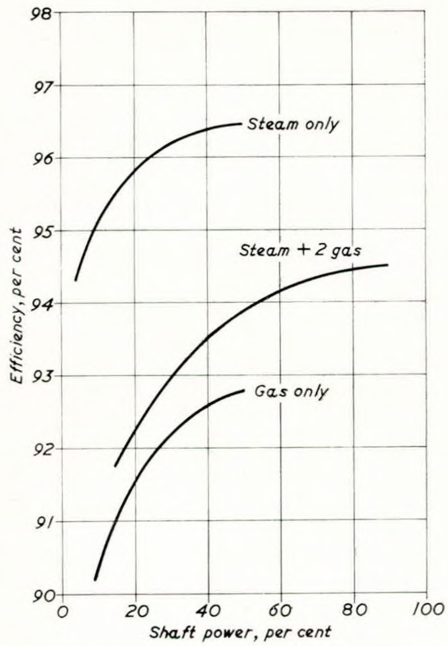


FIG. 37—*Y.102A—Main gearing efficiency*

inery of its type, the Admiralty decided that the essentially new components should be thoroughly developed and tested on shore before installing in ships. Thus the shore trials machinery known as Y.102 was installed at his company's works. Each new aspect of the gearing, bearings, shafting arrangements, clutching and manœuvring gear was thoroughly tested under all expected conditions.

Such aspects as helix correction, alignment, tooth meshing temperature, limiting oil flow to the sprayers, circumferential temperature of bearings and the reliability with wiped bearings were examined for several important meshing pairs with different oils. In addition to proving the operational requirements of the machinery this immense development gave the limits and margins required in machinery of this class to cope with all likely eventualities.

The complete installation was also tested at sea and proved successful both on machinery built by the company and their licencees. The precautions taken in design and manufacture and provision for aligning such a multitude of bearings incorporated in one box proved their worth.

In the opinion of his company there was ample margin for rating up the gear loading. On a life basis the boost gears could go to 700 K and the cruising gears to 500 K with the material used in Y.102. This would require further bearing development.

It would also be advisable to give attention to means for absorbing energy when stopping the machinery, as elegant methods of reversal were available awaiting the solution to this problem.

MR. T. P. JONES (Associate Member) said that in a most interesting and clearly presented paper the most interesting thing for him was to learn of recent results obtained by A.V.G.R.A. on nitrided gear components tested up to K values several times those normally used.

Ten years ago in a paper* presented to the Institute, the speaker and Mr. H. N. G. Allen had stated their belief in the future of nitrided gears. From the ensuing discussion it was clear at that time their view was not widely shared; quite the contrary, in fact! However, their own experience in the use of this method now extended over epicyclic gears aggregating more than two million h.p. with individual gears up to 15,000 h.p. in the high speed range and up to 260,000 lb./ft. torque in the low speed range. The experimental work reported by the authors was therefore already well backed up by service experience, although not in the field of British naval propulsion.

On the Continent there was a similar amount of experience with nitrided epicyclic gears and also considerable experience with large nitrided parallel shaft gears. High powered naval propulsion gears of both types had already been built. It was disappointing to think that the lead had been taken elsewhere, particularly as some of the pioneering work which had made it possible had taken place in this country.

He sensed a tone of regret in the authors' remark that there had been so few new gear designs for the Royal Navy's post-war programme, but what might be lacking in numbers seemed in some cases to have been made up in complexity. Abroad, development had taken place on different lines and the combined gas turbine and Diesel engine propulsion arrangement for the *Köln* class frigates was an example, illustrated in Fig. 38. A combination of epicyclic and parallel shaft gears

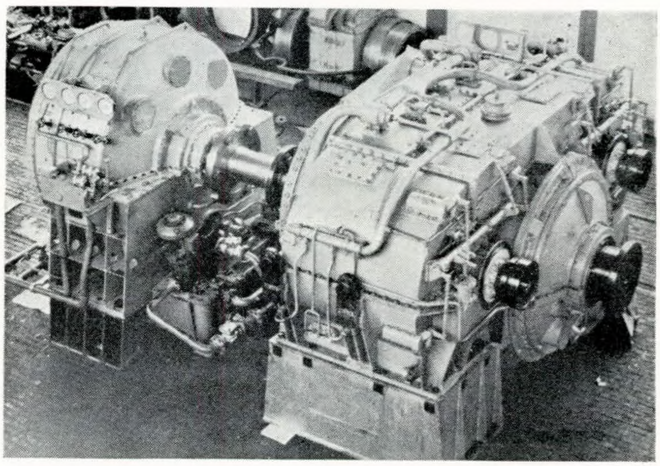


FIG. 38—*Combined gas turbine and Diesel engine propulsion arrangement*

had been used. Two 3,000 h.p. Diesel engines were used for cruising. Their speed was first reduced through two-speed declutchable epicyclic gears and then the power was transmitted by means of a parallel shaft transfer gear to the propeller shaft. For full power a 13,000 h.p. declutchable epicyclic gear was used to connect a gas turbine to the propeller shaft and the second of the two speeds available in the Diesel engine epicyclic gears was used. For changing speed and declutching, brakes were employed which were similar to those used for reversing the *Brave* Class gears, illustrated in Fig. 28. All the hardened gear components were nitrided.

The combination of epicyclic and parallel shaft gearing was something which would, he believed, be increasingly used in the future. Fig. 39 showed a turbine and double reduction gear. It was for the pump of a suction dredger. The transmitted power was 5,500 h.p. and the turbine speed was 4,000 r.p.m. The first reduction epicyclic gear had a ratio of 3·75 : 1 and its planet carrier was overhung from the second reduction pinion. The parallel shaft gear had a ratio of 3·31 : 1 and the output speed was 320 r.p.m. The first reduction train took up little, if any, more space than a gear toothed coupling. It would be easy to use the first reduction gear as a means of declutching the prime mover. Also, possibly combined with a separate propeller shaft brake, an epicyclic unit could be used for reversing. This seemed to be much simpler than some of the arrangements illustrated by the authors and their comments would be appreciated.

Was it possible for the authors to give any information about the efficiency of the gears in Figs. 1, 9, 17 and 21? So far as the gear illustrated in Fig. 28 was concerned it had

* Allen, H. N. G. and Jones, T.P., 1952 "Epicyclic Gears". Trans. I.Mar.E., Vol. 64, p. 79.

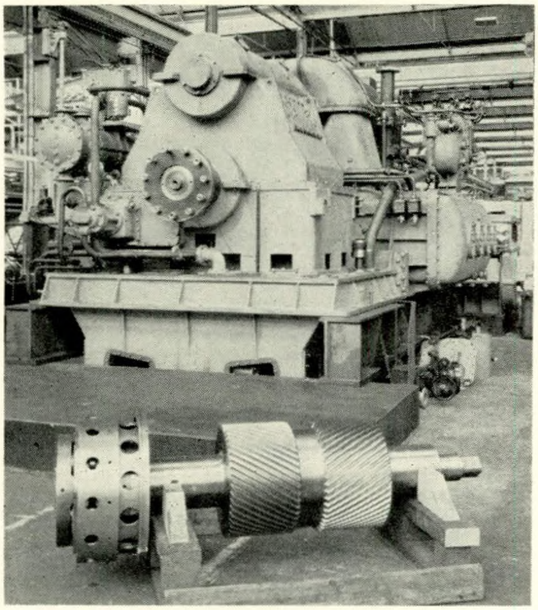


FIG. 39—*Turbine and double reduction gear for the pump of a suction dredger*

been measured when transmitting 3,800 h.p. in both directions and was found to be 97 per cent ahead and 96·4 per cent astern.

He shared the authors' regret that measuring equipment was not generally available for checking large size gears to a very high degree of accuracy. Presumably they had noise in mind when they stated their belief that higher accuracies than those specified in B.S.1807:1952, Grade A1 would be needed in future. Could the authors comment on the noise levels of the various installations which they had described?

It was easily possible to achieve better accuracies than Grade A1 for the sun and planet wheels of even the largest epicyclic gears and measuring equipment was available to check these comparatively small components. An idea of the size of the components of a very conservatively rated epicyclic propulsion gear was given in Fig. 40 which showed a double reduction gear used to reduce the speed of a Deltic from 1,630 r.p.m. to a propeller speed of 180 r.p.m. Two recently installed gears each transmitting 2,200 h.p. had already given 8,000 hours of completely trouble free service in a vehicle ferry

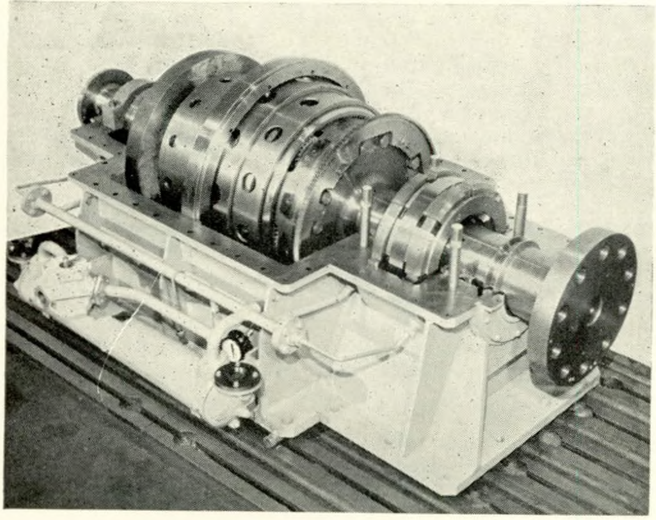


FIG. 40—*Double reduction gear used to reduce the speed of a Deltic from 1,630 r.p.m. to a propeller speed of 180 r.p.m.*

with an extremely high utilization factor. The Michell thrust bearing gave an indication of the size of the gears. The sun and planet wheels were nitrided with a modest K value of 280 in the first train and 220 in the second train.

Had time allowed it had been his intention to give some further information about the gears used in the *Brave* Class. He would submit this in writing but in the meantime he hoped the information he had already given would give the authors even more confidence that nitrided propulsion gears were likely to be used by the Royal Navy in the future, with the advantages which were already being gained from their use elsewhere.

COMMANDER E. B. GOOD, R.N. (Member), said that it was with some trepidation that he addressed the meeting, with so many gear experts present. However, as an ex-user of naval gears and as an active tester of gears and caller for designs of gears he hoped his remarks might prove to be of some interest.

Although the authors had written an extremely comprehensive paper and a valuable addition to gearing literature, it would be useful if they would enlarge on it in their written remarks by covering some of the following areas of gearing design: firstly, gearing efficiency; secondly, gearing noise; thirdly, some remarks on balancing in general, and in particular the balancing of long torque tubes.

Despite this, the paper enabled some extremely useful and broad conclusions to be drawn.

Firstly, the R.N. Y.100 experience in the *Whitby* Class and, even allowing for the wide safety margin, the R.C.N. hardened and ground gearing experience, pointed forcefully to the advantages of specialist manufacturers for gearing and the benefits accruing from taking a large number of gearboxes from the same firm. One previous speaker had, however, drawn quite a different conclusion.

The paper also provided a conclusive demonstration of the need to use hardened and ground gears if really attractive savings in weight and space were to be achieved and here he took Dr. Davis to task. Dr. Davis had mentioned very small savings and wondered whether they were worthwhile. In a naval design it was important to pursue with determination and ruthlessness the attempt to save every single ounce, provided that in so doing the design was not prejudiced. One might only be saving a small net quantity in any particular component but they all added up together and there should never be any excuse for putting weight unnecessarily into a ship.

Finally, the philosophy of steady development of design and the whole evolution of the Y.100 gearing through stages, Mark I to Mark III, pointed the way to the successful production of a reliable advanced gearing design.

It would be useful if the authors could give their precise reasons for selecting 500 K as the limit for future design loadings. This, he thought, might be unduly conservative, and possibly 600 K might be a better figure to choose.

The appendix to the paper, giving details of the S.S.S. clutch, was very valuable, and his own experience in reading a paper* with his co-author, Commander Dunlop, around the country showed that considerable interest and comment were aroused by this particular clutch, and the figures that had been given would be read by many with great interest.

Fig. 36 indicated clearly the torsional vibration which was excited when two elastic shaft systems were connected and showed clearly the reason why in their paper the speaker and Commander Dunlop stated that a marine clutch for main engines could not work satisfactorily without a suitable locking-in arrangement, a lesson which, incidentally, they also learned from the sea trials of Y.100 machinery.

Finally, he associated himself with the authors' views in their conclusions on page 95 that the design of a reversing gearbox for an all-gas-turbine ship could now be made with complete confidence. This would, of course, be based on the

* Dunlop, J. M. C. and Good, E. B. 1963 " Machinery Installations of Guided Missile Destroyers and General Purpose Frigates ". Trans. I.Mar.E., Vol. 75, p. 1.

vast experience now gained in the Royal Navy on this type of design.

MR. J. BILTON said that the information gathered together by the authors on the subject of propulsion gears would form a valuable contribution to the historical record and provide a landmark for future development in this sphere.

It was interesting to note the experiences relating bearing temperatures to the length/diameter ratio and also to diametral clearances. At a conference* arranged by the Institution of Mechanical Engineers in 1958 many references were made to the benefits obtained from shorter bearings, not only from the temperature viewpoint but also from the overall length of the installation.

On page 87 the authors referred to the hydraulic couplings in the Y.102A and Y.111A installations and stated that the output from the gas turbine was restricted in order not to exceed the permitted operating speed. It was quoted that the limitation was imposed mainly by the centrifugal stresses in the hydraulic coupling.

Having had something to do with this he wished to make it clear that the couplings were designed to a speed which was specified and they were tested to 25 per cent above that speed with a very comfortable stress. In actual fact the couplings could run continuously at 20 per cent over the designed speed when they would transmit over 70 per cent more power. With a new design specification and appropriate thicknesses, couplings could be made to enable much higher speeds to be achieved. He believed that it was necessary occasionally for H.M. ships to move quickly and he did not wish the impression to be gained from this paper that the strings attached to couplings were so short that the number of knots was limited!

MR. R. E. SALTHOUSE, B.Sc.Tech. (Associate Member) said that in considering gearcases the authors had rightly stated that this aspect of design required much further investigation. One feature that had been shown to be entirely satisfactory in practice was the use of a three-point support, providing that the gearcase itself was sufficiently stiff and providing that the designated three points were used throughout the manufacturing processes.

The authors had drawn attention to the care that must be taken in protecting surfaces, which were subsequently to be in contact with E.P. oils. Although paints had been expressly forbidden by the Admiralty, the oil wetted surfaces of the Y.102 in shore trials and *Ashanti* gearcases were, in fact, painted before this prohibition came into force. Different paints were used in the two cases being considered and, so far as the contributor was aware, these surfaces remained undamaged after being in contact with hot E.P. oils for many hours. On the other hand, difficulties had certainly arisen in the use of the Admiralty's specification. Alternative and more practical methods of protecting surfaces would appear to be well worth further investigation and the authors' comments on this would be appreciated.

MR. H. A. CLEMENTS said that the appendix to the paper described the maximum relative accelerations likely to occur in the Y.100 machinery, also the Y.102 and Y.111 installations, this maximum acceleration occurring more particularly in a twin-screw vessel under large degrees of helm, in rough sea conditions.

It also described how the S.S.S. clutch engaged at the instant that the input shaft accelerated through synchronism with the output and, when the S.S.S. clutch was moving towards full engagement, the dashpot became effective. As pointed out in the paper, the purpose of the dashpot was to accelerate the slower output member and decelerate the faster input member during the final engaging travel of the clutch.

He had previously shared the authors' view about the advantages of installing the S.S.S. clutch in the second motion shaft as described in the paper, however, on further reflection a high speed clutch could be seen to have certain advantages.

It would be appreciated that, if the relative shaft rotation to engage the clutch was referred to propeller shaft speed, a clutch incorporated in a high speed shaft presented an easier problem than a clutch in a lower speed shaft. If the clutch were incorporated in the high speed shaft the relative acceleration was, of course, greater by the gear ratio, but the torque effect due to the dashpot to synchronize the shaft systems was equally stronger by the gear ratio.

When all these comparisons were considered it could be seen with regard to the relative acceleration between the propeller and the turbine that there was no disadvantage in incorporating the clutch in the high speed shaft. This was confirmed by experience, as the S.S.S. clutch in the high speed shaft of the Y.100 machinery was no less successful than the Y.102 clutch, when subjected to the highest relative acceleration conditions and, in fact, clutch engagement could hardly be detected, even when standing adjacent to the gearbox within about 2ft. of the clutch.

However, a clutch in the high speed shaft line had some disadvantages and the most important disadvantage was pointed out by the authors, which was that the high speed gearing was continually rotating, with continual losses and noise, when its turbine was shut down.

With a high speed clutch special measures must be taken to prevent the build-up of sludge within the clutch and this in fact proved to be the only potential difficulty in the Y.100 S.S.S. clutch. The problem was effectively overcome simply by careful location of the oil inlets and outlets.

MR. H. SINCLAIR said that his interest was more particularly in the appendix to the paper, since it related to S.S.S. clutches. When a certain achievement had been realized in terms of horsepower and speed, it was sometimes said, "Where do we go from here?" It was perhaps interesting to marine engineers to note that in the industrial field a big step forward had recently been taken in the placing of contracts involving gas turbine driven alternators, which came however outside the field of gearing covered by the paper. There would be four S.S.S. clutches each of 100,000 h.p. at 3,000 r.p.m., and eight each of 40,000 h.p. at 3,000 revolutions. Glancing for a moment at Fig. 34 one noticed that the intermediate member which moved along helical splines was directly actuated by the pawls. When one came to higher power and higher speed and larger clutches, it was expedient to use the relay type of S.S.S. clutch in which an S.S.S. unit of small diameter and weight, with corrrespondingly light pawl mechanism, was used to actuate the heavy intermediate member positively and inter-engage the clutch teeth at the moment of passing through synchronism. The relay type of S.S.S. clutch was obviously the correct selection for these large gas turbine driven alternator clutches, which incorporated a dashpot but did not have a locking sleeve control as was described in the appendix to the paper.

MR. N. J. JAMES (Associate Member) said that the company with which he was associated did not manufacture propulsion gears. They had had, however, parallel experiences with turbine generator gears, where similar problems existed, to a much lesser degree, of course. Firstly, with regard to cleanliness, dirt was no longer the arch enemy. This had largely been overcome by the methods stated by Mr. Sampson, i.e. chemical cleaning and meticulous attention to mechanical details during manufacture. No relaxation could, however, be tolerated. Hardened and ground gears were now used exclusively for turbine generator drives. Gas carburized gears were gradually being superseded by nitrided gears, almost entirely due to the distortion problems mentioned by the authors.

His company felt that, commercially speaking, they were approaching the lower physical limits with regard to cylindrical grinding of journals and tooth accuracy, and did not expect much improvement in this direction. Gearcases had received their due share of attention in terms of both accuracy and rigidity.

* I.Mech.E. 1958. Proceedings of the International Conference on Gearing.

They had graduated to thin-walled steel-backed bearings with small L/D ratios but were not entirely happy with bearing design. So called precision interchangeable bearings might easily vary 0·0005in.-0·0006in. from standard thickness and this was not good enough to ensure perfect meshing and associated reduced tooth contact noise. Adjustable bearings were difficult to accommodate in small gearcases and were, they felt, undesirable.

Restricted clearance bearings had not been too successful. He noticed that in the Y.E.A.D.1 proving trials the clearances had to be increased to obtain satisfactory results.

In order to dictate more accurately the position of the pinion at various loadings multi-lobed and hydrostatic bearings were being considered. What were the possibilities, or had the authors any experience of such devices?

Their effect in resisting shock loading due to underwater explosions, short circuiting of turbo-generators and, by no means least, in reducing noise level might be of value.

Mr. S. Archer, M.Sc. (Member) said that the authors had shown clearly that the transition from soft to hardened and ground gearing in naval practice had been by no means free from growing pains, and in particular the attempts to exploit through-hardened alloy steels of greater hardness had been disappointingly unrewarding. After reading this paper, however, he felt sure that all would agree that the results of these intensive research and development efforts had nonetheless been very well worthwhile.

One reason why Lloyd's rules for merchant gearing gave proportionally diminishing credit for increased hardness was no doubt reflected in these rather disappointing results with the higher alloy steels.

On page 78 of the paper there was reference to gear-cutting errors in connexion with the *Whitby* Class ships, and the authors suggested that shaving errors could not be held entirely responsible since pitting had also occurred in ground pinions. Was it not possible that this might be due to errors in the main wheels? Were any noise measurements made to show differences between port and starboard boxes and were these different as regards handling of apices? What sort of percentages at full power had been achieved with these *Whitby* Class vessels? He remembered Mr. Nicholson, when giving his paper*, indicating (in reply to a question by the speaker) that at the time of writing, the average power used in R.C.N. *St. Laurent* Class ships was only 14 per cent. of full power and a mere 2½ per cent or so of the total shaft horsepower-hours at sea had been at full power. Consequently, it would be of interest to know roughly for what proportion of their total service shaft-horsepower-hours the *Whitby* Class gears had operated at full power.

Merchant experience with hardened and ground gears at much lower horsepowers, had not been altogether trouble-free. Some of their defects had been metallurgical leading to tooth failure and even rim fracture and, he suggested, therefore, that the only final test of these highly-loaded case-hardened gears would be prolonged operational experience at full power at sea, as in wartime.

Figures of 160 K for secondary and 200 K for primary gears (page 79) were suggested by the authors for En26 pinions associated with En9 wheels. In the merchant service these would be considered somewhat high unless the pinions were case-hardened and ground, possibly also the primary wheels, and with the maximum possible accuracy for the hobbed and shaved main wheel.

The use of the central drive for the Y.E.A.D.1 secondary pinion was very interesting and logical. One thing that occurred to him as a Lloyd's surveyor was how to examine the pinion quill for cracks in service. He did not know what sort of inspection had been carried out, but it would need to be pretty meticulous in the first place, because the junction with the pinion occurred at probably the worst part of the material.

The interesting analysis of deflexion tests in Fig. 15 intrigued him. He had never realized before that a gearbox could look so much like a piece of jelly! The diagram was exaggerated, of course, for the purpose of illustration, but it had triggered off the idea in his mind that it might be possible to stiffen up the H.P. side, which in the ahead direction seemed to be subject to considerable vertical lifting forces because the measured upward deflexions there were quite large compared with the downward deflexions on the other side and it should not be very difficult to put heavier scantlings and connexions on the H.P. side. What did the authors think of that?

Was he to understand from Fig. 16 (the main structural components of the gearcase of Y.E.A.D.1) that there was no support at all for the transverse frame at the forward end other than through the stiffness of the bottom half structure? In other words, did the chocking in fact stop short where shown in the paper?

Commander J. I. T. Green, O.B.E., R.N. (Member) said that he was very pleased to see in the introduction to the paper a quotation from Captain Tostevin's famous paper on naval gearing. He had known him quite well and felt sure that he would have been greatly interested in the developments described so lucidly in this present paper.

When the Navy had to put up with recurring defects and ships became immobilized a name was usually found for the disease. The early three drum boilers had their "wrapperitis", and this was followed by "condenseritis". With regard to gearing, apart from individual cases, serious as they were, he doubted whether gearing had ever suffered such extensive troubles. The difficulties described in the paper were related to the deliberate policy of cramming more and more power into smaller spaces, leading to a search for perfection.

Examples of this search for perfection were highlighted in Figs. 3 and 4. A helix angle correction of half a minute of arc had to be smoothly joined to an uncorrected length. The very act of shaving introduced new errors, small but important. A point arising from a careful look at Fig. 3 was why the after helix correction ceases at 4in. from the gap. The forward helix uncorrected portion corresponded with the region of maximum total deflexion, but this did not apply to the after helix. It was interesting to read later in the paper that helix correction had been dropped in the case of the gas turbine installations, largely on account of the fact that these had single helical gears. If double helical gears must have this correction would it not be possible to compromise by making it continuous over the full length of each helix? Could this not be applied on first hobbing?

When they came to the Y.100 Mark II there was a remarkable transformation. These gears were finished by grinding instead of shaving, but it was not stated which type of grinding was used, form grinding or generating grinding. He assumed the latter, which he understood was much the best way to carry out profile correction. At any rate, the results in *Rothesay* and *Leander* Classes appeared to justify the change and the best piece of news in the paper was the fact that one set had run quite satisfactorily in a grossly misaligned condition.

Even so, the authors did not convey the impression of a definite decision to use grinding in future. Perhaps they would care to enlarge on this.

Mr. P. Whyatt said that the main synchronizing clutch had been fully described in the appendix. Brief mention should be made of the two other S.S.S. clutches used in the Y.102/Y.111 machinery installations, namely, the gas turbine starting clutch and the gas power turbine turning gear clutch.

The gas turbine starting clutch was used to connect the drive from the air starting motors to the gas turbine compressor shaft. When the gas turbine became self-sustaining the clutch would disengage upon torque reversal and the air motors would be shut down. The output side, carrying the pawl mechanism, could then run up to its maximum operating speed.

The gas power turbine turning gear clutch which was shown in Fig. 17 on page 83 connected the turning gear motor

* Nicholson, D. K. 1961 "Experience with Hardened and Ground Gearing in the Royal Canadian Navy" Canadian Supplement, No. 4, June 1961. (Bound volume Trans. I.Mar.E., Vol. 73, p. 441).

to the gas turbine primary gears in order to barr over the power turbine and was also used to enable the boost and manœuvring manual clutches to be engaged. This clutch had a pawl free position to prevent engagement should the gas turbine shaft be rotated astern during crash astern manœuvres. The clutch automatically disengaged upon torque reversal, the output side running up to its maximum normal operating speed. Both of these clutches were of very simple design and construction.

The principle of the S.S.S. clutch readily lent itself to the Admiralty's policy of both local and remote control of the main machinery and in the barring application it had the advantage of allowing the barring motors to be started at any time, the clutch automatically engaging at barring speed such that the machinery need never come to rest until the barring period had been completed.

This obviated the need for manual operation of dog clutches and also the necessary precautions to adequately safeguard against the maloperation of a dog clutch.

In the transition from the simple single reduction gearbox to the complex double reduction gearboxes of the present machinery installations the accent had been to concentrate on more machinery in a given gearbox space in order to achieve a rigid structure giving maximum stiffness and thus reducing the distortion under load to a minimum. This had led to the sacrifice in many instances of accessibility for maintenance and visible inspection of the components likely to require attention or replacement during the life of the gearbox.

In the paper* by Dunlop and Good stress was placed upon the use of a full scale mock-up of the machinery spaces and of the valuable contribution this made to the final ship installation. When one considered the complex nature of the modern gearbox and its position of prime importance in the overall machinery installation he felt sure that the use of a full scale mock-up could well be extended to the gearbox itself. Although in the case of the Y.102 installation a full size test rig was built from which many useful lessons were learnt, the completion of a test rig of this proportion in a tight production programme naturally arrived rather late to effect design changes to components already in production.

* Dunlop, J. M. C. and Good, E. B. 1963 "Machinery Installations of Guided Missile Destroyers and General Purpose Frigates". Trans. I.Mar.E., Vol. 75, p. 1.

A mock-up should be built at a very early stage in order to establish the correct assembly procedure of components and also to check the feasibility of correct positioning of the datum faces and check points used not only for the initial shop assembly but under conditions that would exist in the ship during repair.

The layout and planning of electrical cables, thermocouples and oil pipes to bearings and controls could also be planned well in advance, which would lead to much shorter assembly times and possibly result in much cleaner gearboxes.

At present many of these points were left to individuals during final assembly when the machinery was at rest, only to cause fouling and possible damage when the machinery was finally turned over on the shop test.

One valuable use of the mock-up would be the simulation of various failures that might occur during the life of the gearbox, the accent here being on the procedures to be adopted for replacement of components using the actual tools and equipment that would be available in the ship and working under the cramped conditions of the machinery spaces.

The principle to be established here was the great need to consider the simplest and most effective method of assembly and repair facilities well in advance, so that as much planning and assistance might be given to the engineer officer always working under a tight programme and inevitably under the most adverse conditions.

COMMANDER H. L. PRATT, R.N., said that he hoped that provision would continue to be made for adjusting the relative alignment of the rotating elements: it was undesirable to commit oneself to running with the alignment as originally set up and in any case the shape of everything on shipboard must be expected to alter during the life of the vessel, even in the absence of such risks as action damage. He noted that the present means of verifying alignment amounted only to aiming at restoration of the original dimensions local to the bearings and suggested that means be provided to verify (and therefore correct) the absolute alignment, in space, of the gearing elements. He also asked for the authors' remarks on the extent and reliability of the provision for monitoring the behaviour when running, of these very elaborate assemblies, primarily in respect of bearing temperatures but also perhaps in respect of axial displacements of the elements.

Correspondence

DR. H. E. MERRITT, M.B.E., in a written contribution said that in the literature of marine propulsion gears, tooth loading was thought of almost exclusively in terms of the K factor. The study of tooth strength had been virtually ignored and tooth breakage regarded as an act of God. The present paper, valuable as it was as a record of some expensive experience, stopped at the point when it was beginning to be clear that fatigue failure under bending stress would have to be taken quantitatively into account.

A great deal of work had been done on this subject in the automotive field and experience in that field showed that in the presence of satisfactory lubrication, tooth strength and not tooth surface breakdown was the limiting factor. This held good for relatively small tooth number combinations and could be expected to apply *a fortiori* to marine propulsion gear having larger tooth numbers.

Subject to fuller investigation of side-effects, it was probable that given more care in tooth design, case-hardened marine gears would carry around 2,500 K for 10^8 cycles. It was important that tests under such loading be not prematurely arrested by tooth breakage and to design the necessary strength into the teeth required, first, an adequate basis for bending-stress calculation and secondly, further study of the fatigue properties of case-hardened or otherwise skin-hardened gear teeth. The need for such studies was of course quite obvious, from the fact that allowable contact loading increased roughly as the square of the hardness, whereas the endurance limit of bending stress increased at a somewhat lower rate than the hardness, some side-effects due to residual stress apart.

In the A.V.G.R.A. tests reported by the authors, the nominal bending stresses under the loads at which tooth breakage occurred were of the order of 35,000 lb./sq. in. in the pinions, and 28,000 lb./sq. in. in the wheels. With the same basis of stress calculation, the endurance limit of case-hardened steel under uni-directional bending was around 60,000 lb./sq. in. The reason why the pinion failures occurred remained to be sought. There were several possible causes. Firstly, an error of tooth alignment over the whole face, equal to twice the static tooth deflexion under the mean loading would double the bending stress. But probably more important, the endurance limit of case-hardened gear teeth was sensitive to several factors, which were difficult to control even in large-volume production and still more difficult when large, individual forgings were in question. Case-hardened teeth also exhibited a greater variety of types of failure, both in fracture and in surface breakdown.

It would therefore be helpful if the authors could give the results of detailed examination of the fractures, the microstructure of the surface layers, the hardness gradient and the carbon gradient. It would also be interesting to learn whether the teeth had been ground over the fillet curves, to their detriment, or shot-peened, to their advantage.

It was also submitted, with the greatest possible respect, that marine engineers, who were beginning to study highly stressed gears, might examine other aspects of the practice of their brethren in the automotive field and tooth design and helix modification in particular.

DR. T. W. F. BROWN, C.B.E., S.M. (Member) wrote that the paper was one of very great interest and that he had been fortunate enough to be concerned in a large number of the tests described by the authors.

All the gearboxes in Fig. 33 were tested as part of full scale trials of complete geared turbine machinery for the Royal Navy, as well as three separate gearboxes for the Royal Canadian Navy. These tests were carried out at Pametrada Research Station, now The British Ship Research Association, Wallsend.

The distortion measurements given graphically in Fig. 15 were measured as accurately as possible and it was remarkable that the distortions measured did not affect the meshing in such a way as to cause tooth defects. Distortion at sea, however, in the Y.100 gearing might well have been contributory to the trouble experienced, as this was considered to be a lightly built gearbox in relation to the other designs shown in the paper. This could be seen in Fig. 1 where a considerable portion of the gearcase was overhung on single wall construction of the main box.

On page 79, item (iii) the authors examined a number of factors to account for variation in performance. It was considered that the gearcase supporting structures used on shore trials were comparable with the structures in the ship.

With reference to item (v) on the same page, a similar occurrence was recorded during the Y.E.A.D.1 trials at Pametrada, the after main wheel bearing being affected. The cause was an unequal rise due to thermal expansion between the main gearcase and the brake, as the allowance made for this difference in lining up cold was not adequate. Although the situation was appreciated during the trials, no trace of any consequential failure in the rest of the gearcase was found. This led to the general conclusion that a set of gears would put up with considerable distortion and misalignment providing that the mating materials were correctly chosen.

Dr. Brown concluded by thanking the authors for an extremely useful paper on the development of naval gears.

REAR-ADMIRAL J. G. C. GIVEN, C.B., C.B.E. (Member) wrote that he would like to congratulate the authors on their informative paper and for carrying on the saga of naval gearing progress and development in the best tradition started by Tostevin and continued by Joughin.

He welcomed the new approach to gearcase design mentioned by the authors which was long overdue and suggested that the designers entrusted with the job should :

a) not have access to any old cast iron designs or fabricated designs following the cast iron principles;
b) initially be told to forget the oiltight envelope required and concentrate on the simplest structure to carry the estimated static and dynamic loads imposed by the gear element journals;
c) finally, get the simplest oiltight envelope which would meet the requirements of accessibility, freedom from resonance and resistant to the transmission of structural and airborne noise.

He hoped that the authors might be able to enlarge a little upon their experience with so-called flexible couplings with particular reference to :

1) the true value of barrelling effects;
2) whether the stage had not been reached where only true axial freedom need be provided;
3) whether they had any information of the transient thrusts in double helical secondary pinions when manœuvring.

Finally, he wished, for historical record, to submit one correction to the paper, namely that the first British designed and manufactured double reduction set of gearing fitted in a British warship was not in the *Daring* Class but in a *Loch* Class frigate which went into service *circa* 1944.

COMMANDER J. H. JOUGHIN, D.S.C., R.N. (Member) wrote that this was indeed an excellent paper and covered a great field of development. He was particularly struck with the work on the successful production and testing of the induction hardened and nitrided gears. This work had encouraged his company to undertake the production of nitrided gears.

The authors had remarked at the end that the problems of maintaining good alignment, throughout the wide range of powers at which naval vessels operated, required further study and that the design and construction of gearcases required further work. Commander Joughin seemed to remember that, in the initial objectives undertaken by A.V.G.R.A., was the study of this whole field of gearcase alignment and distortion under load. Even given readily usable means of measuring distortion of gearcases and the relative position of gear axes, he realized that this was a most complex field, particularly when machinery of the Y.102 and Y.111 type was considered.

He remembered some years ago being shown the gearing of a merchant vessel which had completed some twenty years' successful service and which was fitted with the pinion bearings mounted in a McAlpine frame. This double helical gearing was in excellent condition. This experience had continued to stress for him that, as the authors remarked, it was the even distribution of loading along the length of the gear under load that was required, rather than massive stiffness, particularly since a gearcase was a type of structure which it was very difficult to make torsionally stiff. If the authors had any suggestions regarding the ways and means of studying the maintenance of this gear alignment, he believed they would be most valuable to those concerned in the design and production of gearing.

MR. S. ARCHER, M.Sc. (Member), in a further contribution, wrote that in connexion with Part II of the paper he would like to know if the authors could report any trouble with high speed turbine or gearing thrust bearings of the Michell tilting pad type and, if so, whether such trouble had been associated with the use of any particular grades of oil, especially those of the E.P. type containing chlorine additives.

MR. T. P. JONES (Associate Member), further to his verbal contribution, wrote that the gear illustrated in Fig. 28 had nitrided components with K values up to 430. Fig. 41 showed the overall machinery arrangement of the *Brave* Class. The

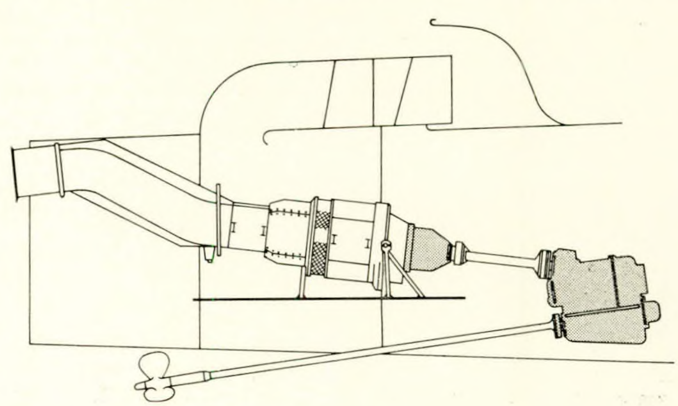


FIG. 41—*Overall machinery arrangement of the* Brave *Class*

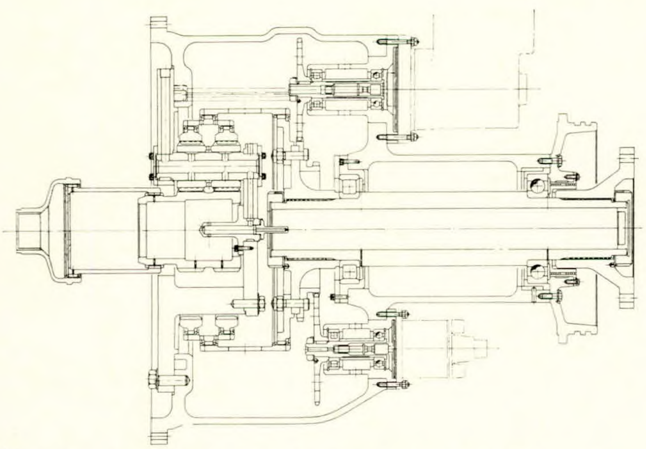


Fig. 42—*Cross-sectional arrangement of the Proteus primary gear*

primary gear, which formed an integral part of the marine Proteus, weighed only 1·5 oz./h.p. Its weight was in fact lower than the original aircraft gear, which it replaced when the Admiralty sponsored the development of the Proteus as a marine engine. In fairness it must be pointed out that the ratio was considerably less than that of the original aircraft gear. A cross-sectional arrangement of this gear, uprated to be suitable for 4,500 h.p., was shown in Fig. 42. The annulus diameter was 10·75in. and the reduction ratio was from 11,600 to 5,250 r.p.m. All the main gear toothed components were nitrided, with K values up to 370.

Mr. J. Bilton wrote, in a further contribution, that the various bearing modifications referred to by the authors, namely, shorter bearings, larger clearance, gutters cut to the ends of the bearings, all had a common result in increasing the flow of oil through the bearing and thus reducing the temperature.

On large fluid couplings at 3,000 to 3,600 r.p.m. it had been found good practice to provide a shaft clearance of between 1·25 and 1·5 thou. per inch. with three equally spaced axial grooves extending to within $\frac{3}{16}$in. from the end of the bearing. These grooves were cut with a milling cutter and had a circumferential width of about one-fifth shaft diameter, where they broke into the journal diameter. In this way a reasonable oil flow could be put through the bearing without having excessive radial clearance.

When three grooves were used care had to be taken to ascertain the position of the load line, since the edge of the groove should be more than 30 deg. away from this line.

Mr. A. D. Newman, B.Sc., wrote that he had read the paper with great interest and pleasure because it recorded a very real increase in the knowledge of an important aspect of marine propulsion and it did this by indicating the full process of research from relatively abstract work through small scale and full scale testing under controlled conditions on land to sea trials and seagoing experience and thence the feedback which led to further research and development.

He was also very pleased to see acknowledged the work of the A.V.G.R.A. research officers because, as a former member of A.V.G.R.A. committees, he had some knowledge of their invaluable efforts. He would like to see the vital part played in this development by the various Admiralty gearing inspectors and their staffs also recognized.

One of the points that emerged from the paper was that there was not enough known about the actual forces and loads on machinery in ships; the different behaviour of various Y.100 sets might be an indication of this. It had seemed to him for some time that the great effort that had been put into full scale gear testing on land had, to some extent been vitiated by this lack of knowledge and that a better balance of effort between full scale testing and the investigation of shipboard conditions might be struck.

He was very interested to see the point made by the authors that the arc between the oil inlet and the load line in a bearing was of considerable significance. He thought that this was an area of practical bearing design that could well do with further work. The seagoing experience with extreme-pressure oils was of very great interest, especially that in which, due to difficulty in obtaining the necessary supplies, certain ships recharged their lubricating oil systems with OM 100 after running satisfactorily on an E.P. oil, this recharging in some cases resulting in further scuffing which was once again arrested by the use of E.P. oils when available. This seemed to him to be very adequate proof indeed, if proof be needed, of the efficacy of extreme-pressure oils. He was very interested to have the authors' views on the flushing of lubricating oil systems, but would like their confirmation that the flushing oil they used was that which would be normally used in the machinery at sea and not a special oil of different viscosity. The points in the paper about cleanliness in general were of very great importance. Reference might be made here to the paper by Braley and Berg* in which the gearing or machinery failures in American warships during the last war were analysed. The main cause of failure was lack of cleanliness. He was slightly amused, incidentally, to see the recommendation that rubber-soled shoes should be used. His only experience of wearing rubber-soled shoes in a ship's engine room was certainly his last because he was lucky to escape with less than a broken neck and vowed never to wear such shoes again. Perhaps checker-plates were not as oily now as they used to be! He was very pleased to see set out the advantages of pre-finished steel shell lined bearings, particularly that of the ease of replacement without fitting work. This Admiralty experience should go a long way to removing the scepticism about this particular design philosophy shown by some engineers. An aspect of bearing performance which was not mentioned in the paper was that of the thrust bearings, especially those in the single helical gear sets which, as far as he knew, operated satisfactorily under relatively high specific loads. It did seem to him however, that this was an area where further work was necessary because the losses in such high-loaded thrust bearings were relatively high at the high speeds involved.

He noticed that the authors had not mentioned any problems with lubricating oil auxiliaries, particularly pumps. He was aware, as no doubt they were, of certain gearing failures due to trouble with lubricating oil pumps, and it seemed to him, as it had for some time, that there was a very good case indeed, particularly in warships, for driving the lubricating oil pumps directly from the main machinery. Some work carried out at Pametrada indicated the feasibility of such lubricating oil supply to bearings and it appeared that except for the abnormal conditions met in certain types of manœuvring of relatively high load at very low speeds or even under stationary conditions in the bearings, this type of lubricating oil supply would be perfectly adequate since to a very high degree the lubricating oil output could be matched to bearing and gearing requirements. Thus, with some form of safeguard for lubricating oil supply at relatively low speeds, which of course mean a relatively low rate of supply, a direct-driven oil pump would be perfectly adequate. He would welcome the authors' views on this.

* "Design and Service Experience with United States Naval Gears" by Cdr. W. W. Braley and M. S. Berg. (Institution of Mechanical Engineers, International Conference on Gearing, 1958.)

Authors' Reply

In replying to the discussion the authors wished to apologize for giving Dr. Davis and perhaps others, the impression that they were attempting to sweep away, rather than describe, all that happened between 1943 and 1948. This had not been their intention and they were aware that some progress in the improvement of gear cutting accuracy had been made during that period. Nevertheless, a large number of the gears fitted in wartime construction had undoubtedly been finished by filing. Filing was not resorted to during manufacture of the *Daring* Class gears and their success was undoubtedly due to the great improvements in gear cutting accuracy which had, by then, been achieved. The first *Daring* Class vessel entered service in 1952.

The authors were grateful to Rear-Admiral Given for pointing out that the *Daring* Class gears were not the first British designed and manufactured double reduction gears to be fitted in a British warship. In fact, double reduction gears of British design and manufacture had been installed in the wartime frigates *Loch Arkaig* and *Loch Tralaig* and went into service in 1945. These sets were of the locked train type with hobbed, double helical gears, 3,000 h.p., reduction ratio 5,000: 198, Lloyd's K factor 82 and they operated successfully for a number of years until both ships were scrapped.

Whilst sharing Dr. Davis' opinions that nitrided gears would come into more general use, the authors felt that they should say a few words in defence of induction hardened gears. Although each tooth was treated individually, operation of the machine was automatic and consistent. Variations in the metallurgical condition of the hardened teeth, as revealed by hardness measurements and etched hardness contours were extremely small. The effect of an electrical power failure during the hardening operation had been investigated and it had been found possible to re-harden, successfully, the tooth concerned. All teeth were crack detected after hardening. It had also been found, in practice, that induction hardened gears were less susceptible to grinding cracks than both carburized and nitrided gears.

The authors agreed that helix corrections should be avoided wherever possible, particularly in warship gears which operated, for the greater part of their lives, at reduced powers when the load was carried mainly on the uncorrected portions of the helices. The Y.100 Mark I gears had corrected helices on both ahead and astern flanks and difficulties had been experienced in checking alignment. It was partly for this reason that corrections were omitted from the astern flanks of the Y.100 Mark II gears.

Mr. Smith had given further information about the choice of materials for the Y.100 Mark I gears and had also assessed their performance in a fair and impartial manner. The authors agreed with that assessment and, in addition, wished to draw attention to the considerable reductions in weight and size which had been achieved when compared with past practice.

They also agreed with Mr. Smith that unground nitrided pinions could now be considered for future gear designs and, should replacements be required, for the original Y.100 vessels.

Mr. Smith had stressed the advantages and need for gear measuring equipment providing autographic records and had said that few marine gear manufacturers in the U.K. possessed such equipment. The Admiralty was most concerned about this and had recently decided to provide A.V.G.R.A. with certain gear measuring equipment, of Continental design and manufacture, which would, it was hoped, permit further advances in the accuracy and quality of naval propulsion gears.

Dr. Shannon had given a most interesting account of his company's experience with carburized, hardened and ground gears for the G.2 and G.4 gas turbines and this experience had undoubtedly been of great value during the design and development of the Y.102A and Y.111A gears.

The authors agreed that, provided satisfactory standards of manufacture had been achieved in all cases, the tooth loadings in those designs could now be regarded as conservative in the light of further full scale gear tests and sea experience. Neverthless, the use of higher tooth loadings in future designs appeared to offer only marginal reductions in size and weight. Root bending stress, rather than surface stress, had become the limiting factor and any large increase in permissible bending stress would necessitate closer control of accuracy in manufacture and assembly and the development of gearcases which would maintain gear alignment more closely. Whilst they did not doubt that, in time, these conditions could be met, they could not, at present, agree that a further large increase in the loading of naval propulsion gears could be safely undertaken.

Dr. Shannon had also referred to means for absorbing energy when stopping the machinery and the authors agreed that there was a need for work in this field and that experience with disc and hydrodynamic brakes and large c.p. propellers must be obtained.

Mr. Jones had mentioned his company's successful experience with epicyclic gears employing nitrided components and the authors were glad to acknowledge that the Admiralty had also benefited from this experience. As stated in the paper, many epicyclic reduction gears were giving satisfactory service in auxiliary machinery fitted in post-war ships but in these gears the nitrided elements were small. In the authors' opinion this experience was not, therefore, directly applicable to the larger gears fitted in high powered marine propulsion gear sets.

Until very recently the largest nitrided gear that could be produced in the U.K. was about 36in. diameter and there was doubt whether the depth of case obtained during nitriding would be sufficient for highly loaded gears in sizes much above 40in. diameter (Page[7]). However, facilities were now available in the U.K. for nitriding gears up to 80in. diameter and the results of the tests of the two 72in. nitrided gear wheels mentioned in the paper should indicate whether or not the depth of nitrided case was adequate.

Dr. Shannon had already stated that epicyclic gears were considered for the Y.102A and Y.111A gas turbine reversing drives but were rejected because suitable brakes were not then available. The use of epicyclic gears in the *Köln* Class frigates had been followed with great interest by the Admiralty and the authors thanked Mr. Jones for describing this application. It was, of course, relevant that those ships were manœuvred by means of c.p. propellers and, unlike the Y.102A and Y.111A installations, the gearing was not required to provide a reverse drive and the epicyclic gear brakes were required only to synchronize, within small speed differentials, the incoming gas turbine.

The authors appreciated the advantages of epicyclic gearing and felt that they were worthy of consideration in future designs.

Dr. Shannon had given figures for gear efficiencies measured on the Y.102 shore trials gear set. For the other designs mentioned by Mr. Jones, the efficiencies measured at full power were as follows:

Daring III	0·981
Y.100	0·976
Y.E.A.D.1	0·98

With regard to gearing noise levels, the authors regretted that they could not quote figures but could say that, notwithstanding the increases in gear loading, improvements in the accuracy of manufacture had resulted in substantial reductions in the noise levels of post-war naval propulsion gears.

Commander Good also had asked for information about gearing noise but would appreciate that the authors were not in a position to discuss this subject freely. With regard to gearing efficiency, the use of shorter journal bearings in the Y.100 and subsequent designs had helped to reduce bearing losses, as also had the use of higher lubricating oil inlet temperatures. The effect of the latter on gearing efficiency was pronounced, particularly at low powers, as shown in Fig. 43.

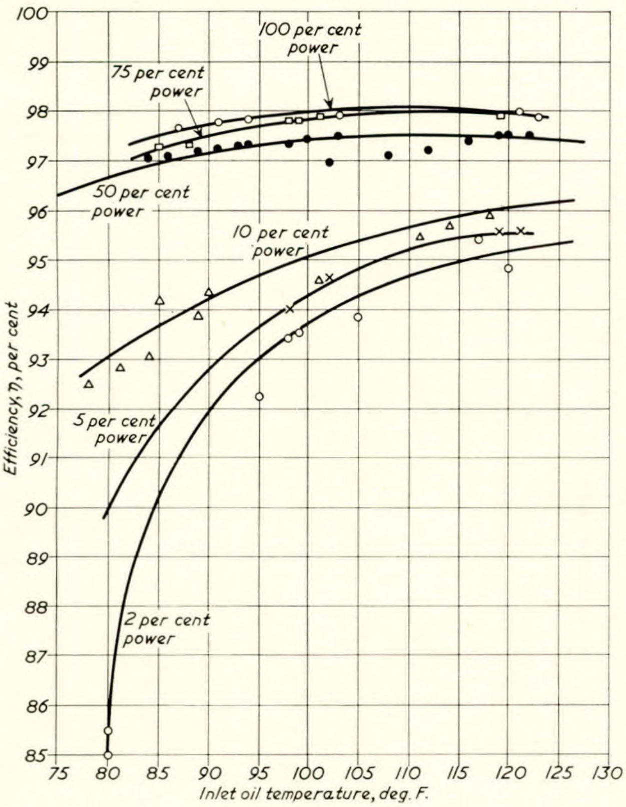


FIG. 43

Dynamic unbalance of gearing components and assemblies had become more important at the higher input speeds used in modern designs and with the increased emphasis on noise reduction. Unbalance could exist in a component itself or arise when two or more components were bolted together. The former depended largely on the accuracy and sensitivity of the balancing machine and on the care taken during the balancing operations. Some of the older balancing machines in general use were unsuitable for modern requirements and the Admiralty had been obliged to specify permissible limits of unbalance for current gear designs.

Commander Good was aware that the long torque tube and flexible coupling assemblies fitted between turbines and gearing in the Y.102A and Y.111A installations had proved difficult to balance to satisfactory standards and had required the use of special adapters and mandrels. The eccentricities of the end flanges, when mounted in the balancing machine were measured and checked after installation in the ship. Although the coupling bolt holes were jig reamed it was frequently impossible to reproduce the same degrees of eccentricity during installation and this had necessitated further "in place" balancing. Corrections were made by drilling radial holes, the size, depth and position being obtained from charts based on the differences between the "run outs" of the flanges when balanced and when installed in the ship.

The authors could not give precise reasons for the selection of 500 K as the limit for future design loadings. They agreed that this figure might be conservative, certainly from the aspect of surface stress and also possibly from that of bending stress. In future designs the limiting factor was likely to be the bending stress and the use of Lloyd's K factor as a criterion for the loading of highly rated marine gears would be less appropriate than it had been in the past.

The authors regretted giving the impression that short strings were necessarily attached to the Y.102A and Y.111A fluid couplings and thanked Mr. Bilton for correcting their misleading statement. The bearing design described in his written contribution to the discussion had been noted with interest.

Mr. Salthouse had referred to difficulties in implementing the Admiralty specification for cleaning and preserving gearcases and had suggested that alternative and more practical methods of protecting surfaces would be worth further investigation. A number of paints had been tested by the Admiralty Materials Laboratory and one of these had been extensively used in the Y.100 gearcases. In the majority of ships this paint had been entirely satisfactory but in others, trouble had been experienced, the paint flaking off and finding its way into the lubricating oil filters. It was clear that the performance of the paint depended very much on careful preparation of the surfaces, some of which were inaccessible and on conscientious, skilful application. It had therefore been decided to ban painting.

The Admiralty was not entirely satisfied with the methods employed at present, involving phosphating of large surfaces and the authors agreed with Mr. Salthouse that further investigations were needed.

Mr. Sinclair and Mr. Clements had given further valuable information about S.S.S. clutches. The successful performance of these clutches was vital to the success of the Y.102A and Y.111A machinery installations and had been most gratifying.

The authors were glad to know that Mr. James had found that the problems of dirt in gearcases could be overcome by the methods they had advocated. Mr. James had mentioned difficulties experienced with thin wall steel shell bearings, arising from variations in thickness. The authors had not experienced similar difficulties in main propulsion gear sets but in these sets the bearing centre distances were, of course, much greater than those referred to by Mr. James.

The authors regretted that they had no experience of multi-lobed or hydrostatic bearings in gearing applications. The conventional, circular journal bearing had not failed under shock loading and, in general, was able to maintain satisfactory gear alignment. The possibilities of maintaining better alignment under all loads and thus perhaps, reducing noise levels by means of restricted clearance bearings were attractive and investigations were needed.

In reply to Mr. Archer the authors agreed that errors produced during shaving of Y.100 main wheels might well have been partly responsible for pitting and this view was, to some extent, supported by the excellent performance of the four main wheels which had been lapped but not shaved. They were unaware of any noise measurements being made to show possible differences between port and starboard boxes and thanked Mr. Archer for mentioning this. As stated in their paper, the starboard gear sets ran with leading aspices whereas, in the port sets, the apices trailed.

It was difficult to obtain precise information on the times steamed at various powers by H.M. Ships but the figures quoted by Mr. Nicholson were similar to those recorded in *Whitby* Class vessels.

Mr. Archer had remarked on failures of hardened and ground gears fitted in merchant ships. The authors had no information on these failures but presumed that the gears had been manufactured on the Continent. It was agreed that the only final test of any naval propulsion gear would be prolonged operational experience at full power at sea, as in wartime. Nevertheless the authors believed that all the evidence obtained so far by A.V.G.R.A. and by the Royal Canadian Navy, indicated that current naval gear designs were conservatively loaded and that failure under wartime conditions was most unlikely.

The tooth loadings suggested for the En26/En9 material combination were applicable to gears working on a similar life cycle to the *Whitby* Class gears. For gears required to work, for most of their lives, at or near full power, some reduction in loading would be made.

The authors agreed that inspection of centre drive pinion bores was not easy. The methods adopted, so far, included ultrasonic examination of the solid forging followed by visual examination with an intrascope after machining. Eight pinions had been made in this way and one forging had been rejected after ultrasonic examination.

Mr. Archer had suggested stiffening the H.P. side of the Y.E.A.D.1 gearbox in order to reduce the large upward deflexions. The authors thought that this was necessary, for in addition to any questions of gear alignment, oil leakage had been experienced at full power, between the flanges of the top and bottom halves of the gearcase near to the forward main wheel bearing.

Throughout the trials the Y.E.A.D.1 gearcase was supported under the three areas shown in Fig. 16, i.e. there were no chocks under the transverse frame at the forward end. Provision had been made to fit chocks had these been found necessary and, in a ship installation, the authors believed that shock requirements would have necessitated fitting some additional support.

With regard to Mr. Archer's written contribution, the authors were pleased to state that no trouble had been experienced with high speed gearing or turbine thrust bearings which could be attributed to the lubricating oil or to the material of the thrust collar. Nevertheless, the troubles experienced in the Merchant Marine were of great interest, particularly so in view of the possible use of En40b steels for nitrided single helical gears in future designs, with Michell type thrust bearings.

Commander Green had asked why the correction on the aft helix ceased 4in. from the gap. The maximum total deflexion (approximately 0·0013in.) occurred at the aft end of the aft helix (the secondary pinion was driven from the aft end) and the ideal helix corrections would form a mirror image of the total deflexion curve. In the original Y.100 pinions the ideal corrections were approached, fairly closely, by the linear corrections shown in Fig. 3, which lie partly above and partly below the origin O. The corrections were applied by shaving, metal being removed from the outer portions of each helix, the relative shift of the uncorrected portions being of no consequence because of the ability of a double helical gear to adjust its axial position until both helices were carrying practically equal loads. As shown in Fig. 3, the correction applied to the aft helix resulted in a high spot at the point 4in. from the gap and during the initial trials as described in the paper, this corresponded with pitting found on the main wheel and the outboard pinion. New pinions were subsequently fitted, in which the correction on the aft helix was extended to a point 2in. from the gap, as shown in Fig. 5, thus removing the high spot.

Practical helix corrections were, inevitably, compromises and care in design and application was essential if overloading of uncorrected portions of the helices at reduced powers was to be avoided.

The improved performance of the Y.100 Mark II gears fitted in the *Rothesay* and *Leander* Classes must be attributed to the use of carburized, hardened and ground pinions rather than to the fact that the helix corrections had been applied by grinding. Swiss gear grinding machines, of the generating type, were used.

With regard to the future, the authors could assure Commander Green that both carburized and induction hardened gears would certainly require grinding. With nitrided gears, the distortion resulting from hardening was small and many large hobbed and shaved nitrided gears, of Continental manufacture had operated successfully without any post hardening finishing treatment such as grinding.

Dr. Davis and Mr. Smith had commented on the importance of unground nitrided gears and the authors wished only to add that A.V.G.R.A. was now actively engaged in the further development of such gears.

The need to limit the length of their paper had resulted in the authors failure to mention the two other S.S.S. clutches fitted in the Y.102A and Y.111A machinery installations and they thanked Mr. Whyatt for supplying this additional information. Mr. Whyatt had also presented a convincing case for the preparation, at a sufficiently early stage, of full size mock-ups in order that erection and maintenance procedures could be planned and difficulties eliminated before the production gear sets were completed. With complex designs, such as Y.102A and Y.111A, the authors entirely agreed that the use of full size mock-ups would be extremely valuable and should pay for themselves in reduced maintenance costs.

Commander Pratt had suggested that means be provided to verify the absolute alignment, in space, of the gearing elements. The authors regretted that they were not aware of suitable methods of doing this and considered that the most reliable method available for checking alignment was on examination of the actual meshing of the gears. Provided it was correctly applied, Talbot Blue lacquer was suitable for this purpose and it was Admiralty policy to check alignment by this method after the initial basin trial and during and after sea trials. Further checks were made periodically throughout the life of a ship.

As stated in the paper, the Y.102A and Y.111A gearing bearings were fitted with thermocouples. Thermocouples were also fitted in the Y.100 Mark I design but proved to be fragile and unreliable. The new design of thermocouple fitted in Y.102A and Y.111A was more robust and, so far, had given satisfactory service. Thermocouple recording instruments had also been unreliable in the earlier ships but a new design fitted in Y.102A and Y.111A was giving very much better service. At present, monitoring of bearing temperatures must be carried out by hand and the large number of bearings fitted in the Y.102A and Y.111A designs caused this to be a lengthy operation. It was, therefore, intended to develop an automatic instrument which would rapidly and continuously scan all bearings and would be fitted with high temperature alarms which could be set for each bearing or possibly, each group of similar bearings.

In addition to thermocouples, mercury-in-steel thermometers had been used but had also been unreliable. This type of thermometer suffered from an appreciable time lag and also was unsuitable for fitting in remote machinery control rooms.

No provision had been made for measuring the axial displacements of gearing elements and the conventional Michell thrust bearings had given very little trouble.

Notwithstanding the traditional uses of the K factor to describe marine gear tooth loading, the authors wished to assure Dr. Merritt that the importance of fatigue failure under bending stress was fully appreciated. Maximum permissible bending stresses, for a number of gear materials, were now specified and had been based on the results of tests on full size and smaller gears and a very large number of Wohler and Schenck fatigue tests, the latter being carried out on gear type test pieces. This work would, of course, continue and it was regretted that shortage of space had prevented the authors including a review of the results obtained so far.

With regard to the En36A carburized and ground secondary pinion (2D.P.) which failed after $21 \cdot 7 \times 10^6$ wheel cycles at 904K, subsequent examination had shown a pronounced banded appearance, consisting of a series of "flats" running along the length of each tooth flank. These had been caused during tooth grinding and were eliminated, in subsequent pinions, by modifications to the grinding machine. Tooth breakage had occurred in fatigue, initiated in a line of macroscopic pits near the pitch line of the ahead flank at the junction of two "flats". Metallurgical examination had shown that carburizing and heat treatment had been satisfactory. The average hardness of the tooth flanks was 652 V.P.N. and effective case depth (to a minimum of 500 V.P.N.) was 0·068in. The root fillets were not ground or shot peened.

With regard to Dr. Merritt's final remarks, the authors took comfort in the fact that one of the leading firms in A.V.G.R.A. designed and manufactured not only marine gears but also very large numbers of gears of other types, covering the entire automotive field.

Dr. Brown had made some interesting observations on the effects of gearcase distortion. The authors believed that, in the present state of the gear designer's art, considerable gearcase distortion was inevitable in all but the largest warships, particularly during high speed manœuvring and rough seas. They agreed therefore, that the choice of gear materials and nominal tooth loadings must be made with such conditions in mind. Nevertheless they believed that fuller investigation of side effects, as mentioned by Dr. Merritt and, in particular, of the art of gearcase design would permit the nominal tooth bending stresses to be raised.

To minimize the effects of thermal expansion and of hull distortion it was the practice to site the first plummer block as far as possible from the gearcase and to align the shafting so that the static loads on main gear wheel bearings were approximately equal and downwards.

Rear-Admiral Given had made valuable suggestions for a new approach to gearcase design and would be interested to know that a design for a 1,000 kW turbo-alternator gearcase had already been prepared, based on these principles and that it was intended to manufacture and test this gearcase.

Although the flexible couplings fitted between the Y.100 turbines and gearing had barrelled, nitrided teeth, they were unable to withstand angular misalignments greater than ± 2 min. (approximately). In general these couplings had given satisfactory service and in the few cases of abnormal fretting which had been experienced, misalignment had invariably been discovered. It had not been possible to ascertain whether such misalignment dated from the original erection of the machinery or had arisen during subsequent service. Relative movement of turbines and gearing could also arise from thermal expansions. The authors therefore considered that provision for angular misalignment of flexible couplings was still necessary. They regretted that they had no information on the transient thrusts in double helical secondary pinions when manœuvring.

It was gratifying to learn that the work done by A.V.G.R.A. had encouraged Commander Joughin's company to undertake the production of nitrided gears and future naval auxiliary machinery would benefit from this decision.

Commander Joughin had asked for suggestions regarding ways and means of studying the maintenance of gear alignment but, as already stated in reply to Commander Pratt, the authors were unaware of suitable methods. It was, however, intended to instrument a Y.100 gear set in one of the frigates in an attempt to ascertain the running positions of the journals under various conditions of operation. The main gear wheel journals of H.M.S. *Ashanti* and a secondary pinion journal in H.M.S. *Devonshire* had already been instrumented in this way and further trials were planned.

The authors fully agreed with Mr. Newman that not enough was known about the actual loads on gearing in ships. Much investigation and development would be required to reduce the gap between the nominal loads which could be safely carried by gears when tested ashore and those which could be permitted in a ship. This would be costly and difficult, particularly as operational ships would be involved but the authors hoped that some progress could be made.

Mr. Newman would be glad to know that an high speed journal bearing test rig had recently been installed at the Admiralty Engineering Laboratory and that it was intended to investigate practical bearing design, with particular reference to size, shape and positions of oil inlets.

Flushing was usually carried out with an oil of similar viscosity to that used in service.

The authors' recommendation that rubber-soled shoes should be worn applied only during the erection of gearing at the manufacturer's works. They agreed that rubber soles were dangerous in a ship's engine room and, in fact, it was a disciplinary offence for R.N. personnel to wear such shoes.

Mr. Newman had suggested that further work was necessary to reduce the losses which occurred in high speed thrust bearings. The authors agreed and, in fact, a thrust bearing investigation had been sponsored by the Admiralty a few years ago but for various reasons had been terminated before any useful results were obtained.

Shaft driven lubricating oil pumps were fitted in the Y.102A and Y.111A installations, in addition to electric motor driven pumps (Good and Dunlop[13]). As Mr. Newman was well aware, lubricating oil systems and pumps were a fruitful field for prolonged discussion and the authors wished to add only that shaft driven pumps were also being considered for future designs.

In conclusion, the authors wished to thank all those who had contributed to this most stimulating discussion and to assure them that their contributions would be of great value to the Admiralty when considering future naval gearing policy.

INSTITUTE ACTIVITIES

Minutes of Proceedings of the Ordinary Meeting Held at The Memorial Building on Tuesday, 11th December 1962

An Ordinary Meeting was held by the Institute on Tuesday, 11th December 1962, when a paper entitled "Progress and Development in Naval Propulsion Gears 1946-1962" by Commander P. D. V. Weaving, R.N. (Member) and W. H. Sampson (Member), was presented by the authors and discussed.

Vice-Admiral Sir Frank Mason, K.C.B. (Chairman of Council) was in the Chair and 59 members and guests were present.

In the discussion which followed fourteen speakers took part.

The Chairman proposed a vote of thanks to the authors which received enthusiastic response.

The meeting ended at 7.40 p.m.

Section Meetings

North East Coast

The Annual Dinner and Dance of the Section was held on Friday, 1st February 1963, at the Royal Station Hotel, Newcastle upon Tyne.

The Section was honoured by the presence of the Senior Vice-President, Mr. J. Calderwood, M.Sc., and Mrs. Calderwood, Vice-Admiral Sir Frank Mason, K.C.B. (Chairman of Council), and Lady Mason, The Right Worshipful The Lord Mayor of Newcastle upon Tyne, Alderman George Jacobson, and Mrs. Jacobson, The Lady Mayoress, Mr. G. R. H. Towers, J.P., President of the North East Coast Institution of Engineers and Shipbuilders, Commander R. F. A. Whately, M.A., Secretary of the North East Coast Institution of Engineers and Shipbuilders, and Mrs. Whately, Mr. J. Stuart Robinson, M.A. (Secretary) and Mrs. Robinson, together with the support of North East Coast talent and a delightful gathering of ladies.

The function was lively and successful and appeared to be thoroughly enjoyed by all.

Scottish

The Ninth Annual Dinner was held on Friday, 15th February 1963, at the Central Hotel, Glasgow, at 7.15 p.m.

Mr. R. Beattie (Chairman of the Section) presided and some 400 members and guests were present.

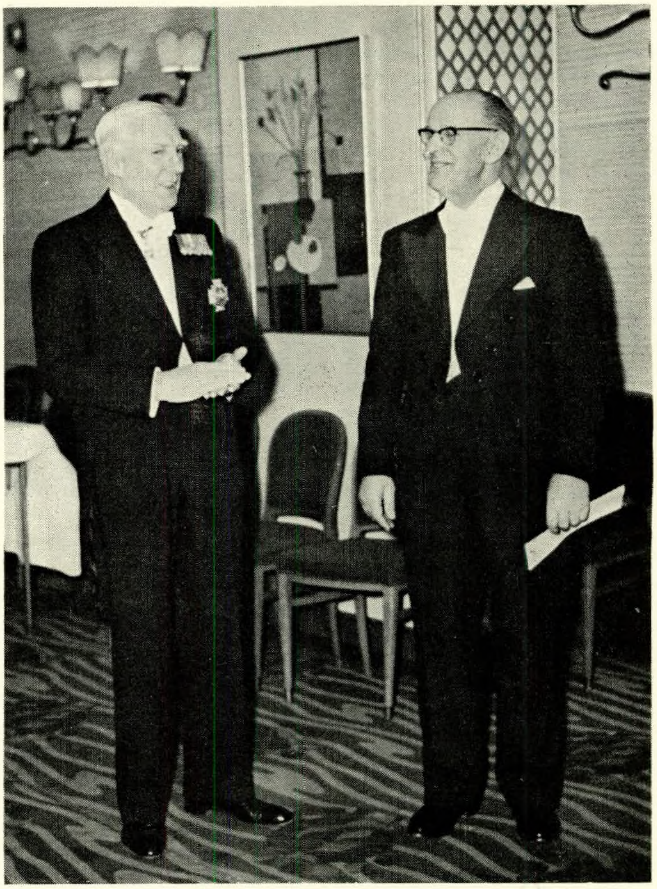
At a reception prior to the Dinner, guests were formally received by Vice-Admiral Sir Frank Mason, K.C.B. (Chairman of Council) and the Chairman, Mr. R. Beattie.

The toast to "The Institute of Marine Engineers in Scotland" was proposed by Mr. I. M. Stewart, B.Sc., President of the Institution of Engineers and Shipbuilders in Scotland, this was replied to by Vice-Admiral Sir Frank Mason, K.C.B.

The Chairman proposed the toast "Our Guests" and the reply was given by Mr. W. Nicholson, President of the Glasgow and Clyde Shipowners Association.

The top table party consisted of representatives of local shipowners, shipbuilders, the Ministry of Transport, educational and sister institutions, Missions to Seamen and office bearers of the Section. The Admiralty was represented by Captain G. D. Pound, D.S.C., R.N., Captain in Charge, Clyde, and the United States Navy by Captain D. N. Syverson, Commander of the Polaris Depot ship *Hunley*. London Headquarters was represented by Past Chairman of Council Mr. W. R. Harvey, O.B.E. (Member of Council) and Mr. J. Stuart Robinson, M.A. (Secretary).

The Dinner was followed by a conversazione until about midnight.



At the Ninth Annual Dinner of the Scottish Section. Vice-Admiral Sir Frank Mason, K.C.B. (Chairman of Council) (left) and Mr. R. Beattie (Chairman of the Section) at the reception before the Dinner

South Wales

Cardiff

A general meeting in conjunction with the Institute of Welding, was held on Monday, 4th February 1963, at the South Wales Institute of Engineers, Park Place, Cardiff, at 7.0 p.m.

Mr. R. A. Simpson (Chairman of the Section) was in the

At the Annual Dinner of the Scottish Section. From left to right: Mr. J. W. Bull (Immediate Past Chairman of the Section), Capt. D. N. Syverson, Commander U.S.S. Hunley, *Mr. J. Stuart Robinson, M.A. (Secretary), Capt. G. D. Pound, D.S.C., R.N., Captain in Charge, Clyde, Mr. W. R. Harvey, O.B.E. (Member of Council), The Rev. G. R. Connock, M.A., Missions to Seamen in Scotland and Mr. W. O. Gardiner (Corresponding Member, Dundee)*

Chair and welcomed the thirty members and guests who were present and thanked them for attending, despite the heavy snow storm. He also made reference to four members who had travelled from Swansea.

The Chairman introduced Mr. A. S. Minton (Associate) who, assisted by his colleagues Messrs. M. Bray and S. McWade, gave a most interesting and illuminating lecture entitled "Non-destructive Testing". The talk was illustrated by slides and numerous instruments for this form of testing were displayed and explained.

A vote of thanks to Mr. Minton and his colleagues was proposed by Mr. F. R. Hartley (Member) and the meeting terminated following the proposal of a vote of thanks to the Chairman by Mr. T. G. Bullen (Member).

Swansea

A very successful meeting was held on Tuesday, 26th February 1963 in the Lecture Theatre of the Showrooms of the Gas Company, Swansea, at 7.0 p.m.

Mr. David Skae (Vice-President), presided at the meeting which was attended by 80 members and guests.

Dr. A. J. Johnson, B.Sc.(Eng.) and Mr. W. McClimont, B.Sc. (Member) gave their lecture entitled "Machinery Induced Vibrations" which was well illustrated by slides and colour film.

Following the discussion at the conclusion of the paper, a vote of thanks to the authors was proposed by Mr. J. Wormald, B.Sc. (Member) and a vote of thanks to the Chairman was proposed by Mr. H. G. Wickett, M.B.E.

West of England

A general meeting of the Section was held on Monday, 11th February 1963, at Smith's Assembly Rooms, Westgate Buildings, Bath, at 7.30 p.m.

Captain R. G. Raper, R.N. (Chairman of the Section) was in the Chair and the audience, which included Mr. D. W. Gelling (Local Vice-President), numbered twenty-eight.

A paper entitled "Progress and Development in Naval Propulsion Gears, 1946-1962" by Commander P. D. V. Weaving, R.N. (Member) and W. H. Sampson (Member), was presented by Mr. Sampson in a most instructive and interesting manner.

The paper which was illustrated by slides, dealt with all the major gearbox designs used in the Royal Navy from 1946 to 1962. The author explained how these gearboxes had behaved in actual service pointing out some of the failures which occurred and the modifications introduced to obviate further troubles.

The speaker made reference to the high tooth loading factor when transmitting such high powers through these compact gearboxes and gave an excellent account of the more modern techniques of gear cutting, the various processes used for teeth hardness treatment, a new arrangement for aligning pinion shafts using adjustable steel bearing housings; he then spoke about lubricating oils used and the necessity for cleanliness in the erection and initial running in of these gears.

The lecture ended with a brief description of the control system for the hydraulic clutches of a combined gas and steam turbine power unit.

In the discussion which followed ten speakers took part and many interesting points of view were expressed all being ably answered by Mr. Sampson.

A vote of thanks to the speaker was given by the Chairman and the meeting ended at 9.40 p.m.

Election of Members.

Elected on the 11th March 1963

MEMBERS

Angus Beck, B.Sc. (Belfast)
Matthew Frank Bennett
Ernest Frederick Hambly Cox
John Davison
Walter Henry Duff
John Garden Gilmour
Trulls Gleditsch
Jack Hodgson
James Henry Hutchinson
James Felix Leach
Walter James Phillips
William Page Shanks

ASSOCIATE MEMBERS

Alexandre Alderson
Frederick William Austin
Colin Nuttall Barlow
Stanley John Bettesworth
George Naismith Buchanan
Herbert Laurence Chappell
Richard Arthur Armytage Dean, Lieut., R.N.
Kenneth William Ferns
Gordon Wilson Graham
Alan Harrogate
Peter Harrold
David Jones
Denis Oswald King
Ian Colquhoun Little
James McAlister
Alexander Imrie Meikle
Lawrence Murray
David Albert Nicholson
Michael John Patton
Joseph Thomson Rankin
Brian William Rees
Thomas Saunders
Ernest Seddon, Eng. Lieut., R.A.N.
Edward Frederick Ridley Shaw
Stewart Mitchell Sherman
Alexander Craig Simpson
Robert Stuart
Thomas Tarpey
Jean Ulrich Thoma, Dr.
Neil Barry Thomas, Lieut., R.N.
Kenneth Walter Thorp
Shrikrishna Keshaw Vijayakar
Glyn G. Weaver
Michael John Antony Webber, Lieut., R.N.
George E. Wilson
Peter Wyatt

ASSOCIATES

Harold Bradley Adams
Bashier Ahmed, Sub. Lieut., P.N.
Stanley George Bloomfield
Robert William Catton
David Cooper
Leung Hu Kwong
Maurice L. Lanouette
Edward E. Olsen
James William Wealleans, B.Sc.(Man.)

GRADUATES

Brian Nicholas Cowell, B.Sc.(Durham)
Derek James Hay
John George Howard
K. V. Kesu
John D'Arcy Langley
Brian Richard Melling
Alan Rutherford
Maung Than Shwe
Moti Lal Tandon

STUDENTS

Kee Ah Bah
Godfrey Uzoka Billy
Teo See Chay
James Chesney
Wong Seng Chow
Gordon Morrice Darroch
Paul Dearnaley
Andrew Allen Dobbs
Tham Yeng Fai
Leong Ah Fatt
Paul Golden
Ahmed Abdullah Harharah
Toh Siong Hoe
Lee Cheng Hong
James Houston
Vincent Wong Bheet Huan
Wee Eng Huat
Joseph Heng Yam Hwee
Foo Ah Kit
Moey Hong Kong
William Briggs Lauchlan
R. Lisle Lysaght
James George MacDonald
Mohamed Omar Mian
Roderick Paterson Morrison
Charles John Revill
Dennis Saw
James Brown Smillie
Lee Gek Soo
Fong Kee Sung
Robert Stuart Thake
Fong Swee Theng
Vans Clark Thomson
Winston Tun
Veerappan Vellaiappan
Khan Hui Ven
Lui Wai Yuen
Ivan G. R. Zobel

PROBATIONER STUDENTS

John Everitt
Stuart Edwin Guthrie
Brian Graham Ingleson
Edward Harry Porter
Frank Raymond Wickman
David Anthony Windsor

TRANSFERRED FROM ASSOCIATE MEMBER TO MEMBER

Charles Francis De Lima
Malcolm Peter Evans
Thomas Leigh Taylor, Cdr., R.N.Z.N.

TRANSFERRED FROM ASSOCIATE TO MEMBER

Kenneth Ernest Sayer

TRANSFERRED FROM GRADUATE TO ASSOCIATE MEMBER

Donald William Albert
Eric Cyril Avery
Dennis Beattie Hayton
Colin Scorer, B.Sc.
Brian Stanley Thompson, B.Sc.
Marnoch Shearer Thomson
Joseph Rory Marshall Wilson

TRANSFERRED FROM STUDENT TO ASSOCIATE MEMBER

Brian Sidney Brown
Gerald Foster
David Westby Hobbis
Roy Mason
Raymond Henry Cunningham Philpot

TRANSFERRED FROM STUDENT TO GRADUATE

Michael A. Baroutakis, B.Sc.(Durham), R.H.N.
William Joseph Brickley
Daniel Greenhalgh Briggs
Derek Ion Holden
Michael David Lacey
K. Ravindran
Terance James Roberts
Colin Francis Rogers
Derek Totton

TRANSFERRED FROM PROBATIONER STUDENT TO ASSOCIATE MEMBER

Bernard Leslie Howe
Peter Gerald Simmons

TRANSFERRED FROM PROBATIONER STUDENT TO ASSOCIATE

Keith Dalton Seiler

TRANSFERRED FROM PROBATIONER STUDENT TO GRADUATE

Peter Edwards
Fergus Toman
Graham Jeffery Wallace

TRANSFERRED FROM PROBATIONER STUDENT TO STUDENT

Edward Green
John David Ireland
William Ridley

OBITUARY

John Nicol

An appreciation by R. K. Craig, C.B.E.
(Honorary Vice-President)

The passing of Mr. J. Nicol on the 29th December 1962, in the Guild House, Littlehampton, at the age of 88 years, removed one of the best beloved names in the industry of marine engineers, that belonged to the pre-war and war era. "John" as he was familiarly known to his many friends in the City and industry, was very definitely a man of great experience and of few words, but who carried out in practice his twelve maxims to the end of his days. These maxims are well worth recording and remembering:

The value of time; the success of perseverance; the pleasure of working; the dignity of simplicity; the worth of character; the power of kindness; the influence of example; the obligation

of duty; the wisdom of economy; the virtue of patience; the improvement of talent; the joy of originating.

Born in 1874 in Aberdeen, he joined the Aberdeen Line as Fourth Engineer on 4th January 1903, and sailed on s.s. *Nineveh,* obtaining his Chief Engineers' Certificate in August 1907. He served in this vessel and s.s. *Moravian* until 1908. He then spent three years with Shaw Savill Company as refrigerating and second engineer until June 1911, when he rejoined the Aberdeen White Star Line as second engineer on s.s. *Themistocles.* He was promoted to Chief Engineer in April 1912, and appointed to s.s. *Marathon* in October 1912, where he remained for four voyages, and then transferred to s.s. *Themistocles* in August 1914, serving as Chief Engineer throughout the 1914-18 conflict, until August 1918, when he came ashore as assistant to Mr. Mannell, Superintendent Engineer, until 1920, when he was appointed Superintendent Engineer.

On the amalgamation of Aberdeen Line and Shaw Savill Company in 1927, Mr. Nicol became assistant to Mr. Adams and succeeded to Chief Engineer Superintendent in 1934, which position he kept until his retirement in 1945.

Mr. Nicol joined the Institute of Marine Engineers on 23rd April 1914, becoming a Member of Council and was co-opted Chairman for a period during his term of office. He resigned from the Institute in 1947.

His retirement to Worthing in 1945 was enjoyed to the full by his good lady and self, for the first six or seven years, when his efforts on the bowling green were much appreciated by the local clubs. Unfortunately, his wife predeceased him by ten years and for the past five years, Mr. Nicol has lived at the Marine Engineers' Guild Home at Littlehampton, where he was able to keep in touch with old shore and seagoing friends.

In extending sympathy to his relatives and many friends, perhaps the quotation from Ibsen, which so aptly applies, would be found appropriate. "So to conduct one's life as to realize oneself". This seems to be the highest attainment possible to a human being. It is the task of one and all of us, but most of us bungle it—John did not.

Donald Eadie (Member 13623) who had been a member of the Institute since 14th January 1952, died after a brief illness, on 25th June 1962, at the age of 79 years. He spent four years from 1898-1902 as an apprentice with the London and South Western Railway, meanwhile attending technical college in Exeter. In February 1902 he commenced eighteen months' sea service as fourth and third engineer in s.s. *Archtor,* after which he joined Palmer's Shipbuilding Co. Ltd. as an apprentice for a little over a year. During the latter period he also received engineering training at the Borough Polytechnic in Jarrow-on-Tyne.

He returned to the sea in 1905, to serve as third and second engineer in s.s. *Rodney* and, obtaining his First Class Steam Certificate in 1907, became second engineer in s.s. *Harpenden* in the same year. He remained a seagoing engineer for the next three years, then accepted an appointment with C. H. Campbell, Dredge and Public Works Contractor. Mr. Eadie continued in this employment until in 1915, he joined the Royal Navy and served throughout the remainder of the First World War as Engineer Lieutenant and Engineer Lieutenant-Commander. He also held the rank of Engineer Lieutenant-Commander in the Special Reserve.

After being demobilized from the Royal Navy he rejoined the firm of C. H. Campbell until he went to Australia in 1927 to become Dredge Engineer and Dockyard Manager to the Melbourne Harbour Trust, a position he held for the next eleven years. He became Engineer Superintendent to Huddart Parker Ltd. in 1938.

Mr. Eadie had been an Associate Member of the Institution of Mechanical Engineers in addition to his membership of this Institute. He died, whilst in the United Kingdom, at Guildford, Surrey.

JOHN CHARLES CODY (Associate Member 16414) was born on 22nd January 1919. From the age of fourteen he served a five year apprenticeship in fitting and turning with the Broken Hill Pty. Co. Ltd. and, whilst indentured, attended the Newcastle (N.S.W.) Technical College, gaining a Certificate of Trade Competency in fitting and machining. Later in his apprenticeship he also gained a Diploma in Mechanical Engineering from the Sydney Technical College.

Mr. Cody began his professional career, in 1938, as an engineer draughtsman and was employed in that capacity first by the company to which he had been indentured, then by Stewarts and Lloyds Aust. Pty. Ltd. and finally by the State Dockyard, Newcastle, N.S.W.

In 1944 he was appointed, by the latter organization, as assistant to the Shipyard Superintendent and two years later became Assistant Ship Repair Superintendent. He went to sea in 1948 and served for four years' as a marine engineer, at the end of which time he had gained a First Class Steam Certificate.

He left the sea to accept appointments as Resident Engineer (South Australia) for Simon-Carves (Aust.) Pty. Ltd. and also as Engineer Representative for their associate company, J. R. Pillars Pty. Ltd. Whilst holding these appointments, he was responsible to the parent company's Chief Engineer for the control by the two firms of all construction work in South Australia.

Mr. Cody was elected an Associate Member of this Institute on 21st June 1955; he was also an Associate Member of the Institution of Engineers, Australia.

GEORGE NELSON (Member 7987) died on 27th July 1962. Born on 13th June 1877, he received his early education at a private school and later at Blyth Technical School. In 1894 he began a five year apprenticeship with the Blyth Foundry and Engineering Co. Ltd. where he gained experience in the pattern shop, foundry, machine shop and also in marine repair work. In addition he studied machine drawing, applied mechanics, mathematics and the principles of steam for seven years, first at Rutherford College and later at Armstrong College, Newcastle upon Tyne.

In 1900 he entered the drawing office of Hawthorn Leslie and Co. Ltd. and, after three years, transferred to the Palmer Shipbuilding and Engineering Co. Ltd., again to a position in the drawing office. Mr. Nelson left Palmer's two years later to return to the Blyth Foundry and Engineering Co. as general manager, a position he held until 1911, when he became a partner in the firm of Niven and Nelson. Whilst sharing the responsibility for carrying on the business of the latter firm, he also acted as technical and sales representative for various firms concerned with marine engineering and at different times was manager, in the Newcastle offices of Hayward Taylor and Co. Ltd. and Broadbent and Son. In 1935, he became Chairman and Joint Managing Director of Niven, Nelson and Matthews Ltd.

Mr. Nelson was a Past President of the Graduate Section of the North East Coast Institution of Engineers and Shipbuilders. He was elected an Associate of this Institute on 7th October 1935 and transferred to full membership on 8th February 1943.

THOMAS CHURCHILL POYNTON (Probationer Student 18385) was born on 20th July 1939. He received his early education at the Blackburn Technical High School, where he gained his G.C.E. In 1956 he became apprenticed to Shell Tankers Ltd. and also commenced studies at the Bolton Technical College, taking the Ordinary National Diploma Course in marine engineering.

Mr. Poynton died as the result of an explosion aboard the s.s. *Verena,* whilst serving at sea off the coast of India.

He was elected a Probationer Student on 12th December 1956.

LIEUTENANT-COMMANDER THOMAS BERTIE ROBINSON, C.D., R.C.N. (Member 15214) died on 2nd December 1962, at the age of 63 years. He had been taken ill almost immediately after the basin trials of the destroyer escort *Saskatchewan* in which he had been superintending the machinery installation since July 1961.

Lieutenant-Commander Robinson was born in Yorkshire and from 1915 to 1921 served an apprenticeship with Messrs. Miller and Dennis, engine and boiler mounting makers of Bradford. In the early twenties he went to Canada and spent three years as a machinist, first with the Ogden Locomotive Works then with the Precision Machine Shop, both of Calgary, Alberta. In 1925 he joined the Royal Canadian Navy as an engine room artificer, fourth class, serving in several H.M.C. ships. After his promotion to Chief E.R.A. he served in H.M.C.S. *Vancouver* and, in 1937, commissioned the original H.M.C.S. *Fraser*. Shortly afterwards he began a long association with Yarrows Ltd. when he "stood by" during the construction of H.M.C.S. *Nootka,* in which vessel he served as Engineer-in-Charge.

Early in the Second World War he was promoted to Warrant Engineer and in 1941 to Commissioned Engineer. He returned to the United Kingdom to commission H.M.C.S. *Sioux* and served in that vessel as Lieutenant (E). Three years later he was overseer in charge of the conversion of H.M.C.S. *Sioux* and *Algonquin* at Yarrows Ltd. in Esquimalt. In 1949 he joined the staff of the Manager of Engineering Department at H.M.C. Dockyard, Esquimalt and in 1952 became Deputy Manager of Engineering Department, when he was promoted Lieutenant-Commander (E). He retired from the R.C.N. in 1958 and later returned to Yarrows with which company he remained in association until he was taken ill in August 1962.

Lieutenant-Commander Robinson was elected a Member of the Institute on 4th October 1954 and at the time of his death was Honorary Treasurer of the Vancouver Island Section.

He leaves a widow.

GEORGE STANLEY RONALD (Member 17959) who was born on the 11th December 1919, died at the age of 43 years, after a short illness. He was the eldest son of the late Captain G. S. Ronald of Manchester Liners Co. Ltd., a recipient of Lloyd's Medal for meritorious service in 1916. Mr. Ronald served his apprenticeship with Morrell Mills Co. Ltd., Manchester, from 1935 until 1940, and started his sea career in 1940 with Manchester Liners Co. Ltd., with whom he remained until 1946. He later transferred to the Canadian Merchant Service from 1946 to 1951, sailing as second and Chief Engineer with March Shipping Co. Ltd. and Western Canada Steamships Co. Ltd.

In 1951, Mr. Ronald joined the Royal Canadian Navy on a Short Service Appointment until 1954. From that year until the time of his death he was the Chief Instructor of the Nova Scotia Marine Engineering School, Halifax, Nova Scotia. He attained the rank of Lieutenant-Commander in the Royal Canadian Navy (Reserve), and was Engineer Officer, H.M.C.S. *Scotian,* being awarded the Canadian Decoration in 1962.

He was also a member of the United Services Institute of Nova Scotia and the Naval Officers' Association. He was elected a Member of the Institute on 29th October 1956 and from 1960 until his passing, was Treasurer of the Atlantic Section of the Canadian Division.

Mr. Ronald leaves a widow, Dorothy, and two sons.

PATRICK SHEA (Member 19858), a Member of the Institute since 10th February 1958, died on 7th February 1963, at the age of 62 years.

In 1916 he joined Worthington-Simpson Ltd., of Newark-on-Trent, as an apprentice and on completion of his indentured service became Resident Engineer for the company, engaged on contracts carried out in the United Kingdom and India. After eleven years he became Chief Trials Engineer to the company in 1932 and was subsequently appointed Assistant Marine Manager in 1945. He held this appointment until the time of his death.

Mr. Shea leaves a widow.

JOSEPH FRANK STRATTA (Member 11239) died suddenly at his home in Langley, Buckinghamshire, on 9th October 1962. He had been ill for some months, but had apparently made a recovery, so that his death came as a great shock to his family.

Mr. Stratta was born on 6th October 1906 and served his apprenticeship, at the Southern Railway's Brighton Locomotive Works from 1922-1928. During this period he attended evening classes at Brighton Municipal Technical College. After completing his apprenticeship he went to sea for eleven years, his last seagoing appointment being as third engineer in s.s. *British Ambassador,* a vessel owned by British Tankers Ltd. His health forced him to relinquish his seagoing career and he accepted a shore appointment on the technical engineering staff of the India Rubber Gutta Percha and Telegraph Works Co. Ltd. Part of his time with that company was spent in charge of installation work at one of their factories.

In 1946 he became works engineer with D. Young and Co. Ltd. and four years later took up a similar appointment with Imperial Chemical Industries Ltd., in their Paints Division at Slough. He held the latter appointment until the time of his death.

Mr. Stratta was elected a Member of the Institute on 4th March 1947. He leaves a widow.

JOHN MATHESON WATSON (Member 9552) was born on 11th March 1903. From 1918-1924 he served an apprenticeship with Scott's Shipbuilding and Engineering Co. Ltd. and remained with the company, on completion of his indentured service, as junior engine draughtsman. He subsequently became senior engine draughtsman with Barclay Curle and Co. Ltd. and later was employed in a similar capacity by John G. Kincaid and Co. Ltd.

In 1932 he was appointed deputy chief draughtsman to British Auxiliaries Ltd. (now British Polar Engines Ltd.) and four years later became personal assistant to the general manager. In 1937 he became chief draughtsman and deputy to the general manager and in 1959 was appointed managing director of the company.

Mr. Watson, who died on 11th February 1963, was elected a Member of this Institute on 2nd March 1943. He leaves a widow.