# Gearing Trends in the Royal Navy

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## SYNOPSIS

*This paper looks at the design features characteristic of a gearbox which might be installed in a modern gas turbine powered warship. Aspects covered include flexible couplings, clutches and brakes; reversing arrangements; gear design and materials; bearings and lubrication and instrumentation, with discussion of recent problem areas where appropiate. Outlined within each topic are the current areas of interest for future development. These are principally related to: (a) the reduction of gear noise, with increasingly tighter targets being set; (b) condition monitoring as a means of both detecting impending failure and minimising unnecessary maintenance activity; and (c) more compact reversing gearboxes as an alternative to the controllable-pitch propeller (CPP), which has been the standard fit in frigate/destroyer size warships operated by most navies.*

## INTRODUCTION

There seems little doubt that the new BS and ISO standards for gearing will prove to be most important contributions to both performance and reliability. The case in support of these standards has been set out in Ref. 1 and 2 and it is not the intention of this paper to take the justification further. However, suffice it to say that it is MoD(N)'s intention to adopt BS 436 for future gearbox designs. Notwithstanding the need for an improved gear rating procedure, it has to be said that the majority of RN main propulsion gearbox failures have no connection whatever with gear tooth design and are of a much more basic engineering nature. Previous papers have covered the development of RN gearboxes from the post-war *Daring* class to the present *Type 21*, *22* and *42* classes of frigates and destroyers.[3,4] This paper is therefore aimed at reviewing the gearbox as a system of components, looking at design practices, naval experience and interests for the future, albeit in fairly general terms.

## GENERAL GEARBOX REQUIREMENTS

The general requirements for naval gearboxes tend to be quite straightforward, *ie* to transmit the stated power from the prime mover(s) to the propeller shaft with maximum efficiency whilst taking up the minimum amount of space and being as quiet as possible. Unfortunately this objective is more easily stated than achieved and it is the designer's responsibility to ensure that inevitable compromises result in the optimum overall configuration.

So far as propulsion engines are concerned a review of current ship designs (Table I) clearly shows the modern trend towards combined plants with cruise and boost units, a practice that is likely to continue in future although possibly with cruise power being provided by a diesel rather than a gas turbine for reasons of fuel efficiency.

Increasing pressure to reduce underwater noise is likely to result in the selection of the lowest shaft speed that the hull configuration will permit and, with the assumption that maximum gas turbine speed remains at about 5700 rev/min, will dictate a consequently increased gear ratio (although unlikely to exceed that obtainable in a double reduction gearbox).

Emphasis on structure-borne noise will also place demands upon both designer and manufacturer to produce gears of the highest accuracy with optimum reliefs. This latter task,

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**Table I: Propulsion plant in current ship designs**

| *Ship Class* | *Propulsion Configuration* | *Boost Engine* | | *Cruise Engine* | | *Gearbox Configuration* |
|---|---|---|---|---|---|---|
| | | *Type* | *Power (MW)* | *Type* | *Power (MW)* | |
| Leander | Steam | Turbine | 11 | | | Single input (A) |
| County | COSAG | G6 (2) | 11 Total | Turbine (2) | 11 Total | Four input (B) |
| Bristol | COSAG | Olympus | 11 | Turbine (2) | 11 Total | Three input (B) |
| Type 21 | COGOG | Olympus | 20 | Tyne | 4 | Twin input (A) |
| Type 42 | COGOG | Olympus | 20 | Tyne | 4 | Twin input (A) |
| Type 22 01-06) 08-10) | COGOG | Olympus | 20 | Tyne | 4 | Twin input (A) |
| Type 22 -07 | COGOG | Spey | 12.75 | Tyne | 4 | Twin input (A) |
| Type 22 11-14 | COGAG | Spey | 12.75 | Tyne | 4 | Twin input (A) |
| Type 23 | CODLAG | Spey | 12.75 | Motor | 1.5 | Single input (C) |
| CVS | COGAG | Olympus | 20 | Olympus | 20 | Twi n input (D) |
| S/M | Steam | Turbine | | | | Twin input (E) |

Notes:
A. Double reduction dual articulated locked train. Double helical
B. Double reduction single articulated train. Single helical. Reversing on GT input
C. Double reduction single articulated train. Double helical primary. Single helical secondary
D. Triple reduction single articulated train. Double helical. Reversing
E. Double reduction dual articulated locked train. Single helical

however, will be complicated by the fact that noise characteristics are important at up to 25% power and not the full power condition for which the design is normally optimised.

The debate over the merits of reversing gearboxes *vis-à-vis* controllable-pitch propellers will undoubtly continue, and it is a distinct possibility that a reversing gearbox will be specified for a future ship of frigate/destroyer size provided that the size penalty is not too great.

The actual duty of the gearbox, in terms of ship operating profile, will be largely unchanged, with relatively short periods at high power (say 5% of time or 200 hours per year) and long periods at 25% power and below. That is not to say that the life of a gearbox is easy since design torque (the critical factor) can readily be exceeded during the frequent accelerations and manoeuvres to which it is subjected: in gas turbine ships by up to 50% if the operator does not react to the indication of over torque (where read out is provided). Also of significance is the time spent in coastal water where shallow water effects result in maximum torque being reached well below maximum speed

## FLEXIBLE COUPLINGS

### High-speed couplings

In steam powered warships, typified by the *Leander* class, the turbine and gearbox are solidly mounted in the same compartment, with the advantages that changes in alignment between them are relatively small and that they are close together. For this duty two fine tooth (gear type) couplings connected by a short steel torque tube have proved quite satisfactory provided that an initial good alignment is maintained and lubrication is adequate. (This is not always the case and seizure of couplings has occured as a consequence.)

With the design of the *Type 21* frigates and *Type 42* destroyers a different approach was required, the basic differences being that turbine and gearbox are sited in separate compartments, necessitating a long torque tube (over 2 m), and that the turbine is mounted on rubber chocks (a constant position mounting systems (CPMS) is in fact used in early *Type 42*s). A further factor to be considered is that the Olympus gas turbine has a maximum permitted connected weight which effectively precludes the use of a steel assembly. The coupling selected was basically similar to that used in the *County* class destroyers,[1] comprising flexible membrane packs at turbine and gearbox ends connected by a torque tube. In this case all but the membranes (steel) were manufactured from Hiduminium. The membrane packs were standard between classes as was the torque tube diameter, although the *Type 21* unit was longer.

Despite appropiate vibration analysis of the complete high-speed line (HSL) having been carried out at the design stage, major problems were experienced in achieving stable performance when it came to setting to work of the propulsion plant in the first *Type 21* (HMS *Amazon*), and as a result the coupling was redesigned with a larger torque tube diameter. Subsequent performance was satisfactory with the specified degree of balance (and hence turbine and gearbox input bearing cap vibration levels) being achieved. (Naval specification is for the peak velocity at rotational frequency to be a maximum of 3.8 mm/s.)

No similar problems were experienced in the first *Type 42* (HMS *Sheffield*) and this class retained the smaller diameter torque tube. In retrospect this was probably the wrong decision as later ships of the class suffered considerable problems during set to work by the shipbuilder, making it clear that stability was marginal. Interestingly, however, very few problems have been experienced in service, even after replacement of the coupling at refit.

Later ship designs, the Olympus powered *Type 22* and *CVS* (*Invincible*) classes, use the same basic coupling as the *Type 21* and have been almost completely trouble free, requiring minimum balance adjustment to meet specified vibration levels. (Balance correction is achieved by adjustment of rings built into the torque tube assembly.) The new Spey powered *Type 23* design is based on a different membrane shape with steel torque tube but otherwise it is similar in principle. It should be stated here that MoD places great importance on a full vibration analysis of the complete high-speed line at the design stage, this being made the responsiblity of the gearbox designer. However, even with the latest analytical programs it is accepted that the results produced can only be a pointer towards the actual response of the line.

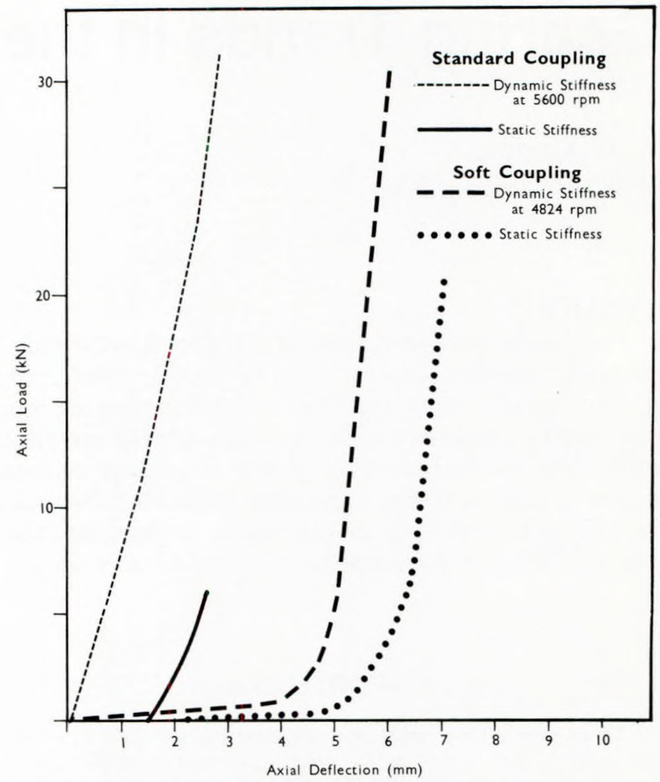


**FIG. 1: Typical stiffness curves for membrane couplings**

One disadvantage of the membrane coupling over the fine tooth type (at least in principle) is that axial displacement causes an axial force to be generated. Typical response curves are shown in Fig. 1. In an Olympus installation, at full power, the combined affects of gearbox and gas turbine movement together with expansion of the Hiduminium torque tube (running at 100+ °C) effectively lengthens the coupling by about 6 mm, equivalent to 3 mm compression on each membrane pack. A 'standard' coupling unit for the required power and speed has a maximum permissible displacement of 2 mm at which it can generate an axial load of 30 kN. Given a common drive arrangement where the turbine is connected directly to the gearbox input pinion *via* the coupling, with axial location being provided by the tubine thrust collar at one end and a location collar on the primary wheel at the other (as for *CVS* and *Type 23* classes), then the effect of this load on double helical gears can be to:

1. Reduce load on the turbine thrust bearing.
2. Overload the wheel location bearing.
3. Bias the torque distribution such that one helix is more highly stressed than the other.

The solution adopted is to install the couplings with a cold pull up (CPU) of about 2 mm per membrane unit (4 mm total), in which case the static loading is about 2 kN and the full power load about 9 kN.

Turbine axial expansion of the Spey engine is considerably more than that of the Olympus and in the Spey powered *Type 22*s it has been necessary to adopt a 'soft' coupling having much reduced axial stiffness (characteristics as in Fig. 1).

Two aspects of high-speed coupling duty are commonly of interest to engineers:

1. Alignment between turbine and gearbox.
2. Transmitted torque.

Neither of these is easy to monitor directly, particularly in view of the environment within the torque tube cover, *viz.* hot (100+ °C) and oil laden. Early attempts to measure misalignment from the deflection of membrane packs, using a simple arrangment of proximity probes, demonstrated that, whilst just adequate for trials purposes, there was much to be desired in terms of a practical system. A trial last year of a commercial product utilising a probe attached to the membrane bank with signals being extracted by a telemetry system proved to have much greater potential although not yet in the realms of a DIY fixture (considerable calibration and tuning was required).

High-speed line torque has until recently been of mainly academic interest, the modern strain gauge torsionmeters fitted to the propeller shaft being perfectly adequate for normal operating purposes. However, with increasing emphasis on engine tuning and balance between engines in multiple installations, trials are shortly to be carried out on a commercial high-speed torsionmeter system built into the coupling torque tube.

As a concluding remark on the subject of high-speed couplings it is appropriate to note that no problems at all have been experienced with membrane units. Initial maintenance philosophy was to replace complete couplings at refit (every 4 years) but this has now been relaxed to *in situ* endoscope inspection of membrane banks.

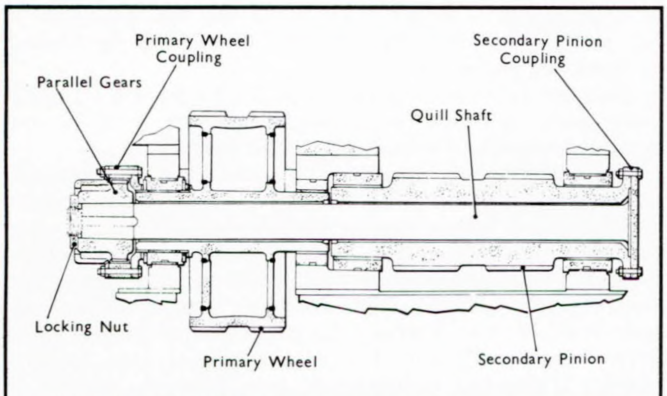


**FIG. 2: Arrangement of intermediate shaft of a *Type 42* gearbox**

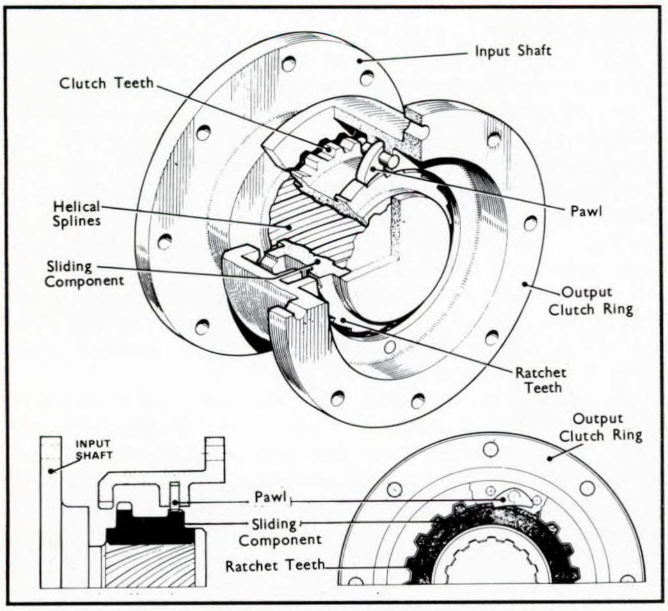


**FIG. 3: Basic features of an SSS clutch**

### Internal gearbox couplings

In gearboxes which have multiple reduction stages, one or more sets of double helical gears and individual location of each gear stage, as in most RN warships, then it is essential that the shafts connecting gear stages incorporate a flexible coupling to allow axial movement between them (thus avoiding the possibility of axial loading of the gears as described earlier). In *Leander*, *Type 21*, *Type 22* and *Type 42* classes this flexibility is provided by fine tooth couplings fitted one at the end of each of the two secondary quill shafts (these being tandem, locked train designs) (see Fig. 2). Whilst overall performance has been satisfactory, a number of problems can arise with this type of unit:

1. Uneven wear of the two couplings can result in one secondary train transmitting substantially more torque than the other. In severe cases this overloading results in deterioration of the affected gears through pitting, scuffing or tooth breakage. In the classes mentioned only one or two such incidents have occured, with damage being limited to relatively minor pitting on the softest gear — the through hardened main wheel.
2. Under high torque considerable axial load is transmitted before the coupling will slide and in the first *Type 21* and *Type 42* ships this stiffness resulted in overloading of the relatively light thrust bearing locating the primary gear train. The problem was resolved by modification of the bearing to incorporate tilting pads.

The Naval Engineering Specification for propulsion gearing (NES 305) now requires the assumption of a friction coefficient of 0.3 in the use of all such couplings.

As a result of these potential problem areas, RN preference is for the use of membrane-type couplings in this application (although these need to be accurately aligned axially if the generation of high forces is to be avoided)

## CLUTCHES AND BRAKES

### Non-slip clutches

In gearboxes with multiple inputs it is highly desirable that engine changeovers are accomplished automatically at any shaft speed within the operating ranges of the incoming and outgoing engines. The clutch adopted for this duty in RN 'all gas turbine' warships is the synchronising self-shifting (SSS) unit, a simple example of which is shown in Fig. 3.

Functioning of the clutch is straightforward. When the input half is running slower than the output the pawls ratchet over the ratchet teeth but as soon as input and output speeds are synchronised the pawls engage and drive a sliding element, which carries the inner main clutch teeth, along helical splines into engagement with the outer main clutch teeth. Phasing between the ratchet teeth and main clutch teeth ensures that the latter engage precisely. As soon as the two halves of the main clutch engage, the pawls disengage from the end of the ratchet ring and thus at no stage transmit any part of main drive torque. Disengagement occurs immediately the clutch output rotates faster than the input, the helical splines taking the sliding member back to its ratchetting position.

Additional features commonly found in RN clutches include:

1. A 'lock-in' which operates automatically as soon as the main clutch teeth are fully engaged. This prevents 'shuttling'

of the clutch which could be caused by torque reversals during manoeuvring *etc.*, although such action is unlikely to result in damage even if left unchecked.

2. A 'lock-out' operated manually, which disengages the pawls from the ratchet ring and permits operation of the gas turbine independantly of the gearbox for testing purposes.

3. A 'lock-out' operated automatically (in reversing gearboxes) to prevent engagement of the clutch during astern operation.

4. A mechanical 'baulk', associated with an automatic lock-out, which prevents the pawls being re-engaged with the ratchet ring when the clutch input is rotating faster than the output. (Early clutches did not incorporate this feature with the result that, if operated incorrectly, pawls and ratchet ring were severly damaged.)

Whilst overall experience with the SSS clutch has been very satisfactory, occasional failures have occured. In almost all cases the failure has been evidenced by an inability to engage the clutch consequent upon damage to the pawls and ratchet ring, the most likely explanation for which is either misalignment between clutch input and output or, for some reason, the pawls not engaging the ratchet ring until the input is running faster than the output.

### Friction clutches

The RN has little recent experience with friction clutches, the last propulsion application being in the *Type 41* and *61* frigates of the 1950s. Two possible future requirements, however, are causing renewed interest in the subject:

1. The use of diesel cruise engines, for which application a friction clutch is better suited than an SSS type.

2. The design of a medium power reversing gearbox, similar to commercial marine reversing gearboxes.

### Brakes

All gearboxes in gas turbine powered warships are fitted with a brake for use in certain manoeuvring situations. The unit adopted in *Type 21*, *22* and *42* classes has a chrome-plated copper disc with three twin caliper units, these being operated from an HP air supply through a metering valve which regulates the rate of application of pressure.

This is one gearbox component that has not met the high reliability standards required in a naval application with numerous failures having been recorded, some of which were quite spectacular, as when a control system malfunction resulted in brake application at high power (not actually a fault of the brake itself). The main problem area has been the metering valve, quite a complex unit, which appears to be unsuited to an environment where it is operated infrequently and, as a consequence, spends much of its time at the end of an often damp air system.

In the *Type 23* propulsion arrangement (see Fig. 4) the gearbox brake plays a much more active role in ship operation, being required to control gearbox speed prior to main clutch engagement or disengagement. High reliability is therefore demanded and, in view of experience with the existing brake, a more basic industrial/marine design operated direct from an LP air supply has been selected.

## GEAR DESIGN AND MATERIALS

### Gear design

The present gear rating procedure for RN gears was evolved through the research and development programme conducted by the Naval Gear Research Association (NAVGRA), which was active from 1946 to 1979. The basic specification is simply defined as a maximum Lloyd's Register of Shipping *K* factor and root bend stress for the particular material and general size of gear (primary or secondary reduction).

Whilst at first sight this might appear far too basic to be reliable, particularly in view of the extensive analysis required by BS 436, it must be remembered that:

1. The range of gear module and diameter for either primary or secondary gears is quite small.

2. The prime movers, whether steam or gas turbine, have very similar characteristics.

3. The operating environment is the same for all applications.

4. Extensive testing (to failure) was carried out on representative primary and secondary gear pairs manufactured in a variety of material combinations.

It might therefore be argued that the numerous factors defined by BS 436 have in fact been rolled up into a standard figure for a standard application.

Notwithstanding the generally very satisfactory experience with the present specification, it has been recognised for some years that a fuller analytical procedure would give a better definition of safety factor and possibly result in smaller gears and hence more compact gearboxes. To this end, in the late 1970s MoD sponsored the development of a rating procedure for naval gears using a similar data base to that for BS 436. The result was not entirely satisfactory in that, when tested against gears in service, failure was predicted for certain applications where experience had been completely trouble free. Further development was intended but, with the decision of BSI to proceed with production of a new British Standard, it was delayed in the hope that RN requirements would be adequately covered.

The present position is that, once again, an evaluation of existing designs is being carried out to the new procedure. If this proves satisfactory then BS 436 (1986) will be adopted for future RN gearbox designs.

Gear accuracy requirements demanded by the use of highly rated gears and low transmitted noise are met by the specification of BS 1807 Grade AO for all elements.

For completeness it should be said that all post-war gearbox designs have used standard involute teeth, although prior to this there were a number of variants including Parsons All Addendum and Vickers Bramley Bostock (VBB). Although it seems unlikely that there will be any major change in the future, a study has been carried out into the application of the Westland design conformal gearing to a marine gearbox. Whilst the potential higher load carrying capacity of this type of tooth has some attraction, the lack of experience in larger sizes combined with uncertain noise characteristics means that a considerable programme of development would be required before serious consideration could be given to naval use.

### Gear materials

The range of materials specified for RN gears together with their maximum permissible *K* factors are shown in Table II.

In the majority of current ship designs the pressure to produce compact gearboxes has resulted in extensive use being made of hardened and ground gears.[5] The high load carrying capacity of carburised teeth is a most attractive feature, although this has to be offset against the problem of distortion during hardening. As a consequence, carburising is most suited to the production of smaller gear elements up to primary wheel size. (Notwithstanding this, carburised main wheels were adopted for the *County* class destroyers and early nuclear submarines, but being single helical distortion could be more readily catered for.)

The production of sound carburised gears places demands upon the designer, material supplier and manufacturer, for failure in any of these areas can completely negate the value of even the best gear rating procedure. It is the designer's responsibility to ensure that the gear construction (particularly if forged) allows free flow of coolant during quenching, thereby avoiding 'soft' areas caused by hot pockets and minimising distortion. The gear manufacturer must, in the grinding process, minimise the variation in final

**Table II: Material combinations and maximum permissible *K* factors for main propulsion gearing**

| *Gear Material* | | *LRS factor* (MN/m²) | |
|---|---|---|---|
| *Pinion* | *Wheel* | *Cruising and Primary Reduction Gears* | *Secondary Reduction Gears* |
| 826 M31<br>826 M40<br>830 M31<br>T, U or V<br>DG Ships<br>6022 | 070 M55 Normalised<br>BS 970: Part 1: 1972 | 1.25 | 1.05 |
| 722 M24<br>Nitrided<br>or<br>897 M39<br>Nitrided<br>DG Ships<br>6019 | 070 M55 Normalised<br>BS 970: Part 1: 1972 | 1.50 | 1.30 |
| | 826 M31<br>826 M40<br>830 M31<br>835 M30<br>DG Ships 6022 | 1.95 | 1.65 |
| | 722 M24 or<br>897 M39<br>Nitrided<br>DG Ships 6019 | 3.10 | SeeCl auseDG<br>075 |
| | 817 M40 T<br>Induction hard<br>DG Ships 6018 | | See Clause<br>0715 |
| | 655 M13 Carburised<br>DG Ships 6017 | 3.10 | 3.10 |
| 655 M13<br>Carburised<br>DG Ships<br>6017 | 070 M55 Normalised<br>BS 970: Part1: 1972 | 1.50 | 1.30 |
| | 826 M31<br>826 M40<br>830 M31<br>835 M30<br>DG Ships 6022 | 1.95 | 1.65 |
| | 722 M24 or<br>897 M39<br>Nitrided<br>DG Ships 6019 | 3.10 | See Clause<br>0715 |
| | 817 M40 T<br>Induction hard<br>DG Ships 6018 | See Clause 0715 | |
| | 655 M13 Carburised<br>DG Ships 6017 | 4.00 | 4.00 |

Clause 0715. Limitations of pitch and maximum permissible *K* factors affecting teeth hardened by nitriding or induction hardening, not given in the table shall be agreed with MOD (PE) for each proposed application

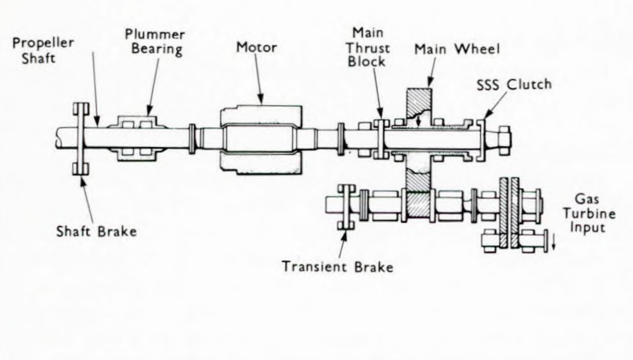


**FIG. 4: *Type 23* propulsion arrangement**

case depth which could result if due allowance is not made for distortion.

The quality of a finished gear is established by the destructive examination of sample material carburised with the gear itself together with checks of surface hardness. Unfortunately there is at present no practical method of establishing case depth non-destructively.

Nitriding is the most attractive gear hardening process from a manufacturing aspect because of the almost total absence of distortion. It is possible, if required, to dispense with final grinding. Nitrided gears have been used in a number of RN designs, applied mainly to primary wheels, although at present carburising is preferred because of its higher load carrying capacity.

Induction hardened gears have the advantage of a load carrying capacity almost comparable to that of carburised gears but with much less distortion. The previously mentioned NAVGRA R and D programme adequately demonstrated the performance of induction hardening applied to larger teeth (8-10 module), typical of those used on secondary reduction gears, but development ceased before the parameters necessary for primary sized gears had been fully established.

In RN applications the process is at present being applied only to main wheels, as in the *Type 23* frigate and later submarines, but primary wheels in some *County* class gearboxes have given trouble free performance.

The quality of induction hardened gears is in large part dependant on the evenness of the hardened layer and this is dictated by the profile of the inductor together with its positioning within the tooth spece. Since, as with carburised gears, the hardness profile cannot be readily measured on finished teeth, use is made of test arcs (identical in form to the actual gear), one treated before and one after on precisely the same machine settings.

Since the 1960s through hardening has been applied only to the main wheels in frigate and destroyer classes (ex *Type 23*) and the main wheel and maneouvring drive wheels in the *CVS* design.

## GEAR PROBLEMS

Gear problems in RN warships are infrequent and, when they do occur, fall into one of three categories.

### Pitting

This only affects through hardened gears and is the result of excessive Hertzian stress on the tooth flanks. Providing the basic design is adequate, pitting is indicative of a fault condition increasing the loading on the gear teeth and, in RN experience, this has been traceable to either axial forces being imposed on the affected element or maldistribution of torque in a locked train gearbox.

Initial pitting has occasionally been experienced on new through hardened gears as high spots are relieved. However, this condition quickly stabilises and does not affect subsequent performance.

### Scuffing

This is a potential hazard in all highly loaded gears and present predictive methods are inadequate to define accurately the safety factor in any specific application. In the RN, where a risk is considered to exist, recourse is made to an extreme pressure oil (OEP69) and as a result of this few problems have been experienced.

Notwithstanding the availibilty of EP oils, there remains a need to understand better the mechanism of scuffing and to develop an accurate analytical procedure. For this reason, MoD are sponsoring R and D at Cardiff and Leicester Universities, Imperial College, London and a gear manufacturer.

## Fracture

### *Nitrided gears*

The first ships of the *Type 21* and *42* classes (HMS *Amazon* and HMS *Sheffield*) both suffered from fracture of teeth on their nitrided primary wheels (*K* factor 450 lbf/in$^2$) and, as a result, a very thorough investigation was made into the design, material properties and manufacturing standard of the gears, but no specific cause could be identified. In the circumstances it was deemed prudent to change the material of later manufacture primary gears to carburised EN36A, with which no problems have been experienced.

Note that a recent reassessment of the original nitrided gears to the new BS 436 rating procedure indicates a safety factor well below that acceptable for the duty.

### *Carburised gears*

Failures of two primary wheels have been experienced in *CVS* class gearboxes. In the first case three teeth were broken, two adjacent and one about 100° away, and in the second case one was broken. In both the characteristics of the fracture were unusual in that the full tooth had been removed just leaving the stubs of end relief (see Fig. 5), the crack in the ahead flank following a line parallel to, but just above, the pitch line.

Again a full investigation has been made into the failures. Metallurgical examination showed variation in case thickness not to be a factor and no other material defects were found (although an independant analysis by Portsmouth Naval Base Laboratory did find evidence of corrosion fatigue resulting from problems with condensation during gearbox flushing by the shipbuilder). Dimensionally all important parameters were within specification and a design assessment confirmed adequate safety factors for the required duty (by naval standards the gears are quite conservatively rated at a *K* factor of 390 lbf/in$^2$). Even assessment to BS 436 has proved satisfactory.

Rather than an inability of the gears to accept specified torque, the alternative explanation was that they had been subject to excessive load. To investigate this possibility sea trials were arranged with one gearbox being instrumented to cover:

1. Primary wheel output torque (propeller shaft torque is a machinery operating parameter, continously displayed at the control position).
2. Axial load on the primary wheel location bearing.
3. Axial deflection of the HSL coupling membrane banks.
4. Pinion and wheel vibration levels.

Results showed no evidence of serious over torquing even during manoeuvres (although this is dependant upon operator reaction) and vibration levels were quite normal. Axial load on the location bearing was, however, recorded at about 18 kN at full power, subsequently calculated as being sufficent to bias the load distribution between helices to 60:40 rather than 50:50 as expected, *ie* a 20% overload on one helix. This was confirmed by measurements across the membrane banks which changed from an initial tension of 3 mm to a compression of 3 mm.

From all the evidence collected it is now believed that the failures did not result from a single cause but a number of factors, including corrosion on tooth flanks, excessive axial load and occasional over torquing. However, even these do not fully explain the incidents.

Corrective measures being taken to reduce the likelihood of a recurrence of the problems are:

1. Increasing cold pull up of coupling to 4.5 mm so that full power compression is limited to 1.5 mm. This will reduce axial loading to 10 kN.
2. In the longer term, converting all couplings to 'soft' membrane packs should mean that full power axial loading will be negligible.

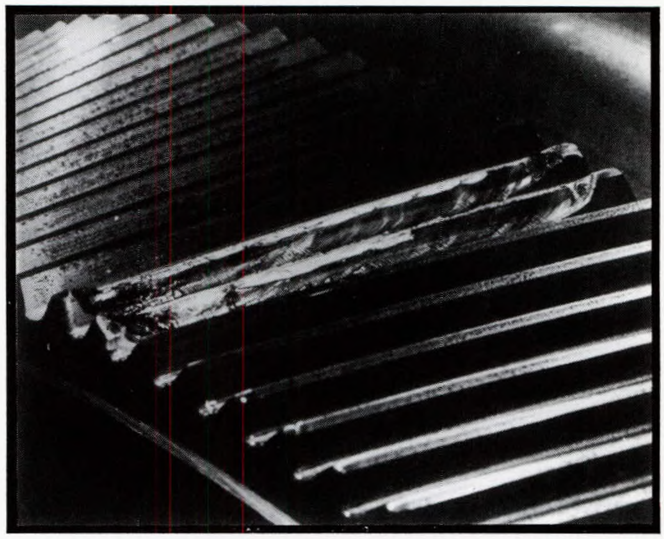

**FIG. 5: Broken teeth on primary wheel of HMS *Illustrious***

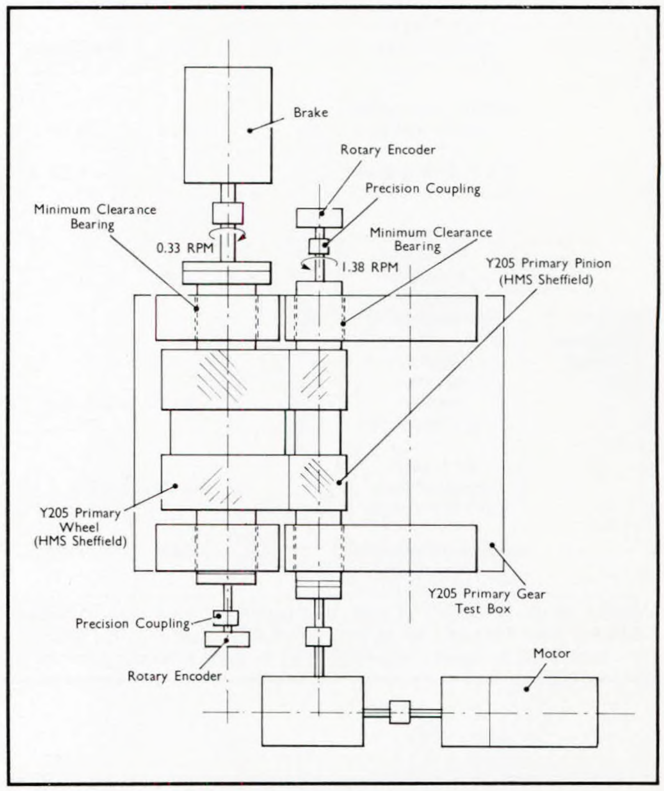


**FIG. 6: Arrangement of transmission error test rig**

## Induction hardened gears

No failures have been experienced with induction hardened gears.

## Through hardened gears

The most comprehensive gear failure of recent years occured in HMS *Torquay*, an elderly *Whitby* class frigate nearing the end of her active life. The gearboxes involved are of dual tandem, locked train, double helical configuration using through hardened gear elements throughout (a full description is contained in Ref. 1) and have given few problems. However, a routine inspection in this ship revealed one secondary pinion with a number of failed teeth. The same pinion and meshing main wheel also exhibited scuffing, gross pitting and almost total dedendum erosion, mainly on one helix.

Given the age of the ship the optimum solution, replacing all secondary gears, was not justifiable and it was found that the next option, replacing the secondary pinions, was not possible because the main wheel teeth were of non-standard height. The decision was therefore taken to dispense with the damaged pinion and resume operation on one secondary train only. In doing so the major concern was over the primary pinion bearings, which would see a much higher load than when balanced by two primary wheels. This loading in fact defined the maximum permissible power transmitted.

Subsequent sea trials were satisfactory, vibration levels being quite acceptable up to the new power limit (demonstrating that dedendums are not really necessary in helical gears) and HMS *Torquay* completed her active career.

The cause of the failure was found not to be wear in the fine tooth couplings, as originally suspected, but slackening of the joint on the other end of the quill shaft on the undamaged pinion, which resulted in it being totally unloaded.

### General note

It is interesting to note that all of the gear failures described have been found as a result of visual inspection, and in no case were ship staff made aware of a potential problem by increased noise and vibration. This is not a reflection on personnel concerned as evidenced by the fact that, following one *CVS* failure, a detailed vibration analysis was carried out on the affected gear but no abnormal characteristics could be found on the signature.

## GEAR RESEARCH AND DEVELOPMENT

Whilst a small amount of R and D into certain aspects of gear design, material and performance is being sponsored, this is largely through the need to investigate aspects of an in-service failure rather than part of a continuing programme. The priority area of planned work is the better understanding of noise/vibration generation and transmission within a gearbox, this being approached through theoretical and practical work on the transmission error between meshing gears. [Transmission error (TE) is here defined as the variation in angular rotation of one member of a gear pair when its mate is turned at perfectly constant speed under constant torque. It is therefore a reflection both of the combined errors of the two gears and the force variation within the mesh.] A fuller appreciation of TE is given in Ref. 6.

Theoretical work carried out over the last decade has resulted in an analysis program which, from details of tooth geometry, loading etc., will produce recommended optimum reliefs and resulting load distribution. This has now been used to modify the corrections on a gearbox in manufacture but it will be some time before it is known whether this has resulted in a noise signature lower than that from earlier gearboxes of original design.

Despite the work put in so far it is recognised that there is still a long way to go before it can be claimed that TE can be accurately predicted.

A programme of practical measurement of TE is being conducted using a test-rig equipped with a primary pinion and wheel from a *Type 42* gearbox (see Fig. 6). A full description of this facility is given in Ref. 7. The essential features are two optical encoders (gratings), one each on the end of the pinion and wheel shafts, which can measure rotation to an accuracy of +/– 0.5 seconds of arc (2 592 000 divisions per revolution). An electronic measuring and analysis system takes the output from the encoders as the pinion is turned slowly, calculates the precise angle turned by both pinion and wheel and, allowing for gear ratio, provides the error between the two at any instant – the transmission error. The overall system has been satisfactorily demonstrated and future trials will investigate the influence of gear alignment and a variety of tooth corrections whilst the gears are running at torque (but not speed) equivalent to cruise power in a destroyer sized warship.

The ideal conclusion from this area of work would of course be an analytical program which accurately predicts the transmission error from a gear pair as measured in the test rig. This is, however, a very distant prospect.

## EPICYCLIC GEARS

Epicyclic gears hold an unfortunate position in relation to RN warship gearboxes – always of interest, occasionally investigated in detail, but never used. Unfortunately this situation is likely to continue until a specific application arises where they have a major advantage over parallel shaft gears.

A study carried out in the 1970s concluded that, in a propulsion plant with two inputs, the epicyclic could show benefit as the final reduction stage and from this started a programme to design, build and back-to-back test a gear capable of transmitting 30 000 hp through a 4.3:1 reduction with an output at propeller shaft speed. Throughout a thorough trials period mechanical performance of the two units was quite satisfactory up to the 42 000 hp overload condition. However, the noise generating characteristics left much to be desired, partly because of the use of spur gears and partly because of the difficulty in achieving high tooth accuracy on a large annulus (although it must be stated that noise was not a major factor in the design specification). Modification to a flexibly mounted annulus improved the situation but, some years after the otherwise successful conclusion of the project, suspicion remains over the noise performance of epicyclics.

Interest is now being renewed over the possible use of epicyclics as input stages to a high reduction gearbox, but it first remains to be demonstrated that these would be no noisier than equivalent duty parallel shaft gears.

## BEARINGS

### Journal bearings

The naval specification for journal bearings has changed little over recent years, the basic requirements being:

1. For main wheel bearings, a length to diameter ratio (*L/D*) of 2/3 to 1 with a specific loading not exceeding 24.5 bar.

2. For all other bearings, an *L/D* ratio of 1/3 to 2/3 with a specific loading not exceeding 34.5 bar.

All bearings are to be thin/medium walled, steel backed and lined with white metal. Exceptionally main wheel bearings may be thick wall and for high-speed duty an aluminium/tin lining is accepted. Oil inlets are to be twin axial groove at the butt line unless the range of journal bearing atitudes necessitates either a single axial groove or circumferential groove.

No active research into bearing design is being carried out at present although an improved design analysis program has recently been part sponsored by MoD. The main requirement for the future is likely to be the demonstration of lower *L/D* ratio and increased specific loading as a contribution to reduction in gearbox length.

### Thrust bearings

Within the gearbox, conventional plain thrust and pivoted pad bearings are used as appropiate to the duty. Flooded type tilting pad bearings are preferred as these at least give the impression of some oil reserve in the event of a supply failure, but in high-speed/high-load applications (such as single helical primary pinions) oil churning can result in reduced efficiency and excessively high temperatures. Directed spray bearings have therefore been accepted for this duty.

As with journal bearings the only recent work in this area has been the development of an improved design program jointly with a maufacturer.

### Bearing problems

Bearing failures in service are infrequent and when they do occur can normally be traced to a lubrication problem, exceptional operating condition or incorrect assembly. Of more concern has been a number of instances of stannic oxide corrosion affecting both journal and thrust bearings, which will be discussed more fully in the section headed 'Lubricants'. Maintainance policy for bearings has until now been specified as 'remove and inspect at refit' (between 4 and 7 years); this is now being reviewed with the prospect of reducing examination to a representative sample.

## THRUST CONES

For the first time in an RN warship, the *Type 23* frigate gearbox uses thrust cones to balance the axial thrust generated by a pair of single helical gears (the main wheel and secondary pinion). The general arrangement is as shown in Fig. 7.

Being a unidirectional unit only the main cone is subject to loading under power, the surge cone acting as a 'stop' in the event of reverse torque under braking.

The principle advantages of thrust cones over conventional bearings are considered to be reduced maintainance, lower risk of failure and a more compact arrangement achievable in some applications. However, they do result in some complication to the design of the pinion. Integral thrust cones cannot be used as these would interfere with gear cutting so separate rings are used, with a suitable location which will prevent any rocking or axial movement when subject to the rotating contact load. This is the critical feature of the design, and for the *Type 23* full-scale testing was carried out at up to 150% full power condition (satisfactorily).

## LUBRICANTS

Until the 1970s an extreme pressure oil (OEP69) was used in all naval gearboxes (and, being on a common supply, the turbines) but at this time a number of instances were reported of stannic oxide corrosion affecting white metal bearings in steam ships. The corrosion appears as a dark staining on the surface of the metal and is very hard. In small areas it is not serious (although it does reduce the ability of the whitemetal to absorb debris) but if left unattended it thickens and eventually flakes, at which time it can have the effect of a machine tool on the journal or thrust collar surface (thrust bearings are more seriously affected than journal bearings).

A full investigation was undertaken into the causes of stannic oxide and this concluded that the principal contributory factors were the presence of water and chlorides in the oil at high temperatures. The chlorides were not in fact the result of sea water contamination but originated from the oil itself, chlorinated wax being the EP additive in OEP69. The solution adopted was to convert as many gearboxes as possible to a straight mineral oil (OM100). Scuffing assessments indicated that this is acceptable in all but reversing gearboxes (*County* and *CVS* classes) and submarines.

For the latter vessels a programme was started, in association with an oil company, to produce a new EP oil of comparable performance to OEP69 but without the chlorine

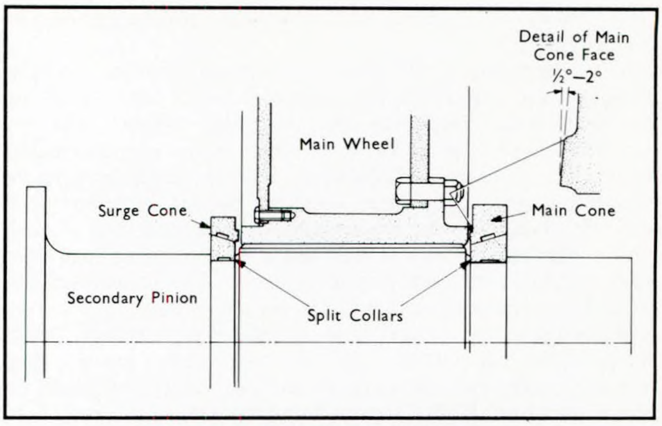


**FIG. 7: Thrust cone configuration**

additive (it having been previously demonstrated that a commerical oil was inadequate when scuffing resulted during trials in a *County* class destroyer). After a lengthy period during which numerous possibilities were evaluated in the laboratory and failed to meet specification, a new oil (OEP80) has been produced, having sulphur and phosphorous based additives, which appears to be the equal of OEP69 in all respects. This is now undergoing service evaluation in one *County* class and one submarine gearbox prior to (hopefully) full acceptance.

## INSTRUMENTATION

The principle instrumentation specified for RN gearboxes comprises:

1. Thermosensors (either resistance of thermocouple) fitted to the back of all thin/medium wall bearings or in the whitemetal of thick wall journal and plain thrust bearings, these being located at the points of minimum oil film thickness (full power). For tilting pad thrust bearings the sensor is in the oil drain.
2. Oil pressure gauges for each group of journal bearing and sprayer supplies.
3. Temperature sensors for all contacting metallic seals.
4. Phonic wheel or tachogenerator for shaft speed and direction.
5. Vibration transducers fitted to all input line bearings with provision for fitting on all other bearings. The transducer on the first input bearing is connected to a continuous monitoring system to give warning of any problem with the high-speed line. This facility in *Type 22* and *CVS* classes (where it indicates only overall vibration) has given some problems whereas there have been none with the actual lines monitored.

## CONDITION MONTIORING

Condition monitoring of gearboxes is at present limited to observation of changes in bearing temperature and periodic vibration surveys of the high-speed line, although studies are being sponsored into improvements in two specific areas:

1. Interpretation of vibration data to identify deterioration in gear condition/performance (as noted previously, conventional diagnosis cannot identify a missing gear tooth). This exercise is using as its basis experience gained in the monitoring of helicopter gearboxes.
2. Methods of establishing journal bearing condition in service. This is in fact the priority area since, if successful, it will obviate the need to open up and inspect bearings on a routine basis. (Such inspections are a major interference in the progress of a refit, leave the gearbox at risk whilst open

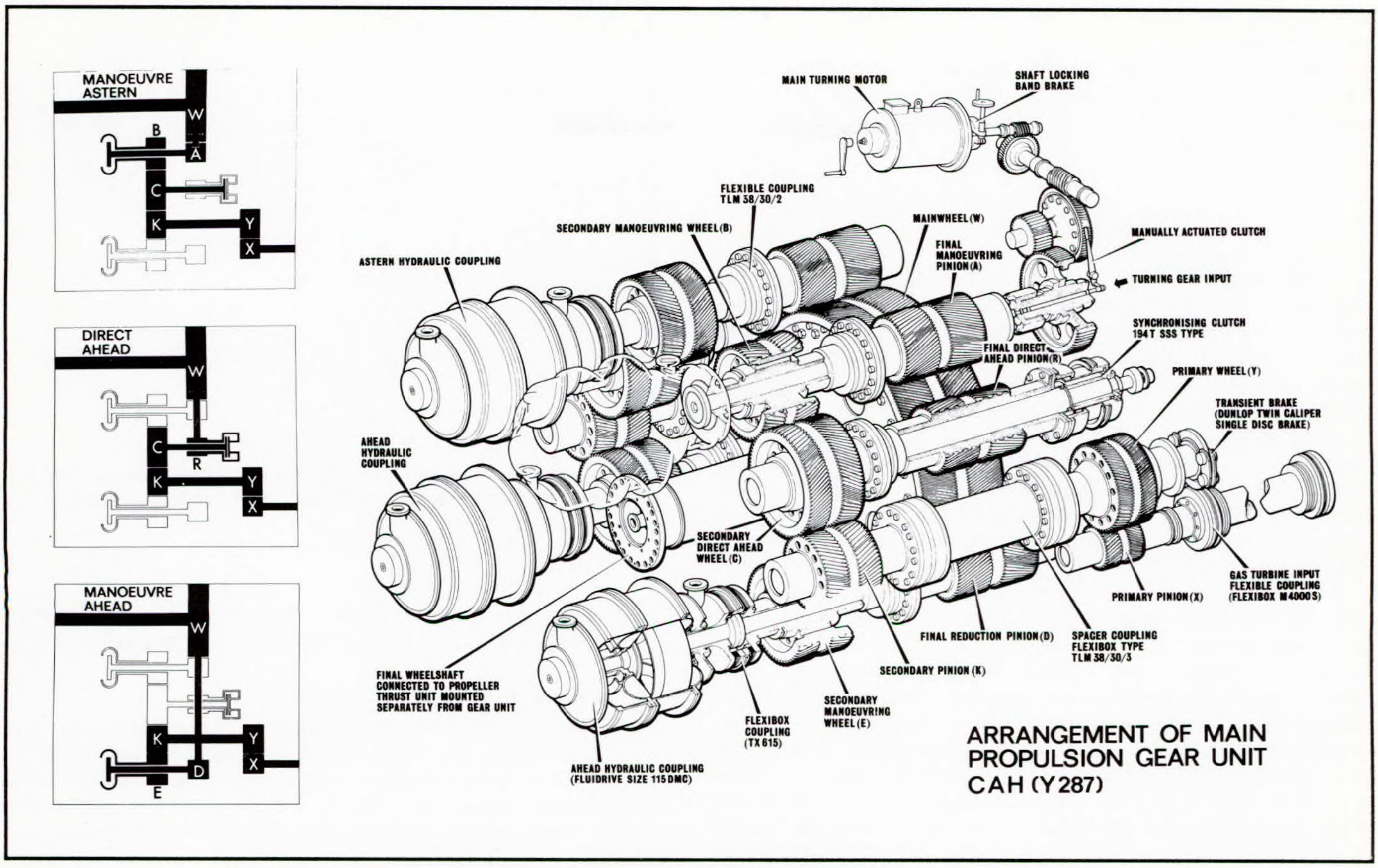


**FIG. 8: Arrangement of *CVS* gearbox**

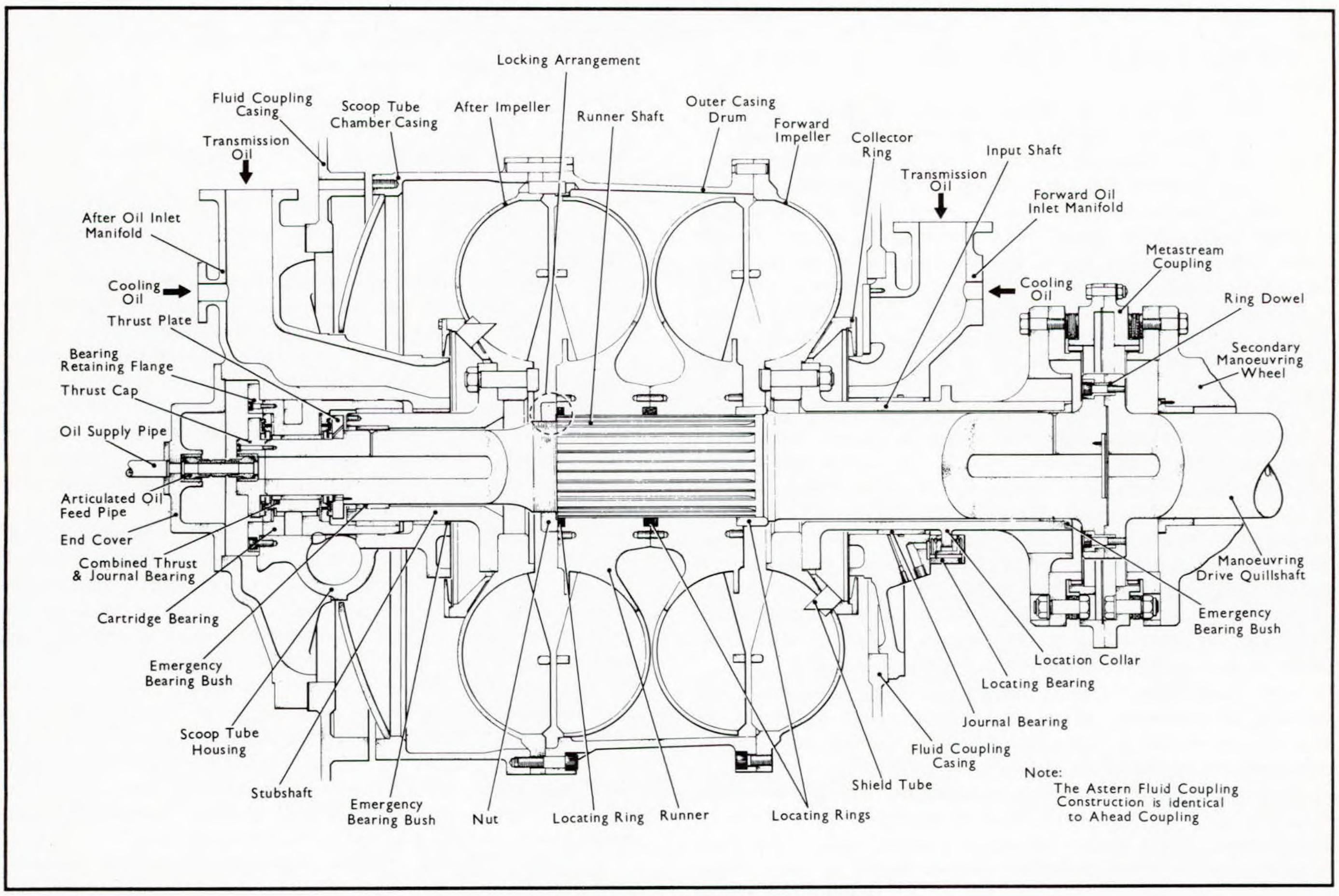


**FIG. 9: Arrangement of *CVS* fluid coupling**

and provide the opportunity for incorrect reassembly, usually for no reason.)

Work has also been carried out into the application of various types of oil debris monitoring systems to naval gearboxes, but none have so far been assesed as of practical value in this specific duty.

## FLUID COUPLINGS

Fluid couplings are used in the manoeuvring drives of reversing gearboxes fitted in the *County* class, HMS *Bristol* and the *CVS* class. In the first two the couplings (one ahead and one astern) on each gas turbine input rotate at engine speed on a common line, resulting in a compact gearbox but complex arrangement of shafts and clutches (a full description of the *County* class gearbox is contained in Ref. 3 and HMS *Bristol* is similar in concept). Some problems were experienced with the couplings during early service, principally because of the high rotational speed, but these were overcome and subsequent performance has been trouble free.

The design of the *CVS* class adopts a different approach with separate gear trains for direct drive, manoeuvring ahead and manoeuvring astern (see Fig. 8). Each of the two Olympus engines drives through a primary reduction stage into a secondary pinion which meshes with both an ahead manoeuvring wheel and a direct drive wheel, the latter also meshing with and driving an astern manoeuvring wheel. The two manoeuvring wheels are connected through their respective fluid couplings to final stage pinions; the direct drive wheel provides the input to an SSS clutch, the output from which drives a third final pinion. The arrangement is symmetrical either side of the main wheel. The direct drive wheel is slightly larger than the manoeuvring wheels so that the input to the clutch runs slower than the input to the couplings by an amount just greater than normal coupling slip.

The fluid couplings are dual circuit, scoop controlled, as shown in Fig. 9.

Gearbox operation is straightforward. In direct drive the clutch is engaged, coupling scoops are 'in' with main oil supply to them isolated, allowing just a small cooling flow. When manoeuvring ahead is selected by the operator, oil from the main lubricating oil system is opened up through a diverter valve to the ahead fluid couplings [a second diverter valve selecting inner, outer or both depending on engine(s) running], at the same time the scoop is withdrawn allowing the working circuits to fill. Once full the coupling output shaft is driven faster than the input to the SSS clutch, which consequently disengages and, as a result of a prior control signal, locks out.

Selection of direct drive reverses the sequence – the clutch moves to its operating state, the scoop is entered and, as the quantity of oil in the coupling reduces, its output speed drops until the clutch engages. Changeover of drive mode therefore occurs automatically upon emptying or filling of the ahead couplings. When astern power is demanded from direct ahead, selection of manoeuvring drive first transfers power to the ahead coupling, the diverter valve then diverts oil supply from this to the astern coupling, the ahead scoop is entered, emptying its circuit, and the astern scoop is withdrawn allowing the astern coupling to fill and transmit power.

The propulsion control system is fully automatic and includes all necessary safeguards to protect the machinery, *eg* from engagement of coupling drive at full power, and to adjust parameters for single and twin engine drive.

In service the manoeuvring arrangement has worked well, the only problem associated with it having been a tendency for the nuts which secure the output (runner) half of the fluid coupling to its drive shaft to slacken back, in three cases permitting sufficient axial movement for contact to occur

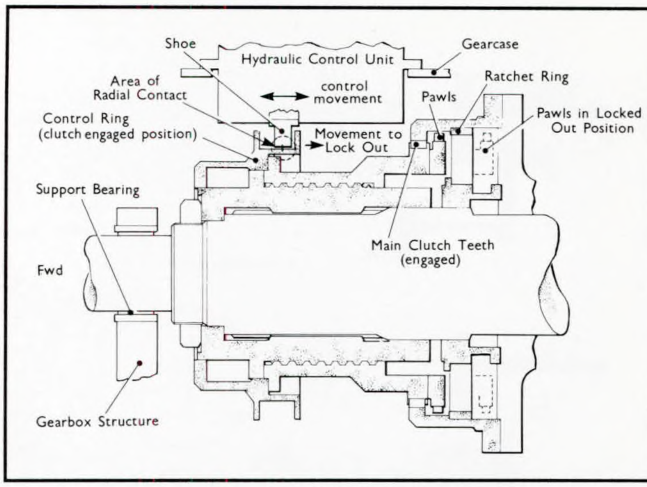


**FIG. 10: Arrangement of *CVS* SSS clutch and actuator**

with the input (impeller) half. Improvement to the locking arrangement will (hopefully) prevent further occurences.

### Future arrangements

Whilst conventional fluid couplings have performed well in RN reversing gearboxes, the requirement for two units, an astern drive train, diverter valves and scoop controls adds considerable complexity and, in the case of *CVS* class, size to the basic gearbox design. The concept of a reversing fluid coupling, eliminating the need for these features, is clearly attractive and such an equipment does exist in a form suitable for marine use, being fitted in the Italian Navy aircraft carrier *Guiseppe Garibaldi*. MoD has acquired a 'second hand' coupling of this type which is now being evaluated in a test facility. This is a subject in its own right which, regrettably, cannot be adequately covered here.

## GEARBOX EXPLOSIONS

In the context of this paper the subject of gearbox explosions is rather out of place and it is included for two reasons only:

1. To provide information on a recent incident in the RN.
2. To emphasise the importance of general engineering detail in gearbox design, manufacture and maintenance.

Gearbox explosions in warships are rare occurences although not unknown.[8,9] The most common cause of such incidents is bearing failure and, in this respect, the standard of instrumentation fitted in modern RN warships, coupled with remote warnings and alarms, gave considerable confidence that any such occurence could be detected before it escalated to a dangerous state.

Unfortunately in April 1986 HMS *Illustrious* did experience an 'explosion' with no prior alarm being recorded. In fact 'explosion' is something of a misnomer – combustion occured within the gearcase and the resulting pressure rise caused the cover joints to rupture, releasing burning oil vapour into the gear room where a sizeable fire started.

An inspection of the gearbox immediately upon return to port revealed that, at least superficially, no damage had been sustained apart from the extensive buckling of the casing, even the internal wiring being untouched. The compartment itself had been less fortunate, with extensive fire damage in the upper level, including external gearbox components such as the brakes, turning gear and local instrumentation.

A more detailed visual examination of the gearbox internals revealed only one potential cause for the explosion, there being obvious overheating of the SSS clutch control shoe as a result of radial contact with the clutch itself (see Fig. 10). However, this could have been a result of the incident as the casing supporting the control was also distorted.

Subsequently the gearbox was completely stripped and gears, bearings and bearing alignment thoroughly checked, with no problems being found. Upon completion of the gearbox rebuild, including new covers, and the major repair of the compartment itself, a full sea trial was carried out satisfactorily and HMS *Illustrious* resumed active service.

The overall conclusion from the investigation into the explosion was that the incident resulted from the previously mentioned contact between the clutch and its control unit, this being attributed to the clutch control seating on the gearbox casing being too close to the centreline of the clutch itself, thereby absorbing the designed clearance (about 4 mm) between static and rotating parts [for reason(s) still to be established].

## CONCLUSIONS

This paper has attempted to cover all aspects of gearbox design and manufacture that, from recent naval experience, have been shown to be important for the achievement of high reliability and availability in service. It follows that, where no mention is made of a particular topic, no significant problem is known to exist. Having said this, it must be admitted that what is probably the single cause of the biggest loss in operational time has not been specifically mentioned, namely the difficulties still experienced with incorrectly fitted or poorly locked screwed fasteners (particularly on rotating assemblies), problems that have afflicted almost all designs of gearbox at some time or other. All of which only serves to emphasise the point that even the best designed gearbox is only as good as its quality of manufacture or maintainance.

In conclusion, it must be said that the opinions expressed in this paper are those of the author and do not necessarily reflect RN policy.

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# Discussion

**I. T. YOUNG:** I have always had a great respect for Lloyd's Rules. Since Tom's revision in 1974, I have regarded the gearing rules in particular as the most straightforward and reliable from any Classification Society and a great improvement on the previous edition. They retain Lloyd's traditional simplicity and openness of approach, while conforming to the international trend in gear rating calculation.

A further revision was bound to come, and Messrs Pomeroy and Koller have presented the new approach very clearly. I am glad that they have aimed at simplification of the ISO formulae, where factors for the marine application and the marine diesel application in particular are likely to have little variation.

One regret I have is that, while they are mentioned, British Standards are largely ignored and the ISO equivalents are preferred, even when the BS has clear advantages. This must be a reflection of the fact that Lloyd's clientelle derives more and more from abroad, for obvious but nonetheless regrettable reasons.

BS 436 Part 3: 1986 is comparatively new, and it is possibly no wonder that Lloyd's have chosen to ignore it and go back instead to the ISO recommendation for their basis of gear rating. When accuracy has to be considered, however, it is lamentable that the authors choose to ignore BS 1807: 1981 and refer instead to ISO Q numbers.

ISO 1328: 1975 lists a framework of permissible errosion in a series of quality (Q) grades, each representative of consistent production from the gear cutting or finishing machine. These bear little relationship to the requirements of the gear in service. Allowable profile error, for example, increases with wheel diameter, whereas from the function point of view it ought to be constant for the mesh and specific loading considered.

BS 1807, the specification for marine main propulsion gears and similar drives, which has existed and been widely repeated since the early 1950s, was completely revised in 1981 and its framework was specifically designed to meet the needs of the marine gear in service. Lloyd's were invited to take part in the work, but unfortunately declined. The Ministry of Defence cooperated euthusiastically and I am glad to see from Mr Cooper's paper that they retain their confidence in the outcome.

Table DI lists some of the unique features of this British Standard. It recognises that marine gears are normally supplied as matched sets. As Mr Yates points out, it is the relative error between pinion and wheel that matters, not the arbtrary precision of each element.

The relative helix error, or to use the old term the 'tooth opening' at no load, is most important and for large gears cannot be sufficently accurately established by measurement of individual gears. BS 1807 recognises the need to match pinion to wheel to ensure that the best meshing conditions are achieved.

**Table DI: Unique features of BS 1807: 1981**

1. A simple basic grade for merchant marine. No further specification required
2. Gears specified as matching pairs
3. Matching errors specified for profile and helix alignment
4. Profile tolerance independent of diameter
5. Helix alignment tolerance independent of face width
6. Relaxation on cumulative pitch error for slower speed gears
7. Pitch error matching limits for high speed double helical gears
8. Special consideration given to the needs of straight spur gears
9. The following are covered:
   Undulations
   Surface texture
   Backlash
   Journal accuracy and alignment
   Bearing clearance variation
   Meshing at no load and at full power

I would urge Messrs Pomeroy and Koller to look again at the effect of accuracy. The incorporation of Q numbers in the ISO formulae gives only the roughest approximation, based on the defects of the ISO 1328 system. Returning to first principles based on the BS 1807 approach on the errors themselves will surely give a more realistic result. Ideally I could wish that Lloyd's would simply specify 'Gears shall be to BS 1807 accuracy' and thus do away with the need to incorporate an accuracy component into their gear rating formulae. Let me assure them that this will not disadvantage Lloyd's internationally. Gearing suppliers have no hesitation in working to any reputable Standard provided that it gains them another order in an increasingly competitive market!

**H. COCKING** (Westland Helicopters Ltd): At last it seems we have a new British Standard 436 for the gearing industry, which I am sure will be welcomed by all gear designers. It is an excellent standard which covers most of the factors involved in gear design. I say most, because unless I have overlooked them there appear to be two factors not covered which experience would make me a little tentative in its use.

First, if I understand correctly, the value for $\sigma_{fo}$ (the rotating bending endurance limit) is obtained in air at ambient temperature, which is considerably different from the environment within a gearbox. Gearbox conditions are oil/oil mist atmospheres at temperatures usually above 50 °C and often above 80 °C. The factor $\sigma_{fo}$ should therefore be obtained under these conditions or two additional factors introduced

One factor (say $Y_t$) to represent the reduction in $\sigma_{fo}$ due to the elevated temperature effect and what is probably more important a second factor (say $Y_o$) to represent the reduction in $\sigma_{fo}$ due to steel being in contact with the lubricating oil. This latter factor varies widely from one oil to another.

I would like to ask if any attention has been paid to these two points, and if so what strength reduction factors were obtained

**R. B. SIGGERS** (Lloyd's Register of Shipping): The papers were read with great interest. On page 9 I wonder if a translation error has occured on Fig. 8, which indicates that ME, MQ, ML quality levels are related to alloying content in the right hand zone, ie high, medium and low alloy steel, whereas they are in fact related to alloy steel quality.

Lloyd's Register of Shipping appears to come in for a little gentle criticism, at which I should like briefly to look. On page 6 it is implied that $K_A$, $K_V$, $K_{H\beta}$ $K_{F\beta}$, $K_{H\alpha}$ and $K_{F\alpha}$ are not in the Society's Rules, but this is not entirely so.

The load application ($K_A$) is $K_2$ and $Y_1$ in the Rules, dynamic ($K_V$) appears as part of an expression for $Z_V{}^2/K_V$ in $5850 + d_1N/3040\ 000$, the longitudinal load distribution $K_{H\beta}$ is $K_3$, the longitudinal load distribution $K_{F\beta}$ is $Y_2$, the transverse load distribution $K_{H\alpha}$ and the transverse load distribution $K_{F\alpha}$ are given the value of unity and a layer safety factor is provided.

On page 12 there is an opinion on the apparently low ratings assigned by LRS. As an exercise we made several assumptions regarding the prime mover of gear 'B', its manufacturing accuracy etc. and found that the LRS ratings are in fact about 32% higher than shown. This is still more conservative than the DIN rating, but our factors of safety do in fact fall just within the ranges for high reliability gears quoted in Table III, page 9.

The input rating of 5874 kW at 1500 rev/min for a single engine (which was one of our assumptions) is also very rare.

Regarding BS 436 itself, it seems to be a great improvement on the old BS 436 (1940), although it still does not guide designers of shrunk gear rims as to how the extra shrink stresses should reduce permissible dynamic root bending stresses

The I. Mech. E. Gearing Working Party received from LRS in 1983 some data showing the sad decline of merchant marine gearing in the UK in 1982. The recent review to 1985 (see Figs D1 and D2) does not show any improvement which could lead one to the view that BS 436 unfortunately has little application in the marine field.

Mr Yates, in his paper, supports the concept of IACS Unified Requirements for gears, a move which could reduce the number of Authorities he has to satisfy.

LRS also examined the available routes when looking to update the Rules. Being international in operation it was clear that ISO was a sound choice.

Thus the paper by Messrs Pomeroy and Koller is the result of a solo effort by LRS, which felt that in 1982 it could not await developments at BS, ISO, DIN, IACS and CIMAC to provide up to date marine standards for consideration.

The IACS work is only just started and if it follows the rough path taken by a similar complex component, *ie* a crankshaft, it will take about 6 years. It should be remembered that the current full membership of IACS comprises nine Societies (ABS, BV, GL, LRS, NK, DNV, PRS, RI and RRS).

**J. D. SMITH** (University of Cambridge): Mr Cooper is to be congratulated on such a clear and comprehensive review of the RN work. The use of thrust cones is an interesting development as single helicals with thrust cones should theoretically give lower distortions than double helical gears but with the corresponding penalty of slightly higher noise levels.

Transmission error measurment is a field in which considerable improvements have occured recently. It is now possible to measure to better than an accuracy of a tenth of a second of arc, allowing satisfactory measurment up to 4 m diameter for either gears in mesh or the manufacturing machine tools. Measurement under full load is achieved routinely at speeds up to 1800 rev/min at the moment and developments in 1986 have extended speeds up to 6000 rev/min. The practical mechanical limits on frequency response have risen to about 1400 Hz, allowing measurements of tooth frequency effects for a 24 tooth pinion up to over 3000 rev/min.

Prediction of transmission error from the basic involute, helix, pitch and casing measurements is a tempting possibility but although computer programs already exist, this is not likely to be a realistic option. The inherent problem lies in the accuracies involved since the uncertainty in each of the involute, helix, pitch or axes alignment is greater than the desired accuracy of tranmission error.

Current trends are for increasing accuracy with checking of bedding and transmission error being necessary to cover both strength and noise requirements

**I. M. MATHIESON** (Yarrow Shipbuilders Ltd): The main peak velocity used on bearing caps of 3.8 mm/s is accepted as being a rough empirical guide based on experience which seems to work. Does Mr Cooper think that it is necessary to have a more refined method of acceptance criteria for high-speed line vibration? I am aware of bearings which have run successfully above this figure.

I would throughly agree with Mr Cooper's statement on the preference of friction clutches for diesel drive as against SSS clutches. Many years ago one of our subcontractors used an SSS clutch on the diesel drive line to the gearbox for a prototype ship with disatrous consequences. The high diesel torque fluctuations at low speed when accelerating the transmission to idling combined with the presence of low-frequency torsional criticals led to severe gear hammer and general shock loading on the transmission. Furtunately, this was rectified fairly easily by installing a friction clutch and

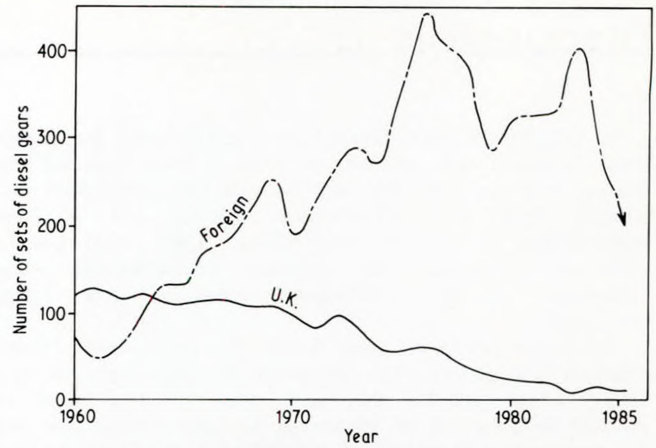


**FIG. D1: Number of sets of UK and foreign diesel gears for the period 1960-1985**

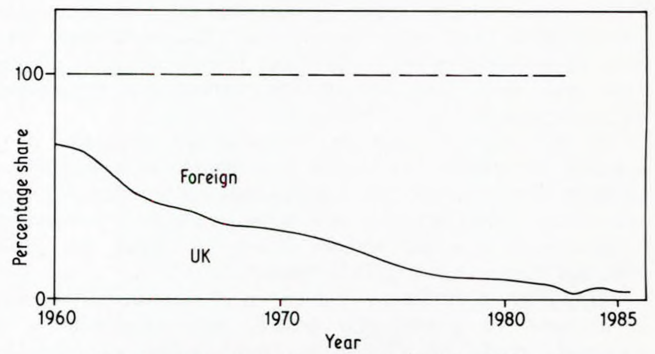


**FIG. D2: Percentage share of market for UK and foreign manufacturers**

bringing the engine up to idling speed by means of the slipping clutch routine, thus avoiding direct coupling between the excitation and the resonances in the transmission system.

Should 'collars' be 'collets' in Fig. 7 of Mr Cooper's paper? I should also like to ask about MOD views on hard-on-soft thrust cones versus hard-on-hard for main propulsion gearboxes, and how do MOD see the future of the application of thrust cones instead of thrust bearings? Finally, could Mr Cooper please comment on the 'state-of-the-art' regarding the use of non-flammable oils in RN gearboxes?

# Authors' replies

## D. A. Hofmann

In reply to Mr Young, I would agree that BS 1807 is quite a good application standard. However, for the production engineer, the specification of gear tolerances to BS 436 (1970) or ISO accuracy grades is more meaningful, since the required accuracy grade will indicate clearly the probable manufacturing difficulty. Personally, I cannot see why there should not be cross-referencing between an application standard such as BS 1807, which requires certain tolerances for functional reasons, and a primary gear tolerance standard such as BS 436 (1970). The required lead, pitch and profile accuracy can each be given in terms of the equivalent BS 436 grade, in the same way as different IT tolerance classes are used in general component tolerancing.

Mr Young is, of course, absolutely correct that 'tooth-opening' at no load is most important, to ensure good tooth contact and a uniform load distribution at running torque.

The required tooth opening can be easily calculated, being the sum of the torsional and bending deflection primarily of the pinion (see for example MAAG Gear Handbook). In most cases, the required 'tooth-opening' will be very much larger than the lead tolerances, and it will be necessary to provide lead correction on the pinion.

Mr Siggers is absolutely correct that the ISO/DIN standard provides for three levels of material quality, ME, MO and ML, for each of the basic material types. However, for the most common materials, the carburising steels, it differentiates, for root bending strength only, between high, medium and low alloy carburising steels of MQ quality. This is correctly shown in Fig. 8.

It was not intended to imply that LRS ratings did not take account of $K_A$, $K_V$, $K_{H\beta}$, $K_{F\beta}$, $K_H$ and $K_F$ (pp. 6-8). Mr Siggers rightly points out that $K_A = K_2$ and $Y_1$, in the LRS rules, and that the other factors are accounted for in some form or other in the LRS rules. The significant difference is surely that in the DIN/ISO and new BS gear standard it is possible to quantify these load factors with some confidence as a direct function of gear accuracy and gear set inertia ($K_V$) and as a function of actual face misalignment, ($K_{H\beta}$, $K_H$ etc).

I cannot comment on the apparent discrepancy between Mr Siggers' calculations and my calculations for the LRS gear rating as shown in Table VI on p. 12. The ratings shown are based on the following factors:

| | | |
|---|---|---|
| $K_1$ | $= 0.34$ | $Y_3 = 1.0$ |
| $K_2 = Y_1$ | $= 1.0$ | $Y_4 = 33.97$ |
| $K_3 = Y_2$ | $= 0.96$ | $Y_5 = 105.0$ |
| $B = \varepsilon_\alpha$ | $= 1.626$ | $Y_6 = 1.0$ |
| $\sigma_s$ | $= 60.0$ | $\sigma_b = 3500.0$ |

## G. C. Mudd and J. M. France

In reply to Mr Cocking the effect of temperature on the rotating bending endurance limit is negligible in the range of ambient temperatures experienced in gear units. A factor, $K_T$, is included in AGMA 218.01, but its value is unity for oil or blank temperatures up to 120 °C. BS, ISO and DIN have all chosen to leave out this factor.

The effect of oil on pitting and scuffing is well known, but I have never heard of normal gear lubricants having any effect on the bending strength. Tests have been performed at DBGI to determine the effect of water contamination in a typical ISO 320 EP gear oil. Even with 10% water in the oil there was no discernible difference in the failure load. It is difficult to imagine then that a sensibly chosen lubricant (eg non-corrosive) will reduce the bending endurance limit.

## R. V. Pomeroy and P. Koller

First we would like to thank all those who have contributed to the discussion.

In reply to Mr Young, his support of the simplified ISO approach presented for a future LRS gearing rule revision is naturally welcomed and of course greatly valued.

A strong case is made for the adoption of BS 1807 (1981) and in this respect Mr Young is in fact preaching to the converted. Unfortunately whilst appreciating the merits of this standard it cannot, as it stands, be fitted into the ISO calculation methods without the loss of an important feature. That is namely the recognition that quality of manufacture does vary and that account should be taken of good quality and also not so good quality when determining maximum permissible gear loadings (see Fig. 4).

BS 1807 defines for merchant marine main propulsion gears one quality level for any design based on the pitch line speed. As has been pointed out ISO 1328 has shortcomings as regards tolerances for matching pinion(s) and wheel and surface texture which BS 1807 covers. Perhaps ideally a combination of the two standards would be the answer.

To tackle some of Mr Young's more detailed points, we agree that profile deviations should only be considered as a function of the module and in this respect BS 1807 is in agreement with DIN 3962. Incidently for the purposes of the ISO calculation methods it is generally accepted that where the DIN quality grade is given:

$$\text{ISO (Q)} \approx \text{DIN (Q)} - 1$$

It should perhaps be said that measuring equipment is available to determine the tooth flank alignment deviation of individual gears, certainly up to 2 m diameter, with a typical accuracy of between 2 and 3 μm. Larger gears can be measured when set up on a grinding machine and here facewidths up to 1100 mm can be checked with a typical accuracy of 4 μm for a 4 m diameter gear. Furthermore it seems reasonable to consider the total tooth flank alignment deviation for a gear, which includes the helix angle deviation, as a function of the facewidth. ISO 1328 and DIN 3962 specify $F_\beta$ for individual gears in this way. BS 1807, as Mr Young points out, does not specify $F_\beta$ for individual gears but the total relative alignment error of pinion and wheel teeth over 80% of the facewidth. It is agreed that this can reasonably be considered as independent of the facewidth.

The matching of pinions and wheels is clearly very important and in this respect BS 1807 provides for marine main propulsion gears the most comprehensive guidance.

Finally, we can report that LRS is now taking a far more active role in BS work in this field and has recently attended at ISO.

## B. C. Cooper

In response to Mr Mathieson's questions, the specified vibration limit was established from trials of a *Type 42* gearbox during which bearing cap vibration was measured for varying degrees of rotor unbalance. The figure of 3.8 mm/s peak velcocity at rotational frequency was adapted as a realistic level for the Olympus high-speed line, and it is a value which relates to one particular installations and cannot be applied generally without due regard for design differences and particular performance requirements. Having said this, experience with other ship classes has shown it to be a good standard value. (An exception is the Tyne clutch input bearing in the *Type 42, 21* and *22* classes which is more flexibly supported, resulting in much higher vibration levels for a given out of balance force, where a limit of 25 mm/s is set.)

As MoD has no direct experience of the use of thrust cones in geaboxes and there is little published information on the subject, it is not possible to make any categoric statements on material selection. In general, the criteria to be considered are similar to those for gear elements and it would seem reasonable to use those material combinations that would be chosen for a gear contact of comparable loading for which soft-on-soft, hard-on-hard or hard-on-soft might all be seen as acceptable. In fact soft-on-soft has been shown to be inadvisable because of the way damage can transfer between cone surfaces in the event of 'dirt' entering the mesh. However, no objection can be seen to either of the other combinations.

The use of the term 'collet' rather than 'collar' in Fig. 7 would probably be more accurate, although I am quite happy with either.

Non-flamable lubricating oils were investigated by the MoD Gearbox Explosion Working Party who concluded that none of the products then available looked sufficiently attractive to warrant adoption for naval gearboxes. The situation has recently been reassessed but most of the problems, including thoses of compatibility with seal materials and high initial cost, are still to be resolved. The subject will remain under review.

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