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A REVIEW OF MARINE GEARBOX EXPLOSIONS

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THE KOOTENAY GEARBOX EXPLOSION

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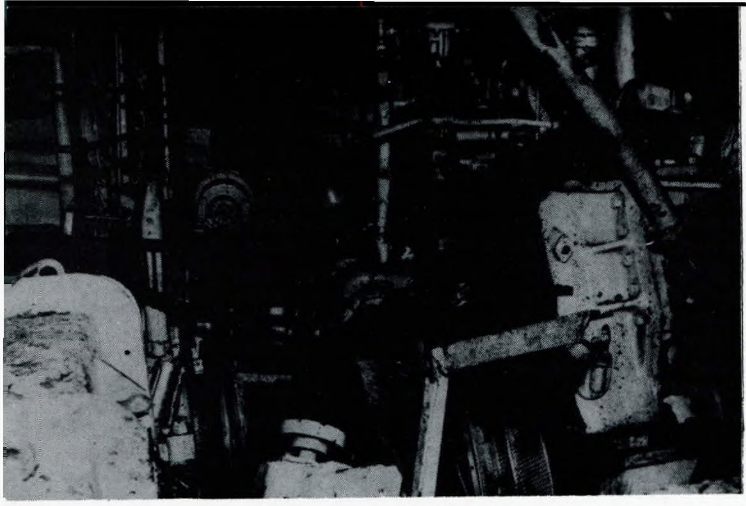

Fig. 1 Gearcase of the 'Seatrain New York'

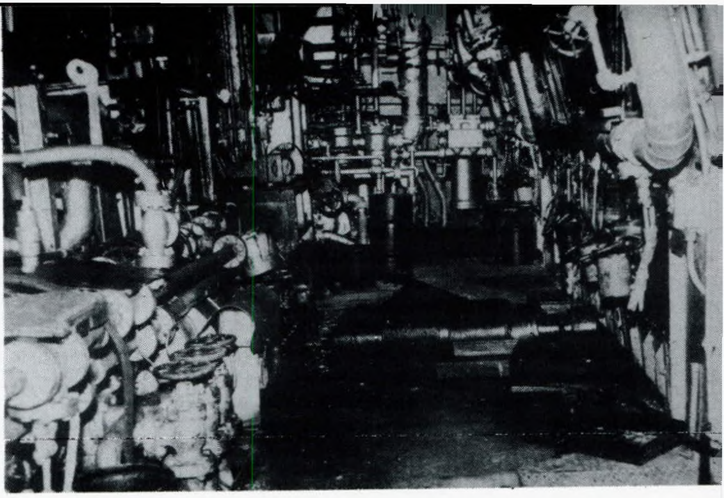

Fig. 2 Fireroom of the 'Seatrain New York'

A Review of Marine Gearbox Explosions

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SYNOPSIS

This is a report on the activities of the Gearbox Explosion Working Party set up in 1970 by the Ministry of Defence, as a result of two major gearbox explosions in naval installations. Twenty incidents are reviewed and causes identified. Investigations of the atmosphere within the gearcase are described and means of limiting subsequent damage examined, as are warning and corrective devices. The conclusion and recommendations of the Working Party are presented.

1. INTRODUCTION

1.1 In October 1969 a major gearbox explosion occurred in the Canadian frigate *Kootenay.* In June 1970 an explosion occurred in the gearbox of a shore test set for a Royal Navy machinery installation.

1.2 Consequently, in July 1970, the Director of Engineering of the Ship Department, Ministry of Defence, set up a Working Party with the following terms of reference:

1.2.1 To investigate previous instances of explosions in gearboxes.

1.2.2. To investigate the causes and mechanisms of such explosions.

1.2.3. To recommend lines of investigations to be carried out to reduce the future risk of an explosion in a gearbox.

1.3 After the first few meetings the membership consisted of representatives of the three major marine gear manufacturers in the UK: David Brown Gear Industries, GEC Marine and Industrial Gears, and Vickers Shipbuilding Group; of the Admiralty Oil Laboratory (now NGTE, Cobham) and Admiralty Materials Laboratory (now AMTE, Holton Heath); of the gearing and scientific advisory sections of the Ship Department; and of Y-ARD. The Gearing Section in the Canadian Defence HQ was kept informed on the progress of the investigations and co-operated in providing information and comment.

1.4 Helpful co-operation was received from the staff of the Safety in Mines Research Establishment at Buxton, Derbyshire and Lloyd's Register of Shipping, and from various individuals in shipping and industrial companies, Universities and other research establishments and in MOD (PE).

1.5 The initial investigations and preliminary conclusions of the Working Party, based on work to end of 1972, were presented in an Interim Report in the Spring of 1973. The final report was published in January, 1980. Most of the work presented in this paper was published in classified reports during the lifetime of the Working Party, and progress to that date was presented to the World Gearing Congress in Paris in June 1977.[1]

1.6 The Working Party initiated a number of activities in parallel and at times could not await the confirmed outcome of one investigation before starting the ensuing one. We have the benefit of hindsight in attempting to present a logical sequence which could not always be followed at the time.

Successive sections of this paper analyse the causes of recorded gearbox explosions and, having identified the predominance of bearing failures, direct attention towards the design and installation of bearings. But why do such failures sometimes cause major explosions, sometimes minor explosions, and sometimes merely oil ignition without explosion?

2. INCIDENTS RECORDED

2.1 Despite intensive enquiries of many authorities in both the industrial and marine fields, relatively little evidence of the causes of gearbox explosions came to light, although it became clear that explosion incidents are not as infrequent as originally thought. Lloyd's Register of Shipping has reported only five gearbox incidents in the period from 1960 to 1978 inclusive.

2.2 Twenty explosion incidents in merchant, naval and industrial environments are tabulated in date order in Appendix 1 and summarised in Table 1. The basic criterion for inclusion as an incident is oil ignition inside the gearbox. Of the 20 incidents recorded, 12 led to minor explosions (those contained within the gearcase) and six led to major explosions involving violent rupture of the gearcase. In the other two incidents oil ignited but there was no subsequent explosion.

2.3 The major explosion in *Kootenay* and the non-explosive incidents in *Skeena, Chaudiere* and *Fraser* are discussed in D.K. Nicholson's companion paper.[2]

2.4 The major explosion in shore test facility in 1970 occurred when the gearbox combining the output of two steam turbines was operating with one turbine only on full power ahead. The main gearwheel's forward journal bearing failed because of a complex inter-relationship of many factors. These included the influence of static and dynamic forces emanating from the main shaft's flexible coupling; and the orientation of the journal attitude relative to the bearing joints and oil gutterways in this particular operating mode.

2.5 The explosion occurred after approximately one hour's running at full power, single-turbine operation. Prior to the explosion, no recorded data indicated impending failure or malfunction of the gearbox, no unusual noise was heard and the airborne noise was low.

Instrumentation fitted to each main gearwheel's journal bearing consisted of one Admiralty-type spring-loaded thermocouple contacting the back of the bearing shell of medium wall thickness. The point of contact was at the position of minimum film thickness for two turbines at full power ahead.

Throughout trials immediately prior to the explosion, the thermocouple in the forward main gearwheel journal bearing recorded a virtually constant temperature, below that of the lubricating oil supply, indicating a possibility that this thermocouple was open-circuited and reading cold junction temperature.

2.6 After the explosion, examination revealed that the forward main gearwheel's journal bearing had suffered a major wipe, exposing the steel backing, with evidence of steel on steel contact on the journal and bearing. The aft bearing showed no signs of wiping. Examination of the thermocouple in the forward bearing did not confirm that it was open-circuited, and when heat was applied to the whitemetal surface, the thermocouple responded; also, when check-calibrated, the thermocouple was apparently satisfactory. The recorded low temperature immediately prior to the explosion has not been fully explained, but it may be associated with the fact that its location was well removed from the maximum temperature position of the bearing under the particular operating condition, and that the form of the thermocouple created a large heat sink short circuit from junction to bearing housing.

2.7 At the time of the explosion an indentical prototype gearbox was also undergoing trials and had apparently operated successfully under similar single-turbine, full-power conditions, for four hours. Examination of this gearbox as a result of the explosion revealed that a minor wipe had occurred in the forward main gearwheel journal bearing, at the same position as the major wipe on that of the exploded gearbox.

2.8 The most serious of the five merchant marine incidents, in terms of loss of life and/or injury were on the *Verena* and *London Pioneer,* but it is interesting to note that there were no casualities in the *Seatrain New York,* despite the spectacular damage, shown in Figs 1 and 2, because machinery was being operated from a remote control room. By comparison, other incidents had been minor, and very little factual evidence was available as to the precise conditions existing at the time. The attitude of many of those concerned with these other incidents was that the malfunction causing the explosion had been readily obvious, design changes had been made and the matter forgotten.

2.9 A better documented case of oil ignition within the gearcase is that in HMS *Zulu* (Figs. 3 and 4) when a lightly loaded steam turbine jackshaft bearing failed. Popping noises and discharge of smoke through joints were reported, followed several minutes later by a heavy grinding noise. Flashing or sparking was also reported at some distance away from the bearing in distress, visible through clutch inspection sight ports.

2.10 An unusual cause was diagnosed for a minor explosion in 1968 during the shore trials of another Royal Navy machinery installation: overheating was caused by windage and churning in the astern fluid coupling when the cooling oil supply was interrupted while the coupling was being operated beyond design conditions. Subsequent inspection revealed no evidence of any rubbing.

2.11 Table I summarises the probable causes of the heat that led to ignition: no differentiation is made between a heat source igniting an already present flammable atmosphere and one that first creates the flammable atmosphere and then ignites it. Of the six major

Item	Date	Installation	Oil Ignition Inside Gearbox	Gearbox Explosion		External Fire	No. of Casualties		Probable Cause
				Minor	Major		Dead	Injured	
1	28FEB48	HMS DEVONSHIRE	YES	YES	NO	NO	NIL	NIL	Bearing and/or gear tooth failure
2	28NOV60	HMS HAMPSHIRE	YES	YES	NO	NO	NIL	NIL	Rub in manual clutch
3	MAR62	SS VERENA	YES	NO	YES	YES	6	NIL	Flexible coupling failure
4	21MAR63	HMS KENT	YES	YES	NO	NO	NIL	NIL	Rub in manual clutch
5	DEC64	SS SEATRAIN NEW YORK	YES	NO	YES	YES	NIL	NIL	Coupling bolt or gear rim failure
6	16SEP65	HMS HAMPSHIRE	YES	YES	NO	NO	NIL	NIL	Bearing failure following oil starvation due to maloperation
7	SEP65	SS MALMOHUS	YES	YES	NO	NO	NIL	NIL	Journal bearing failure
8	AUG66	ROLLING MILL	YES	YES	NO	NO	NIL	NIL	Tapered roller bearing failure
9	15FEB67	RFA REGENT	YES	YES	NO	NO	NIL	NIL	Overheated due to loose component fouling main turning gear clutch
10	1967	CEGB Tilbury 'B'	YES	YES	NO	NO	NIL	NIL	Thrust bearing failure
11	30MAY68	Naval Transmission Test Facility	YES	YES	NO	NO	NIL	NIL	Overheating in fluid coupling due to excessive windage in absence of cooling oil supply
12	23OCT69	HMCS KOOTENAY	YES	NO	YES	YES	9	53	Failure of thrust bearing on primary pinion line
13	30JUN70	Naval Machinery Test Facility	YES	NO	YES	YES	NIL	NIL	Journal bearing failure
14(a)	18SEP69	INDUSTRIAL	YES	YES	NO	NO	NIL	NIL	Tapered roller bearing failures
(b)	31JUL70	PUMP	YES	NO	NO	NO	NIL	NIL	
(c)	21AUG70	DRIVE	YES	YES	NO	NO	NIL	NIL	
15	16APR71	USN CALIENTE	YES	YES	NO	NO	NIL	NIL	Journal bearing failure
16	04JUN71	HMCS SKEENA	YES	NO	NO	NO	NIL	NIL	Journal bearing failure
17	08JUN71	HMCS CHAUDIERE	YES	NO	NO	NO	NIL	NIL	Locating thrust bearing failure
18	26OCT71	MV MAIHAR	YES	NO	YES	YES	NIL	NIL	Rub in vicinity of clutch
19	NOV71	HMS ZULU	YES	YES	NO	NO	NIL	NIL	Journal bearing failure
20	08DEC75	SS LONDON PIONEER	YES	NO	YES	YES	NIL	2	Flexible coupling failure

Table I Summary of gearbox explosion incidents

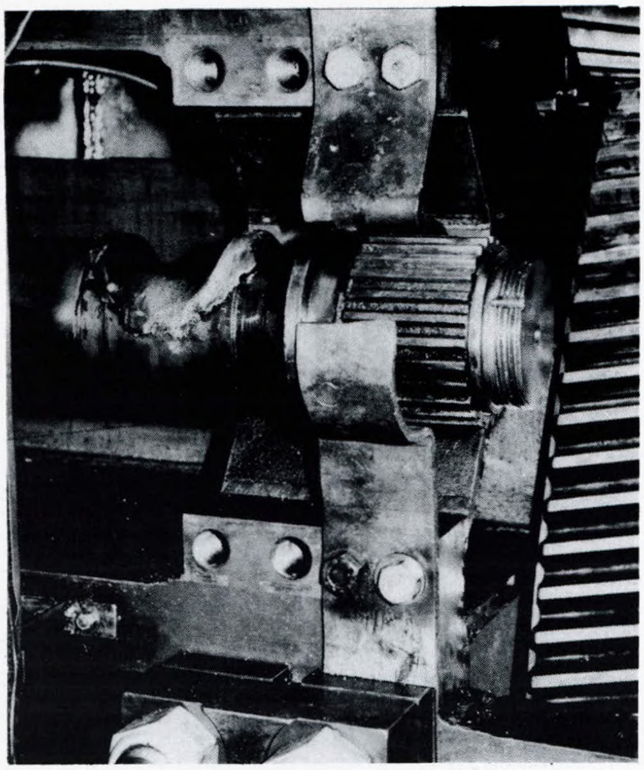

Fig. 3 Jackshaft bearing journal of 'HMS Zulu'

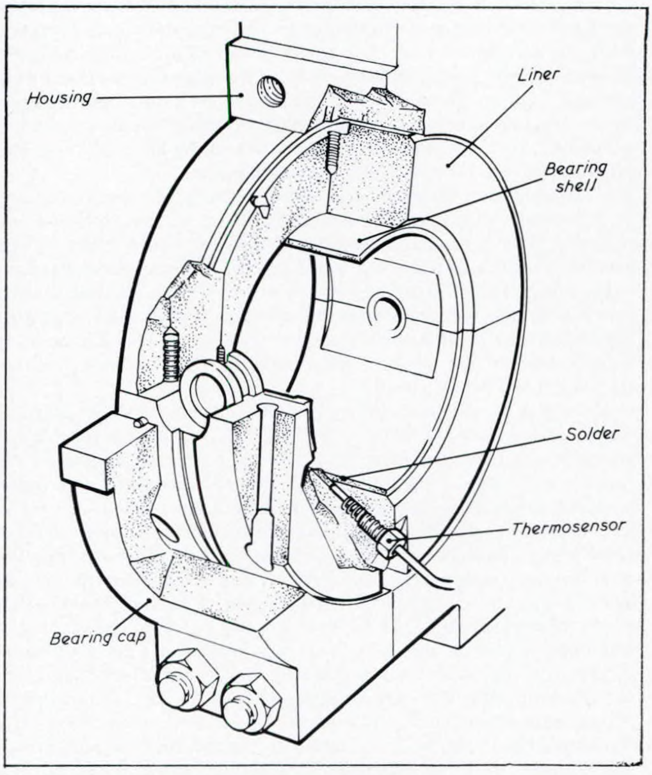

Fig. 5 Typical thermosensor arrangement

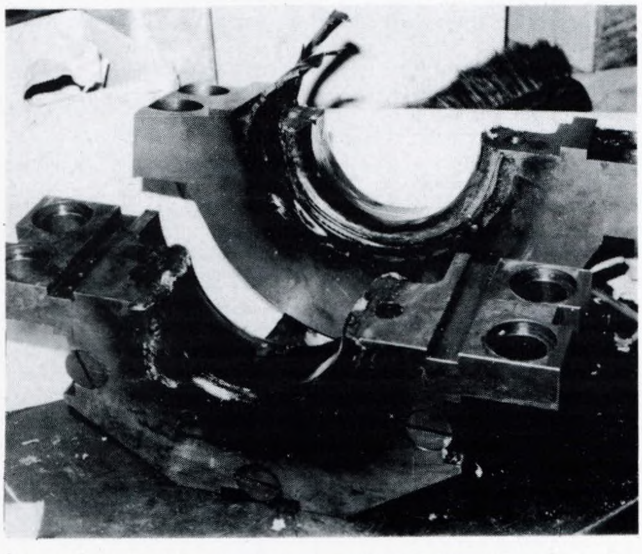

Fig. 4 Jackshaft bearing housing of 'HMS Zulu'

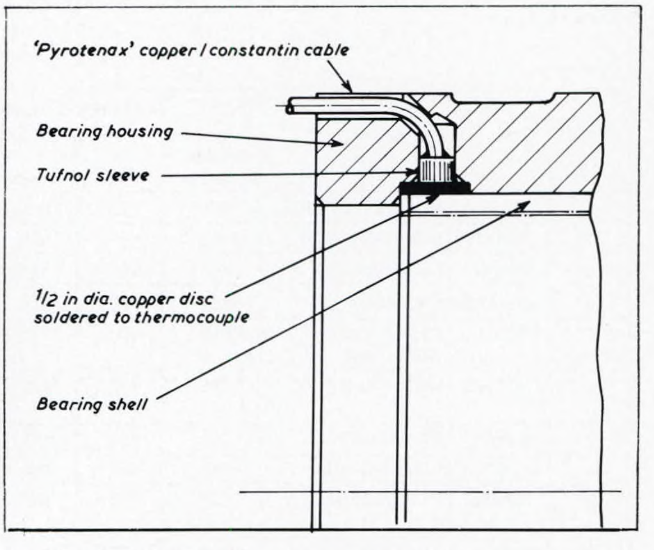

Fig. 6 Typical thermocouple arrangement

explosions, two were probably caused by bearing failure, one by rubbing of a clutch disc carrier and three by failure of mechanical components.

2.12 Of the 12 incidents involving minor explosions, eight were caused by bearing failure, three by rubbing and one by heating due to windage. The two incidents of oil ignition with no subsequent explosion are also associated with bearing failure. The design and installation of bearings are therefore worthy of attention.

3. BEARINGS

3.1 Detailed discussion of bearing design is outside the scope of this paper but, as the causes of recorded oil ignition incidents include wrongly assembled bearings and incomplete recognition of not designed for, and abnormal, operating states, the Working Party recommended that gearing detail design be subjected to an independent audit in respect of calculations and proposals for journal and thrust bearings and shaft seal arrangements. In particular, journal bearing design reviews should include checks of:

3.1.1 position of bearing load lines and lines of minimum oil film thickness under all modes of operation of the prime movers, including no load.

3.1.2 location of oil inlets and line of bearing split.

3.1.3 location, number and type of bearing sensors.

3.1.4 method of location of bearing shells to ensure that bearings cannot be assembled incorrectly, either on initial build, or after opening up for maintenance.

It also recommended that detailed and thorough analysis of the system's dynamic performance be undertaken in the design stage with the object of identifying the steady and transient conditions on all

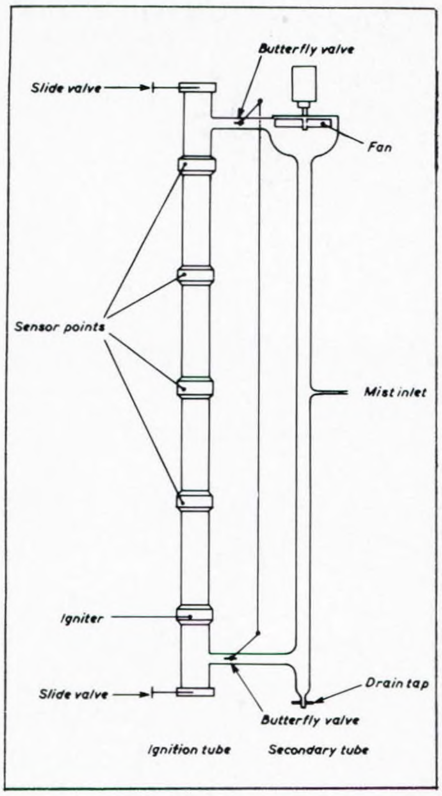

Fig. 7 Mist flammability apparatus

thrust and journal bearings. This should be followed by practical measurement at the prototype trials stage, with further analysis and measurement of any subsequent modifications.

3.2 Some gearboxes in which bearing failure was the probable cause of oil ignition were fitted with bearing temperature indication, but in some of the incidents reported there was no warning of rapid temperature rise.

3.3 The Royal Navy's surface ships of the 1960s used steel shell bearings with medium-thick walls in the gearing with thermocouples spring loaded against the back of the shells. This method introduced a longer path between the surface of the bearing whitemetal and the sensing device than had existed with earlier methods although the introduction of pre-insulated wiring greatly improved the overall reliability of the system compared with earlier ones subject to oil penetration.

However, it is clear from some of the gearbox incidents investigated that there have been cases of low temperature readings due to poor contact on the back of the bearing shells. In the 1970s the Royal Navy changed in some installations to resistance thermometers with the thermosensor spring-loaded to the back of the bearing shell by an arrangement which was dimensionally interchangeable with that of the earlier ships. Fig. 5 illustrates a typical thermosensor arrangement.

3.4 Royal Navy submarines have retained thermocouples in journal and thrust bearings for temperature monitoring. Where thin or medium thick shell bearings are used, the thermocouple junction is made into the form of a disc, illustrated in Fig. 6, rather than the more usual pencil-like probe, and press fitted into the housing so that there is a positive contact with the back of the bearing shell. In journal bearings of the thick-wall, conventional type, the bearing sleeve is drilled to within, or close to, the whitemetal, and a thermocouple having a small junction is secured by a screwed arrangement.

3.5 Propulsion machinery with more than one independent prime mover per shaft permits multiple modes of operation and hence more than one load line, and line of minimum oil film thickness. This makes it difficult to select a bearing oil inlet position and also makes it necessary for the gear designer to choose one particular operating condition for which the thermocouple or electric resistance thermometer will be correctly placed.

Usually the full-power condition is chosen and this means that the response at the cruise, or other, conditions is less than ideal. Current policy is to provide temperature measurement for bearing attitudes at all the major operating modes. There is, moreover, a trend to some redundancy by having additional sensors axially displaced along the bearing, thereby avoiding the need to open up the gearbox just to replace a sensor which has failed; perhaps due to damage caused by previous opening up.

3.6 Thus, in modern naval gearboxes, there is a marked increase in the number of temperature measurements, calling for sophisticated systems to sweep continuously the many measuring points and compare the results with preset values so that warnings or alarms may be initiated when deviations occur.

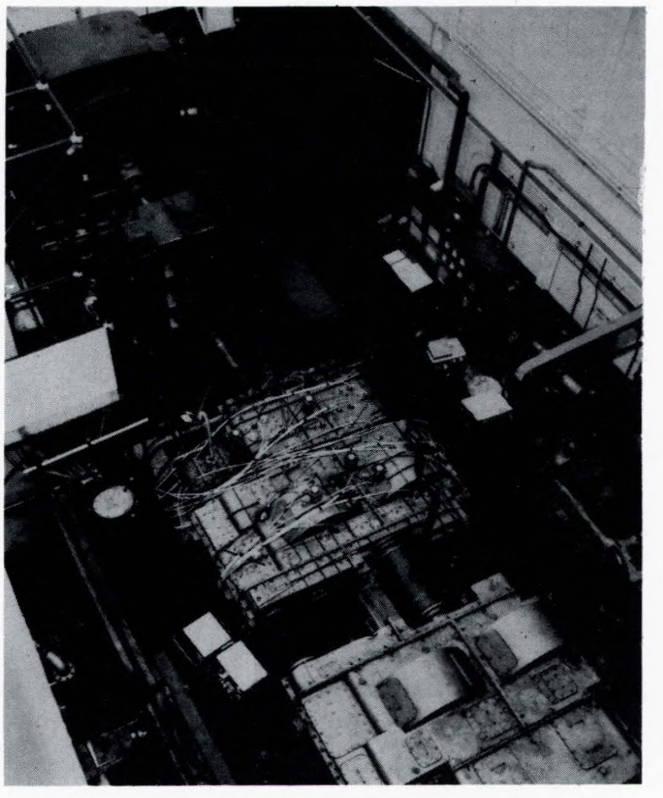

Fig. 8 Marine gearbox atmosphere test rig

4. FLAMMABILITY OF GEARBOX ATMOSPHERES

4.1 Two criteria must be satisfied for the atmosphere inside a gearbox to ignite: the concentration of combustible must lie within its flammable range, and an ignition source must be available. Whenever an ignition has been observed in a gearbox it has been possible to discover a malfunction which could have produced heat. This suggests that either the heat source ignites an existing flammable mixture or it creates such a mixture and then ignites it.

4.2 Mineral oil is not highly flammable and, unless heated well above normal ambient temperature, will not burn in air. However, in certain circumstances flammable mixtures in air can form and an investigation was conducted to study these circumstances.

4.3 Two approaches were used for this investigation: first to make measurements on the atmospheres inside running gearboxes to arrive at the "normal" concentration of combustible material; second to make measurements in the laboratory to study the way in which lubricating oil can be made to burn and to relate this with possible practical gearbox situations.

4.4 Preliminary work sampling gearbox atmospheres on board ship and at shore installations showed that gearboxes running without malfunction contained only small amounts of combustible materials in their atmospheres. This is not to say that oil drops are not plentiful but that the large droplets (which are present in abundance) are not easily ignited.

The other effects of large droplets and their importance were realised later as described.

Measurements of vapour content served to confirm that this is low and related to the expected vapour pressure at the temperature in question.

4.5 In the laboratory, many attempts were made to produce a flammable atmosphere from lubricating oil by generating mist. Various spray nozzles were tried and other mechanical means of mist production, but in no case could a flammable atmosphere be produced. This is not to say that the oil could not be burnt as in a pressure jet burner but this was not considered to be relevant to the interior of a gearbox. When, however, a thermal method of droplet generation was tried, flammable atmospheres were easily obtained.

4.6 Mist was generated in a zone hotter than 400°C. When oil came into the hot zone it evaporated and then recondensed as a mist, with a droplet size of about 3 μm diameter. The condensed mist was substantially at room temperature and optically very dense and persistent in still air.

First, the composition of the oil in the mist was measured to determine any changes that had taken place during its formation. No oxidation or thermal breakdown products were detected. Using the apparatus shown in Fig. 7, the lower flammable concentration limit was then measured and found to be about 48-52 x 10^{-3} kg/m^3. The result did not vary beyond this range for a number of oils of differing viscosity and additive content.

4.7. Using the knowledge gained in these experiments, trials were conducted on a Y100 (Whitby class) gearbox, driven by an electric motor. This is shown in Fig. 8. These trials are described in detail elsewhere [2]. Their objects were: to make a thorough survey of the gearbox atmosphere in terms of mist and temperature; and to inject thermally produced mist into the inerted gearbox to determine its distribution within the gearbox. Mist detectors were fitted to the gearbox at 15 positions so that all the most likely portions of the interior were monitored.

The results confirmed those obtained in earlier trials on other gearboxes, namely that mist contents in normal running were very low, well below those necessary to form a flammable atmosphere. In the second part of the trials, two specially built mist generators were used: each dissipated 12kW and together they produced 0.4m^3/min.

The gearbox was inerted with nitrogen during these experiments. When the gearbox was stationary, with the lubricating system operating normally, the mist detectors registered about 12 x 10^{-3}kg/m^3 when mist was injected. When the gearbox motor was started, the mist content fell almost immediately to the values obtained without mist injection.

Figure 9 shows a mist detector read-out during this period. Note the rapid change when, after one minute, the gearbox was started. This effect was repeatable and was undoubtedly due to the large oil drops absorbing the mist droplets and so removing them from suspension.

4.8 Further laboratory work was done to demonstrate the effect of large drops on the mist. A model of the Y100 gearbox was built to one quarter scale and employing a double-reduction system sketched in Fig. 10. The gear surface speed was variable and included typical Y100 surface speeds. This was fitted with mist detector and a mist generator capable of filling the gearbox with flammable mist while it was stationary.

The same procedure was adopted as with the full scale Y100 tests, ie the gearbox, with lubricating system operating, was filled with the mist to the flammable limit and then the motor was switched on. The time taken for the mist concentration to fall to 10 per cent of the lower flammable limit was noted. The experiments were repeated at different gearbox speeds and it was remarkable that the mist was removed efficiently at speeds down to about one third of full speed.

Thus, in the practical situation, if an abnormal bearing temperature were observed, or some other hot element that could give rise to oil mist, the results above suggest that the safest course for the operator is to reduce power but maintain sufficient gear speed to preserve the valuable "scrubbing" effect of the large droplets.

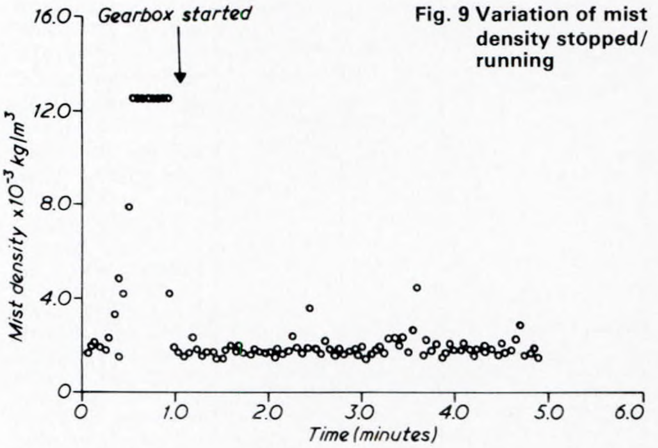

Fig. 9 Variation of mist density stopped/running

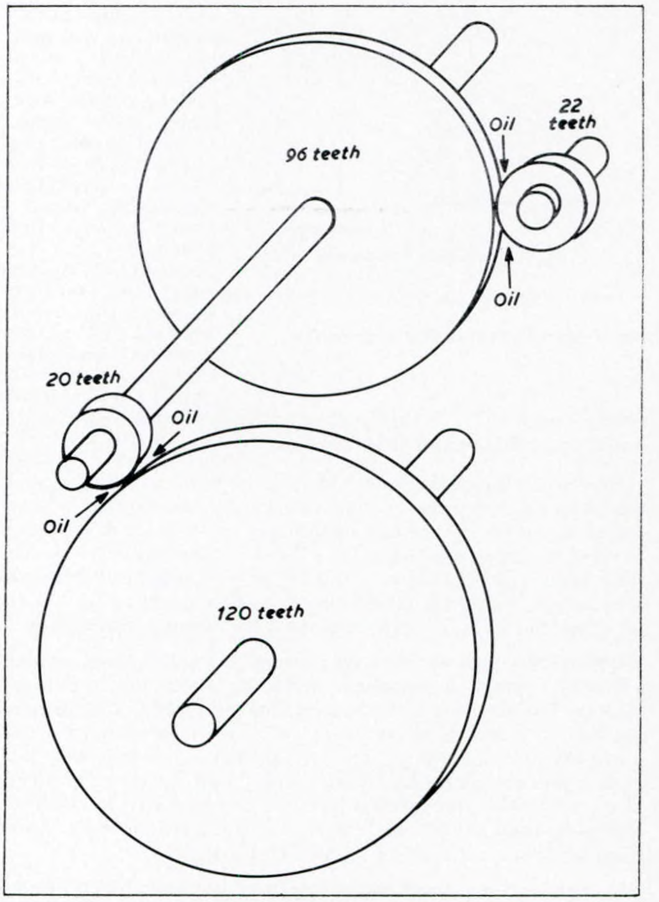

Fig. 10 Arrangement of miniature gearbox

5. FLAME SUPPRESSION, INERTING CONTAINMENT AND RELIEF

5.1 However successful the efforts to eliminate the causes, the possibility of oil ignition must still be acknowledged. The potential defences against consequential disintegration of the gearcase are: to suppress any flame in the immediate vicinity of the ignition; to blanket any flammable atmosphere with inert gas; to contain any explosion within the gearcase; to relieve any rise in gearbox pressure.

5.2 The Working Party sponsored the investigation and tests of flame arresters. These devices were intended to contain any flame generated by a failed bearing within the housing and prevent its spread through the gearbox. An open-celled reticulate foam made from a nickel-chromium alloy was tested and proved effective provided that the clearance between the shafting and fixed arrester wall was 1mm or less. Such a clearance, however, increases the possibility of rubbing between arrester and shaft. It was concluded that, on balance, it would be unwise to fit flame arresters in naval gearboxes operating under normal conditions for fear of doing more harm than good.

5.3 A signal from a bearing temperature or oil mist monitoring device could be used to trigger inert gas injection. The choice between nitrogen and carbon dioxide as an inerting medium depends on three factors.

5.3.1. *Density*: carbon dioxide is about 1.5 times more dense than air, whereas nitrogen has almost the same density; a nitrogen-air mixture would be more stable.

5.3.2. *Storage*: carbon dioxide can be stored as a liquid but storage of liquid nitrogen is more difficult because of its low boiling point. The volume of a nitrogen storage system would be about four times larger than that for the same mass of carbon dioxide.

5.3.3. *Safety*: expanding carbon dioxide produces dry ice crystals which carry electrostatic charges.

Inert gas trials sponsored by the Working Party demonstrated that nitrogen and carbon dioxide were equally effective when used at the same mass flow rate. Hence there is a slight preference for nitrogen

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D.McNeill, BSc, ARCST served a naval dockyard apprenticeship before his national service as a Sub-Lieutenant (E) in the Royal Navy (1945-1948). He studied mechanical engineering at university and gained experience in the electricity supply industry. In 1952 he joined Rolls Royce and specialised in reduction gearing and bearings. In 1954 he joined Y-ARD and in 1968 became manager of the Main Propulsion Machinery Group. Mr. McNeill was on the BSI committee on Mechanical Balancing of Rotating Machinery, and served on the MOD Gearbox Explosion Working Party.

as an inerting, as distinct from a firefighting, medium because of the worries about static charges.

5.4 There remains the problem of choosing the siting and the setting of the triggering signals so as to ensure sufficiently rapid inerting action without too frequent spurious initiation. The Working Party recommended that inert gas injection systems should not under normal circumstances be fitted to naval gearboxes.

5.5 It can be calculated that the maximum pressure generated by a major explosion is of the order of 0.95-1.035 MPa (140-150 lbf/in^2). In the past, gearcase covers have been designed purely as oil-containment skins, and the yield strength of the weakest members of typical naval gearcase covers is of the order of 20 to 34 kPa (3 to 5 lbf/in^2). The weight penalty associated with any attempt to strengthen the gearcase to contain the maximum pressure would be unacceptable. Thus, some form of relief system is required to moderate an explosion should it occur.

5.6 It is common practice in diesels to fit crankcase explosion relief valves. Research work at BICERI has indicated a desirable relief area of $0.685 m^2/m^3$ ($3 in^2/ft^3$) crankcase volume. The relief area allowed by Lloyd's Register rules, namely $0.0115 m^2/m^3$ ($0.5 in^2/ft^3$) of crankcase volume, is based on a containment pressure of about 138 kPa (20 lbf/in^2) and would have to be increased by a factor of at least ten in order to cope with a major gearbox explosion.

If it were decided to fit relief valves a balance would need to be struck between valve area and increased resistance of gearcase covers. Undoubtedly, methods of gearcase-cover design can be formulated so as to achieve more uniform strength: but this may involve a weight penalty.

At the same time, no significant improvement in strength can be made before the problem of pressure containment is transferred to other areas of the gearcase structure such as the oil sump or the oil drain tank. With relief valves, there would be the further problem of ensuring that they did not become flame throwers, particularly in compact naval installations.

5.7 The Working Party agreed that explosion relief valves were not the answer for a naval installation and might engender a false sense of security.

An attempt should be made to design for more uniform strength and to aim at doubling the estimated yield pressure from 34 to 68 kPa (5 to 10 lbf/in^2). It was appreciated and accepted that this would involve a weight penalty compared with earlier designs.

6. WARNING AND CORRECTIVE ACTION

6.1 We have emphasised the importance of early knowledge of temperature rise at the hottest spot in any bearing, and the need for many thermosensors per bearing to ensure that the hottest spot (likely to be different for each operating condition) is covered.

6.2 In addition to bearing temperature monitoring, other parameters which might be covered in a gearbox health monitoring scheme, and thus provide warning, include:

6.2.1 gearbox atmosphere (oil mist detection)

6.2.2 lubricating oil contamination

6.2.3 bearing oil flows

6.2.4 bearing housing vibration

6.2.5 journal vibration

6.2.6 journal attitude

6.2.7 gearbox noise level

6.3 A commercial oil mist detector, tried at the controlled-atmosphere test facility, was found to be unduly sensitive but a simple modification to the electronics gave it an upper limit of $3 \times 10^{-3} kg/m^3$ in its most insensitive form. Whilst this was acceptable for the gearbox at the test facility, a more complex gearbox might well produce transient concentrations of mist exceeding this alarm setting in normal running so that a device with a wider range of alarm level settings would be desirable if oil mist monitoring gained general acceptance.

A compact unit could be designed to sense the gearbox atmosphere at, say, four positions. These would need to be selected as a result of trials on a prototype gearbox with a large number of sensing positions, but the problem remains of choosing alarm settings to ensure timely warning without spurious alarms.

6.4 Measuring the contamination of lubricating oil is more suitable as a long-term indicator of problems than a warning of imminent explosion.

6.5 Whilst bearing oil flow can be measured by flowmeter this is likely to be done on a one-off basis for investigation of a specific problem. If one is looking for indication of a wiped bearing this can more readily be seen by a reduction in the recorded bearing metal temperature or oil outlet temperature, due to the increased oil flow through the increased bearing clearance.

6.6 The measurement of journal vibration in the high-speed line, or main gearwheel shaft attitude in the low-speed line, would be a useful calibration exercise during prototype or first-of-class trials, or trials following a major refit: its usefulness for condition monitoring would depend on whether the permanently fitted instrumentation could retain the required accuracy over lengthy periods without skilled maintenance and calibration.

6.7 The Working Party's enquiries showed that excessive vibration was reported prior to about half of the explosion incidents. In some cases noise and vibration levels three to four times the normal level were reported. Thus, vibration monitoring may be the most useful indicator of the development of serious defects.

7. CONCLUSIONS

7.1 It would be impracticable to design and manufacture a gearbox guaranteed to be free of any failure leading to an explosion hazard.

7.2 The majority of incidents have been due to shortcomings in the basic design of bearings and/or their assembly after refit.

7.3 Means of detecting and inhibiting potential oil mist explosions have been designed, developed and tested.

7.4 Means of containing hazardous oil mist around a bearing have been designed and tested.

7.5 The provision of a reliable and continuous method of scanning gearbox bearing and rubbing seal temperatures would significantly reduce the explosion risk. The instrumentation fitted to past designs has often been inadequate, but improve systems have been incorporated in the more recent naval designs and these appear to be functioning well.

7.6 Measurements on gearboxes, both ashore and afloat, have shown that, under normal running conditions, the oil mist atmosphere is too lean for ignition to occur. The oil mist present is continously reduced by the "scrubbing" action of the relatively coarse oil spray and splashing which occurs in a running gearbox.

7.7 The rupture strength of existing designs of gearboxes is under 34kPa (5 lbf/in^2) whereas the explosion pressure if contained could be expected to rise to about 0.95MPa (140 lbf/in^2). Hence, it would be impracticable to design for total containment, although a doubling of the strength of current designs would give protection against a minor explosion.

7.8 The provision of sufficient relief valve area and associated venting arrangements is impracticable in the naval environment.

7.9 The books of reference containing Operating Instructions for Geared Main Propulation Installations require amendment to include instructions generally similar to those for the avoidance of crankcase explosions in diesel installations.

8. RECOMMENDATIONS

The Gearbox Explosion Working Party made the recommendations listed below. Appropriate action has been taken by the Ship Department's gear design section. A new Naval Engineering Standard, 305, will specify how these aims should be fulfilled by MOD contractors.

8.1 Gearing detail design to be subjected to independent audit in respect of calculations and proposals for journal and thrust bearings and shaft seal arrangements. In particular, journal bearing design reviews should include checks of:-

8.1.1 position of bearing load lines and lines of minimum oil film thickness under all modes of operation of the prime movers, including no load;

8.1.2 location of oil inlets and line of bearing split;

8.1.3 location, number and type of bearing sensors;

8.1.4 method of location of bearing shells to ensure that bearings cannot be assembled incorrectly, either on initial build, or after opening up for maintenance;

8.2 Further work is required on the definition and development of bearing temperature sensing systems to ensure reliability.

8.3 Instrumentation to be developed to provide reliable monitoring of bearing housing vibration, and of vibration and attitude of shaft journals on gearbox input and output lines.

8.4 Following from recommendations 8.2 and 8.3 above, systems are to be developed for gearbox health monitoring with the object of reducing or eliminating the need for routine opening of gearboxes.

8.5 Detailed and thorough analysis of system's dynamic performance is to be undertaken in the design stage with the object of identifying the steady and transient conditions on all thrust and journal bearings. This should be followed by practical measurement at the prototype trials stage, with further analysis and measurement of any subsequent modifications.

8.6 Prototype trials of all new gearing designs should in the gearbox establish typical oil mist concentrations.

8.7 Whilst designs of oil mist detectors, arranged to trigger gas inerting systems, have been developed, and bearing flame-arrester devices have been demonstrated, such devices should be called for only in special circumstances.

8.8 The structural design of future naval gearboxes and their associated oil drain tanks is to be reviewed with the object of raising the rupture strength to about 68kPa (10lbf/in^2).

8.9 The books of reference containing the Operating Instructions for Geared Main Propulsion Installations are to be revised to include instructions to cover procedure in the event of incidents such as overheated bearings. The instructions would be generally similar to those for the avoidance of crankcase explosions in diesel installations. Provided sufficient oil supply is available, the aim would be to keep the gearing turning over at low power for long enough to ensure that the scubbing action is maintained until the heat sources have disappeared.

8.10 There is still a need to gather information, as and when convenient, on the performance of venting systems in existing and new machinery installation designs.

REFERENCES

1. HARRISON, W.H., GREEN, D and HOLNESS, M.H., 1977. Oil mist/vapour content of gearbox atmospheres. Proceedings of World Gearing Congress, Paris, June 1977. pp 541-602.

2. NICHOLSON, D.K., 1981 "The Kootenay gearbox explosion." Trans, IMarE, (TM) Vol 93, Paper 11.

ACKNOWLEDGEMENTS

Although the findings of the MOD Gearbox Explosion Working Party have been quoted the opinions expressed are those of the authors and do not necessarily reflect Naval policy.

The authors wish to acknowledge the influence of previous heads of the Gearing Section and to thank for their assistance and advice various members of the staff of the Director General Ships, Admiralty Laboratories and Y-ARD; and David Brown Gear Industries, GEC Marine and Industrial Gears, and Vickers Shipbuilding Group; and all other individuals and organisations who contributed to this work.

APPENDIX I

LIST OF GEARBOX INCIDENTS

<table>
<tr><th rowspan="2">Item</th><th rowspan="2">Date</th><th rowspan="2">Installation</th><th rowspan="2">Oil Ignition Inside Gearbox</th><th colspan="2">Gearbox Explosion</th><th rowspan="2">External Fire</th></tr>
<tr><th>Minor</th><th>Major</th></tr>
<tr><td>1</td><td>28 FEB 48</td><td>HMS Devonshire</td><td>YES</td><td>YES</td><td>NO</td><td>NO</td></tr>
<tr><td colspan="7">The ship was steaming at 25 knots from Kingston to Belize. Severe vibration occurred in the forward engine room and an explosion was heard in the port outer gearbox. No flames were reported in the engine room but troops on deck saw flames issue for a few seconds from a ventilation trunk which led from above the port outer gearcase. Sometime later the shaft was stopped and locked because of loud noise from gearbox. A tooth had broken from the LP pinion in fatigue and damaged the aft helix of the main wheel and HP pinion.</td></tr>
<tr><td>2</td><td>28 NOV 60</td><td>HMS Hampshire</td><td>YES</td><td>YES</td><td>NO</td><td>NO</td></tr>
<tr><td colspan="7">During shop testing of the starboard set of cross compound steam turbines and main gearing, severe vibration occurred on the HP input line at design speed and speed was reduced. A minor explosion occurred at 68% speed. Subsequent investigation showed heat source to have been a rub in the HP steam turbine manual clutch, the operating fork being severely worn.</td></tr>
<tr><td>3</td><td>MAR 62</td><td>SS Verena</td><td>YES</td><td>NO</td><td>YES</td><td>YES</td></tr>
<tr><td colspan="7">The information on this 12 year old steam turbine tanker was obtained from Lloyd's reports eg 'Tank Steamer Verena: Register surveyor advised main engine L.P. primary pinion flexible claw type coupling failed, smashing upper and lower cast iron casings about the coupling and pinion, resulting in intense local heating and subsequent ignition of lubricating oil vapour in gearcase (six men killed)'.</td></tr>
<tr><td>4</td><td>21 MAR 63</td><td>HMS Kent</td><td>YES</td><td>YES</td><td>NO</td><td>NO</td></tr>
<tr><td colspan="7">During contractors sea trials severe vibration was experienced on the HP steam turbine input line. An inspection revealed only a rub on the input shaft oil seal. A consumption trial at about 80% design speed was resumed. Severe vibration was again experienced and a minor explosion occurred in the port gearbox before instructions to reduce power could be given. Subsequent inspection of the manual clutch operating pads showed they had lost their whitemetal and the steel backing was worn.</td></tr>
<tr><td>5</td><td>DEC 64</td><td>SS Seatrain New York</td><td>YES</td><td>NO</td><td>YES</td><td>YES</td></tr>
<tr><td colspan="7">The owners supplied MOD (N) with reports and excellent pictures of this dramatic incident where a pinion came to rest beside a boiler. The watchkeepers noted a reduction in revs and a slight rumble prior to violent explosion followed by fire. The cause was either a coupling bolt failure or a gear rim may have come off its spider.</td></tr>
<tr><td>6</td><td>16 SEP 65</td><td>HMS Hampshire</td><td>YES</td><td>YES</td><td>NO</td><td>NO</td></tr>
<tr><td colspan="7">During sea trials following a refit a changeover was being made from gas turbine to steam turbine drive. This drill involves closing one oil supply valve for the fluid couplings associated with each gas turbine ie four valves in all. By mischance, the padlock key for one of the fluid coupling supply valves fitted that for the main oil supply to the starboard gearbox ringmain. The oil supply was cut off for 3 to 5 minutes before the fault was detected. After the oil supply was restored and bearing temperatures checked the shaft was restarted. At the end of the forced rolling trial smoke and flames were observed from the port gearbox. The shaft was stopped immediately and fire went out. No excessive bearing temps were observed, probably because the thermocouple in the failed bearing had lost contact with shell due to thermal distortion.</td></tr>
<tr><td>7</td><td>SEP 65</td><td>SS Malmöhus</td><td>YES</td><td>YES</td><td>NO</td><td>NO</td></tr>
<tr><td colspan="7">The first Stal Laval AP 24 installation was in the tanker MALMÖHUS. During setting of the overspeed trip on builders sea trials the speed rose above the rated 112-115% speed to 118% of full speed. There was a recorded increase in vibration level and the inlet oil temperature rose from the normal 45 to 49°C to 53°C. About 15 minutes after these abnormalities were observed there was a muffled detonation from the gearbox followed by yellowish-white smoke from the vent on the inlet end gear casing. The cause was later attributed to inadequate load carrying capacity of the oil film in one of the three journals of the epicyclic gear train on the HP side (see Shipbuilding and Shipping Record, May 25, 1967, P.731).</td></tr>
</table>

continued on p. 9

Item	Date	Installation	Oil Ignition Inside Gearbox	Gearbox Explosion Minor	Gearbox Explosion Major	External Fire
8	AUG 66	Rolling Mill	YES	YES	NO	NO

Repeated failures of a tapered roller bearing occurred in the gearbox drive to a six stand section of a wire rod mill. Before successful modifications were carried out about ten bearings had to be replaced. The last failure gave a minor gearbox explosion.

9	15 FEB 67	RFA Regent	YES	YES	NO	NO

During contractors sea trials an abnormal noise was heard from the gearcase in the vicinity of LP first reduction train. Orders had just been given to stop when a minor explosion occurred and oil vapour and smoke were emitted from cover joints and the vent pipe. Subsequent investigation showed that a bearing ring in the bore of the LP primary pinion (used to locate a balancing mandrel during production) had become dislodged and become trapped between the rotating male member and stationary female member of the main gear clutch. The consequent friction caused severe overheating of the ring.

10	1967	CEGB Tilbury 'B'	YES	YES	NO	NO

Parallel shaft gearbox in drive for 2000 bhp feed pump in power station. Explosion due to thrust bearing failure blew off gearcase covers.

11	30 MAY 68	Naval Transmission Test Facility	YES	YES	NO	NO

A County Class gearbox driven by G6 gas turbines was in use for fluid coupling development trials. Cooling oil was shut off when an astern fluid coupling was being operated at 200% slip. The overheating due to windage and churning led to a minor explosion which blew off some unsecured gearcase covers. One cover went about 40 feet up through the roof of the test house. The subsequent investigation revealed no evidence of any rubbing.

12	23 OCT 69	HMS Kootenay	YES	NO	YES	YES

KOOTENAY had just reached full power but was still building up to full ship speed when a gearbox explosion burst the starboard gearcase cover and caused a severe fire. Six men perished in the engine room and of four others who managed to evacuate all were badly burned and only three survived. The accident was attributed to incorrect fitting of the primary pinion bearing shells in February 1965. Many full power trials had been carried out since then. The ship had no gearing bearing temperature thermometers or thermocouples. Vibration monitoring had commenced 6 months previously when the level found of 117 Vdb was at least 10 Vdb above fleet average levels for the primary reduction bearing positions. The engineer officer reported that the first intimation of trouble was 'a hissing and sparking noise like any oxy-acetylene welding torch followed by a bang and a ball of fire from the starboard after part of the Engine room.' Also, about 2-3 minutes before the explosion, an unusual sound described as a steady organ-like note of 5-10 secs duration was heard by five occupants of the cafeteria above and outboard of the after ER hatch on the starboard side. This noise is believed to be due to the rush of gases in the 8ft long gearcase vent pipe.

13	30 JUN 70	Naval Machinery Shore Test Facility	YES	NO	YES	YES

Selected Temperatures °F from Data Logger

No	Point	TIME 20.16	20.21	20.41	21.00
1	Main wheel thrust drain	125	130	144	150
2	Main wheel ford. brg.	78	81	79	77
3	Main wheel aft brg.	105	107	115	119
33	Main thrust block brg.	91	109	119	94
35	Main LO after cooler P	104	105	106	92
36	Main LO after cooler S	96	96	100	76
37	FW circ. ringmain supply	76	78	80	65

This incident occurred on a production set of gearing comprising two single cylinder steam turbines connected by clutches to a combining gearbox and thence by flexible coupling to a separate main thrust block. The set was undergoing shore trials and had operated for 34 hours under two turbines ahead and astern, and auxiliary drive ahead and astern. The set had then operated under port turbine ahead for one hour at full power when the explosion occurred. Several witnesses reported a build up of sound as a change in the normal running noise immediately prior to the explosion. With the noise of the explosion a column of flame rose to a height of a least 25 feet above the gearbox and lasted about 0.5 seconds. The flame was dull orange or dark red accompanied by thick black smoke. When it died out most witnesses described seeing flames several feet long issuing from a hole in the port side of the gearbox, from the starboard aft corner, and from the auxiliary drive chain case. A data logger with a scanning speed of 100 points in 12 seconds fitted to the set had printed a routine log demand immediately prior to the explosion. All temperatures, as typified by the results shown above, gave no indication of excessive bearing temperatures or impending failure, but the No. 2 thermocouple for the main wheel forward journal bearing was suspected of being open circuited. The damage included:-

(a) broken fibreglass gearcase covers, parts of which had passed through the mesh, damaging gear teeth.
(b) a failed main gearwheel forward journal bearing exhibiting a major wipe
(c) a marked main gearwheel aft journal bearing, particularly at its aft end, bottom half.

14(a)	18 SEP 69	Industrial Pump Drive	YES	YES	NO	NO
(b)	31 JUL 70		YES	NO	NO	NO
(c)	21 AUG 70		YES	YES	NO	NO

In a 500 hp parallel shaft gearbox drive failures of tapered roller bearings caused a minor explosion on two occasions and on the other a fire in the gearcase. The interesting feature is that the gearmaker had taken noise levels originally and just before the first explosion which showed an increase from 95 to 106 dB

15	16 APR 71	USN Caliente	YES	YES	NO	NO

This 30 year old fleet oiler experienced a minor explosion in its port main propulsion gearing. The cause was found to be an overheated forward HP line intermediate bearing which had suffered oil starvation due to the associated needle valve being plugged with rust and dirt.

16	04 JUN 71	HMCS Skeena	YES	NO	NO	NO

Five hours steaming had been achieved on post refit trials and about 70% shaft speed reached. Only primary pinion bearings were being monitored although all bearings were instrumented. Orange flashes indicating lubricating oil ignition were observed through the acrylic inspection windows. A further flash of burning oil was followed by a slight discharge of blackened oil from the gearcase vent. Investigation showed that inverted fitting of the shells of the primary gearwheel aft bearing had shut off the direct oil supply. This bearing is fitted between steel shoulders to axially locate the primary gearwheel-secondary pinion/quillshaft assembly. A major failure in this bearing will cause a galloping expansion—overheating reaction, as the axial clearance is absorbed to produce a dangerous level of heat generation at as low as 70% shaft speed.

17	08 JUN 71	HMCS Chaudiere	YES	NO	NO	NO

After six hours steaming at 90% speed, power was reduced and a screeching noise was heard at 80% speed. The noise disappeared when power was reduced to give less than 50% shaft speed. Vibration readings were taken and found satisfactory. Speed was then increased to about 60% and the primary pinion forward bearing temperature rose about 30°F above normal. An observer using the acrylic inspection ports saw burning particles of oil in the vicinity of the primary pinion forward bearing which is a combined thrust and journal bearing. Later investigation showed that the locating thrust bearing had failed and the primary pinion was rubbing against the aft end of the forward journal bearing—hence the screeching. The thrust bearing failure was attributed to contamination by galvanised zinc particles detaching themselves from the main oil filters. In this ship the thermocouples had detected failure condition.

18	28 OCT 71	MV Maihar	YES	NO	YES	YES

This ship was the second of a pair built by Cunard Brockelbank which were among the first British ocean going cargo-liners to employ geared diesel propulsion. Lloyd's List for week ending 02 NOV 71 reported an engine room explosion and fire whilst the ship was leaving Tokyo Bay. 'The Marine Engineer and Naval Architect' for April 1972, page 143, reported the incident as a gearbox explosion. The cause was a rub by the clutch disc carrier in the input line.

19	NOV 71	HMS Zulu	YES	YES	NO	NO

A major gearbox failure was experienced in this COSAG frigate following failure of a lightly loaded thin shell bearing (4½ inch dia x 2½ inch long) at the aft end of the steam turbine jackshaft line. The ship had been reporting a steady increase in vibration level for some time before the incident. Popping noises were heard in the gearbox and then a grinding noise about 30 minutes later. A few minutes before the grinding noise the watchkeepers saw flashing near the manual clutches (there are windows to observe clutch position) and smoke and oil leaks occurred from gearbox joints. Later inspection showed the jackshaft to have been grooved about 1 inch deep. It was bent in way of the bearing and the attached clutch member had gone into whirl, fouling the gearcase and machining the clutch teeth. This debris passed through the mesh. The aft jackshaft bearing assembly had become welded together. The thermocouple scanner did not indicate an excessive temperature until the time of the grinding noise when over 300°F was indicated. Subsequent metallurgical examination indicated that 950°C had been reached. The records showed that the aft jackshaft bearing had failed twice before, a minor wipe in APR 70 and a total loss of whitemetal in DEC 70. There is a suspicion that the replacement may have wiped soon afterwards.

20	08 DEC 75	SS London Pioneer	YES	NO	YES	YES

The information on this steam bulk carrier, built in 1958, was obtained from Lloyds List cuttings dated 11, 13 and 20 DEC 75 and 21 and 29 JAN 76. Following are quoted. 'Dec 9: London Pioneer had an engine room explosion and fire last night which seriously injured 2 men', 'Dec 11.........high pressure turbine and high pressure flexible coupling also high pressure first reduction pinion destroyed and both second reduction pinions badly damaged on aft ends. Main wheel also damaged on aft end. Would appear oily mist explosion in region of high pressure flexible coupling followed by flash fire......Two seriously burned crewmen were lifted by helicopter from London Pioneer to US Coast Guard cutter MELLON last night'.

The Kootenay Gearbox Explosion

D.K. Nicholson, C Eng, MI MarE, MI Mech E
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SYNOPSIS

The main gearbox explosion in HMCS Kootenay in 1969 remains unprecedented in magnitude and loss of life. It revealed a disturbing lack of knowledge about flammability of gearbox atmospheres and led to extensive investigations. The author describes the installation and conditions leading to the Kootenay explosion; he reviews the results of the initial investigation which failed fully to explain the mechanism of failure. The clue which finally explained this is examined in the light of subsequent failure investigations. These indicated the great heat generated by single-helical primary pinion thrust failures. Finally the author considers how the risk of a heat source causing an explosion can be significantly reduced by the effective use of oil spray. Recommendations are made concerning gear design, instrumentation and maintenance policy.

INTRODUCTION

The main gearbox explosion, which occurred in the Canadian destroyer HMCS *Kootenay* in October 1969 with the loss of nine lives, was not only the first such explosion experienced in the Canadian Navy, it also surpassed in magnitude anything known to most other naval and mercantile authorities. This incident revealed a disturbing lack of knowledge of gearbox explosions and of the nature and flammability of gearbox atmospheres.

While the areas and causes of mechanical failure which led to the Kootenay incident were readily identified in the ensuing investigation, the effectiveness and adequacy of the corrective measures taken could not be fully assessed at the time without a better understanding of the mechanism and risk of gearbox explosions.

The urgent need for knowledge on this subject led to the formation under the auspices of the Royal Navy of a Gearbox Explosion Working Party in 1970. Its task was to investigate the causes and mechanisms of all known gearbox explosions and to initiate or recommend related lines of research into gearbox atmospheres. This work concluded in 1978, was reported by Cooper, Holness and McNeill[1] It has contributed greatly to the investigation on the *Kootenay* explosion conducted by the Canadian Department of National Defence.

The *Kootenay* incident initially gave rise to many and varied theories on the mechanism of failure and explosion. None of these could be proven by means short of initiating or simulating a major bearing failure in an actual gearbox operating at high power. It was no doubt fortuitous that, in the two years following the explosion, three other Canadian ships experienced gearing failures with explosion potential. The analysis of these failures provided the key to understanding the mechanism of failure and the conditions necessary to produce the explosion.

It is the purpose of this paper to trace the development of the understanding and to demonstrate the confidence now felt in the effectiveness of the corrective measures taken.

THE 'KOOTENAY' GEARING

The *Kootenay*, which entered service in 1959, is a 2600 tonne destroyer escort developing 22400 kW (30 000 hp) on two shafts at 225 rev/min. It is fitted with double-reduction, single-helical, hardened and ground gearing described by D. K. Nicholson[3] whose 1961 paper was primarily concerned with the development, manufacture and initial service experience with the hardened and ground gearing fitted in the *St Laurent* Class. It did not deal with the primary pinion bearing problems encountered during the initial years of service nor with the subsequent developments which would ultimately be associated with the cause of the *Kootenay* explosion.

The arrangement of the gearing, as originally fitted for main and cruising turbine drives, is shown in Figs. 1 and 2. The principal design particulars based on a 220 rev/min full-power shaft speed are given in Table I.

The cruising turbines had been found to provide limited fuel economy benefit in service and had been removed from all ships at the time of the explosion. They did, however, have a significant influence on the main gearing pinion-bearing design that contributed to the cause of the explosion.

All mention in this paper of primary pinions and primary pinion bearings will refer to the main turbine drive's primary reduction pinion.

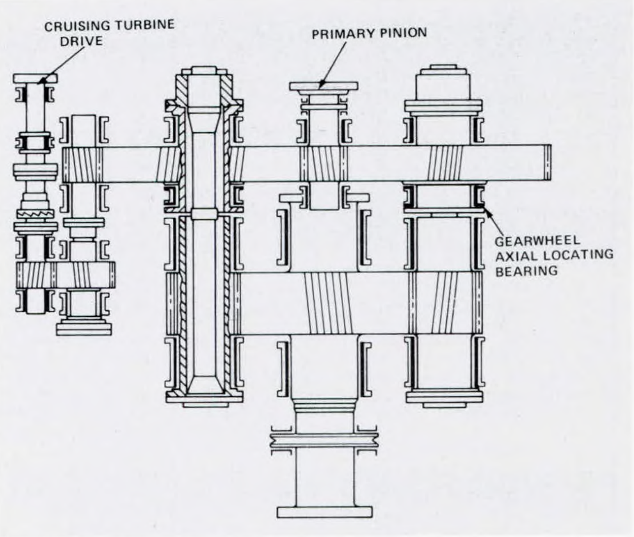


Fig. 1 St. Laurent class main gear arrangement

Table I Kootenay Main Gearing Design Data

	PRIMARY REDUCTION	SECONDARY REDUCTION
Pinion PCD (mm)	228	346
Wheel PCD (mm)	1172	1704
Face width, (mm)	200	350
Pressure angle, (deg)	23	20
Helix angle, (deg).	10	6
K factor, N/mm^2 (lbf/in^2)	2.21 (320)	2.84 (412)
Overall reduction	25.29	
Overall dry weight, (tonne)	17.24	

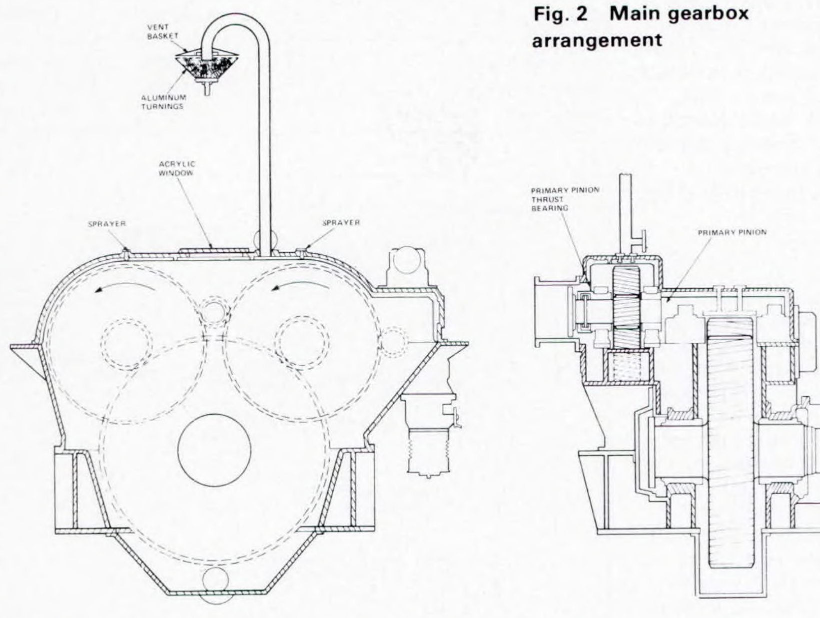


Fig. 2 Main gearbox arrangement

Primary pinion journal

The contractor sea trials conducted on three of the early ships of St. Laurent class revealed a primary pinion bearing problem not experienced in the first ship of the class, nor in the extensive prototype shore trials 2. In each case the routine post-trial gearing and bearing examination showed that in the port unit only, the primary pinions forward bearing, had suffered a white-metal fatigue failure which the bearing themometer had failed to detect. Fig. 3 shows the failed 2.5mm thick white-metal liner, mechanically anchored to a cast iron backing shell, since a sound chemical bond is difficult to obtain between these two materials.

Without a chemical bond, the white metal becomes loose and highly susceptible to fatigue failure under vibratory loads. Such vibratory loading existed in the forward bearing of the primary pinion and was eventually traced to a severe third-order torsional vibration under cruising turbine drive. Since efforts to eliminate this vibration were unsuccessful, it became necessary to find a bearing that could withstand it. This led to the adoption of the precision bronze, medium-wall insert bearing shell with a 0.7mm white-metal lining.

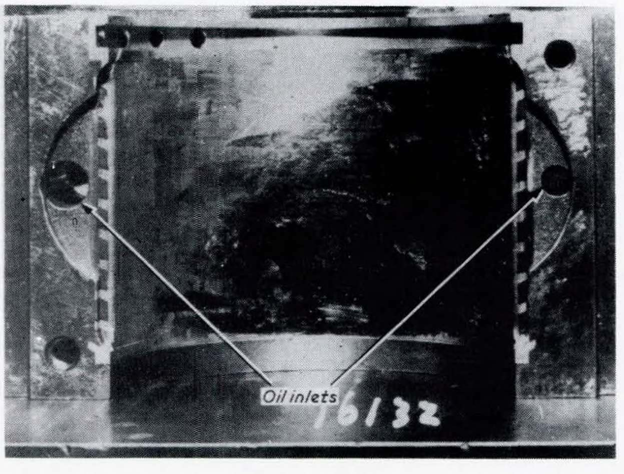


Fig. 3 Primary pinion two-inlet, thick shell bearing failure

The cause of failure was also pursued through an analysis of journal load-lines and attitudes over the full range of powers. This revealed that the original bearing design had been based on the cruising power coming solely from the cruising turbine; whereas in fact the cruising turbine exhausts to the main turbine LP stages, which provide approximately one third of the total cruising power. The lower oil inlet to the pinion bearing was then seen to be badly located for the cruising drive (Fig.4) and was, accordingly, eliminated.

The loaded top half insert has a full annulus with an auxiliary inlet hole located 80° from the bearing split. Extensive ship trials, with the single-inlet insert shells fitted in the primary pinion bearings, were successfully completed in May 1957 under main and cruising turbine drives. Thermocouples fitted in each bearing half, but not penetrating the insert shells, were sufficiently sensitive to register temperature changes due to major helm movements. The complete insert bearing assembly. shown in Fig. 5 was then fitted in all ships of the class.

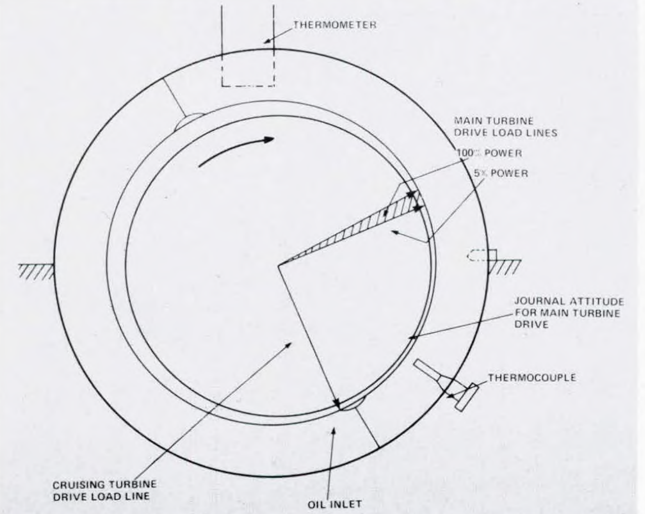


Fig. 4 Primary pinion bearing load lines

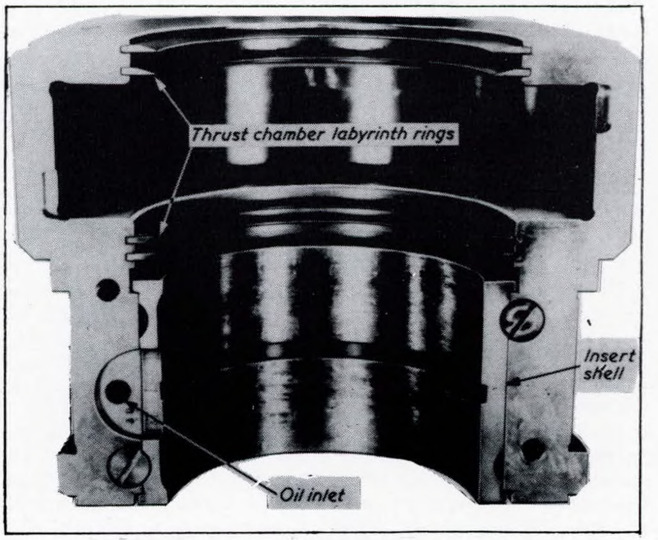


Fig. 5 Primary pinion single inlet insert shell bearing.

Primary pinion thrust bearing

At full power the single-helical main primary pinion exerts a significant axial thrust (Table II), which is absorbed through an integral thrust collar, acting on a tilting pad thrust bearing. Fig. 6 shows the arrangement of the thrust bearing housed in a thrust chamber integral with the primary pinion's forward bearing shell.

A common oil supply feeds the journal and thrust bearings. Labyrinth rings at each end of the thrust chamber maintain flooded lubrication for the 12 pads on each side of the thrust collar. An oil discharge nozzle from the thrust chamber provides lubrication to the aft end of the fine-toothed flexible coupling to the turbine.

Bearing instrumentation

All journal bearings of the gears were originally fitted with mercury-in-steel dial thermometers, shown in Fig. 7. They were assembled in 25.4mm (1in) diameter, bronze, separable sockets or tubes, fitted vertically between the gearcase cover and the bearing. While the bronze sockets penetrated the bearing shell, the mercury bulb could not be placed sufficiently close to the bearing's point of minimum oil-film thickness to sense any significant local increase in temperature.

This instrumentation was in fact quite incapable of detecting the early primary pinion bearing failures already described. While thermocouples had been used in certain bearing installations for special trial purposes, there was a problem in finding a monitoring instrument that was sufficiently rugged and shock resistant to be suitable for continuous shipboard use. A class-wide change-over from dial thermometers to thermocouples was held up pending the successful evaluation of a suitable temperature monitoring instrument..

An installation instrumented with thermocouples and a 40-point continuously sensing monitor was evaluated in two ships in 1960-62 and approved for installation in new ships building during that period. Unfortunately no case could be made at that time for retrofitting in existing ships and replacing the dial thermometers. Eventually the increasing cost and logistic problems of supporting the latter, whose mechanisms did not readily withstand the main gearing vibration, led to an end to further repair and replacement. By 1969, *Kootenay* in common with a number of other ships, had no bearing temperature instrumentation fitted to its gearing.

Gearcase vents

The ship was fitted with a high gooseneck, 100mm bore vent, positioned on the domed part of the gearcase cover over the primary pinion and discharging into a vent basket filled with aluminum turnings which act as an oil trap. The other three ships mentioned in this paper were fitted with the currently approved atmospheric vent system, comprising a short gearcase vent, open to the engine room space; and an atmospheric vent extending from the drain tank. The purpose of the atmospheric vent system is to utilize a differential engine room/atmospheric pressure to reduce the emission of oil fumes to the engine room and to prevent gearbox condensation and corrosion problems by air purging when shutting down main engines.

Inspection windows

Heavy acrylic windows had been fitted over all inspection ports in the aluminum gearcase-cover. These were located over each primary or secondary reduction pinion and had proven most useful in permitting certain checks to be made and conditions observed without the time or risk involved in removing inspection doors. They were also to prove invaluable in detecting two cases of oil ignition.

THE EXPLOSION

In October 1969 *Kootenay,* in company with other ships, was returning from England to Canada to undergo a major refit and had been detached for a scheduled quarterly full-power trial. The shaft speed had been worked up from 186 to 216 rev/min in 70 minutes when "Full Ahead Both Engines" was requested and port and starboard throttles were fully opened.

Some eight minutes later the Engineer Officer walked around the machinery, noting no abnormal conditions and reporting that the gearcase temperatures felt normal to the touch. The shaft speed had reached 221 rev/min or 98 per cent of the normal value reached on previous full-power trials, when, approximately 11 minutes after going to full power, the starboard gearbox exploded, rupturing and

Table II Primary pinion bearing bearing data

	JOURNAL	THRUST
Max. Load kN (lbf)	31.1 (7000)	29.7 (6680)
Pressure, MPa (lbf/in^2)	1.60 (232)	2.66 (386)
Velocity, m/s	49.56	64.91 (mean)

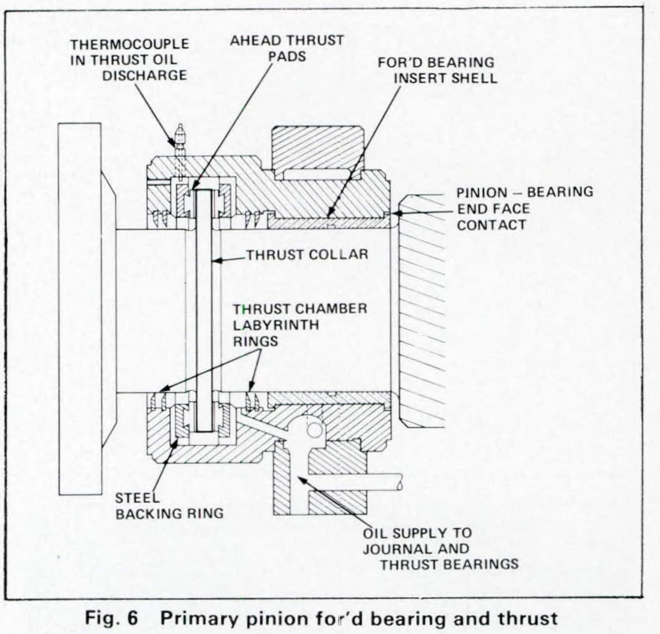
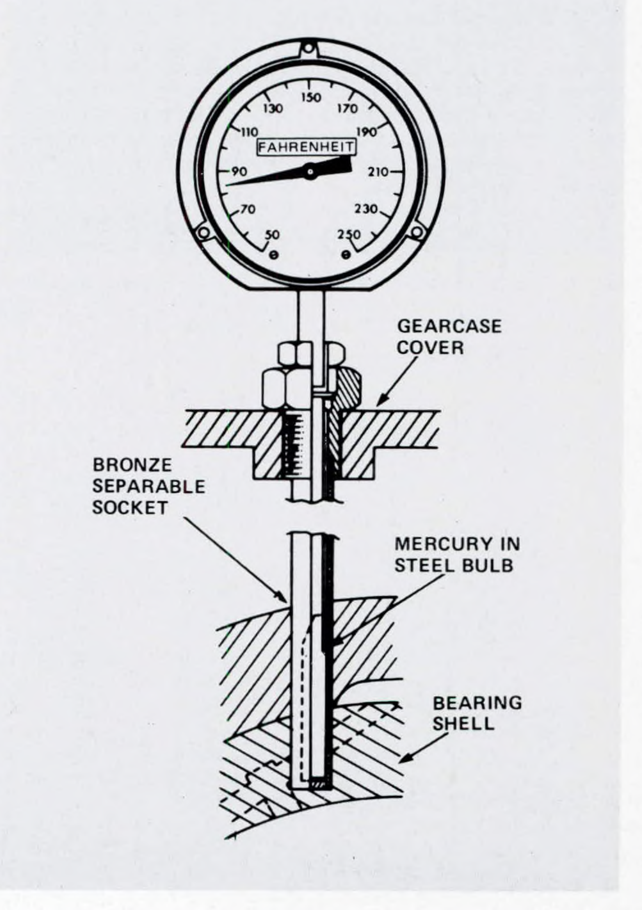


Fig. 6 Primary pinion for'd bearing and thrust



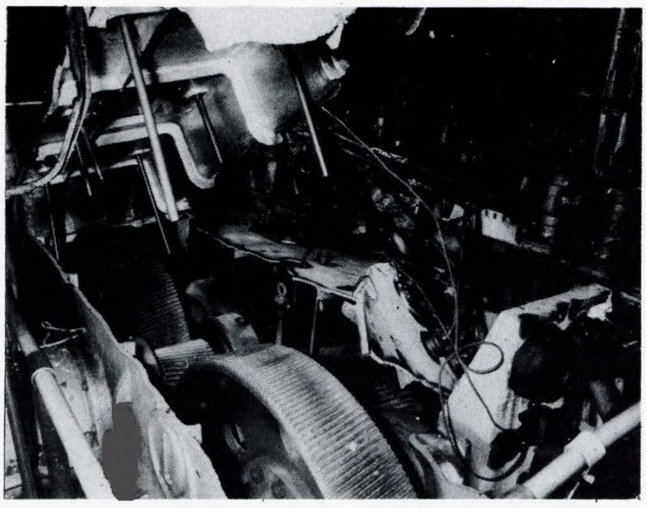


Fig. 8 Kootenay gearbox explosion damage

blowing open the cast aluminum gearcase cover. The explosion produced a rapidly expanding fireball of intense heat which almost instantly enveloped the entire engine room and emerged through the open hatches to the passageway above.

Efforts to close the throttles achieved only three to four turns down from fully open. Of the 10 men in the engine room for the full-power trial, four managed to escape but only three survived. Two more fatalities occurred outside.

The resulting fire and smoke disrupted communication and access to the boiler room, leaving the ship to steam on out of control for about 30 minutes with the engines at more than half power. During this time oil from the sprayers, mounted in the ruptured gearcase cover and supplied by the turbine-drive lube oil pump, continued to feed the fire.

The Engineer Officer, who was one of the three badly burned engine room survivors, reported that the only warning had been "a loud hissing and sparking noise like an oxy-acetylene torch." Experienced off-duty engine room personnel in the cafeteria on the deck, near the engine room hatch, heard a pronounced organ-like note of 5 to 10 seconds' duration, followed by a short pause immediately before the explosion. This noise was later attributed to the discharge of gases under pressure through the gearcase vent.

Post explosion examination

The domed part of the gearcase cover over the primary gear train had ruptured along the upper forward edge and the lower after edge to separate from the rest of the cover, commencing at the inboard end (Fig.8). The inboard end of the separated dome had been blown upwards about one meter and held suspended by the sprayer piping, which remained intact. This appeared to be the primary area of gearcase cover failure, giving rise to all other fractures.

The immediate explosion damage in the gearbox was confined to the cover. All rotating elements were intact, although the primary pinion was displaced forward about 6mm.

The location of the heat source capable of causing the explosion was quickly narrowed down to the primary pinion's foward and aft bearings. The insert bearing shells were found to be installed back to front. With a single main inlet and an auxiliary inlet at 80° from it, this results in a complete blockage of the oil supply, as shown in Fig.9. Both bearings had sustained excessive overheating and wear (Fig. 10). The aft bearing's upper insert had worn 3.8mm and disintegrated under heat into cracked segments.

Metallographic examination showed that the incipient melting temperature of bronze, 800°C, had been reached. The pinion's aft journal had worn 1.27mm and had a honeycomb pattern 0.38mm deep heat cracks, indicating that surface temperatures of 800-900°C had been reached a number of times.

The primary pinion ahead thrust bearing was completely destroyed as shown in Fig.11. The thrust collar had gone through the white-metal-faced bronze bearing pads to make steel-on-steel rubbing contact with the thrust bearing's backing ring. The thrust collar (Fig.12) had been gouged or worn to a depth of 2.8mm with 0.6mm deep heat cracks. Fig 13(a) shows the melted and alloyed bronze-steel-babbit debris overlaying the steel backing ring (lower left) and the bronze thrust pad (lower right). Fig. 13(b), further magnifying a section of 13(a), shows the penetration of the steel base by molten-bronze, suggesting a temperature of 1000°C.

The astern thrust pads suffered relatively little damage and did not reach the melting point of white metal (260°C). The wear in the journal bearings had caused a corresponding 3.4mm wear in the thrust chamber labyrinth rings, thus affecting the flooded lubrication on the thrust pads.

The port gearbox was opened, revealing that its primary pinion's aft journal and ahead thrust bearings had also failed. The port and starboard thrust failures were almost identical, except that the port pinion's thrust collar had worn only 1.2mm.

A micro-examination of the aluminum turnings taken from the centre of the starboard vent basket, directly under the vent pipe, (Fig 14a) showed intergrannular cracking and incipient melting which would not occur below about 650°C. This condition did not exist in

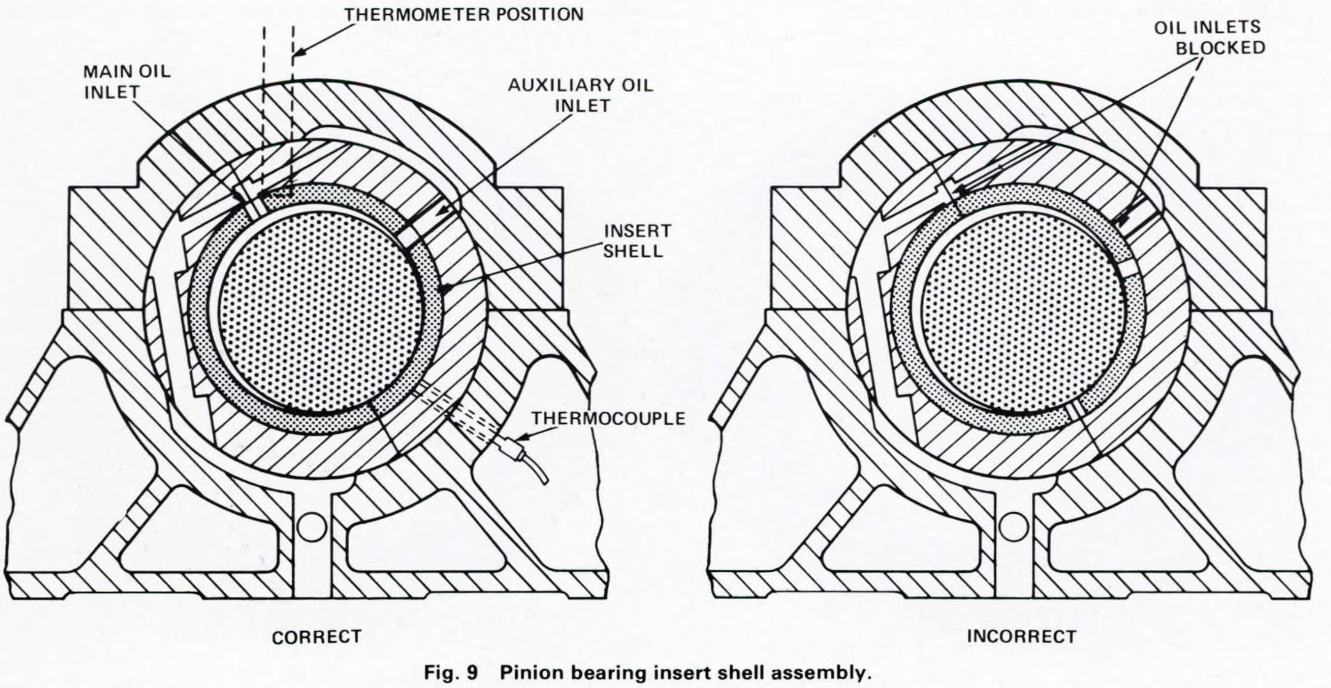


Fig. 9 Pinion bearing insert shell assembly.

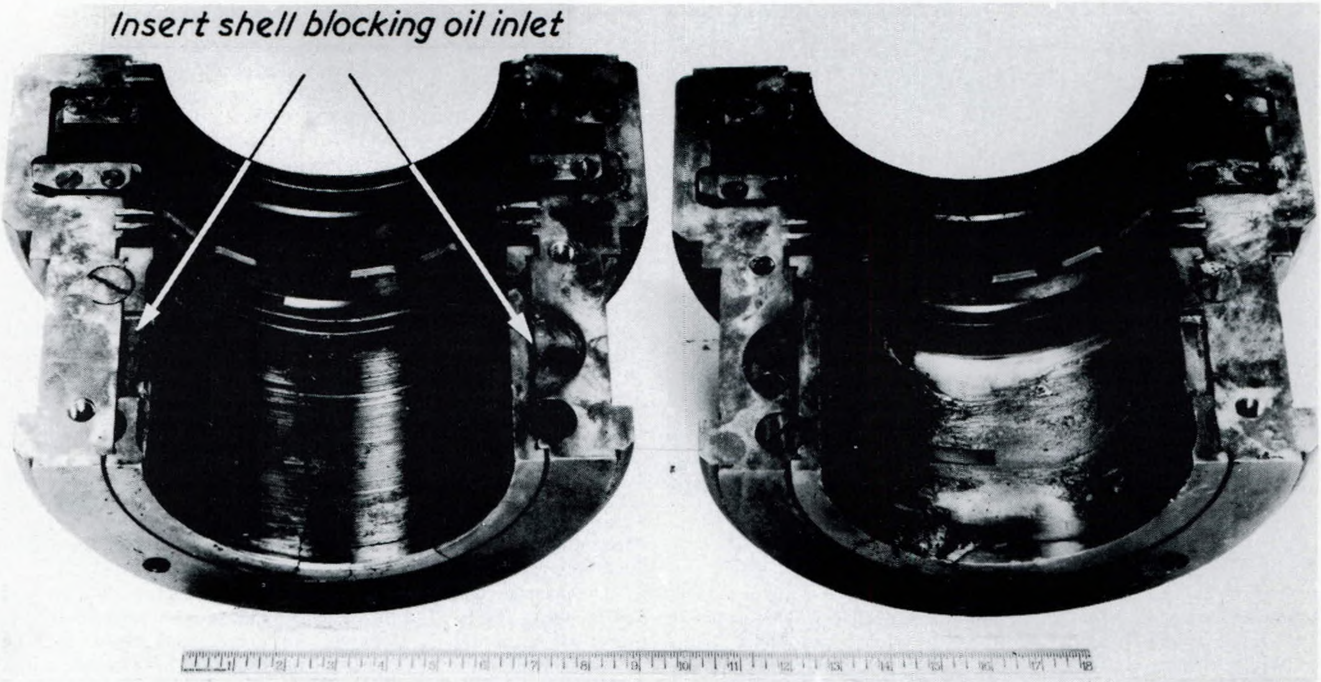


Fig. 10a Kootenay pinion for 'd top (L) and bottom (R) bearing halves

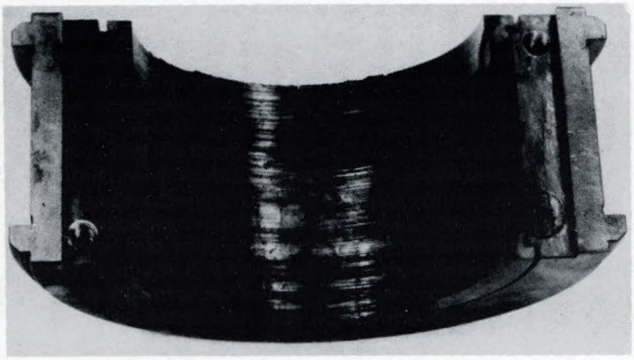


Fig. 10b Kootenay pinion aft bearing top half

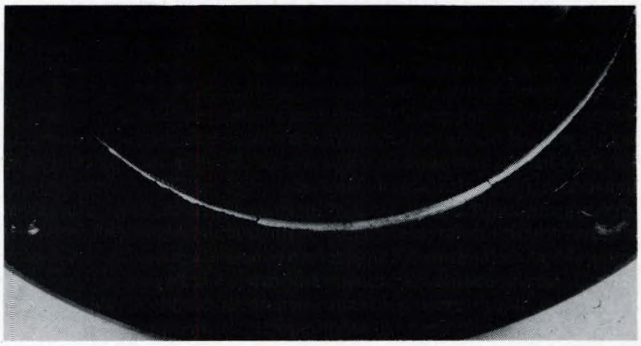


Fig. 10c Kootenay pinion for'd bearing end rubbing contact

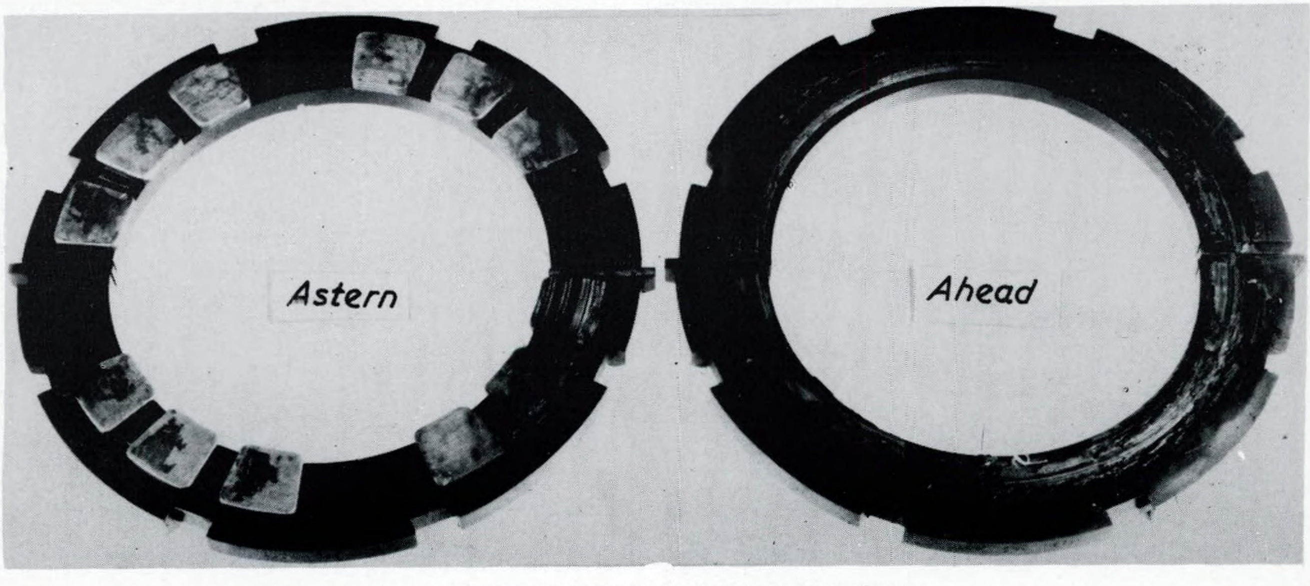


Fig. 11 Kootenay pinion thrust bearing pad assemblies

the turnings taken from the port vent (Fig 14b) or from other areas of the starboard vent basket.

Lubricating oil

A standard marine steam turbine non-EP mineral oil, equivalent to OM100, was used. The analysis of samples taken from *Kootenay* showed that it fully conformed with specification. The flash-point measured by the open cup method was 244-255°C. The fire-point, or minimum temperature for sustained combustion, is typically 290°C and the minimum spontaneous ignition temperature 370°C. Above 250°C the oil will crack into fractions of lower molecular weights and by 500°C it is completely vaporized.

The cross-connected drain tanks were found to contain five parts water and one part oil but allowance must be made for fire hose water entering the starboard gearbox before and after the shafts had been stopped. Separated oil samples examined by gas chromatography and by nuclear magnetic resonance spectrometry showed no evidence of major cracking or degradation, compared to oil samples taken from other ships.

INITIAL FINDINGS AND REMEDIAL ACTIONS

The initial reports attributed the explosion to the gross overheating of the primary pinion journal bearings, due to the blocking of the direct oil supply. This did not, however, explain the fact that the incorrect bearing installation took place four and a half years earlier and that the ship had successfully completed quarterly full-power trials over that period.

The other possible heat source was the pinion thrust failure, which resulted from the journal bearing failures and the subsequent labyrinth ring wear and loss of flooded lubrication. The failure of the starboard pinion thrust prior to the explosion was suspected but could not be proven since it was similar to the port pinion thrusts' failure, which had occurred due to water contamination in the oil after the starboard gearbox had exploded. The direct cause of the explosion was still to be determined.

The evidence of 650°C temperatures in the gearcase vent, together with the report of the organ-like note, confirmed that oil mist was burning and raising the temperature and pressure of the gearbox mixture for at least 20 seconds prior to the explosion. It was initially postulated that the organ-like note and discharge of burning gases had stopped with the exhaustion of the gearbox air and that the subsequent cooling effect of the sprayer oil had allowed an ingestion of replacement air. It was then suggested that the ingestion of air into a hot oil mist might then meet the conditions for spontaneous ignition and explosion. The research however, found little support for the assumption that a gearbox atmosphere is other than lean in terms of the oil mist/air concentration.

Following the *Kootenay* incident, a speed-restriction was imposed on all ships of the class in which the gearing bearings were not fully instrumented with thermocouples and priority was given to installing thermocouples and monitors. Pending the development and supply of new 40-point monitors, ships were fitted with a six-point monitor as an interim measure to sense the three primary pinion bearings, including the thrust, in each gearbox. These bearings were considered to be the most critical.

SIMILAR INCIDENTS

The 'Fraser'

In October 1970 HMCS *'Fraser'* , on night exercises, suffered a total lube-oil failure at 200 rev/min. The turbine lube-oil pump failed and the motor-driven pump failed to cut in automatically. In addition to the sounding of the lube-oil system pressure alarm, the six-point primary pinion bearing monitor registered a temperature alarm.

After stopping and restoring the lube-oil, speed was slowly increased to determine if there was any indication of a change in bearing condition. Speed was briefly increased to 190 rev/min, only to register a high pinion-thrust temperature. The ship subsequently spent approximately six hours at 140 rev/min before returning to port at low speed for a machinery examination.

The turbine and gearing examination revealed a high number of wiped journal bearings and total failure of both port and starboard primary pinion thrust bearings. The port thrust's failure was similar to those in *Kootenay* and had advanced to cause 5.8mm depth of wear in the thrust collar and hard rubbing contact between the end faces of the pinion and the forward bearing.

Fig. 12 Kootenay pinion thrust collar wear

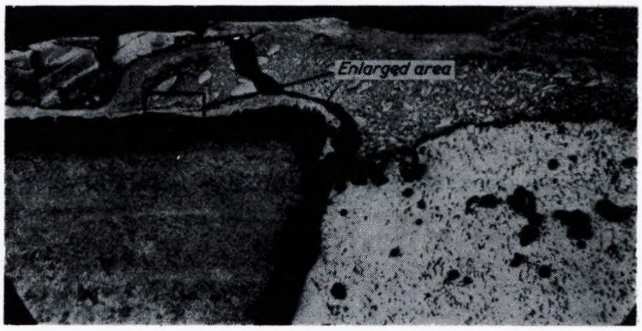


Fig. 13a Melted debris overlaying pinion thrust pad and steel backing ring

Fig. 13b Enlarged area showing penetration of molten thrust bronze into steel backing ring

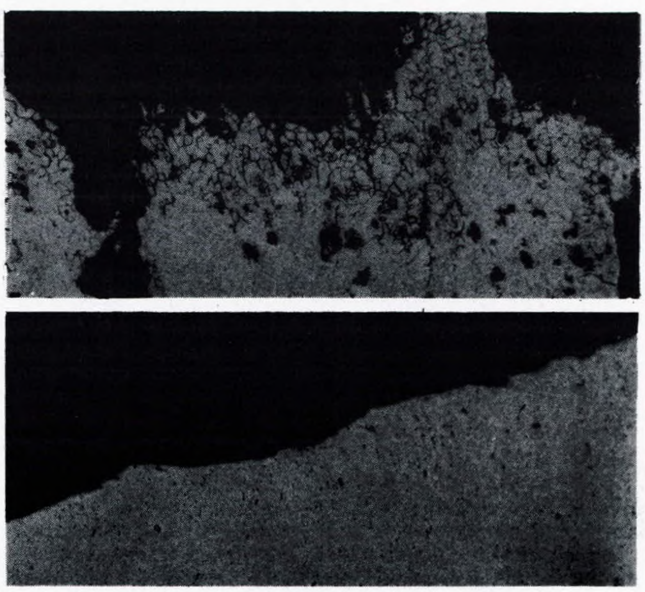


Fig. 14 Micro photos of aluminium turnings (above) from centre of starboard gearbox vent basket (below) from centre of port gearbox vent basket

This incident suggested that a *Kootenay* type thrust failure can be sustained at 200 rev/min without risk of oil ignition or explosion. It supported the initial *Kootenay* findings that excessive overheating of pinion journal bearings, rather than of thrust bearings, presented the greatest explosion hazard.

The 'Skeena'

In June 1971 HMCS *Skeena* commenced post-refit trials, following a gearbox opening for the installation of bearing thermocouples and the six-point monitor. Like all three ships mentioned (other than *Kootenay*), *Skeena* had the atmospheric vent system with the short vent on the gearbox.

From the beginning of the trials there had been a strong smell of overheated oil. After five hours the shaft speed reached 160 rev/min when concern about the cause of the oil smell prompted a reduction in speed. The trials officer investigated the condition of the gearing through one of the acrylic inspection windows. At about 104 rev/min he observed an "orange flash" followed by a small discharge of blackened lube oil from the gearcase vent. *Skeena* had sustained oil ignition and a minor explosion.

The responsible heat source was a failed primary gearwheel aft bearing, which was not covered by the six point monitor. The cause of failure was incorrect installation which resulted in the complete blockage of the oil supply. This bearing axially locates the primary gearwheel/secondary pinion assembly and is positioned between thrust shoulders as shown in Fig.15. An undetected failure in this type of bearing can clearly generate a dangerously high temperature due to the galloping effect of heat causing the absorption of axial clearance which, in turn, increases the heat being generated.

This incident added a new dimension to the assessment of the gearbox explosion risk, since it occurred at a much lower level of power and speed than in *Kootenay*.

The 'Chaudiere'

Within a week of the above incident, a report of oil ignition, resulting from a primary pinion thrust failure in the port gearbox, was received from HMCS *Chaudiere* on the second day of post-refit trials, some 3000 miles away. The gearing had not been opened since December 1969 when thermocouples had been installed in all bearings except the primary pinion thrust. The lube-oil filter canisters had been removed for chemical cleaning and replaced without anyone noticing that the zinc coating had been loosened. The dislodgement of coating downstream of the filter elements was the likely cause of the thrust failure.

On the first day the ship was at low power for five hours before working up to 200 rev/min, the speed limit imposed due to the absence of pinion thrust instrumentation. This speed was maintained for 48 minutes before dropping to 140 rev/min for 50 minutes while returning to harbour.

On the second day the ship worked up to 170 rev/min in three hours, noting a 22°C increase in temperature of the primary pinion's forward bearing and a marked increase in the differential pressure across the filters. On increasing speed to 180 rev/min a loud, high-pitched noise came from the port gearbox. This was recognized as an indication of failure but was not associated with the primary pinion thrust. The noise stopped when speed was reduced.

After six hours at 100-120 rev/min the shaft speed was being increased to 140 rev/min when flashes and burning droplets of oil moving outboard were seen through the primary pinion inspection window. The engines were stopped and the ship returned to harbour on the starboard shaft.

Figs. 16 and 17 show the extent of the pinion thrust failure, together with the overheated area of hard rubbing contact created between the pinion and journal bearing end faces. As in *Kootenay* and *Fraser,* the ahead thrust bearing pads were totally destroyed as the thrust collar had gouged into the steel backing ring, generating considerable heat. The thrust collar was grooved and worn to a depth of 4.6mm.

The loud 'scream' would have occurred immediately the thrust wear had advanced to bring the pinion and bearing end faces into rubbing contact, although the pinion's axial load would still be absorbed mainly at the thrust bearing. The heat generated at the thrust bearing was drastically reduced as the shaft rev/min was dropped to 100-200 rev/min for six hours, so that the pinion length would contract and leave the axial thrust to be absorbed solely at the rubbing end faces. In this condition, the heat generated at 140 rev/min between the rubbing end faces was sufficient to cause oil ignition.

It is a significant that this heat generation was sensed by the

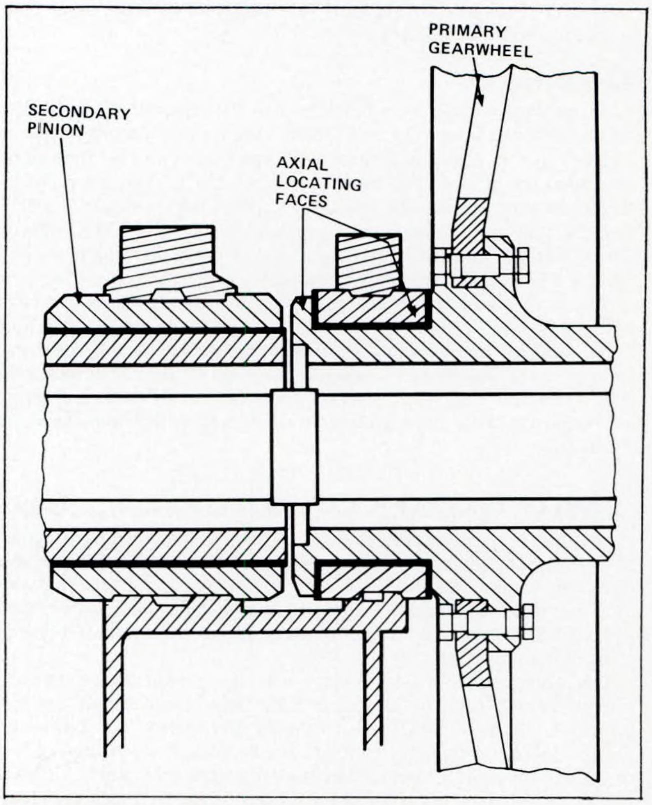


Fig. 15 Gearwheel axial locating bearing

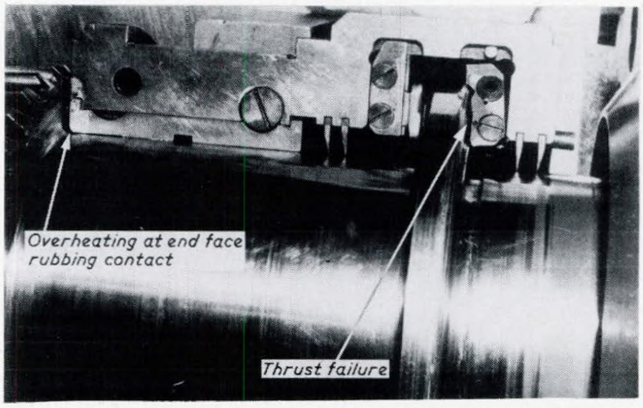


Fig. 16 Chaudiere primary pinion thrust failure

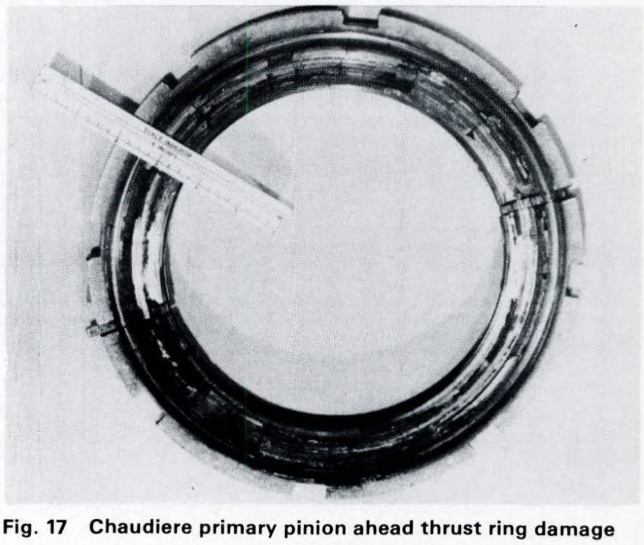


Fig. 17 Chaudiere primary pinion ahead thrust ring damage

thermocouple at the opposite end of the forward journal bearing which was separted from the heat source by an oil annulus.

Pinion thrust bearing friction loss

The Chaudiere incident provided the first indication of the order of frictional power loss caused by a primary pinion's thrust failure. It was concluded that the ship had reached 200 rev/min on the first day of trials with an abnormally high main-boiler steam forcing rate, since it was not expected to exceed 206 rev/min at full power.

The port and starboard turbine receiver and nozzle box pressures generally conformed with a 216 rev/min shaft output. This indicated an overall propulsion machinery power loss at 200 rev/min of the order of 4500 kW (600hp), which had to be associated with the primary pinion thrust bearing failure in the port gearbox.

Problems with the torsionmeters on that day prevented the direct measurement of power transmitted by each shaft. However, assuming equal power being developed by each turbine at a reduced efficiency, and assuming some hydrodynamic power transfer between propellers, the frictional power loss in the port pinion thrust could not be assessed at less than 3730 kW (5000hp).

In this condition, with both shafts at 200 rev/min, the starboard shaft power and torque, and therefore the pinion's axial thrust load, would be approximately 60 per cent higher than those for the port shaft. The coefficient of friction, necessary to produce this 3730 kW power loss, is apparently 3.5, which would have limited the maximum attainable ship speed to 208 rev/min. This may be surprising but is not inconsistent with the high-speed tearing of rubbing steel surfaces generating temperatures of the order of 800°C.

The *Chaudiere* incident showed that a primary pinion thrust failure at 200 rev/min as in *Fraser,* will generate a frictional power loss of 2000-4000 kW which can be sustained for at least 48 minutes in an otherwise normal gearbox condition without risk of oil ignition or explosion.

In fact, these two ship incidents might be taken to indicate that the risk of oil ignition will occur only when the thrust failure advances sufficiently to cause rubbing contact between the pinion and bearing end faces at 140 rev/min shaft speed and above.

'KOOTENAY': THE MECHANISM OF FAILURE

The incorrect installation of the pinion's journal insert bearings, and their continued wear and deterioration over four and a half years, produced wear on the thrust labyrinth ring, which in turn made the pinion thrust bearing failure inevitable. This failure was most likely to occur at high power.

The primary pinion thrust failures in *Kootenay, Fraser* and *Chaudiere,* which all advanced to cause steel-on-steel rubbing contact at the thrust collar, were all similar in character. *Chaudiere* showed that a single pinion-thrust failure would absorb sufficient power to restrict the full-power shaft speed to about 208 rev/min. *Kootenay* was at full power and had reached a shaft speed of 221 rev/min at the time of the explosion. This would not have been possible with a failed pinion thrust bearing. The thrust failure must therefore have occurred at 221 rev/min, immediately prior to the explosion.

Since the ship had previously reached 224-225 rev/min at full power, it must be concluded that the pinion thrust bearing was in the initial stages of failure on reaching 221 rev/min and that the resulting frictional resistance was rapidly building up. Applying the friction coefficient of 3.5, derived from the *Chaudiere* failure, the friction power loss in the starboard primary pinion thrust would reach 6190 kW (8300hp).

Clearly this order of power loss cannot be sustained at 221 rev/min since it must be followed by a reduction in shaft speed and pinion thrust load, until these balance the net propulsion power at the propellers. Thus 6190 kW represents the peak of frictional heat generated in a primary pinion thrust failure at full power.

Since there was no reported reduction in speed, it was concluded that the heat produced by the failed thrust bearing was sufficient to cause immediate oil ignition and explosion

Unlike *Chaudiere,* oil ignition in *Kootenay* was not caused by the rubbing contact of the pinion and forward bearing end faces, although subsequent thrust bearing and collar wear after the explosion did finally just bring those surfaces into contact (Fig. 10c).

Standard mineral lubricating oil of the type in use is stable, non-volatile and not readily rendered inflammable. It must be raised to at least the flash point (243°C for *Kootenay* samples) to vaporise and produce an oil mist, which in turn must attain a concentration in air of about 48mg/l to reach its lower limit of flammability (llf). Oil falling on overheated surfaces will produce oil mist, but if the surfaces are being drenched with oil spray, the resulting scrubbing action will readily disperse the oil mist or keep its concentration below the llf.[1]

The turbulent, oil-drenched atmosphere of a running gearbox, produced by the effects of windage and sprayer oil jets (Fig.2) impinging on fast-moving gear teeth, does not readily support ignition without a heat source of extremely high temperature.

It is considered that the high frictional heat generated by the pinion thrust bearing failure was not only sufficient to form and sustain pockets of flammable oil mist but also to raise it to the spontaneous ignition temperature of 370°C. The temperature and pressure of the burning oil mixture is believed to have increased very rapidly through the process of auto-accelerated combustion. This is borne out by the emission of gasses at 650°C at the gearcase vent and by the reported organ-like note.

The term explosion as applied to the *Kootenay* incident requires qualification. Although the gearbox cover clearly bust open under pressure, it first ruptured along distinct lines of thermal stress concentration, beginning at the inboard side of the primary gearwheel dome. The oil-mist was generated and ignited and carried towards the inboard side of the cover dome by the sprayer oil impinging on the inward rotating gearwheels. The area of highest temperature in the gearbox cover (which was enclosing gases at 650°C) would then be at the inboard end of the dome.

Considering that the coefficient of expansion of the aluminum cover is double that of the steel casing, to which it was bolted, it can be seen that the cover failed primarily under thermal stress at the edges of the dome. The hissing and sparking noises, heard immediately before the explosion, can be explained by the escaping pressurized gases through an initial rupture before the cover blew open. This would also affect the discharge of gases through the gearcase vent and account for the pause in the organ-like note prior to the explosion.

THE RISK OF GEARBOX EXPLOSION

The potential for the ignition and combustion of a gear-box oil atmosphere is created by any bearing failure capable of producing sufficient heat to raise lubricating oil at least to its flash point temperature. Experience nevertheless indicates that the risk of a gearbox explosion is small.

Of the many bearing failures referred to in this paper it is significant that only three produced ignition and combustion. The *Kootenay* primary pinion journal bearings operated in a grossly over-heated state for four and a half years without causing ignition. Major primary pinion thrust bearing failures, involving considerable heat generation, were sustained at shaft speeds up to 200 rev/min in *Kootenay* (port), *Fraser* (port and starboard) and *Chaudiere* (port), all without causing ignition.

On the other hand, oil ignition quickly resulted from a failure in the *Skeena* primary gearwheel journal and axial locating bearing and also from the rubbing contact between the primary pinion and forward bearing end faces in *Chaudiere.* In *Kootenay,* ignition and combustion immediately followed the starboard primary pinion thrust failure at 221 rev/min.

Journal bearings

The ability of a failed bearing to generate and ignite oil mist depends on the nature of the heat source, its magnitude and its intensity or diffuseness.

Safety also depends very much on the effectiveness in ignition suppression of the oil spray, which functions as a sprinkler system dispersing or diluting any oil mist that may be formed. A failed journal bearing may generate sufficient heat to produce oil mist but, under the scrubbing effect of sprayer oil, it is unlikely that the resulting oil mist-air mixture will reach the llf or that the heat source will be sufficiently intense to cause spark ignition. This is borne out in the case of the *Kootenay* pinion journal bearings.

The location of the primary train oil sprayers immediately over each primary gearwheel, as in Fig. 2, rather than at the points of engaging or disengaging meshes, appears to provide better ignition suppression. Approximately 11 per cent of the total oil supply to the gearbox is fed to the sprayers.

No major main gearwheel bearing failures have been experienced in Canadian ships but it would seem that, unless these bearings have the same amount of oil spray protection as pinion bearings, such failures could have serious explosion potential. Such an incident, caused by a main gearwheel bearing failure has been reported.

Pinion thrust and face bearings

Considerably more heat is produced in a non-lubricated rubbing end face situation (as in *Chaudiere* and *Skeena*) than in a journal bearing failure. In these cases a concentrated heat source around the periphery of the rubbing end face will readily ignite the oil mist it is generating, despite the scrubbing and cooling effect of the oil spray. The ignited oil mist will burn around the drops of sprayer oil falling on the bearing to produce the flashes and burning drops of oil reported in these two incidents.

In the case of the primary pinion thrust failures in *Fraser* and *Chaudiere*, a large proportion of the heat generated went into the heat sink of the gearbox structure. At shaft speeds up to 200 rev/min the remaining heat to be dissipated in the gearbox atmosphere was apparently not large enough to cause ignition. At full power, as in *Kootenay*, the heat dissipated to the gearbox atmosphere through the pinion and bearing assemblies becomes sufficiently high to cause the whole oil mist-air mixture to ignite spontaneously. The heat produced in this reaction then causes a combustion wave to advance rapidly through the gearbox with the characteristics of an explosion.

It is difficult, in the light of this experience, to endorse the use of shoulder type axially locating bearings and single-helical pinion thrust bearings. Failures in these two types of bearings at high power, from whatever cause, must be detected immediately to prevent a rapid build-up of heat and a *Kootenay* type explosion. The use of collar type bearings for axial location and of thrust cones to absorb single helical pinions' axial thrust would overcome this problem and avoid any increased risk of explosion in such gearing.

Bearing instrumentation

There can be no question of the need for effective, sensitive and reliable temperature instrumentation in all gearbox bearings to minimize the risk of explosion. Quite clearly there is a problem in the instrumentation of planetary gear bearings, but the need for it should not be lightly dismissed without due consideration of the risk involved.

Monitored thermocouple installations in service have proven to be more sensitive and more reliable than resistance temperature detectors, which may be more compatible with current machinery monitoring information systems. Each bearing temperature alarm circuit is set with a suitable margin over the measured full power reading.

Naval ships do not customarily spend much time at high powers, so it is desirable that an abnormal bearing condition be detected at any power. It is therefore appropriate for alarm levels to be programmed to warn of excessive differentials between bearing and oil inlet temperatures at any power.

Maintenance policy

Main gearing maintenance policy should be based as far as possible on condition monitoring, i.e. gearboxes should be opened only upon reports of specifically identified abnormal conditions. Experience in the Canadian Navy has demonstrated the importance of restricting the occasions for gearbox openings and having them conducted only by designated experienced main gearing inspectors in response to recognized abnormal conditions.

Where bearings are designed to fit in more than one gearbox location and could be fitted the wrong way, it is necessary that the correctness of each particular installation be confirmed by the inspector, and finally checked by proving the discharge of oil.

Oil mist detectors and explosion relief valves

Careful consideration has been given to the question of whether oil mist detectors or explosion relief valves should be fitted to all gearboxes. In view of the understanding now developed of the mechanism and risk of gearbox explosions, and in view of the confidence expressed in the effectiveness of the preventive measures taken, no case for proceeding with the fitting of either device has been found.

CONCLUSION

The Kootenay gearbox explosion was the result of the spontaneous ignition of the oil atmosphere caused by the undetected failure at full power of the primary pinion thrust bearing. This failure is believed to have resulted in the release of several hundred megajoules of energy, culminating in a frictional power loss in the order of 6190kW (8300hp). This was an unprecedented incident which occurred after 20 ships with this type of gearing had been in service for up to 14 years. The effectiveness of the measures subsequently taken to correct the deficiencies of instrumentation and to prevent possible errors of bearing installation have been proven over the last ten years of satisfactory service.

In addition to the provision of reliable and effective bearing temperature instrumentation, future main gearing designs must take into account the risk of explosion and incorporate all reasonable fail-safe features. In particular, oil sprayers should be positioned not only to lubricate and cool gear teeth but also to suppress oil mist formation at all bearings and potential heat sources.

Where the effectiveness of a protective oil spray depends on oil jets impinging on moving gear teeth, an overheated bearing situation will involve a greater risk of explosion when the machinery is stopped. In such a situation, the machinery should be kept turning a low speed until the overheated bearing has been sufficiently cooled.

ACKNOWLEDGEMENTS

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Mr. Nicholson received his engineering education and training at Woolwich Polytechnic and as a student engineering apprentice at the Royal Arsenal, Woolwich. From 1945 to 1947 he served as an engineer officer with the Royal Navy in Ceylon and Singapore. In 1947 he joined the Yarrow English Electric team (forerunner of Y-ARD) to investigate advanced steam power plant in naval construction. This led to the Y100 steam propulsion plant, adopted by the Royal Canadian Navy for the St. Laurent Class destroyers. He joined the Engineer-in-Chief department of the RCN in 1953 to form a Power Transmission Section and became involved with the manufacture of Y100 propulsion equipment in Canada. Since 1973 he has been responsible for marine propulsion systems group at National Defence Headquarters, Ottawa. He is registered as a Professional Engineer in the Province of Ontario.

Discussion

MR I.T. YOUNG (GEC Marine & Industrial Gears Ltd): I have never experienced a gearbox explosion. In the context of the statistics presented by the authors — 20 instances in a total of effectively 15 years — this might not sound altogether surprising. On the other hand, in two of the cases listed, I can remember my own engineer describing to me in detail what had happened, and I can remember inspecting the debris from a third.

The fact that my firm was represented on the Gearbox Explosions Working Party might explain the number of cases with which I am familiar. It also explains my conclusion that such explosions are not as rare as the paper would suggest. Do the authors agree?

The fact, however, that fatal injury occured in only three instances in the 15 year period seems to indicate that the list is realistic in this detail, since death and serious injury are difficult to hide. The minor explosion cases must be much more numerous.

These statistics are the only basis for deciding on positive action. Obviously, defects in design which lead directly to machinery failures should be guarded against, and the authors are right to recommend design reviews to ensure that nothing is forgotten. In the field of monitoring, however, it is difficult to know where to stop, and it is not really true to say that when lives were at risk no expense should be spared. On that basis, no-one would enter a high duty helicopter without a solid 'sky-hook'!

Regarding the design points, no adequate mention was made of venting. Was this no longer considered of importance, or had the authors some recommendation which had a bearing on explosion avoidance?

On the subject of bearings, are not thick rigid shells inherently safer than the thin shell bearing of so many naval applications? Apart from the greater heat sink, an oil annulus cut into the back of the shell with two inlets would tend to make oil starvation much less likely in the event of incorrect assembly. Is this one case, however, where logistics would overrule questions of safety?

Mr Nicholson's mention of bearing location faces, as opposed to tilting pad collar thrusts, seems to be based on inadequate grounds. I see no reason why adequate clearance could not be provided on the inactive face; my experience is that, with proper lubricant supply, well-designed taper-land bearings with moderate loadings can be fully as reliable as the tilting pad variety and indeed as the adjacent journal bearing.

Mr Nicholson's comment on spray oil and its efficacy in dissipating the oil mist before it becomes troublesome raises some practical questions. Does he recommend this as a positive action? To rely on random positioning of sprays designed to cool the gear teeth would seem to be anything but certain in its results. Incidentally, I am surprised by the 11 per cent figure quoted for spray oil. My experience is that, normally, between 30 and 50 per cent of the total would be supplied to the sprayers. Possibly, however, the figure for *Kootenay* referred to primaries only.

Turning now to monitoring, bearing temperature prediction is one of the least reliable arts; actual white metal temperature is so critically dependent on bearing alignment and circularity that comparison between measurement and prediction in general is uncertain within even $\pm 20°C$. A very small local bulge could fool the thermocouple or RTD into predicting a disastrous condition. Too many hours are currently spent arguing out temperatures on test with a customer of fixed ideas.

My conclusion is that for speed of response and accuracy there is nothing to beat the embedded RTD, fitted by the bearing supplier. I grant that it is expensive but it can normally be fitted at the exact angle required: wiring up within the box presents far few problems than the relatively clumsy instruments which have to be screwed in from behind. Thick rigid bearing shells, of course, making fitting of this instrumentation more straightforward.

The manufacturers, who provided gearing to the strict requirements of the American Petroleum Industries' new specification, know to what lengths they have to go to meet vibration tolerances. Non contacting probes are the norm and reliability can only be placed on readings in service if the initial conditions are superlatively good. Would the authors recommend this?

I should be interested to hear the opinions of the authors on the value of gearbox atmospheric pressure monitoring. This might help in the detection of a small explosion which, untreated, could lead to a disaster.

I am puzzled by the recommendation to measure oil mist concentrations during prototype testing. Surely on the basis of evidence presented in the paper there should be no mist whatsoever?

I sympathize with the view that as much damage could be done in routine inspection as could be prevented, but does the author seriously contend that even cursory inspection of the high speed line on *Kootenay* would not have shown evidence of the bearing failure years before it led to an explosion?

Finally, referring to the naval slant to the two papers, are naval conditions such that there is a greater susceptibility to trouble than in merchant marine installations? Alternatively, was it the case that with the cramped engine room conditions the results were more disastrous? The views of Marine Classification Societies on the points raised in the paper would be of great interest.

MR H. SIGG (Maag Gear Wheel Co.): I appreciate that the authors have shed light on oil ignition in gearboxes generally, and on the regrettable accident on board *Kootenay* in particular.

It is important to note that ignition of the lube oil in a gearbox has been observed only after a gear element had broken down and had been allowed to heat up to temperatures far above those existing in gearboxes in normal service.

In the *Kootenay* case a string of events led to the disaster:

(1) The design of the journal bearings made wrong assembly possible.
(2) The high-speed journal bearings *were* wrongly assembled, and completely strangled of oil.
(3) The gear operated in this condition for four and a half years (with the bearings grossly overheated).
(4) Finally, the thrust bearing failed, adding more heat which led to the explosion.

Is it correct in the light of these events to put the entire blame of the disaster on the thrust bearing alone? It seems that this thrust bearing showed great resistance before it gave up in despair.

From experience I know that a well-designed and properly built thrust bearing is as reliable as a journal bearing. There is a large number of single helical pinions in service at speeds two and three times as high as the *Kootenay* pinion and also with powers much in excess. Furthermore, these installations operate under more severe conditions than normally found on board ship.

Generally, a gearbox of any design will suffer destruction if failure of a journal bearing is not detected and the installation is kept in operation. I feel that the only long-term way to safeguard a gear unit from serious damage is to install a reliable and well-maintained monitoring system, linked to an alarm system. Of course, the crew in the control room must react to the alarms promptly and correctly.

Today, extensive monitoring systems are widely used in high-speed industrial applications, especially in the petroleum industry. These one-stage speed increasers are usually instrumented as follows:

(i) Each journal and thrust bearing has a direct-reading thermometer and a thermocouple installed at the hottest point. In two pads of each thrust ring, thermocouples are inserted near the hottest point.
(ii) The lateral vibration at each journal is continously monitored by two radial pick-ups, arranged 90 deg apart. These are inserted from outside and can be replaced whilst the unit is in operation.
(iii) Two axial pick-ups are installed at each thrust bearing, one measuring the vibration, and one detecting the axial location of the thrust collar.
(iv) All thermocouples and vibration pick-ups are connected to an alarm system.

The monitoring of the thrust bearing in these cases is threefold:

(1) temperature;
(2) location, i.e. wear of the white metal of the pads;
(3) vibration.

All three will give an alarm when the preset limit is exceeded.

Such industrial gears nomally have only four journal bearings, and one or two thrust bearings. But a propulsion unit on a warship may have 30 or more bearings. Usually, these gearboxes are of a rather complicated shape making it difficult to install even direct-reading thermometers. In any case it is impossible to reach the hottest point of a bearing with temperature sensors inserted from outside the gearbox. The same applies to vibration pick-ups. All these instruments can be installed or replaced only when the gear covers are lifted.

The sensors available today are quite reliable. But because of the large numbers required, we may be obliged to open up the gear casing rather frequently, just to keep the monitoring system in workable shape. Thus, we should ask ourselves whether the thermocouples for the remote control and alarm system really have to be installed at the hottest point. Is a sensor which can be replaced without opening up the gearbox not more desirable even though it does not measure the hottest point?

The *Kootenay* gear was designed in about 1950. Thirty years ago a suitable remote temperature control was not available. Each journal bearing was equipped with a dial-type mercury thermometer, which measured the temperature close to the white metal but not at the hottest point. It would be interesting to know what temperatures were recorded by these mercury thermometers before and after the incorrect mounting of the high-speed bearings.

It seems to me that these bearings, when totally starved of oil, would have been extremely hot all around, so that an alarmingly high temperature should have shown on a direct-reading thermometer placed at any location.

Do the authors feel that the life expectancy of temperature sensors available today is adequate so that the dismantling of gear covers and bearings does not become necessary too frequently? If the answer is 'yes' I see no other reason not to place one or more sensors near the load lines at the hottest points.

MR W.H.HARRISON (David Brown Gear Industries Ltd): Reports of several incidents involving fires and minor explosions in gearboxes have emerged since the formation of the MOD (N) working party on gearbox explosions. Unfortunately, few investigations went beyond identifying the cause of such malfunction or component failures which had furnished the heat sources. Defects were quickly rectified and machines were returned to service.

The following background information might afford a better appreciation of the very low risk of serious explosions in gearboxes.

On no less than seven ocasions, the rolling mill gear item 10 of Table I suffered failure of tapered roller bearings of the fifth or sixth stands of a six-stand drive. An explosion occurred on the fifth incident of bearing failure.

On opening five gear units of a particular type which had suffered damage in service, it could be seen that the second reduction gears had been very hot. Detailed examination showed that, over more than 50 per cent of their surface area and to a depth of *c*.1.5 mm, the temperature of the gear teeth had risen to more than 600°C. Fire was reported in only one instance.

These incidents indicate that malfunction and the existence of heat sources quite often do not cause ignitions in gearboxes.

In the case of the *Kootenay* explosion, it would appear that had the gear been double helical instead of single helical, or had the gearcase cover been of steel rather than aluminium, the incident might well have been less disastrous.

With double helical gears there would not have been a thrust bearing for the primary pinion. A steel cover would have been less likely to rupture because of differential thermal expansion. Had the cover remained intact there would have been no spread of flame through the engine room and, probably, depletion of oxygen would have extinguished the preliminary fire.

Despite the severe conditions engendered by faulty installation of journal bearings, the primary pinion thrust bearing survived more than four years of operation before damage became so extensive than an explosion ensued. As Mr Sigg pointed out, this shows that combining single helical gears with tilting pad thrust bearings is quite sound. Non-contacting position monitors provide useful back-up for temperature monitors to warn of abnormal conditions.

Designs that enable incorrect installation of components such as bearings, whose malfunction can have dangerous consequences, are, to say the least, most unfortunate. The importance of preventing faulty installation must be impressed continually on designers and maintenance personnel in training.

MR H.A. CLEMENTS (SSS Gears Ltd): I am grateful to the authors for giving so much information regarding operating experiences. Too few of us learn from history and from other people's mistakes. There is a great deal to learn from the events which led up to the gearcase explosions described by Cdr. Cooper and Mr Nicholson.

A gearbox explosion can be an extremely serious occurrence, particularly as happened in HMCS *Kootenay*. The circumstances which occurred are possibly the type that could occur under battle conditions, with which naval ships have to be designed to contend. The ability of engineers to think quickly, recognize the problem and improvise could make all the difference between winning or losing the battle. I fear that, with more complicated designs and automatic controls, this concept is being lost.

The principal requirement is to avoid as many rubbing connections as possible. Unfortunately, there must be bearings to support shafts. In the USA, land-based turbine and compressors use tilting-pad journal bearings to a much greater extent than in Europe. Such a bearing is basically more expensive, but the principal bearing manufacturers have now standardized their designs. If a standard size bearing is selected, the price is not too high compared with a normal sleeve bearing. The tilting-pad bearing was adopted primarily to overcome oil film whirl problems in light-loaded, high-speed applications but this bearing has a number of merits which could be beneficial in marine gearboxes:

(1) The bearing is loaded; therefore, there is less risk of shaft contact under reduced oil flow conditions.
(2) The bearing can accept load in any direction.
(3) The bearing is not critical regarding oil inlet position.
(4) If a central pivot point is used, the pads cannot be wrongly assembled.
(5) During maintenance, only the pads need to be changed and the cost of standard pad is quite low.

The losses may be slightly higher than with a sleeve bearing, but this seems a small price to pay for a bearing able to operate under more adverse conditions. Could the authors state whether such tilting pad bearings have been considered?

The authors mention manual clutches. These are tooth clutches, which are engaged or disengaged at standstill by a gearcase-mounted servo mechanism, shifting the clutch sliding part by a fork mechanism. I believe such mechanisms should be avoided if at all possible, as the forces which can act on the shifting fork with its thrust bearing can be very high. Preferably, the teeth should be locked in engagement in some way and the control fork unloaded. Alternatively, in some cases, the operating cylinder could be within the clutch rotating parts to avoid all rubbing connections.

The authors may know that, with SSS Clutches, positive torque holds the clutch into engagement and it does not rely upon servo mechanisms. I believe this is a most important feature, to give high reliability and to avoid highly-loaded bearing surfaces. Where a locking sleeve is essential, i.e. in bi-directional drives, the sleeve can be held in position by centrifugal oil pressure within the clutch itself and the control fork completely unloaded.

Oil-cooled friction clutches and brakes are likely to give the conditions for a gearcase explosion, particularly under some adverse circumstances, such as reduction of oil flow. For this reason, the hydraulic coupling, as selected by the RN for reversing gearboxes, has considerable merit, having no rubber parts. I am surprised there was an overheating problem in an empty hydraulic coupling with 200 per cent slip. The losses at 200 per cent slip are, in general terms, very low.

Mention is made of the disadvantages of shoulder-type thrust bearings. Normally, the axial clearance is made quite small, whereas, if a large clearance is used the losses are considerably reduced. Therefore, the heat generated is lower.

Presumably, different metals rubbing together promote varying fire hazards. Aluminium is dangerous in this respect. Could the authors comment please?

CDR. D. BAKKER (RNLN, Ministerie van Defensie): I agree with the recommendations of the Working Party, especially with points 8.1.4, 8.2, 8.3, 8.4 and 8.10.

Regarding recommendation 8.8, in my opinion it is more important to raise the ductility and impact toughness of the material of the gearbox cover. For instance, cast aluminium alloys could be replaced by welded AISI 316L.

In addition to recommendation 8.9, the watch-keeping instructions should also be revised, in order to prevent incidents such as overheated bearings, especially for post-refit trials.

On the subject of venting systems, as mentioned in recommendation 8.10, one solution could be to raise the bore of the vent system and place a suction vent directly above the vent basket.

MR F.A. MANNING (MIMarE): The authors are to be congratulated on making public the findings of the MOD Gearbox Explosion Working Party.

Table I lists particulars of gearbox explosions. Item 3 in this table, relating to SS *Verena* (March 1962), alleges that a major explosion occurred with oil ignition inside the gearbox, resulting in external fire.

I have no knowledge of any other explosions but as a member of the team investigating the SS *Verena* fire I can assert that there was no explosion. There is no doubt that the accident was a tragic disaster but the reports of the events specifically avoided mention of 'explosion'.

The broken half of the flexible coupling smashed its way out of the gearbox casing, and a lubricating oil fire then followed. It appears to be a gross exaggeration to claim that ignition of oil spraying to atmosphere and into the open gearbox could be interpreted as even a minor explosion.

It seems uncharitable to say that such matters are forgotten. At the time, a comprehensive investigation was carried out. As a result, design approval and instructions for inspection of gearbox components were revised, to the benefit of the UK merchant shipping industry as a whole.

Authors' Reply

A REVIEW OF MARINE GEARBOX EXPLOSIONS

The authors are grateful for the contributions to the discussion, many of which support the conclusions. In reply to specific questions we have the following additional comments.

Gearbox explosion incidents

Mr Young's question, concerning the frequency of explosion-like incidents, intrigues the authors. A glance at Table I suggests that, statistically, warships seem to have more than their fair share. This might be a function of tight reporting enforced in any Service organization contrasted with the more diffuse data gathering methods of others. It might also be a symptom of the innovative pressures on an armed force compared with the well-tried conservative approach of the commercial market.

The Working Party made a thorough search of all sources of data and only those incidents presented came to light. We believe that the evidence available supports their conclusions but would be glad of any further evidence to lend emphasis.

The authors are also grateful for Mr Manning's contribution concerning the *Verena* incident. As we pointed out in our presentation, the term 'explosion' had been used rather loosely and did not distinguish between a breach of a gearcase as the result of mechanical derangement and a breach due to violent mist combustion. In many cases the evidence available to the Working Party did not permit such a distinction to be drawn.

Bearings
Where thick wall bearings are concerned, the paper highlights one case where a thick wall bearing may have contributed to an explosion. As the result of the large heat sink a dangerously high metal temperature was not indicated by a thermocouple remote from the affected area. Present naval designs feature both types of bearing and we believe there are no significant differences in intrinsic safety between bearing types. The thin shell bearing is therefore preferred by the RN for logistic reasons.

With regard to the comments of Mr Clements, tilting pad journal bearings have advantages where stability is required, particularly in high-speed drive lines, but their selection is affected by the disadvantages of complexity, space and cost which must be taken into account. They have not been used before in naval gearboxes but their use in future will be considered.

The bearings in clutch control forks can certainly be a source of trouble, as Mr Clements suggests, and we have experience of the problem. Where there is an alternative available, avoidance is advocated.

Thrust bearings of the tilting pad or taper land variety are widely used in RN applications and the design rules are well understood. Single helical gears demand a suitable thrust bearing to absorb axial thrust and the reliability of the train is critically dependent on its performance. As Mr Nicholson has pointed out, they are the first to suffer if oil supply fails. The thrust cone is available as an alternative and we believe that this form of thrust/location bearing deserves more consideration.

Gearbox ventilation
Some work on gearbox venting was sponsored by the Working Party in conjunction with work on the Controlled Atmosphere Test Facility. This is reported more fully in Ref. 1 of the paper. There is no conclusive evidence of a link between vent size and disposition and explosion risk. A link was suspected in the case of the *Kootenay* incident but this has since been discounted. Current RN practice is to make provision for the gearbox to 'breathe' rather than to be ventilated. This eliminates the risk of broadcasting potentially flammable vapours more widely. Dehumidification systems are added to ventilate the gearboxes with dry air when shut down only.

Conditioning monitoring
This is a difficult area, particularly in warship gearboxes, and a balanced judgement is necessary. Having taken steps to ensure design quality, condition monitoring in service remains a necessity. A comprehensive outfit of condition monitoring equipment of all types is suggested in order to reduce the need for gearbox strip and inspection and the attendant risk of maintenance-induced defects. Such equipment should fulfil both of the following requirements.

(a) It must give a clear and unambiguous report demanding action on the part of the plant operator, who is assumed to be unskilled in diagnosis and interpretation.
(b) The monitoring equipment should not demand gearbox strip for maintenance and calibration more often than the gear elements.

As Mr Sigg correctly points out, RN gearboxes are seldom simple, involve many bearings and are subject to alternative loading modes.

If routine dismantling of bearings is to be avoided, some form of condition monitoring is essential. In the great majority of cases the measurement of bearing metal temperature will provide all that is required in terms of bearing health. We can confirm that adequate reliability has been achieved both with resistance themometers and with themocouples. We believe that the positioning of the sensor is important, not only as a safeguard against explosion but also as a means of minimizing damage should some form of failure occur. Bearings which are subject to several different loading directions may require more than one sensor for this reason.

The justification for fitting permanent vibration monitoring equipment is less clear and current thinking would limit the use of permanently installed accelerometers to the first input bearing of main gearboxes, where early warning of any flexible coupling deterioration is sought. We do not consider that the widespread use of non-contacting displacement probes is appropriate to health monitoring of RN gearboxes as the cost and the likelihood of false alarms are thought to outweigh the potential benefits. Such instruments have, however, proved to be most powerful tools for investigating vibration and alignment problems in specific installations, such as first of class.

One possible future development concerns the use of periodic vibration analysis. The rapid developments in data processing may soon permit an assessment of gearbox health to be made 'automatically', by a periodic check of vibration at a single point on the gearcase using a sophisticated portable analyser. A datum signature would be recorded on tape during contractor's sea trials. Developments in this area are being watched with interest but for the moment our preference would be to avoid cluttering the gearbox with many permanently wired transducers.

The possibilities of gearcase pressure monitoring suggested by Mr Young have not been explored.

Materials
With reference to the comments of Cdr. Bakker and Mr Clements, aluminium is already a prohibited material in RN gearing installations as a result of its poor high temperature properties; Mr Nicholson's paper gives an example of earlier practice. Cast iron is also prohibited due to its lack of shock resistance. Of the remaining materials left within a gearcase, a failure leading to rubbing contact is a hazard in itself and the material characteristics in this situation are only of second order interest.

THE 'KOOTENAY' GEARBOX EXPLOSION

I am most grateful for all the contributions which clearly bring out many of the aspects of concern or interest to the gearing industry with regard to the risk and avoidance of gearbox explosions. The number of gearbox explosions and significant related incidents which have occurred undoubtedly exceeds those which came to the attention of the Gearbox Explosions Working Party (Ref. 1). As Mr Young suggests, the cases of minor incidents can be expected to be very numerous. On that basis, there can be no question about the need for designers, maintainers and operators to review the adequacy of their current practices.

No evidence has been found that gearbox venting has or can contribute to the risk of explosion, say through the ingestion of air into an overheated oil mist situation. The entrainment of air in the oil discharging from a running gearbox tends to keep the gearbox atmosphere slightly below the engine room pressure. Except where the gearbox vent is also connected to the drain tank, as in *Kootenay*, there is normally a small steady ingress of air through the vent into the gearbox, while there is a corresponding emission from the drain tank.

The ingress of air into a running gearbox, where an overheated bearing is generating oil mist, was originally seen to increase the risk of ignition and explosion. This was on the assumption that the oil mist mixture was deficient in air and above the upper limit of flammability. The ingress of air through the vent was then seen to be capable of bringing the mixture into the flammability zone and triggering spontaneous ignition. It is now known from the work reported in Ref. 1 that, under the scrubbing action of the oil spray, any oil mist formed in a running gearbox will generally be kept below the lower limit of flammability. The ingress of air would just increase the leanness of the mixture.

The use of combined gearbox and drain tank vents was being discontinued in the Canadian Navy prior to the *Kootenay* explosion to avoid the discharge of oil fumes into the engine room. The drain tank, which emits the oil fumes, is vented to atmosphere. Both port and starboard gearboxes in *Kootenay* were known to give a noticeable discharge of oil fumes above half power but this was not reported to be greater than in other ships of the class.

Mr Young's preference for thick shell bearings over precision insert bearing shells is contrary to my experience which has shown the insert shell to be very much more reliable, easier and more economic to support. Certainly the ability to use a common insert shell in a number of bearing locations having different angles of rotation offers significant logistic and support advantages.

Questions of safety regarding the risk or possibility of incorrect installation applies to all types of bearings and involves a responsibility to be thoroughly addressed by the designer and the in-service maintenance authorities. The insert shell design deficiency that permitted the incorrect installation in *Kootenay* was eliminated in the follow-on ships and has given no trouble. Although bearings are designed to prevent incorrect installation, the ingenuity of those intent on using the wrong bearing, when the right one is not available, can be frightening to behold. Under no circumstances can logistic considerations be allowed to compromise safety.

It was interesting to find that none of the contributors were prepared to fault the use of journal bearings with axial locating faces, providing that adequate clearance was allowed to eliminate the risk of overheating and seizure. I do not object to this approach but would merely observe that the axial clearance in this type of bearing, used in locked-train double reduction gearing, is usually kept to the minimum consistent with adequate lubrication.

I am concerned about the apparent unquestioned use of tilting pad type axial thrust bearings in high-speed single-helical pinion applications. In a normally functioning gearbox and lubricating oil system, the tilting pad type axial thrust bearing gives a very reliable performance. However, depending on the speed and specific loading, this type of bearing can be particularly susceptible to failure due to an interruption in the oil supply or to lube oil contamination. Experience in *Kootenay, Fraser* and *Chaudiere* shows how rapidly a primary pinion tilting pad type bearing failure can advance at high power to generate quickly dangerously high levels of heat. Regardless of the instrumentation used to detect any abnormal condition in high speed tilting pad thrust bearings, I would suggest that the maximum PV value be limited to about 110 MPa.m/s to avoid the possible order of heat generation experienced in *Kootenay* in the event of failure.

for the purpose of oil mist dispersal as well as for the lubrication and cooling of gear teeth. My point here is that the oil spray, created by oil jets impinging on fast moving gear teeth, has all along been serving a vital, albeit an unintentional, function of containing and dispersing any build-up of oil mist. It is fortuitous that the greatest concentration of oil spray is created in the upper part of the gearbox where the high speed bearings, being the highest potential heat sources, are usually located.

Since there is now undeniable evidence of the scrubbing action performed by the oil spray, I consider that it is now encumbent on the gear designer to arrange that this protective oil spray is directed to cover all gearbox locations where oil mist could be generated. The need for oil spray protection against oil mist is particularly needed at the main gearwheel bearings and would most probably have prevented incident no. 13 in Appendix I of Ref. 1.

Mr Young's surprise that only 11 per cent of total oil supplied to *Kootenay*-type main gearing is directed to the sprayers is of interest since the same observation has been made by other authorities. While 11 per cent is the design figure, measurements taken during prototype shore testing showed that the primary and secondary sprayers together took up to 13 per cent at full power. Certainly no greater proportion of the oil supply has been found necessary for the satisfactory lubrication of the gear teeth. I would suggest that the delivery of 30-50 per cent of the total oil supply to the sprayers is excessive if the bearing temperatures are considered to be fully satisfactory. I certainly feel that Mr Young should find little difficulty in providing sufficient oil spray to contain oil mist generated by every potential heat source in the gearbox.

Mr Young is correct in commenting on the unreliability of gearing bearing temperature prediction. To minimize the risk of spurious alarms, it is the practice in the Canadian Navy to set temperature alarm levels above the actual accepted full power values measured in each gearbox. However, as already stated, the monitoring of temperature differentials between the white metal and the lube oil supply over the power range is considered to be a better approach to the condition monitoring of naval gearing.

Adequate speed of response and accuracy of temperature measurements have been obtained with RTDs fitted in the bearing backing shell, to bear on the back of the insert shell at or close to the ideal angular position. The incidence of spurious alarms, or failure to detect minor wiping of bearings instrumented with RTDs, has not in my experience been high enough to warrant the increased cost and possible obstruction to bearing inspection, created by having them embedded.

Notwithstanding the possible need to satisfy stringent vibration standards in naval gearing performance trials, the need and value of continuous vibration monitoring might well be questioned, as might the reliability and suitability of available sensors and monitors. On the question of monitoring gearbox atmospheric pressure, it is considered unlikely that this would give a warning sufficiently in advance of a major or minor explosion in a vented gearbox.

Mr Young's question on whether a cursory inspection of the *Kootenay* gearing would not have revealed the pinion bearing failures is difficult to answer. A cursory inspection would involve the lifting of the inspection doors primarily to examine the condition of gear teeth. Considering that the insert bearing shell white metal lining is just 0.7 mm thick, it is most unlikely that any evidence of its failure would readily be seen with the pinion in position.

Full gearbox openings involving the lifting of covers and inspection of selected bearings were not scheduled more frequently than every four years and are now undertaken in response only to unsatisfactory condition reports. It is important, however, to record that a report of a suspected increase in the *Kootenay* starboard gearbox vibration level was received a few months prior to the explosion. Sufficient confidence in vibration analysis techniques in use at that time did not warrant an immediate or emergency gearbox opening prior to that which was already scheduled for the refit later in the year.

The case put forward by Mr Clements for the use of tilting pad type journal bearings would appear to have a limited application in marine gearboxes, except for lightly loaded high-speed bearings supporting drive shafts and clutches where oil whirl problems might occur. They might also be considered where a two-inlet plain journal bearing could not be designed to satisfy a multi load-line situation.

The use of friction clutches and brakes in gearboxes is seen to involve an explosion hazard in the event that the generation of heat and oil mist exceeds the dispersal capability of the available cooling oil. By the same token, I am not convinced that Mr Clements is correct in implying that there is not a similar risk in hydraulic couplings which have no rubbing parts. A lube oil failure tends to create rubbing surfaces where none should exist. A flammable and explosive oil mist mixture can and has been produced in hydraulic couplings, which by their very nature cannot benefit from the scrubbing action of gearing oil spray.

Mr Clements raised the question of the possible fire hazard created by the use of aluminum in rubbing contact with other metals. Aluminum, in common with other low molecular weight metals, when struck with a sharp, harder metallic object tends to release small particles or slivers with sufficient heat energy to ignite a flammable gas mixture. For this reason the use of aluminum is prohibited in potentially explosive atmospheres such as in mines. The same danger does not exist in the oil atmosphere of a marine gearbox, although it would be prudent to avoid the use of aluminum in all gearbox components that could experience rubbing contact.

Although Mr Harrison does not go so far as to compare the risk of explosion between single and double helical gearing, his comments underline the fact that the use of tilting pad bearings to absorb the single helical primary thrust involves a risk that is not found in double helical gearing. It is my view that this risk can be kept to an acceptable level by design and instrumentation or it can be eliminated by the use of primary gear train thrust cones.

While the use of fabricated steel covers in place of aluminum is fully supported, it is unlikely that it could have provided sufficient strength to withstand the rapid pressure build-up that would have occurred in *Kootenay*. This view is substantiated by Cooper, Holness and McNeill (Ref. 1).

Mr Sigg and Mr Harrison each suggest that the primary pinion thrust bearings in *Kootenay* did well to withstand the conditions created by the journal bearing failures for four and a half years. While this is not untrue, it should be understood that the direct cause of failure, being the loss of adequate flooded lubrication in the thrust chamber, occurred in the final minutes of operation. This is borne out by the extent of the port pinion thrust failure which occurred due to water contamination after the starboard gearbox had exploded. The evidence clearly shows that the gearing can tolerate pinion journal bearing failures for several years until they trigger a pinion thrust failure.

Regardless of how the reliability of tilting pad axial thrust bearings is assessed, there is clearly uniform agreement on the need for effective and reliable condition monitorig for all gearing bearings and particularly high-speed thrust bearings. Mr Sigg describes the extensive monitoring system employed in petroleum industry high-speed two-element gearing applications with multi temperature and vibration sensors in each journal and thrust bearing. Unless there is concern for the reliability of either the gearing or the instrumentation, this would seem to be an overkill approach which could not be justified or even physically accommodated in a compact and complex naval gearing installation.

The minimum requirement of one RTD or thermocouple for each journal and thrust bearing should be augmented with one or more additional sensors, depending on the bearing size and maximum load-line angular spread. The use of two thermocouples in high-speed thrust bearings is fully supported. The value of direct-reading bearing thermometers in naval gearing is very questionable on the basis of poor sensitivity and incompatibility with continuous monitoring concepts.

The use of axial location probes to warn of high-speed thrust bearing failures is strongly endorsed. The case for fitting vibration sensors would appear to be justified where anti-friction bearings are used, but for all white metal bearings it is considered that temperature sensing should provide the most reliable method of condition monitoring. Canadian Navy experience does, however, support the case for designing portable vibration probe access points at certain bearing locations.

Mr Sigg questions the need to locate temperature sensors at the bearing's hottest point rather than at locations which would permit sensor replacement without requiring a gearbox opening. In spite of the claims for sensor reliability, the need for their replacement can well exceed the need to replace bearings. From this standpoint, some relaxation in locating for maximum sensitivity to achieve ease of replacement may be justified. This would be less acceptable in bearings which require more than one sensor to give adequate temperature indication over a wide range of load-line conditions. Where sensors cannot be located to achieve ease of replacement without seriously affecting their sensitivity to detect an incipient failure, the fitting of two sensors should be considered.

With regard to the dial-type thermometers originally fitted in *Kootenay* primary pinion bearings, prior to the use of the insert shells, readings at full power were typically 21°C above the oil inlet temperature. Thermocouples positioned as shown in Fig. 4 give typical readings of 44°C above the oil inlet temperature. The dial-type thermometers were measuring something akin to the bearing keep temperature. The fitting of the bearing inserts precluded the use of the originally supplied thermometers and separable sockets. No temperatures can therefore be quoted after the incorrect bearing installation took place. Bearing in mind that they had failed to indicate any of the early primary pinion bearing failures, as in Fig. 3, it is unlikely, had they been fitted, that a highly significant increase in temperature would have been indicated during the initial period of operation. There is of course no doubt that had thermometers been fitted and were still functioning during the pinion thrust bearing failure, that a very significant temperature increase would have been indicated.

In closing it is perhaps appropriate to address Mr Young's question as to whether naval conditions presented greater susceptibility to trouble than applied to the mercantile marine. While naval gearing is generally more complex and is designed for higher load conditions than mercantile gearing, it cannot be said to be less reliable or inherently more prone to trouble. There is, however, a need for naval gearing to be constantly pushing the state of the art to achieve improved performance over a wider range of operating conditions and to accommodate advances in propulsion system concepts.

The state of the art can be advanced only on the basis of objective analysis and understanding of machinery failures such as the *Kootenay* gearbox explosion. Since this requires the willingness of users to disseminate and discuss such information, it follows that incidents involving naval machinery may appear to be more prevalent than mercantile machinery incidents. Although a contribution from a Classification Society would have been most helpful on this matter, it is suggested that merchant ship owners might in general be less interested in reporting and discussing their machinery problems.

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