LUBRICATED ROLLING BEARINGS

ANALYSIS OF DATA FROM FIELD TRIALS AND LABORATORY RIG TESTS

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Introduction

Various authors have described the types of failure which occur in rolling bearings (Refs.1 to 8). Discussion of failure rates and of the importance of any particular cause of failure must be based on a number of failures large enough to be significant. This article discusses the problems involved in obtaining reliable detail from surveys of failures in ships and deals with some of the problems besetting those who endeavour to produce rig test procedures for the evaluation of lubricants to be used with rolling bearings.

Surveys of Bearing Failures

Organization

Two surveys were made to establish the failure pattern in rolling bearings of electrical machinery. The first involved two frigates straight from refit who were instructed to forward all failed bearings to the AOL with details of their use and the nature of the trouble experienced. Only four hundred bearings used in a variety of fans, pumps, generators, etc. were at risk. Therefore, even after a trial period of two and a half years, only a broad picture was obtained.

The second was a major survey of the failures of grease-lubricated rolling bearings of electrical machinery in most of the ships of the fleet. This was conducted for one year and involved the active co-operation of the Ship Maintenance Authority (SMA), who analysed ships' reports, while the AOL examined the bearings and analysed the overall failure rates and failure types (Ref. 8).

Various estimates of bearing failure rates existed, although usually based on a limited number of incidents only; it was desired to arrive at more reliable conclusions as to not only the real overall failure rate but also the failure rate in specific areas. This was appropriate because, after many years using an old style uninhibited soda-base grease, the Navy had adopted a multi-purpose grease XG-274 which contains both oxidation and corrosion inhibitors (See Ref. 9 and Appendix I).

At the time that the first survey was being planned, the following factors were relevant:

(*a*) Some ships were known to have been too enthusiastic with the grease gun, resulting in over-lubrication of those bearings which has inadequate means of venting the excess grease and so a consequent risk of over-heating and early failure.

(*b*) The suspicion that dirt could be introduced through re-greasing under adverse conditions which in consequence was leading to the deliberate avoidance in some cases of periodic relubrication between refits.

(*c*) The interest being aroused by the availability of 'lubricated-for-life' bearings.

The Director of Engineering (Ships), therefore, planned for the minor survey in which two ships were to go to sea after refit with all bearings lubricated with XG-274; one third to be relubricated at regular intervals, one third to be standard bearings without relubrication, and one third to be 'lubricated-for-life' bearings. Only a limited number of 'lubricated-for-life' bearings could in fact be fitted, these mainly in pumps of various types and none in the less accessible axial-flow fans. In the event, of the bearings under trial 214 were to be relubricated periodically and 188, including 50 'lubricated-for-life', were not to be relubricated. Unfortunately not all bearings were packed with fresh grease as had been hoped; all the 'lubricated-for-life' bearings were packed with XG-274, but some of the remainder contained a prototype XG-274 (issued while the specification was being finalized and which did not meet the rust-preventing requirements later laid down) and a few still contained the old soda-base grease.

The ships involved, which served in both home waters and overseas, made adequate reports of failures and returned almost all the suspect bearings. A standard report form was used. This was later modified in the light of experience before the major fleet survey started. The later version of this is shown in Appendix II.

The major survey involved surface ships of all sizes from aircraft carriers to frigates. With 39 000 bearings at risk, it was hoped to do much more than derive an overall failure rate. The SMA were particularly interested in a detailed study of the data.

The Two-Ship Trial

Of the 47 suspect bearings sent in by the ships over a period of two and a half years, 31 were classed as failures of various severity. In addition, three reports of definite failure were not supported by bearings for examination. Thus, a total of 34 failures out of 402 bearings at risk gave a failure rate of 8·5 per cent or 3·4 per cent per annum. Much the greater part of these failures came in the first year, the number of failures in successive six-month periods being 10, 13, 4, 3, and 4. Practically all the returned bearings, the suspect condition of which was not confirmed by the laboratory, were sent in during the first year. There would appear to have been a marked decrease in real failures as the machinery settled down, and perhaps also a more tolerant attitude to minor bearing noises.

Although ships were asked to report running-hours to failure, they were not always able to do so particularly with equipment which runs intermittently. From the figures reported: only one bearing failed under 2000 hours and that was from mechanical faults; sixteen failed between 2001 and 5000 hours, and seven between 5001 and 10 000 hours; two ran for just over 10 000 hours, and one for just over 15 000 hours.

The total number of bearings sent for analysis was too small for other than broad conclusions to be drawn. It was noted that sixteen (7 per cent) of the bearings which had been periodically relubricated had failed against eighteen (10 per cent) of the remainder, of which the fifty lubricated-for-life bearings contributed only two—one from misalignment and the other from severe fretting due to turning in its housing.

The commonest fault was corrosion, the more spectacular examples of which came from an evaporator pump with a failed water seal, a fan at deckhead level operating in tropics, and a bridge supply fan fully exposed to foul weather conditions. Various compressor bearings had suffered from corrosion, and also another severe corrosion failure was the non-drive-end bearing of a fan mounted on an extension shaft in a cold room. This latter failure was aggravated by excessive whip in the shaft. About a third of the bearings received for examination contained grease older than the approved XG-274, and these bearings had a disproportionately large number of corrosion failures due not only to longer usage but also to the effect of grease without a rust inhibitor.

Several bearings failed from mechanical faults such as: misalignment, out-of-balance rotating loads, or excessive axial displacement due to overloading, leading to flaking and fatigue. False brinelling led to failures of:

(*a*) a bearing in a turbine turning motor (FIG. 1).

(*b*) the bearings in a refrigerator pump run week about with another.

(*c*) the bearings of a compressor run on alternate days with another.

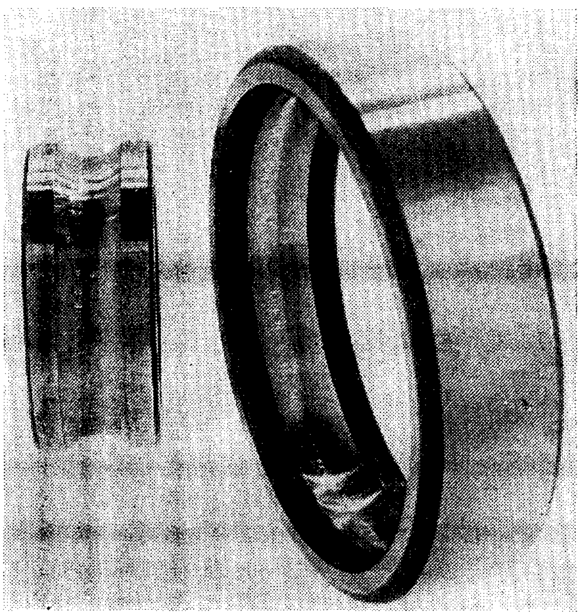

FIG. 1—FALSE BRINELLING CAUSED BY VIBRATION

The damage to the tracks of this bearing from a turbine turning motor occurred while the motor was stationary and was caused by vibration from the rotating turbine.

One of these bearings had a pronounced washboard effect from rust and vibrating loads.

Fretting corrosion in the form of movement between the shaft and the bore, or between the housing and the outer diameter of the bearing was found in a few cases but in only two was it thought to have been a genuine poor fit rather than a result of increased friction in the bearing after it had failed from other causes.

Dirt in various forms found in a number of bearings was a common cause of defect; it led to indentations and complaints of noise rather than to spectacular failures and was often associated with corrosion or other defects.

The various defects expressed as percentages of the suspect bearings received are set out in TABLE I. The numbers were, however, too small for many conclusions to be drawn from them.

TABLE I—*Main causes of failure in bearings received from ship trials*

	Two ship trial		Fleet survey			
	47 received		First 100 received		All 614 received	
	All bearings per cent	Bearings with XG-274 per cent	All bearings per cent	Bearings with XG-274 per cent	All bearings per cent	Bearings with XG-274 per cent
Bearings failed	66	60	97	97	97	98
Bearings failed badly	15	17	45	45	50	48
Bearings too badly damaged for cause of failure to be assessed	0	0	7	6	3	2
Bearings with fractured components	6	6	12	13	6	6
Bearings flaked	15	20	33	30	29	29
Bearings with severe corrosion	4	3	14	14	13	11
Bearings not filled with XG-274 to DGS 6921A	36	—	37	—	29	—
Bearings with following main causes of failure:						
corrosion	28	14	35	35	39	34
dirt	17	17	21	21	18	20
misalignjment	4	7	10	16	12	13
insufficient grease	4	0	10	10	12	13
excessive axial load	9	13	10	11	10	10
hard grease	0	0	4	0	7	1
false brinelling	11	3	4	6	6	6
true brinelling	0	0	3	3	3	5
grease inadequate	0	0	2	3	3	3
natural fatigue	0	0	2	3	3	4
poor fit	4	7	4	3	3	3
rotating radial load	4	7	2	2	2	2
soft grease	0	0	0	0	<1	<1
overheating	2	3	0	0	<1	<1
passage of electric current	0	0	1	2	<1	<1
too much grease	0	0	0	0	<1	<1

The Fleet Survey

The Fleet survey resulted in 614 suspect bearings reaching AOL, 596 of which were confirmed as failures. After some discussion as to what constituted a failure, it was decided by the Ship Department that any bearing which had deteriorated to such an extent that it was likely to confirm the ship's complaints was to be classed as a failure even if it might have run on for some time before complete breakdown. The most common complaint based solely on sound and feel was excessive noise or vibration. No vibration measurements were made by either the ships or the AOL in support of their reports, but advice was taken as to the severity of defect which would give undue noise. The whole of this trial referred to bearings which were only being relubricated at refits, a general instruction to this effect having been given earlier.

With nearly 600 failures, it was possible to arrive at a detailed comparison of the various causes of failure and the incidence of failure in various types and sizes of machines. At first sight, the overall failure rate appeared to be 596 bearings out of 39,000, i.e. 1·54 per cent. It was, however, apparent to the SMA that more failures had occurred than reached the AOL in the form of damaged bearings. The statistician examining the evidence suggested that at best 60 per cent and at worst 40 per cent of suspect bearings in all ships all over the world had been returned to the AOL. This was probably due to the ship's staff having insufficient time to document and send off all bearings which had given trouble and at the same time meet their other commitments. This figure increased the overall failure rate to somewhere between 2·5 per cent and 4·0 per cent per annum.

The high percentage figure for 'bearings failed' shown in TABLE I for the Fleet survey indicates that normally only those bearings with a definite defect were being forwarded to the AOL, although it is also possible that the AOL's standards may have become more stringent. The bearings received had come from a large number of ships.

A comparison of the defect pattern given by the first 100 bearings received with that given by the full 614 shows that the main trend was already well established by the first 100. Although the examination and the assessment of the data was a lengthy process, the continuation of the trial beyond this point was useful to the SMA in relation to failures in specific types of machine.

The main points revealed by the examination of the bearings in this trial were:

(*a*) About one third of the failures occurred in bearings not packed with greases to the specification DGS 6921A for XG-274. In a few cases it was suspected that corrosion or mechanical damage had been caused in transit, storage, or during fitting.

(*b*) The largest proportion of failures came in the first 2000 hours running with later peaks at about 5000 and nearly 10000 hours, the rate decreasing to insignificance thereafter. A graph revealed an uneven rate of decrease with time since fitting, with peaks at the beginning, after about one year, and after about three years. The more glaring defects showed early on, thereafter the periods at which machines were normally inspected have some relevance to when faults, other than the more serious ones, were detected.

(*c*) The largest single cause of failure was corrosion (FIG. 2). Nevertheless this only represents about one per cent or less of the bearings at risk. Misalignment, poor fit, excessive axial load, brinelling, etc. collectively accounted for nearly as many; a point which was duly passed on to the training authorities, although it was realized that in some circumstances it may be very difficult to eliminate these faults.

(*d*) There was a need to eliminate contamination by dirt.

(*e*) Some degree of fretting of the outer diameter and bore occurred in many bearings (See FIG. 3), although this was usually clearly a result rather than a cause of bearing failure.

(*f*) Because of the smallness of the failure rates, it was difficult to attribute a higher incidence of failure to any particular type of machine. The SMA did, however, detect a higher failure rate in a certain type of hoist, although the number at risk was small and most of the failures were accounted by one set of bearings being damaged several times in fitting; rollers had been fitted by brute force methods resulting in brinelling and noisy running.

(*g*) Water pumps and fans, etc. running in exposed positions produced one of the highest failure rates; these accounted for many of the corrosion failures. At that time there seemed little prospect of obtaining a significantly higher standard of corrosion resistance in the grease than had so far been achieved. Although some of the dirt found in bearings could have

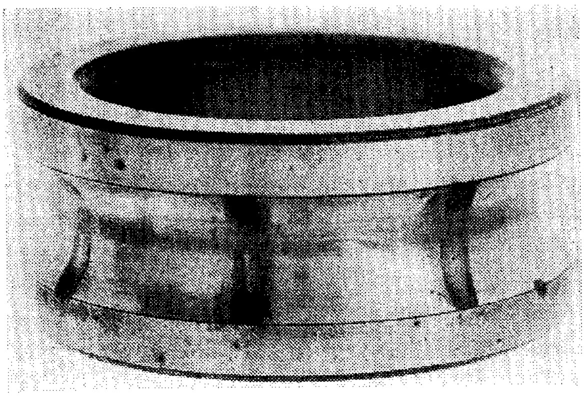

FIG. 2—WATER-MARKING

This is typical of corrosion found in bearings stationary for long periods in moist atmospheres.

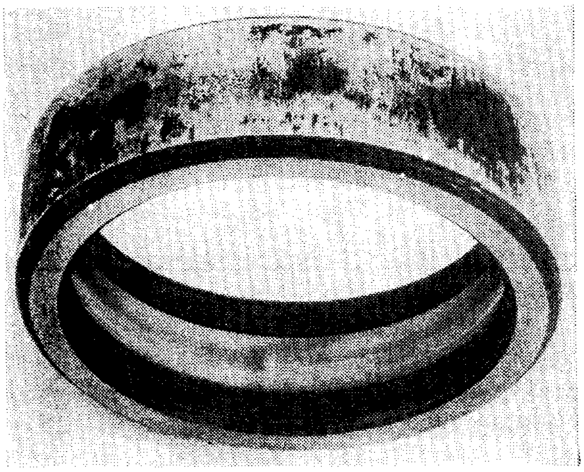

FIG. 3—FRETTING CORROSION

This is more commonly found on the outer diameter than on the bore of a bearing from electrical machinery because the outer race has to be a sliding fit in its housing. It is caused by 'creep' of the race aggravated by poor fit, misalignment and an out-of-balance rotor.

been caused by corrosion, another likely cause is the practice of supplying grease in tins of such a size that they are opened and reopened many times in other than ideal conditions before being used up.

(*h*) Special note was taken of bearings in which the cause of failure could have been inadequate lubrication; although these were few, they were mainly in some types of large bearing. A cross-check on the machines at risk showed that these failures were not significantly more than for other machines. Some bearings contained insufficient grease. Periodic relubrication had been reintroduced for the more highly stressed machines.

Examination of Suspect Bearings

In order to ascertain the cause(s) of failure, it was nearly always necessary to dismantle the bearing completely so that all the internal surfaces could be carefully xamined. The only exceptions were bearings already dismantled on receipt, usually because one or more parts had fractured or were so badly corroded that the balls could not be moved after removal of the cage.

Before dismantling, each bearing was tested for freedom of movement, first in its as-received condition and then after complete removal of the grease, thorough cleaning with solvents, and lubrication with light mineral oil. The quantity of grease was noted and a representative sample taken and examined for identity, consistency, colour and presence of foreign matter. The dismantling procedure usually required the heads of the rivets holding the two halves of the cage together to be drilled away; great care was taken to wash away the drillings before removing the rolling elements so that these and the tracks were not scored or indented. Before and after examination, the bearing parts were kept in a sealed container in which a piece of vapour-phase inhibited paper was placed to prevent the parts corroding.

Most defects could be observed without the aid of a microscope but the latter was necessary to detect small scores and indentations and light corrosion, especially on small bearings. A binocular instrument was used which had a good depth of field and a wide range of magnifications; it was, however, rarely necessary to use magnifications higher than 15 or 25. The detection of a spiral ball path or one with lobes (indicating out-of-roundness) was facilitated by spinning the race on a turn-table.

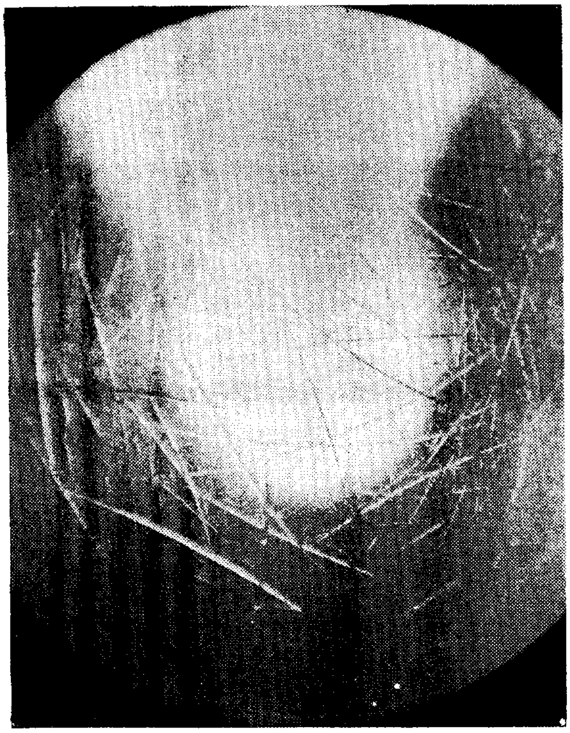

FIG. 4—DAMAGE CAUSED BY DIRT
The deep scores on this ball (magnification × 20) were caused by abrasive dirt in the grease.

The actual assessment was in accordance with the authorities listed in Refs. 1 to 7. The nature of the ball paths indicate the presence or otherwise of defects such as misalignment, out-of-roundness, rotating radial load, and excessive axial load; fretting corrosion and polishing of the outer diameter and bore often match up with peculiarities in the ball path. False and true brinelling and passage of electric current produce characteristic marks on the tracks and the presence of abrasive dirt results in scores on the rolling elements and indentations in the tracks. Sometimes such scores were present in bearings (FIG. 4) although the dirt which caused them was no longer present.

Establishing a Rig Test Procedure

Devising a rig test procedure for rating the performance of a lubricant in rolling bearings involves the specification of an assembly in which a test bearing can be run under the same accurately controlled conditions by different laboratories with conditions of test which will distinguish between lubricants at the desired level of performance in a reasonable time. Although numerous rigs have been devised, few have achieved the level of acceptance for them to be listed as standard methods. There are two Institute of Petroleum standard methods, three American methods and one German standard. This discussion is based on the two Institute of Petroleum methods in the development of which a number of laboratories, including the AOL, took part.

Two of the American methods (Ref. 10), namely Federal Standard 791a methods 331·1 and 333, are both entitled 'Performance Characteristics of Lubricating Grease at Elevated Temperatures'. One incorporates a rig often known as the Annapolis Rig and the other as the Pope Spindle Rig; these rigs are well known over here because they are of importance in establishing greases for aircraft use. The rigs, however, use ball bearings of only 20mm bore (smaller than most industrial or marine bearings) and the procedures rely on the average life to failure of four tests being equal to or above a specified figure. Although this will ensure the grease is of a certain standard, it appears to involve an indeterminate factor since any batch of apparently identical bearings run to failure will give a range of running hours. The U.S. Navy specifications, aimed at shipboard greases, have used variations of the Annapolis Rig with larger bearings on the same hours to failure basis (Ref. 11).

In the U.K. the normal practice has been to run for a fixed number of hours, often 500, and then rate the conditions of bearing and the used grease. Even so several tests are required to take care of unavoidable variations from test to test.

The Institute of Petroleum methods are IP 168 'Rolling Bearing Performance Test for Lubricating Greases' (Refs. 12 and 13), and IP 266 'Rolling Bearing

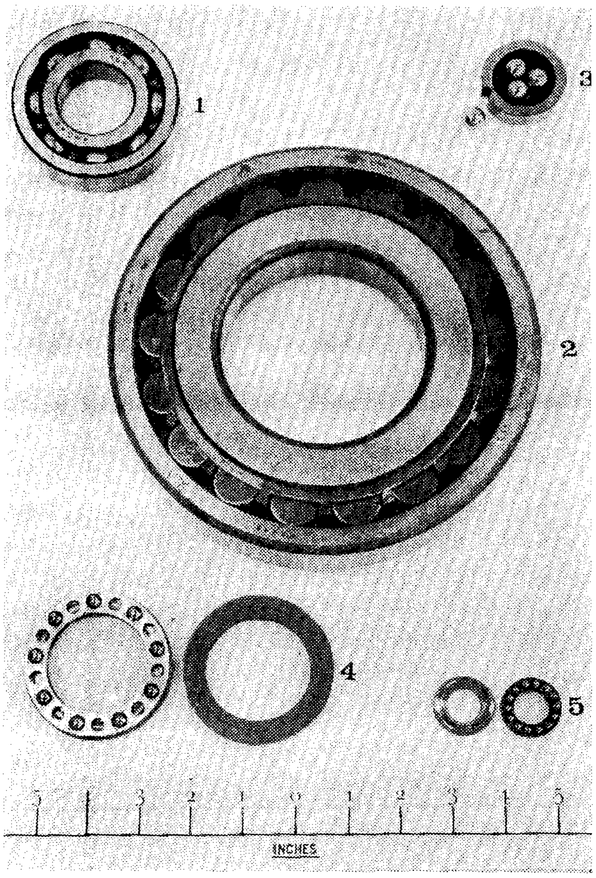

FIG. 5—TEST BEARINGS FROM SOME RIGS

(1) 40 mm bore ball bearing; IP 168 rolling bearing performance test lubricating greases (the IP rig).
(2) 4-inch bore roller bearing; IP 266 rolling bearing grease churning test (Admiralty grease rig).
(3) Test race and balls; IP 300 pitting failure test for oils in a modified 4-ball machine (rolling 4-ball).
(4) Thrust bearing with alternate balls removed; Unisteel rolling fatigue machine.
(5) Thrust bearing; AOL fatigue rigs.

Greases, Churning Tests' (Ref. 13). Both were evolved after a survey of existing tests, especially those used by the bearing manufacturers, and proved by a series of co-operative tests between several laboratories. IP 168 is a 500-hour test using a bearing of average size (Ref. 12). IP 266 is a 50-hour test using a large bearing in which greases are more likely to slump and churn. IP 266 is a modification of the Ransome and Marles Horizontal Rig Test devised by a working group of the Navy Department Fuel and Lubricants Advisory Committee and this version of the rig is known as the Admiralty Rig to distinguish it from the original campany rig. Although churning can be detected in 50 hours, a 500-hour test for overall performance in the Admiralty Rig is also laid down for the Navy grease XG-274.

The bearing used in IP 168 is a 40 mm bore single-row ball bearing of a type commonly used in industry (FIG. 5). As differences in detail existed between bearings made by different manufacturers, it was decided to specify a bearing with a rivetted pressed steel cage and to impose tolerances on the initial radial float of the cage; the amount of wear during the test is gauged by measuring the cage float before and after.

The rig as finally standardized consists of a shaft driven by a belt from an electric motor and mounted in plummer blocks so that a test bearing assembly can be fitted at each end. The rig can be run at up to 10 000 r.p.m. and bearing temperatures of up to 177°C (350°F). The bearings are run-in before each test and are packed with a specified amount of grease which nearly fills the bearing, no grease being placed in the covers.

Co-operative tests at ambient temperatures or 60°C (140°F) and 10 000 r.p.m., 121°C (250°F) and 7000 r.p.m., 150°C (302°F) and 5000 r.p.m., and 177°C (350°F) and 4000 r.p.m. have been run each on several greases of appropriate quality—specially heat-treated bearings must be used for tests above 121°C (250°F). Grease quality is judged by grease leakage, cage wear, visual inspection of the used grease, and the general state of the bearing.

The confidence level is based statistically on the results of the co-operative tests. Up to six tests should be carried out, but if the first four meet the specified requirements this is considered sufficient. This procedure gives a good grease a 7 in 8 chance and a borderline grease a 1 in 8 chance of being accepted.

IP 266 uses a 4-inch roller bearing (FIG. 5) which is run without applied heat at 2000 r.p.m. under 453·6 kg (1000 lb) radial load packed with grease but with

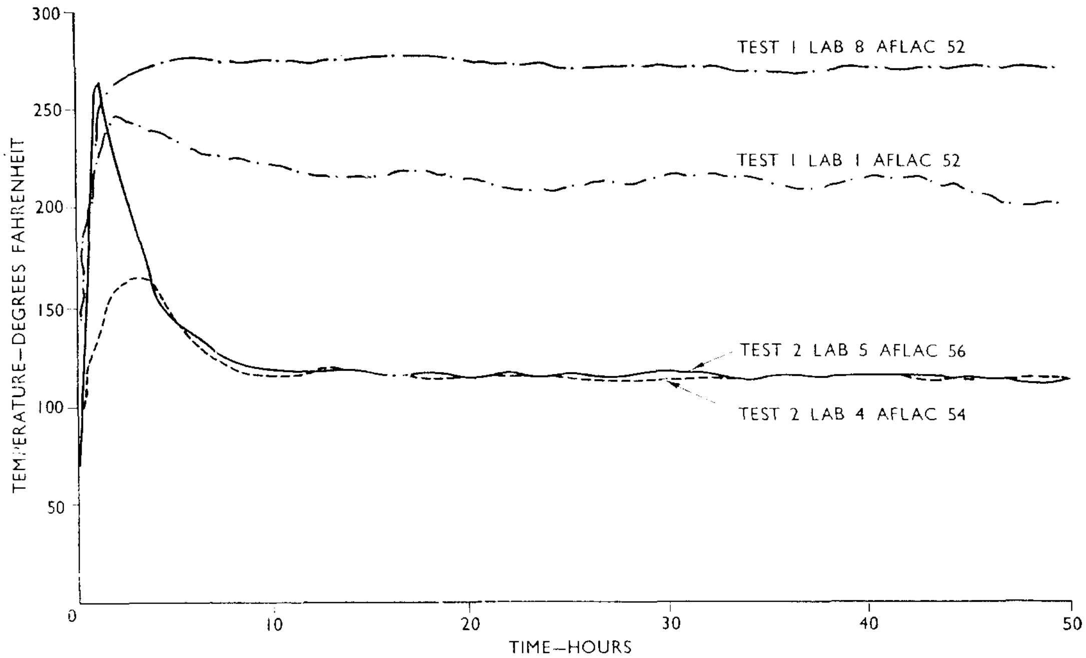

FIG. 6—SOME BEARING TEMPERATURES IN IP 266 (ADMIRALTY RIG)

sufficient space left in the covers to take the grease swept out on start up by the rolling elements. If the grease slumps back and churns, overheating usually accompanied by grease leakage results.

The NAFLAC Working Group standardized the details of the rig, which had been modified over the years, and drew up a standard test procedure. This includes a system of rating the used grease and the condition of the bearing after the test. The working group carried out a preliminary series of 500-hour tests because no correlation work had been done previously to examine the precision of the method. As a result, improvements in detail were made; after this two series of 50-hour tests were run in duplicate in eight laboratories followed by a series of 500-hour tests. If the temperature exceeded 138°C (280°F) these tests were shut down, this being too high a temperature for the bearing; sometimes operators shut down if the temperature was rising rapidly and was in excess of 121°C (250°F).

IP266 is in line with the results of the co-operative work described already. A grease is regarded as having churned if one or other of the following criteria arises:

(*a*) The bearing temperature rises above 121°C (250°F).

(*b*) The bearing temperature stays above 93°C (200°F) for longer than five hours.

(*c*) The test bearing temperature is greater than 64°C (148°F) at 28 hours from the start of the test

(*d*) More than 5 g of grease leaks out of the bearing.

The condition of the grease and the bearing is not included in these criteria but can be reported if required. The 500-hour test uses the rating system to detect wear and other undesirable changes.

Some temperature graphs are shown in FIG. 6; these include two on grease AFLAC 52. In one graph, the temperature reached a peak of 134°C (273°F) and stayed at an average of 131°C (268°F) for the full 50 hours with 133g grease leakage. In the other test shown, the temperature only reached 120°C (247°F) but averaged out at 101°C (213°F) with practically no grease leakage, giving a failure because of prolonged high temperatures. The grease showed churning characteristics almost every time it was tested. AFLAC 56 on the other hand reached initial peak temperatures around 127°C (260°F) and invariably dropped quickly to a satisfactory, steady, running temperature with no leakage. The other line shows a run with a low but well-defined peak temperature.

The initial peak temperatures in the 50-hour runs are given in TABLE II. They show that some greases usually give high, and others usually give low, figures but there are enough differences to make the precision of peak temperatures alone somewhat poor. But when all four criteria of churning listed in IP266 are taken into account, as in TABLE III, a much more reliable pattern emerges. Two to four tests are usually sufficient to show whether a grease is consistently satisfactory or consistently poor. Some greases such as AFLAC 47 and 48 are erratic even under controlled testing in the laboratory. Specification writers aim to accept only consistently good greases.

TABLE II—*Initial peak temperature °F of greases in IP 266 (from correlation programmes on Admiralty rig.)*

AFLAC NO. \ *Lab*	1	3	4	5	6	7	8	9
41	259	239	237	232	225	190	236	166
	242	239	239	234	240	236	235	148
42	270	252	250	244	238	250	250+	232
	264	252	268	247	236	250	250+	228
43	186	124	126	115	140	120	185	134
	122	134	180	120	139	110	135	129
44	233	222	192	224	186	222	206	259
	232	224	197	220	191	210	204	206
45	127	144	170	113	164	123	141	140
	142	135	156	183	138	148	140	128
46	176	186	178	152	133	157	172	168
	177	174	178	163	180	165	181	146
47	271	260	284	261	205	270	250+	252
	273	190	260	259	228	256	250+	258
48	194	252	230	197	248	199	203	182
	217	201	273	186	248	213	250+	246
50	146	154	135	143	130	159	156	199
	154	156	141	157	120	143	153	169
51	234	158	137	137	165	196	—	250
	255	162	140	143	158	152	—	172
52	247	238	255	250+	221	222	273	254
	264	259	255	250+	211	226	280+	253
53	280	264	279	255+	222	270	280+	266
	280+	268	273	256+	244	276	280+	262
54	220	238	153	195	155	280+	284+	204
	280+	271	163	216	143	205	284+	150
56	262	265	261	264	—	—	—	—
	257	264	266	263	—	—	—	—
57	260	176	176	202	—	192	194	152
	185	152	186	193	—	—	136	180
58	136	170	150	165	—	—	145	176
	179	157	150	186	—	—	140	—
59	198	183	155	149	—	161	149	144
	181	179	159	154	—	160	140	144

TABLE III—*Grease churning in IP 266 (from correlation programmes on Admiralty rig)*
(a) High peak temperature (b) Temperature too long above 93°C (200°F)
(c) Temperature too high after 28 hours (d) Over 5g leakage

AFLAC NO. \ Lab	1	3	4	5	6	7	8	9	Total No. of failures
41	abcd abcd	abcd abcd	abcd abcd	abcd abcd	abcd abcd	None abcd	abcd abcd	None None	13
42	abcd abcd	ab–d ab–d	abcd abcd	abcd abcd	a—d a—d	abcd abcd	a—d a—d	abcd abcd	All 16
43	None None	None None	None None	None None	None None	None None	c None	None None	1 slight
44	None None	None None	None None	None None	None None	None c	c c	a None	4 slight
45	None None	None None	None None	None None	None None	None None	None None	None None	0
46	None None	None None	None None	None None	None None	None None	None None	None None	0
47	a a	a None	ab–d None	a None	bc bc	abcd a	a – a –	a abcd	11
	Some others must also have been too long over 93°C (200°F)								
48	None None	abc c	bc abcd	c c	c c	None None	c a	None None	10
50	None None	None None	None None	None None	None None	None None	None None	None None	0
51	c abc	None None	None None	None None	None None	None None	– –	None None	1 definite 1 slight
52	bc abcd	bc abcd	abcd abcd	a – ab–	None None	abcd abd	ab–d ab–d	a—d a—d	14
53	abcd abcd	abcd abcd	abd abcd	ab–d ab–d	b b	abcd ab–d	ab–d ab–d	ab–d ab–d	All 16
54	None ab–	None ab–d	None None	None None	None None	a– None	a– a–	None None	5
56	a a	a a	a a	a a	– –	– –	– –	– –	All 8
57	ab None	None None	None None	None None	– –	None –	None None	None None	1
58	None None	None None	None None	None None	– –	– –	None None	None None	0
59	None None	None None	None None	None None	– –	None None	None None	None None	0

Note: Greases 41 to 48 were tested before it had been decided to include a definite number of hours over 93°C (200°F) as criteria of churning.

Oil Lubricated Bearings

No survey of failures in oil-lubricated rolling bearings has been attempted by the AOL but a study of the causes of bearing failures in warships carried out by the Canadian Naval Engineering Test Establishment embraced all types of rolling bearings however lubricated. It was concluded that 90 per cent of these bearing failures were premature, i.e. failure had or would have occurred from causes other than normal fatigue. Grease-lubricated fan bearings corroded by moisture (including sea-water) were a significant cause of trouble. Oil-lubricated bearings, including oil pumps lubricated by wet oil, were another significant proportion of the failures. The need to keep moisture and dirt out of bearings was clearly shown.

Water in oil can have another effect as well as causing trouble by promoting corrosion: whether or not corrosive action is a contributory cause, water in oil has been shown to lead to a reduction in the fatigue life of bearings. Some synthetic fluids also affect fatigue lives, and other complications are the variations which occur when conditions necessitate the use of other than the usual bearing steel.

The use of synthetic lubricants to obtain adequate performance at unusually high or low temperatures, e.g. in aircraft gas turbines, and the use of fire-resisting fluids created a need for data on fatigue life with different types of fluid. Experiments with actual bearings are time consuming and expensive. Various workers have devised rigs with which to obtain data more quickly under accelerated conditions (Ref. 14).

Lubricants and Fatigue Failures

Some degree of scatter is inherent in the lives of any group of apparently identical articles whether they be rolling bearings or electric bulbs. Weibull (Ref. 15) postulated the relation:

$$\text{Log, Log } \frac{1}{S} = e \text{ Log } \frac{L}{A}$$

where

S = Probability of Survival
e = Dispersion coefficient
L = Life in millions of revolutions
A = Constant

Bearing fatigue lives will plot as straight lines in these co-ordinates commonly known as a Weibull plot. Plotting data from testing a group of apparently identical bearings under the same conditions until fatigue failure occurs gives life at any stage. The B_{10} life, for example, is that reached by all but 10 per cent of the bearings tested or, with greater accuracy, the B_{50} life is the life reached by half the bearings.

Barwell and Scott (Ref. 16), in an endeavour to compare lubricants in a less complex structure than a standard bearing and in less time, adapted a Four Ball Machine to form a Rolling Contact Fatigue Tester. The standard Four Ball Machine consists essentially of three balls clamped together with a rotating ball loaded vertically downwards in contact with them. It measures extreme pressure, friction and wear properties of lubricants as set out in IP 239 'Extreme Pressure Properties: Friction and Wear Tests for Lubricants: Four Ball Machine'. Barwell and Scott realized that if the three lower balls were free to rotate (FIG. 5) conditions of loading could be selected; this led to fatigue failure with pitting of the top ball in the order of 100 minutes. Thus, even allowing for the number of repeat tests needed to give statistically meaningful results, much data could be accumulated in a reasonable time. The procedure is currently under study by an

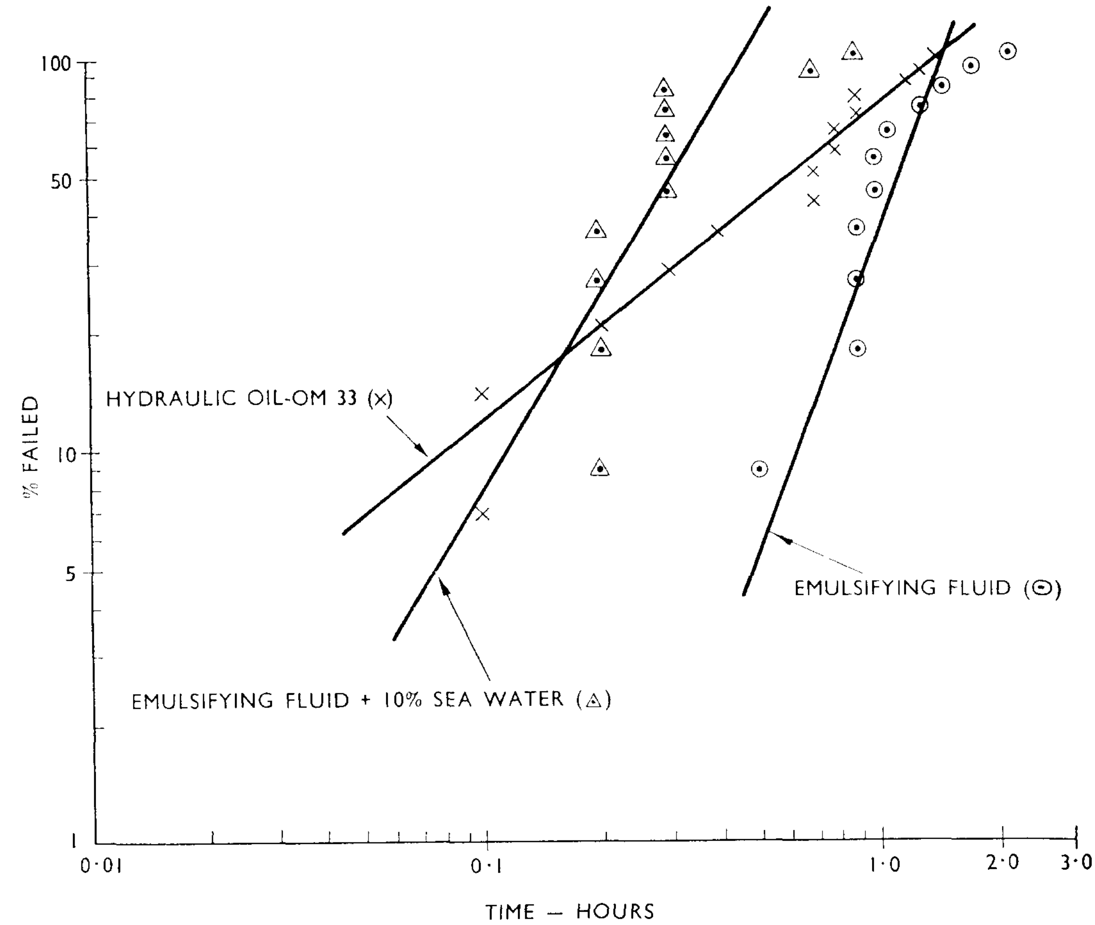

FIG. 7—ROLLING FOUR-BALL FATIGUE TESTS

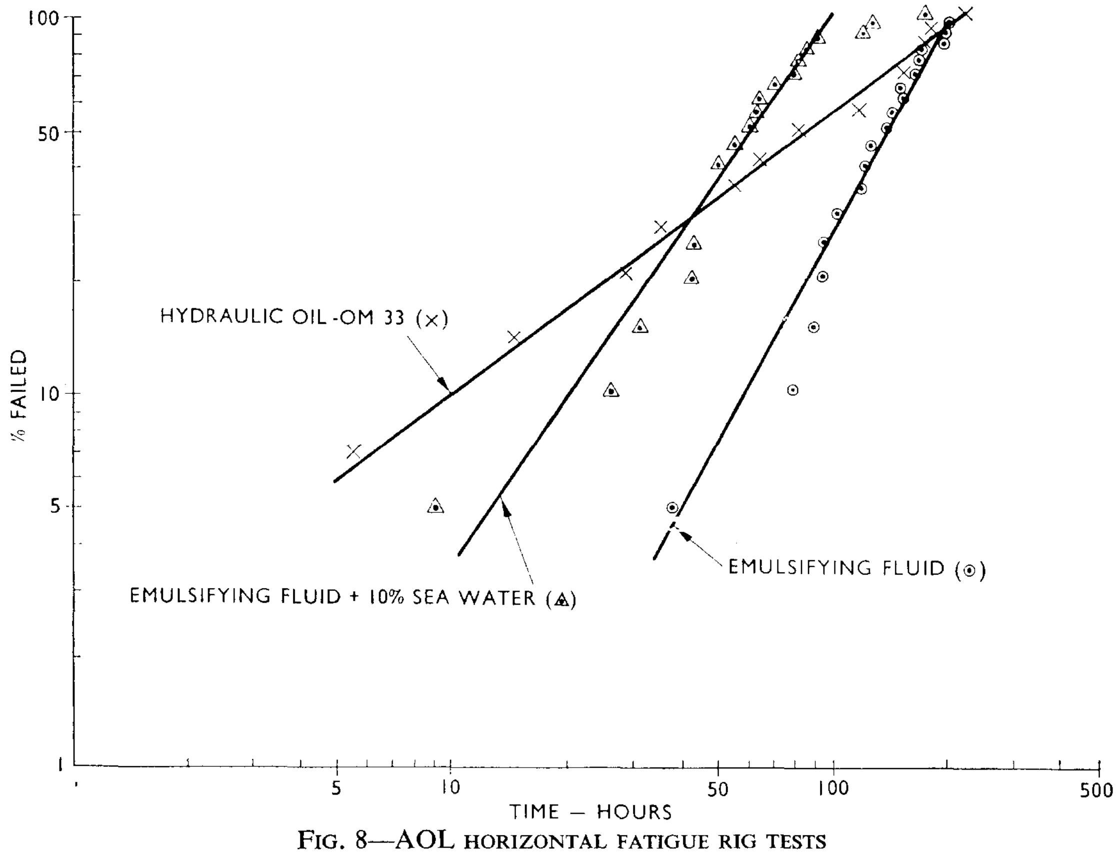

FIG. 8—AOL HORIZONTAL FATIGUE RIG TESTS

Institute of Petroleum Sub Panel in which the AOL takes an active part; a standard method, IP 300, is expected shortly. In the U.S.A., various workers have used a similar machine containing five balls in place of four.

Another rig in use in this country is the Unisteel Machine which consists of a thrust bearing (FIG. 5) running under load without alternate balls to raise the stress and give fatigue failures in a relatively short time. Recently the AOL has built a set of six horizontal fatigue rigs of their own design using small inexpensive thrust bearings (FIG. 5) and this design has led to a more attractive rig, namely: a six-station Admiralty Oil Laboratory vertical fatigue rig.

There is already a mass of literature on the interaction of lubricant, metal, and operating conditions in relation to fatigue failures. Much of the American work published by Anderson and others is aimed at aerospace conditions. Navies, on the other hand, are interested in the effect of water in oils. Grunberg and Scott (Ref. 17) showed that the fatigue life of the EN-31 balls usually fitted in rolling bearings was reduced by water in oil. Stainless steel balls were relatively immune to reduction of fatigue life by water in oil but were inferior to EN-31 in dry oil. Schatzberg and Felsen (Ref. 18) have reported some change in fatigue life even at the level at which only dissolved water was present.

Pumps for hydraulic fluids involve rolling contact and must not fail prematurely. This could occur if slugs of water gain access to the system. Ritchie and Thomson (Ref. 19) have described an emulsifying fluid now used by the R.N. in some hydraulic systems as an insurance against the risk of contamination by water.

In comparison with OM-33, the mineral oil hydraulic fluid used previously, fatigue tests were carried out in rolling four-ball and AOL fatigue rigs with the emulsifying fluid, both dry and containing ten per ecnt of sea-water. Consistent results were obtained although the stresses involved and hence the time scale differed between the rigs (FIGS. 7 and 8). The dry emulsifying fluid gave longer fatigue lives than OM-33 at both the B_{10} and B_{50} levels (10 per cent fail and 50 per cent fail). With 10 per cent of sea-water the emulsifying fluid was still better than dry OM-33 at the B_{10} level but not at the B_{50} level, because the spread of results with OM-33 is greater on both rigs. OM-33 has performed satisfactorily for some years. The emulsifying fluid should take care of appreciable water contamination.

Felsen, McQuaid and Marzani (Ref. 20) have described fatigue studies in hydraulic fluids including emulsifying fluids, using a special rig with a 40mm bore test bearing. Their results are not inconsistent with FIGS. 7 and 8.

Fire-resisting fluids used in coal-mining equipment or on board ship may be water-based, e.g. water emulsions of some kind, polyglycol/water, etc. In general, such water-based fluids lead to lower than normal fatigue lives, but any new fluid offered to the AOL would be treated on its merits. For recent news see Hobbs (Ref. 21) who used various sized bearings in his firm's special rig, and Kenny and Yardley (Ref. 22) who used the Unisteel machine.

Conclusions

Reliable estimates of the incidence and causes of failure in rolling bearings must be based on a systematic approach and a significant number of failures.

Surveys of failures carried out by the AOL have shown that a substantial proportion can be caused by corrosion, dirt, and various mechanical faults rather than by deficiencies in bearings or lubricant.

Although various rig tests for establishing lubricant (usually grease) performance by running in test bearings are known, establishing the precision of such tests requires co-operative programmes between laboratories, and few have achieved the status of standard methods.

It is known that wet oil or some synthetic lubricants, including water-based fluids, can affect the fatigue life of bearings. Several fatigue tests are known and are used in the evaluation of fluids for use in rolling bearings.

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APPENDIX I

MULTIPURPOSE GREASE XG-274

XG-274 is a multipurpose grease primarily intended for the lubrication of rolling bearings in shipboard electrical machinery over the temperature range $-29°C$ to $+121°C$ ($-20°F$ to $+250°F$).

The specification was developed by the AOL and the Grease Panel of the Navy Department Fuel and Lubricants Advisory Committee as a performance specification based on performance tests in rolling bearing test rigs plus tests for water stability, rolling stability, rust preventive properties, oxidation stability, low temperature torque, etc.

The grease is required to consist of mineral oil with a gelling agent, which has normally been a lithium soap, plus oxidation and corrosion inhibitors as necessary and is of a medium consistency.

APPENDIX II

REPORT ON BEARING FAILURE IN ELECTRICAL MACHINERY

This form is to be used for reporting the failure of all bearings in electrical machinery. It is to be completed by Ships Staff, Base Staff, Dockyard or Overseers as appropriate.

The bearing in a clean non-absorbent container, e.g. a polythene bag, and a sample of grease from the end cap in a separate container are to be forwarded together with a copy of this report to: The Superintendent, Admiralty Oil Laboratory, Fairmile, Cobham, Surrey. Additional copies to Administrative Authority (1), The Ship Maintenance Authority, South Terrace, H.M. Dockyard, Portsmouth (1). No covering letter or Form S.2022 is required.

H.M.S. Originating Dept. Date

Ship's Serial No.

Description of Equipment:

(in which the machine is incorporated)

Service Performed	Where fitted
Manufacturer	Serial No.
Admiralty Reference/Pattern No.	Whether Resiliently Mounted

B. *Description of Motor or Generator in which Bearing is fitted*

Manufacturer — Mark/Serial/Type/Pattern No.

Speed — Power — Type of Enclosure

Whether mounted Horizontally, Vertically or Inclined

If Horizontally, whether fore and Aft or Athwartships

C. *Description of Defective Bearing*

Manufacturer — Maker's Part No.

Pattern No. — Driving End or Non-Driving End

D. *Nature of Defect*

(*a*) When was the bearing fitted?

(*b*) When did the fault occur?

(*c*) Estimated running hours.

(*d*) Who fitted the bearing (manufacturer, shipyard, ship's staff, dockyard)?

(*e*) During what periods was the machine idle?

(*f*) Is the machine Continuous/Intermittent/Occasional running?

(*g*) Is a pre-loading spring fitted to this bearing or the opposite end bearing?

(*h*) Is the opposite end bearing ball or roller?

(*i*) Was it defective?

(*i*) Are shaft seals fitted? Were these damaged?

(*k*) How was the failure located, e.g. noise, overheating, vibration, starter tripping, etc.?

(*l*) Quote any previous reports of failure of:

(*i*) this particular motor, or

(*ii*) similar motor fitted to this equipment.

(*m*) Opinion on cause of failure, e.g. grease condition, bearing fit.

Signature.................... Rank................

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