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by Richard J. Parker
Lewis Research Center
Cleveland, Ohio 44135

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ABSTRACT

Rolling-element fatigue tests were run with three synthetic lubricants with and without antiwear additives and with a paraffinic mineral oil at race temperatures of 336 to 353 K (146° to 175° F). The five-ball fatigue tester was used with AISI 52100 steel balls to evaluate the relative fatigue lives with each of six lubricant-additive combinations. The tests were run at 5520 MPa (800 000 psi) maximum Hertz stress, 10 000 rpm shaft speed, and 30° contact angle. The lubricants tested have similar kinematic viscosities at 372 K (210° F) ranging from 0.034 to 0.089 cm^2/sec (3.4 to 8.9 cS). At these conditions, the mode of failure in the five-ball fatigue tester was classical subsurface rolling-element fatigue. The baseline for comparison of fatigue life was the paraffinic mineral oil without additives. The effects of the synthetic lubricants and their additives, which are useful for boundary lubrication, oxidation or foam inhibition, were evaluated.

INTRODUCTION

Synthetic lubricants have provided the greater thermal and oxidative stability required as temperatures increase in applications such as aircraft turbine engines and power transmission systems [1]. Mineral oils, of the specific grade required, still find wide use in lower temperature systems such as in internal combustion engines and industrial power transmission equipment. In the past two decades, much data has been published on the effects of various lubricant base stocks on rolling-element fatigue life for both mineral and synthetic oils [2 to 10]. These data include tests with full-scale bearings [4, 5, 10] and with ball specimens in accelerated fatigue-life test rigs [2, 3, 6 to 9].

In general, the relative life results have been biased because of viscosity variations among various base stocks. Increased lubricant viscosity generally gives longer fatigue lives [2, 6]. Where lubricant viscosity was similar among various base stocks, differences in fatigue life were attributed to the pressure-viscosity coefficient of the lubricants [4,6]. Subsequently, the effects of these lubricant properties have been better understood with increased knowledge of the characteristics of the elastohydrodynamic (EHD) lubricant film present between rolling-elements and raceways. The ratio of EHD film thickness to composite surface roughness has become an acceptable indicator of the effectiveness of the lubricant film within the rolling-element contact zone. This ratio has been experimentally shown to influence the fatigue life of rolling-element bearings [11, 12].

A trend has been apparent in many fatigue tests which shows greater fatigue life with mineral oils than with synthetic lubricants [4, 6, 9, 10]. Data of [8], however, shows several synthetic lubricants with greater life than mineral oils at equivalent viscosities.

Synthetic lubricants such as the diesters and polyesters are used in the majority of aircraft turbine engines and power transmission systems. These lubricants conform to military specifications such as MIL-L-7808 or MIL-L-23699, which impose maximum or minimum limits on viscosity at given temperatures. For systems where maximum viscosity limits are not imposed it is expected that bearing and gear life may be improved with higher viscosity lubricants.

Recently, another class of synthetic lubricants has received attention due to the renewed interest in mechanical transmissions which use the principle of traction for power transfer. These lubricants are generically referred to as traction fluids. They have been developed for better tractive properties (higher traction coefficient) than other lubricants [13, 14]. The fatigue life of tapered-roller bearings [15] was found to be greater with a synthetic cycloaliphatic hydrocarbon traction fluid than with a mineral oil of equivalent viscosity. Traction fluids generally have a relatively high pressure-viscosity coefficient [16], which has a desirable effect on EHD film thickness. This effect may explain the observed fatigue life increase.

Another synthetic lubricant which has shown excellent performance in full-scale bearing life tests at temperatures up to 589 K (600° F) was a synthetic paraffinic oil with an antiwear additive [17]. A lower viscosity (shorter-chain length) version of this 100-percent paraffinic oil, containing an antiwear additive and an oxidation inhibitor has also shown good performance in bearing tests to 492 K (425° F) [18].

The objectives of the research reported herein were (a) to determine the relative rolling-element fatigue lives using two synthetic cycloaliphatic traction fluids, a tetraester with and without additives, and a synthetic paraffinic oil and (b) to compare these relative lives with that obtained using a mineral oil of similar viscosity. Tests were conducted in the NASA five-ball fatigue tester at a maximum Hertz stress of 5520 MPa (800 000 psi), a contact angle of 30°, and a shaft speed of 10 700 rpm.

TEST LUBRICANTS AND MATERIALS

Properties of the synthetic lubricants and the baseline mineral oil used in this study are listed in table I. Viscosity-temperature characteristics are shown in figure 1.

The tetraester was a neopentyl polyol ester (type II fluid) which meets the specification MIL-L-23699A and is widely used in U.S. commercial and military aircraft turbine engines. The formulated tetraester contained additives including oxidation and corrosion inhibitors as well as an antiwear additive. The base fluid without the additive package was also tested.

The high traction fluids were from the cycloaliphatic hydrocarbon family. These lubricants possess a relatively high coefficient of traction, approximately 50 percent greater than mineral oils [19]. The two traction lubricants evaluated contained different additives and had different viscosity characteristics due to the presence of a polymethacrylate viscosity index (VI) improver in one fluid, although both were formulated from the same base stock. The lower viscosity sample, referred to herein as traction fluid 1, contained only an oxidation inhibitor. Traction fluid 2 contained the oxidation inhibitor, the viscosity index improver, and antifoam and antiwear additives. The antiwear additive, zinc dialkyl dithiophosphate is quite common to many automotive oils and transmission fluids.

The synthetic paraffinic oil was a 100-percent paraffinic fluid of a type which has provided good bearing life in full-scale bearing tests at temperatures up to 589 K (600° F) [17]. The fluid used in this investigation was from the same class of fluids but with a shorter chain length molecule and hence, a lower viscosity. It contained a substituted organic phosphonate antiwear additive and an aromatic amine type oxidation inhibitor. This lower viscosity lubricant has shown good bearing performance at temperatures to 492 K (425° F) [18].

The baseline lubricant for these tests was a paraffinic mineral oil containing no additives. It was chosen to have a viscosity within the range of that of the synthetic lubricants tested.

The 12.7-millimeter (0.500-in.) diameter test balls used in this study were made from a single heat of carbon-vacuum - deoxidized AISI 52100 steel. The balls were AFBMA Grade 10 and were through-hardened to Rockwell C hardness of 61.

APPARATUS AND PROCEDURE

Five-Ball Fatigue Tester

The NASA five-ball fatigue tester was used for all tests conducted. The fatigue tester, fully described in [20], is shown in figure 2. It consists essentially of an upper test ball pyramided on four lower support balls that are positioned by a separator and are free to rotate in an angular-contact raceway. System loading and drive are supplied through a vertical drive shaft. For every revolution of the drive shaft the upper test ball receives three stress cycles.

The upper test ball and raceway are analogous in operation to the inner and outer races of a b. ring, respectively. The separator and the lower balls function in a manner similar to the race and the balls in a bearing.

Fatigue Testing

Before they were assembled in the five-ball fatigue tester, all test-section components were flushed and scrubbed with ethyl alcohol and wiped dry with cheesecloth. The specimens were examined for imperfections at a magnification of ×15. After examination all specimens were coated with the test lubricant to prevent corrosion and wear at startup. A new set of lower balls was used with each upper test-ball specimen. The speed, outer-race temperature, and oil flow were monitored and recorded at regular intervals. After each test the outer race of the five-ball system was examined visually for damage. If any damage was observed, the race would be replaced before further testing.

Method of Presenting Fatigue Results

The total test time for each specimen was recorded and converted to total stress cycles. The statistical methods of [21] for analyzing rolling-element fatigue data were used to obtain a plot of the log log of the reciprocal of the probability of survival as a function of the log of stress cycles to failure (Weibull coordinates). For convenience, the ordinate is graduated in statistical percent of specimens failed. From these plots, the number of stress cycles necessary to fail any given portion of the specimen group may be determined. Where high reliability is of paramount importance, the main interest is in early failures. For comparison, the 10-percent life on the Weibull plot was used. The 10-percent life is the number of stress cycles within which 10 percent of the specimens can be expected to fail; this 10-percent life is equivalent to a 90-percent probability of survival. The failure index indicates the number of specimens that failed out of those tested.

RESULTS AND DISCUSSION

Fatigue Results

Rolling-element fatigue tests were run in the five-ball fatigue tester with two traction fluids, a tetraester with and without additives, a synthetic paraffinic oil and a paraffinic mineral oil. Standard test conditions for all tests consisted of a maximum Hertz stress of 5520 MPa (800 000 psi), a contact angle of 30°, and an upper ball speed of 10 700 rpm. All tests were run at room temperature (i.e., no heat added). Outer-race temperatures varied from 339 to 353 K (150° to 175° F) for the synthetic lubricants and averaged approximately 336 K (146° F) for the mineral oil.

The results of the rolling-element fatigue tests are shown on Weibull coordinates in figure 3 and are summarized in table II. The paraffinic mineral oil gave longer lives than any of the synthetic lubricants tested. The 10-percent life with the mineral oil was 34.8 million stress cycles whereas the lives with the synthetic lubricants ranged from 9.5 million cycles to 17.9 million cycles. The statistical significance of the differences in these lives from that of the mineral oil is reflected by the confidence numbers given in table II. The numbers, based on the experimental data, are calculated by methods of [21]. They indicate the percent of time that components lubricated with the mineral oil would show fatigue lives superior to the fatigue lives of identical components lubricated with each of the synthetic lubricants. A confidence number greater than 95 percent, which is equivalent to a 2σ confidence level, indicates a high degree of certainty. Although all of the synthetic lubricants show lives less than the mineral oil, only the lives with the tetraester base and the synthetic paraffinic oil approach statistically significant differences at the 10-percent life level. At the 50-percent life level, all experimental lives with the synthetic lubricants are significantly less than the mineral oil. These data tend to support and confirm the data of [4, 6, 9, 10]. These results do not, however, agree with the results of [15] wherein a similar traction fluid showed greater bearing life than did a mineral oil.

Although good correlation has been repeatedly demonstrated between the relative fatigue life performance of lubricants tested in the five-ball apparatus and that actually experienced in full-scale bearing tests (e.g., see [9], it is conceivable that unforeseen chemical-stress effects may alter the aforementioned life rankings of the test lubricants at the lower contact stress levels associated with typical bearing and traction drive applications. Undoubtedly the safest course to follow, particularly for critical applications, would be to conduct fatigue life tests for unconfirmed lubricants at the contact stress level of interest. This of course is not always a practical option.

All fatigue failures considered in the present analysis appeared to be a result of classical subsurface fatigue; that is, each of the failed balls had a single spall located in the running track. These spalls (one of which is shown in fig. 4(a)) are typical of those obtained in five-ball fatigue experiments.

Lubricant Viscosity Effects

The operating temperature was not a controlled variable in these five-ball fatigue tests. The room temperature and the lubricant mist temperature were controlled, and the test system was allowed to seek a stabilized temperature. As mentioned previously, the race temperature varied somewhat with each of the lubricants tested. The average race temperature during the fatigue tests with each lubricant are shown in table III. Because of this variation in test temperature and small variations in viscosities of the lubricants, the operating viscosities varied among the test lubricants.

Lubricant viscosity is known to influence rolling-element fatigue life. In [7] and [22] it is reported that rolling-element fatigue life is proportional to kinematic viscosity raised to powers from 0.2 to 0.3. The viscosity of the lubricant as it enters the contact is the proper parameter to use in this relation. Unfortunately, it was impractical to measure the temperature of the lubricant at this point. Considering the relative viscosities of the test lubricants at the measured race temperatures and their effect on fatigue life in accordance with the aforementioned relation, however, life adjustment factors can be estimated for tests that might have been run at equal lubricant viscosities. These tests would, of course, require additional lubricant cooling for the hotter running lubricants. Correcting for lubricant viscosity differences would result in the adjusted 10-percent fatigue lives shown in table III. The adjustments are seen to be small and have not made significant differences in the relative lives.

Tetraester

The tetraester, both with and without the additive package, gave lives less than the paraffinic mineral oil. The confidence numbers (table II) indicate that the differences are approaching the significant level. The formulated tetraester is a highly developed lubricant with excellent thermal and oxidative stability and chemical and physical characteristics for bearing operation up to 505 K (450° F). Since the specific tetraester formulation used in this program is a commercially available lubricant, the specific chemical composition and additive types are considered proprietary by the manufacturer. The effects of the additives on the fatigue life in these tests were small and are considered to be insignificant.

Synthetic Paraffinic Oil

The life with the synthetic paraffinic oil was not statistically different from the formulated tetraester. In full-scale bearing tests at 492 K (425° F) [18], this same synthetic paraffinic oil showed a fatigue life about twice that of the same tetraester with a confidence of 84 percent. The present tests with insignificant life difference and the bearing tests with borderline significance indicate that the synthetic paraffinic oil has little if any fatigue life advantage over the widely used tetraester at temperatures up to 492 K (425° F). The synthetic paraffinic oil (higher viscosity version) does show promise for higher temperature operation, giving good thermal stability and bearing life up to 589 K (600° F) [17].

Traction Fluids

The results of these tests with the traction fluids were initially reported in [23]. The traction fluids stabilized at a race temperature of about 347 K (160° F) (table III) which was higher than the other lubricants with the exception of the synthetic paraffinic oil, although the operating viscosity tended to be lower than or equal to that of the others. The observed higher temperatures may be attributed to increased spinning frictional heat in the contact zone as a result of the traction fluids' relatively high traction coefficient. Because of the adverse effect of the higher temperature on EHD film formation, contact spin heating can be of concern for designers who specify traction fluids.

In the case of traction fluid 2, some difficulties were encountered in establishing and maintaining test operating conditions. Several tests had to be discontinued within minutes of initiation because of severe wear and overheating. A pronounced wear track is quite evident on the upper ball specimen of figure 4(b), which was removed from the tester after just 30 minutes of operation. Test balls from these aborted tests generally showed a high degree of discoloration from overheating. Those tests that failed to achieve stable operation within 1 hour were terminated, and eliminated from statistical consideration. Eight tests fit this category.

In 10 cases, test specimens lubricated with traction fluid 2 did achieve and maintain stable running conditions, but after several hours of testing, unexpectedly began to run roughly and overheat. Figure 4(c) shows an extreme case of an upper ball that ran smoothly for more than 50 hours before it began to overheat and wear drastically. The appearance of this ball with its excessive surface distress when contrasted against that of a ball operated under identical conditions which reached the 100-hour cut-off time (fig. 4(d)) exemplifies the erratic behavior of traction fluid 2.

Examination of the test rig's oil jets before and after failures showed them to be in good working order. It is considered unlikely that temporary lubricant deprivation is the proper explanation. Tests that were halted for excessive surface distress, that is, nonfatigue failures, were treated as suspensions in the statistical analysis.

The behavior of traction fluid 2 is even more perplexing in view of the fact that surface distress was not encountered in any of the tests with traction fluid 1. Traction fluid 1 not only appeared to be less viscous but did not have the benefit of an antiwear additive as did fluid 2.

The role played by surface reactive antiwear additives under the relatively high loading conditions that exist in the five-ball fatigue tester is not well defined. Under conventional loading conditions for bearings and gears, that is, maximum Hertz stresses typically less than 2070 MPa (300 000 psi), antiwear additives form chemical surface films, which presumably minimize asperity contact and, consequently, surface distress. At relatively high pressures, chemical effects become a significant factor and could conceivably have an adverse effect on rolling-element life. Tests reported in [24], performed in a four-ball fatigue tester under an unusually high maximum Hertz stress loading of 8300 MPa (1.2×10^6 psi), showed an increase in surface distress with the addition of an antiwear additive in about half of the material-additive tests conducted. Similarly, tests conducted with the NASA five-ball fatigue rig [25] showed that the presence of several surface active additives were generally detrimental to rolling-element fatigue life, although only a chlorinated wax additive caused test ball surface distress.

The antiwear additive present in traction fluid 2 is zinc dialkyl dithiophosphate. Such additives aid in protecting the sliding contacts between the cage and the balls or rollers and between the cage and the race guiding lands. They also protect the ball-raceway or roller-raceway contacts in bearings operating under high-temperature, high-speed conditions where lubrication conditions are marginal.

The results of four-ball fatigue tests [26] with a mineral oil containing the zinc dialkyl dithiophosphate additive showed significant life reductions from the additive free oil, although the authors did not differentiate fatigue related failures from those caused by gross surface distress and smearing.

In the study reported in [27], the load-carrying performance of a range of metal dialkyl dithiophosphates were studied in a four-ball wear apparatus under both antiwear and extreme pressure conditions. The major distinction between antiwear and extreme pressure regions of lubrication is in terms of the severity of the load and the attendant contact operating temperatures. A significant conclusion of [25] is "that good performance by an additive in the antiwear region is not necessarily accompanied by good performance in the extreme pressure region." In fact, the zinc dithiophosphate additive, which exhibited the best antiwear performance of all the metal dithiophosphates tested in [25], provided the poorest protection under extreme pressure conditions.

Although the aforementioned data are far from conclusive, they do suggest that overreactive chemical effects of the antiwear additive under high contact pressure might be responsible for the erratic behavior observed in the tests with traction fluid 2.

Another difference between the traction fluids tested was the presence of a polymeric additive known as methacrylate in traction fluid 2, which as a viscosity index improver, gave the fluid less viscosity-temperature sensitivity. However, the degree of viscosity enhancement actually achieved by the addition of such a long chain polymer is often dependent on the operational shear rate to which the oil blend is subjected.

Thus, at the high shear rates that exist in the five-ball fatigue tester (greater than 10^6 sec^{-1}), it is likely that the actual viscosities of both traction fluids were nearly the same. In view of this, it is not surprising that both traction fluids had approximately the same 10-percent fatigue lives even though fluid 2 enjoyed about a 60 percent apparent viscosity advantage over fluid 1.

SUMMARY

Rolling-element fatigue tests were conducted in the five-ball fatigue tester with two traction fluids, a tetraester with and without additives, a synthetic paraffinic oil, and a paraffinic mineral oil. The following results were obtained:

1. All the synthetic lubricants gave rolling-element fatigue lives less than the paraffinic mineral oil. At the 10-percent life level, the life differences between the traction fluids and the mineral oil were not statistically significant.

2. The rolling-element fatigue life with the formulated tetraester was not significantly different from that with the tetraester base oil.

3. The rolling-element fatigue life with the synthetic paraffinic oil was not significantly different from that with the formulated tetraester.

4. Erratic test behavior was observed for the traction fluid that contained a zinc dialkyl dithiophosphate antiwear additive. Some tests were terminated prematurely because of excessive surface distress and overheating. It is speculated that overreactive surface chemistry at the high contact pressures may be partially responsible for this behavior.

5. Slightly higher race temperatures were recorded for the traction fluids than for the other lubricants of similar viscosity. This difference is believed to be due to greater lubricant spin heating within the contact zone as a result of the traction fluids' relatively high coefficients of traction.

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TABLE I. - TEST LUBRICANT PROPERTIES

Property	Lubricant description					
	Paraffinic mineral oil	Formulated tetraester	Tetraester base	Traction fluid 1	Traction fluid 2	Synthetic paraffinic oil
Additives	None	Antiwear Oxidation inhibitor Corrosion inhibitor	None	Oxidation inhibitor	Antiwear Oxidation inhibitor Antifoam Viscosity index improver	Antiwear Oxidation inhibitor
Kinematic viscosity, cm^2/sec (cS) at						
311 K (100° F)	0.284 (28.4)	0.29 (29)	0.29 (29)	0.23 (23)	0.34 (34)	0.60 (60)
372 K (210° F)	0.049 (4.9)	0.053 (5.3)	0.053 (5.3)	0.037 (3.7)	0.056 (5.6)	0.089 (8.9)
Flash point, K ($^\circ$F)	480 (405)	533 (500)	533 (500)	422 (300)	435 (325)	538 (510)
Fire point, K ($^\circ$F)	505 (450)	-----------	-----------	435 (325)	447 (345)	575 (575)
Autoignition temperature, K ($^\circ$F)	-----------	694 (800)	694 (800)	589 (600)	600 (620)	645 (700)
Pour point, K ($^\circ$F)	261 (10)	214 (-75)	214 (-75)	230 (-45)	236 (-35)	225 (-55)
Specific heat at 311 K (100° F), J/kg·K (Btu/lb·$^\circ$F)	-----------	1920 (0.467)	1920 (0.467)	2130 (0.51)	2130 (0.51)	2090 (0.50)
Thermal conductivity at 311 K (100° F), J/m·sec·K (Btu/hr·ft·$^\circ$F)	-----------	0.16 (0.094)	0.16 (0.094)	0.10 (0.06)	0.10 (0.06)	0.14 (0.08)
Specific gravity at 311 K (100° F)	-----------	0.977	0.977	0.886	0.889	0.824

TABLE II. - ROLLING-ELEMENT FATIGUE LIFE OF AISI 52100 BALLS LUBRICATED WITH ONE OF SEVERAL SYNTHETIC LUBRICANTS OR A PARAFFINIC MINERAL OIL IN THE FIVE-BALL FATIGUE TESTER

[Maximum Hertz stress, 5520 MPa (800 000 psi); shaft speed, 10 700 rpm; contact angle, 30°.]

Lubricant	Rolling-element fatigue life, millions of upper-ball stress cycles		Weibull slope	Failure index[a]	Confidence number,[b] percent	
	10-percent life	50-percent life			10-percent life	50-percent life
Paraffinic mineral oil	34.8	214	1.04	15 out of 37	--	---
Formulated tetraester	14.3	71.1	1.17	35 out of 39	83	>99
Tetraester base	10.5	46.2	1.27	38 out of 40	92	>99
Traction fluid 1	17.9	115.9	1.01	21 out of 32	69	94
Traction fluid 2	17.3	72.7	1.31	15 out of 28	72	>99
Synthetic paraffinic oil	9.5	91.9	.83	26 out of 36	87	>99

[a]Number of failures out of total number of tests.

[b]Percent of time that the fatigue life with the mineral oil will be greater than that with the specific synthetic lubricant.

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TABLE III. - TEN-PERCENT FATIGUE LIFE ADJUSTED FOR LUBRICANT VISCOSITY DIFFERENCES AT OPERATING TEMPERATURE

Lubricant	Experimental 10-percent fatigue life, millions of upper-ball stress cycles	Average race temperature, K (°F)	Kinematic viscosity at race temperature, cm^2/sec (cS)	Relative life factor[a]	Ten-percent fatigue life adjusted for viscosity of mineral oil, millions of upper-ball stress cycles
Paraffinic mineral oil	34.8	336 (146)	0.113 (11.3)	1.0	34.8
Formulated tetraester	14.3	339 (150)	.105 (10.5)	1.02	14.6
Tetraester base	10.5	344 (160)	.098 (9.8)	1.04	10.9
Traction fluid 1	17.9	347 (165)	.060 (6.0)	1.21	21.6
Traction fluid 2	17.3	347 (165)	.101 (10.1)	1.03	17.8
Synthetic paraffinic oil	9.5	353 (175)	.14 (14.0)	.94	8.9

[a]Based on life proportional to viscosity raised to a power of 0.3.

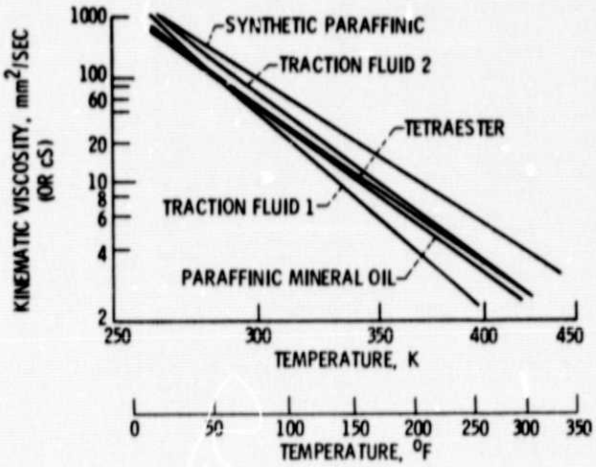

Figure 1. - ASTM chart of test lubricant kinematic viscosity as a function of temperature.

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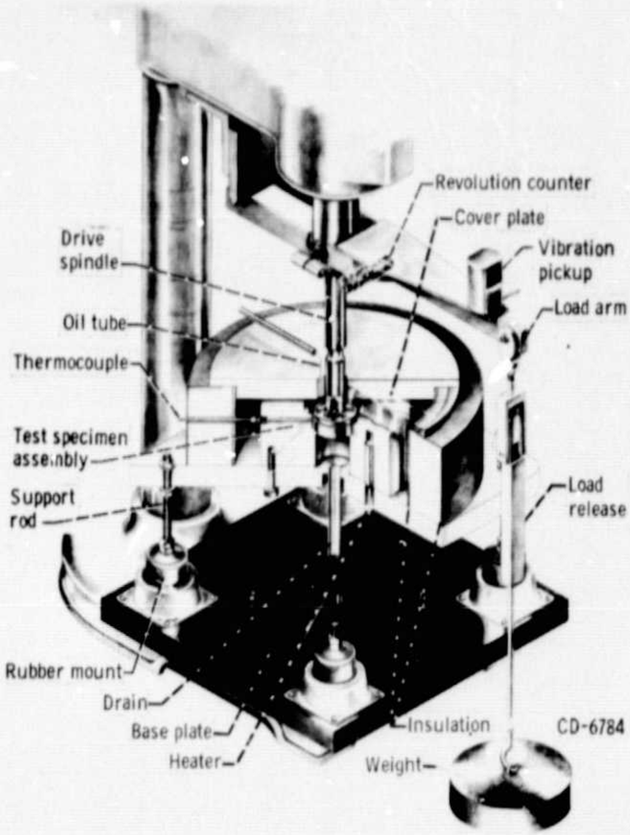
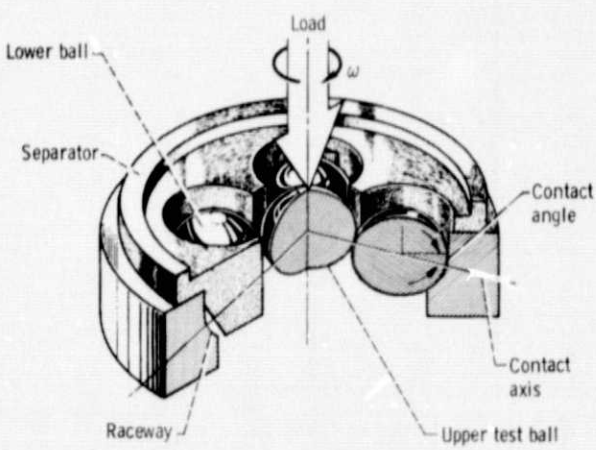


(a) Cutaway view of five-ball fatigue tester.



(b) Five-ball test assembly.

Figure 2. - Test apparatus.

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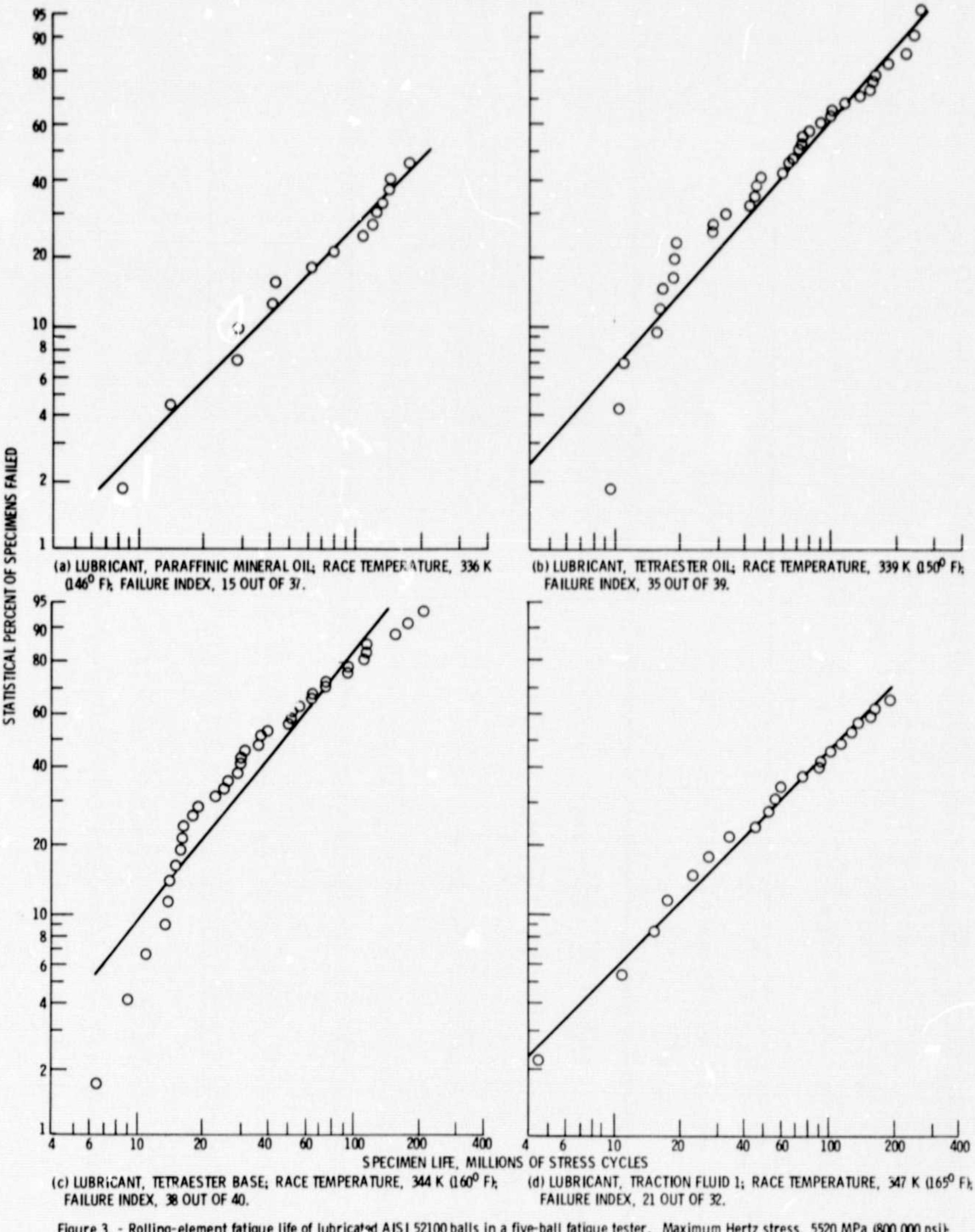


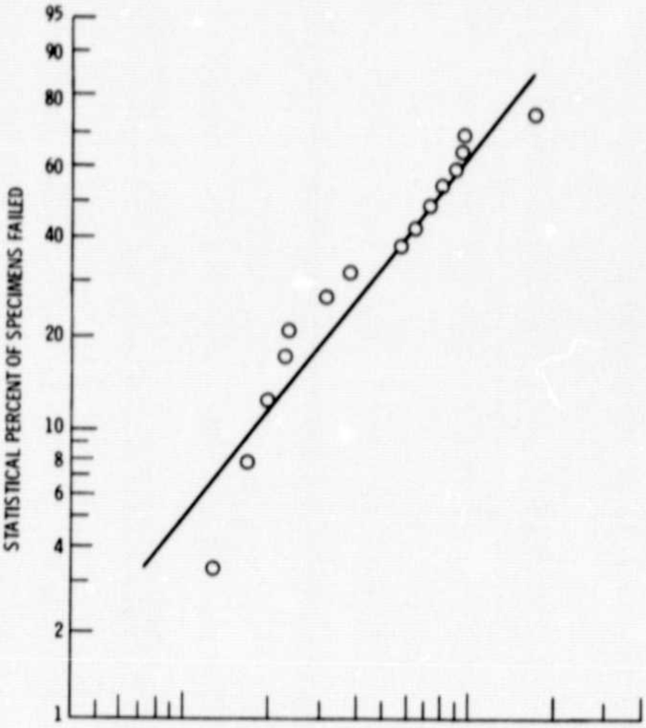
(a) LUBRICANT, PARAFFINIC MINERAL OIL; RACE TEMPERATURE, 336 K (146° F); FAILURE INDEX, 15 OUT OF 37.

(b) LUBRICANT, TETRAESTER OIL; RACE TEMPERATURE, 339 K (150° F); FAILURE INDEX, 35 OUT OF 39.

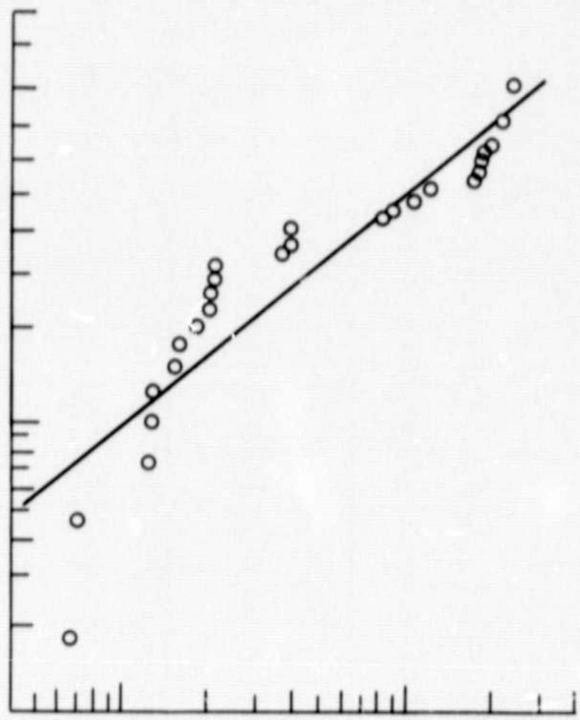
(c) LUBRICANT, TETRAESTER BASE; RACE TEMPERATURE, 344 K (160° F); FAILURE INDEX, 38 OUT OF 40.

(d) LUBRICANT, TRACTION FLUID 1; RACE TEMPERATURE, 347 K (165° F); FAILURE INDEX, 21 OUT OF 32.

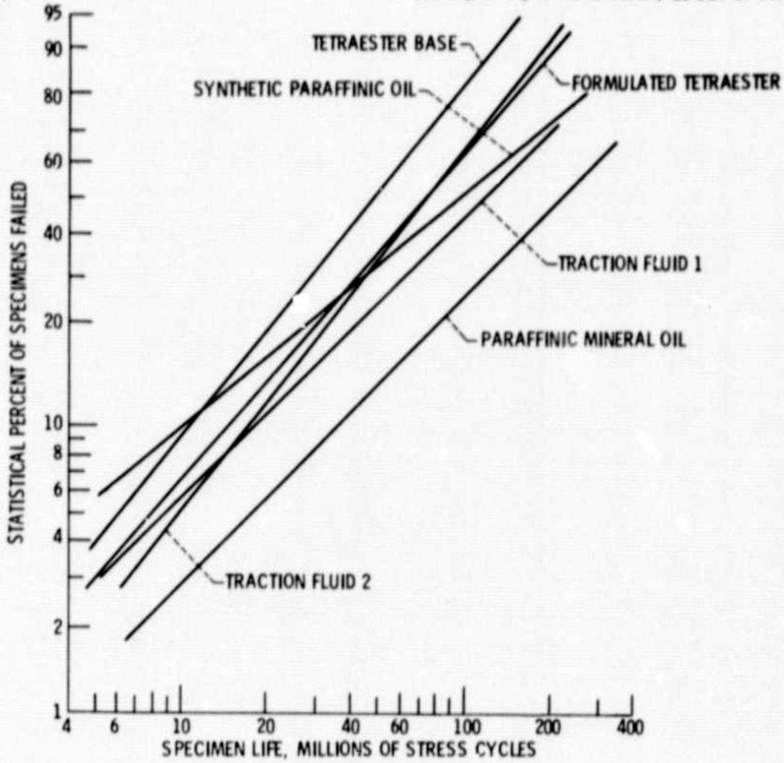
Figure 3. - Rolling-element fatigue life of lubricated AISI 52100 balls in a five-ball fatigue tester. Maximum Hertz stress, 5520 MPa (800 000 psi); shaft speed, 10 700 rpm; contact angle, 30°.



(e) LUBRICANT, TRACTION FLUID 2; RACE TEMPERATURE, 347 K (165° F); FAILURE INDEX, 15 OUT OF 28.



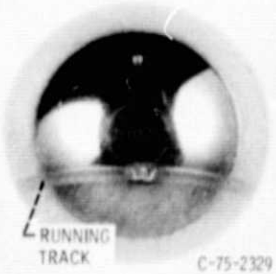
(f) LUBRICANT, SYNTHETIC PARAFFINIC OIL; RACE TEMPERATURE, 353 K (175° F); FAILURE INDEX, 26 OUT OF 36.



(g) SUMMARY.

Figure 3. - Concluded.

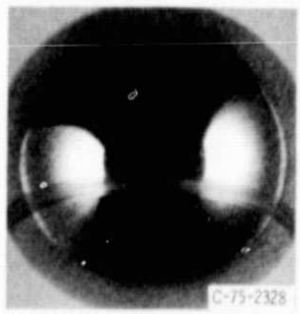
E-8751



(a) TYPICAL FATIGUE SPALL; 48.3 TEST HOURS (93 MILLION UPPER BALL STRESS CYCLES).

(b) TEST ABORTED AFTER 0.5 TEST HOUR (1.0 MILLION UPPER BALL STRESS CYCLES).

(c) TEST SUSPENDED AFTER 52.6 HOURS (101 MILLION UPPER BALL STRESS CYCLES).



(d) TEST RUNOUT TO 101.3 HOURS (195 MILLION UPPER BALL STRESS CYCLES).

Figure 4. - Appearance of upper balls from five-ball fatigue tester lubricated with traction fluid 2.