N71-14786

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ANTIWEAR AND EP ADDITIVE EFFECTS DURING ROLLING AND SPINNING CONTACT WITH A SYNTHESIZED HYDROCARBON OIL

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TECHNICAL PAPER proposed for presentation at
Conference on Chemical Effects at Bearing Surfaces
sponsored by the Institution of Mechanical Engineers
Swansea, England, January 6-8, 1971

E-5411

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ABSTRACT

Tests were conducted to determine the effect of antiwear and extreme-pressure (EP) additives in a synthesized hydrocarbon oil on bearing spinning torque and endurance. These additives were stearic acid, oleic acid, oleyl phosphate, oleyl phosphite, zinc dithiophosphate and a substituted organic phosphonate. Under conditions of pure spinning, elastohydrodynamic conditions prevailed without significant surface interaction. The addition of the antiwear and EP additives in concentrations of 0.1, 1.0 and 10 percent did not change the spinning torque over that obtained with the base fluid in the tests with pure spinning and no rolling. The viscoelastic properties of the base fluid were not changed by the additives tested. The substituted organic phosphonate additive proved effective to temperatures of 600° F in preventing gross surface damage with full-scale angular-contact ball bearings. Under a low oxygen environment at 600° F, 120-mm bore angular-contact ball bearings had a life nearly 14 times AFBMA-predicted (catalog) life with the fluid having the organic phosphonate additive.

INTRODUCTION

In bearing and gear applications, where thin film or boundary lubrication conditions may exist, antiwear and extreme-pressure (EP) additives are used in the lubricating fluid. In many gear applications, the addition of antiwear or EP additives to the lubricant can prevent excessive wear of the gear teeth. In a ball bearing, the ball spins and rolls in an angular-contact raceway. Bearing power loss is due to a number of factors: shearing of the lubricant in the bearing cavity; rubbing of the ball against the cage pocket, rubbing of the cage against one of the raceways, and spinning of the ball in the raceway. Antiwear and EP additives could decrease this power loss by reducing friction in the ball-cage contact and the race-cage contact. In addition, the additives should also decrease wear in these sliding contacts. The question remains whether friction is reduced in the ball-race contact.

The addition to the lubricant of certain reactive materials, in a sliding condition where boundary lubrication exists, may decrease the coefficient of friction over that obtained with a nonreactive lubricant. However, a search of the literature reveals little or no diffinitive work which establishes with certainty final reaction products of various additives with bearing steels. Additives such as stearic acid and oleic acid on steel form metallic soap films [1,2] that shear easily and reduce sliding friction [3-5]. The phosphate additives such as tricresyl phosphate [6] and oleyl phosphate probably form surface films of ferric phosphate on the surface of steel, causing a reduction of wear and friction under boundary sliding conditions.

Under elastohydrodynamic (EHD) conditions, a rolling element is separated from a mating surface by a thin lubricant film [7]. This thin film may be the same order of thickness as the boundary film formed by the antiwear or EP additives. Where there is complete separation of the surfaces, antiwear and EP additives should have very little or no effect on wear but may have some effect on friction due to a change in the lubricant shear behavior, that is, a change in lubricant rheology. This rheological change could cause a change in the EHD film thickness. A decrease in film thickness should result in an increase in measured torque [8]. Where there is significant asperity interaction, an additive of the antiwear or EP type should have some measurable effect on friction and wear.

Analysis [9] has indicated that, for a ball spinning in a nonconforming groove without rolling, an elastohydrodynamic film can be formed. However, it was not determined whether the film was thick enough to prevent surface interactions. If surface interactions did occur, the values of torque measured should be significantly affected by the presence of an antiwear or EP additive.

When the advent of gas turbine powerplants first pushed aircraft engine bearing operating temperatures beyond the capabilities of the mineral oils, the development of synthetic lubricating fluids was undertaken. The guiding philosophy of most lubricant development was to obtain high-temperature thermal and oxidative stability in fluids having relatively limited viscosity variation over wide temperature ranges while at the same time reducing lubricant volatility [10]. Among the classes of fluids which have shown superior high-temperature characteristics are the synthetic paraffinics and super-refined mineral oils [11,12]. These fluids can have useful liquid ranges from -20° to 700° F. In addition, they will accept additives which improve their boundary lubricating characteristics.

The research reported herein, which is based, in part, on the work reported initially in [13-15] was undertaken to determine the effect of antiwear and extreme-pressure (EP) additives in a synthesized hydrocarbon oil (100-percent paraffinic) on bearing spinning torque and endurance. These additives were stearic acid, oleic acid, oleyl phosphate, oleyl phosphite, zinc dithiophosphate and a substituted organic phosphonate. All experimental results were obtained with lubricant from the same batch and specimens from the same heat of material unless otherwise specified.

APPARATUS, SPECIMENS, AND PROCEDURE

Three test apparatuses were used in the tests reported. These were a "spinning torque apparatus", a "high-speed, high-temperature bearing test apparatus", and a "high-temperature fatigue test apparatus".

Spinning Torque Apparatus

A spinning torque apparatus (see Fig. 1 as described in [16,17]) essentially consists of a turbine drive, a pneumatic load device, an upper and lower test specimen, a lower test-housing assembly incorporating a hydrostatic air bearing, and a torque measuring system. In operation, the upper test specimen is pneumatically loaded against the lower test specimen through the drive shaft. As the drive shaft is rotated, the upper test specimen spins in the groove of the lower test specimen. This causes an angular deflection of the lower test-specimen housing which is restrained by a torsion wire. This angular movement is sensed optically by the torque-measuring system and is converted into a torque value. During a test, the torque is continuously recorded on a strip chart.

The upper test specimen is a conventional 1/2-inch diameter bearing ball made of SAE 52100 steel (table I) having a nominal Rockwell C hardness of 61 and a surface finish of 2 microinches rms. The lower test specimen (Fig. 2) is a 1/2-inch diameter ball (from the same heat of material as the upper test specimen) which is modified by grinding a flat on one side and a cylindrical groove of radius R_g (Fig. 2) on the other. The groove simulates the race groove of a bearing. The axis of the groove is parallel to the flat. The groove radius expressed as a percentage of the upper-ball diameter is defined as the ball-race conformity. The specimens used in these tests were ground to a ball-race conformity of 55 percent. The surface finish of the cylindrical groove was approximately 2 to 6 microinches rms.

High-Speed, High-Temperature Bearing Tester

A high-speed, high-temperature bearing test apparatus (Fig. 3) was used for the endurance tests with the 25-mm bore angular-contact ball bearings. In this apparatus, two thrust-loaded test bearings are mounted on a drive shaft. Integral with the drive shaft are two screw pumps which supply lubricant to the test bearings. The oil sump and the test housing are inerted by nitrogen gas. Temperatures are monitored by thermocouples on the test-bearing outer races, in the oil sump, in the test housing, and on the pump liner. Cartridge-type heaters maintain the desired temperature in the test housing and oil sump. The oil sump temperature can be controlled within $\pm 13^{\circ}$ F. The mean temperatures of the two test bearings are not controlled separately. Experience has shown that the bearing mean temperatures in any specific test usually differ by less than 17° F.

Instrumentation provides for automatic failure detection due to increased bearing temperature or vibration. The apparatus is capable of temperatures to 1000° F, bearing thrust loads to 1000 lb, and speeds to 50 000 rpm. A more detailed description of this tester can be found in [18].

Test bearings used in the high-speed, high-temperature bearing tester were 7205-size (25-mm bore) angular-contact ball bearings of ABEC-5 specification having counterbored outer races (table II). The bearings were manufactured from consumable-electrode vacuum-melted (CVM) AISI M-1 steel (table I). The inner and outer races were black oxide coated to improve their resistance to surface distress due to marginal lubricating conditions. Nominal Rockwell C hardnesses for the balls and races were 64 and 65, respectively. The inner-race-riding retainer was silver-plated AISI M-1 steel of nominal Rockwell C hardness 58. Further specifications are shown in table II.

High-Temperature Fatigue Tester

The high-temperature fatigue tester used in the rolling-element fatigue tests with the 120-mm bore angular-contact ball bearings is shown in figure 4. This tester is described in detail in [19]. Essentially the tester consists of a shaft to which two test bearings are attached. Thrust loading is applied through a system of 10 springs. Drive of the test rig is accomplished by a flat belt on a crowned spindle (not shown in the illustration).

Lubrication is provided to the test bearings through a jet-feed lubrication system which uses a pump immersed in a heated oil reservoir. The reservoir has approximately a 3-gallon capacity. The pump is capable of circulating the oil through the system at 3 gallons per minute at 600° F. Gravity drainage for the lubricant is provided by a single exit under each bearing as well as from the bellows in the center of the bearing assembly.

Instrumentation provides for automatic shutoff by monitoring and recording bearing temperatures, oil temperature, bearing vibration, nitrogen flow rate and pressure, as well as support bearing lubricant flow and pressure. A change in any of these parameters from those programmed for the test conditions will terminate the test. Oxygen content within the bearing housing assembly is monitored frequently during operation. An infrared pyrometer was used to measure inner-race temperature, using a sapphire window sight tube aimed at the inner race of the front test bearing.

The test bearings were ABEC-5 grade, split inner-race 120-mm bore angular-contact ball bearings having a nominal contact angle of 20 degrees (table II). The inner and outer races were manufactured from one heat of CVM-AISI M-50 steel (table I), and the balls were manufactured from a second heat. The nominal Rockwell C hardness of the balls and races was 63 at room temperature. The cage was a one-piece outer-land riding type made of a nickel-base alloy (AMS 4892) having a nominal Rockwell C hardness of 33. The retained austenite content of the ball and race material was measured at less than 3 percent. All components in each bearing with the exception of the cage were matched within ±0.5 point Rockwell C. This matching assured a nominal differential hardness (i.e., the ball hardness minus the race hardness, commonly called ΔH) of zero in each bearing.

Lubricant

The lubricant used for the test was a synthesized hydrocarbon oil (straight paraffinic chain). Properties of the oil are given in table III. It was necessary to maintain this fluid in a low-oxygen environment (less than 1-percent oxygen by volume) at 600° F in order to prevent excessive oxidation.

The lubricant was modified for the tests by the addition of 0.1, 1.0, or 10 percent of the additive (i.e., stearic acid, oleyl acid, oleyl phosphate, oleyl phosphite, zinc dithiophosphate or a low concentration of a substituted organic phosphonate). The stearic acid reacts with the steel to form iron stearate [1]. It is speculated that the oleic acid forms iron oleate with steel [2]. These films reduce boundary friction by reducing the shear strength of the surface layer [3-5]. Based upon the work of [20], it is speculated that the oleyl phosphate and oleyl phosphite may react with iron to form low shear strength iron-phosphorus compounds. The phosphonate additive may react with steel in a similar manner to tricresyl phosphate which probably forms a low shear strength iron phosphate film [6] or an iron oxide film [21] on the bearing surface. These films also reduce the shear strength of the surface and thus reduce friction [6]. The interaction of zinc dithiophosphate with ferrous surfaces results in formation of surface films which reduce friction and wear [22].

RESULTS AND DISCUSSION

Spinning Torque Tests

Tests were conducted with SAE 52100 steel 1/2-inch diameter balls in the spinning torque apparatus against lower grooved test specimens with a conformity of 55 percent. The resulting torques due to ball spinning were measured. The results were evaluated with respect to the type and amount of additive and the maximum Hertz stress.

The results of the tests with the synthesized hydrocarbon oil without any additives are shown in figure 5(a) as a function of stress. These results compare with data obtained under similar conditions and reported in [16,17]. These data show increasing spinning torque with increasing Hertz stress. The rate and magnitude of this increase can be predicted based on elastohydrodynamic (EHD) theory [9].

The condition in the Hertzian contact is either one of boundary lubrication with metal to metal contact or of EHD lubrication with a continuous film. In the former case the addition of the additives used in this program will decrease the torque through its influence on the coefficient of friction under boundary lubrication conditions [1-6,10]. In the latter case, there could be an effect on torque only if the anti-wear or EP additive changes the viscoelastic properties of the lubricant. A change in the viscoelastic properties might alter the lubricant's shear sensitivity and film thickness, and thus the torque.

Tests were conducted with the synthesized hydrocarbon fluid containing varying percentages of stearic acid. The results of these tests are presented in figure 5(b). The broken line is the curve from figure 5(a) for the oil without the additive. The results for the stearic acid indicate that there is no effect of the additive on the resultant torque. This result suggests that there exists a complete elastohydrodynamic (EHD) film throughout the contact area and that the lubricant viscoelastic properties are not affected by the additive in a manner that would affect spinning torque.

The results with the oleic acid, oleyl phosphate, oleyl phosphite, and zinc dithiophosphate additives, which are shown in figures 5(c) to (f), were the same as those for the previous additive. No significant change occurred in torque due to the addition or amount of additive contained in the oil.

A typical grooved test specimen that ran with a smooth low torque value did not show evidence of gross metal-to-metal contact but had a slightly smoother surface where the contact ellipse was located. This type of surface and the torque trace indicate the existence of an elastohydrodynamic film over the complete contact ellipse.

If a spinning torque test were run for an extended length of time, the torque trace would become erratic and higher than normal indicating that, because of side leakage and shearing of the lubricant, gross metal contact had occurred. (The lubricant supply was not replenished during a test.) It was found that very often the first contact between the spinning ball and the groove occurred at the edge of the inscribed circle within the contact ellipse (see Fig. 6). This phenomenon can be explained as follows. Upon loading the ball onto the groove, the oil is trapped in the contact zone. The Hertzian pressure, being higher at the contact center, causes a higher viscosity (because of pressure) to exist with a resulting thicker "squeeze film" at the center. Upon rotation, a film is maintained in the area outside the inscribed circle by hydrodynamic action. The greatest shear rate occurs in the "squeeze film" in the inscribed circle at the edge of the circle. Because of increased temperature due to shearing effects and nonNewtonian behavior, the lubricant viscosity decreases. As a result, the first film break-through occurs at the edge of the inscribed circle.

Since the ball is spinning on a thin EHD film, the spinning torque is dependent on the rheological properties of the lubricant. Because the spinning torque was unchanged by the additives in the lubricant, it can be concluded that the rheological properties of the lubricant were unchanged by the additives.

The aforementioned results would imply that where a rolling-element bearing is designed to operate in a predominant EHD range, an EP or antiwear additive should have no significant effect on bearing torque. The question remains whether the additive would affect high-temperature, high-speed bearing endurance.

25-mm Bore Bearing Endurance Tests

An EHD criterion which is referred to as the film parameter was reported in [23] where

$$\Lambda = \frac{H}{\sigma}$$

and

$$\sigma = \left(\sigma_1^2 + \sigma_2^2\right)^{1/2}$$

where

H	minimum film thickness, μ in.
σ	composite surface roughness, μ in., rms
σ_1 and σ_2	surface roughnesses of contacting bodies, μ in., rms

Experimental results indicate that where $\Lambda \lesssim 1$, gross surface distress will occur and boundary lubrication prevails in the entire contact. Where $1 \lesssim \Lambda \lesssim 1.5$, surface distress in the form of glazing and superficial pitting as well as wear will occur. Under this condition, mixed lubrication exists, that is, a combination of boundary and EHD lubrication, with boundary lubrication being the predominant mode. In the range $1.5 \lesssim \Lambda \lesssim 3$, some surface glazing and wear might occur. For this range, mixed lubrication also occurs; however, EHD lubrication is the predominant mode. For $3 \lesssim \Lambda \lesssim 4$, nearly full film separation or EHD lubrication occurs with no visible surface damage or measurable wear. Some insignificant surface asperity contact will occur. At $\Lambda > 4$, full lubrication occurs.

Most rolling-element bearings operate in the range $1.5 \lesssim \Lambda \lesssim 3$. Inasmuch as the EP and antiwear additives do not affect lubricant rheology, they would not be expected to grossly affect bearing endurance in this Λ range. However, they may be important at times of bearing start-up, shutdown during intermittent EHD film breakdown and where values of are 1.5 or less. In order to determine the additive effect in full scale bearing operation at high temperatures and high speeds, tests were conducted with 25-mm bore angular-contact ball bearings (table II). For the test conditions listed in table II, $\Lambda \approx 2.7$ and $\Lambda \approx 1.5$, were calculated for the inner and outer races at 600° F, respectively, using the methods of Harris [24].

Eighteen 7205-size (25-mm bore) bearings were tested with the synthesized hydrocarbon oil without additives at 42 800 rpm and 459-lb thrust load at 500° to 600° F outer-race temperature in a low oxygen environment (less than 1 percent by volume). These bearings exhibited many early failures both by smearing and by fatigue. Only two bearings reached the calculated AFBMA 10-percent (catalog) life of 240 million revolutions (~90 hr at 42 800 rpm and 459-lb load). Ten additional bearings were tested with this lubricant containing the substituted organic phosphonate in a very low concentration under the aforesaid conditions.

All ten bearings ran to twice the AFBMA 10-percent (catalog) life without failure. No gross surface distress or wear was apparent in these bearings although some glazing was apparent. In addition, the bearings and all parts of the tester were free of decomposition products. The clean, undamaged components of one of these bearings are shown in figure 7. It is apparent that the additive provides effective boundary lubrication thus inhibiting gross surface damage.

Bearing Fatigue Tests at 600° F

Based upon the results with the 25-mm bore bearing, it was concluded that an antiwear additive must be provided in a lubricant in order to assure long term reliability under high-temperature, high-speed rolling-element conditions. It is desirable to determine the effectiveness of the synthesized hydrocarbon oil with the substituted organic phosphate additive under conditions simulating advanced turbine engine operation. In order to accomplish this, tests were run with 120-mm bore angular-contact bearings in the high-temperature bearing fatigue test apparatus at a temperature of 600° F and a speed of 12 000 rpm under a low oxygen environment (less than 0.1 percent oxygen by volume). Four bearings were initially run at a thrust load of 7000 lb, which produced a maximum Hertz stress of 342 000 psi on the inner race. While the bearings did not fail from fatigue, ball diametrical wear averaging 0.002-0.003 in. occurred in each bearing for operating times up to 165 hr (118×10^6 inner-race revolutions). These results indicated that mixed lubrication occurs at this stress level, that is, a combination of boundary and EHD lubrication with boundary lubrication being the predominant mode ($\Lambda \leq 1.5$). As a result, a thrust load of 5800 lb was selected for the remaining fatigue tests in order to assure a predominant EHD lubrication regime. The calculated values of Λ for the inner and outer races were 2.9 and 3.3, respectively. The measured value of Λ for the ball inner-race contact at this condition using X-ray techniques under simulated conditions was approximately 1.8 [25]. The fatigue-life results for the 26 bearings tested are shown in figure 3. The failure index (i.e., the number of fatigue failures out of the number tested) was 6 out of 26. The ten-percent life is nearly 14 times the AFBMA-predicted (catalog) life at this load condition (also given in figure 8 for comparative purposes).

Typical fatigue spalls occurring on the balls of a bearing run with the synthesized hydrocarbon oil containing the substituted organic phosphonate additive can be seen in figure 9. Metallurgical examination of the bearings indicated that failure was by classical rolling-element fatigue. The fatigue spalls were of subsurface origin, initiating in the zone of resolved maximum shearing stress. An inner-race failure is shown in figure 10.

An unfailed bearing run to suspension (500 hr of operation) is shown in figure 11.

Surface profile measurements of the bearing components made after testing, together with pretest and post-test weight measurements, indicated that there was essentially no measurable wear or weight change in the bearing components. Some of the bearings tested, however, had a slight glazed appearance, which indicated at least occasional asperity contact.

CONCLUDING REMARKS

The results of the research reported herein would indicate that, in general, EP and antiwear type additives in concentrations normally used do not affect the lubricant's rheological properties. These results further indicate that, under extreme operating conditions, the lubricant requires an antiwear or EP additive to help protect the rolling-element surfaces. (Extreme operating conditions can be defined in terms of combinations of high-temperature, high-loads, and high-speeds.) These additives are considered to be important, specifically at times of bearing start-up, shutdown, during intermittent EHD film breakdown and where values of Λ are 1.5 or less.

Under conditions of predominant EHD lubrication, that is, Λ greater than 1.5, it would not be expected that these additives would have any appreciable effect on rolling-element fatigue life. However, at lower Λ values and during start-up and shutdown, the additive can make the difference between reasonably successful operation (that is, the bearing reaching its design life) or catastrophic failure. The lubricant manufacturer must match the additive to the bearing operating temperature, lest the additive evaporate or suffer from thermal instability during high-temperature bearing operation.

Results with the bearings run with the synthesized hydrocarbon oil containing the substituted organic phosphonate indicated no measurable wear on the race lands, cage pocket or rails. For the 25-mm bore bearings, this may be attributed to black oxide coated races and silver plated cages.

However, the 120-mm bore bearings had no protective coatings. It is therefore conceivable that, under the extreme conditions reported herein, the antiwear or EP additive can play a role in inhibiting wear on the rubbing surfaces where no protective coatings are initially provided.

At Λ values less than 1, significant asperity interaction occurs (more than 80 percent of the asperities are in contact) [23]. At these values and under the test conditions reported for the 25-mm and 120-mm bearings, the antiwear or EP additives investigated herein would not be effective in preventing gross surface damage. However, under less severe conditions, it is conceivable that relatively long bearing endurance can be achieved.

SUMMARY

Tests were conducted to determine the effect of antiwear and extreme-pressure additives in a synthesized hydrocarbon oil on bearing spinning torque and endurance. These additives were stearic acid, oleic acid, oleyl phosphate, oleyl phosphite, zinc dithiophosphate and substituted organic phosphonate. The tests were run with a ball spinning without rolling in a nonconforming groove and with 25-mm and 120-mm bore angular-contact bearings to 600° F. The following results were obtained:

1. Under conditions of pure spinning, elastohydrodynamic conditions prevailed with no significant surface interaction.

2. The addition of the antiwear and EP additives in concentrations of 0.1, 1.0 and 10 percent to the synthesized hydrocarbon oil did not change the spinning torque over that obtained with the base fluid.

3. The viscoelastic properties of the base fluid were apparently not changed by the additives tested.

4. Under marginal elastohydrodynamic conditions with the 25-mm and 120-mm bore angular-contact ball bearings, the substituted organic phosphonate additive proved effective in preventing gross surface damage at 600° F.

5. Under a low oxygen environment and at a maximum Hertz stress of 323 000 psi, the 120-mm bore bearings run with the synthesized hydrocarbon oil with the phosphonate additive had a life approximately 14 times the AFBMA-predicted (catalog) life. The fluid provided adequate elastohydrodynamic lubrication whereby no measurable wear occurred.

ACKNOWLEDGEMENTS

The author acknowledges the assistance obtained from the engineers and chemists at the Mobile Research and Development Corporation, Paulsboro, New Jersey, USA, SKF Industries, Inc., King of Prussia, Pennsylvania, USA, General Electric Company, Cincinnati, Ohio, USA, and NASA-Lewis Research Center, Cleveland, Ohio, USA. Mobile Research and Development Corporation manufactured the synthesized hydrocarbon oil and blended the additives. Testing with the 25-mm bore angular-contact ball bearings was conducted by SKF Industries, Inc. under NASA Contract NASw-492. The General Electric Company conducted the tests with the 120-mm bore angular-contact ball bearings under NASA Contract NAS3-7261. The spinning torque tests were conducted at the NASA-Lewis Research Center.

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TABLE 1. - CHEMICAL COMPOSITION OF BEARING STEELS

Material	Carbon	Manganese	Phosphorus	Sulfur	Silicon	Chromium	Molybdenum	Vanadium	Iron
	Composition, wt. percent								
SAE 52100	1.05	0.35	0.025 (max)	0.025 (max)	0.25	1.45	----	----	Bal.
AISI* M-1	0.80	0.30	0.030	0.030	0.30	4.00	8.00	1.00	Bal.
AISI M-50	0.802	0.24	0.008	0.003	0.18	3.95	4.36	0.93	Bal.

*Also contains 1.50 percent tungsten.

TABLE 2. - SPECIFICATIONS OF ANGULAR-CONTACT BALL BEARINGS AND TEST CONDITIONS

Bore size	25-mm	120-mm
Nominal contact angle	19^{o}	20^{o}
Number of balls	12	15
Ball diameter, in.	5/16	13/16
Conformity, percent of ball diameter		
Inner race	52.2	54
Outer race	52.4	52
Material	AISI M-1	AISI M-50
Bearing thrust load, lb	459	5800
Maximum Hertz stress, ksi		
Inner race	244	323
Outer race	250	268
Speed, rpm	42 800	12 000
Temperature, ^{o}F	500 - 600	600

TABLE 3. - TEST LUBRICANT PROPERTIES

	Spinning torque tests - batch 1	25-mm bearing tests - batch 2	120-mm bearing tests - batch 3
Kinematic viscosity, cs, at 100° F 210° F 500° F	 448 43 3.3	 314 32 2.9	 443 40 3.0
Flash point, $^{\circ}$F	515		
Fire point, $^{\circ}$F	600		
Autoignition temperature, $^{\circ}$F	805		
Pour point, $^{\circ}$F	-35	-45	-35
Volatility, weight percent	14.2		
Specific heat at 500° F, Btu/lb/$^{\circ}$F	0.695		
Thermal conductivity at 500° F, Btu/hr/ft/$^{\circ}$F	70×10^{-3}		
Specific gravity at 500° F	0.71		
Antiwear or E-P additive	As indicated	Substituted organic phosphonate	Substituted organic phosphonate
Anti-foam agent	None	None	Silicone

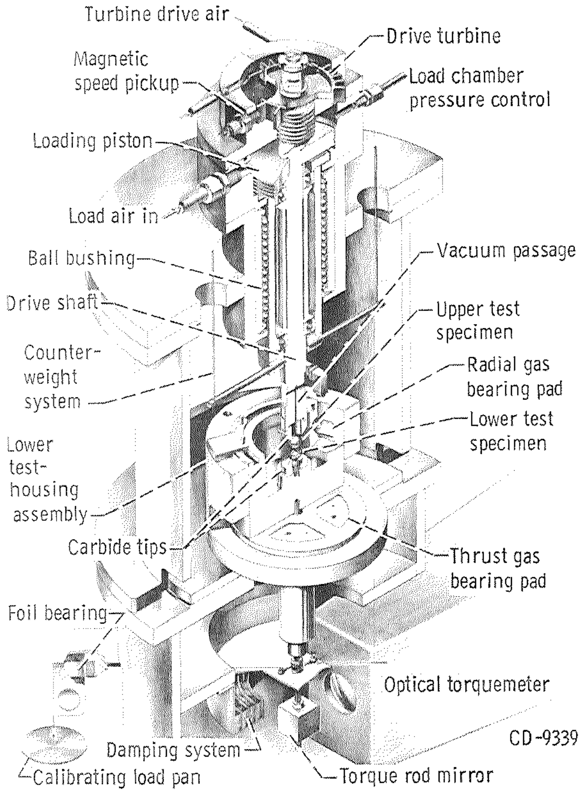


Figure 1. - Spinning torque apparatus.

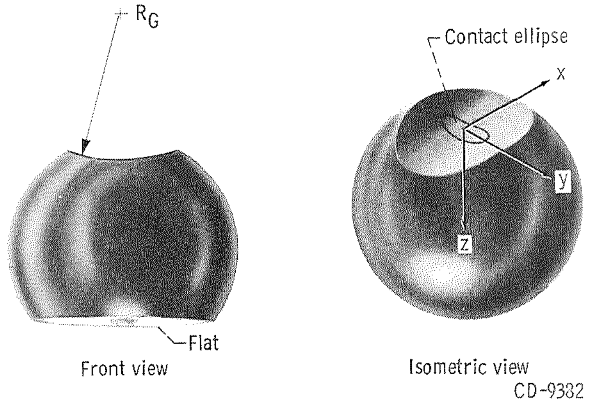


Figure 2. - Lower test specimen for spinning torque tests.

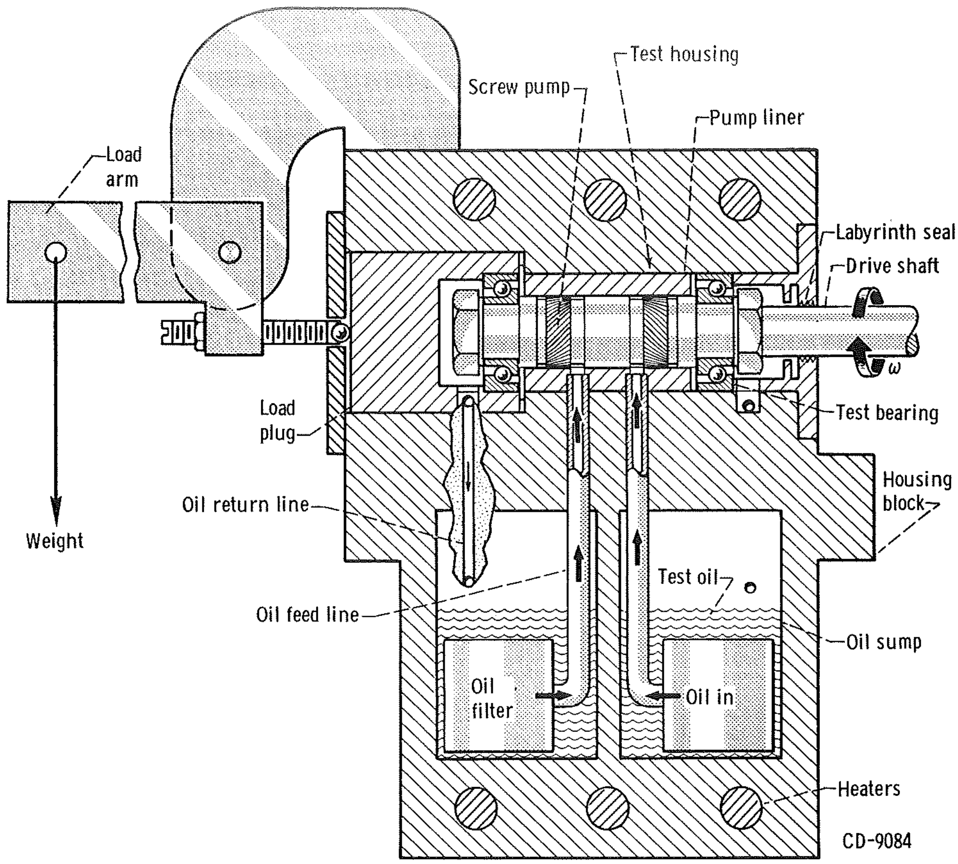


Figure 3. - High-speed, high-temperature bearing tester.

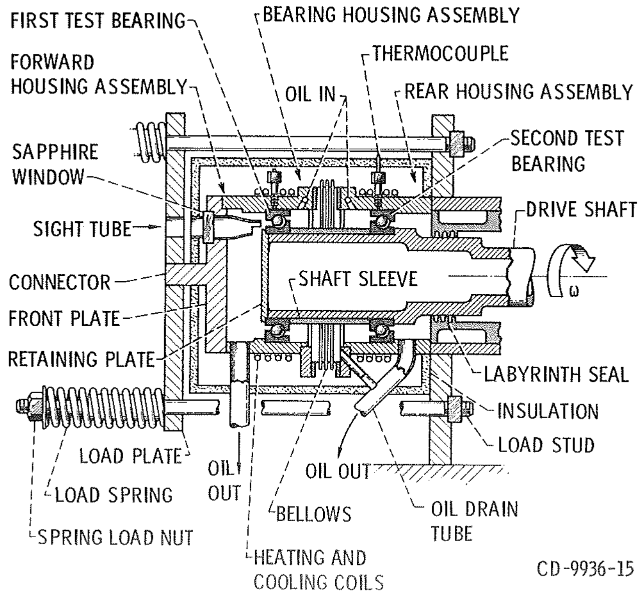


Figure 4. - High-temperature bearing fatigue test apparatus.

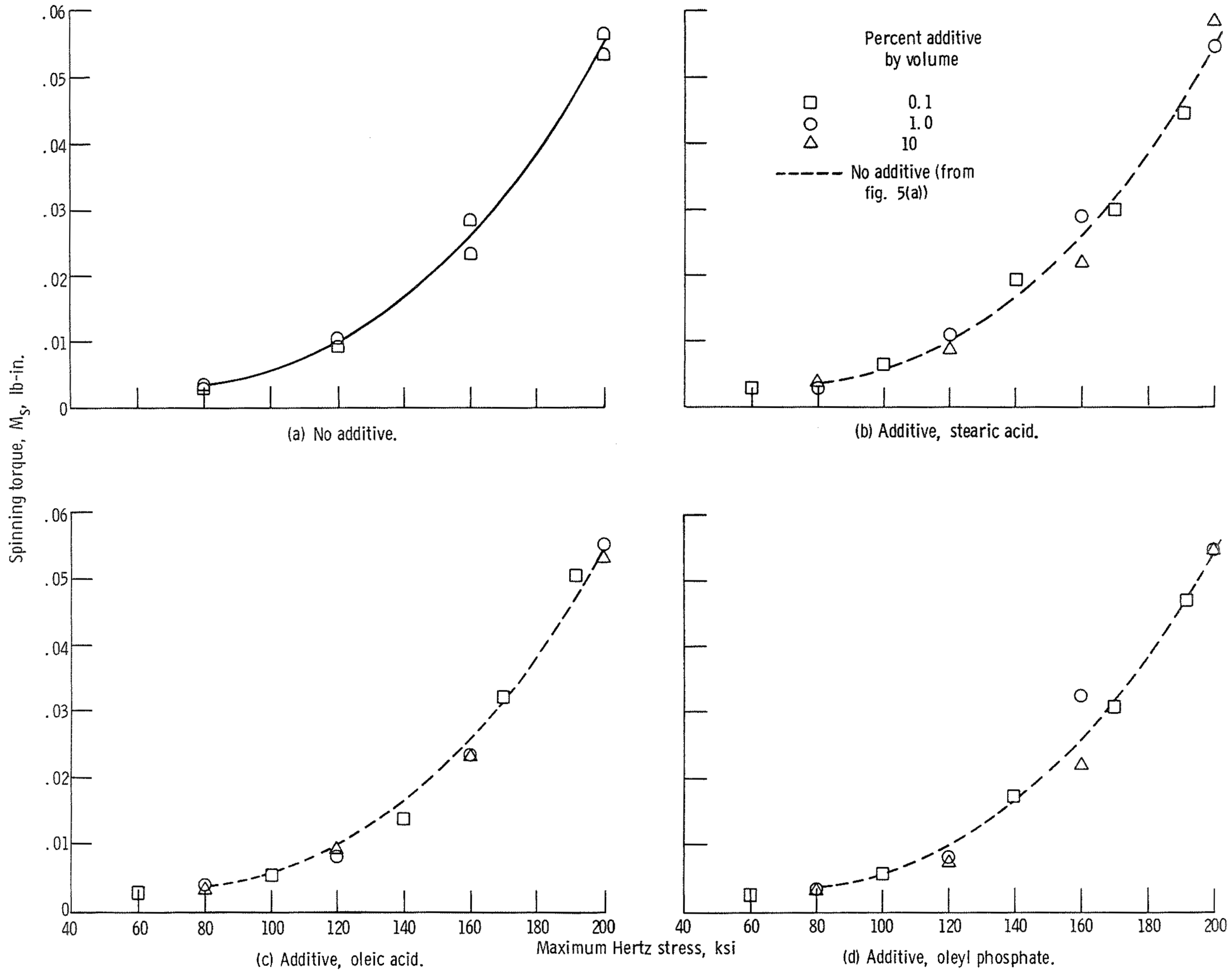
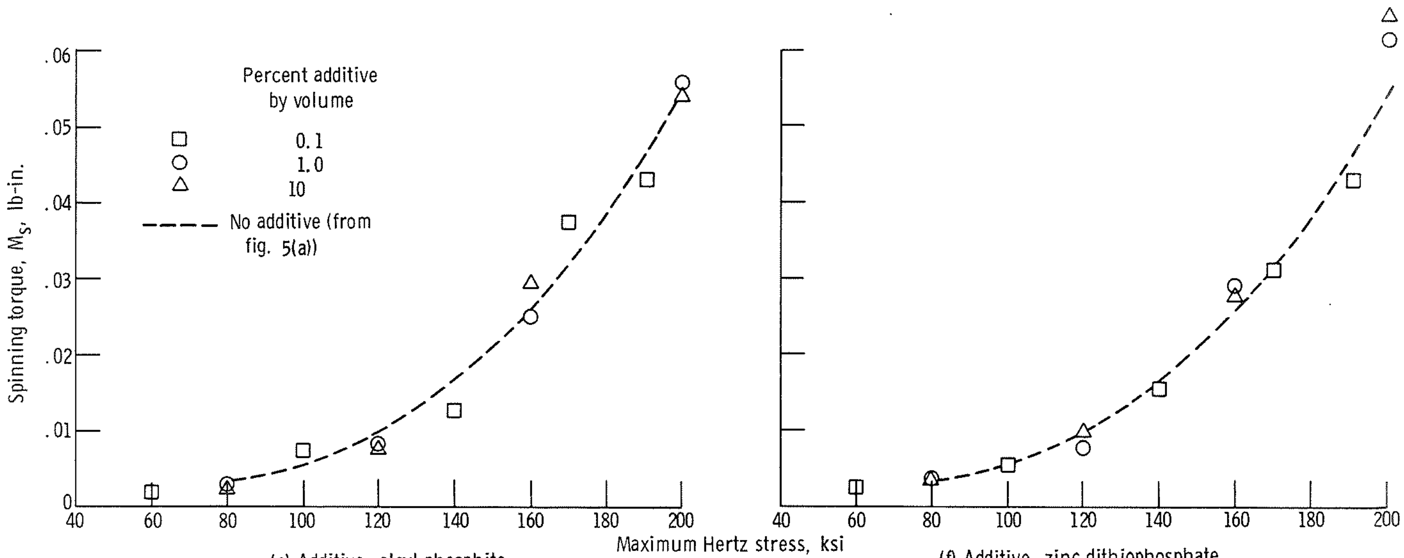


Figure 5. - Spinning torque as function of maximum Hertz stress for synthesized hydrocarbon oil with several additives in varying amounts. Spinning speed, 1000 rpm; conformity, 55-percent; room temperature (no heat added).



(e) Additive, oleyl phosphite.

(f) Additive, zinc dithiophosphate.

Fig. 5. - Concluded.

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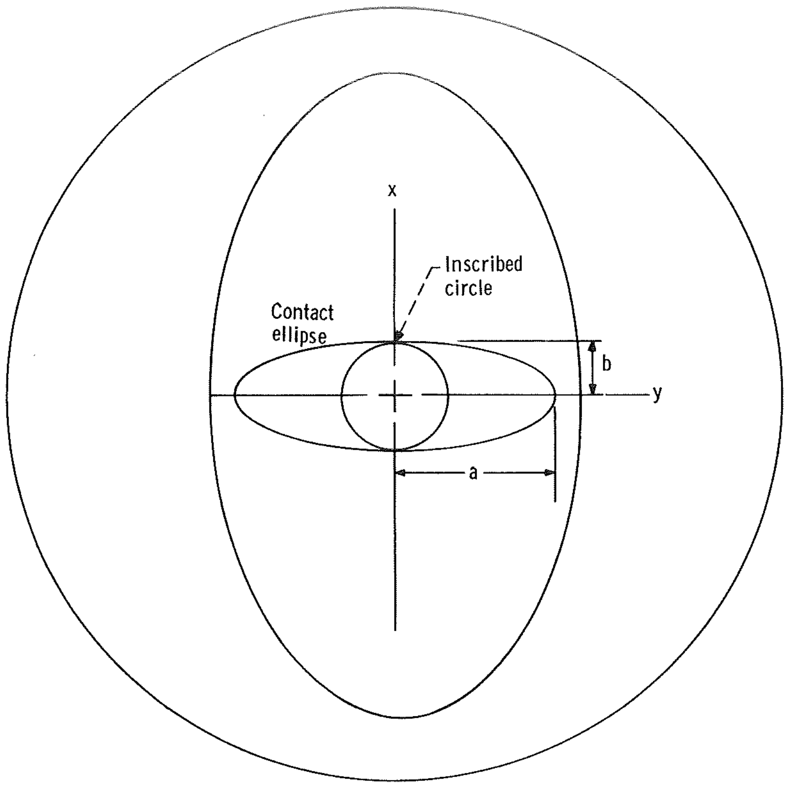

Figure 6. - Contact ellipse for ball in nonconforming groove (see fig. 2).

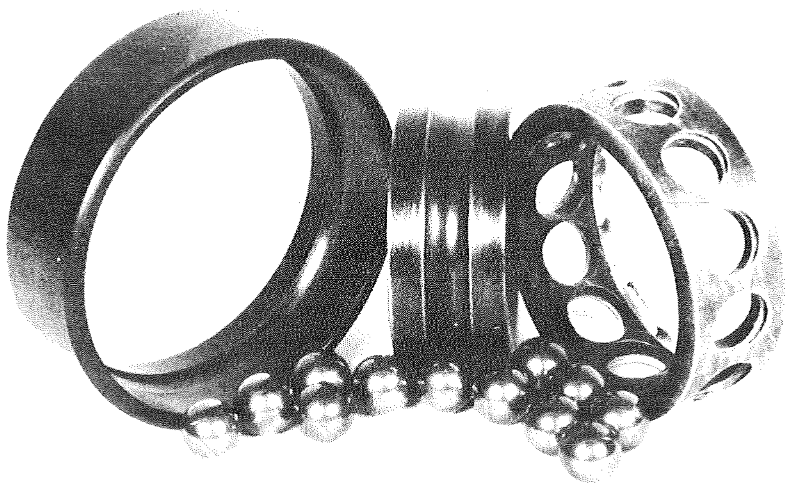

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Figure 7. - Unfailed 25-mm angular-contact ball bearing run with synthesized hydrocarbon oil containing substituted organic phosphonate showing clean, undamaged components after 283 hrs. of operation. Material, AISI M-1 steel; thrust load, 459 lb; speed, 42,800 rpm; temperature, 600° F; low-oxygen environment.

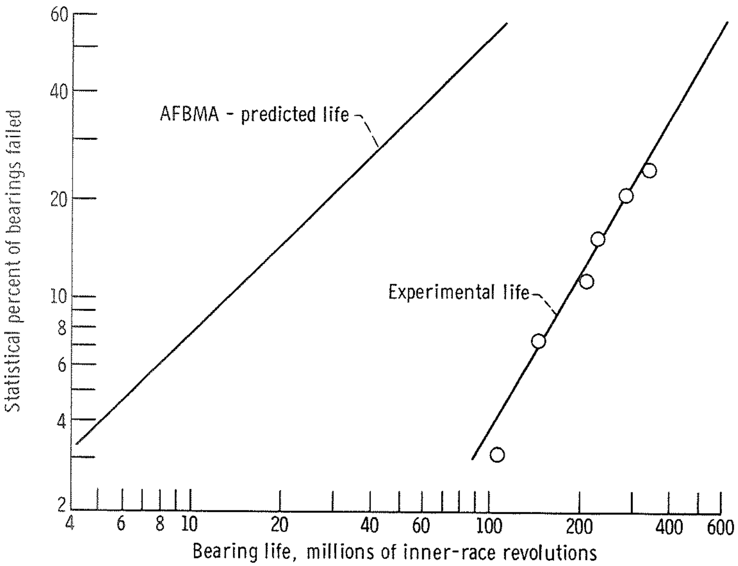

Figure 8. - Rolling-element fatigue life of 120-mm bore angular-contact ball bearings with synthesized hydrocarbon oil containing substituted organic phosphonate. Material, AISI M-50 steel; thrust load, 5800-lb; 12,000 rpm; temperature, 600° F; low oxygen environment; failure index, 6 out of 26.

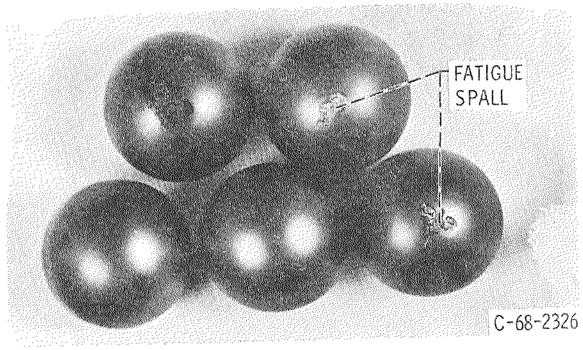

Figure 9. - Typical fatigue spalls on balls of 120-mm angular-contact ball bearing run with synthesized hydrocarbon oil containing substituted organic phosphonate. Material, AISI M-50 steel; thrust load, 5800 lb; speed, 12,000 rpm; temperature, 600° F; low-oxygen environment; running time, 375 hrs. (270×10^6 inner-race revolutions).

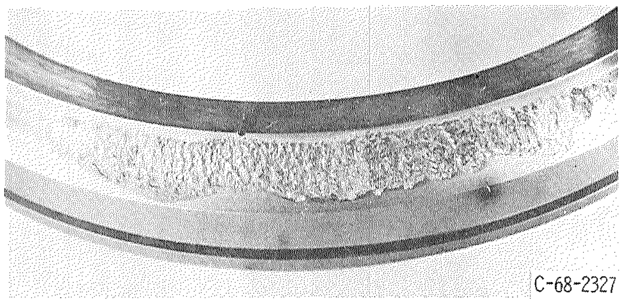

Figure 10. - Fatigue failure on inner race of 120-mm angular-contact ball bearing run with synthesized hydrocarbon oil containing substituted organic phosphonate. Material, AISI M-50 steel; thrust load, 5800 lb; speed, 12,000 rpm; temperature, 600° F; low-oxygen environment; running time, 189 hrs. (136×10^6 inner-race revolutions).

Figure 11. - Unfailed 120-mm angular-contact ball bearing run with synthesized hydrocarbon oil containing substituted organic phosphonate. Material, AISI M-50 steel; thrust load, 5800 lb; speed, 12,000 rpm; temperature, 600° F; low-oxygen environment; running time, 500 hrs. (360×10^6 inner-race revolutions).