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EFFECT OF CONTACT ANGLE ON ROLLING-ELEMENT FATIGUE LIFE WITH A FLUORINATED ETHER LUBRICANT AT A CRYOGENIC TEMPERATURE OF 170 K (305° R)

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16. Abstract

Rolling-element fatigue tests were conducted at a temperature of 170 K (305° R) at contact angles of 20°, 30°, and 40° in a modified five-ball fatigue tester with AISI 52100 steel balls. A fluorinated ether fluid was used as the lubricant. The maximum Hertz stress was 5500 MN/m^2 (800 000 psi). Test results indicate that there is a decrease in fatigue life with increased contact angle. The differences in fatigue life were not statistically significant. The decrease in life with contact angle in fluorinated ether fluid at 170 K (305° R) is identical to the trend with a diester fluid at 328 K (590° R). Analysis indicates that this decrease in fatigue life is caused by decreased EHD film thickness and decreased ball hardness due to increased contact temperature. The increase in contact temperature was of the same order of magnitude for the tests with a fluorinated ether at 170 K (305° R) as with a diester at 328 K (590° R).

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EFFECT OF CONTACT ANGLE ON ROLLING-ELEMENT FATIGUE

LIFE WITH A FLUORINATED ETHER LUBRICANT AT A

CRYOGENIC TEMPERATURE OF 170 K (305° R)

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SUMMARY

Rolling-element lubrication tests were conducted with 12.7-millimeter (1/2-in.) diameter AISI 52100 steel balls. The tests were run in the NASA five-ball fatigue tester modified for cryogenic temperature testing. Test conditions included a drive shaft speed of 4750 rpm, a maximum Hertz stress of 5500 MN/m^2 (800 000 psi), an outer-race temperature of 130 K (235° R); a lubricant temperature of 170 K (305° R); and contact angles of 20°, 30°, and 40°. A fluorinated ether fluid was used as the lubricant.

There was a decrease in fatigue life with increased contact angle. However, the differences among the fatigue lives obtained at 20°, 30°, and 40° contact angles were not statistically significant. This decreasing trend in fatigue life with increased contact angle in fluorinated ether fluid at 170 K (305° R) is identical to the trend with a diester fluid at contact angles of 20°, 30°, and 40° at 328 K (590° R). Analysis indicates that the decrease in fatigue life with the fluorinated ether fluid is caused by decreased elasto-hydrodynamic (EHD) film thickness and decreased ball hardness caused by increased contact temperature. The change in contact temperature with contact angle calculated for the fluorinated ether at 170 K (305° R) is the same order of magnitude as that measured in tests with a diester fluid at 328 K (590° R).

INTRODUCTION

Rolling-element bearings are used in most engineering designs with rotating components. If a rolling-element bearing is properly designed so that it has adequate load capacity and sufficient lubrication and cooling, it will ultimately fail by rolling-element fatigue.

A rolling-element fatigue failure occurs when the surface material of the rolling-elements or races of a bearing spalls thereby producing a pit. Vibration and temperature levels in the bearing increase, and the bearing is no longer useful for the purposes for which it was designed.

The fatigue life of a rolling-element bearing is affected by many variables. Among these are operating variables such as load, speed, and temperature and design variables such as bearing material, lubricant type, and bearing geometry. One important geometrical aspect of the bearing design is the contact angle. Jones (ref. 1) defines free bearing contact angle as, "The angle made by a line passing through the points of contact of the ball and both raceways with a plane perpendicular to the axis of the bearing when both races are centered with respect to each other and one race is axially displaced with respect to the other without the application of measurable force." The actual contact angle in a running bearing will vary from the free contact angle as a function of the bearing speed and load. It may also be different at the inner and outer races of the bearing. This variation due to load and speed is considered by Jones (refs. 1 to 3) and Harris (ref. 4). The general variations in contact angle are shown in figure 1.
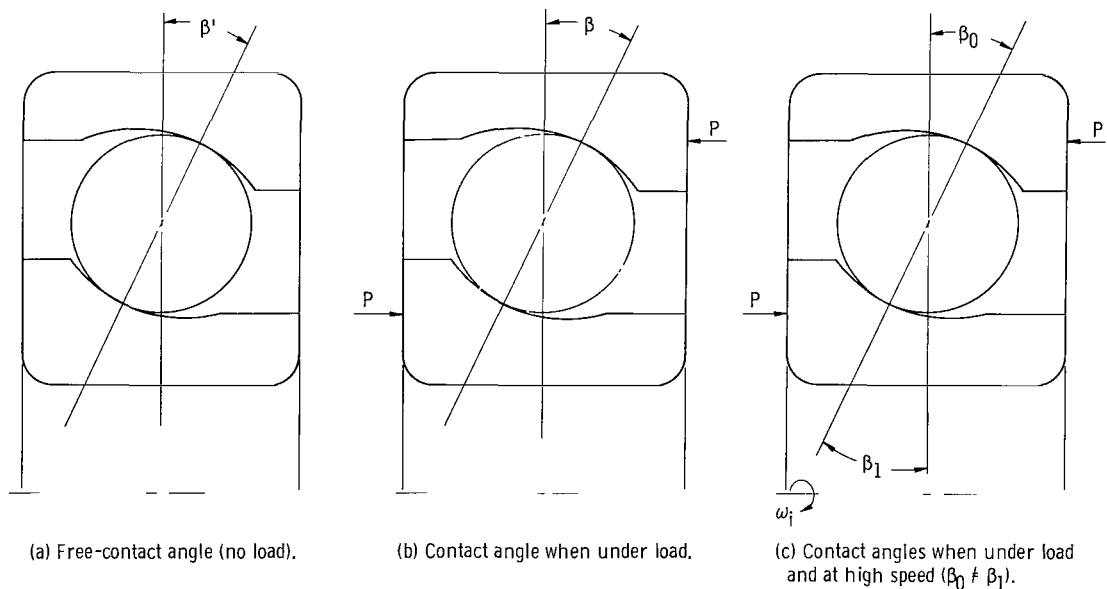

(a) Free-contact angle (no load).   (b) Contact angle when under load.   (c) Contact angles when under load and at high speed ($\beta_0 \neq \beta_1$).

Figure 1. – Change in contact angle with load and speed.

From the analysis of Lundberg and Palmgren (ref. 5)

$$L = K_1 \left(\frac{1}{N}\right)^p$$

where L equals the 10-percent fatigue life, N equals the normal ball load in pounds, K_1 is a constant, and p is generally accepted as three for point contact. From this analysis, for a constant normal ball load N, no change in life should occur as the contact angle increases. However, the data of reference 6 indicate that, for a constant normal ball load, fatigue life decreases as contact angle increases.

The authors of reference 6 indicate that the discrepancy between their data and the analysis of reference 5 may be due to thermal stress effects caused by sliding (or spinning) in the ball-race contact. The sliding velocity increases as contact angle increases thus causing higher local temperatures. The increased temperature will, in turn, affect the viscosity of the lubricant, and thus the thickness of the elastohydrodynamic (EHD) film between the rolling-element surfaces.

For space applications such as the proposed space shuttle and large orbiting space stations, bearings may be operated under liquid lubrication conditions at cryogenic temperatures in fuel and oxidizer systems, fuel cell systems, and life support systems. In order to accomplish long duration bearing operation at cryogenic temperatures, fluorinated ether lubricants were investigated in reference 7. These fluids showed excellent lubrication properties at outer-race temperatures as low as 89 K (160° R). At these operating temperatures, bulk rolling-element temperature increased only slightly because of the increasing ball-spin velocity as contact angle increased from 10° to 40°. However, the local temperature or the temperature in the contact zone may be considerably higher than the bulk ball temperature. Therefore, local temperature gradients may have considerable effect on the fluid viscosity (which may in turn affect the shearing stresses in the bearing material) and on the EHD film thickness.
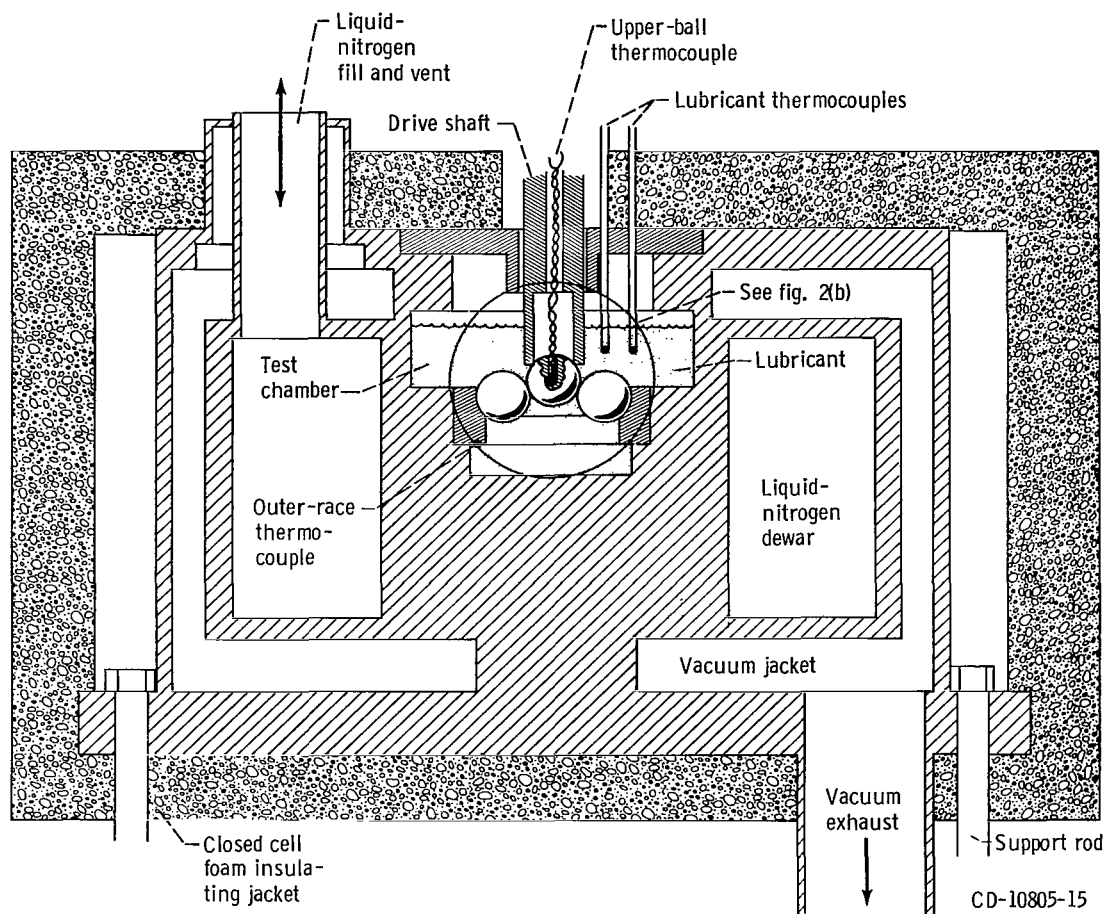
The objectives of the research reported herein were (1) to investigate the effect of contact angle on rolling-element fatigue life at cryogenic temperatures, (2) to compare these results with those of similar tests previously conducted at 328 K (590° R), and (3) to conduct an analysis to determine the effects of EHD film thickness and bearing material hardness on rolling-element fatigue life. In order to accomplish the objectives, rolling-element fatigue tests were run in the NASA five-ball fatigue tester with AISI 52100 12.7 millimeter (1/2-in.) diameter steel balls in a fluorinated ether lubricant. Test conditions include a lubricant temperature of 170 K (305° R), an outer-race temperature of 130 K (235° R) (both controlled for these tests), a shaft speed of 4750 rpm, a maximum Hertz stress of 5500 MN/m^2 (800 000 psi), and contact angles of 20°, 30°, and 40°. These test conditions were chosen so that results could be compared with data of reference 6.
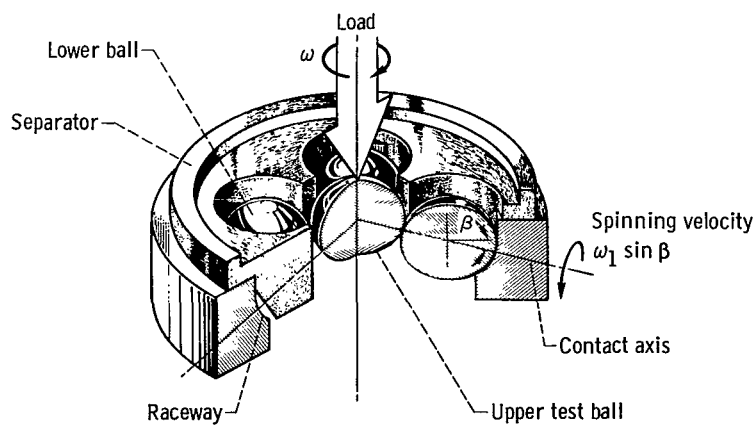
SYMBOLS

E	Young's modulus, N/m^2; psi
e	constant
K_1	constant
L	10-percent rolling-element fatigue life, stress cycles
m	exponent
m'	exponent
N	normal ball load, N; lbf
n	exponent
p	exponent
T	temperature, K; ^{O}R
ΔT	temperature difference between ball running track and remainder of ball, K; ^{O}R
z/b	normalized depth, dimensionless
α	coefficient of thermal expansion, cm/cm/K; in./in./^{O}R
ν	kinematic viscosity, centistokes
σ_c	thermal compressive stress, N/m^2; psi
$\sigma_{x,y,z}$	stress in x, y, and z planes, respectively, N/m^2; psi
τ	shearing stress, N/m^2; psi
τ_c	shearing stress as modified by the thermal compressive stress, N/m^2; psi

APPARATUS

The test apparatus used in these experiments was a modified NASA five-ball fatigue tester (fig. 2(a)). The tester comprises an upper test ball and four lower test balls. The upper ball is analogous to the inner race of an angular-contact bearing. The upper ball is pyramided on the four equally spaced lower test balls that are positioned by a retainer and are free to rotate in an outer race (fig. 2(b)). The upper test ball is driven by, and axially loaded through, a drive shaft. The lower test block, which contains the outer race, is supported by rubber mounts to minimize stresses due to vibratory loads and minor misalinements. The test block includes an annular vacuum-

Liquid-
nitrogen
fill and vent

Upper-ball
thermocouple

Drive shaft

Lubricant thermocouples

See fig. 2(b)

Test
chamber

Lubricant

Outer-race
thermo-
couple

Liquid-
nitrogen
dewar

Vacuum jacket

Vacuum
exhaust

Closed cell
foam insula-
ting jacket

Support rod

CD-10805-15

(a) Simplified cross section of low-temperature NASA five-ball fatigue tester.

Load

ω

Lower ball

Separator

Spinning velocity
$\omega_1 \sin \beta$

β

Contact axis

Raceway

Upper test ball

(b) Five-ball assembly.

CD-6838-15

Figure 2. - Test apparatus.

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jacketed liquid-nitrogen dewar (fig. 2(a)). In this application, the liquid nitrogen acts as an infinite heat sink with a temperature of 78 K (140° R). The five-ball assembly is completely submerged in the fluid. The fluid acts as both a heat-transfer medium and a lubricant. The lower test block, which is constructed of stainless steel, is covered with an insulating jacket of polystyrene foam to insulate it against undue heat leakage from the environment. The sources of heat within the test assembly region are from the heat leak down the drive shaft, heat generation due to the rolling and sliding contacts, and heat generation due to the viscous shearing of the lubricant in the test chamber.

SPECIMENS AND PROCEDURE

The test specimens were 12.7-millimeter (1/2-in.) diameter AISI 52100 steel balls with a nominal Rockwell C hardness of 61 at room temperature. Based on data of reference 7 and the temperature measurement of some randomly chosen test specimens, the upper-ball bulk temperature at a lubricant temperature of 170 K (305° R) was 193 K (345° R). Therefore, from figure 3, the nominal upper-ball hardness at test condi-
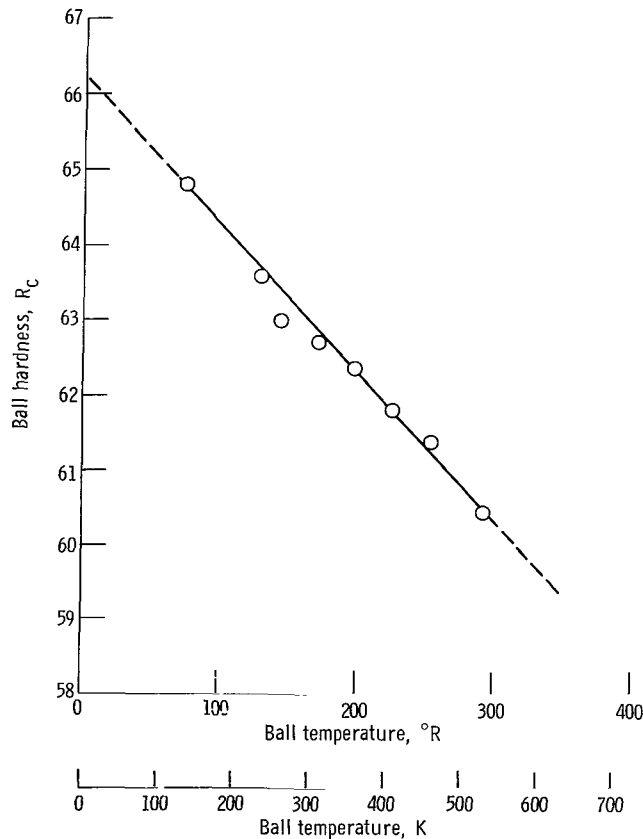
Figure 3. - Ball hardness of AISI 52100 steel as function of temperature (from ref. 13).

tions was Rockwell C 62.5. The balls were thoroughly cleaned by immersion in a 95-percent ethyl alcohol solution. They were removed from the cleaning solution and air dried. The balls were inserted in the test block, and enough lubricant was added to entirely cover the five-ball assembly. An axial load was applied to the five-ball system to produce a maximum Hertz stress of 5500 MN/m^2 (800 000 psi).

Liquid nitrogen was added to the dewar to cool the system to the operating temperature. As the outer-race temperature reached 214 K (385° R), the drive motor was started, and a drive shaft speed of 4750 rpm was maintained. Temperatures were measured and recorded continuously, and the test continued until either a fatigue failure (upper or lower ball) or a suspended test (>150 hr) was obtained.

The lubricant temperature was measured at two different stations in the test cavity approximately 6.4 millimeters (1/4 in.) apart radially (see fig. 2(a)). The upper-ball temperature of some randomly chosen specimens was measured during operation by means of a thermocouple inserted in the center of the ball. The thermocouple electromotive force (emf) was obtained through a slip-ring - brush assembly.

TEST LUBRICANT

The test lubricant used was one of a family of fluorinated ethers having the general formula

$$F(\underset{\underset{CF_3}{|}}{CFCF_2O})_n CHFCF_3$$
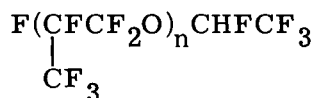
The subscript n represents the degree of polymerization of the polymer so that, at a given temperature, the viscosity of the lubricant increases as the degree of polymerization increases. The value of n for the lubricant used in these tests was 1. Lubricant properties are summarized in table I. The viscosity of the lubricant as a function of temperature is shown in figure 4.

TABLE I. - PROPERTIES OF FLUORINATED ETHER LUBRICANT (REF. 7)

[General formular for family of fluorinated ethers, $F(CFCF_2O)_n CHFCF_3.$]

$$\underset{CF_3}{F(CFCF_2O)_n CHFCF_3.}$$

|	
Degree of polymerization of polymer, n	1
Molecular weight	286.03
Boiling point, K (°R)	312 (562)
Compressibility at 298 K (537° R) and 500 atm, percent	8.20
Heat of vaporization at boiling point, J/kg (Btu/lbm)	96×10^3 (41.4)
Approximate pour point (0.2 m^2/sec or 200 000 cS), K (°R)	118.6 (214)
Density at 298 K (537° R), g/ml (lb/gal)	1.538 (13.2)
Specific heat, C_p, J/(kg)(K) (Btu/(lb)(°R))	1025 (0.254)
Thermal expansion, m^3/(kg)(K) (ft^3/(lb)(°R))	1.07×10^{-6} (10×10^{-6})
Vapor pressure at 325 K (585° R), N/m^2 (psia)	163×10^3 (23.7)
Viscosity at 296 K (537° R), m^2/sec (cs)	0.3×10^{-6} (0.3)

[a]Estimated values.
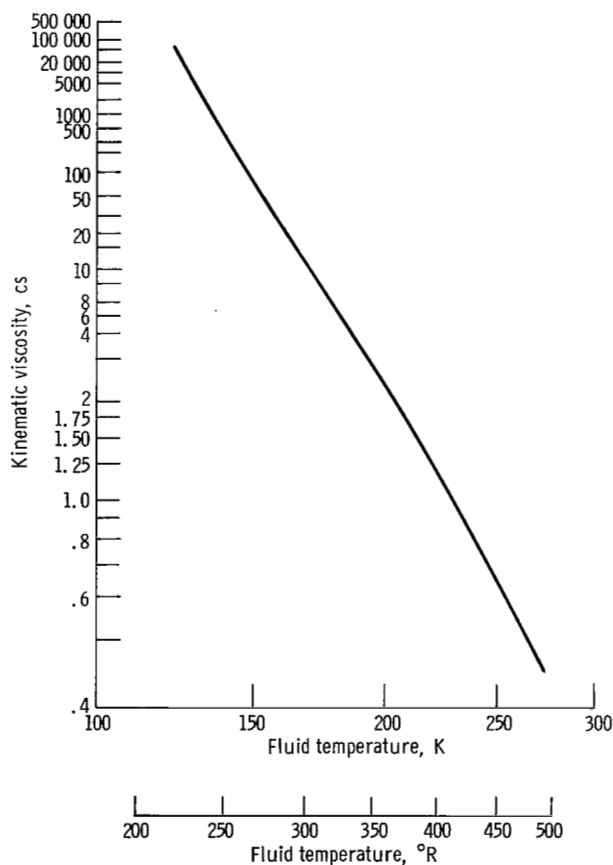
Figure 4. - Kinematic viscosity of fluorinated ether as function of temperature (ref. 7).

RESULTS AND DISCUSSION

Fatigue Results

Rolling-element fatigue tests were run on 12.7-millimeter (1/2-in.) diameter AISI 52100 steel balls in a modified five-ball fatigue tester. The test conditions was as follows: maximum Hertz stress, 5500 MN/m^2 (800 000 psi); shaft speed, 4750 rpm; lubricant, florinated ether; outer-race temperature, 130 K (235° R); lubricant temperature, 170 K (305° R); and contact angles of 20°, 30°, and 40°. The results of the experimental data are shown in figure 5 on Weibull coordinates.

To accurately compare rolling-element fatigue lives of ball specimens at varying contact angles, corrections should be made for the stressed volume (ref. 8). The stressed volume is defined by the width multiplied by length of the running track on the upper-ball specimen contact and by depth under the track to the plane of maximum shearing stress. In the five-ball fatigue tester, as contact angle is increased, the length of the upper-ball running track decreases. The stressed volume is proportional to the track length, which, in turn, is proportional to the cosine of the contact angle.

If S_1 is the probability of survival of a specimen with a track length X_1, then the probability of survival S_2 of a specimen with a track length X_2, tested under the same conditions, is $S_2 = (S_1)^{X_2/X_1}$. For example, the running track of a 20° specimen is 1.22 times as long as the track of a 40° specimen. Referring to figure 5(c), the 40° contact angle data are corrected for the stressed volume at a contact angle of 20° by using the preceding equation at the 10-percent life (90-percent probability of survival). Thus,

$$\left.\begin{aligned}
(S_{40})_{corrected} &= (S_{40})^{1.22} \\[6pt]
(S_{40})_{corrected} &= (0.9)^{1.22} \\[6pt]
(S_{40})_{corrected} &= 0.880
\end{aligned}\right\} \tag{1}$$

Therefore, the 10-percent life for the 40° contact angle represents the $(1 - 0.880) \times 100$ or the 12-percent life when corrected to the 20° track length. The Weibull line of figure 5(c) is, therefore, redrawn as the dashed line, which can now be compared directly with the 20° data. Similarly, the 30° data are corrected to 20° as shown by the dashed line in figure 5(b). The corrected data are summarized in figure 5(d).

(a) Contact angle, 20°.
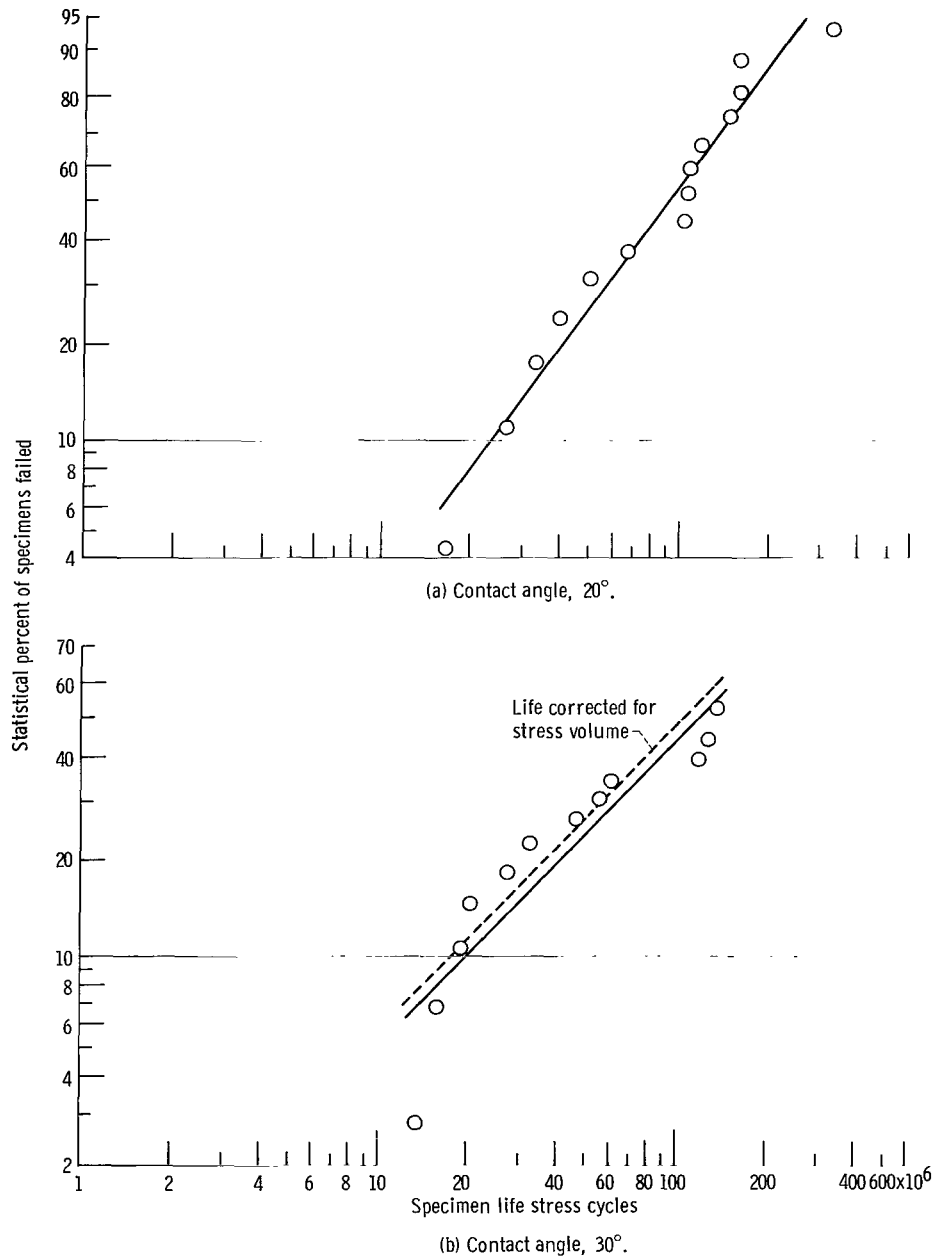
(b) Contact angle, 30°.

Figure 5. – Rolling-element fatigue life of AISI 52100 steel. Speed, 4750 rpm; maximum Hertz stress, 5500 MN/m^2 (800 000 psi); outer-race temperature, 130 K (~225° F).

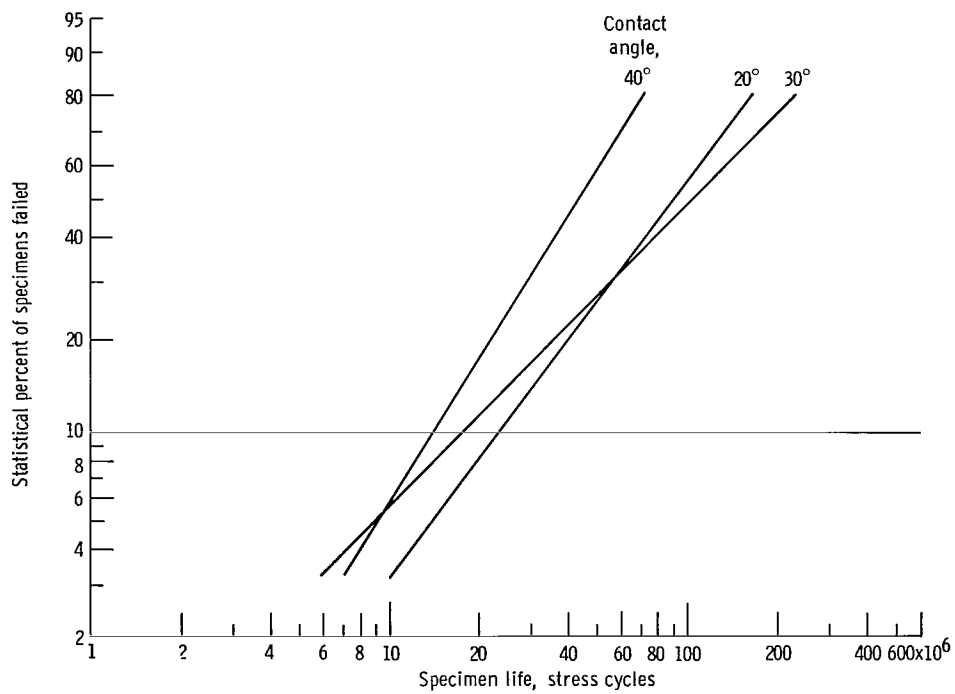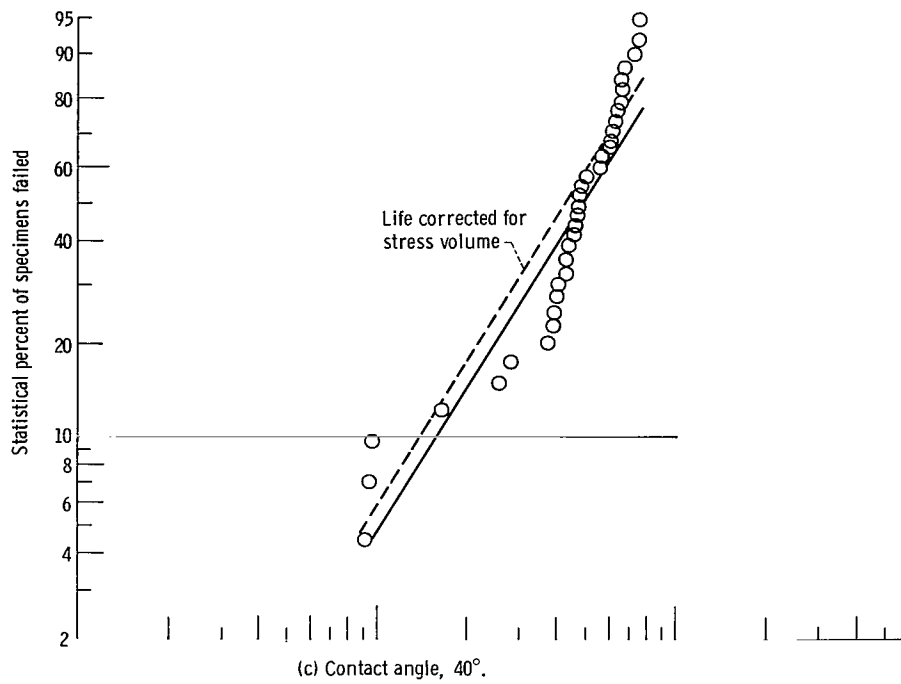(c) Contact angle, 40°.

(d) Summary showing corrected lines.

Figure 5. - Concluded.

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In aircraft and space system design where high reliability is required, early failures are of prime interest. In general, the 10-percent life is used for comparative purposes. In figure 5(a) the 10-percent life for the 20° contact angle is 23.4×10^6 stress cycles. In figure 5(b) the corrected 10-percent life for the 30° contact angle is 17.7×10^6 stress cycles, a decrease of 24-percent. In figure 5(c), the corrected 10-percent life for the 40° contact angle is 14.3×10^6 stress cycles, a decrease of approximately 40-percent of the life for the 20° contact angle. The corrected data for 20°, 30°, and 40° contact angles are shown in figure 5(d).

In comparing several populations of statistical data, we must be able to predict the reliability of the ranking of the populations. Reference 9 provides a method of computing a confidence number. A confidence number of 95-percent means that 95 times out of 100 the population with that number will have the relative rank shown. The confidence numbers for the 40° and 30° contact angle data with respect to the 20° data are 80- and 62-percent, respectively. A 68-percent confidence is approximately equal to a one sigma
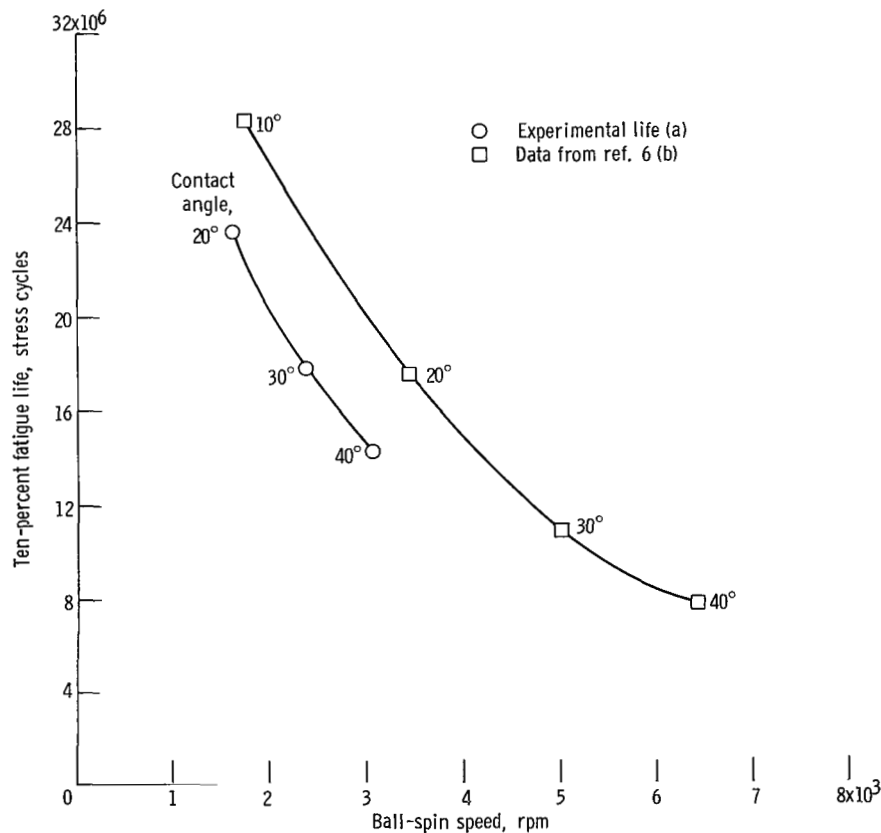

Figure 6. - Ten-percent life as function of ball-spin speed. (a) Material, AISI 52100 steel; lubricant, fluorinated ether; bearing speed 4750 rpm; maximum Hertz stress, 5500 MN/m^2 (800 000 psi). (b) Material, AISI M-1; lubricant, synthetic diester fluid; bearing speed, 10 000 rpm; maximum Hertz stress, 5500 MN/m^2 (800 000 psi).

TABLE II. - COMPARATIVE BEARING TEST CONDITIONS

	Experimental data	Data from ref. 6
Material	AISI 52100	AISI M-1
Lubricant	Fluorinated ether	Synthetic diester
Bearing speed, rpm	4750	10 000
Maximum hertz stress, MN/m^2 (psi)	5500 (800 000)	5500 (800 000)
Temperature (outer race), K (^{O}R)	130 (235)	328 (590)

deviation, which, for statistical purposes, is considered to be too insignificant to allow us to conclude that there is any difference in life between two population groups. Although the data of figure 5 show no statistically significant differences, there is a noticeable trend with contact angle. In figure 6, the test data are compared with those of reference 6 and are plotted as a function of ball-spin velocity. The data compare well and show a decreasing life trend with increasing contact angle. Table II shows the test conditions for the tests of reference 6 and those reported herein. Specimen size, maximum Hertz stress, and contact angles were the same; test material, lubricant composition, speed, and temperature were different. In spite of these differences, figure 6 shows a clearly decreasing life trend with increasing contact angle for both lubricants. More important, however, the slopes for the two sets of data are practically identical.

The exact mechanism that accounts for the difference in life with contact angle is not known. However, as contact angle increases, ball-spin velocity increases; thus, local heating increases in the contact zone. As a result, the reduction in fatigue life may be the result of thermal stresses induced in the surface of the ball, of reduced material hardness, and/or of a reduction in the EHD film thickness. The following analysis will show that the differences can be accounted for by elastohydrodynamic effects and by effects due to changes in the material hardness with temperature.

Thermally Induced Stress Effects

In a five-ball fatigue tester, as the upper test ball rotates, the lower balls spin about an axis normal to the upper-ball - lower-ball contact (see fig. 2) as well as roll. As a result, heat is generated at the upper-ball - lower-ball contact. This heat generation can cause temperature gradients in the upper-test ball and to a lesser extent the lower test balls. It was speculated in reference 6 that these temperature gradients can cause thermal stresses that can reduce life.

If it is assumed that the metal under the contact zone, which is experiencing a thermal compressive stress from an instantaneous rise in the local metal temperature, is in
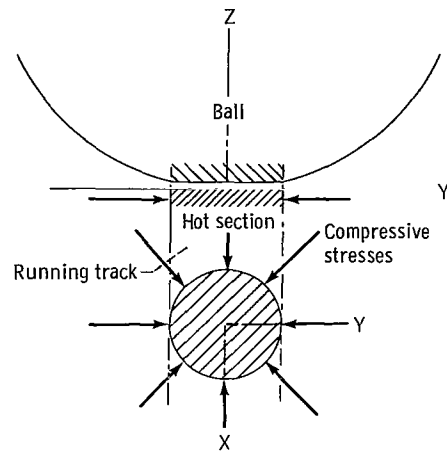
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Figure 7. - Sections in ball and running track subject
to thermal stresses from transient temperature
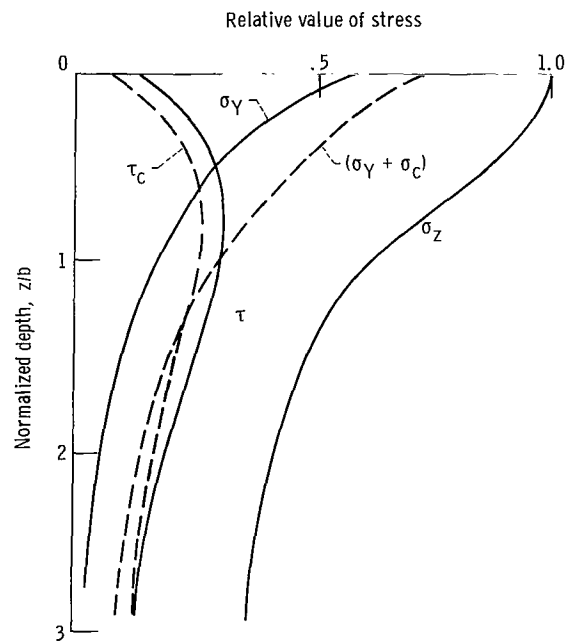increases in contact zone.

Figure 8. - Relative value of compressive and shearing
stress as affected by thermal compressive stress.

a fully restrained condition due to the cooler surrounding metal (as illustrated in fig. 7), then, the thermal compressive stress is given by

$$\sigma_c = -E\alpha \, \Delta T \tag{2}$$

where

E   Young's modulus, N/m^2 (psi)

α coefficient of thermal expansion, cm/cm/K (in./in./ $^{)}R$)

ΔT temperature difference (between running track contact area and rest of ball) K (R^o)

This thermal compressive stress would be additive to the principal stresses as illustrated in figure 8. As a result, the shearing stress on a 45° plane would decrease to τ_c. Because $L \propto (1/\tau)^9$, the compressive thermal stress actually should increase fatigue life rather than decrease it. The experimental results, of course, showed a decrease in life with increased contact angle. Accordingly, other explanations such as elastohydrodynamic and hardness effects must be considered to account for the reduction in life.

Elastohydrodynamic (EHD) and Hardness Effects

The EHD film thickness is affected by lubricant temperature and viscosity, contact pressure (Hertz stress), and the elastic characteristics and velocities of the contacting bodies.

Rolling-element fatigue life can be related to viscosity by the following expression (refs. 10 and 11)

$$L \propto \nu^n \tag{3}$$

where

L   ten-percent fatigue life

ν kinematic viscosity, cs

n   exponent

From figure 4, the viscosity can be related to temperature as follows:

$$\nu \propto T^p \tag{4}$$

where

T   temperature, K (^{O}R)

p   exponent

Combining equations (3) and (4) results in

$$L \propto T^{pn} \quad \text{or} \quad \frac{L_1}{L_2} = \left(\frac{T_1}{T_2}\right)^{pn} \tag{5}$$

From references 10 and 11 $n = 0.25$, and from figure 4 $p \approx -7$ for the test fluid.

Using equation (5), the required lubricant temperature change to cause a reduction in the 10-percent fatigue life as shown in figure 5(d) can be calculated. The life ratios from figure 5 are

$$\left.\begin{aligned} \frac{L_{30}}{L_{20}} &= 0.755 \\\\ \\\\ \frac{L_{40}}{L_{20}} &= 0.605 \end{aligned}\right\} \tag{6}$$

and

From equations (5) and (6)

$$\left.\begin{aligned} T_{30^o} &= \frac{T_{20^o}}{\left(\dfrac{L_{30^o}}{L_{20^o}}\right)^{4/7}} = 199 \text{ K } (358^O \text{ R}) \\\\ \\\\ T_{40^o} &= \frac{T_{20^o}}{\left(\dfrac{L_{40^o}}{L_{20^o}}\right)^{4/7}} = 266 \text{ K } (407^O \text{ R}) \end{aligned}\right\} \tag{7}$$

and

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where $T_{20^O} = 170$ K (305° R) is the measured temperature at the 20° contact angle and is assumed to be, for purposes of calculation, the temperature of the lubricant at the entrance to the contact zone.

From equation (7) the temperature increases required to reduce the 10-percent lives (fig. 5(d)) at the 30° and 40° contact angle tests, based on viscosity effects, were 29 K (53 R^O) and 57 K (102 R^O), respectively.

From figure 3, for AISI 52100 steel, hardness decreases approximately one (1) point Rockwell C per 55 K (100° R). Therefore, there arises the probability that some hardness effect on fatigue life might occur. From reference 13,

$$\frac{L_2}{L_1} = e^{m\left(R_{c_2} - R_{c_1}\right)} \tag{8}$$

where

m   exponent

R_{c_1}, R_{c_2} Rockwell hardnesses

For AISI 52100 steel $m = 0.1$ (ref. 13). Also, from figure 3, the hardness of AISI 52100 steel as a function of temperature can be expressed as follows:

$$R_c = 66.2 - 0.0108\ T \tag{9}$$

Substituting equation (9) into (8)

$$\frac{L_2}{L_1} = e^{m'(T_1 - T_2)} \tag{10}$$

where

$$m' = 0.00108$$

Combining equations (5) and (10) will produce a relation that accounts for the effects of temperature on lubricant viscosity and material hardness for the fluorinated ether and the AISI 52100, respectively, where

$$\frac{L_2}{L_1} = \left(\frac{T_1}{T_2}\right)^{7/4} e^{0.00108(T_1 - T_2)} \tag{11}$$

Using an iteration process with equation (11), the increased temperature required to reduce the experimental life can be determined for the 30° contact angle as follows:

$$\frac{L_{30^O}}{L_{20^O}} = 0.755 = \left(\frac{T_{20^O}}{T_{30^O}}\right)^{7/4} e^{0.00108(T_{20^O} - T_{30^O})} \tag{12}$$

where

$$T_{20^O} = 170 \text{ K } (305^O \text{ R})$$

From equation (13)

$$T_{30^O} = 193 \text{ K } (347^O \text{ R})$$

Similarly, for the 40° contact angle

$$T_{40^O} = 215 \text{ K } (388^O \text{ R})$$

The temperature difference required to reduce the 10-percent fatigue life as presented in figure 5(d) from both viscosity and hardness effects is 23 K (42 R^O) for the 30° contact angle and 46 K (83 R^O) for the 40° contact angle. From equation (11) the effect on the 10-percent fatigue life of hardness and viscosity can be separated. When this is done it is found that the life is reduced by hardness changes approximately 5 percent and by viscosity changes approximately 20-percent at the 30° contact angle and 8 and 32 percent, respectively, at the 40° contact angle. These data are summarized in table III.

In reference 6, temperature measurements were made at the edge of the running track at several contact angles in a test apparatus similar to the one used in the tests reported herein. In reference 6 the temperature difference between the 20° and 30° contact angles was 11 K (19 R^O) and between the 20° and 40° contact angles, 21 K (37 R^O). It would be expected that the temperature in the track itself would be somewhat higher than at the edge of the track. From these data and considering the high contact stress (550 MN/m^2, 800 000 psi) and the increased spin velocity at the higher contact angle, the calculated temperature changes appear to be reasonable.

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TABLE III. - REDUCED LIFE AT DIFFERENT CONTACT ANGLES FROM INCREASED

TEMPERATURE AND ITS EFFECT ON VISCOSITY AND HARDNESS

Life ratio	Experimental percent of 20° contact angle life	Temperature change required to cause total life reduction from viscosity effects only		Combined effects of viscosity and hardness			
				Temperature change required		Percent life reduction	
		K	°R	K	°R	Viscosity effects	Hardness effects
$L_{30}L_{20}$	75.5	29	53	23	42	20	5
L_{40}/L_{20}	60.5	57	102	46	83	32	8

SUMMARY OF RESULTS

Rolling-element fatigue tests were run with AISI 52100 12.7-millimeter (1/2-in.) diameter steel balls in a fluorinated ether lubricant. Test conditions were a lubricant temperature of 170 K (305° R), an outer-race temperature of 130 K (235° R), a shaft speed of 4750 rpm, a maximum Hertz stress of 5500 MN/m^2 (800 000 psi), and contact angles of 20°, 30°, and 40°. An analysis was conducted to determine the effects of elastohydrodynamic and hardness effects on fatigue life. Results were compared with data obtained at 328 K (590° R) with a diester lubricant. The following results were obtained:

1. There were no statistically significant differences among the 10-percent fatigue lives at contact angles of 20°, 30°, and 40°. However, there was a trend of decreasing life with increasing contact angle.

2. The trend of decreasing life with increasing contact angle at 170 K (305° R) with fluorinated ether lubricants is similar to the trend reported with a diester fluids at 328 K (590° R).

3. Analysis indicates that the decrease in 10-percent fatigue life with increasing contact angle can be estimated from thermal effects on lubricant viscosity and rolling-element material hardness.

4. The temperature changes calculated for varying contact angles at 170 K (305° R) were of the same order of magnitude as the temperature differences measured in tests conducted with a diester at 328 K (590° R) at the same contact angles.

Lewis Research Center,
    National Aeronautics and Space Administration,
        Cleveland, Ohio, March 19, 1971,
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