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REDUCED-PRESSURE ENVIRONMENT EFFECTS ON ROLLING-ELEMENT FATIGUE LIFE WITH SUPER-REFINED MINERAL OIL LUBRICANT

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SUMMARY

A modified NASA five-ball fatigue tester was used to investigate the effect of a reduced-pressure environment on rolling-element fatigue life, deformation, and wear of SAE 52100 steel balls with a super-refined naphthenic mineral oil used as the lubricant. Tests were conducted at atmospheric pressure, at a 20° contact angle, with a thrust load of 60 pounds (267 N) to produce an initial maximum Hertz stress of 800 000 psi (5.5×10^{9} N/m^{2}) at a shaft speed of 4900 rpm with no heat added. The atmospheric-pressure tests served as reference data for tests conducted at a reduced pressure, approximately the vapor pressure of the lubricant. For these reduced-pressure tests, the ball specimens were lubricated by a quasi-mist method and by immersion in the lubricant. In both cases, all other test conditions were the same as those in the atmospheric-pressure tests.

No significant difference in the fatigue lives was observed for tests conducted at atmospheric pressure and those conducted at reduced ambient pressure. The amounts of deformation and wear for the two pressure conditions differed little, regardless of the lubrication mode employed, which was an indication of a sufficient elastohydrodynamic lubricating film at the reduced-pressure levels.

INTRODUCTION

High reliability is required in aerospace applications where rotating machinery must function unattended for periods of up to 10 000 hours. Among the rotating machinery components are rolling-element bearings. Bearings will be used in space turboelectric power generating systems necessary for long-term-life support systems and electric propulsion systems (ref. 1). When all factors, such as design, lubrication, and operating conditions, for a specific application have been optimized, fatigue failure will be the life-limiting factor (refs. 1 and 2). In these aerospace applications, rolling-element bearings will be used in semisealed systems having an environmental pressure less than atmospheric pressure, near that of the lubricant vapor pressure.

A problem that might be encountered in a reduced-pressure environment is the removal of surface oxide films by wear more rapidly than they can be reformed (ref. 1). For example, it was reported (refs. 3 and 4) that, for a polyphenyl ether lubricant, the wear rate under predominantly boundary lubrication was much greater at reduced pressure then at atmospheric pressure. However, where elastohydrodynamic conditions exist and complete separation of the rolling-element surfaces is effected, surface oxides would be of small importance in inhibiting wear for long-term operation.

A second problem that might be encountered in the reduced-pressure environment is the evaporation of the lubricating fluid, which limits the useful time the bearing can be operated. With properly designed static and dynamic seals, however, lubricant evaporation can be reduced to a negligible amount for extended operation (refs. 5 and 6).

Rolling-element fatigue life is affected by lubricant bulk viscosity. A 50-percent decrease in the apparent bulk viscosity of a super-refined mineral oil and in an ester-base oil was observed when the lubricants were saturated with nitrogen gas (ref. 7). Since most lubricants contain air or gases at atmospheric pressure, lubricant viscosity may be increased by degassing. In general, as lubricant viscosity increases, so does the fatigue life of a rolling-element bearing (ref. 8).

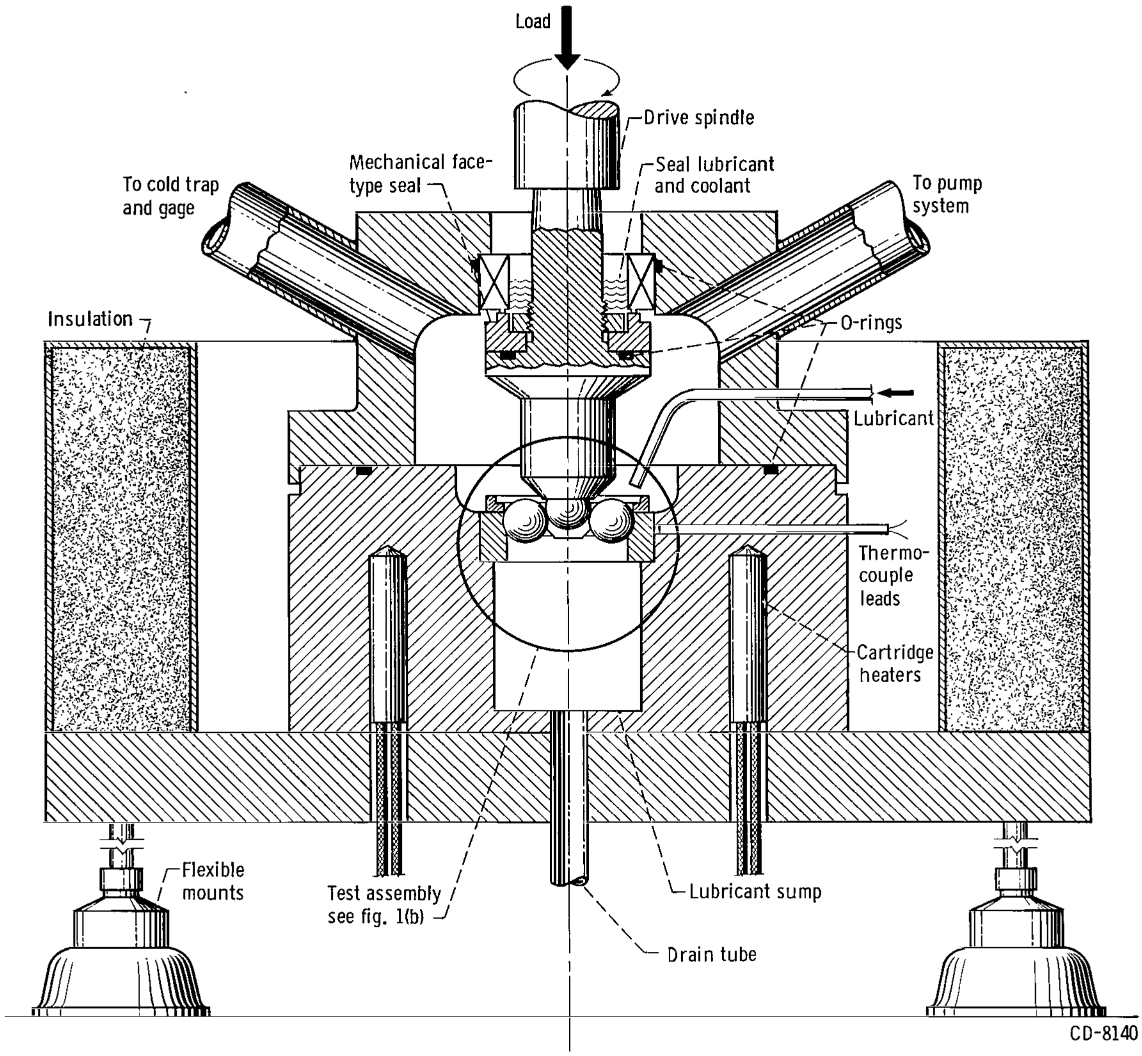
The present investigation was conducted to determine the effect of a reduced-pressure environment on rolling-element operation with a mineral oil lubricant. The objectives were to determine experimentally

(1) If a relation exists between ambient pressure and rolling-element fatigue life
(2) If, under reduced-pressure conditions, rolling-element fatigue life is different with quasi-mist (partial flashing of the lubricant on exposure to the reduced pressure) lubrication and total immersion of the rolling elements in the lubricant
(3) If differences exist in the deformation and wear of rolling-elements run under normal atmospheric conditions and in a reduced-pressure environment

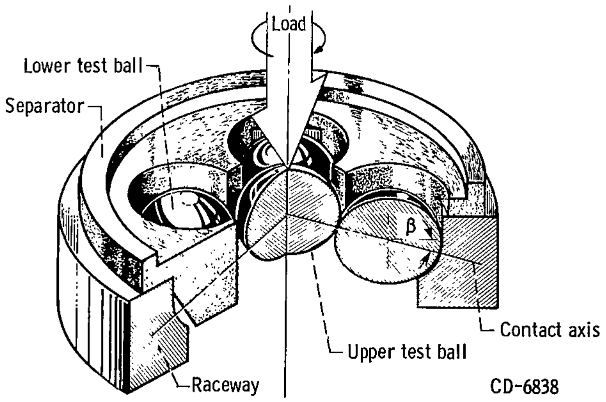
Tests were conducted with SAE 52100 steel 1/2-inch- (1.27-cm-) diameter balls of nominal Rockwell-C hardness 62.5 at atmospheric pressure and at the approximate vapor pressure of the lubricant (less than 10^{-5} torr or 1.33×10^{-3} N/m^2). Test conditions were a temperature of 130° F (327° K); a 20° contact angle; a thrust load of 60 pounds (267 N), which produced a maximum Hertz stress of 800 000 psi (5.5×10^9 N/m^2); and a speed of 4900 rpm with no heat added. All experimental results were obtained with ball specimens from the same heat of material and lubricant from the same lubricant batch.

APPARATUS

The NASA five-ball fatigue tester, modified for reduced-pressure testing, was used for this investigation. The modified five-ball apparatus is described in reference 3 and is shown in figure 1(a). Briefly, it comprises an upper test ball pyramided on four lower



(a) Section view showing modifications for lubrication tests at reduced pressures.



(b) Test assembly.

Figure 1. - Five-ball fatigue tester.

test balls which are positioned by a separator and are free to rotate in an angular-contact raceway (fig. 1(b)). System loading and drive are supplied through a vertical drive shaft. Failure detection and automatic shutdown instrumentation permitted long-term unmonitored tests.

The apparatus modification comprises access tubes to the test chamber for a vacuum pumping system and its accompanying pressure monitoring instrumentation, and a mechanical shaft seal of the spring-loaded, carbon-face type. The seal was lubricated and cooled with the test lubricant. Its performance at 4900 rpm was such that no leaks could be detected with a helium leak detector.

The vacuum pumping system was a commercial system capable of attaining pressures of 10^{-6} torr (1.3×10^{-4} N/m^2). It comprises a mechanical forepump and a 2-inch (5.08-cm) oil diffusion pump with a liquid-nitrogen cold trap. An automatic liquid-nitrogen-dispensing system was provided to make possible the necessary long-term tests. A schematic diagram of the vacuum system is shown in figure 2.

Test lubricant was supplied to the test chamber from a reservoir that had a cover gas of nitrogen under a slight pressure. The lubricant flow rate to the test assembly was controlled by a valve.

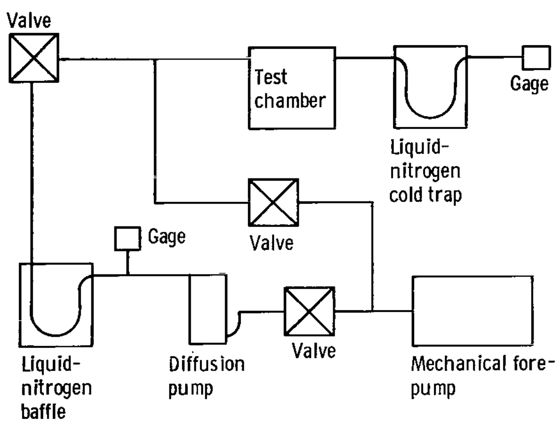


Figure 2. - Schematic diagram of vacuum system.

PROCEDURE

The upper and lower test balls in the NASA five-ball fatigue tester were 1/2-inch-(1.27-cm-) diameter SAE 52100 steel balls of nominal Rockwell-C hardness 62.5, all from the same heat of material. The lower balls were positioned in a raceway having a 20° contact angle. A thrust load of 60 pounds (267 N) was applied which produced a maximum Hertz stress of 800 000 psi (5.5×10^{9} N/m^2) in the upper-lower ball contact. A

TABLE I. - PHYSICAL PROPERTIES OF

TEST LUBRICANT, SUPER-REFINED

NAPHTHENIC MINERAL OIL

[Manufacturer's data.]

Density, g/cu cm, at -	
0^{o} F (255^{o} K)	0.908
100^{o} F (311^{o} K)	.873
200^{o} F (366^{o} K)	.838
300^{o} F (422^{o} K)	.802
400^{o} F (477^{o} K)	.768
500^{o} F (533^{o} K)	.732
Vapor pressure (extrapolated), mm of Hg (N/m^2), at -	
125^{o} F (325^{o} K)	$<10^{-5}$ (1.3×10^{-3})
300^{o} F (422^{o} K)	.07 (9.3)
400^{o} F (477^{o} K)	2.0 (2.6×10^{2})
500^{o} F (533^{o} K)	17.0 (22.7×10^{2})
Viscosity, cs, at -	
0^{o} F (255^{o} K)	10 000
30^{o} F (271^{o} K)	1 500
100^{o} F (311^{o} K)	79
210^{o} F (372^{o} K)	8.4
500^{o} F (533^{o} K)	1.1
700^{o} F (644^{o} K)	.6
Cleveland open cup flash point, ^{o}F (^{o}K)	445 (502)
Cleveland open cup fire point, ^{o}F (^{o}K)	495 (530)
ASTM pour point, ^{o}F (^{o}K)	-30 (239)

super-refined naphthenic mineral oil was used as the test lubricant. Table I gives the properties of the lubricant.

Prior to each test, the ball specimens and test block were cleaned with a solvent, flushed with ethyl alcohol, and wiped with clean cheesecloth. The ball specimens were placed in the angular-contact raceway, subjected to the load, and run at a shaft speed of 4900 rpm until failure or for a period of 250 hours, whichever occurred first. (For each revolution of the shaft, the upper ball specimens received three stress cycles.) The total running time of the tests was recorded.

At the completion of the tests under the atmospheric-pressure conditions, the tests were repeated at reduced pressure under the same test conditions. At the reduced pressure, an adjustment of the applied load was necessary in order to compensate for the added forces from the pressure differential across the seal. (The Hertzian stress in the test system was the same as that in the atmospheric tests.)

For the reduced-pressure testing, the lubricant was degassed. The degassing procedure comprised placing the lubricating fluid in a flask that was connected to a mechanical vacuum pump through a liquid-nitrogen cold trap. External heat was applied to the flask, which was agitated occasionally during the pumping process. Initially, the lubricant foamed violently when agitated. After a period of approximately 2 hours, only a few small gas bubbles remained from agitation, at which time the lubricant degassing process was considered completed. The line to the vacuum pump was closed off, and the flask was connected to the test chamber through a needle valve and pressurized with nitrogen gas (see the section, APPARATUS). The lubricant was thus sealed from the air atmosphere.

Prior to starting a test, the test chamber was pumped down to approximately 10^{-6} torr (1.3×10^{-4} N/m^2), and the lubricant was allowed to enter the chamber through the needle valve. Because of the reduced pressure, the lubricant "flashed" into a quasi-mist maintaining the pressure in the test chamber at approximately the vapor pressure of the lubricant. The lubricant flow rate was the same as that of the atmospheric tests, which provided sufficient elastrohydrodynamic lubrication.

A second series of reduced-pressure environment tests was conducted under an immersed lubricant condition. In this series of tests, all conditions and procedures were the same as those previously described with the exception that the lubricant drainage was modified. This modification resulted in a lubricant buildup such that the test specimens were immersed in the lubricant. The pressure in the test chamber was approximately that of the lubricant vapor pressure.

RESULTS AND DISCUSSION

Rolling-element fatigue tests were conducted in a five-ball tester modified to permit the investigation of the effect of a reduced-pressure.

Tests were conducted with SAE 52100 steel 1/2-inch- (1.27-cm-) diameter balls of nominal Rockwell-C hardness 62.5 at atmospheric pressure and at the approximate vapor pressure of the lubricant (less than 10^{-5} torr or 1.3×10^{-3} N/m^2). Test conditions were a temperature of 130° F (327° K); a 20° contact angle; a thrust load of 60 pounds (267 N), which produced a maximum Hertz stress of 800 000 psi (5.5×10^9 N/m^2); and a speed of 4900 rpm with no heat added. All experimental results were obtained with ball specimens from the same heat of material and lubricant from the same lubricant batch.

Effect of Reduced Pressure on Lubricant and Fatigue Life

The results of the fatigue tests are presented in figure 3. The statistical methods outlined in reference 9 were used to analyze the fatigue data. A summary of these fatigue data is presented in table II.

Research reported in reference 7 indicated that, for a super-refined mineral oil of the type used in the tests reported herein, an approximate 50-percent decrease in viscosity can occur when the fluid is saturated with nitrogen gas. Thus, a two-to-one increase in apparent bulk viscosity can occur when the lubricant is degassed. For the fatigue tests at atmospheric pressure, the lubricant was introduced into an air atmosphere. In addition, the lubricant was used as received from the manufacturer. Experience has shown that a fluid in this condition can be highly saturated with air. Thus, for the reduced-pressure tests, the fluid was degassed prior to usage. In addition, any entrapped gases remaining in the lubricant would have a tendency to be eliminated when the lubricant enters the reduced pressure of the test chamber. It was therefore assumed that the fluid in the reduced-pressure tests had an apparent viscosity twice that in the atmospheric-pressure tests. Investigators have shown that, as the viscosity at atmospheric pressure of a mineral-oil lubricant is increased, the fatigue life of a rolling-element also increases (refs. 1, 8, and 10). The accepted relation between fatigue life L and lubricant viscosity μ is $L = K\mu^n$, where K is a constant and n equals 0.2 to 0.3 (refs. 8 and 10). Therefore, if n is taken as 0.25, the fatigue life would be expected to increase under the reduced-pressure condition by approximately 19 percent based on a two-to-one increase in viscosity. The 10-percent life is that life in which 90 percent of the specimens tested will survive. For comparative purposes, the 10-percent life is generally chosen throughout the bearing industry. From figure 3(d) and table II, the 10-percent fatigue life with quasi-mist lubrication at reduced pressure indicated an increase in fatigue life of approximately 50 percent over that obtained with the atmospheric-pressure tests. However, the immersed-lubrication reduced-pressure tests indicate a reduction in fatigue life by approximately 20 percent. Therefore, in order to come to any conclusions regarding these results, the data must be analyzed on a statistical basis.

The confidence that can be placed in the experimental results was determined statistically using the methods given in reference 9. Each of the reduced-pressure results was compared with the results of the atmospheric tests, and confidence numbers for the 10-percent life were calculated and are given in table II. These confidence numbers indicate the percentage of the time that the 10-percent life, obtained with each series of tests at the reduced-pressure condition, will have the same relation to the 10-percent life obtained with the atmospheric tests. Thus, a confidence number of 60 percent means that 60 out of 100 times the specimens tested at the quasi-mist reduced-pressure condition will be ranked as in table II. A 68-percent confidence is approximately equal to a

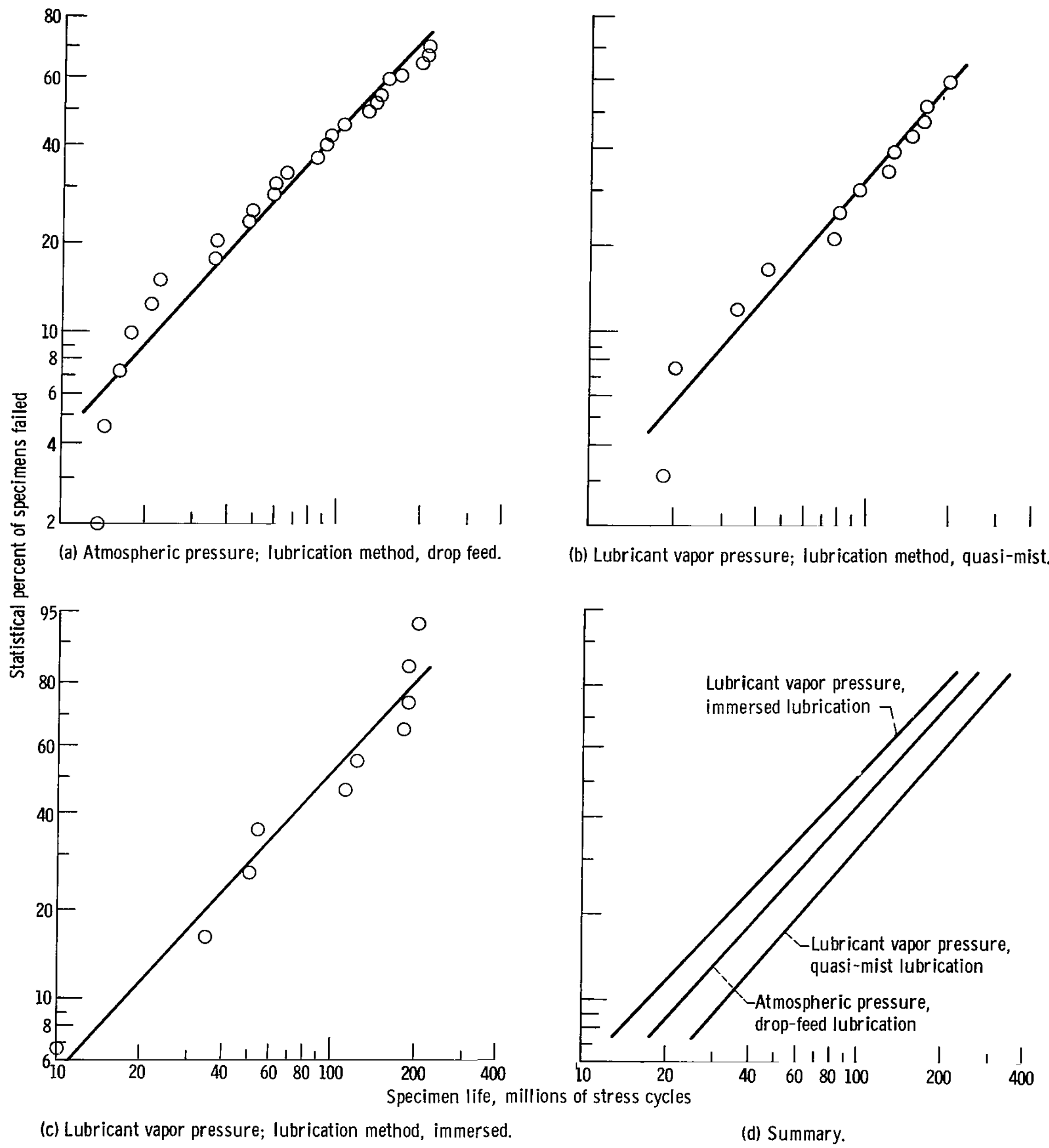


Figure 3. - Rolling-element fatigue life at atmospheric and reduced pressures in five-ball fatigue tester. Shaft speed, 4900 rpm; contact angle, 20°; race temperature, 130° F (327° K); maximum Hertz stress, 800 000 psi (5.5×10^{9} N/m^2); lubricant, superrefined naphthenic mineral oil.

TABLE II. - RESULTS OF FATIGUE TESTS CONDUCTED IN NASA

FIVE-BALL FATIGUE TESTER

[Maximum Hertz stress, 800 000 psi (5.5×10^{9} N/m^{2}); lubricant, super-refined naphthenic mineral oil; contact angle, 20°; ball material, SAE 52100 steel; shaft speed, 4900 rpm; race temperature, 130° F (327° K).]

Pressure environment	Lubrication	Fatigue life, millions of stress cycles		Weibull slope	Confidence number at 10-percent life level, percent (a)	Failure index (b)
		10-percent life	50-percent life			
Atmospheric	Drop feed	23	122	1.1	--	25 out of 38
Reduced[c]	Quasi-mist[d]	34	168	1.2	60	13 out of 22
Reduced[c]	Immersed	18	98	1.1	58	10 out of 10

[a]Percentage of time that 10-percent life obtained at reduced-pressure condition will have the same relation to the 10-percent life obtained at atmospheric-pressure condition.

[b]Number of fatigue failures out of total number tested.

[c]Lubricant vapor pressure, less than 10^{-5} torr (1.3×10^{-3} N/m^{2}).

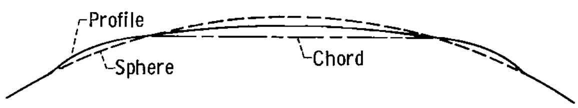
[d]Lubricant was drop fed; upon entrance to the test chamber, lubricant flashed into a quasi-mist because of reduced pressure.

one sigma deviation, which, for statistical purposes, is considered insignificant to conclude that there is any difference in early life between the reduced-pressure tests and those conducted under atmospheric conditions.

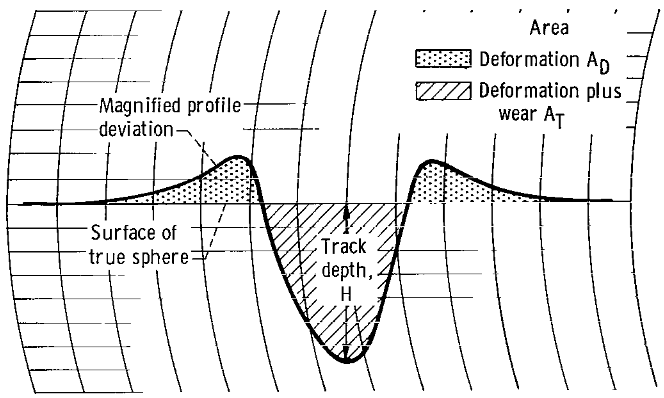
While no statistical significance can be related to the differences in fatigue life between the quasi-mist reduced-pressure tests and those conducted under atmospheric conditions, the immersed-lubrication reduced-pressure tests had a life approximately 50 percent less than those run under reduced pressure with the quasi-mist lubrication. The calculated confidence number between these two reduced-pressure tests is approximately 70 percent at the 10-percent life level. Again, no significance can be attributed to these life differences. Therefore, it can be concluded that, while there may be some effects of a reduced-pressure environment and method of lubrication on fatigue life, these effects are statistically insignificant where early failures are of importance.

Deformation and Wear

Transverse profile traces of the test specimen running tracks were made with a surface profile tracer to determine whether there was an effect of the reduced pressure or the mode of lubrication on the deformation and wear of the rolling elements. Schematic diagrams of a transverse section of a ball running track and a surface trace are shown in figure 4. The crosshatched area A_T in figure 4(b) represents deformation and wear. The shaded area A_D represents the plastic deformation. The difference in the two areas $(A_T - A_D)$ is the wear. The depth of the running track is represented by H in figure 4(b).



(a) Schematic diagram of transverse section of ball surface.

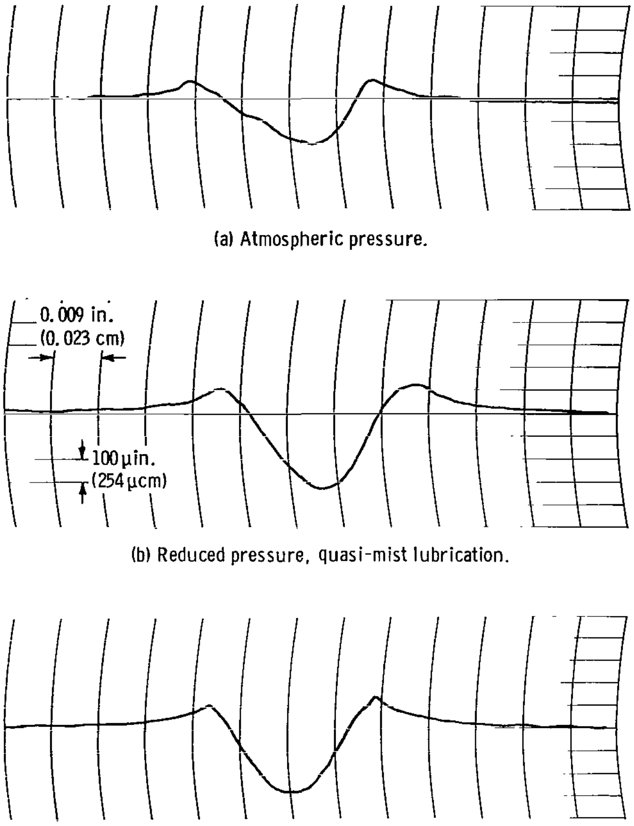


(b) Transverse profile trace of ball running track at high magnification.

Figure 4. - Transverse section of ball running track.

Profile traces were made at six randomly chosen locations across the running tracks of each of 10 upper test balls selected from each of the three test series. These upper test balls were selected to encompass the full range of running times. Typical profile traces for upper test specimens for each of the test series are shown in figure 5. Experience has shown that a lack of deformation area A_D is indicative of gross wear and, thus, a lack of elastohydrodynamic lubrication. Conversely, the presence of the deformation area A_D indicates minimal wear and the presence of an elastohydrodynamic film.

The areas of deformation A_D and deformation plus wear A_T were measured and averaged for each test series. The areas of deformation A_D and of wear A_T-A_D are



(a) Atmospheric pressure.

(b) Reduced pressure, quasi-mist lubrication.

(c) Reduced pressure, immersed lubrication.

Figure 5. - Typical profile traces of upper-ball track for various ambient pressures and lubrication modes. Shaft speed, 4900 rpm; contact angle, 20°; race temperature, 130° F (327° K); maximum Hertz stress, 800 000 psi (5.5×10^{9} N/m^2); lubricant, super-refined naphthenic mineral oil.

TABLE III. - EFFECT OF DEFORMATION AND WEAR ON MAXIMUM HERTZ STRESS FOR SAE 52100 STEEL BALLS

[Initial maximum Hertz stress, 800 000 psi (5.5×10^{9} N/m^2); lubricant, super-refined naphthenic mineral oil; contact angle, 20°; shaft speed, 4900 rpm; race temperature, 130° F (327° K).]

Lubrication method	Pressure environment	Deformation area from surface trace, A_D		Wear area from surface trace, $A_D - A_T$		Calculated profile radius, r		Effective maximum Hertz stress (a)		Calculated theoretical-life ratio (b)	Experimental-life ratio (b)
		in.^{2}	cm^2	in.^{2}	cm^2	in.	cm	psi	N/m^2		
Drop feed	Atmospheric	1.2×10^{-6}	7.7×10^{-6}	1.2×10^{-6}	7.7×10^{-6}	0.38	0.97	7.71×10^{5}	53×10^{8}	1.0	1.0
Quasi-mist	Reduced[c]	2.4	15.5	1.5	9.7	.65	1.65	7.25	50	1.7	1.5
Immersed	Reduced[c]	1.6	10.3	1.4	9.0	.57	1.45	7.32	51	1.6	.8

[a]No deformation of lower test balls assumed.

[b]Ratios are relative to lives obtained for atmospheric-pressure tests.

[c]Lubricant vapor pressure, less than 10^{-5} torr (1.3×10^{-3} N/m^2).

summarized in table III. As can be seen, there is very little difference in wear among the test series, which was not unexpected. When an elastohydrodynamic film is present, most of the wear occurs at startup and shutdown. However, there appears to be a greater amount of deformation at the reduced-pressure conditions. This permanent alteration of the geometry (plastic deformation and wear) affects the calculated theoretical maximum Hertz stress. An effective maximum Hertz stress can be calculated using a deformed transverse-profile radius. This deformed profile radius (ref. 11) can be expressed as

$$r = \frac{\left(\frac{A_T}{H}\right)^2}{2\left[R - \sqrt{R^2 - (A_T/H)^2} - H\right]}$$

where

A_T deformation plus wear area scaled from surface trace, in.^{2}; cm^2

H depth of running track scaled from surface trace, in.; cm

R radius of ball, in.; cm

r effective radius of ball profile after deformation plus wear, in.; cm

The deformed profile radii and the effective maximum Hertz stresses are given in table III. The recalculated stresses show reductions from the initial maximum Hertz stress of 800 000 psi (5.5×10^9 N/m^2) ranging from about 4 to about 9 percent for the atmospheric-pressure tests and the reduced-pressure quasi-mist tests, respectively. The effect of these stress differences on fatigue life can be calculated using the commonly accepted stress-life relation $L \propto \left(\frac{1}{S}\right)^9$. The theoretical-life ratios calculated with this relation are given in table III. The experimental-life ratios are also shown in table III. Because the theoretical- and experimental-life differences are so small, no real significance can be placed on the deformed profile radii beyond that which has been discussed.

Based on the results presented herein, there is no significant effect of operation at a reduced-pressure environment on fatigue life, material deformation, rolling-element wear, and, hence, on elastohydrodynamic lubrication. Therefore, for the lubricant-material combination studied, it may be concluded that the criteria governing the load capacity of rolling-element bearings at atmospheric-pressure environments may be successfully employed for space applications that utilize a semisealed system where the ambient pressure is the vapor pressure of the lubricating fluid.

SUMMARY OF RESULTS

A modified NASA five-ball fatigue tester was used to investigate the effect of a reduced-pressure environment on rolling-element operation. Tests were run with SAE 52100 steel balls at atmospheric pressure; at a 20° contact angle; with a thrust load of 60 pounds (267 N), which produced a maximum Hertz stress of 800 000 psi (5.5×10^{9} N/m^{2}); and at a speed of 4900 rpm with no heat added. Two additional series of tests were run under the same test conditions with the exception that the ambient pressure was approximately that of the vapor pressure of the lubricant. The first series of reduced-pressure tests was conducted with the lubricant introduced into the test chamber in a quasi-mist form, and the second series was conducted with the ball specimens immersed in the lubricant. The following results were obtained:

1. The difference in fatigue life between tests conducted at atmospheric pressure and those conducted at reduced-pressure is statistically insignificant, regardless of the lubrication mode employed. Therefore, for the lubricant-material combination used in these tests, the load capacity employed for rolling-element bearings used for atmospheric-pressure operation may be employed successfully for rolling-element bearings used in a semisealed system when the ambient pressure is that of the lubricant vapor pressure.

2. The amount of wear was essentially the same for tests conducted at atmospheric-pressure conditions and for those conducted at reduced-pressure conditions. A greater amount of plastic deformation occurred at the reduced-pressure condition. However, no real significance can be placed in these differences.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, October 6, 1967,
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