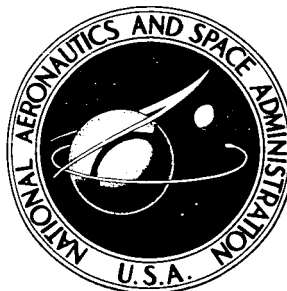


NASA TECHNICAL NOTE



NASA TN D-3948

NASA TN D-3948

N67-26543

FACILITY FORM 602

(ACCESSION NUMBER)	(THRU)
22	7
(PAGES)	(CODE)
	22
(NASA CR OR TMX OR AD NUMBER)	(CATEGORY)

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GPO PRICE \$ \_\_\_\_\_

CFSTI PRICE(S) \$ 3.00

Hard copy (HC) \_\_\_\_\_

Microfiche (MF) 65

ff 653 July 65

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NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

# BEARING TORQUE AND FATIGUE LIFE STUDIES WITH SEVERAL LUBRICANTS FOR USE IN THE RANGE 500<sup>0</sup> TO 700<sup>0</sup> F

by Richard J. Parker, Eric N. Bamberger\*, and Erwin V. Zaretsky

Lewis Research Center

## SUMMARY

The NASA high-temperature bearing torque apparatus, with 204 size angular-contact ball bearings made from AISI M-10 steel was used to determine bearing operating characteristics with seven lubricants to 900<sup>0</sup> F bearing outer-race temperature. The General Electric rolling-contact fatigue tester was used to determine the fatigue lives obtainable with three of these lubricants at a test bar temperature of 600<sup>0</sup> F. Classes of lubricants investigated included a polyester, a super-refined mineral oil, a synthetic paraffinic oil, a polyphenyl ether, and a fluorocarbon. The synthetic paraffinic oil and the polyphenyl ether were evaluated with and without additives.

Bearing torque decreased with increasing outer-race temperature up to 700<sup>0</sup> F for all seven lubricants. Increasing torque was observed with increased lubricant viscosity. Of four lubricants tested between 700<sup>0</sup> and 900<sup>0</sup> F outer-race temperature and lubricant-mist temperatures of 450<sup>0</sup> to 620<sup>0</sup> F, only the synthetic paraffinic oil with the antiwear additive showed no abrupt torque increase as the outer-race temperature was increased. Furthermore, increasing the lubricant-mist temperature had no effect on bearing torque. These results indicated that, in these tests, the lubricant mist had little cooling effect on the bearing. Apparent elastohydrodynamic lubrication, based on running track appearance, was indicated with all seven lubricants to outer-race temperatures of 700<sup>0</sup> F when sufficient lubricant flow was maintained.

Fatigue tests were conducted at 600<sup>0</sup> F with a synthetic paraffinic oil plus an antiwear additive, a polyphenyl ether with an oxidation inhibitor, and a fluorocarbon with no additives. The synthetic paraffinic oil exhibited 10-percent fatigue lives 2 and 3 times greater than the lives obtained with the fluorocarbon and the polyphenyl ether lubricants, respectively.

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\* General Electric Company, Cincinnati, Ohio.

## INTRODUCTION

Proper lubrication of a bearing is as vital to its successful operation and endurance as proper design and material selection. The precise role of a lubricant in rolling-element fatigue (and thus load capacity) is not fully understood. As technology advances, higher bearing temperatures (in the range from 500<sup>o</sup> to 700<sup>o</sup> F) dictate the need to develop new lubricants with better thermal and oxidative stability (ref. 1).

Oxidation is a common cause of lubricant failure in bearings that operate at high temperatures. This oxidation results in formation of resins, sludges, and acids that can cause deposits on critical bearing surfaces, corrosion of bearing and lubrication system metal parts, clogging of screens, small clearances and passages in the lubrication system, and an increase in lubricant viscosity (ref. 2).

Lubricant type and rheology also affect bearing fatigue life. Lubricants form elastohydrodynamic films in the ball-race contacts during bearing operation. These films affect the stress distribution on the races and balls. In essence, the lubricant film reduces the maximum stress from that calculated for a dry static contact (ref. 3). A reduction as small as 10 percent in stress can mean more than a twofold increase in fatigue life. In general, bearings tested with higher viscosity fluids produce the higher fatigue lives (ref. 4). When the viscosity of the lubricant becomes so low that boundary lubrication occurs, the bearing will probably not fail by fatigue but by wear or superficial surface pitting.

The objectives of the research reported herein were to determine (1) the operating capability of several lubricants in rolling-element bearings at temperatures from 400<sup>o</sup> to 900<sup>o</sup> F and (2) the rolling-contact fatigue characteristics with three representative high-temperature lubricants at 600<sup>o</sup> F.

Seven lubricants, which are considered to be of interest for high-temperature bearing application, were investigated. Each of these lubricants was run with 204 size angular-contact ball bearings made from AISI M-10 steel in the NASA high-temperature bearing torque apparatus to determine the effect of the lubricants on torque characteristics and running-track appearance at temperatures in the range of 400<sup>o</sup> to 900<sup>o</sup> F.

The effects of three of these lubricants on the rolling-contact fatigue life of test bars at 600<sup>o</sup> F with a maximum Hertz stress of 700 000 psi was determined in the General Electric rolling-contact apparatus.

## APPARATUS

### High-Temperature-Bearing Torque Tester

The test apparatus shown in figure 1 was used to test the seven lubricants at outer-race temperatures from 400<sup>o</sup> to 900<sup>o</sup> F. The test bearing is held in a housing that is

heated by eight resistance-type cartridge heaters. The housing is supported by an externally pressurized gas bearing which, in turn, is supported by four rubber vibration mounts. The effects of minor misalignments and oscillations are thus minimized. The rotating inner race is mounted on a shaft driven by a gear belt and induction motor. A thrust load is applied through a load arm system with a dead weight.

The gas bearing allows torque measurement. A cable was attached at one end to the test-bearing housing and at the other end to a bridge-type strain-gage force transducer. The output of the force transducer (which is proportional to bearing torque) was monitored on a cathode-ray oscilloscope.

The test lubricant was introduced in a mist form with a nitrogen carrier gas at a lubricant flow rate of about 0.06 cubic inch per minute (0.002 lb/min at a specific gravity of 0.9). The lubricant mist could be preheated before entering the test bearing with a resistance heating coil wrapped around the lubricator tube.

Temperatures were measured during operation with thermocouples at several locations: the outside diameter of the outer race, the ambient gas in the test bearing chamber, and the lubricant mist before it contacted the bearing surfaces. The inner-race temperature was measured by stopping the apparatus rotation and thrusting a low-mass thermocouple against the inner race. The outer-race temperature and the mist temperature could be manually controlled by adjusting the voltage applied to the heaters previously mentioned.

The design of the apparatus does not allow a totally inert atmosphere to be used in the bearing chamber; however, the use of nitrogen gas to carry the lubricant mist reduces the oxygen content in the chamber. A paramagnetic oxygen analyzer was used to determine the oxygen content in the bearing chamber with a sensitivity of 0.1 percent in the range of zero to 25 percent.

## Rolling-Contact Fatigue Tester

The rolling-contact fatigue tester is shown in figure 2. A cylindrical test bar is mounted in a precision chuck. A drive means is attached to the chuck thereby driving the bar which, in turn, drives two large idler disks. The disks are 7.5 inches in diameter and have a crown radius of 0.25 inch. The disks are mounted on double-row ball bearings that are supported by a massive pendulum yoke. The load is applied by closing the disks against the test bar with a micrometer-threaded turnbuckle and a calibrated load cell. The disks in contact with the test bar produce a compressive stress pattern similar to that of a ball loaded on the inner race of a ball bearing. Lubrication is supplied by a drip feed system with a needle valve to control the flow rate. Several test runs can be made on one test bar by moving the bar position in the axial direction relative to the disk

contacts. The test bar is rotated at 12 500 rpm thus receiving 25 000 stress cycles per minute. Test temperature is maintained by induction heating coils around the test bar on each side of the crowned disks. The temperature is measured and controlled by an infra-red pyrometer. The lubricant can be preheated to the testing temperature prior to dropping onto the test bar. The test bar is open to an air atmosphere.

## MATERIALS AND LUBRICANTS

### Test Bearings

The bearings used in the high-temperature bearing test apparatus were 204 size (20-mm bore) angular-contact ball bearings of ABEC-3 specifications (Annular Bearing Engineers Committee). The bearing material was AISI M-10 steel (balls and races) with a nickel-alloy inner-race riding retainer. Rockwell C hardnesses were  $60 \pm 1$  for the outer races,  $62.5 \pm 1$  for the inner race, and  $61 \pm 2$  for the balls. The nickel alloy retainer was Rockwell B 72 to 74. Further specifications of the bearings are as follows:

Contact angle, deg . . . . .	18±1
Number of balls . . . . .	8
Ball diameter, in. . . . .	5/16
Conformity, percent:	
Inner race . . . . .	51
Outer race. . . . .	52

### Test Bars

The 3-inch long cylindrical test bars for the rolling-contact apparatus fatigue tests were fabricated from three separate heats of consumable-vacuum-melted AISI M-50 steel. The bars were heat treated to achieve a Rockwell C hardness of  $63 \pm 0.5$ . The contacting disks were machined from a fourth heat of consumable-vacuum-melted AISI M-50 steel and heat treated to the same hardness as the bars. The test bars were then ground to a diameter of 0.375 inch with a surface finish of 5 to 8 microinches root mean square. Similarly, the disks were ground to a disk diameter of 7.5 inches and a crown radius of 0.25 inch. The surface finish of the disks was the same as the test bars.

## Test Lubricants

Seven test lubricants were evaluated that were considered of interest for high-temperature bearing applications. The lubricants and their properties are summarized in table I. An ASTM standard temperature-viscosity chart is shown in figure 3. These properties were obtained from the manufacturer of each lubricant. Classes of lubricants investigated included a polyester, a super-refined mineral oil, a synthetic paraffinic oil, a polyphenyl ether, and a fluorocarbon. The synthetic paraffinic oil and the polyphenyl ether were evaluated with and without additives.

## PROCEDURE

### Bearing Tests

Each bearing to be tested was cleaned with solvent, dried, and installed in the test apparatus. The nitrogen cover gas was turned on, and the heaters were energized. When the initial test temperature was reached, lubricant flow was started, the load was applied, and the apparatus motor was started. Periodic readings were made of bearing torque, outer-race temperature, test chamber ambient temperature, and lubricant-mist temperature. Race temperature and lubricant-mist temperature were varied independently during the tests. Testing time at each temperature varied from 10 to 60 minutes depending on the test temperature. After shutdown, the flow of nitrogen cover gas was maintained until the test chamber had cooled to room temperature. The bearing was removed from the test chamber, photographed, and visually inspected for varnish deposits and wear.

### Fatigue Tests

Fatigue testing was performed in the rolling-contact fatigue tester. The test bar was placed in the tester and adjusted in the drive chuck for maximum runout of 0.001 inch. The disks were brought against the bar by using a mechanical turnbuckle. A load was applied sufficient to allow the test bar to drive the contacting disks. The test bar was subsequently run at full speed. The test temperature stabilized within 30 minutes of operation.

When the disks and test bar were in thermal equilibrium, the full test load was applied. When a fatigue failure occurred, the apparatus and related instrumentation were automatically shut down. The axial position of the test bar in the drive chuck was changed in order to use a new running track, and then testing was resumed.

A bar from each of the heats of material was used (see section Test Bars) with each lubricant tested. The fatigue data were statistically evaluated by using the Weibull distribution with the methods of reference 5. A straight line was drawn through the points by the least-squares method. The data for each lubricant tested are a composite of data from the three heats of AISI M-50 steel.

## RESULTS AND DISCUSSION

Seven bearings were run in the first phase of testing, each with one of the seven prospective high-temperature lubricants listed in table I. Each bearing was run for 1 hour at each of four outer-race temperatures ( $400^{\circ}$ ,  $500^{\circ}$ ,  $600^{\circ}$ , and  $700^{\circ}$  F), a speed of 10 600 rpm, and a thrust load of 100 pounds (maximum Hertz stresses at inner and outer races were 192 000 and 165 000 psi, respectively).

Based on unpublished test results in contract NAS 3-7261 with the General Electric Company, maximum Hertz stress values as low as 195 000 pounds per square inch may be necessary so that long-term bearing operation without gross wear may be attained at temperatures of about  $600^{\circ}$  F. (In these tests a thrust load of 4365 lb on a 120-mm-bore bearing produced a maximum Hertz stress at the inner race of 195 000 psi.) Lubricant flow rate in the once-through mist-lubrication system was held constant at approximately 0.06 cubic inch per minute at each temperature for all lubricants tested.

### Bearing Temperature Distribution

The temperature distribution in the test bearing chamber is shown in table II for outer-race temperatures from  $400^{\circ}$  to  $900^{\circ}$  F. The highest bulk temperatures in the bearing are at the outer race inasmuch as the major source of heat was from the cartridge heaters in the bearing test housing. The inner-race temperature was about  $50^{\circ}$  F below the outer-race temperature over the range from  $400^{\circ}$  to  $700^{\circ}$  F. The lubricant-mist temperature and the chamber ambient-gas temperature were both below the measured bearing temperatures.

### Torque as Affected by Temperature and Viscosity

The average value of the bearing torque for each lubricant is plotted in figure 4 over the outer-race temperature range from  $400^{\circ}$  to  $700^{\circ}$  F. A general trend of decreased torque with increasing outer-race temperature to  $700^{\circ}$  F existed with all seven lubricants.



The bearing torque was plotted in figure 5 against the absolute viscosity of the lubricant at the outer-race temperature. The data in this figure suggest that torque decreased with decreasing lubricant viscosity.

## Deposit Formations

After testing, the bearings were visually inspected for deposit formations, race wear, track appearance, and cage wear. Deposits were formed in various amounts with each of the test lubricants. Heaviest deposits were observed on the bearing run with the NA-XL-24 synthetic paraffinic oil with the antiwear additive; the least deposits accumulated on the bearing run with the NA-XL-20 fluorocarbon lubricant. Photographs of these bearings are shown in figure 6. The other five lubricants produced varying amounts of deposits falling between these two extremes.

## Ball Track Appearance

No significant differences in cage wear, race wear, and track appearance were observed for the seven lubricants. A portion of the inner-race groove of a typical test with a maximum outer-race temperature of 700<sup>0</sup> F is shown in figure 7. The photograph shows one edge of the running track. The finishing marks on the running track surface that can readily be seen indicate that an elastohydrodynamic film was apparently present in the contact zone. In contrast, an inner-race running track is shown in figure 8 where the groove surface has obviously been destroyed. Some smearing and superficial pitting has occurred, and the original finish marks have been obliterated. This race is from a test bearing in which the NA-XL-23 lubricant flow was inadvertently low for a period of about 30 minutes. The maximum race temperature was 900<sup>0</sup> F which would suggest that not only must a lubricant have a reasonable viscosity at the operating temperature, but also a minimum oil flow is required for elastohydrodynamic lubrication.

## Tests to Outer-Race Temperature of 900<sup>0</sup> F

Four of the lubricants with their respective bearings used initially were tested at outer-race temperatures above 700<sup>0</sup> F in an effort to determine an operating temperature limit at which each could be used. It was expected that an increase in bearing torque would result if a temperature were reached where the lubricant was no longer effective in providing an elastohydrodynamic film or began to form excessive deposits. Figure 9

shows the results of these tests. The solid symbols indicate the results of the initial torque tests previously described. Each open symbol represents the average torque for about 10 to 15 minutes at each temperature in the range from 700<sup>o</sup> to 900<sup>o</sup> F. For each lubricant, except the NA-XL-24 synthetic paraffinic oil with an antiwear additive, an increase in torque occurred at some temperature.

The NA-XL-10 super-refined mineral oil showed increased torque at the lowest temperature, about 750<sup>o</sup> F, but leveled off above 750<sup>o</sup> F. The NA-XL-20 fluorocarbon and the NA-XL-23 polyphenyl ether showed increased torque at 900<sup>o</sup> and 850<sup>o</sup> F, respectively.

Examination of the bearings after these tests show that some surface distress had occurred in the race grooves in all four test bearings. In three of the bearings, this surface distress was characterized by an obscuring of the original grinding marks in the inner-race ball track. The fourth bearing that was tested with the NA-XL-23 showed more severe smearing and pitting (fig. 8) which probably was a result of the period of lubricant starvation discussed in the previous section.

No significant differences were noted in the conditions of the running tracks of the bearing races tested with NA-XL-10, 20, and 24 lubricants. The bearings tested with NA-XL-20 and 23, however, had considerably more deposits than the other two tests.

### Effect of Lubricant Preheat on Torque

For each of the four lubricants, NA-XL-10, 20, 23, and 24, at outer-race temperatures of 600<sup>o</sup> and 700<sup>o</sup> F, the effect of lubricant-mist temperature on bearing torque was measured. The lubricant-mist temperature was increased to about 550<sup>o</sup> and 600<sup>o</sup> F at outer-race temperatures of 600<sup>o</sup> and 700<sup>o</sup> F, respectively. In all tests with these four lubricants at both outer-race temperatures of 600<sup>o</sup> and 700<sup>o</sup> F, no significant change in bearing torque was observed as the lubricant-mist temperature was varied over the range from 350<sup>o</sup> to 600<sup>o</sup> F. The lubricant mist had little cooling effect on the bearings and quickly became heated to the bulk temperature of the bearing components.

### Oxygen Content of Gas in Bearing Chamber

The volume-percent of oxygen contained in the ambient gas in the test bearing chamber was monitored during one test with the NA-XL-23 lubricant. Measurements of oxygen content were made over the outer-race temperature range from 400<sup>o</sup> to 900<sup>o</sup> F and lubricant-mist temperatures from 250<sup>o</sup> to 600<sup>o</sup> F. In general, the oxygen content was in the range from 0.5 to 1.7 percent by volume; a slight tendency toward higher oxygen percentages with higher temperatures existed.

## Fatigue Results

Fatigue tests were conducted with three of the lubricants. These were the fluorocarbon, NA-XL-20; the polyphenyl ether with an oxidation inhibitor, NA-XL-23; and the synthetic paraffinic oil with an antiwear additive, NA-XL-24. Each of these lubricants was tested in the rolling-contact fatigue tester with consumable-vacuum-melted M-50 test bars at 600° F bar temperature, 700 000 psi maximum Hertz stress, 12 500 rpm, and 500±20° F lubricant inlet temperature in an air atmosphere. The results of these tests are presented in figure 10 and summarized in table III. Figure 10 consists of three Weibull plots, one for each lubricant, and a summary (fig. 10(d)). The 10-percent lives from the Weibull plots (the life which 90 percent of the test bars will survive) were used for comparison purposes.

The improvement in fatigue life of the test bar with the synthetic paraffinic oil over the fluorocarbon and the polyphenyl ether was about 2 and 3 times, respectively.

The confidence numbers for the 10-percent lives are given in table III. These confidence numbers indicate that 98 and 83 percent of the time the NA-XL-24 lubricant will have a 10-percent fatigue life greater than the NA-XL-23 and NA-XL-20 lubricants, respectively.

## SUMMARY OF RESULTS

Seven lubricants that can be classified as a polyester, a super-refined naphthenic mineral oil, a synthetic paraffinic oil, a polyphenyl ether, and a fluorocarbon were investigated in a high-temperature bearing torque tester. The synthetic paraffinic oil and the polyphenyl ether were evaluated with and without additives. Tests were run with 204 size angular-contact ball bearings of AISI M-10 steel, at a thrust load of 100 pounds, an inner-race speed of 10 600 rpm, and outer-race temperatures from 400° to 900° F. Lubrication was accomplished by a mist-lubrication system. The lubricants were compared with respect to bearing torque and the ability to provide an elastohydrodynamic film. Three of the lubricants were tested with AISI M-50 bar specimens in the rolling-contact fatigue tester at a 600° F test bar temperature. The following results were obtained:

1. Tests with the synthetic paraffinic oil with an antiwear additive (NA-XL-24) exhibited the highest fatigue life. At the 10-percent life level, tests with the synthetic paraffinic oil with an antiwear additive (NA-XL-24) exhibited 2 and 3 times the lives of the fluorocarbon (NA-XL-20) and the polyphenyl ether with an oxidation inhibitor (NA-XL-23), respectively.

2. Examination of the bearing running tracks indicates that elastohydrodynamic lubrication apparently existed in all tests up to an outer-race temperature of 700° F. Above

700<sup>o</sup> F there appeared to be boundary lubrication.

3. Bearing torque decreased with increasing outer-race temperatures up to 700<sup>o</sup> F for all seven lubricants. Above 700<sup>o</sup> F, abrupt increases in torque were exhibited except with the synthetic paraffinic oil with an antiwear additive (NA-XL-24). A general trend was noted showing increased bearing torque with increasing lubricant viscosity.

4. Increasing the lubricant-mist temperature had little or no effect on bearing torque which indicates that the lubricant in mist form had little cooling effect on the bearings and that it quickly became heated to the bulk temperature of the bearing.

Lewis Research Center,

National Aeronautics and Space Administration,

Cleveland, Ohio, January 25, 1967,

720-03-01-01-22.

## REFERENCES

1. Zaretsky, Erwin V.; Anderson, William J.; and Parker, Richard J.: Effect of Nine Lubricants on Rolling-Contact Fatigue Life. NASA TN D-1404, 1962.
2. Bisson, Edmund E.; and Anderson, William J.: Advanced Bearing Technology. NASA SP-38, 1964, pp. 175-202.
3. Zaretsky, E. V.; Sibley, L. B.; and Anderson, W. J.: The Role of Elastohydrodynamic Lubrication in Rolling-Contact Fatigue. J. Basic Eng., vol. 85, no. 3, Sept. 1963, pp. 439-450.
4. Anderson, William J.; and Carter, Thomas L.: Effect of Lubricant Viscosity and Type of Ball Fatigue Life. ASLE Trans., vol. 1, 1958, pp. 266-272.
5. Johnson, Leonard G.: The Statistical Treatment of Fatigue Experiments. Rep. No. GMR 202, General Motors Corp., Apr. 1959.

TABLE I. - PROPERTIES OF TEST LUBRICANTS

Lubricant designation	Base stock	Additive content	Flash point, °F	Fire point, °F	Kinematic viscosity, cs, at -			Specific heat at 500° F, Btu/(lb)(°F)	Density at 500° F, lb/ft <sup>3</sup>
					100° F	210° F	500° F		
NA-XL-10	Super-refined naphthenic mineral oil	(a)	445	495	79	8.4	1.1	0.660	45.7
NA-XL-13	Synthetic paraffinic oil	None	530	580	314	32	<sup>b</sup> 2.9	-----	----
NA-XL-16	Polyester - hindered type	(c)	480	540	27.6	5.2	<sup>b</sup> 1.0	-----	----
NA-XL-20	Fluorocarbon	None	---	---	335	29	2.1	.317	94
NA-XL-22	5P4E polyphenyl ether	None	550	660	363	13.1	1.2	.496	63.1
NA-XL-23	5P4E polyphenyl ether	(d)	550	660	363	13.1	1.2	.496	63.1
NA-XL-24	Synthetic paraffinic oil	(e)	530	580	314	32	<sup>b</sup> 2.9	-----	----

<sup>a</sup>Oxidation inhibitor, extreme pressure additive, and antifoam agent.

<sup>b</sup>Extrapolated.

<sup>c</sup>Oxidation inhibitor and dispersant.

<sup>d</sup>Oxidation inhibitor.

<sup>e</sup>Antiwear additive.

TABLE II. - TYPICAL TEMPERATURE DISTRIBUTION IN THE TEST SYSTEM

Outer-race temperature, °F	Inner-race temperature, °F	Ambient chamber gas temperature, °F	Lubricant mist temperature, °F
400	355	330	~250
500	445	420	280 to 305
600	555	525	350 to 400
700	655	625	450 to 500
800	<sup>a</sup> >700	650 to 700	500 to 550
900	<sup>a</sup> >700	720 to 780	580 to 620

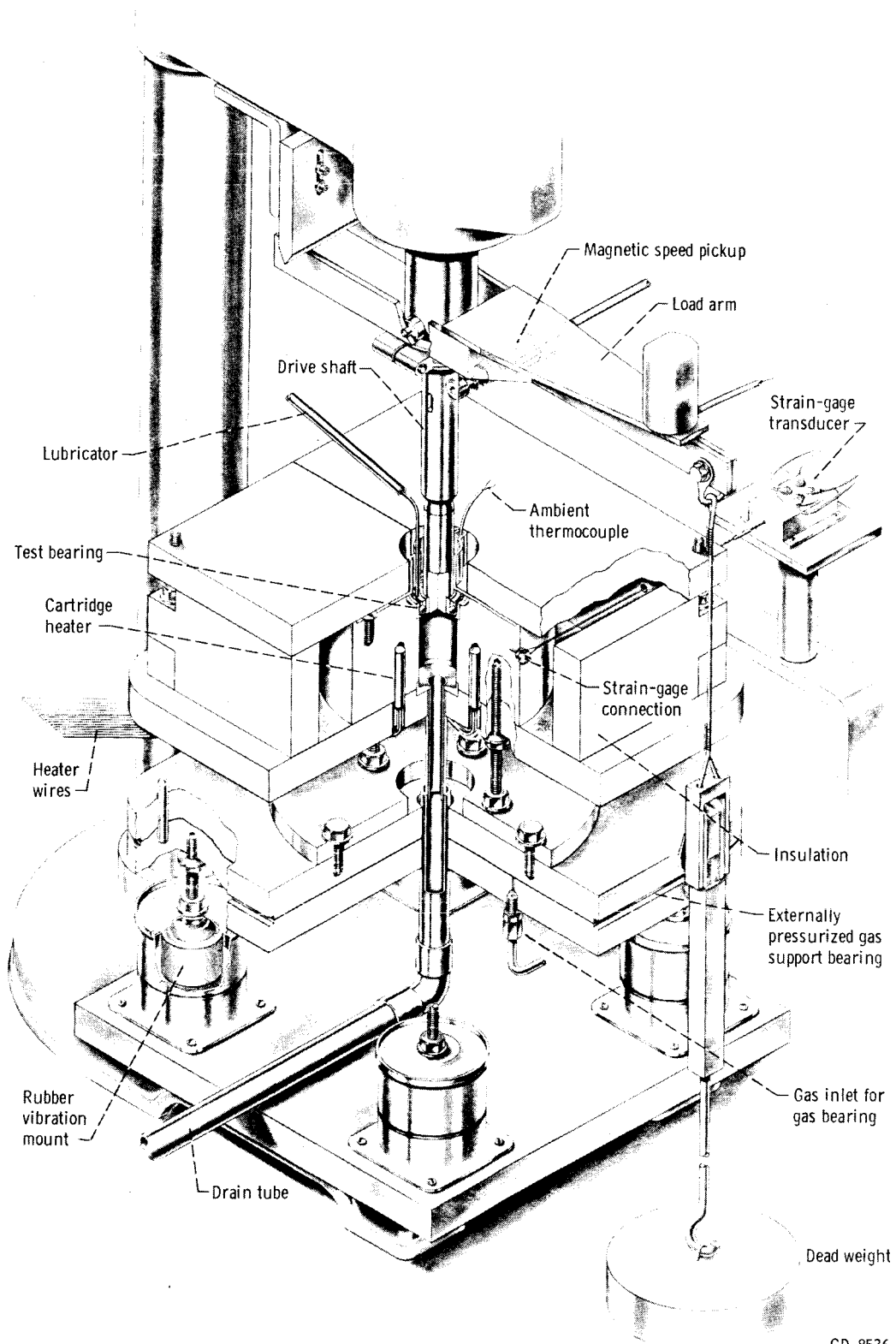
<sup>a</sup>Beyond limit of measurement.

TABLE III. - ROLLING-CONTACT FATIGUE RESULTS WITH THREE LUBRICANTS  
 IN ROLLING-CONTACT FATIGUE APPARATUS

[Bar specimens, consumable vacuum melted M-50; bar temperature, 600° F;  
 maximum Hertz stress, 700 000 psi.]

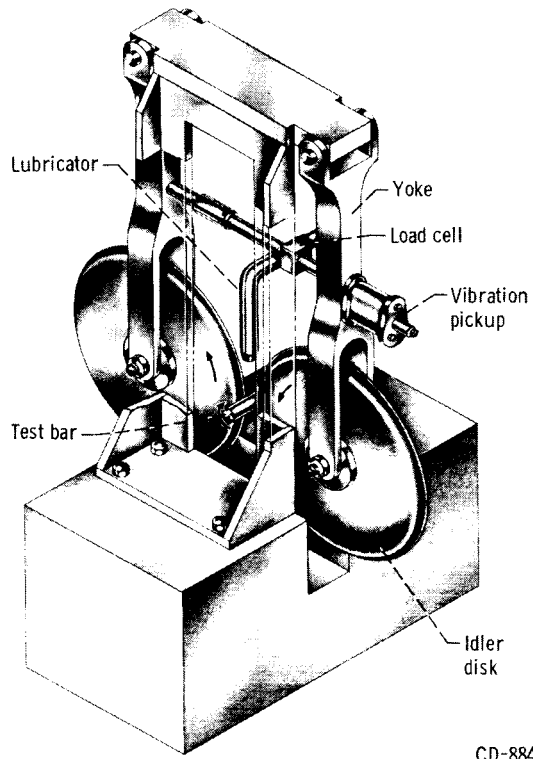
Lubricant designation	Lubricant type	Additive	10-Percent life, stress cycles	50-Percent life, stress cycles	Weibull slope	Number of tests	Number of failures	Confidence number, <sup>a</sup> percent
NA-XL-20	Fluorocarbon	None	1.04×10 <sup>6</sup>	6.76×10 <sup>6</sup>	1.0	15	15	83
NA-XL-23	Polyphenyl ether	Oxidation inhibitor	.69	2.70	1.4	28	28	98
NA-XL-24	Synthetic paraffinic oil	Antiwear additive	2.35	7.26	1.7	19	19	--

<sup>a</sup>Percentage of time that 10-percent life with NA-XL-24 will be greater than the 10-percent life with other lubricants.



CD-8536

Figure 1. - High-temperature bearing test apparatus.



CD-8840

Figure 2. - Rolling-contact fatigue apparatus.

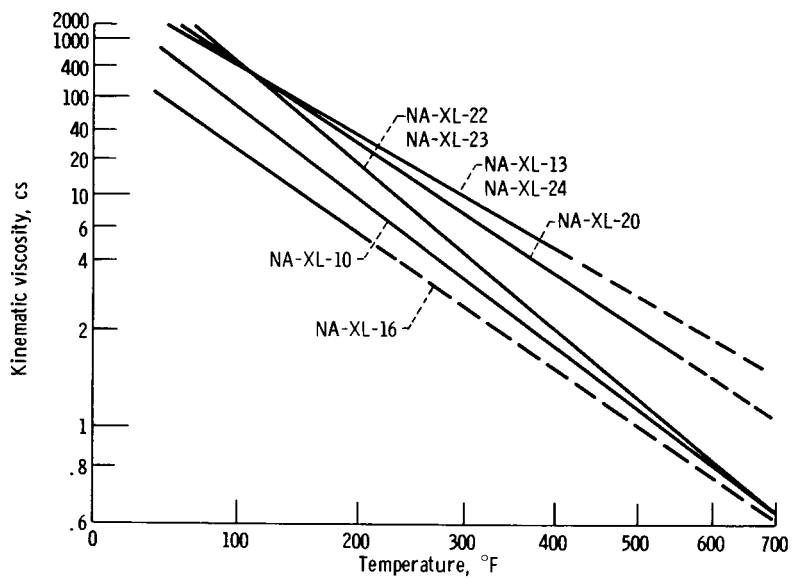


Figure 3. - Temperature-viscosity characteristics of seven test lubricants. (Data from lubricant manufacturer. For description of lubricants, see table I.)



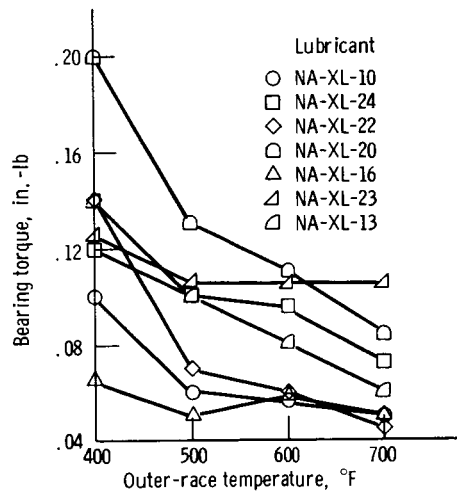


Figure 4. - Effect of outer-race temperature on measured bearing torque for seven lubricants. Thrust load, 100 pounds; test duration at each temperature, 1 hour; inner-race speed, 10 600 rpm. (For description of lubricants, see table I.)

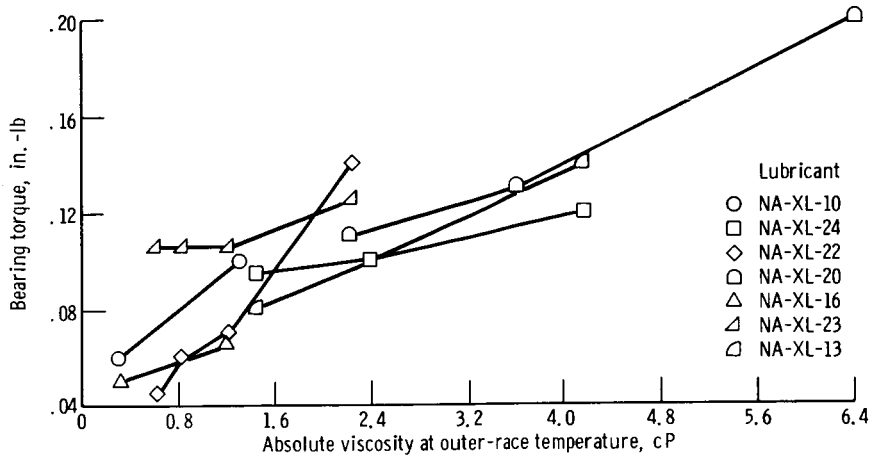
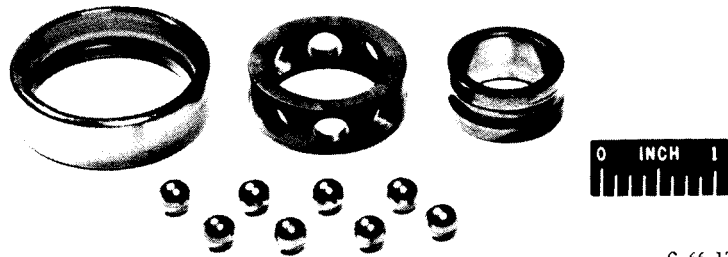
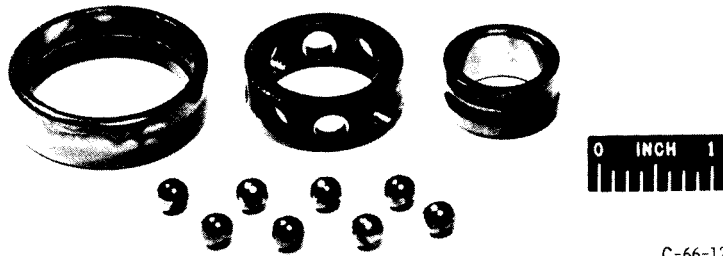


Figure 5. - Bearing torque as function of absolute viscosity at outer-race temperature. Thrust load, 100 pounds; test duration, 1 hour at each point. Inner-race speed, 10 600 rpm. (For description of lubricants, see table I.)



C-66-1250

(a) Fluorocarbon lubricant (NA-XL-20).



C-66-1244

(b) Synthetic paraffinic oil lubricant with antiwear additive (NA-XL-24).

Figure 6. - Best and worst case deposit formations on test bearings run for 4 hours at outer-race temperature from 400° to 700° F.

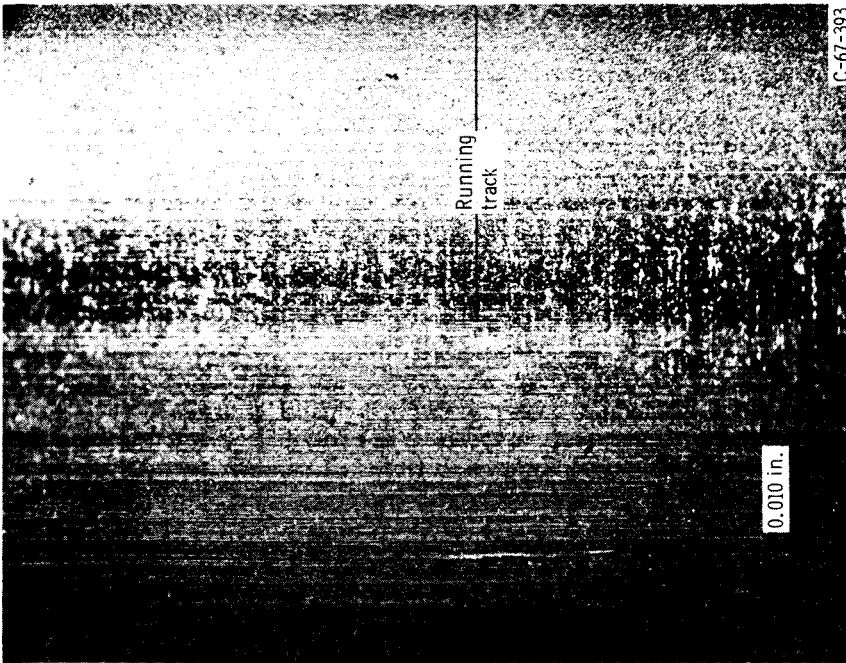


Figure 7. - Typical inner race groove after test at outer-race temperature of 700° F with synthetic paraffinic oil lubricant (NA-XL-13).

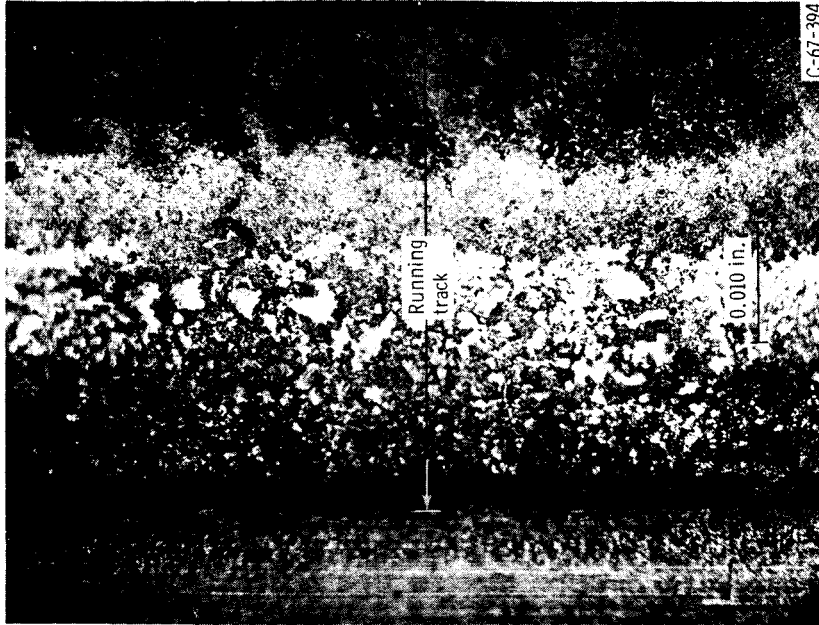


Figure 8. - Pitting in running track of bearing run with polyphenyl ether with oxidation inhibitor (NA-XL-23) at maximum outer-race temperature of 900° F.

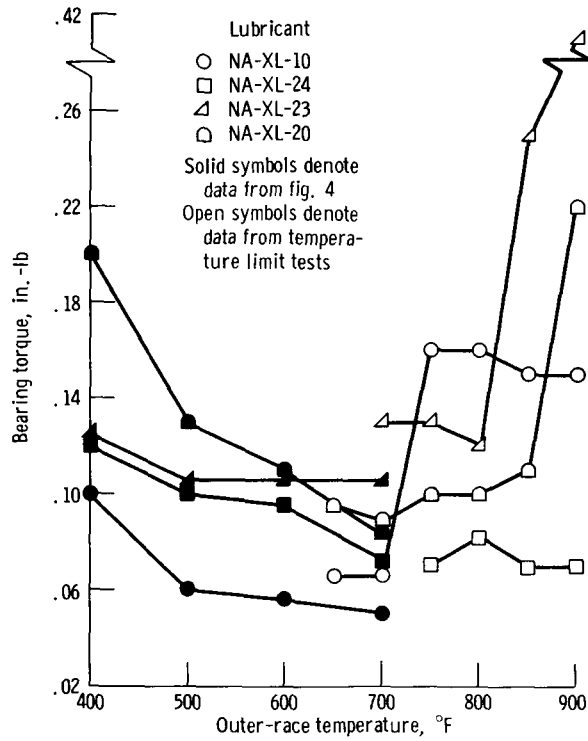
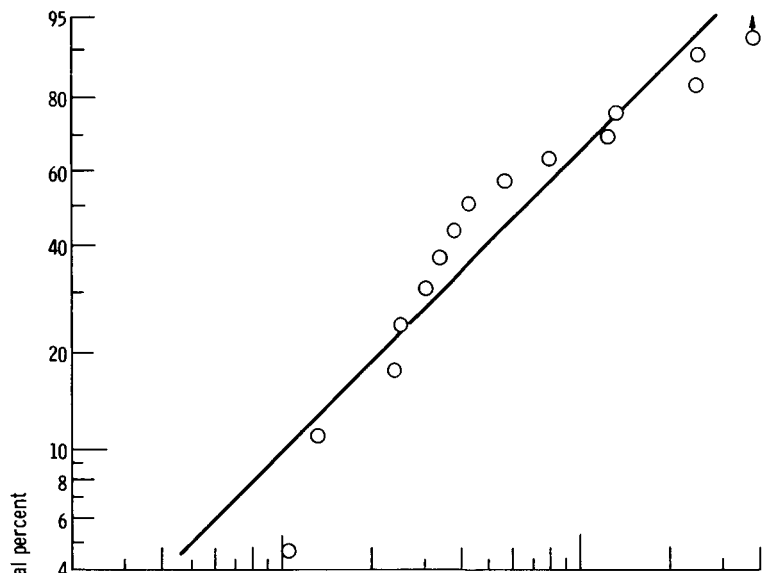
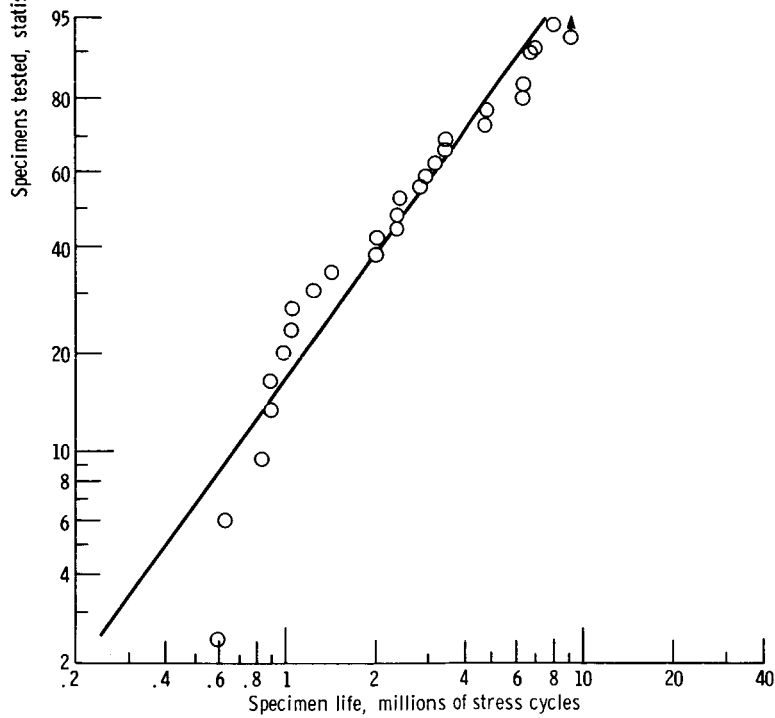


Figure 9. - Effect of outer-race temperatures over 700° F on bearing torque for four lubricants. Thrust load, 100 pounds; test duration at each temperature, 10 minutes; inner-race speed, 10 600 rpm. (For description of lubricant, see table I.)

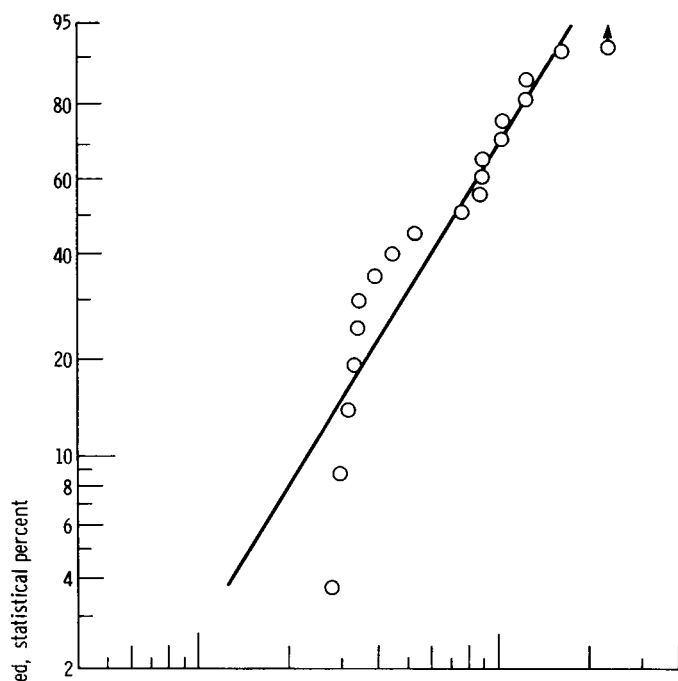


(a) Fluorocarbon (NA-XL-20).

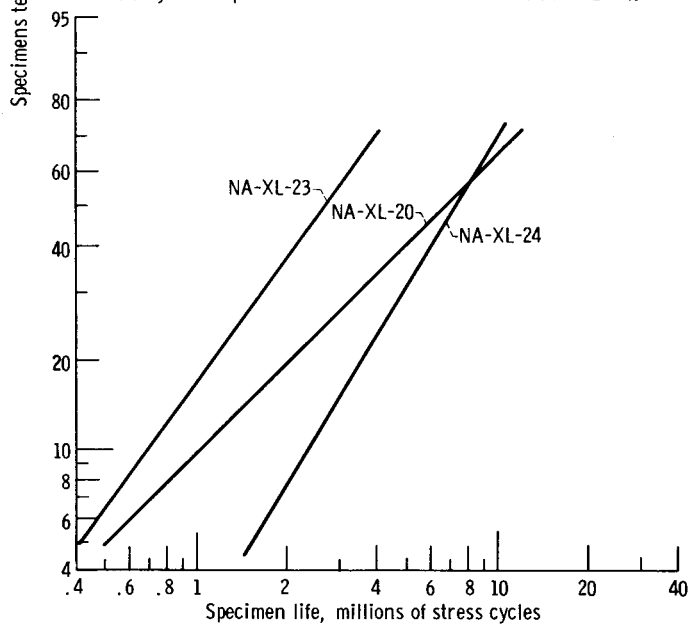


(b) Polyphenyl ether with oxidation inhibitor (NA-XL-23).

Figure 10. - Rolling-contact fatigue life of AISI M-50 test bars in rolling-contact fatigue apparatus with various lubricants. Test bar temperature, 600° F; maximum Hertz stress, 700 000 pounds per square inch.



(c) Synthetic paraffinic oil with antiwear additive (NA-XL-24).



(d) Summary.

Figure 10. - Concluded.