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TECHNICAL NOTE 4161

EFFECT OF LUBRICANT BASE STOCK ON ROLLING-CONTACT FATIGUE LIFE

By Thomas L. Carter

Lewis Flight Propulsion Laboratory
Cleveland, Ohio

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EFFECT OF LUBRICANT BASE STOCK ON ROLLING-CONTACT FATIGUE LIFE

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SUMMARY

Five lubricants of different base stock were tested using groups of 1/2-inch air-melt AISI M-1 tool-steel balls under rolling-contact fatigue conditions in the fatigue spin rig. A methyl silicone, a mineral oil, a glycol, a sebacate, and an adipate were used. The particular fluids chosen all had about the same atmospheric pressure viscosity (10 centistokes) at the test temperature of 100° F. Other test conditions such as ball loading were held constant. Ball loading was held at that level necessary to produce a maximum theoretical Hertz stress of 725,000 pounds per square inch in compression at the contacting surfaces and 225,000 pounds per square inch in shear at a depth of 0.009 inch below the surfaces. This investigation studies the effect of lubricant base stock upon rolling-contact fatigue life and correlates any observed differences in life results with unique properties of the different base-stock fluids.

The tests showed differences in rolling-contact fatigue life for the five different base-stock fluids tested. The observed lives appeared to correlate with the pressure-viscosity coefficients of lubricants of the same base stock. Lubricants whose viscosities were increased the greatest by pressure produced the longest fatigue lives. However, other lubricant properties, such as bulk modulus and chemical activity, may well influence fatigue life, and more data are required to determine their relative importance. Chemical activity did not appear to be significant in these tests, and the spalls obtained in all tests compared closely with those obtained in full-scale bearings. Metallurgical transformation in the material was consistent for all test runs.

INTRODUCTION

The ever increasing operating temperatures of aircraft gas-turbine engines have created a demand for rolling-contact bearings capable of sustaining more severe operating conditions than possible with long established materials and lubricants. Development of new lubricant-material combinations must be carried out in order to meet successfully present and anticipated demands of engine designers for satisfactory bearing performance under more severe operating conditions. A number of

available lubricants and steels retain their mechanical properties throughout the anticipated temperature range necessary for future bearing operation. However, another consideration in bearing development is rolling-contact fatigue life. In addition to the bearing design and the loading and the materials used in the rolling elements and races, the lubricant used has an important role in rolling-contact fatigue life.

Two of the more important and most commonly considered functions of the lubricant are the reduction of sliding friction and the cooling of the bearing. Another function of the lubricant that appears to be very important is its influence on contact stresses through hydrodynamic action. Since fatigue life varies inversely with about the tenth power of maximum contact stress in the fatigue spin rig (ref. 1), a small change in the stresses borne by the contacting surfaces can have a significant effect upon fatigue life.

An evaluation of this hydrodynamic effect of the lubricating film will provide valuable information for selection of the more promising high-temperature lubricants from a number of possible choices. The literature available in this category is very limited. In an attempt to increase the understanding of the hydrodynamic effect of the lubricating film, AISI M-1 tool-steel balls were run in the fatigue spin rig with a series of common paraffinic-base mineral oils of different viscosity as measured by standard methods at atmospheric pressure. This work is reported in reference 2. A continuous increase in fatigue life was observed with increasing lubricant viscosity indicating that the lubricant may have acted to reduce contact stresses. This effect is also reported in reference 3, where a variation of temperature was required to change viscosity. A reduction in life at higher temperatures by factors other than lubricant viscosity is reported in reference 4. In the previous work reported in reference 2 and in the work covered by this report, temperature was held constant.

The effect of lubricant viscosity on fatigue life observed in reference 2 is a correlation of life with that viscosity which is measured at atmospheric pressure. At a constant temperature, viscosity is known to increase very substantially with pressure (ref. 5). Since the lubricating fluid in the contact zone of the rolling elements is under very high pressures, the actual viscosity in the contact zone would be much higher. Since the viscosities of fluids of different base stock increased at different rates with pressure, fluids with the same viscosity at atmospheric pressure but of different base stock should produce different fatigue lives.

This report concerns the effect of various lubricants on the life of AISI M-1 tool steel. These lubricants were of different base stocks but had the same viscosity at atmospheric pressure at 100° F. Five lubricant fluids, a methyl silicone, a paraffinic mineral oil, a glycol, a sebacate, and an adipate, were chosen for this investigation.

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APPARATUS

In the interest of brevity, only brief descriptions of the apparatus and procedure are given here. A more detailed presentation can be found in references 1 and 6 as well as in appendix A. Figure 1(a) is a cutaway view of the rolling-contact fatigue spin rig. Air jets caused the two test balls to revolve in a horizontal plane on the bore surface of a hardened tool-steel cylinder (fig. 1(b)). The loading on the balls was produced by centrifugal force, and the stress was calculated according to the methods of reference 7. Approximately 15 milliliters per hour of lubricant were introduced in droplet form into the drive airstream between the two guide plates above and below the air jets. The fast-moving airstream had an atomizer effect, and the lubricant was reduced to a fine mist that adhered to surfaces to provide a lubricating film.

Orbital speed was measured by counting the pulses from a photoamplifier on an electronic tachometer; a pulse was generated each time a ball interrupted a light beam. A ball or race failure resulted in increased vibration and hence the generation of an increased signal from a velocity pickup attached to the rig. This signal, when amplified, actuated a meter relay that shut down the system.

Temperature was controlled by mixing heated air with the normal drive air supply. The test temperature and temperature-control signal were taken from thermocouples on the top and bottom of the race cylinder.

All ball test specimens were from the same heat of AISI M-1 air-melt tool steel and had a nominal 1/2-inch diameter. The running track on the balls was predetermined by grinding two diametrically opposed 3/16-inch flats on the ball surface. Race cylinders were AISI M-1 vacuum-melt tool steel.

PROCEDURE

Before the test all race cylinders were given dimensional surface-finish and hardness inspections. All test balls were weighed and given a surface examination at a magnification of 36. A record was kept of any abnormalities in surface conditions at the running track. Prior to inspection and use, test specimens were flushed and scrubbed with 100 percent alcohol and clean cheesecloth. During storage they were protected by a corrosion-resistant-oil film. Care was taken during assembly not to scratch the running surfaces. The bore surface and test balls were coated with the test lubricant during assembly.

The rig was brought up to operating speed as rapidly and as smoothly as possible. About 3 minutes were required for the hot airstream to heat the test cylinder to the test temperature when running at 100° F. Speed,

air pressure, temperature, and vibration levels were recorded during the test. Total running time was recorded and converted into total stress cycles on the ball specimen. A post-test surface examination at a magnification of 36 was made to observe track conditions.

Failure data were plotted on Weibull paper, which is a distribution of the log log of the reciprocal of the portion of the sample surviving against the log of the stress cycles to failure. This distribution function developed by Weibull fits the observed scatter in the fatigue lives of rolling-contact bearings (ref. 8). Because of the usually small sample (about 30 bearings) involved, the data cannot be fitted reliably into a frequency curve. Instead, the cumulative form of the distribution is used. The cumulative distribution function (Weibull) is as follows:

$$\log \frac{1}{S(L)} = GL^e$$

where $S(L)$ is the fraction of the sample surviving the first L stress cycles and G and e are positive constants.

Figures presenting ball life results use special probability paper on which the Weibull distribution becomes a straight line of slope e. The ordinate represents log-log $1/S(L)$ but is graduated in terms of the fraction failed at L stress cycles.

A set of data is ordered according to life, and each succeeding life is given a rank (statistical percentage) and is plotted on Weibull paper. If the median rank is used, a line is drawn that takes the general direction of the array of points and splits the array in half. A median rank is an estimate of the true rank in the population that has an equal probability of being too large or too small.

A table of median ranks for sample sizes up to 20 and formulas for calculation of the median rank values for any order position in any sample size are given in reference 8.

As with any measurement, the confidence in this data is limited by its statistical reliability. With rolling-contact fatigue data, the wide scatter normally encountered necessitates large sample sizes in order to establish accurately the life against percent survival relation manifested in the Weibull plot. At the same time, the expense and the duration of each test limit the practical number of specimens that can be evaluated. Confidence limits for the data produced in this program were calculated by the method of Lieblein (ref. 9).

RESULTS AND DISCUSSION

Groups of 16 to 28 1/2-inch air-melt AISI M-1 tool-steel balls were run in the rolling-contact fatigue spin rig with each of five lubricants of different base stock. The lubricants tested were a methyl silicone, a paraffinic mineral oil, a glycol, a sebacate, and an adipate. Properties of the lubricants are summarized in table I. The M-1 ball material was obtained from Latrobe heat 13-801 and contained nominally 0.8 percent carbon, 4 percent chromium, 1 percent vanadium, 1.5 percent tungsten, and 8.5 percent molybdenum. The kinematic viscosity of each lubricant fluid was close to 10 centistokes at the test temperature of 100° F. In all tests except those in which the sebacate was used, ball loading was held at that level necessary to produce a maximum theoretical Hertz stress of 725,000 pounds per square inch in compression at the contacting surfaces and 225,000 pounds per square inch in shear 0.009 inch below the surfaces. In the sebacate tests a maximum Hertz stress of 650,000 pounds per square inch was maintained, and the results were corrected to the stress level of 725,000 pounds per square inch.

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Weibull plots for these test runs are given in figure 2. A summary of the Weibull plots is given in figure 3. These results are condensed in the following table:

Lubricant	Pressure-viscosity coefficient, psi^{-1}	Sample size	Failures	Runouts	10-Percent failure life, stress cycles	50-Percent failure life, stress cycles
Methyl silicone	8.35×10^{-5}	26	5	21	106×10^{6}	[a]17000×10^{6}
Paraffinic mineral oil	5.38×10^{-5}	28	14	14	18×10^{6}	660×10^{6}
Glycol	3.28×10^{-5}	16	11	5	7.6×10^{6}	167×10^{6}
Sebacate	2.52×10^{-5}	22	15	7	5.2×10^{6}	200×10^{6}
Adipate	Unavailable	26	12	14	2.4×10^{6}	1000×10^{6}

[a]Extrapolated.

The number of failed balls in the silicone group is fewer than that usually considered desirable for an evaluation of a lubricant-material combination. However, extremely long life was observed with this lubricant. This life was superior to that with the other lubricants studied. A greater number of failures would have entailed multiplying already excessive test run times. Even though only a limited number of failures was obtained, the Weibull plot for those points was close to a straight line. In view of this and the large number of balls tested without failure after over one billion stress cycles, the data are considered conclusive to the extent that the methyl silicone is a superior lubricant in rolling-contact fatigue under these test conditions.

The di(2-ethylhexyl)sebacate data were originally produced for a purpose other than this series of evaluations (ref. 4). In the sebacate data, the Hertz stress was 650,000 pounds per square inch as compared with 725,000 pounds per square inch in the tests of the other lubricants, but, since life is a function of the reciprocal of the tenth power of stress (ref. 1), the life which should result at the stress level of 725,000 pounds per square inch can be calculated and is presented in figure 2(d). Ball specimens were identical with the other test runs reported in this group.

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Failure type and appearance were consistent for all fatigue failures observed in this investigation. No difference was observed between failures produced with the various lubricants. These failures were consistent with other failures observed in the spin rig and in full-scale bearings. Both are limited in depth and localized in area and usually originated from subsurface shear cracking. The wide range in life results is readily apparent in figure 3 and the previous table. The results as indicated by the 10-percent lives have a range of about 40 to 1 from the silicone to the adipate. Since all test conditions were held constant except for the lubricating fluids, and all the fluids had essentially the same atmospheric viscosity, differences in chemical or physical properties of the lubricants themselves must be responsible for this range in life results.

The variation in viscosity as measured at atmospheric pressure (8.75 to 13.5 centistokes) of the fluids used (table I) would account for a life range of only about 1.05 to 1 based upon the relation of life against viscosity established in reference 2. This range can be considered negligible when compared with the observed range of 40 to 1.

Another possible cause of variation in fatigue life from one lubricant to another could be the variation in type and intensity of chemical action at the contacting surfaces. Since the five lubricants are distinctly different compound types, they could produce widely different decomposition products and chemical attack at the contacting surfaces. However, no significant evidence of chemical attack on the ball track surfaces was observed in this series of test runs. The running tracks were darkened because of the formation of an oxide film in all but the shortest lived balls. The only variation among these five fluids was a less pronounced track darkening for the glycol run. This darkening of the running track is normal for specimens run in the spin rig, and no significant contrasts were noted among the five lubricant-material groups observed in this test series. No evidence of corrosion pitting or stress corrosion cracking was observed in any of the test specimens during the post-run inspection.

Metallurgical transformations in the subsurface zones of maximum shear in the contacting bodies are characteristic of materials subjected to a large number of rolling-contact stress cycles. This condition is discussed in detail in reference 10. This alteration of the subsurface material

due to deterioration of the metallographic structure could affect fatigue life. However, only slight amounts of transformation were observed at 100° F and no contrasts were observed among balls run with the different lubricants. It is not felt that metallurgical transformation was a significant factor in the variation in rolling-contact fatigue life observed with the five different lubricants studied.

Since differences in chemical activity, metallurgical transformation, and atmospheric-pressure lubricant viscosity do not appear to be significant factors in causing the wide variation in rolling-contact fatigue life, the effect may be caused by differences in the physical properties of the five lubricants.

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During high-speed rolling contact the lubricant probably affects the pressure distribution existing in the contact zone through hydrodynamic action. The theoretical calculations of stress as described in appendix A are for static loading only. At high rolling speeds these may not be entirely correct. A precise three-dimensional analysis of this phenomenon would be very complicated, but a two-dimensional analysis for an infinitely long roller on a race is given in reference 11. The analysis of reference 11 indicates that appreciable reductions in contact stress can be accomplished when a viscous lubricant is present. However, the assumption of constant viscosity in reference 11 may necessitate qualifying the results.

If lubricant viscosity has an effect on fatigue life, the viscosity at the pressures existing in the contact zone is the important variable. Figure 4 shows that the rate of increase in viscosity with pressure may be significantly different for fluids of different base stock. This relation can be approximated by the form

$$\eta_P = \eta_o 10^{\alpha P}$$

where

η_P viscosity at pressure, centipoises

η_o viscosity at 1 atm, centipoises

α constant, psi^{-1}

P pressure, lb/sq in.

In this relation, α depends on the fluid and η_P is influenced by both η_o and α. Since η_o is constant for these fluids, the controlled variable is α. Thus, fatigue life would be influenced by the pressure-viscosity characteristics of the lubricant. If this theory is correct, a

correlation would exist between rolling-contact fatigue life and the pressure-viscosity coefficient α.

Pressure-viscosity data were not available for the actual batches of lubricants used in this investigation. Elaborate procedures and equipment are needed for measurement of viscosity at high pressure. Pressure-viscosity data for fluids of the same base stock as those used in this report were evaluated in reference 5. From this source the data for the particular fluids which most closely resembled the fluids used to obtain fatigue data were selected, and the pressure-viscosity coefficients were calculated using an average slope (fig. 4). Details of this selection and calculations are given in appendix B.

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A plot of 10-percent life against pressure-viscosity coefficient α is presented in figure 5(a). The same plot for 50-percent life is given in figure 5(b). No point is included for the adipate run because no pressure-viscosity data were available for this fluid. The correlation between the pressure-viscosity coefficient and the 10-percent life is good and that for the 50-percent life is fair. This is in contrast to reference 12 in which the authors attempted the same correlation with inconclusive results.

Evidently other lubricant properties also influence fatigue life. Perhaps bulk modulus or relative chemical activity might be of importance. Although no evidence of chemical activity was found, the least active lubricants (silicone and mineral oil) gave the best life, while the most active (adipate and sebacate) gave the poorest life. It is interesting to note that long life was obtained with the lubricants having low neutralization numbers and relatively short life was obtained with those having high neutralization numbers.

Still another factor might be the dynamic response of the viscosity to a sudden change in pressure. At 30,000 rpm in the spin rig, a stress cycle on an elementary volume occurs in approximately 8×10^{-6} second. In reference 13 it is shown experimentally that, 0.001 second after the application of a high pressure, the viscosity of a lubricant is still appreciably below its equilibrium value at the same pressure. Therefore, it seems logical to conclude that the time rate of change of viscosity after the application of a high pressure might be an important property. This time response of viscosity to a pressure change might be significantly different for different base stocks and may not correlate with the ultimate equilibrium pressure-viscosity values. More extensive data and further rheological studies are needed before definite conclusions can be drawn.

Interpretation of the correlation of the pressure-viscosity coefficient of a lubricant with rolling-contact fatigue life must be tempered by the confidence in the statistical reliability of the data. No confidence limits could be calculated by the method of reference 9 for figure

2(a) because of the large number of unfailed balls. However, an estimate based on the line is given. For the sample sizes of 28, 16, 22, and 26 balls each, used to produce the data in figures 2(b) to (e), these confidence limits are wide in relation to the observed differences in lives. The standard deviations for the individual groups cover a range of about 8 to 1 while the range between the shortest and longest lived groups is about 40 to 1. Thus, the possible variation within the individual groups is not small when compared with the over-all range of results. From this standpoint, the results may not be statistically significant. However, if no effect due to lubricant base stock exists, the probability of the four life plots falling in order of ascending pressure-viscosity coefficients is only 1 in 24. This is so because the lines were calculated by the least-squares, best-fit technique so that they are objective. Thus, the results reported in figure 5(a) have a 96-percent probability not to have been caused by chance.

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SUMMARY OF RESULTS

Five lubricants of different base stock having approximately the same viscosity at atmospheric pressure were tested under rolling-contact fatigue conditions using groups of 1/2-inch air-melt AISI M-1 tool-steel balls. A methyl silicone, a paraffinic mineral oil, a glycol, a sebacate, and an adipate were tested. All other test conditions were held constant with a test temperature of 100° F and a maximum theoretical Hertz stress of 725,000 pounds per square inch in compression. The results of this investigation are as follows:

1. A variation in rolling-contact fatigue life with the various lubricants was noted which appeared to correlate with the pressure-viscosity coefficients of lubricants of the same base stock as those used in the tests. Longer fatigue lives were observed for balls run with lubricants such as a silicone and a mineral oil whose viscosities increased at a greater rate with pressure. However, other lubricant properties, such as bulk modulus and the dynamic response of viscosity to a pressure change, may influence fatigue, and more data are required to determine their relative importance.

2. Chemical action did not appear to play a significant role in fatigue life at a test temperature of 100° F. No significant differences in this respect were observed among the lubricants studied although it was noted that lubricants with the lowest neutralization numbers showed the best fatigue lives.

3. Failure type and appearance were the same for balls run with all the five fluids studied.

Lewis Flight Propulsion Laboratory
National Advisory Committee for Aeronautics
Cleveland, Ohio, October 9, 1957

APPENDIX A

APPARATUS AND PROCEDURE

Test Rig

As was previously stated, figure 1(a) is a cutaway view of the rolling-contact fatigue spin rig. The test specimens are the two balls revolving in a horizontal plane on the bore surface of a hardened tool-steel cylinder (fig. 1(b)). Air at pressures to 100 pounds per square inch is introduced through the nozzles to drive the balls at high orbital speeds. The nozzle system and the cylinder are held in place by upper and lower cover plates fastened by three removable bolts. The rig assembly is supported from a rigid frame by three flexible cables. In order to keep external constraints at a low value, the drive air is introduced into the rig through a 6-foot-long flexible metal hose. 4683

Operation. - The two test balls separate and maintain relative positions 180° apart above the critical frequency. A detailed analysis of the rig operation is given in reference 6.

Loading. - The only loading on the balls is that produced by centrifugal force. No contact is made with the ball test specimen except by the race cylinder at the contact ellipse. The load can exceed 700 pounds for a 1/2-inch steel ball revolving in a 3.25-inch bore race cylinder at an orbital speed of 30,000 rpm. At this speed a maximum Hertz stress of approximately 750,000 pounds per square inch compression will be developed at the center of the contact ellipse.

The introduction of fluid lubricant was accomplished by introducing droplets of the lubricant into the drive airstream between the guide plates (fig. 1(a)). The rotating airstream atomizes the droplets and carries the lubricant to all surfaces. Lubricant flow rate is controlled by regulating the pressure upstream of a long capillary tube. The pressure drop through the capillary was sufficient to give excellent control of the flow for small flow rates. The lubricant flow rate used in this series of tests was approximately 15 milliliters per hour.

Instrumentation. - Three major instrumentation systems provide for speed measurement and control, temperature measurement and control, and failure detection and shutdown.

Orbital speed of the balls is measured by counting the pulses from a photoamplifier on an electric tachometer. The pulses are generated by the two test balls interrupting a light beam focused on the photocell. A voltage proportional to the frequency of the photocell output is fed into an electronic controller which automatically regulates the drive air

pressure to maintain the desired orbital ball speed. Temperature is measured with an iron-constantan thermocouple which is in contact with the top of the race cylinder (fig. 1(a)). This is the closest practical location of the thermocouple with relation to the ball running track. A calibration with a thermocouple in the airstream surrounding the balls showed a variation of less than 2 percent of the test temperature. It can be assumed that the race, balls, and surrounding air are all maintained within a narrow temperature range. A second thermocouple contacting the cylinder top provides the signal for the automatic temperature controller. For tests at elevated temperatures, this controller blends room-temperature air with air heated by a 25-kilowatt heater in the proportion necessary to maintain the desired test temperature.

Failures are detected by comparing the amplified signal from a velocity vibration pickup (attached to the rig, see fig. 1(a)) against a predetermined signal level preset on a meter relay. The large vibration amplitude resulting from a ball or cylinder fatigue spall trips the meter relay and results in shutdown of the test and all instrumentation.

Air supply. - The drive air is dried to less than 30-percent relative humidity and then is filtered before being used in the rigs. A pressure of 125 pounds per square inch is maintained by a central centrifugal compressor system.

Test Specimens

Cylinders. - The dimensions of the test cylinders are as follows: outside diameter, 4.750 inches; length, 3.00 inches; initial nominal inside diameter, 3.250 inches. The bore surface finish was 2 to 3 microinches for all cylinders. Roundness of the bore was held to 0.0001 inch and bore taper was held to a maximum of 0.0003 inch. Hardness measurements were taken on the cylinder ends. Each cylinder was uniform within two hardness numbers, although average hardness varied from Rockwell C-60 to C-64 for different cylinders.

Between 10 and 15 tests may be run on a bore surface. The bore is then reground to 0.060 inch larger and refinished. Since this new surface is about 0.022 inch below the location of the maximum shear stress of the previous tests, the effects of prior stressing are considered to be negligible. Failure positions on one cylinder surface do not correlate with failure positions of the previous test surface.

Test balls. - By taking advantage of the fact that a rotating body free to adjust itself will rotate about the axis of maximum rotational inertia, test balls may be modified so that they will rotate about any fixed axis. All test balls in this series of tests were air-melt AISI M-1 tool steel hardened to Rockwell C-63 to C-64. The axis of rotation of each ball

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was preselected by grinding two diametrically opposed 3/16-inch flats. This facilitated preinspection of the running track and restarting the unfailed balls. The axes were selected in a random manner in order to reduce the effect of fiber flow orientation previously reported in reference 10.

Pretest Inspection

Cylinders were given dimensional surface-finish and hardness inspections. This was followed by a magnetic particle inspection for both cracks and large subsurface inclusions and a visual inspection for deep scratches and other mechanical damage.

All test balls were weighed and given an inspection at a magnification of 36. The presence of excessive scratches or pitting, and any cracks, laminations, or flat spots was noted in a permanent record.

Prior to inspection and use, test specimens were flushed and scrubbed with 100 percent ethyl alcohol and clean cheesecloth. This procedure left a thin film of grease on the surface, but this was considered desirable to minimize possible corrosion.

Starting and Running Procedure

The rig and test specimens were cleaned and assembled with care to prevent scratching of the bore surface. The bore surface and test balls were coated with the test lubricant. The rig was brought up to operating speed as rapidly and as smoothly as possible. The rig could then be switched to automatic speed control or left on manual control. On manual control the rig speed must be corrected at intervals to compensate for the speed increase due to run-in of the test specimens. Run-in is rapid for the first few minutes and is practically complete after the first 2 hours. Rig temperature reached equilibrium at the 100° F test temperature in about 3 minutes. The control system automatically maintained the desired temperature.

Speed, temperature, and oil flow were monitored regularly. Speed, temperature, air pressure, and vibration levels were recorded at each reading. The test was continued until a predetermined number of stress cycles had been exceeded or until a ball or race failure actuated the meter relay which shut down the rig.

Stress Calculations

With ball weight, speed, and orbital radius of rotation of the test balls known, the load can be calculated. The stress developed in the

contact area was calculated from the load and specimen geometry by using the modified Hertz formulas given in reference 7.

Post-Test Inspection

After failure or a predetermined limiting number of stress cycles, the ball running tracks were examined with a microscope at a magnification of 36. Any abnormalities which correlated with the fatigue life results were noted and followed up with a further metallographic investigation. Specimens were mounted in Bakelite, ground to the desired cross section, and polished and etched to reveal subsurface metallographic structure. Some inspections of the running track surfaces after a polish with diamond dust were made to determine corrosion pitting and its relation to crack formation.

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Lubricants

Data on lubricant properties (table I) were obtained with unused lubricant before the tests were started and also unused lubricant after the conclusion of the tests. This was done because unpublished work indicated that some lubricants showed storage instability.

APPENDIX B

PRESSURE-VISCOSITY DATA

The common expression of fluid viscosity refers to that viscosity which is measured at a given temperature and at atmospheric pressure. However, in a rolling-contact bearing the lubricating fluid is subjected to much higher pressures up to the maximum contact pressure existing between the rolling elements. If lubricant viscosity has a significant effect upon rolling-contact fatigue life, the viscosity at the pressure in the contact zone is the important variable to be considered, not the commonly measured viscosity at 1 atmosphere. 4683

The measurement of the viscosity of a fluid at high pressures requires special equipment able to maintain the pressure and take readings at that pressure. Thus, the pressure-viscosity data available on lubricant fluids are limited. No data were taken on the samples of the actual fluids used in this investigation, but reference 5 contains data on a large number of fluids. Data for those fluids having the closest similarity to the fluids used in the fatigue life tests of this report were selected to correlate pressure viscosity with fatigue life. These data are reproduced in figure 4. In all cases the base stock was the same and the atmospheric pressure viscosity was as close as possible to the 10-centistoke weight used in this investigation. However, the rate of increase of viscosity with pressure is the important consideration, and this rate is reasonably constant for all weights of fluids of a given base stock.

The pressure-viscosity relation can be represented approximately by the form

$$\eta_P = \eta_o 10^{\alpha P}$$

where

η_P viscosity at pressure, centipoises

η_o viscosity at 1 atmosphere, centipoises

α constant, psi^{-1}

P pressure, lb/sq in.

or

$$\log \eta_P = \log \eta_o + \alpha P$$

Thus, α is the slope of the plot of $\log \eta$ against pressure in figure 4. This slope is reasonably constant over the entire pressure range. By approximating the average slope of the line, an average value for α can be computed which is a good approximation of α at any pressure. The value of α is obtained from the relation

$$\alpha = \frac{\log \eta_P - \log \eta_o}{P}$$

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For example, with the mineral oil,

$$\alpha = \frac{\log 9000 - \log 64}{40,000}$$

$$= 5.38\times10^{-5}\ \text{psi}^{-1}$$

These values are tabulated in table I and are used as the abscissa in figure 5. No data were available for the adipate-base fluid.

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TABLE I. - LUBRICANT PROPERTIES

Viscosity, centistokes						Viscosity index			Neutralization number		Pressure-viscosity coefficient[b], psi^{-1}
Before		After (a)		Average		Before	After (a)	Average	Before	After (a)	
100° F	210° F	100° F	210° F	100° F	210° F						
Methyl silicone											
10.27	4.42	10.34	4.43	10.31	4.42	270	270	270	0.05	0.05	8.35×10^{-5}
Paraffinic-base mineral oil											
10.14	2.49	10.37	2.54	10.26	2.52	70.5	74.3	72.4	0.05	0.05	5.38×10^{-5}
Water-soluble polyalkylene glycol											
8.62	2.27	8.75	2.28	8.69	2.28	75.7	72.7	74.2	1.03	1.05	3.28×10^{-5}
Di-(ethylhexyl) sebacate											
13.5	4.91					232				0.15	2.52×10^{-5}
Di-isooctyl adipate											
9.80	2.76	9.73	2.76	9.77	2.76	141	143	142	1.32	1.35	Unavailable

[a]Obtained from unused lubricants retained in storage and taken after test.

[b]Obtained from ref. 5 for lubricants similar to those used herein.

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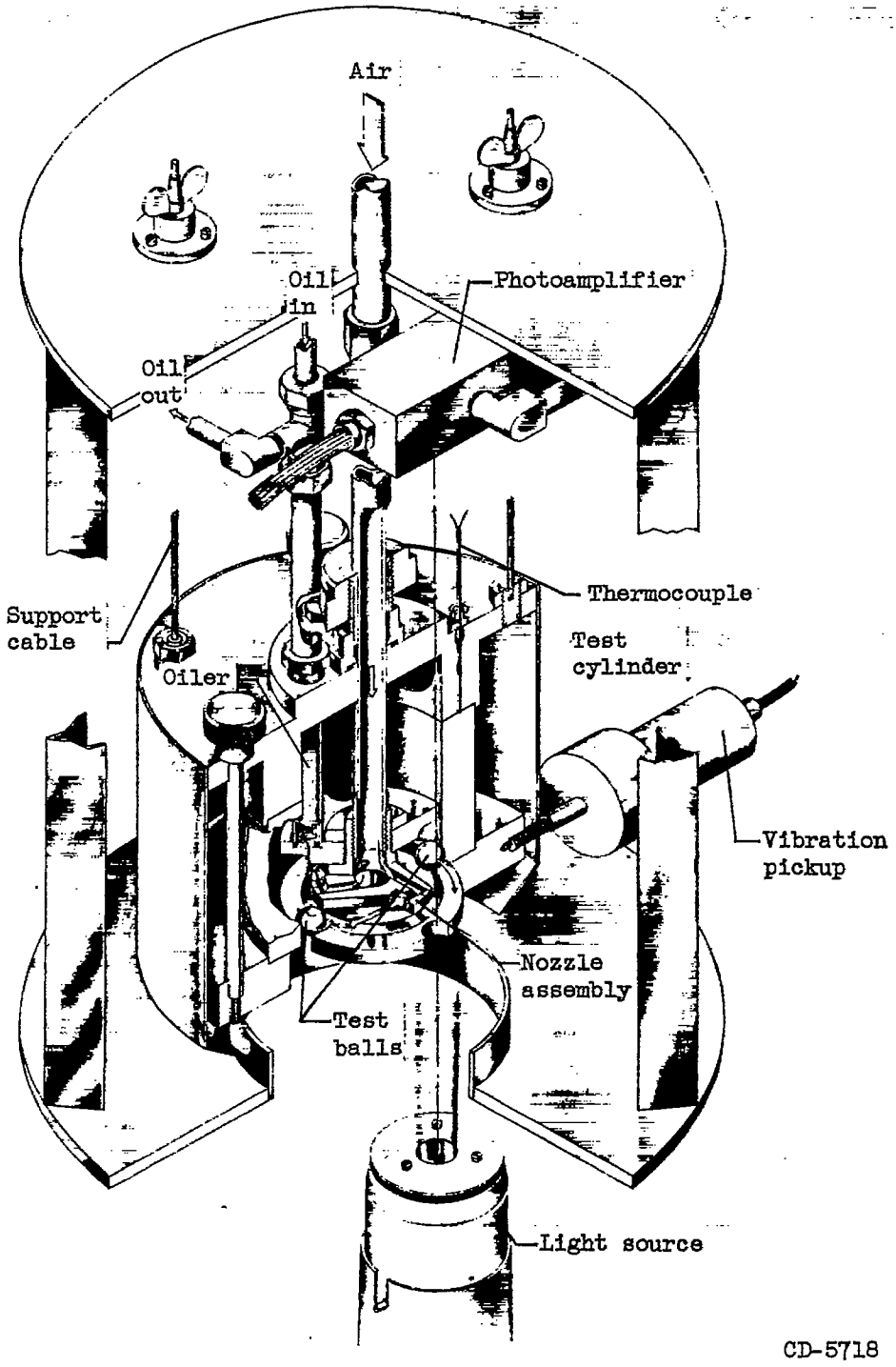

(a) Cutaway view.

Figure 1. - Rolling-contact fatigue spin rig.

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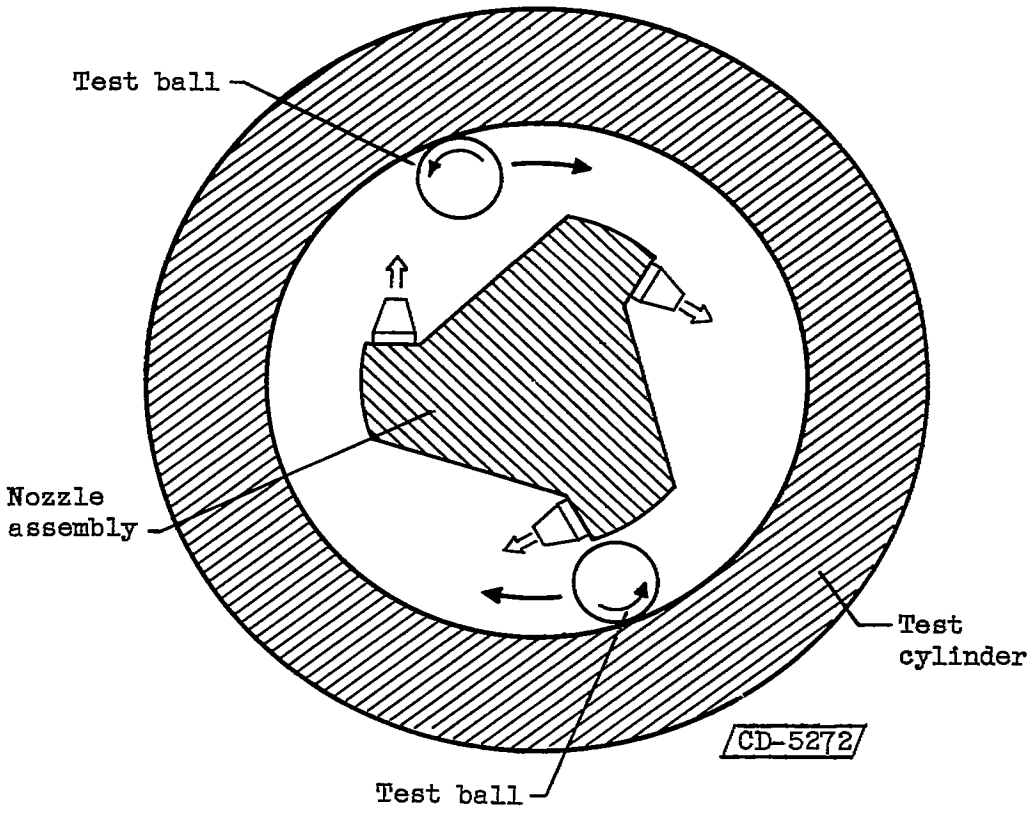
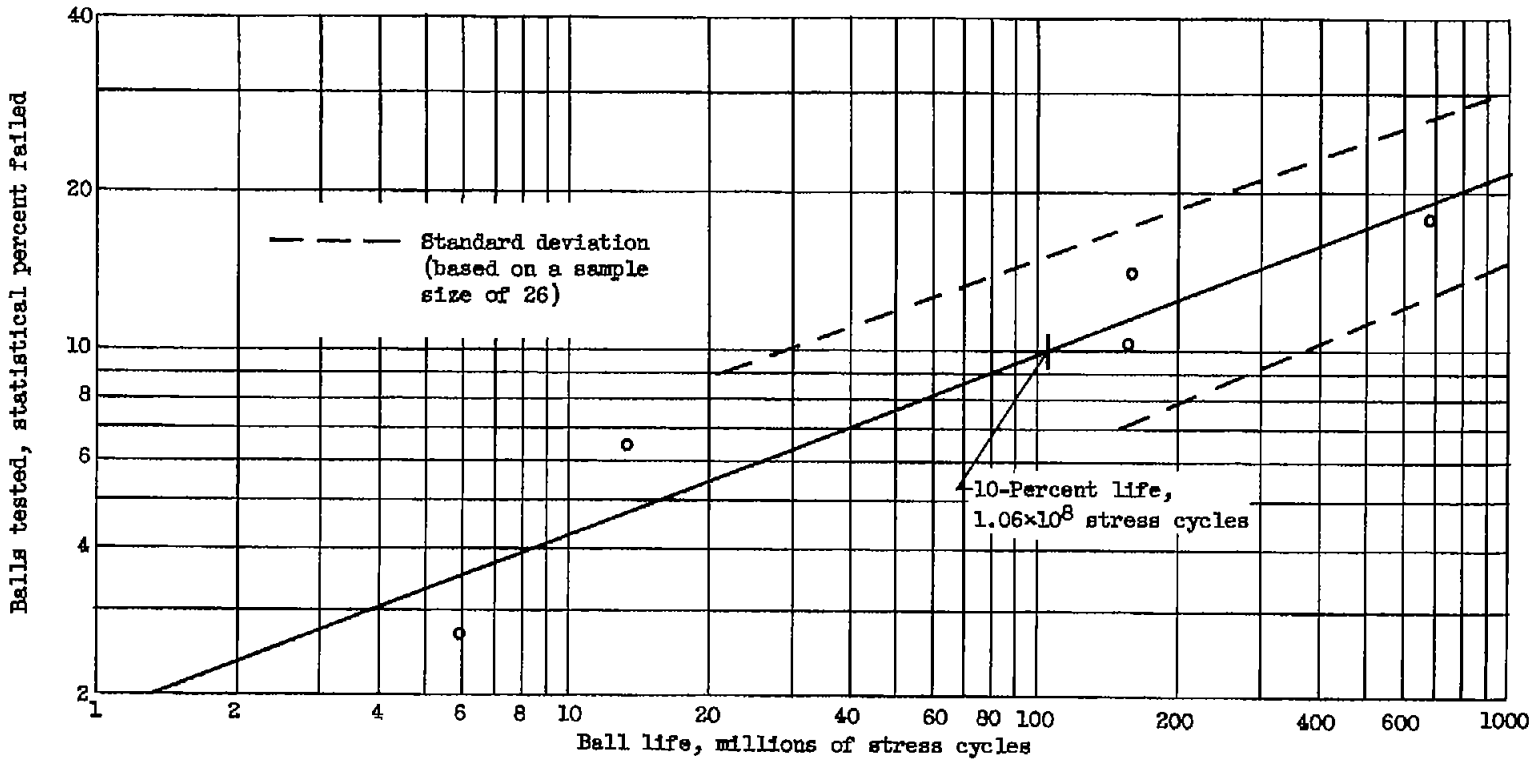
CH-3 back



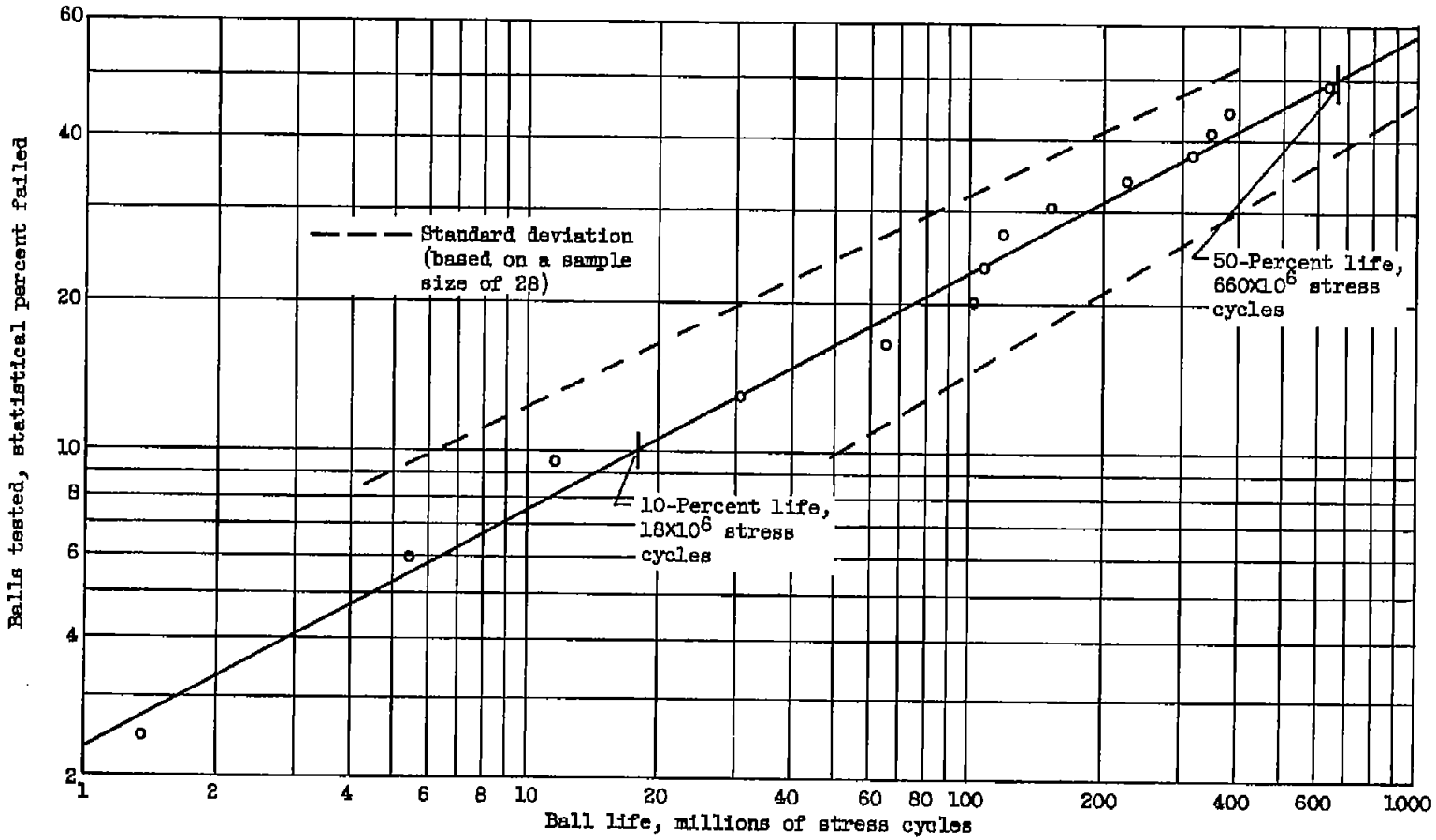
(b) Schematic diagram.

Figure 1. - Concluded. Rolling-contact fatigue spin rig.



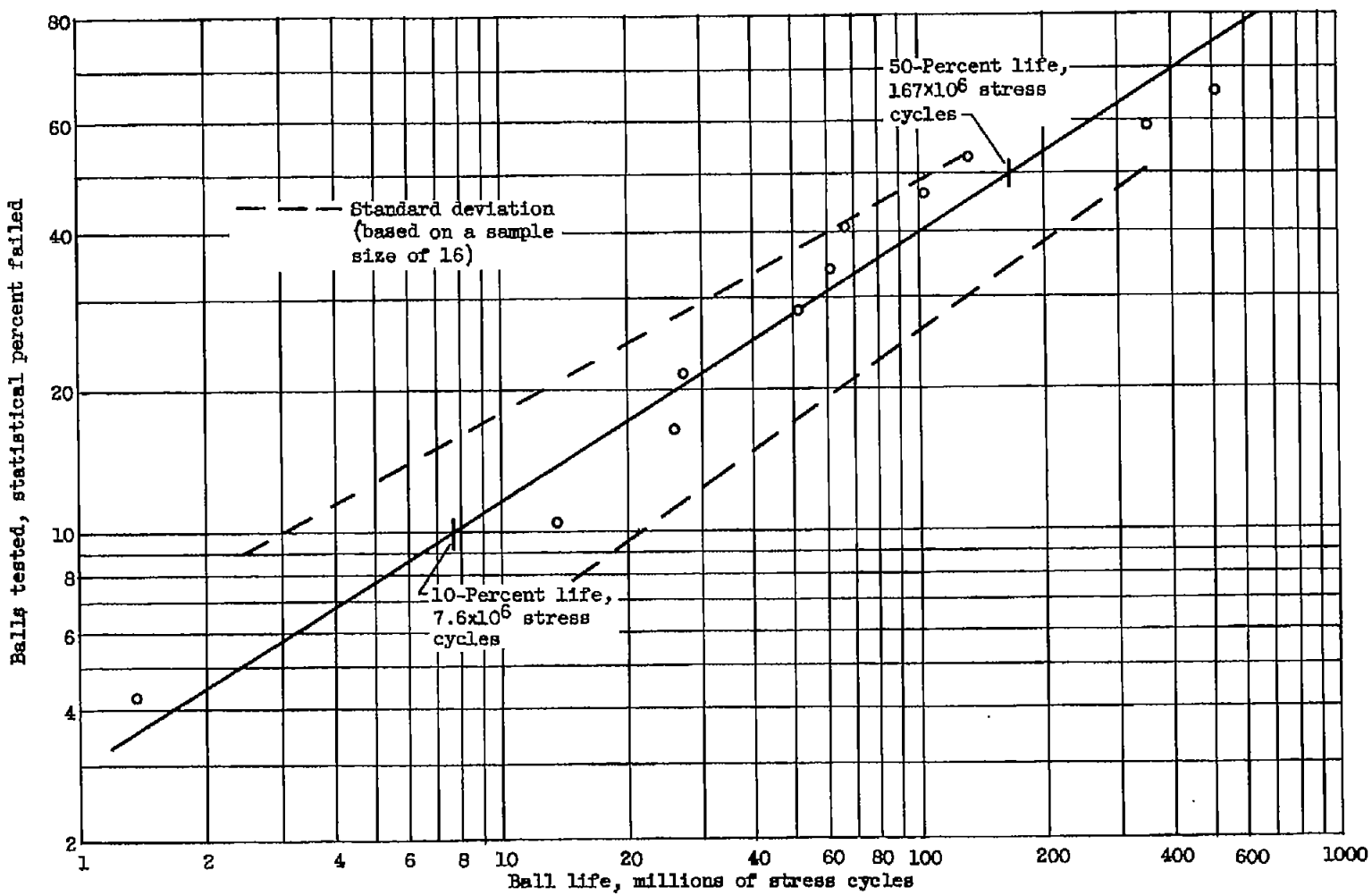
(a) Methyl silicone. Maximum Hertz compressive stress, 725,000 pounds per square inch.

Figure 2. - Fatigue life of 1/2-inch AISI M-1 tool-steel balls lubricated with various base-stock fluids. Test temperature, 100° F.



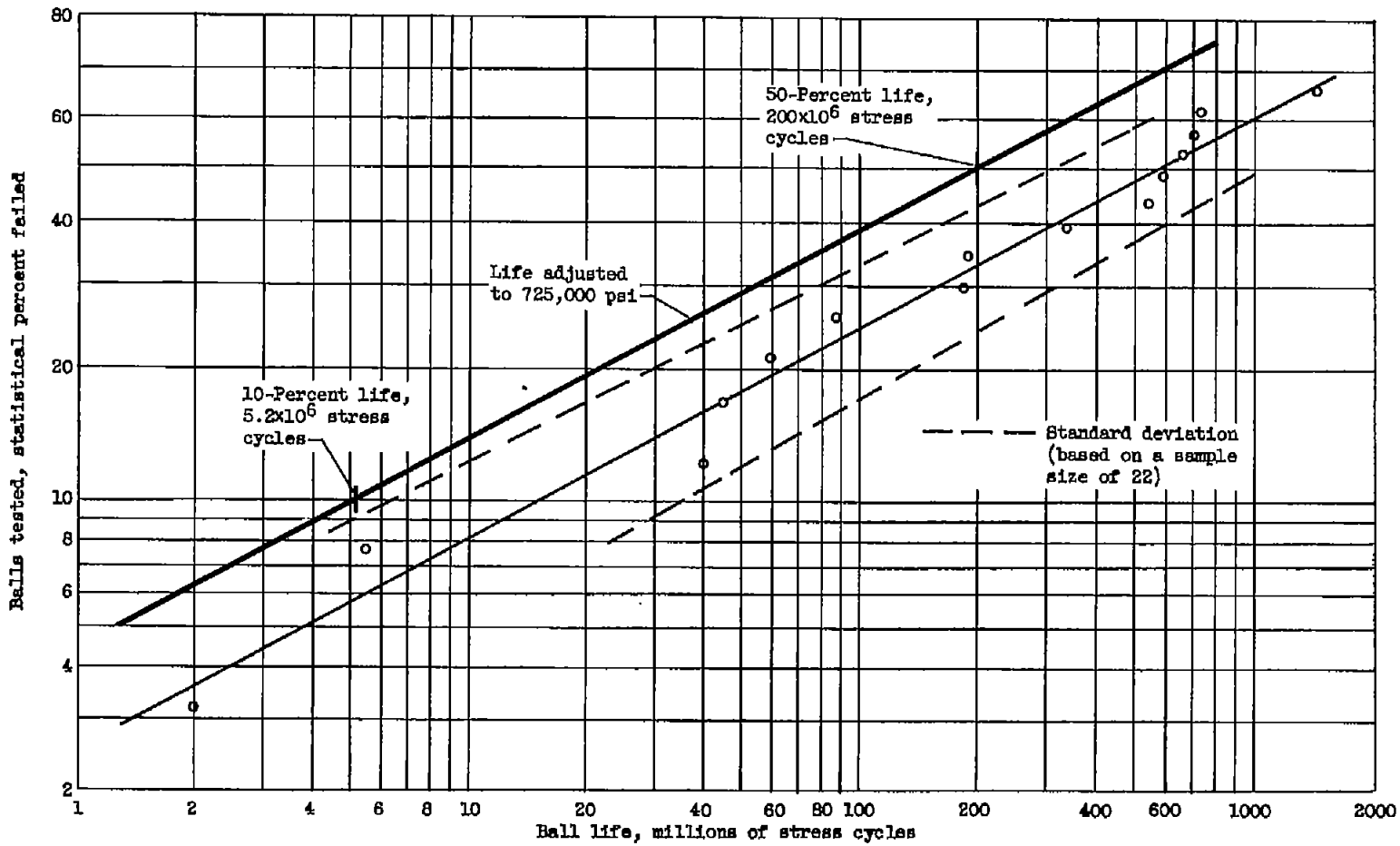
(b) Paraffinic mineral oil. Maximum Hertz compressive stress, 725,000 pounds per square inch.

Figure 2. - Continued. Fatigue life of 1/2-inch AISI M-1 tool-steel balls lubricated with various base-stock fluids. Test temperature, 100° F.



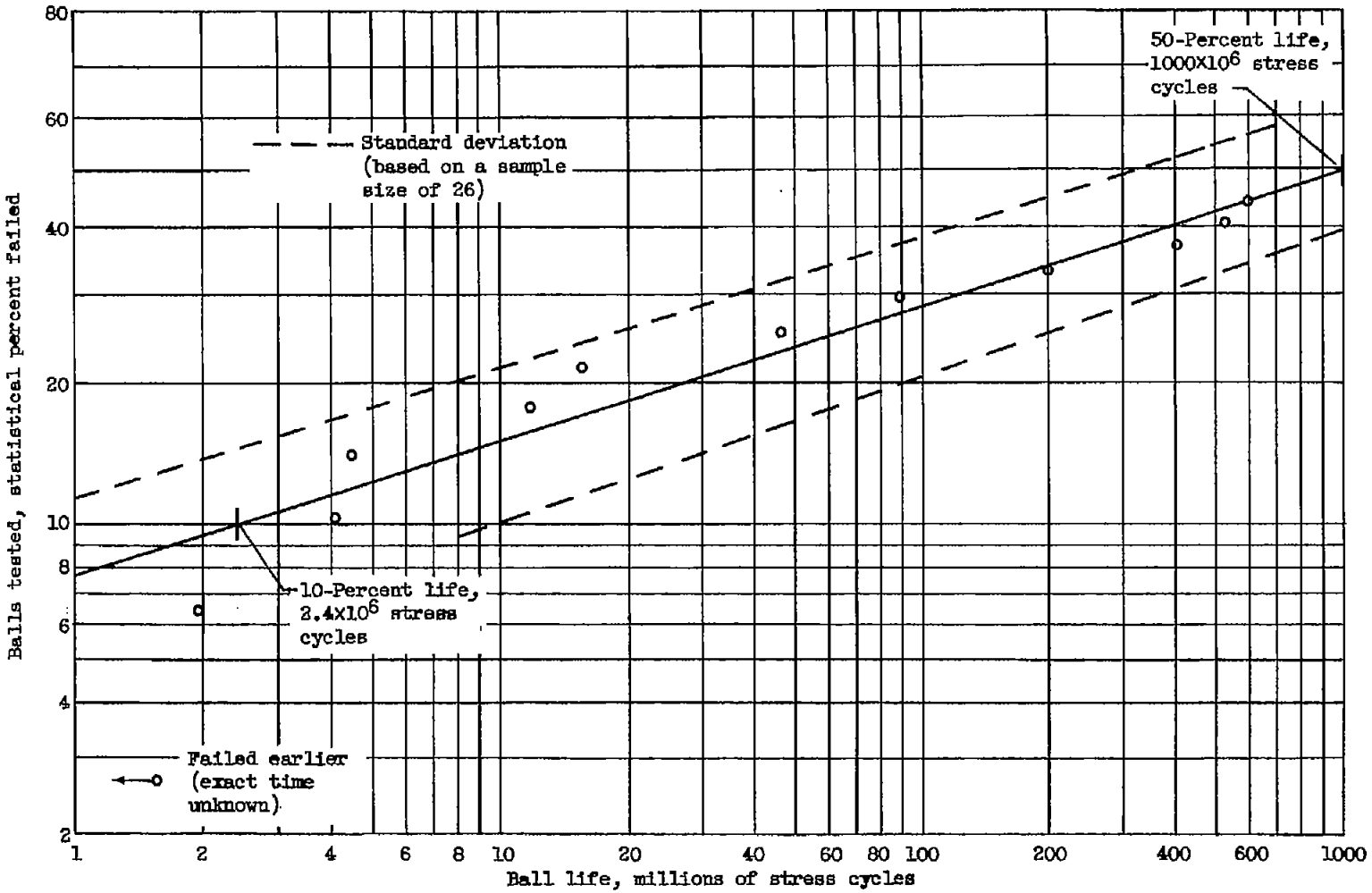
(c) Water-base glycol. Maximum Hertz compressive stress, 725,000 pounds per square inch.

Figure 2. - Continued. Fatigue life of 1/2-inch AISI M-1 tool-steel balls lubricated with various base-stock fluids. Test temperature, 100° F.



(d) Di(2-ethylhexyl)sebacate. Maximum Hertz compressive stress, 650,000 pounds per square inch.

Figure 2. - Continued. Fatigue life of 1/2-inch AISI M-1 tool-steel balls lubricated with various base-stock fluids. Test temperature, 100° F.



(e) Diisooctyl adipate. Maximum Hertz compressive stress, 725,000 pounds per square inch.

Figure 2. - Concluded. Fatigue life of 1/2-inch AISI M-1 tool-steel balls lubricated with various base-stock fluids. Test temperature, 100° F.

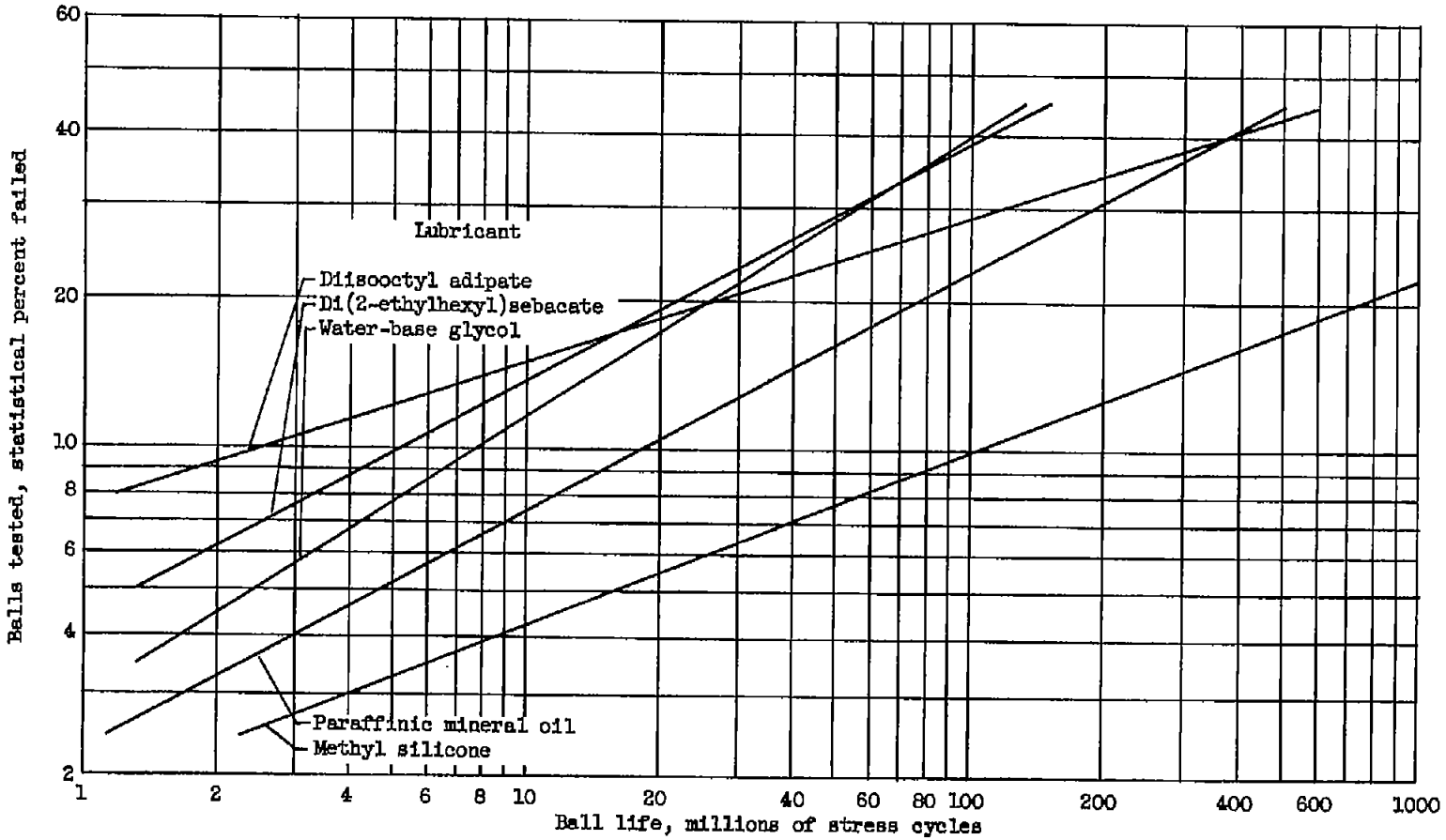


Figure 3. - Summary of fatigue lives of 1/2-inch AISI M-1 tool-steel balls lubricated with various base-stock fluids. Test temperature, 100° F; maximum Hertz compressive stress, 725,000 pounds per square inch.

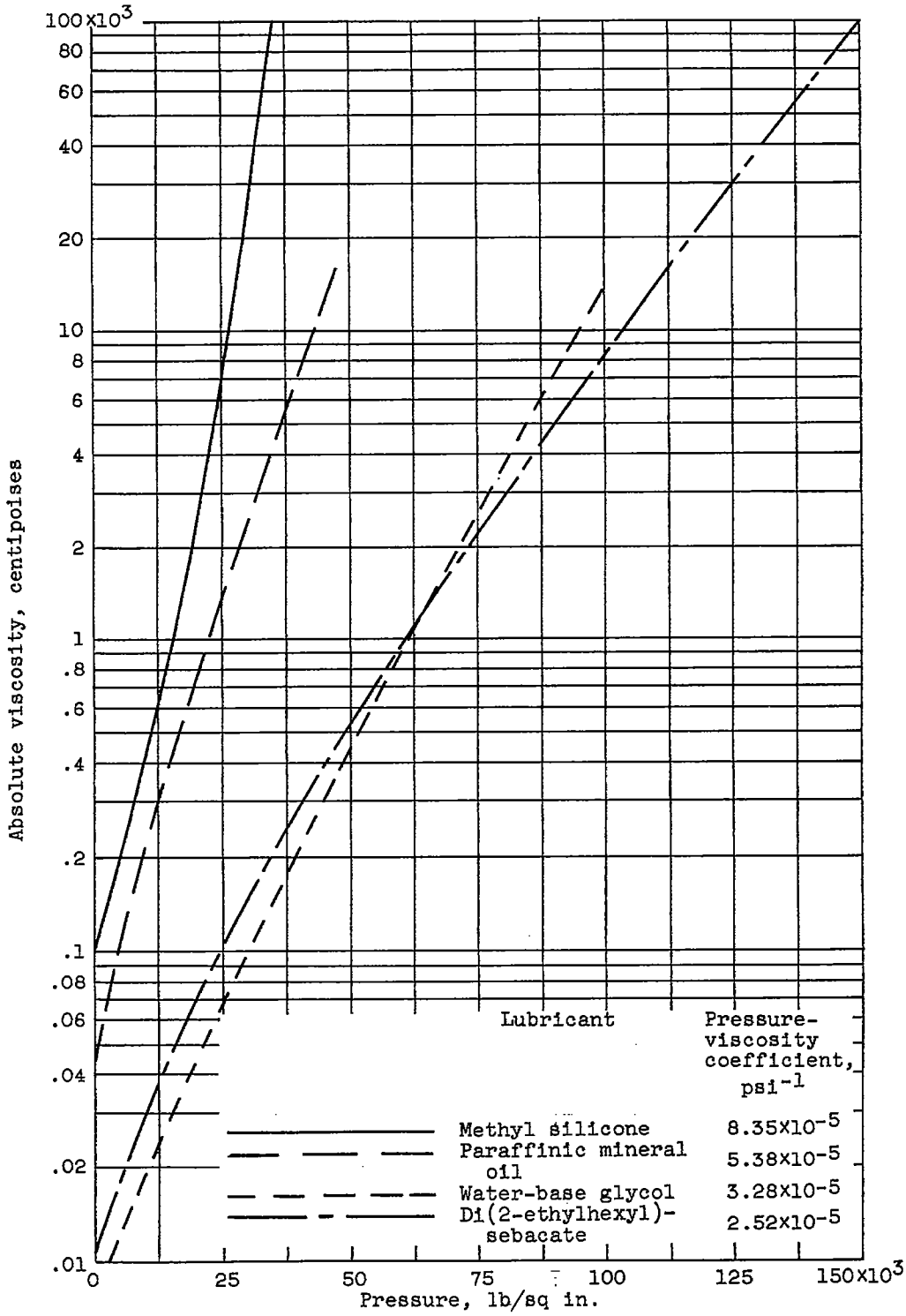


Figure 4. - Viscosity as function of pressure for various lubricants (ref. 5). Test temperature, 100° F.

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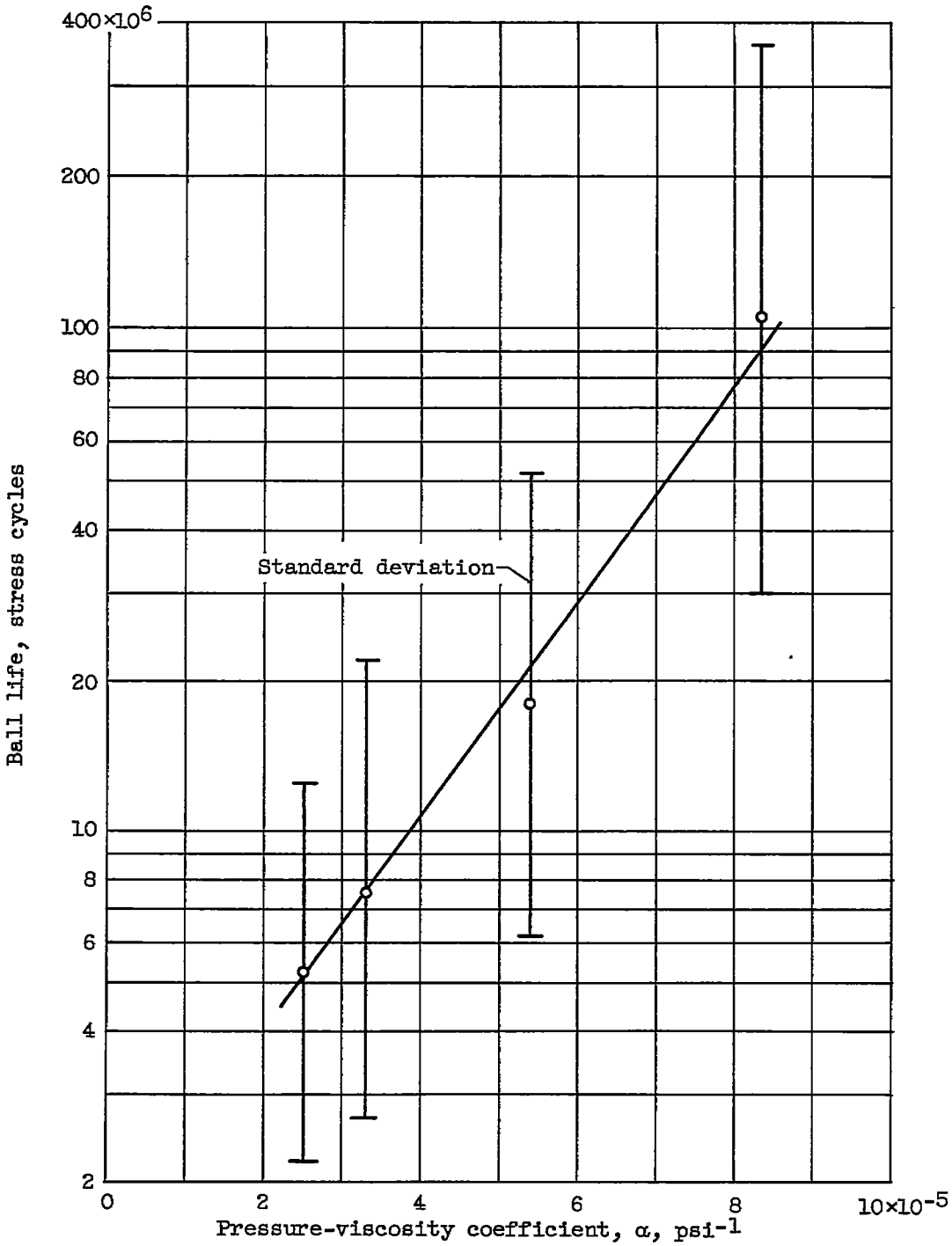
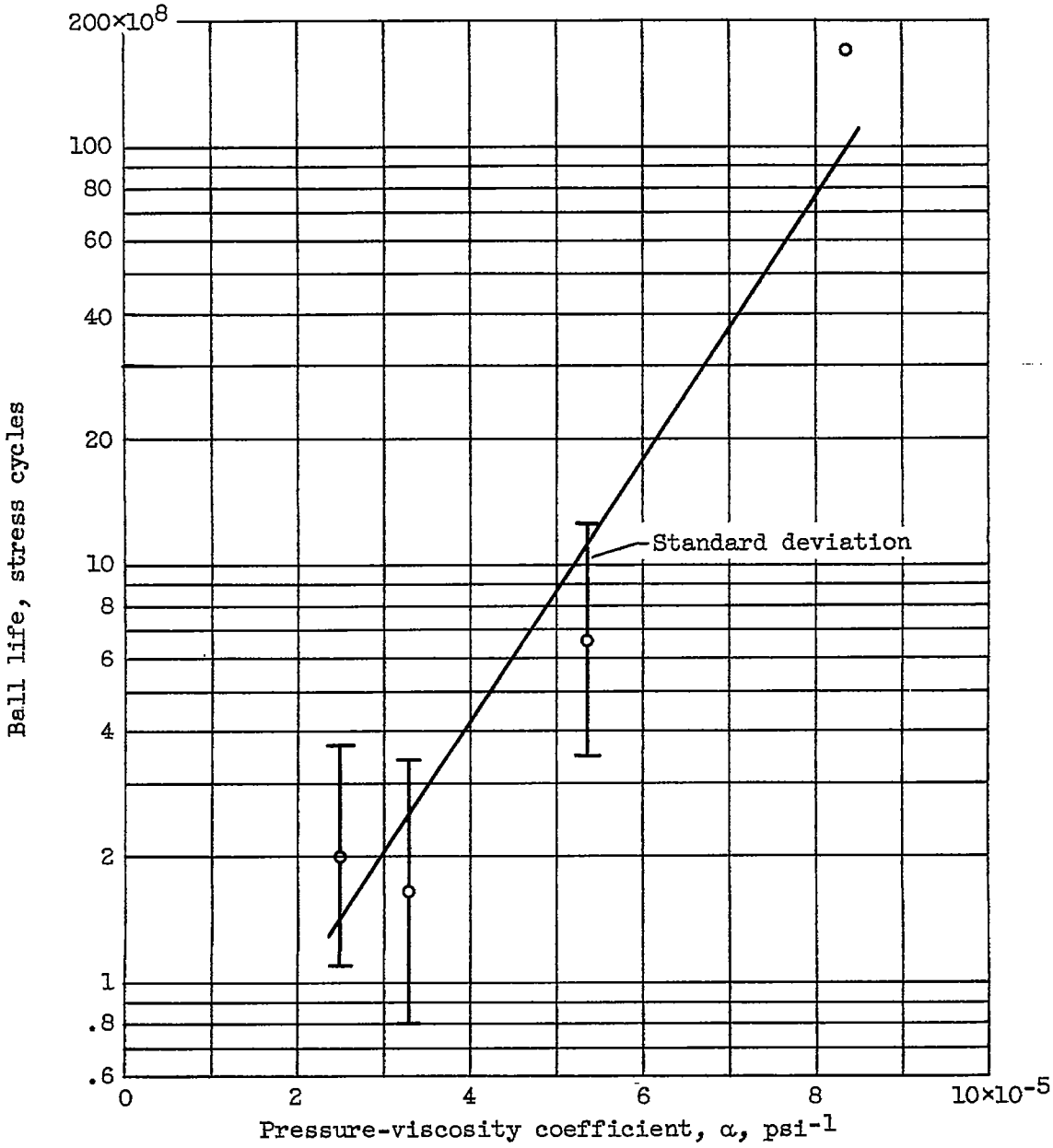


(a) 10-Percent life.

Figure 5. - AISI M-1 tool-steel ball life as function of pressure-viscosity coefficient. Test temperature, 100° F; maximum Hertz compressive stress, 725,000 pounds per square inch. $\eta_p = \eta_o \times 10^{\alpha p}$.



(b) 50-Percent life.

Figure 5. - Concluded. AISI M-1 tool-steel ball life as function of pressure-viscosity coefficient. Test temperature, 100° F; maximum Hertz compressive stress, 725,000 pounds per square inch. $\eta_P = \eta_0 \times 10^{\alpha P}$.