NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

TECHNICAL NOTE 4101

EFFECT OF LUBRICANT VISCOSITY ON ROLLING-CONTACT FATIGUE LIFE

By Thomas L. Carter

Lewis Flight Propulsion Laboratory
Cleveland, Ohio

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SUMMARY

A series of rolling-contact fatigue tests was conducted in a bench rig developed at the NACA Lewis laboratory. Four paraffin-base mineral oils of varying viscosity (at atmospheric pressure) were used as lubricants. Ball specimens were AISI M-1 tool steel (air melt). Test temperature was 100° F, and a calculated Hertz compressive stress level of 725,000 psi was maintained.

A continuous trend toward longer life was observed with increasing lubricant viscosity over the range studied (5 to 120 centistokes at 100° F). This trend holds at any percentage of specimens failed. The life scatter remained about constant at each viscosity level studied.

A plot of log of life at any survival level against log of lubricant viscosity produces a reasonably straight line. This line indicates that rolling-contact fatigue life is a function of approximately the 0.2 power of lubricant viscosity.

INTRODUCTION

One of the primary considerations in developing a bearing capable of sustaining the high temperatures encountered in present and anticipated aircraft gas-turbine engines is the rolling-contact fatigue life of the bearing elements. Aside from bearing design and loading, the fatigue life is affected by the materials used in the bearing elements and the substance used to provide lubrication.

During high-speed rolling contact, the lubricant, in addition to reducing sliding friction and cooling the bearing, affects the pressure distribution in the contact zone through hydrodynamic action. The theoretical calculations of stress 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 general discussion will illustrate the point.

Figure 1(a) is a cross section of the contact zone and pressure distribution for a ball running without a lubricant. The integral of pressure and the contact area would be the total ball loading; thus, for a given load the maximum pressure (i.e., compressive stress borne by the ball) would depend on the size of the contact area. Figure 1(b) shows how the presence of a lubricant extends the effective load-carrying area and thus reduces the maximum pressure. The lubricant film ahead of the ball must be reduced to the minimum film thickness in the contact zone. For a given speed this requires a proportional rate of shear in the lubricant ahead of the ball to remove the excess fluid from the ball path. The force necessary to maintain this rate of shear depends upon the viscosity of the fluid. Thus a more viscous fluid would require a greater shear force. Since this shear force is resolved from the pressure between the adjacent rolling-element surfaces, a portion of the ball load is borne by that portion of the fluid which is outside of the contact area that would exist if no lubricant were present. Thus the effective contact area is increased and the maximum contact pressure is reduced. For a given rolling speed the maximum pressure would decrease with increasing lubricant viscosity. Since life is inversely proportional to the tenth power of maximum stress, the effect of lubricant viscosity in high-speed rolling contact could be significant. A two-dimensional mathematical analysis of this effect is given in reference 1.

The literature available on this role of the lubricant in rolling-contact fatigue is limited. In references 2 and 3 the test temperatures are varied to vary the viscosity. Metallurgical transformations in the materials are significant within the temperature range necessary to produce an adequate viscosity range (refs. 4 and 5). Any chemical reactions at the contact surface and in the lubricant would be influenced by temperature.

Reference 6 shows a linear increase in life with lubricant viscosity. Though not stated, it appears that the lubricants are all paraffin-base stocks and the tests were conducted at room temperature. This is a more controlled evaluation of the effect of lubricant viscosity on rolling-contact fatigue life, although the low scatter in life (2:1 or less) is not characteristic of bearing-fatigue data and may indicate overloading and crushing of the balls.

Another method of varying viscosity is to use lubricants of different base stock. This also introduces variables other than normal (atmospheric pressure) viscosity such as pressure viscosity effects and chemical reactions in the lubricant. Normal viscosity is that viscosity which is measured by standard tests at atmospheric pressure. Viscosity tends to increase significantly at the high pressure existing in bearing lubricating films (ref. 7). This effect is of varying degree in various base stocks. For this reason different base stocks may have different effects on fatigue life even though the measured (normal) viscosity is constant.

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The best method of observing the effects of lubricant viscosity on rolling-contact fatigue life would probably be by use of a series of fluids of the same base stock (i.e., same temperature and pressure viscosity characteristics) but with a range of viscosity as measured at atmospheric pressure. All other test conditions could then remain constant. The controlled variable would thus be the normal lubricant viscosity.

With this objective in mind, a group of paraffin-base mineral oils was selected. Properties of this group of oils are given in table I. Rolling-contact fatigue-life data were secured for each fluid under consistent test conditions.

APPARATUS

Only brief descriptions of the apparatus and procedure are given here. A more detailed presentation can be found in reference 8 and appendix A. Figure 2(a) is a cutaway view of the rolling-contact fatigue spin rig. The test specimens were two balls revolving in a horizontal plane on the bore surface of a hardened tool-steel cylinder (fig. 2(b)). Air at pressures up to 100 pounds per square inch was introduced through the nozzles to drive the balls at high orbital speeds. The loading on the balls was that produced by centrifugal force, and the stress was calculated according to the methods of reference 9. Approximately 15 milliliters per hour of lubricant were introduced in droplet form into the drive airstream between the guide plates. 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 ball or race failure resulted in increased vibration and hence in an increased signal from a velocity pickup attached to the rig. This signal when amplified actuated a meter relay which shut down the system.

Temperature was controlled by mixing heated air with the normal drive-air supply. This heated drive air surrounded the test balls and the inner surface of the race cylinder. The air was then exhausted over the outer surface of the race cylinder. The test temperature and temperature control signal were taken from thermocouples on the top and bottom of the race cylinder. A calibration showed that this temperature did not deviate more than 2° from the temperature of the airstream surrounding the test balls.

All ball test specimens were from the same heat of AISI M-1 air-melt tool steel and were a nominal 1/2-inch diameter. Nominal composition is given in table II. The running track on the balls was predetermined by grinding two diametrically opposed 1/8-inch flats on the ball surface. Because of the preferred moment of inertia produced, the balls would run on a track which was the great circle perpendicular to the diameter terminating at the ground flats. This procedure simplified preinspection and permitted restarting of the balls when necessary. Race cylinders were AISI M-1 vacuum-melt tool steel. The outside dimensions were 4.750 inches diameter and 3.000 inches long, and the bore diameters ranged from 3.310 to 3.550 inches.

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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 60. 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 ethyl alcohol and clean cheesecloth. During storage they were protected by a corrosion-resistant oil film. Care was taken during rig assembly not to scratch the running surfaces. The bore surface and test balls were coated with lubricant during assembly.

The rig was brought up to operating speed as rapidly and smoothly as possible. About 3 minutes were required for the hot airstream to heat the 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 60 was made to observe track conditions. Failure data were plotted on Weibull paper, which is a plot of the log of the reciprocal of the portion of the sample surviving against the log of the stress cycles to failure. Lines were fitted to these data by the least-squares method. The method of presenting the data is discussed in detail in appendix B.

RESULTS AND DISCUSSION

Life Data with Four Mineral Oils

The data produced in this program represent the effect of lubricant viscosity on rolling-contact fatigue life. Four common paraffin-base mineral oils with viscosities ranging from 5 to 119 centistokes at 100° F were used. With these life data, rolling-contact fatigue life and lubricant viscosity as measured at atmospheric pressure were correlated.

A Weibull plot is presented in figure 3(a) for AISI M-1 tool-steel balls lubricated with 1005 mineral oil (5.1 centistokes) at an ambient temperature of 100° F and a centrifugal-force loading that produced a 725,000-psi maximum Hertz compressive stress and a maximum shear stress of 225,000 psi 0.009 inch below the running track. Plots for mineral oils 1010 (10.3 centistokes), 3042 (24.2 centistokes), and 1065 (119.1 centistokes) at the same test conditions are shown in figures 3(b), (c), and (d), respectively. The life plots all have the same general appearance, but they shift toward the right (i.e., life increases) with increasing viscosity, while the slopes (i.e., scatter) remain about the same.

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The results of figures 3(a) to (d) are summarized in figure 3(e). The shift toward longer rolling-contact fatigue life with increased normal (atmospheric pressure) viscosity of the lubricating fluid film is easily observed. This effect exists at any level of survival (i.e., long or short lives). The life scatter (i.e., Weibull slope) is about the same for all four curves; thus, the only significant change in appearance of the Weibull plot is the shift toward longer life as the lubricant viscosity is increased.

Fatigue spalls for all four test groups were consistent in appearance with each other and with spalls produced in the fatigue spin rig with other fluid lubricants. They closely resembled fatigue failures characteristic of full-scale bearings, being limited in area and depth and originating in subsurface shear cracking. A ball fatigue spall is compared with one from a full-scale bearing inner race in figure 4. The groups of balls in this investigation had the metallographic transformation in the subsurface zone of maximum shear stress characteristic of specimens subjected to rolling-contact stressing (ref. 4).

No significant conditions unique to this group of tests were observed in the post-run surface examination. All balls had the characteristic darkening of the running track observed with most fluid lubricants and AISI M-1 tool steel. Chemical activity resulting in corrosion of the track surfaces did not appear to be present to any significant degree.

The variable studied in this investigation was the lubricating fluid viscosity as measured by standard methods at atmospheric pressure. The lubricant base stock and all test conditions remained constant. The increase in fatigue life with lubricant viscosity noted is consistent with the effect reported in references 2, 3, and 6. However, test conditions were not held constant in references 2 and 3, where temperature was varied over a wide range to achieve the desired viscosity range. The high ball loading of reference 6 did not produce any appreciable scatter; hence there is questionable similarity of the test conditions to those of a full-scale bearing where scatter is much higher.

Since there is a continuous trend toward longer rolling-contact fatigue life with increased lubricant viscosity within the range studed (5 to 120 centistokes), it is possible to plot a curve for this relation. Figure 5 is a plot of the log of fatigue life (10% and 50% failures) against the log of viscosity. This plot is almost linear; straight lines have been drawn for 10- and 50-percent failures by the least-squares method. These lines resolve with the following relation: 4615

$$L = K\mu^{n}$$

where L is fatigue life in millions of stress cycles, and μ is lubricant viscosity in centistokes at 100° F. Then

$$10\%\ \text{fatigue life} = 11.3\mu^{0.237}$$

$$50\%\ \text{fatigue life} = 245\mu^{0.167}$$

These relations indicate that the rolling-contact fatigue life is a function of about the 0.2 power of lubricant viscosity. This is in contrast to the results of references 2, 3, and 6, where the life is given a linear relation to viscosity, although in all cases the rate of change of life with viscosity is about the same over the 5- to 120-centistoke range.

A more complex but more versatile equation can be fitted to the results for life against lubricant viscosity from consideration of the extreme-value theory. Since each failure results from the weakest point on the running track, all other points on the track are of necessity stronger; thus the fatigue lives observed are a series of extreme values for all the infinitesimal areas of the running tracks. An extreme-value analysis results in an equation of the form

$$Q = e^{-\left(\frac{L\times 10^{-8}}{5+0.42\sqrt{\mu}}\right)^{(0.625+0.007\sqrt{\mu})}}$$

where Q is the probability of survival. In this form the relation can be used to calculate the life as a function of viscosity at any level of survival. This curve is plotted for 10- and 50-percent lives in figure 5. This form would, upon extrapolation, give a finite fatigue life at zero viscosity where the simple power function would not. For this reason it is considered to be a more reasonable form of the life-viscosity function, although a greater range of viscosity would be needed to establish it accurately at the low viscosity levels.

Discussion

The results of this investigation indicate that lubricant viscosity has a significant effect on rolling-contact fatigue life. Since viscosity is greatly increased at the high pressures existing in bearings, the pressure-viscosity and compressibility characteristics of the fluid may also have a very significant influence on life. In this series of tests, characteristics of the fluid were held constant by using the same base stock in order to evaluate the effect of atmospheric pressure viscosity (i.e., molecular weight) of the fluid, but a systematic evaluation of the effect of base stock (i.e., pressure-viscosity characteristics) on fatigue life is necessary in order to complete the evaluation of the effect of viscosity on rolling-contact fatigue life.

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 relation of life against percent survival in the Weibull plot. At the same time, the expense and duration of each test limit the practical number of specimens that can be evaluated. Confidence limits for the data in figure 3 were calculated by the method of Lieblein (ref. 10) and are presented in figure 5. For the sample sizes of 15, 18, 19, and 19 balls used to produce figure 3, these confidence limits are wide in relation to the range observed between the results for the different viscosity fluids. However, if no effect due to lubricant viscosity exists, the probability that the four life plots will fall in order of ascending viscosity is only 1 in 24. This is so because each of the lines was calculated by a least-squares best-fit technique so that they are unbiased. Thus, the results in figure 5 have a 96-percent probability not to have been caused by chance.

SUMMARY OF RESULTS

Rolling-contact fatigue studies were made in the rolling-contact fatigue spin rig on 1/2-inch-diameter balls of AISI M-1 air-melt tool steel at a test temperature of 100° F and a maximum Hertz compressive stress of 725,000 psi. A series of paraffin-base mineral oils identical in all respects except viscosity were used as lubricants. The results of these studies are as follows:

1. A continuous trend toward longer rolling-contact fatigue life is shown with increasing lubricant viscosity throughout the range studied (5 to 120 centistokes at 100° F). This trend holds at any percentage of failed specimens. Scatter remains about the same for all lubricants observed.

2. A plot of the log of life at any survival level against log of lubricant viscosity produces a reasonably straight line. This line indicates that fatigue life is a function of approximately the 0.2 power of lubricant viscosity.

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APPENDIX A

APPARATUS AND PROCEDURE

Test Rig

A cutaway view of the rolling-contact fatigue spin rig is shown in figure 2(a). The test specimens are the two balls revolving in a horizontal plane on the bore surface of a hardened tool-steel cylinder (fig. 2(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 constraint at a low value, the drive air is introduced into the rig through a 6-foot-long flexible metal hose.

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 8.

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 3.5-inch-bore race cylinder at an orbital speed of 30,000 rpm. At this speed a maximum Hertz stress of approximately 750,000-psi 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. 2(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 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 when the two test balls interrupt a light beam focused on the photocell.

A voltage proportional to the frequency of the photocell output is fed into a Swartwout 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. 2(a)). This is the closest practical location of 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° 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. This controller blends room-temperature air with air heated by a 25-kilowatt heater in the proportion necessary to maintain the temperature of the drive-air supply at the proper level for the desired test temperature. This drive air surrounds the test balls and is then exhausted over the outer surface of the race cylinders.

Failures are detected by comparing the amplified signal from a velocity vibration pickup (attached to the rig, see fig. 2(a)) with a predetermined signal level present 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 filtered before being used in the rigs. A pressure of 125 pounds per square inch is maintained by a central centrifugal compression 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 increasing to 3.550 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 to a maximum of 0.0003 inch. Hardness measurements were taken on the cylinder ends. Each cylinder was uniform within 2 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 0.060 inch larger and refinished. This new surface is about 0.022 inch below the location of the maximum shear stress of the previous tests, and so the effects of prior stressing are considered negligible. Failure positions on one cylinder surface do not correlate with failure positions of the previous tests surface.

Test lubricants. - The lubricants used were all paraffin-base mineral oils. The only difference among them was normal viscosity. Normal viscosity is that viscosity which is measured by standard tests at atmospheric pressure. Viscosity tends to increase significantly at the high pressures existing in bearing lubricating films. This effect is of varying degree in different base stocks. The lubricating-oil characteristics for the fluids used is given in table I.

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. The axis of rotation of each ball was preselected by grinding two diametrically opposed 1/8-inch flats. This facilitated preinspection of the running track and restarting of the surviving balls. The axes were selected in a random manner in order to reduce the effect of fiber-flow orientation previously reported in reference 11. All test balls were air-melt AISI M-1 tool steel hardened to Rockwell C-62 to C-63. Ball material is analyzed in table II.

Procedure

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 60. 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. A very thin film of grease was left on the ball surface by this procedure, but this was considered desirable to minimize corrosion and was not heavy enough to impede surface inspection.

Assembly of rig. - 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 lubricant. The test position in the cylinder was set by loosening the collet (fig. 2(a)) and moving the nozzle assembly and the test balls axially to the test station and then retightening the collet. The rig was mounted in the support frame and then leveled. Oil lines, air supply lines, thermocouples, and the vibration pickup were connected. The final step was to check the alinement of the light beam on the photocell.

Starting and running procedure. - 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 it 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. 4615

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 9.

Post-test inspection. - After failure or a predetermined number of stress cycles, the ball running tracks were examined at a magnification of 60 with a microscope. Any abnormalities which correlated with the fatigue-life results were noted and followed up with a futher 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-trace surfaces were made after a polish with diamond dust to observe possible corrosion pitting and its relation to crack formation.

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APPENDIX B

PRESENTATION OF BEARING FATIGUE DATA

A distribution function developed by Weibull fits the observed scatter in the fatigue lives of rolling-contact bearings (ref. 12). Because of the usually small sample (about 30 bearings) involved, the data cannot reliably be fitted into a frequency curve. Instead, the cumulative form of the distribution is used. The cumulative distribution function (Weibull) is as follows:

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

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

Special probability paper is used for figures 3 on which the Weibull distribution becomes a straight line of slope e. The ordinate represents log-log $1/Q(L)$ 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 which takes the general direction of the array of points and which 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-rank values 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 12.

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TABLE I. - LUBRICANT PROPERTIES[a]

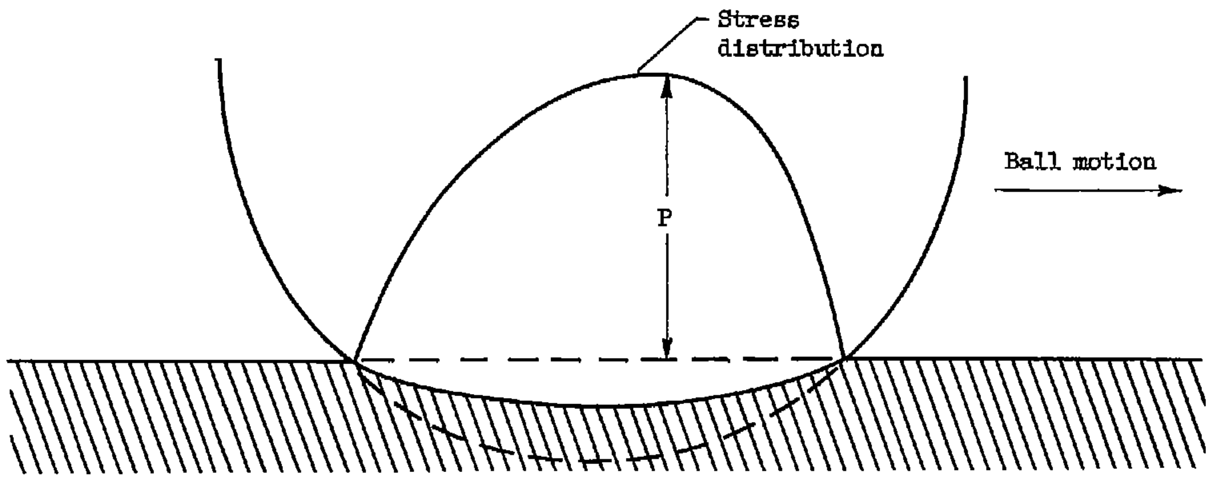
Viscosity, centistokes				Average viscosity, centistokes		Viscosity index, average	Neutralization number	
Before		After					Before	After
100° F	210° F	100° F	210° F	100° F	210° F			
MIL-0-6081B-2, grade 1005								
5.07	1.68	5.13	1.72	5.10	1.70	104.5	0.05	0.05
MIL-0-6081B-2, grade 1010								
10.14	2.49	10.37	2.54	10.26	2.52	73.2	0.05	0.05
MIL-L-15016A-2, grade 3042								
24.20	4.51	24.26	4.57	24.23	4.54	114.0	0.05	0.05
MIL-L-6082B, grade 1065								
118.6	11.94	119.56	12.27	119.1	12.11	99.4	0.06	0.06

[a]Additives: All fluids had pour-point depressants and may have had antifoam agents and oxidation inhibitors. None had viscosity index improvers. No further details on additives are available, since this information is regarded as proprietary by manufacturers. Specimens were bulk samples taken from lubricant supply at beginning and end of test run.

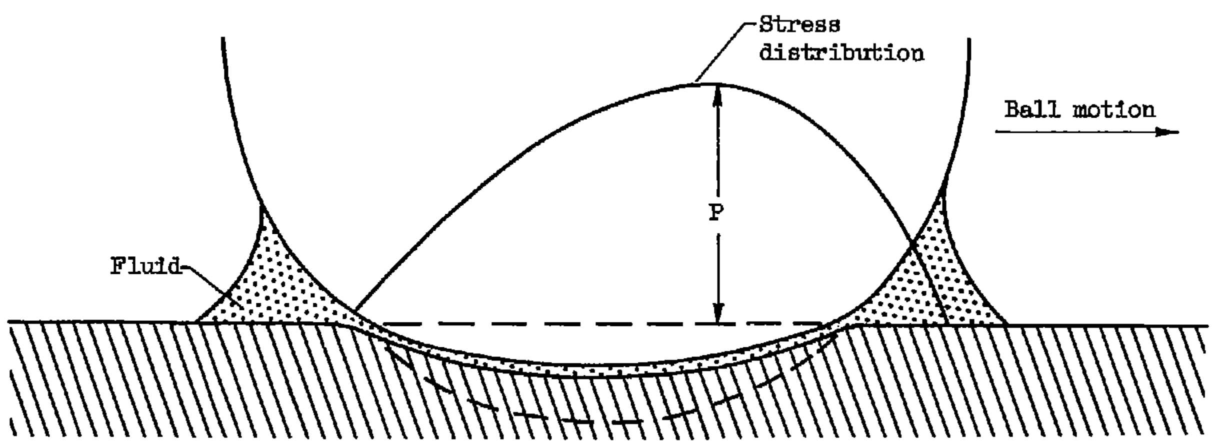
TABLE II. - MATERIAL ANALYSIS

	C	Si	Mn	S	P	W	Cr	V	Mo
Nominal AISI M-1 composition, %	0.80	0.23	0.23	Min.	Min.	1.50	4.00	1.00	8.00
Actual[a] composition, %	0.79	0.28	0.27	0.014	0.020	1.51	3.82	1.08	8.42

[a]Latrobe heat, 13-801; air-melt AISI M-1 tool steel.



(a) Dry.



(b) Lubricated.

Figure 1. - Diagram of stress distribution in a rolling sphere.

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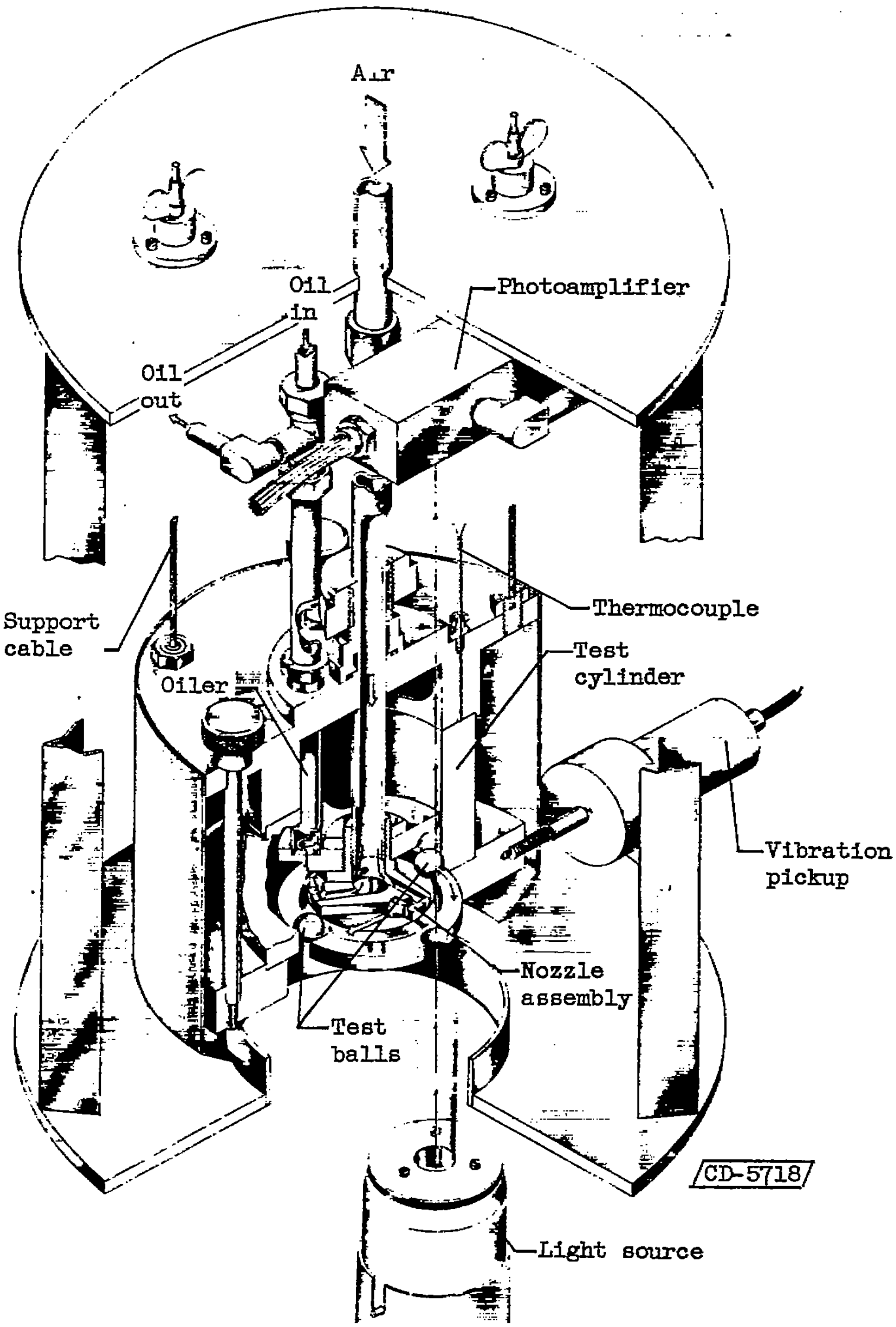

(a) Cutaway view.

Figure 2. - Rolling-contact fatigue spin rig.

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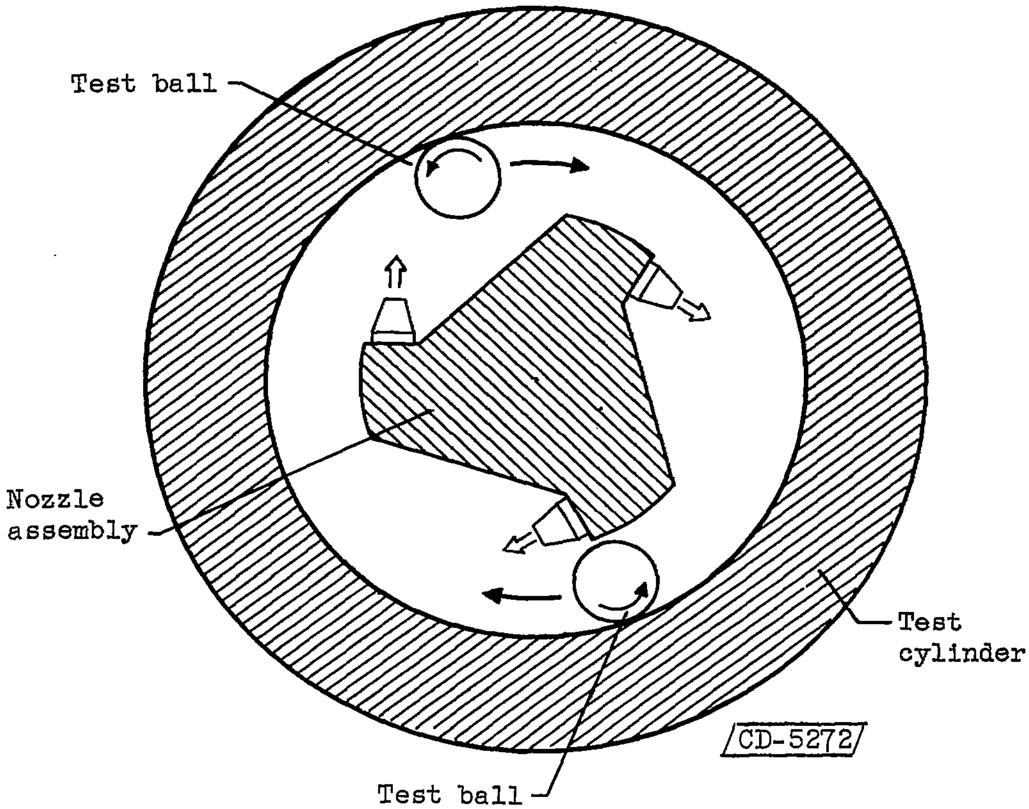
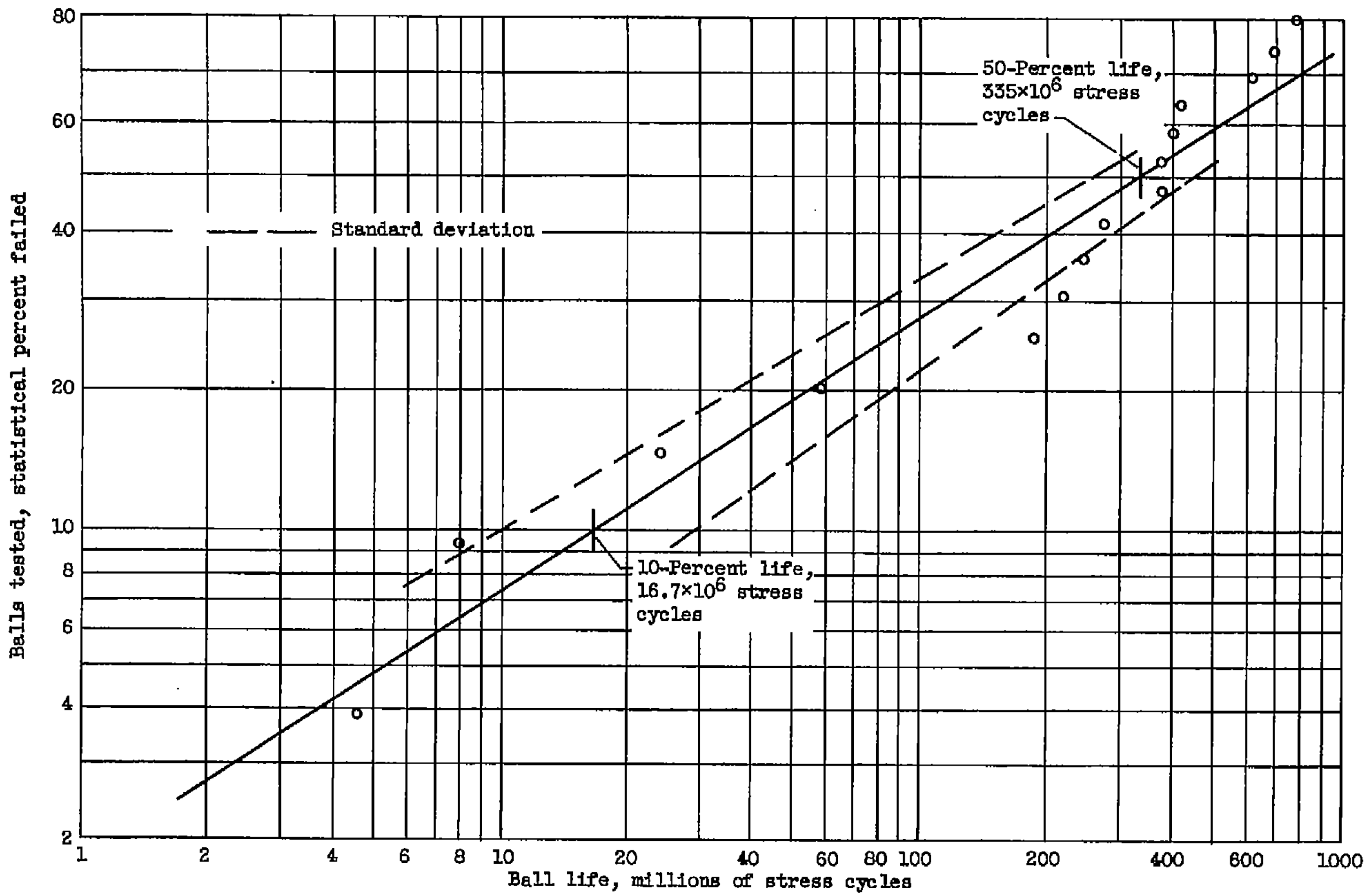


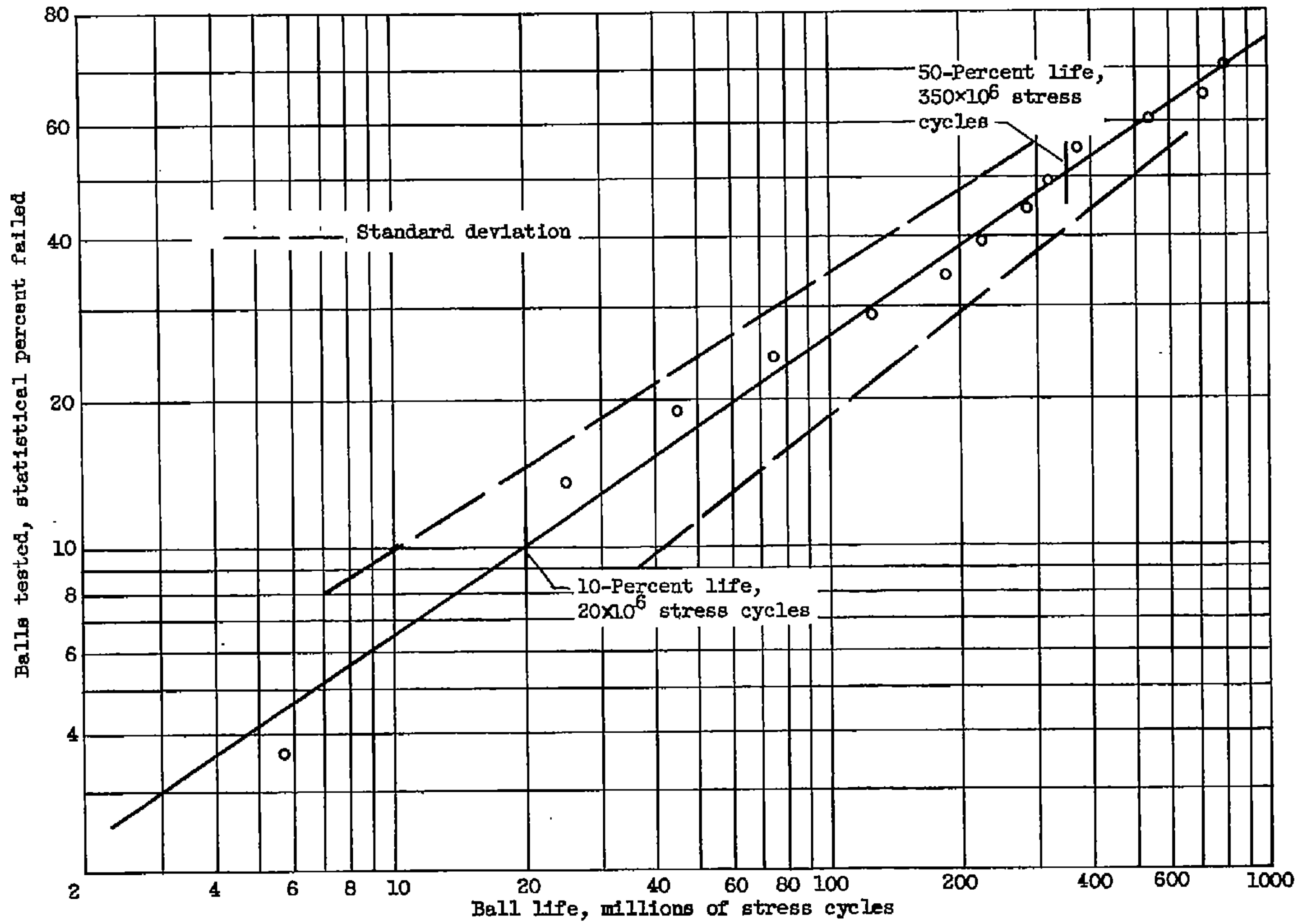
(b) Schematic diagram.

Figure 2. - Concluded. Rolling-contact fatigue spin rig.



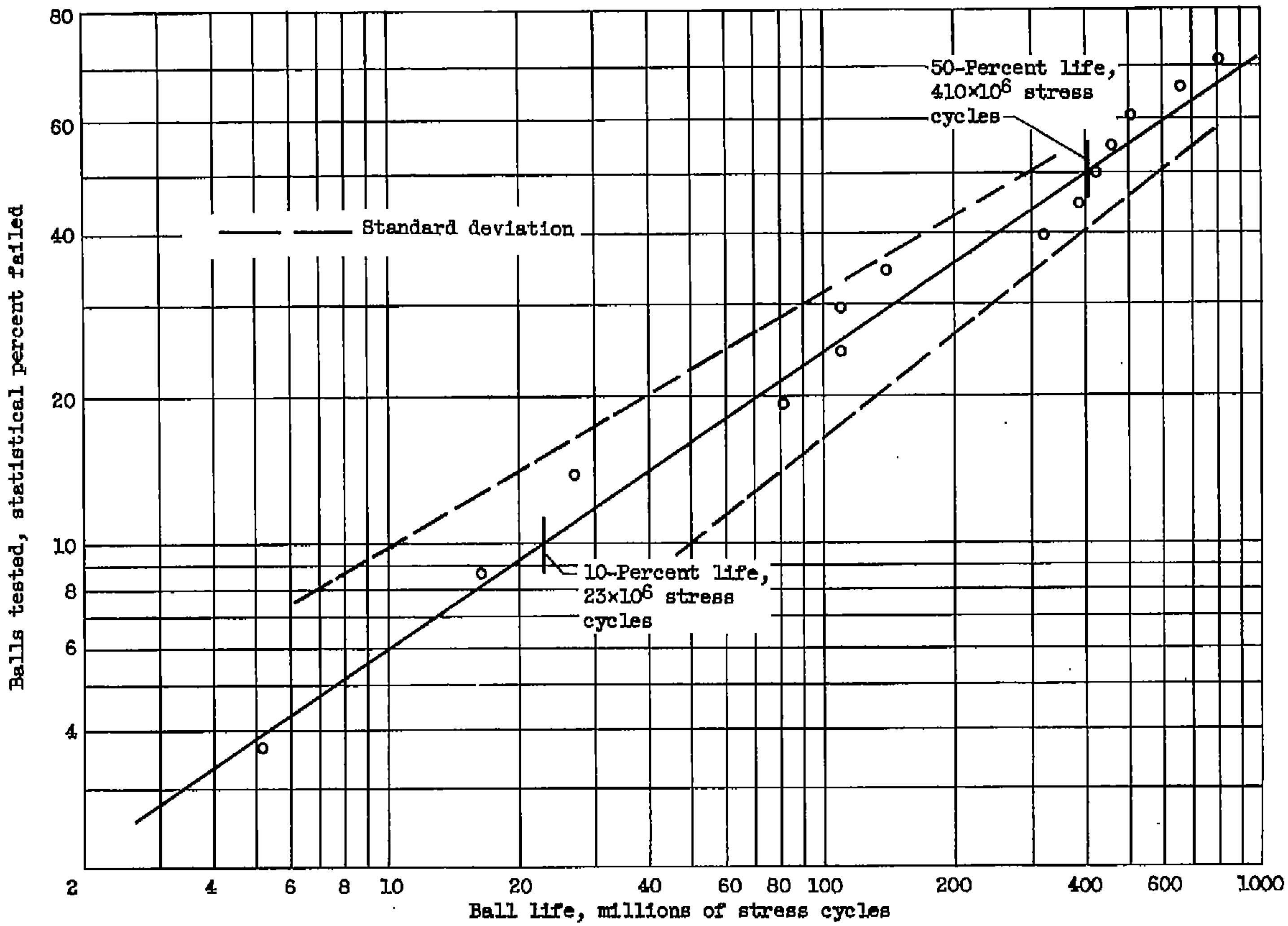
(a) Lubricant, MIL-0-6081B-2, grade 1005 mineral oil; viscosity, 5.1 centistokes.

Figure 3. - Rolling-contact fatigue life of AISI M-1 tool-steel balls with various lubricants. Test temperature, 100° F; maximum Hertz compressive stress, 725,000 psi.



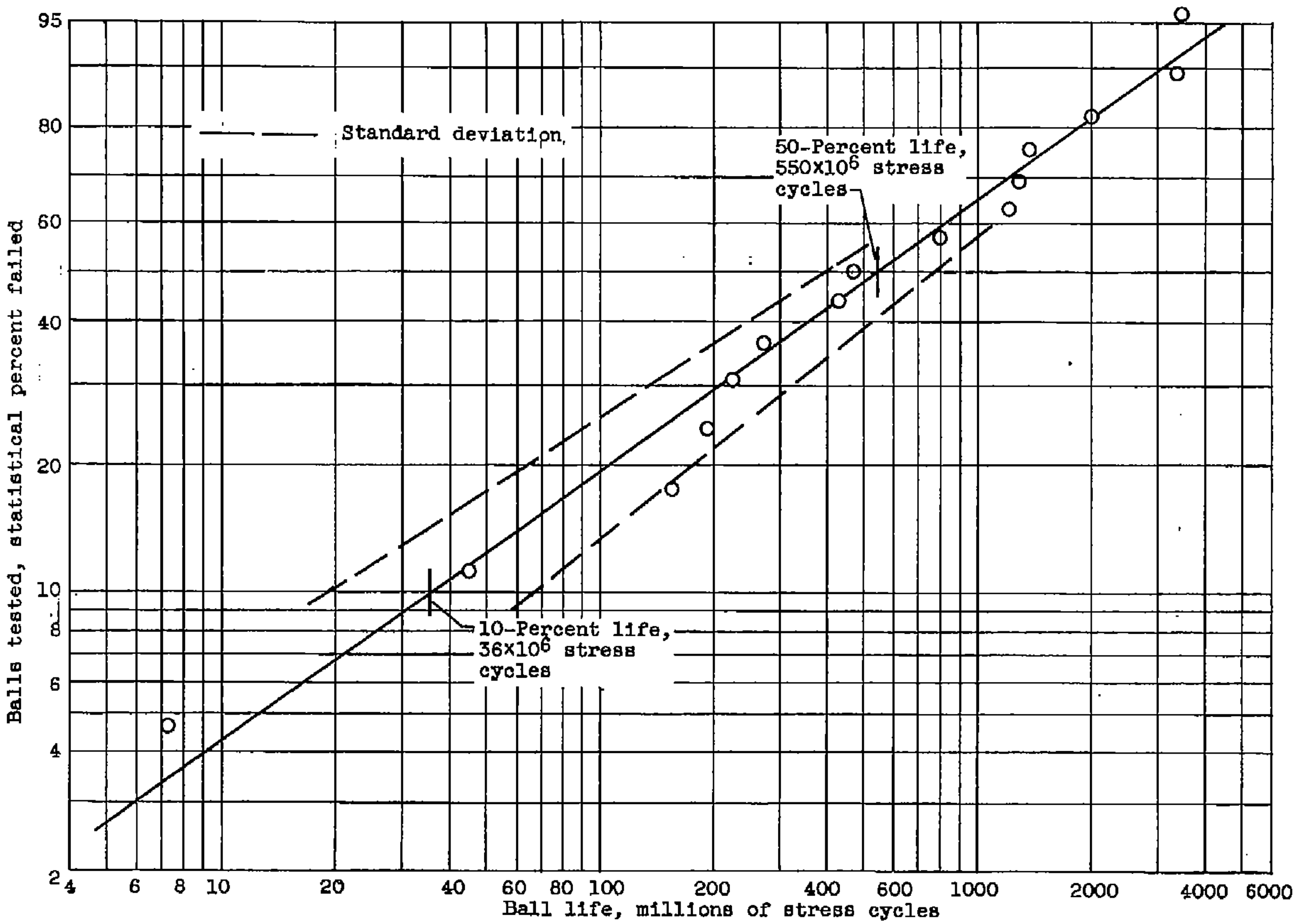
(b) Lubricant, MIL-O-6081B-2, grade 1010 mineral oil; viscosity, 10.3 centistokes.

Figure 3. - Continued. Rolling-contact fatigue life of AISI M-1 tool-steel balls with various lubricants. Test temperature, 100° F; maximum Hertz compressive stress, 725,000 psi.



(c) Lubricant, MIL-L-15016A-2, grade 3042 mineral oil; viscosity, 24.2 centistokes.

Figure 3. - Continued. Rolling-contact fatigue life of AISI M-1 tool-steel balls with various lubricants. Test temperature, 100° F; maximum Hertz compressive stress, 725,000 psi.



(d) Lubricant, MIL-L-6082B, grade 1065 mineral oil; viscosity, 119.1 centistokes.

Figure 3. - Continued. Rolling-contact fatigue life of AISI M-1 tool-steel balls with various lubricants. Test temperature, 100° F; maximum Hertz compressive stress, 725,000 psi.

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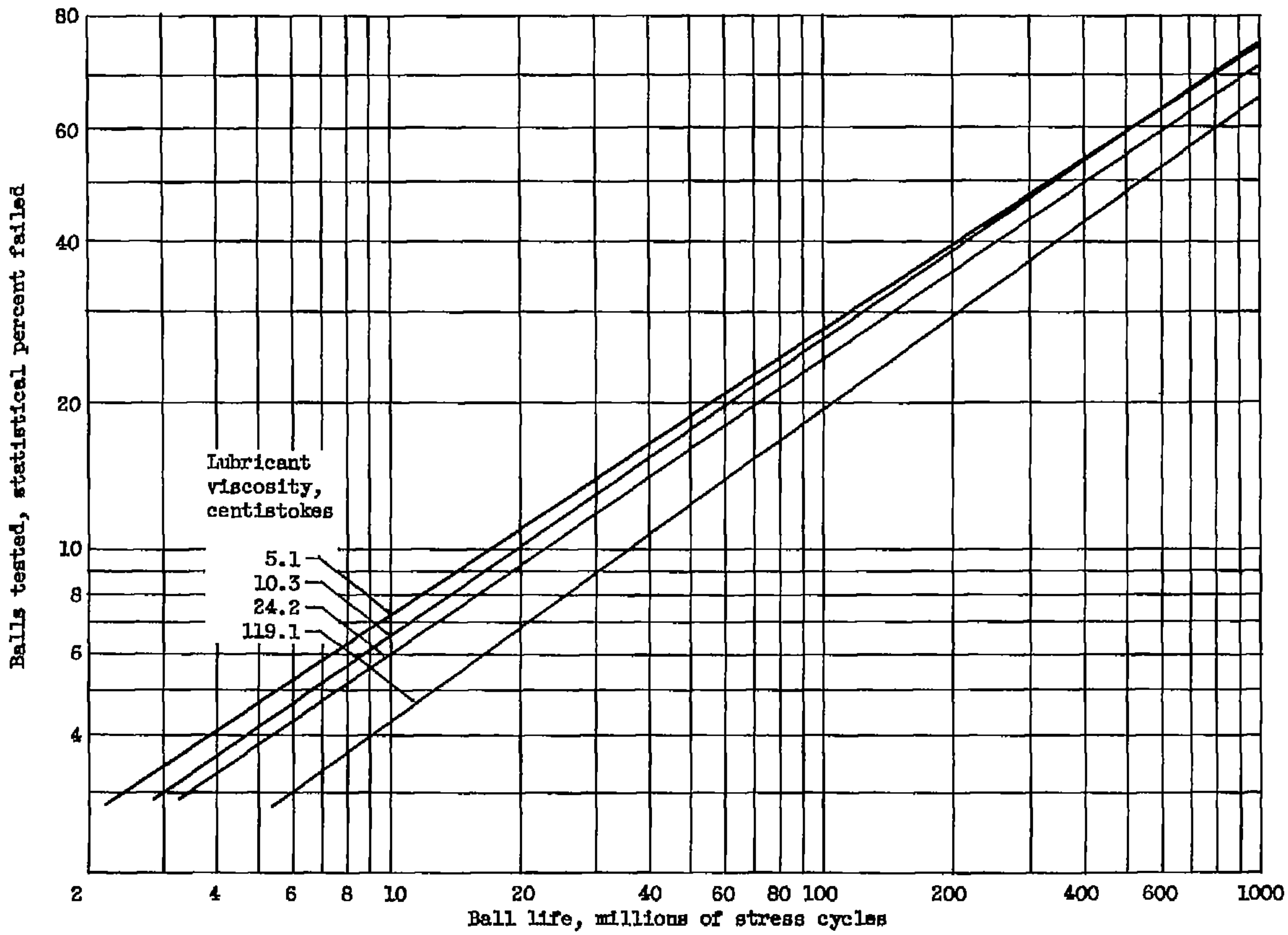
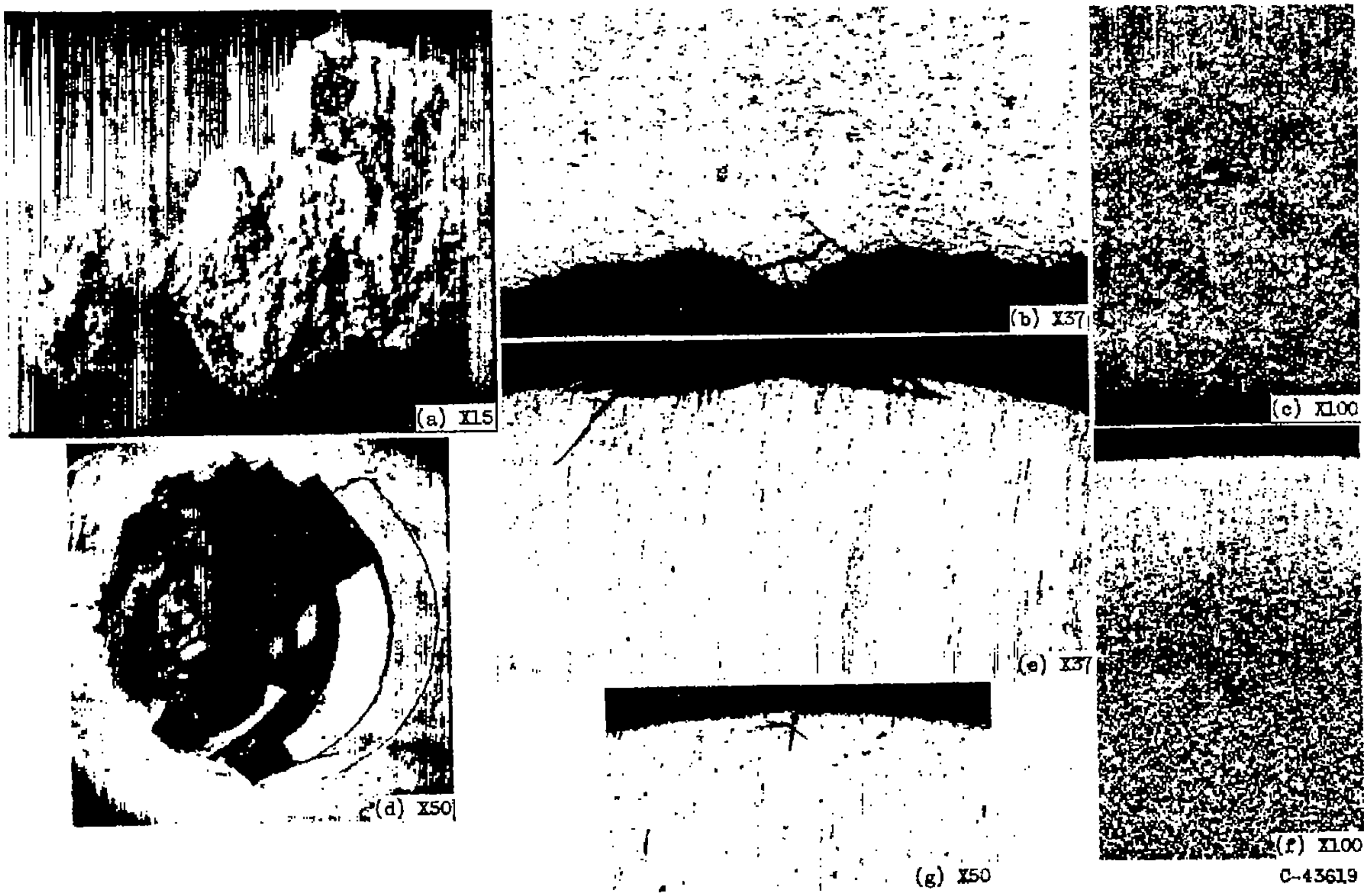


(e) Summary of fatigue lives with four mineral-oil lubricants.

Figure 3. - Concluded. Rolling-contact fatigue life of AISI M-1 tool-steel balls with various lubricants. Test temperature, 100° F; maximum Hertz compressive stress, 725,000 psi.



(a) Small spall on inner race of M-1 (222) bearing.
(b) Section view of part of same spall.
(c) Incipient failure near point of maximum shear stress.
(d) Typical spall, spin-rig-tested ball.
(e) Section view of spall.
(f) Incipient failure near point of maximum shear stress.
(g) Section view of early failure on spin-rig-tested ball.

Figure 4. - Comparison of failures of a bearing inner race and balls tested in rolling-contact fatigue spin rig.

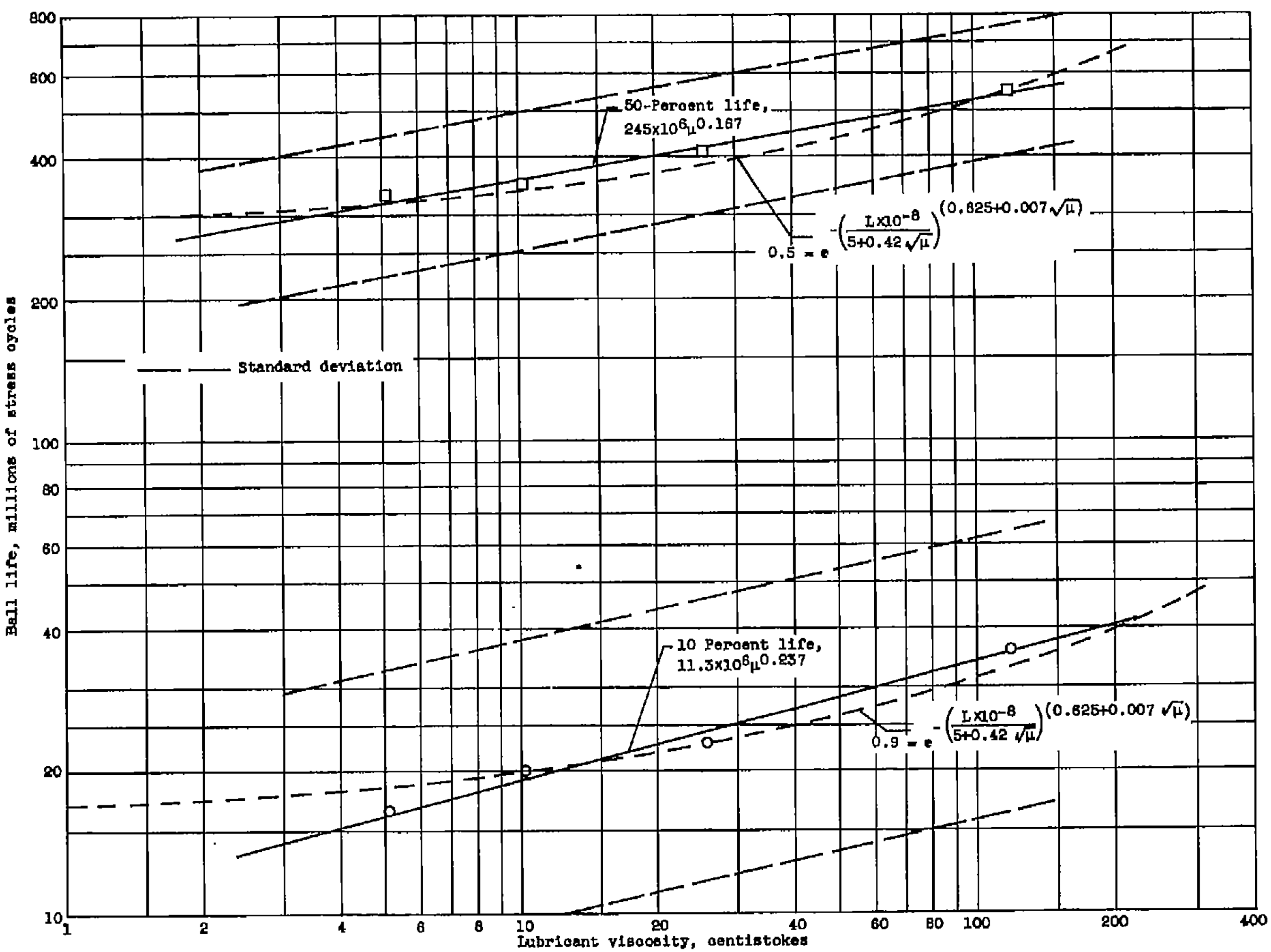


Figure 5. - Variation of ball life with lubricant viscosity. Test temperature, 100° F; maximum Hertz compressive stress, 725,000 psi.