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SUMMARY

Elastohydrodynamic film thicknesses and asperity contact-time fractions were measured for a 20-millimeter-bore ball bearing by using the capacitance and conductance techniques. The bearing was thrust loaded to 90, 445, and 778 newtons (20, 100, and 175 lb). The corresponding maximum stresses on the inner race were 1.28, 2.09, and 2.45 gigapascals (185 000, 303 000, and 356 000 psi). The test speed ranged from 400 to 15 000 rpm. The test bearing was mist lubricated with an MIL-L-23699A turbine oil. The test temperature was 27° C (80° F).

The experimental results were correlated to give the percent film (no-contact-time fraction) as a function of measured film thickness. Measurements were made for three test series that represented the test bearing in various conditions of run-in. The measurements show that the percent film changes with bearing run-in time. The experimental results agreed well with theoretical predictions based on surface trace analysis for a new bearing. For the run-in state, they agreed with previously reported experimental results. The results show that asperity contact existed at a film thickness-roughness ratio Λ of 6.0 or less for a new bearing. After run-in, no asperity contact occurred at a Λ of 3.5 or greater.

INTRODUCTION

Elastohydrodynamic (EHD) lubrication of concentrated mechanical contacts has been investigated theoretically and experimentally for several decades (refs. 1 to 16). EHD lubrication theory has practical applications in ball and roller bearings, gears, and traction-drive transmissions.

Two significant effects occur in elastohydrodynamic lubrication: (1) high pressure influences the viscosity of the lubricant and (2) the elastic contacts are substantially deformed locally. The net result is the formation of a thin lubricant film that is beneficial in preventing seizure and rapid wear of contacting parts.

EHD film thickness is difficult to measure because the films are so thin: They are generally of the same order of magnitude as the rms surface roughness. Various measurement methods have been developed and used (ref. 17). The X-ray method is restricted to applications where there is an open path for X-rays to the EHD contact, such as in the disk machine. The optical interference method is used for a ball on a transparent disk and in cases where the bearing outer race can be filled with a transparent window. The capacitance method is suitable for measuring EHD film thickness in an operating rolling-element bearing, as well as in the disk machine.

Most theoretical models of EHD lubrication assume that smooth surfaces are in contact. However, real engineering requirements of manufacture and economy of production result in surfaces that are rough to greater or lesser degrees. There have been experimental and theoretical studies performed for EHD lubrication of rough surfaces. Reference 18 presents a theoretical model for the effect of roughness on average film thickness. Reference 19 reports on the experimental determination of film thickness on disks in rolling contact. The theory and experiment showed that surface roughness with longitudinal lay has a deleterious effect on film thickness. If the EHD film becomes too thin, metal-to-metal contact occurs at the surface asperities. Reference 20 describes a model for the average asperity contact-time fraction. This model makes it possible to predict the amount of asperity contact as a function of film thickness and surface roughness.

Asperity contact-time fraction is defined as the expected time fraction during which there is metal-to-metal contact in the EHD zone. A contact-time fraction of zero corresponds to full film lubrication, with no contact between surface asperities. In connection with this definition, it is important to realize that time averaging rather than spacial averaging is used. A very small percentage of area contact can result in a very large asperity contact-time fraction. Indeed, if there were only one asperity contact in the EHD zone at all times, the asperity contact-time fraction would be unity, but the percentage of area contact would be almost zero.

One conclusion of reference 20 was that measuring film thickness by the conductance (asperity contact) technique is not practical. Notwithstanding this conclusion, conductance measurements do reveal asperity contact events and are most useful in determining the integrity of EHD films under varying conditions of operation and surface roughness.

Reference 21 describes tests of a 20-millimeter-bore ball bearing at speeds from 400 to 15 000 rpm, at temperatures to 121° C (250° F), and at contact stresses to 2.45 gigapascals (356 000 psi) in four oils. In those tests, the film thickness was measured by the capacitance method and compared with theoretically predicted film thickness. In the present work, the asperity contact-time fraction has been measured by the conductance method and the results have been correlated with measured film thickness. The film-thickness data for test series 3 of this report are the same as those presented in reference 21. The general objective of the present work was to explore the regime in which asperity contact takes place. Therefore, neither an extensive range of test temperature nor a great variety of test oils were used.

The specific objectives of the research described in this report were (1) to measure the asperity contact-time fraction for an operating ball bearing, (2) to correlate the results of the asperity contact-time fraction measurement with measured EHD film thickness, and (3) to compare the asperity contact-time fraction measurements with the theory of reference 20.

To accomplish these objectives, we measured EHD film thicknesses and asperity contact-time fractions for a ball bearing by the capacitance and conductance techniques. The test bearing was a 20-millimeter-bore ball bearing with three balls. It was run in the NASA EHD bearing test rig at speeds from 400 to 15 000 rpm. The test bearing was thrust loaded to 90, 445, and 778 newtons (20, 100, and 175 lb); this gave maximum contact stresses on the inner race of 1.28, 2.09, and 2.45 gigapascals (185 000, 303 000, and 356 000 psi). The lubricant, an MIL-L-23699 turbine oil, was supplied through a mist system. The tests were conducted at room temperature.

APPARATUS, SPECIMENS, AND PROCEDURE

EHD film-thickness test rig. - The experiments were conducted in the test rig shown in figure 1 and described in reference 21. The test rig consists of a rotating driveshaft supported by two bearings running against a test bearing, which is located in a vibration-isolated housing. The shaft is driven with an air turbine. Shaft speed is sensed with a magnetic pickup located near six slots that were milled in the shaft. The speed is held constant during a test with an automatic controller. The load is applied pneumatically. The linear ball bushing allows the thrust load to be transmitted from the pneumatic loader through the shaft to the test bearing. The thrust load is determined from the air pressure acting on the loading cylinder area. All tests reported herein were conducted at room temperature (27° C; 80° F). The bearing temperature remained constant within ± 3 degrees C (± 5 deg F) during the tests. Temperatures were sensed by thermocouples placed on the outer bearing race and in the inlet lubrication stream. The support bearings and the test bearing were mist lubricated with separate systems.

Test bearing. - The test bearings for this experiment were 20-millimeter-bore angular-contact bearings with three 7.15-millimeter- (9/32-in.-) diameter balls. The bearing specifications are given in table I. The bearing cage was polyimide, and the outer race was electrically isolated from the housing. The spring-loaded thermocouple at the outer race and the slip rings on the drive shaft provided electrical contact for measurements across the test bearing.

Test lubricant. - The lubricant chosen for this test program is of interest for bearing and gear applications. It is an improved type II synthetic, aircraft gas-turbine lubricant that meets MIL-L-23699A specifications. This lubricant is used in aircraft gearboxes and helicopter transmissions as well as in gas-turbine engines. The properties of this lubricant are shown in table II.

Measurement scheme. - The lubricant film thickness was measured by using the electrical capacitance method, as described in references 3 and 6. A schematic is shown in figure 2(a). The electrical capacitance between the inner and outer races of the test bearing is a function of the thickness of the lubricant film and the size of the Hertzian areas of contact (table III). The capacitance across the test bearing was measured with a capacitance bridge, and the film thickness was read from figure 3. During the tests the capacitance-bridge balancing signal was observed on an oscilloscope. Any short circuiting of the test-bearing capacitance as a result of asperity contact could be observed.

A separate measurement system was used to determine the asperity contact-time fraction. The measurement arrangement is shown in figure 2(b). A 200-millivolt-at-200-hertz half-wave rectified signal was applied across the test bearing. The voltage was visually observed on the oscilloscope for asperity contact. An asperity contact caused the voltage to drop to zero. The signal was also converted in a voltage-to-frequency converter and read on a pulse counter. The counter was set up to read 100 000 hertz for a known 100-percent lubricant film (no asperity contacts). Any asperity contacts reduced the average voltage across the bearing, and less than 100 000 hertz would be registered by the counter. The counter reading divided by 1000 is the percentage of film lubrication. This reading has been commonly called percent film. However, this may be an unfortunate choice because of a likely confusion with percentage of area contact through the lubricant film. A very small percentage of area contact can result in a very high reading for contact-time fraction. Nevertheless, we shall use the present terminology, that is, Percent film = (1 - Contact-time fraction) x 100. The conductance measurement concept has been described in reference 6 and used in references 13 and 22.

Test procedure. - Thrust loads applied to the bearing were 90, 445, and 778 newtons (20, 100, and 175 lb). The shaft speed was varied from 400 to 15 000 rpm. Temperatures remained constant, with a ± 3-degree C (± 5-deg F) tolerance during each test. The test temperature was nominally 27° C (80° F).

During the course of some prior experiments, the percent film was periodically measured. In preparation for running a new bearing in a test series, we noted that, for 100-percent film to be developed in the new bearing, it had to be run two orders of magnitude faster than the used bearing. We concluded that there must be considerable asperity contact in a new bearing and that asperity contact diminishes with running time. Therefore, in the present series of tests the percent film for the test bearing was measured at various run-in times.

Three series of tests were run. The test data are summarized in table IV. The first series was started with a new bearing. The shaft speed was very quickly increased to a speed high enough to produce a 100-percent film with the 90-newton (20-lb) load. The speed was then slowly decreased to the point where 70-percent film occurred. The testing continued by gradually increasing the load at each test speed to obtain a series of data points for film thickness and percent film. Care was taken not to operate with less than 70-percent film for test series 1.

The second series of tests was run immediately after the first series. In the second test series, speed was decreased in steps from 15 000 rpm. Test loads were applied at each speed until percent-film readings as low as 50 percent were recorded.

After the second test was completed, the bearing was run at 27°, 65°, and 121° C (80°, 150°, and 250° F) so that film-thickness data could be obtained for a range of loads and speeds. The results of these tests are reported in reference 21.

After the testing at higher temperatures was completed, a third test series for percent film was run, again at room temperature. In the third test series, the speed was rapidly increased to 15 000 rpm with minimum load. Then the load was increased at each speed selected. Speed was decreased in steps down to the minimum speed of 400 rpm. The data recorded at each test point for the three series of tests are given in table IV. Approximately 1 hour of run time was accumulated on the bearing before test series 3 began.

RESULTS AND DISCUSSION

After test series 3 was concluded, the test bearing was removed from the rig and examined visually and under the light microscope for evidence of wear. It was expected that flattening of the asperities would be seen. None could be seen under X100 magnification. Then high-resolution surface profile traces of the bearing inner race were made at several locations in the cross-groove direction.

Figure 4 is a typical trace. Since the bearing was an angular-contact type, the right side of the trace shows the run track and the left side shows the unrun portion of the surface. Although space does not permit showing all the traces, generally there was no observable difference in surface character between the run and unrun portions. There was no indication of any surface run-in, such as wear or diminishing of asperities. The large curvatures in the trace are due to the inability of the surface tracer to exactly follow the radius of the inner race. The surface finish, 0.025 micrometer (1 μin.) rms, was much better than the bearing manufacturer's specification given in table I.

The film thickness measured by the capacitance method is compared in figure 5 with the film thickness predicted by the Hamrock-Dowson EHD film-thickness equation (ref. 16). Figure 5(a) presents the data from test series 1 and figure 5(b), the data from test series 3. The greater scatter in the test-series-1 data is due to difficulties in accurately balancing the capacitance bridge in the presence of asperity contacts. Under these conditions there was a weak bridge sensitivity. It was impossible to take a capacitance reading for less than 90-percent film.

There is little experimental scatter in the test-series-3 data (fig. 5(b)). The capacitance bridge was easy to balance, since there was a definite response null. The data show approximately the expected theoretical sensitivity of film thickness to load. The measured film thickness was approximately one-third of the theoretically predicted film thickness. An inaccurate value of dielectric constant, an error in the pressure-viscosity exponent, or surface roughness effects may have caused this disagreement between theory and experiment.

If a calibration error is taken into account, the data generally follow the theoretically predicted trend in film thickness with speed. At high speeds the leveling off of film thickness with speed is predominantly caused by lubricant film starvation (ref. 21). At low speeds, the film levels off at a thickness approaching the rms level of the combined surface roughness. This effect is in general agreement with results presented in reference 23.

The percent-film trends with speed and load are shown for the three test series in figure 6. As expected, there is an order-of-magnitude speed difference for equivalent percent-film readings between test series 1 and test series 3. Test series 2, which was run immediately after test series 1 without the rig being shut down, shows definite change in percent film with speed. Thus, physical changes in the bearing that affect percent film seem to occur immediately after a new bearing is put into operation.

The percent film is correlated with measured EHD film thickness for the three test series in figure 7. For a new bearing, asperity contact and film breakdown begin to occur at a ratio of film thickness to rms surface roughness Λ of 4.25, for the lightest load (fig. 7(a)). After run-in (test series 3, fig. 7(c)) the test results show no film breakdown at Λ = 3.5 for even the heaviest load, which put a 2.45-gigapascal (356 000-psi) contact series on the inner race contact.

Results from other similar work (refs. 20, 23, 24, and 25) are shown in figure 8 and are compared here with the present results. In reference 23, Sidik and Coy used surface trace data, stored on magnetic tape in digitized form, to statistically predict percent film as a function of EHD film thickness. The traces were made from data taken on new surfaces. The computed results compare very well with the data taken in test series 1 (fig. 7(a)) for the new bearing. In reference 20, Coy and Sidik show that low-pass filtering of the surface trace data shifts the percent-film curves to the left. Low-pass filtering would decrease the asperity density in the surface records and thus simulate the expected effect of "running-in" the bearing. That this trend appears in the present data verifies that the percent-film curves shift toward the left to lower film thicknesses with increased run time. Data are also shown for the experimental results of other researchers (refs. 24 and 25) as reported in reference 26. Their data agree well with the present data from test series 3. In general, the experimental observations reported herein agree well with the theoretical prediction of reference 20; and with increased run-in, the test data approach that of references 24 and 25.

SUMMARY OF RESULTS

Film thicknesses and percent films were measured by the capacitance and conductance methods for a thrust-loaded 20-millimeter-bore ball bearing operating under a range of speeds and loads. The measurements were made on the bearing when it was new and again after it had been run through some higher temperature test sequences. The test parameters were thrust loads of 90, 445, and 778 newtons (20, 100, and 175 lb), which gave maximum contact stresses of 1.28, 2.09, and 2.45 gigapascals (185 000, 303 000, and 356 000 psi) on the inner race, and speeds ranging from 400 to 15 000 rpm. The bearing was mist lubricated with an MIL-L-23699A lubricant and run at room temperature (27° C, 80° F).

The experimental results of film-thickness and percent-film measurements were correlated and compared with other measurements and theoretical predictions. The following results were obtained:

1. In a new bearing, asperity contact existed at a film thickness-roughness ratio Λ of 6 or less. For a run-in bearing with a heavy load, less than 2.45-GPa (356 000-psi) Hertz stress, there was no asperity contact for a Λ of 3.5 or greater.

2. The amount of asperity contact in a new bearing rapidly diminished with run time, an indication of wear-in. But under visual examination at X100 magnification, there were no obvious physical changes in the test bearing surface.

3. There was good agreement between asperity contact theory and the measured percent film for new bearings; however, the mechanism of asperity contact reduction is unknown.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, July 12, 1979,
505-04.

REFERENCES

1. Grubin, A. N.: Fundamentals of the Hydrodynamic Theory of Lubrication of Heavily Loaded Cylindrical Surfaces. Investigation of the Contact of Machine Components, Kh. F. Ketova, ed., Translation of Russian Book No. 30, Central Scientific Institute for Technology and Mechanical Engineering, Moscow, 1949, Chapter 2. (Available from Department of Scientific and Industrial Research, Great Britain, Transl. CTS-235, and Special Libraries Association, Transl. R-3554.)

2. Dowson, D.; and Higginson, G. R.: A Numerical Solution to the Elasto-Hydrodynamic Problem. J. Mech. Eng. Sci., vol. 1, no. 1, June 1959, pp. 6-15.

3. Dyson, A.; Naylor, H.; and Wilson, A. R.: The Measurement of Oil-Film Thickness in Elastohydrodynamic Contacts. Proc. Inst. Mech. Eng., (London), vol. 180, pt. 3B, paper 10, 1965-66, pp. 119-134.

4. Archard, J. F.; and Cowking, E. W.: Elastohydrodynamic Lubrication at Point Contacts. Proc. Inst. Mech. Eng. (London), vol. 180, pt. 3B, 1965-66, pp. 47-56.

5. Dowson, D.; and Higginson, G. R.: Elastohydrodynamic Lubrication. Pergamon Press, Inc., 1966.

6. Allen, G. E.; Peacock, L. A.; and Rhoads, W. L.: Measurement of Lubricant Film Thickness in Hertzian Contacts. (SKF-AL-68T075, SKF Industries, Inc.; NASA Contract NAS3-7912.) NASA CR-105378, 1968.

7. Foord, C. A.; Hammann, W. C.; and Cameron, A.: Evaluation of Lubricants Using Optical Elastohydrodynamics. ASLE Trans., vol. 11, no. 1, Jan. 1968, pp. 31-43.

8. Zaretsky, E. V.; and Anderson, W. J.: EHD Lubrication. Mach. Des., vol. 40, Nov. 7, 1968, pp. 167-173.

9. Cheng, H. S.: A Numerical Solution of Elastohydrodynamic Film Thickness in an Elliptical Contact. J. Lubr. Technol., vol. 92, no. 1, Jan. 1970, pp. 155-162.

10. Tallian, T. E.: Elastohydrodynamic Hertzian Contacts - Part 1. Mech. Eng., vol. 93, no. 11, Nov. 1971, pp. 14-18.

11. Tallian, T. E.: Elastohydrodynamic Hertzian Contacts - Part 2. Mech. Eng., vol. 93, no. 12, Dec. 1971, pp. 17-22.

12. Parker, Richard J.; and Kannel, Jerrold W.: Elastohydrodynamic Film Thickness Between Rolling Disks with a Synthetic Paraffinic Oil to 589 K (600° F). NASA TN D-6411, 1971.

13. Parker, Richard J.; and Kannel, Jerrold W.: Elastohydrodynamic Film Thickness Measurements with Advanced Ester, Fluorocarbon, and Polyphenyl Ether Lubricants to 589 K (600° F). NASA TN D-6608, 1971.

14. Kannel, J. W.; and Bell, J. C.: Interpretations of the Thickness of Lubricant Films in Rolling Contact. 1 - Examination of Measurements Obtained by X-rays. J. Lubr. Technol., vol. 93, no. 3, Oct. 1971, pp. 478-484.

15. Cheng, H. S.: Isothermal Elastohydrodynamic Theory for the Full Range of Pressure-Viscosity Coefficient. J. Lubr. Technol., vol. 94, no. 1, Jan. 1972, pp. 35-43.

16. Hamrock, B. J.; and Dowson, D.: Isothermal Elastohydrodynamic Lubrication of Point Contacts. Part III - Fully Flooded Results. J. Lubr. Technol., vol. 99, no. 2, Apr. 1977, pp. 264-276.

17. Seth, B. B.; and Willis, T.: Techniques for Film Thickness Measurements in Elastohydrodynamic Lubrication. ASME paper 76-DET-79, Sept. 1976.

18. Chow, L. S. H.; and Cheng, H. S.: The Effect of Surface Roughness on the Average Film Thickness Between Lubricated Rollers. J. Lubr. Technol., vol. 98, no. 1, Jan. 1976, pp. 117-124.

19. Dow, T. A.; Kannel, J. W.; and Parker, R. J.: Effect of Longitudinal Surface Finish on Elastohydrodynamic Lubrication. NASA TN D-7967, 1975.

20. Coy, J. J.; and Sidik, S. M.: Two Dimensional Random Surface Model for Asperity Contact in Elastohydrodynamic Lubrication. Presented at the International Conference on Metrology and Properties of Engineering Surfaces, (Leicester, England), April 18-20, 1979.

21. Coy, J. J.; Gorla, R. S. R.; and Townsend, D. P.: Comparison of Predicted and Measured EHD Film Thickness in a 20-Millimeter-Bore Ball Bearing. NASA TP-1542, 1979.

22. Kannel, J. W.; and Snediker, D. K.: Elastohydrodynamic Lubrication in an Instrument Ball Bearing. J. Lubr. Technol., vol. 98, no. 2, Apr. 1976, pp. 244-250.

23. Sidik, S. M.; and Coy, J. J.: Statistical Model for Asperity - Contact Time Fraction in Elastohydrodynamic Lubrication. NASA TP-1130, 1978.

24. Tallian, T. E.; et al.: Lubricant Films in Rolling Contact of Rough Surfaces. ASLE Trans., vol. 7, no. 2, Apr. 1964, pp. 109-126.

25. Poon, S. Y.; and Haines, D. J.: Friction Behavior of Lubricated Rolling-Contact Elements. Proc. Inst. Mech. Engrs. (London), vol. 181, 1966-67, pp. 363-376.

26. Johnson, K. L.; Greenwood, J. A.; and Poon, S. Y.: A Simple Theory of Asperity Contact in Elastohydrodynamic Lubrication. Wear, vol. 19, no. 1, Jan. 1972, pp. 91-108.

TABLE I. - TEST BEARING SPECIFICATIONS

Material, balls and races	AISI M-1
Inside diameter, mm (in.)	20 (0.7874)
Outside diameter, mm (in.)	47 (1.8504)
Width, mm (in.)	14 (0.5512)
Pitch diameter, mm (in.)	35.5 (1.4)
Nominal contact angle, deg	17
Inner race curvature, percent	53
Outer race curvature, percent	54
Number of balls	3
Ball diameter, mm (in.)	7.15 (9/32)
Rockwell C hardness (inner race, outer race, and balls)	62 - 64
Surface finish (rms), m (in.):	
Races	0.15 (6)
Balls	0.025 - 0.05 (1 - 2)
Tolerances	ABEC-5

TABLE II. - TEST LUBRICANT SPECIFICATIONS

[Texaco 7730 A2 - improved type II synthetic, aircraft gas-turbine lubricant (MIL-L-23699A).]

Pour point, °CV	-60
Flashpoint, °C	262
Fire point, °C	---
Autogenous ignition temperature, °C	426
Density, g/cm^3:	
At 100° C (212° F)	0.93
At 38° C (100° F)	0.98
Pressure-viscosity exponent, m^2/N:	
At 100° C (212° F)	1.00×10^{-8}
At 38° C (100° F)	1.37×10^{-7}
Kinematic viscosity, cS:	
At 100° C (212° F)	5.2
At 38° C (100° F)	28
Specific heat, J/kg K:	
At 100° C (212° F)	2000
At 38° C (100° F)	1913
Thermal conductivity, W/m K:	
At 100° C (212° F)	0.152
At 38° C (100° F)	0.162

TABLE III. - HERTZIAN CONTACT CONDITIONS AT INNER AND OUTER RACES FOR THREE THRUST LOADS

[Width of rolling track is determined by major axis width.]

Race	Contact condition	Thrust load, N (lbf)		
		90 (20)	445 (100)	778 (175)
Inner	Maximum Hertzian stress, GPa (ksi)	1.28 (185)	2.09 (303)	2.45 (356)
	Semimajor axis, cm (in.)	0.0510 (0.0200)	0.0840 (0.0330)	0.0990 (0.0390)
	Semiminor axis, cm (in.)	0.0066 (0.0026)	0.0110 (0.0043)	0.0130 (0.0051)
Outer	Maximum Hertzian stress, GPa (ksi)	1.13 (164)	1.85 (269)	2.19 (318)
	Semimajor axis, cm (in.)	0.0460 (0.0180)	0.0740 (0.0290)	0.0860 (0.0340)
	Semiminor axis, cm (in.)	0.0086 (0.0034)	0.0140 (0.0055)	0.0170 (0.0065)

TABLE IV. - TEST RESULTS

(a) Test series 1.

Speed, rpm	Load		Capacitance, pF	Film thickness		Percent film
	N	lb		μm	in.	
3 000	89	20	-----	----	----	70
4 000	89	20	48.2	0.11	4.5	100
6 000	89	20	41.2	.14	5.5	100
	445	100	-----	----	----	70
8 000	89	20	30.2	.21	8.2	100
	445	100	-----	----	----	88
	534	120	-----	----	----	75
10 000	89	20	37.7	.15	6.1	100
	445	100	41.2	.39	15.5	98
	779	175	-----	----	----	75
15 000	89	20	39	.15	5.9	100
	445	100	102.2	.13	5.3	100
	779	175	-----	----	----	66

(b) Test series 2.

Speed, rpm	Load		Capacitance, pF	Film thickness		Percent film
	N	lb		μm	in.	
15 000	779	175	-----	----	---	66
	445	100	102.2	0.13	5.3	100
	89	20	39	.15	5.9	100
10 000	89	20	44.2	.1	5.1	100
	445	100	88.2	.16	6.2	100
	779	175	-----	----	---	88
8 000	89	20	43.2	.13	5.2	100
	445	100	138.2	.09	3.7	97
	779	175	-----	----	---	71
6 000	89	20	53.2	.10	4.0	100
	445	100	97.2	.14	5.5	93
	779	175	-----	----	---	68
4 000	89	20	52.2	.10	4.1	100
	445	100	137.2	.09	3.7	93
	779	175	-----	----	---	55
3 000	89	20	52.2	.10	4.1	100
	445	100	-----	----	---	69
2 000	89	20	67.7	.07	2.9	100
	445	100	-----	----	---	50

(c) Test series 3.

Speed, rpm	Load		Capacitance, pF	Film thickness		Percent film
	N	lb		μm	in.	
15 000	89	20	37.7	0.15	6.1	100
	445	100	85	.16	6.4	100
	779	175	139.2	.14	5.4	98
10 000	89	20	43.4	.13	5.1	100
	445	100	93.4	.15	5.8	
	779	175	150.2	.13	5.0	
8 000	89	20	41.2	.14	5.5	
	445	100	106.6	.13	5.0	
	779	175	161.9	.12	4.7	
6 000	89	20	47.4	.12	4.6	
	445	100	107.4	.12	4.9	
	779	175	181.2	.10	4.1	
4 000	89	20	50.4	.11	4.3	
	445	100	119.2	.11	4.3	
	779	175	200.2	.09	3.6	98
3 000	89	20	56.9	.09	3.7	100
	445	100	131.8	.10	4.0	100
	779	175	217.8	.08	3.3	98
2 000	89	20	72	.07	2.7	100
	445	100	165.2	.08	3.0	98
	779	175	227.2	.08	3.1	89
1 500	89	20	81	.06	2.3	100
	445	100	208.2	.06	2.3	97
	779	175	-----	----	---	92
1 000	89	20	103.2	.04	1.75	100
	445	100	275.2	.05	1.8	95
	779	175	-----	----	---	45
800	89	20	100.2	.05	1.85	100
	445	100	292.2	.04	1.7	94
	779	175	-----	----	---	10
600	89	20	105.2	.04	1.75	100
	445	100	288.2		1.65	97
400	89	20	107.2		1.65	100
	445	100	315.2		1.45	97

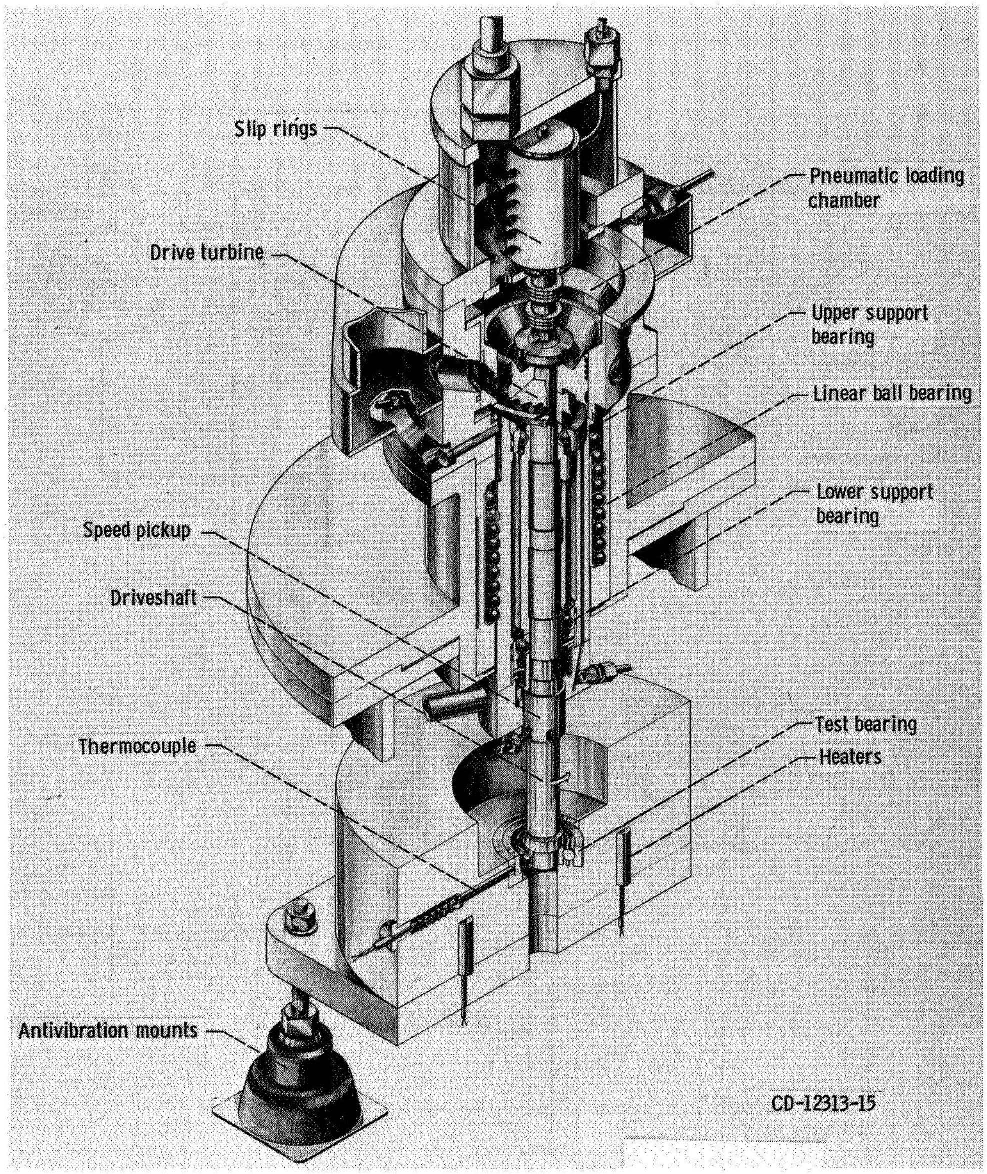
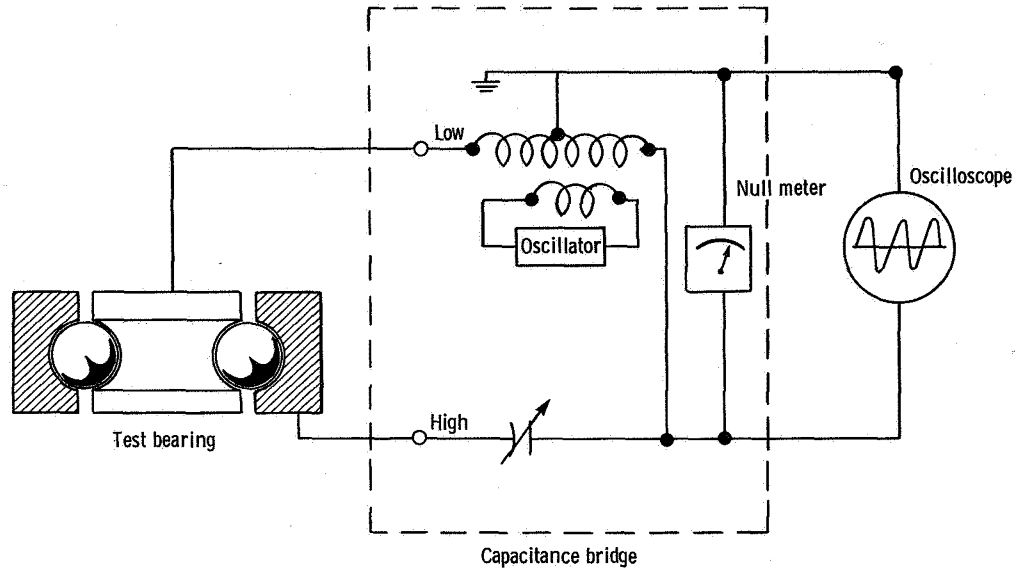
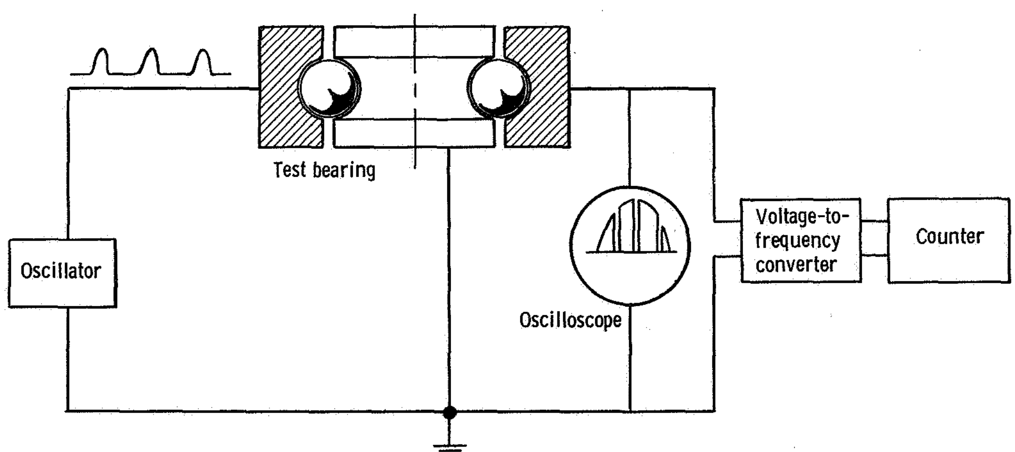


Figure 1. - EHD film thickness measurement rig.



(a) Film thickness.



(b) Percent film.

Figure 2. - Measurement schematics.

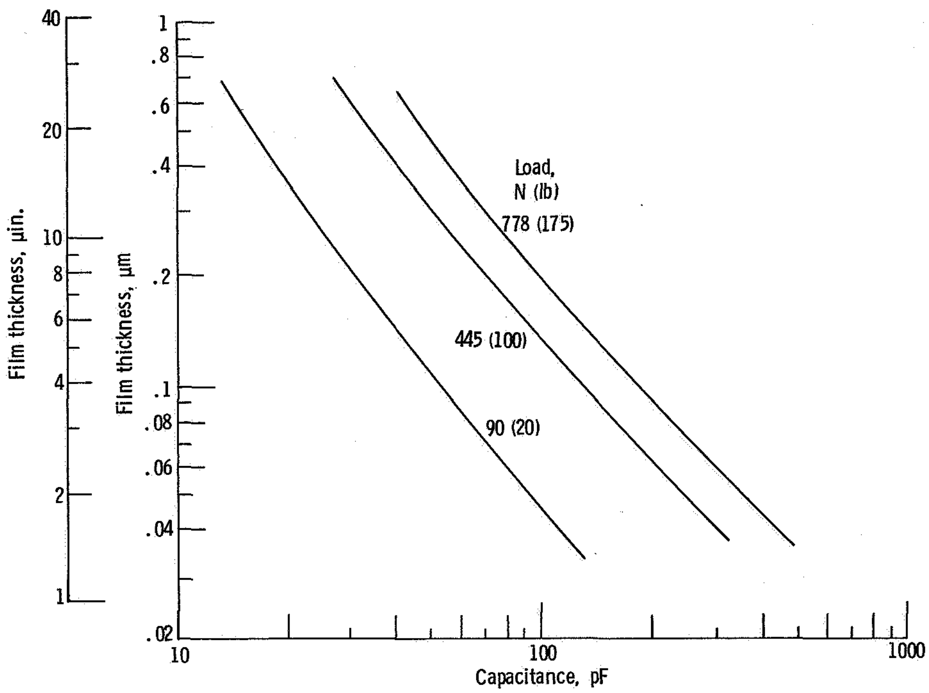


Figure 3. - Calculated capacitance for test bearing as function of load and film thickness.

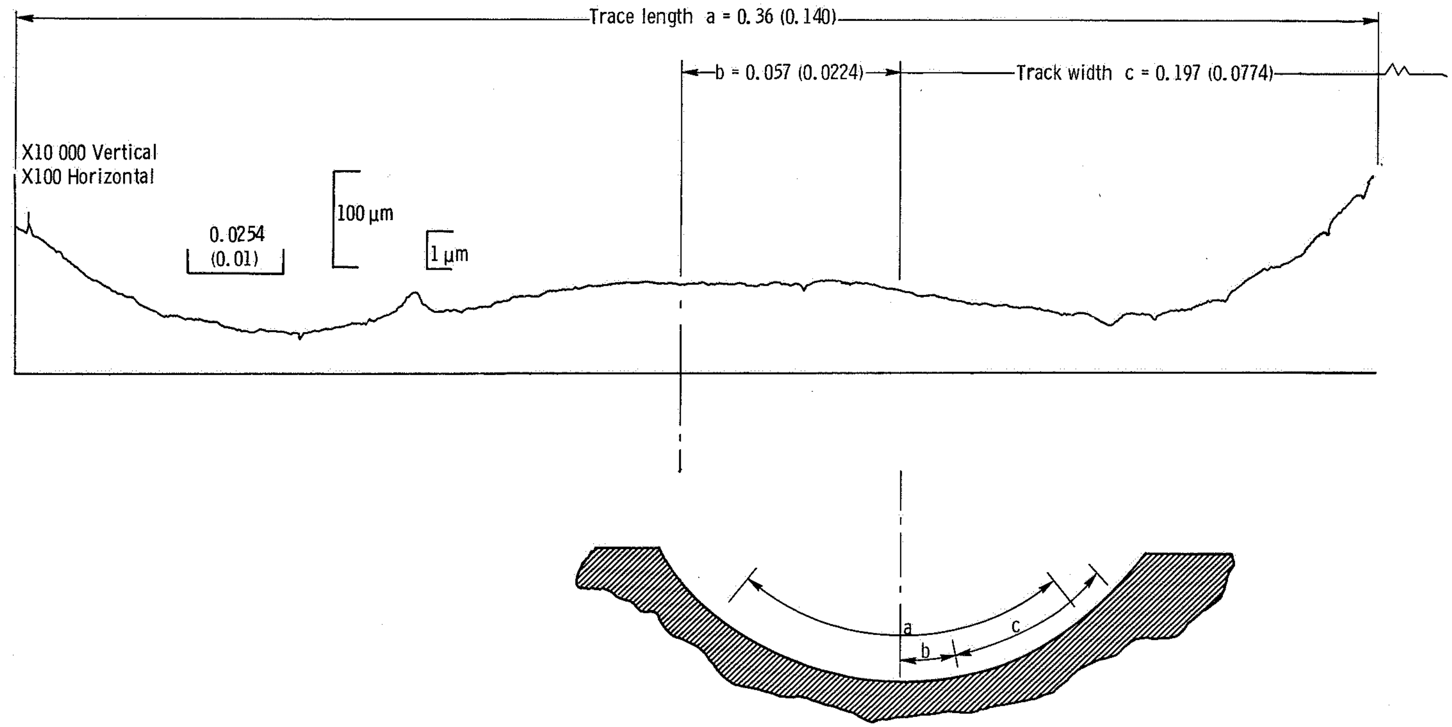


Figure 4. - Surface profile trace in cross-groove direction. (Dimensions are in cm (in.).)

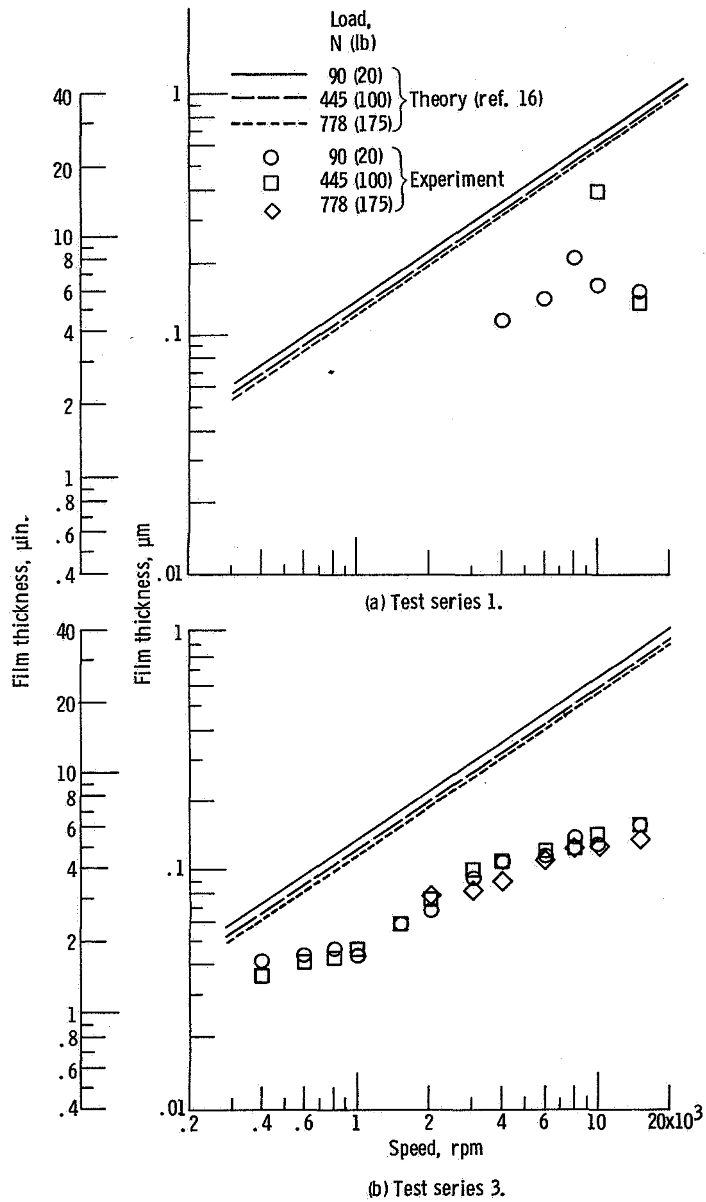
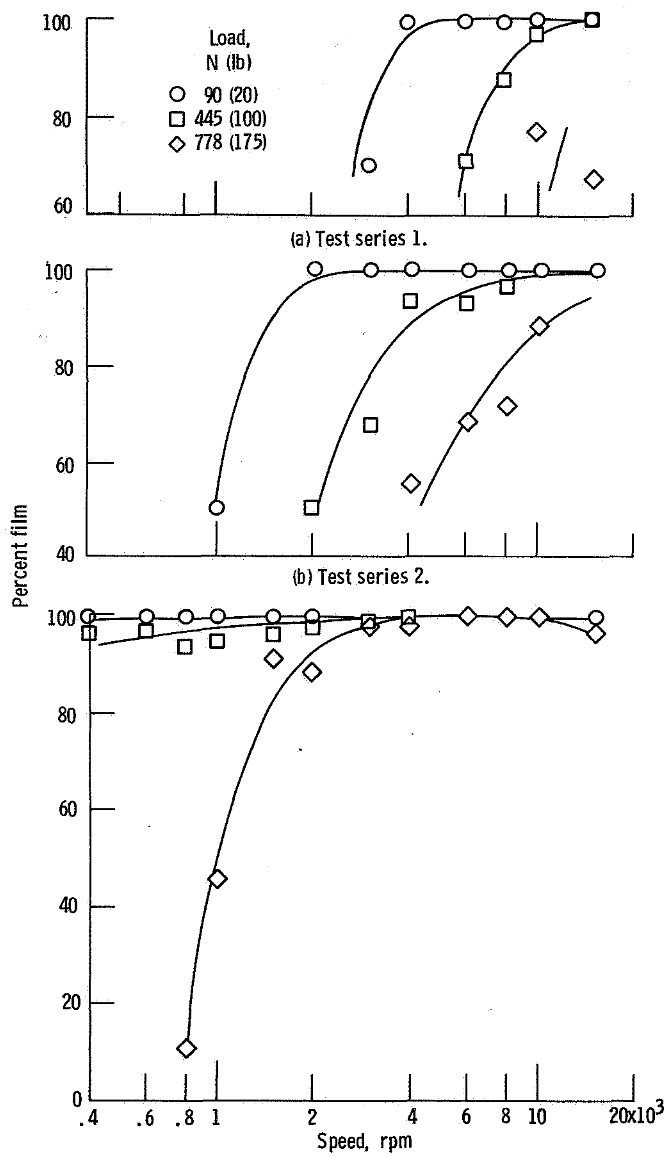
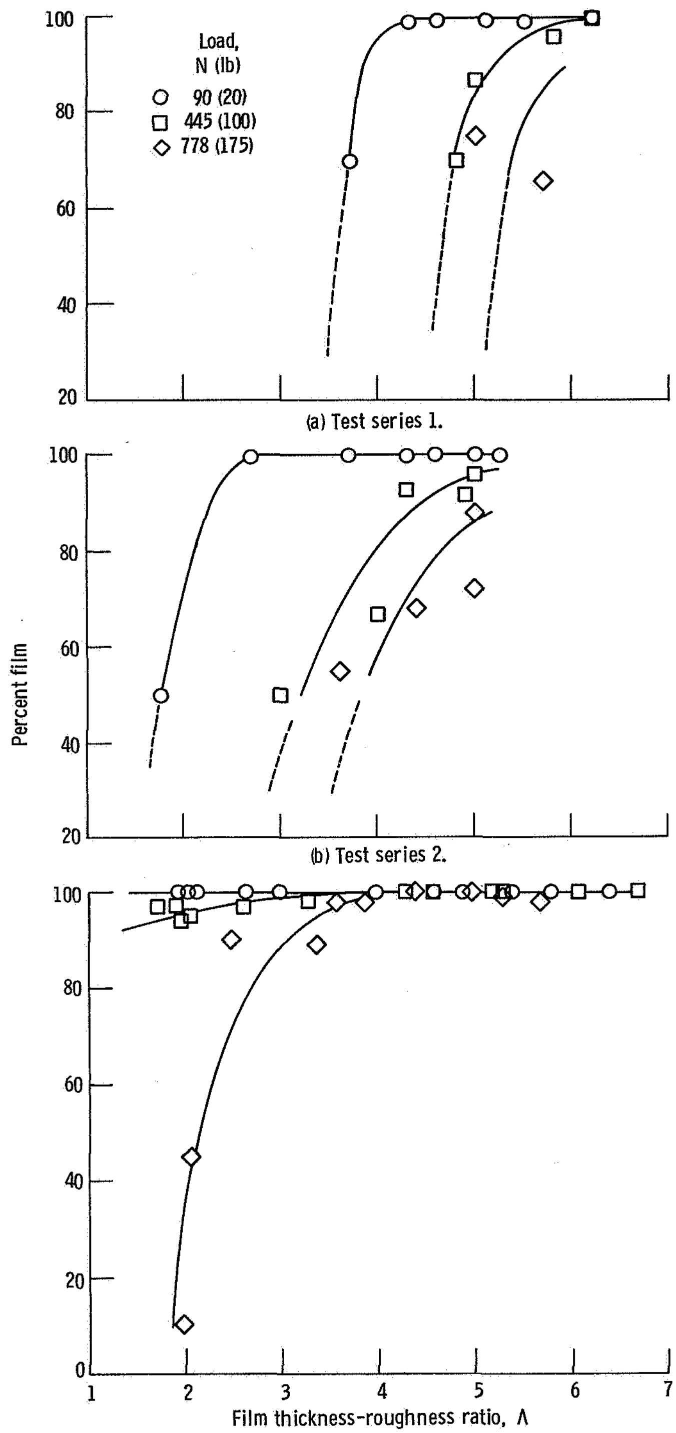


Figure 5. - Film thickness measurements compared with theory.



(c) Test series 3.

Figure 6. - Percent film measurements.



(c) Test series 3.

Figure 7. - Percent film correlated to film thickness.

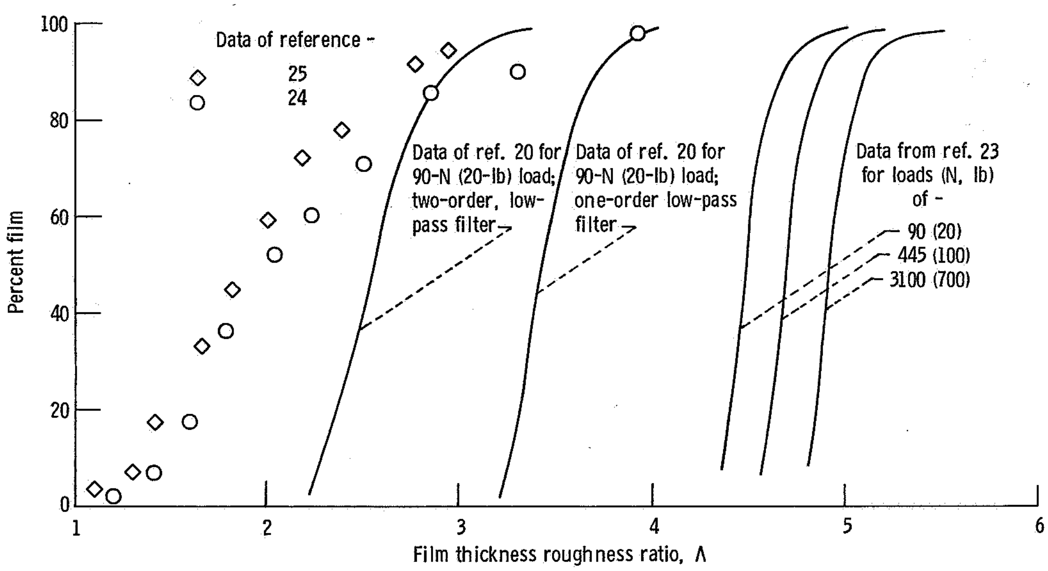


Figure 8. - Percent film - theoretical results for different thrust loads and amounts of filtering. Experimental points included for comparison with figure 7.

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15. Supplementary Notes

16. Abstract

Elastohydrodynamic film thicknesses and asperity contact-time fractions were measured for a 20-mm-bore ball bearing by using the capacitance and conductance techniques. The bearing was thrust loaded to 90, 445, and 778 N (20, 100, and 175 lb). The corresponding maximum stresses on the inner race were 1.28, 2.09, and 2.45 GPa (185 000, 303 000, and 356 000 psi). The test speed ranged from 400 to 15 000 rpm. The test bearing was mist lubricated with an MIL-L-23699A turbine oil. The temperature was 27° C (80° F). The experimental results were correlated to give the percent film (no-contact-time fraction) as a function of measured film thickness. Measurements were made for three test series that represented the test bearing in various conditions of run-in. The measurements show that the percent film changes with bearing run-in time. The experimental results agreed well with theoretical predictions based on surface trace analysis for a new bearing. For the run-in state, they agreed with previously reported experimental results. The results show that asperity contact existed at a film thickness-roughness ratio Λ of 6.0 or less for a new bearing. After run-in, no asperity contact occurred at a Λ of 3.5 or greater.

17. Key Words (Suggested by Author(s))
Bearings; Lubrication; Elastohydrodynamics; Surface roughness

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