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EVALUATION OF IMPREGNATED LUBRICANTS IN BALL-BEARING RETAINERS AT 10^{-6} TORR

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Lewis Research Center
Cleveland, Ohio

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NASA Lewis Research Center, Cleveland, Ohio
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1. High viscosity mineral oil was the best of the four lubricants for this application on the basis of low torque, minimum torque variation and endurance.

2. Lubrication by bearing retainer impregnation was found inadequate for high vacuum conditions.

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SUMMARY

Four liquid lubricants, a polyphenyl ether, a polysiloxane, a sebacate, and a high-viscosity mineral oil, were evaluated as lubricating impregnants in cotton-cloth phenolic retainers of 20-millimeter-bore ball bearings operating in a vacuum of 10^{-6} torr. The effect of the cloth weave on the lubricating effectiveness of the best of the lubricants was also studied. The test bearings were run at 3550 rpm under axial loads of 50 to 100 pounds (22.7 to 45.4 kg) for 1 hour, or until the torque exceeded and remained above 20 inch-ounces (14.1 cm-N).

Although none of the lubricants provided good lubrication in the vacuum environment of this system, the most acceptable lubricant was the mineral oil. It provided the lowest torque levels and torque roughness and carried the maximum load for the full duration of the test. High initial torques occurred frequently with all of the lubricants, which indicated inadequate lubrication during the initial part of the run. Comparing bearing performance obtained with only impregnated retainers to that obtained with both impregnated retainers and prelubrication showed that the impregnated retainers used did not feed the lubricant fast enough to provide good lubrication at the conditions of this investigation. In additional tests with the high-viscosity mineral oil, varying the weave of cloth in the retainer produced no significant improvements in the bearing torque.

INTRODUCTION

Rolling-element bearings used in open or partially sealed spacecraft mechanical systems must operate at ambient pressures below 10^{-6} torr (mm Hg). Although most conventional lubricants evaporate rapidly at these low pressures, several liquid lubricants having low vapor pressures might be used effectively in certain vacuum bearing applications (refs. 1 and 2).

Since both weight and space are at a premium in any space vehicle, bearings using self-contained lubricant reservoirs would be most desirable. In this lubrication technique the retainer pores or other voids in the bearing, filled with low-vapor-pressure oil or grease, may function as the lubricant reservoir. During bearing operation the lubricant flows from the interior of the retainer to its surfaces. Lubrication at the bearing sliding and rolling surfaces is provided by transfer from the retainer to the balls. This type of lubricating system was used in instrument-motor bearings of references 3 and 4. Further studies are required to determine the range of applicability of these low-vapor-pressure lubricants in larger size bearings operating at higher loads.

The objectives of this investigation are (1) to evaluate, in vacuum, the lubricating effectiveness of four liquid lubricants in 20-millimeter-bore, size 204, bearings, and (2) to determine the importance of retainer-material weave by conducting tests using the best lubricant from the previous tests with cotton-cloth phenolics having fine, medium, and coarse weaves.

The four lubricants used were (1) mix-bis(mix-phenoxyphenoxy) benzene (polyphenyl ether), (2) methyl chlorophenyl polysiloxane, (3) di(2-ethylhexyl) sebacate (Mil-L-7808C), and (4) high-viscosity mineral oil. All tests were conducted at ambient pressures of approximately 10^{-6} torr, thrust loads of 50 to 100 pounds (22.7 to 45.4 kg), and a speed of 3550 rpm, for periods of 1 hour or until failure occurred.

BEARING LUBRICATION ANALYSIS

The distribution of the lubricant from the retainer to the different parts of the bearing proceeds as follows: (1) the lubricant flows from the interior of the impregnated retainer to the surface; (2) the oil is transferred from the retainer to the balls and to the retainer locating surface; (3) the balls transfer the lubricant to the race grooves; and (4) the lubricant is lost from the exposed surfaces of the bearing parts by evaporation and other mechanisms.

The effectiveness of lubrication depends upon maintaining an adequate lubricant film on the bearing surfaces where rolling or sliding occurs. The ability to maintain the lubricant film on the bearing surfaces depends in part upon (1) the rate of lubricant flow from the retainer and (2) the rate of evaporation from these surfaces.

The cotton-cloth phenolic material used in the retainers can be considered a porous material. But the "porosity" here is not that of open voids, as in a sintered material or a sponge, but of cotton threads acting as wicks woven into a cloth running through a plastic matrix. The threads on the surface are exposed by the machining of the retainer, which opens paths along which the lubricant flows into the retainer. The cloth, therefore, provides a network of capillary paths or wicks in which the lubricant is stored.

Once the lubricant reaches the bearing surfaces, its dwell time is determined partially by its evaporation rate. Table I shows free-surface evaporation rates at several temperatures for these lubricants. The mineral oil seems better than the other materials because it has the lowest evaporation rate of the four lubricants and would thus remain on the surface for the longest time.

For a given matrix, ideal lubrication will exist when the lubricant moves to the surface of the retainer at just the rate necessary to replenish the amount lost from the bearing surfaces by the various mechanisms. If such a low-evaporation-rate lubricant can be found which provides acceptable torque and roughness, a long-lived lubricant system will be established.

APPARATUS

Bearing Test System

A cutaway view and a cross section of the test-bearing arrangement are shown in figure 1. The test bearing was operated in vacuum and loaded axially by pressurizing the load bellows with compressed air. A stainless steel wire connected the load bellows to the bearing housing. The load bellows was calibrated with a strain-gage ring. The load capacity of the system was approximately 200 pounds (91 kg).

The test-bearing shaft was driven by a 1/4-horsepower canned induction motor inside the bell jar, which rotated the test shaft at a nominal constant speed of 3550 rpm. A magnetic speed pickup was used to monitor the motor speed.

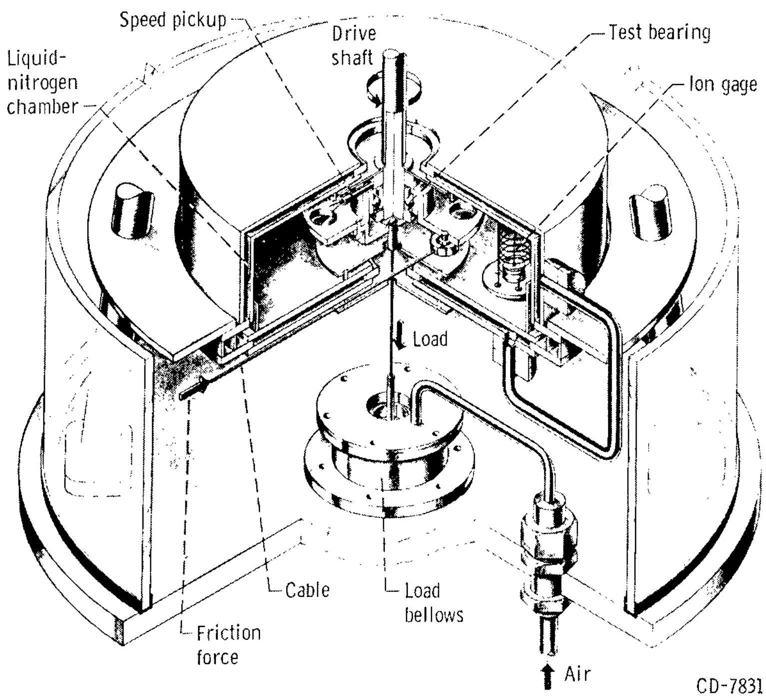
The test shaft was mounted vertically and supported by the drive-motor bearings. The drive-motor bearings and the stator were oil cooled to prevent overheating in vacuum.

A force transducer connected to the free bearing housing measured the test-bearing running torque. In the earlier tests an unbonded 48-ounce- (133-N-) capacity strain gage mounted inside a steel housing was used as a force transducer (fig. 2(a)). In operation, since the heat transfer from the elements in the vacuum environment was inadequate, the elements heated excessively and burned out. As a result of this excessive heating problem, the unbonded strain-gage force transducer was replaced (following the test of bearing 39) by a capacitance force transducer (fig. 2(b)), consisting of an accurately machined cantilever beam and a capacitance probe. The capacitance readout instrument had a voltage output proportional to the beam deflection. This output was suitable for a strip-chart recorder, oscilloscope, and digital counter. The amplitude, frequency, and waveform of the bearing torque were observed on an oscilloscope. The test-bearing torque could be measured within an average accuracy of 0.07 inch-ounce (0.05 cm-N).

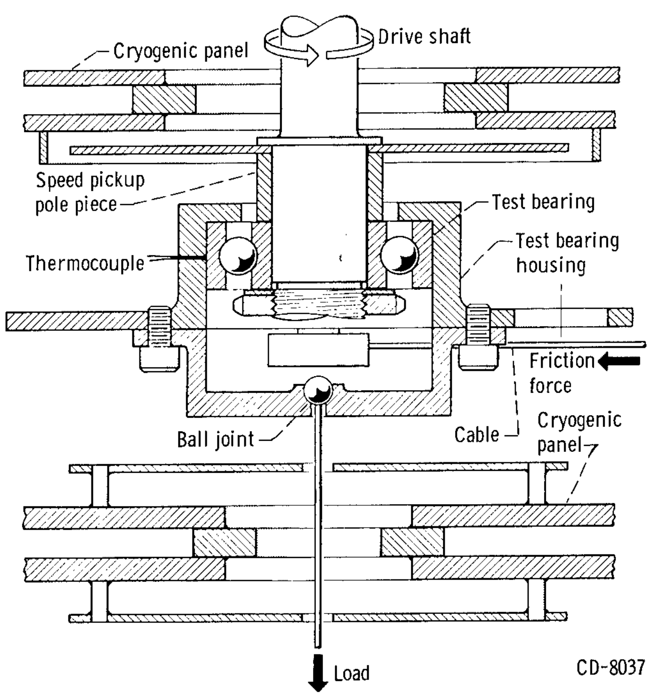
A thermocouple welded in the bearing housing and pressed against the outer race of the test bearing was used to indicate the relative bearing temperature. The thermocouple output was continuously recorded on a strip-chart recorder.

Vacuum System

The test apparatus was mounted in an 18-inch- (46-cm-) diameter bell-jar vacuum system using a 15-cubic-foot-per-minute (7.1-liter/sec) mechanical forepump and a 4-inch- (10-cm-) diameter oil-diffusion pump. The diffusion pump has a water-cooled baffle and a liquid-nitrogen cold trap. The test-bearing housing was mounted inside a double-walled chamber through which liquid nitrogen was circulated (fig. 1(a)). The double-walled chamber was designed to in-

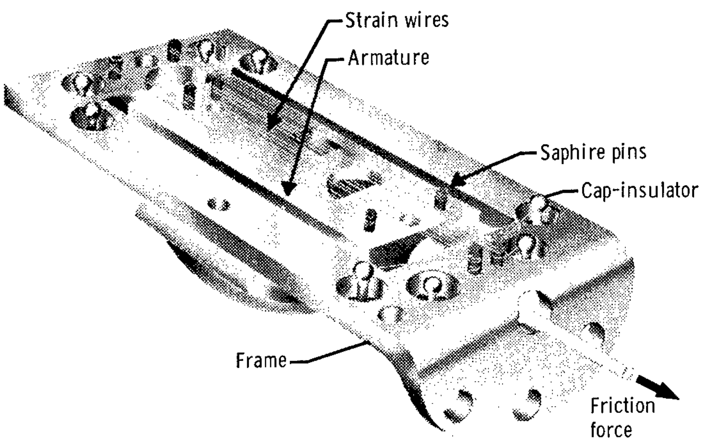


(a) Cutaway view.

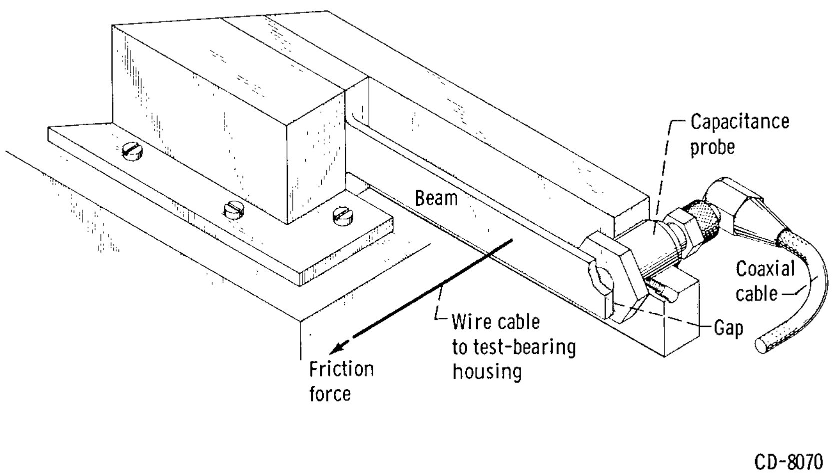


(b) Detailed view of test-bearing installation.

Figure 1. - Test apparatus.



(a) Unbonded strain-gage type.



(b) Capacitance type.

Figure 2. - Force transducers.

hibit backstreaming of diffusion-pump oil vapors to the test bearing and to capture oil evaporating from the bearing.

This cooling-chamber design simulated the environment of outer space, where an evaporated oil molecule would not be reflected back to the bearing. The pressure measurements in the system were made with a Bayard-Alpert nude ionization gage mounted inside the double-walled liquid-nitrogen chamber (fig. 1(a)).

Lubricants

For the initial series of tests, four liquid lubricants, (1) mix-bis-(mix-phenoxyphenoxy) benzene (polyphenyl ether), (2) methyl chlorophenyl polysiloxane, (3) di(2-ethylhexyl) sebacate, and (4) high-viscosity mineral oil, were selected for evaluation. A summary of some properties of these four lubricants is presented in table I (refs. 2, 5, and 6). (The evaporation rates for the mineral oil were determined on the apparatus of refs. 2 and 6.)

The sebacate was selected because knowledge of its established performance in air provides a basis for comparison with its operation in the vacuum environment (ref. 5). The other three lubricants have fairly low evaporation rates (table I). Also, these fluids have been tested (ref. 7) as lubricants in small instrument bearings subjected to light loads in a vacuum of the order of 10^{-6} torr. The results of the tests (ref. 7) indicated that the latter three fluids should be studied further for application in systems using larger size bearings.

Test Bearings

All tests were performed with size 204 (20-mm-bore), SAE 52100 steel bearings with an average hardness for the test races and balls of Rockwell C-63. One of the outer-race shoulders was removed to make the bearings separable. The inner- and outer-race conformities were 0.53 and 0.51, respectively. The test bearings were ABEC grade 5 with an average radial clearance of 0.0017 inch (0.0043 cm) (see table II). The various retainers were located on the inner-race with an average clearance of 0.016 inch (0.041 cm) and a ball pocket clearance of 0.009 inch (0.023 cm). Specific clearances for each bearing are given in table II. The bearing retainers were cotton-cloth phenolic of three different weaves. Some properties of the weaves, designated as fine, medium, and coarse, are given in table III.

PROCEDURE

Test Procedure

The ball and race sets were degreased initially in a solvent (1,1,1-trichloroethane), dried in a vacuum oven, weighed and measured, and the dimensions were recorded. The test bearing retainers were cleaned in three solvents, 1,1,1-trichloroethane, acetone, and alcohol, and then baked out, weighed, and put in a vacuum-impregnation chamber. The chamber was provided with a valved pumping line and a valved lubricant line. At a pressure of approximately 10^{-3} torr the pump valve was closed and the lubricant valve was opened so that the lubricant could flow into the chamber and completely immerse the retainer. After 1 hour at atmospheric pressure, the retainers were removed from the chamber and the excess lubricant was wiped off with an absorbent tissue. Preimpregnation and postimpregnation retainer weights were recorded to determine the amount of lubricant impregnated into the retainer (table IV). The other test bearing components were then recleaned by the procedure described previously for the retainers, the bearing was assembled, and the radial clearance was measured.

The test bearing was installed, and the test chamber was pumped down to the operating pressure of approximately 10^{-6} torr with liquid nitrogen circulating through the double-walled chamber. The torque measuring system was set at zero, the drive motor was started, and the bearing was operated at an initial load of approximately 35 pounds (16 kg). After approximately 5 minutes of operation the full axial load was applied, and the bearing was operated at this test condition for 1 hour unless the torque exceeded 20 inch-ounces (14.1 cm-N), in which case the test was terminated prematurely.

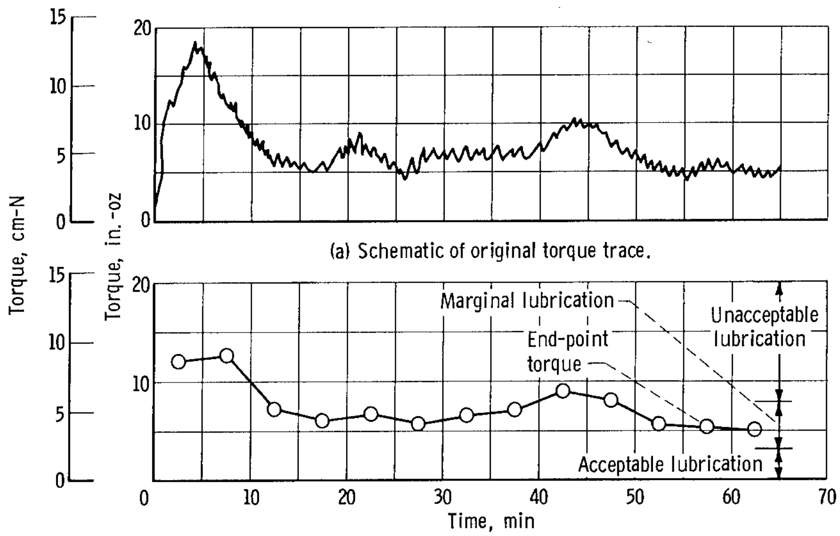
After each test, the test bearing components were measured and weighed and the data were recorded. A visual inspection of the bearing components was performed at magnifications of 10 to 25 to detect indications of wear and surface damage.

Torque Data Reduction

In order to achieve some understanding of the bearing performance, a method of reducing the original torque traces (fig. 3(a)) to a simpler, averaged form was used. The average-torque value for 5-minute intervals of the original torque trace was calculated and plotted at the midpoint of the interval (fig. 3(b)). For the purpose of evaluating the lubricants and retainer cloth weaves, limits for acceptable and unacceptable torque levels were chosen (fig. 3(b)), on the basis of a study by Nemeth and Anderson (ref. 8).

In this study, various liquid lubricants were evaluated in air at various temperatures for very small quantities of lubricant flow. The level of acceptable bearing torque was set at about 3.2 inch-ounces (2.3 cm-N). Therefore, any torque level in the present investigation below 3.2 inch-ounces (2.3 cm-N) was considered to be acceptable lubrication, as shown in figure 3. (This torque value corresponds to a coefficient of friction of 0.003 at 100 lb (45.4 kg), but at this load a coefficient of friction of 0.001 would represent a good lubrication level for an oil-lubricated rolling-element bearing.)

For the other limit, unacceptable lubrication, a torque level of 8 inch-ounces (5.65 cm-N) was chosen as the lower boundary, as suggested by the graphs in reference 8. (This torque level at 100 lb (45.4 kg) corresponds to a coefficient of friction of 0.008, eight times the value for a good lubrication level.) The region between these two limits is termed marginal lubrication.



(b) Average torque for 5-minute intervals.

Figure 3. - Illustration of torque averaging process and of defined terms.

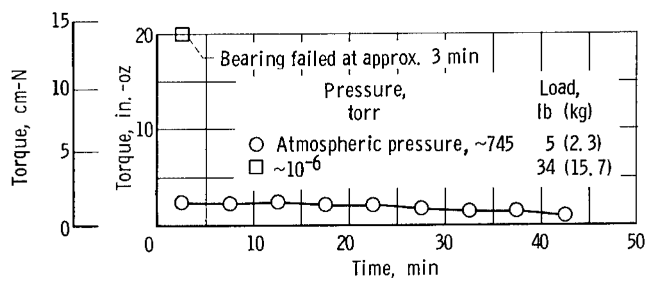


Figure 4. - Comparative performance of two dry, unlubricated ball bearings with cotton-cloth phenolic retainers. Speed, 3550 rpm.

DISCUSSION OF RESULTS

Unlubricated Bearing Performance

Figure 4 shows the average friction torque for clean, unlubricated bearings run at atmospheric pressure and at 10^{-6} torr. The figure shows clearly that, at atmospheric pressure, the bearing torque falls entirely within the acceptable lubrication band, whereas the bearing torque at 10^{-6} torr is totally unacceptable and, in fact, the bearing failed after 3 minutes. This shows the need for lubrication in a vacuum.

Experiments with Four Lubricants

Torque characteristics. - In the first series of experiments, the four lubricants were impregnated into the same retainer material. The torque histories of this first series are shown in figure 5. In general, the mineral-oil runs appear to have the lowest bearing torque among the four lubricants. Although the torque values were initially higher than for the other three lubricants, the mineral oil was the only lubricant which completed the full 60-minute running time at the three loads.

End-point torque. - End-point torque is defined as the 5-minute-average torque for the time interval 55 to 60 minutes (fig. 3(b)). The end-point torques for each of the four lubricants are plotted against load in figure 6. A general increase in torque with increasing load for all four lubricants is observed. This plot indicates, however, that the mineral oil provides torque as low as, or lower than, any of the other lubricants over the entire load range studied.

Torque roughness. - Another characteristic of the original torque trace, which is filtered out by the averaging process, is the torque roughness. Roughness is defined as the peak-to-peak amplitude of the bearing torque excursions. The magnitude of the roughness can provide, on a relative scale, an indication of the lubricating effectiveness of this retainer-impregnation technique. A portion of the original torque trace, showing high, moderate, and low roughness, has been reproduced in figure 7. These three roughness levels are indicated at the top of the average-torque traces in figure 5.

Effect of Cloth Weave

The bearings lubricated with the mineral oil ran with generally lower average torque, end-point torque, and torque roughness than did the bearings lubricated with the other fluids. Therefore, the mineral oil was chosen for the investigation of the effects of cloth weave.

Since the amount of lubricant available on the surface of the retainer depends largely on the balance between evaporation and the rate of flow from the interior of the retainer, some effect on bearing torque is expected when the quantity of lubricant available on the surface is varied by modifying its rate

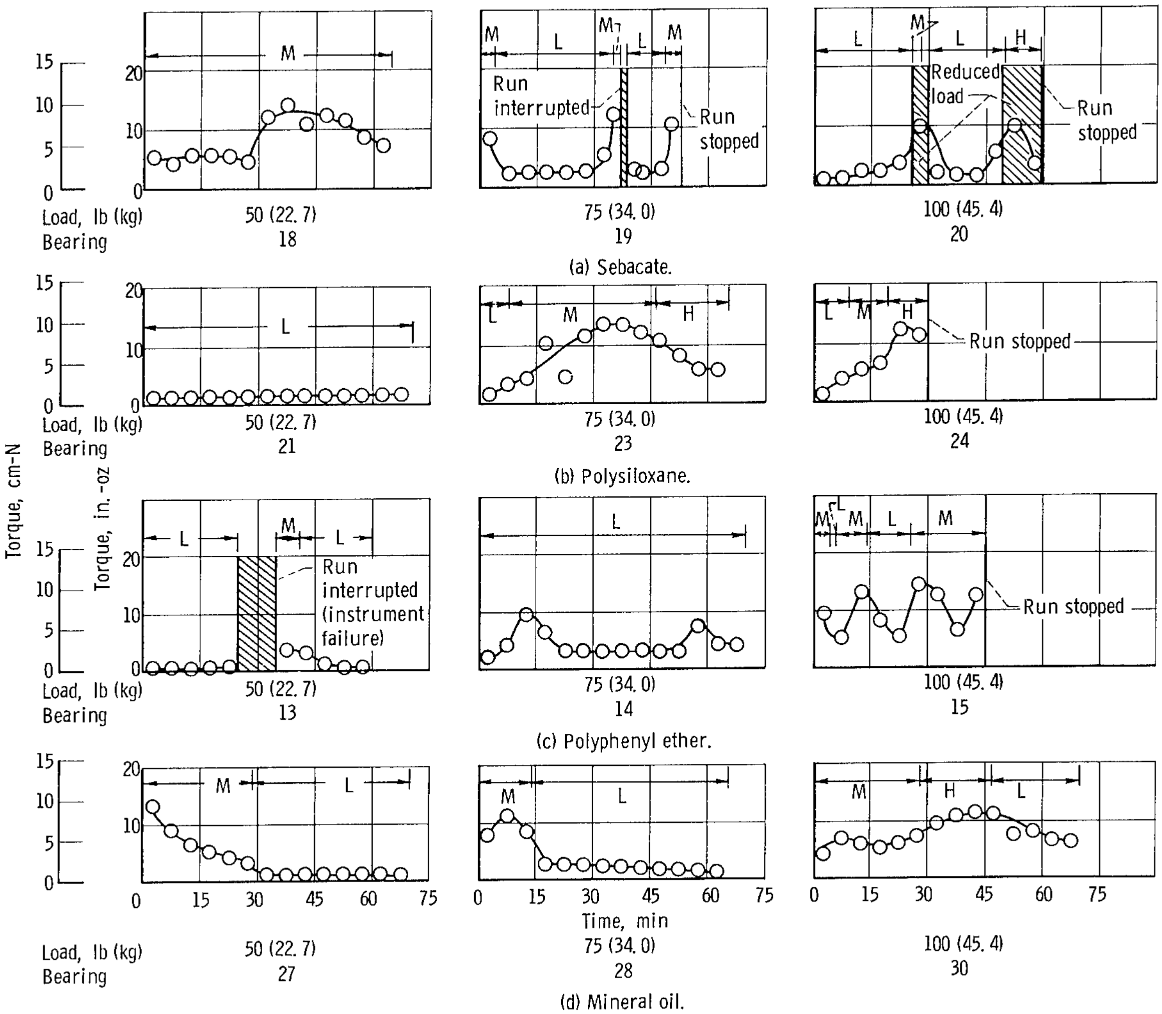


Figure 5. - Effect of lubricant on bearing torque. Four impregnants in fine-weave cotton-cloth phenolic retainer; pressure, approximately 1×10^{-6} torr; speed, 3550 rpm. Roughness levels: low (L), 0 to 1 inch-ounce (0 to 0.7 cm-N); moderate (M), 1 to 7 inch-ounces (0.7 to 4.9 cm-N); high (H), 7 to 20 inch-ounces (4.9 to 14.1 cm-N).

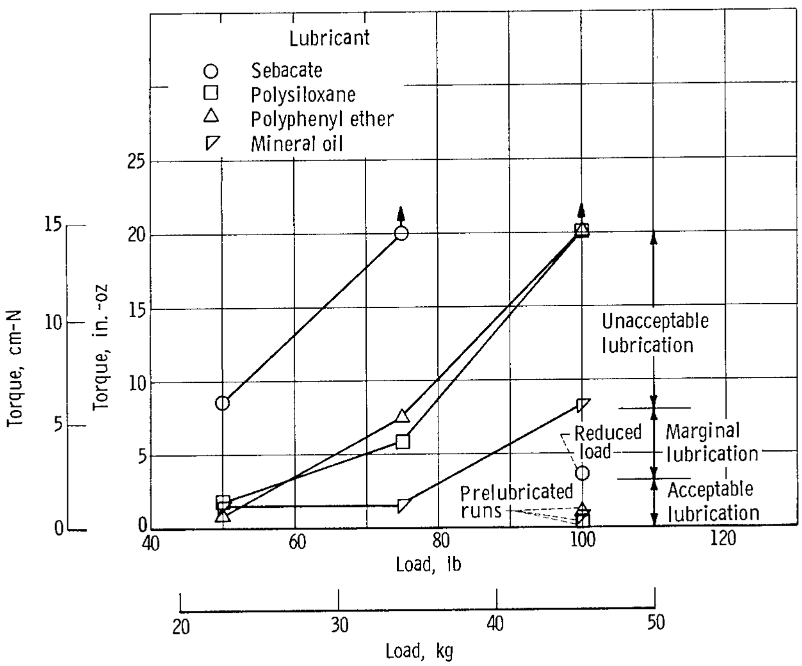


Figure 6. - Dependence of end-point torque on load for different lubricants. Fine-weave retainers.

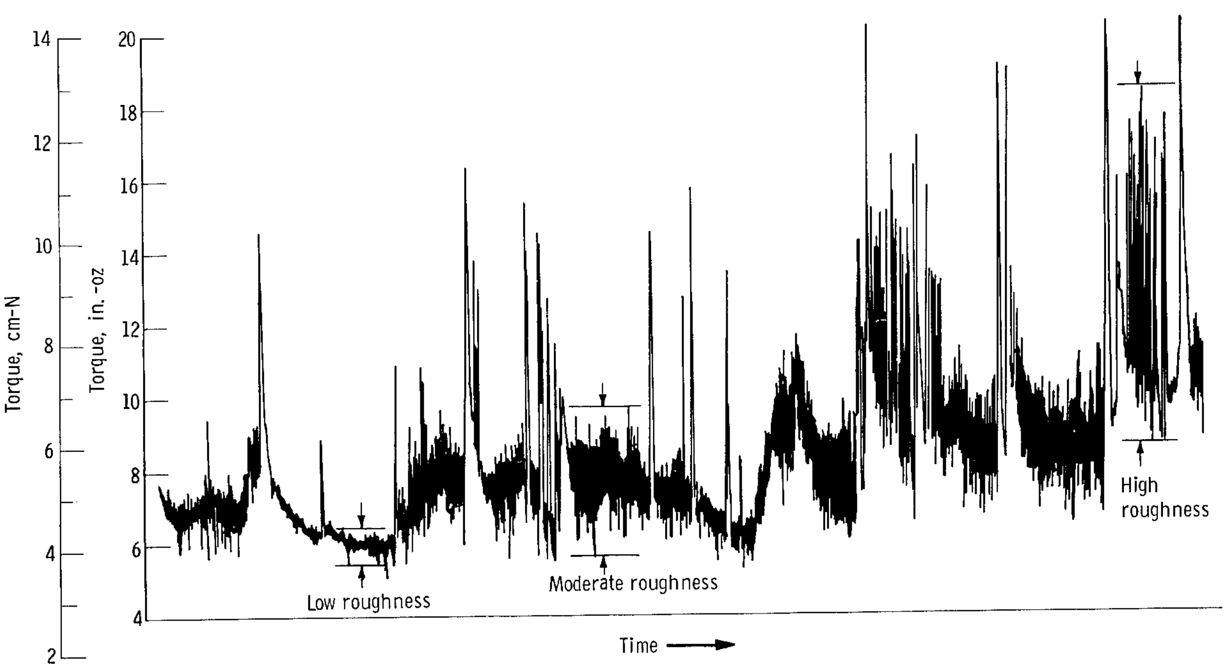


Figure 7. - Torque trace illustrating three levels of roughness.

of flow. Changing the weave of the cloth used in the laminate may vary the size of thread available for lubricant flow by capillary action.

Three cotton-cloth, phenolic-laminate materials, having different cloth weaves (fine, medium, and coarse), were made into retainers for the size 204 bearings. The manufacturer's data on some properties of the cotton cloths used are given in table III. More thread volume is available to the impregnant in the coarse-weave material than in the fine-weave material. However, examination of the lubricant weight column in table IV shows no systematic difference in the amount of impregnant in retainers of different weaves.

The torque histories for the series of runs made with the three cloth weaves are presented in figure 8. No significant differences in bearing torque were observed. Figure 9 shows that the end-point torque for the three weaves is approximately the same. Essentially, no differences in roughness levels for the three weaves were noted.

Prelubricated Runs

In all experiments discussed previously, the bearings were installed in the test rig with clean, unlubricated balls and races. The high torque frequently observed during the early part of the tests could be attributed to an initially poor lubricant distribution within the bearing. To verify the relation of the early, high torque with a poor lubricant distribution and to give some data on well lubricated bearings operating in vacuum, three prelubricated bearings were run at a load of 100 pounds (45.4 kg) for 60 minutes. These three bearings were prepared in a manner similar to all the others, except that each bearing was prelubricated with a few drops of the lubricant in the outer-race groove. The bearing was rotated slowly a few times to distribute the lubricant evenly around the balls and in both race grooves. For this set of runs, the three lubricants with lowest vapor pressures (mineral oil, polyphenyl ether, and polysiloxane) were impregnated into the coarse-weave cotton-cloth phenolic retainers.

The very low end-point torques shown in figure 6 of approximately 2.6 inch-ounces (1.8 cm-N) represent the bearing torque developed throughout the run, since the torque was nearly constant.

The roughness was negligible for all three lubricants during the period of full loading.

These results indicate that the impregnated case alone does not feed the lubricant at a rate fast enough to provide good lubrication at the bearing surfaces, particularly during the initial phase of operation. The impregnated retainers used were evidently not optimum for this application.

A porous material designed to hold and feed a fluid lubricant effectively can probably be developed. The use of such a material for bearing retainers would permit the lubrication technique explored here to be more effective.

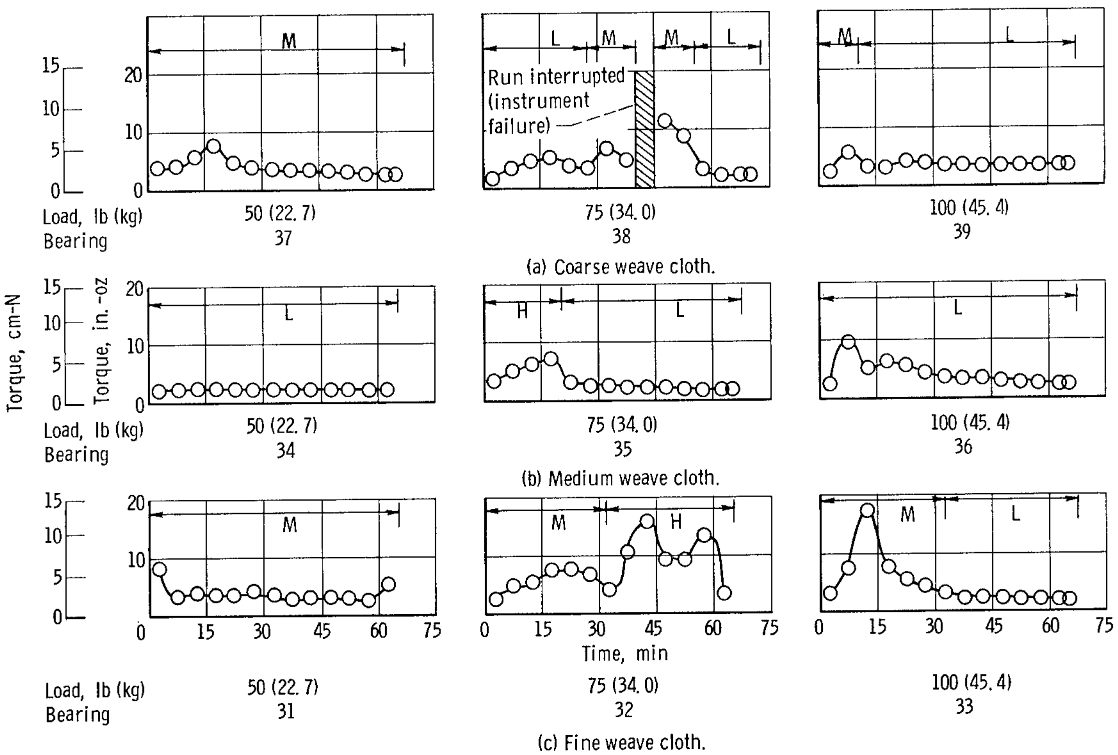


Figure 8. - Effect of cloth weave on bearing torque. Mineral-oil impregnant in cotton-cloth phenolic retainer; pressure, approximately 1×10^{-6} torr; speed, 3550 rpm. Roughness levels: low (L), 0 to 1 inch-ounce (0 to 0.7 cm-N); moderate (M), 1 to 7 inch-ounces (0.7 to 4.9 cm-N); high (H), 7 to 20 inch-ounces (4.9 to 14.1 cm-N).

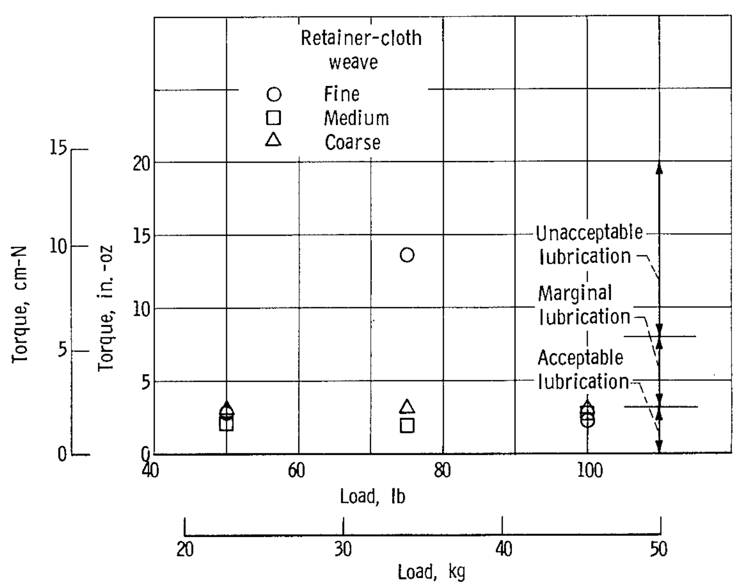


Figure 9. - Dependence of end-point torque on load for different retainer-cloth weaves. Mineral-oil lubricant.

Torque-Temperature Correlation

The torque level is also reflected in the temperature recorded by a thermocouple against the outer race (shown in fig. 1(b)). Figure 10 is a plot of the torque and temperature histories of a typical bearing at a 100-pound (45.4-kg) load. Figure 10 shows that, for low torque, the slope of the temperature-time plot is low; as the torque increases, so does the slope of the temperature-time plot. There is a noticeable time lag between the increase in torque and the increase in slope on the temperature curve because of the time required for heat to flow to the outer-race surface. The temperatures measured are to be considered only as relative temperature indications, since the thermocouple was welded to the bearing housing and thus the indicated temperature depended on the heat transfer from the race to the housing.

Bearing-Clearance Effects

A temperature rise in a bearing usually reduces its operating clearance. If a bearing has insufficient radial clearance for a specified running condition, it will seize. On the other hand, too much radial clearance may permit the balls to ride up on the shoulders of the race grooves and adversely affect bearing operation. The average initial radial clearance for the bearings of this investigation was 0.0017 inch (0.0043 cm), which is three to nine times the

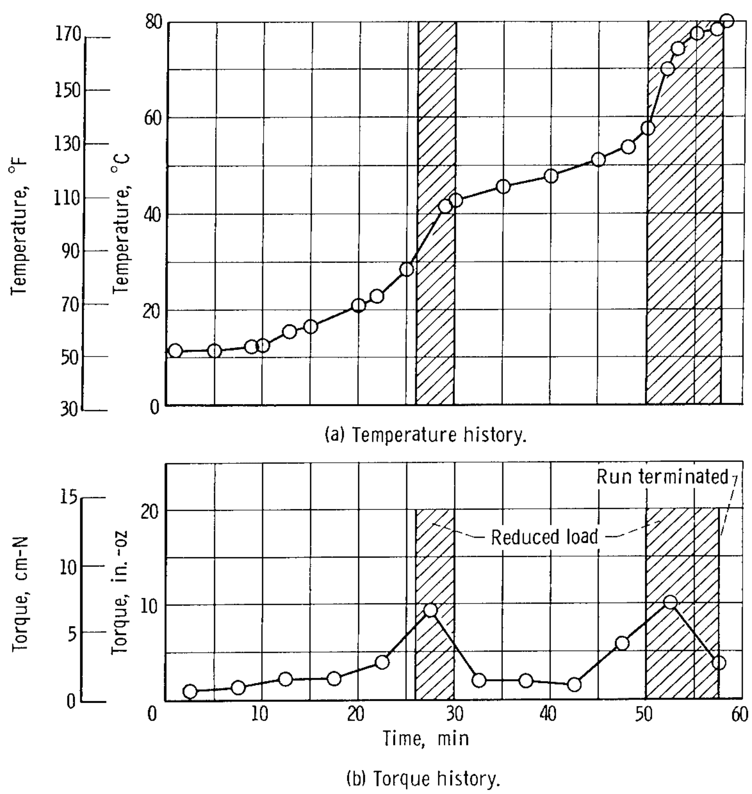


Figure 10. - Comparison of torque and temperature histories for bearing 20. Load, 100 pounds (45.4 kg); pressure, 10^{-6} torr; speed, 3550 rpm.

radial clearance usually used in off-the-shelf 20-millimeter-bore bearings. None of the bearings tested seized, nor was there any evidence of the balls riding on the race-groove shoulders. Therefore, the clearance range of the bearings was satisfactory.

Heat Generation and Dissipation

In a vacuum, the problem of dissipating the heat generated in a bearing is more severe than in a normal atmospheric environment. Of the three modes of heat transfer, only conduction and radiation can operate in the absence of a gaseous environment. As shown in table IV, the final temperatures of the test bearings, measured at the outer race, were sometimes high (above 200° F (93° C)) for the loads applied. An estimation of the radiation heat transfer showed that less than 1 percent of the heat generated was rejected by radiation. Therefore, the principal mode of heat rejection is conduction. In a vacuum, however, the conduction heat-transfer coefficient across an interface can be reduced considerably, so that even the conduction mode is not as efficient in vacumm as it is in air. These heat-transfer deficiencies point to the need to design a suitable heat sink for the frictional heat generated in a bearing operating in a vacuum environment.

SUMMARY OF RESULTS

Four liquid lubricants (a polyphenyl ether, a polysiloxane, a sebacate, and a high-viscosity mineral oil) were evaluated in 20-millimeter-bore ball bearings with cotton-cloth phenolic retainers at pressures of about 10^{-6} torr. Three different cloth weaves (fine, medium, and coarse) were used with the best of the four lubricants. The bearings were operated at 3550 rpm with thrust loads of 50 to 100 pounds (22.7 to 45.4 kg) for 1 hour. The investigation produced the following results:

1. The high-viscosity mineral oil was the best of the four lubricants studied, on the basis of the lower end-point torque, lower torque roughness, and longer satisfactory operation obtained with this fluid.

2. No significant differences in bearing torque were found among the three cloth weaves when the mineral oil was used as the impregnated lubricant.

3. Lubrication by the retainer-impregnation technique was found inadequate under the conditions of this investigation. Heat accelerates the distribution of lubricant to the rolling and sliding surfaces by decreasing lubricant viscosity and thereby reduces bearing torque during operation. The runs with prelubricated bearings show much better performance than those with nonprelubricated bearings, which indicates that the rate of lubricant flow out of the cotton-cloth-phenolic cage material is not fast enough to provide good lubrication.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, November 30, 1965.

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TABLE I. - TYPICAL LIQUID-LUBRICANT PROPERTIES

Property	Lubricant			
	Polyphenyl ether (mix-bis(mix-phenoxyphenoxy) benzene)	Polysiloxane (methyl chlorophenyl polysiloxane)	Sebacate (di(2-ethylhexyl) sebacate with additives)	Mineral oil (high-viscosity)
Molecular weight	446.5	800 to 6000	426.7	1620
Specific gravity	[a]1.204	[b]1.050	[b]0.914	[a]0.921
Pour point °F (°C)	40(4)	-100(-73)	<-75(-59)	50(10)
Viscosity, cS, at Room temperature 100° F (38° C) 210° F (99° C)	 [c]1000 363 13	 [d]70 52 16	 [e]21.0 [e]11.4 [e]3.2	 [f]18 100 8 000 190
Free-surface evaporation rate, g/(cm^2)(sec), at ~10^{-7} torr and 100° F (38° C) 200° F (93° C) 300° F (149° C)	 [g]5×10^{-8} [g]80 [g]2000	 [g]30×10^{-8} [g]80 [g]300	 [h,i]0.6×10^{-8} [h]500 -----------	 0.7×10^{-8} 7 10

[a]At 68° F (20° C) relative to water at 68° F (20° C).
[b]At 77° F (25° C) relative to water at 77° F (25° C).
[c]At 80° F (27° C).
[d]At 77° F (25° C).
[e]Data from ref. 5.
[f]At 70° F (21° C).
[g]Data from ref. 2.
[h]Data from ref. 6.
[i]At 55° F (13° C).

TABLE II. - BEARING SPECIFICATIONS

[Angular-contact 20 mm-bore ball bearings with one-piece cotton-cloth phenolic-laminate retainers and SAE 52100 balls and races; inner-race conformity, 0.53; outer-race conformity, 0.51; ball diameter, 9/32 in. (7.144 mm); number of balls, 11; nominal contact angle, 25°.]

Bearing	Radial clearance		Retainer locating-surface clearance		Ball-pocket clearance	
	in.	cm	in.	cm	in.	cm
9	0.0018	0.0046	-----	-----	0.009	0.023
10	.0014	.0036	0.005	0.013	.017	.043
13	.0012	.0030	.022	.056	.016	.041
14	.0014	.0036	.021	.053	.016	.041
15	.0014	.0036	.022	.056	.019	.048
18	.0016	.0041	.018	.046	.017	.043
19	.0012	.0030	.020	.051	.009	.023
20	.0017	.0043	.021	.053	.006	.015
21	.0015	.0038	.021	.053	.007	.018
22	.0013	.0033	.021	.053	.010	.025
23	.0016	.0041	.021	.053	.016	.041
24	.0015	.0038	.020	.051	.007	.018
27	.0016	.0041	.022	.056	.007	.018
28	.0014	.0036	.022	.056	.008	.020
29	.0015	.0038	.023	.058	.006	.015
30	.0016	.0041	.025	.064	.006	.015
31	.0019	.0048	.012	.030	.008	.020
32	.0017	.0043	.011	.028	.004	.010
33	.0021	.0053	.013	.033	.007	.018
34	.0022	.0056	.009	.023	.006	.015
35	.0021	.0053	.016	.041	.006	.015
36	.0021	.0053	.011	.028	.006	.015
37	.0026	.0066	.015	.038	.006	.015
38	.0024	.0061	.011	.028	.006	.015
39	.0017	.0043	.010	.025	.007	.018
40	.0020	.0051	.010	.025	.007	.018
41	.0018	.0046	.008	.020	.005	.013
45	.0017	.0043	.008	.020	.004	.010

TABLE III. - COTTON CLOTH PROPERTIES

Weave	Thread count (a)		Cloth layer count (b)		Cloth weight (a)		Relative thread volume (c)
	Threads/in.	Threads/cm	Layers/in.	Layers/cm	Oz/yd^2	g/m^2	
Fine	80 by 80	31.5 by 31.5	125.2	49.3	3.00	101.7	1.00
Medium	48 by 48	18.9 by 18.9	74.4	29.3	6.00	203.4	1.19
Coarse	38 by 39	15.0 by 15.4	73.6	29.0	8.00	271.2	1.57

[a]Manufacturer's data.
[b]Actual count.
[c]Calculated with constant density assumed for thread.

TABLE IV. - SUMMARY OF BEARING TEST CONDITIONS

Bearing	Load, lb	Load, kg	Cotton-cloth phenolic retainer material: Weave	Cotton-cloth phenolic retainer material: Manufacturer	Lubricant	Lubricant weight, mg	Pressure range, torr	Final outer-race temperature, °F	Final outer-race temperature, °C	Comments
9	34	15.6	Fine	A	Polyphenyl ether	----	0.2 to 100×10^{-6}	203	95	
10	33	15.0				18.6	.2 to 300	158	70	
13	50	22.7				32.5	.3 to 20	75	24	Test interrupted at 28 and 34 min
14	75	34.0				40.3	.2 to 40	186	86	
15	100	45.4				36.8	.3 to >600	235	113	Test stopped at 42 min because of excessive torque and pressure
18	50	22.7	Fine	A	Sebacate	45.9	0.2 to 30×10^{-6}	259	126	
19	75	34.0				33.3	.07 to >10	156	69	Test stopped at 52 min because of excessive torque
20	100	45.4				18.0	.1 to >10	181	83	Test stopped at 57 min because of excessive torque
21	50	22.7	Fine	A	Polysiloxane	----	0.1 to 10×10^{-6}	106	41	
22	75	34.0				26.8	.1 to 600	376	191	
23	75	34.0				----	.3 to 70	269	132	
24	100	45.4				18.7	.2 to >10	138	59	Test stopped at 28 min because of excessive torque
27	50	22.7	Fine	A	Mineral oil	28.6	0.2 to 2×10^{-6}	121	49	
28	75	34.0				46.8	1 to 4	152	67	
29	100	45.4				63.1	.1 to 10	287	142	
30	100	45.4				70.2	.1 to 7	226	108	
31	50	22.7	Fine	B	Mineral oil	63.7	0.3 to 4×10^{-6}	144	62	
32	75	34.0				57.3	.3 to 4	164	73	
33	100	45.4				67.7	.2 to 5	221	105	
34	50	22.7	Medium	B	Mineral oil	36.3	0.4 to 1×10^{-6}	116	47	
35	75	34.0				----	.4 to 3	149	65	
36	100	45.4				58.7	.4 to 5	196	91	
37	50	22.7	Coarse	B	Mineral oil	55.6	0.4 to 2×10^{-6}	142	61	
38	75	34.0				60.7	.3 to 7	134	57	Test interrupted at 45 min
39	100	45.4				55.5	.3 to 3	180	82	
40	100	45.4				44.8	1 to 7	205	96	Run continued as endurance test; support bearings failed at 8.6 hr; outer-race temperature, 242° F (117° C)
41	75	34.0				23.8	3 to 5	100	38	
45	34	15.7	Coarse	B	None	----	1 to 7×10^{-6}	48	9	Test stopped at 3.5 min because of excessive torque

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