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ABSTRACT

A parametric study was performed with 120-mm bore angular-contact ball bearings having a nominal contact angle of 20°. The bearings either had an inner- or an outer-race land riding cage. Lubrication was by recirculating oil jets. The oil jets either had a single or dual orifice. Thrust load, speed, and lubricant flow rate were varied. Test results were compared with those previously reported and obtained from bearings of the same design which were under-race lubricated but run under the same conditions. Jet lubricated ball bearings were limited to speeds less than 2.5×10^{6} DN. Bearings having inner-race land riding cages produced lower temperatures than bearings with outer-race land riding cages. For a given lubricant flow rate dual orifice jets produced lower bearing temperatures than single orifice jets. However, under-race lubrication produced under all conditions of operation lower bearing temperatures with no apparent bearing speed limitation.

INTRODUCTION

Rolling-element bearings for advanced airbreathing aircraft engine applications are anticipated to approach speeds of 3 million DN within end of the next decade (DN is defined as the bearing bore in mm multiplied by the shaft speed in rpm). For large size bore bearings these speeds are already attainable with long bearing life [1]. In order to overcome the detrimental centrifugal effects in extreme speed applications, oil is introduced directly to the interior of the bearing through radially directed passageways in the inner race [2]. The combination of passages provide lubricant directly to the balls and the cage riding surfaces.

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Another method of lubricating bearings is oil-gas mist or once-through vapor lubrication. A variety of mist lubricators and lubrication systems are commercially available. These systems offer the advantages of reduced complexity because they eliminate pumps and scavenge lines and reduce operating torque. Their disadvantages lies in their inability to remove heat from the bearings. An oil-gas mist obviously does not have the heat removal capability of high flow rate recirculating systems so that a mist lubricated bearing will run hotter. In high-speed, highly-loaded bearings, mist flow requirements make them uneconomical even when thermal equilibrium can be maintained [3].

Recirculating jet or nozzle systems require more complex hardware than do oil-gas mist systems. Recirculating jet lubrication is most commonly used on airbreathing turbojet engines today.

There have been many studies conducted to determine optimum lubricant jet arrangements. Typical of these is [4] in which the efficiency of single, multiple and multiple-opposed jet arrangement was studied.

The effects of oil-mist and recirculating oil-jet lubrication on bearing operating temperatures and torque was studied in [5]. For a conventional 75-mm bore ball bearing running at 20,000 rpm (1.5 million DN), the outer-race temperature was 394 K (250° F) with recirculating oil jet lubrication and 447 K (345 F) with oil-air mist lubrication. However, the torque was 1.3 newton-meter (11.5 in.-lb) with oil jet lubrication and only 0.4 newton-meter (3.5 in.-lb) with oil-air mist lubrication.

Recent work [6] with 30-mm bore ball bearings studied the effect of cage location as well as jet lubrication. Even with optimum jet arrangements, there is a definite limiting DN value above which a jet lubrication system is no longer adequate. Unless significant cooling was provided bearing failure due to overheating occurred at high speeds. For inner-race and outer-race riding cages, the limiting speed was 1.5 million DN and 2.4 million DN, respectively.

The research reported herein was undertaken to investigate the performance of jet lubricated 120-mm bore angular-contact ball bearings made from vacuum-induction melted, vacuum-arc remelted AISI M-50 steel. The bearings were fitted with either an inner- or outer-race riding cage. The objectives were to (a) determine the limiting speed of large bore ball bearings with jet lubrication, (b) determine the effect of cage location and lubricant jet on bearing temperature and power loss, and (c) compare the operating characteristics of jet lubricated and underrace lubricated ball bearings.

HIGH-SPEED BEARING TESTER

A schematic of the high-speed, high-temperature bearing tester used in these tests is shown in Fig. 1. This tester is described in detail in [7, 8]. Essentially the tester consists of a shaft to which two test bearings are attached. Loading is supplied through a system of 10 springs which thrust load the bearings. Drive of the test rig is accomplished by a flat belt (not shown in the illustration).

Lubrication is provided to the test bearings through a jet feed lubrication system. Two lubricant jets positioned 180° apart were used for each bearing. The jets had either a double or single orifice as shown in Fig. 2. There was a 3.1 mm (0.122 in.) gap distance from the face of the outer-race to the orifice. The orifice diameter was 3.2 mm (0.125 in.) and 1.6 mm (0.062 in.) for the single and double orifice jets, respectively. The pump is capable of circulating the oil through the system at rates up to 2.8×10^{-2} cubic meter (7.5 gal) per minute. Gravity drainage for the lubricant is provided by a single exit located under each test bearing as well as by an exit from the bellows area in the center of the bearing assembly.

Instrumentation provided for automatic shutoff by monitoring and recording bearing temperatures, oil temperature, and bearing vibration, as well as support bearing lubricant flow and pressure. A change in any of these parameters from those programmed for the test conditions, terminated the test. An infrared pyrometer was used to measure inner-race temperature, using a sapphire window sight tube aimed at the inner race of the first test bearing. The oil flow was established by a flowmeter. The bearing outer-race and lubricant inlet and outlet temperatures were measured by thermocouples and continuously recorded by a strip chart recorder.

TEST BEARINGS

The test bearings were ABEC-5 grade, split inner-race 120 mm bore ball bearings. The inner and outer races, as well as the ball, were manufactured from one heat of double vacuum-melted (vacuum-induction melted, vacuum-arc remelted- AISI M-50 steel. The chemical analysis of the particular heat is shown in Table 1. The nominal hardness of the balls and races was Rockwell C-63 at room temperature. Each bearing contained 15 balls, 2.0638 cm (13/16 in.) in diameter. The cage which could be replaced was of one-piece construction and could be either inner-race or outer-race land riding depending on the test parameter to be studied (see Fig. 2). The cage was balanced within 3 gm-cm (0.042 oz-in.).

The retained austenite content of the ball and race material was less than 3 percent. The inner- and outer-race curvatures were 54 and 52 percent, respectively. The nominal contact angle was 20°. All components with the exception of the cage were matched within ±1 Rockwell-C point. This matching assured a nominal differential hardness in all bearings (i.e., the ball hardness minus the race hardness, commonly called H) of zero [9]. Surface finish of the balls was 2.5 cm (1 microin.) AA, and the inner and outer raceways were held to a 5 cm (2 microin.) AA surface finish.

LUBRICANT

The oil used for the parametric studies as well as for the subsequent long-time high-speed (3 million DN) endurance tests, was a 5-centistoke neopentylpolyol (tetra) ester. This is a type II oil, qualified to MIL-L-23699 as well as to the internal oil specifications of most major aircraft-engine producers. The major properties of the oil are presented in Table 2 and a temperature-viscosity curve is shown in Fig. 3.

TEST PROCEDURE

The test procedure was adjusted according to the test conditions to be evaluated. Generally, a program cycle was defined which would allow the evaluation of a number of conditions without a major interruption. With the exception of speed, all test parameters such as load, lubricant flow rate, and oil temperature could be adjusted while the tester was in operation. During operation, the tester was allowed to reach equilibrium condition before the data were recorded.

Power loss per bearing was determined by measuring line-to-line voltage and line current to the test-rig drive motor. Motor drive power was then calculated as a function of line current, reflecting bearing power usage at the various operating speeds.

Data were recorded at three bearing thrust loads, these being 6672, 13, 340, and 22, 240 newtons (1500, 3000, and 5000 lb). Oil inlet temperature was maintained constant at 394 K (250° F).

RESULTS AND DISCUSSION

Effect of Cage Location - The effect of cage location on bearing temperature and power consumption are shown in Figs. 4 to 8. Figures 4(a) and (c) show the effect of load and bearing inner and outer race temperatures for nominal speeds of 12, 000 and 16, 700 rpm and at flow rates of 1.9×10^{-3}, 3.8×10^{-3}, 5.7×10^{-3}, and 8.3×10^{-3} cubic meter (0. 5, 1. 0, 1. 5, and 2. 2 gal) per minute for a bearing with an inner-race land riding cage.

In general, at the 16, 700 rpm speed (Fig. (4(c)) the inner-race temperature is as much as 20 K (35 F) higher than the outer race. The minimum required flow to maintain temperature stability for the test conditions at this speed was 3.8×10^{-3} cubic meters (1. 0 gal) per minute.

At the lower speed of 12, 000 rpm (Fig. 4(a)) temperature stability could be maintained at flow rates as low as 1.9×10^{-3} cubic meters (0. 5 gal) per minute. There was generally not more than 5 K (9 F) difference in temperature between the inner and outer races at this speed. However, as the load increased, the inner-race temperature increased at a greater rate than the outer race temperature whereby the highest temperature occurred at the inner race at the higher load conditions.

Results of tests with the outer-race riding cage are shown in Figs. 4(b) and (d). For the 12, 000 rpm speed (Fig. 4(b)) the difference in temperature between the inner and outer races ranged from approximately 12 to 22 K (22 to 40 F). The inner-race temperature was higher than the outer-race temperature for all four lubricant flow rates. The inner-race temperature at a flow rate of 1.9×10^{-3} cubic meters (0. 5 gal) per minute generally exceeded 472 K (390° F). At the flow rate of 8.3×10^{-3} cubic meters (2. 2 gal) per minute, the inner-race temperature was approximately 433 K (320° F).

For the outer-race riding cage the bearings could only be run over the load range with lubricant flow rates 5.7×10^{-3} and 1.3×10^{-3} cubic meters (1. 5 and 2. 2 gal) per minute. At a flow rate of 5.7×10^{-3} cubic meters (1.5 gal) per minute the inner-race temperature ranged from approximately 466 to 489 K (380° to 420° F). Hence, at 16,700 rpm the lubricant flow rate was approximately three times that required at 12,000 rpm to maintain the same temperature. For the conditions reported, the inner race exhibited the higher temperatures.

A comparison of inner-race temperatures for bearings with outer- and inner-race land riding cages are shown in Figs. 5 and 6 as a function of lubricant flow. In all cases the bearings with the inner-race land riding cage had lower temperatures than bearings with outer-race land riding cages. The difference in temperature was most pronounced at the lower speed of 12,000 rpm and at the lower loads and flow rates. The maximum difference in bearing temperature between the bearings having an inner-race land riding cage and those having an outer-race riding cage was approximately 40 K (72 F) at a bearing thrust load of 6672 newtons (1500 lb) for both speed conditions.

A comparison of the power generation between the bearings having the outer-race and inner-race land riding cages as a function of lubricant flow rate is shown in Fig. 7. At both the 12,000 and 16,700 rpm speed conditions with the outer-race land riding cage there was little difference in power loss per bearing with load. However, for the inner-race and riding cage, the difference was as much as 3 kW (4 hp) between the 6672 and 22,240 newtons (1500 and 5000 lb) thrust loads.

Power generation with the bearings having the inner-race land riding cages appeared, in general, to be less than that with the bearings having the outer-race land riding cages. At 12,000 rpm the difference in power loss was approximately 2 kW (2.7 hp) at the lower thrust loads. However, at the higher thrust loads of 22,240 newtons (5000 lb) the inner-race land riding cage bearing power loss was approximately 2 kW (2.7 hp) higher than the outer-race land riding cage bearing. At the higher speed of 16,700 rpm the differences in power losses was more marked. The power loss with all three thrust loads for the bearing with the inner-race land riding cage was as much as 3 kW (4 hp) lower than the bearing with the outer-race land riding cage.

The heat transferred to the from the bearings is shown in Fig. 8 as a function of jet lubricant flow rate. As would be expected, the total heat transfer to the lubricant increased with increasing lubricant flow. The marked distinction in power loss between bearings with inner- and outer-race land riding cages is not as apparent from these data as those recorded for bearing power loss shown in Fig. 7.

In [10] it was reported that there was as much as a 3 kW (4 hp) higher heat rejection rate to the lubricant than was indicated by the bearing power loss data (which is based upon shaft horsepower measurements). It was concluded in [10] that there could be sufficient inaccuracies in the determination of tare losses in the test rig and/or shaft horsepower measurements to account for the power differences. The data of Fig. 8 was compared with those of Fig. 7. At 12,000 rpm, the measured bearing loss encompassed most of the data representing the heat transferred to the lubricant. However, at the 16,700 rpm speed, the heat transfer to the lubricant was approximately $3\frac{1}{2}$ kW (4.7 hp) greater than the measured power loss. This difference correlates with the possible inaccuracies in the bearing power measurements previously reported in [10].

Effect of Jet Location - It has been reported that the number of lubricant jets, jet location, orifice size, jet distance from the bearing and jet velocity affect bearing temperature and power loss [4, 6]. It may be concluded that bearing temperature and power loss is a function of the amount of lubricant which penetrates the bearing cavity. In the tests reported in the previous section, two double orifice jets were used for each bearing. It is general practice for some users, however, to employ single orifice jets aimed at the largest gap or spacing between the race and the cage (Fig. 2(c)). Further, for higher speed operation, there are some bearing users who generally specify an outer-land location cage. Since the bearing with the outer-race land riding cage had the higher race temperatures and somewhat higher power losses with the dual orifice jet, the tests were rerun with two single jets per bearing as before except that each jet had a single orifice of 3.2 mm (0.125 in.) diameter (Fig. 2(c)). The orifice diameter for the dual orifice jets was 1.6 mm (0.062 in.).

Race temperatures as a function of lubricant flow for the single orifice jet lubricated outer-land riding cage bearings are shown in Fig. 9(a). The outer-race temperatures are less than the inner-race temperatures by approximately 17 to 22 K (30 to

40 F). Comparing the single orifice jet with the dual orifice jet lubricated bearing in Fig. 9(b), the dual orifice jet lubricated bearings having outer-race land riding cages were as much as 10 to 17 K (18 to 30 F) cooler than the single orifice jet lubricated bearings at the high lubricant flow rates. The most marked difference in race operating temperatures occurred between the single orifice jet lubricated outer-race land cage bearing and the dual orifice jet lubricated inner-race land riding cage bearing (Fig. 9(c)). The inner-race land cage riding bearing was as much as 36 K (65 F) lower in temperature than the outer-land riding cage bearing.

Power loss as a function of lubricant flow is given in Fig. 10 for the single orifice jet lubricated bearings. These data when compared with those of Fig. 7 would suggest no significant difference between the bearing power loss of the single and dual orifice jet lubricated bearings. However, at the very low flow rates there is a suggestion that the dual orifice jet lubricated bearings may have a higher power loss than the single orifice jet lubricated bearings. If this were so, it may be possible to conclude that more lubricant was able to penetrate the bearing cavity using dual orifice jets. Thus, for a given lubricant flow rate, lubricant efficiency may be enhanced with the use of dual orifice jets.

It is stated in [6] that the velocity of an oil jet should be between 10 and 20 meters (394 and 788 in.) per second in order to lubricate and operate a 30-mm bore ball bearing at ultra-high speeds. For the tests reported herein the jet velocity was 4 and 17.8 meters (159 and 701 in.) per second for flow rates of 1.9×10^{-3} and 8.3×10^{-3} cubic meter (0.5 and 2.2 gal) per minute, respectively, for the dual orifice jets. For the single orifice jets, the jet velocity was 2 and 8.8 meters (78 and 345 in.) per second, respectively. The higher jet velocity for the dual orifice jets probably accounts for the larger volume of lubricant penetrating the bearing cavity resulting in lower bearing temperature.

Speed Limitations - A question which remains to be answered with definite certainty is "What is the speed limitation for reliable operation of jet lubricated, large-bore, angular-contact ball bearings?" Having established that inner-race land riding cages lubricated with dual orifice jets produced the lowest race temperature at 12,000 and 16,700 rpm, further tests were conducted at a shaft speed of 20,800 rpm with the inner-race land riding cage. The criteria for successful operation comprised the ability of the bearing to operate over the spectrum of bearing thrust loads which may be reasonably expected in actual turbojet engine applications. The results of these tests indicated that successful operation at a thrust load of 22,240 newtons (5000 lb) and a speed of 16,700 rpm (2 million DN) could only be obtained at a flow rate of 3.8×10^{-3} cubic meter (1.0 gal) per minute or higher. At a speed of 20,800 rpm (2.5 million DN), the bearing temperature could be stabilized at 495 K (431° F) at a thrust load of 6672 newtons (1500 lb) and a flow rate of 8.3×10^{-3} cubic meter (2.2 gal) per minute (see Fig. 11). At the lower flow rates, the bulk bearing temperature began to exceed the estimated operating temperature limitations of the lubricant (495 K (431° F)) [11]. This temperature limitation is based upon the lubricants thermal and oxidative stability and its ability to form an adequate elastohydrodynamic film at the bearing operating temperature.

From these test results, it may be concluded that the limiting speed of the bearings at the optimal lubricant conditions with an inner-race land riding cage is less than 2.5 million DN. However, high oil jet velocities might extend the maximum permissible operating bearing speeds to values greater than 2.5 million DN. This would be because of better oil penetration and more efficient cooling. Practical limits of jet size and supply pressure would have to be determined.

Effect of Under-Race Lubrication - Previous research reported in [1, 10, 12] showed that bearings could be operated reliably for very long time periods at speeds to 3 million DN with lubrication of the bearing through annular passages extending radially through the bearing split inner race referred to as "under race lubrication" (Fig. 12). The concept of "Bearing Thermal Management" was proposed [1, 10] as the proper technological approach to high-speed bearing operation. The basis of this is the recognition that total and flexible thermal control over all the bearing components is essential to achieve a reliable high-speed, highly-loaded bearing. Results reported in the previous section show that bearings operating with jet lubrication under certain conditions are limited to speeds generally less than 2.5 million DN.

Is there, however, a cross-over where it becomes more advantageous to use one lubrication system over another?

Data from [11] obtained with the same bearing design with under-race lubrication was selected for comparison with the data reported herein. The inherent difference in the operating conditions was the oil inlet temperature which was 428 K (310° F) in 12 and 394 K (250° F) for the tests reported herein. As a result, based upon the work of [10, 12], the measured bearing race temperatures of [12] were decreased by 34 K (60 F) in order to compare them with the results of the current study. The inner-race temperatures for under-race lubricated bearings are presented in Fig. 11 along with the data for the jet lubricated bearings under the same conditions. The under-race lubricated bearings also were provided with outer-race cooling. However, outer-race cooling generally had an insignificant effect on the inner-race temperature [10]. The results shown in Fig. 11(a) indicated that at all operating conditions the under-race lubricated bearings had lower temperatures than the dual-orifice jet lubricated bearings. At 12,000 rpm (1.44×10^6 DN) the temperature difference was approximately 22 K (40 F) and at 16,700 rpm (2×10^6 DN), the temperature difference is approximately 44 K (80 F). Beyond 2 million DN, the bearing temperature with under-race lubricant increases only nominally while the temperature of the jet lubricated bearings increases at an accelerated rate. Hence, proper thermal management using jet lubrication is not achievable at the higher speeds. From the above, it is easily concluded that under-race lubrication results in lower operating temperatures.

The data of Fig. 11(b) compares power loss for the two different lubrication systems. As was reported in [1, 10], power loss is a function of the amount of lubricant penetrating the bearing cavity. This is due to viscous drag and lubricant churning [13]. From Fig. 11(b), the power loss with under-race lubricated bearings is higher than with the jet lubricated bearings. At 12,000 rpm (1.44×10^6 DN) the under-race lubricated bearing power loss was approximately 1 kW (1.3 hp) greater than the jet lubricated bearings. At 16,700 rpm (2×10 DN), the difference was approximately 2.3 kW (3.1 hp). The power loss with the under-race lubricated bearing with a flow rate of 4.9×10^{-3} cubic meter (1.3 gal) per minute was equivalent to a jet lubricated flow rate of approximately 6.8×10^{-3} cubic meter (1.8 gal) per minute. If bearing power loss is a function of lubricant flowing through or in the bearing cavity, then it can be reasonably concluded that, for a given jet lubricant flow, approximately 70 percent of the lubricant penetrates the bearing cavity at speeds to at least 2 million DN. At higher speeds this percentage probably decreases due to centrifugal force and windage effects.

SUMMARY

A parametric study was performed with 120-mm bore angular-contact ball bearings having a nominal contact angle of 20°. The bearings either had an inner- or an outer-race land riding cage. Lubrication was by recirculating oil jets. The oil jets either had a single or dual orifice. Thrust load, speed and lubricant flow rate were varied. Test results were compared with those previously reported and obtained from bearings of the same design which were under-race lubricated and run under the same conditions. The following results were obtained:

1. Jet lubricated ball bearings were limited to speeds less than 2.5×10^{6} DN for the jet velocities and supply pressures studied.

2. Bearings having inner-race land riding cages produced lower temperatures than bearings with outer-race land riding cages. The maximum difference in bearing temperatures between the bearings having an inner-race land riding cage and those having an outer-race land riding cage was approximately 40 K (72 F).

3. For a given lubricant flow rate dual orifice jets produced lower bearing temperatures than single orifice jets. The difference in temperatures was as much as 10 to 17 K (18 to 30 F). Difference in bearing power loss was not considered significant.

4. Under race lubrication produced, under all conditions of operation, lower bearing temperatures than jet lubricated bearings. Additionally, there is no apparent speed limitation for bearings operated with this lubrication mode.

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TABLE 1. - CHEMICAL ANALYSIS

OF VACUUM INDUCTION MELTED,

VACUUM ARC REMELTED AISI

M-50 BEARING STEEL

Element	Composition, wt. % Races and balls
Carbon	0.83
Manganese	0.29
Phosphorus	0.007
Sulfur	0.005
Silicon	0.25
Chromium	4.11
Molybdenum	4.32
Vanadium	0.98
Iron	Balance

E-8400

TABLE 2. - PROPERTIES OF

TETRAESTER LUBRICANT

Additives	Antiwear, oxidation inhibitor, antifoam
Kinematic viscosity, cS, at -	
311 K (100 deg F)	28.5
372 K (210 deg F)	5.22
477 K (400 deg F)	1.31
Flash point, K (deg F)	533 (500)
Fire point, K (deg F)	Unknown
Autoignition temperature, K (deg F)	694 (800)
Pour point, K (deg F)	214 (-75)
Volatility (6.5 hr at 477 K (400 deg F)), wt. %	3.2
Specific heat at 477 K (400 deg F), J/(kg)(K) (Btu/lb)(deg F))	2340 (0.54)
Thermal conductivity at 477 K (400 deg F), J/(m)(sec)(K) (Btu/hr)(ft)(deg F))	0.13 (0.075)
Specific gravity at 477 K (400 deg F)	0.850

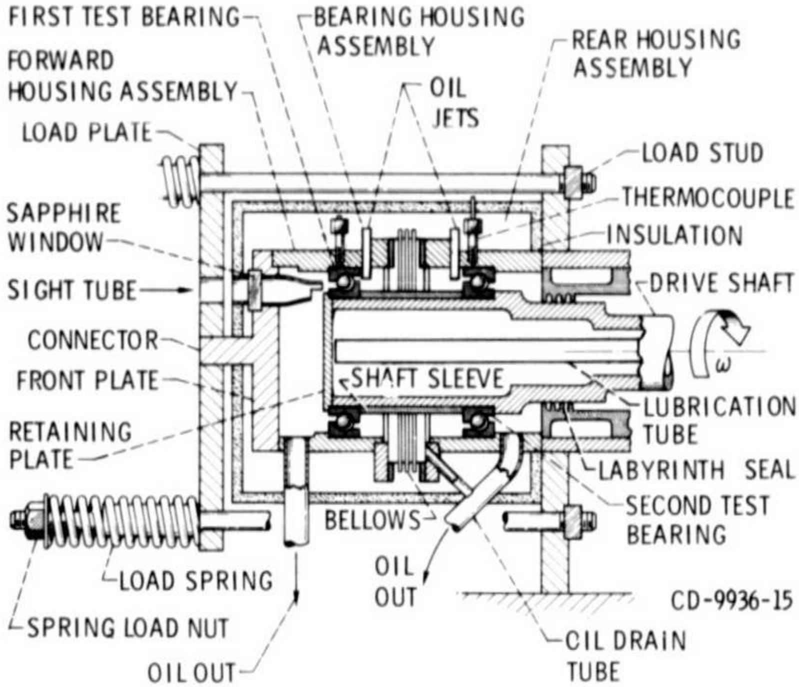

Figure 1. - High-speed, high-temperature bearing test apparatus.

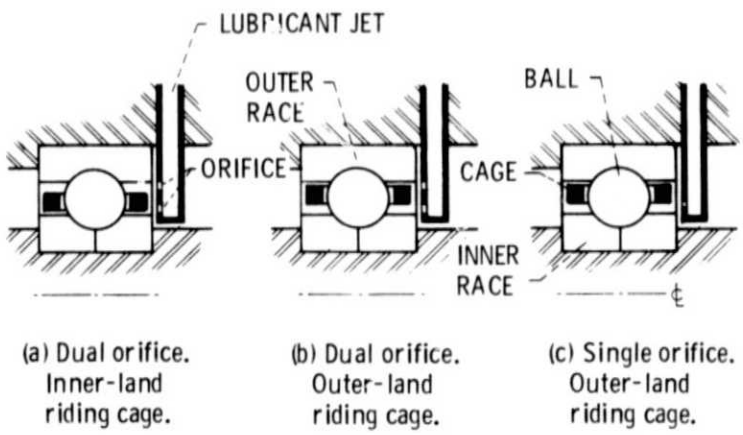

Figure 2. - Bearing lubrication for inner and outer-race land riding cages. Number of jets: 2 per bearing.

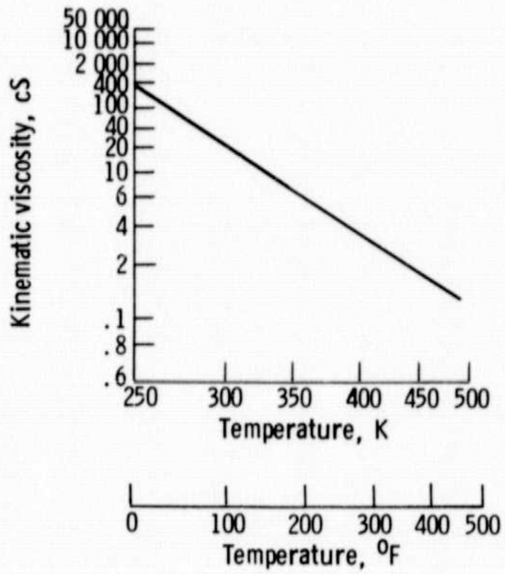

Figure 3. - Viscosity as function of temperature for tetraester (type II) lubricant.

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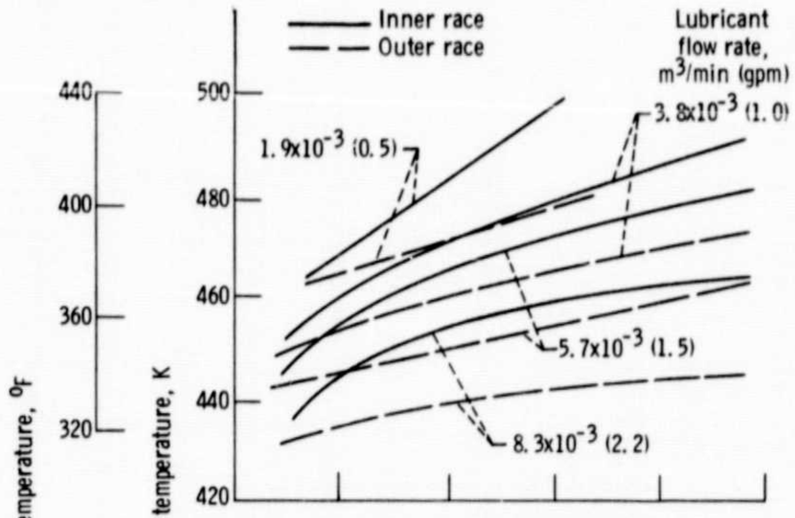
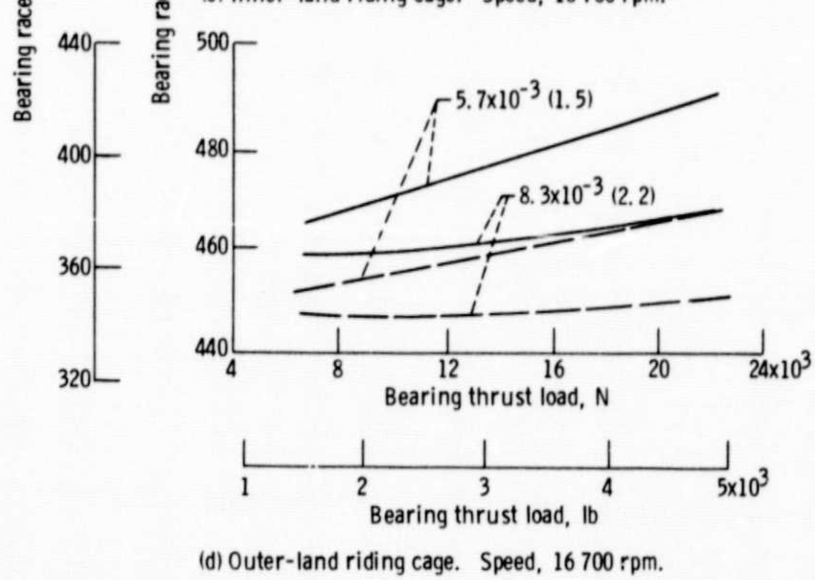


(c) Inner-land riding cage. Speed, 16 700 rpm.



(d) Outer-land riding cage. Speed, 16 700 rpm.

Figure 4. - Concluded.

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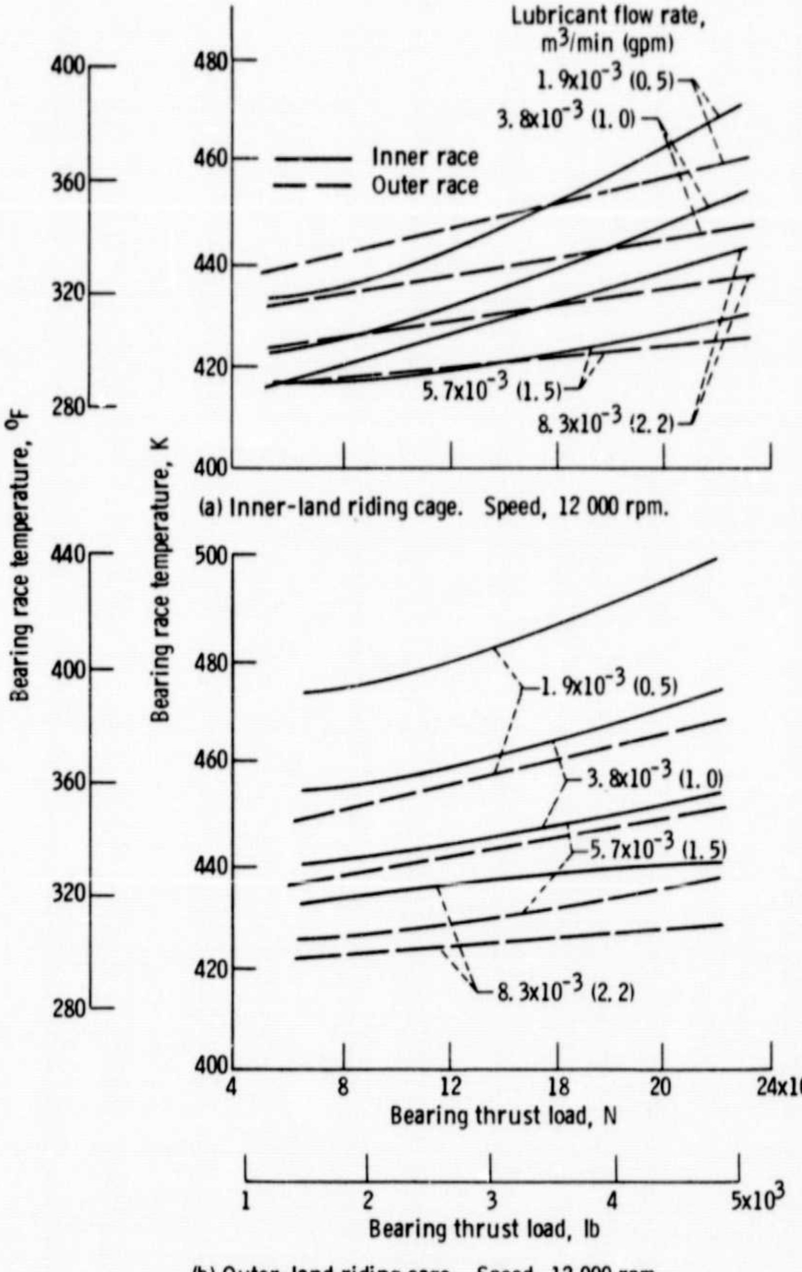

Figure 4. - Bearing race temperature as a function of thrust load for jet lubrication. Bearing type, 120-mm bore angular-contact ball bearings; lubricant jet, dual orifice; number of jets, 2 per bearing; oil inlet temperature, 394 K (250° F); contact angle, 20°.

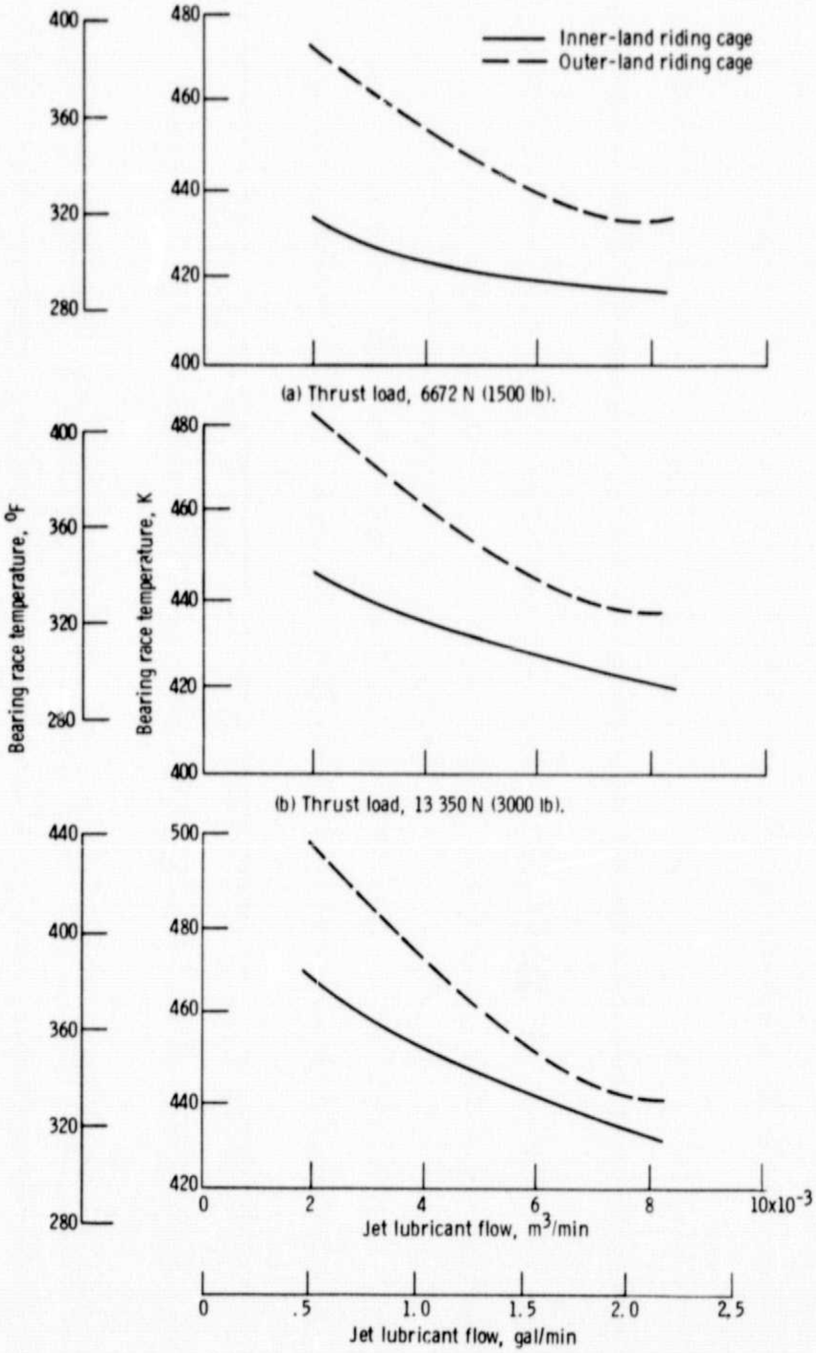

Figure 5. - Comparison of inner-race temperature of bearing operating at 12 000 rpm having inner and outer-land riding cages as a function of lubricant flow rate for various thrust loads. Bearing type, 120-mm bore angular-contact ball bearing; lubricant jet, dual orifice; number of jets, 2 per bearing; oil inlet temperature, 394° K (250° F); contact angle, 20°.

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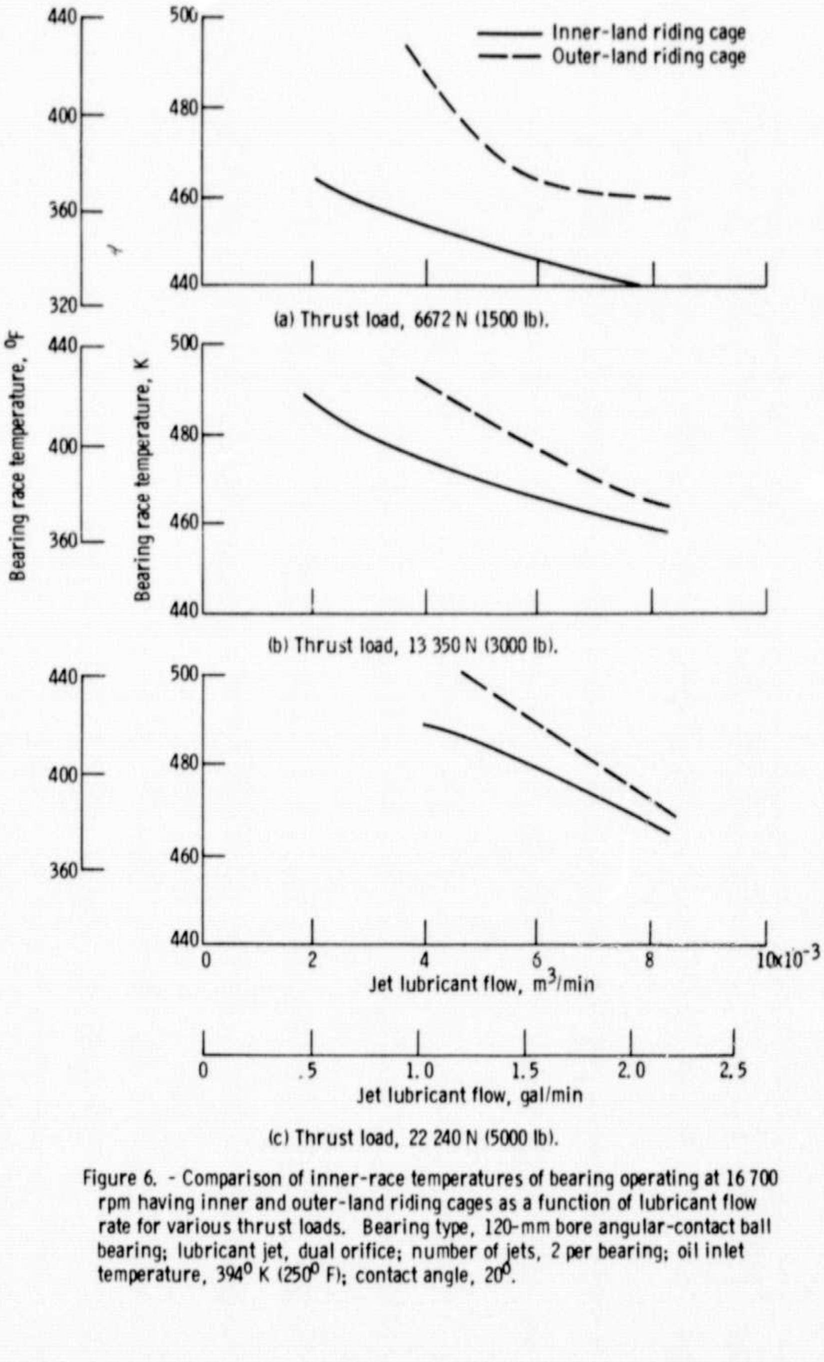
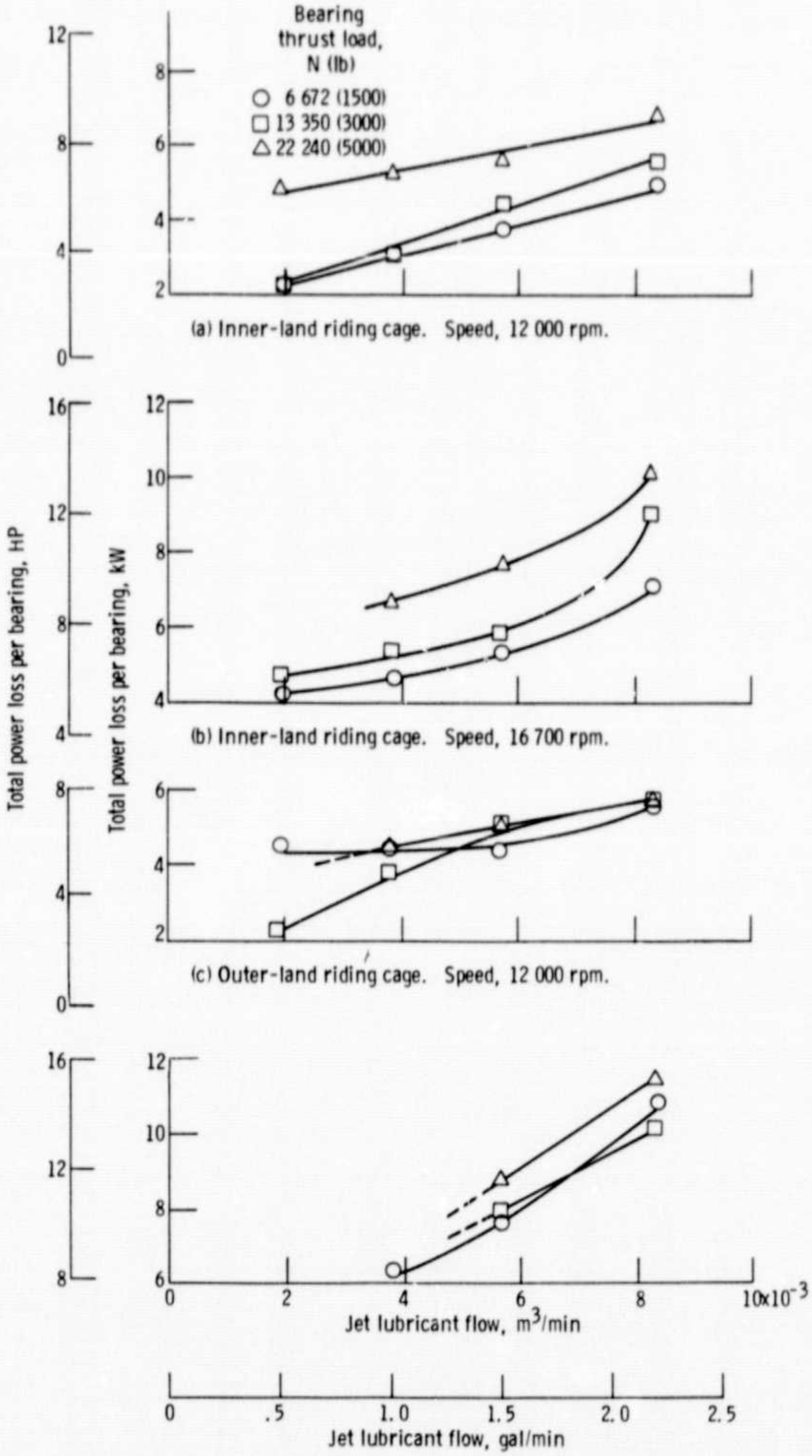


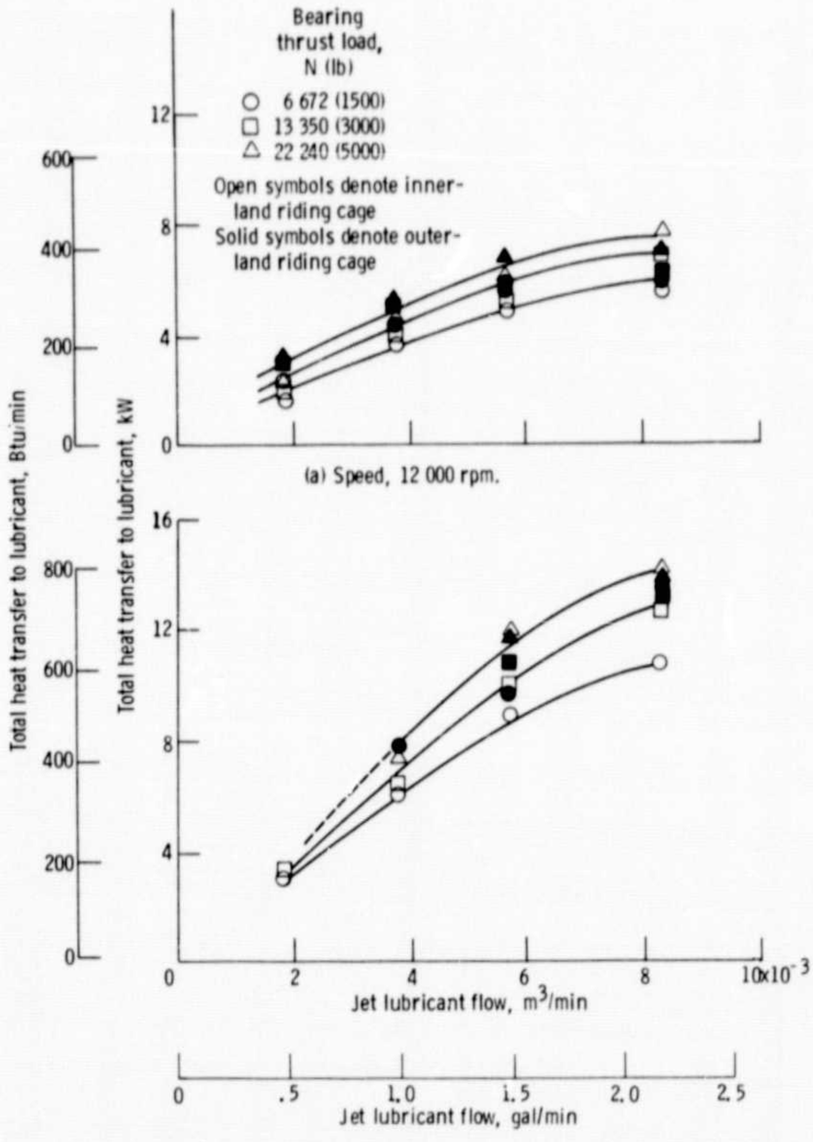
(c) Thrust load, 22 240 N (5000 lb).

Figure 6. - Comparison of inner-race temperatures of bearing operating at 16 700 rpm having inner and outer-land riding cages as a function of lubricant flow rate for various thrust loads. Bearing type, 120-mm bore angular-contact ball bearing; lubricant jet, dual orifice; number of jets, 2 per bearing; oil inlet temperature, 394° K (250° F); contact angle, 20°.



(d) Outer-land riding cage. Speed, 16 700 rpm.

Figure 7. - Bearing power loss as a function of lubricant flow rate for varying thrust loads. Bearing type, 120-mm bore angular-contact ball bearings; lubricant jet, dual orifice; number of jets, 2 per bearing; oil inlet temperature, 394 K (250° F); contact angle, 20°.



(b) Speed, 16 700 rpm.

Figure 8. - Heat generated by a bearing transferred to lubricant as a function of lubricant flow rate. Bearing type, 120-mm bore angular-contact ball bearings; lubricant jet, dual orifice; number of jets, 2 per bearing; oil inlet temperature, 394 K (250° F); contact angle, 20°.

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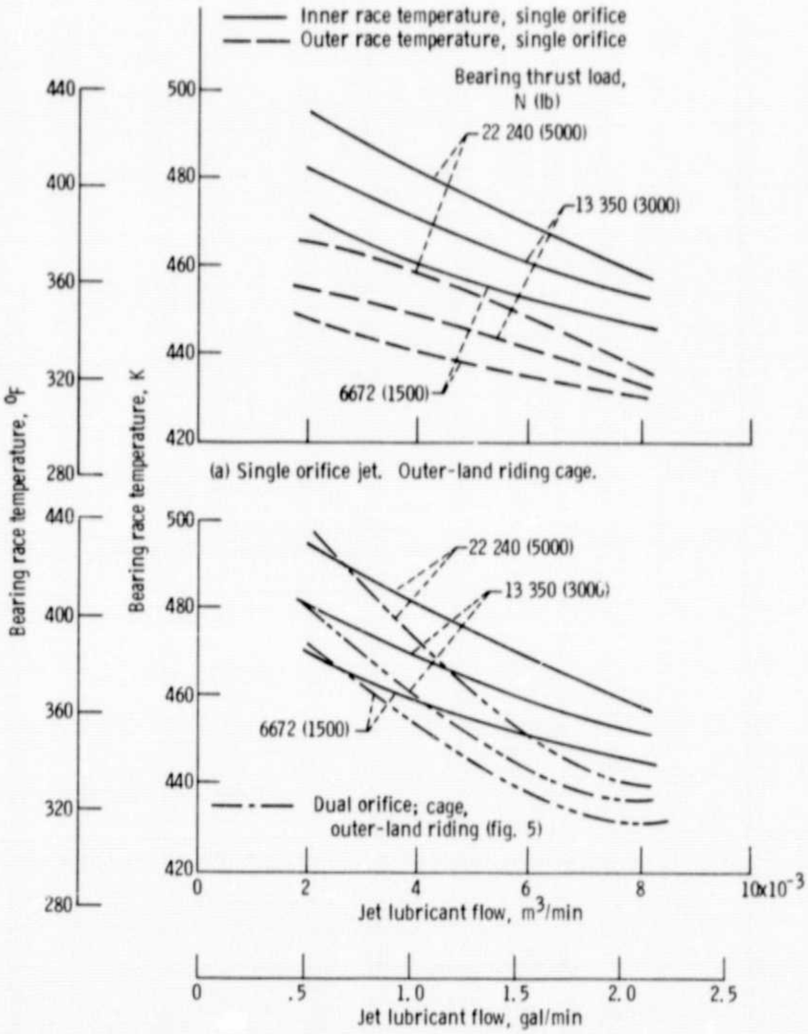

(b) Comparison of inner-race temperatures obtained with single and dual orifice jets. Outer-land riding cage.

Figure 9. - Bearing race temperature as a function of lubricant flow for varying thrust loads. Bearing type, 120-mm bore angular-contact ball bearings; number of jets, 2 per bearing; oil inlet temperature, 394 K (250° F); contact angle, 20°; speed, 12 000 rpm.

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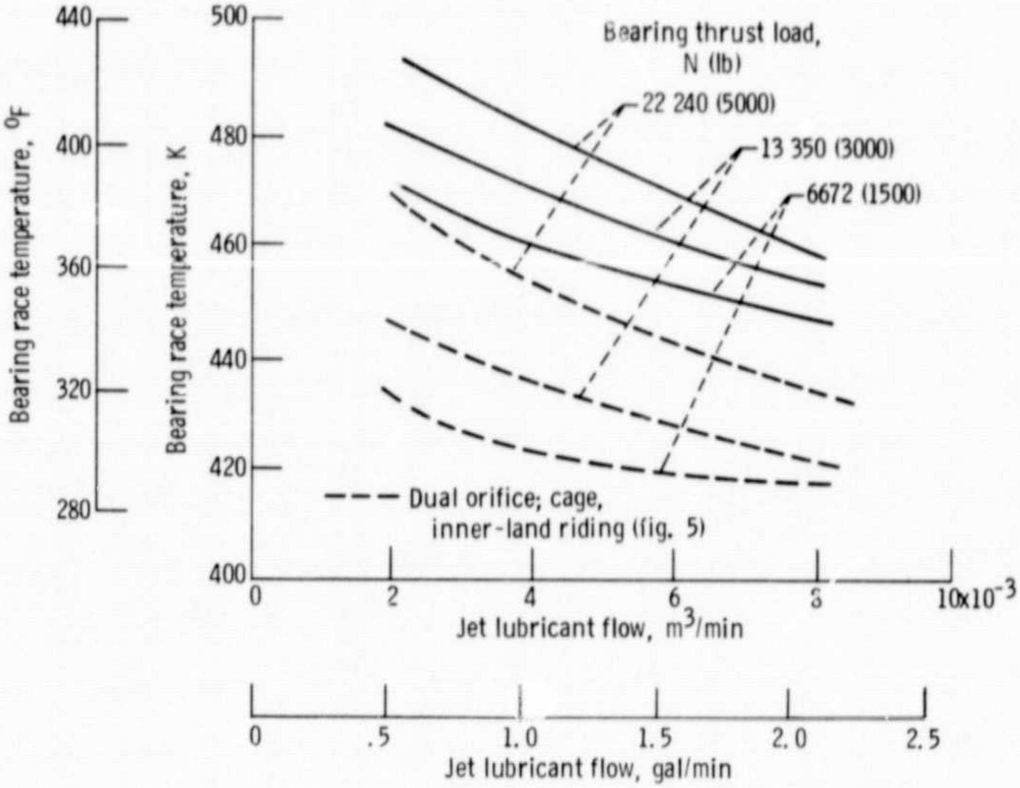

(c) Comparison of inner-race temperatures obtained with single orifice jet lubricated outer-land riding and dual orifice jet lubricated inner-land riding cages.

Figure 9. - Concluded.

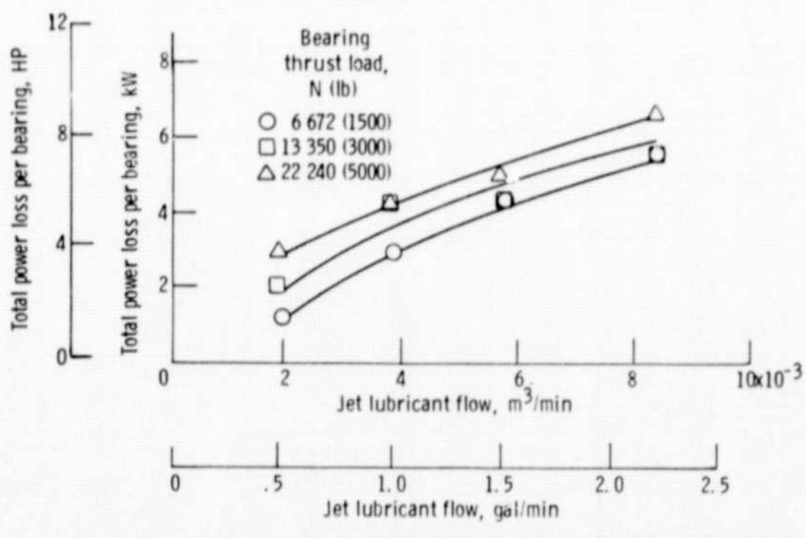

Figure 10. - Bearing power loss as a function of lubricant flow rate for varying thrust loads. Bearing type, 120-mm bore angular-contact ball bearings; lubricant jet, single orifice; number of jets, 2 per bearing; oil inlet temperature, 394 K (250° F); contact angle, 20°; speed, 12 000 rpm.

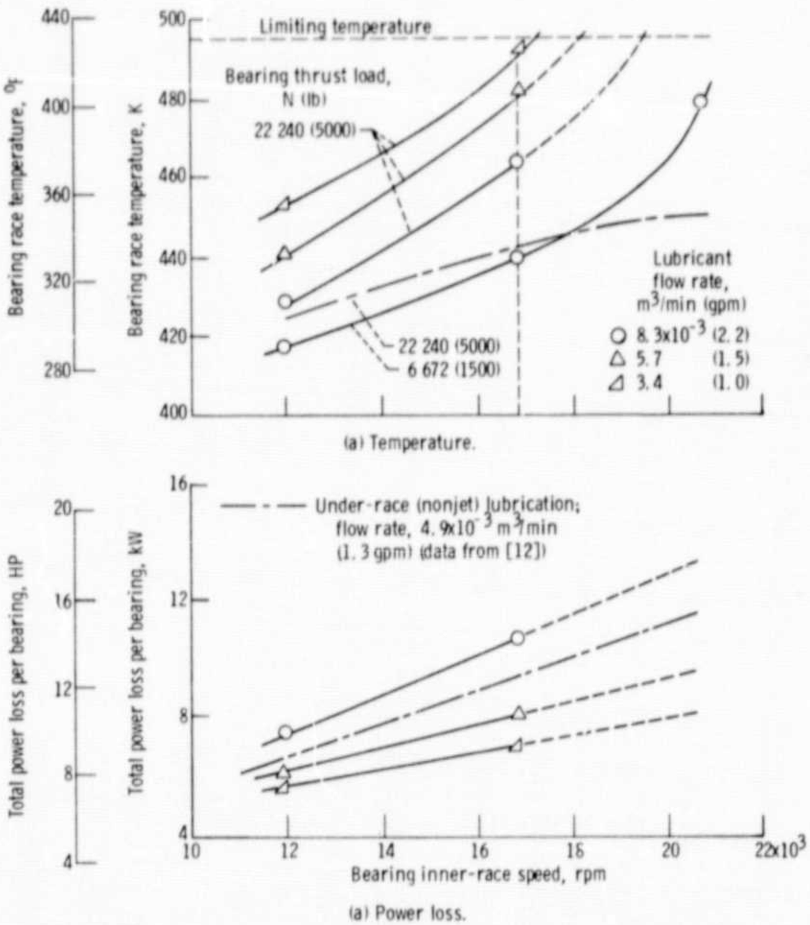

Figure 11. - Bearing inner-race temperature and power loss as a function of speed for varying thrust loads and lubricant flow rates. Bearing type, 120-mm bore angular-contact ball bearings; lubricant jet, dual orifice; number of jets, 2 per bearing; oil inlet temperature, 394 K (250° F); contact angle, 20°.

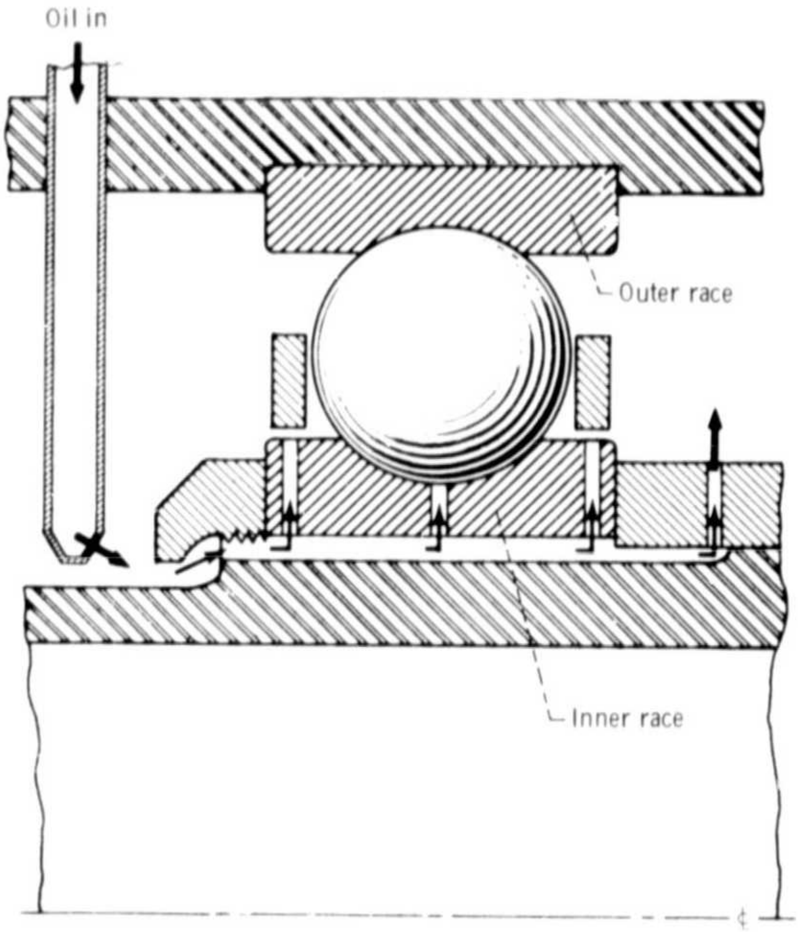

Figure 12. - Under-race lubrication.

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