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Lubrication and Performance of High-Speed Rolling-Element Bearings

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LUBRICATION AND PERFORMANCE OF HIGH-SPEED ROLLING-ELEMENT BEARINGS

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ABSTRACT

Trends in aircraft engine operating speeds have dictated the need for rolling-element bearings capable of speeds to 3 million DN. A review of high-speed rolling-element bearing state-of-the-art performance and lubrication is presented. Through the use of under-race lubrication and "Bearing Thermal Management" bearing operation can be obtained to speeds of 3 million DN. Jet lubricated ball bearings are limited to 2.5 million DN for large bore sizes and to 3 million DN for small bore sizes. Current computer programs are able to predict bearing thermal performance.

INTRODUCTION

Current production engine bearings generally operate at speeds less than 2.3 million DN[1] and at temperatures generally less than 490 K (425 °F). Because compressor or turbine blade tip speeds and disk burst strengths begin to limit the maximum speed of rotating components, a bearing speed of 3 million DN is equivalent to the practical limit of aircraft engine operation. Conventional lubrication methodology generally limited bearing speeds to less than 2 million DN.

A problem associated with operating bearings at high speed is the need to adequately cool the bearing components because of relatively high heat generation. One method used successfully to 3 million DN is to apply cooling lubricant under the race (ref. 1). Lubricant is centrifugally injected through the split inner race and shoulders of an angular-contact ball bearing through a plurality of radial holes. As a result, both the cooling and lubricant function is accomplished.

Recirculating jet lubrication is most commonly used on airbreathing turbojet engines today. There have been many studies conducted to determine optimum lubricant jet arrangements - single-, multiple-, and multiple-opposed jets (ref. 2).

Research (ref. 3) with 30-mm bore ball bearings studied the effect of cage location as well as jet lubrication to speeds up to 3 million DN. Unless significant cooling was provided, bearing failure was due to overheating at high speeds. Bearing limiting speed was dependent on bearing design, jet arrangement, and oil flow rate and velocity (ref. 3).

Tapered-roller bearings are being used in some helicopter transmissions to carry combined radial, thrust, and moment loads and, in particular, those

[1]DN is a speed parameter defined as the bearing bore in millimeters multiplied by its speed in rpm.

loads from bevel gears such as high-speed input pinions (ref. 4). Tapered-roller bearings have greater load capacity for a given envelope or for a given bearing weight than the combinations of ball and cylindrical roller bearings commonly used in this application. Speed limitations have restricted the use of tapered-roller bearings to lower speed applications relative to ball and cylindrical roller bearings. The speed of tapered-roller bearings is limited to approximately 0.5 million DN (a cone-rib tangential velocity of approximately 36 m/s (7000 ft/min) unless special attention is given to lubricating and designing the cone-rib/roller-large-end contact. At higher speeds centrifugal effects starve this critical contact of lubricant (refs. 4 and 5).

In 1959 NASA began a research program to encompass the projected speed requirements for large-bore ball and roller bearings. In the 1970's NASA expanded its program to encompass tapered-roller, and small-bore ball and roller bearings. The reported work is a review of high-speed rolling-element bearing state-of-the-art performance and lubrication with emphasis on NASA contributions.

LUBRICANT SELECTION

The criteria for a liquid lubricant to function in a rolling-element bearing are that (a) it be thermally and oxidatively stable at the maximum bearing operating temperature, and (b) it form an elastohydrodynamic (EHD) film between the rolling surfaces. The EHD film, which is generally dependent on lubricant base stock and viscosity, is 0.1 to 2.5 μm (5 to 100 μin) thick at high temperatures (ref. 6). When a sufficiently thick EHD film is present, rolling-element bearings will not usually fail from surface distress. Instead, they fail from rolling-element fatigue which usually manifests itself, in the early stages, as a shallow spall with a diameter about the same as the contact width.

A requirement for long-term high temperature bearing operation is that the EHD film thickness, h, divided by the composite surface roughness, $(\sigma_1^2 + \sigma_2^2)^{1/2}$, equal 1-1/2 or greater, where σ_1 and σ_2 are the surface finishes of the raceway and rolling elements, respectively (ref. 6). The EHD film thickness is a function of several lubricant and bearing operating variables (ref. 7). However, as a general rule, the minimum viscosity required of a lubricant is 1 cSt at operating temperature (ref. 8). This same research indicated that the ester based lubricants (table 1) meeting the MIL-L-23699 specification could provide the necessary lubrication requirements to 490 K (425 °F) in an air environment. While other base stock lubricants could give satisfactory operation to 590 K (600 °F) (ref. 8), they were precluded from further consideration because of their cost and/or commercial availability. Further, at temperatures above approximately 500 K (450 °F) a low oxygen environment would be required to minimize lubricant oxidation for most of the lubricant types studied (refs. 8 and 9).

LUBRICATION METHODS

Jet Lubrication

For aircraft engines and transmissions, where speeds are too high for grease or simple splash lubrication, jet lubrication is used to both lubricate and control bearing and gear temperatures by removing generated heat. In jet lubrication the placement and number of nozzles, jet velocity, lubricant flow rates, and removal of lubricant from the bearing and immediate vicinity are all very important for satisfactory operation.

The placement of jets should take advantage of any natural pumping ability of the bearings. This is illustrated in figure 1 for a ball bearing with relieved races and for a tapered-roller bearing. Centrifugal forces aid in moving the oil through the bearing to cool and lubricate the elements (ref. 10).

Directing jets in the radial gaps between the races and the cage is beneficial. The design of the cage and the lubrication of its surfaces sliding on the shoulders of the races greatly affects the high-speed performance. The cage has been typically the first element to fail in a high-speed bearing with improper lubrication.

It has been shown (ref. 3) that with proper bearing cage design, nozzle placement, jet velocities, and adequate scavenging of the lubricant, jet lubrication can be successfully used for small-bore ball bearings at speeds to 3 million DN. For large (120 mm bore) ball bearings (ref. 11), speeds to 2.5 million DN are attainable. For large (120.65 mm bore) tapered-roller bearings, jet lubrication was successfully demonstrated to 1.8 million DN (ref. 5), although a high lubricant flow rate of 0.0151 m^3/min (4 gpm) and a relatively low oil-inlet temperature of 350 K (170 °F) were required.

Under-Race Lubrication

A more effective and efficient means of lubricating rolling-element bearings is under-race lubrication. Conventional jet lubrication fails to adequately cool and lubricate the inner-race contact as the lubricant is thrown centrifugally outward. Unfortunately, increased flow rates add to heat generated from oil churning. An under-race lubrication system used in turbofan engines for ball and cylindrical roller bearings is shown in figure 2. Lubricant is directed under the inner race and centrifugally forced out through a plurality of holes in the race to cool and lubricate the bearing. Some lubricant may pass completely through under the bearing for cooling only as shown in figure 2(a). Although not shown in the figure, some radial holes may be used to supply lubricant to the cage side lands.

This lubricating technique has been thoroughly tested for large-bore ball and roller bearings up to 3 million DN (refs. 1 and 12 to 14).

Under-race lubrication has been applied to tapered-roller bearings to lubricate and cool the critical cone-rib/roller-end contact. As described in reference 15, 88.9-mm (3.5-in) bore tapered-roller bearings were run under combined radial and thrust loads to 1.42 million DN with cone-rib lubrication (the term used to denote under-race lubrication in tapered-roller bearings).

RESULTS AND DISCUSSION

Angular-Contact Ball Bearings

Large bore. - A parametric study was performed with AISI M-50 120-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 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 million 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.

Research reported in references 14, 16, and 17 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. 2(b)). The concept of "Bearing Thermal Management" was proposed (refs. 14 and 16) 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.

Data for both under-race lubricated and jet lubricated bearings are presented in figure 3. The under-race lubricated bearings were provided with outer-race cooling. However, outer-race cooling generally had an insignificant effect on the inner-race temperature (ref. 14).

The results shown in figure 3(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 million DN) the temperature difference was approximately 22 K (40 °F) and at 16 700 rpm (2 million DN), approximately 44 K (80 °F). Beyond 2 million DN the bearing temperature with under-race lubrication increases only nominally, while the temperature of the jet lubricated bearings increases at an accelerated rate. Hence, proper thermal management using jet lubrication was not achievable in these tests at the higher speeds. From the above it was concluded that under-race lubrication results in lower operating temperatures.

The data of figure 3(b) compare power loss for the two different lubrication systems. As was reported in references 14 and 16 power loss is a function of the amount of lubricant penetrating the bearing cavity. This is due to viscous drag and lubricant churning (ref. 18). From figure 3(b) the power loss with under-race lubricated bearings is higher than with the jet lubricated bearings. At 12 000 rpm (1.44 million DN) the under-race lubricated bearing power loss was ~1 kW (1.2 hp) greater than the jet lubricated bearings. At 16 700 rpm (2 million DN), the difference was ~2.3 kW (3.1 hp). The power loss with the under-race lubricated bearing with a flow rate of 4920 cm^3/min (1.3 gal/min) was equivalent to a jet lubricated flow rate of ~6812 cm^3/min (1.8 gal/min). 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, ~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.

Small bore. - Applying under-race lubrication to small-bore ball bearings is more difficult than with the larger bore sizes. This is because of limited space available for grooves and radial holes, the means of getting the lubricant under the race. For a given DN value, centrifugal effects are more severe with small bearings since centrifugal force varies with the square of the speed. Heat generated per unit of surface area is much higher and heat removal is more difficult.

Parametric tests were performed on AISI M-50 35 mm bore angular-contact ball bearings (refs. 19 to 24). Figure 4 shows the bearing configurations tested and the means of lubrication. The bearings had a nominal, unmounted contact angle of 24° and either a single- or a double-outer-land-guided cage. The investigation included lubrication by oil jets or through passages in the bearing inner race. When inner race lubrication was used, the oil was channeled through axial grooves and radial holes in the bearing inner race. In some tests, 50 percent of the oil supplied to the inner race was introduced into the bearing for lubrication and 50 percent cooled the inner race exterior surfaces. In other tests the distribution was 25 percent lubrication and 75 percent cooling. In selected tests the bearing outer diameter was cooled with a constant oil flow of 1700 cm^3/min (0.45 gal/min).

Test conditions included nominal shaft speeds from 48 000 to 72 000 rpm, a radial load of 222 N (50 lb) and/or a thrust load of 667 N (150 lb). Lubricant flow to the bearing ranged from 300 to 1900 cm^3/min (0.08 to 0.50 gal/min) at an inlet temperature of 394 K (250 °F). All bearings were successfully run at speeds to 2.5 million DN. Generally, increasing the lubricant flow decreased bearing race temperatures but increased bearing power loss.

Figure 5 shows significantly cooler inner-race temperatures with under-race lubrication at speeds to 72 500 rpm (2.5 million DN) than with jet lubrication. Lowest power loss was generally obtained with inner-race lubricated bearings having an inner-race guided cage. The highest power loss of 327 kW (4.39 hp) occurred at 72 500 rpm for a jet lubricated bearing with outer-race cooling.

The effect of outer-race cooling with the 35-mm bore ball bearings is shown in figure 6. Outer-race cooling for both oil-flow distribution and for the jet lubricated bearings resulted in a substantial decrease in outer-race temperature but had a minimal effect on inner-race temperature. The same effects were obtained with the large bore angular-contact bearings.

Cylindrical Roller Bearings

Early NACA (now NASA) research (refs. 25 and 26) was conducted on jet lubricated 75-mm cylindrical roller bearings. The maximum DN value reached was 1.65 million at which speed the inner-race riding cage failed. The general trends in temperature with oil flow reported hereinabove for the ball bearings were reported for these 75-mm roller bearings. For all values of oil flow and all oil-jet diameters investigated, the outer-race maximum temperature increased linearly with an increase in DN value; however, the inner-race temperature increased with DN value at a rate slightly greater than linear.

For the oil-jet positions investigated, the inner- and outer-race bearing temperatures were found to be at a minimum when the oil was directed at the cage-locating surface perpendicular to the bearing face.

Parametric tests were conducted in a high-speed bearing tester on an AISI M-50 118-mm bore roller bearing with a round outer ring (refs. 27 and 28). Test parameters were radial loads of 2200, 4400, 6700, and 8900 N (500, 1000, 1500, and 2000 lb) and nominal speeds of 10 000, 15 000, 20 000, and 25 500 rpm. The oil-inlet temperature (for both the lubricating and the cooling oil) was 366 K (200 °F). Oil was supplied through the inner ring for lubrication and inner-ring cooling at flow rates from 1.9×10^{-3} to 10.2×10^{-3} m^3/min (0.5 to 2.7 gal/min). Outer-ring cooling flow rates were 0 to 4.3×10^{-3} m^3/min (0 to 1.13 gal/min).

The 118-mm bore roller bearing ran successfully to 3.0×10^6 DN with radial loads of 2200 to 8900 N (500 to 2000 lb) and no outer-ring cooling. At a maximum total flow rate to the inner ring of 9.5×10^{-3} m^3/min (2.5 gal/min) and no outer-ring cooling, the maximum bearing temperature was 466 K (380 °F).

No cage slip occurred at 3.0×10^6 DN primarily because of the tight clearance or even slight interference within the bearing at this high speed. Bearing temperatures varied inversely with cage slip for all conditions. There was no effect of load on either bearing temperature or cage slip as load was varied from 2200 to 8900 N (500 to 2000 lb) over a speed range of 10 000 to 25 500 rpm.

Cooling the outer ring decreased its temperature but increased the inner-ring temperature. With no outer-ring cooling, bearing temperatures decreased with increasing total oil flow rate to the inner ring. Outer-ring temperatures were always higher than inner-ring temperatures over the shaft speed range of 10 000 to 25 500 rpm. Heat rejected to the lubricant (power loss within the bearing) increased with both shaft speed and total oil low rate to the inner ring.

Tapered Roller Bearing

Tapered-roller bearings have been restricted to lower speed applications than have ball and cylindrical roller bearings. The speed limitation is primarily due to the cone-rib/roller-end contact which requires very careful lubrication and cooling consideration at higher speeds. The speed of tapered-roller bearings is limited to ~0.5 million DN (a cone-rib tangential velocity of ~36 m/s (7000 ft/min)) unless special attention is given to lubricating and designing this cone-rib/roller-end contact. At higher speeds centrifugal effects starve this critical contact of lubricant.

The technique of under-race lubrication has been applied to tapered-roller bearings to lubricate and cool the critical cone-rib/roller-end contact. As described in (ref. 15), 88.9-mm (3.5 in) bore tapered-roller bearings were run under combined radial and thrust loads to 1.42 million DN with cone-rib lubrication (the term used to denote under-race lubrication in tapered-roller bearings).

A comparison of cone-rib lubrication and jet lubrication was reported in reference 5 for 120.65-mm (4.75 in) bore tapered-roller bearings under combined radial and thrust loads. These bearings were standard catalog bearing design except for the large end of the roller, which had a different spherical end radius to provide more favorable contact with the cone rib (ref. 29). Those bearings that used cone-rib lubrication also had holes drilled through from a manifold in the cone bore to the undercut at the large end of the cone (fig. 7). The results of reference 5 show very significant advantages of cone-rib lubrication (fig. 8). At 15 000 rpm (1.8 million DN) the bearing with cone-rib lubrication had a cone face temperature 34 K (62 °F) lower than one with jet lubrication. Furthermore, reference 5 shows that the tapered-roller bearing would operate at 15 000 rpm with cone-rib lubrication at less than half the flow rate required for jet lubrication at that speed.

Further work has shown successful operation with large-bore tapered-roller bearings at even higher speeds. Long-term operation of 107.94 mm (4.24 in) bore tapered-roller bearings under pure thrust load to 3 million DN with a combination of cone-rib lubrication and jet lubrication was reported in (ref. 30). Successful operation of optimized design 120.65 mm (4.75 in) bore tapered-roller bearings under combined radial and thrust load with under-race lubrication to both large (cone-rib) and small ends to speeds up to 2.4 million DN was reported in references 31 to 33.

COMPUTER ANALYSIS

There are currently several comprehensive computer programs that are capable of predicting rolling-element bearing operating and performance characteristics. These programs generally accept input data of bearing internal geometry (such as sizes, clearance, and contact angles), bearing material and lubricant properties, and bearing operating conditions (load, speed, and ambient temperature). The programs then solve several sets of equations that characterize rolling-element bearings. The output produced typically consists of rolling-element loads and Hertz stresses, operating contact angles, component speeds, heat generation, local temperatures, bearing fatigue life, and power loss. The critical assumptions currently necessary in the use of these programs are the form of the lubricant traction model and the lubricant volume percent (the assumed volume percent of the bearing cavity occupied by the lubricant). These programs were used to predict the performance of the 120-mm angular-contact ball bearings (ref. 34) and the 118-mm cylindrical roller bearing (refs. 27 and 28) discussed in the previous sections. Representative data are shown for the ball bearing in figure 9 and for the roller bearing in figure 10 as a function of lubricant flow rate. The correlation between the prediction and experiment is reasonably good for these bearings.

Another correlation of predicted and experimental data for cylindrical roller bearings is shown in reference 35. Experiments were performed with 124.3 mm bore bearings at speeds up to 3 million DN, and the results were compared with analysis using the computer program described in reference 12. Predictions of both heat rejection to the oil and outer-race temperature were within 10 percent of the experimental values.

As was mentioned above, one of the critical assumptions necessary in the use of these computer programs is the lubricant volume percent on the bearing cavity. Bearing heat generation and temperature are dependent on this factor. The lubricant percent volume varies with bearing type, size, design, lubrication method, lubricant flow rate, and shaft speed. An equation was derived for the lubricant percent volume in the bearing cavity as a function of lubricant flow rate through the cavity, shaft speed, and bearing pitch diameter (ref. 36). The predicted bearing heat generation, inner- and outer-race temperatures, and oil-out temperatures agreed very well with the experimental data obtained from three sizes of ball bearings (35-, 120-, and 167-mm bore) over a speed range from 1.0 to 3.0 million DN.

The equation derived appears to be valid over the range of shaft speeds and lubricant flow rates for the three bearing sizes investigated. The predicted trends of increasing temperature and heat generation with increased shaft speed and of decreasing temperature and increasing heat generation with increased lubricant flow rate were verified by the experimental data.

SUMMARY

Conventional lubrication methodology generally limited rolling-element bearing speeds to less than 2 million DN. Jet and under-race lubrication were applied to ball, roller, and tapered-roller bearings in order to define bearing performance and speed limitations. Bearing bore sizes ranged from 35 to 120 mm. Lubricant flow rate and distribution together with bearing load and speed were varied with temperature and power loss determined. Experimental results were compared to computer predictions. Operating speed limitations were defined.

The following results were obtained:

1. Through the use of under-race lubrication and "Bearing Thermal Management," reliable highly-loaded bearing operation can be obtained to speeds of 3 million DN.

2. Jet lubricated ball bearings are limited to 2.5 million DN for large bore sizes and to 3 million DN for small bore sizes.

3. Current computer programs are able to predict bearing thermal performance. An equation was derived for the lubricant percent volume in the bearing cavity as a function of lubricant flow rate through the cavity, shaft speed, and bearing pitch diameter.

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TABLE I. - PROPERTIES OF TETRAESTER LUBRICANT

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

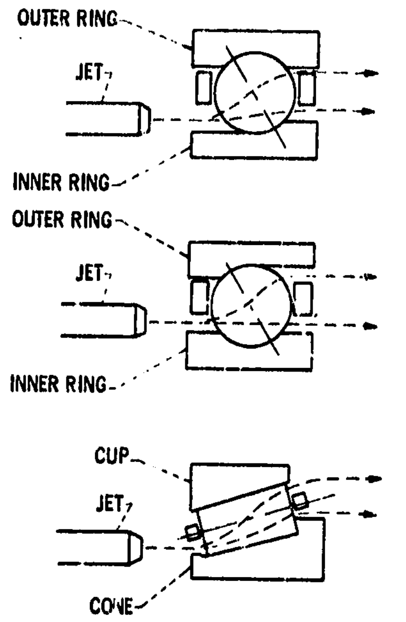


Figure 1. - Placement of jets for ball bearings with relieved rings and tapered-roller bearings.

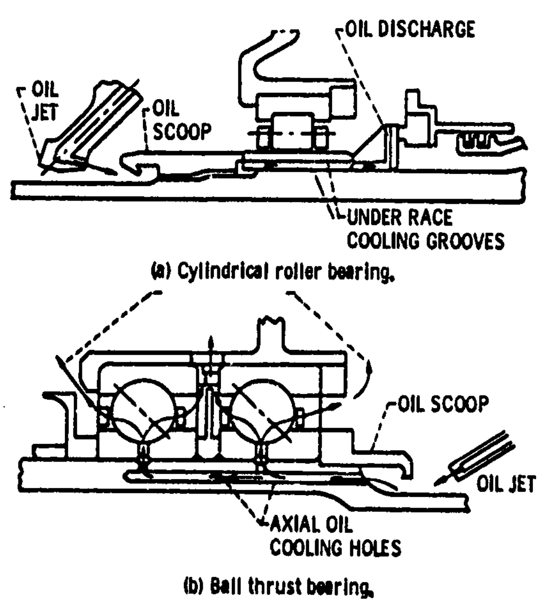


Figure 2. - Under-race lubrication system for main shaft bearings on turbofan engine.

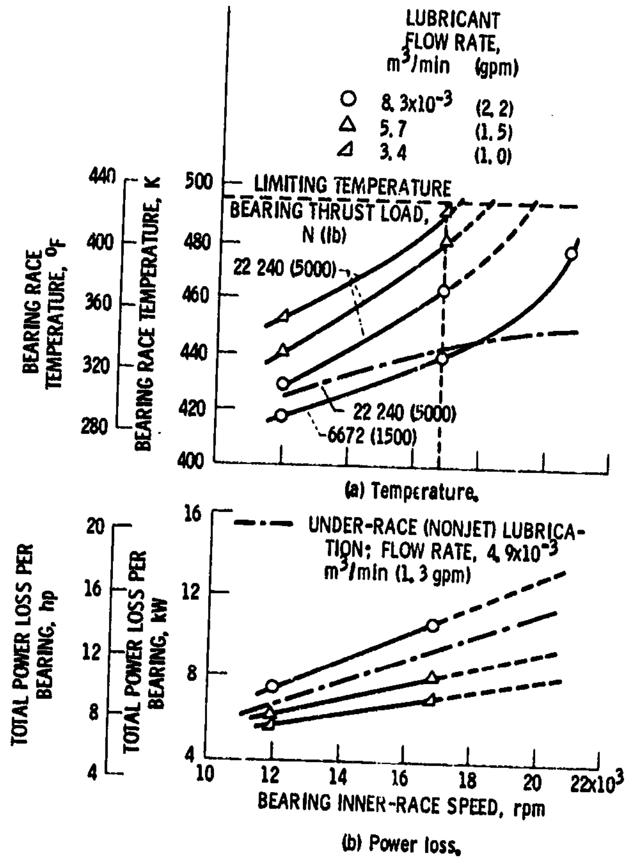
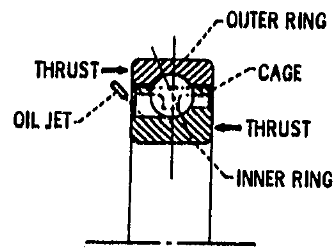
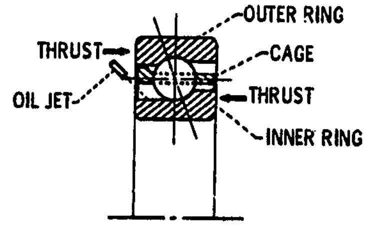


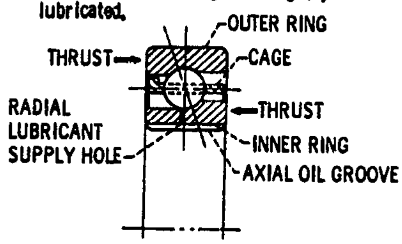
Figure 3. - Bearing inner-race temperature and power loss with jet lubrication as a function of speed for varying thrust loads and lubricant flow rates. Bearing type, 120-mm-bore angular-contact ball bearing; lubricant jet, dual orifice; number of jets, 2 per bearing; oil inlet temperature, 395 K (250 °F); contact angle, 20°.



(a) Double-outer-land-guided cage; jet lubricated.



(b) Single-outer-land-guided cage; jet lubricated.



(c) Single-outer-land-guided cage; inner-ring lubricated.

Figure 4. - Angular-contact ball bearing.

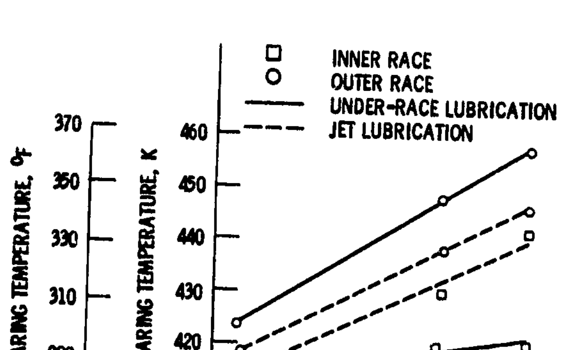


Figure 5. - Effect of under-race lubrication with 35-mm-bore angular-contact ball bearings. (Total oil flow rate, 1318 cm^3/min (0.348 gpm); oil-in temperature, 395 K (250 °F).)

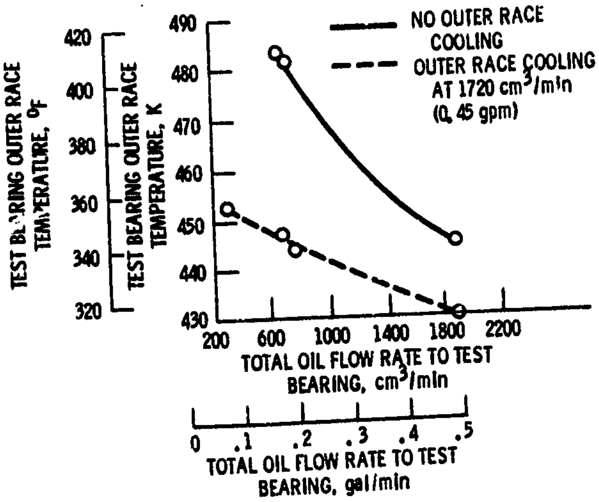


Figure 6. - Effect of outer-race cooling on outer-race temperature of 35-mm-bore ball bearings at 72 300 rpm. Oil-in temperature, 395 K (250 °F).

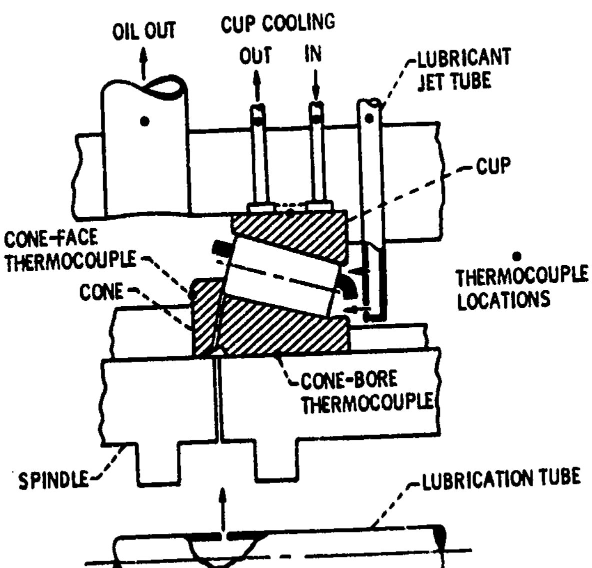


Figure 7. - Tapered-roller bearing with cone-rib and jet lubrication.

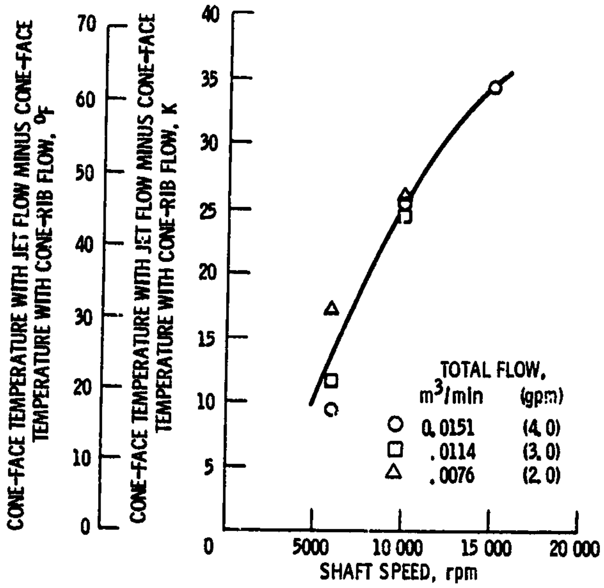


Figure 8. - Effect of shaft speed on cone-face temperature with jet lubrication minus that with cone-rib lubrication for 120.65-mm-bore tapered-roller bearing. Oil-in temperature, 350 K (170 °F); thrust load, 53 400 N (12 000 lb); radial load, 26 700 N (6000 lb).

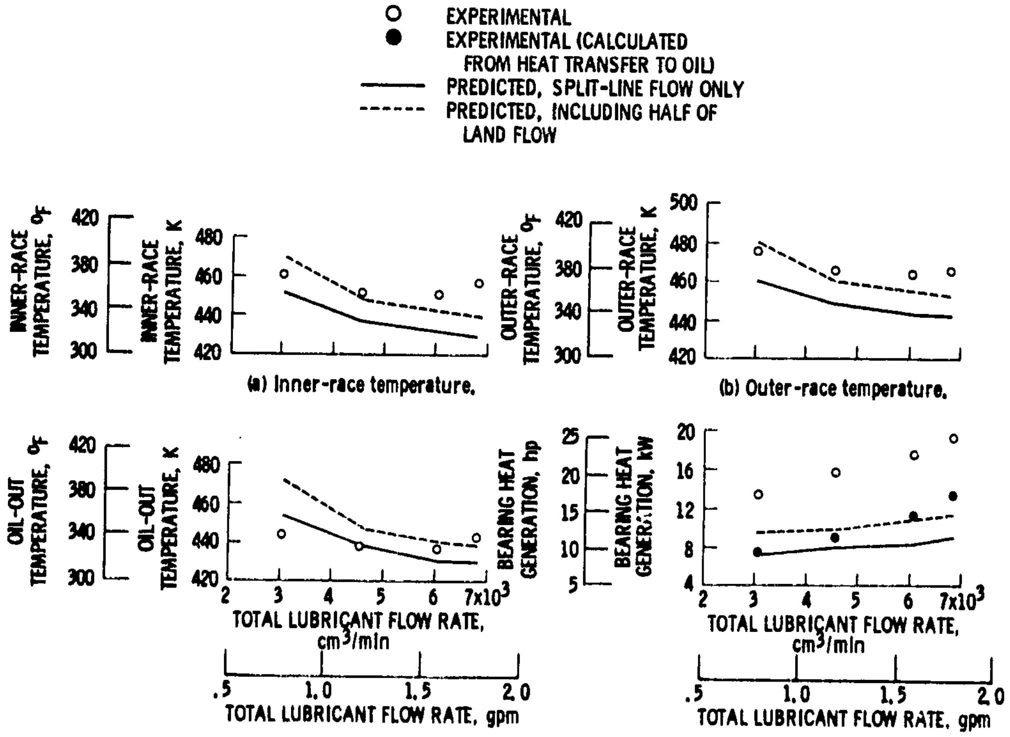
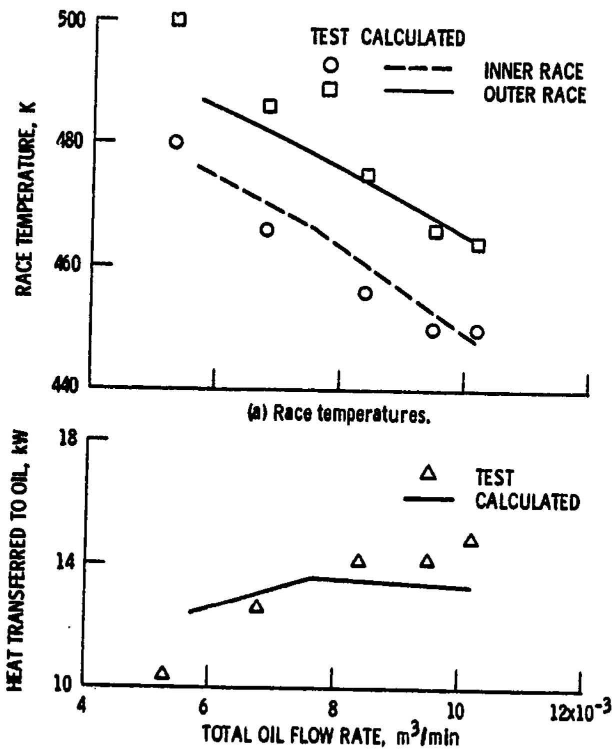


Figure 9. - Predicted and experimental thermal performance of 120-mm-bore ball bearing as function of total lubricant flow rate. Thrust load, 22 240 N (5000 lb); oil-in temperature, 394 K (250 °F); shaft speed, 25 000 rpm.



(b) Heat transferred to lubricating oil.

Figure 10. - Calculated and experimental 118-mm-bore roller bearing operating characteristics with an input cold diametral clearance of 0.09 mm in the computer program - shaft speed, 25 500 rpm. Radial load, N (2000 lb); lubricant volume, 2 percent.