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PERFORMANCE OF LARGE-BORE TAPERED-ROLLER BEARINGS UNDER COMBINED RADIAL AND THRUST LOAD AT SHAFT SPEEDS TO 15 000 RPM

Richard J. Parker and Hans R. Signer

Lewis Research Center

Cleveland, Ohio 44135

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Page 15: The curves in figures 10(a) and 11(a) are in error. These figures should be replaced with the attached figures.

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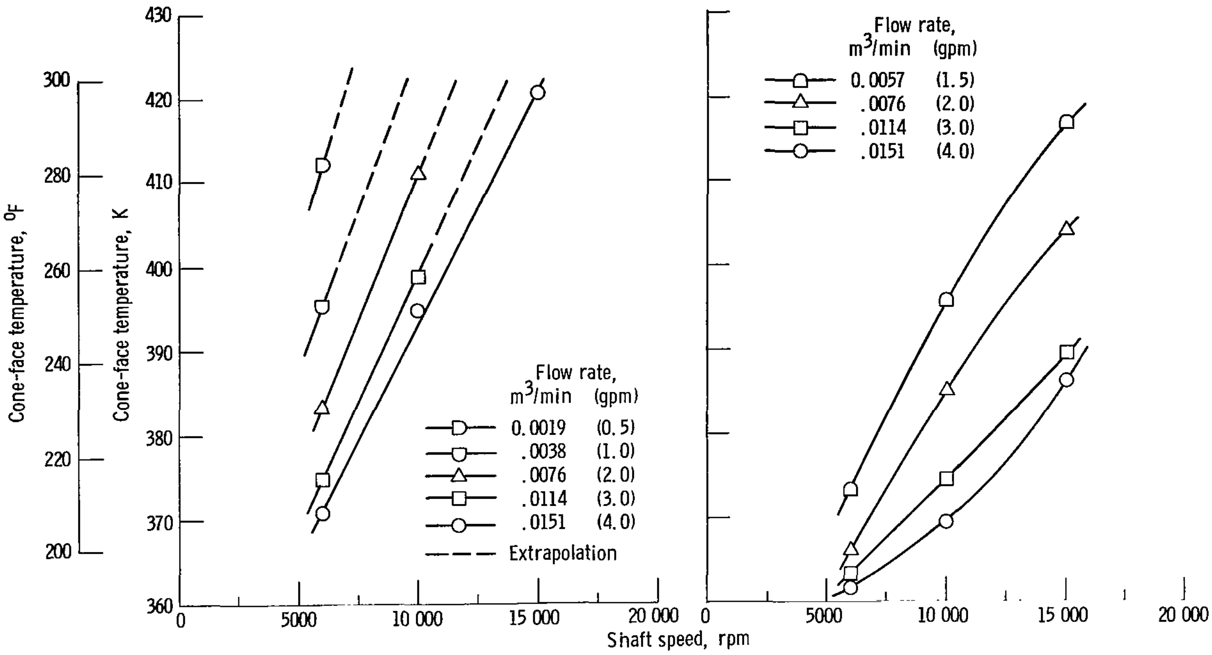
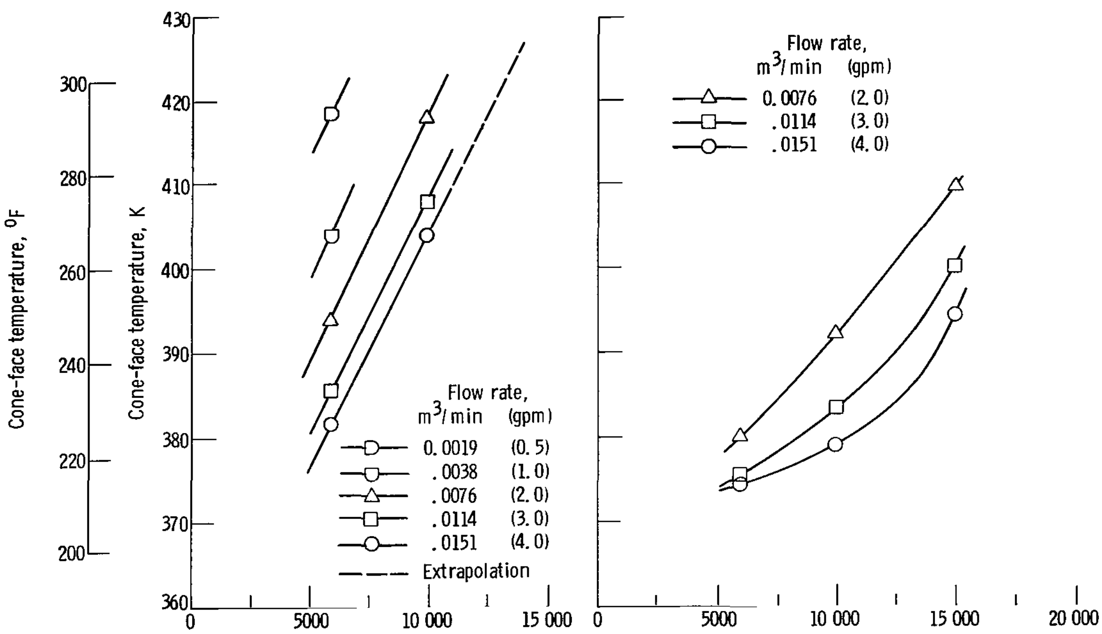


(a) Lubrication by jets only.

(b) Lubrication through cone rib with 0.0038 m^3/min (1.0 gpm) by jets.

Figure 10. - Effect of shaft speed and flow rate on cone-face temperature for oil-in temperature of 350 K (170° F). Thrust load, 53 400 N (12 000 lb); radial load, 26 700 N (6000 lb).



(a) Lubrication by jets only.

(b) Lubrication through cone rib with 0.0038 m^3/min (1.0 gpm) by jets.

Figure 11. - Effect of shaft speed and flow rate on cone-face temperature for oil-in temperature of 364 K (195° F). Thrust load, 53 400 N (12 000 lb); radial load, 26 700 N (6000 lb).

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16. Abstract

The performance of 120.65-mm- (4.75-in.-) bore tapered-roller bearings was investigated at shaft speeds up to 15 000 rpm (1.81×10^6 DN). Temperature distribution and bearing heat generation were determined as a function of shaft speed, radial and thrust loads, lubricant flow rate, and lubricant inlet temperature. Lubricant was supplied by either jets or by a combination of holes through the cone directly to the cone-rib contact and jets at the roller small-end side. Cone-rib lubrication significantly improved high-speed tapered-roller bearing performance, yielding lower cone-face temperatures and lower power loss and allowing lower lubricant flow rates for a given speed condition. Bearing temperatures increased with increased shaft speed and decreased with increased lubricant flow rate. Bearing power loss increased with increased shaft speed and increased lubricant flow rate.

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by Richard J. Parker and Hans R. Signer*

Lewis Research Center

SUMMARY

The performance of 120.65-mm- (4.75-in. -) bore tapered-roller bearings was investigated at shaft speeds up to 1.81 million DN (cone-rib tangential velocities up to 130 m/sec (25 500 ft/min)). Temperature distribution and bearing heat generation were determined as a function of shaft speed, radial and thrust loads, lubricant flow rate, and lubricant inlet temperature. Lubricant was supplied by either jets or by a combination of holes through the cone directly to the cone-rib contact and jets at the roller small-end side. Test conditions included shaft speeds from 6000 to 15 000 rpm, radial loads from 13 300 to 26 700 N (3000 to 6000 lb), thrust loads from 26 700 to 53 400 N (6000 to 12 000 lb), lubricant flow rates from 0.0019 to 0.0151 m^3/min (0.5 to 4.0 gpm), and lubricant inlet temperatures of 350 and 364 K (170° and 195° F).

Cone-rib lubrication significantly improved tapered-roller bearing performance at high speeds. Higher shaft speeds were attainable at lower lubricant flow rates when cone-rib lubrication was used. Cone-face temperatures ranged from 14 K (25° F) lower at 6000 rpm to 34 K (62° F) lower at 10 000 rpm when cone-rib lubrication was used rather than lubrication by jets only for the same total flow rates. For stable tapered-roller bearing operation at 15 000 rpm, a total flow rate as low as 0.0057 m^3/min (1.5 gpm) with cone-rib lubrication was successful, whereas 0.0151 m^3/min (4.0 gpm) was required where lubrication by jets only was used. The portion of bearing power loss determined by heat transfer to the lubricant was less with cone-rib lubrication than with jet lubrication for equivalent flow rates. Bearing temperatures and power loss increased with increased shaft speed. With increased lubricant flow rate, bearing temperatures decreased and bearing power loss increased.

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*Industrial Tectonics, Inc., Compton, California.

INTRODUCTION

Tapered-roller bearings are being used in some helicopter transmissions to carry combined radial, thrust and moment loads and in particular, those loads from bevel gears such as high-speed input pinions. For this application, tapered-roller bearings have greater load capacity for a given envelope or for a given bearing weight than the more commonly used ball and cylindrical roller bearings. Speed limitations have restricted the use of tapered-roller bearings to lower speed applications relative to 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 high speeds. The speed of tapered-roller bearings is limited to approximately 0.5 million DN (a cone-rib tangential velocity of approximately 36 m/sec (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.

Several means of supplying lubricant directly to this cone-rib contact were investigated at higher speeds in reference 1. Results of the work in reference 1 indicate the most successful means was to supply lubricant to the cone-rib contact through holes from the bore of the cone. Additionally, the radius of the spherical large end of the roller was optimized at 75 to 80 percent of the apex length. Development of a large, high-speed tapered-roller bearing for a heavy-lift helicopter transmission was reported in reference 2. The feasibility of tapered-roller bearings for the high speed and nearly pure thrust load conditions of turbine engine main-shaft bearings was reported for large and small bores in references 3 and 4, respectively.

The use of computer programs can increase the capability of designing and analyzing tapered-roller bearings for such high-speed applications. These programs, described in references 5 and 6, take into account the difficulty of lubricating the contacts in high-speed tapered-roller bearings and consider the effects of the elastohydrodynamic (EHD) films in these contacts. Reference 7 discusses the effects of the EHD films in tapered-roller bearing contacts. Experimental data at higher speeds are needed to verify the predictions of these computer programs.

The research reported herein was undertaken to investigate the performance of 120.65-mm- (4.75-in.-) bore tapered-roller bearings at speeds up to 15 000 rpm (1.81×10^6 DN). The maximum cone-rib tangential velocity at this speed was 130 m/sec (25 500 ft/min). The objective of this program was to determine the operating characteristics, including temperature distribution and heat generation, of this bearing as a function of shaft speed, radial and thrust load, lubricant flow rate, and lubricant inlet temperature. Lubrication was applied either by jets or by a combination of holes through the cone directly to the cone-rib contact and jets at the roller small-end side. Test conditions included shaft speeds of 6000, 10 000, 12 500, and 15 000 rpm, radial loads of 13 300 to 26 700 N (3000 to 6000 lb), thrust loads of 26 700 to 53 400 N (6000 to 12 000 lb), lubricant flow rates from 0.0019 to 0.0151 m^3/min (0.5 to 4.0 gpm); and lubricant-in temperatures of 350 and 364 K (170° and 195° F).

APPARATUS AND PROCEDURE

Test Rig

Mechanical arrangement. - Two test bearings are mounted on a spindle as shown in figure 1. The cup of each test bearing is mounted in a test head assembly. The right test head assembly is mounted rigidly to the machine frame; whereas the left test head assembly is axially movable and is supported on ball and roller ways. This arrangement allows thrust loading of the test bearings with a hydraulic actuator. Radial load is applied to the test bearings by a hydraulic actuator that exerts a force on a center housing containing two cylindrical roller support bearings mounted on the spindle.

A flat-belt-pulley system is used to drive the test spindle from a 3600-rpm, 75-kilowatt (100-hp), 460-volt, three-phase electric motor. A reduced voltage motor started is used to select the desired acceleration rate during startup. Test spindle

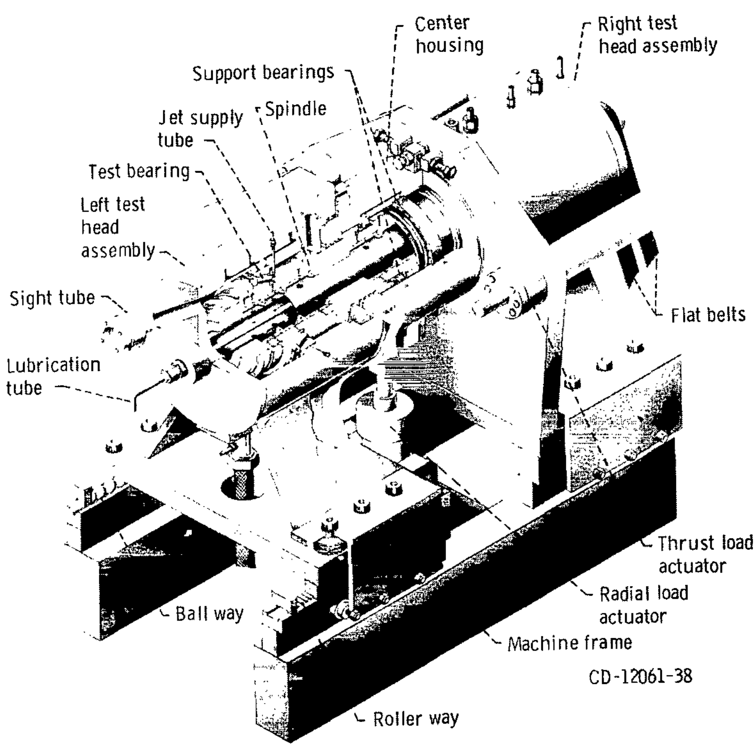


Figure 1. - Pictorial view of high-speed tapered roller bearing test rig.

speeds of 6000, 10 000, 12 500, 15 000, and 20 000 rpm are chosen by exchanging drive pulleys on the motor. The pair of flat belts are guided by an idler pulley arrangement which maintains controller preload on the slack side of the belts. The test head assemblies, hydraulic actuator, drive motor, and idler pulley are shown in the photograph in figure 2.

The test spindle is hollow and contains contoured inserts with annular grooves to distribute lubricant to radial holes for cone-rib lubrication of the test bearing and for lubrication of the load bearing. A stationary lubrication tube delivers the desired lubricant flow to the annular grooves. For jet lubrication of the test bearings, two supply tubes are located 180° apart at the roller small end of each test bearing. Each tube contains two holes directed at the test bearing as shown in figure 3. The hole diameters were varied to maintain desired jet velocities and maintain a ratio of two-to-one of the flow directed inboard of the cage to that directed outboard of the cage.

Lubrication system. - The lubrication system contains five lubrication circuits supplied from a single 0.03-cubic-meter (8-gal) sump. Three of the circuits supply lubricant to the test bearings for cone-rib lubrication, jet lubrication, and cup (outer ring) cooling. The remaining two lubrication circuits direct lubricant to the cylindrical roller support bearings for under-race lubrication and cooling and outer ring cooling. The flow through each circuit is metered with variable flow control valves and measured with a

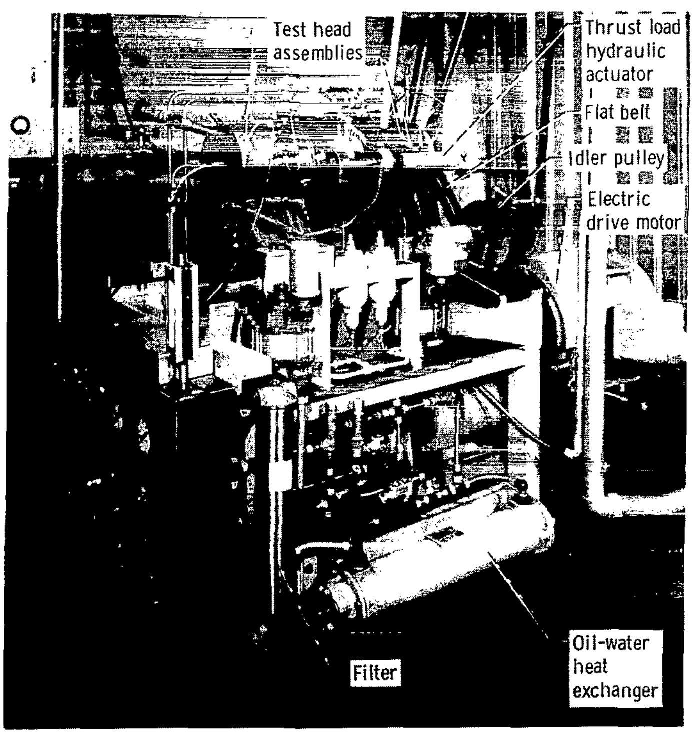


Figure 2. - Photograph of high-speed tapered roller bearing test rig.

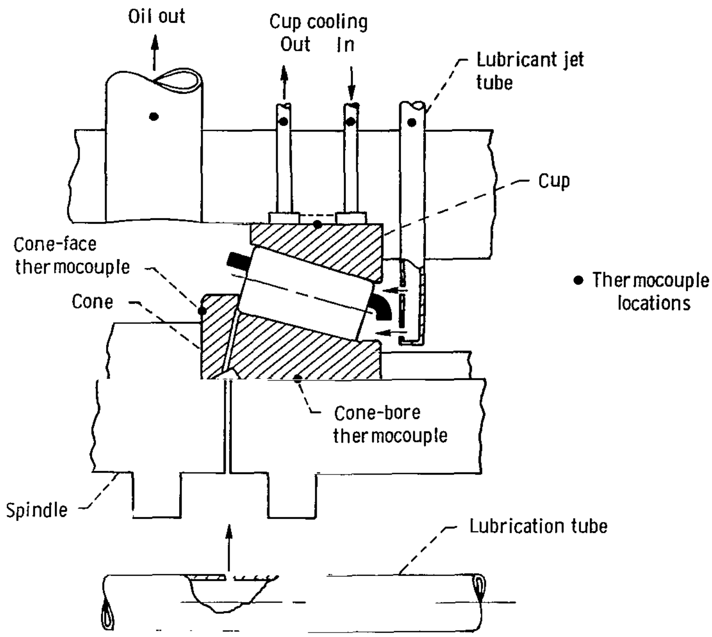


Figure 3. - Test bearing lubrication and thermocouple locations.

flowmeter. A pump supplies a total flow rate of 0.0454 m^3/min (12 gpm) at a manifold pressure of 0.55 MPa (80 psi). A high capacity 10-micron filter, flow and level switches, relief valves, and pressure gages protect the hydraulic circuits. An oil-water heat exchanger is used with appropriate mixing valves to control the oil-in temperature to the desired level. Large capacity gravity drain lines scavange oil from the test bearing and cylindrical roller bearing areas. Hydraulic pressure for the thrust and radial load actuators is supplied from a high pressure, 41.4-MPa (6000-psi) system separate from the lubrication system.

Instrumentation. - Thermocouples are installed for temperature measurements of each test bearing cup, each cylindrical load bearing outer ring, and oil inlet and outlet temperatures or both test and load bearings. Temperatures of the cone bore and cone face of the test bearing on the drive end of the test spindle were measured with thermocouples and either a slip-ring system or an FM telemetry system. The thermocouple on the cone face was located approximately 2.3 mm (0.09 in.) from the outside diameter of the cone rib. An alternate means of cone-face temperature measurement was with an infrared pyrometer aimed through an air-purged sight tube assembly. However, all bearing temperatures reported herein were measured with thermocouples on the bearing at the drive end of the test spindle.

The machine vibration level is measured with piezoelectric accelerometers which automatically shut down the test when machine vibration exceeds a predetermined level due to bearing failures. Proximity probes measure shaft excursion in two planes as well as shaft speed and test bearing separator speed. Meters to measure drive motor line voltage and current were incorporated to monitor machine power requirements. Preset safety flow switches and oil level switches were used to shut down the test machine in the event of lubrication system malfunction.

Test Bearings

The tapered-roller test bearings had a bore of 120.6 mm (4.75 in.) and an outside diameter of 206.4 mm (8.125 in.). The cup angle was 34°, and the roller included angle was $3^{\circ}10'$. The bearing contained 25 rollers with a large end diameter of 18.29 mm (0.720 in.) and an overall length of 34.17 mm (1.3452 in.). The rollers were fully crowned with a crown radius of 25.4×10^{3} mm (1000 in.) and had a spherical end radius equal to 80 percent of the apex length.

The material of the cup, cone, and rollers was case-carburized consumable-electrode vacuum-melted AISI 4320 steel. The one piece, roller-riding cage was silver plated AISI 1010 steel. The hardnesses, case depth, and surface finish specifications are shown in table I.

The cone contained forty oil holes, 1.016 mm (0.040 in.) in diameter, drilled through from a manifold on the cone bore to the undercut at the large end of the cone, as shown in figure 3.

TABLE I. - TEST BEARING SPECIFICATIONS

Cup, cone, and roller material	AISI 4320
Case hardness, Rockwell C	58 to 64
Core hardness, Rockwell C	25 to 48
Case depth (to 0.5 % carbon level after final grind), cm (in.):	
Cup and cone	0.086 to 0.185 (0.034 to 0.073)
Roller	0.091 to 0.201 (0.036 to 0.079)
Surface finish[a], μm(μin.) rms:	
Cone raceway	0.15 (6)
Cup raceway	0.20 (8)
Cone-rib	0.18 (7)
Roller taper	0.13 (5)
Roller spherical	0.15 (6)

[a]Measured values.

The basic dynamic load ratings for this bearing are 74 700 N (16 800 lb) radial load and 58 700 N (13 200 lb) thrust load. (The thrust or radial load which gives 10 percent life of 90 million cone revolutions.) The Antifriction Bearing Manufacturers Association (AFBMA) basic dynamic capacity is 288 000 N (64 800 lb).

Lubricant

The oil used for these studies was a 5-centistoke neopentylpolyol (tetra) ester. This type II oil is qualified to MIL-L-23699 as well as to the internal oil specifications of most major aircraft turbine engine manufacturers. Properties of the oil are presented in table II. A temperature-viscosity curve is shown in figure 4.

TABLE II. - 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)
Fire point, K (°F)	Unknown
Autoignition temperature, K (°F)	694 (800)
Pour point, K (°F)	214 (-75)
Volatility (6.5 hr at 477 K (400° F)), wt. %	3.2
Specific heat at 372 K (210° F), J/(kg)(K) (Btu/(lb)(°F))	2140 (0.493)
Thermal conductivity at 372 K (210° F), J/(m)(sec)(K) (Btu/(hr)(ft)(°F))	0.15 (0.088)
Specific gravity at 372 K (210° F)	0.931

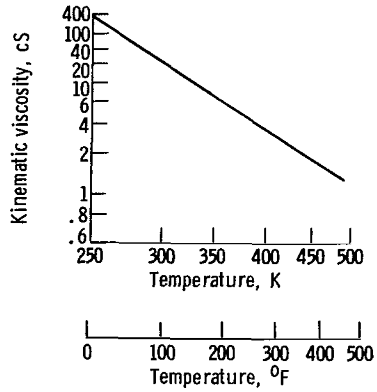


Figure 4. - Viscosity as function of temperature for tetraester (type II) lubricant.

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. Test parameters such as load, speed, and oil inlet temperature were maintained constant while the tester was in operation. Lubricant flow rates were adjusted during operation. The test bearings were allowed to reach an equilibrium condition before data were recorded and the next test condition was sought.

RESULTS AND DISCUSSION

Effect of Lubricant Flow on Bearing Temperatures

The effect of lubricant flow rate either through jets (jet lubrication) or through holes in the cone (cone-rib lubrication), was determined for a variety of speeds, loads, and oil-in temperatures. Temperatures of the 120.65-mm- (4.75-in.-) bore tapered-roller test bearing at the drive end of the test spindle were measured on the cone bore and the cone face as well as on the outer surface of the cup. Oil-out temperature was also measured. Test spindle speeds ranged from 6000 to 15 000 rpm. Thrust loads varied from 26 700 to 53 400 N (6000 to 12 000 lb). Radial loads varied from 13 300 to 26 700 N (3000 to 6000 lb). The range of calculated maximum Hertz stresses at several of these conditions are shown in table III. Lubricant flow rate was varied from 0.0019 to 0.0151 m^3/min (0.5 to 4.0 gpm). When cone-rib lubrication was used, a constant 0.0038 m^3/min (1.0 gpm) of jet flow was also used to assure some lubrication of the roller small end.

TABLE III. - CALCULATED MAXIMUM HERTZ STRESSES AT SEVERAL CONDITIONS OF TAPERED-ROLLER BEARING PERFORMANCE TESTS

Shaft speed, rpm	Thrust load		Radial load		Maximum Hertz stress at most heavily loaded roller					
					Cup		Cone		Cone rib	
	N	lb	N	lb	MPa	psi	MPa	psi	MPa	psi
6 000	26 700	6 000	26 700	6000	951	138 000	1010	146 000	156	22 700
10 000	40 000	9 000	↓	↓	1062	154 000	1080	156 000	187	27 100
10 000	53 400	12 000			1160	168 000	1190	173 000	194	28 200
15 000	53 400	12 000			1160	168 000	1090	158 000	220	31 900

Test bearing temperatures and oil-out temperatures, measured at these test conditions, are shown in figures 5 to 9. Figures 5 and 6 show no effect of radial load and very little effect of thrust load on cone face temperature. This data is typical throughout the range of data taken. That is, regardless of speed, oil-in temperatures, or flow rates, load had little effect on bearing or oil-out temperatures. Therefore, the data shown in figures 7 to 9 are for only one load condition, that is, 53 400 N (12 000 lb) thrust load and 26 700 N (6000 lb) radial load.

Figure 7 shows the general decrease in bearing and oil-out temperatures with increased flow rate at a shaft speed of 6000 rpm for both jet and cone-rib lubrication and for oil-in temperatures of 350 and 364 K (170° and 195° F). With jet lubrication (figs. 7(a) and (b)), temperatures are decreased by an average of approximately 28 K (50° F) as flows are increased from 0.0019 to 0.0078 m^3/min (0.5 to 2.0 gpm). At higher flow rates, the rate of temperature decrease diminishes.

Similar effects are seen in figures 7(c) and (d) for cone-rib lubrication, where the total flow rate includes 0.0038 m^3/min (1.0 gpm) of lubricant through jets at the roller small end of the bearing. Thus, the data points at 0.0038 m^3/min (1.0 gpm) are for zero cone-rib flow rate.

Figure 8 shows flow rate effects on bearing and oil-out temperatures at a shaft speed of 10 000 rpm. The trends are similar to the 6000-rpm data. Data at jet flow rates below 0.0078 m^3/min (2.0 gpm) were not obtained, since the first test at a jet flow rate of 0.0038 m^3/min (1.0 gpm) at an oil-in temperature of 350 K (170° F) resulted in surface distress of the cone rib on one of the test bearings prior to reaching equilibrium. The test bearing on which the cone-face and cone-bore temperatures were measured was not damaged. Further tests at this flow rate and below were not run, including those at the higher oil-in temperature.

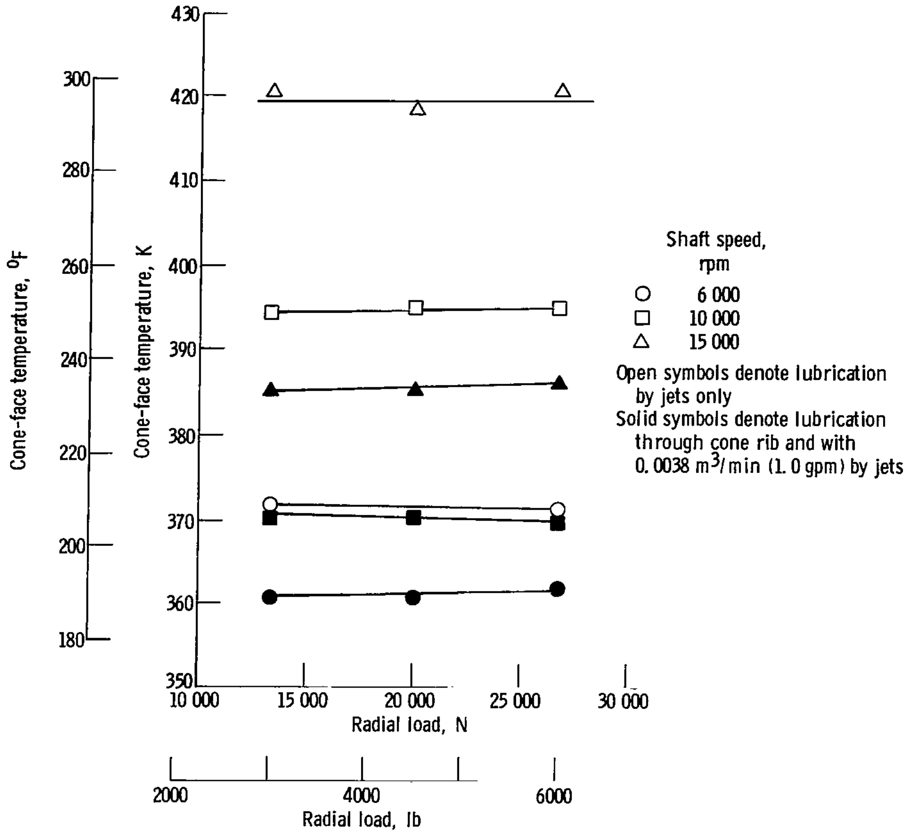


Figure 5. - Effect of radial load on cone-face temperature. Thrust load, 53 400 N (12 000 lb); oil-in temperature, 350 K (170° F); total oil flow, 0.0151 m³/min (4.0 gpm).

Extrapolation of the cone-face temperature in figure 8(a) to a flow rate of 0.0038 m^3/min (1.0 gpm) shows that a temperature in excess of 433 K (320° F) could have existed. The temperature of the cone-rib contact with the roller large end would undoubtedly be even higher. If the severity of this rolling/sliding contact and the temperature limitations of the AISI 4320 material of the cone and rollers are considered, the occurrence of surface damage at this condition is not surprising.

Data for cone-rib flow rates as low as 0.0019 m^3/min (0.5 gpm) (total flow rate of 0.0057 m^3/min (1.5 gpm)) were obtained at 10 000 rpm for both oil-in temperatures. Maximum cone-face temperatures at this flow rate was only 403 K (267° F) at an oil in temperature of 364 K (195° F) (fig. 8(d)).

The effects of flow rate at a shaft speed of 15 000 rpm are shown in figure 9. Only one test condition was run at 15 000 rpm with jet flow. Those data, at 0.0151 m^3/min (4.0 gpm), are shown as the solid data points in figure 9(a). It was anticipated that lower jet flow rates at this shaft speed may allow excessive cone-rib temperatures and cause subsequent surface distress.

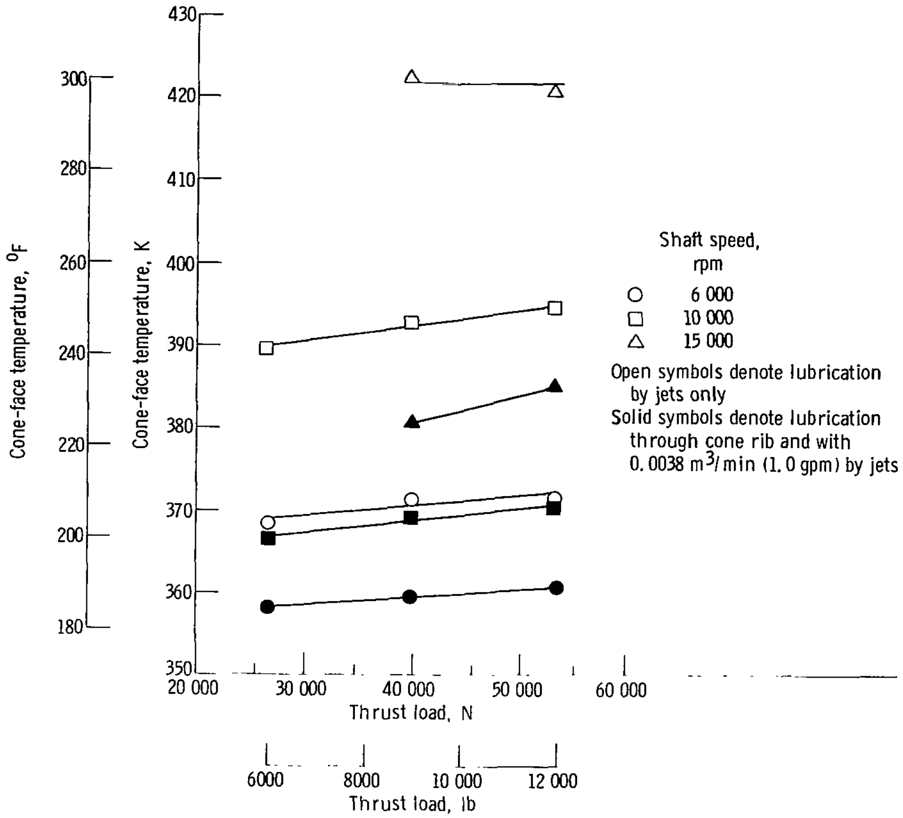


Figure 6. - Effect of thrust load on cone-face temperature. Radial load, 13 400 N (3000 lb); oil-in temperature, 350 K (170° F); total oil flow, 0.0151 m^3/min (4.0 gpm).

Data were obtained for all desired flow rate conditions with cone-rib flow at 350 K (170° F) oil-in temperature (fig. 9(a)). Data at an oil-in temperature of 364 K (195° F), figure 9(b), were not obtained for total flow rates less than 0.0076 m^3/min (2.0 gpm) due to temperature limitations. (Data shown at 0.0076 m^3/min (2.0 gpm) were obtained at 20 000 N (4500 lb) radial load.) An increased effect of flow rate on oil-out temperature is seen at this higher shaft speed of 15 000 rpm, whereas the flow rate effect on bearing temperatures at the higher speed is not significantly different from that at the lower speeds.

In general, the effects on bearing temperatures of flow rates above 0.0114 m^3/min (3.0 gpm) are small. The use of flow rates greater than this for these bearings at these conditions does not appear to be warranted, especially where cone-rib lubrication is used. Additionally, as will be shown later, bearing power loss increases as lubricant flow rates are increased.

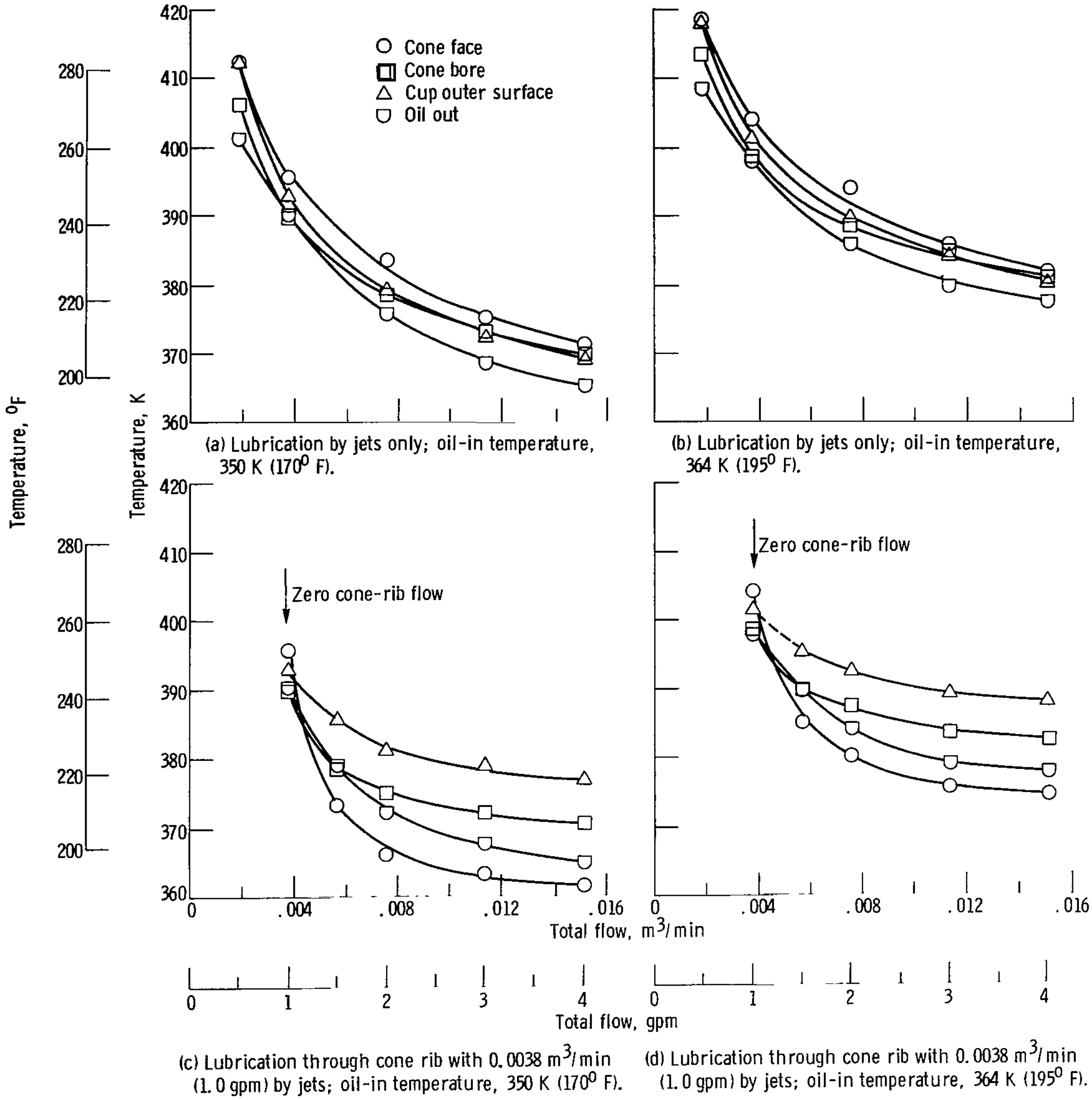


Figure 7. - Temperature as a function of flow rate at shaft speed of 6000 rpm. Thrust load, 53 400 N (12 000 lb); radial load, 26 700 N (6000 lb).

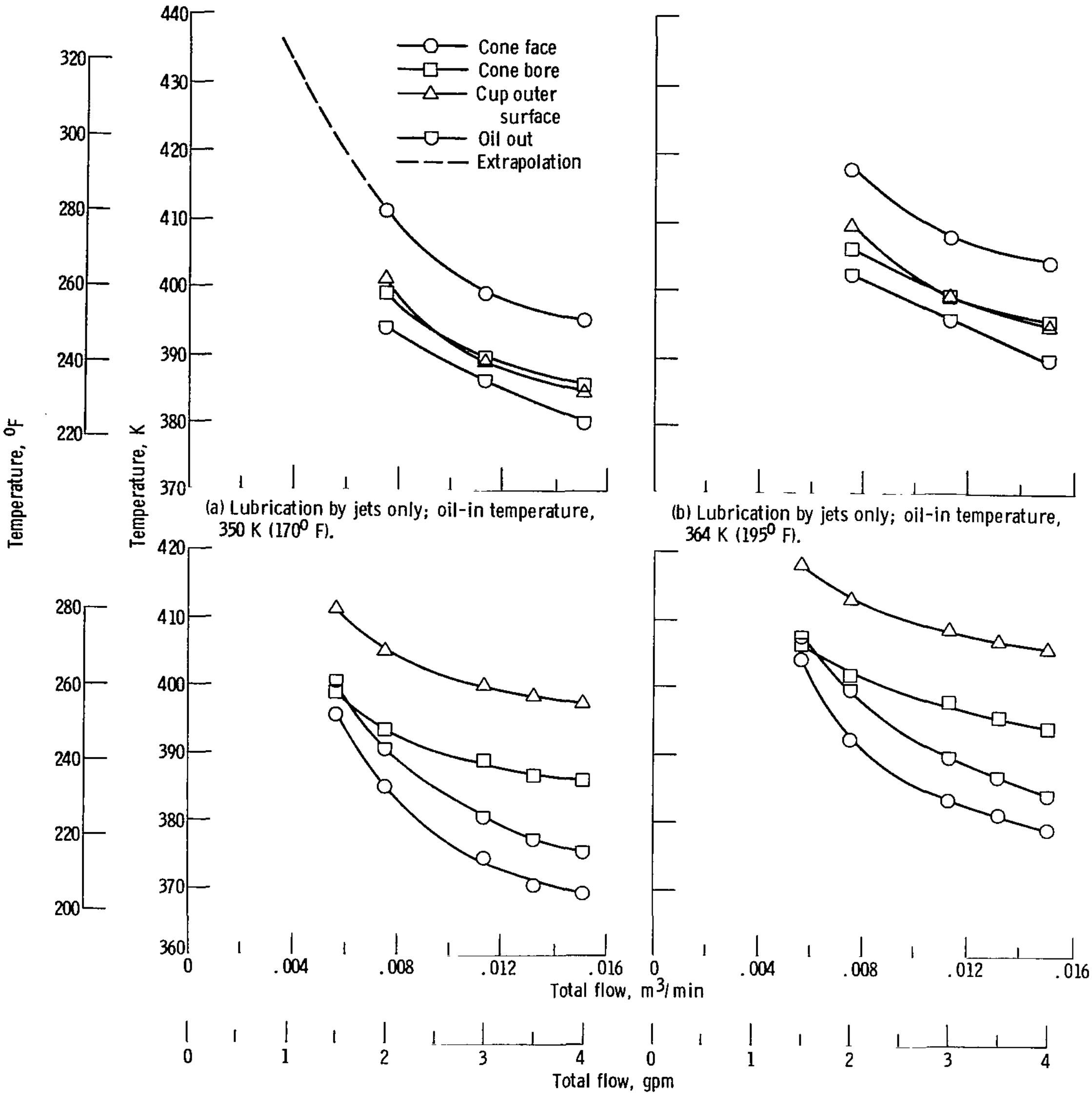


Figure 8. - Temperature as a function of flow rate at a shaft speed of 10 000 rpm. Thrust load, 53 400 N (12 000 lb); radial load, 26 700 N (6000 lb).

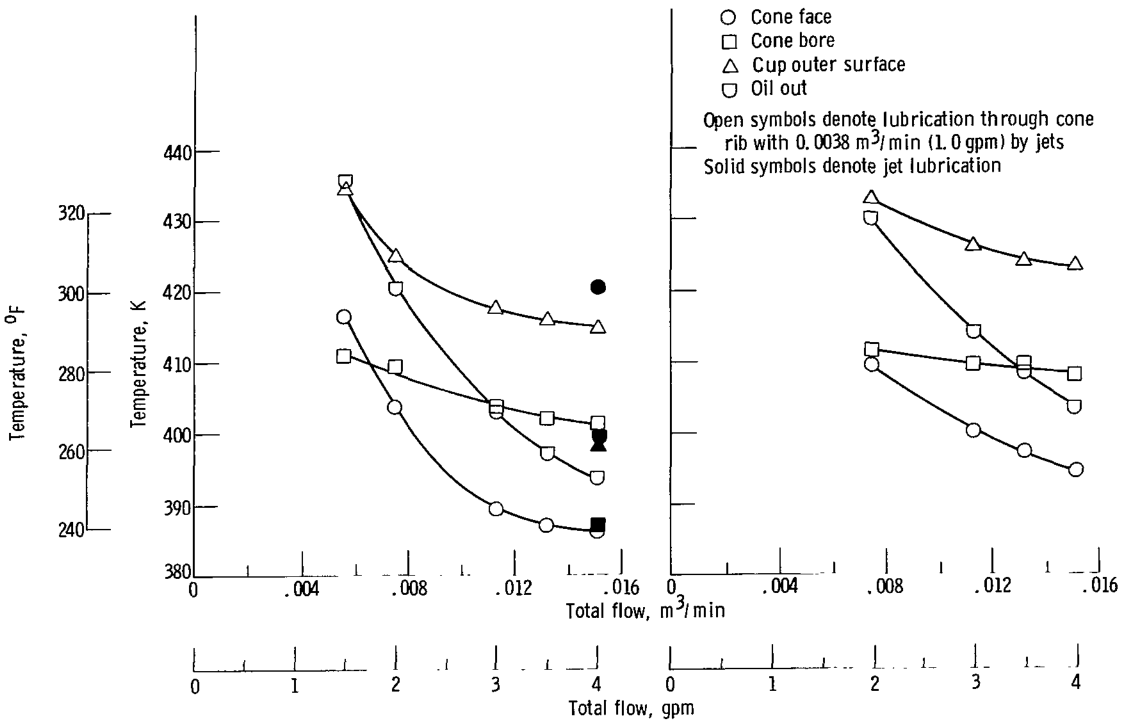
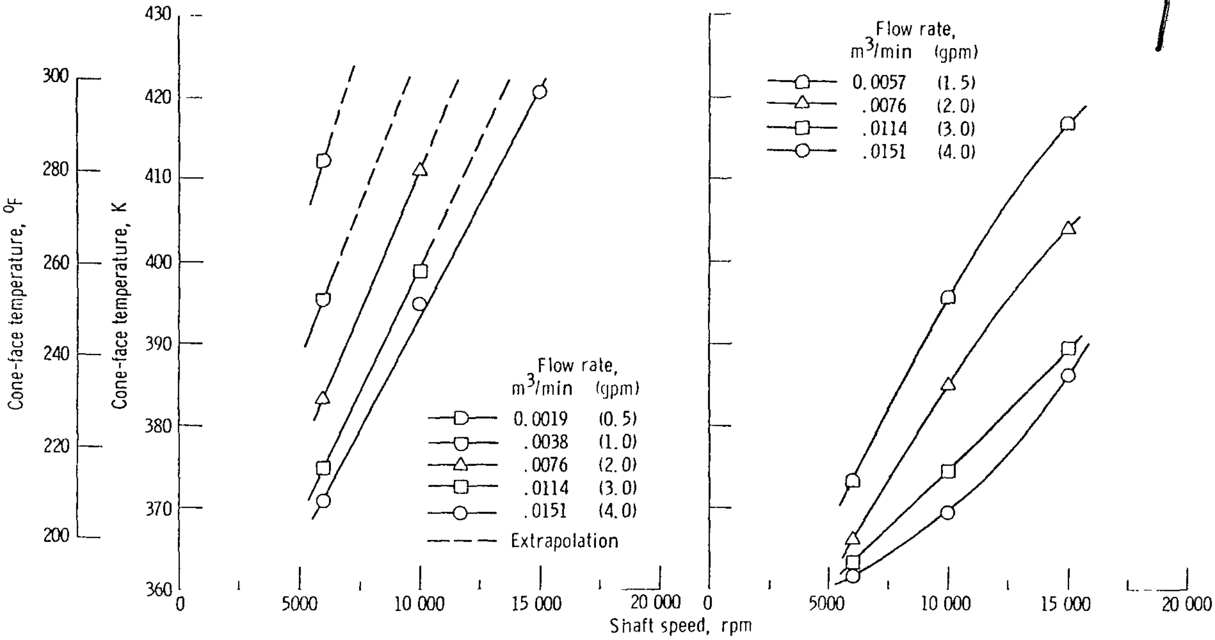


Figure 9. - Temperature as function of flow rate at shaft speed of 15 000 rpm. Thrust load, 53 400 N (12 000 lb); radial load, 26 700 N (6000 lb).

Effect of Shaft Speed and Cone-Rib Lubrication

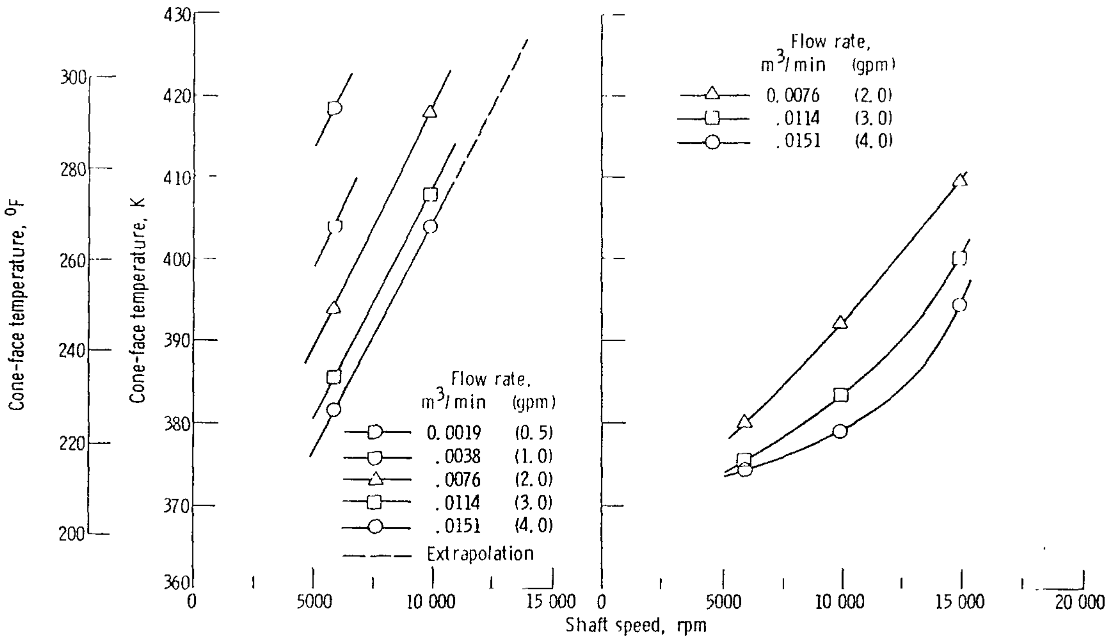
The effect of shaft speed on cone-face temperature is shown in figures 10 and 11 for oil-in temperatures of 350 and 364 K (170° and 195° F), respectively. Increasing the shaft speed from 6000 to 15 000 rpm increases cone-face temperature by as much as 49 K (89° F). Shaft speed has a lesser effect on cone-face temperature where cone-rib lubrication is used rather than jet lubrication. It is apparent that extrapolation of the data in figure 10(a) to 15 000 rpm for jet flow rates less than 0.0076 m^3/min (2.0 gpm) at 350 K (170° F) oil-in temperature would give cone-face temperatures in excess of 430 K (320° F). Similarly, the data of figure 11(a) extrapolate to greater than this temperature for all the flow rates at 364 K (195° F) oil-in. For lubrication through the cone rib (with 0.0038 m^3/min (1.0 gpm) jet flow), satisfactory cone-face temperatures were obtained at 15 000 rpm with total flow rates as low as 0.0057 m^3/min (1.5 gpm) for 350 K (170° F) oil-in temperature (fig. 10(b)) and 0.0076 m^3/min (2.0 gpm) at 364 K (195° F) oil-in temperature (fig. 11(b)).



(a) Lubrication by jets only.

(b) Lubrication through cone rib with 0.0038 m^3/min (1.0 gpm) by jets.

Figure 10. - Effect of shaft speed and flow rate on cone-face temperature for oil in temperature of 350 K (170^0 F). Thrust load, 53 400 N (12 000 lb); radial load, 26 700 N (6000 lb).



(a) Lubrication by jets only.

(b) Lubrication through cone rib with 0.0038 m^3/min (1.0 gpm) by jets.

Figure 11. - Effect of shaft speed and flow rate on cone-face temperature for oil-in temperature of 364 K (195^0 F). Thrust load, 53 400 N (12 000 lb); radial load, 26 700 N (6000 lb).

The advantage of cone-rib lubrication is further illustrated in figure 12. The difference in the temperature of the cone-face with jet lubrication and that with cone-rib lubrication increases with shaft speed. At 15 000 rpm, the difference is 34 K (62° F). Even at the lower speed of 6000 rpm, the temperature imorovement is an average of approximately 13 K (23° F).

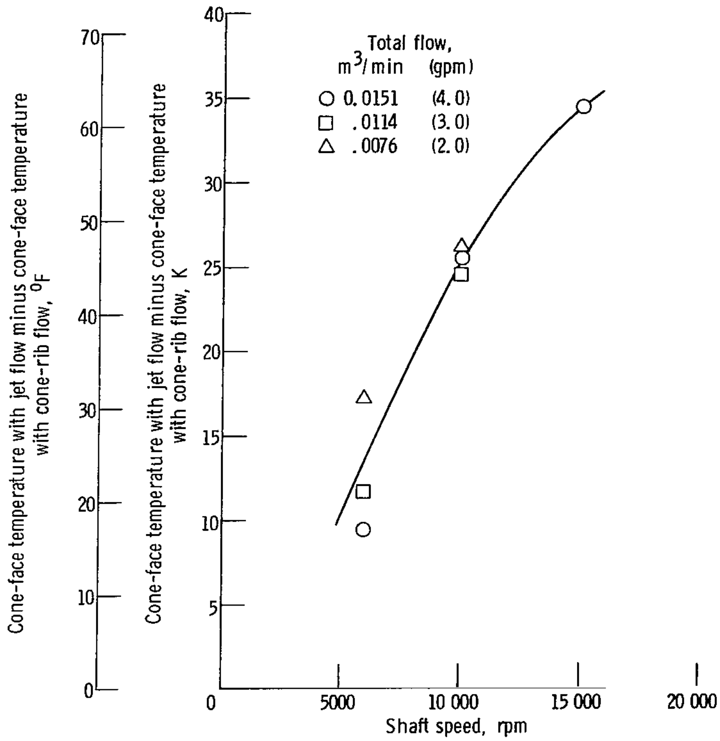


Figure 12. - Effect of shaft speed on cone-face temperature with jet lubrication minus that with cone-rib lubrication. Oil-in temperature, 350 K (170° F); thrust load, 53 400 N (12 000 lb); radial load, 26 700 N (6000 lb).

It can be observed from figures 7 to 9 that, when cone-rib lubrication is used, the highest bearing temperature measured at each condition is at the cup outer surface. When jet lubrication alone is used, the highest measured temperatures were on the cone face. This effect is further illustrated in figure 13 where the temperatures are plotted against shaft speed for an oil-in temperature of 364 K (195° F) and a total oil flow of 0.0114 m^3/min (3.0 gpm). Cone-bore and oil-out temperatures for jet lubrication and for cone-rib lubrication are not significantly different. It is believed that, when cone-rib lubrication is used, less oil is thrown centrifugally outward to cool the cup before it leaves the bearing. Also, the oil that is directed through the cone rib and does contact the cup has been heated somewhat in cooling the cone rib. Thus, a somewhat greater cup

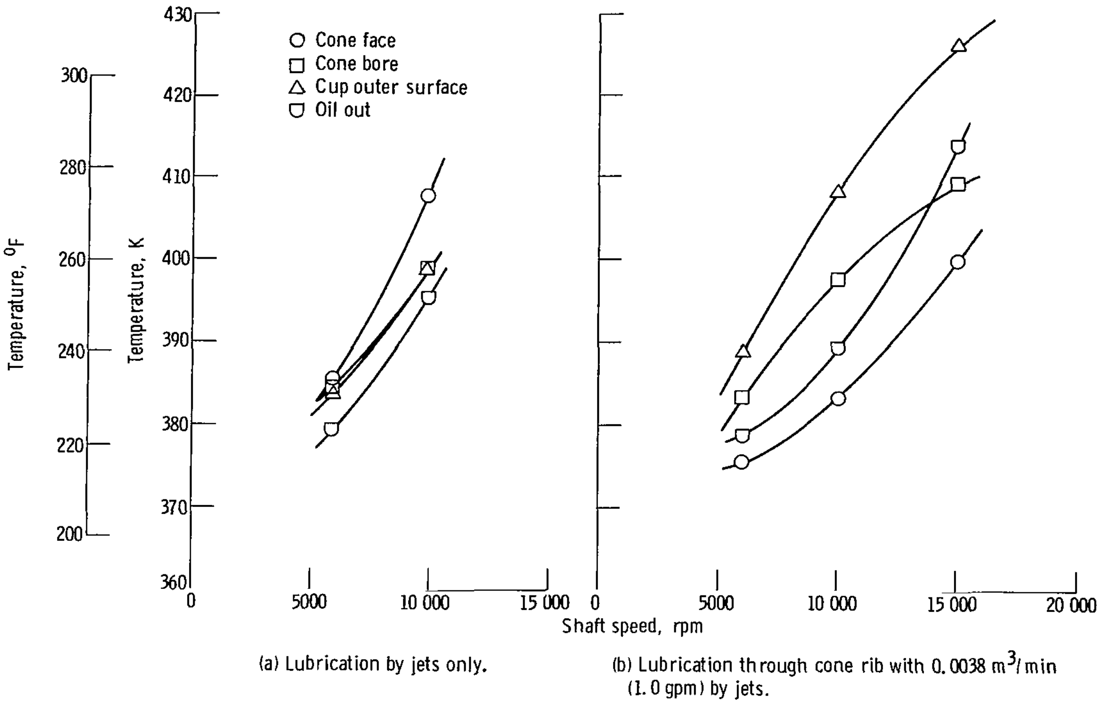


Figure 13. - Effect of jet lubrication and cone-rib lubrication on bearing and oil-out temperatures. Thrust load, 53 400 N (12 000 lb); radial load, 26 700 N (6000 lb); oil-in temperature, 364 K (195° F); total oil flow rate, 0.0114 m^3/min (3.0 gpm).

temperature has accompanied a cooler cone rib, but because of the critical nature of the cone-rib contact in high-speed tapered roller bearings, this small sacrifice appears justified.

The higher cup temperatures may be decreased with the use of cup cooling oil flowing in the cup housing in contact with the outer surface of the cup. Figure 14 includes some additional temperature data obtained at a shaft speed of 12 500 rpm and 0.0057 m^3/min (1.5 gpm) total oil flow (cone-rib flow of 0.0019 m^3/min (0.5 gpm) plus jet flow of 0.0038 m^3/min (1.0 gpm)). With the addition of 0.0026 m^3/min (0.7 gpm) cup cooling flow (solid symbols in fig. 14), the cup outer surface temperature is decreased 14 K (25^o F) without significant change in cone-face and cone-bore temperatures. Oil-out temperature was 6 K (11^o F) lower due to the quantity of heat removed by the 364 K (195^o F) cup cooling oil which was measured at 380 K (225^o F) upon exit from the cooling passages.

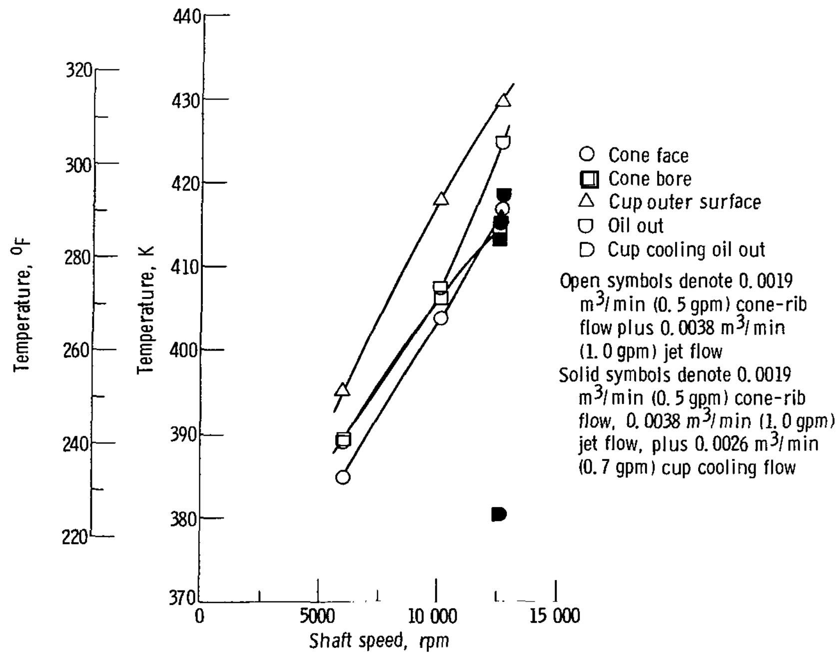
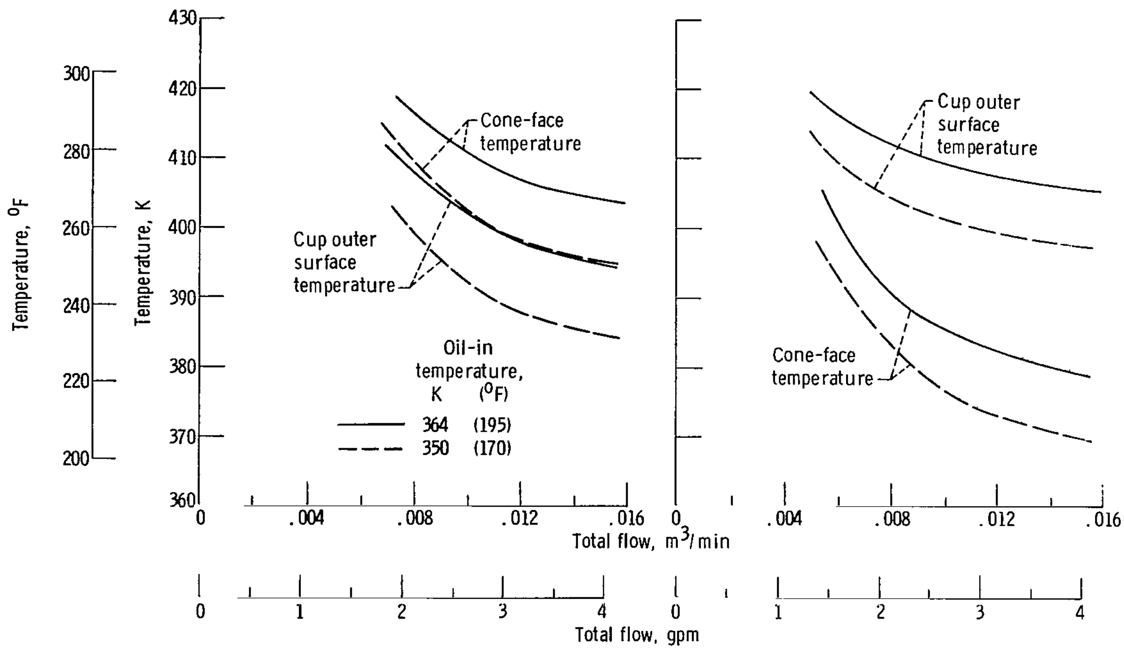


Figure 14. - Effect of cup cooling on bearing and oil-out temperatures. Thrust load, 53 400 N (12 000 lb); radial load, 26 700 N (6000 lb); oil-in temperature, 364 K (195° F).

Effect of Oil-in Temperature

The effect of oil-in temperature on cone-face and cup outer surface temperatures at a shaft speed of 10 000 rpm is shown in figure 15. At an oil-in temperature of 364 K (195° F), the cup outer surface and cone-face temperatures are from 7 to 10 K (12° to 18° F) higher than for an oil-in temperature of 350 K (170° F) for both jet lubrication and for cone-rib lubrication. There is a slight indication that oil-in temperature has a greater effect on cone-face temperature at the higher flow rates. These effects and trends were similar for shaft speeds of 6000 and 15 000 rpm and for the cone-bore and oil-out temperatures. In general, the net change in bearing temperatures is on the order of 50 to 75 percent of the change in oil-in temperature.



(a) Lubrication by jets only.

(b) Lubrication through cone rib with 0.0038 m^3/min (1.0 gpm) by jet.

Figure 15. - Effect of oil-in temperature on cone-face and cup outer surface temperatures. Shaft speed, 10 000 rpm; thrust load, 53 400 N (12 000 lb); radial load, 26 700 N (6000 lb).

Effects of Speed and Lubricant Flow on Bearing Power Loss

The power loss from the bearing is dissipated in the form of heat by conduction to the lubricant and by convection and radiation to the surrounding environment. Lubricant outlet temperature from the bearing was measured for all conditions of flow. Heat transferred to the lubricant was calculated using the following standard heat transfer equation:

$$Q_T = MC_p(t_{out} - t_{in}) \tag{1}$$

where

Q_T total heat transfer to lubricant, J/min (Btu/min)

M lubricant mass flow, kg/min (lb/min)

C_p specific heat, J/(kg)(K) (Btu/(lb)(°F))

t_{out} oil outlet temperature, K (°F)

t_{in} oil inlet temperature, K (°F)

The result of these heat transfer calculations are shown in figure 16 as a function of

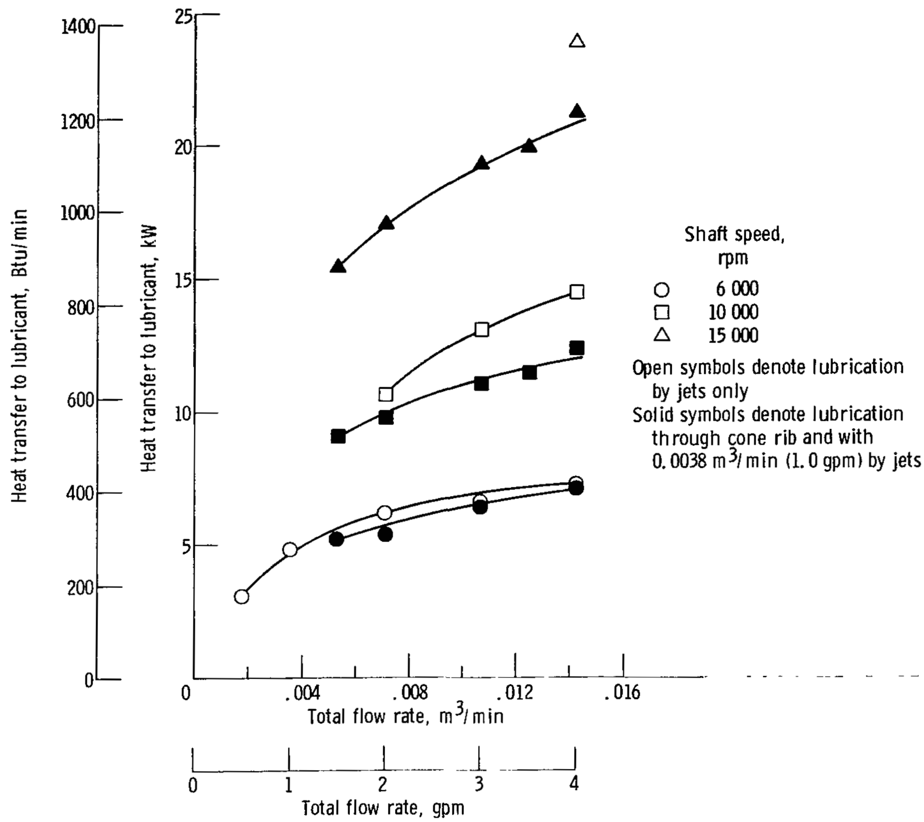


Figure 16. - Heat transferred to lubricant as function of total flow rate. Thrust load, 53 400 N (12 000 lb); radial load, 26 700 N (6000 lb); oil-in temperature, 350 K (170° F).

shaft speed and total flow rate. (For convenience, heat values were converted from J/min to kW.) The heat transferred to the lubricant increases with both increased shaft speed and increased lubricant flow rate. These increases are expected due to increased lubricant drag or churning. These heat quantities are a portion of the heat generated in the test bearings and do not include heat transferred from the bearing by conduction, convection, and radiation. At higher bearing temperatures, the heat transferred by these latter forms should become a greater portion of the total.

An equation for calculating operating torque of tapered-roller bearings has been developed by Witte (ref. 8). The equation is based on a dimensional analysis of the operating and geometric variables involved. Experimental data are used to obtain exponents and constants. The equation for combined radial and thrust loads on tapered-roller bearings as published in reference 8 is

$$M = AG(S\mu)^{1/2}\left(\frac{f_T F_r}{K}\right)^{1/3} \qquad (2)$$

where

M total bearing torque, N-m (lb-in.)

A constant determined from test data of ref. 8

G bearing geometry factor given in ref. 8

S shaft speed, rpm

μ lubricant absolute viscosity at atmospheric pressure, cP

f_T equivalent thrust load factor given as a function of ratio of thrust-to-radial load in ref. 8

F_r bearing radial load, N (lb)

K bearing K factor = ratio of basic dynamic radial load rating to basic dynamic thrust load rating

Operating torque for the test bearing used in this program was calculated using this equation and converted to power loss for comparison with the heat transfer data. This comparison is shown in table IV. At the lower speeds the calculated bearing power loss and the heat transferred to the lubricant are very similar in magnitude. At the higher speeds, the heat transferred to the lubricant is less than the calculated power loss. This effect may be expected, since at higher speeds, and thus higher temperatures, losses due to convection and radiation to the surrounding environment would be greater. The

TABLE IV. - HEAT TRANSFERRED TO LUBRICANT AND BEARING POWER LOSS FOR SEVERAL SPEED AND LOAD CONDITIONS

[Total flow rate, 0.0114 m^3/min (3.0 gpm); oil-in temperature, 350 K (170° F); lubrication, jet only for 6000 rpm and cone-rib lubrication for 10 000 rpm and above.]

Shaft speed, rpm	Thrust load		Radial load		Heat transferred to lubricant calculated from equation (1)		Bearing power loss calculated from equation (2)	
	N	lb	N	lb	kW	hp	kW	hp
6 000	26 700	6 000	26 700	6000	5.6	7.5	5.5	7.4
6 000	40 000	9 000	26 700	6000	6.2	8.3	7.2	9.6
6 000	53 400	12 000	26 700	6000	6.6	8.9	7.2	9.6
10 000	26 700	6 000	13 300	3000	11.3	15.1	9.8	13.2
10 000	40 000	9 000	26 700	6000	12.4	16.7	12.4	16.7
10 000	53 400	12 000	26 700	6000	13.0	17.5	12.4	16.7
12 500	53 400	12 000	26 700	6000	12.6	[a]16.9	17.9	24.0
15 000	40 000	9 000	8 900	2000	17.2	23.1	22.0	29.5
15 000	53 400	12 000	13 300	3000	19.2	25.8	24.2	32.4
15 000	53 400	12 000	26 700	6000	19.2	25.8	24.2	32.4

[a]Data for total flow rate of 0.0076 m^3/min (2 gpm).

agreement at lower speeds is reasonable, although the power loss equation (eq. (2)) does not account for effects of lubricant flow rates. Significant increases in heat transferred to the lubricant with increased lubricant flow rate are shown in figure 16.

SUMMARY OF RESULTS

The performance of 120.65-mm (4.75-in.) bore tapered-roller bearings was investigated at shaft speeds up to 15 000 rpm (cone-rib tangential velocities up to 130 m/sec (25 500 ft/min)). Temperature distribution and bearing heat generation was determined as a function of shaft speed, radial and thrust loads, lubricant flow rate, and lubricant inlet temperature. Lubricant was supplied by either jets or by a combination of holes through the cone directly to the cone-rib contact and jets at the roller small-end side. Test conditions included shaft speeds from 6000 to 15 000 rpm, radial loads from 13 300 to 26 700 N (3000 to 6000 lb), thrust loads from 26 700 to 53 400 N (6000 to 12 000 lb), lubricant flow rates from 0.0019 to 0.0151 m^3/min (0.5 to 4.0 gpm), and lubricant inlet temperatures of 350 and 364 K (170° and 195° F). The following results were obtained:

1. Direct cone-rib lubrication significantly improved tapered-roller bearing performance at high speeds. With cone-rib lubrication, total flow rates as low as 0.0057 m^3/min (1.5 gpm) provided stable bearing operation at 15 000 rpm, whereas with jet lubrication alone, a flow rate of 0.0151 m^3/min (4.0 gpm) was required. Bearing power loss was less with cone-rib lubrication than with jet lubrication at the same lubricant flow rates.
2. Bearing temperatures and power loss increased with increasing shaft speed.
3. With increased lubricant flow rate, bearing temperatures decreased and bearing power loss increased.
4. Bearing power loss calculated from a published equation for operating torque of tapered-roller bearings showed very good agreement with the portion of the bearing power loss as determined experimentally from heat transfer to the lubricant.
5. At 6000 rpm, flow rates as low as 0.0019 m^3/min (0.5 gpm) provided stable bearing operation for all conditions tested.
6. Increasing oil-in temperature from 350 to 364 K (170° to 195° F) increased bearing temperatures approximately 7 to 10 K (12° to 18° F).
7. The effect of load on bearing temperatures was very small relative to speed and lubricant flow rate effects.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, October 29, 1976,
505-04.

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