https://ntrs.nasa.gov/search.jsp?R=19800008082 2020-03-21T19:08:47+00:00Z

N O T I C E

THIS DOCUMENT HAS BEEN REPRODUCED FROM MICROFICHE. ALTHOUGH IT IS RECOGNIZED THAT CERTAIN PORTIONS ARE ILLEGIBLE, IT IS BEING RELEASED IN THE INTEREST OF MAKING AVAILABLE AS MUCH INFORMATION AS POSSIBLE

NASA Technical Memorandum 81414

(NASA-TM-81414) PERFORMANCE OF COMPUTER-OPTIMIZED TAPERED-ROLLER BEARINGS TO 2.4 MILLION DN (NASA) 29 p HC A03/MF A01 CSCL 13I
N80-16342
Unclas
G3/37 47036

PERFORMANCE OF COMPUTER-OPTIMIZED TAPERED-ROLLER BEARINGS TO 2.4 MILLION DN

R. J. Parker
Lewis Research Center
Cleveland, Ohio

and

S. I. Pinel, and H. R. Signer
Industrial Tectonics, Inc.
Compton, California

Prepared for the
International Lubrication Conference
cosponsored by the American Society of Mechanical Engineers
and the American Society of Lubrication Engineers
San Francisco, California, August 18-21, 1980

RECEIVED
NASA STI FACILITY
ACCESS DEPT.

PERFORMANCE OF COMPUTER-OPTIMIZED TAPERED-ROLLER BEARINGS TO 2.4 MILLION DN

by R. J. Parker
Lewis Research Center
Cleveland, Ohio

and

S. I. Pinel and H. R. Signer
Industrial Tectonics, Inc.
Compton, California

E-332

ABSTRACT

The performance of 120.65-mm- (4.75-in.-) bore high-speed design tapered roller bearings was investigated at shaft speeds of 20 000 rpm (2.4 million DN) under combined thrust and radial load. The test bearings design was computer optimized for high-speed operation. Temperature distribution and bearing heat generation were determined as a function of shaft speed, radial and thrust loads, lubricant flow rates and lubricant inlet temperature. The high-speed design tapered roller bearing operated successfully at shaft speeds up to 20 000 rpm under heavy thrust and radial loads. Bearing temperatures and heat generation with the high-speed design bearing were significantly less than those of a modified standard bearing tested previously. Cup cooling was effective in decreasing the high cup temperatures to levels equal to the cone temperature.

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. 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/sec (700 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.

By supplying lubricant directly to this critical contact through holes in the cone, tapered-roller bearings have been successfully operated to 1.81 million DN under heavy combined radial and thrust loads (1).[1] Under thrust load only conditions (2) speeds as high as 3.5 million DN have been successfully attained.

The internal geometry of the bearings used in these tests (1 and 2) were standard catalog series designs with modifications of the roller spher-

[1]Numbers in parenthesis designate references at end of paper.

ical ends and lubrication holes through the cone. Also, some of the bearings used in (2) had cages designed for high-speed operation, and the rollers, cones and cups were made of CBS 1000M material.

The use of computer programs can increase the capability of designing and analyzing tapered-roller bearings for high-speed applications. These programs, described in (3 and 4), 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. The analysis of (3) was used to select the internal geometry for the high-speed design tapered-roller bearing used for the research reported herein.

The objective of this research was to determine the operating characteristics, including temperature distribution and heat generation of a high-speed design 120.65-mm- (4.75-in.-) bore tapered-roller bearing at speeds to 20 000 rpm (2.4 million DN). The cone-rib tangential velocity at this highest speed was 158 m/sec (31 200 ft/min). Independent test variables were shaft speed, radial and thrust load, lubricant flow rate, and lubricant inlet temperature. The results were compared with the operating characteristics of modified standard test bearings reported in (1).

APPARATUS AND PROCEDURE

High-Speed Tapered-Roller Bearing Test Rig

The test rig used for this effort was that used in (1 and 5) except for an improved spindle design with closer spacing of the two test bearings. The 75 kilowatt (100-hp) motor was replaced with a 93 kilowatt (125-hp) motor with improved starting torque characteristics required for the higher speed testing.

One of two test bearings mounted on a spindle is shown in Fig. 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 pair of hydraulic actuators. Radial load is applied to the test bearings by a hydraulic actuator that exerts a force on a center housing containing two cylindrical roller load bearings mounted near the center of the spindle.

A flat-belt-pulley system is used to drive the spindle from an electric motor. The desired spindle speeds are chosen by exchanging drive pulleys on the motor. The test spindle is hollow and contains annular grooves to distribute lubricant to radial holes at both the large end and the small end of the test bearing (fig. 2) and for lubrication of the load bearing. A stationary lubrication tube delivers the desired lubricant flow to the annular grooves.

The lubrication system contains separately controlled circuits for cone-rib (large end) lubrication, small end lubrication, cup (outer ring) cooling, and load bearing lubrication. The lubricant flow through each circuit is metered with a variable flow control valve. A 10 micron filter and an oil-water heat exchanger are common to all the separate circuits.

Thermocouples are installed for temperature measurements of each test bearing cup outer surface, each cylindrical load bearing outer ring, and oil inlet and outlet of both test and load bearings. Temperatures of the cone bore and cone face of the test bearing (fig. 2) on the drive end of the test spindle were measured with thermocouples and transmitted with an FM telemetry system. A more detailed description of this test rig is given in (5).

Test Bearings

The tapered-roller test bearings had a bore of 120.65 mm (4.75 in.). The remainder of the bearing internal geometry and external dimensions were selected by computer optimization with due consideration for economical manufacturing utilizing existing tooling.

The computer analysis of (3) was used to select optimum cup and roller angles and roller size with a fixed bore of 120.65 mm (4.75 in.) for a load and speed condition of 26 700 N (6000 lb) radial and 53 400 N (12 000 lb) thrust and a range of shaft speeds from 12 500 to 20 000 rpm. The analysis of (3) requires complete definition of a tapered roller bearing and its immediate environment. The optimization process using this analysis consisted of evaluating over 30 selected combinations of cup and roller angles and roller diameters, each defining a unique bearing internal geometry. These combinations were all within the constraints fixed by good design practice for tapered-roller bearings (3). Criteria for comparison were bearing fatigue life, total bearing heat generation, and cone-rib contact stress and heat generation. Greater consideration was given to those combinations which have lower stress and heat generation at the cone-rib contact. Further details of the optimization variables are given in the appendix.

The geometry of the computer optimized design is given in table 1. This geometry was reviewed by a bearing manufacturer to consider use of existing tooling for more economical manufacturing. As a result, some compromise of the geometry was reached and checked with computer analysis to be satisfactory. This selected geometry is shown in table 1, along with the geometry of the standard bearing tested and reported in (1).

The rollers in the selected bearing were fully crowned with a crown radius of 25.4 m (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 CBS 1000M. The cage was one-piece machined AISI 4340 steel with silver plating and was guided by lands on the cone. The hardness, case depth, and surface finished specifications are shown in table 2.

The cone contained 40 oil holes, 1.016 mm (0.040 in.) in diameter at each end, drilled through from a manifold on the cone bore to the undercuts at each raceway end, as shown in Fig. 2. In addition, six oil holes were drilled at each end to lubricate the cage land riding surfaces.

The basic dynamic load ratings for this bearing are 70 700 N (15 900 lb) radial load and 51 600 N (11 600 lb) thrust load. (The thrust or radial load which gives 10 percent life of 90 million cone revolutions.) The Anti-friction Bearing Manufacturers Association (AFBMA) basic dynamic capacity is 255 000 N (57 400 lb). This bearing has approximately 10 percent less capacity than the standard bearing of (1), due to its optimization of performance at higher speeds.

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 was determined for a variety of speeds, loads and oil-in temperatures. Lubricant was supplied through holes at both the small end and the large end of the cone as shown in Fig. 2. No jet lubrication was used. 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 temperatures were also measured. Test spindle speeds ranged from 6000 to 20 000 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). Total lubricant flow rates varied from 0.0038 to 0.0151 m^3/min (1.0 to 4.0 gpm). The lubricant used was a 5-centistoke neopentypolyl (tetra) ester, MIL-L-23699.

Test bearing temperatures and oil-out temperatures, measured at these conditions, are shown in Figs. 3 to 6. Figure 3 shows very little effect of radial load on cone-face temperatures. The effects of thrust load were likewise small. This data is typical throughout the range of variables. That is, regardless of speed, oil-in temperature, or flow rates, load had little effect on bearing or oil-out temperature. Therefore, all the data shown in the remainder of this report are for one load condition, that is, 53 400 N (12 000 lb) thrust load and 26 700 N (6000 lb) radial load.

Figure 4 shows the general decrease in bearing and oil-out temperatures with increased lubricant flow rate to the large end and a constant 0.0038 m^3/min (1.0 gpm) to the small end. At the higher speeds, for example 18 500 rpm in Fig. 4(d), the effect of the lubricant flow rate is much greater than at the lower speeds. Bearing temperatures at 18 500 rpm are decreased as much as 28 K (50° F) as total flow rate is increased from 0.0076 m^3/min (2.0 gpm) to 0.0151 m^3/min (4.0 gpm). The temperature of the cup outer surface is greater than the cone bore or cone face throughout the range of flow rate and speed conditions.

Figure 5 shows the effect of increasing flow rate to the small end of the bearing from 0.0019 m^3/min (0.5 gpm) to 0.0057 m^3/min (1.5 gpm) with a constant 0.0076 m^3/min (2.0 gpm) at the large end. Greater effects of small-end flow rate occurred at the higher speeds. Comparison of Figs. 4 and 5, shows that small-end flow rate has a greater effect on cup temperature than on cone-face temperature, and the opposite is true for large-end flow rate. This effect should be expected since the lubricant introduced at the small end is pumped through the bearing to the large end and in doing so is thrown centrifugally to the cup raceway surface thus cooling the cup. Lubricant introduced through the large end cools the cone rib and quickly exits the bearing with less cooling effect on the cup. The large effect of small-end flow rate on cone-bore temperature at the higher speeds (figs. 5(d) and (e)) indicates its effectiveness in cooling the cone.

It is apparent that, for speeds in the range of 15 000 to 20 000 rpm, flow rates of 0.0076 m^3/min (2.0 gpm) at the large end and 0.0038 m^3/min (1.0 gpm) at the small end are near optimum for this design and size of bearing. Smaller flow rates give significantly higher temperatures, and large flow rates only increase bearing heat generation as will be shown later.

Effect of Shaft Speed

The effect of shaft speed on bearing and oil-out temperatures is shown in Fig. 6 for a total flow rate of 0.0114 m^3/min (3.0 gpm). The greater effect of speed on cup outer surface temperature than on cone-bore temperature is readily apparent. The oil-out temperature increases with speed at about the same rate as the cup outer-surface temperature. The cage is guided on lands on the cone, and greater heat generation at those guiding surfaces at higher speeds is expected, along with higher cone temperature. Apparently the lubricant flow through the holes in the cone (both large and small ends) very adequately cools the cone, and by the time it reaches the cup raceway, it has been heated appreciably and has less cooling capacity. As will be shown later, cup cooling can be used to decrease the high cup temperatures at the higher speeds.

The test bearing cage speed was measured at all shaft speed conditions. The ratio of cage speed to shaft speed did not vary more than 1 percent from a nominal value of 0.445 throughout the range of speed, load and lubricant flow conditions.

Stable operating temperatures were achieved at all speed, load and lubricant flow rate conditions. Some of the lower lubricant flow rate points were not attempted at 20 000 rpm due to anticipated excessive temperatures. Observations of the bearings after tests at 20 000 rpm showed no signs of surface distress on the roller-raceway or roller/cone-rib contacting surfaces. Likewise, cage contacting surfaces showed no abnormal wear or distress.

Effect of Oil-In Temperature

The effect of oil-in temperature on bearing and oil-out temperatures is shown in Fig. 7. Temperatures at an oil-in temperature of 350 K (170° F) and 18 500 rpm shaft speed are shown compared with data from Fig. 4(b) at 364 K (195° F) oil-in temperatures. At the lower oil-in temperature, bearing and oil-out temperatures generally were 6 to 14 K (10° to 25° F) less than at the higher oil-in temperature. The cone-bore temperature was decreased the full 14 K (25° F) of the decrease in oil-in temperature. Least effected was the cup outer-surface temperature.

Similar data was also obtained at a shaft speed of 12 500 rpm where the temperatures were generally 6 to 10 K (10° to 18° F) lower for the lower oil-in temperatures. Least effected was the cone-bore temperature. Generally, for both shaft speeds, the bearing and oil-out temperatures were effected slightly more by oil-in temperature at the higher flow rates.

Effect of Cup Cooling

As was shown in Fig. 6, the highest bearing temperatures measured were at the cup-outer surface, and the difference from cone temperatures was increasingly greater at higher speeds. 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 8 shows the effects of cup cooling flows up to 0.0028 m^3/min (0.75 gpm) at 18 500 rpm and 350 K (170° F) oil-in temperature. As expected, greatest effects were at the cup-outer surface where temperatures were decreased 11 K (20° F) with a cup cooling flow of 0.0019 m^3/min (0.5 gpm). Higher cooling flows had little or no effect.

Limited data was also obtained at 18 500 rpm and an oil-in temperature of 364 K (195° F). As shown in table 3, a cup cooling flow rate of 0.0038 m^3/min (1.0 gpm) was sufficient to lower the cup outer surface 14 K (25° F) or to a level equal to the cone-bore temperature.

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 Fig. 9 as a function of total flow rate (For convenience, heat values were converted from J/min to kW.). The heat transferred to the lubricant increases with increased lubricant flow rate. The effect is greater at higher speeds.

Figure 10 shows an even greater effect of shaft speed on heat transferred to the lubricant. The effect of shaft speed can be closely approximated by the relation $Q_T \propto N^{1.35}$ where N is the shaft speed in r.m.

These increases are expected due to increased lubricant drag or churning. These heat quantities are only a portion of the heat generated in the test bearing 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.

Comparison with Standard Design Bearing

In (1 and 5) data from tests with a modified standard catalog series tapered-roller bearing of the same bore (120.65 mm (4.75 in.)) was presented at speeds up to 15 000 rpm. Loads and total flow rates were over the same range as those reported herein for the high-speed design bearing. Major differences in the two bearing designs were smaller cup angle, smaller pitch and outside diameters, and fewer rollers in the high speed design (table 1). The cage of the high speed design bearing was made to be guided by lands on the cone whereas the standard bearing had a stamped steel cage made to be guided by the rollers. The latter design cage has lesser strength for high-speed operation.

Although lubricant to the standard bearing was directed through holes in the large end of the cone to the cone-rib surface, lubricant at the small end was through a pair of jets directed at the small end of the rollers. Comparisons herein were made with equal flow rates for both bearings at both the large end and the small end. At the small end, a constant of 0.0038 m^3/min (1.0 gpm) was used, and that was fed through jets for the standard design and through the cone small end for the high-speed design.

Materials with higher temperature capabilities were used for the high-speed design bearings. Cups, cones, and rollers were made of CBS-1000M, and the cage was AISI 4340. The standard design bearing had AISI 4320 cups, cones, and rollers and AISI 1010 cages.

In Fig. 11, cone-face temperatures of the high-speed design from Fig. 4 are compared with data from (5). The symbols are used to identify the curves at each flow rate. At 15 000 rpm, the high-speed design bearing operates 8 to 11 K (15° to 20° F) cooler than the standard design. The improvement at 6000 rpm is much less. The cup outer-surface temperature is also lower for the high-speed design bearing as shown in Fig. 12. Here, the improvement is slightly less than that at the cone face.

Also shown in Fig. 9 are data for the standard design bearing showing heat transferred to the lubricant at 6000 and 15 000 rpm. The high-speed design bearing has lower heat generation, as represented by heat transferred to the lubricant, than the standard design bearing at both speeds. As shown in Fig. 10, the improvement with the high-speed design is approximately 16 percent throughout the range of 6000 to 15 000 rpm.

SUMMARY

The performance of 120.65-mm (4.75-in.) bore computer-optimized tapered-roller bearings was investigated at shaft speeds up to 20 000 rpm (cone-rib tangential velocities up to 158 m/sec (31 200 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 through the cone at both the large end and the small end of the roller. Test conditions included shaft speeds from 6000 to 20 000 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 0.0038 to 0.0151 m^3/min (1.0 to 4.0 gpm), and lubricant inlet temperatures of 350 and 364 K (170° and 195° F). The following results were obtained:

1. The computer-optimized high-speed design tapered-roller bearing operated successfully at shaft speeds up to 20 000 rpm under heavy thrust and radial loads. Stable temperatures were reached and no surface distress was observed.

2. Bearing temperatures and heat generation with the high speed design bearing were significantly less than those of a modified standard bearing tested previously.

3. Bearing temperatures and heat generation increased as expected with increased shaft speed. The highest temperatures measured were at the cup outer surface.

4. With increased lubricant flow-rates (either large end or small end), bearing temperatures decreased and heat generation increased. A total flow rate as low as 0.0114 m^3/min (3.0 gpm) gave stable operating temperatures at 20 000 rpm without excessive bearing heat generation.

APPENDIX

The computer optimization utilized the program described in (3). The bearing bore diameter was fixed at 120.65 mm (4.75 in.). Over 30 cases were evaluated using cup half angles, α_o of 12°, 15° and 18°, roller half angles, υ of 1.0°, 1.5° and 2.0°, and roller large end diameters ranging from 13.3 to 27.9 mm (0.525 to 1.100 in.).

Other dimensions as fixed by good design practice (3) were:

A = 5.08 mm (0.20 in.)

B = 5.08 mm (0.20 in.)

C = 0.15 W

E = 0.635 mm (0.025 in.)

$R_s = 0.8\ L_A$

where R_s = roller spherical end radius

L_A = apex length

$T_w = 0.2\ D_m$

where T_W = cage web thickness

D_m = roller mean diameter

h = 2.54 mm (0.10 in.)

where h = roller end-flange contact height

$1.0 < L/D < 2.6$

where L = total roller length

D = roller large end diameter

These dimensions are defined in Fig. 13.

References

1. Parker, R. J. and Signer, H. R., "Lubrication of High-Speed, Large Bore Tapered Roller Bearings," Journal of Lubrication Technology, Vol. 100, No. 1, Jan. 1978, pp. 31-38.

2. Orvos, P. S. and Pressler, G. J., "Tapered-Roller Bearing Development for Aircraft Turbine Engines," AFAPL-TR-79-2007, Air Force Aero Propulsion Lab., Wright-Patterson AFB, Ohio, Mar. 1979.

3. Crecelius, W. J. and Milke, D. R., "Dynamic and Thermal Analysis of High Speed Tapered Roller Bearings Under Combined Loading," SKF-AL73P010, SKF Industries, Inc., King of Prussia, Penn.; NASA CR-121207, 1973.

4. Cornish, R. F., Orvos, P. S., and Gupta, S. R., "Development of High Speed Tapered Roller Bearing," IR-1, Timken RollerBearing Co., Canton, Ohio; AFAPL-TR-73-85, Air Force Aero Propulsion Lab., Wright-Patterson AFB, Ohio, Oct. 1973.

5. Parker, R. J. and Signer, H. R., "Performance of Large-Bore Tapered-Roller Bearings Under Combined Radial and Thrust Load at Shaft Speeds to 15,000 rpmn." NASA TN D-8404, 1977.

TABLE 1. - TEST BEARING GEOMETRY

Dimension	Computer optimized bearing	Selected bearing	Standard bearing[a]
Cup half angle	15°	15°53'	17°
Roller half angle	1°30'	1°35'	1°35'
Roller large end diameter, mm (in.)	18.42 (0.725)	18.29 (0.720)	18.29 (0.720)
No. of rollers	22	23	25
Total roller length, mm (in.)	37.31 (1.469)	34.18 (1.3456)	34.17 (1.3452)
Pitch diameter, mm (in.)	155.5 (6.123)	155.1 (6.105)	166.8 (6.569)
Bearing outside diameter, mm (in.)	192.3 (7.571)	190.5 (7.500)	206.4 (8.125)

[a]Bearing used in tests of (1 and 5).

TABLE 2. - TEST BEARING SPECIFICATIONS

Cup, cone, and roller material	CBS 1000M
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	0.061 to 0.185 (0.024 to 0.073)
Roller	0.091 to 0.201 (0.036 to 0.079)
Surface finish,[a] μm (μin.), rms:	
Cone raceway	0.10(4)
Cup raceway	0.10(4)
Cone-rib	0.41(16)
Roller taper	0.25(10)
Roller spherical	0.08(3)

[a]Measured values.

TABLE 3. - EFFECT OF CUP COOLING ON BEARING AND OIL-OUT TEMPERATURES

[Shaft speed, 18 500 rpm; thrust load, 53 400 N (12 000 lb); radial load, 26 700 N (6000 lb); oil-in temperature, 364 K (195° F); large-end flow, 0.0076 m^3/min (2.0 gpm); small-end flow, 0.0038 m^3/min (1.0 gpm).]

Cup cooling flow rate, m^3/min (gpm)	Temperature, K (°F)				
	Cone face	Cone bore	Cup outer surface	Oil-out	Cup cooling oil-out
0	389(240)	423(302)	438(329)	426(307)	--------
0.0038(1.0)	391(245)	424(303)	424(304)	426(308)	386(235)

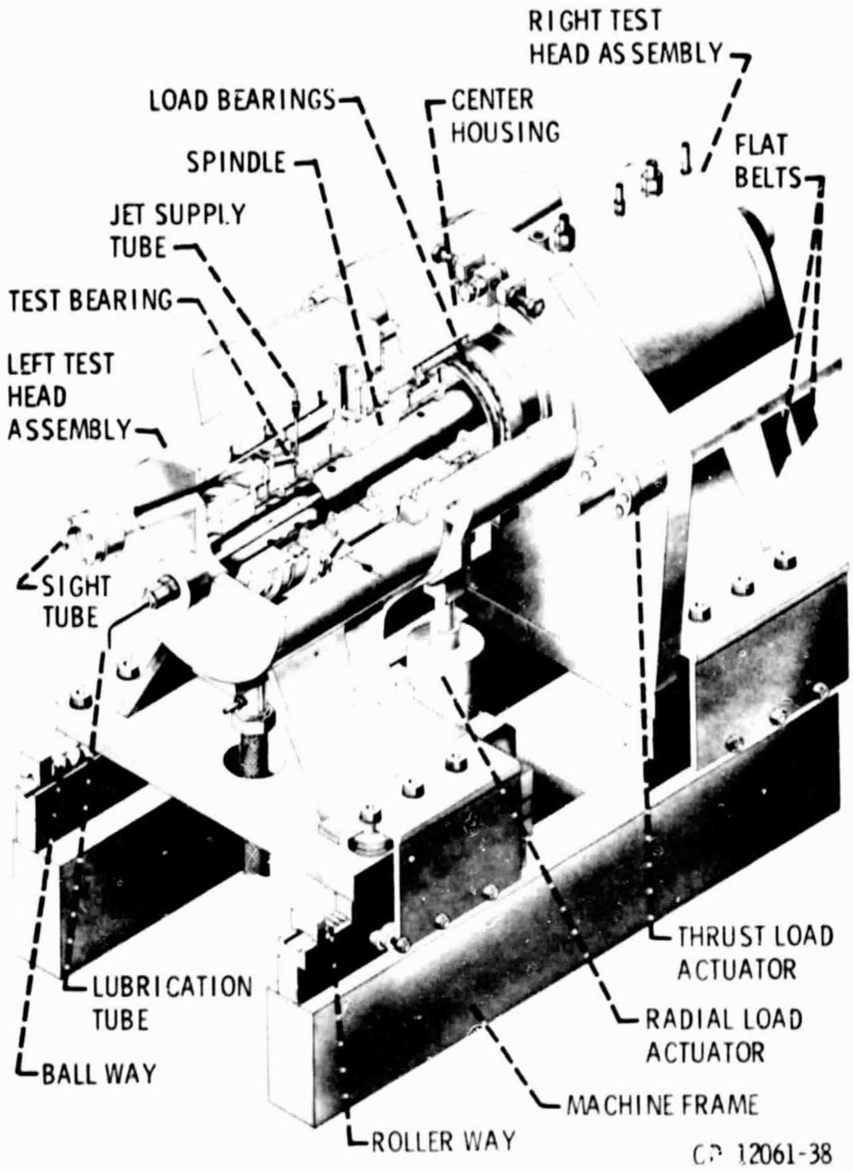


Figure 1. - Pictorial view of high-speed tapered roller bearing test rig.

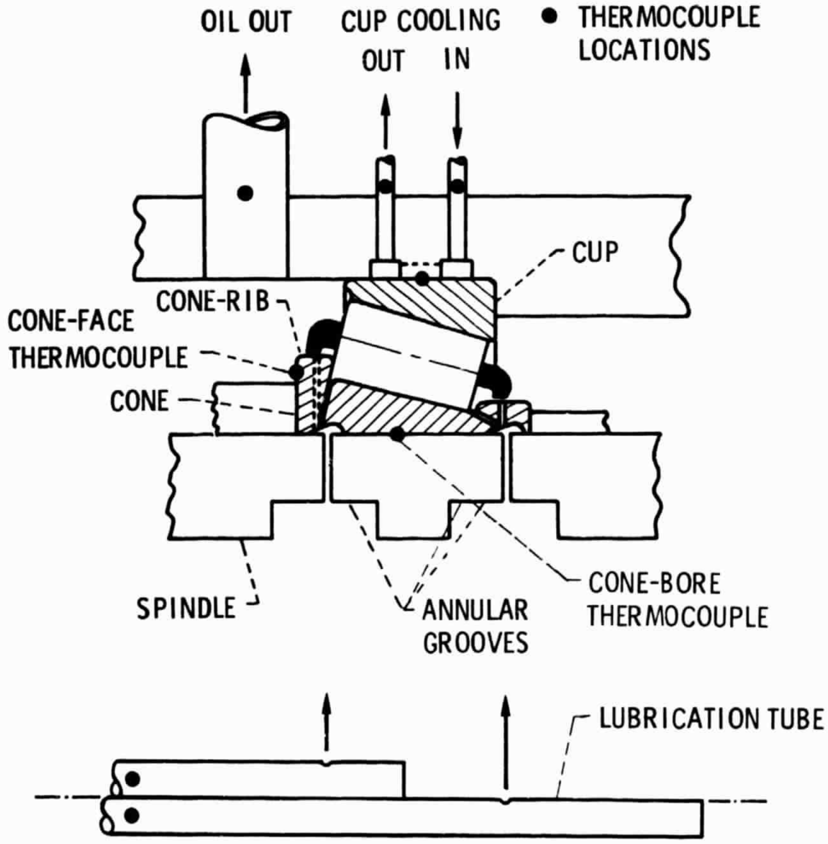


Figure 2. - Test bearing lubrication, cooling, and thermocouple locations.

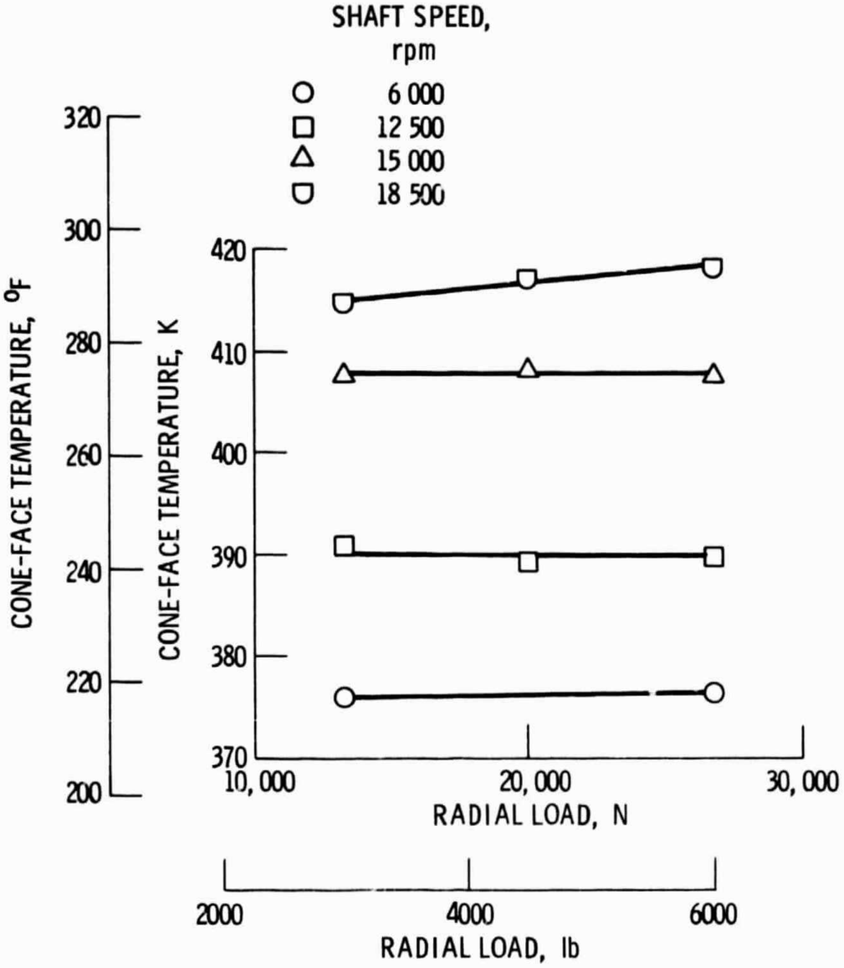


Figure 3. - Effect of radial load on cone-face temperature. Thrust load, 53400N (12 000 lb); oil-in temperature, 364 K (195° F); total oil flow, 0.0076 m^3/min (2.0 gpm).

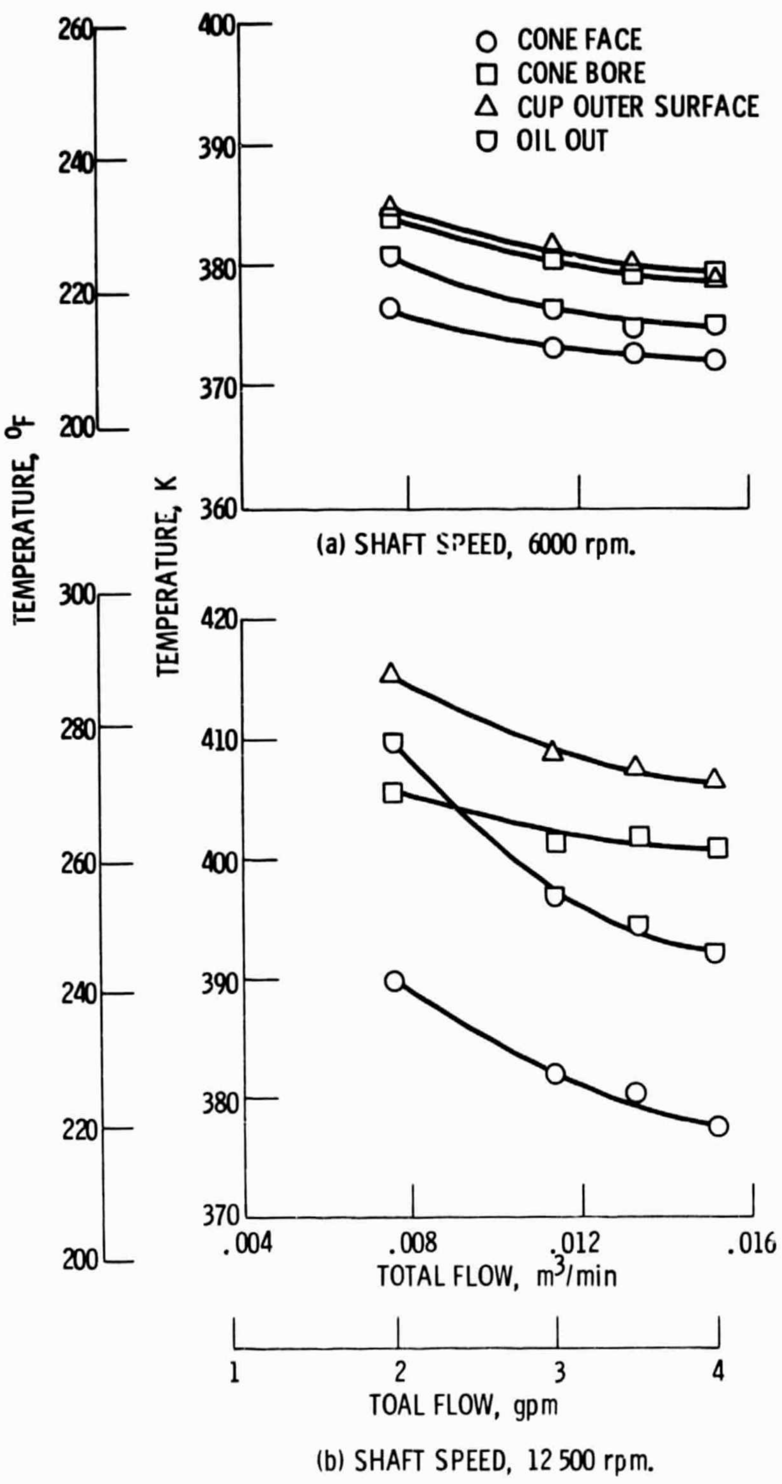
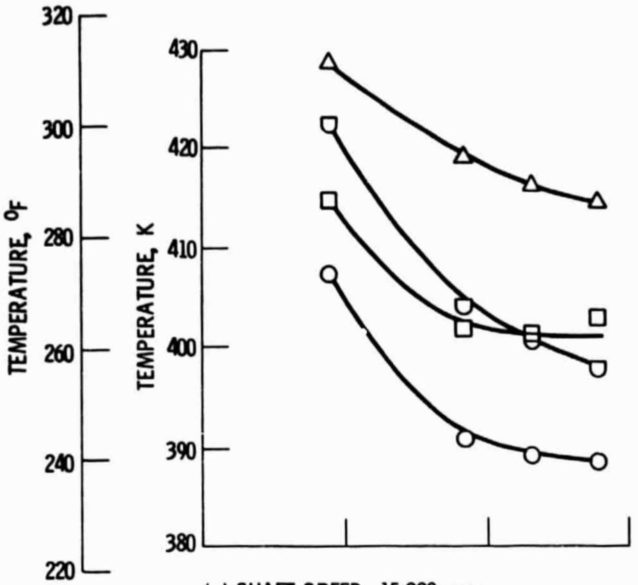
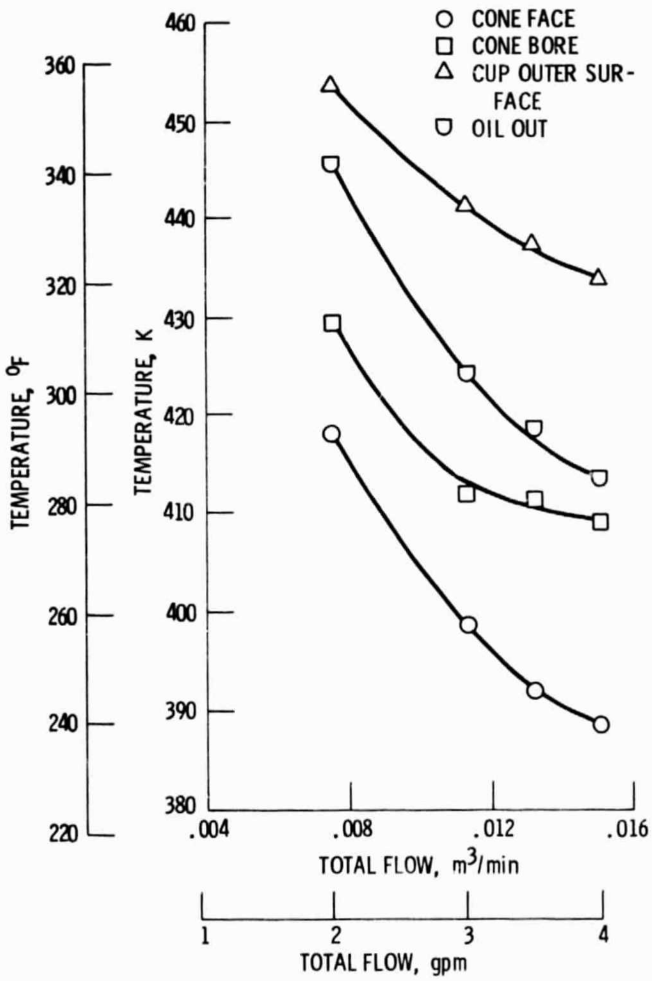


Figure 4. - Temperature as a function of total flow rate with variable large end flow. Small end flow rate, 0.0038 m^3/min (1.0 gpm); thrust load, 53 400 N (12 000 lb); radial load, 26 700 N (6000 lb); oil-in temperature, 364 K (195^0 F).

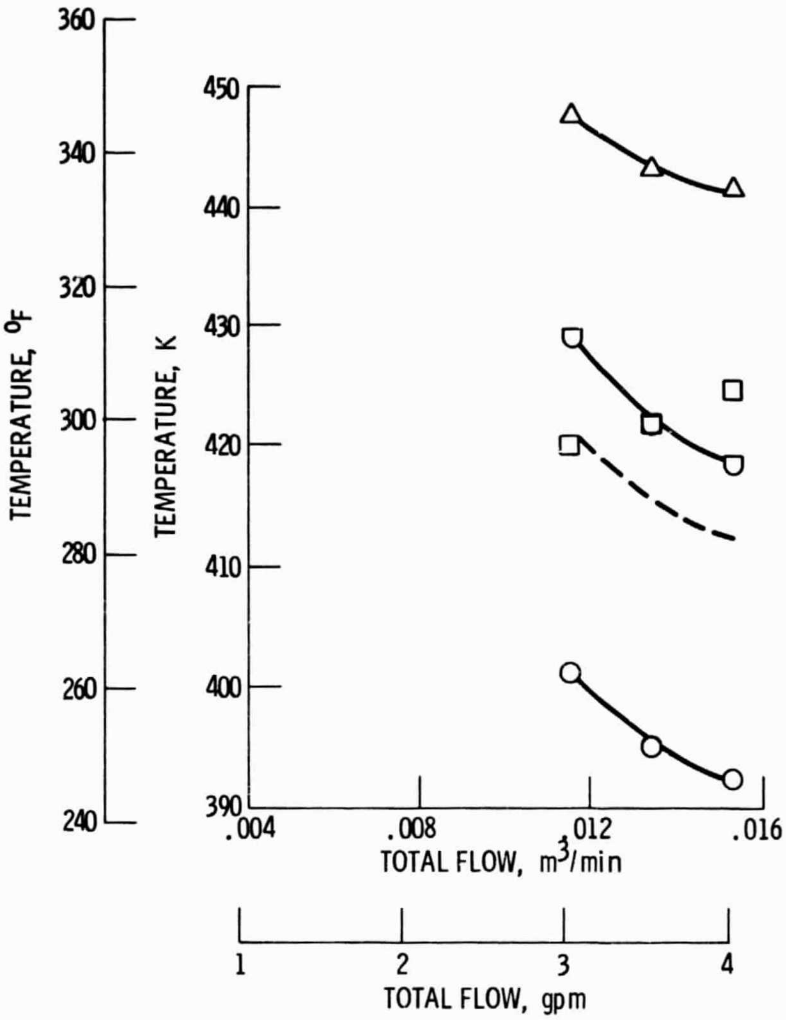


(c) SHAFT SPEED, 15 000 rpm.



(d) SHAFT SPEED, 18 500 rpm.

Figure 4. - Continued.



(e) SHAFT SPEED, 20 000 rpm.

Figure 4. - Concluded.

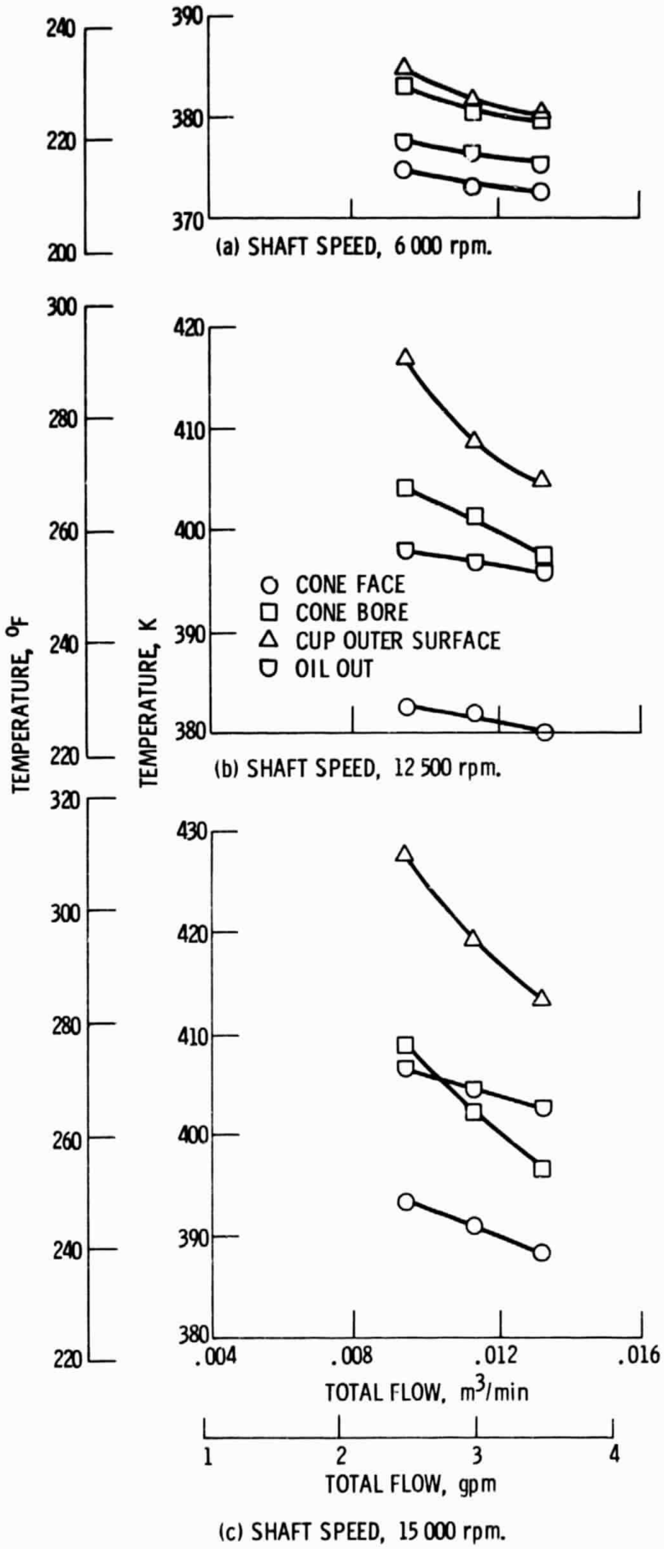
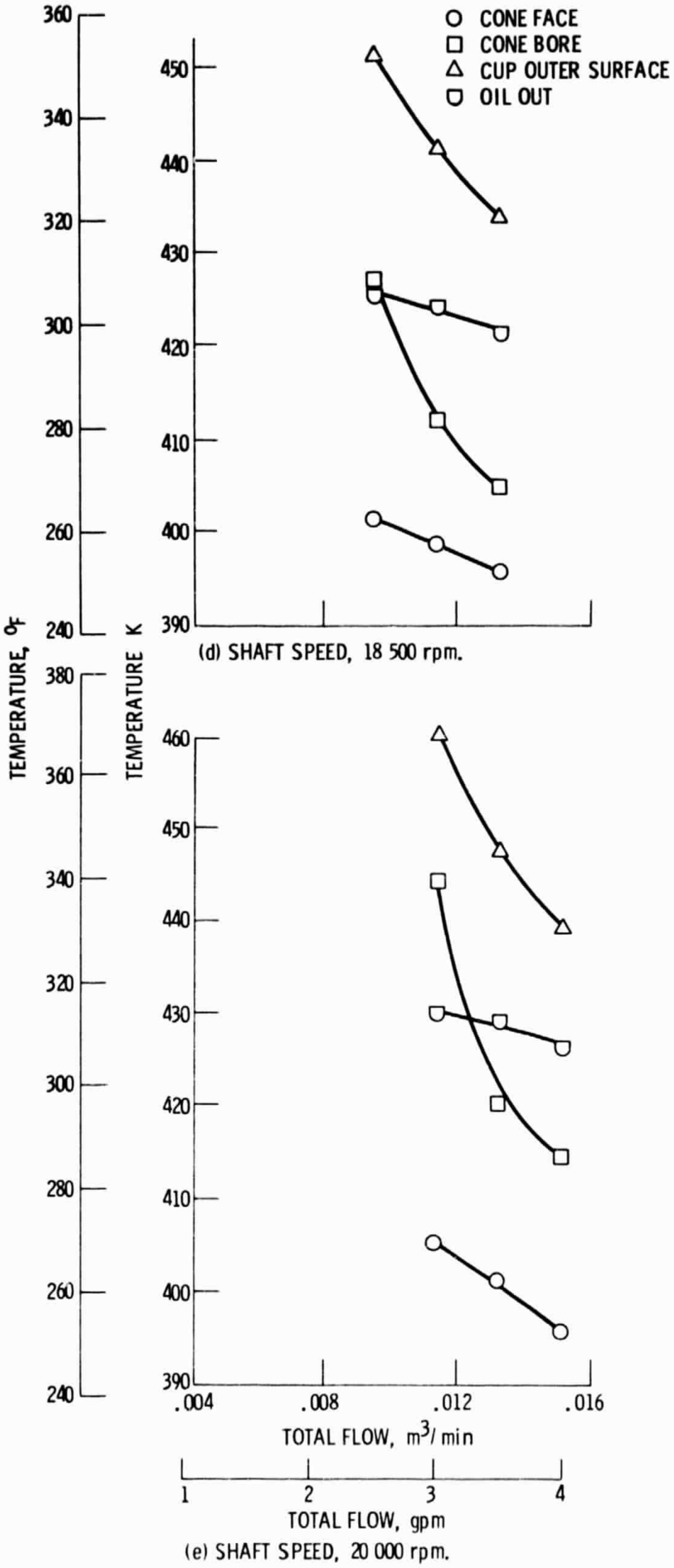


Figure 5. - Temperature as a function of total flow rate with variable small end flow. Large end flow rate, 0.0076 m^3/min (2.0 gpm); thrust load, 53400N (12 000 lb); radial load, 26700N (6000 lb); oil-in temperature, 364 K (195° F).



(d) SHAFT SPEED, 18 500 rpm.

(e) SHAFT SPEED, 20 000 rpm.

Figure 5. - Concluded.

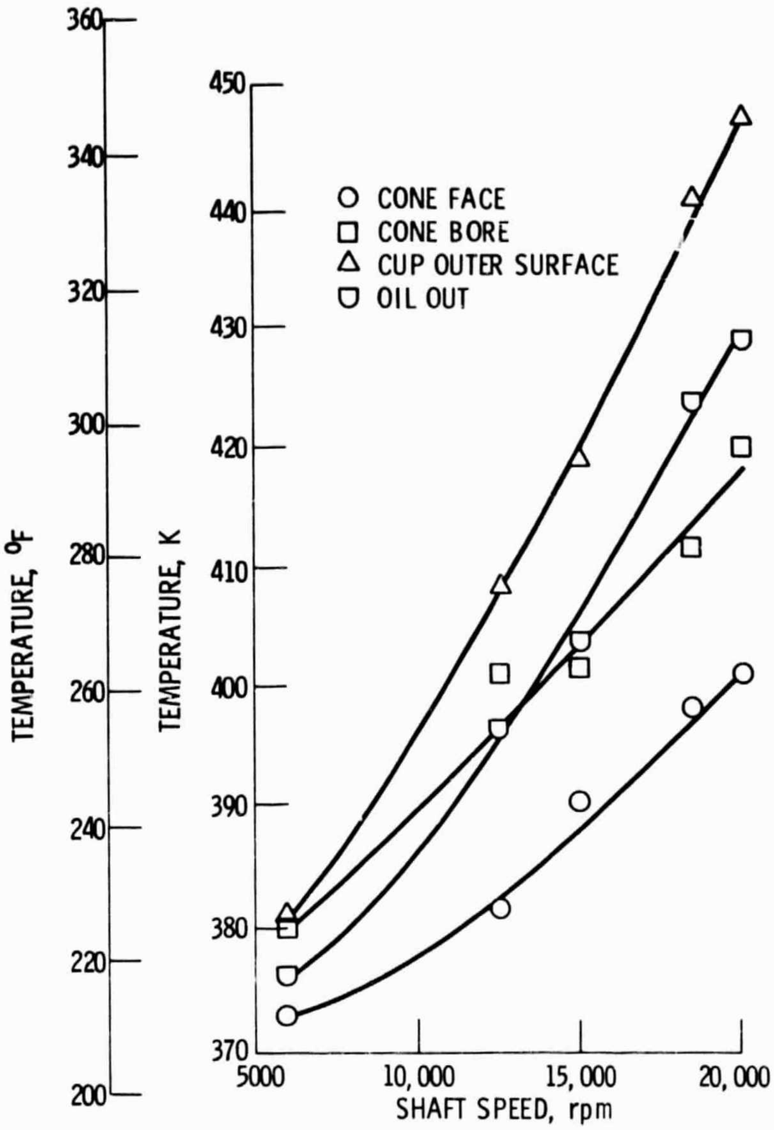


Figure 6. - Effect of shaft speed on bearing and oil-out temperatures. Thrust load, 53 400 N (12,000 lb); radial load, 26 700 N (6000 lb); oil-in temperature, 364 K (195° F); large end flow rate, 0.0076 m^3/min (2.0 gpm); small end flow rate, 0.0038 m^3/min (1.0 gpm).

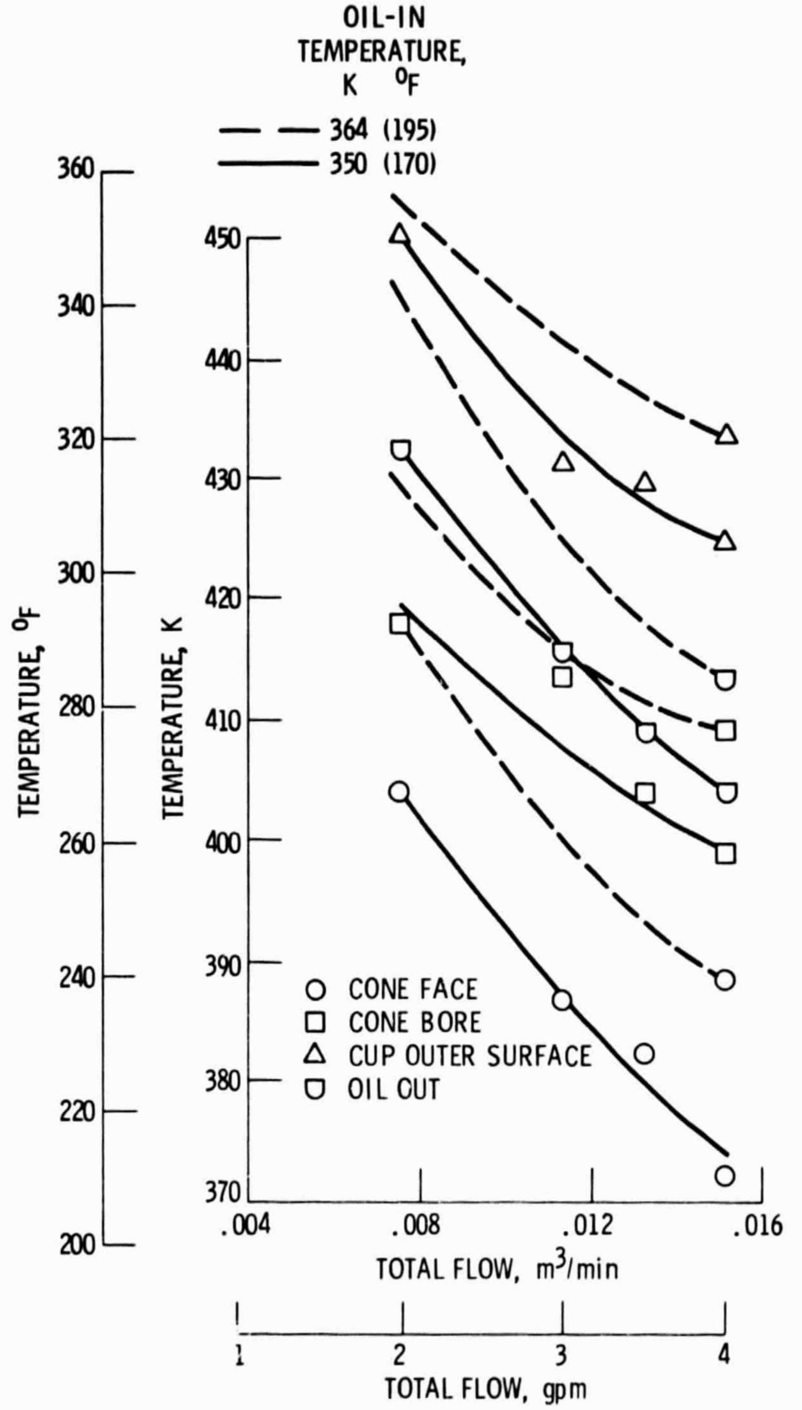


Figure 7. - Effect of oil-in temperature on bearing and oil-out temperatures. Thrust load 53 400 N (12 000 lb); radial load, 26 700 N (6000 lb); shaft speed, 18 500 rpm; small end flow rate, 0.0038 m^3/min (1.0 gpm).

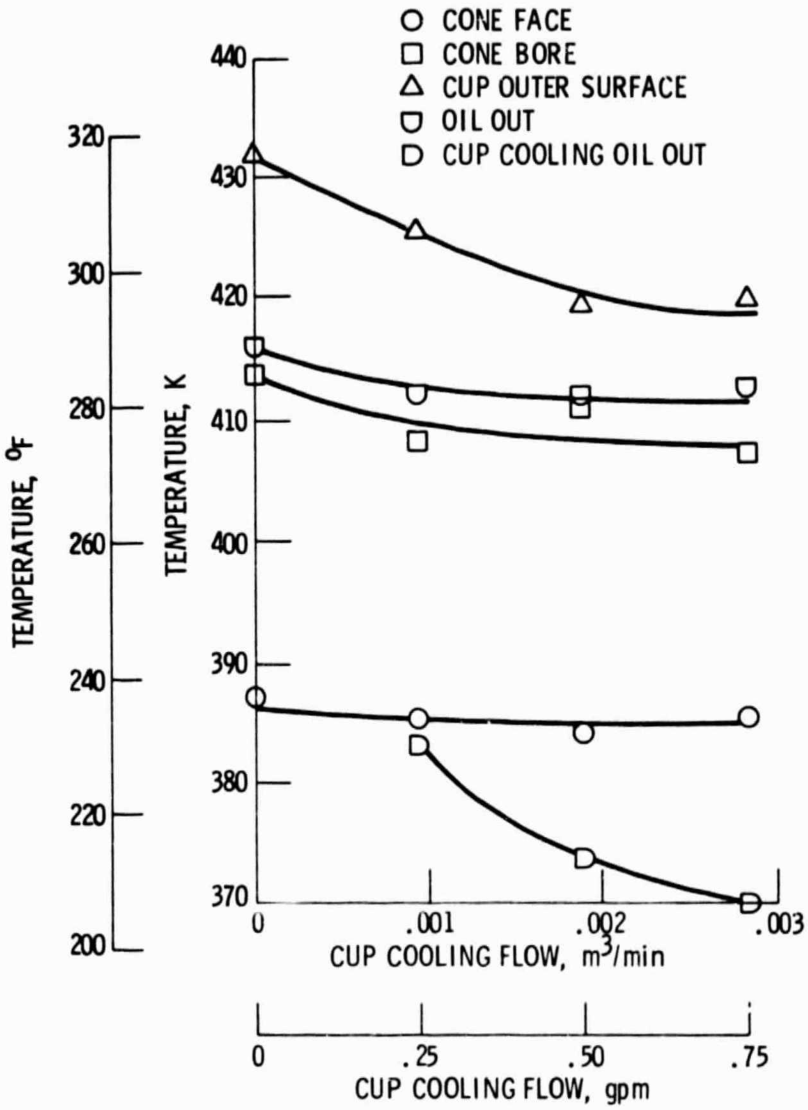


Figure 8. - Effect of cup cooling flow rate on bearing and oil-out temperatures. Thrust load, 53 400 N (12 000 lb); radial load, 26 700 N (6000 lb); shaft speed, 18 500 rpm; large end flow rate, 0.0076 m^3/min (2.0 gpm); small end flow rate, 0.0038 m^3/min (1.0 gpm); oil-in temperature, 350 K (170° F).

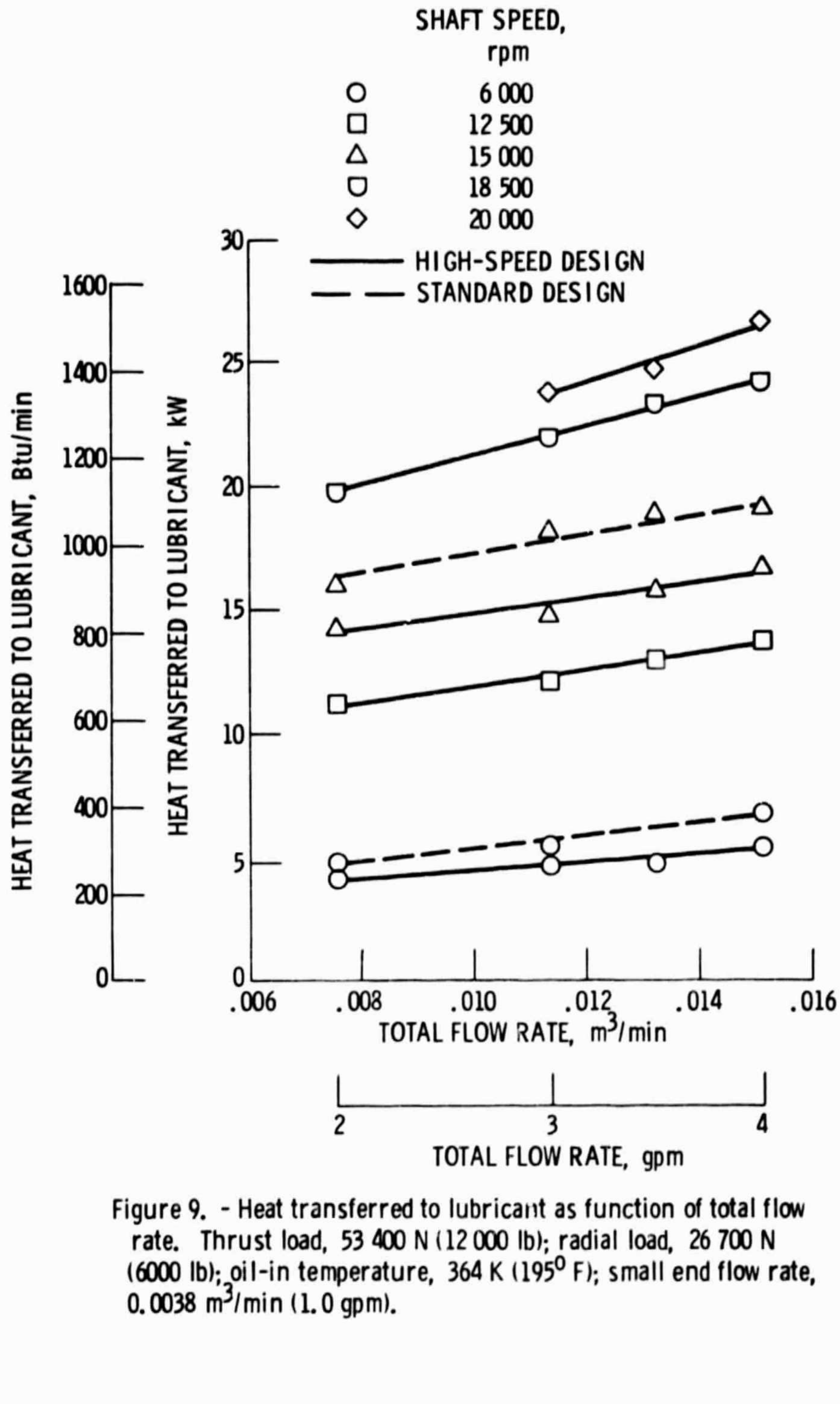


Figure 9. - Heat transferred to lubricant as function of total flow rate. Thrust load, 53 400 N (12 000 lb); radial load, 26 700 N (6000 lb); oil-in temperature, 364 K (195° F); small end flow rate, 0.0038 m^3/min (1.0 gpm).

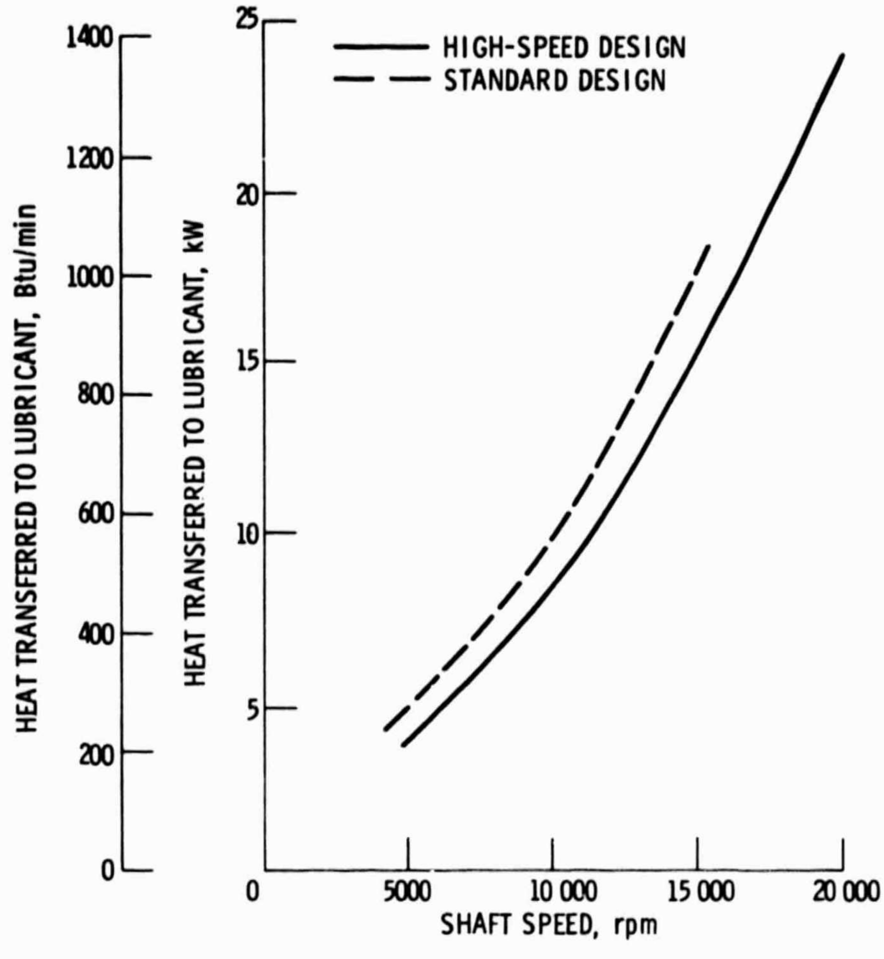


Figure 10. - Heat transferred to lubricant as function of shaft speed. Thrust load, 53 400 N (12 000 lb); radial load, 26 700 N (6000 lb); oil-in temperature, 364 K (195° F); large end flow rate, 0.0076 m^3/min (2.0 gpm); small end flow rate, 0.0038 m^3/min (1.0 gpm).

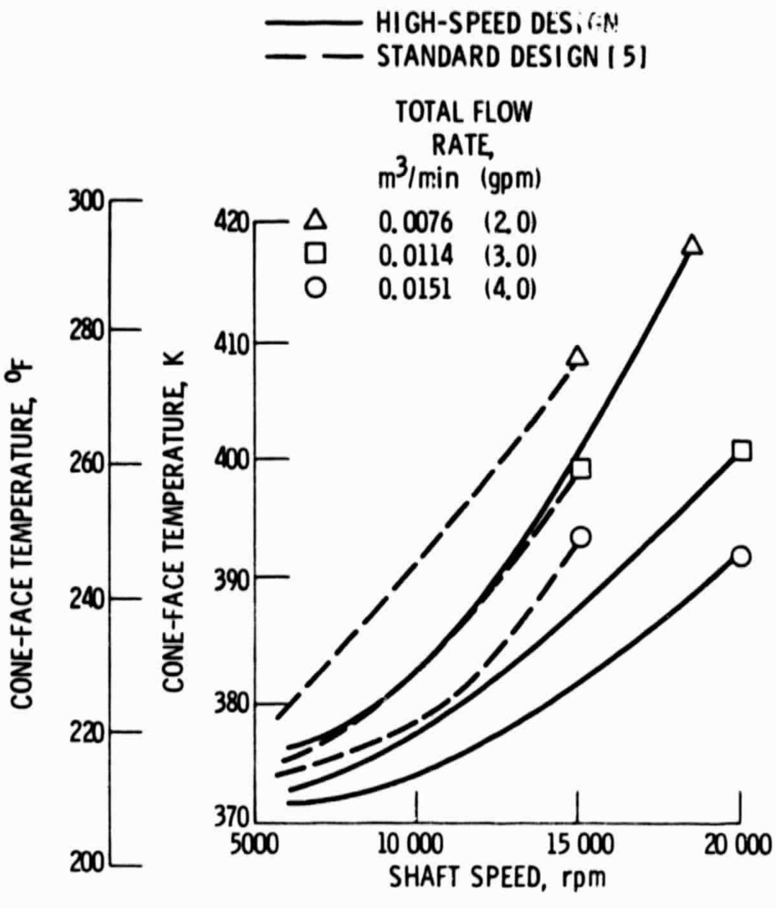


Figure 11. - Cone face temperature with high-speed design compared to that with standard design. Thrust load, 53 400 N (12 000 lb); radial load, 26 700 N (6000 lb); oil-in temperature, 364 K (195° F); small end flow rate, 0.0038 m^3/min (1.0 gpm).

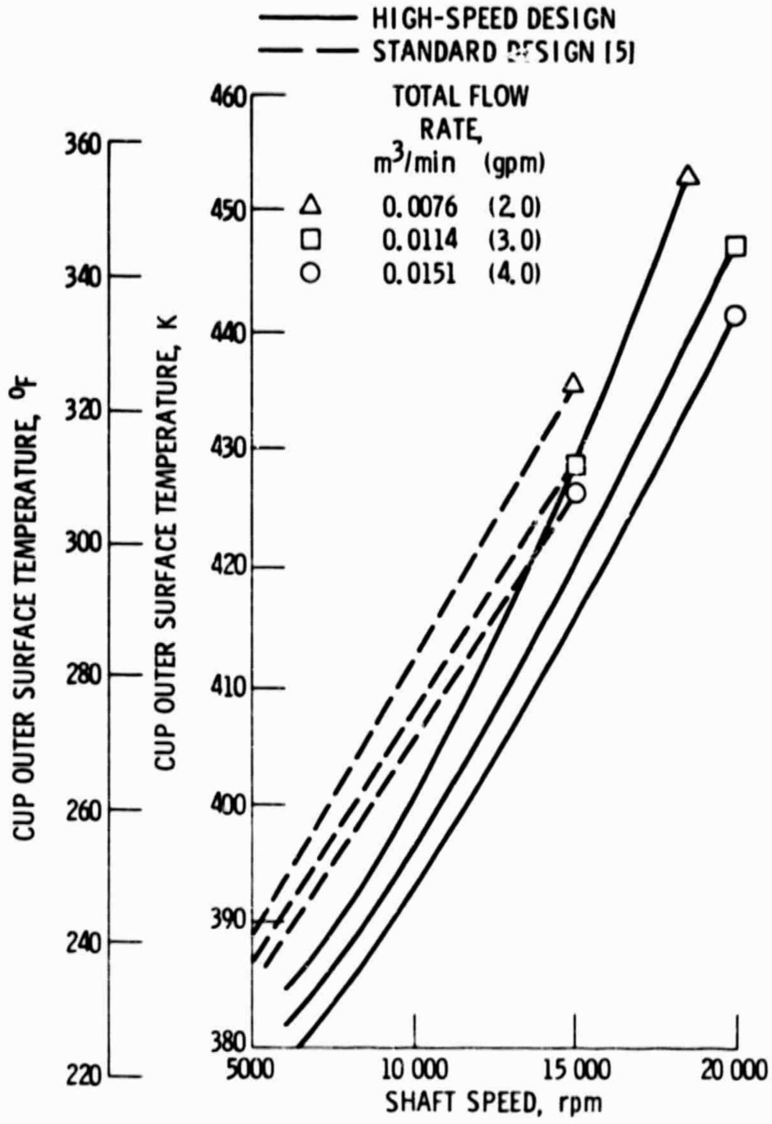


Figure 12. - Cup outer surface temperature with high-speed design compared to that with standard design. Thrust load 53 400 N (12 000 lb); radial load, 26 700 N (6000 lb); oil-in temperature, 364 K (195° F); small end flow rate, 0.0038 m^3/min (1.0 gpm).

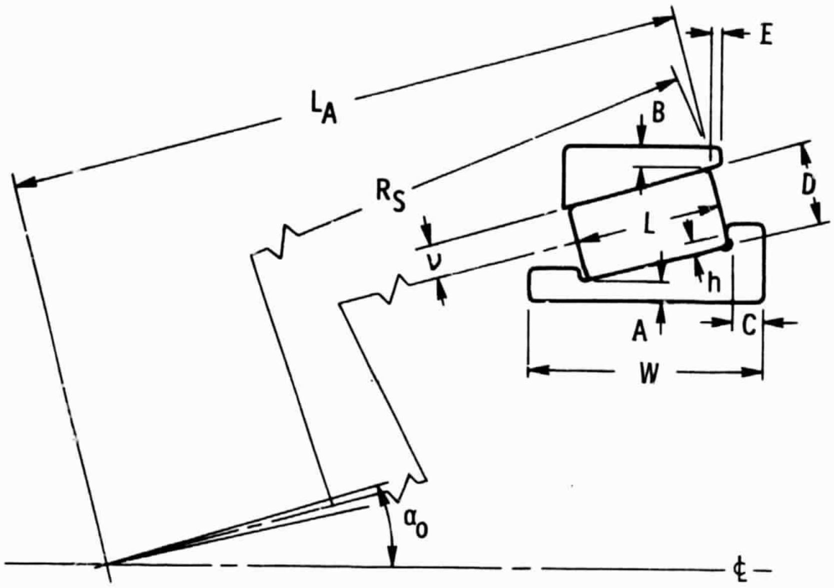


Figure 13. - Tapered-roller bearing geometry.

1. Report No. NASA TM-81414	2. Government Accession No.	3. Recipient's Catalog No.
4. Title and Subtitle PERFORMANCE OF COMPUTER-OPTIMIZED TAPERED-ROLLER BEARINGS TO 2.4 MILLION DN		5. Report Date
		6. Performing Organization Code
7. Author(s) R. J. Parker, S. I. Pinel, and H. R. Signer		8. Performing Organization Report No. E-332
		10. Work Unit No.
9. Performing Organization Name and Address National Aeronautics and Space Administration Lewis Research Center Cleveland, Ohio 44135		11. Contract or Grant No.
		13. Type of Report and Period Covered Technical Memorandum
12. Sponsoring Agency Name and Address National Aeronautics and Space Administration Washington, D.C. 20546		14. Sponsoring Agency Code

15. Supplementary Notes

R. J. Parker, NASA Lewis Research Center, S. I. Pinel and H. R. Signer, Industrial Tectonics, Inc., Compton, California.

16. Abstract

The performance of 120.65-mm- (4.75-in.-) bore high-speed design tapered roller bearings was investigated at shaft speeds to 20 000 rpm (2.4 million DN) under combined thrust and radial load. The test bearing design was computer optimized for high-speed operation. Temperature distribution and bearing heat generation were determined as a function of shaft speed, radial and thrust loads, lubricant flow rates and lubricant inlet temperature. The high-speed design tapered roller bearing operated successfully at shaft speeds up to 20 000 rpm under heavy thrust and radial loads. Bearing temperatures and heat generation with the high-speed design bearing were significantly less than those of a modified standard bearing tested previously. Cup cooling was effective in decreasing the high cup temperatures to levels equal to the cone temperature.

17. Key Words (Suggested by Author(s))	18. Distribution Statement
Rolling element bearing; Roller bearings; Tapered-roller bearing; High-speed bearings	Unclassified - unlimited STAR Category 37

19. Security Classif. (of this report)	20. Security Classif. (of this page)	21. No. of Pages	22. Price*
Unclassified	Unclassified		

* For sale by the National Technical Information Service, Springfield, Virginia 22161