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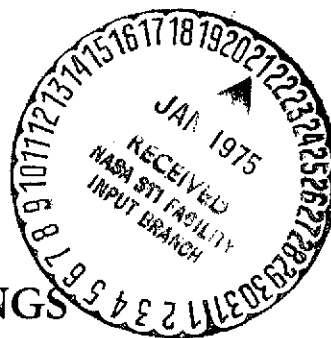
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16. Abstract A study was undertaken to determine the effects of speed and load on the operation of 120-mm bore angular-contact ball bearings at speeds to 25 000 rpm (3×10^6 DN) and thrust loads to 22 240 newtons (5000 lb). Bearing temperature and power consumption increased with increases in load and/or speed. The effect of load on temperature and power consumption was small relative to the speed effect. Actual measurements of bearing operating contact angle were in excellent agreement with theoretical predictions. Skidding occurred in the bearing in various amounts, generally increasing with speed at a given load. The highest amount of skidding, 6 percent, occurred at the highest speed, 25 000 rpm. No visible damage to the bearing surfaces occurred due to the skidding.			
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EFFECT OF SPEED AND LOAD ON ULTRA-HIGH-SPEED BALL BEARINGS

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SUMMARY

Parametric tests were conducted in a high-speed, high-temperature bearing tester with optimally designed 20° and 24° contact angle 120-millimeter-bore angular-contact ball bearings. The bearings were manufactured from double vacuum-melted AISI M-50 steels having a room-temperature Rockwell C hardness of 63. Test parameters were thrust loads of 6672, 13 350, and 22 240 newtons (1500, 3000, and 5000 lb) and nominal speeds of 12 000, 16 000, 20 000, and 25 000 rpm. Test conditions included an oil-inlet temperature of 428 K (310° F), a lubricant flow rate through the inner race of 1.2×10^{-3} cubic meter per minute (0.313 g/min), an inner-race cooling flow rate of 3.6×10^{-3} cubic meter per minute (0.94 g/min), and an outer-race cooling flow rate of 1.9×10^{-3} cubic meter per minute (0.5 g/min). The lubricant was a neopentylpolyol (tetra) ester.

Bearing inner- and outer-race temperatures increased with increases in load and/or speed. However, the effect of load on temperature was small relative to the speed effect. The difference between the inner- and outer-race temperature was negligible.

Bearing power consumption increased with speed. The increase in power consumption with thrust load was considered negligible relative to the effect of speed for most practical engineering design applications.

Actual measurements of bearing operational contact angle were in excellent agreement with theoretical predictions with the exception of the inner-race contact angle at the maximum speed and load condition.

A comparison of measured cage speed to theoretical predictions indicates that, with the exception of the 12 000 rpm, 20° contact angle data, the bearings operated with some skidding. The highest amounts of skidding (6 percent of the theoretical cage speed) occurred at the 25 000 rpm (3×10^6 DN) test conditions. (DN is defined as the speed of the bearing in rpm multiplied by the bearing bore in millimeters.) Skidding decreased with increasing thrust load and decreasing speed. No visible damage to the bearing surfaces was observed.

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INTRODUCTION

Advances in bearing technology allow angular-contact ball bearings to operate at speeds to 3×10^6 DN (ref. 1). (DN is defined as the speed of the bearing in rpm multiplied by the bearing bore in millimeters.) The speeds are those anticipated in advanced airbreathing engines well into the 1980's. To effect these high speeds in a bearing, bearing operating temperature, differences in temperature between the inner and outer races, and bearing power consumption can be tuned to the desirable operating requirement by varying four lubricant and cooling parameters. These parameters are outer-race cooling, inner-race cooling, lubricant flow to the inner race, and oil inlet temperature (ref. 1). The cooling rate at the inner race can be controlled by varying the lubricant flow to the bearing through a number of radial holes in the center and shoulders of the bearing's split inner race (refs. 1 and 2). Cooling of the outer race is achieved by flowing the lubricant into grooves around the outer race (ref. 1).

In order to adequately design for proper bearing thermal management, the effect of bearing operating parameters such as load and speed on operating temperature and power loss must be assessed.

The effect of operating speed and load on bearing operation and life has been analytically considered in references 3 to 5 without considering the effect of the lubricant. These analyses were expanded in references 6 and 7 to include the effect of lubricant within the bearing. However, these analyses do not consider the effect of inner- and outer-race cooling. The experimental work of reference 1 comprised a study of cooling rate on bearing temperatures and power loss at only one speed and load (3×10^6 DN and 22 240 newtons (5000 lb)). The objectives of the research reported herein are to determine the parametric effects of speed and load on bearing performance.

In order to accomplish the aforesaid objective parametric tests were conducted in a high-speed, high-temperature bearing tester with optimally designed 20° and 24° contact angle 120-millimeter-bore angular-contact ball bearings (ref. 8). The bearings were manufactured from double vacuum-melted AISI M-50 steel having a room-temperature Rockwell C hardness of 63. Test parameters were thrust loads of 6672, 13 350, and 22 240 newtons (1500, 3000, and 5000 lb) and nominal speeds of 12 000, 16 000, 20 000, and 25 000 rpm. Test conditions included an oil-inlet temperature of 428 K (310° F), a lubricant flow rate through the inner race of 1.2×10^{-3} cubic meter per minute (0.313 g/min), an inner-race cooling rate of 3.6×10^{-3} cubic meter per minute (0.94 g/min), and an outer-race cooling flow rate of 1.9×10^{-3} cubic meter per minute (0.5 g/min). These flow rates were held constant for all tests. The lubricant was neopentylpolyol (tetra) ester meeting the MIL-L-23699 specification.

APPARATUS, MATERIALS, AND PROCEDURE

High-Speed Bearing Tester

A schematic of the high-speed, high-temperature bearing tester used in these tests is shown in figure 1. This tester is described in detail in references 9 and 10 and has been subsequently modified to operate at speeds of 25 000 rpm (ref. 1). The tester consists of a shaft to which two test bearings are attached. Loading is supplied through a system of 10 springs which apply a thrust load to the bearings. Dual flat belts drive the test spindle from a 75-kilowatt (100 hp) fixed speed electric motor. The drive motor is mounted to an adjustable base so that drive pulleys for 12 000 to 25 000 rpm can be used with the same drive belts. The drive motor is controlled by a reduced voltage auto-transformer starter which permits a selection of the motor acceleration rate during startup. This control effectively protects the bearings from undesirable acceleration during startup.

The lubrication system of the test rig delivers flow rates up to 2.8×10^{-2} cubic meter per minute (7.5 g/min). There are three lubricant loops in the system. The oil flow in each loop is metered by adjustable flow control valves and can be individually measured by a flow rate indicator without interruption to the machine operation. Two of these loops are shown in figure 2. The first of these loops (C_o) supplies cooling oil to the test bearing outer race. The second loop is divided by a lubricant manifold which feeds individual annular grooves or channels at the shaft internal diameter proportioning the amount of oil which is to lubricate and/or cool each bearing inner race. The oil flow to the bearing through a plurality of radial holes in the center of the split inner race is designated L_i . The lubricant used to cool the bearing inner race and lubricate the contact of the cage with the race land through a number of radial holes in the inner-race shoulder is designated C_i . A selection of various lubricant schemes is permitted by the lubricant system: these include bearing lubrication through the inner-race split, lubrication of the cage-race shoulder contact region, the application of inner- and/or outer-race cooling, and a selection of any desired flow ratio for cooling and lubrication as well as the conventional lubrication through jets. The third lubricant loop is fed into the slave bearing which supports the shaft (not shown in figs. 1 and 2). By the system of valves and manifolds previously discussed an unlimited number of combinations of oil flows can be achieved to evaluate various conditions. Consequently, values of L_i , C_i , and C_o can be independent of each other.

The machine instrumentation includes the standard protective circuits which shut down a test when a bearing failure occurs or if any of the test parameters deviate from the programmed conditions. Measurements were made of bearing inner-race speed, bearing cage speed, test spindle axis amplitude of motion, oil flow, test bearing inner-

and outer-race and lubricant temperatures, and machine vibration level. The inner-race and cage speeds and spindle excursion measurements were made with magnetic pickups and proximity probes, respectively. The oil flow was established by a flow-meter. Bearing outer-race and lubricant inlet and outlet temperatures were measured by thermocouples and continuously recorded by a strip chart recorder. The inner-race temperature of the front test bearing was measured with an infrared pyrometer.

Test Bearings

The test bearings were ABEC-5 grade, split inner-race 120-millimeter-bore ball bearings. The inner and outer races, as well as the balls were manufactured from one heat of double vacuum-melted (vacuum-induction melted consumable electrode vacuum remelted) AISI M-50 steel. The chemical analysis of the particular heat is shown in table I. The nominal hardness of the balls and races was Rockwell C 63 at room temperature. Each bearing contained 15 balls, 2.0638 centimeters (13/16 in.) in diameter. The cage was a one piece inner-land riding type made of an iron base alloy (AMS 6415) heat-treated to a Rockwell C hardness range of 28 to 35 and having a 0.005 centimeter (0.002 in.) maximum thickness of silver plate (AMS 2410). The cage balance was 3 grams per centimeter (0.042 oz/in.).

The retained austenite content of the ball and race material was less than 3 percent. The inner- and outer-race curvatures were 54 and 52 percent, respectively. The hardnesses of all the components with the exception of the cage were matched within ± 1 Rockwell C point. This matching assured a nominal differential hardness in all bearings (i. e., the ball hardness minus the race hardness, commonly called ΔH) of zero (ref. 11). Surface finish of the balls was 2.5×10^{-6} centimeter (1 μ in.) AA and the inner and outer raceways were held to a 5×10^{-6} centimeter (2 μ in.) AA maximum surface finish.

A photograph of the test bearing is shown in figure 3. The bearing design permitted under-race lubrication by virtue of radial slots machined into the halves of the split inner races. It had been shown in reference 2 that this was the most reliable technique for lubricating high-speed bearings. Provision was also made for inner-race land to cage lubrication by the incorporation of several small diameter holes radiating from the bore of the inner race to the center of the inner shoulder.

Lubricant

The oil used for the parametric studies was a 5-centistoke neopentylpolyol (tetra) ester. This is a type II oil, which is qualified to MIL-L-23699 as well as to the internal

oil specifications of most major aircraft-engine producers. The major properties of the oil are presented in table II and a temperature-viscosity curve is shown in figure 4.

Test Procedure

The test procedure was adjusted according to the test conditions to be evaluated. Generally, a program cycle was defined which would allow the evaluation of a number of conditions without a major interruption. Test parameters such as load, speed, and oil inlet temperature were maintained constant while the tester was in operation. The lubricant flow rate was also constant during operation. The total lubricant flow rate selected was based on the results reported in reference 1. This flow ($6.7 \times 10^{-3} \text{ m}^3/\text{min}$ (1.75 g/min)) assured that the maximum bearing temperature at 25 000 rpm and 22 240 newtons (5000 lb) did not exceed 494 K (430° F). During operation the tester was allowed to reach the equilibrium condition before the data were recorded.

Power loss per bearing was determined by measuring line to line voltage and line current to the test-rig drive motor. Motor drive power was then calculated as a function of line current. The motor power less estimated test rig tare losses at the various operating speeds reflected bearing power usage. At 25 000 rpm and 22 240 newtons (5000 lb) load, this determination of power consumption may be 3 kilowatts (4 hp) low based on the data presented in reference 1.

RESULTS AND DISCUSSION

Effect of Speed and Load on Bearing Temperature

The effects of load on bearing inner- and outer-race temperatures of the 120-millimeter-bore angular-contact ball bearings are shown in figures 5 and 6. The 20° and 24° contact angle bearings were run at nominal speeds of 12 000, 16 000, 20 000, and 25 000 rpm. Test conditions included an oil at 428 K (310° F), an inner race flow rate L_i of 1.2×10^{-3} cubic meter per minute (0.313 g/min), an inner race cooling rate C_i of 3.6×10^{-3} cubic meter per minute (0.94 g/min) or $3 L_i$, and an outer race cooling flow C_o of 1.9×10^{-3} cubic meter per minute (0.5 g/min).

Bearing temperatures increased nearly linearly with load for most of the speeds. For any given speed the temperature rise ranged from 4 to 17 K (8 to 30 F deg) at both inner and outer races when the load was increased from 6672 to 22 240 newtons (1500 to 5000 lb). At any given speed and load the difference in temperature between the inner and outer race for the 24° contact-angle bearings (fig. 6) was insignificant and generally

within ± 3 K (± 5 F deg). For the 20° contact-angle bearings (fig. 5) the difference between inner- and outer-race temperatures was largest at the lower loads for all four speeds. The maximum difference in temperature was 8 K (15 F deg) at 20 000 rpm for the 6672 newton (1500 lb) thrust load.

The effect of speed on the inner- and outer-race temperatures of the 120-millimeter-bore angular-contact ball bearings are shown in figures 7 and 8 for the three thrust loads. Over the range of speeds investigated both inner- and outer-race temperatures ranged from 441 to 494 K (335° to 430° F) or an increase of 53 K (95 F deg). In general, the 24° contact angles bearing (fig. 8) ran approximately 3 to 8 K (5 to 10 F deg) cooler than 20° contact angle bearings (fig. 7) over the entire speed range. These differences in temperature were not considered significant.

Since the effects of bearing contact angle and load appear to be second order effects, the effect of bearing speed on temperature can be considered independently. The difference between inner-race temperature and lubricant inlet temperature as a function of speed is shown in figure 9. (A single line is drawn through the array of data points regardless of contact angle and/or load.) The resulting temperature of the inner race can be approximated independently of load and contact angle by the following empirical relation:

$$T_{Bi} = T_{oil\ in} + K_1 N^m \quad (1)$$

where

T_{Bi}	inner-race temperature, K ($^\circ$ F)
$T_{oil\ in}$	lubricant inlet temperature, K ($^\circ$ F)
K_1	proportionality factor, $(K/rpm)^m$ ($(^\circ$ F/rpm) m)
N	speed, rpm
m	exponent

For these data, the value for m can be taken as 1.36. For temperature in degrees Kelvin, K_1 equals 6.3×10^{-5} , for temperature in degrees Fahrenheit, K_1 equals 1.2×10^{-4} .

From this it may be concluded that within the range of data presented the bearing inner- and outer-race temperatures are more sensitive to changes in speed than changes in thrust load or contact angle. The bearing speed parameter combined with lubricant flow rate into the bearing (ref. 1) can give a reasonable engineering approximation of bearing inner-race temperature.

Effect of Speed and Load on Bearing Power Loss

The power loss for both bearings over the speed range is shown in figures 10 and 11. Bearing power consumption increases linearly with speed for the three loads shown. At a thrust load of 22 240 newtons (5000 lb) and a speed of 25 000 rpm, the power loss per bearing was approximately 15 kilowatts (20 hp) for both the 20° and 24° contact angles. At 12 000 rpm and a thrust load of 6672 newtons (1500 lb), the power loss was approximately 3 kilowatts (4 hp) or 20 percent of the power loss at the high-speed, high-load condition.

For two contact-angle bearings at a 22 240-newton (5000-lb) load and at low speeds, the 24° contact-angle bearing had a power loss approximately 25 percent greater than the 20° contact-angle bearing. At the higher speeds and for all three loads, the difference in power loss between the two contact angles was essentially insignificant.

Bearing power loss as a function of load for the two contact angles is shown in figures 12 and 13. The greatest increase in power consumption from the 6672-newton (1500-lb) thrust load to the 22 240-newton (5000-lb) thrust load occurred at 12 000 rpm and was approximately 3 kilowatts (4 hp). The increase in power consumption with thrust load may be considered negligible relative to the effect of speed for most practical engineering design applications.

From the data of figures 10 and 11 a bearing coefficient of friction can be determined:

$$f = \frac{K_2 P_B}{\pi D T_L N} \quad (2)$$

where

- D bearing pitch diameter, m (ft)
- f bearing coefficient of friction
- K_2 constant, m-N/kW-min (ft-lb/hp-min)
- N bearing speed, rpm
- P_B bearing power loss, kW (hp)
- T_L bearing thrust load, N (lb)

For power in terms of kilowatts, K_2 equals 18 244; for power in terms of horsepower, K_2 equals 33 000.

The calculated values of f were essentially independent of speed and contact angle. They were, however, a function of load. Average values of f were 0.008, 0.005, and 0.003 for loads of 6672, 13 350, and 22 240 newtons (1500, 3000, and 5000 lb), respectively.

Effect of Speed on Contact Angle

Another effect of speed on bearing operation is the centrifugal effect on outer-race load or stress and dynamic contact angle. The effect of speed on contact stress and operating contact angle using the methods of Harris (refs. 6 and 7) is shown in table III. For these calculations, a bearing operating temperature of 478 K (400° F) was assumed. These calculated data show that there is a more marked increase in stress with speed at the outer-race ball contact at the 6672-newton (1500-lb) thrust load than at the 22 240-newton (5000-lb) thrust load. This difference in the relative increase in stress at the outer-race ball contact probably accounts for the slightly higher rate of increase in power consumption with speed with the lower thrust loaded bearing tests. Bearing running track location was measured directly on bearings having an initial contact angle of 24° and tested at 12 000, 16 000, and 25 000 rpm with a 22 240-newton (5000-lb) load. Trigonometric relations were then employed to calculate the actual contact angles. These contact angles were in excellent agreement with the theoretical predictions made in table III, with the exception of the inner-race contact angle at the maximum speed condition. At this speed the measured angle was approximately 25° against a predicted operating angle of 34°. The reason for this discrepancy is not understood at the present time.

Effect of Load and Inner-Race Speed on Cage Speed

Designers of high-speed turbomachinery are concerned with potential skidding problems which may occur in rolling-element bearings. Skidding is generally experienced in lightly loaded, high-speed ball bearings. For angular-contact ball bearings there is controversy as to whether there is a negative effect of skidding. Harris (ref. 6) considers skidding occurring when "no line of instant centers occurs in the ball inner race contact. This condition may well occur on the descending portion of the curve of cage/shaft (inner-race) speed ratio against load. Therefore, the critical load, i. e., the load below which skidding occurs, may not necessarily be indicated by the curve maximum, although this is probably a safe design point."

A plot of cage/shaft speed ratio as a function of thrust load for the 120-millimeter-bore angular-contact ball bearings is shown in figure 14. Both the experimental and theoretical (ref. 6) cage speeds are shown. With the exception of the 20° contact angle bearing at a shaft speed of 12 000 rpm, the experimental values of cage/shaft speed ratio are less than theoretically predicted. Based on the Harris definition of skidding (ref. 6), it must be concluded that with the exception of the 12 000 rpm, 20° contact angle data, the bearings operated in a skidding regime. Visual examination of the bearing surfaces after running showed no damage to the raceways or the balls.

The amount of skidding occurring in the bearing can be estimated by comparing the theoretical cage speed with the experimental values. These data are presented in figure 15. In general, as the bearing thrust load is increased the experimental cage speed approaches the theoretical estimates. The greatest deviation between experimental and theoretical values occur at the high-speed, low-load conditions. For the 12 000 rpm (1.44×10^6 DN) condition, skidding in the bearing is negligible. However, at the 25 000 rpm (3×10^6 DN) condition there is approximately 5 to 6 percent skidding in the ball inner-race contact as determined by the difference between the calculated and experimental cage speeds. However, this amount may be actually less due to the difference between the measured and theoretical inner-race contact angle.

At the 25 000 rpm speed the elastohydrodynamic (EHD) film thickness divided by the surface composite roughness (commonly called Λ) is approximately 4 ($\Lambda \approx 4$) (ref. 11). This condition is sufficient to give full EHD lubrication. That is, there should be essentially no asperity contacts within the ball-race contacts. Hence, it can be concluded that where skidding takes place under a full EHD film, no damage to the rolling-element surfaces will occur.

SUMMARY OF RESULTS

Parametric tests were conducted in a high-speed, high-temperature bearing tester with optimally designed 20° and 24° contact angle 120-millimeter-bore angular-contact ball bearings. The bearings were manufactured from double vacuum-melted AISI M-50 steels having a room temperature Rockwell C hardness of 63. Test parameters were thrust loads of 6672, 13 350, and 22 240 newtons (1500, 3000, and 5000 lb) and nominal speeds of 12 000, 16 000, 20 000, and 25 000 rpm. Test conditions included an oil-inlet temperature of 428 K (310° F), a lubricant flow rate through the inner race of 1.2×10^{-3} cubic meter per minute (0.313 g/min), an inner-race cooling rate of 3.6×10^{-3} cubic meter per minute (0.94 g/min), and an outer-race cooling flow rate of 1.9×10^{-3} cubic meter per minute (0.5 g/min). The lubricant was a neopentylpolyol (tetra) ester. The following results were obtained:

1. Bearing inner- and outer-race temperatures increased with increases in load and/or speed. However, the effect of load on temperature was small relative to the speed effect. The difference between inner and outer race temperatures was negligible.
2. Bearing power consumption increased with speed. The increase in power consumption with thrust load was small relative to the effect of speed for most practical engineering design applications.
3. A comparison of measured cage speed to theoretical predictions indicates that, with the exception of the 12 000 rpm, 20° contact angle data, the bearings operated with some skidding. The highest amounts of skidding (6 percent of the theoretical cage speed)

occurred at the 25 000 rpm (3×10^6 DN) test conditions. Skidding decreased with increasing thrust load and decreasing speed. No damage to the bearing surfaces under conditions of skidding was observed.

4. Actual measurements of bearing operating contact angle were in excellent agreement with the theoretical prediction with the exception of the inner-race contact angle at the maximum speed and load condition.

Lewis Research Center,
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TABLE I. - CHEMICAL ANALYSIS OF VACUUM
INDUCTION MELTED, CONSUMABLE-ELECTRODE
VACUUM REMELTED AISI M-50 BEARING STEEL

Element	Composition of races and balls, wt %
Carbon	0.83
Manganese	.29
Phosphorus	.007
Sulfur	.005
Silicon	.25
Chromium	4.11
Molybdenum	4.32
Vanadium	.98
Iron	Balance

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 477 K (400° F), J/(kg)(K) (Btu/(lb)(°F))	2340 (0.54)
Thermal conductivity at 477 K (400° F) J/(m)(sec)(K) (Btu/(hr)(ft)(°F))	0.13 (0.075)
Specific gravity at 477 K (400° F)	0.850

TABLE III. - CALCULATED OPERATING CONTACT ANGLES AND STRESSES
AS FUNCTION OF INITIAL CONTACT ANGLE, SPEED, AND LOAD

Speed, rpm	Thrust load		Contact angle, deg			Outer race		Inner race	
	N	lb	Unloaded	Operating		Maximum Hertz stress			
				Outer race	Inner race	N/m ²	ksi	N/m ²	ksi
12 000	6 672	1500	20	13	26	1365×10 ⁶	198	1420×10 ⁶	206
	13 350	3000	20	18	26	1572	228	1806	262
	22 240	5000	20	21	27	1779	258	2130	309
	6 672	1500	24	14	30	1344	195	1351	196
	13 350	3000	24	19	30	1531	222	1731	251
	22 240	5000	24	23	31	1731	251	2048	297
16 000	6 672	1500	20	9	27	1517×10 ⁶	220	1400×10 ⁶	203
	13 350	3000	20	14	28	1682	244	1779	258
	22 240	5000	20	18	28	1855	269	2096	304
	6 672	1500	24	9	32	1503	218	1330	193
	13 350	3000	24	15	32	1648	239	1703	247
	22 240	5000	24	19	32	1813	263	2020	293
20 000	6 672	1500	20	6	28	1682×10 ⁶	244	1386×10 ⁶	201
	13 350	3000	20	11	29	1813	263	1758	255
	22 240	5000	20	15	29	1958	284	2068	300
	6 672	1500	24	6	33	1682	244	1324	192
	13 350	3000	24	12	33	1923	260	1682	244
	22 240	5000	24	16	33	1924	279	1993	289
25 000	6 672	1500	20	5	29	1889×10 ⁶	274	1365×10 ⁶	198
	13 350	3000	20	8	30	1993	289	1731	251
	22 240	5000	20	12	30	2110	306	2041	296
	6 672	1500	24	5	34	1862	270	1310	190
	13 350	3000	24	8	34	1993	289	1662	241
	22 240	5000	24	12	34	2096	304	1965	285

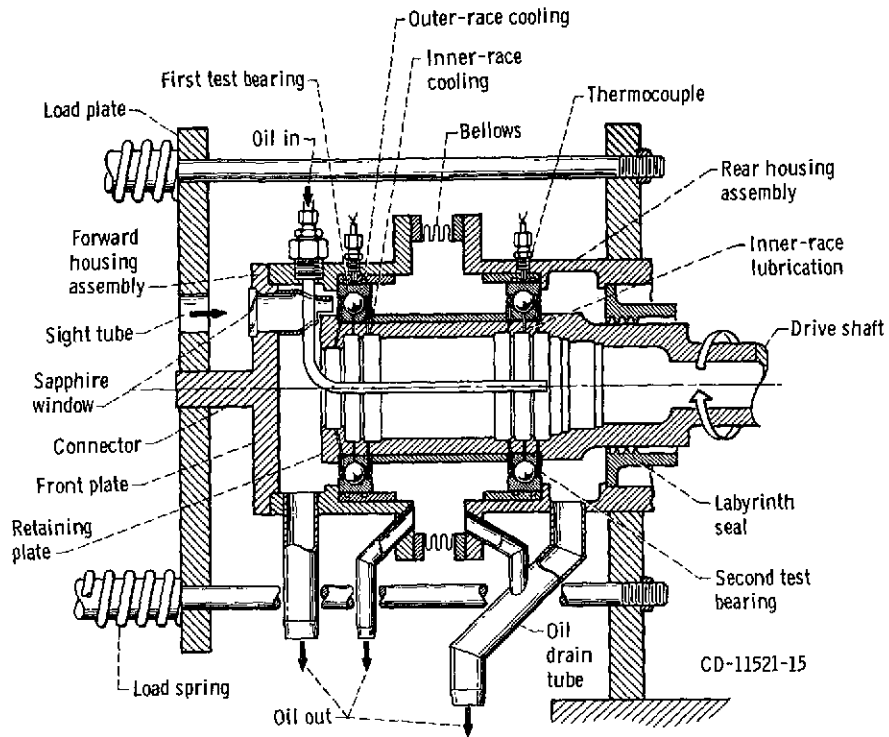


Figure 1. - High-speed, high-temperature bearing test apparatus.

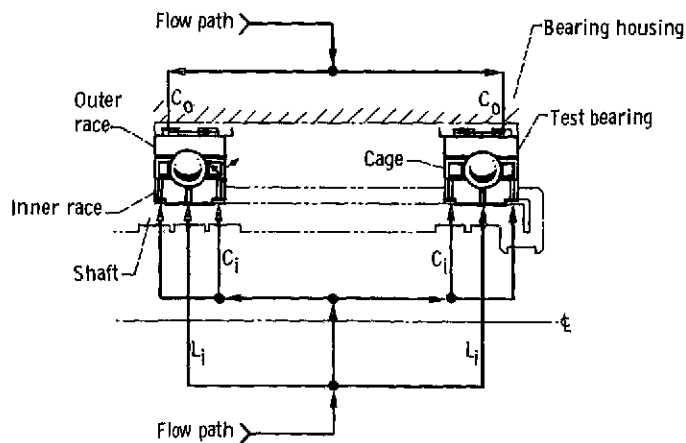


Figure 2. - Lubricant system for test bearings.

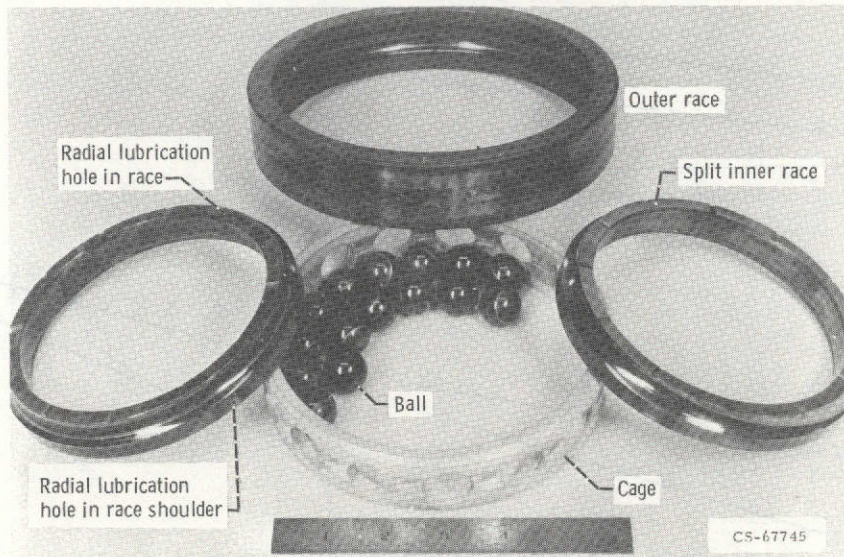


Figure 3. - Unfailed 120-millimeter-bore angular-contact high-speed test ball bearing. Running time, 1000 hours; speed, 25 000 rpm (3 million DN); temperature, 492 K (425° F); thrust load, 22 240 newtons (5000 lb).

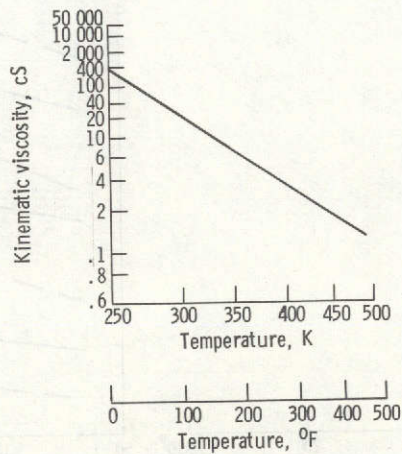


Figure 4. - Viscosity as function of temperature for tetra-ester (type II) lubricant.

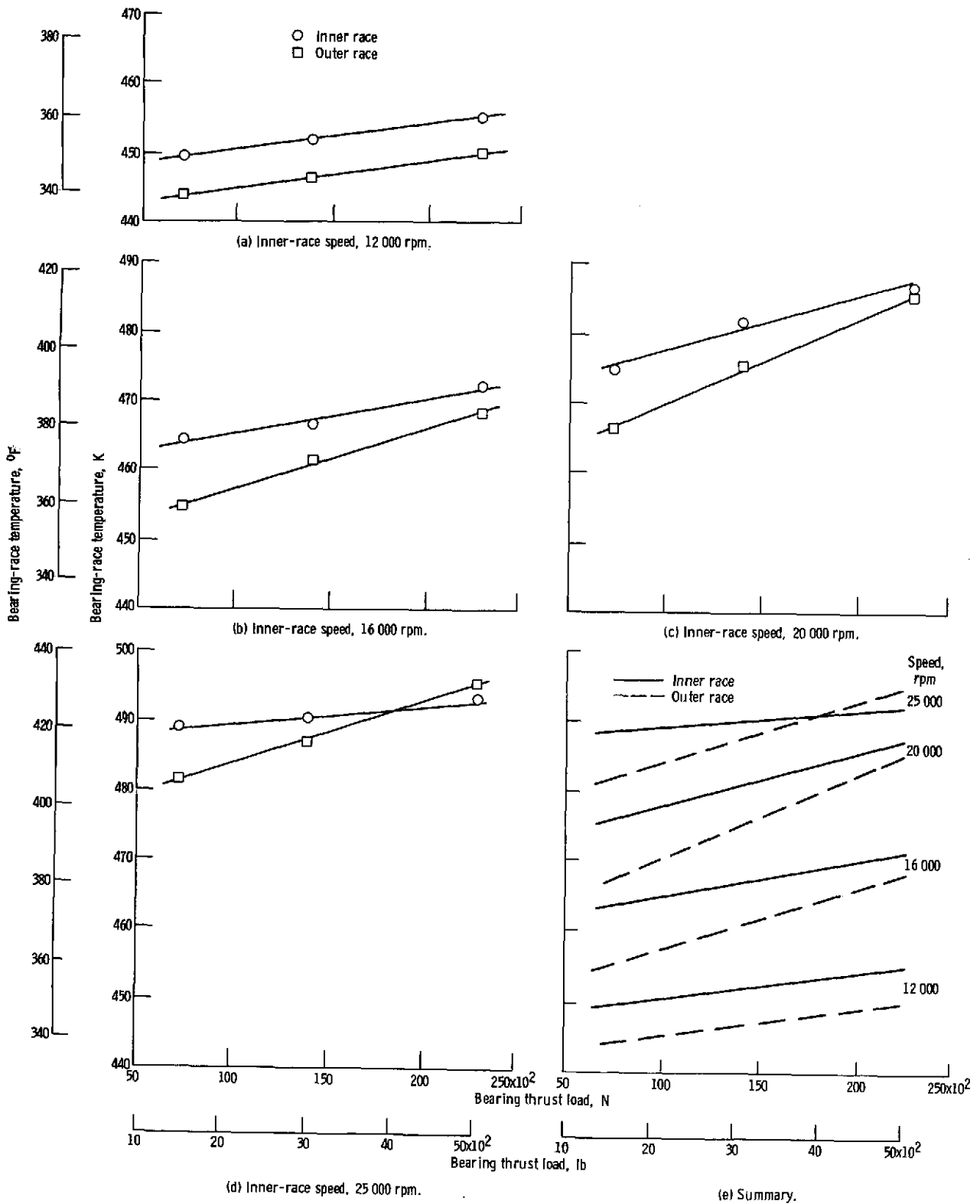


Figure 5. - Temperature of 20° contact angle bearing as function of bearing thrust load for various speeds. Bearing type, 120-millimeter-bore angular-contact ball bearing; lubricant flow to inner race, 1.2×10^{-3} cubic meter per minute (0.313 g/min); inner-race cooling flow, C_i , 3.6×10^{-3} cubic meter per minute (0.94 g/min); outer-race cooling flow, C_o , 1.9×10^{-3} cubic meter per minute (0.5 g/min); oil inlet temperature, 428 K (310° F).

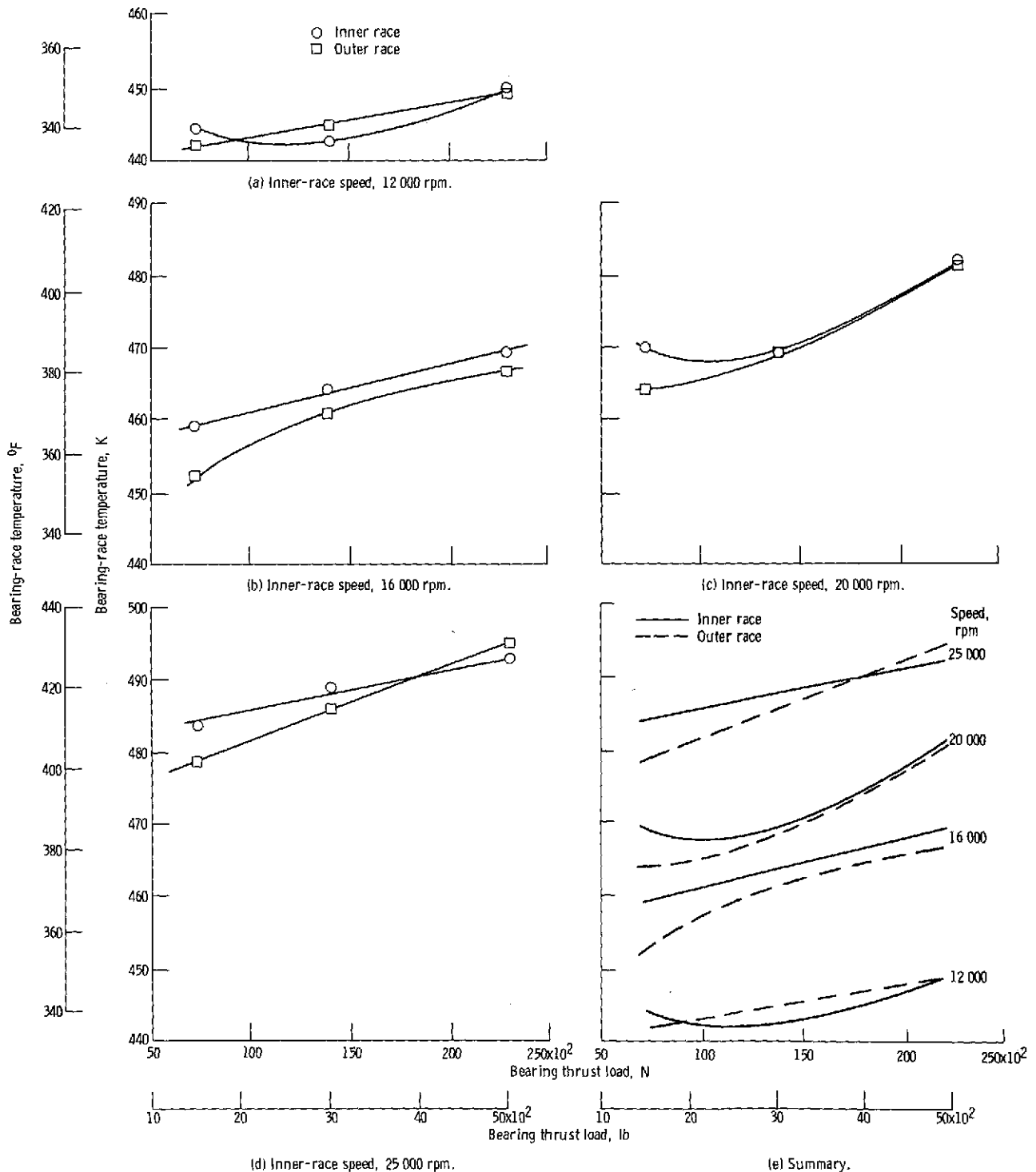


Figure 6. - Temperature of 24° contact angle bearing as function of bearing thrust load for various speeds. Bearing type, 120-millimeter-bore angular-contact ball bearings; lubricant flow to inner race, L_i , 1.2×10^{-3} cubic meter per minute (0.313 g/min); inner-race cooling flow, C_i , 3.6×10^{-3} cubic meter per minute (0.94 g/min); outer-race cooling flow, C_o , 1.9×10^{-3} cubic meter per minute (0.5 g/min); oil inlet temperature, 428 K (310° F).

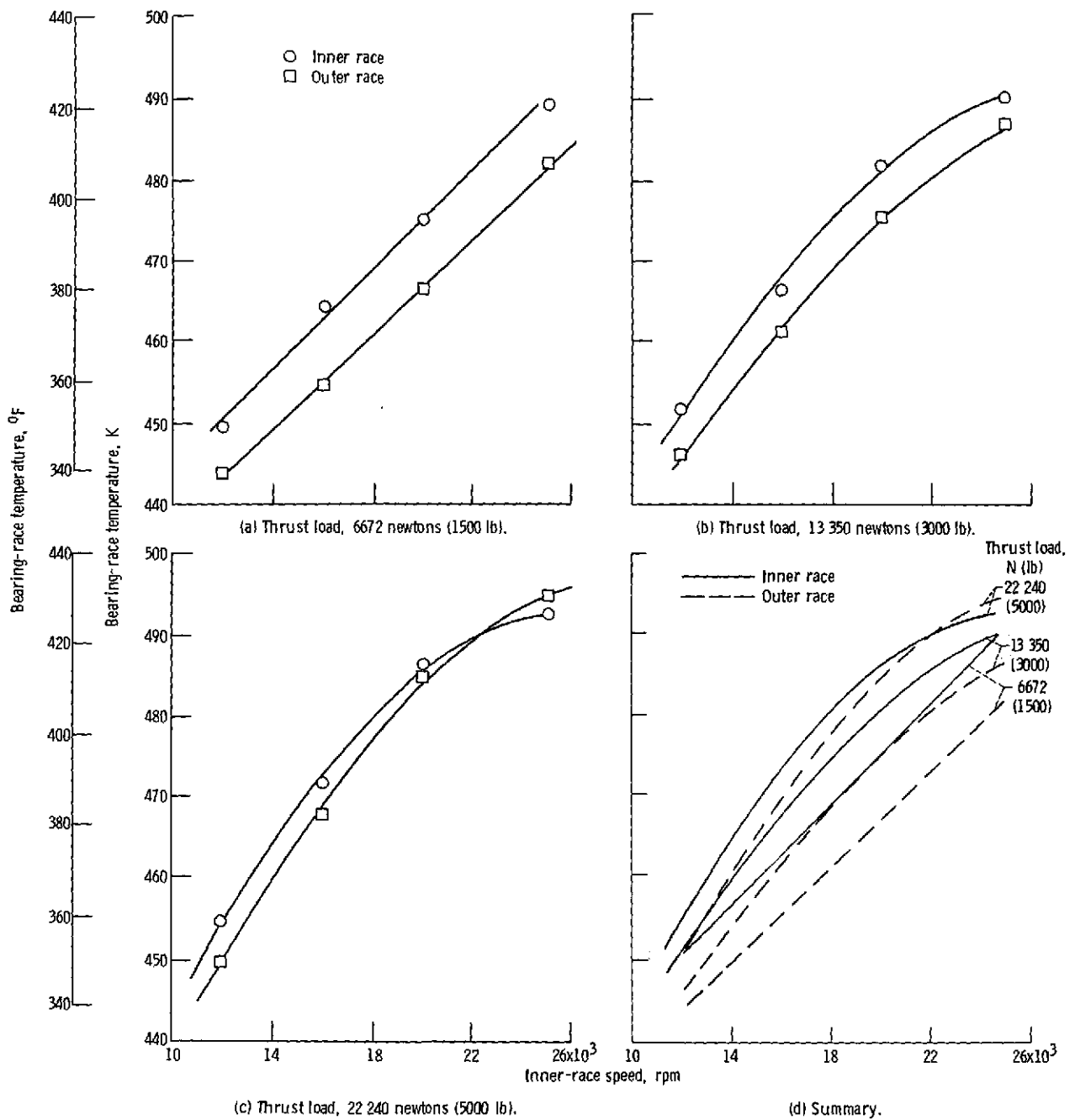


Figure 7. - Temperature of 20° contact angle bearing as function of speed for various bearing thrust loads. Bearing type, 120-millimeter-bore angular-contact ball bearings; lubricant flow to inner race, L_i , 1.2×10^{-3} cubic meter per minute (0.313 g/min); inner-race cooling flow, C_i , 3.6×10^{-3} cubic meter per minute (0.94 g/min); outer-race cooling flow, C_o , 1.9×10^{-3} cubic meter per minute (0.5 g/min); oil inlet temperature, 428 K (310° F).

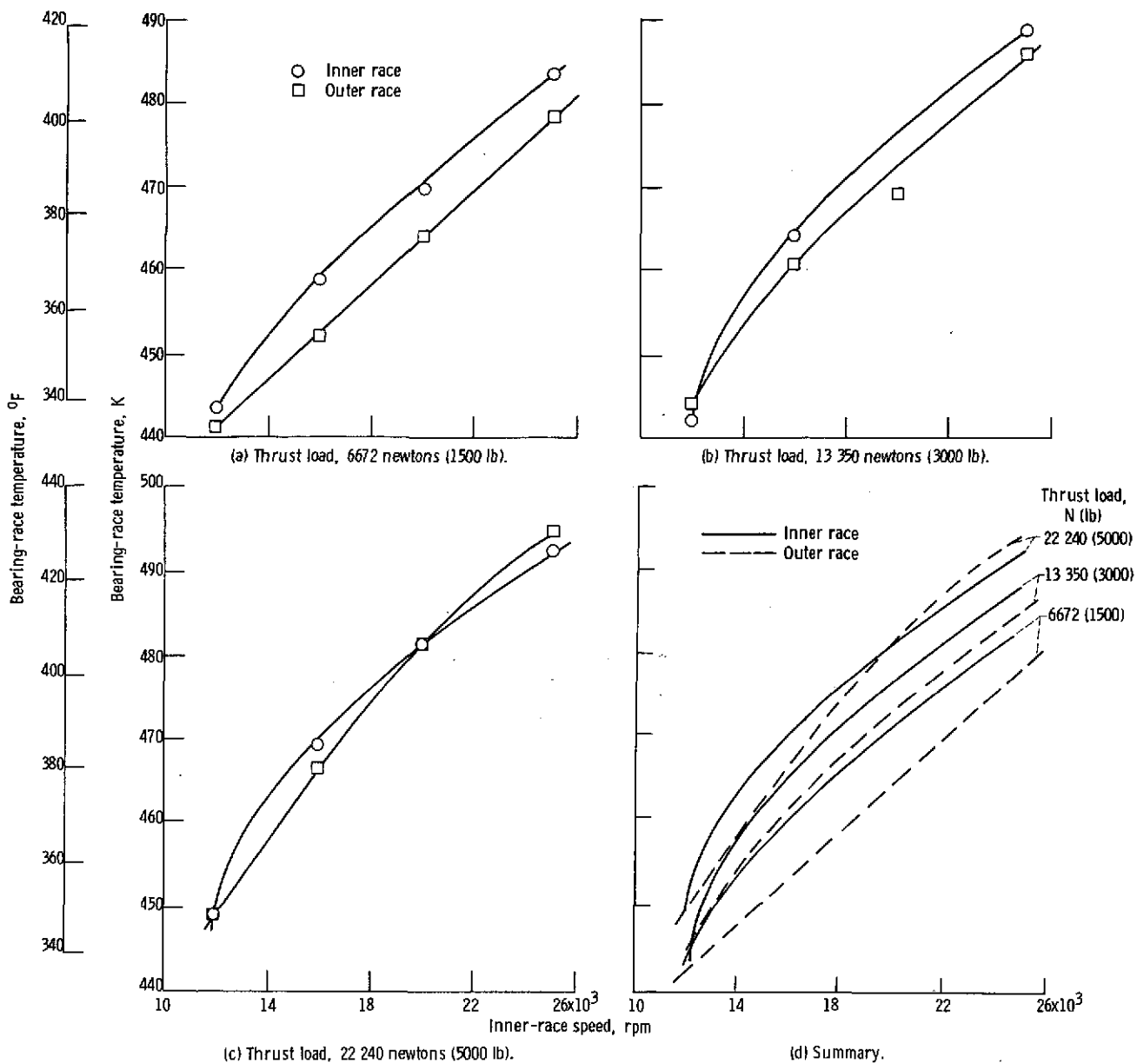


Figure 8. - Temperature of 24° contact angle bearing as function of speed for various bearing thrust loads. Bearing type, 120-millimeter-bore angular-contact ball bearing; lubricant flow to inner race, L_i , 1.2×10^{-3} cubic meter per minute (0.313 g/min); inner-race cooling flow, C_i , 3.6×10^{-3} cubic meter per minute (0.94 g/min); outer-race cooling flow, C_o , 1.9×10^{-3} cubic meter per minute (0.5 g/min); oil inlet temperature, 428 K (310° F).

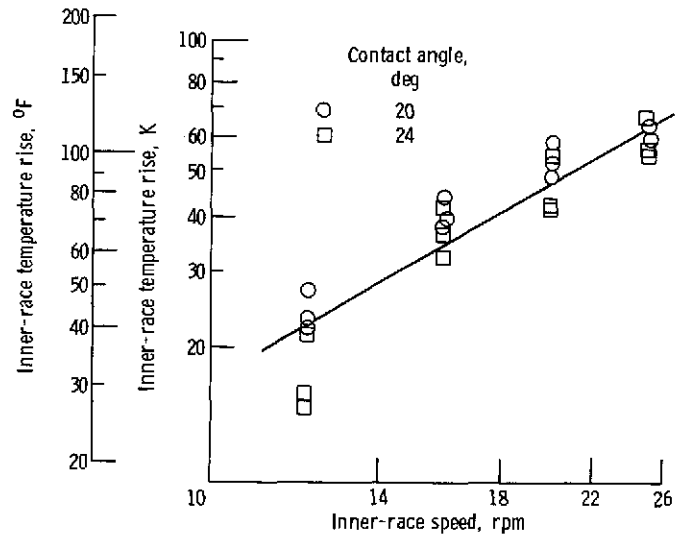


Figure 9. - Difference between inner-race temperature and oil inlet temperature as function of speed. Bearing type, 120-millimeter-bore angular-contact ball bearing; lubricant flow to inner race, L_i , 1.2×10^{-3} cubic meter per minute (0.313 g/min); inner-race cooling flow, C_i , 3.6×10^{-3} cubic meter per minute (0.94 g/min); outer-race cooling flow, C_o , 1.9×10^{-3} cubic meter per minute (0.5 g/min); oil inlet temperature, 428 K (310° F).

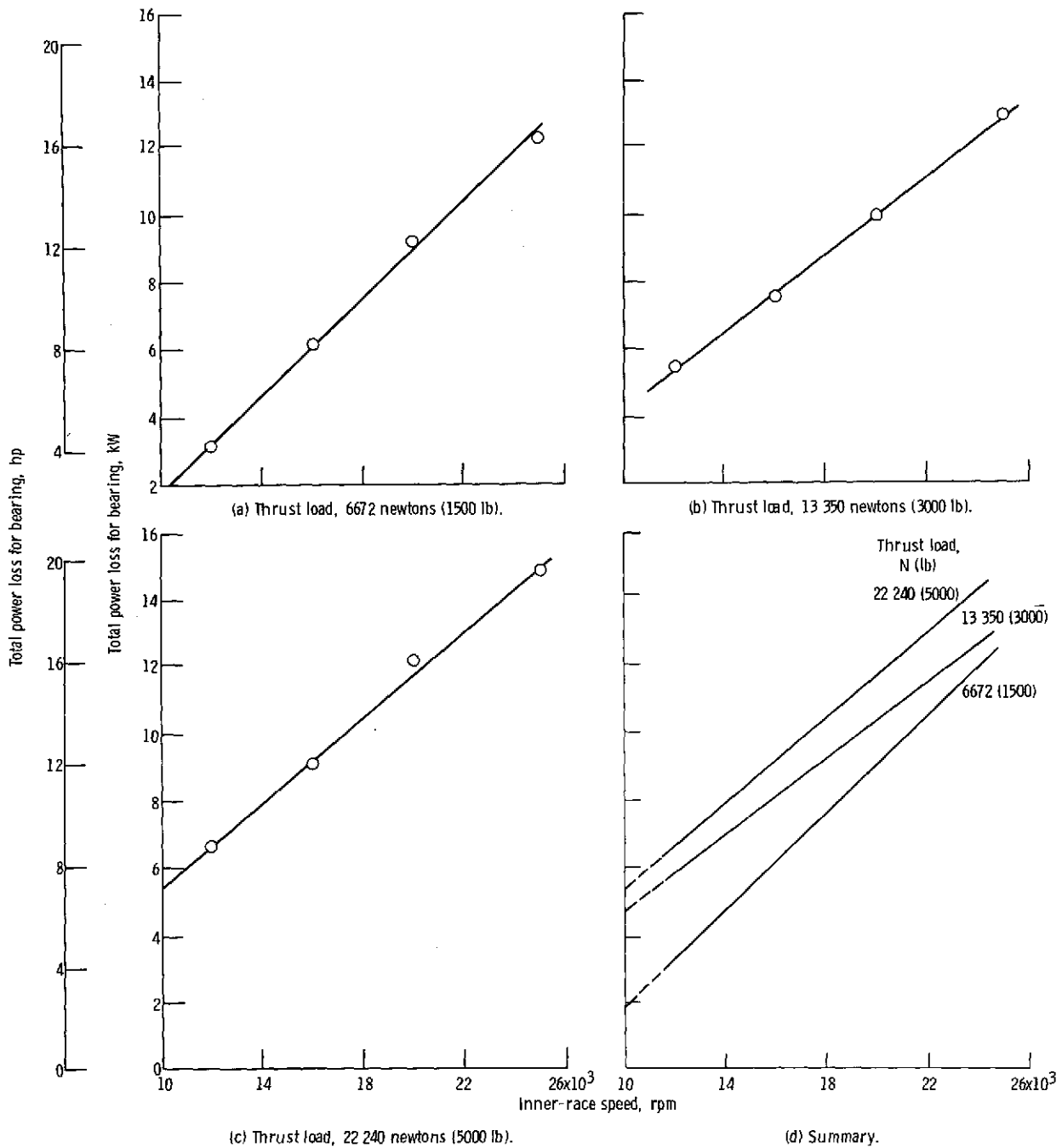


Figure 10. - Bearing power loss for 20° contact angle bearing as function of speed for varying bearing thrust loads. Bearing type, 120-millimeter-bore angular-contact ball bearing; lubricant flow to inner race, L_i , 1.2×10^{-3} cubic meter per minute (0.313 g/min); inner-race cooling flow, C_i , 3.6×10^{-3} cubic meter per minute (0.94 g/min); outer-race cooling flow, C_o , 1.9×10^{-3} cubic meter per minute (0.5 g/min); oil inlet temperature, 428 K (310° F).

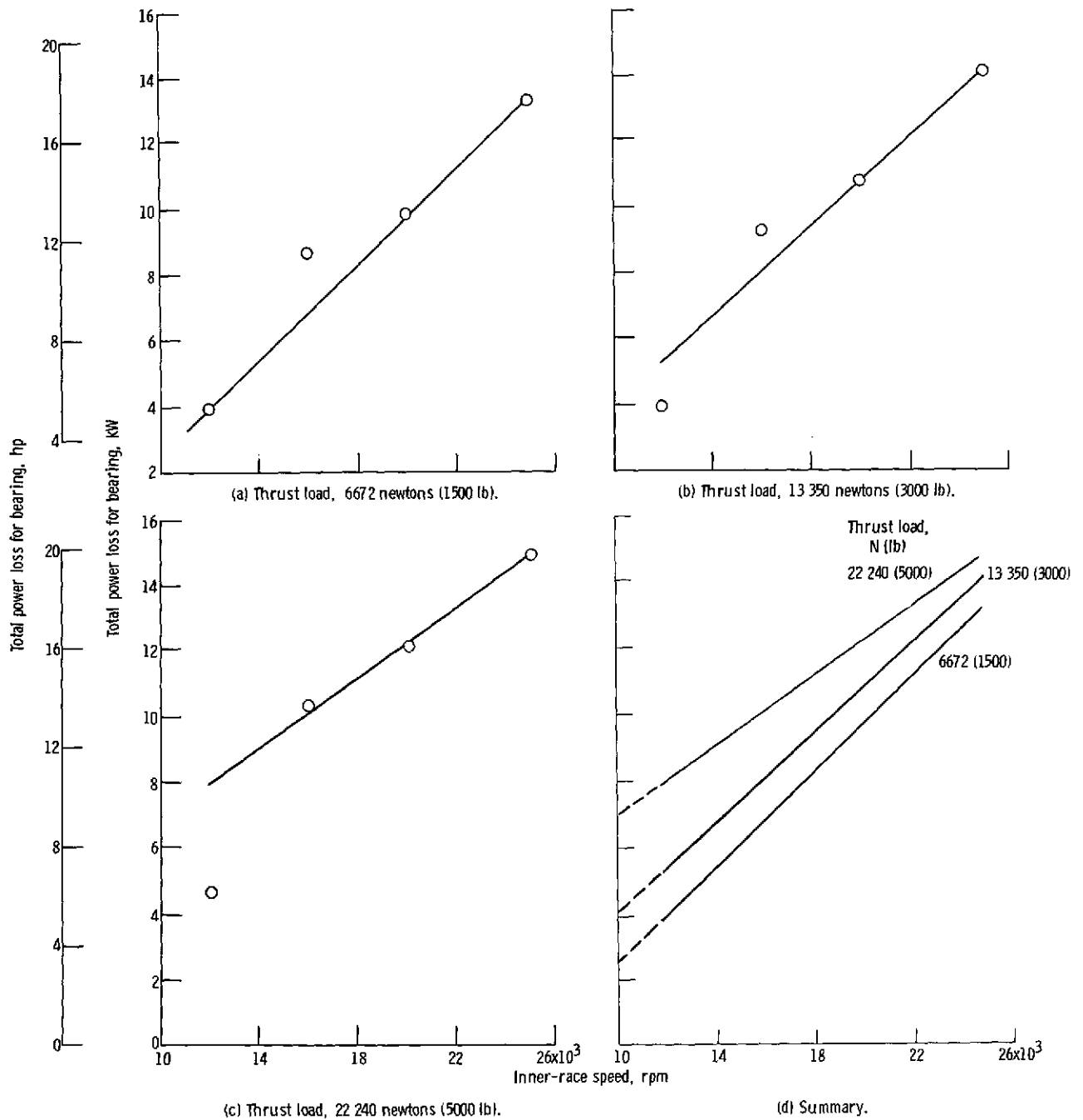


Figure 11. - Bearing power loss for 24° contact angle bearing as function of speed for varying bearing thrust loads. Bearing type, 120-millimeter-bore angular-contact ball bearing; lubricant flow to inner race, L_i , 1.2×10^{-3} cubic meter per minute (0.313 g/min); inner-race cooling flow, C_i , 3.6×10^{-3} cubic meter per minute (0.94 g/min); outer-race cooling flow, C_o , 1.9×10^{-3} cubic meter per minute (0.5 g/min); oil inlet temperature, 428 K (310° F).

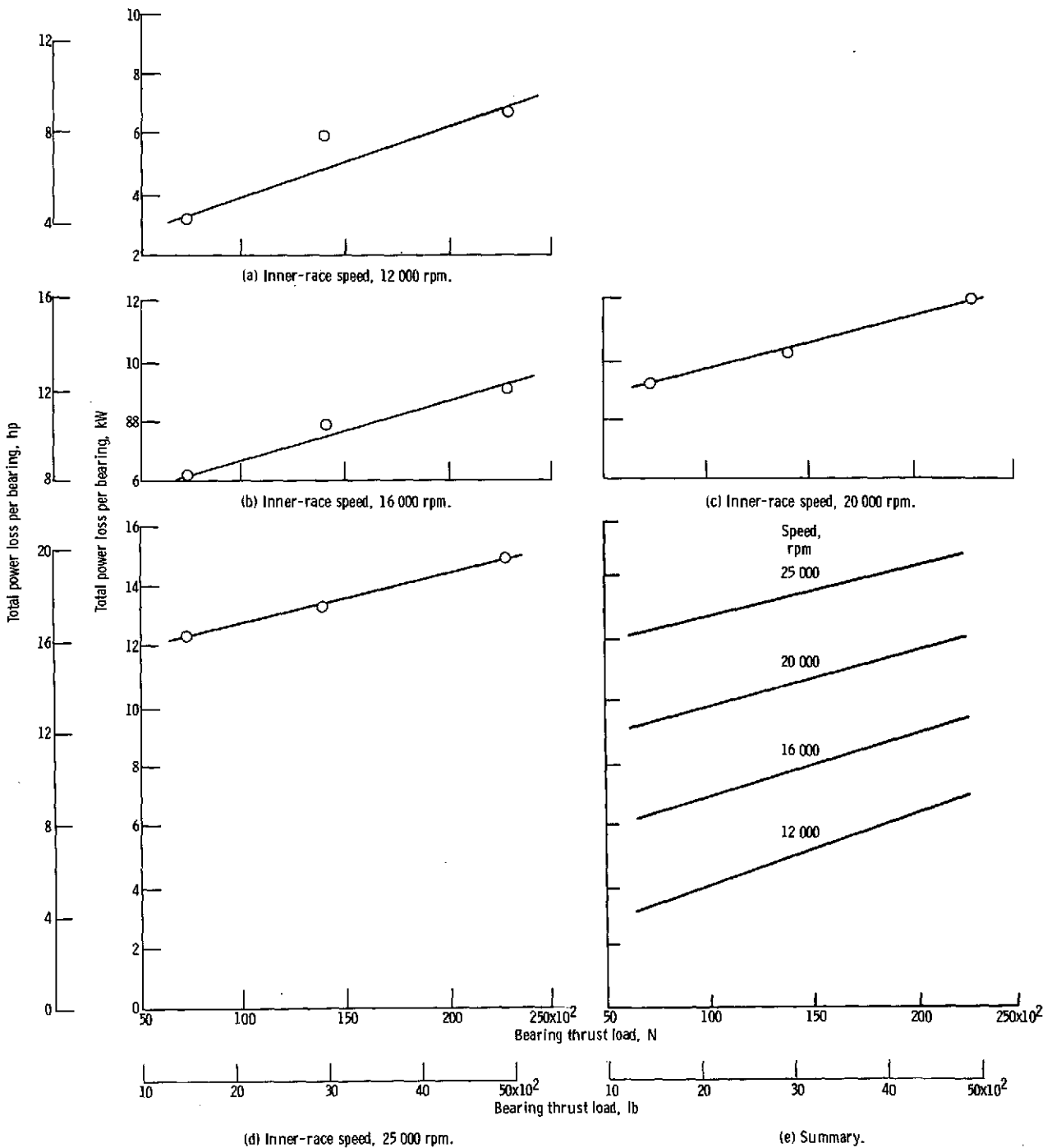


Figure 12. - Bearing power loss for 20° contact angle bearing as function of bearing thrust load for various speeds. Bearing type, 120-millimeter-bore angular-contact ball bearing; lubricant flow to inner race, L_i , 1.2×10^{-3} cubic meter per minute (0.313 g/min); inner-race cooling flow, C_i , 3.6×10^{-3} cubic meter per minute (0.94 g/min); outer-race cooling flow, C_o , 1.9×10^{-3} cubic meter per minute (0.5 g/min); oil inlet temperature, 428 K (310° F).

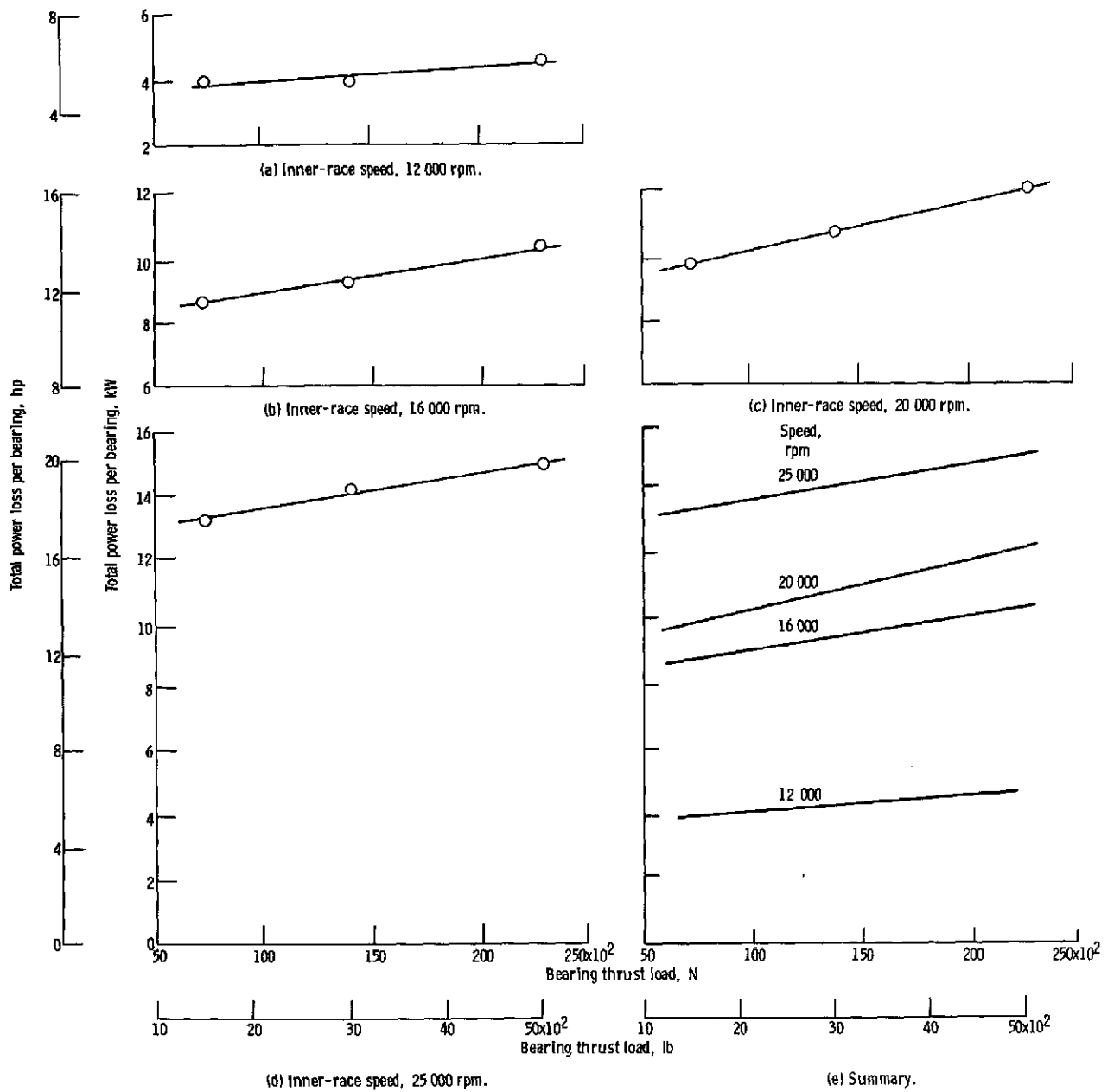


Figure 13. - Bearing power loss for 24° contact angle bearing as function of bearing thrust load for various speeds. Bearing type, 120-millimeter-bore angular-contact ball bearing; lubricant flow to inner race, L_i , 1.2×10^{-3} cubic meter per minute (0.313 g/min); inner-race cooling flow, C_i , 3.6×10^{-3} cubic meter per minute (0.94 g/min); outer-race cooling flow, C_o , 1.9×10^{-3} cubic meter per minute (0.5 g/min); oil inlet temperature, 428 K (310° F).

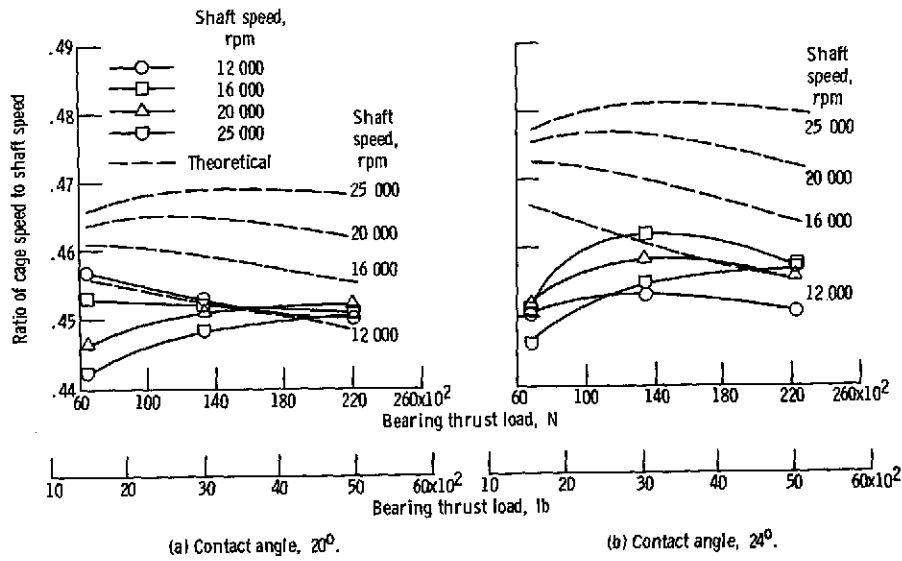


Figure 14. - Bearing cage speed relative to shaft speed as function of bearing thrust load for various inner-race speeds. Bearing type, 120-millimeter-bore angular-contact ball bearing; lubricant flow to inner race, L_i , 1.2×10^{-3} cubic meter per minute (0.313 g/min); inner-race cooling flow, C_i , 3.6×10^{-3} cubic meter per minute (0.94 g/min); outer-race cooling flow, C_o , 1.9×10^{-3} cubic meter per minute (0.5 g/min); oil inlet temperature, 428 K (310° F).

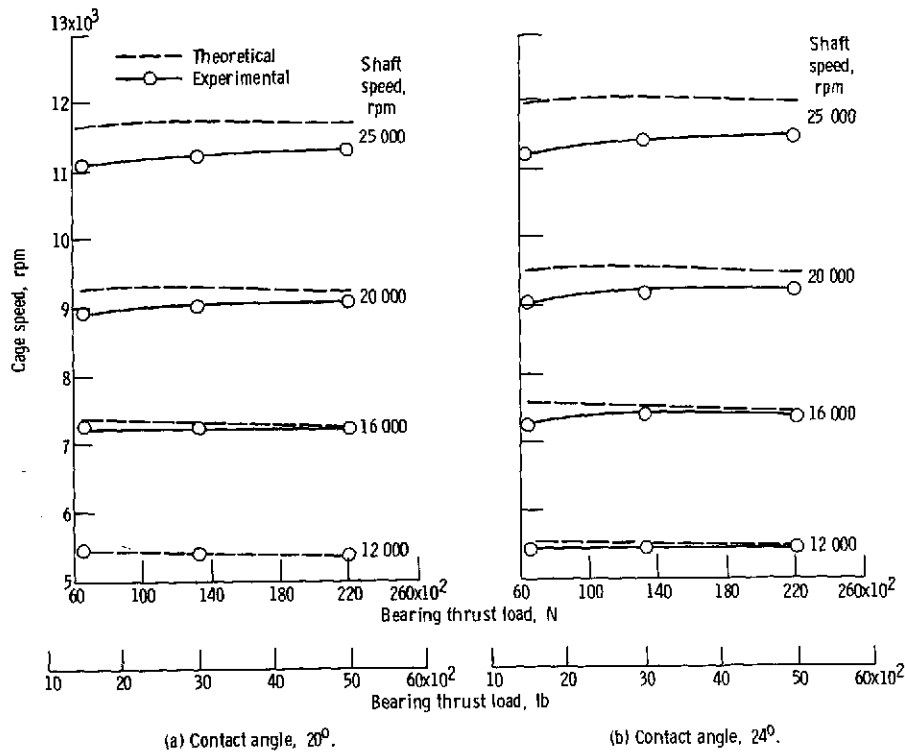


Figure 15. - Bearing cage speed as function of bearing thrust load for various inner-race speeds. Bearing type, 120-millimeter-bore angular-contact ball bearing; lubricant flow to inner race, L_i , 1.2×10^{-3} cubic meter per minute (0.313 g/min); inner-race cooling flow, C_i , 3.6×10^{-3} cubic meter per minute (0.94 g/min); outer-race cooling flow, C_o , 1.9×10^{-3} cubic meter per minute (0.5 g/min); oil inlet temperature, 428 K (310° F).