REPORT 1084

COMPARISON OF HIGH-SPEED OPERATING CHARACTERISTICS OF SIZE 215 CYLINDRICAL-ROLLER BEARINGS AS DETERMINED IN TURBOJET ENGINE AND IN LABORATORY TEST RIG[1]

By E. FRED MACKS and ZOLTON N. NEMETH

SUMMARY

A comparison of the operating characteristics of 75-millimeter-bore (size 215) cylindrical-roller one-piece inner-race-riding cage-type bearings was made by means of a laboratory test rig and a turbojet engine. Cooling correlation parameters were determined by means of dimensional analysis, and the generalized results for both the inner- and the outer-race bearing operating temperatures are compared for the laboratory test rig and the turbojet engine.

Inner- and outer-race cooling-correlation curves were obtained for the turbojet-engine turbine roller bearing with the same inner- and outer-race correlation parameters and exponents as those determined for the laboratory test-rig bearing. A method is given that enables the designer to predict the inner- and outer-race turbine roller-bearing temperatures from single curves, regardless of variations in speed, load, oil flow, oil inlet temperature, oil inlet viscosity, oil-jet diameter, or any combination of these parameters.

The turbojet-engine turbine roller-bearing inner-race temperatures were 30° to 60° F greater than the outer-race-maximum temperatures, the exact values depending on the operating condition and oil viscosity; these results are in contrast to the laboratory test-rig results, for which the inner-race temperatures were less than the outer-race-maximum temperatures.

The turbojet-engine turbine roller-bearing maximum outer-race circumferential temperature variation was approximately 30° F for each of the oils used.

The effect of oil viscosity on inner- and outer-race turbojet-engine turbine roller-bearing temperatures was found to be significant. With the lower-viscosity oil [6×10^{-7} reyns (4.9 centistokes) at 100° F; viscosity index, 83], the inner-race temperatures were approximately 30° to 35° F less than those obtained with the higher-viscosity oil [53×10^{-7} reyns (42.8 centistokes) at 100° F; viscosity index, 150], whereas the outer-race-maximum temperatures were 12° to 28° F lower with the lower-viscosity oil over the DN (product of bearing bore in mm and shaft speed in rpm) range investigated in spite of the lower flows (at a given DN value) of the lower-viscosity oil.

INTRODUCTION

The lubricating and cooling function of the oil is critical in high-speed roller-bearing operation (reference 1, Gurney's discussion in reference 2, and reference 3). High-speed rolling-contact bearings used in aircraft gas-turbine engines are generally lubricated with oil flowing from a single jet directed between the bearing races. The effects of load speed, and cage design on roller-bearing operating temperatures at high speeds are described in reference 3. For minimum bearing temperatures of inner-race-riding cage-type bearings, the lubricant should be directed at the cage-locating surface and perpendicular to the bearing face (reference 4). The effects of variations in oil flow and oil-jet diameter on roller-bearing operating temperatures at high speeds are given in reference 4 for single-jet lubrication. The effect of oil inlet temperature and oil inlet distribution on roller-bearing operating temperatures at high speeds is described in reference 5; a cooling-correlation analysis, by which the test results of references 3 to 5 are generalized, is also presented in reference 5.

The data of references 3 to 5 were obtained in a test rig in which the heat-flow paths and the ambient temperatures did not exactly simulate engine conditions; a comparison of test-rig and turbojet-engine data is therefore desirable. The investigation reported herein, which is a continuation of the work reported in references 3 to 5, was conducted at the NACA Lewis laboratory in 1950 to compare the operating characteristics of the laboratory test bearing and the turbojet-engine turbine roller bearing.

Cylindrical-roller bearings currently used as turbine roller bearings in commercial aircraft gas-turbine engines were used as test bearings in this investigation. These bearings were of 75-millimeter bore (size 215), 25-millimeter width, and 130-millimeter outside diameter and were equipped with one-piece inner-race-riding brass cages. The ranges of controlled variables for the test-rig bearing were: load, 7 to 1113 pounds; *DN* (product of bearing bore in mm and shaft speed in rpm), 0.3×10^6 to 1.2×10^6; oil inlet viscosity, 1.5×10^{-7} to 53×10^{-7} pound-second per square inch (1.23 to 42.8 centistokes); oil inlet temperature, 100° to 205° F; oil flow, 0.6 to 12.9 pounds per minute. The ranges of controlled variables for the turbojet-engine bearing were: *DN*, 0.3×10^6 to 0.862×10^6; and oil inlet viscosity, 1.5×10^{-7} to 36×10^{-7} pound-second per square inch (1.23 to 24.2 centistokes).

The cooling-correlation analysis presented in reference 5 was applied to turbojet-engine bearing data in order to provide a means of comparing test-rig and engine-bearing data as well as to provide a means of estimating the engine-bearing-temperature change due to a change in such operating variables as *DN*, oil flow, oil inlet temperature, oil-jet diameter, and oil inlet viscosity.

[1] Supersedes NACA RM E51I05, "Comparison of High-Speed Operating Characteristics of Size 215 Cylindrical-Roller Bearings as Determined in Turbojet Engine and in Laboratory Test Rig," by E. Fred Macks and Zolton N. Nemeth, 1951.

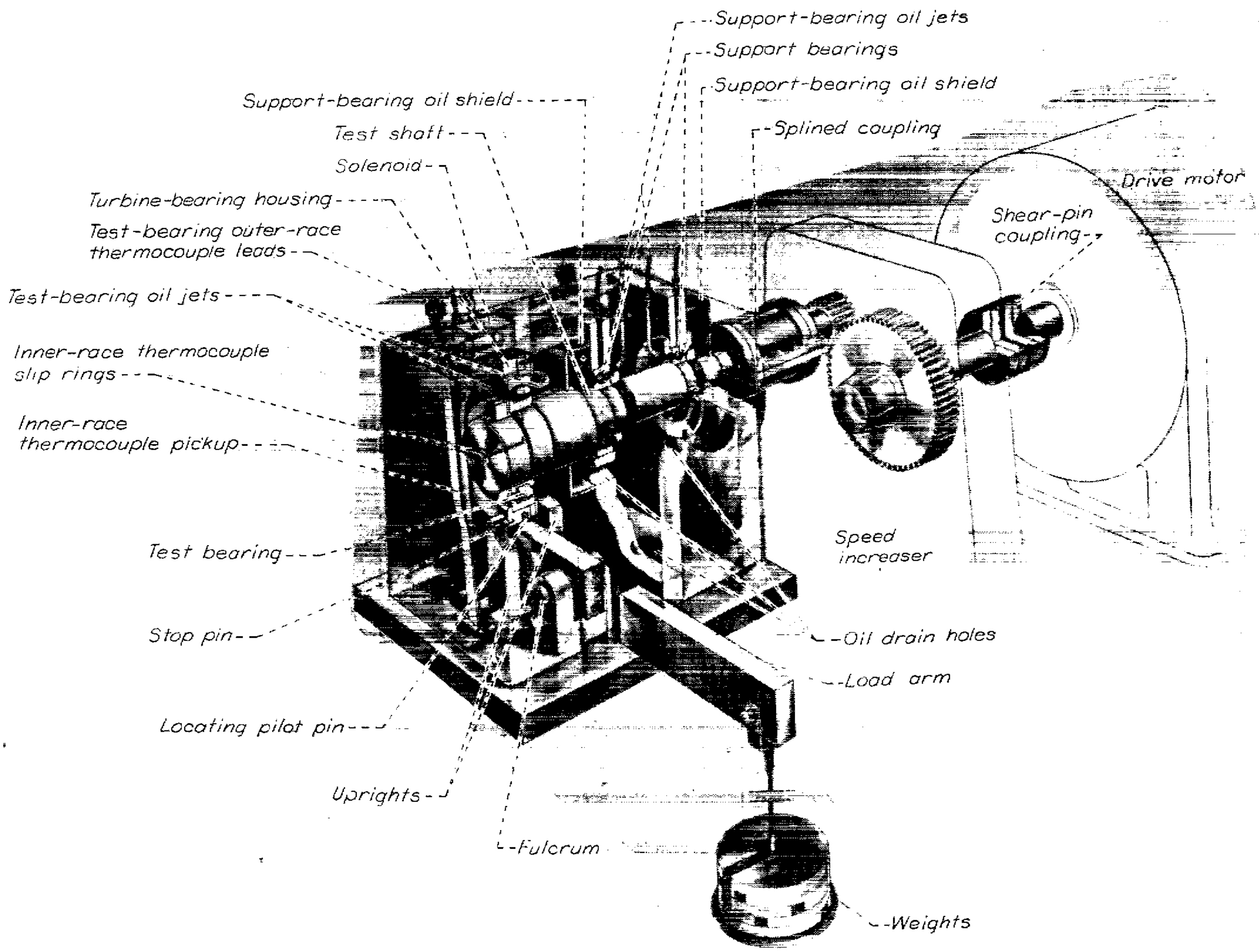


FIGURE 1.—Cutaway view of laboratory test rig.

APPARATUS

BEARING RIG

The bearing rig (fig. 1) used in this investigation is described in references 3 to 5. The bearing under investigation was mounted on one end of the test shaft, which was supported in a cantilever manner, for observation of bearing component parts and lubricant flow during operation. Radial load was applied to the experimental bearing by means of a lever and dead-weight system in such a manner that the outer race of the experimental bearing was essentially unaffected by small shaft deflections or by small shaft and load-arm misalinements.

Drive equipment.—The drive equipment, which is described in references 3 to 5, had a possible test-shaft speed range of 800 to 50,000 rpm.

Temperature measurements.—The method of temperature measurement is fully described in references 3 and 4. Briefly, for measuring outer-race test-bearing temperatures, six iron-constantan thermocouples were located at 60° intervals

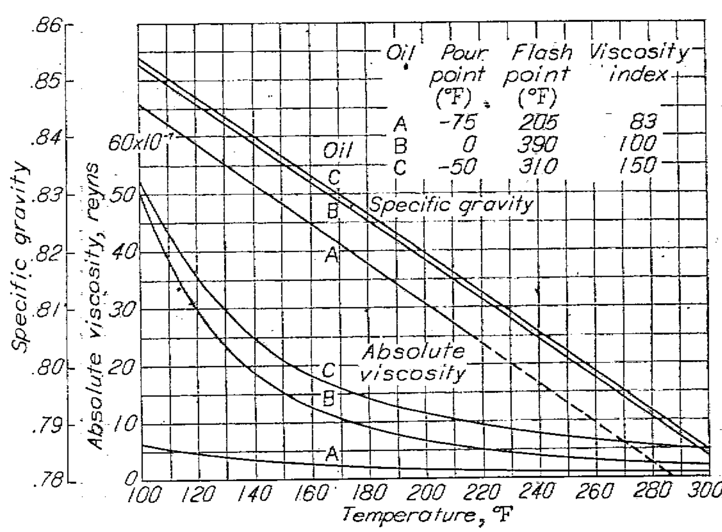


FIGURE 2.—Absolute viscosity and specific gravity of oils A, B, and C.

around the outer-race periphery at the axial center line. For measuring inner-race test-bearing temperatures, a copper-constantan thermocouple was pressed against the bore of the inner race at the axial midpoint of the test bearing, the voltage being transmitted from the rotating shaft by means of slip rings (according to the method of reference 6).

Lubrication system.—The lubrication system was the same as that described in references 3 to 5. The support bearings were lubricated in the manner described in references 3 to 5. The oil was supplied to these bearings at a pressure of 10 pounds per square inch through 0.180-inch-diameter jets. The temperature of the oil to the support bearings was the same as the temperature of the oil to the test bearing for all runs.

The properties of the lubricating oils used in the test rig (oils A and C) are given in figure 2. Oil C is the same type as that reported in references 3 to 5 and is a commercially prepared blend of a highly refined paraffin base with a small percentage of polymer added to improve the viscosity index.

TURBOJET ENGINE

The effect of engine speed on inner- and outer-race turbine roller bearing temperatures for a given oil was determined by static test-stand engine operation (fig. 3). The rear turbine bearing (bearing 11, table I) of a commercial gas-turbine engine was instrumented to obtain inner- and outer-

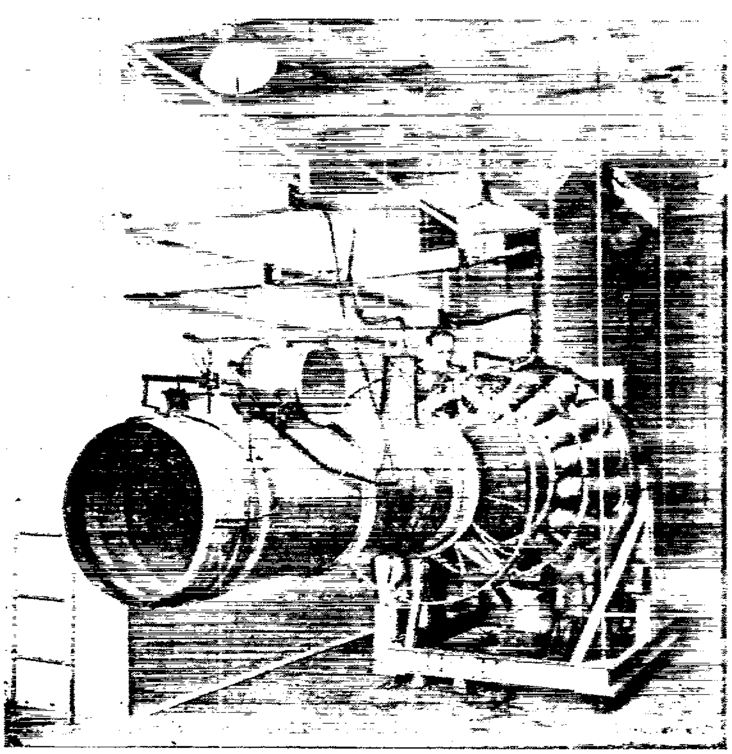


FIGURE 3.—Turbojet-engine static test-stand installation.

race bearing temperatures. Six outer-race thermocouples were located at 60° intervals around the bearing circumference. The thermocouples were mounted radially through the outer-race housing so that the sensitive element made contact with the outer race (the method used in the test rig). The inner-race bearing temperature was obtained by means of a chromel-alumel thermocouple pressed against the inner-race bore. The voltage was transmitted from the rotating shaft by means of slip rings. The brushes, through operation of a solenoid, contacted the slip rings only at the time a reading was made. All the thermocouples were mounted at the axial center line of the bearing. A special oil inlet thermocouple and pressure take-off was installed immediately before the oil-jet nozzle of the engine to obtain more accurate values of oil inlet conditions than are given by engine sump temperature and engine pressure. Ambient-air temperature in the vicinity of the test bearing (approximately ¼ in. from the outer-race bearing housing at its axial center line) was obtained by means of an oil-shielded thermocouple. This bearing was lubricated by a single oil jet of 0.052-inch diameter. The estimated static load on the bearing was 375 pounds.

The properties of the lubricating oils used in the turbojet engine are given in figure 2 (oils A, B, and C).

TEST BEARINGS

The three test bearings used in this investigation were cylindrical-roller bearings of the type currently used as turbine roller bearings of commercial aircraft gas-turbine engines. The bearing dimensions were 75-millimeter bore, 130-millimeter outside diameter, and 25-millimeter width. The bearings were equipped with inner-race-riding brass (bronze) cages. The operating conditions imposed on this bearing in engine service are: DN, 0.3×10^6 to 0.862×10^6; approximate gravity load, 375 pounds; oil flow, 0.53 to 1.98 pounds per minute. Additional physical characteristics of the test bearings are given in table I. The test bearings were measured in the manner described in references 3 to 5.

The test bearings are numbered from the first part of the high-speed investigation on 75-millimeter roller bearings; that is, bearing 10 of this report is the same as bearing 10 of reference 5.

PROCEDURE

LABORATORY TEST RIG

The effects of speed, load, oil flow, oil inlet temperature, and oil-jet diameter on inner- and outer-race test-bearing temperatures were investigated in the laboratory test rig for a given oil. For each data point, a specific operating condition was maintained until equilibrium bearing temperatures were reached. Values of DN from 0.3×10^6 to 1.2×10^6 and loads from 7 to 1113 pounds were investigated. The lubricant was supplied to the bearings under investigation through a single jet directed at the cage-locating surface and perpendicular to the bearing face. The lubricant was supplied at inlet temperatures from 100° to 205° F and flows from 0.6 to 12.9 pounds per minute through oil jets from 0.023 to 0.129 inch in diameter.

TURBOJET ENGINE

For each data point, a specific power level was maintained until equilibrium bearing temperatures were obtained. The turbojet engine was run at speeds of 4000, 6000, 8000, 10,000, and 11,000 rpm when lubricated with oils B and C; and 4000, 6000, 8000, 10,000, 11,000, and 11,500 rpm when lubricated with oil A.

The tail-cone setting was such that when the engine was run at a speed of 11,500 rpm and lubricated with oil A the tail-cone temperature (average temperature of 14 thermocouples located in a circle 2 in. inward from the inner wall at the exhaust-cone outlet) was approximately 1250° F. This same tail-cone setting was maintained at the lower speeds when oil A was used. For oils B and C, the tail-cone setting was adjusted at each speed to give approximately the tail-cone temperature that was obtained with oil A at the corresponding speed.

RESULTS AND DISCUSSION

The results of the experimental investigation are presented in figures 4 to 12. Bearing temperature was chosen as the principal criterion of operation inasmuch as, in the final analysis, temperature is an over-all indication of the effects of all the operating variables. An example of the reproducibility of results is given in reference 3.

OPERATING CHARACTERISTICS OF TEST-RIG BEARING

The operating characteristics of the test-rig bearing are described in references 3 to 5. The results were generalized by means of dimensional analysis and a cooling-correlation technique; this cooling correlation is given in reference 5 for both the inner- and outer-race bearing temperatures. The cooling-correlation curves and a discussion thereof, for both the inner- and the outer-race bearing temperatures as determined in the test rig, are given in the section entitled **"COMPARISON OF TEST-RIG BEARING AND TURBOJET-ENGINE BEARING OPERATING CHARACTERISTICS."**

OPERATING CHARACTERISTICS OF TURBOJET-ENGINE TURBINE ROLLER BEARING

Effect of speed and oil viscosity on inner- and outer-race temperatures.—The effect of *DN* on inner- and outer-race equilibrium temperatures of bearing 11, in turbojet engine on static test stand, is shown in figure 4 for oils A, B, and C. The inner- and outer-race bearing temperatures increased with an increase in *DN* at an increasing rate. The inner-race temperatures were 30° to 60° F higher than the outer-race maximum temperatures, the exact value depending on the operating condition and the oil type.

The effect of oil type on inner- and outer-race bearing temperatures was appreciable over the engine operating range. With the oil of lower viscosity [A, 6×10^{-7} reyns (4.9 centistokes) at 100° F; viscosity index, 83], the outer-race maximum temperatures were 12° to 28° F lower, and the inner-race temperatures were 30° to 35° F lower than were the respective temperatures with the oil of higher viscosity [C, 53×10^{-7} reyns (42.8 centistokes) at 100° F; viscosity index, 150].

The engine was not operated at maximum speed (11,500 rpm) for the oils of higher viscosity (B and C) inasmuch as the temperature data indicated that the equilibrium bearing temperatures would be at a critical value (above 375° F for the inner race). Experience has shown that an incipient bearing failure may occur with oil C at maximum bearing operating temperatures above 375° F (reference 3).

Effect of engine speed and oil viscosity on oil inlet temperature, oil inlet pressure, ambient-air temperature, and oil flow.—The effect of engine speed on oil inlet temperature, ambient-air temperature, oil inlet pressure, and oil flow for bearing 11 when lubricated with oils A, B, and C is shown in figure 5. The oil inlet temperature increased with *DN* at an increasing rate, there being little effect of oil viscosity on the oil inlet temperature. The oil inlet temperature varied from 113° to 212° F, depending on the operating condition.

The ambient-air temperature increased more rapidly than a linear function from a *DN* of 0.3×10^6 to 0.6×10^6, but at approximately a constant rate from a *DN* of 0.6×10^6 to the maximum engine speed (*DN*, 0.862×10^6). Little difference in ambient-air temperature was apparent for the three grades of oil investigated. The ambient-air temperature varied from 146° to 328° F, depending on the operating condition.

The oil inlet pressure increased linearly with an increase in *DN* for each of the oils. The oil inlet pressure was 1 to 4 pounds per square inch lower for the lower-viscosity oil than

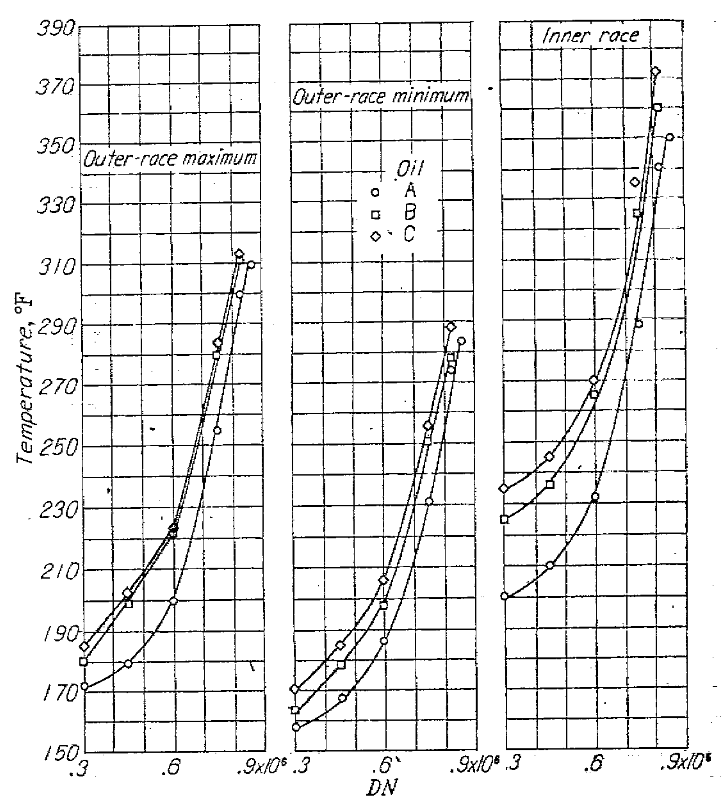
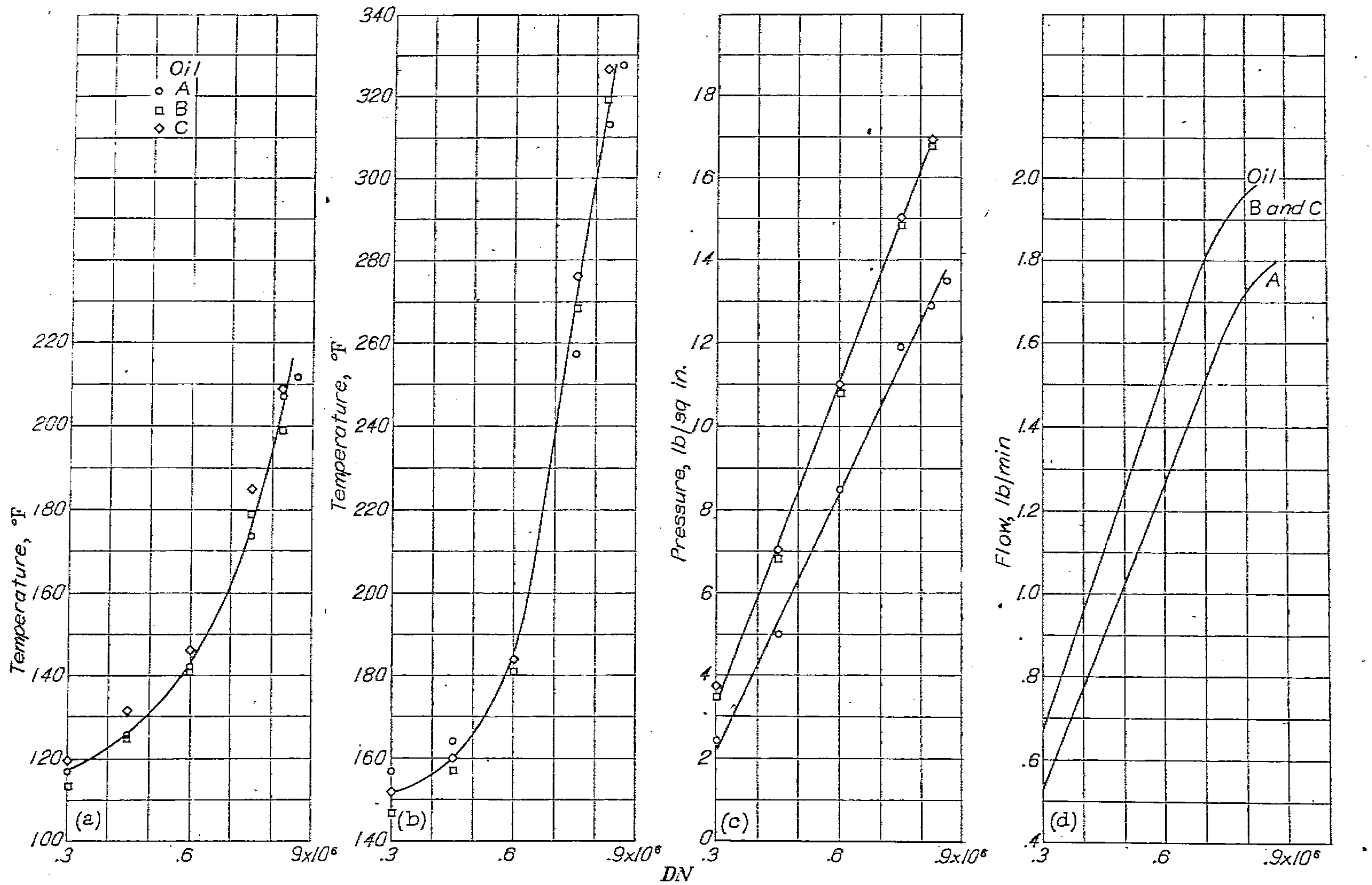


FIGURE 4.—Effect of *DN* on inner- and outer-race temperatures of bearing 11 (turbine roller bearing in turbojet engine) lubricated with three oils. *DN*, 0.3×10^6 to 0.862×10^6; oil-jet diameter, 0.052 inch; load on engine bearing, 375 pounds.



(a) Oil inlet temperature.
(b) Ambient-air temperature.
(c) Oil inlet pressure.
(d) Oil flow.

FIGURE 5.—Effect of DN on oil inlet temperature, ambient-air temperature, oil inlet pressure, and oil flow of bearing 11 (turbine roller bearing in turbojet engine) for three oils. DN, 0.3×10^6 to 0.862×10^6; oil-jet diameter, 0.052 inch; load on engine bearing, 375 pounds.

for the two higher-viscosity oils, presumably because of lower pumping efficiency with the lower-viscosity oil. The oil inlet pressures varied from 2.4 to 17 pounds per square inch, depending on the operating condition and oil used.

The oil flow increased linearly with increase in DN for each of the oils up to a DN value of 0.7×10^6 and was 0.15 to 0.29 pound per minute lower for the lighter oil than for the heavier oils. The oil flow varied from 0.53 to 1.98 pounds per minute, depending on the operating condition and the oil used. The flow curves in figure 5(d) were obtained from an oil flow pressure-calibration curve which had been determined in a setup having the previously obtained equilibrium oil-inlet-temperature and oil-pressure values. [Inasmuch as oil flow does not vary significantly with a change in viscosity for a given jet size (reference 4), oils A, B, and C have essentially the same relation of flow with pressure.]

Circumferential temperature variation.—Polar diagrams showing the outer-race circumferential temperature variation and the inner-race temperatures for bearing 11 at a DN of 0.825×10^6 are given in figure 6 for the three oils investigated. The maximum outer-race circumferential temperature variation was little affected by the oil viscosity, its magnitude being approximately 30° F for each of the oils used.

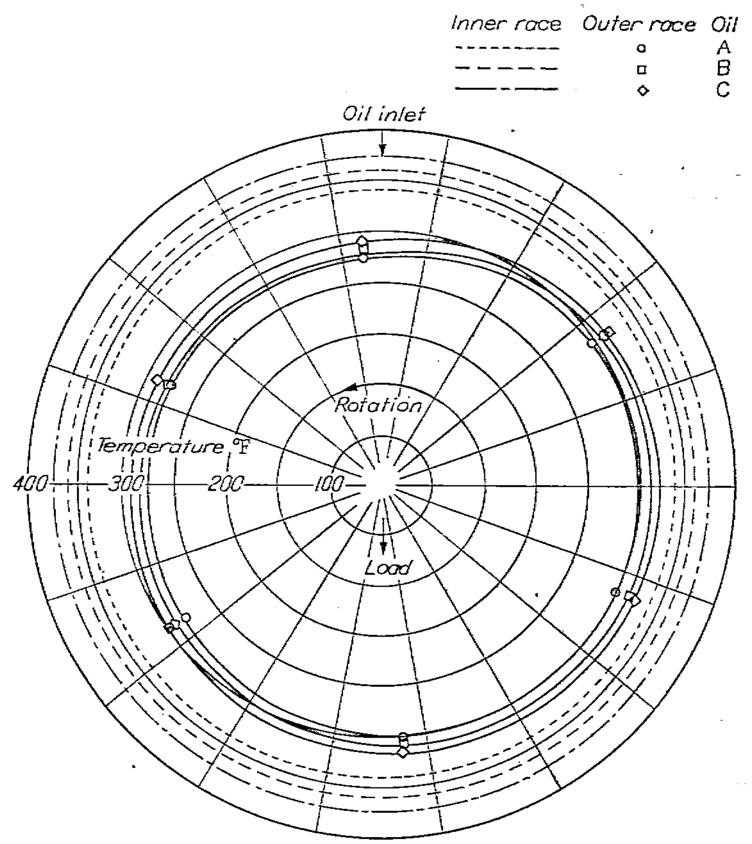


FIGURE 6.—Circumferential temperature distribution for inner and outer race of bearing 11 (turbine roller bearing of turbojet engine) for three oils. Engine speed, 11,000 rpm (DN) 0.825×10^6); oil-jet diameter, 0.052 inch; load on engine bearing, 375 pounds.

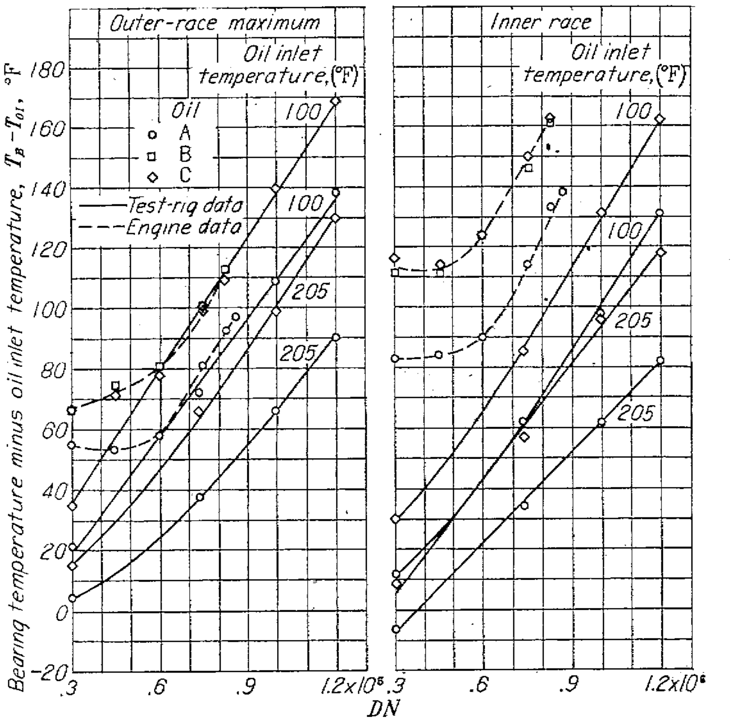


FIGURE 7.—Effect of DN on bearing temperature minus oil inlet temperature for bearing 18 (test-rig bearing) and bearing 11 (turbine roller bearing in turbojet engine) for three oils. For test rig: DN, 0.3×10^6 to 1.2×10^6; oils, A and C; oil inlet temperatures, 100° and 205° F; oil flow, 2.75 pounds per minute; oil-jet diameter, 0.050 inch; load, 368 pounds. For turbojet engine: DN, 0.3×10^6 to 0.862×10^6; oils, A, B, and C; oil inlet temperatures, 113° to 212° F; oil flow, 0.53 to 1.98 pounds per minute; oil-jet diameter, 0.052 inch; load, 375 pounds.

Bearing operating temperature above oil inlet temperature.—The effect of DN on T_B-T_{OI} (bearing temperature minus oil inlet temperature) of bearings 11 and 18 is shown in figure 7 with oil type as parameter for both the turbojet engine and the test-rig inner-race and outer-race maximum bearing temperatures.

For turbojet-engine operation, the increase in bearing temperature was approximately equal to the increase in oil inlet temperature up to a DN of 0.5×10^6. Above a DN of 0.6×10^6, T_B-T_{OI} increased rapidly. These results are in contrast to those obtained in the test rig where, for constant oil inlet temperature and constant oil flow, the inner- and outer-race bearing temperatures above the oil inlet temperature increased approximately as a straight line with an increase in DN value except at very low loads (references 3 to 5). This difference between the engine and the test-rig data is due to the variation of oil inlet temperature, oil flow, and external heat with DN for the engine bearing (fig. 5).

There was little difference in T_B-T_{OI} for the two higher-viscosity oils (B and C); however, the lower-viscosity oil resulted in an appreciably lower T_B-T_{OI} over the entire speed range. This fact indicates that the oil-cooler requirements are reduced if the engine is operated with a lighter oil.

Bearing operating temperature above ambient-air temperature.—The effect of DN on T_B-T_{AA} (bearing temperature minus ambient-air temperature in the vicinity of the test bearing) is shown in figure 8 with oil type as parameter for both the turbojet engine and the test-rig inner-race and outer-race maximum bearing temperatures.

For turbojet-engine operation, the increase in bearing temperature was approximately equal to the increase in ambient-air temperature up to a DN of about 0.6×10^6. Above a DN of 0.6×10^6, T_B-T_{AA} decreased and for the outer-race-maximum temperatures rapidly reached negative values. These data indicate how much more rapidly the bearing ambient-air temperature increased than did the bearing temperature with increase of engine power output. These results are in contrast to those obtained in the test rig where for constant oil inlet temperature and constant oil flow the inner- and outer-race bearing temperatures above the ambient-air temperature increased approximately as a straight line with an increase in DN. This difference between the engine and test-rig data is due to the variation of oil inlet temperature, oil flow, and external heat with DN for the engine bearing (fig. 5). There was not much difference in T_B-T_{AA} for the two higher-viscosity oils (B and C); however, use of the lower-viscosity oil again resulted in an appreciably lower T_B-T_{AA} over the entire speed range. It is significant to note that whereas the bearing outer race is heated by the ambient air at the higher DN values, the inner race is cooled by the ambient air over the entire speed range. The operation of the inner race at higher temperatures than the outer race in the engine, but at lower temperatures than the outer race in the test rig, is apparently due mainly to the flow of heat from the turbine wheel to the bearing inner race. This phenomenon may not be representative of all turbojet engines.

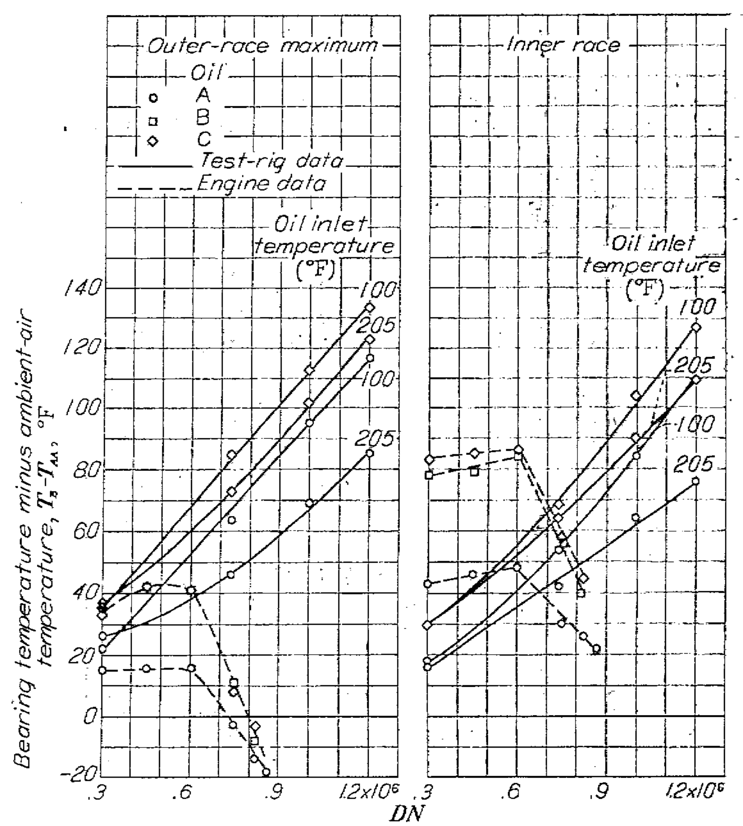


FIGURE 8.—Effect of DN on bearing temperature minus ambient-air temperature of bearing 18 (test-rig bearing) and bearing 11 (turbine roller bearing in turbojet engine) for three oils. For test rig: DN, 0.3×10^6 to 1.2×10^6; oils, A and C; oil inlet temperatures, 100° and 205° F; oil flow, 2.75 pounds per minute; oil-jet diameter, 0.050 inch; load, 368 pounds. For turbojet engine: DN, 0.3×10^6 to 0.862×10^6; oils, A, B, and C; oil inlet temperatures, 113° to 212° F; oil flow, 0.53 to 1.98 pounds per minute; oil-jet diameter, 0.052 inch; load, 375 pounds

COMPARISON OF TEST-RIG BEARING AND TURBOJET-ENGINE BEARING OPERATING CHARACTERISTICS

A comparison of the operating characteristics of the test-rig bearing and the turbine roller bearing of the jet engine must be made on as nearly an equivalent basis as is possible. Such a comparison is complicated by the facts that oil inlet temperature, oil flow, and the external heat vary with turbojet-engine speed. A generalization of the results by the application of heat-transfer parameters from a cooling-correlation analysis is used for this comparison.

THEORY

A determination of cooling-correlation parameters for high-speed bearings for which the source of heat is that generated by friction in the bearing is given in reference 5. These parameters are used in reference 5 to determine the cooling-correlation curves for both the inner- and outer-race test-rig-bearing temperatures; the derivation may be summarized as follows:

The general equation relating the significant operating variables to the operating temperature at any point on a high-speed rolling-contact bearing was obtained by dimensional analysis (reference 5):

$$\frac{C\Delta T}{(DN)^2}=\Omega\left[\frac{\rho D(DN)}{\mu}, \frac{k}{\mu C}, \frac{d}{D}, \frac{v}{DN}, \frac{W}{\mu D(DN)}, \frac{X}{D}, \frac{Y}{D}, \frac{Z}{D}, n\right] \quad (1)$$

where

C	specific heat of oil at oil inlet temperature
ΔT	temperature rise of bearing above oil inlet temperature
D	bearing bore
N	shaft speed
Ω	function
ρ	mass density of oil at oil inlet temperature
μ	viscosity based on oil inlet temperature
k	thermal conductivity of oil at oil inlet temperature
d	oil-jet diameter
v	oil inlet velocity (v is proportional to M/d^2, where M is mass flow of oil)
W	bearing load
X, Y, Z	space coordinates in temperature field
n	number of jets

A particular solution of equation (1) may be written (reference 5) for the following limiting conditions: (a) a specific point on a specific bearing, (b) a specific lubrication arrangement, (c) lubricants for which the effects of the variations of specific heat, density, and thermal conductivity are small in comparison with the effect of viscosity, and (d) bearing load between 300 and 1100 pounds. (It is shown in reference 3 that little variation occurs in bearing operating temperature over the load range 300 to 1100 pounds.)

$$\frac{\Delta T}{(DN)^a}=B\left(\frac{d^x}{M}\mu^z\right)^n \quad (2)$$

where a, B, x, z, and n are constants, the values of which are again dependent on the specific bearing system.

It is shown in reference 5 that equation (2) is applicable to both the inner- and the outer-race test-rig-bearing temperatures.

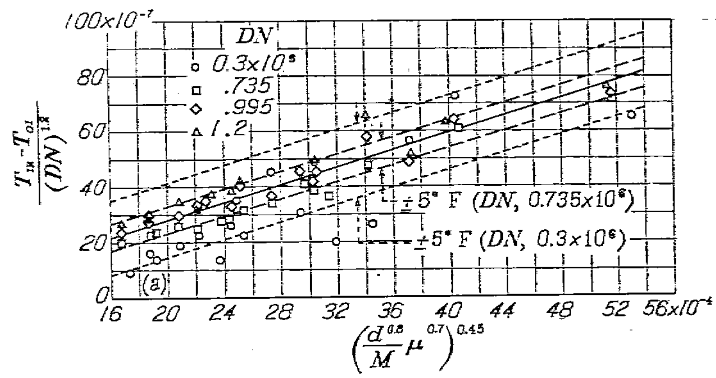
Under actual operating conditions in an engine, the turbine bearing is heated both by friction and by heat conducted from the hot engine parts and gases. The parameters given in equation (2) are used to correlate the test data obtained on an engine. Any difference that may exist between the correlation curves for the engine and the test rig is an indication of the inadequacy of the correlation parameters when external heat is a factor. A more detailed analysis of the heat-transfer process may result in more suitable correlation parameters.

APPLICATION OF THEORY

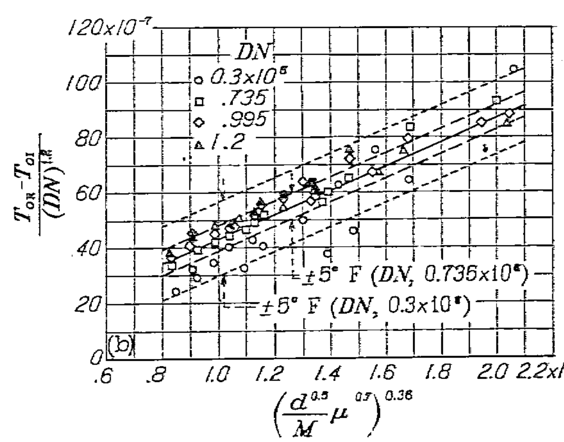
Generalization of test-rig results.—The significance of equation (2) is demonstrated in figure 9, which presents the cooling-correlation curves for the inner-race and outer-race-maximum bearing temperatures for the test-rig bearing. The method of determining the constants is described in reference 5.

The constants of equation (2) for both the inner- and the outer-race test-rig-bearing cooling-correlation curves are listed in the following table:

Constant	Inner race	Outer race
a	1.2	1.2
B	15.4×10^{-4}	4.22×10^{-4}
x	.8	.5
z	.7	.7
n	.45	.36



(a) Inner-race temperature.



(b) Outer-race-maximum temperature.

FIGURE 9.—Cooling-correlation curve as determined in test rig for temperatures of bearing 10. (Temperature T, °F; DN, bearing bore in mm times shaft speed in rpm; viscosity μ, lb-sec/sq in.; oil-jet diameter d, in.; oil flow M, lb/min.) (Data from reference 5.)

The curves of figure 9 are from reference 5 and have been reproduced herein to allow a convenient comparison to be made with the engine cooling-correlation curves which follow. The short dashed lines indicate a ±5° F deviation from each cooling curve at a DN of 0.3×10^6, whereas the long dashed lines indicate a ±5° F spread at a DN of 0.735×10^6.

The estimated range of applicability of the test-rig cooling-correlation curves is as follows: DN, 0.3×10^6 to 1.2×10^6; oil inlet temperature, 100° to 205° F; oil flow, 2 to 10 pounds per minute (in order to stay within the range of oil inlet pressures from 5 to 400 lb/sq in., the oil inlet velocity must be between 10 and 150 ft/sec as calculated from $V=0.0574\ M/d^2$); oil inlet viscosity, 1.5×10^{-7} to 53×10^{-7} reyns; oil-jet diameter, 0.023 to 0.129 inch; load, 300 to 1100 pounds.

GENERALIZATION OF TURBOJET-ENGINE-BEARING RESULTS

Operating variables.—The specific operating variables of speed, oil type, oil flow, oil inlet temperature, tail-cone temperature, and oil-jet diameter used to correlate the turbojet-engine-bearing results are given in the following table:

Operating condition	DN	Oil flow (lb/min)			Oil inlet temperature (°F)			Tail-cone temperature (°F)			Oil-jet diameter (in.)
		Oil			Oil			Oil			
		A	B	C	A	B	C	A	B	C	
1–3	0.3×10^6	0.54	0.67	0.70	117	113	120	1046	1048	1050	0.052
4–6	.45	.84	1.04	1.06	126	125	131	1015	1008	1015	.052
7–9	.6	1.24	1.51	1.54	142	141	146	1007	998	995	.052
10–12	.75	1.64	1.89	1.90	174	179	185	1120	1132	1125	.052
13–15	.825	1.74	1.96	1.99	207	199	210	1230	1225	1217	.052
16	.862	1.79	------	------	212	------	------	1250	------	------	.052

Inner-race cooling correlation.—The inner-race cooling-correlation curve for the engine bearing (fig. 10(a)) was obtained by use of equation (2) and the constants thereof as determined from the laboratory test-rig inner-race data. Although the data for the lowest speed (DN 0.3×10^6) lie far from the curve (see following table), the remaining data correlate reasonably well. This curve should therefore be used only for DN values of 0.45×10^6 and above. The additional data for figure 10(a) at a DN of 0.3×10^6 are:

$\left(\frac{d^{0.8}}{M}\mu^{0.7}\right)^{0.45}$	$\frac{T_{IR}-T_{OI}}{(DN)^{1.2}}$
71.9×10^{-4}	307×10^{-7}
78.5	299

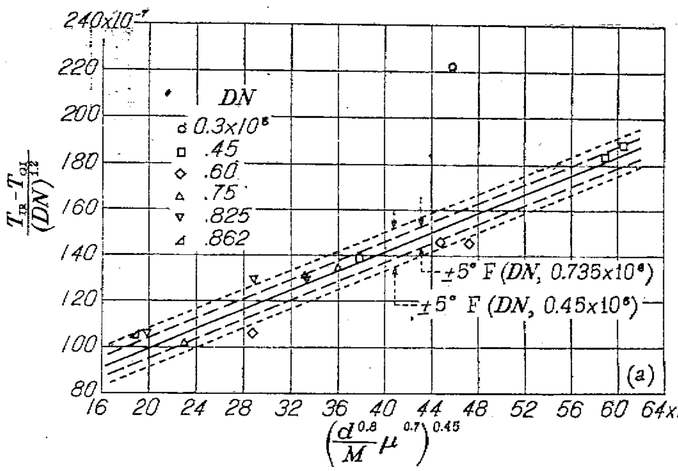
The short dashed lines indicate a ±5° F deviation from the cooling curve at a DN of 0.45×10^6, where the long dashed lines indicate a ±5° F spread at a DN of 0.735×10^6. This analysis is not the final answer to the generalization of engine-bearing data, particularly since external heat flow has not been considered. The cooling correlation is presented, however, to illustrate the extent of the usefulness of the theory as it stands as well as to enable an approximate comparison to be made of the engine and the test-rig data.

The engine correlation curves should also apply for variation in oil-jet diameter from 0.023 to 0.129 inch and for variation in oil flow to about 10 pounds per minute (reference 5).

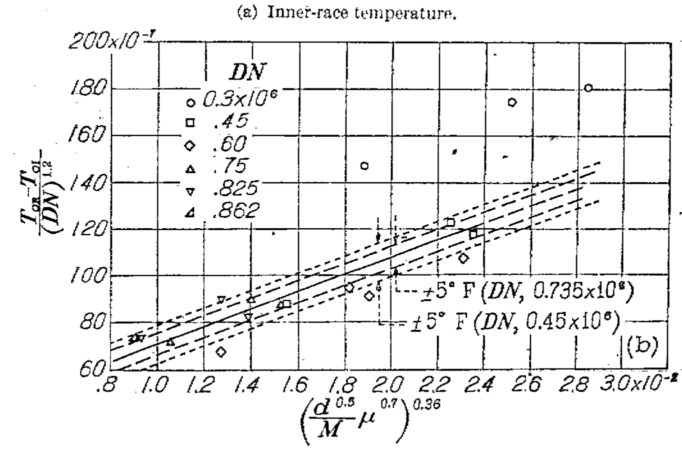
The estimated range of applicability of the engine cooling-correlation curve (fig. 10(a)) for the engine bearing investigated is as follows: DN, 0.45×10^6 to 0.86×10^6; oil inlet temperature, 110° to 210° F; oil flow, 0.5 to 10 pounds per minute (in order to stay within the range of oil inlet pressures from 5 to 400 lb/sq in., the oil inlet velocity should be between 10 and 150 ft/sec as determined by $V=0.0574\ M/d^2$); oil inlet viscosity, 1.5×10^{-7} to 36×10^{-7} reyns; oil-jet diameter, 0.023 to 0.129 inch.

Outer-race cooling correlation.—The outer-race cooling-correlation curve for the engine bearing (fig. 10(b)) was obtained by means of equation (2) and the constants thereof, as determined from the laboratory test-rig outer-race data. Although the data for the lowest speed again lie far from the curve, the remaining data correlate fairly well. As for the inner race, this curve should be used only for DN values of 0.45×10^6 and above. The short dashed and long dashed lines parallel to the cooling curve again indicate a ±5° F deviation from the cooling curve at DN values of 0.45×10^6 and 0.735×10^6, respectively.

The estimated range of applicability of the correlation in figure 10(b) is the same as that of figure 10(a).



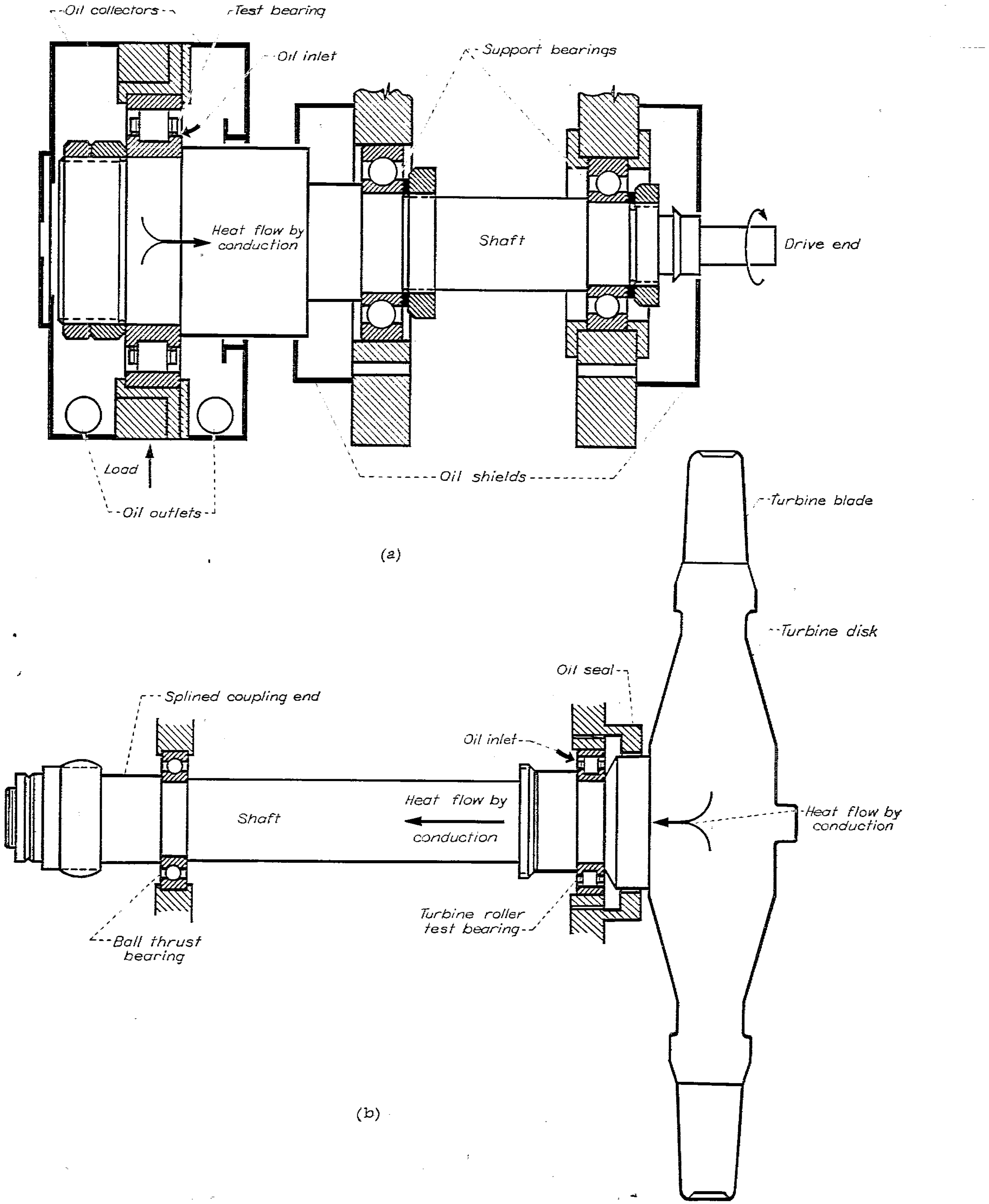
(a) Inner-race temperature.



(b) Outer-race-maximum temperature.

FIGURE 10.—Cooling-correlation curve for temperatures of bearing 11 (turbine roller bearing in turbojet engine). (Temperature T, °F; DN, bearing bore in mm times shaft speed in rpm; viscosity μ, lb-sec/sq in.; oil-jet diameter d, in.; oil flow M, lb/min.)

The inner- and outer-race cooling-correlation curves were utilized in making a comparison of the bearing operating characteristics as determined in the laboratory test rig and in the turbojet engine. A schematic diagram of the laboratory test-rig and turbojet-engine-bearing arrangements is given in figure 11 for purposes of comparison.



(a) Laboratory test rig.
(b) Turbojet-engine wheel and shaft.
FIGURE 11.—Schematic diagram of laboratory test-rig and turbojet-engine bearing arrangements.

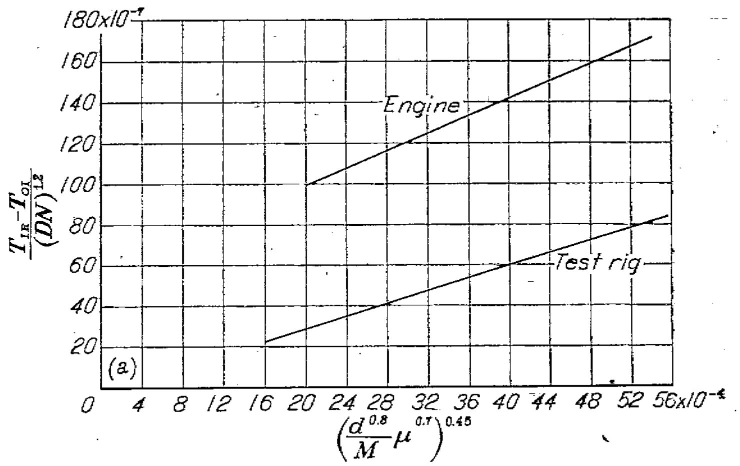
Inner-race comparison.—The inner-race cooling-correlation curves for the test-rig data and the engine data are given in figure 12(a) over the range of operating variables investigated. The cooling curve for the engine data lies appreciably above the cooling curve for the test-rig data. The slope of the cooling curve for the engine data is 36.6 percent greater than the slope of the curve for the test-rig data; the values of slope are 21.2×10^{-4} and 15.4×10^{-4}, respectively. The differences in the inner-race operating characteristics of the test rig and the engine bearing are mostly due to the flow of heat from the bearing inner race out through the shaft in the case of the test rig as contrasted to a flow of heat through the shaft to the bearing inner race from the turbine disk in the case of the engine (as shown in fig. 11). The test-rig and the engine inner-race temperature rise ΔT for two values of the variable $\left(\frac{d^{0.8}}{M}\mu^{0.7}\right)^{0.45}$ are given in the following table for DN values of 0.5×10^6, 1.0×10^6, and 1.5×10^6:

$\left(\frac{d^{0.8}}{M}\mu^{0.7}\right)^{0.45}$	$\frac{T_{IR}-T_{OI}}{(DN)^{1.2}}$		$T_{IR}-T_{OI}$ (°F)					
	Test rig	Engine	Test rig			Engine		
			DN			DN		
			0.5×10^6	1.0×10^6	1.5×10^6	0.5×10^6	1.0×10^6	1.5×10^6
20×10^{-4}	28.0×10^{-7}	99.5×10^{-7}	20	44	73	69	158	256
54	80.5	171.5	56	127	208	118	272	441

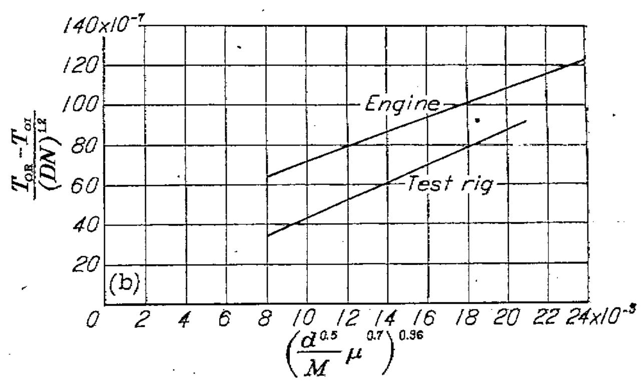
From the foregoing comparison, it is evident that the engine bearing operated at about 3.5 times the value of inner-race ΔT of the test-rig bearing at a $\left(\frac{d^{0.8}}{M}\mu^{0.7}\right)^{0.45}$ of 20×10^{-4} and at about twice the value of inner-race ΔT of the test-rig bearing at a $\left(\frac{d^{0.8}}{M}\mu^{0.7}\right)^{0.45}$ equal to 54×10^{-4}. It is evident that the engine bearing would overheat ($T_{IR}-T_{OI}=272°$ F) if operated to a DN of 1.0×10^6 with $\left(\frac{d^{0.8}}{M}\mu^{0.7}\right)^{0.45}$ equal to 54×10^{-4}; however, if this variable was set equal to 20×10^{-4} the cooling curve indicates that the engine bearing would operate at the more reasonable value of $T_{IR}-T_{OI}$ equal to approximately 158° F.

Outer-race comparison.—The outer-race cooling-correlation curves for the test-rig and the engine data are given in figure 12(b) over the range of operating variables investigated. The cooling curve for the engine data lies slightly above the cooling curve for the test-rig data. In contrast to the results for the inner race, the slope of the outer-race cooling curve for the engine data is 17 percent less than the slope of the curve for the test-rig data, these values of slope being 3.66×10^{-4} and 4.42×10^{-4}, respectively. The differences in level and slope of the outer-race cooling curves for the engine and the test-rig data were due primarily to the external heat associated with engine operation.

A comparison of the test-rig and the engine outer-race maximum temperature rise for two values of the variable $\left(\frac{d^{0.5}}{M}\mu^{0.7}\right)^{0.36}$ are given in the following table for DN values of 0.5×10^6, 1.0×10^6, and 1.5×10^6:



(a) Inner-race temperature.



(b) Outer-race temperature.

FIGURE 12.—Comparison of cooling-correlation curves for temperatures of test-rig and engine bearings.

$\left(\frac{d^{0.5}}{M}\mu^{0.7}\right)^{0.36}$	$\frac{T_{OR}-T_{OI}}{(DN)^{1.2}}$		$T_{OR}-T_{OI}$ (°F)					
			Test rig			Engine		
			DN			*DN*		
	Test rig	Engine	0.5×10^6	1.0×10^6	1.5×10^6	0.5×10^6	1.0×10^6	1.5×10^6
8×10^{-3}	34.5×10^{-7}	64.0×10^{-7}	24	55	89	44	102	165
20	87.5	108.0	60	139	226	74	171	278

It is evident from the foregoing comparison that the engine bearing operated at about 1.8 times the value of outer-race ΔT of the test-rig bearing at a $\left(\frac{d^{0.5}}{M}\mu^{0.7}\right)^{0.36}$ of 8×10^{-3} and at about 1.3 times the value of $T_{OR}-T_{OI}$ of the test-rig bearing at a $\left(\frac{d^{0.5}}{M}\mu^{0.7}\right)^{0.36}$ of 20×10^{-3}. It is indicated that the engine bearing outer-race-maximum temperature rise may be reduced from 171° to 102° F at a *DN* of 1.0×10^6 merely by reducing the value of $\left(\frac{d^{0.5}}{M}\mu^{0.7}\right)^{0.36}$ from 20×10^{-3} to 8×10^{-3}.

The foregoing comparisons of the test-rig and the engine data illustrate the usefulness of generalizing the results of relatively few operating conditions. It should be mentioned here that whereas the magnitudes of temperature rise may be approximately estimated by means of the foregoing generalized results, another use of the correlation lies in the prediction of the trend effect of various combinations of the operating variables.

It has thus been illustrated how the engine designer can predict, in a general fashion, the bearing operating temperatures for increased power conditions of present or future turbojet engines. Just what variables are significant and the approximate relative importance of each variable in controlling the bearing temperature are also indicated by equation (2) and the constants involved.

SUMMARY OF RESULTS

The following results were obtained in an experimental investigation that was conducted to determine the effects of several operating variables on the operating characteristics of 75-millimeter-bore one-piece inner-race-riding cage-type roller bearings, determined in a laboratory test rig and in a turbojet engine over a range of high speeds; generalizations and comparisons of the data were made by means of a cooling-correlation analysis:

1. Inner- and outer-race cooling-correlation curves were obtained for the turbojet-engine turbine roller bearing with the same inner- and outer-race correlation parameters and exponents as were determined for the laboratory test-rig bearing.

2. The relation between the operating variables and the bearing temperature rise above oil inlet temperature T_B-T_{OI} for either the test-rig or the engine bearing has the following form:

$$\frac{\Delta T}{(DN)^a}=B\left(\frac{d^x}{M}\mu^z\right)^n$$

where *DN* is the product of the bearing bore in millimeters and the shaft speed in rpm, d is the oil-jet diameter in inches, M is the oil flow in pounds per minute, and μ is the oil inlet viscosity in pound-seconds per square inch; the constants are given in the following table:

Constant	Laboratory test-rig bearing		Turbojet-engine bearing	
	Inner race	Outer race	Inner race	Outer race
a	1.2	1.2	1.2	1.2
B	15.4×10^{-4}	4.22×10^{-4}	21.2×10^{-4}	3.66×10^{-4}
x	.8	.5	.8	.5
z	.7	.7	.7	.7
n	.45	.36	.45	.36

3. The engine turbine roller bearing operated at about 3.5 times the value of inner-race temperature rise above oil inlet temperature of the test-rig bearing at $\left(\frac{d^{0.8}}{M}\mu^{0.7}\right)^{0.45}$ of 20×10^{-4} and at about twice the value of inner-race temperature rise above oil inlet temperature of the test-rig bearing at a $\left(\frac{d^{0.8}}{M}\mu^{0.7}\right)^{0.45}$ equal to 54×10^{-4}.

4. The engine turbine roller bearing operated at about 1.8 times the value of outer-race temperature rise above oil inlet temperature of the test-rig bearing at a $\left(\frac{d^{0.5}}{M}\mu^{0.7}\right)^{0.36}$ of 8×10^{-3} and at about 1.3 times the value of outer-race temperature rise above oil inlet temperature of the test-rig bearing at a $\left(\frac{d^{0.5}}{M}\mu^{0.7}\right)^{0.36}$ of 20×10^{-3}.

5. The difference in the cooling-correlation curves for the laboratory test rig and the turbojet engine was mainly due to the different heat-flow paths to and from the bearings under consideration.

6. The inner-race temperatures of the turbojet-engine turbine roller bearing were 30° to 60° F greater than the outer-race-maximum temperatures, the exact value depend-

ing on the operating condition and oil viscosity; this was in contrast to the laboratory test-rig results, for which the inner-race temperatures were less than the outer-race-maximum temperatures.

7. The effect of oil viscosity on inner- and outer-race turbojet-engine roller-bearing temperatures was significant. With the lower-viscosity oil [6×10^{-7} reyns (4.9 centistokes) at 100° F; viscosity index, 83], the inner-race temperatures were approximately 30° to 35° F less than those obtained with the higher-viscosity oil [53×10^{-7} reyns (42.8 centistokes) at 100° F; viscosity index, 150], whereas the outer-race-maximum temperatures were 12° to 28° F lower with the lower-viscosity oil over the *DN* range investigated in spite of the lower flows (at a given *DN* value) of the lower-viscosity oil.

8. The maximum outer-race circumferential temperature variation of the turbojet-engine turbine roller bearing was approximately 30° F for each of the oils used.

9. In contrast to the test-rig results, the engine-bearing inner- and outer-race temperatures increased with *DN* more rapidly than a linear function.

LEWIS FLIGHT PROPULSION LABORATORY
NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS
CLEVELAND, OHIO, *September 1, 1951*

REFERENCES

1. Dawson, J. G.: Lubricating Problems of the Gas Turbine Engine. Shell Aviation News, no. 133, July 1949, pp. 14–22.
2. Wilcock, Donald F., and Jones, Frederick C.: Improved High-Speed Roller Bearings. Lubrication Eng., vol. 5, no. 3, June 1949, pp. 129–133; discussion, vol. 5, no. 4, Aug. 1949, p. 184.
3. Macks, E. Fred, and Nemeth, Zolton N.: Investigation of 75-Millimeter-Bore Cylindrical Roller Bearings at High Speeds. I—Initial Studies. NACA TN 2128, 1950.
4. Macks, E. Fred, and Nemeth, Zolton, N.: Investigation of 75-Millimeter-Bore Cylindrical Roller Bearings at High Speeds. II—Lubrication Studies—Effect of Oil-Inlet Location, Angle, and Velocity for Single-Jet Lubrication. NACA TN 2216, 1950.
5. Macks, E. Fred, and Nemeth, Zolton N.: Lubrication and Cooling Studies of Cylindrical-Roller Bearings at High Speeds. NACA Rep. 1064, 1952. (Supersedes NACA TN 2420.)
6. Tarr, Philip, R.: Methods for Connection to Revolving Thermocouples. NACA RM E50J23a, 1951.

TABLE I—PHYSICAL CHARACTERISTICS OF TEST BEARINGS

Bearing	10		11		18	
Construction	One-piece inner-race-riding cage		One-piece inner-race-riding cage		One-piece inner-race-riding cage	
Number of rollers	18		18		18	
Length-diameter ratio of roller	1		1		1	
Pitch diameter of bearing, in.	4.036		4.036		4.036	
	Before	After	Before	After	Before	After
Total running time, hr	0	86.8	0	25.5	Unknown	86.5
[a] Severity factor	0	3.22×10^5	0	1.80×10^5	Unknown	6.28×10^5
Roller diameter, in.	[b] 0.5513	--------	[b] 0.5513	0.5509	[b] 0.5513	0.5513
Roller length, in.	[b] 0.5510	--------	[b] 0.5510	0.5507	[b] 0.5510	0.5500
Diametral clearance between cage and roller, in.	[b] 0.0087	--------	[b] 0.0087	0.0065	[b] 0.0087	0.006
Axial clearance between roller and inner-race flange, in.	[b] 0.002	--------	[b] 0.002	0.001	[b] 0.002	0.003
Axial clearance between roller and cage, in.	[b] 0.006	--------	[b] 0.006	0.007	[b] 0.008	0.008
Unmounted bearing:						
[c] Diametral clearance, in.						
Bearing	0.0018	--------	0.0017	[d] 0.0014	0.0018	[d] 0.0015
Cage	.013	--------	.013	.012	.014	.0215
[e] Eccentricity, in.	.0001	--------	.0003	.0004	.0001	.0000
Mounted bearing:						
[f] Diametral clearance, in.						
Bearing	0.0007	0.0008	--------	0.0010	0.0004	[d] 0.0004
Cage	.013	--------	--------	--------	.013	.0205
[g] Eccentricity, in.	--------	.0006	--------	--------	--------	--------
Remarks:	Satisfactory operation. Shut off oil and ran bearing to incipient failure.		Satisfactory operation in turbojet engine.		Satisfactory operation in test rig. Slight wear on cage-locating surfaces.	

[a] Summation of products of difference between equilibrium bearing temperature and oil inlet temperature for each operating condition and corresponding operating time in min at that particular condition.
[b] Measurements obtained from sample bearing.
[c] Measurement obtained in fixture with dial gage.
[d] Diametral clearance actually decreased because of apparent growth of inner race.
[e] Measurement obtained in fixture with dial gage, inner race rotating and outer race stationary.
[f] Measurements obtained as mounted in test rig with dial gage.
[g] Measurements obtained as mounted in test rig with dial gage with inner race rotating and outer race stationary.