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Lewis Research Center

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SUMMARY

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Minimum-oil-flow tests were conducted with 30-millimeter-bore, deep-groove ball bearings over a DN range (product of bearing bore in mm and shaft speed in rpm) of 0.825×10^6 to 0.975×10^6 and a temperature range of 170° to 400° F at a thrust load of 265 pounds with naphthenic-mineral-oil - air mist lubrication.

Minimum required oil flow for continuous operation increased with increasing DN and bearing temperature. This increase agreed with previously published data for 75-millimeter-bore bearings.

A generalized correlation of the data obtained herein and with the 75-millimeter-bore bearing indicates that minimum required oil flow can be expressed as a single function of bore size, load, DN, and bearing temperature. The correlation was developed for 30- and 75-millimeter-bore bearings at thrust loads of 265 to 3000 pounds, DN values of 0.6×10^6 to 0.975×10^6, and temperatures of 225° to 500° F.

Airflow, which was controlled independently of oil flow, was found to affect the minimum required oil flow at specific operating conditions. Airflow apparently affects the efficiency with which the bearing can utilize the oil droplets fed to it, and there appears to be an optimum mist velocity for a given DN or rotative speed.

Author

INTRODUCTION

The extension of bearing operating temperatures in supersonic aircraft and space vehicle applications has raised questions about the feasibility of using organic lubricants because of limitations in their oxidative and thermal stabilities. The extent of lubricant degradation is a function of the availability of oxygen and of the time and the temperature to which the lubricant is exposed. Since thermal and oxidative degradation are functions of time and temperature, the use of a "once-through" lubrication system may be advantageous in some high-temperature applications. Lubricant degradation could be minimized by exposing the lubricant to the high temperature for the shortest possible time.

Since the lubricant is discarded after one pass in a once-through system, economy of use becomes of paramount importance. Thus, the oil-flow requirements of a bearing must be known before the economic feasibility of a once-through system for a specific application can be evaluated.

Some work has been done to define quantitatively the actual oil-flow requirements of ball bearings. References 1 and 2 report some data on the relation of running time to failure as a function of the quantity of oil, applied prior to the test, for 30- and 50-millimeter-bore ball bearings. In reference 2, steady-flow minimum oil requirements were obtained from an analysis of the experimental data; these data were used to estimate quantitatively the required bleed rate of greases. Some data on minimum-oil-flow requirements of 25-millimeter-bore ball bearings at a DN (product of bearing bore in mm and shaft speed in rpm) value of 0.9×10^6 are reported in reference 3.

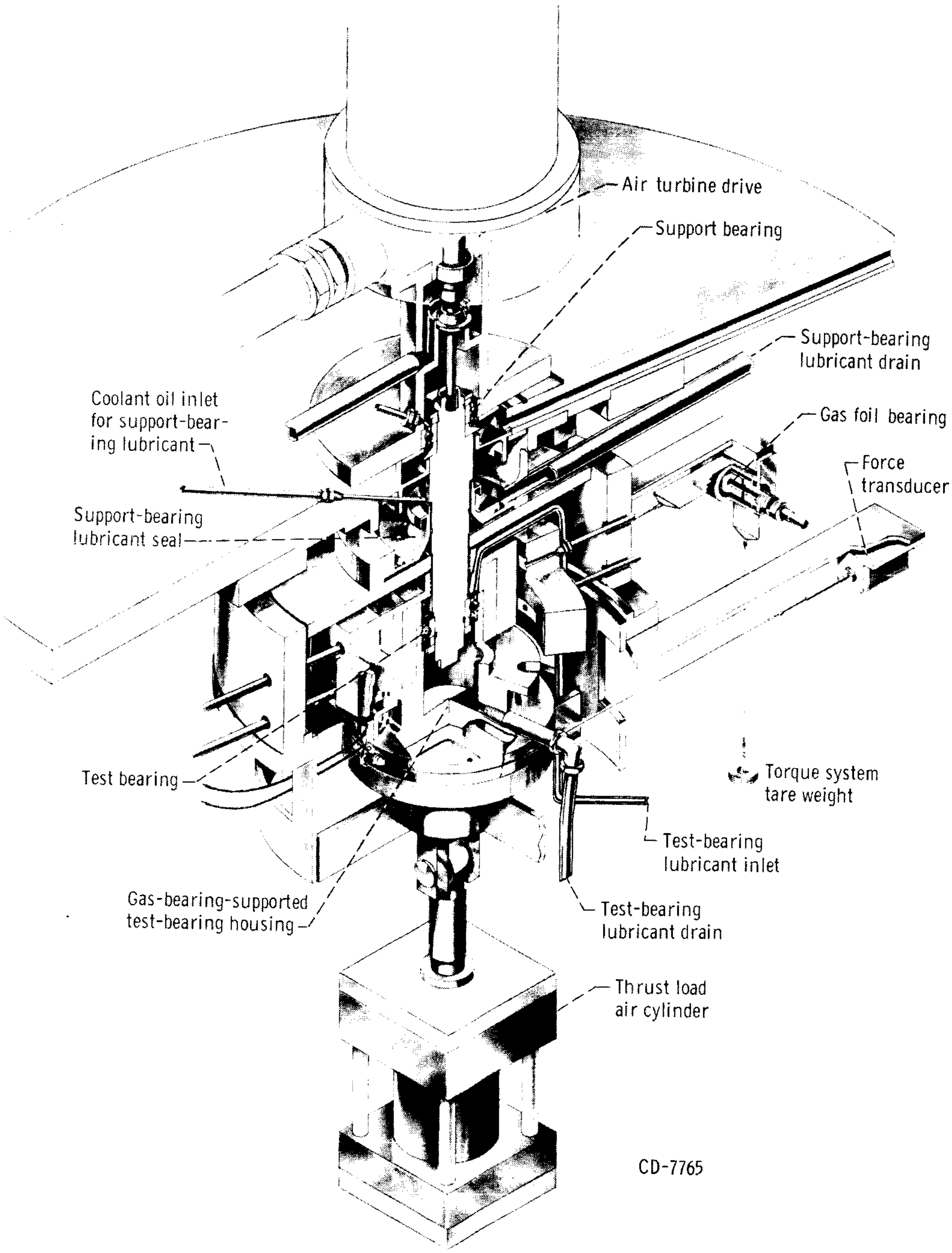
References 4 and 5 report extensive minimum-oil-flow data with 75-millimeter-bore ball bearings over a wide range of temperatures, loads, and DN values with two oils. In reference 4, an MIL-L-7808 diester oil was used in tests conducted at bearing temperatures to 500° F, DN values to 0.975×10^6, and thrust loads to 3000 pounds. Reference 5 contains data obtained with a highly refined naphthenic mineral oil at temperatures to 800° F, DN values to 1.2×10^6, and thrust loads to 3000 pounds. From references 4 and 5, data are available for a broad range of temperatures and loads for 75-millimeter-bore bearings. Therefore, obtaining minimum-oil-flow data with a second size bearing seemed advisable so that the effect of bearing size could be assessed.

The object of this investigation was to obtain minimum-oil-flow data for 30-millimeter-bore, deep-groove ball bearings and to attempt to correlate these data with those of references 4 and 5. Data were obtained over a DN range of 0.825×10^6 to 0.975×10^6, a temperature range of 170° to 400° F, a thrust load of 265 pounds with a naphthenic mineral oil lubricant. A thrust load of 265 pounds on a 206-size (30-mm-bore) bearing results in a ratio of load capacity to equivalent load equal to that for a 215-size (75-mm-bore) bearing under a 1000-pound thrust load.

APPARATUS

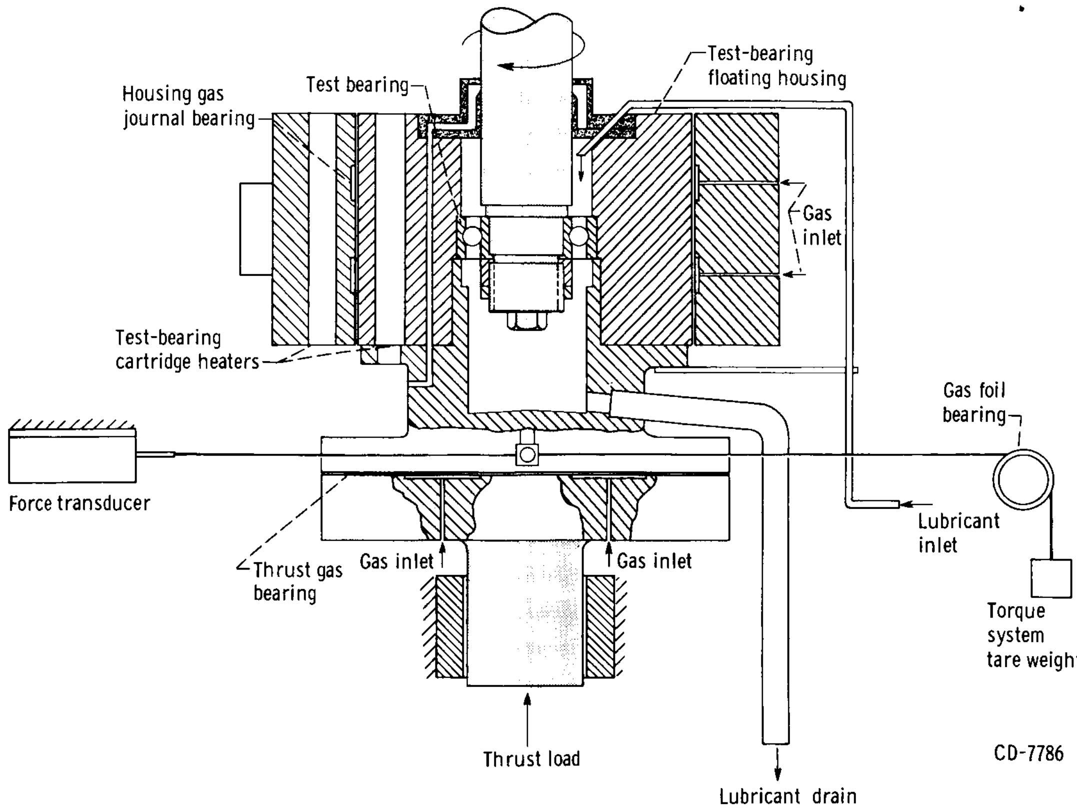
Test Apparatus

A cutaway view of the test apparatus and a cross section of the test bearing installation are shown in figure 1. Figure 1(a) illustrates the configuration of the test apparatus. The test shaft was vertically mounted with a support ball bearing at its upper end



(a) Cutaway view.

Figure 1. - Test apparatus.



(b) Detailed view of test-bearing installation.

Figure 1. - Concluded. Test apparatus.

and the test bearing at its lower end. The test shaft was driven by a 4-inch air turbine through a quill shaft. Maximum speed capability of the test apparatus was about 60 000 rpm.

The test bearing was located in a housing which was floated in a pressurized gas journal bearing (fig. 1(b)). Thrust load, applied by pressurizing a pneumatic cylinder, was transmitted to the test bearing housing through a pressurized flat-plate gas thrust bearing. The thrust load reaction was taken by the upper support bearing.

Test bearing torque was measured by a force transducer (a wire-wound resistance strain gage), which was connected to the periphery of the test bearing housing assembly. Torque was recorded continuously on a photoelectric potentiometer. Since the test bearing housing floated freely in the pressurized gas journal and thrust bearings, the test bearing torque could be measured directly by the force transducer.

Temperature Control and Measurement

A ring of eight equally spaced cartridge heaters was located in the test bearing housing. These heaters were used to supply heat to the test bearing to maintain the desired

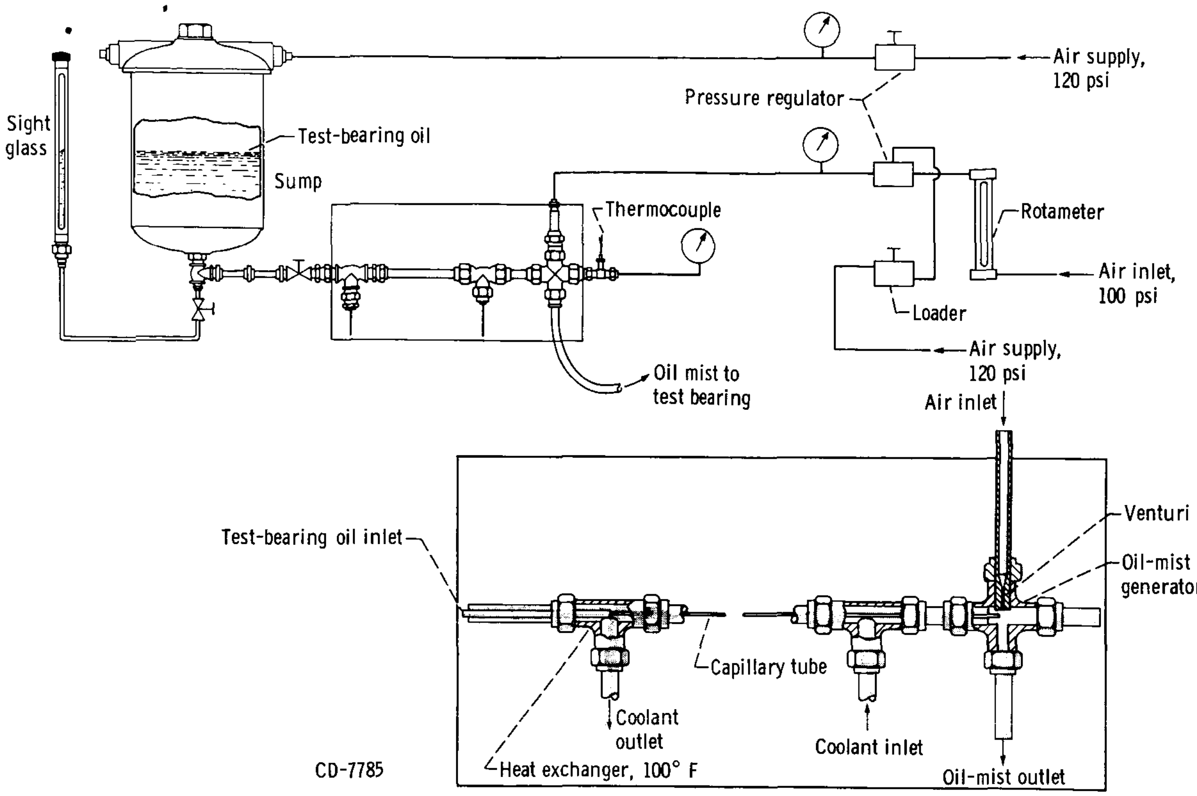


Figure 2. - Oil-gas mist supply system.

temperature. A thermocouple located at the test bearing outer race supplied a millivolt signal to a control instrument, which automatically maintained the bearing outer housing at a preset temperature. An additional ring of eight cartridge heaters was located in the frame around the test bearing housing. These heaters could be used to supply additional heat to the test bearing. Two additional thermocouples located at the test bearing outer race were used to measure and to record test bearing temperature.

Lubrication Systems

The test bearing was lubricated with an air-oil mist in all tests reported herein. The oil used was a highly refined naphthenic mineral oil designated as either MLO 7243 or MLO 7277 (both designations refer to the same fluid; viscosity, 79 cS at 100° F and 8.4 cS at 210° F, ref. 6). Oil in mist form was used to maintain a steady flow at extremely low oil-flow rates (flows below 1 g/min). The oil-mist generator used in this investigation is shown in figure 2. The oil mist was generated by a controlled flow of high-velocity air which atomized a metered flow of oil (ref. 7). Metering of the oil was accomplished by forcing it through a calibrated capillary tube held at constant temperature. Oil flowed from the pressurized sump through the capillary, which passed through a heat exchanger, to the exit region of a Venturi (fig. 2). The oil flowing through the capillary was held at 100° F by the heat exchanger. Oil flow was controlled by regulating the pressure drop across the capillary.

Airflow was controlled independently of oil flow by regulating the pressure drop across the Venturi. Airflow was measured by a rotameter.

This type of mist generator is extremely versatile because a wide range of oil flows (of the order of 10^{-5} to 10^{-2} lb/min) can be obtained by changing the capillary geometry and because the oil flow and airflow can be controlled independently. High-temperature mists could also be generated by including an oil heat exchanger of sufficient capacity and by preheating the air.

The upper support ball bearing was jet lubricated with approximately 100 grams per minute of SAE 10 oil. The turbine-shaft ball bearings and the spline couplings were lubricated with the same SAE 10 oil at lower flow rates. The support bearing oil was recirculated and cooled.

Oil draining from the upper support ball-bearing cavity was prevented from flowing down the shaft to the test bearing by a series of slingers, a close clearance seal, and a flow of cool oil that was introduced to help condense oil vapors.

Test Bearings

All the test bearings in this investigation were 206-size deep-groove ball bearings manufactured to ABEC 5 tolerances. Races and balls were of consumable-electrode vacuum-melted M-2 tool steel. Outer-race-riding machined retainers of fully annealed M-2 tool steel were used. The average diametral clearance of the bearings was 0.0015 inch and the average axial clearance 0.012. The conformity of both races was 55 percent.

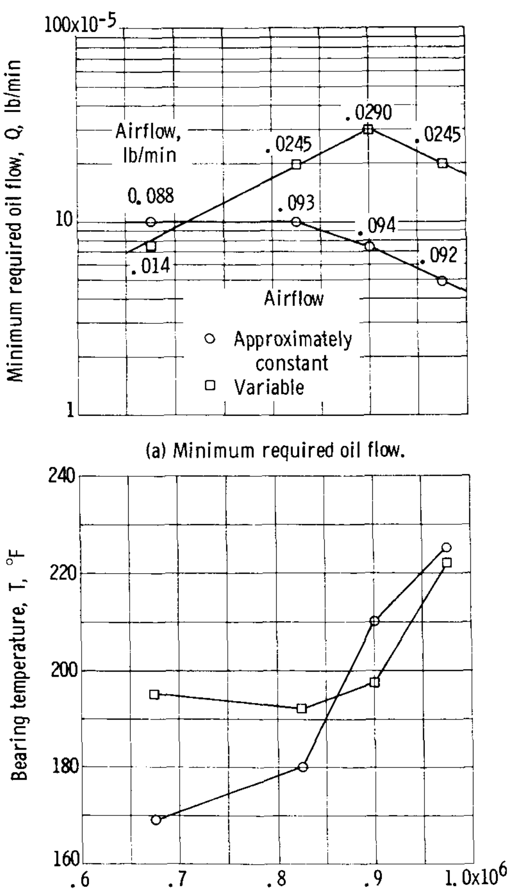
PROCEDURE

In each test the bearing was run at speeds of 5000, 10 000, 15 000, and 20 000 rpm, and then the bearing was run at speed increments of 2500 rpm until the desired speed was reached. The bearing was run at each speed without external heat addition until temperature equilibrium was reached (approximately 10 to 15 min). The oil flow at each operating speed was maintained considerably above the minimum required flow.

When the desired test speed was reached, the bearing temperature was brought up to the desired level by external heat addition. The test bearing oil flow was then reduced hourly until bearing torque increased sharply.

The test apparatus shut down automatically when a sharp increase in test bearing torque occurred; this prevented prolonged running under marginal lubrication conditions that would seriously damage the test bearing. After an incipient failure, the test bearing was allowed to cool down. It was then run-in at reduced temperature and 10 000 rpm with a high oil flow. Test bearing torque level after run-in was used as a criterion to determine whether or not permanent damage had occurred during an incipient failure. The automatic shutoff system proved to be quite reliable because no detectable test bearing damage occurred in the tests reported herein.

The airflow used to supply the air-oil mist to the test bearing was found to have a slight effect on the minimum required oil flow at a specific load, speed, and temperature. Therefore, airflows were adjusted to match those used in references 4 and 5, since attempts would be made to correlate the data reported herein with those reported in references 4 and 5. Oil flows in these experiments varied from 0.00005 to 0.003 pound per minute and airflows from 0.014 to 0.034 pound per minute.



(a) Minimum required oil flow.

(b) Bearing outer-race temperature.

Figure 3. - Effect of DN on minimum required oil flow and bearing outer-race temperature for constant and variable airflow.

RESULTS AND DISCUSSION

The results of this investigation are presented in figures 3 to 7. The effects of DN and temperature on the minimum-oil-flow requirements of 30-millimeter-bore, deep-groove ball bearings were determined at a thrust load of 265 pounds. A general correlation of these data and similar data obtained with 75-millimeter-bore ball bearings is presented.

Effect of Airflow

A series of exploratory tests without heat addition to the bearing was conducted to determine approximately the minimum required oil flow over a range of DN values. The results obtained are illustrated in figure 3(a). The circles represent the initial tests, which were conducted using airflows in the range 0.088 to 0.094 pound per minute. In the light of the results obtained in references 4 and 5, which showed that minimum required oil flow increased with DN, the trend shown by the circles was quite unexpected. Airflow was then adjusted to match approximately the airflows used in the tests of references 4 and 5. These results are given by the squares in figure 3(a). The trend here indicates an increase in the minimum required oil flow at DN values from 0.675×10^6 to 0.9×10^6, but then a decrease in going from 0.9×10^6 to 0.975×10^6. These results indicate that airflow affects the efficiency with which the bearing can utilize the oil droplets that are directed at its face. It is conceivable that, for a given bearing design and rotative speed, there is an optimum mist velocity which will result in maximum utilization. If the velocity is too high, some of the oil droplets may get through the bearing without striking any surface (fixed or moving). As the rotative speed or DN value increase, two things happen:

(1) The ability of the rotating parts to catch oil droplets increases, but their ability to throw oil out of the bearing also increases. The net effect may partly explain why the oil-flow requirements apparently decreased with increasing DN at the high airflow (fig. 3(a)).

(2) The bearing temperature rises (fig. 3(b)), and the amount of oil required to wet the bearing increases because of increased evaporation. The temperature factor should be less important at the lower airflows because of the more nearly constant temperature.

Since airflow affects the minimum required oil flow, the 30-millimeter-bore-bearing controlled-temperature tests were run with airflows closely matching those of references 4 and 5 to facilitate comparisons of the data.

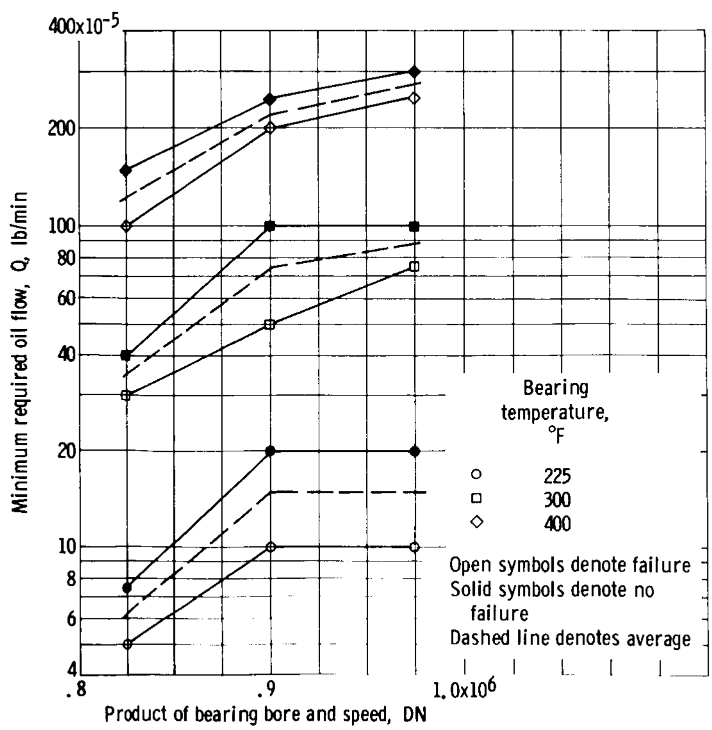


Figure 4. - Effect of DN on minimum required oil flow for 265-pound load and three bearing temperatures.

Effect of DN

The results of tests conducted at DN values of 0.825×10^6, 0.900×10^6, and 0.975×10^6 and temperatures of 225°, 300°, and 400° F are shown in figure 4. For each of the three temperature levels, bearing temperature was controlled by adjusting the heat input to the bearing. Minimum required oil flows for no failure, as indicated by the solid data points, increased with increasing DN, although they were equal for the two highest DN values at 225° and 300° F. This equality could be due to nonoptimum airflows or to the fact that tests were not run at close enough flow intervals. For example, at 225° F and DN values of 0.9×10^6 and 0.975×10^6, satisfactory operation was obtained at an oil flow of 2×10^{-4} pound per minute while failure occurred at an oil flow of 1×10^{-4} pound per minute. Thus, the flow at which failure occurred was one-half that of the minimum required oil flow. In general, oil flows were reduced by factors of 1/3 to 1/2, so that the true minimum required oil flow (as estimated by the average curves of fig. 4) may be intermediate between the lowest flow at which successful operation was obtained and the flow at which failure occurred. Actually, determination of minimum required oil flows with greater precision is not necessary because large factors of safety (one order of magnitude or more) must be used in applying data of this type.

Effect of Temperature

The effect of temperature can best be seen by cross plotting the data of figure 4. Minimum required oil flow is shown as a function of temperature in figure 5. The rate of increase of minimum required oil flow with temperature is quite high. An increase in temperature from 225° to 400° F requires an increase in oil flow of 15 to 20 times. The evaporation rate of the oil evidently plays a major role in determining the oil flow needed to wet the bearing.

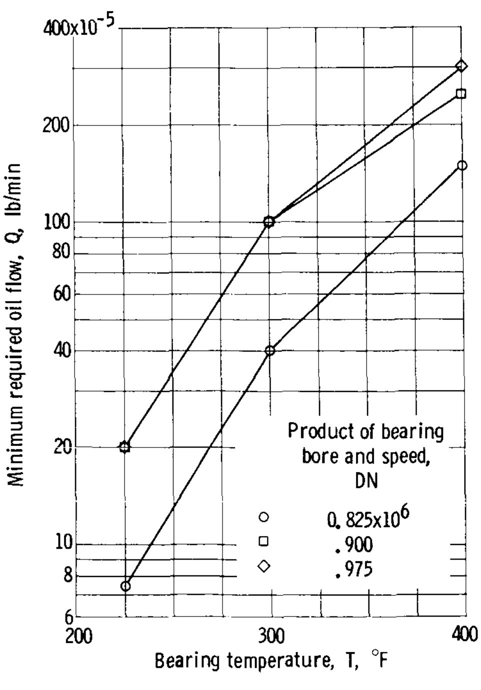


Figure 5. - Effect of bearing temperature on minimum required oil flow for 265-pound load and three DN values.

Generalized Correlation

A preliminary comparison of data was made with data from references 4 and 5 at the same DN value (0.9×10^6) to determine if the data could be put into a more generally useful form. The results of this preliminary comparison, shown in figure 6, were promising. Considering that the data were obtained with two bearing sizes and two lubricants over a temperature range of 180° to 500° F in two different test apparatus, it is surprising to see that they fit a single curve fairly well. Therefore, all the controlled temperature data from references 4 and 5 and from this investigation were correlated by means of a dimensional analysis. Some 57 data points were included in the analysis. The ranges of pertinent variables were as follows:

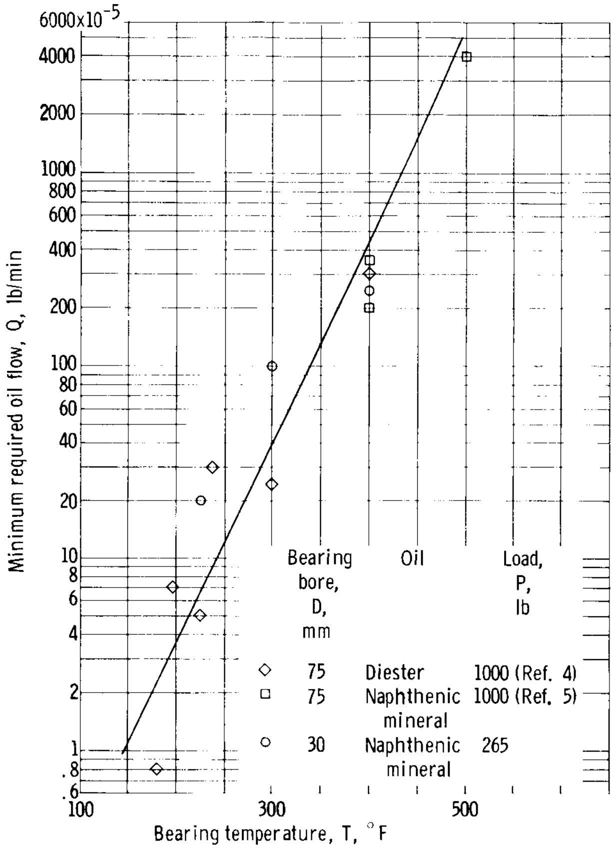


Figure 6. - Effect of bearing temperature on minimum required oil flow for 30- and 75-millimeter bore bearings with two lubricants at DN of 0.9×10^{6}.

Bore, D, mm . . . . 30 and 75
Load, P, lb thrust . . . . 265 to 3000
DN . . . . 0.6×10^{6} to 0.975×10^{6}
Bearing temperature, T, °F . . . . 225 to 500

Details of the analysis are given in the appendix.

The results, shown in figure 7, indicate a fair degree of correlation between $(Q/PN)\times10^{10}$ and $PN^{5}T^{8}D^{3}\times10^{-48}$, where

Q minimum required oil flow, lb/min

P load, lb

N speed, rpm

T bearing temperature, °F

D bearing bore, mm

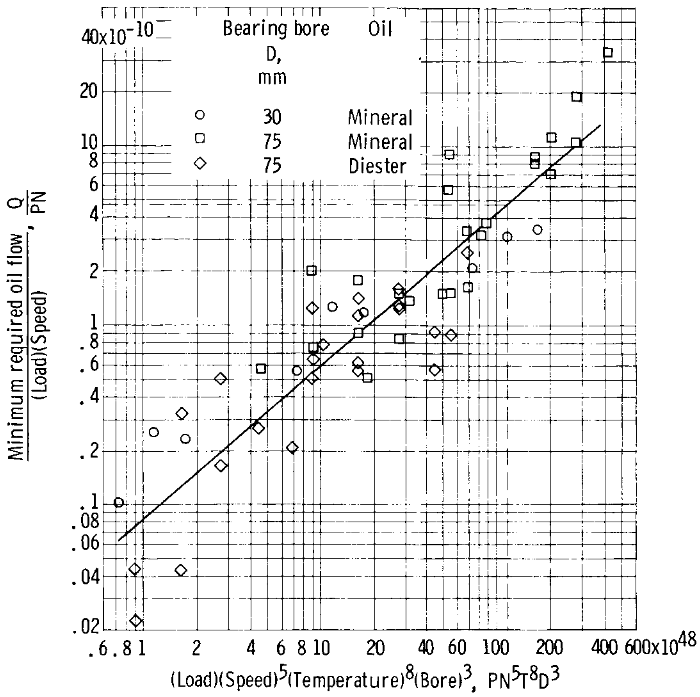


Figure 7. - Generalized correlation of minimum-required-oil-flow data for two bearing sizes and two lubricants. DN range, 0.6×10^6 to 0.975×10^6; thrust load range, 265 to 3000 pounds; bearing temperature range, 225° to 500° F. Example (dashed line): load, P, 265 pounds; speed, N, 30 000 rpm; bearing temperature, T, 400° F; bearing bore, D, 30 millimeters; $PN^5T^8D^3 = 114\times10^{48}$; $Q/PN = 4.65\times10^{-10}$; minimum required oil flow, Q, 3.7×10^{-3} pounds per minute.

The resulting curve may be expressed by the equation

$$\log \frac{Q}{PN} = -51.982 + 0.852 \log PN^5T^8D^3$$

The individual data points are scattered so that, at specific values of the abscissa variable they range from a minimum of $\frac{1}{3}$ to a maximum of $3\frac{1}{2}$ times the flow indicated by the best fit line. These flow ratios, however, are not much greater than the flow intervals used to determine the minimum required oil flows in individual tests.

If values of P, D, N, and T are known, the value of the abscissa variable $PN^5T^8D^3\times10^{-48}$ can be calculated. From the best fit line the value of the ordinate variable $(Q/PN)\times10^{10}$ can be determined so that Q can be calculated. For safety a Q value order of one magnitude higher than that calculated should be used. It is not recommended that the correlation of figure 7 be used if any of the pertinent variables range beyond the limits of those used to obtain figure 7, especially for bearing temperatures above 500° F.

As an example, calculate the minimum required oil flow for a 30-millimeter-bore bearing operating at a load of 265 pounds, a speed of 30 000 rpm ($DN = 0.9\times10^6$), and a temperature of 400° F:

$$PN^5T^8D^3 = 114\times10^{48}$$

$$\log(PN^5T^8D^3) = 50.057$$

$$\log\left(\frac{Q}{PN}\right) = -51.982 + 0.852(50.057)$$

$$\log\left(\frac{Q}{PN}\right) = -9.333$$

$$\frac{Q}{PN} = 4.65\times10^{-10}$$

$$Q = (4.65\times10^{-10})(265)(30\ 000)$$

$$Q = 3.7\times10^{-3}\ \text{lb/min}$$

SUMMARY OF RESULTS

A series of minimum-oil-flow tests was conducted with 30-millimeter-bore deep-groove ball bearings over a DN range (product of bearing bore and speed) of 0.825×10^6 to 0.975×10^6 and a temperature range of 170° to 400° F at a thrust load of 265 pounds with naphthenic-mineral-oil - air mist lubrication. The following results were obtained:

1. A correlation of the data obtained herein and previously published data for 75-millimeter-bore bearings indicates that minimum required oil flow can be expressed as a single function of bore D, load P, DN, and bearing temperature T, where $\log(Q/PN) = -51.982 + 0.852 \log(PN^5T^8D^3)$ for the following ranges of these variables: bore, 30 and 75 millimeters; load, 265 to 3000 pounds thrust; DN, 0.6×10^6 to 0.975×10^6; and temperature, 225° to 500° F.

2. Airflow, which was controlled independently of oil flow, was found to affect the minimum required oil flow at a specific load, DN, and temperature. Airflow apparently affects the efficiency with which the bearing can utilize the oil droplets fed to it, and there appears to be an optimum mist velocity for a given DN or rotative speed.

3. As with previously published data, the minimum required oil flow for continuous operation increased with increasing DN and bearing temperature.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, April 20, 1965.

APPENDIX - CORRELATION OF MINIMUM REQUIRED OIL FLOW

The characteristic equation for oil-flow requirement of a rolling-element bearing may be deduced by using a dimensional analysis technique (ref. 8). The nondimensional group containing the dependent variable (in this case Q) becomes equal to an unknown function of the remaining nondimensional groups. As an approximation, the unknown function is assumed to be the product of the independent nondimensional groups, each raised to an empirically determined power. Second-order variables such as the specific heat, density, thermal conductivity, and viscosity of the oils are assumed constant because the two oils have similar thermal and viscous properties. When this assumption is made, the characteristic equation (ref. 9) becomes

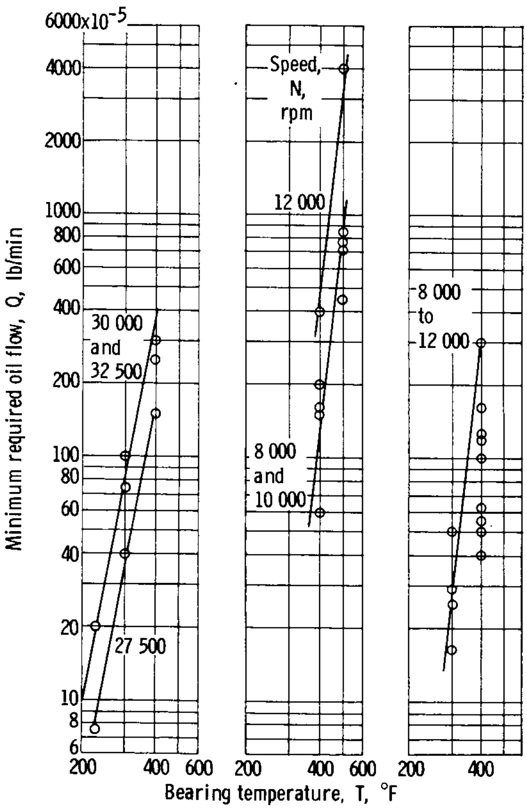
$$Q = KP^aN^bT^cD^d$$

where

Q	minimum required oil flow, lb/min
K, a, b, c, d	constants dependent on bearing system
P	load, lb
N	speed, rpm
T	bearing temperature, $^{\circ}F$
D	bearing bore, mm

Experimental data are used to determine the values of the exponents. To determine the influence of the variables on the dependent variable, the variables must be taken singly and in an order and manner such that only one variable affects the dependent variable.

The exponent c is obtained first by plotting Q against T on log-log coordinates (fig. 8). The slopes are 8.27, 8.0, and 8.18 for the two bearing sizes and two different oils. The approximate equality of slopes for the two oils indicates that it was valid to assume that the small differences in the thermal



(a) Bearing bore, 30 millimeters; lubricant, mineral oil. (b) Bearing bore, 75 millimeters; lubricant, mineral oil. (c) Bearing bore, 75 millimeters; lubricant, diester oil.

Figure 8. - Influence of bearing temperature on minimum required oil flow.

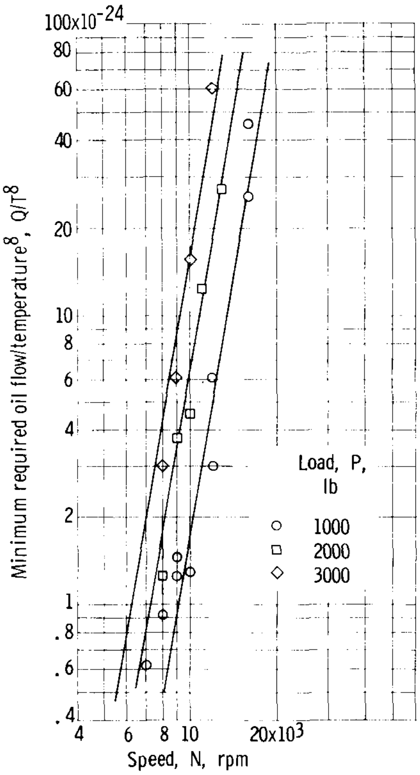


Figure 9. - Influence of speed and load on minimum required oil flow for 75-millimeter-bore bearings.

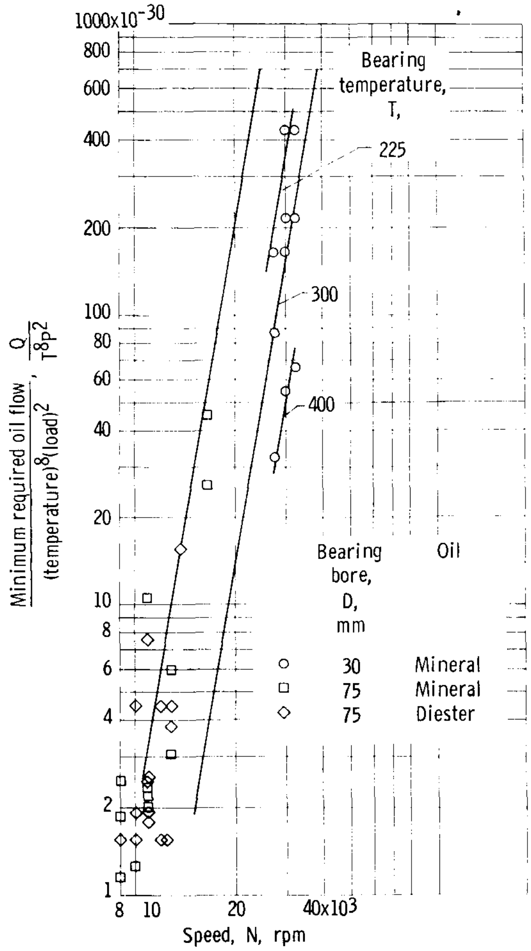


Figure 10. - Influence of speed and bearing bore on minimum required oil flow.

and the viscous properties of the two oils could be neglected. The exponent c is taken as 8. The exponents b and a are obtained next by plotting Q/T^8 against N on log-log coordinates for three loads (fig. 9). The slope of the curve gives the exponent b. The actual slope is 5.85 and is taken as 6. The exponent a is obtained by noting that Q is proportional to the second power of the load at a given value of N.

The exponent d is obtained by plotting Q/T^8P^2 against N (fig. 10). The effect of load is eliminated, and the influence of N and D remains. The slope is 5.85, and the exponent b is checked for the 30-millimeter-bore bearing. There is some scatter because of temperature for the 30-millimeter-bore bearing. The exponent d is obtained by noting that Q is proportional to the third power of bearing size at a given value of N.

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