The results of these tests and many others show that for lightly loaded, high-speed aero/space applications where slip is resulting in skidding damage the elliptical, preloaded bearing should be considered as a viable solution. There are thousands of engines both in military and commercial planes as testimonial for the success of the elliptical preload principle.

Conclusion

While the out-of-round concept has proven itself adequate in many engine applications of all major engine builders, it has not been without problems. In recent years, and as late as this past fall, both intershaft bearings and main shaft bearings have all had roller/raceway skid problems with elliptical bearings. This points out that though we have a fundamental analytical tool to predict roller slip, there is no detailed understanding of the conditions at which damage results when slip occurs. Under extremely stable lubrication and operating conditions considerable slip may occur without surface damage. However, under unfavorable lubrication conditions the thermal or mechanical shock of the slip can collapse, or cause total or partial absence of oil film with resulting surface damage. A predictive criterion is not available to predict when skid damage will occur.

In order to permit design optimization of cylindrical roller bearings to prevent skid damage, a more analytical program is needed to 1) improve prediction of lubricant film conditions using more up-to-date lubrication models, and 2) determine a damage criterion for roller/raceway contacts and use it to predict regimes in which skid damage may occur.

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Solid Lubricated Bearing Inter-Element Heat Transfer Mechanisms

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Solid lubricated bearings suffer from the absence of a good heat transfer medium such as provided in conventional bearings by the lubricated oil. The frictional heat generated in localized areas in the bearing produces large temperature gradients, some of them periodic, which undoubtedly contribute to failure. A need exists for better knowledge of the internal heat transfer between bearing elements to aid in accurate computation of the element temperature distributions. In this paper, results are reported for a series of experiments made with actual bearings, both annular and pellet separator types, to explore the characteristics of the heat transfer occurring within the bearing operating in an air environment. Measured element temperatures reveal a strong propensity of the bearings to isolate themselves from the adjacent ambient air. Overall ring-to-ring conductances are evaluated from the measured temperatures and electrically imposed heat fluxes. The separate roles of convection and conduction are explored with tests on modified bearings and with a tentative model of the transient conduction at the ball-raceway contact.

Nomenclature

A = inner surface area of the outer bearing ring
k = thermal conductivity
l = length of ball-race contact ellipse
N = bearing speed
$\dot{q}, \dot{q}_{r,o}$ = electrically supplied heat transfer rate
$\dot{q}_1, \dot{q}_2, \dot{q}_3$ = thermal circuit heat transfer rates (see Fig. 6)
R_1, R_2, R_3 = thermal circuit resistances (see Fig. 6)
$s_b, s_{r,o}$ = frictionally generated energy rate (see Fig. 6)
t_b = bearing ball temperature
t_p = bearing pellet temperature
$t_{r,i}$ = bearing inner ring temperature
$r_{r,o}$ = bearing outer ring temperature
t_s = bearing annular separator temperature
U = thermal conductance
w = width of ball-race contact ellipse
α = thermal diffusivity
θ = ball-race contact time

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Introduction

TYPICAL rolling element bearing applications in gas turbine engine technology utilize a continuous recirculating oil flow for both lubricating and cooling purposes. There are many instances, however, where the necessary complexity of the oil system and the requirement of relatively low oil temperatures make the conventionally lubricated bearing quite unattractive. A possible alternative, particularly for applications which require a relatively short service life but long unattended shelf life, is the solid lubricated bearing where the lubricant is contained within the bearing in a solid, wearable element.

The service life of bearings, like most machine components, is in part dictated by the operating temperature distribution, especially when the temperature gradients are large and periodic. In conventionally lubricated bearings, a major function of the oil, in addition to providing the lubricant film, appears to be its ability to remove heat rapidly from the localized contact where frictional heat generation occurs. The oil not only removes the generated energy from the bearing; but in the process redistributes thermal energy among the bearing components, reducing the temperature gradients.

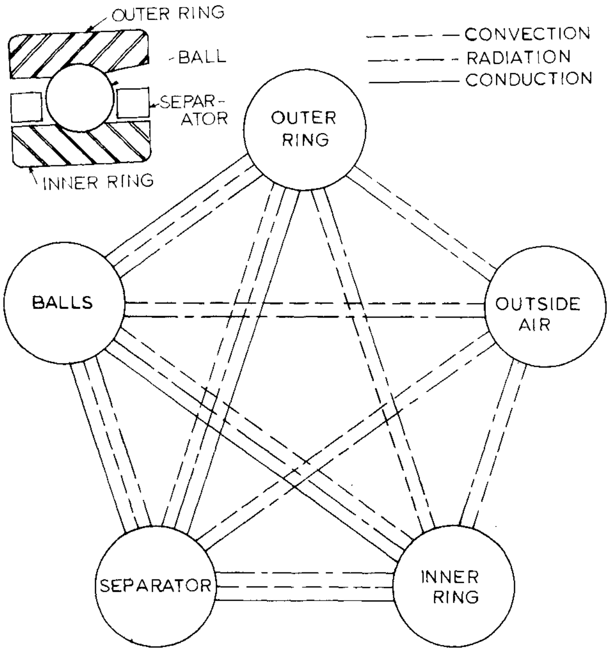

Fig. 1 Possible thermal coupling of bearing elements.

In a self-lubricated, solid film bearing, the oil and its heat transfer capabilities are absent, and the relatively short service lives reported for this type of bearing are probably associated to some extent with large temperature gradients within the bearing. This is compounded by the fact that solid lubricated contacts operate in the boundary lubrication regime with metal-to-metal contact; the heat generation is higher than that produced by the elastohydrodynamic regime of conventionally lubricated bearings.

Improvements in solid lubricated bearing performance thus appear to depend in part on the ability of the designer to predict the detailed operating temperature distribution in the bearing, correlate this with bearing life, and then modify to a more favorable distribution through material and geometrical changes and/or deliberate auxiliary cooling of the bearing components. The bearing component temperature distributions, although relatively easy to calculate numerically, are highly dependent on the convective and radiative conditions at the component boundaries, and very little information is available that is directly applicable to heat transfer in the bearing geometry. The problem is potentially quite complex because of the multiple elements in the bearing and the possibility that convective, radiative, and conductive energy exchange can take place between most combinations of the elements.

This is illustrated schematically in Fig. 1, which illustrates a typical bearing cross section with an annular separator piloted on the inner ring. The separator is the lubricating element. Temperature differences between the various elements are produced by frictional heat generated at the contacts and by temperature differences between the rotating shaft and the outer ring mounting structure. For the annular separator configuration shown, the direct conduction coupling between the separator and outer race would not exist. In addition, the direct convective and radiative coupling between the inner and outer rings would probably be negligible. However, for the case of pellet separators, which are individual cylinders of solid lubricant between the balls, all of the energy exchange mechanisms shown in Fig. 1 may be present.

The present paper summarizes the methods and results of an experimental investigation conducted to observe the behavior of this inter-element heat transfer by imposing a known heat flux through one of the elements, in this case, the inner surface of the outer ring. The tests were conducted with relatively small temperature differences between the elements so that radiation heat transfer is negligible; thus the measured conductances are combinations of conduction and convection terms alone. In this connection, it should be noted that a radiation exchange model for the bearing, while geometrically quite complex, poses no particular difficulties and can be as accurate as necessary within the bounds imposed by uncertainties in the surface radiation properties of the bearing elements. On the other hand, the convection exchange is largely a function of the fluid flow patterns in the bearing, of which very little is known.

Experimental Apparatus

The general idea of the test program is simply to impose a known heat flux at the inner surface of the outer ring by mounting an electrical heater on the outside and insulating it so that virtually all of the generated heat flux must pass to the inside of the bearing. Size 205 bearings with both annular and pellet graphite separators were prepared and mounted as shown in Fig. 2. The outside surface of the outer ring of each bearing was wrapped with small diameter resistance wire between two layers of high temperature insulating tape. The bearing was then pressed into a machined ½ in. deep annulus of balsa wood. The bearing inner race was pressed onto a special bearing holder machined to screw into a commercially available precision two-bearing grinding quill. Each bearing holder is machined to final dimensions while in place on the quill to assure concentricity. The center cavity of the bearing holder can be cooled or heated by air impingement provided through a concentric supply and exhaust tube arrangement: this provides control of the inner race temperature.

A schematic of the overall test facility is shown in Fig. 3. The bearings are rotated through a flat belt drive to an autotransformer controlled electric motor. Reflected light from the partially blackened quill pully is viewed by a photoelectric cell whose output is fed to a counter for continuous speed monitoring. Outer ring temperatures are

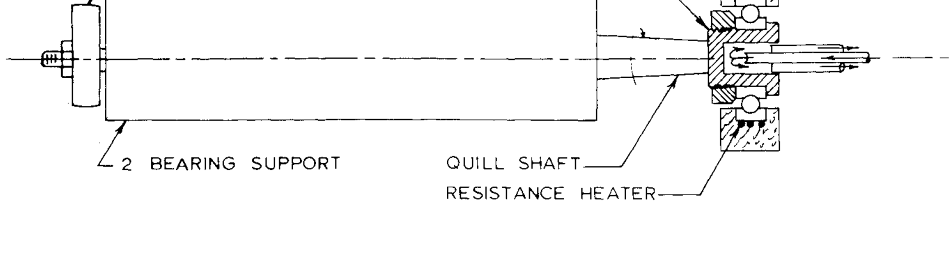

Fig. 2 Test bearing mounting.

measured with conventional thermocouples, ice bath reference, and potentiometric recorder. The temperatures of the bearing holder, separator, and, in the case of the pellet bearings, the balls are measured with a Leeds and Northrup "Surtemp" surface temperature probe and control unit. This probe has a thermocouple mounted flush in a ceramic tip containing a second thermocouple and guard heater. When the probe is touched to a surface the control unit provides a precise amount of current to the guard heater to maintain an adiabatic condition at the measuring tip, insuring proper surface temperature readings. Measurements with this probe are made after steady-state conditions are reached for a given bearing speed and outer ring heating rate by switching off the drive motor and touching the probe tip to the bearing element as soon as it stops rotating. The deceleration rate and probe response time are sufficiently fast that the measurement is routinely recorded within six sec of the motor shutoff time. A transient conduction analysis of the bearing components shows that this is fast enough to record the steady-state operating temperatures within less than 1°F. The accuracy of the probe is ±3°F. Further details of the test equipment and procedures are available in Ref. 1.

Test Results and Discussion

Measurements of Bearing Element Temperatures

Figure 4 shows typical measured values of the outer and inner ring temperatures and separator temperatures for the case of a steel ball, annular graphite separator bearing at various speeds, without electrical heating applied at the outer ring. In the case of the inner ring, it is not possible to measure its temperature directly with the surface temperature probe because of the configuration of the bearing holder. The values presented are actually measurements of the bearing holder temperature. The low heat flux levels present in these tests, together with the good thermal contact achieved by pressing the bearing onto the holder, result in identical inner ring and holder temperatures within a fraction of a degree.

In all cases, the thermal coupling between the outer and inner ring is unexpectedly strong as evidenced by the dependence of the outer ring and separator temperatures on the temperature level maintained at the inner ring. These same observations apply to similar measurements made with electrical heating applied at the outer ring. Thus the convection heat transfer between the outer race and the ambient environment of the bearing appears to be very weak compared to the combined conduction and convection heat transfer occurring between the rings.

Figure 5 shows typical measured bearing element temperatures for a steel ball, graphite pellet separator bearing operating at 3000 rpm for three values of electrical heating rates applied at the outer ring. Again, the strong influence

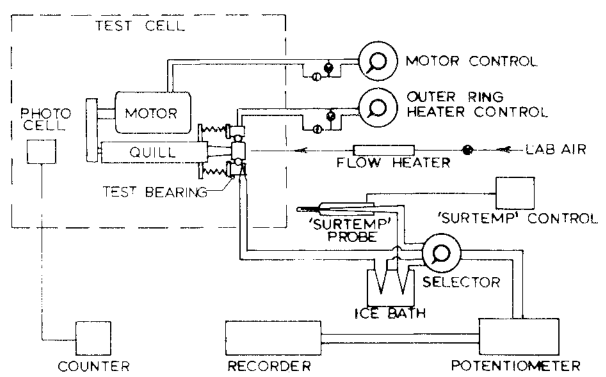

Fig. 3 Apparatus schematic.

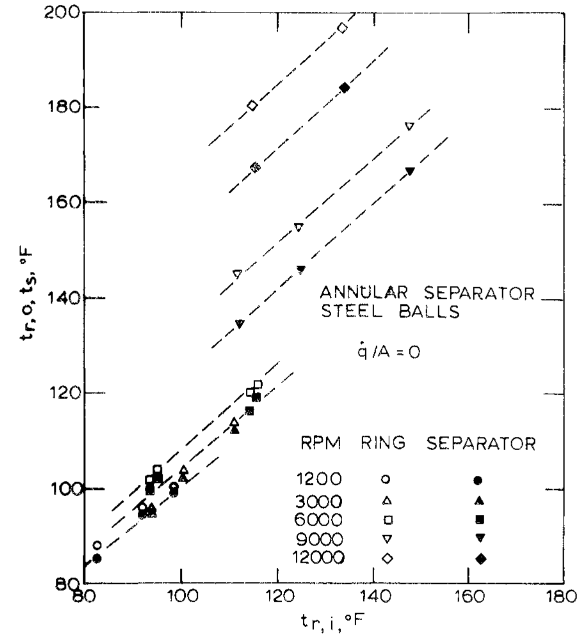

Fig. 4 Steel ball/annular separator temperature with $\dot{q}/A = 0$.

of the inner ring temperature is apparent. In this case, both the ball and pellet temperatures can be measured by the surface temperature probe, and the two are essentially equal for all conditions. These same trends, including the equality of ball and pellet temperatures, were also observed at a bearing speed of 6000 rpm.

In an effort to determine whether conduction or convection effects are responsible for the strong dependence on the inner ring temperature, additional tests were conducted with the annular separator configuration with both alumina and silicon nitride balls. These were chosen to give ball thermal conductivity values both higher and lower than that of the steel balls. Subsequent data from the manufacturer, however, indicate that the thermal conductivity of both new materials were nearly the same (nominal values of approximately 20 Btu/hr ft F compared with approximately 14 Btu/hr ft F for the steel

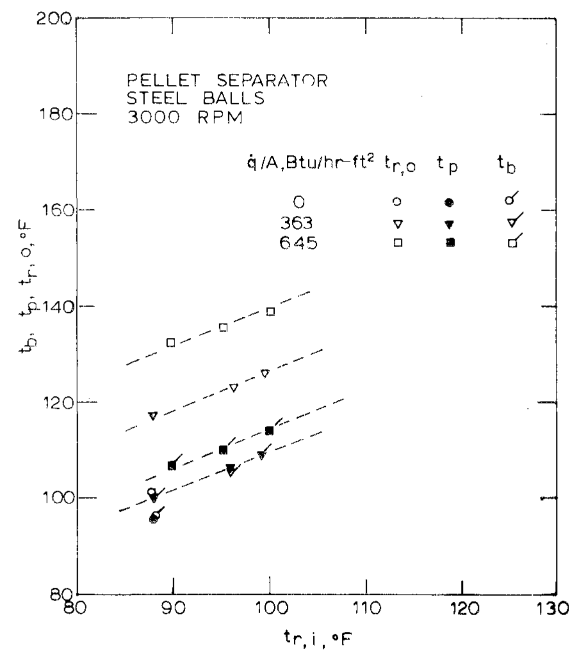

Fig. 5 Steel ball/pellet separator temperatures at 3000 rpm.

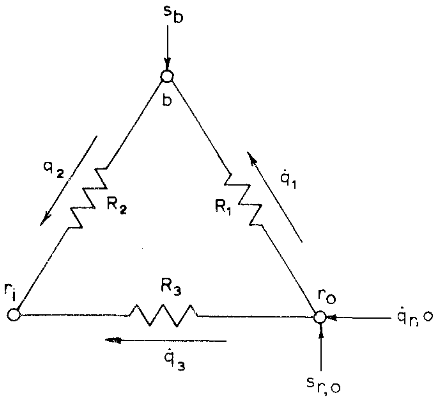

Fig. 6 Lumped thermal circuit for bearing.

balls). All of the testing to this point was done with a small thrust loading of approximately 1 lb imposed on the bearing in order to keep the frictional heat generation as small as possible. A few additional tests were conducted with the steel ball/annular separator combination with a higher thrust loading of 11 lbs. The measured element temperatures with both the new thrust level and the new ball materials exhibit qualitatively the same behavior shown in Fig. 4, that is, a strong dependence on inner ring temperature.

Determination of the Overall Ring-to-Ring Conductance

An interpretation of these measured bearing element temperatures can be made by considering the simplified, lumped thermal circuit depicted in Fig. 6. Here the interaction with the outside air is assumed to be negligible as indicated by the measured temperature behavior. Nodes r_o and r_i represent the outer and inner rings, and node b represents a lumped combination of the balls and separator. R_1 is an overall thermal resistance between the outer ring and the ball-separator combination, R_2 is the corresponding overall resistance between the ball-separator combination and the inner ring, and R_3 is a possible convection resistance between the rings directly. Both R_1 and R_2 can be parallel combinations of separate conduction and convection terms. The term s_b represents a thermal energy source somewhere in the ball-separator combination due to frictional generation. Similarly, $s_{r,o}$ represents a source term at the outer ring created by friction. The variable electrical heating at the outer ring is represented by $\dot{q}_{r,o}$.

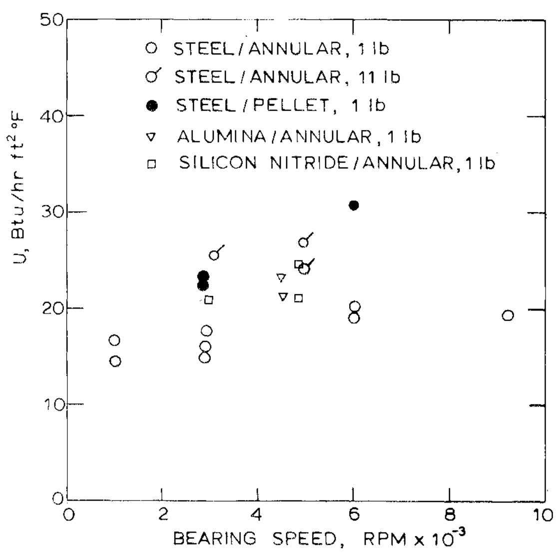

Fig. 7 Bearing ring-to-ring conductance.

For small temperature differences the circuit will be linear with the thermal potentials and currents related as follows:

$$t_{r,o} - t_b = \dot{q}_1 R_1;\ t_b - t_{r,i} = \dot{q}_2 R_2;\ t_{r,o} - t_{r,i} = \dot{q}_3 R_3 \tag{1}$$

$$\dot{q}_1 + \dot{q}_3 = s_{r,o} + \dot{q}_{r,o};\ \dot{q}_2 = \dot{q}_1 + s_b \tag{2}$$

These relations can be combined to yield:

$$(t_{r,o} - t_{r,i})(R_1 + R_2 + R_3)/R_3 = \dot{q}_{r,o}(R_1 + R_2) + s_{r,o}(R_1 + R_2) + s_b R_2 \tag{3}$$

If the resistances and frictional generation terms are functions of bearing speed and load alone, then two tests a) and b) made with two different electrical heating rates but with identical inner ring temperatures can be used to evaluate the overall ring-to-ring conductance:

$$\frac{(\dot{q}_{r,o})_a - (\dot{q}_{r,o})_b}{(t_{r,o})_a - (t_{r,o})_b} = \frac{R_1 + R_2 + R_3}{R_3(R_1 + R_2)} = U \tag{4}$$

where U is the overall ring-to-ring conductance.

Figure 7 presents the calculated overall conductances for the various ball-separator combinations tested vs the bearing speed. Here U is based on the entire inner area of the outer ring. For all of the combinations except the steel ball annular separator, the values were obtained from a single test bearing. For the steel/annular combination, the conductances presented were obtained from two separate bearings.

"Zero-Conduction" Tests

Although the calculated conductances are somewhat higher for the balls with higher thermal conductivity, the effect is not as pronounced as might be expected if the conduction mechanism alone dominated the heat transfer from ring to ring. This leads to the speculation that convection is an important inter-element heat transfer mechanism despite the fact that convection coupling to the outside air appears to be very weak. In an effort to further clarify this point one of the annular separator bearings was disassembled, and the regular balls were replaced with simulated balls shaped from balsa wood. These were cemented into the separator spaces, and the separator in turn was cemented to the inner ring. The outer diameter of this assembly was made slightly smaller than normal, so that when slipped into the outer ring no actual contact is made between the balls and outer raceway if it is positioned concentrically. For these tests the electrically heated, insulated outer race was supported in a separate fixture.

Inner and outer ring temperatures were measured for this configuration with various electrical heating rates for several speeds from 1200–11,700 rpm. The outstanding feature over the entire test range is the retention of the one-to-one correspondence between the inner and outer ring temperatures. Thus, the strong thermal coupling is present even without conduction through the balls. This convection thermal coupling can be explained in terms of a strong, confined secondary flow of Taylor-like vortices in the annular space between the outer ring and the separator. The confinement could be reinforced by the air flow pumped radially outward on the lateral surfaces of the inner ring and separator, acting as a curtain separating the annular space from the outside environment.

Figure 8 shows the values of ring-to-ring conductance calculated from the temperatures and heating rates measured during the "Zero Conduction" tests. In the actual bearing the retainer rotates at about 40% of the shaft speed, so these zero-conduction tests are probably representative of bearing speeds some 2.5 times the test speed. If the conductance values at these equivalent bearing speeds are compared with the overall conductances of Fig. 7, it is seen that they both appear to increase at about the same rate with increasing speed; however, the zero-conduction values are only 40–50% of the overall values. It should be kept in mind that the simulation achieved in these zero-conduction tests is not perfect. The separator rotates at the inner ring speed, and the balls do not rotate with respect to the separator. Nevertheless, the modeling in the annular space between the separator and outer ring should be fairly good.

Race-to-Ball Conduction

Since the evidence at this point suggests both convection and conduction are important mechanisms for transferring heat across the bearing, a logical next step would be to run the test bearings inside an evacuated enclosure to eliminate convection. Unfortunately, this is not possible with the present test equipment and procedures. However, it is possible to make some general comments about the nature of the conduction mechanism, based on the particular geometry of the bearing and classical conduction theory.

As the ball travels around the raceway, each point on the contact path of the ball actually makes contact with the raceway for only a very short time, which can be estimated easily from the bearing kinematics and knowledge of the contact deformation. During this short contact time, heat will be transferred by conduction between the race and the particular point on the ball, if a temperature difference exists. In between the race contacts, the point on the ball will be under the influence of convection and conduction to adjacent points on the ball. Thus, even during steady-state bearing operation, the conduction mechanism between the raceway and balls is transient in nature.

The elliptically shaped pressure surfaces resulting from a Hertzian contact model have been calculated for the 205 size bearings of the present study in Ref. 2, using the methods outlined by Jones.[3] They show that the contact time will usually be less than 0.01 of the noncontacting time. Thus most of any local temperature increase or decrease on the ball surface associated directly with the contact will be eliminated by convection and internal ball conduction before the next contact is made. The basic conduction mechanism, then, is similar to that of a semi-infinite solid encountering a step change in its boundary temperature. The solution to this conduction problem is well known[4]; the cumulative heat flux over the contact time is proportional to the size of the temperature step and to the product $k(\theta\alpha)^{1/2}$, where θ is the contact time and k and α are the solid thermal conductivity and diffusivity, respectively. The contact area for conduction per unit time in the ball-race contact can be shown to be proportional to the contact area length, l, and the bearing speed, N. The contact time can be shown to be proportional to the contact area width, w, and inversely proportional to the bearing speed. If, in addition, the temperature difference that the ball surface sees as it comes in contact with the race is proportional to the overall ring-to-ring temperature difference, then the conduction contribution to the overall conductance will have the following functional form:

$$U_{\text{conduction}} \propto kl(wN/\alpha)^{1/2} \tag{5}$$

A preliminary comparison of this model has been made with the present experiments by evaluating the proportionality constant to match the observed results for the steel balls/annular separator at 5000 rpm and 1 lb thrust. At these conditions, the overall conductance is approximately 18 Btu/hr ft^2 F from Fig. 7. The convective contribution to this total indicated by the zero-conduction results of Fig. 8 is about 11 Btu/hr ft^2 F. This leaves 7 Btu/hr ft^2 F for $U_{\text{conduction}}$ and the proportionality constant was chosen to yield this number with the appropriate values of k, l, w, N, and α. The same proportionality constant was used to calculate $U_{\text{conduction}}$ at other speeds and loads; these were added to the zero-conduction results to produce the predicted trends shown in Fig. 9. The limited experimental results and the approximate nature of the analysis make any conclusions drawn from a comparison of the two very speculative. However, the predicted increase in conductance at higher speeds appears to be greater than the measurements indicate. Since this has important implications for uses of these bearings in high-speed turbomachinery, more refined analyses and tests under vacuum conditions to isolate the conduction mechanism are desirable.

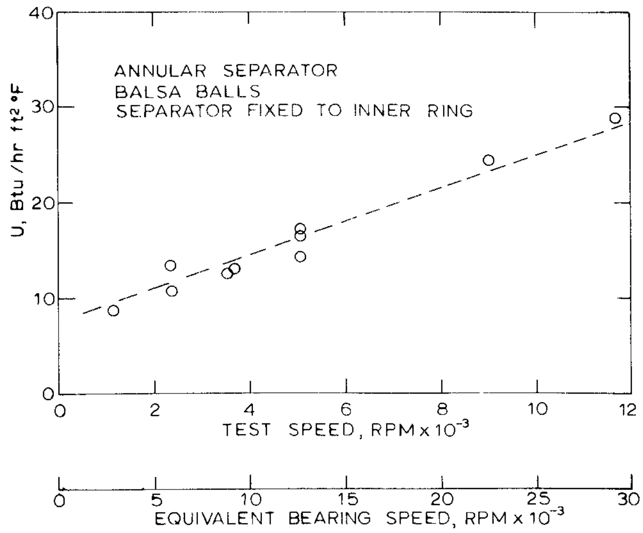

Fig. 8 Conductances for the zero conduction tests.

Conclusions

The limited results of the present series of experiments do not provide all of the information needed for accurate modeling of the conduction and convection heat transfer occurring inside the bearings. They do, however, provide measured element temperatures under controlled condi-

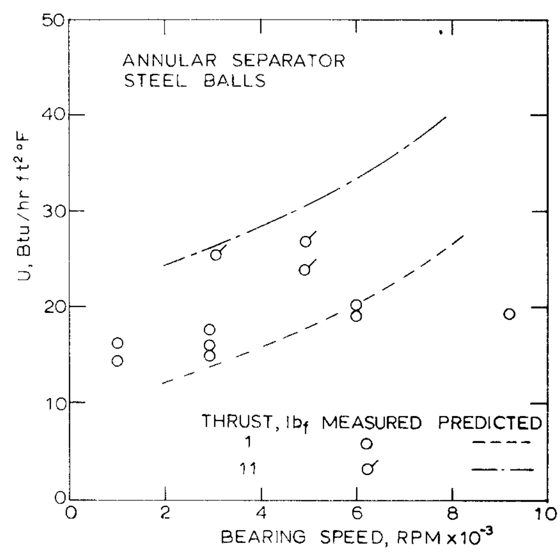

Fig. 9 Comparison of measured and predicted trends.

tions that can be used to check proposed bearing thermal models. In addition, the overall ring-to-ring conductances evaluated from the measured temperatures and heat fluxes appear reasonable, and any proposed thermal model should be structured so that its equivalent overall conductance is in agreement over the range of tested conditions.

The tendency for the bearing to isolate itself from the ambient air appears to be very strong, and both conduction and convection mechanisms appear to contribute to this fact. It should be remembered, however, that all of the present tests were conducted with zero pressure difference across the bearing; thus there was no net cross flow. Cross flow should be expected to disrupt the convection coupling between the inner and outer rings, and should be a subject for further testing.

One implication of the present results is that reduction in the operating temperature of one of the bearing rings through deliberate cooling will be felt throughout the bearing. This leads to the speculation that relatively uncomplicated under-ring cooling schemes for the stationary ring may be useful in some situations for lowering operating temperatures and extending bearing life.

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