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EXPERIMENTAL EVALUATION OF THE SERIES-HYBRID ROLLING-ELEMENT BEARING

by Richard J. Parker, David P. Fleming, William J. Anderson, and Harold H. Coe

Lewis Research Center
Cleveland, Ohio 44135

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16. Abstract

One approach to the high speed rolling-element bearing problem is the series-hybrid bearing. This concept couples a fluid-film bearing with a rolling-element bearing such that the rolling-element bearing inner race runs at a fraction of shaft speed. A series-hybrid bearing was run at thrust loads from 100 to 300 pounds (445 to 1340 N) and speeds from 4000 to 30 000 rpm. The lowest speed ratio (ratio of ball bearing inner-race speed to shaft speed) obtained was 0.67. Agreement between theoretical and experimental speed sharing was good; deviations were greatest at highest speeds. The series-hybrid bearing has the potential of substantially increasing high-speed bearing fatigue life.

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SUMMARY

A series-hybrid bearing composed of a fluid-film bearing coupled in series with a ball bearing was run at thrust loads from 100 to 300 pounds (445 to 1340 N) and speeds from 4000 to 30 000 rpm. One element of the fluid-film bearing rotated at shaft speed. The second element of the fluid-film bearing rotated with the inner race of the ball bearing at a speed less than the shaft speed. The ball bearing outer race was stationary. The fluid-film bearing consisted of an orifice-compensated annular thrust bearing and a self-acting journal bearing. Lubricant for the fluid-film bearing was supplied through the shaft center. Centrifugal effects were utilized to provide pressurized lubricant for the fluid-film bearing. The rolling-element bearing was a 115-size deep-groove ball bearing.

At shaft speeds where the centrifugally generated supply pressure was great enough to separate the fluid-film thrust bearing surfaces under the applied thrust load, the inner race of the ball bearing rotated at a speed less than the shaft speed. In this manner, speed sharing between the ball bearing and the fluid-film bearing was accomplished. The lowest speed ratio (ratio of inner race speed to shaft speed) obtained was 0.67. This corresponds to an approximate reduction in DN value of 1/3. For a ball bearing in a 3 million DN application, fatigue life would theoretically be improved by a factor as great as 8.

Agreement between theoretical and experimental liftoff speeds and speed sharing was good. Experimental liftoff speeds were only slightly higher than predicted. Deviation of experimental from theoretical speed sharing was greatest at highest speeds.

Torque of the series-hybrid bearing was greater after liftoff of the fluid-film bearing and was in the same range as that of the ball bearing alone when jet lubricated.

INTRODUCTION

Recent developments in gas turbine engines - such as higher thrust-to-weight ratios, advanced compressor design, high temperature materials, and increased power output - have resulted in a requirement for larger shaft diameters and higher main shaft bearing speeds (ref. 1). Bearings in current production aircraft turbine engines operate in the range from 1.5 to 2 million DN (bearing bore in mm times shaft speed in rpm). Engine designers anticipate that during the next decade turbine bearing DN values will have to increase to the range of 2.5 to 3 million. It is speculated that after 1980 turbine engine developments may require bearing DN values as high as 4 million.

When ball bearings are operated at DN values above 1.5 million, centrifugal forces produced by the balls can become significant. The resulting increase in Hertz stresses at the outer-race ball contacts can seriously shorten bearing fatigue life. There are several possible approaches to solve this problem.

The first and most obvious approach is to optimize the bearing internal geometry for maximum life using a high-speed ball bearing dynamics computer program (ref. 2). This will yield the optimum ball diameter and number, race curvatures, and contact angle. However, this approach will not yield more than a small, incremental improvement in life over that of bearings now in use. As a result, less conservative approaches must be considered.

One such approach is to reduce the mass of the ball (ref. 3). A 50-percent reduction in ball weight, while maintaining the ball diameter constant, can theoretically result in an increase in life of four to six times at DN values greater than 3 million. However, experimental results with hollow balls with 50-percent weight reduction indicate that they fail in flexure fatigue resulting in lower lives than obtained with solid balls (refs. 4 and 5).

Another approach to reducing ball mass is the use of drilled or cylindrically hollow balls. The balls are constrained from misorientation in the bearing raceways by guide pins or studs (ref. 6).

Two other techniques that can be used to improve high-speed ball bearing life involve the coupling of a fluid-film bearing with the ball bearing. The first approach couples the fluid-film and ball bearings in parallel so they share the thrust load. This concept is called the "hybrid-boost" bearing (ref. 7).

In theory, a more effective extreme speed hybrid bearing can be obtained by coupling the ball bearing and the fluid-film bearing in series. In the series hybrid bearing, each component bearing carries the full system load but the two bearings share the speed. One element of the fluid-film bearing rotates at shaft speed. The second element of the fluid bearing rotates with the inner race of the ball bearing at a speed less than shaft speed. The ball-bearing outer race is mounted in a stationary housing. The intermediate member rotates at a speed such that the torques of the fluid-film and ball bearings are equal.

The object of this investigation was to experimentally evaluate the series hybrid rolling-element bearing concept. Tests were performed with a 75-mm bore angular-contact ball bearing having a fluid-film bearing attached to the inner race. Speeds and thrust loads were varied to determine the magnitude of speed reduction that could be achieved.

The hybrid series rolling-element bearing was evaluated in a high-speed, turbine-driven bearing test rig. Test conditions included shaft speeds of 4000 to 30 000 rpm, bearing thrust loads from 100 to 300 pounds (445 to 1340 N), and a type II ester fluid as the lubricant. The rolling-element bearing was a 115-series ball bearing. Test results were evaluated with respect to speed differential between the drive shaft and the ball-bearing inner race.

BACKGROUND

The magnitude of the high-speed bearing problem is evident from the curves of figure 1. Figure 1 illustrates the effect of DN on the fatigue life of a thrust loaded 150-mm bore ball bearing at four values of thrust load. These curves are based on the analysis of references 2 and 8. An increase in speed from a DN of 1.8 million to 4.2 million results in a reduction in life of 98.8 percent at 1000 pounds (4450 N) load and 96 percent at 4000 pounds (17 800 N) load. This is the approximate range of thrust loads which such a bearing would carry in an aircraft turbine engine.

Since high centrifugal forces created by the extreme speeds is largely responsible for the drastic reduction in predicted fatigue life at high DN values, it is logical to consider methods for reducing ball mass. It is presumed that the geometry optimization has yielded the optimum ball diameter so that life cannot be further improved by reducing ball diameter to obtain lower ball mass. Further improvements in life can be achieved only by maintaining the ball diameter at its optimum, and by reducing ball mass by hollowing the ball.

The theoretical analysis of references 2 and 8 indicates that considerable improvement in fatigue life can be obtained at high DN values by reducing ball mass. Figure 2 illustrates the effect of a 50-percent reduction in ball weight (constant ball diameter) for a 150-mm bore ball bearing. At 2000 pounds (8900 N) thrust load, life is improved by a factor of 4.2 at 3 million DN and by a factor of 5.5 at 4 million DN. Life improvement factors at 4000 pounds (17 800 N) thrust load are 2.5 at 3 million DN and 4.2 at 4 million DN. It is evident from this theory that hollow balls are most effective in improving fatigue life at high speeds and low loads. This is further illustrated in figure 3. Again,

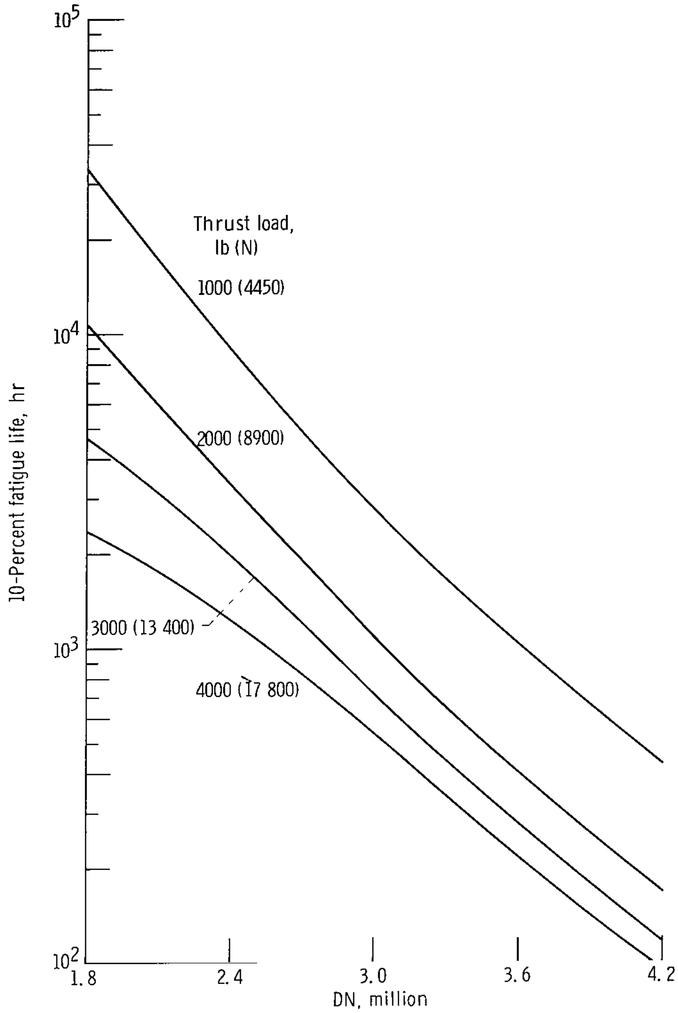


Figure 1. - Theoretical fatigue life of a thrust-loaded 150-mm bore ball bearing (based on analysis of refs. 2 and 8).

these curves are based on the analysis of references 2 and 8. Life improvement ratios approach 6 for 50 percent hollow balls at the highest DN (4.2 million) and the lowest load (1000 lb (4450 N)).

Thin wall, spherically hollow balls have been fatigue tested (ref. 3) and also evaluated in full-scale bearings (refs. 4 and 5). These balls were made by electron-beam welding two hemispheres. In reference 3 the 0.5-inch- (12.7-mm-) diameter, 0.1 inch- (2.5 mm-) wall thickness balls exhibited a rolling-element fatigue life comparable to that of solid balls made from the same heat of material. In reference 4, however, 0.687-inch- (17.5-mm-) diameter, 0.060-inch- (1.5-mm-) wall thickness balls failed in flexure fatigue because of the stress concentrations at the inside diameter in the weld area. It is questionable at this time whether thin wall hemispheres can be joined with perfect

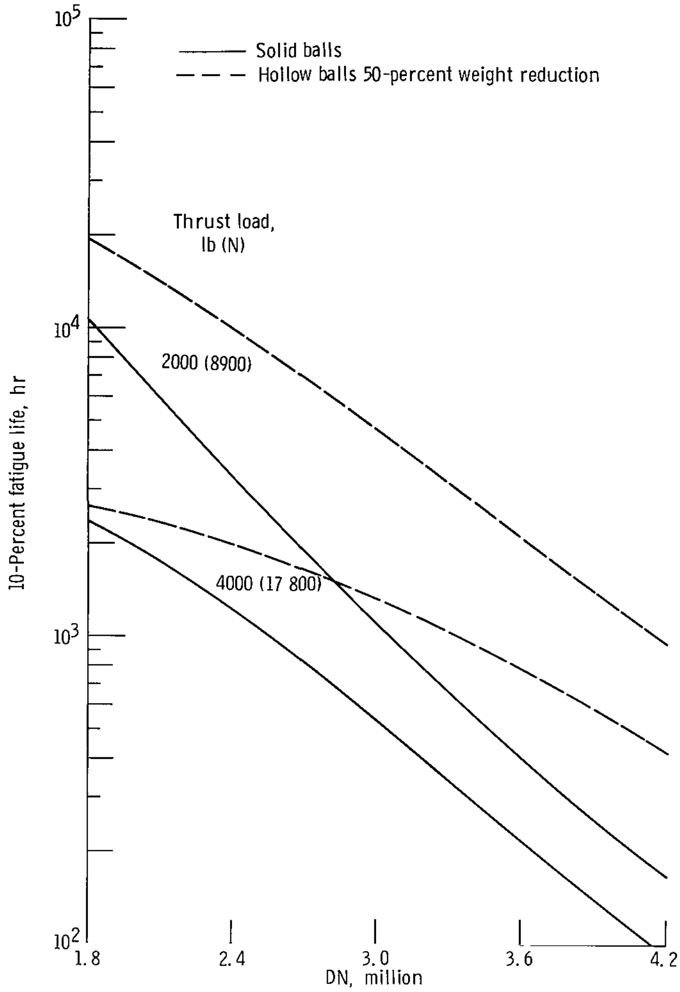


Figure 2. - Theoretical fatigue life of a thrust-loaded 150-mm bore ball bearing with solid and with 50-percent hollow balls (based on analysis of refs. 2 and 8).

weld integrity and completely free of any internal stress concentrations. Further work on joining techniques needs to be done. In addition, even if all the fabrication problems can be solved, there remains the possibility of standing or travelling waves causing flexure failures of the wall at high frequencies of stressing. An analysis of this potential problem has not as yet been made.

Another approach to reducing ball mass is the use of drilled or cylindrically hollow balls. Precise, concentric holes that are free of scratches or other stress concentrations can easily be made in finished balls. The disadvantage of this concept is that drilled balls require guide pins or studs to prevent ball misorientation. Ball guide pin contact wear may limit life. Cylindrically hollow balls may also be subject to flexure

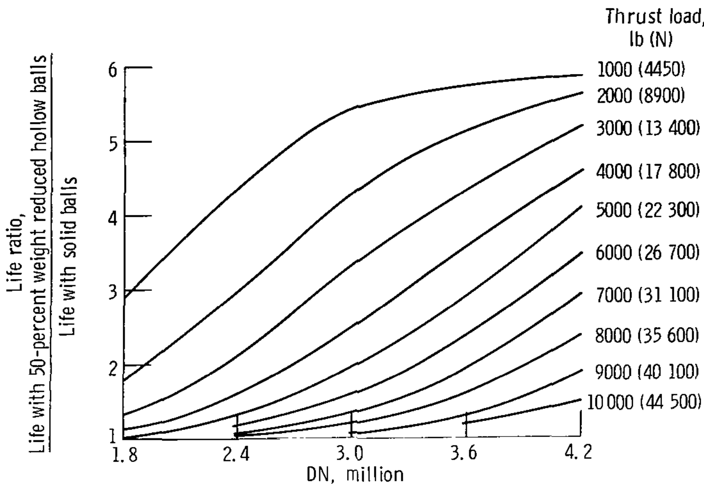


Figure 3. - Theoretical fatigue life improvement of 150-mm bore ball bearings having balls with 50-percent weight reduction (based on analysis of refs. 2 and 8).

failures at high frequencies of stressing. Ball bearings fitted with drilled balls have been successfully run at DN values to 2.1 million (ref. 6). Some guide pin wear was observed, but there were no signs of abnormal ball-race tracking.

Two other techniques that can be used to improve high-speed ball bearing life involve the coupling of a fluid-film bearing with the ball bearing. The first approach couples the fluid-film and ball bearings in parallel so they share the thrust load. Since the ball bearing carries a lower load, its life is increased. This concept, called the "hybrid-boost" bearing, was evaluated in reference 7. The primary disadvantage of this concept is the high power loss in the fluid-film bearing. This high power loss is critical in aircraft turbine engines where the ability to reject heat is limited. In addition, its effectiveness at very high speeds and low system thrust loads is limited. Turbine engine thrust bearings are lightly loaded under cruise conditions. Furthermore, there is a lower bound of safe ball bearing load at the point of impending skidding.

In theory a more effective extreme speed hybrid bearing can be obtained by coupling the ball bearing and the fluid-film bearing in series. In the series hybrid bearing, each component bearing carries the full system load but the two bearings share the speed. One element of the fluid-film bearing (fig. 4) rotates at shaft speed. The second element of the fluid-film bearing rotates with the inner race of the ball bearing at a speed less than shaft speed. The outer race of the ball bearing is mounted in the stationary housing. The intermediate member rotates at a speed such that the torques of the fluid-film and ball bearings are equal.

The potential benefits of the series-hybrid bearing are illustrated in figure 5. These curves were obtained from those of figure 2. For the hollow ball bearing the life ratio

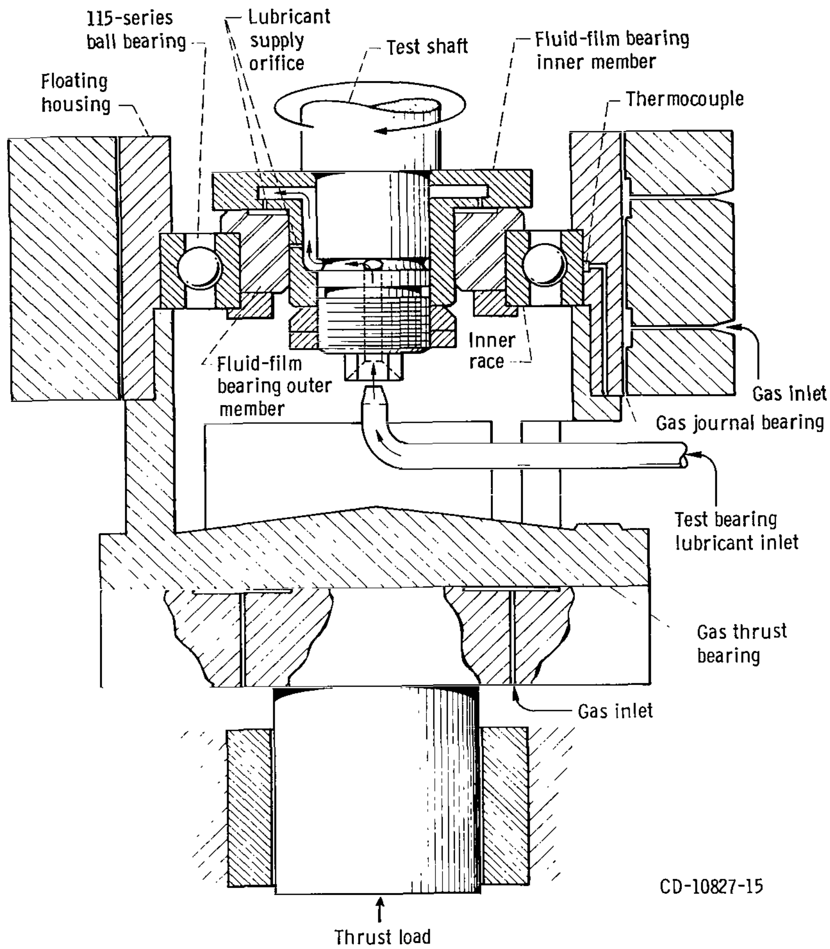


Figure 4. - Cross section of series-hybrid bearing installation.

at each DN is obtained by comparing the 4000 pound (17 800 N) dashed and solid line curves of figure 2. For the parallel hybrid bearing, the 2000 and 4000 pound (8900 and 17 800 N) solid line curves of figure 2 are compared. For the series-hybrid bearing the 4000 pound (17 800 N) solid line curve is used, with the lines at a particular DN and at 30 percent lower DN used to obtain each life ratio.

At 3 million DN with 30-percent speed reduction the series-hybrid bearing improves life 3.2 times, and at 4 million DN the life improvement factor is 5.7. With hollow balls at 3 million DN the life improvement factor is 2.5, and with 50-percent load reduction with the parallel hybrid bearing life improves only 1.8 times. The series-hybrid bearing is considerably more effective than either hollow balls or the parallel hybrid bearing. This concept does introduce considerable mechanical complexity, but its potential for

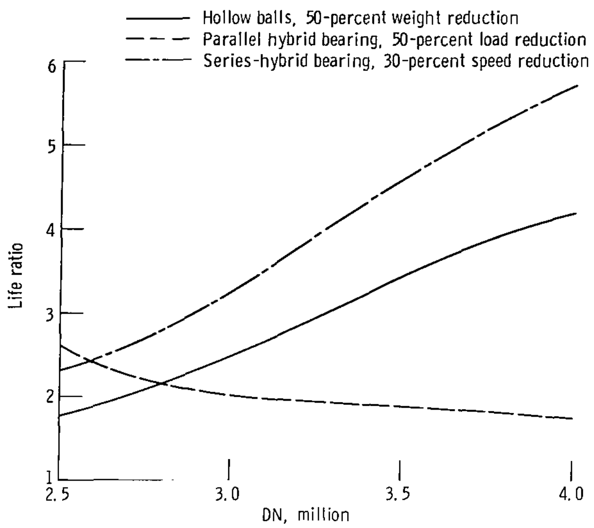


Figure 5. - Theoretical life improvement factors as function of DN for a 150-mm bore ball bearing with hollow balls, a parallel-hybrid bearing, and a series-hybrid bearing. Thrust load, 4000 pounds (17 800 N).

TABLE I. - APPROACHES TO HIGH-SPEED BALL-BEARING PROBLEM

Approach	Problems and limitations
Hollow balls	
Spherically hollow	(1) Fabrication difficulties (weld integrity, internal stress concentrations, unbalance) exists. (2) Wall flexibility and reduced stiffness occur at weight reductions greater than 50 percent. (3) Possibility exists of standing or travelling waves causing flexure failures of the wall at high frequencies of stressing.
Drilled or cylindrically hollow	(1) Balls require guide pins or studs to prevent misorientation. Ball-guide pin contact wear may limit life. (2) Balls may be subject to flexure failures at extreme speeds.
Parallel hybrid or hybrid boost	(1) High power loss exists in fluid-film bearing. (2) Effectiveness at very high speeds and low system loads is limited; lower bound of ball bearing load is governed by skidding. (3) Some mechanical complexity exists. (4) Bearing sensitive to oil system failure. (5) Bearing not completely failsafe - must maintain separation in the fluid-film bearing at all times.
Series hybrid	(1) Considerable mechanical complexity exists. (2) Speed reduction is limited by torque characteristics of fluid-film bearing. (3) Moderate power loss exists in fluid-film bearing. (4) Bearing sensitive to oil system failure.

extreme speed applications makes it worthy of investigation. Table I lists some of the problems and limitations of each of the approaches discussed.

APPARATUS AND PROCEDURE

Test Bearing

The series-hybrid bearing tested in this program consists of a fluid-film bearing (with both thrust and journal surfaces) attached to the inner race of a 75-mm bore (115-series) deep-groove ball bearing (fig. 4). The inner member of the fluid-film bearing rotates with the shaft. The outer member of the fluid-film bearing rotates with the inner race of the ball bearing at part speed. The outer race of the ball bearing is stationary.

Design of the fluid-film thrust bearing. - The centrifugal force on the oil in the rotating shaft can be used to generate hydraulic pressure to supply a pressurized thrust bearing. As soon as enough hydraulic pressure is generated to lift the applied thrust load, sliding will commence within the fluid-film bearing; the starting torque is zero. The relative rotational speed will increase until the torque required equals the torque of the ball bearing. Zero starting torque is particularly advantageous since the torque available is limited to that of the ball bearing.

The analysis and computer program of reference 9 were used to evaluate fluid-film thrust bearing designs. The analysis is for an orifice-compensated annular thrust bearing. A line feed and laminar conditions are assumed, and rotational effects are included. The lubricant supply pressure was taken as the pressure that could be developed from centrifugal effects at the radius of the orifices.

Because the test program constituted an initial effort to evaluate a concept, optimization of the fluid-film thrust bearing for the anticipated test conditions (300 lb (1340 N) thrust load and 30 000 rpm) was not attempted. Therefore, the bearing had ample thrust load capacity, but it could not attain the target speed split (50-50) between the ball bearing and fluid-film bearing. A small self-acting journal bearing was provided to take any radial load that might be present.

After the first data were taken, the intermediate member of the fluid-film bearing was reduced in diameter from 3.35 to 2.80 inches (85 to 71 mm) in order to lower the running torque and increase the relative speed of the fluid-film bearing. The journal bearing was also shortened and two small lubricant supply holes were provided for the journal bearing.

Dimensions of the fluid-film bearing for the original and modified series-hybrid bearing designs are given in table II.

TABLE II. - FLUID-FILM BEARING SPECIFICATIONS

	Original design	Modified design
Thrust bearing outside diameter, in. (mm)	3.35 (85)	2.80 (71)
Thrust bearing inside diameter, in. (mm)	1.75 (44.4)	1.75 (44.4)
Number of orifices	4	4
Orifice diameter, in. (mm)	0.010 (0.254)	0.010 (0.254)
Orifice locating diameter, in. (mm)	2.4 (61)	2.4 (61)
Journal bearing diameter, in. (mm)	1.75 (44.4)	1.75 (44.4)
Journal bearing diameter clearance, in. (mm)	0.002 (0.051)	0.002 (0.051)
Journal bearing length, in. (mm)	0.86 (21.8)	0.61 (15.5)

Rolling-element bearing. - The rolling-element bearing portion of the series-hybrid bearing is a 115-series deep-groove ball bearing. One shoulder of the outer race was relieved to make the bearing separable. The ball and race material was consumable-electrode vacuum melt 52100 steel. The one-piece, machined, inner-race-located retainer was silver-plated bronze. Specifications of the ball bearing are as follows:

Bearing tolerance ABEC-5
Internal radial clearance, in. (mm) 0.0027 (0.069)
Number of balls 14
Ball diameter, in. (mm) 0.500 (12.7)
Nominal contact angle, deg. 15
Conformity, percent
  Inner race 54
  Outer race 54
Retainer-land diametral clearance, in. (mm) 0.025 (0.63)

Test Apparatus

A cutaway view of the test apparatus is shown in figure 6. A 4-inch (10-cm) air turbine drives the test shaft through a quill shaft. The test shaft was mounted vertically with a support ball bearing at the upper end and the test bearing at the lower end. Maximum speed capability of the apparatus was about 60 000 rpm. A pneumatic cylinder loaded the bearing through an externally pressurized gas thrust bearing.

The bearing torque was measured by an unbonded strain-gage force transducer connected to the periphery of the test bearing housing assembly, as shown in figure 6. The

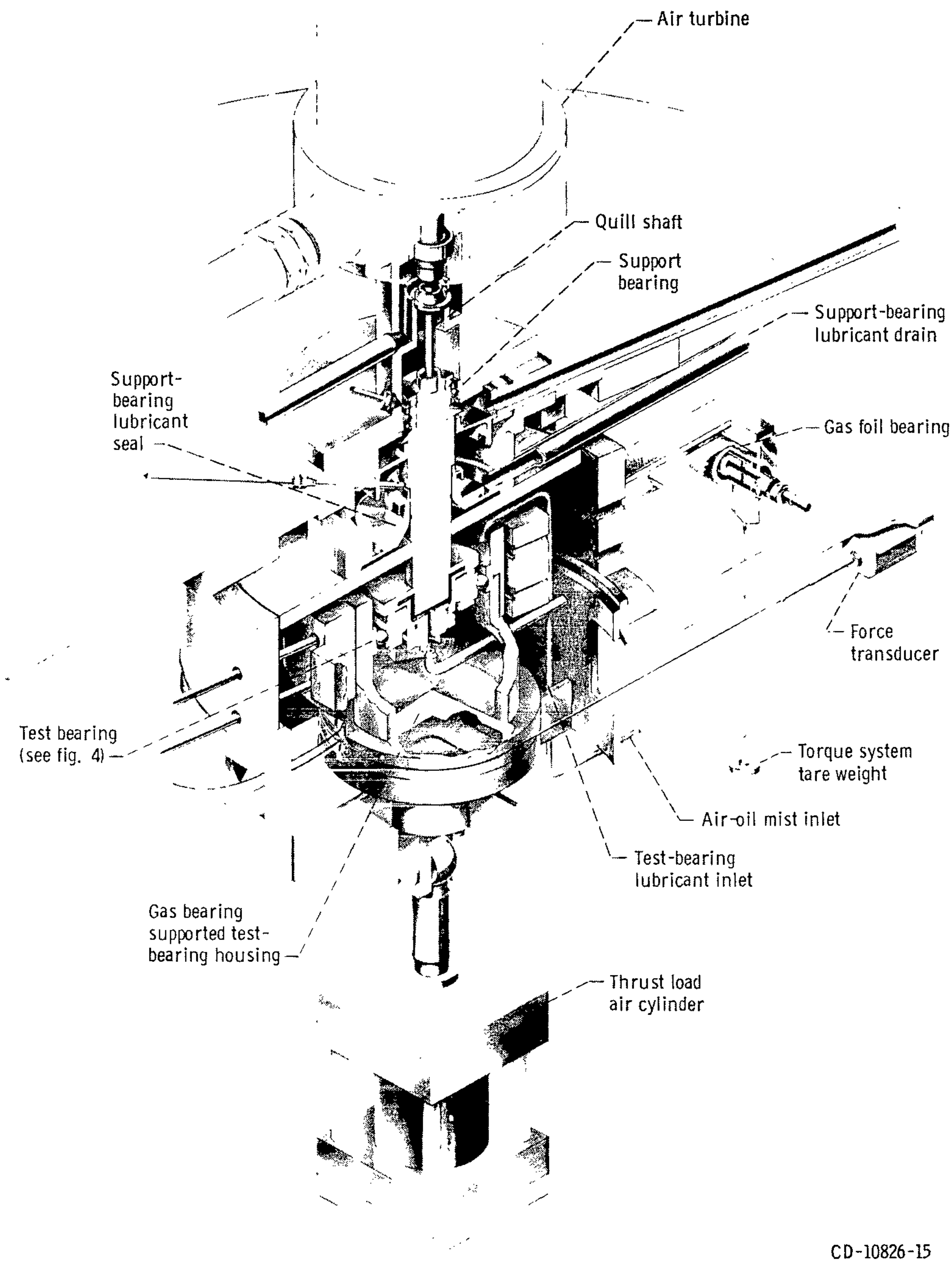


Figure 6. - Test apparatus.

ball bearing outer-race temperature was measured with a thermocouple embedded in a copper bead in the test bearing housing.

The lubricant used for these tests was a type II ester. The test bearing was lubricated as shown with a 0.10-inch- (2.5-mm-) diameter jet guiding oil into the shaft. The shaft rotation pumped the oil to the fluid-film bearing. The oil leaving the fluid-film thrust bearing after liftoff flowed to the ball bearing. In addition, the ball bearing was lubricated with an air-oil mist.

Procedure

During assembly, all surfaces of the ball bearing and the fluid-film bearing were coated with oil. After assembly and checkout of all systems, the test shaft was slowly brought up to a speed less than 4000 rpm while system temperatures stabilized. The oil tank heaters were energized, and the oil-in temperature was controlled to 140° F (330 K). When all temperatures stabilized (after about 90 min), conditions were set for the first data point at either 4000 or 5000 rpm. Data were subsequently taken as shaft speed was increased in 1000- to 2000-rpm increments up to 30 000 rpm. Conditions of temperature equilibrium were reached after about 10 to 20 minutes at each speed. The ball-bearing inner-race speed and cage speed were recorded at each point. Bearing torque and outer-race temperature were continuously recorded on strip-chart recorders. The shaft speed and the ball-bearing inner-race speed were also observed on a dual-trace cathode-ray oscilloscope.

RESULTS AND DISCUSSION

Experimental Results with Initial Design

The series-hybrid rolling-element bearing was run at thrust loads from 100 to 300 pounds (445 to 1340 N) and speeds from 4000 to 30 000 rpm. At each equilibrium condition (described previously) the ball-bearing inner-race speed was either the same as or less than the shaft speed. The intermediate speed member of the hybrid bearing rotates with the inner race of the ball bearing, such that the difference in speed between the shaft and the inner race is the speed of the fluid-film bearing.

Ball-bearing inner-race speed is plotted in figure 7 as a function of shaft speed for each of four thrust loads. For each load, the inner race initially rotates at the shaft speed. As shaft speed increases, a point is reached where the fluid-film bearing lifts

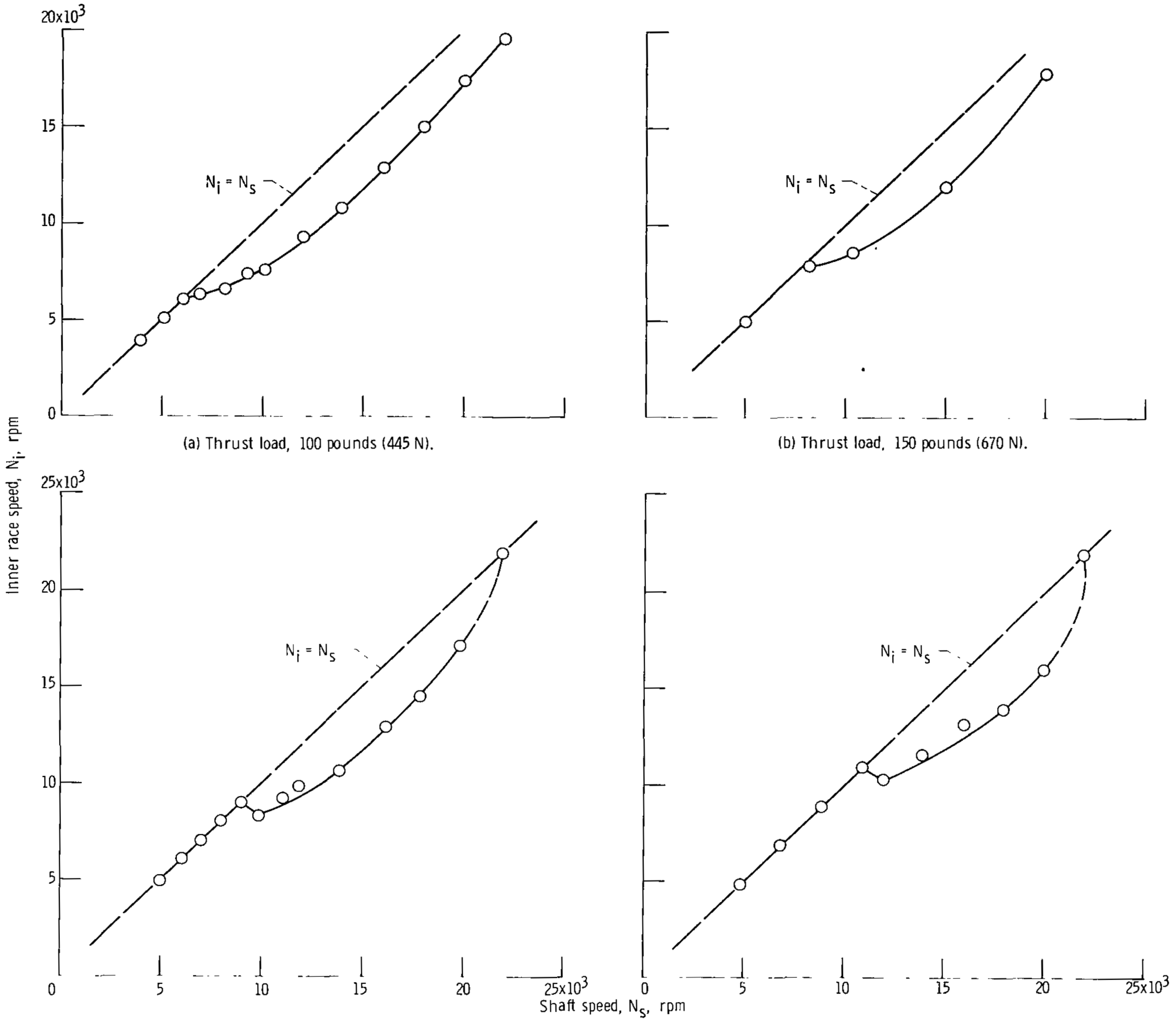


Figure 7. - Inner-race speed as function of shaft speed for original series-hybrid bearing.

off. These liftoff points are seen in figures 7(a) to (d) where the data indicate inner-race speed and shaft speed are no longer equal.

As thrust load increases, the liftoff speed increases as shown in figure 8. In this figure, the speed ratio N_i/N_s is plotted against shaft speed (N_i is inner-race speed and N_s is shaft speed). The trend of higher liftoff speed with higher thrust load is expected since the load capacity of the fluid-film bearing depends on the lubricant supply pressure. This pressure increases as the square of the shaft speed. Thus, for higher thrust loads, greater hydrostatic pressures (and thus higher shaft speeds) are required for liftoff.

Also shown in figure 8 is the variation in speed ratio N_i/N_s as shaft speed increases beyond the liftoff range. The speed ratio drops rapidly after liftoff but in all cases reaches a minimum value and then begins to increase. Minimum speed ratios were in the range of 0.76 to 0.80. For example, with a shaft speed of 16 000 rpm, the inner-race speed would be about 12 000 rpm. Or, in terms of speed sharing, the ball bearing is rotating at 12 000 rpm, and the effective fluid-film bearing speed is 4000 rpm. This speed of 4000 rpm appears to be an upper limit for this fluid-film bearing, conceivably because of turbulence in the fluid film. Turbulence can increase the drag of the fluid-film bearing (ref. 10).

In the tests at 200- and 300-pound (890 and 1340 N) thrust loads, the fluid-film bearing ceased to operate when the shaft speed reached 22 000 rpm. At this point, the ball-bearing inner-race speed increased to the shaft speed. This effect was accompanied by an oscillation in bearing torque which indicated a possible critical speed of the shaft assembly. (This oscillation was also noted at 22 000 rpm in later tests with a modified bearing.) After testing, the bearing was disassembled, and a slight scuffing of the journal bearing surface was noted which could have been a result of this operating condition.

The torque of the series-hybrid bearing is plotted as a function of inner-race speed in figure 9. When the fluid-film bearing lifts off, an abrupt increase in torque occurs.

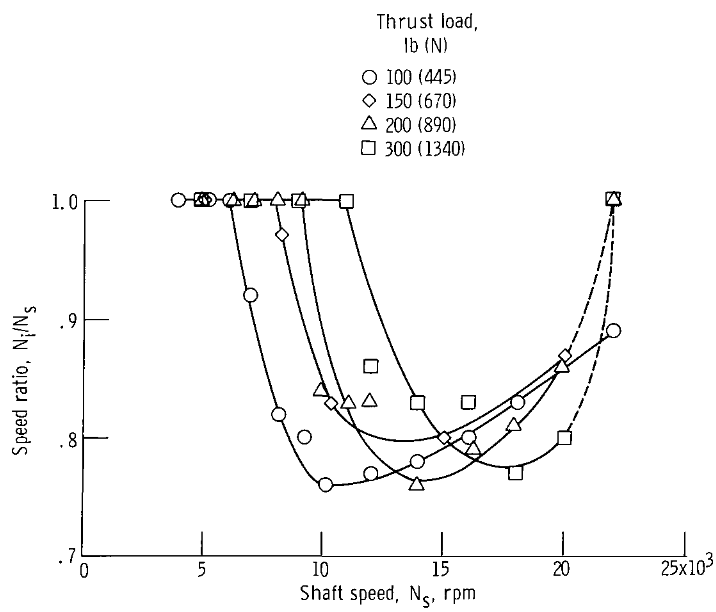
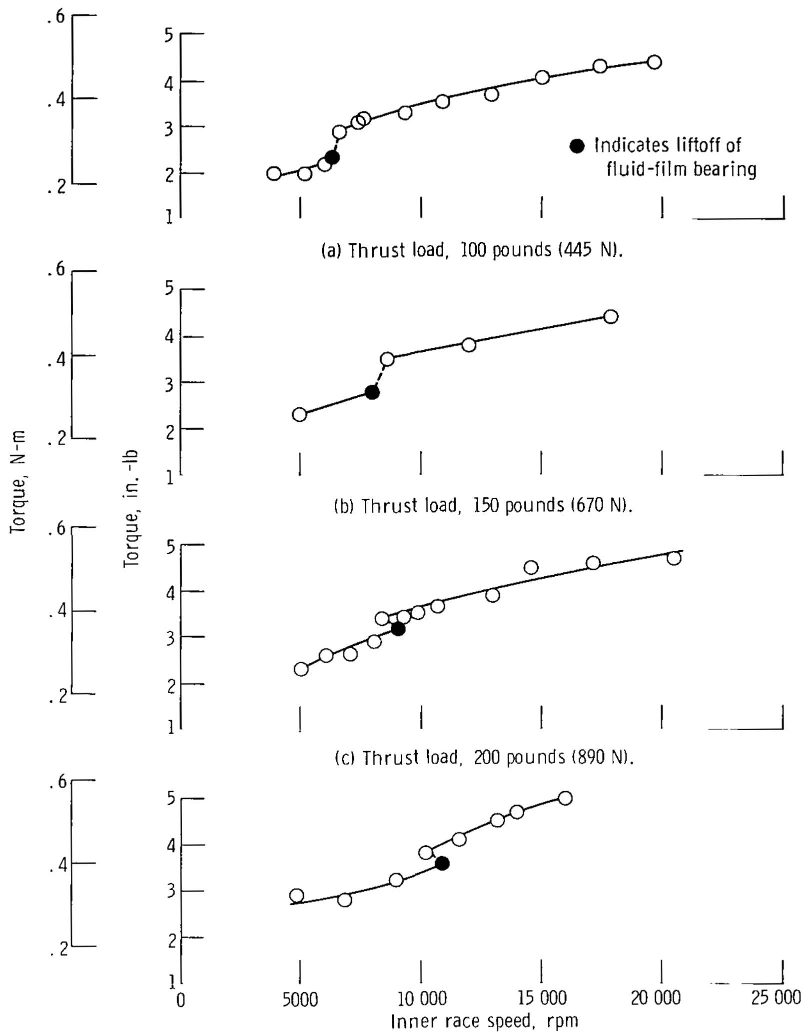


Figure 8. - Ratio of inner-race speed to shaft speed as function of shaft speed for original series-hybrid bearing.



(d) Thrust load, 300 pounds (1340 N).

Figure 9. - Torque of original series-hybrid bearing as function of speed of ball-bearing inner race.

The lubricant leaving the fluid-film thrust bearing also passes through the ball-element bearing. This flow increases greatly when the fluid-film bearing lifts off. The increase in oil flow through the ball-element bearing results in increased torque due to oil drag. The torque increase at liftoff appears to be greater at lower loads than at higher loads. The magnitude of the torque increase should be a function of the oil flow and the viscosity which are dependent on liftoff speed and load.

The temperature at the outer race of the ball bearing is plotted as a function of inner-race speed in figure 10 for each equilibrium condition. Fluid-film bearing liftoff affects this temperature as shown by the apparent discontinuity at the liftoff speed. This effect

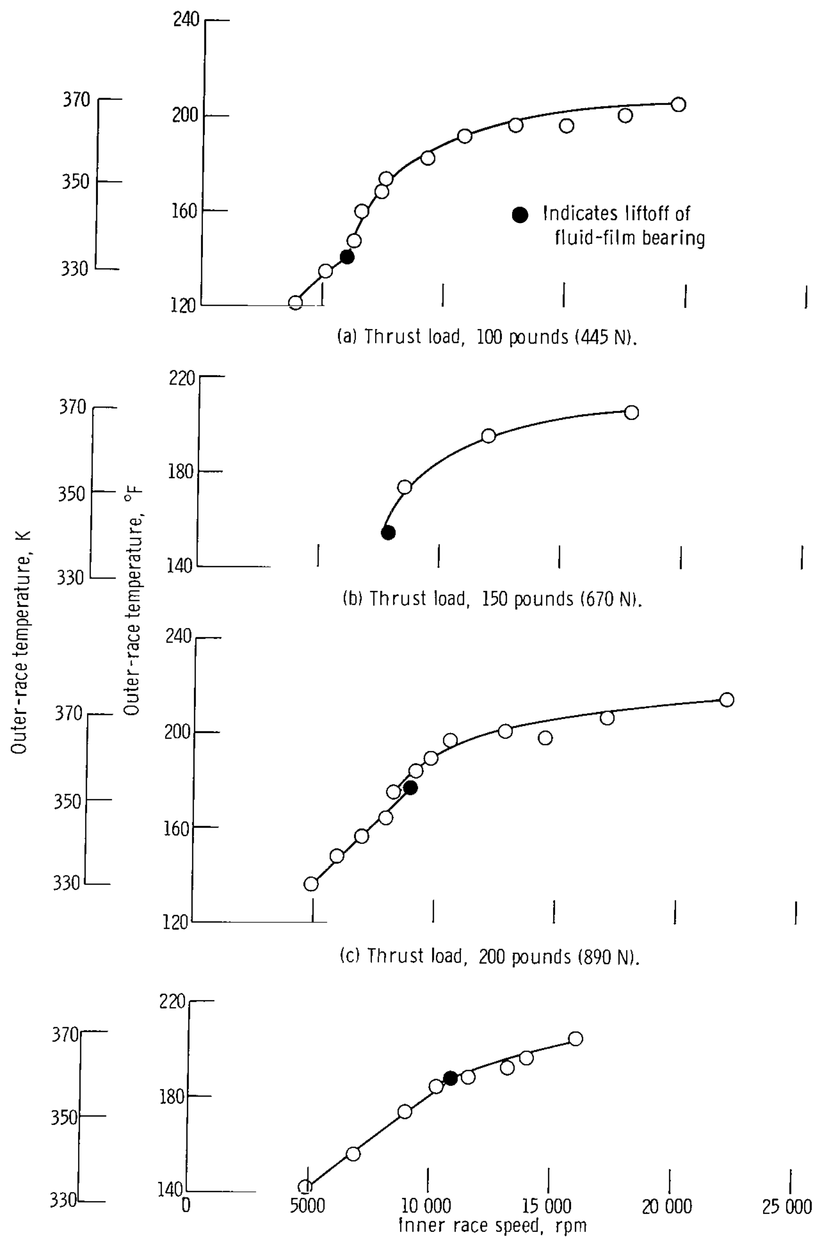


Figure 10. - Equilibrium temperature as measured at outer race of bearing as function of inner-race speed for original series hybrid bearing. Oil inlet temperature, 140 F (330 K).

appears to be greater with lower loads than with the higher loads, corresponding to the effect of liftoff on bearing torque. The effect is not seen for the 150-pound (670-N) thrust load because an outer-race temperature was not recorded below the liftoff speed.

Table III shows bearing torque and outer-race temperature data measured with the ball bearing alone with mist lubrication and with jet lubrication. The torque of the series-hybrid bearing (fig. 9) approximates that shown in table III for jet lubrication. The outer-race temperature of the series-hybrid bearing above liftoff speeds fell between those measured for the ball bearing alone with mist lubrication and with jet lubrication.

TABLE III. - RESULTS OF TESTS WITH 115-SERIES BALL BEARING TYPE II ESTER LUBRICANT

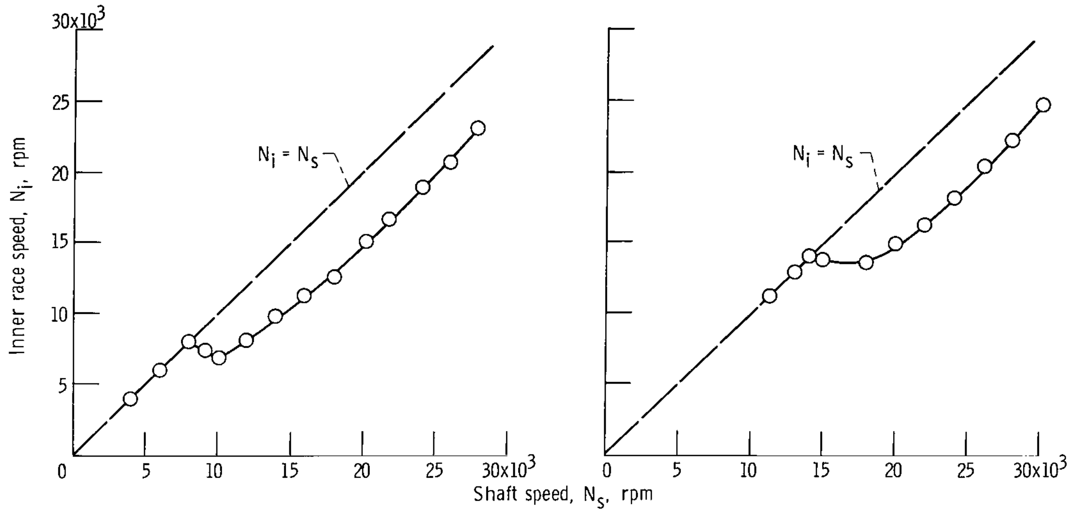
Oil in temperature, 100° F (310 K)				
Shaft speed, rpm	Thrust load, lb (N)	Lubrication system	Torque, in.-lb (N-m)	Outer-race temperature, $^{\circ}$F (K)
10 000	200 (890)	air-oil mist[a]	2.8 (0.32)	145 (336)
20 000	150 (670)	air-oil mist[a]	1.4 (0.16)	165 (347)
20 000	300 (1340)	air-oil mist[a]	1.6 (0.18)	183 (357)
10 000	150 (670)	jet[b]	4.2 (0.48)	161 (345)
15 000	150 (670)	↓	4.2 (0.48)	200 (366)
15 000	300 (1340)		4.4 (0.50)	211 (373)
20 000	150 (670)		3.7 (0.42)	221 (378)
20 000	300 (1340)		3.4 (0.38)	234 (385)

[a]Air-oil mist lubrication system; oil flow rate, 0.002 lb/min (0.015 g/sec).

[b]Jet lubrication; single nozzle; oil flow rate, 0.5 lb/min (3.8 g/sec).

Experimental Results with Modified Design

The fluid-film bearing was modified for subsequent tests. The modification consisted of a smaller diameter thrust bearing, a shorter journal bearing length, and lubricant feed holes for the journal bearing. This modified bearing proved to be superior to the initial design in speed sharing capabilities when tested in the same range of test conditions. The results are shown in figures 11 and 12. Ratios of ball-bearing inner-race speed to shaft speed as low as 0.67 were obtained, and the speed of the ball bearing was reduced nearly 6000 rpm below the shaft speed at one point. (With the original series-hybrid bearing, the maximum speed reduction was 4000 rpm.)



(a) Thrust load, 100 pounds (445 N).

(b) Thrust load, 300 pounds (1340 N).

Figure 11. - Inner-race speed as function of shaft speed for modified series-hybrid bearing.

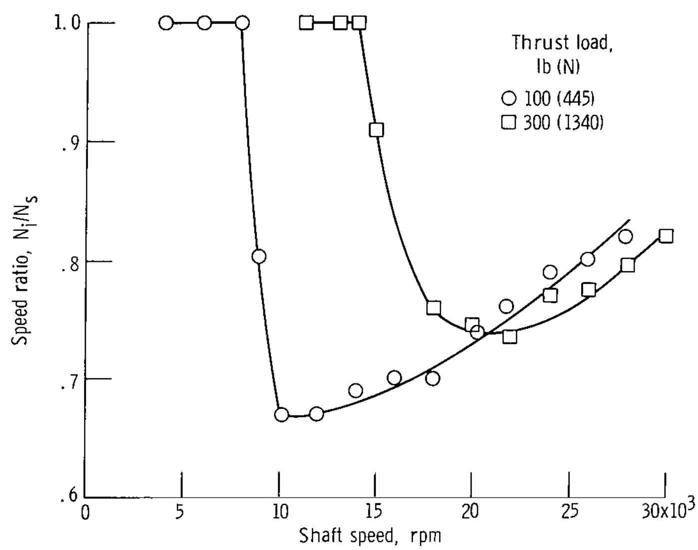


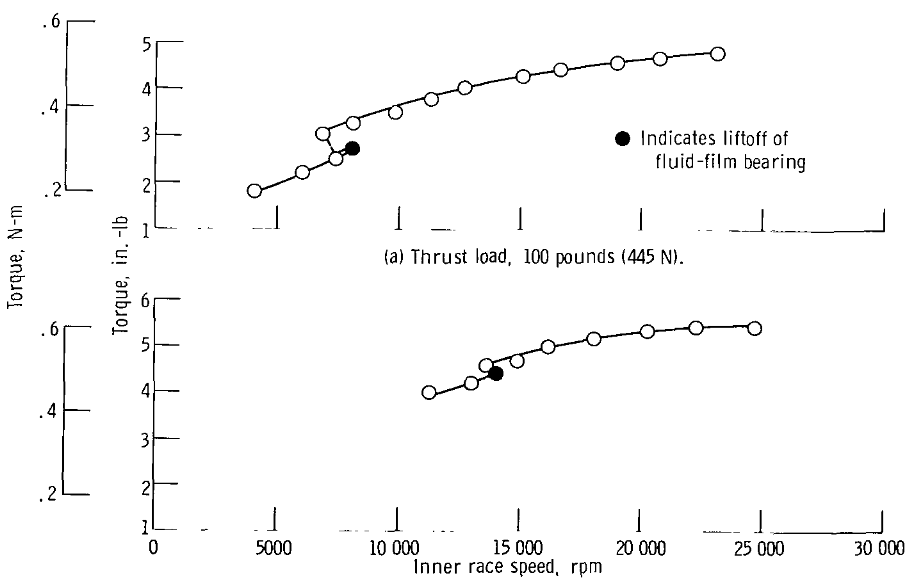
Figure 12. - Ratio of inner-race speed to shaft speed as function of shaft speed for modified series-hybrid bearing.

Liftoff speed was higher with the modified bearing, and the maximum speeds attained were greater (i.e., 30 000 rpm shaft speed). No significant differences in the torque and outer-race temperature measurements were noted between the initial and the modified versions (figs. 13 and 14). The same effects of liftoff on torque and temperature are apparent.

Comparison of Theoretical and Experimental Results

Figures 15 and 16 show the bearing inner-race speed plotted against shaft speed for the original (fig. 15) and modified (fig. 16) bearings. The theoretical curves (from the analysis and computer program of ref. 9) were calculated from the measured bearing torque and an oil viscosity corresponding to the measured ball-bearing outer-race temperature.

When the liftoff speed is reached, according to the analysis, the inner-race speed drops abruptly. Inner-race speed varies little with applied load, but is sensitive to lubricant viscosity. The 300-pound (1340-N) curve lies below the 100-pound (445-N) curve in figure 16 since the oil was hotter (hence, less viscous) during the 300-pound (1340-N) run. For the experimental data in figure 15 temperatures did not vary as widely with applied load. A mean temperature was used in the calculations. Liftoff speeds are higher for the modified bearing (fig. 16) because of its smaller area.



(b) Thrust load, 300 pounds (1340 N).

Figure 13. - Torque of modified series-hybrid bearing as function of inner-race speed.

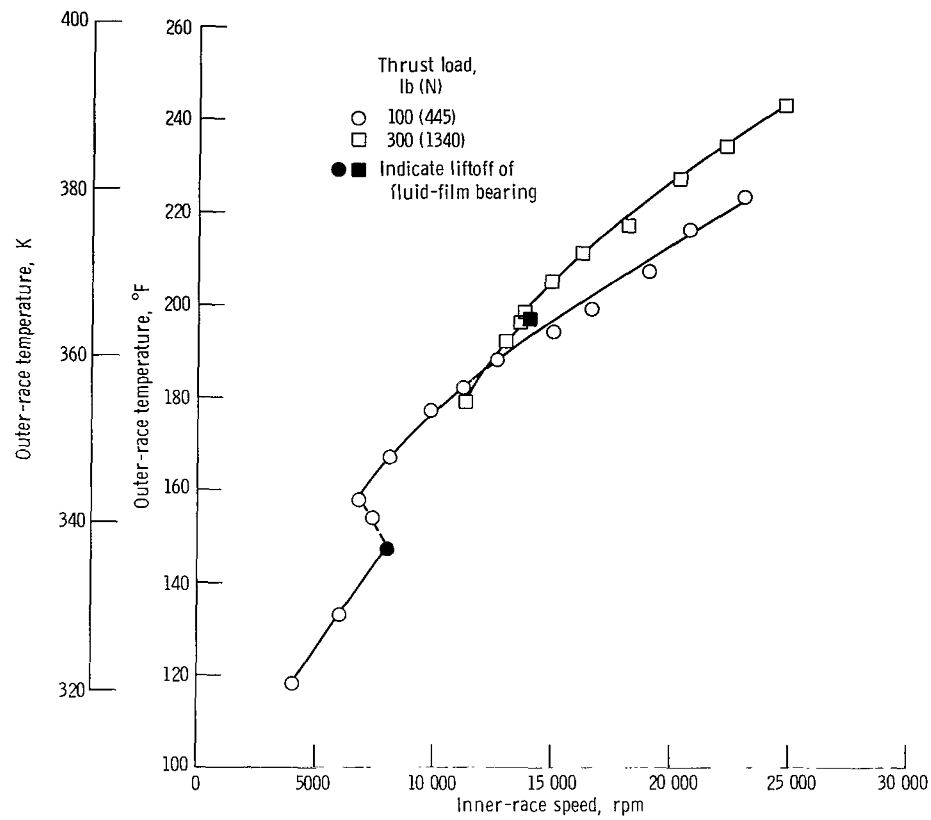


Figure 14. - Equilibrium temperature as measured at outer race of the ball bearing as function of inner-race speed with modified series-hybrid bearing.

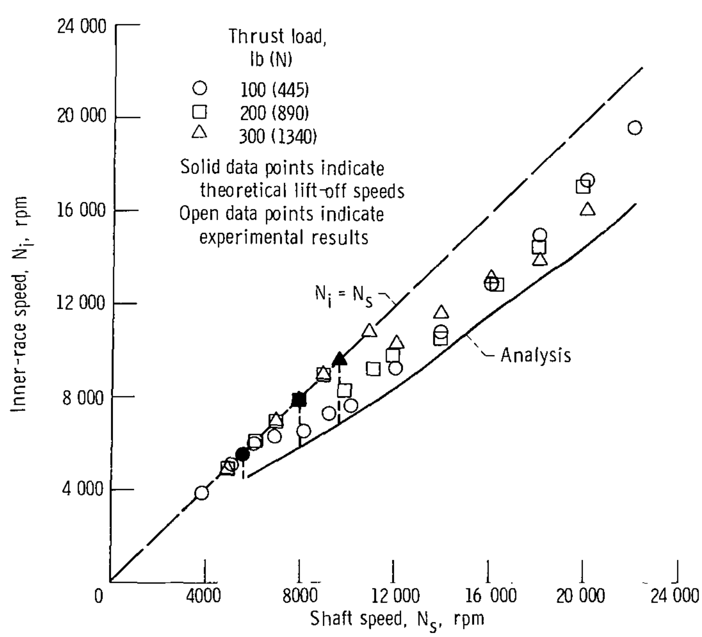


Figure 15. - Comparison of theoretical and experimental speed sharing results for original series-hybrid bearing.

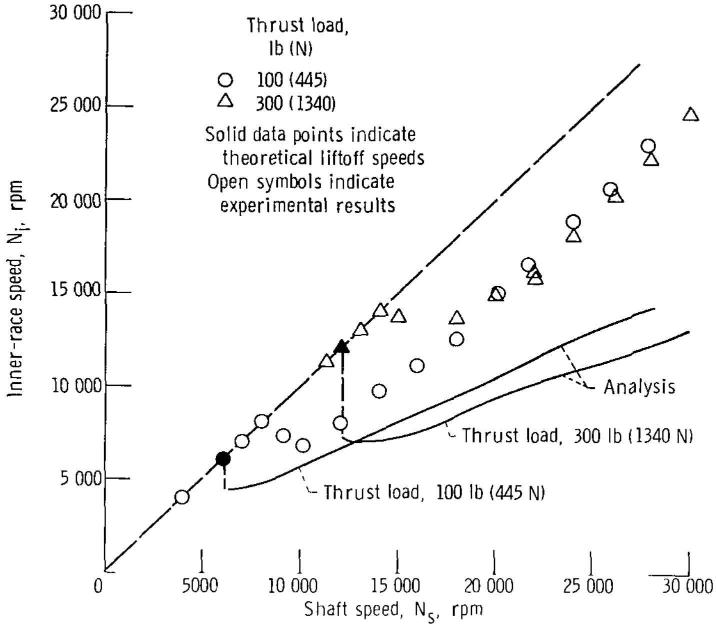


Figure 16. - Comparison of theoretical and experimental speed sharing results for modified series-hybrid bearing.

Experimental liftoff speeds and inner-race speeds are higher than predicted by theory. Also, the experimental points indicate a nearly constant difference between shaft speed and inner-race speed, whereas the analysis shows the difference increasing with shaft speed. Agreement between theory and experiment is better for the original series-hybrid bearing than for the modified bearing, though, as expected, inner-race speeds are lower for the modified bearing. Higher experimental thrust loads generally resulted in lower inner-race speeds, showing the effect of higher temperatures on the fluid-film bearing.

Possible reasons for the disparity between analysis and experiment are the following:

(1) The analysis assumes a line source of pressurized lubricant, whereas the actual bearing has only four orifices.

(2) The fluid-film bearing may be cooler than the ball-bearing outer race.

(3) There may be some turbulence in the fluid film at higher speeds. The highest Reynolds number, based on the clearance in the bearing recess, was 1240. Reference 11 gives a critical Reynolds number of 1000 for thrust bearings.

Analysis of Fatigue Life Effects

The potential increase in bearing fatigue life due to a reduction in the effective DN value can easily be determined from figure 1. If the 150-millimeter bore ball bearing of figure 1 is operating at 3 million DN and 1000 pounds (4450 N) thrust load, its expected 10-percent fatigue life (the life which will be exceeded by 90 percent of the bearings) is 2750 hours. If the same bearing is operated in a series-hybrid arrangement with a speed ratio of 0.67, then its effective DN is approximately 2 million and its 10-percent fatigue life 21 000 hours. The life improvement factor is thus 7.6.

If the DN value in a bearing application is 4 million, the thrust load 1000 pounds (4450 N), and the speed ratio 0.67, then the life improvement factor is approximately 9.

The bearing tested is, of course, a concept and the difficulties of achieving comparable or even lower speed ratios at extreme DN values have not been examined in detail. In addition, the mechanical complexity poses a problem in applying the concept, and decreases in shaft diameter required to pass the shaft through the ball-bearing location have not been accounted for.

SUMMARY OF RESULTS

A series-hybrid bearing composed of a fluid-film bearing coupled in series with a ball bearing was run at thrust loads from 100 to 300 pounds (445 to 1340 N) and speeds from 4000 to 30 000 rpm. One element of the fluid-film bearing rotated at shaft speed. The second element of the fluid-film bearing rotated with the inner race of the ball bearing at a speed less than the shaft speed. The ball-bearing outer race was stationary.

The fluid-film bearing consisted of an orifice-compensated annular-thrust bearing and a self-acting journal bearing. Oil for the fluid-film bearing was supplied through the shaft center. Centrifugal effects were utilized for pressurization of the fluid-film bearings. The rolling-element bearing was a 115-series (75-mm bore) deep-groove ball bearing. Experimental speed sharing results were compared with theoretical calculations. The following results were obtained:

1. At shaft speeds great enough to lift off the applied thrust load, the inner race of the ball bearing rotated at a speed less than the shaft speed.

2. The lowest speed ratio (inner-race speed to shaft speed) obtained was 0.67. This corresponds to an approximate reduction in DN value of 1/3. For a ball bearing in a 3 million DN application, fatigue life would be improved by a factor as great as 8.

3. Experimental speed ratios were smaller than theoretical predictions. Deviation of experimental from predicted speed ratios was greatest at higher speeds.

4. Experimental liftoff speed increased with increased thrust load and were only slightly higher than theoretical predictions.

5. Above the liftoff speed, the experimental speed ratio generally increased, but the experimental speed difference remained essentially constant.

6. Torque of the series-hybrid bearing was greater after liftoff of the fluid-film bearing and was in the same range as that of the ball bearing alone when jet lubricated.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, September 3, 1970,
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