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TECHNICAL PAPER proposed for presentation at Annual Meeting of the American Society of Lubrication Engineers Cleveland, Ohio, May 6-9, 1968

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PRELIMINARY EVALUATION OF GREASES TO 600 F AND SOLID LUBRICANTS TO 1500 F IN BALL BEARINGS

by Harold E. Sliney and Robert L. Johnson

ABSTRACT

A special apparatus designed for the evaluation of high temperature lubricants in 20 mm bore ball bearings is described. The results of bearing runs at temperatures up to 1500 F in air are reported.

Of the greases evaluated, a fluorocarbon grease, provided bearing lubrication for longer times at 450 and 600 F than either polyphenyl ether or silicone greases thickened with dyes and/or MoS_2. Cobalt alloy ball bearings, lubricated either with barium fluoride-calcium fluoride coatings bonded to the cages or with porous metal cages impregnated with these fluorides, ran successfully at 1200 and 1500 F under a thrust load of 30 pounds and at a shaft speed of 5000 rpm.

INTRODUCTION

The attainment of adequate wear life and acceptable reliability are two of the most difficult problems in the area of solid lubrication. When a solid lubricant is used as a bonded coating, its wear life is finite; when the coating wears out, the lubricant cannot be readily replaced. Self-lubricating composites often extend wear life capabilities but are susceptible to distortion and cracking. Therefore, after basic studies have demonstrated that a solid has promising friction and wear characteristics, it is then of interest to determine the performance of the candidate lubricant in actual bearings.

In this paper, a new type of bearing test apparatus, which was designed for evaluating solid lubricants or greases in 204 size (20 mm bore) ball bearings, is described. The preliminary test results for greases at bearing temperatures from 325 to 600 F and for solid lubricants at bearing temperatures from 1200 to 1500 F are presented.

The solid lubricants were applied as bonded coatings on the bearing cage, or in some cases, the cages were fabricated from self-lubricating composites consisting of porous, sintered nickel-chromium alloy (Inconel) impregnated with a CaF_2-BaF_2 eutectic filler. All tests were in air. A thrust load of 30 pounds and shaft speeds of 2000 or 5000 rpm were used. The calculated maximum Hertz stress at 1200 F for a 30 pound thrust load was 93,000 psi.

BACKGROUND

Ball bearings lubricated with solid lubricants offer several important advantages; (1) they can be operated at temperature extremes beyond the capabilities of oil or grease-lubricated bearings (Refs. 1 and 2); (2) because dry-lubricated bearings do not require cooling, recirculating oil systems with their associated pumps and heat exchangers are not needed, and (3) rotating shafts can be shortened because dry-lubricated bearings can be located closer to heat sources.

These advantages are of importance in general machine design, but are of particular importance in aerospace equipment which must be of light weight and minimum complexity.

Several techniques for the use of solid lubricants in ball bearings have been studied. One method that has shown promise is powder lubrication; whereby the solid film is continuously deposited on the bearing surfaces from a suspension of solid particles in a carrier gas. Some early studies, which proved the feasibility of lubricating rolling contact bearings in this manner employed MoS_2 or graphite with air as the carrier gas (3, 4). MoS_2 provided lubrication to about 700 F before decomposition of the lubricant occurred; graphite provided lubrication to 1000 F. At 1000 F, graphite oxidized to harmless gaseous products but was replenished by the continuous supply of lubricant. Other experiments showed that MoS_2 could be effectively used to at least 1000 F if an inert carrier gas (nitrogen) were employed (5).

It has been shown that the variation in the friction coefficient of graphite over a large temperature range can be reduced by the addition of certain metal oxides; cadmium oxide was a particularly beneficial additive (6). Cadmium oxide-graphite powder in an air carrier has lubricated cobalt-alloy ball bearings at temperatures from ambient to 1000 F (7). Very careful metering is required to provide adequate lubrication while avoiding lubricant starvation or excessive lubricant build-up in the bearings.

Ball bearings have been effectively lubricated in air at temperatures up to 750 F with MoS_2 bonded to the bearing cage (8). In the same work, it was also shown that the wear life can be substantially improved by providing a number of depressions or lubricant reservoirs in the ball pockets and the locating lands of the bearing cages.

Lead oxide-lead silicate solid lubricant coatings have lubricated ball bearings at 1000 and 1200 F (9). The melting point of the lead oxide-lead silicate coatings limits their maximum useful temperature to about 1250 F.

Another class of solid lubricant coatings for severe environments is the series of fused fluorides reported in (10). The barium fluoride (BaF_2) calcium fluoride (CaF_2) coatings, which are evaluated as lubricants for bearing cages in this study, have much higher melting points than PbO coatings and have been shown to lubricate at temperatures up to 1500 F. These coatings also have somewhat better friction and wear life characteristics than PbO coatings at temperatures below 1000 F but have considerably higher friction coefficients than MoS_2 below about 700 F.

Solid lubricants may also be incorporated into a composite bearing material. Some examples are: molybdenum disulfide (MoS_2) with silver (11), PTFE, molybdenum diselenide ($MoSe_2$) with gallium and indium (2), and MoS_2 with a metal matrix of iron or platinum (13).

The composites described in (12) were evaluated as cage materials for ball bearings. The bearing performed satisfactorily in vacuum over a temperature range of -180 to 300 F (14). WSe_2 and gallium composite cages (2) provided lubrication in air to 950 F. The composites of porous nickel chromium alloy (Inconel) impregnated with BaF_2-CaF_2 eutectic, which is evaluated as a self-lubricating bearing cage material in this study, has shown promise in fundamental sliding-friction and wear studies for uses at temperatures to 1500 F (15).

TEST BEARINGS

The solid-lubricated bearings used are 204 size, angular contact ball bearings. The ball set consists of twelve 9/32-in. diameter balls. Both inner and outer raceway curvatures are 54 percent and the nominal contact angle is 20°. The internal clearance (radial play) is 0.002-in. Ball and race material is a cobalt-chromium alloy (Stellite 6B) hardened to Rockwell C-42. The cage is inner race-located. Some of the cages are a nickel-chromium alloy (Rene 41) coated with 0.001-0.002 in. thickness of fluoride solid lubricant (60 wt % CaF_2-40 wt % BaF_2); others are self-lubricating composites of porous, sintered nickel-chromium alloy (Inconel) impregnated with an eutectic fluoride (38 wt % CaF_2 -

62 wt % BaF_2). Cage-ball and cage-race diametral clearances are variable from 0.010 to 0.030 in. In each bearing, the cage clearances are held to a tolerance of 0.0005 in.

The grease lubricated bearings are hardened 440-C steel, deep groove ball bearings. The cages are stamped steel and the bearings are provided with two press fit steel grease shields.

BEARING TEST APPARATUS

A cutaway view of the bearing test head is given in Fig. 1. The test head can be completely assembled as a unit prior to mounting on the drive system; it consists of a double concentric bellows assembly provided with a bearing housing at the front and back for the test bearing and the slave bearing, respectively. The annular space between the two bellows is gas-tight and pneumatic pressure is used to apply a uniform thrust load to the outer races of the two bearings. As indicated on the drawing, the inner races are located on the shaft by means of a tubular spacer between the two bearings. The bellows were calibrated on the fixture shown schematically in Fig. 2. The calibration curve is given in Fig. 3.

Pressurized gas is supplied to the bellows through a long section of small diameter (1/16-in. O.D.) tubing which does not significantly interfere with torque measurements.

The assembled test head is mounted on a motor-driven support bearing spindle. Alignment and shaft balance are achieved during fabrication by performing final machining and balancing of the shaft while mounted on the same spindle. During a bearing test, the combined torques of the test bearing and the slave bearing apply a rotational moment to the outer races and hence to the bearing housing. Rotation is prevented by a swiveled linkage between a torque arm on the housing and a strain ring which continuously measures the combined torque. A variable-speed drive provides a range of shaft speeds from 500 to 5000 rpm.

Heating is provided by a high frequency (475,000 cps) induction coil. The test bearing housing and a hub which extends into the bore of the inner race are heated directly, then conduct heat into the bearing races. Bearing outer race temperatures are measured with chromel alumel thermocouples. Both outer and inner race temperatures are monitored with an infrared pyrometer capable of measuring temperatures from 250 to 4500 F.

The bearing test head was carefully designed to minimize heat transfer from the heated test bearing to the slave bearing by keeping the cross sectional area for conductive heat transfer to a minimum.

A cross-section through the bearing test head and its temperature profile are given in Fig. 4. In addition to thermocouples and the infrared pyrometer, temperatures were determined with temperature indicating marking sticks. These merely bracketed temperature within 100 F increments, but were useful at the cooler end of the test head where the surfaces were unoxidized and highly reflective, therefore of uncertain infrared emissivity. At a test bearing outer race temperature of 1200 F, the rear bearing outer race temperature was about 400 F. Fig. 5 gives the temperature of the slave bearing for test bearing temperatures up to 1300 F.

RESULTS AND DISCUSSION

Grease Lubricated Bearings

Several greases, which are useful to 600 F have been reported (16); therefore, greases were considered for lubrication of the slave bearing. In our studies, long duration runs were made with a number of high temperature greases to gain some idea of the life and torque characteristics of these greases in the 440-C steel bearings of the type used as slave bearings in the test section.

The greases evaluated were;

(a) a 5P4E polyphenyl ether fluid thickened with an organic dye,

(b) a polyphenyl ether (unknown molecular weight) - silicone blended fluid thickened with an organic dye and MoS_2,

(c) a perfluorinated alkyl ether fluid thickened with an organic dye, and

(d) a perfluorinated alkyl ether fluid thickened with a fluorocarbon telomer (intermediate molecular weight polymer).

Long Duration Tests of Grease-Packed Bearings

The failure criterion for the long duration tests was arbitrarily set at a bearing torque of 10-in-oz. This corresponds to about 5 times the typical combined torque of the two bearings during normal operation.

The results of these studies are summarized in Table 1. Both polyphenyl ether greases tended to harden during prolonged operation at bearing temperatures from 325 to 600 F.

The hardening of the grease was accompanied by an increase in bearing torque. The bearings did not fail from loss of lubricating material, but rather by stiffening of the grease. No bearing damage was noted, but the high torque and rough operation after less than 50 hours at bearing temperatures of 385, 450, and 600 F made these bearing-grease combinations unsatisfactory for purposes of this program.

The two fluorocarbon greases performed in a more satisfactory manner. Lives in excess of 200 hours were obtained at bearing temperatures up to 450 F with both the dye-thickened and the telomer-thickened grease. At higher temperatures, life was limited by evaporation of the lubricating fluid.

At 500 F the telomer-thickened fluorocarbon grease lubricated for 33 hours. Inspection of the bearing, showed that most of the fluid had evaporated or bled out of the bearing. The residue was a thin, white film, of what appeared to be the telomer thickener on all of the internal surfaces of the bearing. No bearing damage was apparent except a small amount of wear on several of the cage ball pockets where the protective film had worn through. At 600 F, the dye-thickened grease lubricated for 20 hours. Inspection of the bearing indicated that the fluid had evaporated and lubrication was apparently provided by the dye thickener which remained in the bearing.

The telomer-thickened fluorocarbon grease was chosen as the slave bearing lubricant. It was chosen in favor of the polyphenyl ether greases because of longer bearing life and more stable torque characteristics. It was chosen in favor of the dye-thickened fluorocarbon grease only because it is clean and convenient to handle.

Grease Torque Characteristics

In these experiments, both the test and the slave bearing were packed with 3 cc of telomer-thickened fluorocarbon grease. Fig. 6 gives the test bearing torque at 2000 and 5000 rpm, and at outer race remperatures up to 500 F. With no external heat addition to the bearings, the bearing temperature and torque stabilized at 160 F and 2 in-oz. at 2000 rpm and at 210 F and 3-1/2 in-oz. at 5000 rpm. Under these conditions the temperatures of the test and the support bearings were equal within 20 F. Therefore, the torque per bearing was assumed to be one-half the combined torque of the two bearings.

The bearings were then stopped and allowed to cool to room temperature. After cooling, they were restarted and the test bearing was rapidly induction heated to a stable, elevated temperature. The slave bearing, which was heated only by frictional heat and by conduction from the test bearing, reached a stable temperature at a much lower rate than the test bearing. The combined torque was measured while the slave bearing temperature was still within the range measured with no external heat addition to either bearing. The torque of the heated test bearing was then determined by difference. Repeating this process at several test bearing temperatures, the torque of the grease lubricated bearing was determined over the temperature range shown in Fig. 6. Fig. 6 was used in subsequent tests as a calibration curve to correct test bearing torque for the torque contribution of the slave bearing.

Solid-Lubricated Bearings

Coatings. - The ball bearings in this series of tests were lubricated with a 0.002-in. thick coating of 60 wt % CaF_2+40 wt % BaF_2 bonded to the bearing cages. The tests were run at 1200 and 1500 F. The results are summarized in Table 2. Test bearing torque was in the range of 1 to 4 in-oz. and therefore, comparable to the torque of the grease-lubricated bearings. The bearings were considered failed when the torque increased to 10 in-oz. which is the same failure criterion that was chosen for the grease lubricated bearings.

The potential for long life is indicated by the results of Exp. 1 (Table 2) in which a bearing heated to 1200 F ran for 220 hrs. at 2000 rpm and an additional 710 hrs. at 5000 rpm and did not fail. The earliest failure at 1200 F for a bearing equipped with an inner race located cage, occurred after 56 hrs. at 2000 rpm.

Bearing life at 1500 F was shorter than at 1200 F (Exp. 5), but the bearing performed well for 12 hrs. at 2000 rpm and an additional 24 hrs. at 5000 rpm. The balls and races were in good condition with a highly polished appearance on the rolling contact surfaces. However, the solid lubricant coating on the cage was rough and porous. It was earlier observed (10) that the coatings were relatively unaffected by long duration exposure to 1200 F air, but at 1500 F, the coatings slowly deteriorated due to oxidation of the base metal at the coating bond line. The coating is not adversely affected by exposure to 1500 F in non-oxidizing environments such as hydrogen or argon.

Examination of the bearings after the tests indicated that all failures were associated with the generation of wear debris from the balls and the raceways. The coatings in the ball pockets were apparently worn away but the coatings on the race-located surfaces of the cages were highly polished and apparently in excellent conditions.

These experiments were preliminary in nature, and insufficient in number to give a reliable indication of the bearing life that can be reasonably expected. However, the results do indicate that it is definitely possible to lubricate cobalt-chromium alloy ball bearings with fluoride solid lubricants at 1200 and 1500 F for useful periods of time. These experiments defined the main problem area - to improve the coating in the ball pockets of the bearing cage.

Self-Lubricating Composite Cages. - Ball bearings equipped with self-lubricating composite cages were also tested. The results are summarized in Table 3. In nine experiments at bearing temperatures from 1200 to 1500 F, only one bearing failed from lack of lubrication. This was during Exp. 6 (Table 3) where the composite cage was used in the as-machined condition. During machining, the metal phase of the composite material was smeared over the lubricant phase. Because of the lack of lubricant at the surface, the bearing immediately ran at a torque in excess of 10 in-oz.

Several surface treatments were effective in re-exposing the lubricant after machining; these included acid etches, wet honing with waterproof sandpaper, and heat treatment to promote fluoride exudation to the etched or honed surface. The heat treatment was performed in a nitrogen or argon atmosphere. In general, better results were obtained in 1200 F bearing tests with cages that had been heat treated in argon. The lowest wear of all bearing elements was obtained when the etched cage was coated with a thin overlay of CaF_2-BaF_2 (Exp. 5).

In Exp. 1 (Table 3), the bearing ran at 1200 F and 5000 rpm for 149 hrs with no failure and with very low wear. Another bearing (Exp. 9) ran at 1500 F and 2000 rpm for 20 hrs and at 5000 rpm for an additional 50 hrs. The other bearings ran for shorter periods of time with the shortest duration at 23 hrs. In all cases except (Exp. 6) failure was caused by closing of cage clearances due to dimensional instability of the composite material. This instability occurred as a result of a gradual uniform swell of the composite during exposure to air at high temperatures, and does not appear to be associated with the mechanical stresses on the cage during operation of the bearing. Because no lubrication failure occurred when composite cages were used, very long endurance life should be attainable if the problem of dimensional instability can be solved.

Figs. 7 and 8 give the torque-time characteristics under various conditions for bearings equipped with self-lubricating cages. The torque of an unlubricated bearing with a machined nickel-chromium alloy (Rene 41) cage is shown for comparision.

Very high and erratic bearing torque was characteristic of the unlubricated bearing. However, examination of the bearing after this test showed that in spite of the unfavorable torque, the bearing was still in good condition. The bearing was run under essentially a pure thrust load and was carefully aligned. Under these conditions, the forces acting on the cage at the cage-ball and cage-race sliding contact areas are small (17); and the naturally-formed oxide films on Stellite 6B and Rene 41 are apparently adequate to prevent surface damage (wear). In air, at elevated temperatures, therefore, the primary purpose of the fluoride lubricant is to provide a low and relatively steady bearing torque.

CONCLUDING REMARKS

The new type of bearing test apparatus described in this paper was satisfactory for evaluating high-temperature greases or solid lubricants in thrust-loaded ball bearings.

Ball bearings with Stellite 6B balls and races were lubricated at 1200 and 1500 F with CaF_2-BaF_2 solid lubricants. These lubricants were effective either as bonded coatings on Rene 41 alloy bearing cages or as fillers in composite bearing cages.

Perfluorinated alkyl ether greases with telomer or organic dye thickeners were satisfactory lubricants up to 500 F for the 440-C tool steel slave bearings used in the test rig.

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TABLE 1. - RESULTS OF TESTS WITH BALL BEARINGS

LUBRICATED BY HIGH-TEMPERATURE GREASES

[Bearings, deep groove, eight ball, stamped cage, material, 400 C; lubricant, three cm^3 of grease (double press fit grease shield used); conditions, air atmosphere, 30 pound thrust load; failure criterion, increase of bearing torque to 10 in. -oz.]

Grease description		Temperature bearing races	Shaft speed	Test duration (b)	Typical torque (c)
Fluid	Thickener				
		^{o}F	rpm	hr	in. -oz
5P4E Polyphenyl ether	Organic dye	325	2000	220	2.0
		385	5000	46	1.5
		450	5000	45	1.0
Polyphenyl ether-silicone blend	Organic dye and M_0S_2	325	5000	148	1.2
		385	5000	23	1.0
		600	5000	3	1.8
Fluorocarbon oil	Organic dye	450	5000	384	1.0
		600	5000	20	3.0
Fluorocarbon oil	Fluorocarbon telomer	325	[a]2000	134	1.2
		400	2000	215	1.0
		325	[a]5000	734	1.4
		450	5000	209	1.0
		500	5000	33	.8

[a]No failure in times indicated.
[b]Lubricant failed in times indicated unless otherwise noted.
[c]Stable torque before onset of failure.

TABLE 2. - PERFORMANCE OF HIGH-TEMPERATURE BALL BEARINGS

EQUIPPED WITH FLUORIDE-COATED CAGES

[Bearing description, 204 size (20 mm bore), angular contact (20^{o} contact angle), 12 balls; bearing material, balls and races, Stellite 6B (age-hardened cobalt-chromium alloy), cages, Rene 41 (age-hardened nickel-chromium alloy) coated with ~0.002-inch CaF_2 - BaF_2 solid lubricant; test conditions, 30 pounds thrust load, air atmosphere; failure criterion, increase of bearing torque to 10 in.-oz.]

Experiment no.	Cage clearance		Test conditions		Test duration (a)	Typical torque	Weight change (rate)		
	Inner race	Ball pocket	Temperature	Speed			Inner race	Outer race	Balls
	in.	in.	^{o}F	rpm	hr	in.-oz	mg/hr	mg/hr	mg/hr
1	0.010	0.016	1200	2000	220	2	------	-----	------
	-----	-----	----	5000	[b]710	3	-0.01	-0.02	-0.06
2	0.010	0.016	1200	2000	56	1	-18.0	-7.6	-6.2
3	0.025	0.020	1200	2000	140	1	-0.06	-0.04	[c]-0.04
4	[d]0.013	0.016	1200	588	5	$1\frac{1}{2}$	------	-----	------
	------	-----	----	1500	20	2	------	-----	------
	------	-----	----	2000	2	4	------	-----	------
	------	-----	----	5000	6	4	------	-0.26	-0.10
5	0.010	0.016	1500	2000	12	3	0.9	-0.16	0.16
	-----	-----	----	5000	24	4	-2.0	-1.0	-3.2

[a]Failed in time indicated unless otherwise noted.
[b]No failure in times indicated.
[c]Hastelloy-C (nickel-chromium alloy) balls.
[d]Outer race-cage clearance.

TABLE 3. - PERFORMANCE OF HIGH-TEMPERATURE BALL BEARINGS EQUIPPED WITH SELF-LUBRICATING

COMPOSITE CAGES

[Bearing description, 204 size (20 mm bore), angular contact (20° contact angle), 12 balls; bearing materials, balls and races, Stellite 6B (age-hardened cobalt-chromium alloy); cages, sintered porous Inconel (nickel-chromium alloy) of 35 vol. % porosity vacuum-impregnated with CaF_2 - BaF_2 eutectic; test conditions, 30 pounds thrust load, air atmosphere; failure criterion, increase of bearing torque to 10 in.-oz.]

Experiment no.	Cage pretreatment				Cage clearance		Test conditions		Test duration (a)	Typical torque	Weight change (rate)		
	Surface treatment	Heat treatment			Inner race	Ball pockets	Temperature	Speed			Inner race	Outer race	Balls
		Atmosphere	°F	hr	in.	in.	°F	rpm	hr	in.-oz	mg/hr	mg/hr	mg/hr
1	Acid etch	Argon	1500	2	0.020	0.016	1200	5000	b149	2.6	-0.04	-0.03	-0.02
2	Acid etch	Argon	1500	2	0.015	0.018	1200	2000	41	1.0	-----	0.02	0
	----------	-------	----	---	-----	-----	----	5000	15	5.0	-0.03	-.05	0
3	Acid etch	Nitrogen	1600	2	0.015	0.016	1200	2000	60	4.5	-0.03	-0.03	-0.08
4	Acid etch	Nitrogen	1600	2	0.010	0.016	1200	5000	23	4	-0.17	-0.03	-----
5	Acid etch CaF_2 - BaF_2 overlay	Nitrogen	1700	0.3	0.027	0.015	1200	2000	27	1.5	0	0	0
6	None	None	----	---	0.020	0.016	1200	2000	0	10	-----	-----	-----
7	Acid etch and hone	None	----	---	0.030	0.020	1300	2000	34	1	-2.1	-0.9	-0.01
8	Acid etch	Nitrogen	1600	2	0.020	0.016	1400	2000	35	1	-0.01	0.06	0
9	Acid etch	Argon	1500	2	0.020	0.016	1500	2000	b20	0.6	0.13	0.16	0.09
	----------	-------	----	---	-----	-----	----	5000	b50	1.4	-.17	-.04	-.32

[a]Failed in times indicated unless otherwise noted.
[b]No failure in times indicated.

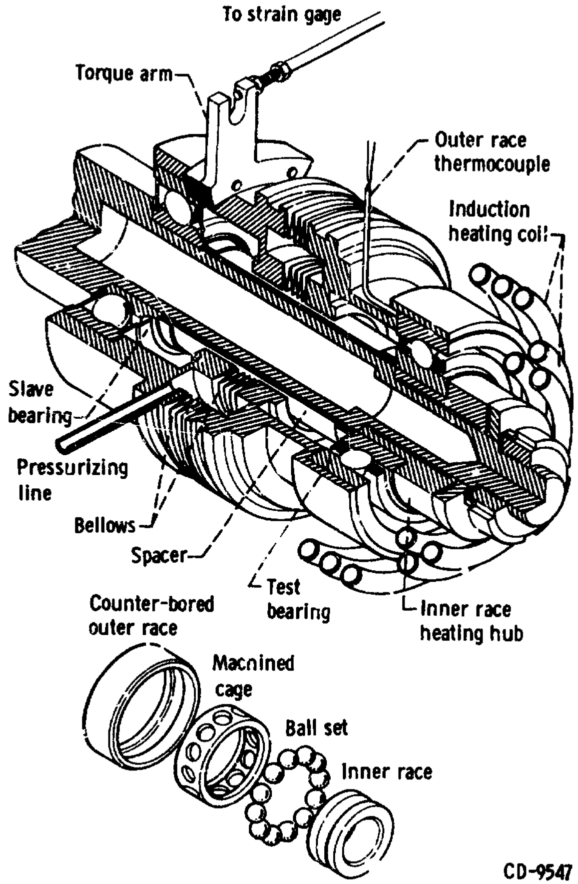

Fig. 1. - Bearing test head and exploded view of test bearing.

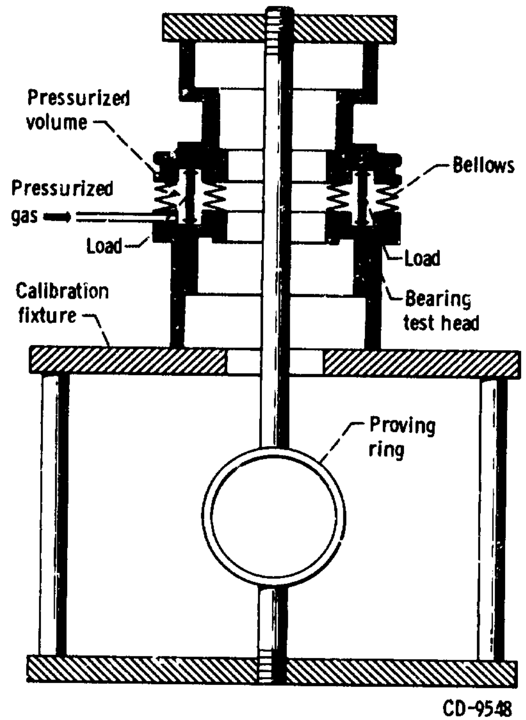

Fig. 2. - Bellows calibration fixture.

E-4159

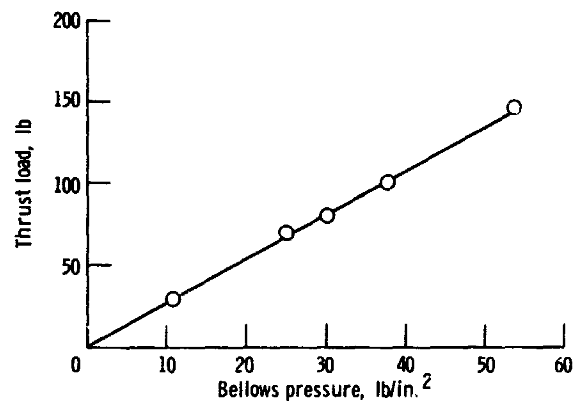

Fig. 3. - Calibration curve for bellows thrust-loading device.

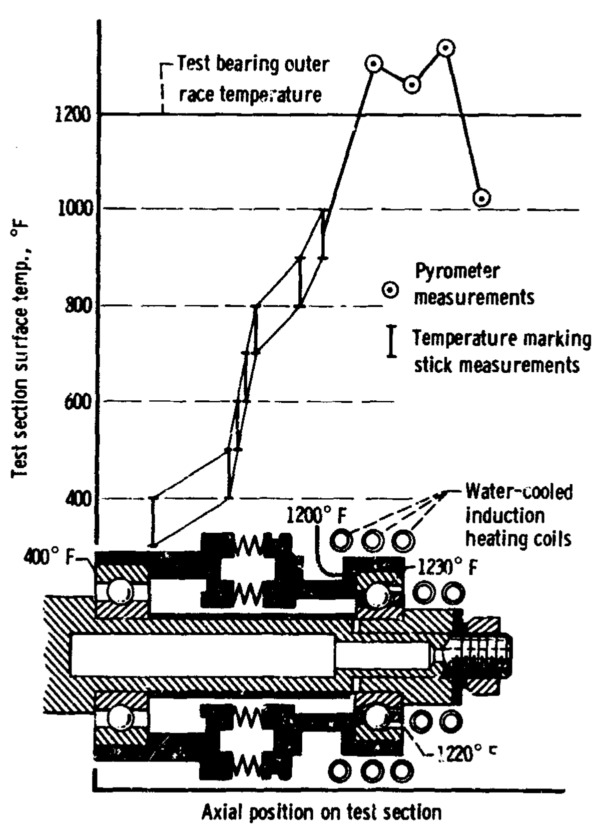

Fig. 4. - Temperature distribution of bearing test section.

E-4159

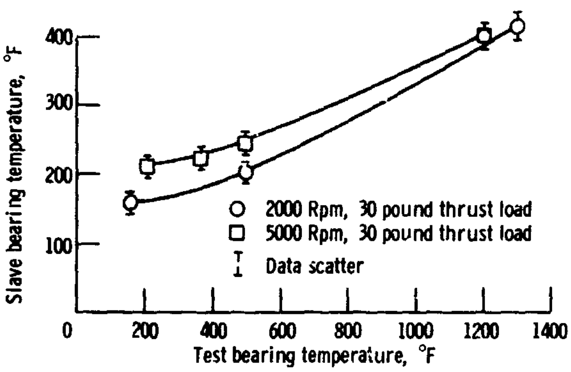

Fig. 5. - Relationship of slave bearing temperature to test bearing temperature.

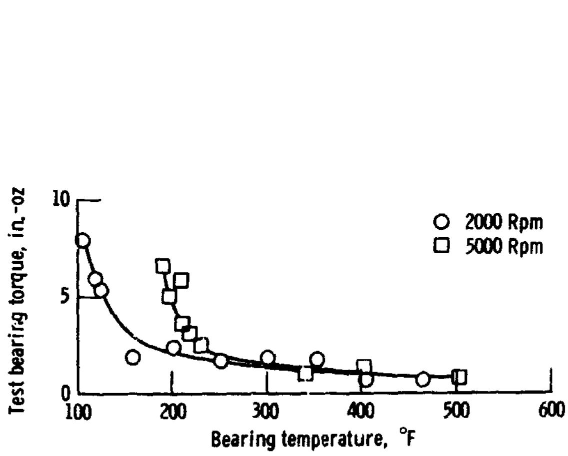

Fig. 6. - Torque of 20 mm ball bearings lubricated with a fluorocarbon grease. Load, 30 lb thrust.

E-4159

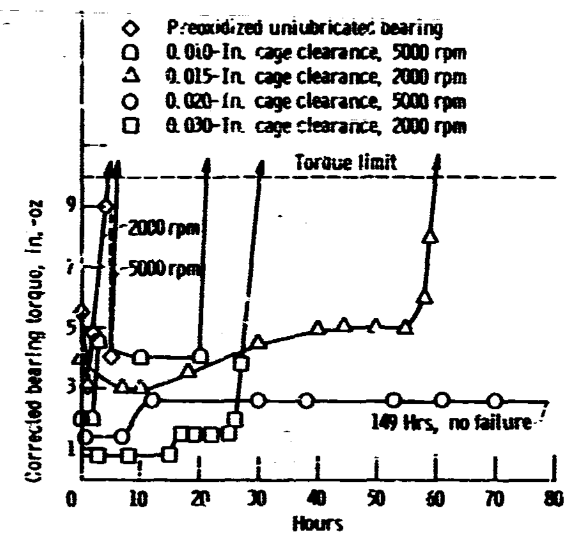

Fig. 7. - Bearing performance at 1200° F. Cobalt alloy bearings with self-lubricated fluoride-Inconel composite cages 30 pound thrust load, inner race-located, 0.016 inch ball pocket clearance.

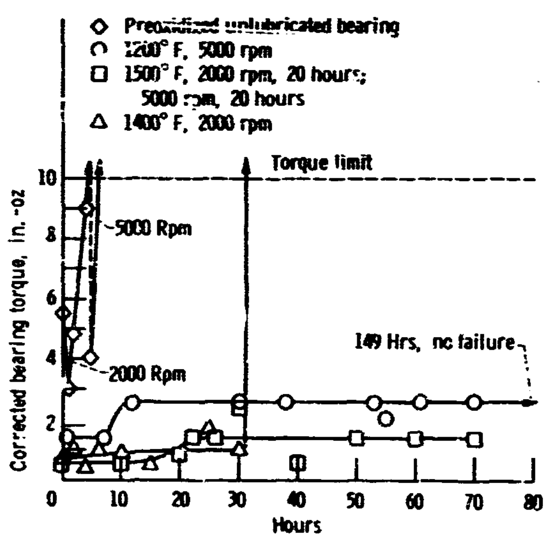

Fig. 8. - Influence of bearing temperature and speed on the performance of cobalt alloy ball bearings with self-lubricated fluoride-Inconel composite cages. 30-Lb thrust load; inner race-located cage, 0.020-in.; cage-race clearance; 0.016-in. ball pocket clearance.