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Development of a Ceramic Rolling Contact Bearing for High Temperature Use[1]

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This paper describes the development of a ceramic ball bearing for use at temperatures of about 1000 to 1500 F. Several selected combinations of ceramic and cermet materials were screened in simple sliding and rolling experiments, and the most promising materials were selected for fabrication into full-scale bearings. Bearing experiments at temperatures of 1000 and 1500 F demonstrated the feasibility of bearing operation at these temperatures with little or no wear under moderate load and speed.

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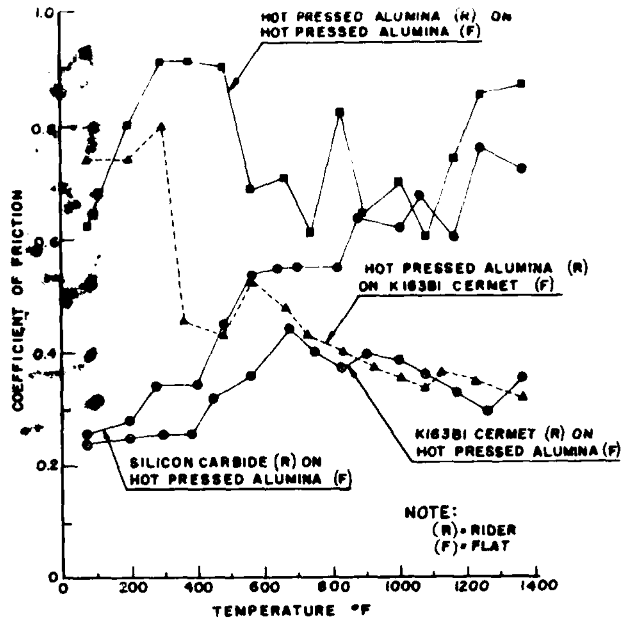

Fig. 1 Coefficient of friction of ceramic and cermet materials

In recent years the demand for increased performance of machine components has forced the use of substitute materials for rolling-contact bearings to enable satisfactory operation at high temperatures. For example, some aircraft-turbine rotors have ball and roller bearings constructed of tool steel with high hot hardness for improved performance at high gas-turbine temperatures. Even these special bearings need extensive cooling, however, and aircraft-component designers would welcome the development of reliable bearing materials and lubricants for even higher temperatures. In addition, the efficient use of atomic energy will demand the operation of high-performance bearings in the special corrosive and high-temperature environments of nuclear power reactors. Current military applications already require the operation of machines at temperatures up to 2000-2500 F. The importance of corrosion resistance and hot hardness in bearing materials at these temperatures has focused attention on ceramics for such applications.

The object of this study was to develop a ceramic rolling-contact bearing for operation at temperatures higher than is feasible with metal bearings. The procedure was to conduct preliminary wear and friction screening tests on several combinations of ceramic materials possessing desirable physical properties, followed by a bearing design study, and finally full-scale, ball-bearing tests on the two most promising combinations of materials. Some of the full-scale tests were made with limited application of dry lubricants, and some with continuous application of the same lubricants.

EXPERIMENTAL PROCEDURE AND RESULTS

Selection and Preparation of Materials

Ball and roller bearings require materials with high hardness and strength at operating temperatures. Stresses at the contacts between rolling elements and races exceed 200,000 psi for many practical bearing loads. An operating temperature of about 1000 F is the practical limit of present alloys with the highest hot hardness (high-speed tool steels, hardenable stainless steels, nickel and cobalt-base alloys).

Selection of ceramic and cermet materials for this study was based upon the following considerations:

1 High strength and good oxidation resistance at use temperatures.

2 Hard, essentially non-porous bodies capable of taking a good finish.

3 Resistance to spalling by thermal or mechanical shock.

On the foregoing bases, the following materials were selected for experimental study: (a) Hot-pressed alumina, (b) hot-pressed silicon carbide, and (c) K163B1 titanium carbide cermet.

The hot-pressed alumina was 99 per cent Al_2O_3 with chief impurities of magnesia, silica, and iron oxide. The hot-pressed silicon carbide contained approximately 97 per cent silicon carbide and a total of about 3 per cent of silicon, aluminum, iron, and oxygen. The chemical analysis of the titanium-carbide cermet, K163B1, as supplied by the manufacturer, was as follows:

TABLE 1
Some Physical Properties of Bearing Materials
Selected for Evaluation

Property at Room Temperature	Hot-Pressed Alumina	Hot-Pressed Silicon Carbide	Titanium Carbide Cermet (K163B1)
Modulus of Rupture $\times 10^3$ psi	70-100	70	200
Modulus of Elasticity $\times 10^6$ psi	60-80	60-80	55
Compressive Strength $\times 10^3$ psi	400	400	300
Tensile Strength $\times 10^3$ psi	35-40	35-40	100
Hardness, Knoof K = 100	2000	2400	1800 (Av.)
Bulk Density % of Theoretical	98.5-100	98-100	99-100

TABLE 2
Wear Rate in Sliding Contact at 1000 F

Test No.	Material Combination	Wear Rate g./kg./cm.*
1	Hot-Pressed Alumina Rider	150×10^{-9}
	Hot-Pressed Alumina Flat	650×10^{-9}
2	Hot-Pressed Silicon Carbide Rider	250×10^{-9}
	Hot-Pressed Silicon Carbide Flat	32×10^{-9}
3	K163B1 Cermet Rider	3.3×10^{-9}
	K163B1 Cermet Flat	3.7×10^{-9}

*Grams of material lost per kilogram of load per centimeter of distance traveled.

	Per cent
Titanium	44.1
Columbium . . .	4.5
Tantalum	0.3
Carbon	11.1
Nickel	33.3
Molybdenum . . .	6.7

Typical physical properties of the three materials are shown in Table 1.

No particular difficulty was encountered in grinding the balls, and specimens were obtained of about 1 microin (rms) surface finish with close tolerances as to roundness and uniformity in size. However, considerable difficulty was experienced in grinding the experimental bearing races. The difficulty stemmed mainly from inability to obtain diamond wheels of sufficiently close dimensional tolerances for plunge grinding. The problem was solved by the use of oscillating grinding which produced quite satisfactory races. These will be discussed in more detail under "Bearing Design."

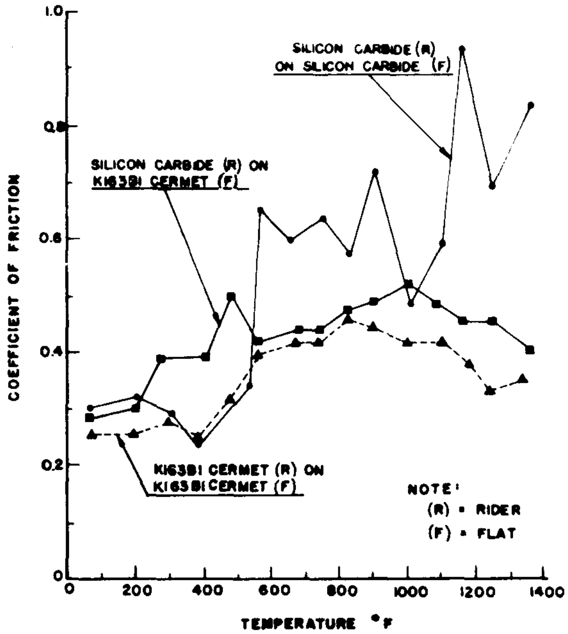

Fig. 2 Coefficient of friction of ceramic and cermet materials

Preliminary Tests

1 Sliding Friction and Wear Tests. These tests consisted in measuring friction and wear rate occurring when a 1/4-in-diam cylindrical rider was pressed against a rotating plate (1).[2] The tests were carried out in air without lubrication at temperatures up to 1300 F.

In the friction tests, the load on the rider was 500 grams, and the linear speed was 3 cm per sec. Each of the three materials (hot-pressed alumina, hot-pressed silicon carbide, and K163B1 cermet) was tested against itself and in combination with the other two materials. In the case of the one combination, hot-pressed alumina against K163B1 cermet, two tests were made; in one test the alumina was the rider and in the second test the K163B1 was the rider.

From Figs.1 and 2 it can be seen that of the materials run against themselves, the K163B1 cermet gave the lowest friction over the entire temperature range. Of the combinations of two materials, friction values were lowest for those in which K163B1 was one of the materials; however, it is difficult to choose between K163B1-alumina and the K163B1-silicon carbide combinations.

The conditions under which friction measurements were made were not severe enough to indicate relative wear resistance; therefore, wear was evaluated under more drastic conditions. The hot-

[2] Numbers in parentheses designate References at the end of the paper.

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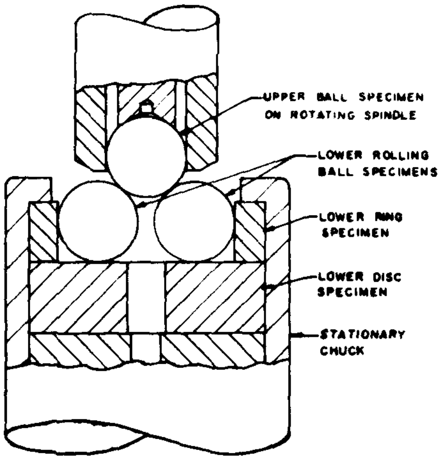


Fig. 3 Sketch of the rolling four-ball tester

pressed alumina and hot-pressed silicon carbide were tested under a load of 1600 grams, a speed of 27 cm per sec, a temperature of 1000 F, and a running time of 30 min. The K163B1 cermet which had shown slight evidence of superiority to wear under conditions of the friction tests, was evaluated under still more severe conditions by increasing the load to 2400 grams and the time to 60 min, but keeping the temperature at 1000 F, and the speed at 27 cm per sec, as with the other materials.

The results, shown in Table 2, indicate wear in grams of material lost per kilogram of load per centimeter of distance traveled when each material was tested against itself. Under conditions of the tests, the sliding wear resistance of K163B1 cermet is definitely superior to that of the other materials. The relative rating of hot-pressed alumina and hot-pressed silicon carbide, however, is not clear, but based on the combined weight loss of the rider and flat, the silicon carbide is somewhat superior.

Four-Ball Wear Tests. This test consists of four balls in a pyramidal arrangement, the lower three contained in a cup, the upper ball attached to a chuck, rotated and pressed into the lower three. In some tests the lower balls were held stationary; in other tests they were allowed to rotate (see Fig.3 and reference 2). In the latter case the cup was formed by a hollow cylinder and flat disk, both made of K163B1 cermet.

Table 3 shows data for tests on K163B1 titanium-carbide cermet, hot-pressed alumina, and hot-pressed silicon-carbide balls where the lower three balls were held stationary and the top rotating ball was the same material as the three lower stationary balls. The test was run dry in air at a sliding speed of 120 fpm.

TABLE 3
Four-Ball Wear Evaluations

Each material sliding dry against itself in air at 120 ft./min.

Test No.	Material	Ambient Temp., °F.	Total Load, kg.	Time min.	Average Wear Rate on Each Lower Ball,(1) in^3/min.	Average Coefficient of Friction
1	K163B1	Room	1-6(2)	4	0.6×10^{-6}	0.5
			52(3)	10	3.3×10^{-6}	0.3
1A		500	1-6	4	0.3×10^{-6}	0.4
			37(3)	10	3.0×10^{-6}	0.3
1B		1000	1-6	4	0.1×10^{-6}	0.1
			21(3)	10	2.2×10^{-6}	0.2
2	Al_2O_3	Room	1-6	4	33×10^{-6}	0.3
2A		500	1-6	4	100×10^{-6}	0.6
2B		1000	1-6	4	110×10^{-6}	0.4
3	SiC	Room	1-4	3	150×10^{-6}	0.4
3A		500	1-2	2	150×10^{-6}	-(4)
3B		1000	1-4	3	220×10^{-6}	-(4)

(1) Wear rates were calculated from measured wear scar diameters, assuming that the wear volume was a spherical segment of the 1/2-inch diameter ball specimens.

(2) A total of 1 kg. before wear produces a calculated mean Hertz load of 97,000 psi with K163B1 balls; the total load was 1 kg. the first minute, 2 kg. the second, 4 kg. the third, and 6 kg. the fourth.

(3) Total load was calculated to produce a unit load of 50,000 psi, based on the average wear scar area at the beginning of the 10-minute run.

(4) The vibration was so severe, reaching a magnitude equivalent to a coefficient of friction equal to 8, that no meaningful average coefficient of friction could be determined.

The results show that the K163B1 cermet is far superior to the other compositions when each is operated against itself. The cermet showed not only much lower wear rate but also a decrease in friction and wear rate with increase in temperature. Furthermore, friction coefficient generally decreased with increasing load. Since the hot-pressed alumina and hot-pressed silicon carbide showed high wear rates after the 4-min run-in period, they were not tested in the 10-min run. In all of these experiments wear on the top rotating ball was light.

In general, these results confirm those obtained in the sliding friction and wear tests when each of the foregoing materials was run against itself. However, good performance was also observed in the sliding-friction tests when hot-pressed alumina or hot-pressed silicon carbide was run against the K163B1 cermet. It was, therefore, decided to make four-ball tests using a K163B1 cermet ball in the chuck and hot-pressed alumina and hot-pressed silicon carbide as the three lower balls. In this series, the four-ball machine was modified slightly to test the materials under rolling conditions. Ring and disk specimens of K163B1 were installed in the four-ball machine to form the cup in which the three lower balls were allowed to roll instead of slide against the upper rotating ball to simulate more closely the contact conditions between the balls and races in a bearing. The wear surfaces of the ring-and-disk specimens were ground, lapped, and polished to a finish comparable to that of the ball specimens. The results, shown in Table 4, indicate possibly some superiority of the alumina-K163B1 combination over the silicon carbide-K163B1.

TABLE 4
Rolling Four-Ball Wear Evaluations

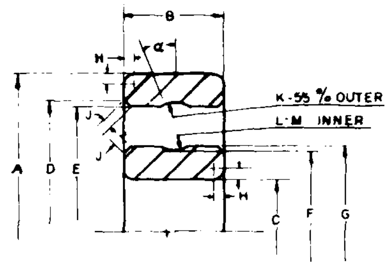
Upper ball, K163B1, rotating at 10,000 rpm. Lower balls rolling on a K163B1 flat disc and cylindrical ring. Load calculated to provide a maximum contact-stress on upper ball of 275,000 psi; on flat disc, 162,000 psi.

Lower Ball Material	Lubricant[1]	Ambient Temperature, °F.	Total Load, kg.	Time, min.	Average Width of Wear Track, mils. On Upper Ball	On Disc
K163B1	None	1000	2.2	10	-	29
		1000	2.2	10	-	33
		1300	2.2	9(2)	42	37
K163B1	PbO	1000	2.2	10	-	-(3)
		1000	2.2	10	-	14
		1300	2.2	20	13	20
Al_2O_3	None	1000	1.7	10	-	23
		1000	1.7	10	-	25
		1300	1.7	20	21	26
Al_2O_3	PbO	1000	1.7	10	-	-(3)
		1000	1.7	10	-	12
		1300-1500	1.7	20	15	17
SiC	None	1000	1.7	10	-	29
		1000	1.7	10	-	31
		1300	1.7	20	61	36

(1) Powdered lubricant was supplied to the contacting surfaces at the copius rate of 0.13 grams per minute. The powder was suspended in nitrogen gas and blown into the test chamber in four, 2-second puffs per minute.

(2) This experiment ended prematurely owing to severe wear and vibration.

(3) No visible wear track other than a band relatively free of lubricant.



SPECIAL 6204 SIZE BEARING(17mm x 52mm x 16mm) WITH 7-5/16 Dia.BALLS

DIM.	DESIGN #1	DESIGN #2
A (mm)	52	52
B (mm)	16	16
C (mm)	17	17
D ±.0005"	1.6393	1 6393
E (mm)	38.940 / 38.810	38.940 / 38.810
F ±.0005"	1.0168	1.0168
G (mm)	28.190 / 28.170	28.190 / 28.170
H (mm)	1.0	1.0
J (mm)	0 8	0 8
K ±.001"	0.1719	0.1719
L ±.001"	0 1625	0 1697
M	52 %	54¼ %
α	23°	20°
Radial Looseness (in.)	0.0035	0 0035
Axial Looseness (in)	0.017	0.020

Fig. 4 Bearing design data

Bearing Design

In view of preliminary sliding friction and wear tests and the "four-ball" tests, the two combinations of materials chosen for full-scale bearing tests were (a) K163B1 titanium-carbide cermet rings and balls, and (b) K163B1 titanium-carbide cermet rings and hot-pressed alumina balls. The full-scale tests were conducted using a special 6204 size (17 mm by 52 mm by 16 mm) bearing with seven 5/16-in-diam balls.

Based upon the physical properties of the materials, two experimental bearings were designed as shown in Fig.4. These differed from conventional designs for metal bearings mainly in having looser conformity to make possible the use of solid lubricants, to facilitate elimination of wear debris, to minimize sliding friction and to accommodate differences in thermal expansion when two different materials were used in combination. In Design 1, the ball-race conformity of the inner ring is closer than that of the outer ring, calculated to make the maximum contact stresses the same on both races, while in Design 2 the conformity of the inner and out rings is nearly the same, calculated to equate the ball-spin torque on both races. Thus in Design 1, the inner race controls the friction spinning moment of the balls while in Design 2, neither race imparts a controlling moment.

The cages for the full-scale bearings tested at 1000 F were made of M-1 tool steel except in one instance where Monel S was tried. For the tests at 1500 F, cages of a molybdenum alloy were used. Some considerations in making the cages were (a) moderate clearance on guiding lands, (b) maximum opportunity for expulsion of wear products, and (c) moderately loose retainer pockets.

The components of a typical bearing tested at 1000 F are seen in Fig.5.

Full-Scale Bearing Tests at 1000 F

These experiments were conducted using the equipment illustrated in Fig.6. The test bearing, mounted on a stainless-steel shaft and located in a Kovar® housing, was thrust-loaded by means of a lever and dead-weight system. Electric heaters of 750 watts capacity, located on each side of the test-bearing housing, maintained the required temperature.

In experiments where dry lubricants were used, the lubricant was introduced into the bearing as a finely suspended powder by means of a gas carrier consisting of 5 liters of nitrogen per min.

The first experiments, which were strictly exploratory, were made at 1000 F without lubricants at 2450 rpm and 25 lb thrust load. It was found that after about 10 to 15 min of operation, under these conditions, vibration became noticeable and moderate wear was experienced.

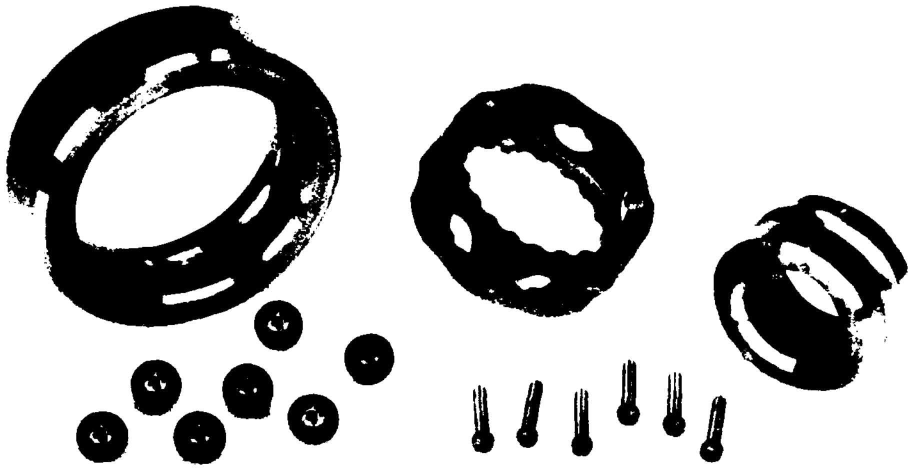


Fig. 5 Components of bearing tested at 1000 F

Several runs were then made with limited lubrication. These bearings were run-in at 1000 F, 2450 rpm and 25 or 50 lb thrust load for a period of 1-hr with metal-free phthalocyanine lubricant prior to testing without lubrication. At the end of the run-in period, the various bearing components were cleaned, inspected and measured for wear. The bearings were then reassembled and operated for an additional 5 min with metal-free phthalocyanine. The test was then continued without lubricant until either the bearing axial looseness increased approximately 0.010 in. or further running was not advisable as the result of excessive vibration. Bearing composition, cage material, bearing design, and test results are shown in Table 5.

TABLE 5
Tests on Full-Scale Bearings at 1000°F With Limited Lubrication

Races, K163B1 cermet; maximum deviation of groove surfaces from true radius, 7 microns. Cages, Monel S in Runs 2 and 2A; M-1 tool steel in other experiments.

Run No.	Oscillation of Races, % Inner Race	Outer Race	Ball Material	Operating Conditions Speed, rpm	Load, lb	Lubri-cant	Time, hr	Total Weight Loss of Brg.* arts, mg/hr.
2	53	55.5	Al_2O_3	2450	50	PCH_2**	1.0	0
2A				2450	25	None	1.9	29
2B				2450	25	None	1.0	95
3	53	56.5	Al_2O_3	2450	25	PCH_2	1.0	2
3A				2450	25	None	1.4	120
4	52.9	55.5	K163B1	2450	25	PCH_2	1.0	2
4A				2450	25	None	1.6	400

*Brg. = Bearing.
**PCH_2 = metal-free phthalocyanine.

In all instances, practically no wear was noted as the result of the 1-hr run-in with metal-free phthalocyanine. In the subsequent runs without lubricant, however, wear rate was relatively high, the bearing lasting only 1.4 to 2.9 hr. Significantly greater wear was obtained with the K163B1 titanium-carbide cermet balls than with the hot-pressed alumina balls when running without lubricant.

The Monel S cages used in Runs 2 and 2A galled and smeared on the inner ring land riding guide surfaces and displayed considerable ball pocket wear. In contrast, the M-1 hardened steel cages at no time showed excessive wear, smearing or galling, and were used with good success in the tests up to 1000 F.

A series of runs was next made at 1000 F with continuous lubrication. The procedure was as follows: After running-in the bearings for 1-hr at 2450 rpm and 50 lb thrust, each bearing was operated at 5350 rpm under 75 lb thrust until axial looseness or excessive vibration terminated the test. The bearing was then disassembled, inspected for wear, reassembled when possible with new balls (this was done only in those bearings having hot-pressed alumina balls since extra K163B1 balls were not available), the rings reversed to bring new ball groove test surfaces into use, run-in as previously mentioned, and then operated at 5350 rpm with a thrust load of 175 lb until it was necessary to terminate the test. Results are summarized in Table 6.

Referring to Table 6, it can be seen that no significant wear occurred during the 1-hr run-in periods at 50 lb thrust load and 2450 rpm with either metal-free phthalocyanine or molybdenum

TABLE 6
Tests on Lubricated Full-Scale Bearings at 1000°F

Races, K163B1 cermet; maximum deviation of groove surfaces from true radius, 8 microns. Cages, M-1 tool steel.

Run No.	Oscillation of Races, % Inner Race	Outer Race	Ball Material	Operating Conditions Speed, rpm	Load[1] lb.	Lubricant	Average Flow Rate g/min.	Time, hr	Total Weight Loss of Bearing Parts, mg/hr	Increase in Bearing Axial Looseness mils/hr
7	52.1	56.1	Al_2O_3	2450	50	PCH_2[2]	.047	1	1	0
7A				5350	75		.024	8	39	1.4
7B				2450	50		.027	1	1	0
7C				5350	175	Bearing destroyed - cage not properly fastened.				
8	54.6	55.5	K163B1	2450	50	PCH_2	.032	1	4	0
8A				5350	75		.014	12	30	.9
8B				2450	50		.033	1	5	1.0
8C				5350	175		.029	28	22	.8
9	51.8	56.4	Al_2O_3	2450	50	MoS_2	.165	1	0	0
9A				5350	75		.076	3.5	142	4.0
9B				2450	50		.117	1	3	0
9C				5350	175		.113	8.5	33	1.0
10	54.8	55.6	K163B1	2450	50	MoS_2	.315	1	3	1.0
10A				5350	75		.076	4.5	180	4.9
10B				2450	50		.148	1	5	7.0
10C				5350	175		.086	11.3	28	0.4

(1) Approximate maximum ball-race contact stresses under the given bearing thrust loads, before wear, were as follows: 50 lb. - 240,000 psi,
75 lb. - 270,000 psi,
175 lb. - 360,000 psi.

(2) PCH_2 = metal-free phthalocyanine.

disulfide lubricant. At 75 lb thrust load and 5350 rpm, the rate of wear is much higher with the molybdenum disulfide. However, at 175 lb thrust load, the wear rate with the molybdenum disulfide lubricant decreased to such an extent that their was no significant difference between the two lubricants. Bearing vibration seemed much less severe at 175 lb thrust load than at 75 lb, and this effect may have contributed to the lower recorded wear rates at high load. When lubricants were used at 1000 F, there did not appear to be any significant difference in wear rate between bearings fitted with hot-pressed alumina balls and K163B1 titanium-carbide-cermet balls. However, it should be recalled that without lubricant, the wear rate was faster with K163B1 balls.

The difference in the operation of the bearings with and without lubricant is striking. The longest run without lubricant, even after the bearing had been run-in with lubricant and the test continued without removing the residue, was 2.9 hr at 25 lb thrust load and 2450 rpm. With lubricant, one run lasted for 28 hr at 175 lb thrust load and 5350 rpm.

Full-Scale Bearing Tests at Temperatures Higher Than 1000 F

The real potential of ceramic and cermet ball bearings is at temperatures higher than 1000 F, since tool steel and Stellite alloy bearings have been operated successfully at 1000 F on an experimental basis (3).

Bearing Evaluation Equipment and Procedures for Temperatures Higher Than 1000 F. A machine that had been used to evaluate ceramics and cermets in high-speed sliding contact at temperatures to 1800 F, using a pad thrust-bearing specimen configuration (4), was adapted to evaluate ball bearings under a thrust load. The ball-bearing adapters, shown in Fig.7, were fabricated to replace the slider pad-bearing specimen configuration. Inconel "X" alloy, stress-equalizing heat-treated for dimensional stability and corrosion resistance up to 1800 F, was used for most of these parts. A self-aligning thrust-loading mechanism consisting of a ceramic ball and flat was also fabricated. Dead-weight thrust loads were applied to the test bearing and a speed of 8000 rpm was supplied from an electric motor and timing-belt drive. The test bearing was heated by passing high-velocity high-temperature air (10 per cent free oxygen at 1800 F) from a natural gas-burning combustion apparatus over and around the test-bearing housing. Conduction losses through the spindles were minimized by thin-walled sections, radiation shields, and stagnant air spaces built into the spindle heads. Bearing

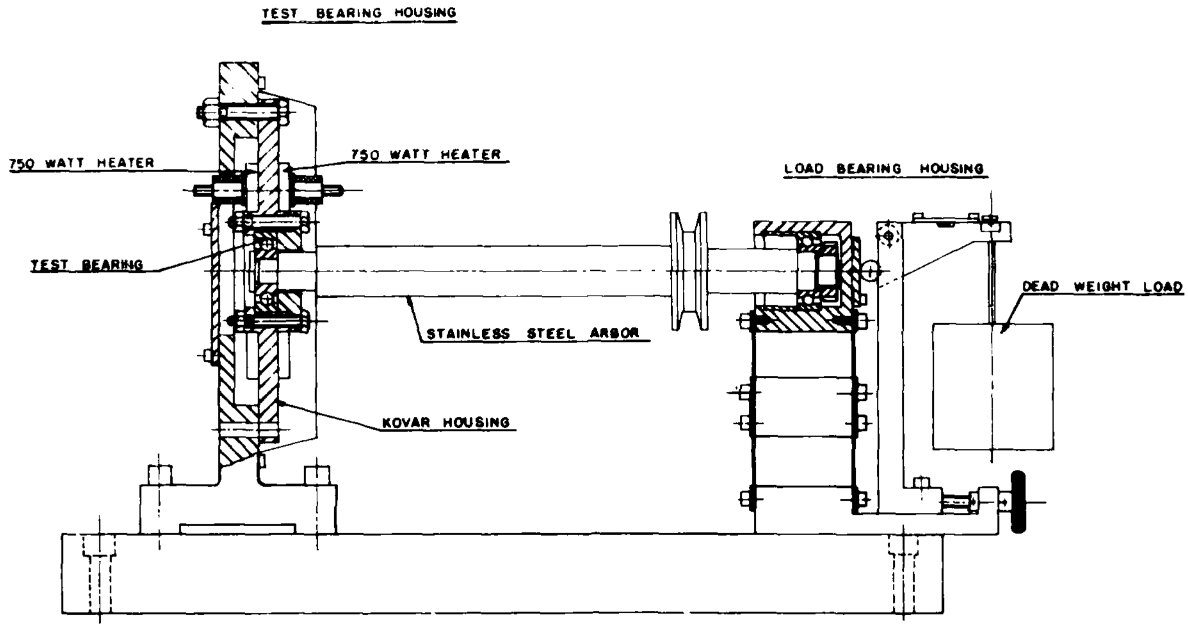


Fig. 6 Full-scale bearing tester used at 1000 F

friction torque and vibration were monitored by a cantilever arm fitted with strain gages that restrained the stationary spindle in torsion. A micrometer head mounted on the loading system was used to measure the axial displacement of the inner ring with respect to the outer ring during operation. Bearing temperature was measured with thermocouples pressed against the bearing outer ring. Powdered lubricant (for high-temperature operation) suspended in a gas stream was supplied through a tube in the bearing housing and was exhausted through a labyrinth seal on the opposite side of the bearing. Thus, the test-bearing environment was separated from the heating-air environment and was controlled by the lubricant carrier gas.

A consideration of possible lubricants for use at the anticipated temperatures led to the choice of molybdenum disulfide since lowest friction and wear of any fairly well-known lubricant at 1200 F has been obtained with MoS_2 in a N_2 carrier (5). Molybdenum disulfide is reported not to decompose thermally below about 1800 F if protected from an oxidizing atmosphere, and also lubricates well over the entire temperature range down to at least room temperature. Because nitrogen is reactive at temperatures above about 1200 F, particularly toward the bearing retainer, an inert gas, argon, was used as the lubricant carrier in the higher temperature experiments. The microfine MoS_2, dried and screened, was supplied to the bearing at an average rate of about 0.16 gram per min.

The retainer materials used in the bearings operated at 1000 F have definite temperature limitations owing to dimensional and structural instability. The AISIType M-1 tool steel softens rapidly and undergoes structural transformations at temperatures higher than 1200 F. Molybdenum plus 0.5 per cent titanium alloy was selected for the retainer in the bearing to operate above 1000 F for several reasons. This alloy retains sufficient hardness, stability, and strength in an inert atmosphere at all temperatures within the capability of the equipment (up to 1800 F), and partial surface oxidation in a blanket inert environment seems to inhibit galling. In addition, its thermal expansivity approaches that of the ceramics and cermets, thus permitting closer clearances and control of geometry over a wide temperature range. A simple retainer design was selected; diametral clearances of 0.005 in. were used both on the retainer outer ring land guide surfaces and in the ball pockets.

The bearing used in this series of tests had hot-pressed alumina balls and K163B1 titanium-carbide-cermet races. The oscillation of the outer race groove was 59.4 per cent and of the inner race groove, 53.1 per cent.

After ultrasonic cleaning of all the bearing parts and adapters, the bearing inner ring adapter was mounted on the end of the precision rotating spindle such that the runout of the axial and radial bearing mounting surfaces was less than 0.0002 in., as determined with a dial indicator, Fig.7. The radial land on the adapter was machined

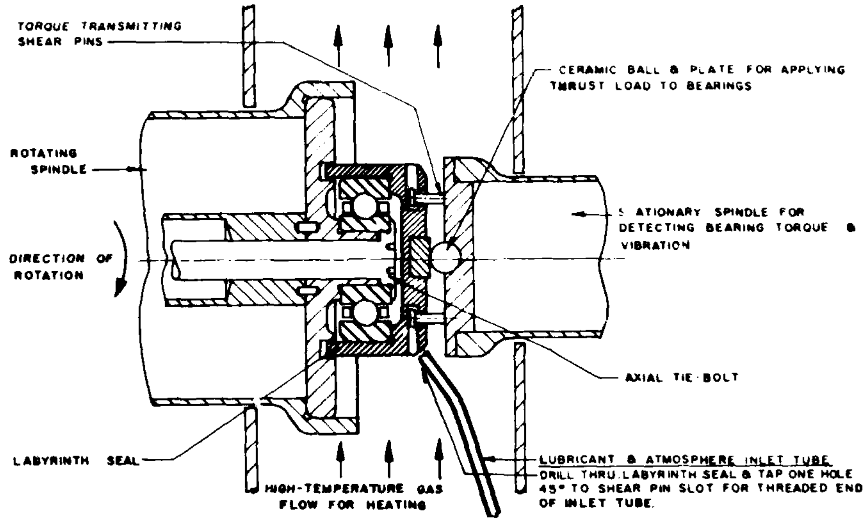


Fig. 7 Machine used for evaluation of ceramic ball bearings at 1500 F

TABLE 7
Summary of Data From Full-Scale Bearing Evaluation at Temperatures Above 1000°F (1)

Bearing Outer Ring Temp., °F.	Total Operating Time at Temp., hr.	Average Flow Rate of Argon, (2) SCFH	Average Coefficient of Friction at Pitch Diameter	Total Variation in Friction Torque, in-lb.
1000	2.1	8	0.02	0.07
1200	1.1	9	0.01	0.05
1350	0.5	12	0.006	0.03
1500	8.2	12	0.006	0.07-0.17(3)
1000-1500 (total)	11.9			

(1) Bearing had K163B1 rings, hot-pressed Al_2O_3 balls, and a Mo+0.5% Ti retainer. It was operated at 8000 rpm under 50 pounds thrust load (calculated maximum Hertz stress on outer race greater than 300,000 psi)

(2) The flow rate of MoS_2 lubricant powder depended on both the flow rate of the argon carrier gas and the level of the powder in the lubricator. A total of 137.3 g of MoS_2 lubricant was used at an average rate of 0.16 g/min during the operating time of the lubricator.

(3) The variation, or vibration, in bearing friction torque increased near the end of the experiment although the level of bearing friction remained essentially constant.

with enough clearance from the bore of the bearing inner ring to just make up for the differential thermal expansion between the adapter and the bearing ring from room temperature up to 1500 F. The bearing inner ring was clamped axially in compression by an Inconel X tie-bolt (threaded into the relatively cool spindle shaft a few inches outside the heating chamber) in a centered position with both axial and radial runout of less than 0.0002 in.

Since test bearings were operated successfully at 1000 F and 5350 rpm, the high-temperature evaluations were started at 1000 F, 50 lb thrust load, and 8000 rpm, the minimum speed of the equipment. Once it was shown that the bearing operated satisfactorily at 1000 F, operation at higher temperatures was attempted.

Results of Bearing Evaluation at Temperatures Higher Than 1000 F. The results of the evaluation for 11.9 hr at temperatures above 1000 F, 8.2 hr of which were at 1500 F, are summarized in Table 7. The experiment was interrupted and the bearing was dismantled for examination after the first 2 hr of operation at temperatures up to 1350 F. The appearance of the bearing was quite encouraging, as the wear tracks on the rings were smooth with no sign of wear. The balls were metallic in appearance instead of gray-white as they were before the experiment, and for the most part, were smooth and very reflective. Careful measurements with an electrolimit comparator indicated that the balls were from 60 to 100 millionths of an inch larger in diameter than when they were new. It appeared that a very even coating of metal, probably molybdenum from the retainer, was deposited on the hot-pressed alumina balls during bearing operation. The retainer sustained the most wear of any part of the bearing. The retainer guide land surfaces were worn 0.0038 in. on the diameter and the pockets had wear scars 0.08 in. wide, on the average. However, the worn surfaces on the retainer were fairly smooth and did not appear to be severely damaged.

Since this bearing was certainly not considered to have failed, but merely to have been nicely worn in, the parts were thoroughly cleaned, including the deburring of the sharp worn edges of the retainer, and reassembled in the machine in as nearly as possible the same identical position in which it was operated before. The bearing

Fig. 8(a) Outer-race pitted area

Fig. 8(b) Outer-race unpitted area; X50 (reduced approximately 22 per cent for reproduction)

was operated again at 8000 rpm under 50 lb thrust load. The temperature was increased to 1500 F and the bearing operated at this temperature for over 8 hr with low friction torque and no sign of failure. Although the average bearing torque remained quite low (about 0.006 coefficient of friction at the pitch line), the variation in friction torque (or vibration) increased noticeably during the 8-hr run at 1500 F.

Upon disassembly, the bearing appeared, on first examination, to be in as good condition as after the first 2-hr run; in fact, the surfaces of the balls, retainer pockets and retainer guide land seemed more smooth and polished than before. No increase in wear on the balls, races, or retainer could be measured. The race wear tracks were not significantly wider than the major axes of the ball contact ellipses calculated by elasticity theory. However, on close inspection, it was found that some spotty superficial pitting damage had occurred in the wear track on the outer race, which was overstressed owing to the large conformity of the outer groove on the balls in the bearing. Even at only 50 lb thrust load, the 59 per cent outer-race conformity, together with the high elasticity of the ceramic balls, produced a maximum contact stress of over 300,000 psi. The partially pitted area extended about 45 deg around the ring. Figs.8(a, b, and c) show the condition of the race wear surfaces at the termination of the test.

CONCLUSIONS AND DISCUSSION

Sliding friction and wear tests and four-ball wear tests show that when lubricants are not used, the performance of K163B1 titanium-carbide cermet operating against itself or against hot-pressed alumina or hot-pressed silicon carbide is superior to other possible two-component combinations of these three materials.

Full-scale ball bearing tests at 1000 F demonstrate that bearings consisting of K163B1 rings and either K163B1 or hot-pressed alumina balls with M-1 tool steel cages can be operated for significant periods (about 10-28 hr) at 5350 rpm and 175 lb thrust load with either metal-free phthalocyanine or molybdenum-disulfide lubricant. Without lubricant, life of the bearing was much shorter (1.4 to 2.9 hr).

A full-scale test at 1500 F on a bearing composed of K163B1 titanium-carbide-cermet rings and hot-pressed alumina balls, with a molybdenum plus 0.5 per cent titanium-alloy cage, has shown that operation at 8000 rpm and 50 lb thrust load is possible for at least 8 hr without significant wear or damage when using molybdenum-disulfide lubricant in an inert atmosphere.

Although the present investigation has demonstrated that rolling-contact bearings made of certain ceramic-cermet materials have the potentiality of operating for significant periods under rather severe conditions of temperature, unit load, and speed, the full capabilities of these bearings have not yet been explored. Furthermore, there are possible approaches for improvement based on observations made during the present investigation.

The improvement in performance of highly refractory ceramic or cermet bearings through the use of lubricants, as demonstrated in the tests at 1000 F, is striking. Also, in the tests at 1000 F it was observed that the elimination of

Fig. 8(c) Inner race; X50 (reduced approximately 50 per cent for reproduction)

vibration, even at the expense of greatly increasing the load, markedly increased the life of the bearing. In the test at 1500 F, bearing torque and vibration were found to be extremely sensitive to both dynamic runout and lubricant flow rate. Therefore, adequate lubrication, internal bearing precision and precise alignment of the bearing during operation appear to be essential to satisfactory performance.

The low wear rate observed in the test at 1500 F suggests that if ceramic bearings are made with high precision and carefully aligned during operation, closer ball-race comformity could be used, thus increasing bearing load-carrying capacity within the contact-stress limitations of the ceramic material. With low wear rate, large clearances to expel wear debris would not be necessary. However, with close conformities it would, of course, be necessary to give careful attention to the thermal-expansion characteristics of the materials chosen for various components of the bearing. In addition, close conformities increase the sliding velocities in the high-load contact regions, thus placing more stringent requirements on the lubricant to maintain an effective lubricating film and reduce harmful tractive and thermal stresses in the contact regions that result from this sliding.

In regard to the superficial pitting that occurred in the outer race wear track of the 1500 F bearing, shown in Fig.8(a), this damage appears to have been caused by a structural breakdown of the surface material subjected to the high contact stresses (over 300,000 psi). Such damage may be typical of the beginning of the attritional-type wear detected in the other bearing experiments. Other studies of cermet wear at high temperatures (6) indicate the existence of partially cracked ceramic grains in these materials that may contribute to surface failure under friction conditions. These results serve to emphasize the importance of the homogeneity and surface integrity of ceramics for use in rolling-contact bearings.

On the basis of the results reported here, the successful operation of ceramic ball bearings at temperatures of 1000 to 1500 F, appears entirely feasible. With the development of the more refractory ceramics and lubricants for bearings, satisfactory operation at even higher temperatures should be possible.

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