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PART I

SOLID LUBRICATED BEARING TECHNOLOGY, PART I — SOLID LUBRICATED HIGH SPEED BALL BEARINGS

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FOREWORD

This report was prepared by Westinghouse Research Laboratories, Beulah Road, Churchill Boro, Pittsburgh, Pennsylvania, 15235, under USAF Contract Number F-33615-70-C-1736. The work is being administered under the direction of the Air Force Aero Propulsion Laboratory, Air Force Systems Command, with Mr. M. A. Sheets acting as project engineer. The Air Force Task Number is 304806, Project 3048.

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This report covers work conducted from July 1, 1970 to June 30, 1971.

This technical report has been reviewed and is approved.

Howard F. Jones, Chief
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ABSTRACT

This report describes the first year's effort of a two year program that has as its objective the development of improved self-contained, solid lubricated ball bearings capable of at least 25 hours operation at speeds approaching 60,000 rpm. The program encompasses a temperature range of -40°F to 600°F and a maximum load combination of 75 lb thrust/25 lb radial. The work is a follow-on effort to the completed Air Force funded program F33615-68-C-1250.

During this reporting period, a high speed test stand was designed and manufactured that is capable of functionally evaluating the performance of solid lubricated, 207 ball bearings over a 30,000 to 60,000 rpm speed range. The device incorporates a 40 H.P., variable speed motor driving a transmission having a 15:1 gear ratio. A test head containing two test bearings is coupled to the output shaft of the transmission.

The functional test portion of the program has thus far achieved a maximum life of 15 hours on a solid lubricated ball bearing operating at 30,000 rpm and a temperature of 200°F.

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I. INTRODUCTION

Current trends toward high mach number operation and high thrust to weight ratios in the development of propulsion and power generation systems dictate the requirement for reliable, long-term lubrication under severe operating conditions. The upper operating temperature of load-bearing systems is limited by the capabilities of available lubricants. In addition, the lubricant problem is greatly complicated if an oxidizing environment coexists with a high-temperature operating requirement. The upper practical limit of conventional fluid lubrication is about 600°F. In order to maintain liquid lubricant temperatures at acceptable levels in many high mach number systems, severe penalties would be imposed on the system with respect to high heat load and weight. As a result, other lubrication techniques must be developed to satisfy this requirement. One such technique is the use of a solid lubricating retainer in a ball bearing, thereby providing a self-contained load bearing system. Over the past five years, efforts have been focused specifically in this laboratory on the development of self-contained, solid lubricated ball bearings for use under combined high speed, high load, and high temperature operating conditions. The work was sponsored under contracts from the Aero Propulsion Laboratory, Wright-Patterson Air Force Base.[1,2]

Under Air Force Contract F33615-70-C-1736, a two year program has been undertaken to extend solid lubricated bearing technology from that which was achieved during the earlier programs to capabilities that more nearly match the requirements of current and future applications. During this effort, functional tests on experimental, self-lubricated ball bearings are being performed over the following range of operating conditions:

Temperature: -40 to +1200°F

Speed: Zero to 60,000 rpm

Load: 75 lb thrust/25 lb radial

Atmosphere: From atmospheric characteristics at sea level to those simulating 100,000 ft altitudes.

Bearing size: 204 and 207

The primary goal of the program is the development of a self-contained, solid lubricated ball bearing system capable of 25 hours life at a speed of 60,000 rpm.

In support of the high speed functional test portion of this effort, the initial phase of the program involves the design and construction of a device capable of driving one or more test bearings at a speed ranging from 30,000 to 60,000 rpm. The bearings are to be of the 207 size and carry a load of 75 lb thrust/25 lb radial.

II. BACKGROUND DISCUSSION

A. Material Considerations

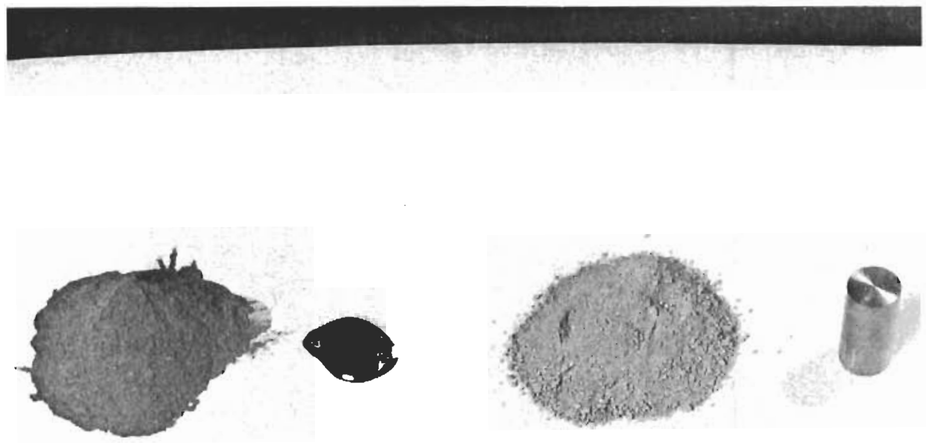
In an inert environment such as nitrogen or high vacuum, most solid lubricants of the MoS_2 type are chemically stable to at least 2000°F. The introduction of oxygen to their environment, however, limits their long-term usefulness to temperatures of about 600°F. Their oxidation produces metal oxides which, in most instances, are quite abrasive. In all cases the lubricating ability of the oxide cannot compare with that of its lubricant counterpart. At the onset of the oxidation process, higher friction coefficients, greater wear rates, and general deterioration of performance occurs in any bearing system utilizing these materials as lubricants. This oxidation problem is considered one of the most serious obstacles in the development of a long-term bearing system for operating temperatures of 600°F and higher.

In applications encompassing ultrahigh speeds and moderate temperatures, such as the zero to 60,000 rpm and 600°F requirements of this program, it is imperative that the ball bearing retainer, employed as the lubricating component, possess mechanical properties capable of tolerating the centrifugal forces imposed upon it. It is well known that metal matrix, self-lubricating composites are inherently weaker, structurally, than a truly sintered metallic composite formed through powder metallurgy techniques. This is due to the fact that in order to impart low friction-wear characteristics, it is necessary to incorporate into their matrix relatively large quantities of solid lubricants such as PTFE or MoS_2. This lack of high mechanical strength, therefore, is considered the major obstacle in the development of an ultrahigh speed, moderate temperature, solid lubricated bearing system.

1. Oxidation Resistant Lubricating Composites WSe_2/GaIn

One of the basic self-lubricating composites used in this program is composed of tungsten diselenide (WSe_2) interacted with an alloy of gallium-indium. A complete description of the physical, chemical and lubricating characteristics of this material has been presented in the literature.[3,4,5] These references also describe in detail the results of efforts designed to optimize this composite with respect to mechanical strength, lubricating properties, and oxidation resistance. For this reason, the following description of the material will be kept brief.

The technique for fabricating these composites involves the inter-mixing of the solid lubricant (WSe_2) with the metal gallium or a gallium-indium alloy. It is performed with the solid lubricant in powdered form (-200 + 325 mesh) and the metal in the molten state, as shown in Figure 1. The mechanism of the interaction between tungsten diselenide and the gallium-indium alloy is such that from 5% to 20% (wt) of the metal is accepted by the solid lubricant. Upon acceptance of the metal, the powdered lubricant/metal aggregate is charged to a die and compacted at room temperature and high pressure. After stripping, the compact is subjected to a carefully controlled high temperature cure of 8 hrs at 450°F, 4 hrs at 650°F, and 4 hrs at 1030°F. The resultant piece is a hard, non-porous material, capable of being machined or drilled, that can resist oxidizing environments at temperatures two to three times higher than the parent lubricant. In Figure 2 the oxidation rates of a number of solid lubricants and seal materials are compared with that of a WSe_2/GaIn composite at temperatures up to 1500°F. The WSe_2/GaIn composite contained 10% (wt) gallium-indium (75-25) alloy. All materials were in pellet form of about the same surface area and were held at each temperature level for one hour periods. The results show that a 10% gallium-indium pretreatment provided a WSe_2 compact that remained relatively unaffected by an environment of 1400°F air. In comparison, untreated tungsten diselenide exhibited a 6% weight loss/hour at 575°F.



Left Tungsten Diselenide with Gallium-Indium (75-25)
Center Tungsten Diselenide Powder after Amalgamation
Right Pressed, Cured Pellet of Tungsten Diselenide Amalgam

Fig. 1 - Steps in Forming the Tungsten Diselenide/Gallium-Indium Composite

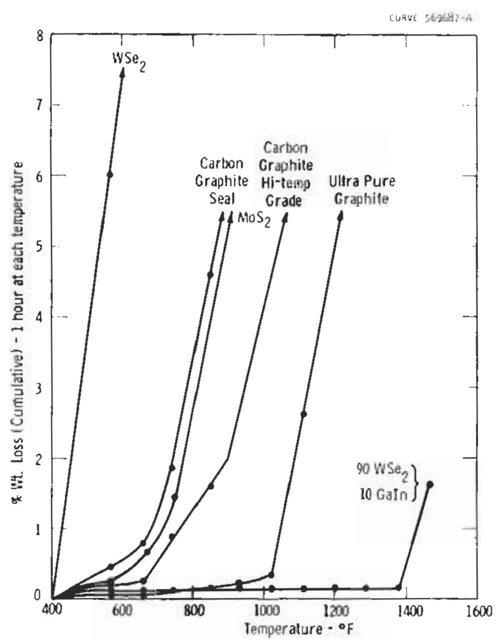


Fig. 2 - Oxidation Characteristics of Solid Lubricants

Optimization studies showed clearly that a composite consisting of 80% WSe_2 - 20% GaIn (wt %) exhibited the best overall performance with regard to friction-wear characteristics and mechanical strength. Table I presents the physical and lubricating characteristics of this composite. In addition, optimum fabricating and curing conditions are given.

2. Silver Matrix Composites

A second type of self-lubricating composite developed during the performance of the previous contract[(2)] is comprised of a silver or silver-mercury matrix. Throughout this matrix is distributed a mixture of molybdenum diselenide ($MoSe_2$) and polytetrafluoroethylene (PTFE) particles. The composites are formed by compacting the thoroughly blended metal-resin-solid lubricant aggregate in a tool steel die under high heat and pressure. A complete description of the general physical and lubricating characteristics of these composites, as well as the technique used to fabricate them, can be found in the literature.[(6)] Chemical modification of the PTFE used in the composite significantly reduces both the wear rate and the friction coefficient of these materials. The physical properties of the PTFE are changed in such a way that it exhibits a sharp melting point at ∿600°F. As a result, its use in a self-lubricating load bearing composite at this temperature provides a thin, fluid film of the resin on the metal components with which it is in contact. In this way, dynamic friction coefficients and wear, of itself and of bearing components, are reduced. The resin is modified in the following manner:

Approximately 200 grams of powdered PTFE (∿100 mesh) are placed in a porcelain dish and placed in a furnace at about 1100°F for a period of 1-1/2 hours. Following this exposure, the material is removed and placed in a second furnace for 3 hours at a temperature of 700°F. During this second firing, the modified PTFE changes in color from brownish-gray to pure white due to the removal of certain volatiles and carbonaceous components. When pulled from the furnace, the resin is in the molten state and quite viscous. Upon cooling, the solid material is then ground to a 40-50 mesh particle size. Subsequent to the grinding

TABLE I

PREPARATION,

PHYSICAL PROPERTIES AND LUBRICATING CHARACTERISTICS OF 80% WSe_2 - 20% GaIn (75% Ga - 25% In) wt %

Preparation	200 mesh WSe_2 ball-milled with molten alloy
Fabrication (Pressing)	Room temperature 25,000 psi - single action die
Cure (Air Environment)	8 hours at 450°F 4 hours at 650°F 4 hours at 1030°F
Friction Coefficient	
(80°F - 70 fpm) 1000 psi	0.06
(600°F - 140 fpm) 500 psi	0.05
Wear - gms/hr	
(80°F - 70 fpm) 1000 psi	0.004
(600°F - 140 fpm) 500 psi	0.006
Compressive Strength - psi	19,980 (average of six tests)
Tensile Strength - psi	2,075
Shear Strength - psi	1,260
Flex Modulus - psi	0.25×10^6

operation, the modified PTFE is then mixed with the silver and solid lubricant components of the composite. Finally, the aggregate is formed into a machinable body in the manner described earlier. Figure 3 is a typical macrostructure of one such composite having the composition: 70% Ag - 22-1/2% PTFE - 7-1/2% $MoSe_2$. (vol)

Table II presents the friction-wear characteristics of a number of silver matrix composites employing modified PTFE and compares them with similar composites containing the standard resin.

TABLE II

EFFECT OF MODIFIED PTFE ON THE FRICTION-WEAR** CHARACTERISTICS OF COMPOSITES

	75°F		600°F	
Composition Vol %	Friction Coefficient	Scar mm	Friction Coefficient	Scar mm
70 Ag-20 PTFE-10 $MoSe_2$	0.21	2.0	0.45	12.0
70 Ag-20 WTFE* -10 $MoSe_2$	0.19	1.5	0.12	3.5
80 Ag-15 PTFE-5 $MoSe_2$	0.16	2.0	0.19	6.0
80 Ag-15 WTFE* -5 $MoSe_2$	0.23	3.0	0.09	3.3
70 AgHg-20 PTFE-10 $MoSe_2$	0.11	2.0	0.06	4.0
70 AgHg-20 WTFE* -10 $MoSe_2$	0.16	2.0	0.01	3.0

* Modified PTFE - m.p. ∿590°F
** 2550 fpm - 240 psi on 1/4" face, line contact

B. Lubricant Application Technique

The primary purpose of any lubricant in a load-bearing system is to prevent metal-to-metal contact and thereby prevent wear. In a solid lubricated load-bearing system, therefore, it is imperative that the lubricant not only thoroughly films the load-bearing surfaces, but also is available for replenishing purposes as the lubricant is lost due to wear. In a ball bearing, metal-to-metal contact occurs between

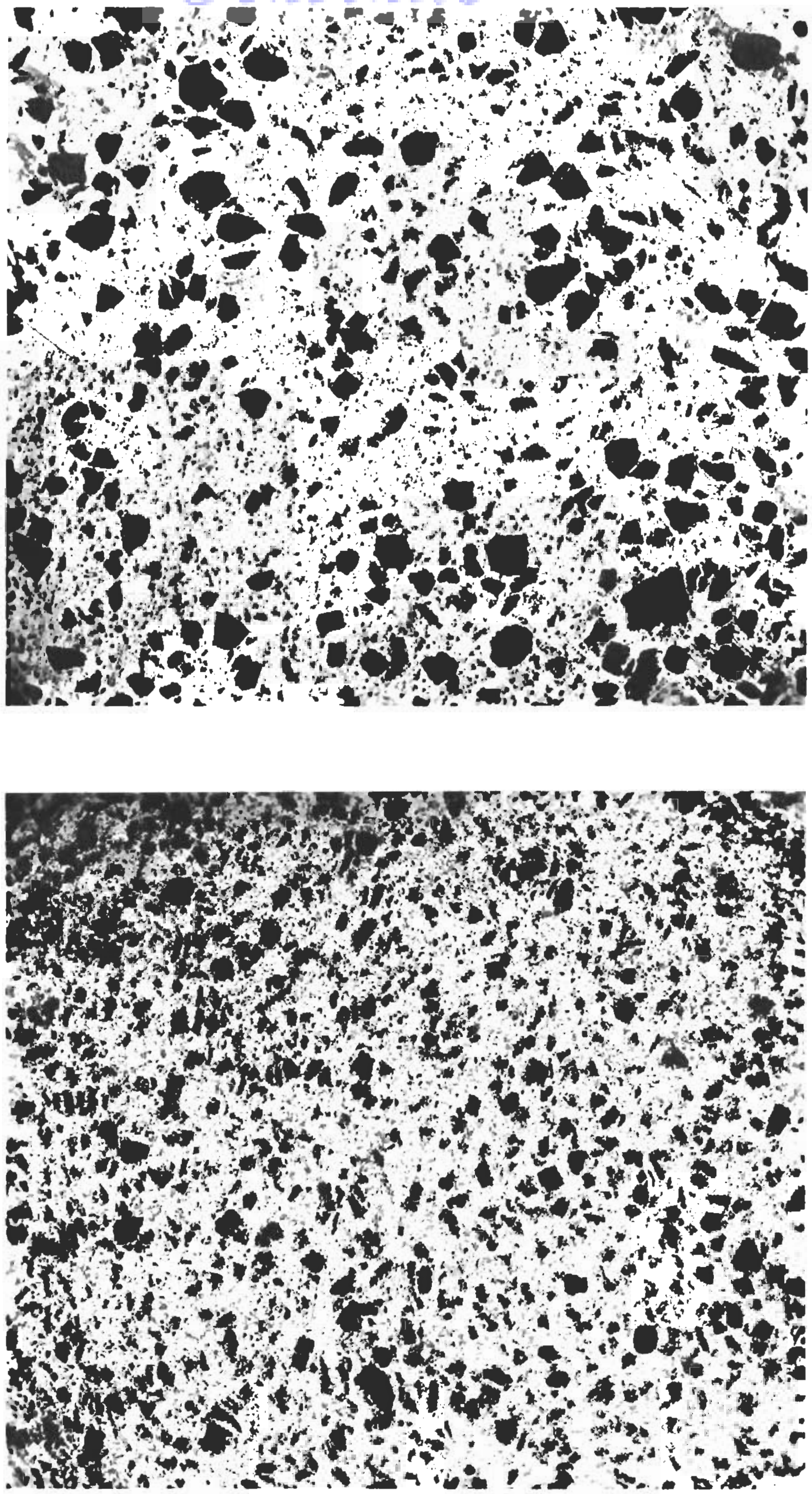
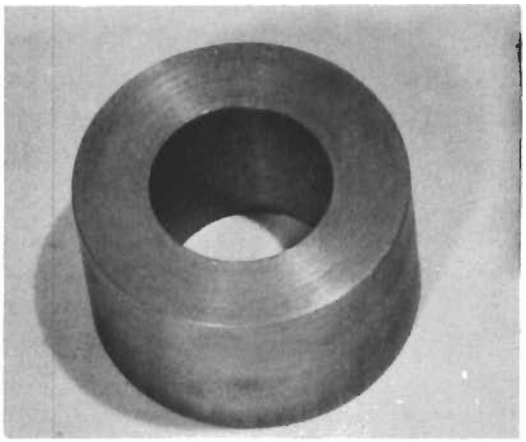


Fig. 3 —Typical macrostructures of hot pressed self lubricating composites

the balls and the retainer (sliding) and the balls and the inner and outer races (primarily rolling). The balls are in sliding or rolling contact with every other component of the bearing system. If, therefore, a solid lubricating film can be established on these balls and replenished as needed, solid lubrication of the entire system can be achieved through what is called a film transfer mechanism. This mechanism is simply the transference of a solid lubricating film from one metal surface to another with which it is in sliding or rolling contact.

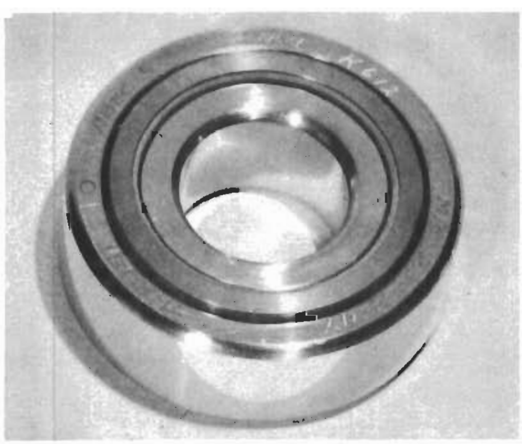
The technique employed in the program for accomplishing this type of lubrication in a ball bearing is to replace its metal retainer with a welded, double-shrouded retainer fabricated from a self-lubricating composite. Figure 4 is a photograph illustrating the various steps involved in the fabrication and assembly of one such bearing system.

In operation, minute quantities of lubricants are continuously metered to the balls in sliding contact with the retainer. The balls, in turn, transfer this lubricant to the race groove in which they are rolling. In this way, all critical load bearing surfaces are coated with a thin tenacious lubricating film. Since the balls are continuously sliding against the self-lubricating retainer, the film can be replenished if lost through wear. It will be noted from data presented in Table III that the application of this technique during previous programs[7] proved quite successful.



Tungsten diselenide-gallium indium blank for 204 size bearing

Double, titanium shrouded retainer machined from blank



204 size ball bearing equipped with shrouded retainer

Figure 4 — Application technique for WSe_2/GaIn double-shrouded composite retainers in ball bearings.

Table 3 — Pertinent Functional Test Results, Self-lubricating Retainer, 204 and 207 Ball Bearing

Bearing Size	Test Temp., °F	Speed, rpm	Load, lb Thrust	Load, lb Radial	Life, hr	No. of Balls	Retainer Material[a]
204	600	10,600	50	50	215	8	80 WSe_2-20 GaIn[b]
204	600	10,600	50	50	337	8	93 WSe_2/GaIn-5Ag-2CaF_2[b]
204	600	10,600	50	50	370	8	70 AgHg-20 WTFE-10 $MoSe_2$
204	600	20,000	100	25	101	8	80 WSe_2-20 GaIn[b]
204	750	10,600	50	50	140	8	CR-82 Graphite
204	900	10,600	50	25	80	8	Clevite 300
204	1,200	10,600	50	10	145	8	Clevite 300
204	1,500	10,600	50	10	10	8	Clevite 300
204[c]	-225 to +100	10,600	50	50	>50	8	80 WSe_2-20 GaIn[b]
207[d]	75	3,400	50	50	>17,000	9	70 AgHg-20 WTFE-10 $MoSe_2$
204[e]	150	3,400	10	5	5,000	8	58 AgHg-34 WTFE-8 $MoSe_2$
207	200	10,000	50	50	7,030	9	70 AgHg-20 WTFE-10 $MoSe_2$
207	450	10,000	50	50	1,400	9	70 AgHg-20 WTFE-10 $MoSe_2$
207	600	10,000	50	50	456	9	70 AgHg-20 WTFE-10 $MoSe_2$
207	600	8,000	50	30	660	9	70 AgHg-20 WTFE-10 $MoSe_2$
207	600	10,000	250	250	86	9	80 WSe_2-20 $MoSe_2$[b]

a. All percentages are by volume unless otherwise noted
b. Wt %
c. Test stopped for inspection purposes.
d. Test remains in progress at this time.
e. Environment simulating ∿200,000 ft altitude.

III. MATERIAL STUDIES

A. Test Facility

All friction-wear screening tests performed on candidate self-lubricating composites were made on the High Temperature Wear and Friction Test Apparatus shown in Figure 5. The test specimen was held in the left side of two shoes and rubbed against the edge of a rotating tool steel disc (1.375 in. in dia.) and shaft assembly. Sliding velocities of 230 and 1650 ft/min. were employed. The shoes were mounted on a torque cone and bearing assembly that pivoted in a concentric arc about the disc shaft. The load was applied to the test specimen through the left shoe with a lever and dead weight. Wear, friction values, and temperature were observed on the test specimens. The wear was measured as the chord of the arc worn on the 0.25 in. face. Since the wear area varied for the different specimens, the unit force varied. The results of tests were observed as scar width in mm and reported as wear for various loads in lb. per sq-in. The friction was determined at 5 min intervals by observing the strain gage deflection on an SR-4 strain indicator as transmitted from a dynamometer ring. The temperature was monitored with a thermocouple, the bead of which was located directly behind (1/16 inch) the line of contact between disc and specimen. In most cases the specimens were fabricated directly to the finished dimension (0.25" x 0.75" x 0.50") in a tool steel rectangular die. Test duration was 30 min.

B. Ferrous Alloy Friction-Wear Studies

Functional testing during a previous program* had demonstrated that the major cause of failure for 204 ball bearings, operating in the temperature range of 1200° to 1500°F, was severe retainer and bearing component wear. For example, at 1200°F, 10,600 rpm and 50 lb thrust/

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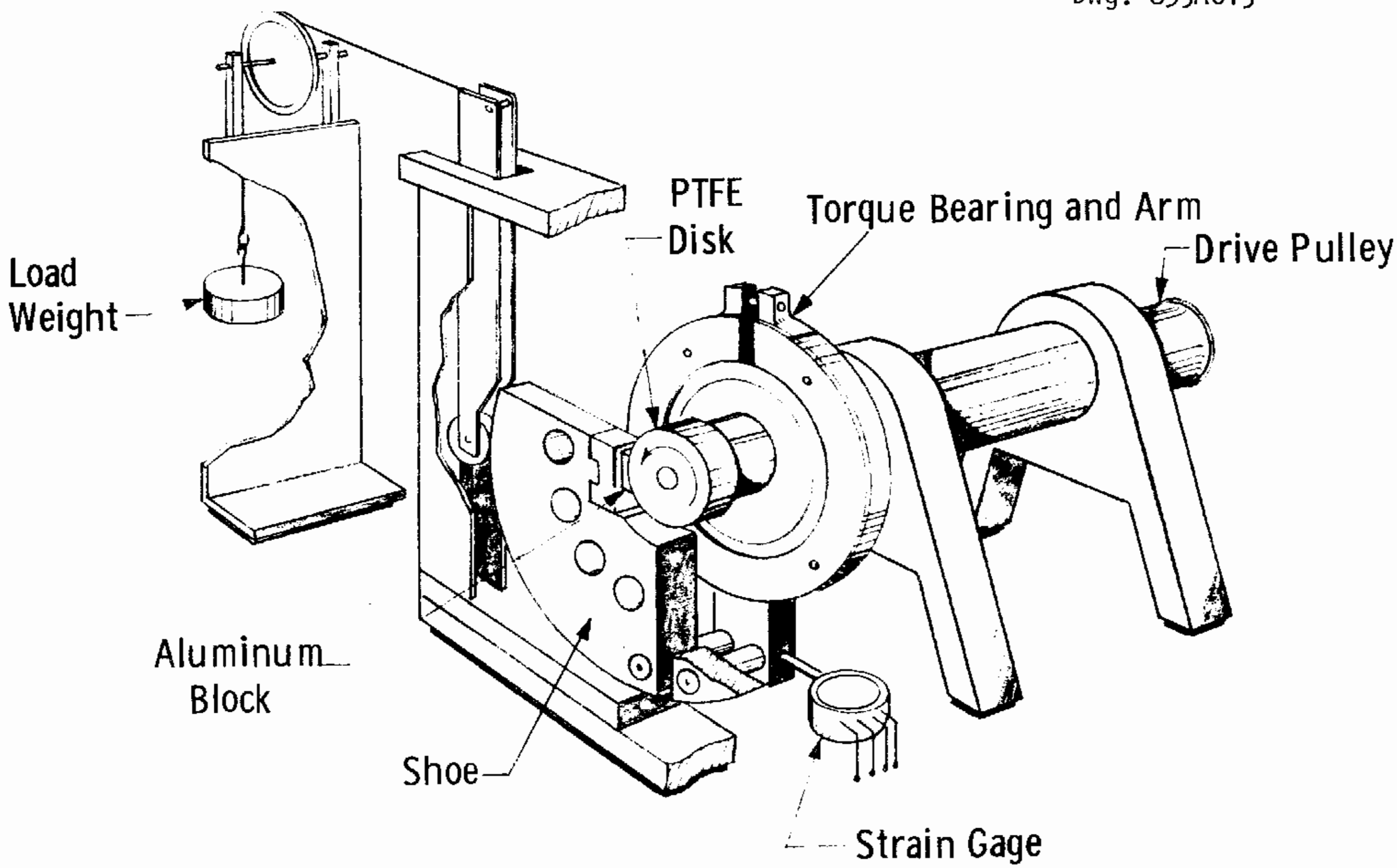

Figure 5 — High temperature wear and friction test apparatus.

25 lb radial loading, a Clevite 300 retainer provides an operating life of 100 hours before failure due to bearing component wear. As mentioned previously, one of the tasks of this program was to conduct exploratory functional testing at 1200°F in an attempt to improve on this life. In support of this task, a series of metal powders was obtained from the Pfizer Minerals, Pigments, and Metals Division and evaluated - after green pressing and sintering - with respect to their friction and wear characteristics. The materials are basically iron powders with varying copper content. Micro-examination of the powder shows each particle to be composed of iron infiltrated with copper having a microstructure similar to that produced by conventional infiltration. Pfizer's technique, however, provides materials of substantially higher Cu content than can be achieved by normal infiltration techniques. The powders contained 1% (wt) graphite and were sintered in dissociated ammonia at 2050°F for a period of 35 minutes at temperature.

The results of the friction-wear measurements made on these materials are summarized in Table 4. The tests were performed over a 900°F temperature range at a surface velocity of 1650 fpm. The wear resistance of the three materials under all test conditions was excellent. It was noted, however, that only that material having the highest copper concentration exhibited acceptable friction coefficients over the full range of temperatures evaluated. Only at the elevated temperature of 900°F did all materials exhibit moderate to low friction coefficients. The data indicate that these materials are functioning as high temperature, self-lubricating composites in a manner similar to that of Clevite 300; i.e., formation of a self-lubricating film in a high temperature — oxidizing environment. Because of the high mechanical strength of these composites, an evaluation of their performance as self-lubricating retainers appeared warranted. The results of these tests are discussed in a subsequent section of this report.

A second series of experiments was undertaken to investigate the friction - wear characteristics of a series of ferrous alloys composed of the same metallic components of Clevite 300 but in varying concentrations. Specimens of the same composition were green pressed at 75,000 and

TABLE 4 - Friction-Wear Characteristics of Prefiltron*
Iron-Copper Composites

Grade	Copper** Content-Wt.%	Test Temp. °F	Friction Coefficient	Wear MM
220107	22	75	0.24	3.2
	22	600	0.29	3.3
	22	900	0.08	2.1
12042	12	75	0.36	3.1
	12	600	0.31	3.0
	12	900	0.14	2.0
04103	4	75	0.20	2.9
	4	600	0.16	3.0
	4	900	0.17	2.1

* Trademark -- Pfizer

** Balance - iron + 1% graphite

Test Conditions: 1650 fpm at 3 lb. load on 1/4" face-line contact.

100,000 psi and subsequently sintered in a vacuum environment at temperatures of 600, 800, and 1000°C. The results of the friction-wear tests are given in Table 5. Relatively high friction coefficients were experienced on all samples tested, although wear rates, for those samples containing 20% and 30% molybdenum and sintered at 800°C or higher, were quite acceptable. High wear and friction coefficients were experienced in those cases where the lowest sintering temperature (600°C) was employed as well as where a molybdenum content $< 20\%$ was present. Because of the high friction coefficients exhibited by these materials, no functional evaluation of them as ball bearing retainers was undertaken.

TABLE 5 - EFFECT OF FABRICATING PRESSURE AND SINTERING TEMPERATURES ON FRICTION-WEAR* PROPERTIES OF FERROUS ALLOYS

Pellet No.	Composition - Wt.%	Pressing Load - psi	Sintering Temp. °C	600°F Friction Coef.	600°F Wear mm
1	60Fe-30 Mo-10 Co	75000	600	0.34	15
13		100000	600	0.15	13
5		75000	800	0.42	3.5
17		100000	800	0.22	3.4
9		75000	1000	0.28	3.3
21		100000	1000	0.49	2.9
2	60Fe-20 Mo-20 Co	75000	600	0.38	11
14		100000	600	0.28	10.5
6		75000	800	0.48	3.4
18		100000	800	0.35	4.3
10		75000	1000	0.28	4.2
22		100000	1000	0.28	4.0
3	60Fe-10 Mo-30 Co	75000	600	0.35	8.0
15		100000	600	0.29	8.4
7		75000	800	0.32	9.0
19		100000	800	0.21	6.5
11		75000	1000	0.18	7.3
23		100000	1000	0.32	6.4
4	60Fe-15Mo-15Co-10MoSe$_2$	75000	600	0.58	--
16		100000	600	0.56	7.3
8		75000	800	0.44	4.9
20		100000	800	0.43	4.8
12		75000	1000	0.59	4.6
24		100000	1000	0.38	3.1

*Load - 240 psi; Speed - 1600 fpm

IV. HIGH SPEED TEST FACILITY
60,000 rpm TEST STAND DESIGN

Phase I of this program involves the design and construction of a facility capable of functionally evaluating the performance of 207 size, solid lubricated ball bearings over a 30,000 to 60,000 rpm speed range. Evaluation of various propulsion units capable of driving a ball bearing test spindle over this speed range resulted in two promising approaches. The first utilized a liquid cooled, variable frequency, motorized high speed precision spindle. Power to this device would be provided by a 30 KVA motor-generator set providing variable frequency over a 500 to 1000 cycle range. Power output from the 2 pole, motorized spindle would be approximately 3 horsepower. Since the solid lubricated test bearings would be employed to support this spindle, fail-safe cut-out devices would be a necessity in this design to prevent severe stator damage in the event test failure permits the rotor to contact the stator under high speed operating conditions.

The second approach - and that which was finally selected - was to employ a high speed transmission driven by a variable speed propulsion unit. Coupled to the high speed output shaft of the transmission is a test cartridge containing two preloaded, 207 size test bearings. Figure 6 is an overall schematic view of the test apparatus. The propulsion unit is a Westinghouse 22-1000, 40 HP adjustable speed, DC drive. It is capable of operating at a maximum speed of 4150 rpm regulated to within ±1% by means of a separate tachometer unit. The unit includes a solid state Thyrister controller and operator's station.

The high speed transmission is a Model 215D, modified by the Cotta Transmission Company for operation at 60,000 rpm. The 15:1 ratio unit is equipped with a pressure lubrication system furnishing oil mist on bearings and spray jet on gears. A water to oil heat exchanger is an integral part of the unit. Coupled to the high speed, output shaft of this transmission is the bearing test head which was designed and fabricated in this laboratory. Figure 2 is a schematic of the test fixture. The dual, 207 size test bearings are preloaded by means of six springs contained in a cartridge that in turn is loaded against the lead bearing's outer race. The springs provide a total load of 70 lbs. in thrust. The test bearing shaft is coupled to the high speed output shaft of the transmission by means of a quill containing a central shear section and an internal spline on either end. Use of this double spline design in conjunction with relatively loose fits between the spline provides a relatively flexible coupling system in which the concentricity of the two shafts and total run-out are no longer critical parameters. The design of the quill is such that it will shear through its central section upon reaching an operating torque less than that required to stall the output shaft (test bearing seizure). In this way, catastrophic failure of the output shaft and integral gearing is avoided.

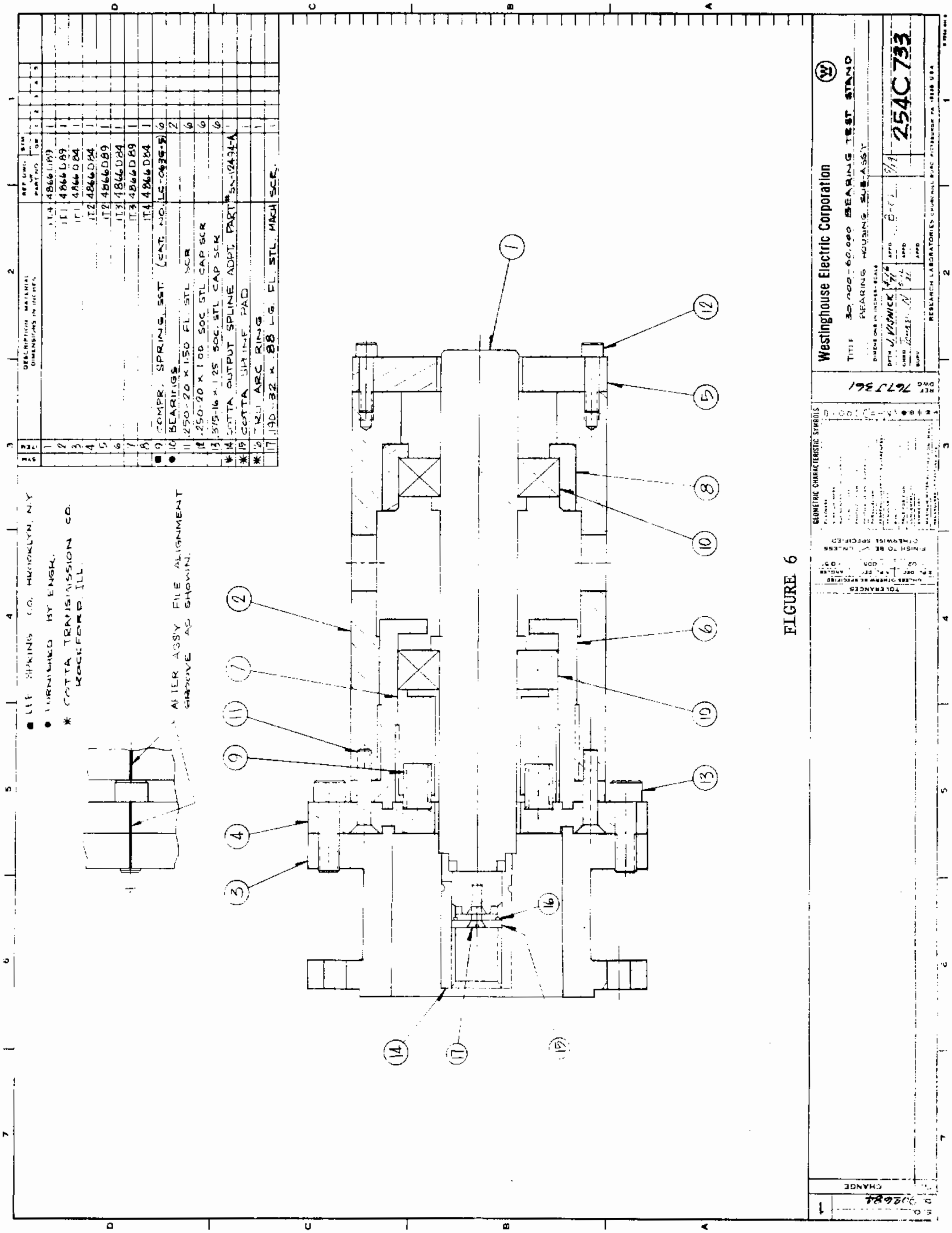
Westinghouse Electric Corporation
30,000-60,000 BEARING TEST STAND
BEARING HOUSING SUB-ASSY
254C733
RESEARCH LABORATORIES
COMPR. SPRING SET
BEARINGS
COTTA OUTPUT SPLINE ADPT. PART
COTTA SHIM PAD
TRU ARC RING
LEE SPRING CO. BROOKLYN, N.Y.
FURNISHED BY ENGR.
COTTA TRANSMISSION CO. ROCKFORD, ILL.
AFTER ASSY FILE ALIGNMENT GROOVE AS SHOWN.

FIGURE 6

V. FUNCTIONAL TEST RESULTS

A. Test Facilities

Nine testing stations are available for functionally evaluating 204 and 207 bearings at speeds of 10,600, 21,500, and 30,000 rpm. Three stations operate at each speed level, with one of each capable of simulating an altitude of 200,000 ft. Figure 7 is an overall view of the test laboratory with one vacuum shroud and one oven installed for illustration purposes. Spindle bearings and drive motors are water cooled. The 10,600 and 21,500 rpm spindles are identical with the exception of drive motors.

A hot worked M-2 tool steel shaft, supported by 207 bearings at both the drive end and the test end, is direct-driven by a water cooled electric motor. Use of 4-pole aircraft motors, which normally provide one horsepower at 11,200 rpm, and variation of the number of poles and frequency of the motors permit variation in speed. Two generators, one providing 360 cycle, 180 volt, 3-phase current, and one providing 510 cycle, 250 volt, 3-phase current, provide an independent power source for the drive motors. One bank of three test stations operating at 10,600 rpm, and one bank of three test stations operating at 21,500 rpm are powered by a generator furnishing 360 cycle current to drive motors which are identical, except for the number of poles employed. A 4-pole motor is used on the former bank; a 2-pole motor is used on the latter. One additional bank of three test stations operating at 30,000 rpm employs a second type 2-pole motor powered by the second generator furnishing 510 cycle, 250 volt, 3-phase current.

Test bearings are directly loaded. It should be pointed out that a compensating bellows arrangements is used for the vacuum

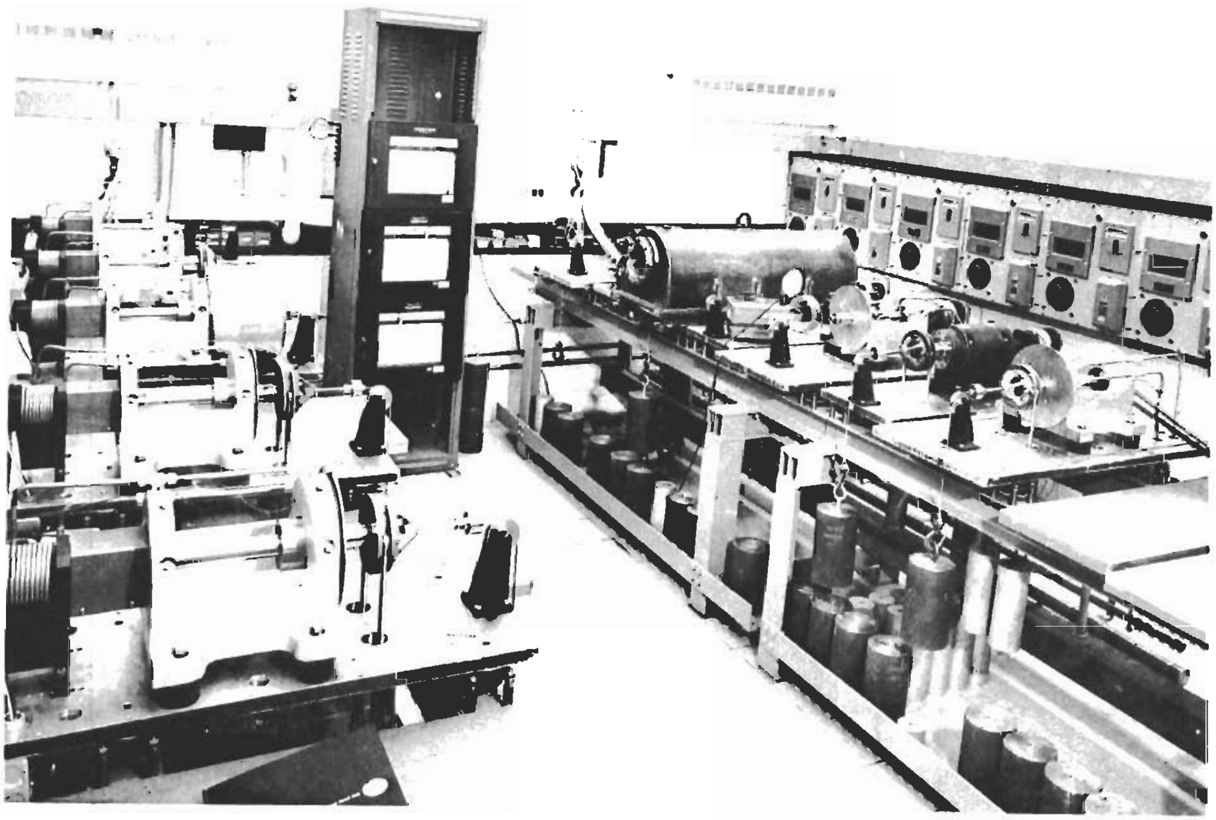

Figure 7 — High speed bearing test laboratory.

test stations where the loads are applied externally. The following data can be obtained from a test run: bearing temperature, oven temperature, environmental pressure, bearing speed, and bearing component wear. All shafts, housings, load devices and other components directly associated with test bearings operating up to 900°F are fabricated from M-2 tool steel hardened to R_C 58-60. All such components employed in 1200° and 1500°F functional tests are fabricated from Rene' 41.

In the majority of cases, all functional tests described in this report were permitted to operate to failure. The time of bearing failure was selected as the point at which the operating temperature of the bearing exceeded the test temperature by approximately 30°F. This temperature rise usually occurred quite abruptly and invariably indicated that contact had occurred between one or more of the balls and the reinforcing metal shrouds. This, in turn, indicated either excessive pocket wear or retainer fracture in the ballpocket bridge.

B. Functional Test Results:

1. 204 size - 10,600 rpm - 900° to 1200°F

A total of eleven tests was performed on the 204 size bearing system under the above conditions. A twelfth test was performed at a speed of 3,500 rpm in a vacuum environment simulating ~240,000 ft. altitude. The bearings were fabricated from M-2 tool steel and equipped with self-lubricating retainers having an 8 ball complement. Pertinent design parameters of this bearing are as follows:

Race Curvatures:	
a. Inner -	0.57
b. Outer -	0.56
Free Contact Angle	15°
Ball Diameter -	0.281"
Internal Clearance -	0.0045"
Ball Complement	8

The primary objective of these tests was to improve upon, if possible, the life achieved during the previous program* on this bearing

*F33615-68-C-1250

system at 1200°F. This life was 140 hours at a speed of 10,600 rpm under a 50 lb. thrust/10 lb. radial load. The results of these tests are summarized in Table 6. Runs 1 thru 6 were performed for the purpose of evaluating the effect on life of various fits between a Clevite 300 retainer and the various bearing components. It will be noted that an extremely tight fit of 3 mils, between the retainer and bearing balls, causes a sharp reduction in operating life. Only by increasing the fit to the original 10 mils was the operating life of approximately 130 hours, achieved in prior work, repeated. Bearing failure in all cases was caused by severe bearing wear and rough operation. In Run 6, it should be noted that an increase in the radial load from 10 lb. to 25 lb. caused a reduction in life from 130 hrs. to 50 hrs.

Runs 18, 19, and 20 were performed for the purpose of evaluating the performance of the ACF-10 grade of graphite supplied by Poco Graphite, Inc. The material exhibits a high strength to density ratio and a very fine and uniform grain structure. As seen from Table 6, under a 50 lb. thrust/50 lb. radial load and a speed of 10,600 rpm, a maximum life of 12 and 2.5 hrs. was obtained at 900° and 1200°F, respectively. A 50% reduction in radial load in Run 20 resulted in only a slight increase in 1200°F life. This life is singificantly less than that obtained from two grades of carbon-graphite* supplied by Stackpole Carbon Company and evaluated during the previous program. No further high temperature work was therefore undertaken with this material

The final two high temperature runs listed in Table 6 were performed on two different grades of Prefiltron described in Section III-B of this report. Prior to assembly, the retainers were preoxidized at a temperature of 500°C for 5 hours. That retainer fabricated from the Prefiltron grade containing 22% Cu (#220107) operated for a period of 3 hours before excessive vibration caused test shutdown. Severe bearing wear was observed in all bearing components. Upon reducing the Cu content in the retainer from 22% to 12% (wt.), it was found that bearing life dropped sharply, with bearing component wear and vibration again causing test failure. Since these materials do not apparently offer any improve-

*Grades CR-82 and 2066

ment in life over that already achieved with certain carbon-graphite grades and Clevite 300 under these test conditions, their performance will not be evaluated further.

Run 10 was conducted on 204 bearings installed in a 3/4 HP motor operating in an environment simulating an altitude of ~240,000 ft. (1 micron pressure). To insure adequate lubrication in this environment, it had been necessary to employ a AgHg retainer containing 14% (vol.) more PTFE than normally used in this composite. The bearings operated for a period of 5058 hours prior to shut-down for inspection purposes. Examination revealed that failure was imminent in both the front and rear bearing locations. Ball-shroud contact had occurred in most of the retainer pockets, and heavy wear was observed on its inner surface. The results of this test are again indicative of the more severe operating conditions imposed on load bearing systems required to operate in a vacuum environment. While an operating life of 5000 hours under these conditions certainly is significant, the fact remains that it is only 28% of that already achieved under the same operating speed, but at higher loads and with less lubricant available in the composite. (See following section.) Figures 8 and 9 are photographs of the front and rear bearings taken after 1000 hrs. and 5058 hrs., respectively.

2. <u>Carbon-Graphite Retainers - 30,000 rpm</u>

A total of fourteen functional tests were performed at 30,000 rpm on 204 and 207 size bearings equipped with double shrouded retainers fabricated from various carbon-graphite formulations. Pertinent design parameters of the M-2 tool steel, 207 size bearing are as follows:

Race Curvature	
Inner	0.57
Outer	0.535
Free Contact Angle	15°
Ball Diameter -	0.375 "
Internal Clearance -	0.0055 "
Pitch Diameter -	2.117 "
Ball Complement	9

TABLE 6

204 Functional Test Results - 10,600 rpm

Run No.	Load-lb. Thrust	Load-lb. Radial	Temp. °F	Cage Clearance - mils Inner Race	Cage Clearance - mils Ball	Cage Composition	Life Hr.
1	50	10	1200	1	3	Clevite 300[a]	9.5
2	50	10	1200	4	3	Clevite 300	13.5
3	50	10	1200	4	3	Clevite 300	18.5
4	50	10	1200	5	10	Clevite 300	9
5	50	10	1200	5	10	Clevite 300	130
6	50	25	1200	5	10	Clevite 300	50
18	50	50	900	18	10	ACF-10 Graphite[b]	12
19	50	50	1200	18	10	ACF-10 Graphite	2.5
20	50	25	1200	18	10	ACF-10 Graphite	4
27	50	25	900	20	30	Prefiltron 220107[c]	3 min
28	50	25	900	20	30	Prefiltron 12042	5 min
10*	10	5	150	10	10	58AgHg-34WTFE-10MoSe$_2$	5058

(a) Clevite Corporation

(b) Poco Graphite, Inc.

(c) Pfizer

* Test operated at 3,500 rpm in vacuum environment simulating 240,000 ft. altitude.

Fig. 8—Run 10 — 204 size bearings from 3/4 H. P. motor after 1000 hrs. at 3, 400 rpm and ~ 240, 000 ft. altitude

Fig. 9—Run 10 — 204 size bearings from 3/4 H. P. motor after 5058 hrs. at 3,400 rpm and ~ 240,000 ft. altitude

The primary purpose of this test series was to determine if low density, carbon-graphite retainer materials exhibited any potential with respect to achieving the goal of a 25 hour operating life in the 30,000 to 60,000 rpm speed range. The results of these tests are summarized in Table 7.

Runs 9 and 11 were performed on the 30,000 rpm test spindles in an effort to evaluate the performance of two different grades of carbon-graphite having unusually good mechanical properties. Both retainers were reinforced with heli-arc welded, double-shrouded, titanium shrouds. It will be noted that in both cases the starting torque was of a sufficient magnitude that the 0.7 HP drive motor could not attain operating speed. In both cases, high torque was encountered regardless of the thrust to radial load ratio employed. In addition, a reduction in operating speed to 21,500 rpm (Run #12) did not result in a corresponding reduction in starting torque. In view of the high torques experienced with these materials, it was decided to sacrifice some mechanical strength in the retainer and employ a more highly graphitized grade of carbon-graphite in an attempt to significantly reduce sliding friction coefficients, and thereby torque. Runs 15, 16, and 17 were therefore performed on 207 size bearings equipped with retainers fabricated from materials which exhibited lower frictional characteristics than those tested previously. Although all three tests reached speed within 5 seconds, those runs employing the CR-83* grade material failed due to extremely rough operation within 3 minutes. Run 15, operating under a 50 lb. thrust/25 lb. radial load, required shut-down in 3 minutes due to severe vibration. Increasing the thrust load to 75 lb. in Run 16 resulted in an even earlier shut-down (1.5 min.). Although no retainer fracture occurred, poor lubricant filming of the balls and races was evident. Run 17 utilized a CR-81* grade of carbon-graphite with the bearing carrying a reduced load of 25 lbs. thrust/ 25 lbs. radial and operated for 20 minutes prior to failure. Test shut-down was again caused by high torque and rough operation at that time, when the outer race temperature reached 275°F. Prior to this, bearing operation was quite smooth. No retainer fracture occurred.

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* Stackpole Carbon Company

TABLE 7

FUNCTIONAL TEST RESULTS - CARBON GRAPHITE RETAINERS
30,000 rpm - Room Temperature

Run No.	Bearing Size	Load - lb. Thrust	Load - lb. Radial	Internal Clearance-mil	Max. Outer Race Temp. °F	Life Hrs.	Retainer Fit - mils I.D.	Retainer Fit - mils Balls	Retainer Material
9[a]	207	75	25	5.5	-	(c)	20	15	P-5Ag Graphite[d]
		50	25		-	(c)			
		25	10		-	(c)			
11	207	50	25	5.5	245	(c)	20	15	P-55 Graphite[d]
12[b]	207	25	10	5.5	110	(c)	20	15	P-55 Graphite
15	207	50	25	5.5	145	3 min.	20	15	CR-83 Graphite[e]
16	207	75	25	5.5	105	1.5 min.	20	15	CR-83 Graphite
17	207	25	25	5.5	275	20 min.	20	15	CR-81 Graphite[e]
23	207	50	25	9.5	150	3 min.	20	15	CR-81 Graphite[e]
24	207	50	25	9.5	-	3 min.	25	15	ACF-10 Graphite[f]
25	207	50	25	9.5	-	2 min.	25	15	ACF-10 Graphite
33	204	50	25	5.5	340	10 min.	18	10	ACF-10 Graphite
34	204	50	25	5.5	275	8 min.	8	10	ACF-10 Graphite
35	204	50	25	5.5	400	7 min.	8	10	ACF-10 Graphite
36	204	50	25	5.5	230	23 min.	18	20	Grade 2060 Graphite[e]
37	204	50	25	5.5	270	1 hour	18	20	Grade 2066 Graphite[e]

[a] Same bearing system used under 3 different load conditions.
[b] Test operated at 21,500 rpm.
[c] High torque prevented drive motor from reaching speeds.
[d] Pure Carbon Company.
[e] Stackpole Carbon Company.
[f] Poco Graphite, Incorporated.

Runs 23, 24, and 25 were performed under a load of 50 lb. thrust/25 lb. radial to determine the effect of larger bearing internal clearances on operating life. The inner race grooves of the 207 size test bearings were ground to an extent sufficient to increase their radial clearance from 5.5 to 9.5 mils. In all tests, bearing operation was extremely rough from the moment of start-up, indicating that for this particular bearing design, a 9.5 mil clearance cannot be tolerated with no significant temperature differential across the inner and outer rings. Failure in all cases was caused by severe chipping of the carbon-graphite retainers in the ball pocket areas and high torque.

The final five runs listed in Table 7, runs 33 through 37, were performed on the 204 size, M-2 ball bearing system at a speed of 30,000 rpm and room temperature conditions. The tests were undertaken to evaluate (a) the high speed performance of a number of grades of carbon-graphite in the smaller size bearing system, and (b) the capability of the 204 system to carry the 50 lb. thrust/25 lb. radial load at 30,000 rpm. It will be noted that attempts to operate bearings equipped with retainers of ACF-10* grade graphite were unsuccessful, with failure caused by cage chipping and fracture. Runs 36 and 37, however, were performed with retainers fabricated from two grades of Stackpole carbon-graphite and provided the longest life thus far achieved with carbon-graphite retainer materials during this program at 30,000 rpm. A retainer fabricated from grade 2066 graphite provided a life of 1 hour before failure occurred due to pocket wear and ball/shroud contact. The maximum temperature reached during this period was 270°F. While it is true that a life of 1 hour at 30,000 rpm remains far short of the 25 hour program goal, this test does indicate that the smaller, 204 size bearing is capable of handling a 50 lb. thrust/25 lb. radial load under high speed conditions. Figure 10 is a post-test photograph of this bearing and clearly illustrates the degree of pocket wear occurring during 1 hour operation.

*Poco Graphite, Incorporated

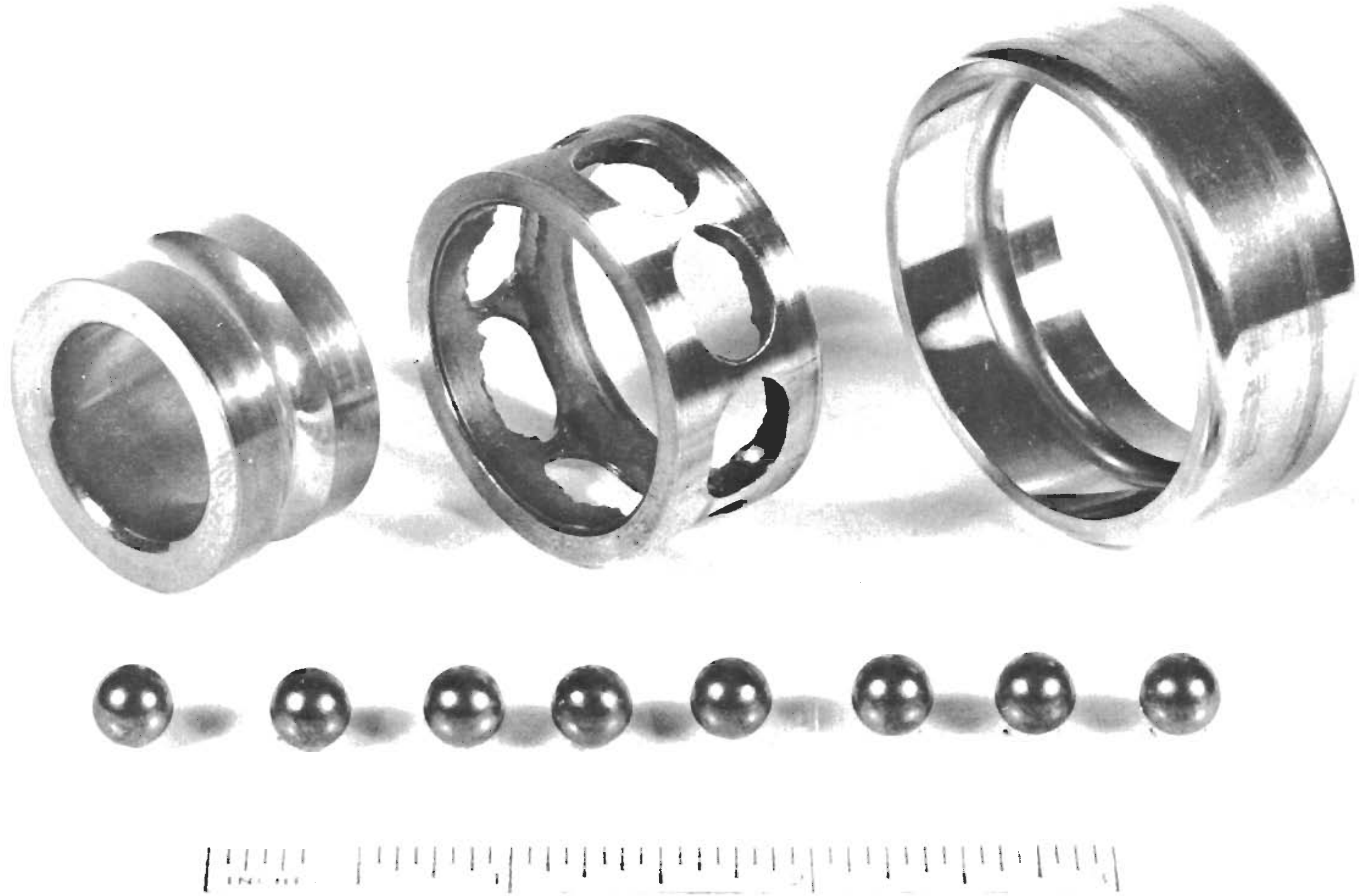

Fig. 10—Run 37 — 204 size; 1 hour at 30,000 rpm, 50 lbs. thrust/25 lbs. radial, room temperature

3. Metal Matrix Retainers - 30,000 rpm

A total of sixteen functional tests was performed at 30,000 rpm utilizing self-lubricating retainers fabricated from metal matrix composites containing various lubricant fillers. In addition, a life test (Run 173) was continued from the prior program and operated throughout the first year of the current program. The room temperature test is being performed on a 207 size ball bearing operating at 3,500 rpm and equipped with a retainer fabricated from the 70AgHg-20WTFE-10MoSe$_2$ (vol. %) composite. The bearing is carrying a load of 75 lb. thrust/25 lb. radial and thus far has accumulated 18,500 hrs. of operation. Figure 11 is a photograph illustrating the condition of the bearing after 8,706 hrs. and 17,410 hrs. of operation. It is apparent from this photograph that both the retainer as well as the bearing components have suffered negligible wear during this period of operation.

Table 8 summarizes the results of these functional tests. The first five experiments were performed on the 207 bearing system equipped with the standard silver-mercury composite (70AgHg-20WTFE-10MoSe$_2$; vol. %). All tests were performed at 30,000 rpm. In Run 8 the bearing carried a load of 75 lb. thrust/25 lb. radial and operated for a period of 1 hour. At this point, a surge in operating temperature indicated contact between one of the bearing components and the retainer shroud. As a result, the test was terminated. Inspection revealed that the retainer's ball pockets had suffered only minor wear. The inside surface of the retainer (guiding surface), however, showed preferential wear over an area of approximately 30°. This localized wear of the retainer's inside diameter eventually allowed contact between the retainer shroud and the outer ring of the bearing. The problem appeared to be identical to that experienced in previous work on the 204 system at 21,500 rpm. Run 13 was performed under a slightly reduced load of 50 lb. thrust/25 lb. radial but at an ambient temperature of 300°F rather than the room temperature condition of the previous test. Within 15 minutes, bearing temperature exceeded 600°F as evidenced by excessive melting of the WTFE. Test failure occurred due to retainer seizure on the inner race of the 207 size bearing. Although no fractures occurred, retainer wear was excessive. Figure 12

Fig. 11—Run 173 — 207 size; 75 lb. thrust/25 lb. radial,
3,500 rpm - room temperature
Top - after 8,700 hrs. operation
Bottom - after 17,410 hrs. operation

TABLE 8

FUNCTIONAL TEST RESULTS - METAL MATRIX RETAINERS
30,000 rpm - Room Temperature

Run No.	Bearing Size	Load-lb. Thrust	Load-lb. Radial	Internal Clearance-mils	Max Outer Race Temp °F	Life Hrs.	Retainer Fit-mils I.D.	Retainer Fit-mils Balls	Retainer Material
173	207	75	25	5.5	90	18,500*	20	15	70AgHg-20WTFE-10$MoSe_2$
8	207	75	25	5.5	310	1	20	15	70AgHg-20WTFE-10$MoSe_2$
13	207	50	25	5.5	440	15 min	20	15	70AgHg-20WTFE-10$MoSe_2$
14	207	50	25	5.5	350	25 min	20	15	70AgHg-20WTFE-10$MoSe_2$
21	207	50	25	5.5	390	15 min	20	15	70AgHg-20WTFE-10$MoSe_2$
22	207	50	25	9.5	375	12 min	20	15	70AgHg-20WTFE-10$MoSe_2$
30	207	50	25	5.5	360	10 min	20	15	80WSe_2-20GaIn
31	207	50	25	7.5	380	30 min	20	15	80WSe_2-20GaIn
32	207	50	25	7.5	500	32 min	20	15	80WSe_2-20GaIn
38	207	50	25	5.5	410	45 min	10	15	80WSe_2-20GaIn
39 (a)	207	50	25	7.5	460	22 min	20	15	80WSe_2-20GaIn
40 (a)	207	50	25	7.5	350	10 min	20	15	80WSe_2-20GaIn
44 (b)(c)	207	70	4	7.5	185	3 hr/15 min	20	15	58AgHg-34WTFE-8$MoSe_2$
45 (b)(c)	207	70	4	7.5	205	3 hr/15 min	20	15	58AgHg-34WTFE-8$MoSe_2$
41 (b)	205	100	4	5.0	250	15 hr/10 min	5	10	58AgHg-34WTFE-8$MoSe_2$
42	205	50	25	7.0	235	2 hr/10 min	5	10	58AgHg-34WTFE-8$MoSe_2$
43 (b)	205	100	4	5.0	175	2 hr/10 min	5	10	80WSe_2-20GaIn

* Test remains in operation at this time. Speed - 3,500 rpm.
(a) Insert type retainer.
(b) Bearing used as spindle bearing in 30,000 rpm test stand.
(c) Bearings have not as yet failed.

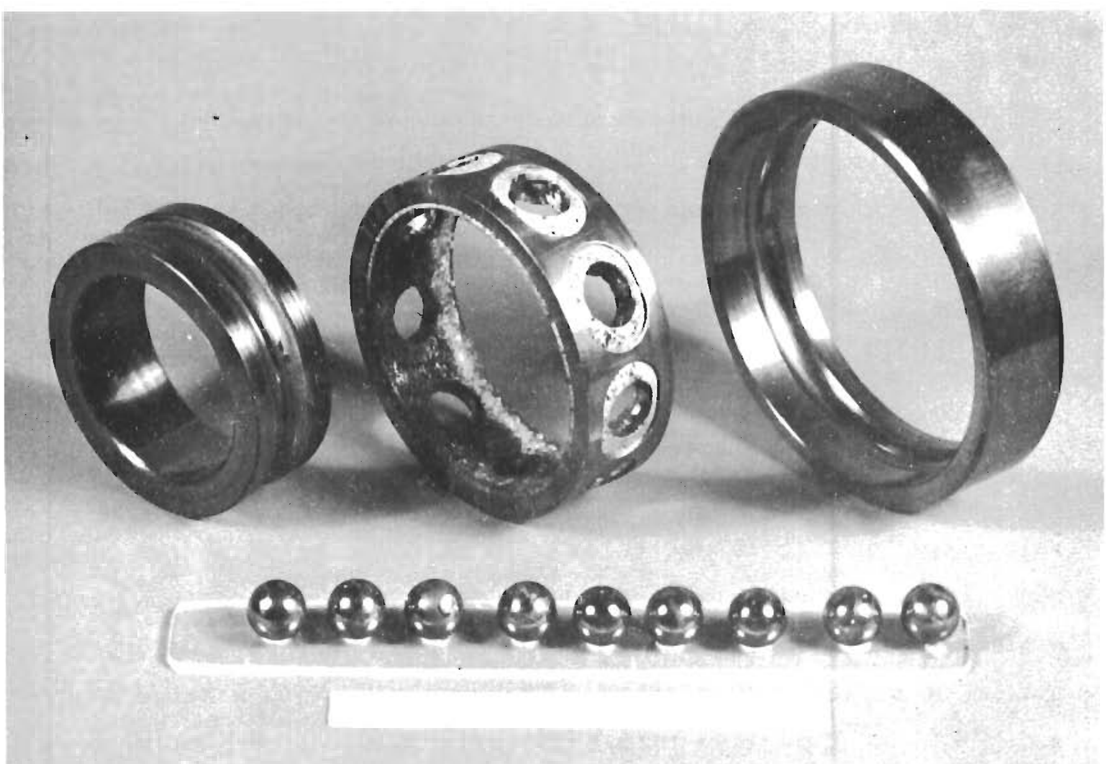

Fig. 12—Run 13 — 207 size; 15 min. at 30,000 rpm, 50 lbs. thrust/25 lbs. radial, 300°F

contains a photograph of the dismantled bearing after test as well as a close-up view of the retainer which shows clearly the tracks of WTFE left on the shroud as the PTFE material was thrown from the retainer.

Run 14 was started at room temperature under a load of 50 lbs. thrust/25 lbs. radial. The bearing operated for a period of 25 minutes before excessive torque resulted in test shut-down. At test termination, the outer race of the bearing had reached a temperature of 350°F, with no external heating applied. Upon dismantling the test bearing, it was noted that the inner race steel of the bearing, as well as the balls, were quite discolored, indicating that they had reached a temperature considerably higher than the 350°F monitored on the bearing's outer race during the test. In addition, excessive melting of the WTFE present in the retainer had occurred, as shown in Figure 13, and indicates that the temperature of this component had surpassed 600°F. The evidence obtained from this test suggested strongly that a temperature differential of at least 300°F was generated between the inner and outer races of the 207 bearing system as it operated at 30,000 rpm with the AgHg lubrication system. It is pointed out that high torque - and not retainer fracture or shroud rub-caused failure. The evidence indicated that a steadily increasing temperature differential between the inner and outer races during the test eventually resulted in a complete loss of internal bearing clearance. This, in turn, caused high ball loading as the inner race temperature continued climbing, greater frictional heating, increased torque, and subsequent test failure.

In an attempt to verify this theory, Run 21 was operated under the same conditions as those employed in the previous test, but on a bearing whose inner race had been treated with various temperature sensitive paints designed to change color upon reaching temperatures of 650, 750, and 850°F. The purpose of the test was to obtain, if possible, a better estimate of the temperature reached by the inner race of the bearing during a test of at least 20 minutes. The experiment was started at room temperature under a load of 50 lbs. thrust/25 lbs. radial. The bearing operated for a period of 15 minutes before excessive torque resulted in test shut-down. During this period, outer race temperature reached 390° and was still

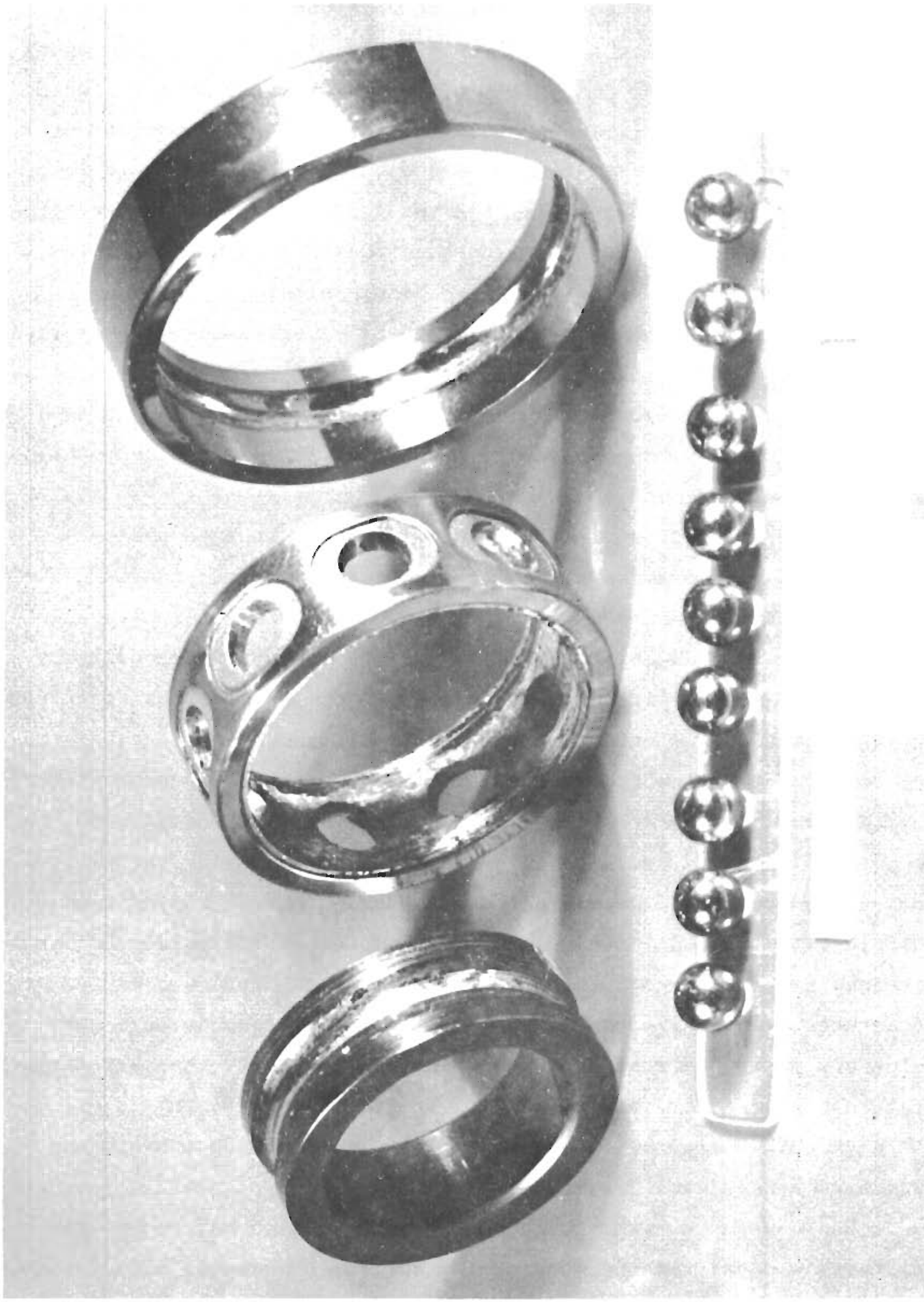

Fig. 13—Run 14 — 207 size; 25 min. at 30,000 rpm 50 lbs. thrust/25 lbs. radial, room temperature

climbing at test termination. No external heating was applied during the run. Upon dismantling the test bearing, it was found that all temperature sensitive paints had changed color, indicating that the bearing's inner race temperature had exceeded 850°F. In view of the fact that the outer race reached only 390°F at test termination, a temperature gradient of at least 460°F existed between the inner and outer bearing rings. This finding supports the hypothesis proposed earlier that an extreme temperature differential between the inner and outer bearing races is causing ball loading, high torque, and eventual bearing seizure.

The final run in this first series of experiments was performed on a 207 bearing whose internal clearance had been increased to 9.5 mils. As was the case when a carbon-graphite retainer was employed, bearing operation was extremely rough upon start-up, with failure caused by high torque after 12 minutes of operation.

The second series of experiments listed in Table 8 was performed at 30,000 rpm on 207 size bearings equipped with retainers fabricated from the $80WSe_2$-20GaIn composite. The tests were operated under a 50 lb. thrust/25 lb. radial load at room temperature conditions. In Run 30, a life of 10 minutes was obtained before test failure occurred due to high torque. The outer race of the bearing had reached a temperature of 360°F at the time of failure. In an attempt to reduce the temperature differential between the inner and outer rings, and in so doing reduce the ball preload and high torque caused by this differential, the internal clearance of the bearings used in Runs 31 and 32 was increased from 5.5 to 7.5 mils. In both instances a life of approximately 30 minutes was obtained prior to test failure. Failure in these runs was caused by cage fracture in the ballpocket areas. Operation of the bearings was significantly improved throughout the runs, however, with the operating temperature of the outer race at time of failure slowly rising to approximately 380°F. Figures 14 and 15 are post-test photographs of the bearings from Runs 31 and 32, respectively.

Runs 39 and 40 were performed on the 207 size bearing system equipped with a metal retainer containing WSe_2/GaIn inserts. Its design is shown in Figure 16. The retainer itself is fabricated from titanium

Fig. 14—Run 31 — 207 size; 30 min. at 30,000 rpm, 50 lbs. thrust/25 lbs. radial, room temperature

Fig. 15—Run 32 – 207 size; 32 min. at 30,000 rpm, 50 lbs. thrust/25 lbs. radial, room temperature

Dwg. 2943A76

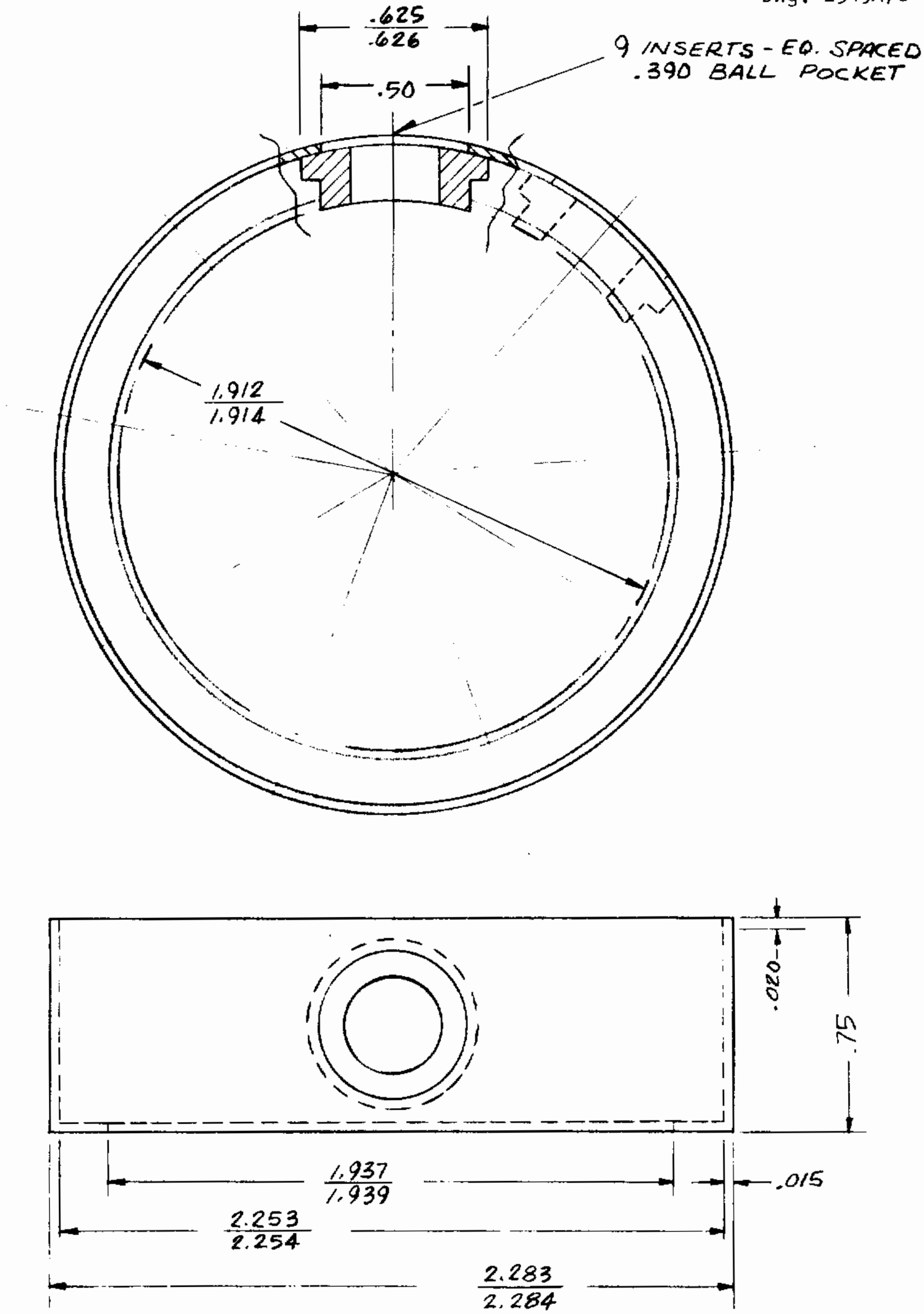

Figure 16 — Size 207 insert type retainer.

and is equipped with 9 oversize ball pockets into which are pressed inserts of WSe_2/GaIn. A thin titanium shroud is then placed over the outside diameter of the retainer and welded in place, thus locking in the self-lubricating inserts. The bearings were operated at 30,000 rpm under a 50 lb. thrust/25 lb. radial load and room temperature conditions. Failure in both cases was caused by insert fracture in one ball pocket. No significant improvement in operation was observed due to the reduced weight of the insert-equipped titanium retainer. Outer race temperature at the time of failure had reached 400°F.

Run 41 is by far the most significant functional test performed during this reporting period. The test was performed at 30,000 rpm on a 205 size bearing equipped with a double shrouded retainer fabricated from 58AgHg-34WTFE-8$MoSe_2$. The bearing operated for a period of 15 hours before test failure occurred due to ball-shroud contact. The life obtained on this bearing is 20 times greater than any run performed previously, and falls only 10 hours short of the 25 hour program goal. It should be remembered, however, that the bearing was operating at 30,000 rpm rather than the 60,000 rpm objective. In the writer's opinion, the primary reason for this significant life extension is the fact that the 205 bearing was mounted as one of the spindle bearings in the 30,000 rpm test facility rather than at the normal test bearing location. It is believed that the elimination of the overhung load on the test bearing was a major factor in the improved performance. Bearing operation was quite smooth throughout the experiment, with the outer race temperature holding steadily at ∿200°F. No melting of WTFE occurred, indicating that the inner race temperature was well below 600°F. The bearing was loaded with a 100 lb. pre-load (spring) and carried only that radial load caused by mechanical imbalance of the spindle. Retainer fits with the inner race and balls were 5 and 10 mils, respectively. These tighter fits may also have contributed to the bearing's improved performance characteristics. Figure 17 is a post-test photograph of the test bearing.

Run 42 was performed on an identical 205 bearing system but with the bearing located in the normal, over-hung load condition rather

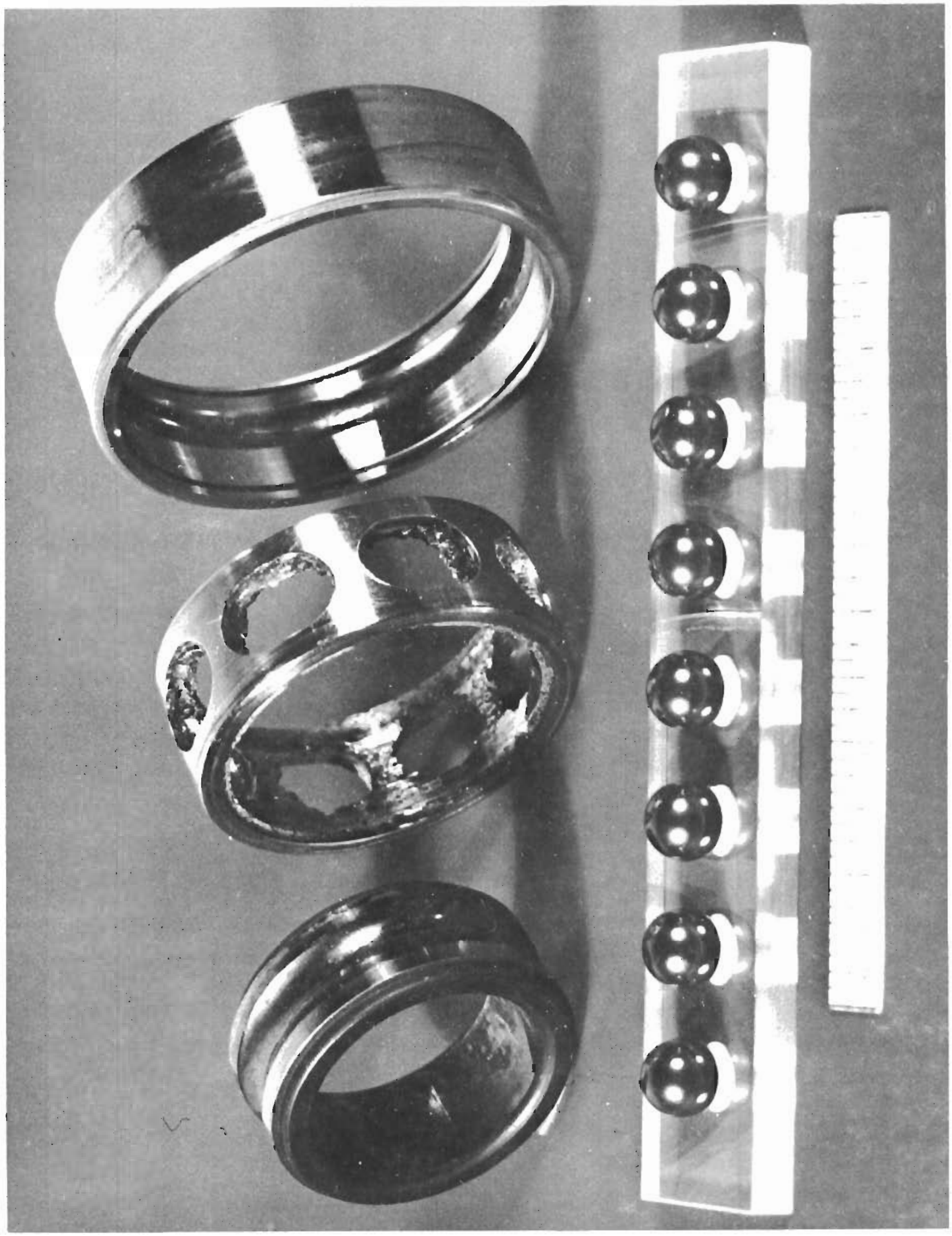

Fig. 17—Run 41 – 205 size; 15 hrs. – 10 min. at 30, 000 rpm, 100 lbs. thrust/4 lbs. radial, room temperature

than in the spindle bearing position employed in Run 41. It will be noted that life was reduced by a factor of approximately 7 by exposing the bearing to the overhung load conditions. While it is true that the bearing carried a radial load 20 lbs. greater than that of Run 41, the writer does not believe its significance would warrant a reduction in life of the magnitude observed.

In Run 43, a 205 size bearing equipped with a WSe_2/GaIn retainer operated for a period of 2 hrs. - 20 min. at 30,000 rpm prior to ball/shroud contact. The bearing was operated in the spindle bearing position. When operated in the overhung load position on a 207 ball bearing, this retainer material provided a maximum life of 40 minutes before cage fracture and ball/shroud contact. Figure 18 is a photograph of the bearing taken after the test was completed.

Runs 44 and 45 were performed simultaneously on 207 ball bearings employed as both the front and rear spindle bearings of the 30,000 rpm test stand. The bearings were equipped with retainers fabricated from the AgHg composite containing 34% (vol.) WTFE. The use of the larger 207 bearings as spindle bearings on the 30,000 rpm stand caused a higher than normal electrical load to be placed on the drive motor. As a result, motor winding failure occurred after 3 hrs. - 15 minutes of operation. Inspection revealed that both bearings remained in excellent condition. Maximum life obtained on an identical bearing system in the overhung load mode was approximately 30 minutes. Shaft modifications underway at the present time will allow the use of a grease lubricated 205 bearing in the rear spindle position while maintaining the 207 self-lubricated bearing in the front location. It is hoped that this change will reduce motor load to normal operating values and permit continuation of the tests.

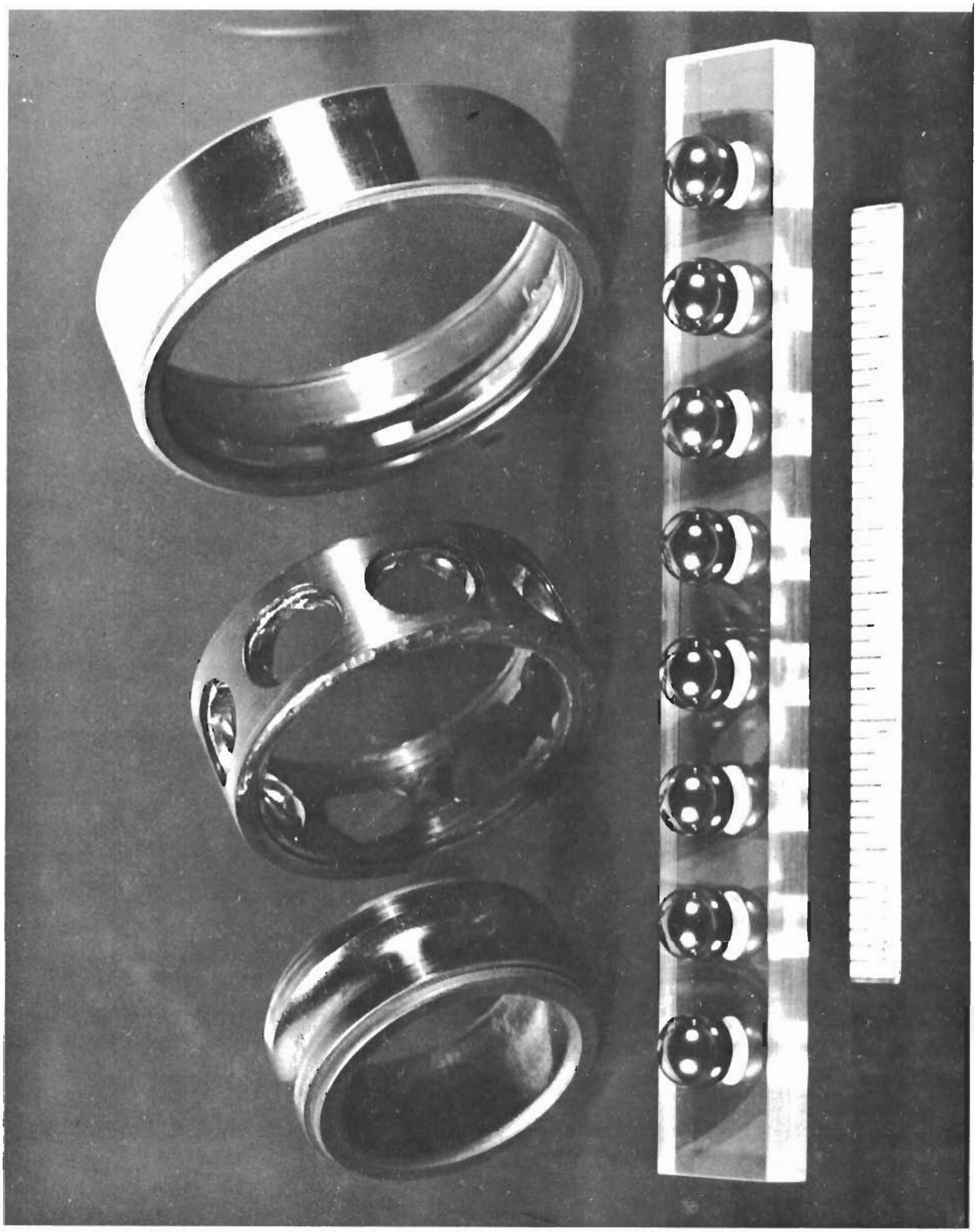

Fig. 18—Run 43 – 205 size; 2 hrs. – 10 min. at 30,000 rpm, 100 lbs. thrust/4 lbs. radial, room temperature

VI. SUMMARY

The following summarizes the results of work performed during the first year of this contract.

1. Attempts were unsuccessful to improve upon the 140 hours life obtained during the previous program[2] at 1200°F, 10,600 rpm and a 50 lb. thrust/10 lb. radial load. No life improvement was achieved by either retainer design modifications or the utilization of a number of new, self-lubricating retainer materials.

2. A high speed ball bearing tester has been designed that is capable of operating 207 size bearings over a 30,000 to 60,000 rpm speed range. The device utilizes a 40 HP, variable speed motor driving a 15:1 ratio high speed transmission. The output shaft of the transmission is coupled to a cartridge containing the two test bearings.

3. A maximum life of 15 hrs. - 10 min. has thus far been achieved on a 205 size bearing operating at 30,000 rpm and 200°F. The bearing carried a 100 lb. thrust/4 lb. radial load and was operated as a slave bearing in the 30,000 rpm test spindle. It was equipped with a 58AgHg-34WTFE-8$MoSe_2$ (vol. %) composite retainer.

4. A life greater than 3 hrs. - 15 min. is possible on a 207 size bearing operating at 30,000 rpm and 200°F. The bearing was again equipped with the AgHg composite retainer and operated as a slave bearing in the test spindle under a load of 70 lbs. thrust/4 lbs. radial.

5. Subjecting a solid lubricated test bearing to an overhung radial load condition at 30,000 rpm results in a reduction in operating life by a factor of approximately seven.

6. Under 30,000 rpm operating conditions, retainers from silver matrix composites provide significantly longer life than those prepared from carbon-graphite or WSe_2/GaIn composites.

7. Ball bearings of the 207 size equipped with silver matrix composite retainers have provided a life in excess of 18,500 hours continuous operation without failure. The bearing carries a load of 75 lbs. thrust/25 lbs. radial and is operating at 3,500 rpm. and room temperature.

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13. ABSTRACT

This report describes the first year's effort of a two year program that has as its objective the development of improved self-contained, solid lubricated ball bearings capable of at least 25 hours operation at speeds approaching 60,000 rpm. The program encompasses a temperature range of -40°F to 600°F and a maximum load combination of 75 lb thrust/25 lb radial. The work is a follow-on effort to the completed Air Force funded program F33615-68-C-1250.

During this reporting period, a high speed test stand was designed and manufactured that is capable of functionally evaluating the performance of solid lubricated, 207 ball bearings over a 30,000 to 60,000 rpm speed range. The device incorporates a 40 H.P., variable speed motor driving a transmission having a 15:1 gear ratio. A test head containing two test bearings is coupled to the output shaft of the transmission.

The functional test portion of the program has thus far achieved a maximum life of 15 hours on a solid lubricated ball bearing operating at 30,000 rpm and a temperature of 200°F.

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Ball Bearings						