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EMERGENCY AND MICROFOG LUBRICATION AND COOLING OF BEARINGS FOR ARMY HELICOPTERS

BY

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16. Abstract

An analysis and system study was performed to provide design information regarding lubricant and coolant flow rates and flow paths for effective utilization of the lubricant and coolant in a once-through oil-mist (microfog) and coolant air system. A system was designed, manufactured, coupled with an existing rig and evaluation tests were performed using 46mm bore split-inner ring angular-contact ball bearings under 1779N (400 lb.) thrust load. An emergency lubrication aspirator system was also manufactured and tested under lost lubricant conditions.

A total of fourteen step-speed tests and two extended period tests were performed with the mist and cooling air system. Bearing speeds as high as 3×10^6DN were obtained in the step-speed tests. No problems were encountered except at speeds above 2.5×10^6DN where cage instability and excessive cage to land wear were encountered in several tests. Successful operation was obtained with an oil flow rate as low as 51 cc/hr (3.1 in^3/hr). In another test a total air flow of only 0.283 scmm (10 scfm) supplied at a temperature of 359°K (185°F) was found adequate to maintain the bearing temperature below 505°K (450°F).

Extended period tests of 50 and 100 hours were performed at 2.5 and 2.0×10^6 DN respectively with good bearing performance obtained.

The testing demonstrated the feasibility of using a mist oil and cooling air system to lubricate and cool a high speed helicopter engine mainshaft bearing. The testing also demonstrated the feasibility of using an emergency aspirator lubrication system as a viable survivability concept for helicopter mainshaft engine bearing for periods as long as 30 minutes.

17. Key Words (Suggested by Author(s))

Bearing Mist Lubrication; Bearing Microfog Lubrication; Helicopter Engine Bearing Lubrication; High Speed Bearing Lubrication; Mist Lubrication; Microfog Lubrication; Bearing Lubrication

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FOREWORD

The research described herein, conducted by the SKF Industries, Inc. Technology Services Division, was performed under NASA Contract NAS3-17343. The work was completed under the management of the NASA Project Manager, Mr. William R. Loomis, Fluid System Components Division, NASA Lewis Research Center.

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SUMMARY

The research performed on this program had the following two objectives:

1) Demonstrate the feasibility of using an oil aspirator as an emergency lubrication device for a helicopter engine mainshaft ball bearing (approximately 46mm bore) operating at speeds and load corresponding to present engine mainshaft conditions.

2) Design and manufacture a full time oil mist lubricating system and air cooling system and demonstrate their feasibility for lubricating and air cooling of a helicopter engine mainshaft bearing operating at high speeds (DN of 1.75×10^6 and higher).

The feasibility of an emergency aspirator lubrication system was demonstrated as a viable survivability concept for helicopter engine mainshaft bearings for periods as long as 30 minutes after cessation of the recirculating oil supply. Without the emergency aspirator system, test bearings failed within 30 seconds after cessation of the recirculating oil. With the aspirator, the bearings operated for 2.5 minutes on a 10cc (0.61 in^3) reservoir of oil in the oil manifold. Extended operation to 30 minutes was achieved by refilling the reservoir every 2.5 minutes. Therefore, the duration of the emergency aspirator lubrication was limited only by the size of the reserve oil reservoir, at least within the 30 minute time boundary. A type II ester oil (MIL-L-23699) was used at an inlet temperature of 438°K (330°F) with room temperature air supplied to the aspirator at a flow rate of 0.024 scmm (0.84 scfm). The bearing speed was 38,000 rpm (1.75×10^6 DN) with a thrust load of 1779 newtons (400 lbs.) applied.

Analytical and experimental studies were performed to determine the lubricant and coolant flow rates and flow paths needed in an oil-mist, air cooled lubrication system. A system was designed and manufactured and coupled with an existing SKF owned test rig to perform evaluation testing. The bearings were aircraft quality 45.923 mm bore split-inner ring angular-contact ball bearings.

The oil mist system was designed to supply air supported oil particles through eight converging reclassifying nozzles to one side of the test bearing. The oil mist was directed by the nozzles to the gap between the cage bore and inner-ring land and on to an enlarged chamfer on the inner-ring face from which it was centrifugally pumped into the bearing. The cooling air system was designed to supply three flow paths; directly through the bearing from eight straight nozzles, circumferentially around the bearing housing, and through the hollow shaft.

A total of fourteen step-speed tests and two extended period tests were performed with the mist lubrication applied to the test bearing. Three mist tests, one each, were performed with a domestic refined Type I ester (MIL-L-7808H), a specially formulated polyphenylether, and a foreign refined Type I ester (MIL-L-7808, NATO-O-148). The remaining tests were performed with a Type II ester (MIL-L-23699).

The results of the step-speed tests proved the feasibility of using a once-through oil-mist and cooling-air system to lubricate and cool current high speed helicopter engine mainshaft bearings. The extended period tests (50 hrs. at 55,000 rpm and 100 hrs. at 43,600 to 45,000 rpm) demonstrated the feasibility of long term bearing operation with mist lubrication. All tests were performed with cooling air supplied through the bearing only. Housing cooling air and shaft cooling air were not required.

A majority of the tests were performed with a mist oil flow rate of 442 to 492 cc/hr (27 to 30 in^3/hr) and at total mist and cooling air flow rate of 0.439 to 0.586 scmm (15 to 20 scfm). However, in one step-speed test an oil flow rate as low as 51 cc/hr (3.1 in^3/hr) was shown to be adequate. In a different step-speed test a total mist and cooling air flow rate as low as 0.283 scmm (10 scfm) supplied at a temperature of 359°K (185°F) was found adequate to maintain the bearing temperature below 505°K (450°F). In another step-speed test an appreciably simplified mist system (drip/mist system) where oil drops were fed directly into the cooling air flow stream within the rig demonstrated the feasibility of operating with a significantly reduced vulnerability profile and mechanical complexity of the mist system.

No problems were encountered in any of the mist lubrication tests up to speeds of approximately 55,000 rpm (2.5×10^6 DN). At speeds between 55,000 and 65,000 rpm (3×10^6 DN) two problems were frequently encountered; 1) cage instability and 2) excessive wear and drag between the outer ring guide land and cage rail on the downstream side of the bearing. Changes in cage design were only partially successful in eliminating these problems.

1.0 INTRODUCTION

1.1 Objective

The high speed mainshaft bearings in helicopter gas turbine engines and the input pinion bearings in helicopter transmissions are normally lubricated by recirculating oil, supplied through pressurized jets, impinging on the sides of the bearings or oil pumped through holes in the bearing rings. The oil performs two functions; lubrication of the bearing elements contacting surfaces and removal of the bearing generated heat. Previous studies conducted at the SKF Technology Center (1)* established that the major portion of the lubricating oil supplied to high speed, highly loaded bearings is required for the removal of the bearing generated heat, while only a small portion is required to provide the desired elastohydrodynamic (EHD) lubrication. The bearing generated heat of which a large portion results at high operating speed from the churning of the oil supplied to cool the bearing, represents a power loss to the engine.

The circulating oil systems required to lubricate and cool the critical high speed helicopter engine and transmission bearing are highly vulnerable to ballistic damage which can result in the cessation of oil flow. The loss of oil to the bearings precipitates a bearing seizure, generally within 1 to 3 minutes, which can result in a forced landing or the crash of the aircraft. Based on the vulnerability of the recirculating lubrication system, the U. S. Army now specifies that the next generation of helicopter engines and transmissions shall be designed to operate for 30 minutes following oil cessation to permit emergency landing of the helicopter away from enemy occupied territory.

There are several approaches which can be employed which offer possibilities of providing feasible backup or emergency bearing lubrication and cooling systems to obtain the desired emergency operating life. These systems would include:

1. A redundant recirculating oil system.
2. An aspirated oil system utilizing residual oil retained in the recirculating oil manifold.
3. A once through oil mist and cooling air system.

Redundant systems are frequently employed in aircraft electronics to improve reliability and provide an emergency operating mode. In these cases, the penalty of increased weight and complexity is generally insignificant compared to the advantages gained.

*Numbers in parentheses refer to List of References at the end of this report.

However, a redundant oil system would add an appreciable amount of weight and complexity and add only marginally to the emergency operating mode since the redundant system would be as vulnerable to ballistic damage as the primary system and may be inoperative when needed.

An aspirated oil lubrication system could be devised which would be continuously operative, simple in construction, minimally vulnerable to ballistic damage, and carry a very small weight and complexity penalty. The effectiveness of such a system depends on the oil supply rate and the quantity of compressor bleed air required to aspirate the oil and provide some cooling under emergency operating conditions.

The use of an emergency oil mist and cooling air system, which is basically an extension of an aspirator system that supplies a greater quantity of oil and cooling air, could be incorporated which would automatically be actuated by the loss of recirculating oil pressure. Again the higher reliability expected from such a system relative to that of an aspirator system would carry a weight and complexity penalty. However, the effect on engine efficiency due to bleed off of compressor air would be considered insignificant under emergency conditions and possibly offset by deactivating the recirculating oil system.

A once through mist oil and cooling air system also represents a possible replacement of the primary recirculating oil system and could provide the following advantages:

1. Reduced vulnerability to ballistic damage since large volume pressurized oil tanks, an air-oil heat exchanger, and oil supply and scavenge pumps would not be necessary.

2. Reduced bearing heat generation rate at high operating speeds from oil churning.

3. Higher temperature capability, since the lubricant is discarded after use and thermal degradation is of less concern.

4. Reduced weight and system complexity, since large capacity supply and scavage pumps, and heat exchangers may not be required.

It is noted that these advantages could possibly offset the reduction in engine efficiency from the use of bleed-off air from the compressor or some other source to supply the mist and cooling air requirements.

The purpose of the research performed on this program was directed at demonstrating the feasibility and practicability of using an aspirator type misting device as an emergency lubrication method for high speed helicopter engine and transmission bearings and to demonstrate the feasibility of lubricating and cooling high speed bearing (DN* of 1.75 million and higher) with a once through oil mist (microfog) and cooling air system using lubricants presently specified for military aircraft engines and transmissions. The program consisted of two major tasks.

Task I - Design, manufacture, and test an oil aspirator as an emergency lubrication device for a full-scale aircraft quality engine ball bearing (approximately 46mm bore) operating at speeds and loads corresponding to present engine mainshaft conditions.

Task II - Design and manufacture a once through oil mist and cooling air application system and demonstrate the feasibility of lubricating a high speed (DN of 1.75×10^6 and higher)*, approximately 46mm bore, aircraft quality engine bearing as modified for use with the lubrication system.

1.2 Background

The use of mist-oil lubrication for bearings is not new, the principles were first developed by a European bearing manufacturer in the late 1930's. The problem that nurtured this development was the inability to satisfactorily lubricate high-speed spindle bearings on grinders and similar equipment. Using the principles developed at that time, the generation of microfog oil, its delivery and insertion to lubricate bearings was further developed by various companies and used in a broad range of industrial applications. These applications, however, are less severe (lower DN values, loads and temperatures) than those presently envisioned for aircraft engine bearings of the near future.

Early studies and evaluation testing of mist lubrication of aircraft turbine engine ball bearings was performed by SKF(2). Several different lubricants were evaluated in tests up to 1.75×10^6 DN. These tests showed a definite potential; however, these studies were basically conducted with bearings and lubricant supply components which were designed for recirculating oil systems. In particular, the bearings and oil-mist, cooling air

*The DN value is the product of the bearing bore in mm and the shaft speed in RPM.

supply configuration was not optomized for delivery to bearing areas where the lubricant is required for effective utilization. Thus, initial attempts to operate large mainshaft size angular-contact ball bearings with mist lubrication under advanced turbine powerplant speeds, loads, and temperatures were unsuccessful with most candidate lubricants and only partially successful with one oil (Mobil XRM-177F) at 1.75×10^6 DN.

As the result of these tests, basic oil mist studies were conducted by Mobil Oil Company (3, 4) to determine oil-mist particle size distribution, mist reclassification nozzle operation and wetting efficiencies, heat transfer coefficients through wetted films, and a variety of related basic phenomena underlying oil-mist lubrication technology.

A synopsis of the most important results from the Mobil studies is as follows:

1. Larger mist particle size (11 microns) gave greater wetting rates when the oil mist was impinged on flat rotating disks. Previous reference data had indicated optimum particle size range of 2 to 3 microns for most effective wetting.

2. Centrifugal forces enhance wetting. Wetting rates increased with increasing temperatures and speeds of a rotating disk.

3. Convective heat transfer to the mist is primarily a function of the gas phase. Oil in the mist had little effect on the heat transfer coefficient (e.g.,4 percent increase). Heat transfer coefficients were established to be approximately 340 W/m^2-°F).

4. The use of a 150-mesh screen inserted into the inlet of the mist nozzle increased the wetting rate of the mist by 80 percent.

5. The synthetic paraffinic lubricant was found to have the best wetting characteristic of five fluids studied.

6. Rates of oil output were found to increase with decreasing kinematic viscosity of the oils and with increasing gas flow rate.

Based on these results and the potential advantage of oil-mist lubrication, further studies were performed at SKF (4). The approach used to evaluate mist lubrication of helicopter engine mainshaft bearings and the results obtained are the subject of this report.

2.0 MATERIALS TESTED

2.1 Test Bearings

The test bearing design selected for this program is a split-inner-ring, angular-contact ball bearing; the type most widely used in aircraft propulsion turbines. This design, SKF 464539VAA, which permits a maximum ball complement by virtue of the separable inner-ring halves, can support high thrust loads in either direction. The separable ring feature also permits the use of a precision-machined one-piece cage which is required for high-speed, high-temperature operation. The cage is outer-ring piloted with a nominal diametral clearance of 0.33mm (0.013 in.).

The test bearings have a bore diameter of 45.923 mm and a nominal unmounted design contact angle of 32° 30". The basic bearing is illustrated in Figure 1. The ring and balls are manufactured from consumable electrode vacuum melted (CVM) M50 tool steel to provide high temperature hardness. The cage is silver plated AMS 6414 steel according to the latest jet-engine bearing practice to minimize the friction and wear at the ball and ring land contacts.

The basic bearing design was modified for the major portion of the mist test by incorporating a 45 degree chamfer (1.8 x 1.8 mm) between the inner-ring face and land surface on the side opposite the puller groove. This chamfered surface was provided to pump plated out mist oil particles by centrifugal action into the bearing. In addition a taper of 3 degrees was machined on both sides of the cage bore which sloped radially outward toward the center line of the bearing to cause the oil plated on the bore surface to be centrifugally pumped into the bearing ball pockets.

In addition to the described mist cage design, four additional redesigned cages were tested to determine if improved bearing performance could be obtained at high speeds (2.5 to 3.0 x 10^6 DN). One of the modifications consisted of increasing the diametral clearance. The clearance of one rail was increased 0.05 mm (0.002in.) above the other rail which was machined to provide the maximum design tolerance dimension of 0.40 mm (0.016in.).

Two cages were specially designed to provide the pumping of lubricant and cooling air into the rail land interface. One cage, shown in cross section in Figure 2, incorporated a bore tapered radially outward from the center to an oil retaining lip beneath each rail. At the base of each

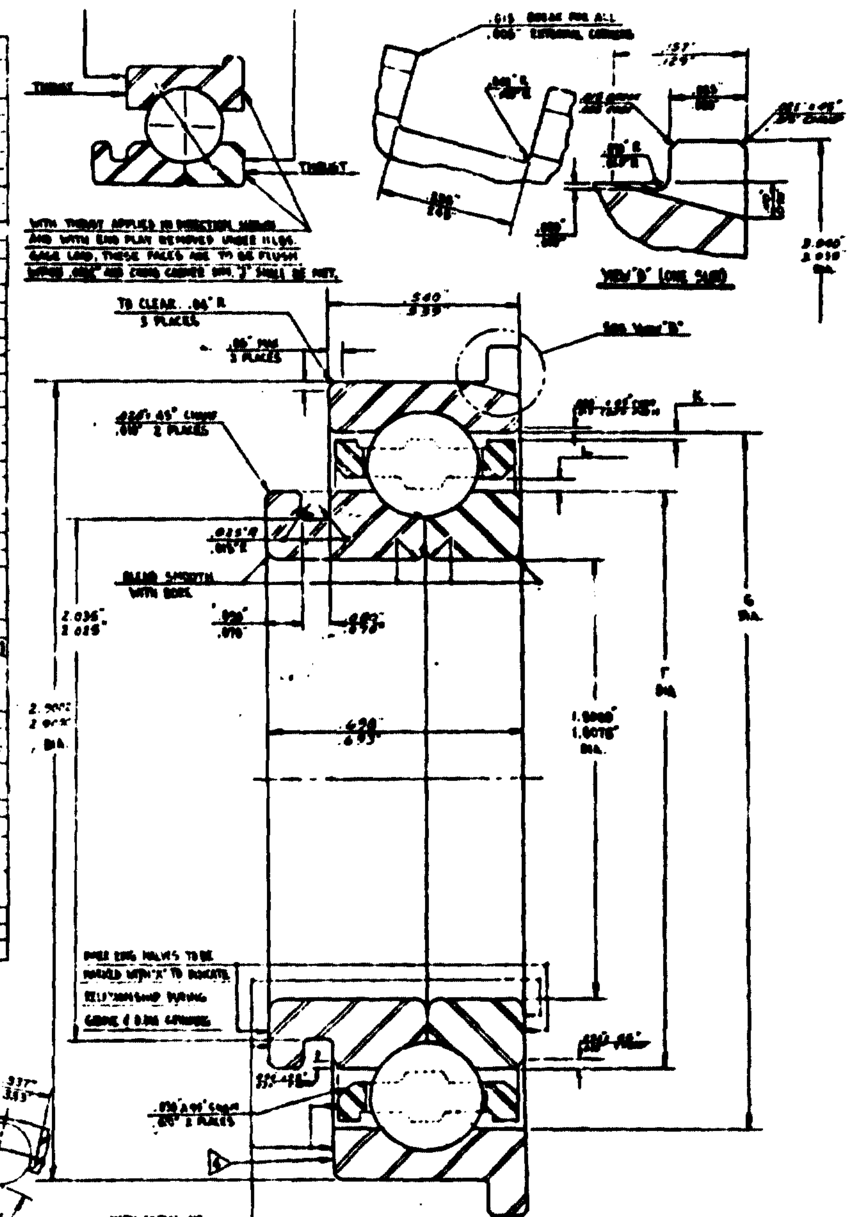
ORIGINAL PAGE IS OF POOR QUALITY

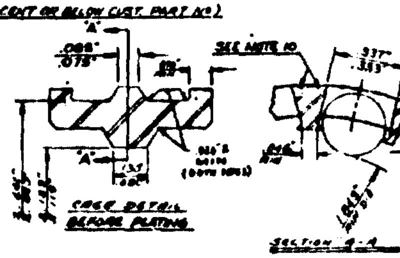
FIGURE 1

TEST BEARING

	Engineering Ref Data			
1	VENDOR PART IDENT NO			464537 VAA
2	VENDOR'S DRAWING NO			464537 VAA
3	BALL DIA (AFBMA GRADE 10)			.3125"
4	PITCH DIA			2.350"
5	RATIO · RADIUS OF INNER RING CURVATURE TO BALL DIA. MIN · MAX			.5296-.5325
6	RATIO · RADIUS OF OUTER RING CURVATURE TO BALL DIA. MIN · MAX			.5100-.5131
7	SPECIFIC DYNAMIC CAPACITY · LB			3320#
8	STATIC CAPACITY · LB			2720#
9	WEIGHT · LB			.25#
10	FOR SPLIT RING DESIGN ONLY	A	SHIM ANGLE	18°3'
		B	SHIM THICKNESS	.006
	ENGINEERING REQUIREMENTS			
1	NUMBER OF BALLS			16
2	RADIAL INTERNAL CLEARANCE UNDER CAGE LOAD	A	CLEARANCE MIN. · MAX.	.0030"-.0034"
		B	GAGE LOAD · LB	NO LOAD
3	END PLAY UNDER GAGE LOAD	A	END PLAY · MAX	.009"
		B	GAGE LOAD · LB	11
4	DIAMETRAL CAGE CLEARANCE AT K MIN · MAX			.010"-.016"
5	DIAMETRAL CAGE CLEARANCE AT L MIN. · MAX			[illegible]
6	DIAMETRAL BALL TO POCKET CLEARANCE MIN. · MAX.			.016" .023"
7	BALL MAT'L			B50TF105 CL·A
8	RING MAT'L			B50TF105 CL·A
9	CAGE MAT'L			AMS 6414
10	RIVET MAT'L			—
11	CAGE PLATING ON FUNCTIONAL RUBBING CONTACT SURFACES OTHER SURFACE OPTIONAL	A	SPECIFICATION	AMS 2412
		B	THICKNESS MIN. · MAX	.001"-.002"
		C	CORNER BUILD UP MAX.	.003"
12	BEARING HEAT TREATMENT TO BE STABILIZED FOR OPERATION AT THE FOLLOWING TEMP · °F MIN.			-65° TO 800°
13	BALL & RING HARDNESS ROCKWELL C MIN MAX.			60-64
14	CAGE WIDTH MIN MAX			.495"-.501"
15	CAGE FACE TO BEARING FACE CLEARANCE · MIN · END PLAY REMOVED			.003" (PROTRUSION)
16	BALL POCKET LOCATION IN CAGE · DIA OF TRUE POSITION			.006"
17	CAGE FACES TO BE PARALLEL WITHIN (TOTAL)			.002"
18	CAGE O DIA. TO BE CONCENTRIC TO CAGE BORE WITHIN			.002"
19	OUTER RING SHOULDER DIA "G" MIN MAX			2.5494"-2.5504"
20	RUNOUT OF DIA "G" WHEN O D OF OUTER RING IS USED AS DATUM · F I R			.0005"
21	INNER RING SHOULDER DIA "F" MIN MAX			2.076"-2.086"
22	RUNOUT OF DIA "F" WHEN I D OF INNER RING IS USED AS DATUM · F I R			.001"
23	CONTACT ANGLE UNDER GAGE LOAD	A	ANGLE MIN MAX	30°18' 34°46'
		B	GAGE LOAD · LB	11
24	SURFACE ROUGHNESS	A	BALLS	2 RMS
		B	RACEWAYS	6 RMS
		C	LAND RIDING SHOULDER DIA "G"	12 RMS
25	FOR SPLIT RING DESIGN ONLY		RUNOUT OF SPLIT SURFACES WHEN I.D OF SPLIT RING IS USED AS DATUM F.I.R	.0001"
26	CROSS CORNER DIM J	A	MAXIMUM	.9402"
		B	MINIMUM	.9386"
27	CAGE HARDNESS ROCKWELL C MIN·MAX			26-32

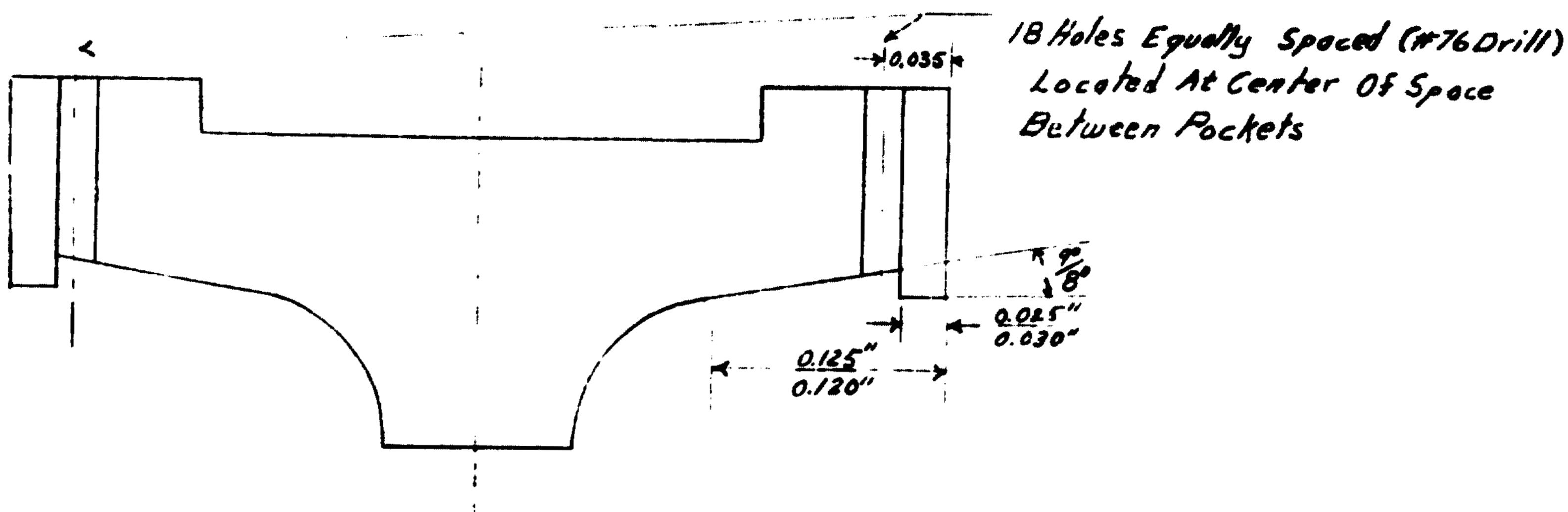
1. BEARING TOLERANCE - ABEC 7 (UNLESS OTHERWISE SPECIFIED)
2. 100% VISUAL INSPECTION (470659)
3. CAGE TO BE 100% FLUORESCENT PENETRANT INSPECTED (470900)
4. BEARING TO BE PERMANENTLY MARKED WITH CUST PART NO (MARK "J" ADJACENT OR BELOW CUST. PART NO.)
5. 100% ETCH INSPECTION (471637)
6. 100% MAGNAFLUX INSPECTION (471600)
7. CAGE TO BE ONE PIECE MACHINED AND DRILLED, OUTER LAND RING SEPARABLE, BALLS NON-SEPARABLE FROM CAGE.
8. HIGH POINT OF RADIAL RUNOUT SHALL BE PERMANENTLY MARKED ON EACH INNER RING FACE BY MEANS OF A CIRCULAR BURNISH MARK.
9. ALL CAGE DIMENSIONS AND ENGINEERING REQUIREMENTS ARE AFTER PLATING UNLESS OTHERWISE SPECIFIED
10. MIN RADIUS AT BOTTOM OF LINE INDENTING TO BE .002"
11. CAGE UNBALANCE NOT TO EXCEED [illegible] @ 500 RPM WITH LAND RIDING SURFACE AS REF. CAGES ARE NOT TO BE [illegible] TO MEET THIS REQUIREMENT
12. MATERIAL IDENTIFICATION & CONTROL PER SPEC 471006





<u>FIGURE 2</u>

RADIAL OIL PUMPING CAGE CROSS SECTION



lip, eighteen 0.635 mm (0.025 in.) diameter holes equally spaced around the bore extend to the rail OD. The second cage design incorporates 27 pumping grooves on each rail OD equally spaced around the circumference. The (0.050 in.) wide grooves extend from the side to the center of the rail at an angle of 20° with the face and tapered radially outward, see Figure 3.

The final cage configuration shown in Figure 4, is an inner ring piloted cage based on standard aircraft bearing design guide lines and tolerances. All modified cages are manufactured from AMS 6414 steel and silver plated.

2.2 Lubricants

The selection of the test lubricants for this program was made from military specification lubricants used in recirculating oil systems for aircraft jet engine and transmission bearings. The four fluids used in the test program were:

1. A domestic refined Type I ester which meets the requirements of MIL-L-7808H specification.

2. A Type II ester which meets the requirements of MIL-L-23699.

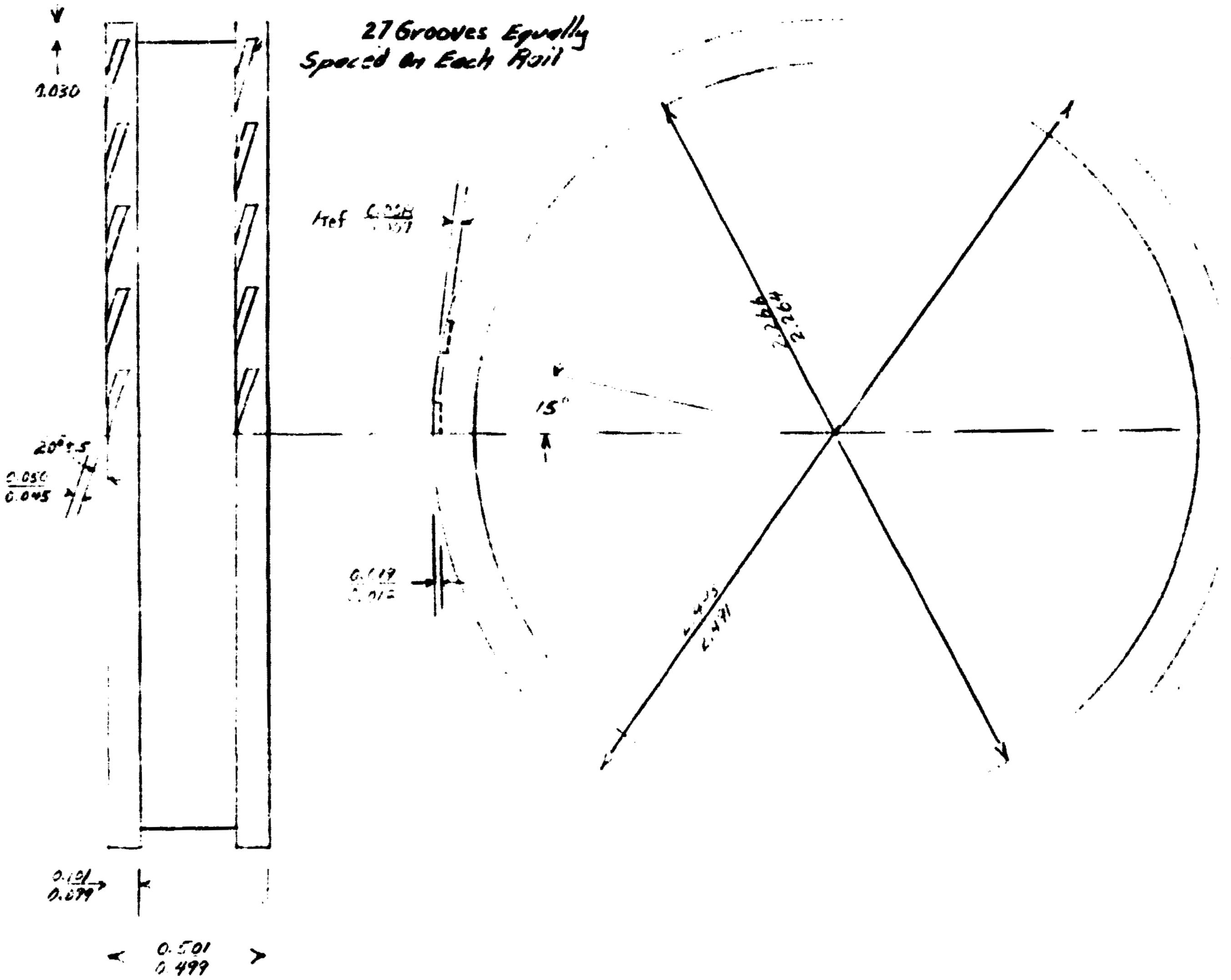
3. A specially formulated polyphenylether.

4. A foreign refined Type I ester which meets the requirements of MIL-L-7808 and NATO-0-148 specifications.

Identification by chemical type, and the physical properties for each lubricant are listed in Table I.

FIGURE 3

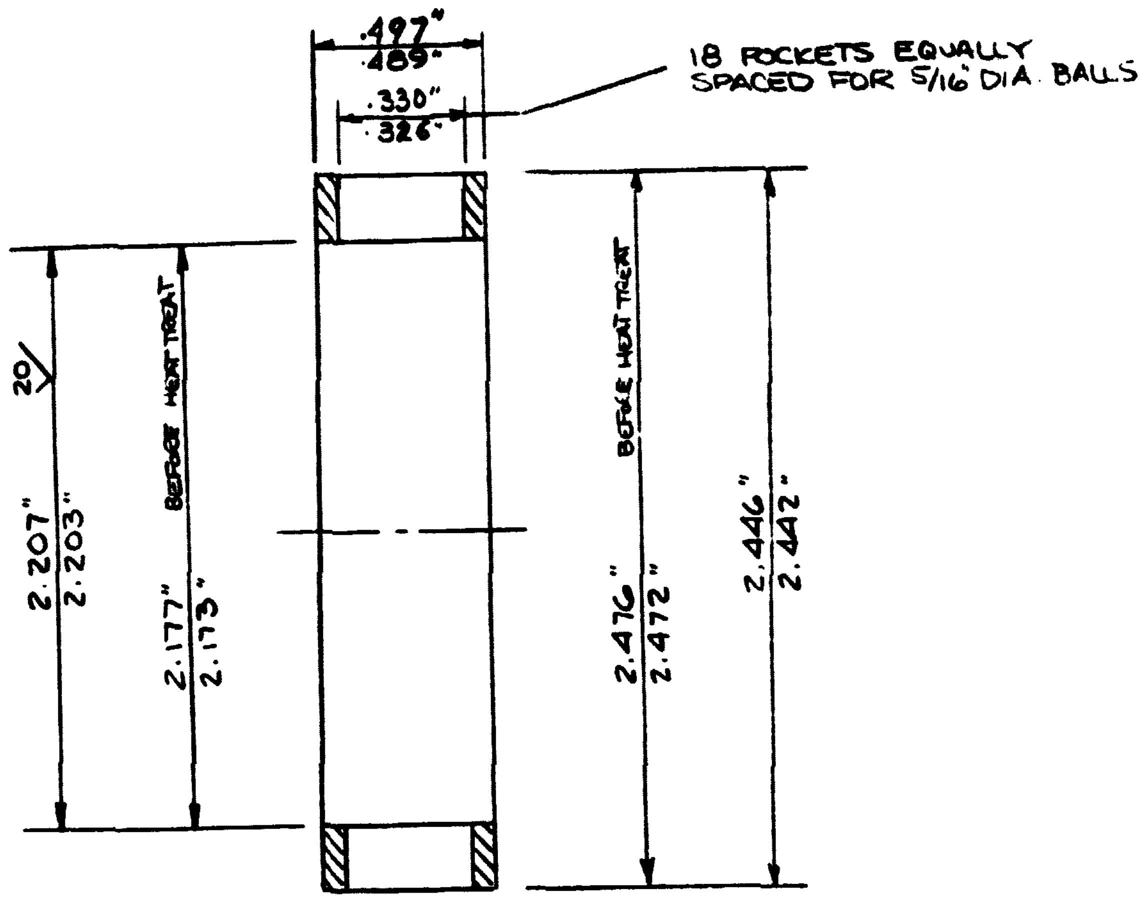
GROOVED CAGE DESIGN



1. All Dimensions in inches
2. Silver Plate Per Spec. 471069
   0.0005-0.001 Inches Per Surface
3. Cage Balance Less than 2 gr-cm

FIGURE 4

INNER RING RIDING CAGE DESIGN



Dimensions Shown are Before Silver Plating.

Material: AMS 6415 or AMS 6416 (Ref. SKF Specs. 471847 and 471711

Heat Treat Per Spec. 471464

Silver Plate Per Spec. 471069, .001"-.002" Per Surface

Cage Balance Requirement Per Spec. 471848

Except As Noted, Dimensional and Surface Requirements Per ¶2 and ¶3 of Spec. 470939

TABLE I - PHYSICAL PROPERTIES OF TEST FLUIDS

Chemical Type	Domestic Refined Type I Ester	Type II Ester	Polyphenylether	Foreign Refined Type I Ester
Flash Point, °F	450	485	565	426
Fire Point, °F		545	660	473
Kinematic Viscosity, cs				
@ 400°F		1.29		1
@ 210°F	3.3	5.15	12.9	3.3
@ 100°F	13.5	28.30	353	13.3
Density @ 20°C, g/cm^3	0.955	0.93	1.2	0.940
Surface Tension @ 25°C, dynes/cm			49.9	30
Specific Heat @ 300°F Btu/lb°F	0.56	0.55	0.43	0.56

3.0 TEST FACILITY

All tests were performed in an existing SKF owned high speed bearing test facility modified to accomodate the requirements for emergency aspirated and mist lubrication of the test bearing. The basic test equipment consists of the following components:

Test Rig
Mist and Bearing Cooling Air Systems
Rig Bearing Recirculating Lubrication System
Instrumentation

3.1 Test Rig

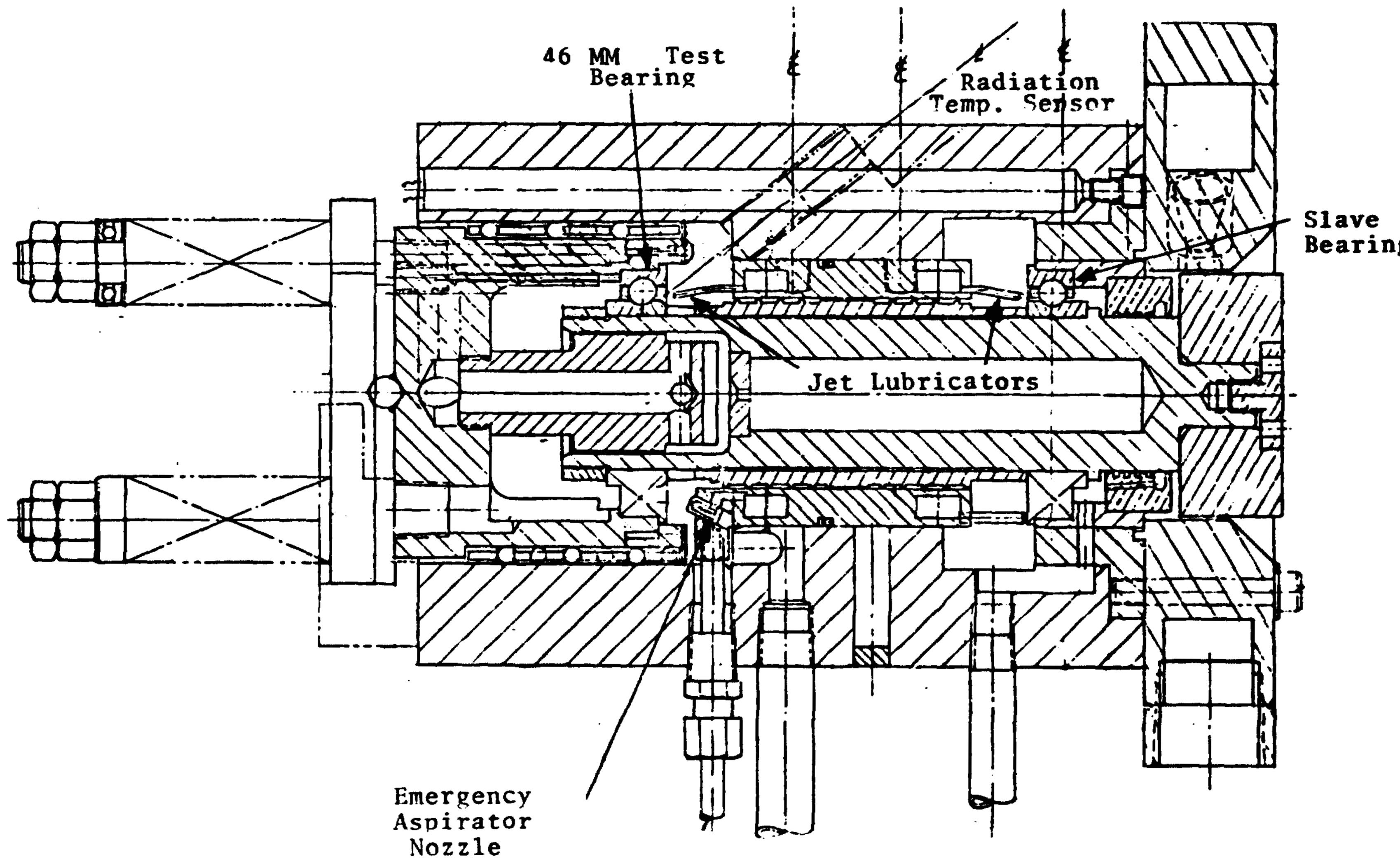
The test rig, layout drawing shown in Figure 5, consists of a 8.5 x 6.5 x 9 inch rectangular rig housing with a stepped bore to accomodate the hollow shaft, support bearings, bearing housings, and oil delivery manifolds. The housing also contains access holes for cartridge heaters, oil and air inlet lines, and oil scavenge lines.

The shaft is supported on two 45.923 mm bore angular contact ball bearings (described in Section 2) which are thrust loaded against each other. The test bearing, located on the end of the shaft opposite the drive turbine, is mounted in an extended housing "load plug" which is supported within the rig housing on a ball sleeve assembly. The ball-cage assembly permits both free axial movement and angular rotation of the load plug and thus provides a means of applying thrust to the test bearing and measurement of the bearing drag torque. The thrust load is applied directly to the center of the load plug through a ball-to-flat contact by a spring loaded beam. The load beam, which reacts through two rod extensions to the rig housing, is equipped with a temperature compensating strain gage system to provide thrust measurement capability. The thrust load, impacted in this manner to the shaft through the test bearing, is reacted by the rig bearing on the other end of the shaft.

Test bearing torque measurement is achieved since the load plug assembly, containing the bearing outer ring, is free to rotate in the supporting ball sleeve and at the ball-to-flat thrust load application point. During operation of the rig, the load plug assembly is restrained from rotating by a strain-gaged, flexible "torque arm" attached to the rig housing and reacted against a pin in the face of the load plug.

FIGURE 5

LAYOUT DRAWING OF TEST RIG



The inboard end of the ball sleeve is sealed by a flat teflon washer supported by a steel washer of slightly smaller O.D. attached to the load plug end. This seal is incorporated to eliminate the flow of cooling air and mist through a path away from the bearing. To minimize the loss of cooling air and mist to the rig bearing, a viscous pumping seal is located between the two bearings by incorporating a grooved sleeve on the shaft. The stationary surface of the viscous pump is formed by the bore of the lubricant manifold.

The shaft is driven by an impulse air turbine located at the end of the rig opposite the load plug. The buckets are machined into an aluminum disk attached to the end of the shaft. The nozzle block, containing nine converging-diverging nozzles, is attached to the end of the rig housing. The turbine is designed to provide a variable speed range of 0 to 65,000 rpm with a maximum output power of 25 horsepower at 65,000 rpm.

To accomodate the various types of lubrication tests performed on the program, three differently configured lubrication manifolds were provided. All manifolds are designed to provide recirculating oil to the rig bearing through three 0.81 mm I.D. jets equally spaced on the turbine-drive-end face of the manifold. One jet is positioned to direct oil into the gap between the outer ring and the cage and the other two to the inner-ring gap. The lubricant manifold used in the lost lubricant and emergency aspirated oil tests is identical on the test bearing side except for the incorporation of an atomizing nozzle (standard tip from a De Vilbiss 127 atomizer). The tip is mounted in a special connector to permit forced air to aspirate oil from the supply annulus and direct it into the test bearing.

The manifold configuration incorporated to provide mist lubrication to the test bearing consists of an open annulus with a cross-sectional area of 3.9 cm^2 (0.6 in^2) into which the mist is supplied. Eight converging nozzles, with an exit diameter of 2.0mm (0.080 in), positioned in an equally spaced circular pattern extend from the annulus to impinge mist on the chamfer on the inner ring and into the gap between the inner ring and cage bore. Exterior to the mist cavity, a second annulus is formed by a flange on the manifold and the rig housing wall to serve as the manifold for the through bearing cooling air which is directed into the bearing by eight straight 3.2 mm (.125in) diameter nozzles. The air nozzles are equally spaced between the mist nozzles.

To accomodate the drip/mist system test, the mist manifold was modified by plugging six of the mist nozzles and inserting an oil drip tubes in the other two nozzles. The drip tubes are positioned to feed drops of oil into two through bearing cooling air streams located 180° apart, see Figure 6.

3.2 Mist and Bearing Cooling Air Systems

The mist and cooling air system is shown schematically in Figure 7. The air flow commences with the air compressor which has a rated output of 2.57 scmm (91 scfm) at 1.38 newtons/meter^2 (200 psig). Air feeds directly to a dryer and filter column which reduces the moisture content to a 227°K(-50°F) dew point and the hydrocarbons to 13 parts per million. This clean, dry air then flows to a 0.57 cubic meter (20 cu. ft.) receiver where the supply pressure is maintained between 6.55×10^5 and 7.24×10^5 newtons per square meter (95 to 105 psi).

The air leaving the receiver divides, one portion going to the mist system and the remaining portion to the cooling air system. A pressure regulator, flowmeter, and thermocouple are located in the mist air line to control and measure the air flow rate. The air then enters a 3 kw heater where it is heated to 478°K before entering the mist generator. The mist generator consists of a .38m^3/min. mist generating head housed in a rectangular mist oil retainer where the mist oil supply is maintained at 356 to 367°K (180 to 200°F) by a thermostat controlled electrical heater attached to the container. Leaving the generator, the mist flows through a 25.4 mm (1 in.) I.D. stainless tube to the mist manifold in the test rig. A thermocouple is located just prior to the rig housing to determine the mist inlet temperature. The description of the mist manifold accompanying eight nozzles are described in Section 3.1.

The cooling air passes through a pressure regulator and 4 kw heater and then divides into three separate paths; 1) shaft cooling air, 2) outer ring cooling air, and 3) through bearing cooling air. Each path contains a flowmeter, valve, and two

FIGURE 6

LAYOUT DRAWING OF DRIP/MIST SYSTEM

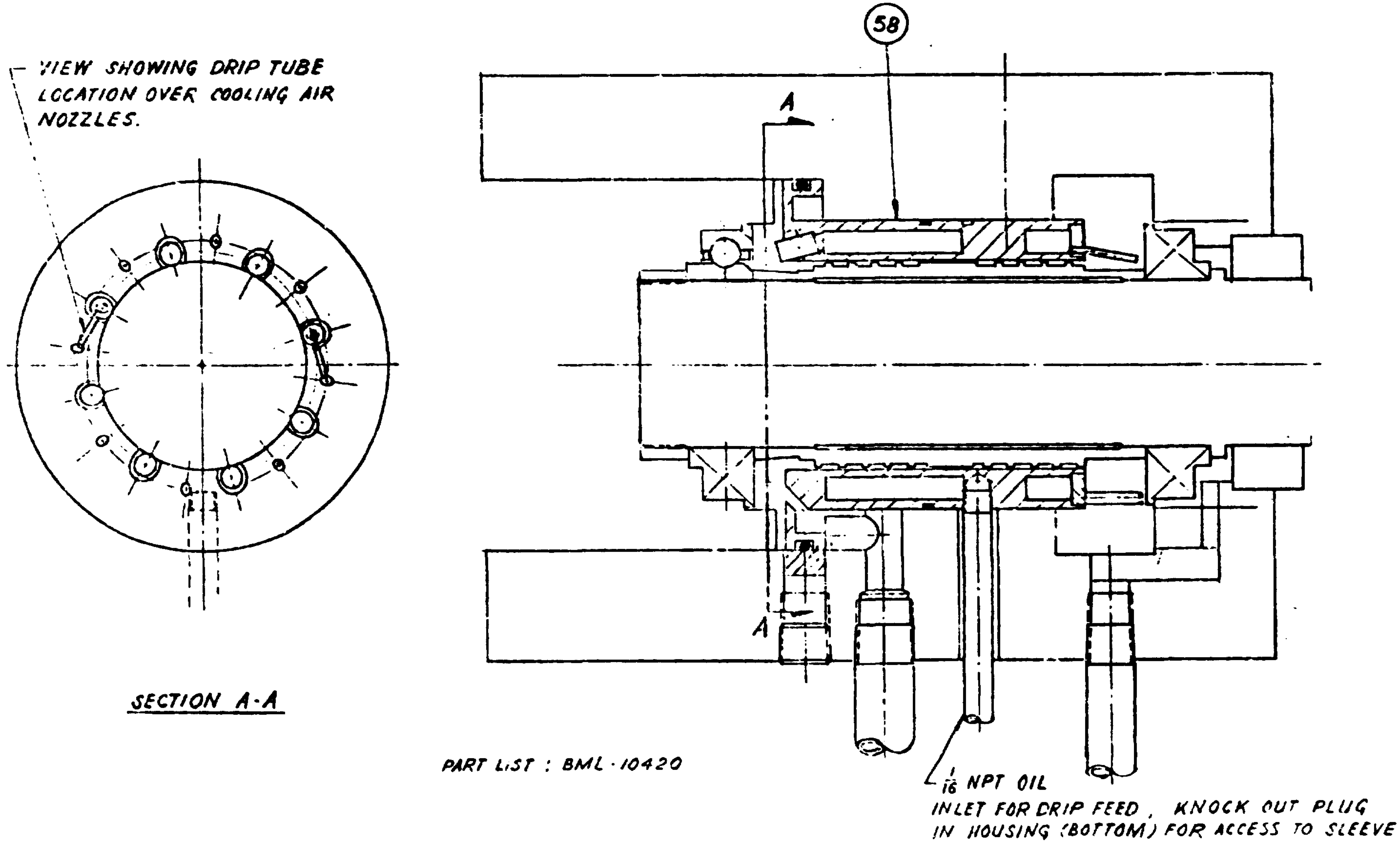
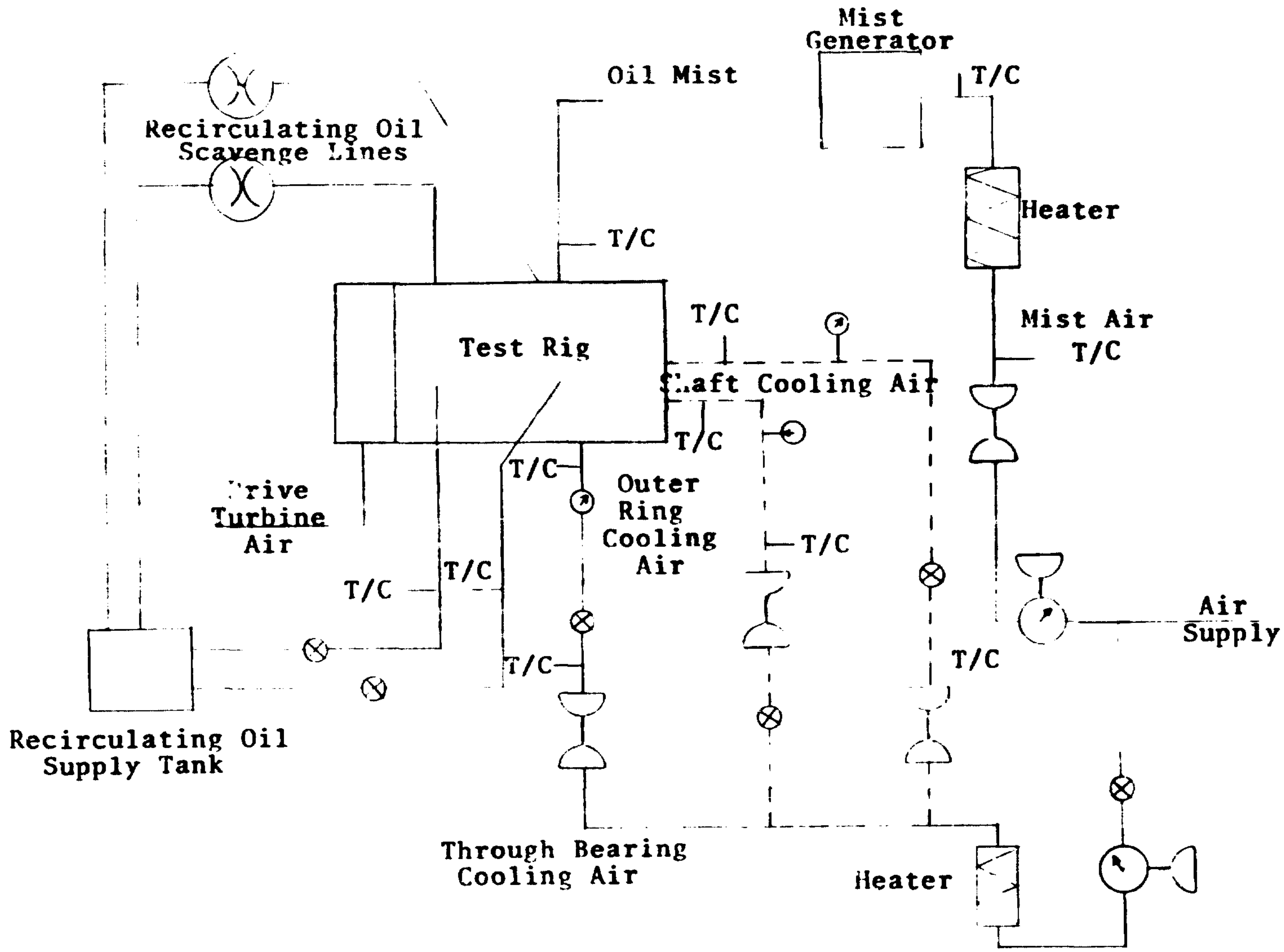


FIGURE 7

TEST RIG, RECIRCULATING AND MIST OIL, AND COOLING AIR FLOW SCHEMATIC



thermocouples to control and measure the air flow rate and the temperature of the air entering the rig.

The shaft or inner ring cooling air enters the hollow shaft through a 12.7 mm (0.5 in) I.D. stationary tube attached to the load plug. It is forced radially outward from this manifold through four 6.35 mm (0.25 in.) diameter holes located 90 degrees apart. The air then sweeps axially outward through the annulus formed by the shaft bore and manifold outer surface. A thermocouple is located at the end of the annulus to sense the exhaust air temperature.

The bearing housing , outer-ring cooling air also enters through a hole in the load plug and passes through an annulus radially outward from the outer-ring seat. The air is exhausted through a hole located 180 degrees from the entrance where the temperature is sensed by a thermocouple.

The through bearing cooling air enters through the rig housing into the annulus located in the mist manifold. It then passes through the 8 air nozzles described in Section 3.1 and impinges on the test bearing. The combined mist and through bearing cooling air is forced through the bearing and the exhaust temperature sensed by a thermocouple located slightly downstream of the bearing before it leaves the rig through a hole in the load plug.

3.3 Rig Bearing Recirculating Lubrication System

Oil circulation to the rig bearing is provided by an internal gear type pump through a filter unit, flowmeter and flow control valve to the test rig oil manifold. The oil is recovered through drain holes in the test rig and returned to the storage tank by a scavange pump. A by-pass line is incorporated downstream of the pump to permit the excess flow to be pumped directly back into the supply tank. The filter unit accepts fiber glass elements having a specific pore size of 20 microns and deliberately has excessive flow capacity in order to secure low pressure drops and long life, even with an oil that is undergoing some thermal degregation. The oil in the storage tank is heated by a thermostatically controlled electric immersion heater capable of maintaining the supply oil at a temperature in excess of 450°K (350°F) if desired.

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3.4 Instrumentation

The desired temperature measurements are sensed by shielded iron-constantan thermocouples. The test bearing inner and outer ring temperatures are recorded on a continuous strip chart recorder, and the rig bearing temperature on a temperature indicator. The following temperatures are automatically recorded on a Esterline Angus Model E-6704 multipoint recorder.

Mist air at flowmeter and entering mist generator

Mist entering rig

Shaft cooling air at flowmeter, entering shaft and leaving shaft

Bearing housing cooling air at flowmeter, entering rig, and leaving housing

Through bearing cooling air at flowmeter and entering rig

Combined mist and through bearing cooling air out

Rig bearing oil in and out

Test bearing housing

Rig housing

In addition the following data are recorded manually:

Oil level in mist supply tank

Oil temperature in mist supply tank

Rig bearing outer ring temperature

Shaft speed

Mist air flow rate

Shaft cooling air flow rate

Bearing outer-ring cooling air flow rate

Through bearing cooling air flow rate

Pressure at mist air flowmeter

Pressure at each cooling air flowmeter

Mist generator adjustment screw position

Recirculating oil flow indication

The test bearing drag torque and thrust load are sensed by strain gaged beams and recorded on a Hewlett Packard Model 7702B strip chart recorder. The shaft speed is detected by a magnetic pickup and presented on a Hewlett Packard Model 531CR electronic counter. All air flow measurements are made by Brooks flowmeters. All air flow rate measurements are corrected for the pressure and temperature values at the flowmeters.

4.0 ANALYTICAL STUDY

In preparation for designing the mist lubrication and cooling air system for a helicopter gas turbine mainshaft bearing, several different methods of applying the mist oil and cooling air were reviewed. The evaluation of these various methods and the final selection were performed on NASA Contract NAS3-16826 which was performed concurrently with this program. (5). The selected procedure was then refined considering the theoretically required mist oil and cooling air flow rates. Methods initially considered included:

1. Injection of oil mist and cooling air axially to both sides of the test bearing through oil mist and cooling air nozzles located in a circular pattern similar to that used in many aircraft engine recirculating jet lubrication systems.

2. Injection of mist oil radially through holes in hollow shaft and bearing inner ring with cooling air forced axially through the bearing and through a circumferential path in the outer ring housing.

3. Injection of oil mist and cooling air through nozzles located in a circular pattern on one side of the bearing allowing the momentum and small pressure build up to carry the mist into the bearing and exhaust through a port on the opposite side.

4. Same as method three plus modification of the bearing inner ring and cage geometry to aid in pumping the lubricant into needed locations within the bearing. In addition, cooling air would be forced axially through a hollow shaft to aid in the removal of heat from the inner ring, and circumferentially through the bearing housing to remove heat from the outer ring.

Method #4 was selected for more detailed evaluation. This method of application had the advantages of permitting cooling air to be inserted in three different locations which could be independently controlled. In this manner, temperature differentials in the bearing assembly could be properly adjusted. The volume of air, and indeed the need for each flow path could be established through testing. In addition, the possibility of the mist and cooling air flows bucking each other, as may occur if method 1 was incorporated, was eliminated. The housing and shaft cooling air flow paths also provide alternatives to forcing all the cooling air through the bearing where high air velocities could possibly sweep useable oil from the cage, rings and ball surfaces, out of the bearing.

4.1 Evaluation of Cooling Air Flow Rate Requirements

One of the major questions with respect to mist lubrication and air cooling of high speed helicopter engine bearings was the feasibility of removing the bearing generated heat by the use of mist and cooling air. To establish the expected quantity of air required, the heat generated by the bearing was calculated at various shaft speeds and specified loads. This is the amount of heat which must be transferred from the bearing to obtain a steady-state thermal condition. From the established bearing heat generation rate,the quantity of air required was calculated.

The expected heat generation rate of a 46 mm bore angular contact ball bearing with a thrust load of 1779 Newtons (400 lbs.) applied and lubricated by mist oil was initially calculated using SKF Computer Program AE79Y003. The applicable output values from the computer runs are listed in the following table:

COMPUTER OUTPUT DATA
(Bearing-464539)

Shaft Speed (RPM)	Heat Generation Inner (BTU/HR)	Outer (BTU/HR)	Lube (BTU/HR)	Cage (BTU/HR)	Total-Lube (BTU/HR)	Drag Torque (IN-LB)	Contact Angle Inner (Degrees)	Outer (Degrees)	Max. Hertz Stress Inner (PSI)	Outer (PSI)	Major Contact Length Inner (IN)	Outer (IN)
25,000	517	453	287	28	938	1.27	44.5	31.0	1.9×10^5	1.6×10^5	.046	.074
35,000	1800	893	600	53	2746	2.37	45.9	24	1.9×10^5	1.7×10^5	.046	.080
45,000	4211	1238	1000	84	5533	3.6	45.7	19	1.9×10^5	1.8×10^5	.046	.086
55,000	7736	1475	1504	118	9329	4.87	44.7	15	1.9×10^5	1.97×10^5	.046	.092
65,000	12157	1647	2077	156	13950	6.1	43	12	1.9×10^5	2.1×10^5	.046	.10

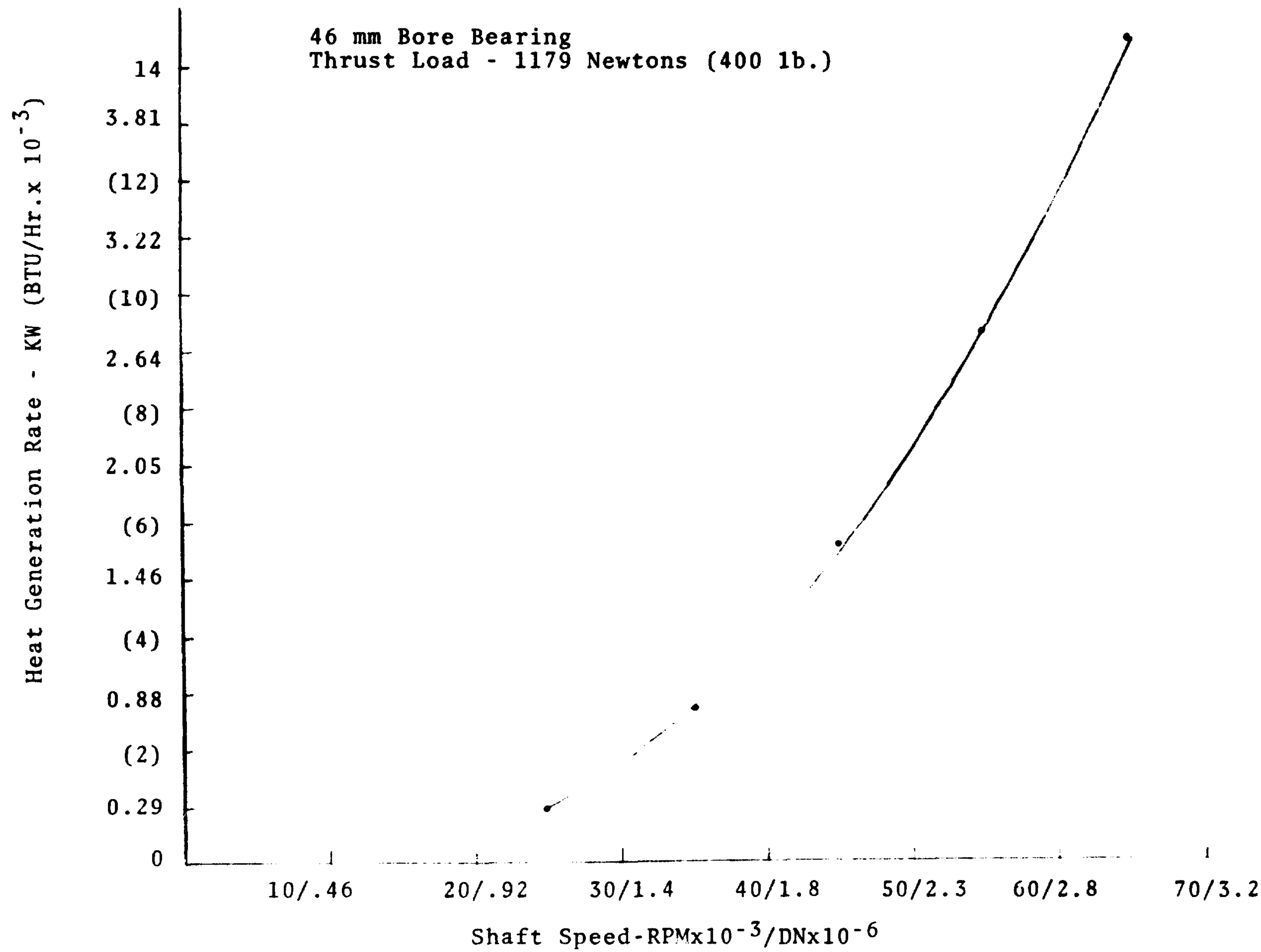
This data presents the theoretical heat generation by the inner and outer race contacts, the cage, lubricant churning and thetotal heat generated less the lubricant churning. The latter value represents the heat generated when mist lubrication is utilized and is presented graphically in Figure 8 .

This projected curve of heat generation was utilized to determine the quantity of cooling air required to maintain a steady state thermal condition in the bearing at an operating temperature of 533°K (500°F) at the maximum anticipated speed, i.e., 65,000 rpm (3 x 10^6 DN). The calculations are presented in Appendix I which indicate that a total air flow of approximately 2.04 scmm (72 scfm) would be required when the air inlet temperature was 366°K (200°). Based on the conceived method of supplying the mist and cooling air as presented earlier, it was considered logical that 50 percent of the air should be passed through the bearing to cool the race and cage contact surfaces. The remaining air would be applied external to the bearing cooling the outer housing and the hollow shaft at a volume ratio equal to that of the heat generation rates at the inner and outer ring contacts.

The computer program was also utilized to establish the expected bearing drag torque, the major contact lengths formed between the races and rolling elements, and the maximum contact (Hertz) stresses at the contact for various shaft speeds from 25,000 to 65,000 rpm with a 1779 Newtons (400 lbs.) thrust load applied to the bearing.

FIGURE 8

CALCULATED TEST BEARING HEAT GENERATION RATE



4.2 Evaluation of Mist Oil Supply Rate Requirements

Elastohydrodynamic lubrication theory was applied to calculate the quantity ofoil necessary to lubricate the bearing inner-ring contacts. These contacts are considered to be the most critical with respect to oil replenishment as the oil displaced from the inner ring will have a tendency to flow to the cage pockets and outer ring due to centrifugal forces. The maintenance of an elastohydrodynamic oil film relies on a sufficient supply of lubricant in the inlet of the contact between the contacting surfaces. Insufficient lubricant supply results in a phenomenon called "EHD film starvation" characterized by the reduction in EHD film thickness and in a lessening of the distance from the front edge of the contact to the miniscus line at which the fluid pressure begins to rise (6).

The calculations of required oil replenishment rate were based on the assumption that all of the oil displaced out of the inner-ring track as a ball passes must be replaced by the mist supply, a very conservative assumption. It is known that some of the oil displaced from the track does flow directly back into the depleted track by the action of surface tension forces after the rolling element has passed. Some of the oil which is lost is also recirculated in the bearing by centrifugal, gyroscopic, and windage forces and finds its way back to the contact area thus decreasing the amount of replenishment oil required. It has been shown that, at high speeds, values of starvation can be relatively high and adequate EHD lubrication still be obtained.

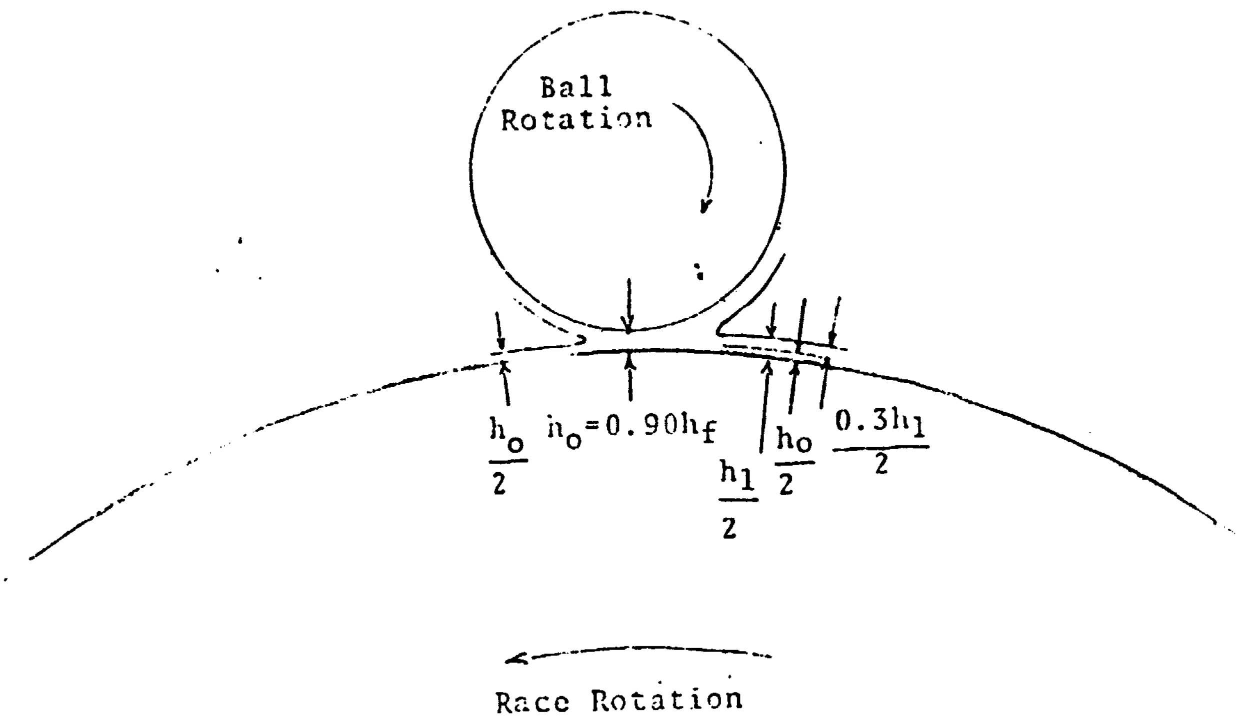
In performing the oil flow rate calculations the following assumptions were made:

(1) The contact is only slightly starved, i.e. the film thickness (h_o) is 90% of the unstarved film thickness (h_f) calculated from classical theory.

(2) The half-films clinging to the ball and race ahead of the contact, see Figure 9, are to be increased in thickness by oil mist directed onto the track or ball.

FIGURE 9

INNER RACE BALL CONTACT SHOWING
OIL FILM THICKNESS AT VARIOUS LOCATIONS



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(2) At each ball passage, the oil on the track is displaced out of the track except for a thin layer of film thickness $h_{o/2}$.

Starvation theory developed in Reference 5 shows that in order to maintain a ratio $h_o/h_f = 0.90$, the half film height $(h_1)/2$ at the contact inlet must satisfy the following conditions:

$$h_o/h_1 = 0.70 \qquad (1)$$

$$\frac{u \alpha \eta R_x}{h_1^{3/2}(3+2K)} = 0.10 \qquad (2)$$

where u = rolling velocity at the contact (in/sec)

η = viscosity of oil at the inlet of contact (lb-sec/in^2)

α = pressure-viscosity coefficient (in^2/lb)

R_x = equivalent radius of curvature in the rolling direction (in)

R_y = equivalent radius of curvature transverse to rolling direction

k = R_x/R_y = 0.027 for a typical ball-raceway contact with contact ellipse axis ratio equal to 10.

h_o = plateau film thickness in the contact (in).

The amount of increase of inlet film thickness between two successive contacts due to spray is

$$h = h_1 - h_o = 0.3h_1 \qquad (3)$$

The time interval between two contacts is t = s/u where s is the rolling element spacing. The contact frequency is

$$f = \frac{1}{t} = \frac{u}{s} \qquad (4)$$

The rate of oil supply to the rolling track is given by

$$q = L \Delta h \; w \frac{f}{Z} \qquad (5)$$

where w is the effective width of the rolling track and ℓ is the length of rolling track.

For z rolling elements,

$$\ell = sZ$$

From the SKF Computer Program AE70Y003 the inner race major axis contact is approximately 0.046 inch for all speeds from 25,000 to 65,000 rpm with a thrust load of 1,779 Newtons (400 lbs.).

The factor 1/2 in the above equation for q denotes that the layer thickness varies linearly with distance from $h_0/2$ to $h_1/2$ at the inlet. Thus, after substituting for Δh from (3), one has

$$q = 0.15\ h_1\ w\ u\ z \qquad (6)$$

Substituting for h_1 from (2) into (6) and using k = 0.027 gives,

$$q = 0.15\ w\ u\ z\ (3.28 \mu \eta \alpha R_x^{1/2})^{2/3} \qquad (7)$$

For the test bearing, SKF No. 464539VAA, one has

$$Z = 18$$
$$u = 0.0663N \text{ in/sec.}$$
$$N = \text{shaft speed (rpm)}$$
$$R_x = 0.1362$$

The properties of a Type II ester and the calculated q values at N = 65,000 rpm are presented in the following table.

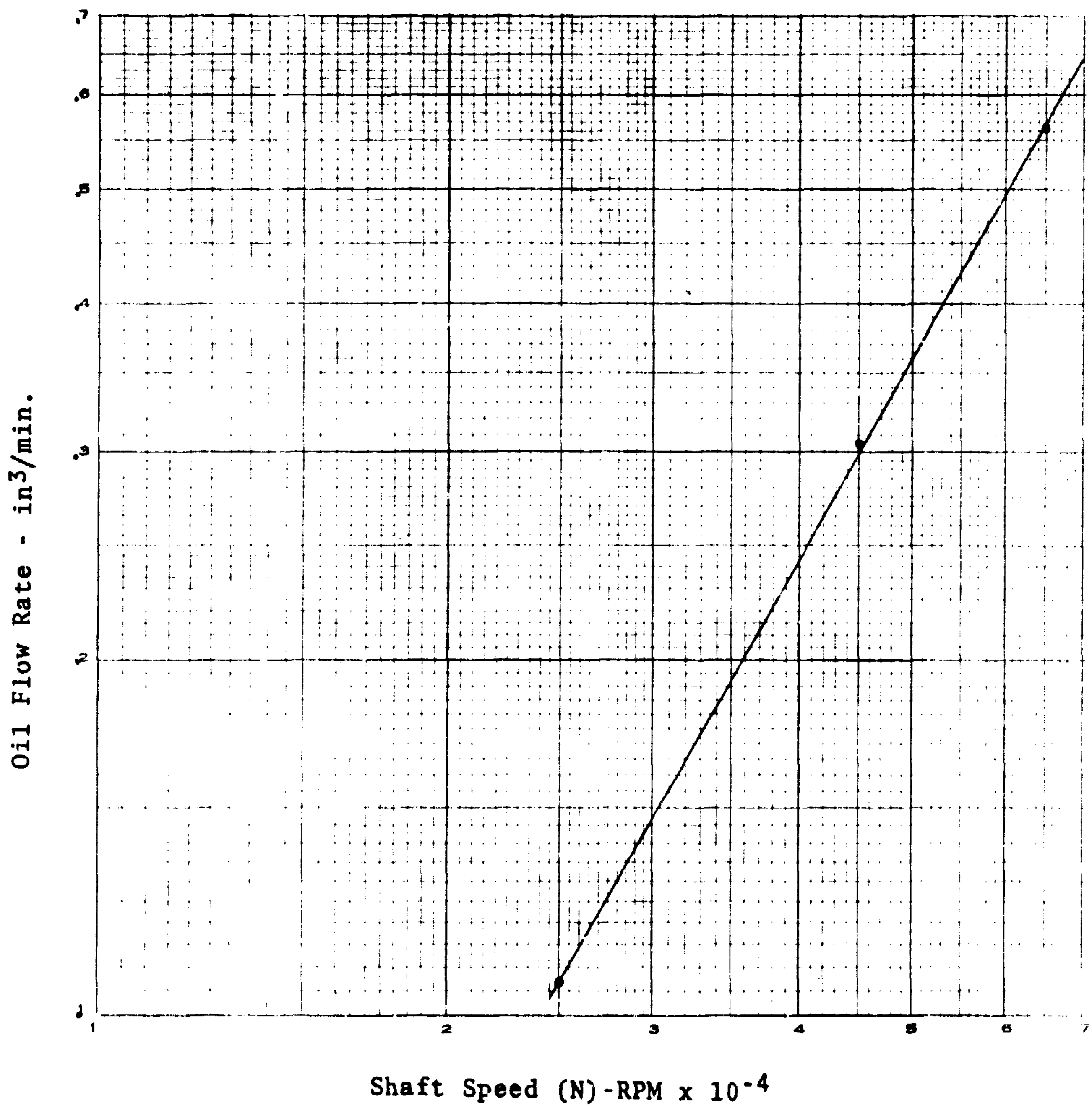
	Type II Ester MIL-L-23699 (Jet No. II)
Sp Gr @ 20°C	0.99 gm/cm^3
Pressure-Viscosity Coeff.(α)	1.1×10^{-4} in^2/lb
Kinematic Viscosity @ 500°F	0.88 c.s.
Absolute Viscosity @ 500°F	0.87 c.p.
Absolute Viscosity @ 500°F	1.26×10^{-7} $lb\text{-}sec/in^2$
Absolute Viscosity	1.39×10^{-11} sec
	0.565 in^3/min.

Figure 10 is a plot of the variation of the calculated required oil supply rate for $h_o/f_f = 0.90$ with respect to shaft speed.

The calculations show that oil replenishment rates of 3.6cc/min (0.22 in^3/min) and 9.2cc/min (0.56 in^3/min) are required when operating at 38,000 and 65,000 rpm respectively. A comparison of the mist lubricant requirement rate as calculated in the same manner for a 125 mm bore ball bearing with mist flow established by testing shows that the calculated value is conservative, approximately 8 times the actual quantity required (5). However, the calculated values were used as a guide line for the mist lubrication testing on the program.

FIGURE 10

CALCULATED OIL REPLENISHMENT RATE REQUIREMENT



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4.3 Mist and Cooling Air Delivery System

In establishing the physical components of oil mist and cooling air delivery system several design parameters were considered. These included the selection of the ratio of air passed through the bearing to the externally applied air; quantity of mist air, mist generator size selection; sizing of mist transfer piping and manifold; number, design and location of mist nozzles; point of application of the mist, and modifications to the bearing to assist in distributing the oil to the desired locations within the bearing.

Due to the apparently large quantity of mist and cooling air flow required to cool the test bearing, as established in Section 4.1, the selection of a system to supply mist and cooling air through the bearing and cooling air around the bearing was considered essential even though the complexity may be difficult to apply in all gas turbine mainshaft bearing mounting configurations. To take full advantage of the heat transfer surfaces, both internal and external of the bearing, it was concluded that an equal split between the total required flow would be reasonable. Thus, considering the maximum desired operating speed of 65,000 rpm, as the design speed, an air flow rate of 1.02 scmm (35 scfm) would need to be passed through the bearing and the same amount external to the bearing. It was also considered reasonable to divide the external cooling air proportionally, based on the ratio of the calculated heat generated at the inner ring and outer ring, and direct the flows along the surfaces of a hollow shaft and through a circumferential groove machined in the bearing housing.

The air passing through the bearing would consist of a combination of mist air and auxiliary cooling air with the additional cooling air applied externally.

Based on the estimated mist oil flow requirement determined in Section 4.2 an Alemite high volume oil-mist generator (Model 3720-BS) was selected with a 13 cfm mist head. The mist head has a rated mist oil delivery rate of 0.4 in^3/hr per cfm air flow. Experience with a similar mist generator had shown that appreciable greater flow rate existed when heated oil and preheated air was used. Thus for the purpose of determining the mist air flow rate necessary to provide the desired oil flow, an oil delivery rate value of 2.7 in^3/hr. per scfm was used.

The maximum estimated oil flow rate requirement of 557 cm^3/hr. (34 in^3/hr.) could be obtained with a mist air flow rate of 0.354 scmm (12.5 scfm). The remaining through bearing cooling air 0.665 scmm (23.5 scfm) would be supplied as auxiliary cooling air which would contain no oil.

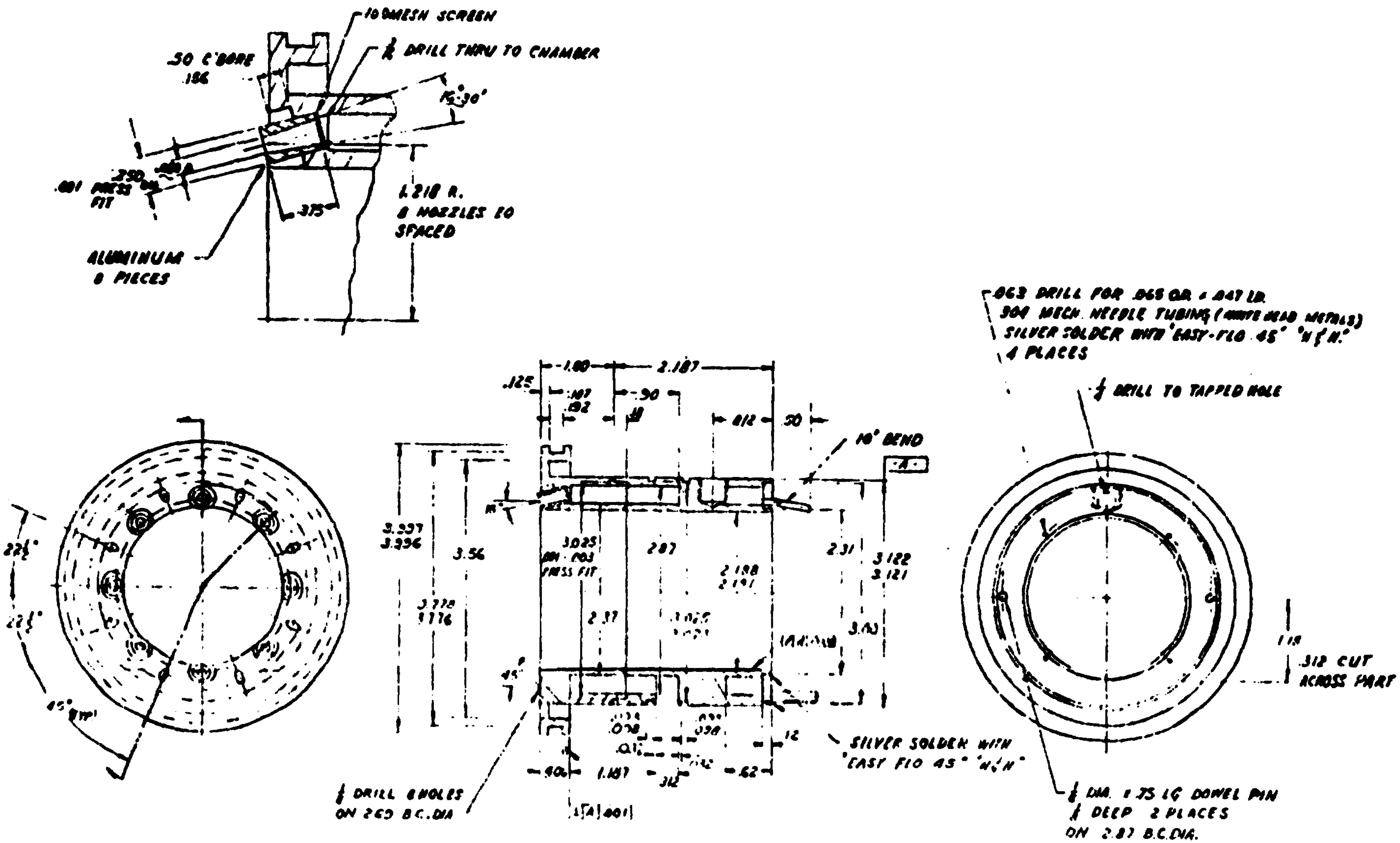
The oil-mist research, conducted by the Mobil Oil Company, reported in References 3 and 4, established that the best oil wetting occurred on a rotating disk when the application nozzle was of a reclassifying design. The reclassifying nozzle increases the mist oil particle size thus increasing the momentum of the oil particles which decrease the effect of air currents from dispersing the particles away from the desired surfaces. Commercial mist generators produce oil particles in the 4 micron range which minimize the premature plating out of the oil in the transfer lines. The reclassifying nozzle design found most effective by Mobil consisted of a screen located in the entrance of the application nozzle on which oil is collected and larger particles emitted.

Mobil also established that maximum wetting of the desired surface occurred when the nozzle was located 4 to 8 nozzle diameters from the surface and optimum efficienty of the oil supplied is obtained when a secondary air nozzle, concentric and outside the mist nozzle, is provided to blow any oil dripping from the primary nozzle into the bearing.

Based on this information the mist nozzles were designed to be converging with a throat diameter of 2.0 mm (.080 in) and having a screen at the entrance to increase the mist particle size. A 100 mesh screen using 0.076 mm (0.003 in) diameter stainless steel was utilized which provides a 50 percent open area, see Figure 11. A total of eight nozzles was selected to be positioned in a circular pattern to provide a well distributed oil flow pattern to the bearing; thus, attempting to eliminate hot spots. The design of the mist nozzles is presented in Appendix II. Due to space limitation it was not possible to incorporate the cooling air nozzles concentric with the mist nozzles. Therefore, eight straight air nozzles were incorporated on the same circle and equally spaced between the mist nozzles.

FIGURE 11

MIST AND COOLING AIR NOZZLE CONFIGURATION



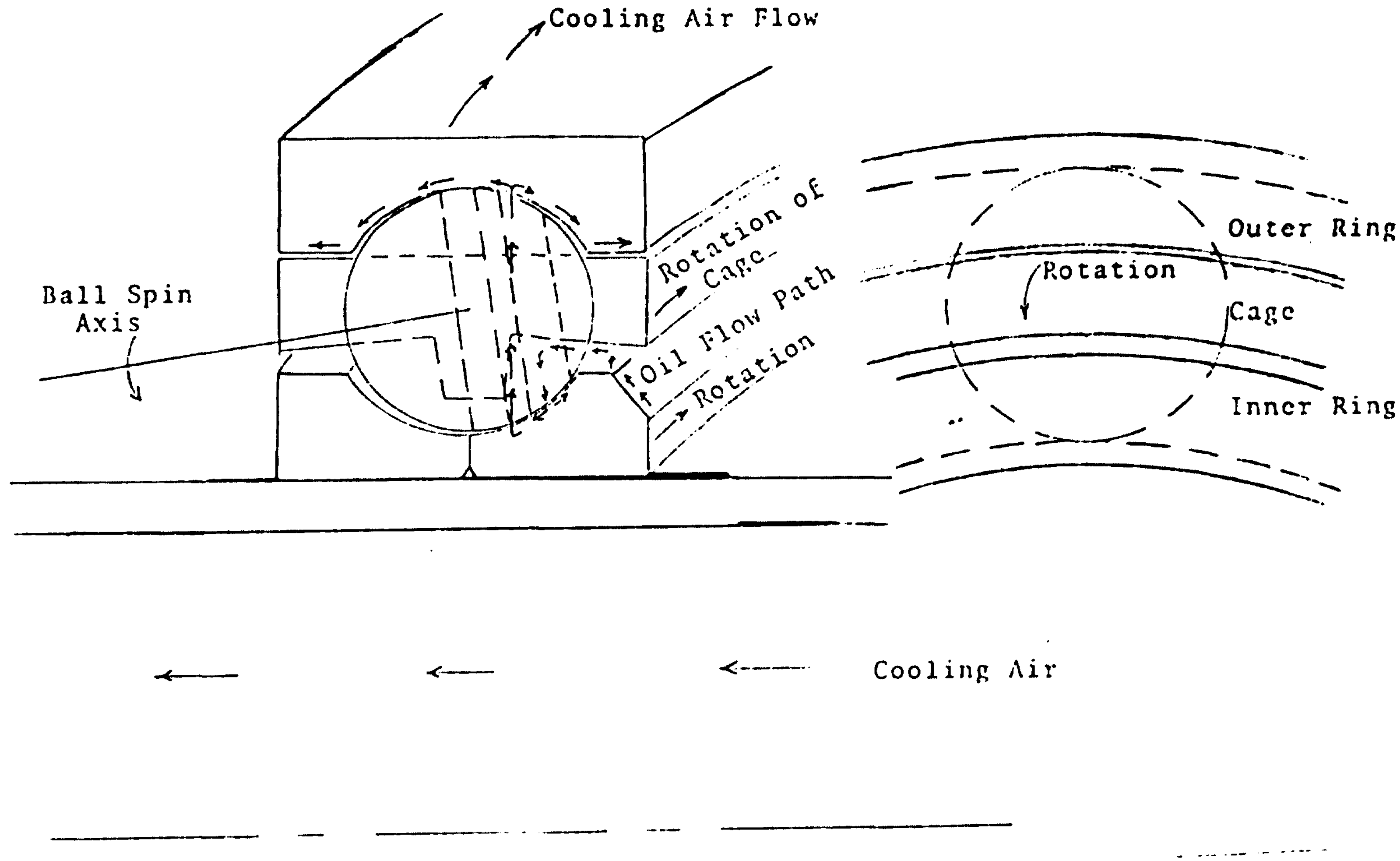
In the lubrication of a ball bearing, there are four major types of contact areas which must be plated with oil: 1) the contact area between the ball and inner ring (inner ring ball track), 2) the contact between the ball and the outer ring (outer ring ball track), 3) area between the cage and the land on which it rides and 4) the contact area between the ball and cage pocket. The ball tracks can be lubricated by applying oil to either the ball or the races.

Since the oil entering a bearing migrates toward the outer ring, the most difficult area to lubricate is the inner ring ball track. It is also noted from the calculated bearing heat generation rate values presented in Section 4.1 that proportionally more heat is generated at the ball inner-ring contacts than at the ball outer-ring contacts as speed increases. This condition results from differences in the contact angle at the inner and outer rings. At high speeds the outer-race contact angle decreases and the inner-ring contact angle increases thus producing less sliding at the outer ring. In addition to being more difficult to lubricate, more heat must be conducted from the inner ring to obtain a thermally stable operating bearing.

Thus it was decided that the dual mist and cooling air nozzles should be directed from the inboard side of the bearing toward the space between the inner ring and cage and against a chamfer machined on the inner ring side face. In this manner the air, at its lowest temperature, impinges on the inner ring on the loaded half to obtain maximum cooling of this area. A portion of the oil or mist enters the bearing inner cavity where it is available to be picked up by the ball (collision between the mist particles and ball) as it orbits the shaft. An additional portion of the oil is plated out on a chamfered surface, See Figure 12, provided for this purpose. The chamfer is configured to insure that the location of the intersection of the of the sloped surface and the ring land is well within the outer edge of the cage. The oil on the chamfer migrates along the surface to the corner due to centrifugal force where it is thrown off and impinges on the cage bore or carried further into the bearing by the cooling air flow. The cage is modified to form a radially inward slope on its inner diameter surface along which the oil flows, again due to centrifugal forces, to the ball pocket where it lubricates the contact area between the ball and cage and wets the inner ring contact path on the ball.

FIGURE 12

LUBRICANT FLOW PATH



The lubricant on the ball provides lubrication to the inner ring and ball contact area. As the ball rotates into the inner race that portion of the oil which does not form the EHD film is forced to both sides of the major ellipse. Some of the extruded oil remains on the ball and some on the race where it has a tendancy to migrate toward the outer race due to the centrifugal force generated by the rotation of the ball and inner ring. In so doing, part of the oil flows back into the inner ring ball track where it is again available to form the EHD film. Surface tension and cohesion forces also tend to draw the extruded oil back into the ball track area. In addition, according to EHD lubrication theory, that oil which forms the EHD film is equally divided on the ball and race and aids in forming the next contact EHD film. That oil which migrates to the outer stationary ring is available to lubricate the outer ring contact area and the outer ring land and cage contacts.

5.0 TEST PROCEDURE

Three basic types of tests were performed during the program; these were:

1. Lost lubricant and emergency lubrication tests
2. Step-speed tests
3. Extended period tests

In all tests a thrust load of 1779 newtons (400 lb.) was applied to the test bearing. The test load was selected to provide a maximum Hertzian stress of approximately 1.28×10^9 newtons per square meter (200,000 psi) at speeds from 25,000 to 65,000 rpm. The theoretical contact stress and the contact angle, calculated using SKF Computer Program AE70Y003, at various test speeds are presented below:

Shaft Speed (rpm)	Contact Stress/Contact Angle (ksi)/(degrees)	
	Inner Ring	Outer Ring
25,000	190/44.5	160/31
35,000	190/45.9	170/24
45,000	190/45.7	180/19
55,000	190/44.7	197/15
65,000	190/43	210/12

In the emergency and lost lubricant tests, both the primary lubrication system, either recirculating or mist, and the emergency lubrication system were energized prior to rotating the shaft. In test runs where the rig housing heaters were being utilized, the heaters were also energized prior to shaft rotation and the housing heated to the desired temperature. After the desired lubricant temperature and flow rate were obtained, air was supplied to the drive turbine and the shaft accelerated at a relatively uniform rate to a test speed of 38,000 rpm (1.75×10^6DN) in approximately 30 minutes and operated at this speed until thermal stabilization was obtained. The primary lubrication was then shut off and operation continued until a bearing failure was imminent. In cases where it was feasible, the primary lubrication was reinitiated to prevent a catastrophic failure and permit additional checks to be performed.

In tests where a secondary or emergency method of lubrication was being supplied, the test was continued after cessation of the primary oil supply until the desired period of operation was obtained or a bearing failure was imminent.

All step-speed tests utilized a mist oil and cooling air system to lubricate and cool the test bearing. With the thrust load applied, the cooling air and mist air were turned on and allowed to pass through the rig while the oil in the mist supply tank was heated to the desired temperature, 266°K (200°F). The rig-bearing circulating oil system was also activated and the oil heated to the desired temperature while being circulated through the rig. In all cases, the same type oil was used in the recirculating system as in the mist system. During the warm up period, adjustments were made to the air heaters and pressure valves to obtain the operating temperature and flow rates.

Having obtained the desired operating conditions in the lubrication and cooling air systems, the shaft was accelerated to a relatively low speed. The test bearing temperature was allowed to stabilize and test data recorded. The shaft speed was then increased in incremental steps,allowing the bearing temperature to stabilize after each increase and data recorded. This procedure was continued until the maximum desired speed was obtained or a bearing failed. In all step-speed tests in which the rig housing heaters were not used, the test bearing generated heat transferred to the mist and cooling air was calculated by using the equation $q = \Delta t C_p W$ where:

q = heat transfer rate

Δt = inlet and outlet air temperature difference

C_p = specific heat of air at constant pressure

W = air flow rate

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The extended period tests procedure was similar to that used in the step-speed tests with the exception that the shaft was accelerated directly to the desired speed at a relatively uniform rate in approximately 30 minutes. The rig was then run for a desired period, shut off and allowed to cool to room temperature. The cycle was repeated until the pre-established period of operation at speed was obtained. One test was performed at 55,000 rpm (2.5×10^6DN) for 50 hours and another at 44,000 rpm (2.0×10^6DN) for 100 hours.

Following all tests, the rig was disassembled and the test bearing visually inspected using a light microscope with magnification up to 30X.

6.0 TESTING RESULTS AND DISCUSSION

The testing performed on this program, for the purpose of this discussion, is subdivided into two categories of type and classification. Type refers to the general method of running the test as described in Section 5, and class is a subdivision referring to the general purpose of the test.

The three types of tests performed are; (I) emergency lubrication, (II) step-speed, (III) extended period. Type I included two classes; 1) baseline and 2) emergency lubrication. The baseline tests were performed to determine the length of time that the test bearing would run after the cessation of lubricant supply under a given set of conditions. The emergency lubrication tests were performed to determine if the emergency methods applied were adequate to obtain the desired operating period of 30 minutes. A total of six Type I tests were performed.

Four classes of tests were included in the Type II category. These were: 1) relative evaluation of different lubricants used in a mist and cooling air system 2) evaluation of the effects of changes in the controlled variables of the mist and cooling air system 3) evaluation of the effects of changes in the mist system configuration 4) evaluation of cage changes to eliminate wear between the cage OD and guide land. A total of 14 tests of Type II were performed.

Two Type III tests were performed, both of the same class, to demonstrate an extended period of operation with a mist oil and cooling air system used to lubricate and cool the test bearing.

6.1 Emergency Lubrication Tests - Type I

The initial testing on the program was performed to establish a baseline for the program and evaluate two approaches of applying lubrication to the test bearing after cessation of oil from the primary system. The goal was to obtain 30 minutes of emergency operation at a bearing speed of 38,000 rpm (1.75 x 10^6 DN) and a thrust load of 1779 newtons (400 lb.). These conditions are representative of those experienced by contemporary helicopter engine mainshaft bearings. All tests were performed with an oil meeting MIL-L-23699 specifications.

The first approach, which has the advantage of being readily incorporated into many engine bearing mounting configurations, was an attempt to utilize the residual oil adhering to the shaft to extend the lost lubricant operation period. To accomplish this, a tapered sleeve was incorporated on the shaft adjoining the bearing inner ring on the oil injection side. The major OD of the sleeve taper was mated flush with the corner chamfer on the inner ring such that oil migrating up the sleeve, due to centrifugal force, would continue to flow along the chamfer. The oil reaching the inner corner of the chamfer would continue to migrate into the inner race due to adhesion forces or be dispersed radially outward onto the bore of the cage. This latter surface was also tapered to permit centrifugal pumping of the oil into the ball pockets where it could be distributed to the contact surfaces.

The second approach was the incorporation of an atomizing nozzle connected through a special mounting fixture to permit the residual oil in the primary recirculating oil system manifold following shut-off to be aspirated and sprayed directly into the test bearing.

6.1.1 Baseline Tests

Several tests were performed to establish a baseline value of the time between oil cessation and imminent bearing failure without supplying emergency lubrication. A bearing failure was considered to be imminent when the bearing outer ring temperature reached 617°K (650°F) or the speed appreciably decreased with the air flow to the drive turbine held constant. The high sensitivity of shaft speed to bearing drag torque would generally permit reinstating the lubricant flow or terminating the test before appreciable bearing damage occurred. Thus, the location of the failure initiation area could be determined.

The first two baseline runs (Test 1, Run 1 and 2) were performed with recirculating lubrication. In run 1, the oil was supplied at an inlet temperature of 460°K (370°F) and a flow rate of 757 cc/min. (0.2 gpm). The rig housing heaters were energized to heat the housing to 505°K (450°F). The shaft was accelerated to speed and the bearing outer ring temperature stabilized at 510°K (460°F). Following oil cessation, the bearing continued to operate for 6.9 minutes before the outer ring temperature reached 617°K (650°F). The inner ring temperature, measured with an optical pyrometer after cessation of oil flow, increased from 516° to 624°K (470 to 665°F). The second run was performed with an oil inlet temperature 422°K (300°F) without the rig housing heaters energized. The bearing temperature stabilized at 444°K (340°F) at the test speed prior to oil shut off. The bearing ran for 5.8 minutes before a failure was indicated by a speed decrease. The inner ring temperature increased more rapidly than the outer ring with a total difference of 50°K (90°F) present at the time of failure. The inspection of the bearings following the tests showed that the bearing used in the first run had not seized but appreciable wear had occurred between the cage rails and the outer ring guide lands. The bearing used in run 2 had experienced a thermal imbalance failure or binding due to the unequal thermal growth between the inner and outer rings.

In general, the baseline tests with recirculating oil lubrication resulted in a longer operating period after oil cessation than was expected. However, the relatively uniform temperature change of the inner and outer ring, allowing the bearing radial clearance to remain fairly constant, was consistant with the extended period of operation. Further consideration suggested that the oil was not being scavenged properly from the bearing chamber, ie. oil was building up around the bearing and was continuing to lubricate the bearing for a period following oil cessation. A scavenge pump was therefore added in the return line prior to the conduct of additional lost and emergency lubricant testing.

The second two baseline test runs (Test 2, Run 1 and 2) were performed with mist oil being the primary lubrication system. In the first run, since it was the initial attempt to operate the test bearing with the mist oil and cooling air system, the primary goal was to obtain temperature stabilization at test speed and determine if sufficient lubrication was being supplied. Thus, the tests were terminated for bearing inspection without oil cessation.

Utilizing the values obtained in the design of the mist and cooling air system analysis as a guide, a total of 0.83 scmm (29.3scfm) mist and cooling air was supplied to the bearing. A flow of 0.12 scmm (4.4 scfm) of preheated air was passed through the mist generator and entered the rig at a temperature of

378°K (220°F). The mist oil was preheated to 366°K (200°F) and an average of 126cm^3/hr transferred by the mist air to the bearing. The remaining 0.70scmm (24.9scfm) of air was supplied at 350°K (170°F) to the various cooling air paths: through bearing cooling air-0.31scmm (11scfm), bearing housing cooling air-0.18 scmm (6.3 scfm), shaft cooling air-0.21 scmm (7.6 scfm).

No problems were encountered in obtaining thermal stabilization at the test speed of 38,000 rpm. A test bearing outer ring temperature of 412°K (285°F) approximately 55°K above the average air inlet temperature was recorded. Inspection of the test bearing indicated good lubrication was obtained during the test.

In the second run (Test 2 Run 2) after thermal stabilization was obtained at 38,000, the mist oil was shutoff while maintaining the mist air and cooling air flows. An imminent bearing failure, indicated by a shaft speed decrease, occurred after 1.6minutes following the mist oil cessation.

6.1.2 Utilization of Residual Oil on Shaft

With the modification described in Section 6.1, emergency lubricant tests were performed to determine if residual oil on the shaft could be utilized to extend the bearing life following oil loss. A total of three test runs (Test 3, Run 1, 2 and 3) were performed with recirculating oil as the primary system and one (Test 4) performed with mist oil.

The runs with the recirculating oil system were performed under conditions similar to those used in the second baseline run with the exception that a scavenge pump was attached to the drain line. By reinitiating the oil flow when a failure was indicated by a decrease in shaft speed a total of seven different checks were performed without any catastrophic bearing failures. During Test 3, Run 1 three checks were performed with the inlet oil temperature at 455°K (360°F) and a bearing outer ring temperature of approximately 505°K (450°F). In the first two checks, indications of test bearing problems occurred within 20 seconds. During the third check a bearing problem was observed 40 seconds after the oil was shut off.

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It was reasoned that the appreciably shorted period of operation, compared to the second baseline test, may have resulted partially from higher oil inlet temperature in addition to the fact that the oil was being scavenged more effectively. Thus, the second and third runs were performed with oil inlet temperatures maintained at 432°K (320°F) and 442°K (335°F) respectively. In the second run, two checks were performed with the bearing temperature at approximately 488°K (420°F). Bearing problems occurred after 40 seconds of operation in both cases. An inspection of the test bearing following Run 2 indicated that the problem, increased drag torque, resulted from a binding between the cage and the guide lands on the outer ring. This condition was considered to have occurred due to a thermal growth of the cage resulting from the increase in the heat generation rate produced by the increased friction coefficient following the loss of lubrication. In run 3, the same results were obtained except the failures occurred within 26 seconds in the two checks performed.

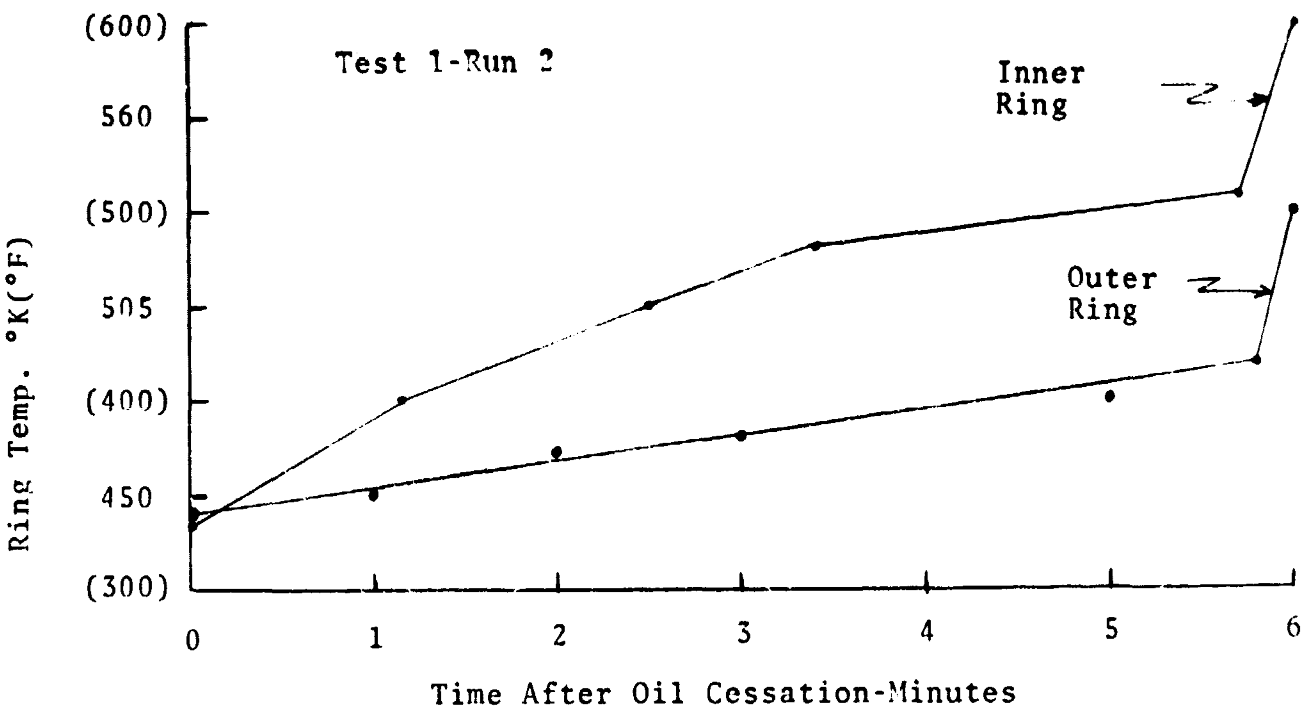
Plots of the inner and outer ring temperature following oil cessation in Test 1, Run 2 and Test 3, Run 2 are presented in Figure 13. The plot of Test 1, Run 2 data shows that the inner ring temperature increased more rapidly than the outer producing the thermal imbalance failure between the inner and outer races. The plot of Test 3 Run 2 data shows that the two rings increased in temperature essentially uniformly for 30 seconds and then the outer ring temperature increased rapidly indicating the time when the cage started to bind. The increase in inner ring temperature shortly thereafter indicates the time when the heat generated by the increasd sliding of the balls on the inner ring occurred due to spin velocity decrease resulting from driving the binding cage.

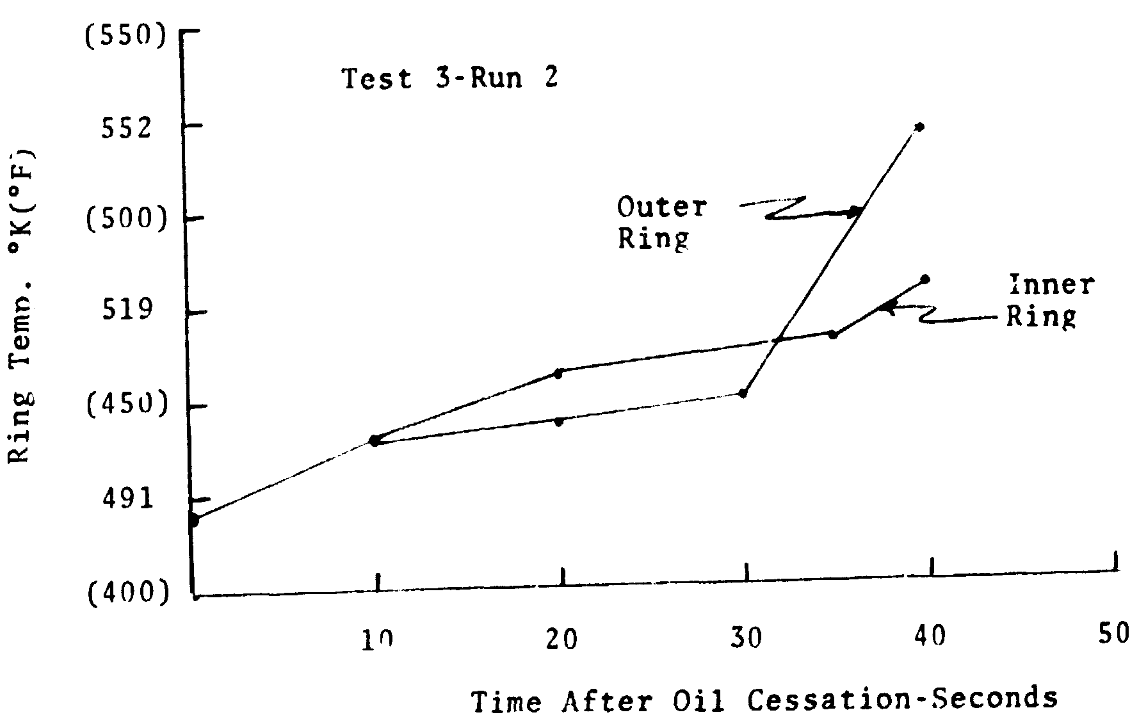
The test performed with mist lubrication and the modifications incorporated to pump residual oil from the shaft into the bearing was carried out in a similar manner to the second baseline test (Test 2, Run 2). Following mist oil cessation, with the mist air and through bearing cooling air retained, the bearing ran for 2.6 minutes before indications of bearing problems.

A review of the test data resulting from the first two sets of tests performed (Test 1 through Test 4) indicated that the relatively long period of operation obtained during the baseline tests with recirculating oil (Test 1) results from an oil buildup

FIGURE 13

BEARING TEMPERATURE AFTER OIL CESSATION





in the bearing chamber which continued to lubricate and cool the bearing until it drained. The short time periods obtained in Test 3 definitely indicated that the scheme to pump residual oil from the shaft into the bearing was unsuccessful or the residual oil was inadequate to produce a significant effect on bearing life. Therefore, the operating period following cessation of oil in Test 3 is considered to represent the time before a bearing problem starts when recirculating oil is lost and the bearing sump is well scavenged under the test conditions and rig configuration employed.

The slight extension of operation obtained when mist oil was used as the primary system of lubrication is attributed to the cooling supplied by the mist and cooling air which was continued after oil shut off. However, the extension was insignificant in light of the desired goal. An increase in the cooling air flow rate was not evaluated as it was considered to have little potential in adding greatly to the bearing life.

6.1.3 Emergency Lubrication by Aspirated Residual Oil

The purpose of this testing was to determine the feasibility of obtaining the desired goal of 30 minutes of operation following recirculation oil cessation by aspirating the oil retained in the lubricant reservoir and injecting it into the bearing as mist. To accomplish these tests, an atomizing tip (standard tip from a De Vilbiss 127 atomizer) was incorporated in conjuction with an air supply line to aspirate, atomize, and direct the residual oil into the test bearing. The tip was assembled to the oil manifold, see Figure 5 , by a special connector.

Preliminary checks were performed with the aspirator system prior to the emergency lubrication testing to determine the air pressure and flow rate required to aspirate and atomize the oil. It was determined that an air supply pressure of 40 psi easily aspirated and atomized the oil into a fine mist. The air flow rate produced was 0.024 scmm (0.84 scfm) and the residual oil of 10cc (0.61 in^3) retained in the reservoir below the lowest jet nozzle was depleted in 2.5 minutes; thus producing an oil flow rate of 239cc/hr. (14.6 in^3/hr.). Due to the quick depletion of the residual oil, which was limited bythe oil manifold configuration, the oil in the reservoir was replenished every 2.5 minutes. In this manner, continuous aspirator operation could be obtained.

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Two emergency lubrication tests were performed (Test 5 and 6) using the aspirator system. Both tests were initiated under conditions similar to those in the prior emergency lubrication tests with the exceptions that a standard bearing, no modification to the inner ring chamfer or cage bore was used and both the recirculating oil and aspirator systems were employed at the initiation of the test. Room temperature air was used to activate the aspirator.

After the shaft speed of 38,000 rpm was obtained in Test 5, with an oil inlet temperature and flow rate of 438°K (330°F) and 757 cc/min. (0.2 gpm) respectively the outer ring temperature stabilized at 508°K (455°F). The recirculating oil was then shut off and operation continued with the aspirated oil only. After 2.5 minutes, the aspirator reservoir was refilled while operation continued. The refilling cycle was repeated each 2.5 minutes until a total time of 30 minutes of operation was obtained after cessation of the recirculating oil.

During the period of emergency lubrication there was no indication of improper operation of the test bearing which suggested that even a longer period of operation could have been obtained. A maximum outer ring temperature increase of only 19°K (35°F) was recorded which was considerably lower than anticipated.

To insure that no oil was reaching the test bearing from the rig bearing supply and the test bearing chamber was being properly scavenged four additional checks were performed. In the first two checks, the aspirator was used without replenishing the reservoir supply. In both checks, shaft deceleration was observed in approximately 2.5 minutes after recirculating oil shut off and the bearing outer ring temperature increased to 590°F before recirculation of the oil was reestablished. In the second two checks, the aspirator system was not activated and in both cases bearing problems were encountered after 40 seconds. These four checks thus provided further assurance of the effectiveness of the aspirator.

In the second test (Test 6) utilizing the aspirator system, cooling air was supplied at 350°K (170°F) through the shaft and bearing housing at a rate of 0.28 and 0.11 scmm (10 and 4 scfm) respectively. The cooling air supply was initiated approximately one minute after the cessation of the recirculating oil. Shortly thereafter, bearing problems were encountered as evidenced by a rapid temperature increase and speed decrease. The recirculating oil was reinstated with the cooling air retained. Four additional checks were performed by turning off the recirculating oil. In all cases, bearing problems resulted between 1.6 minutes and 3 minutes. Oil was replenished in the aspirator reservoir when the operating period dictated.

These results indicated that the cooling air supplied to the shaft and bearing housing at the rates used was detrimental to the bearing performance. To further varify this assumption, an additional check was performed without the cooling air. After ten minutes of operation in this mode (replenishing the aspirator oil every 2.5 minutes) with no indication of bearing problems the test was terminated having concluded that the assumption was correct. The bearing outer ring temperature stabilized at 572°K (570°F) during this check which was more reasonable and indicated the measurement in run one was erroneous.

It is concluded from the results of these two tests that it is feasible to appreciably extend the operating life, up to more than 30 minutes, of helicopter engine or transmission bearings following the loss of oil from the primary system by aspirating a small quantity of oil from a reservoir and injecting it into the bearing in the form of mist. It is also concluded that the use of cooling air, at rates applied in Test 6 through the shaft and bearing housing, is detrimental in extending the bearing life. Changes in the cooling air flow rates or location of the cooling air could be beneficial; however, this was not verified by testing.

In general, the lost lubricant and emergency tests showed that bearing failure can be expected in approximately two minutes or less when no emergency lubrication is supplied. With an aspirator type system used to supply emergency lubrication, operation periods of 30 minutes or greater can be obtained.

6.2 Step Speed Tests - Type II

The first set of step speed tests (Test 7-10) were performed with four different lubricants to obtain a relative evaluation of their performance when used in a mist lubrication system. Three of the lubricants, described in Section 2, meet specific military specifications and all four are presently being used in recirculating oil systems to lubricate bearings in commercial or military gas turbine engines. It should be noted that these four tests were also an evaluation of the initial design of the mist and cooling air systems. Prior to performing the tests, the intent was to evaluate the lubricant performance by comparing the bearing contacting surfaces after testing was complete. However, since bearing failures were encountered which were not necessarily related to the lubricant used, no attempt was made to rank the performance of the oils.

The second set of step-speed tests (Test 11-13) was performed to establish the effects of changes in the controlled variables and the third set of tests (Test 14-16) was performed to evaluate basic changes in the system configuration. During many of these step speed tests, two basic problems were encountered (cage instability and excessive cage rail wear and binding between the downstream rail and guide land on the outer ring) at speeds between 53,000 to 65,000 rpm (2.5×10^6 to 3.0×10^6 DN) Thus, the fourth set of step speed tests (Test 17-20) were performed with different modificatons to the cage in an attempt to eliminate these problems.

6.2.1 Lubricant Evaluation Tests

Standard bearings (SKF 464539VAA) with the modifications described in Section 2.1 and the removal of the ball retaining tangs to permit easy disassembly of the bearing for inspection purposes were used in all lubricant evaluation tests. A bearing load of 1779 newtons (400 lb.) thrust force was used in all testing.

The first test (Test 7) was performed with a Type II Ester meeting MIL-L-23699. The shaft was accelerated to 26,000 rpm and then 40,000 rpm and data recorded. At this speed the cooling air flow rates through the shaft, circumferentially through the bearing housing, and directly through the bearing were individually changed to establish their effect on the bearing temperature and behavior. It was found that little or no immediate effect was produced by the shaft or housing cooling air, but the through bearing cooling air had an appreciable effect on the bearing outer ring temperature. No reliable inner ring temperature values could be measured due to interference of the pyrometer line-of-sight by the oil mist. However, it was assumed that the through bearing cooling air had the same or greater effect on the inner ring temperature since it was directed to impinge on or pass across the inner ring surface. The total air flow was then set at 0.439 scmm (15.3 scfm) (mist air-9.7 scfm and through bearing cooling air-5.6 scfm) and the shaft speed increased in steps to a maximum speed of 65,000 rpm (3×10^6 DN). Since the bearing outer ring temperature did not exceed the pre-established maximum limit of 533°K (500°F) no further increase in air flow was made. After operating at 65,000 rpm for approximately 30 minutes without the bearing temperature completely stabilizing, the temperature suddenly increased and the shaft decelerated terminating the test. From a speed of 55,000 to 65,000 an intermittent noise, interpreted as being produced by cage instability, was observed. During the test, a mist oil flow rate of 492 cc/hr. (31 in^3/hr) was supplied to the bearing. The inspection of the test bearing showed that the cause of failure had resulted from excessive rubbing between the cage rail and the outer ring guide land on the downstream side of the bearing. Test data, including the calculated heat transferred from the bearing to the mist and cooling air, is presented for this and subsequent tests in Appendix III.

Based on the good results obtained in this test, it was decided that the housing and shaft cooling air would not be used in any subsequent tests. The elimination of two of the cooling air flow paths greatly simplified the cooling air system and thus increased the number of possible applications of a mist and cooling air system by including those where housing and shaft cooling could not be incorporated.

Based on the good results obtained in this test, operations up to 3 x 10^6 DN with appreciably less air than theoretically determined necessary without the bearing temperature reaching 533°K and the use of only through bearing mist and cooling air without a thermal imbalance between the rings producing a failure, it was decided that the housing and shaft cooling air would not be used in any subsequent tests. The elimination of two of the cooling air flow paths greatly simplified the cooling air system initially envisioned as being necessary and thus increased the number of possible applications of a mist and cooling air system by including those where housing and shaft cooling could not be incorporated.

The next three tests (Test 8-10) were performed with a Type I Ester meeting MIL-L-7808H specifications, a second Type I Ester which was refined in a foreign country, and a polyphenylether respectively. The tests were performed in the same manner as Test 7 with the exceptions that mist air flow rates were increased from 40 to 55 percent and moderate changes in the mist and through bearing cooling air temperature as noted in data presented in Appendix III.

In Test 8, a total of 0.543 scmm (19.2 scfm) of air was passed through the bearing with 0.385 scmm (13.6 scfm) being mist air. The average mist oil flow rate was 590 cc/hr. (36 in^3/hr.). A maximum speed of 60,000 rpm was obtained prior to a decrease in shaft speed indicating a bearing problem. In Test 9, a total air flow of 0.586 scmm (20.7 scfm) was supplied at the maximum speed obtained and an average mist oil flow rate of 492 cc/hr (30 in^3/hr.) was supplied. The desired speed of

65,000 was obtained and operation continued for 40 minutes before the test was terminated due to excessive noise. In Test 10, a similar air flow was supplied with a mist oil flow rate of 442 scmm (27 in^3/min). A maximum speed of 50,000 rpm was obtained before a sudden temperature excursion occurred and the shaft decelerated terminating the test.

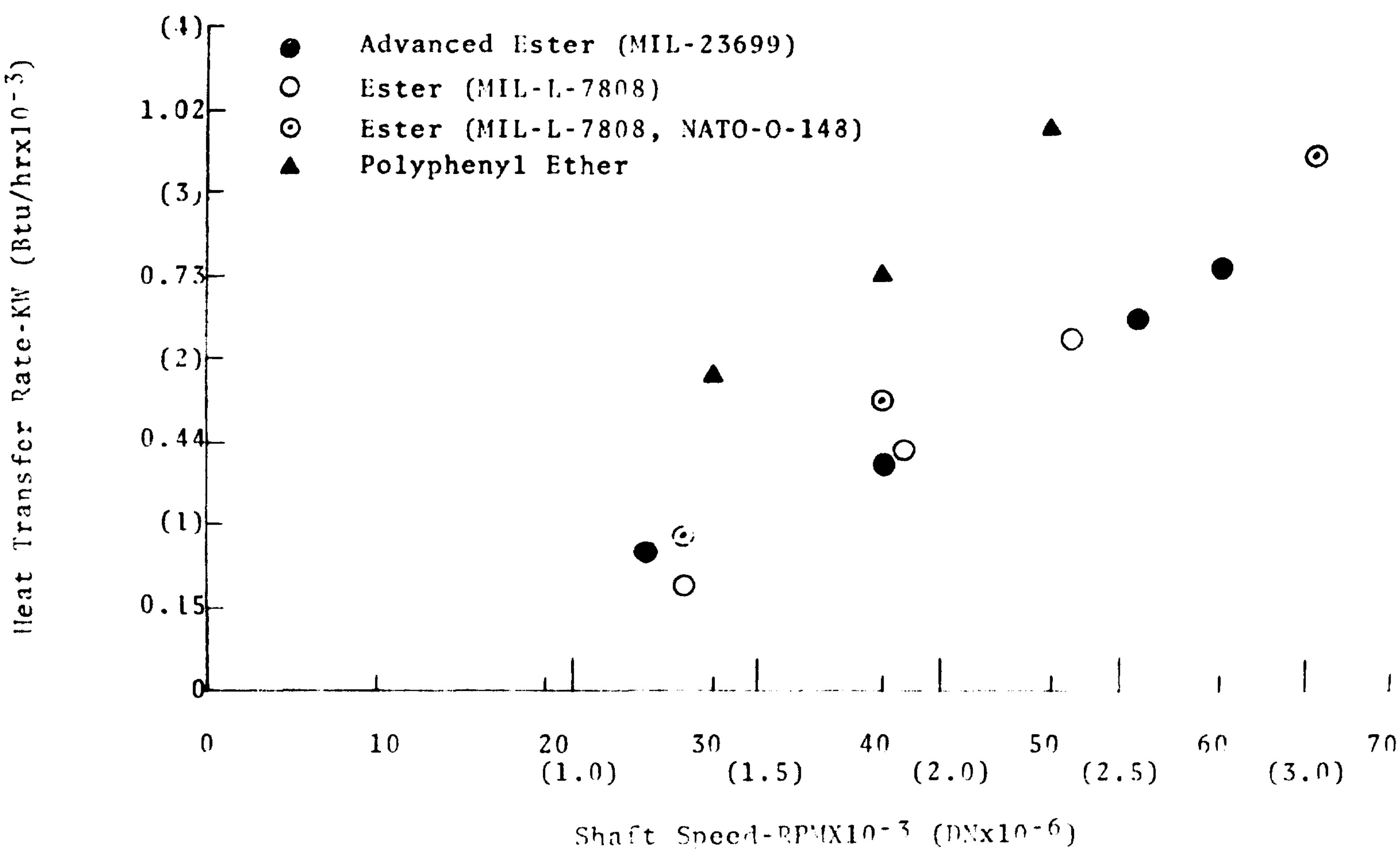
In each of the three tests, there was indication of cage instability noted by an intermittent rattle starting at speeds from approximately 50,000 to 55,000 rpm. Inspection of the test bearings indicated that the cause of the bearing failures encountered were produced by excessive friction between the downstream cage rail and the outer ring guide land which suggested that insufficient lubrication was present at this location. This condition was diagnosed from the excessive wear on the rail and land, and the heavy wear on the leading face of the ball pockets caused by the balls trying to drive the cage.

The results of the mist lubricant evaluation indicated that any of the four lubricants tested could be utilized with a mist and cooling air system to provide adequate lubrication of helicopter engine mainshaft bearings operating at speeds up to 2×10^6 DN. Although there were differences in the maximum speeds obtained with the different lubricants before problems were encountered, these differences are not necessarily a function of the lubricating qualities of the oils. The problems were considered to be the result of inadequate lubrication and cooling to the downstream side of the cage. The cage instability, which is a function of the cage configuration and mass, and damping, occurred in the mist lubricated bearing and not the recirculating jet lubricated bearing due to the decreased oil damping provided by the mist system.

A plot of the heat transferred from the bearing to the air in each test is presented in Figure 14. This plot shows that the bearing heat generation rate was quite similar during the testing of three of the lubricants. However, an appreciable difference (approximately 60 percent higher) was obtained with the polyphenylether. This difference is considered, at least in part, to result from the higher viscosity.

FIGURE 14

LUBRICANT EVALUATION TESTS (7-10)
HEAT TRANSFER RATE TO MIST AND COOLING AIR



The MIL-L-23699 lubricant was selected for the remaining bearing tests. This selection was based on its good performance in the early tests and its widespread use in military aircraft engines in this country.

6.2.2 Changes In Controlled Variable Tests

The purpose of the controlled variable tests was to determine the possibility of operating under more severe conditions with respect to the mist oil flow rate, air flow rate, and air temperature. All tests were performed with MIL-L-23699 oil, and mist and through bearing cooling air only. Test 11 was performed to evaluate the possibility of operating with appreciably less mist oil supplied to the bearing. Tests 12 and 13 were performed to evaluate the effects of operating with a reduced air flow rate and a higher average air temperature respectively.

Prior to Test 11, an attempt was made to calibrate the mist generator. For the mist generator being used, the manufacturer does not guarantee mist generation for adjustment screw setting below 180° open when 294°K (70°F) temperature air is utilized. At this opening, the oil flow was greater than the minimum value at which testing was desired since the air was being supplied to the mist generator at a temperature of 478°K (400°F). Further checks at screw settings less than 180° produced inconsistencies. Thus, the calibration could only be used as a guideline and the actual oil supply rate established after the test by measuring the quantity of oil used.

A total of four runs were performed in Test 11, each for a duration of approximately 4 hours. The first two runs were performed with the same oil flow rates, 328 cc/hr. (20 in^3/hr.) or approximately 66 percent of what was supplied in the lubrication evaluation tests. The total air flow rate in both runs was 0.458 scmm (16.2 scfm), 0.260 scmm (9.2 scfm) as mist air and 0.198 scamm (7 scfm) as cooling air, at an average temperature of approximately 350^{o}K (172^{o}F). In both runs, the maximum desired speed of 65,000 was obtained with the only problem observed being an intermittent noise (rattle) interpreted to be cage instability, at speeds from 60,000 to 65,000 rpm. A cursory inspection of the test bearing following the second run indicated that the bearing was in excellent condition.

In run 3, the mist flow adjustment screw was opened only 80 degrees. The mist and cooling air flow rates were set at the same values as in the prior runs and the average air temperature was 355°K (178°F). The shaft was accelerated to 65,000 rpm in steps. No problems were encountered except for intermittent noise at the higher speeds. The calculated oil flow rate was only 51 cc/hr. (3.1 in. ^{3}/hr.) or approximately 0.11 pints/hr.

Run 4 was essentially a repeat of run 3 with the exception that the mist generator oil flow adjustment screw was opened only 60 degrees. At the 60° opening, the first visual indication of mist flow was observed at the rig exhaust port. After temperature stabilization at 55,000 rpm, the shaft was accelerated to 60,000 rpm at which point the test was terminated following a speed decrease and a rapid increase in the test bearing temperature.

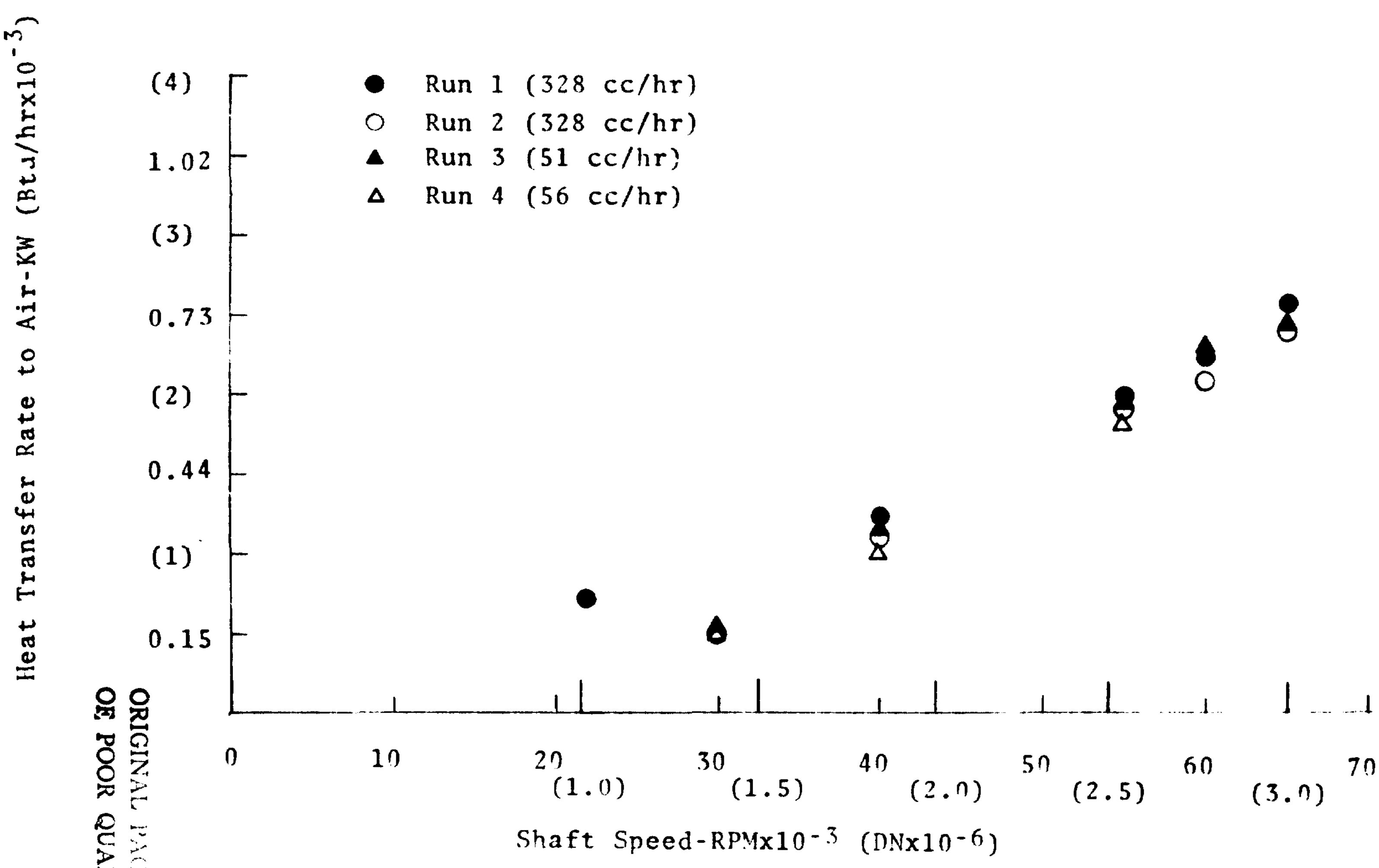
Without any changes in the mist and cooling air flow rates, following the shaft deceleration, no mist could be visually detected at the exhaust port. Although this did not definitely indicate that the mist had stopped flowing, it did indicate that the flow had decreased from the initiation of the test. Measurement of the oil used during the test showed that the average flow rate was 56 cc/hr. (3.4 in^3/hr.) which indicated that the flow rate was fluctuating. A plot of the heat transferred from the bearing to the air is presently in Figure 15 for all runs. This graph shows that no appreciable change resulted from varying the oil flow rate.

The inspection of the test bearing showed that the problems resulted from excessive friction between the cage rail and outer ring guide land on the downstream side of the bearing. Since this type of failure had occurred in prior tests when the mist oil flow rate was ten times as great, it was thus reasoned that the failure was not the result of the lower flow rate, but the distribution of the oil in the bearing. It was, therefore, concluded that the same quality of lubrication was obtained with the much reduced oil flow rate and that adequate lubrication was provided with as little as 51 cc/hr (3.1 in^3/hr.) to permit operation to speeds as high as 2.5 x 10^6 DN.

In Test 12, the reduced air flow rate test, the testing was initiated with oil and air flow rates and temperatures similar to those used in the lubricant evaluation tests. After allowing temperatures to stabilize at a speed of 30,000 rpm, the cooling air flow rate was slowly decreased to zero without any bearing problems produced. With a mist air flow rate of 0.283 scmm (10 scfm)

FIGURE 15

TEST II REDUCED OIL FLOW RATE
HEAT TRANSFER RATE TO MIST AND COOLING AIR
LUBRICANT-ADVANCED ESTER



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and inlet temperature of 359°K (185°F) the speed was increased to 65,000 rpm in steps. Before temperature stabilization at 65,000 rpm, a rapid increase in the bearing temperature occurred and the test terminated. The average oil flow rate during the test was 442 cc/hr. (27 in^3/hr.). The bearing inspection revealed that smearing had occurred between the downstream cage rail and outer ring land as in prior tests.

The results of the test showed that 0.283 scmm of air would provide adequate, uniform cooling of all bearing components except at the highest test speed where problems also resulted at much higher air flow rates. Figures 16 and 17 show the bearing outer ring temperature and heat transfer rate to the mist and cooling air respectively, compared with that obtained in run 1 of Test 11 where the higher quantity of cooling air was supplied. The graph shows that at speeds of 30,000 to 40,000 rpm, the increase in bearing temperature resulting from the lower air flow was approximately 22°K (40°F). At the higher speeds, 55,000 to 60,000 rpm, the temperature difference had increased, as would be expected, to approximately 44°K (80°F). However, even with the lower flow rate, the temperature was still well below that where property changes in the bearing material would effect the endurance life.

Test 13 was performed with a higher average air temperature, approximately 388°K (238°F), entering the rig to cool the bearing. The through bearing cooling air was supplied at 402°K (265°F) and the mist air at 349°K (170°F). To obtain the high average total air temperature, at approximately the same flow rate as used in Test 11, the through bearing cooling air flow rate was increased from 0.198 to 0.359 scmm (7 to 12.7 scfm) and the mist air decreased from 0.260 to 141 scmm (9.2 to 5 scfm). The decrease in the mist air flow rate resulted in an average mist oil supply rate of 236 cc/hr. (14.4 in^3/hr.) which was approximately half that supplied in Test 12, but well above that found to be adequate in Test 11.

A speed of 60,000 rpm was obtained in incremental steps without problems. However, a failure occurred shortly after reaching a speed of 65,000 rpm. The inspection of the test bearing showed that the failure was essentially the same as that which had occurred in Test 12, gross wear between the downstream cage rail and outer ring guiding land.

FIGURE 16

TEST 12-DECREASED COOLING AIR FLOW RATE
TEST BEARING OUTER RING TEMPERATURE
LUBRICANT-ADVANCED ESTER

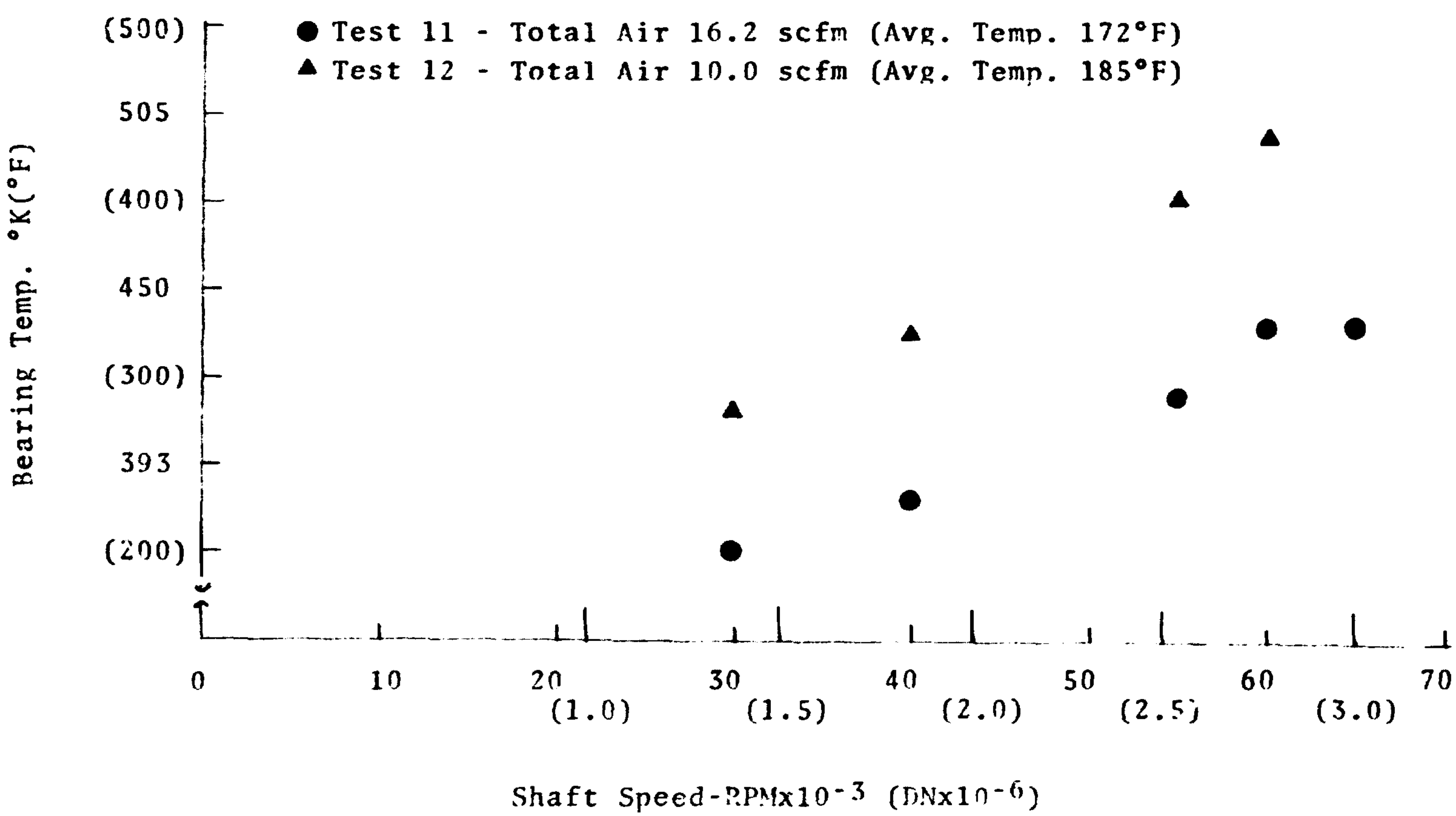
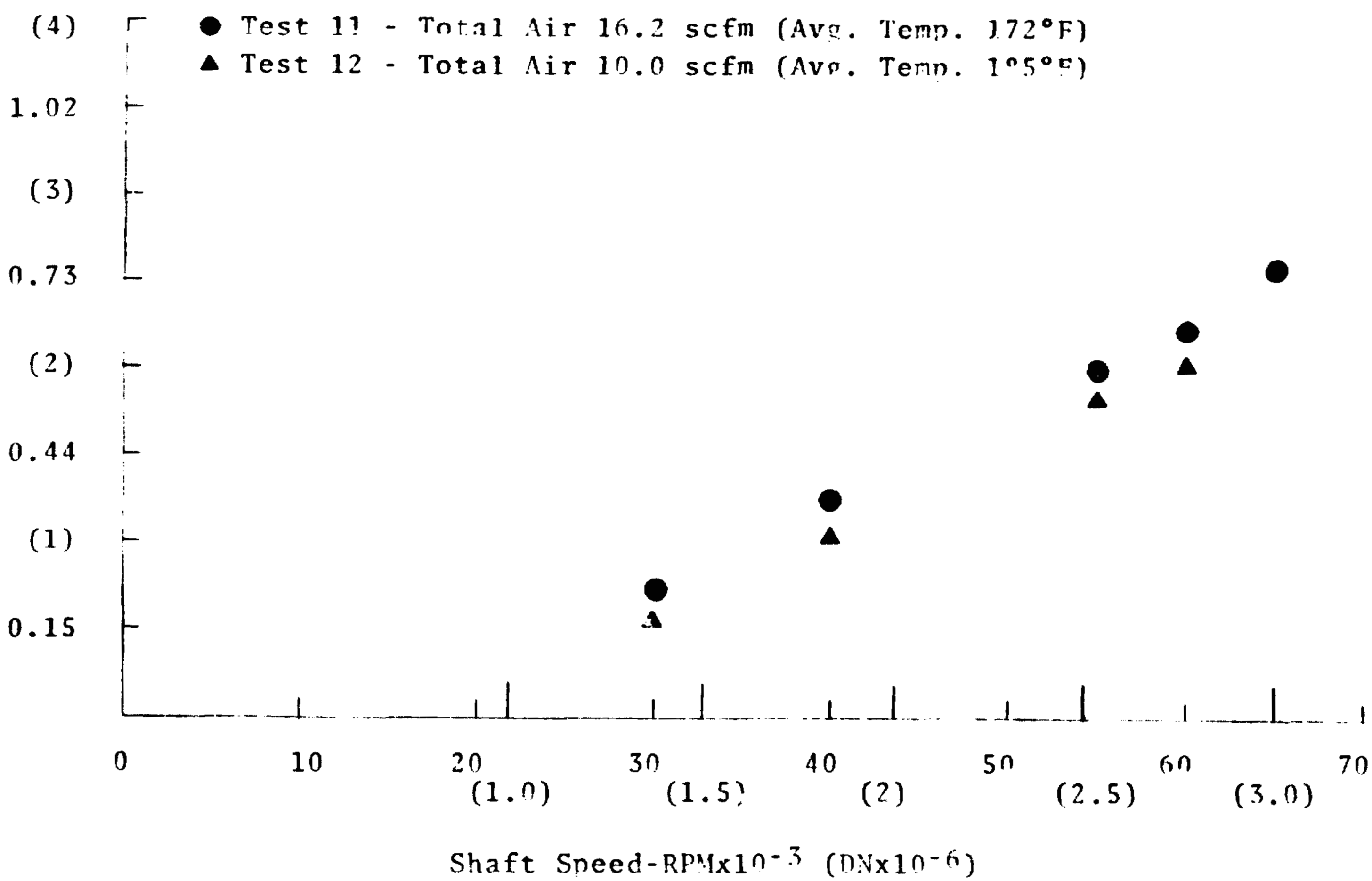


FIGURE 17

TEST 12-DECREASED COOLING AIR FLOW RATE
HEAT TRANSFER RATE TO AIR
LUBRICANT-ADVANCED ESTER



The test bearing outer ring temperature and the heat transfer rates at the various test speeds are presented in Figures 18 and 19 , respectively, along with the data from Tests 11 and 12. As would be expected at lower speeds the test featuring a higher cooling air temperature results in higher bearing outer ring temperatures than the test with a decreased flow rate. However, at higher speeds where considerably more heat is generated by the bearing, the effect of air flow volume on the bearing temperature is larger than the effect of air temperature. Although bearing failures occurred at high speeds in both tests, (the increased bearing temperatures were measured prior to failure) the failures are not considered to be the result of the higher bearing temperatures as failures were typical of those in prior tests when the outer ring temperatures were considerably lower.

As shown in Figure 18 the increase in the air temperature of 39°K above that used in Test 11, resulted in approximately the same temperature increase in the test bearing over the full speed range. This is considered to be reasonable since the same Δt between the bearing and cooling air was maintained to remove approximately the same quantity of bearing generated heat.

6.2.3 Changes In the Mist System Configuration Tests

A total of three tests (Tests 14-16) were performed with modification to the mist system. All tests were performed with the standard thrust load of 400 lbs. and MIL-L-23699 oil. The first two modifications were relatively simple and involved the test cavity configuration near the bearing or the mist nozzle configuration. The third change simplified the mist system by replacing the mist generator with a drip mist system.

In preparation for Test 14, a baffle plate was manufactured, see Figure 20, to be located on the upstream side of the bearing as shown in Figure 21. Two tabs were formed on the bearing outer ring flange to permit positioning and assembly of the plate. The bore of the baffle plate was chamfered at a 45° angle from the outer surface and a 0.152 mm (0.006 in) lip machined on the inner surface which extend into the bearing and reduced the gap formed with the cage face. The dimension of the bore were established to permit the high velocity cooling air emitted through the eight nozzles to flow directly across the chamfer and produce a pressure drop in the annulus formed between the cage and baffle plate.

FIGURE 18

TEST 13-INCREASED COOLING AIR TEMPERATURE
TEST BEARING OUTER-RING TEMPERATURE
LUBRICANT-ADVANCED ESTER

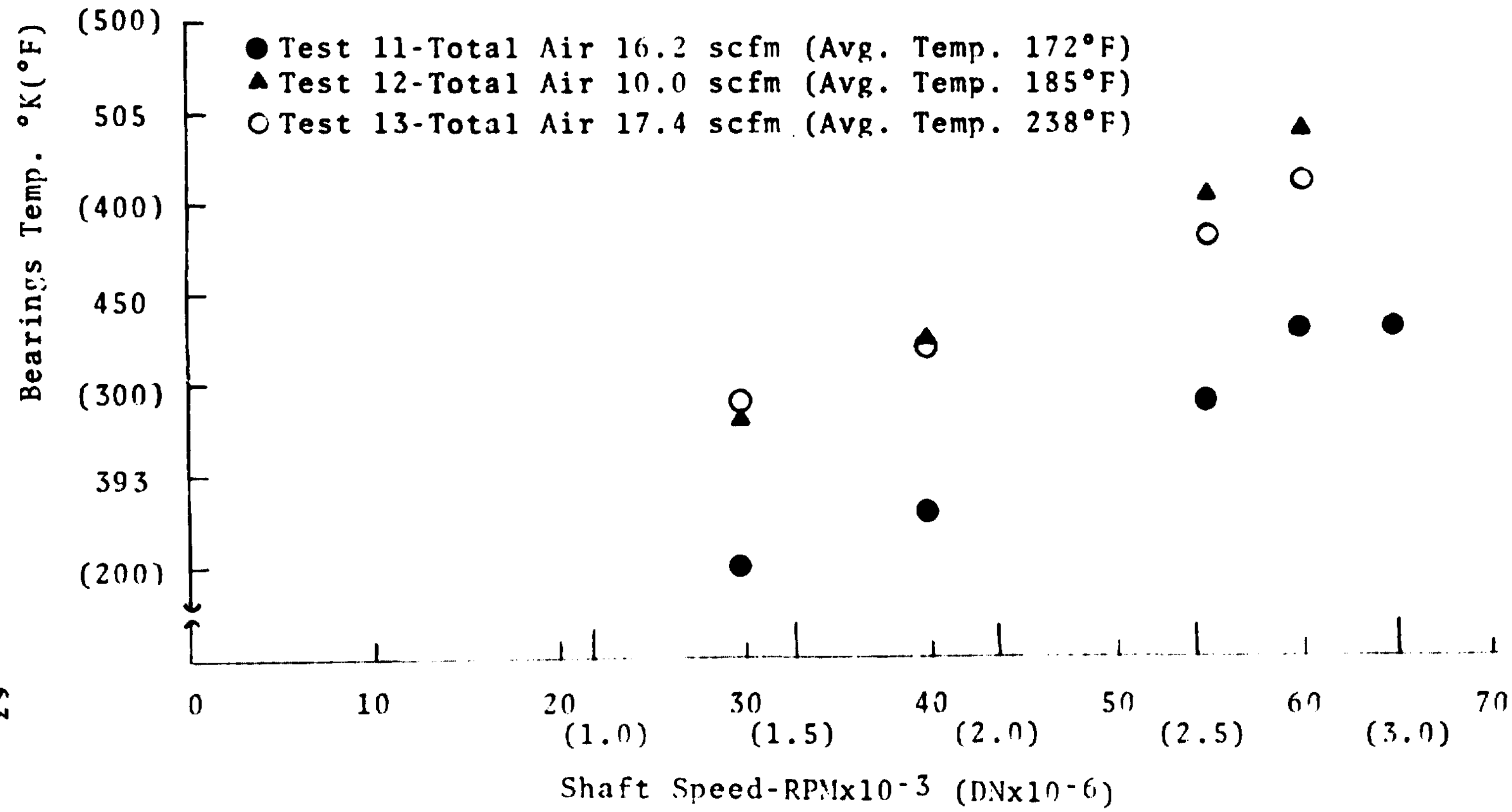


FIGURE 19

TEST 13-INCREASED COOLING AIR TEMPERATURE
HEAT TRANSFER RATE TO AIR
LUBRICANT-ADVANCED ESTER

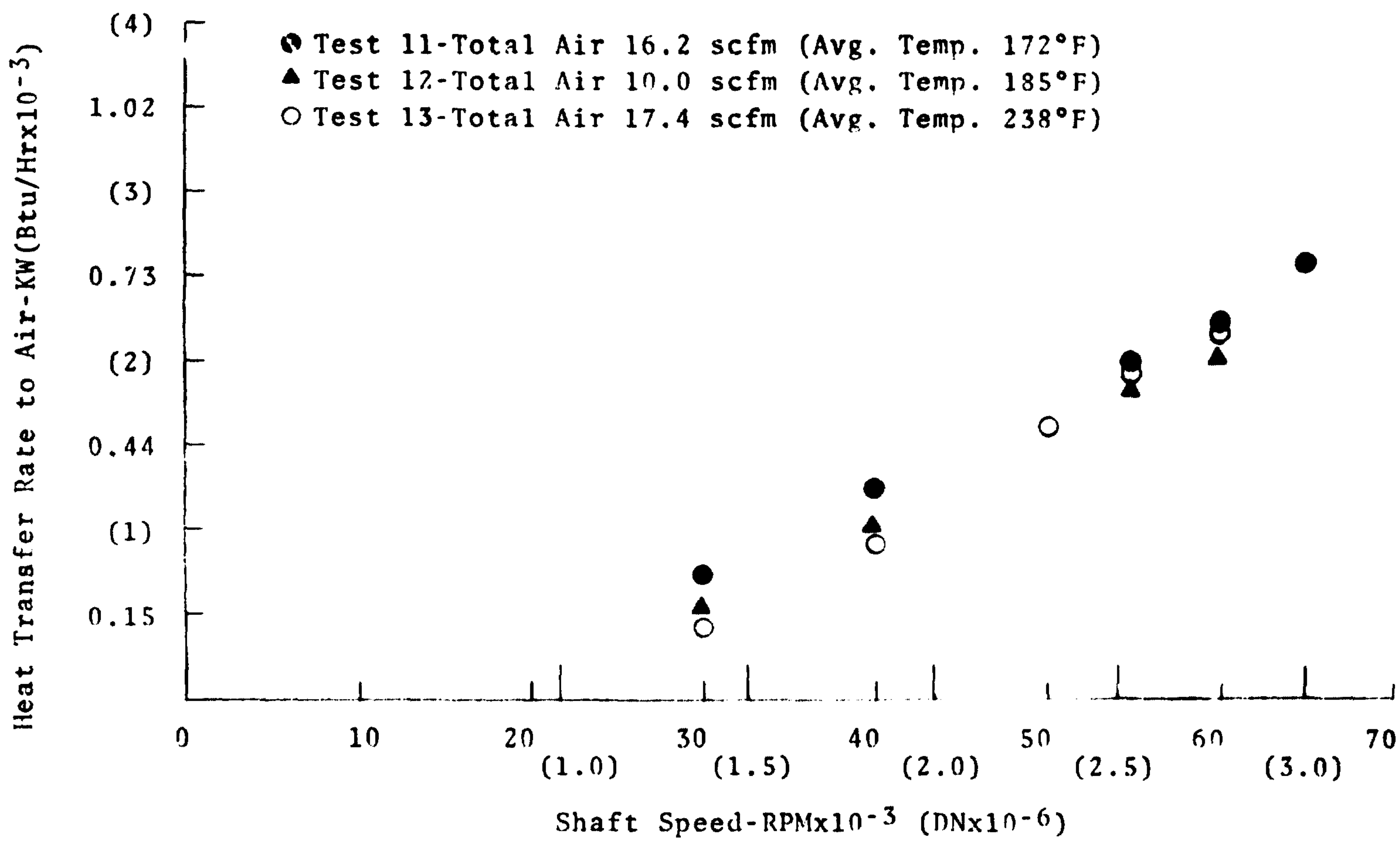


FIGURE 20

BAFFLE PLATE DRAWING

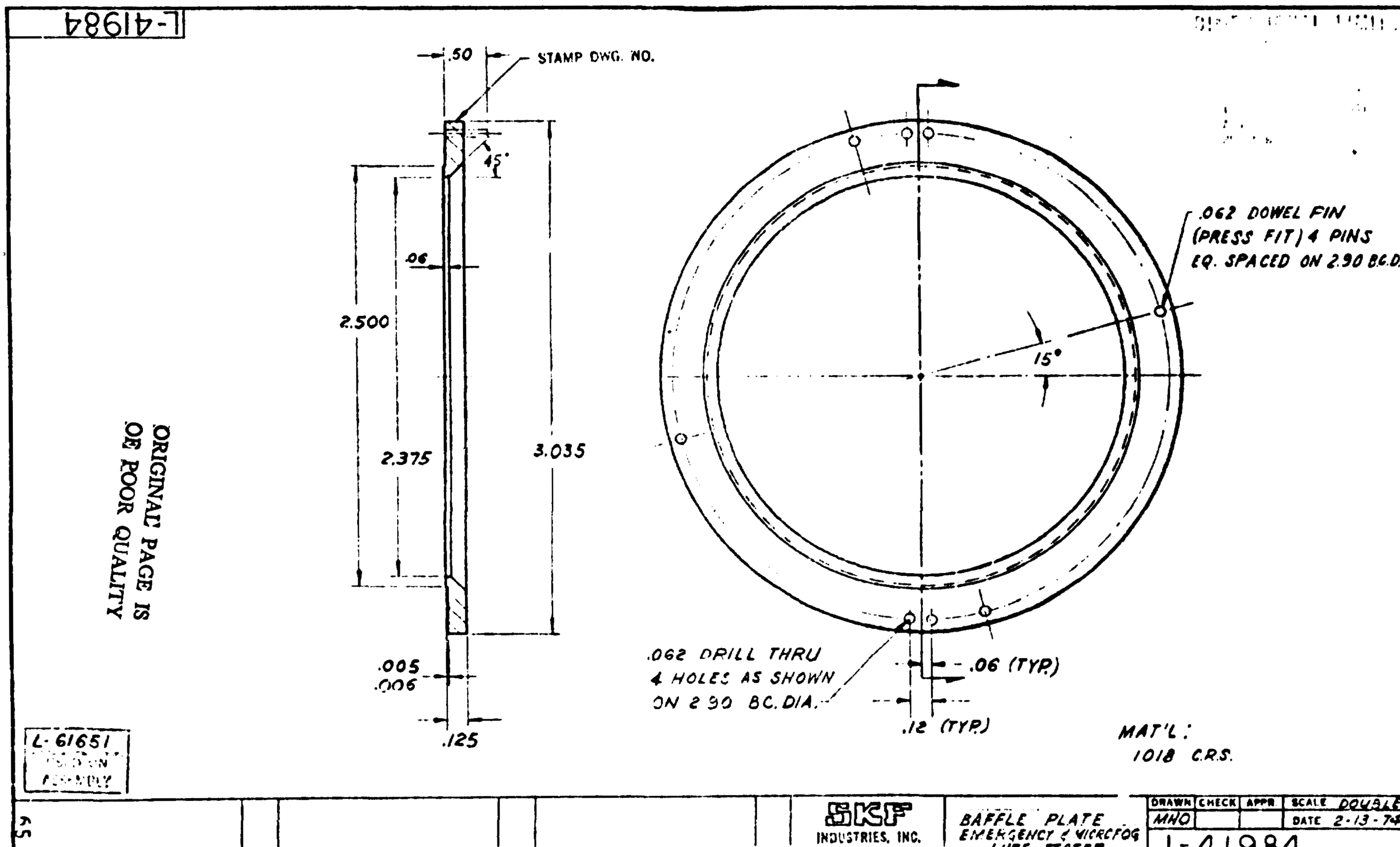
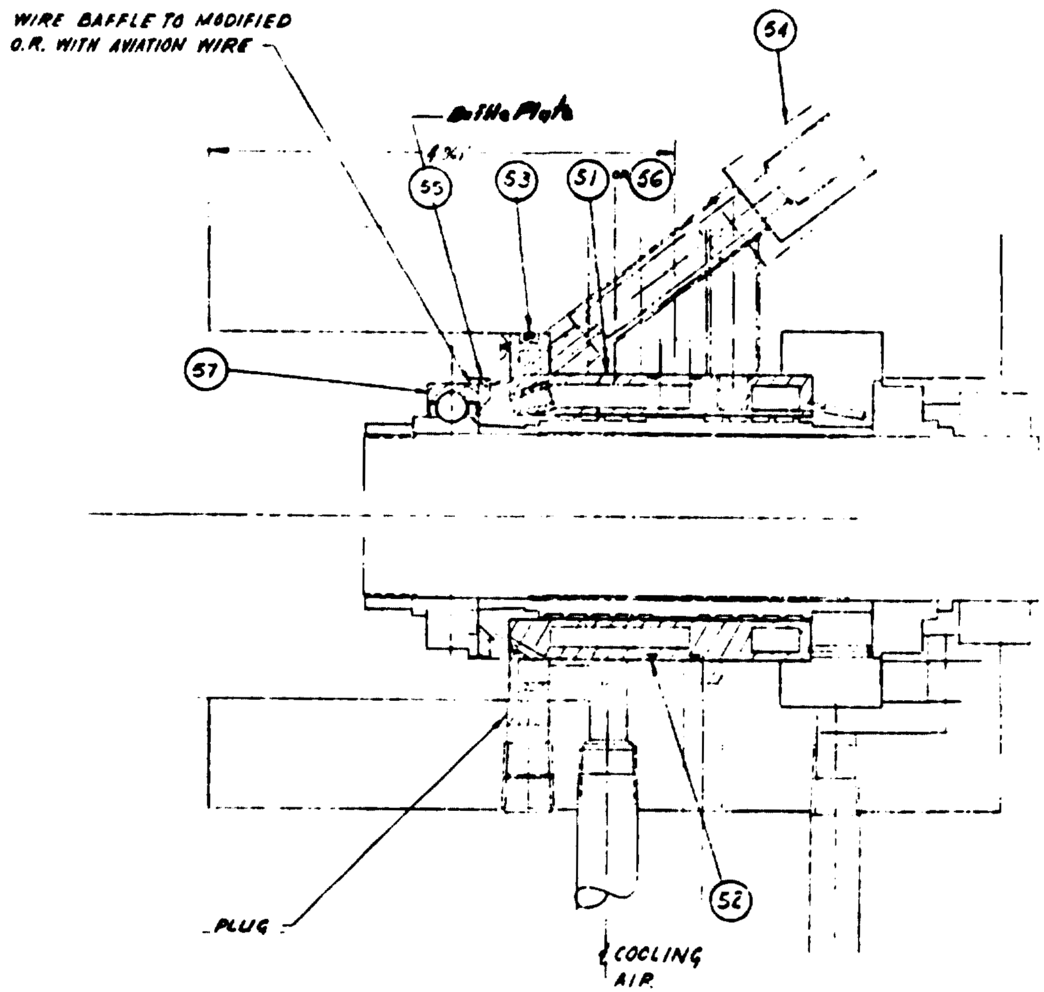


FIGURE 21

LAYOUT DRAWING OF RIG SHOWING
BAFFLE PLATE LOCATION



The incorporation of the baffle plate was aimed at improving the cooling and lubricant flow paths within the bearing by modifying the pressure distribution. It was hoped that this would provide better lubrication and cooling of the downstream cage rail and thus eliminate the excessive wear and binding which had occurred in several of the prior tests at speeds in the 3 x 10^6 DN range.

The test was performed in a manner similar to prior step-speed tests with a mist air flow rate of 0.321 scmm (11.34 scfm) and through bearing cooling air flow rate of 0.190 scmm (6.7 scfm). The average air temperature was 350°K (171°F). The mist oil flow rate was 451 cc/hr (27.5 in^3hr.). The maximum desired speed of 65,000 rpm was obtained and operation continued for 30 minutes before test termination. Intermittent noise, considered to be produced by cage vibration, was observed at both 60,000 and 65,000 rpm which had no measurable effect on the bearing performance.

The test bearing was in excellent condition following the test with minor polishing of both cage rails and slightly heavier polishing and wear in the ball pockets. A thin layer of degraded oil and fine wear particles of silver had collected on the cage bore which had not been present following any prior tests. This condition indicated that the mist and cooling air flow paths through the bearing had been altered.

Test 15 was performed to evaluate the effect of replacing the converging reclassifying nozzles with straight nozzles without screens at the entrance to reclassify (increase the oil particle size) oil droplets. The eight straight nozzles incorporated had an inner diameter equal to the minor diameter of the converging nozzles (2.0mm). The baffle plate was retained and the same bearing used to obtain additional test results with the altered flow paths in the bearing.

The test was successfully performed to a maximum speed of 65,000 rpm and operated at that speed for 30 minutes before termination. The mist air, oil, and cooling air flow rate and temperature were essentially the same as in Test 14. Cage instability was again indicated at both 60,000 and 65,000 rpm. The test bearing was in excellent condition with only minor changes in wear of the cage lands and pockets as evidenced by the minor buildup of degraded oil and silver wear particles occurring again on the cage bore. No changes could be detected on the contacting surfaces.

The results of these two tests indicated that the incorporation of the baffle plate was beneficial in minimizing the heavy wear and binding of the cage experiences in several prior tests at the higher speeds. No improvement in the apparent cage instability resulted from using the baffle plate. Insufficient testing was performed to establish the realibility of the scheme employed and the generation of the silver wear particles from the cage indicated that metal to metal contact did still occur to some degree which is not abnormal even in a bearing lubricated with recirculating oil.

Under the conditions used in Test 15 (relatively high oil flow rate, 29 in^3/hr., compared to those used in Test 11) no detrimental effects resulted from the change in nozzle configuration. However, at lower oil flow rates where the efficient plating out of the oil is more important, the nozzle configuration could possibly have an appreciable effect.

The purpose of Test 16 was to establish the feasibility of using a simplified mist system (drip/mist system) which would greatly reduce the vulnerability to ballistic damage, and decrease weight and complexity by eliminating the mist generator and the relatively large mist transfer line required to minimize oil plating out in transit. The idea was to provide oil drops in the flow path of the cooling air as it exited the nozzles where it would be misted and directed into the bearing.

To accommodate this scheme, six of the eight mist nozzles were plugged and drip tubes, 0.040 in. ID, were attached to the remaining nozzles to position oil drops in the flow path of two of the through bearing air jets, see Figure 6 . An external oil reservoir was provided to supply the oil. The oil flow rate (drip rate) was controlled by the hydrostatic head, due to the elevated position of the oil supply reservoir, and a feedback pressure from the chamber into which the oil was fed.

The test was performed in a similar manner to prior step-speed test, with the exception that the cooling air flow rate was increased to 0.495 scmm (17.5 scfm) to compensate for the elimination of the mist air. The cooling air temperature was 338°K (147°F) and the oil flow rate varied between 328 and 524 cc/hr. (20 and 32 in^3/hr) during the test.

The maximum speed of 65,000 rpm was successfully obtained. At 65,000 rpm a noise, interpreted to be caused by cage instability, occurred which produced variations in speed from 62,000 to 68,000 rpm. After operating at this speed for approximately 20 minutes, a much louder noise was observed and the test terminated. It was observed during the disassembly of the rig that the two drip tubes had loosened from the initial press fit and were laying in the cavity between the oil manifold and bearing. Heavy dents in the tubes indicated that they had been battered back and forth between the shaft and rig housing probably causing the noise heard just prior to test termination.

The inspection of the test bearing, which was also used in the two prior modified configuration tests was found to be in good to excellent condition. The ball tracks were difficult to detect with the original surface finish mark still observable. Further polishing of the rail on the downstream cage rail had occurred.

The results of the test indicated that sufficient lubrication was being provided by the drip/mist system and thus the simplified mist system was shown to be a feasible method of lubricating the bearing. Although the method used to supply the oil drops (hydrostatic head plus pressure feedback) was adequate for the test, observed changes in the flow rate (524 to 327 cc/hr) could not be tolerated if a lower initial oil flow rate was used. Therefore, a constant rate or infusion type pump would have to be used to maintain a constant oil flow rate when the simplified mist system was incorporated in an application.

6.2.4 Cage Design Modification Tests

Although excellent bearing performance was obtained with mist lubrication and air cooling in all prior step speed tests up to speeds in excess of 2×10^6 DN, either one or both of two major problems occurred in the majority of tests at speeds in the range of 2.5×10^6 to 3.0×10^6 DN. The most severe of these two problems was the excessive wear on the downstream cage rail and binding of the same rail with the outer ring guide land. The other problem was what appeared to be cage instability.

The excessive wear and binding problem occurred at a location where such a problem might be expected. The design gap between the rail and land is quite small, 0.13 to 0.20 mm (0.005 to 0.008 inches), thus limiting the flow of oil and air. In addition, the cooling air is greatly increased in temperature (essentially exhaust temperature) when it reaches the downstream rail; thus, its cooling effect is less, allowing the rail to increase in temperature and diameter. The downstream cage rail, therefore, provides the total guidance since the upstream rail can not contact the land.

The existence of this problem was easily detected by inspection of the tested bearings. When excessive wear did occur on the guide rails, it always occurred on the downstream rail with little or no signs of contact existing on the upstream rail. When the more severe condition of binding occurred, the wear was uniform over the full circumference of the rail. Also, excessive wear was present in the leading face of the ball pockets where the ball pressed hard to force movement of the restricted cage. Photographs of a typical bearing which had failed in this manner are presented in Figure 22.

Such failure suggest that insufficient oil is present at the rail-land interface thus permitting metal-to-metal contact. The resulting increased friction coefficient produces a greater heat generation rate which increases the thermal growth of the cage and thus further decreasing the flow of oil to the interface. This cycle can thus repeat until binding occurs.

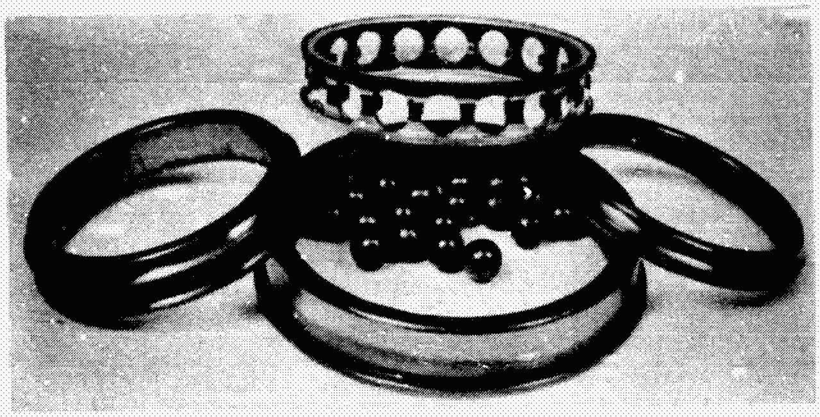
The detection of the other problem is more difficult and the assumption that the intermittent noise resulted from cage instability was based on experiencing similar problems in other applications. Cage instability is often encountered in grease lubricated bearing and bearing operating in a vacuum where low damping forces exist. Since the quantity of mist oil present in the test bearing is relatively small compared to the rig bearing (lubricated with recirculating oil). where the problem was not experienced, the presence of cage instability was considered likely.

Since both problems were considered to result in some degree from insufficient cage lubrication, the testing under this classification was performed to evaluate the possibility of eliminating the problem at high speeds by modifying the cage design to increase the oil and cooling air in the critical location.

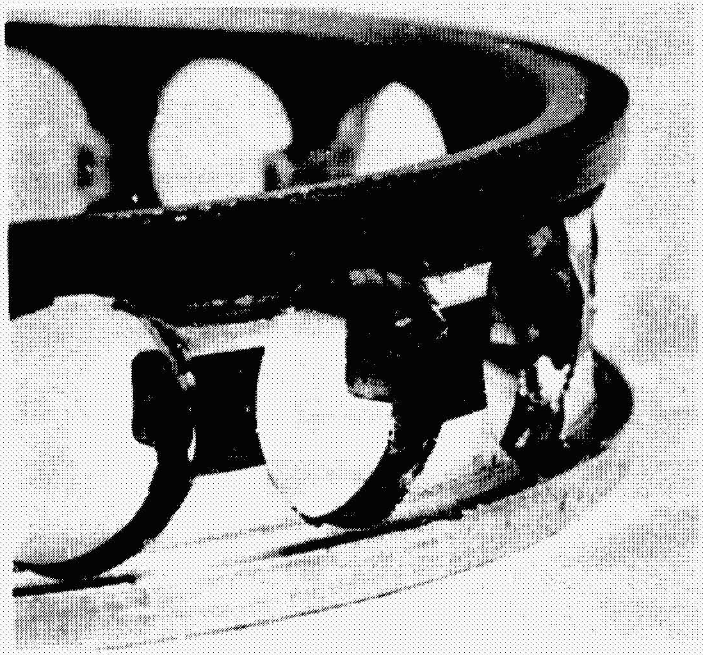
A total of four tests (Tests 17-20) were performed, each with a different cage design, aimed at increasing the oil flow to the cage land interface. Test 17 was performed with an increased cage land clearance, Test 18 with a cage designed to pump oil from the bore to the interface, Test 19 with an inner

FIGURE 22

TYPICAL BEARING FAILURE



Bearing Elements



Cage Showing Heavy Wear on Upstream
Rail and Leading Face of Ball Pockets

Outer Ring Showing Smearing on Upstream Land ring riding cage, and Test 20 with pumping grooves machined in the cage rails.

The increased gap evaluated in Test 17 was accomplished by increasing the outer ring land diameters. In the standard bearing design, the cage land gap can vary from 0.13 to 0.20 mm (0.005 to 0.008 in). Measurements performed on several of the bearings previously tested showed that the gap was at the low end of the tolerance limits. The bearing used in this test was modified by increasing the outer ring land diameter to provide a 0.008 inch gap on the upstream side and a 0.009 inch gap on the downstream side. The difference in the gaps was incorporated to compendate for the difference in the thermal growth of the two cage rails resulting from the variations in cooling capacity noted at the two locations. The cage in the modified bearing was projected to ride equally on both rails when operating at high speeds.

In addition, a change was incorporated in the test procedure to aid in obtaining a better evaluation of the cause of the intermittent noise at high speed should it occur. The **change, using the housing heaters to maintaining the rig housing temperature within 28°K (50°F) of the bearing outer ring temperature, was included to insure complete freedom of the load plug and thus** permit the **observation of minor changes in bearing drag torque** that would be produced **by cage vibrations.**

The test was performed in the same manner as prior step-speed tests with the exception noted. The mist oil flow rate, and the mist air and cooling air flow rates maintained throughout the test were 426 cc/hr (26 in^3/hr), 0.274 scmm (9.7 scfm) and 0.25 sfmm (8.0 scfm) respectfully. The shaft was accelerated to 35,000 rpm with no indication of problems. After accelerating to 40,000 rpm, a soft intermittent noise was noted which was accomplished by minor vibrations of the drag torque trace. As the speed was increased to 55,000 rpm, the intermittent noise level increased as did the drag torque vibration level. It was also noted that the temperature of the test bearing outer ring would increase shortly after the vibration started and decrease when it stopped. These observations confirmed that the intermittent noise was the result of cage instability. While attempting to accelerate to 60,000 rpm, the shaft suddenly slowed indicating a failure.

Inspection of the bearing revealed heavy wear had occurred between both rails and the corresponding guide lands. Thus it was concluded that the difference in the two gaps had permitted the cage to be guided by both lands. However, no observable improvement was obtained in lubricating the interface area and the increased gap appeared to be detrimental with respect to cage instability as the problem occurred at appreciably lower operating speeds.

The cage used in Test 18 was modified to incorporate a bore profile which was small at the center and tapered radially outward to oil retaining lips located under each rail. Eighteen 0.65 mm (0.025 in.) diameter holes were drilled from the cage bore to the OD of the rail just inside the retaining lip. The holes were equally spaced circumferentially and located equal distance between the ball pockets. A drawing of the cross section of the cage is presented in Figure 2. This modification was incorporated to pump the oil plated out on the cage bore to the cage-land interface by centrifugal force.

Two test runs were performed. The first run was performed with the following conditions which were relatively conservative with respect to air flow rate and cooling air temperature.

Mist air flow rate	0.38 scmm (13.6 scfm)
Mist inlet temperature	366°K (200°F)
Mist oil flow rate	410 cc/hr (25 in^3/hr)
Through brg. cooling air flow rate	0.29 scmm (10.4 scfm)
Through brg. cooling air inlet temp.	300°K (80°F)

The shaft was accelerated in steps to 65,000 rpm without any indication of cage instability. While operating at this speed for approximately 10 minutes only two short periods, 3 to 5 seconds, was cage instability observed. A cursive examination of the test bearing, without removing it from the shaft, showed no indication of wear.

The second run was initiated with conditions similar to those used in run 1 with the intent to decrease the oil flow and increase the cooling air temperature to determine if any major changes in bearing performance occurred. The shaft was accelerated to 45,000 rpm without any indication of problems. While accelerating to 55,000 rpm, cage instability was encountered at 52,000 rpm which continued for approximately 2 minutes and then stopped. The shaft was then accelerated to 65,000 rpm.

During the speed change, the instability was noted several times for a period of 1 to 2 seconds. The intermittent vibration continued for approximately 2 minutes and did not occur again during the next 7 minutes.

The mist air flow rate was decreased from 0.38scmm (13.6 scfm) to 0.24 scmm (8.6 scfm) which reduced the oil flow from 459 cc/hr (28 in^3/hr.) to 246 cc/hr. (15 in^3/hr.). After operating at this condition for 20 minutes, intermittent cage instability began. The mist air flow rate was returned to its former value and operation continued for 7 minutes without any effect on the cage vibration. The test was then terminated after operating at 65,000 rpm for 1.7 hours. The run was terminated to permit an examination of the bearing before a failure initiated by the cage vibration occurred.

The post bearing examination showed the bearing to be in excellent condition. Only very slight polishing of the downstream cage rail had occurred which would be considered normal even for a bearing lubricated with recirculating oil. Post test photographs of the test bearing cage are presented in Figure 23.

The results of the test indicated that the cage modification had been effective under the conditions evaluated to minimize cage-land wear and prevent cage seizure. The modification also decreased the cage instability, but was not successful in eliminating the problem.

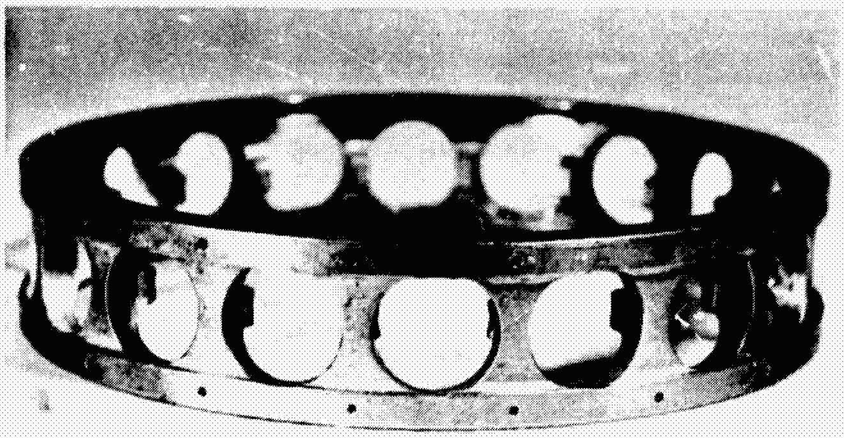
Test 19 was performed with an inner ring guided cage. The cage was designed to have a pilot clearance on the diameter of 0.13 to 0.40 mm (0.005 to 0.016 in.) and a ball pocket clearance on the diameter of 0.23 to 0.40 mm (0.009 to 0.016 in).

The test was performed with a mist oil flow rate of 524 cc/hr (32 in^3/hr.) and a combined mist and cooling air flow rate of 0.64 scmm (22.6 scfm). The speed was increased in steps to 55,000 rpm without incident. While increasing speed to 65,000 rpm a sudden increase in bearing outer ring temperature indicated failure and the test was terminated.

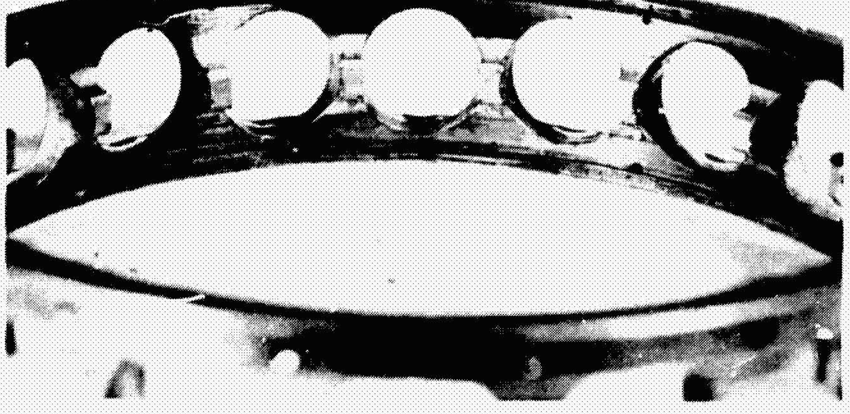
Inspection of the test bearing showed excessive wear between both cage rails and inner ring lands over an arc of 140°, see Figure 24. The wear pattern indicated that the cage was forced radially to one position and continued to be piloted

FIGURE 23

POST TEST PHOTOGRAPHS OF MODIFIED CAGE USED IN TEST 18



Cage Showing Radial Holes in Rails



Cage Showing Taper and Oil Retaining Lip in Bore

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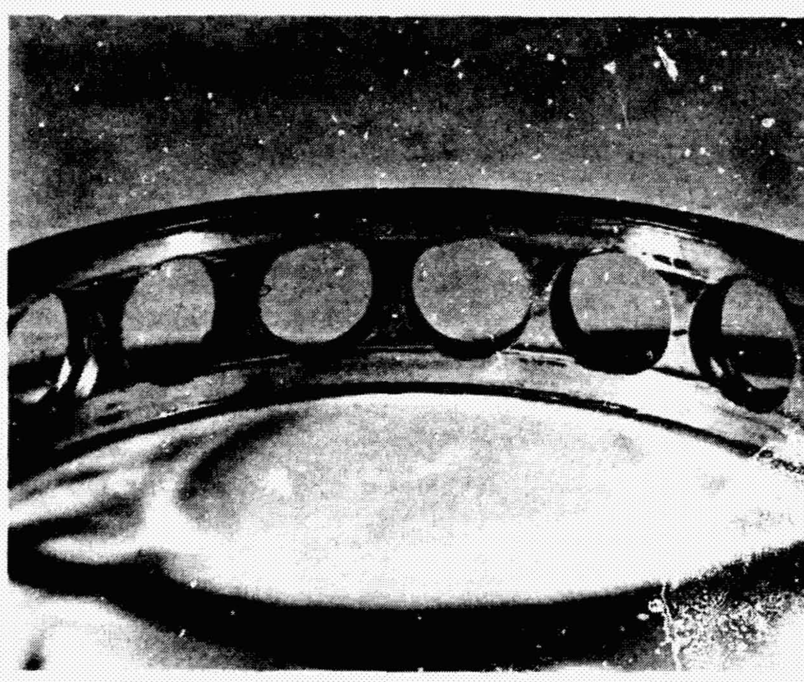
FIGURE 24

POST TEST PHOTOGRAPH OF INNER RING GUIDED CAGE USED IN TEST 19



at this contacting area. The fact that the bearing heat generation rate at all speeds was approximately twice that measured in previous tests indicated that the condition was present over the major portion of the run.

The cage balance was checked prior to testing and the existing unbalance was less than the standard requirement for aircraft engine bearing cages, ie 3 gr-cm. At a shaft speed of 60,000 rpm (cage speed of approximately 25,000 rpm) this would produce an unbalance force of approximately 47 pounds. An additional unbalanced force of approximately 4.6 pounds is produced from the eccentric position of the cage mass center with respect to the center of rotation when the cage rides on the land. In the worst condition, these two unbalanced forces would be added to produce a total unbalance force of 51.6 pounds. This force is not considered to be excessive if adequate lubrication is present. Since the pressure at the contact is appreciably less than at a ball race contact. Therefore, it was concluded that insufficient oil was present on the rail-land interface surface at high speeds to provide adequate lubrication.

The modification to the cage used in Test 20 consisted of 27 pumping grooves in each rail OD surface equally spaced around the circumference and extending from the inboard side to the center of the rail. The grooves were 0.050 inches wide forming an angle of 20° with the face to produce an axial pumping force and tapered radially to perform centrifugal pumping to the rail pad located between the grooves. The gap between the cage rail and the guiding land of the outer ring was 0.13 mm (0.005 in). The cage design is shown in Figure 3.

The speed was increased in steps up to 50,000 rpm without any indication of abnormal performance. At approximately 53,000 rpm, a short period of cage instability was observed. While accelerating from 60,000 to 65,000 rpm the bearing temperature suddenly increased and the speed decreased which resulted in the termination of the test.

The post test inspection of the test bearing revealed that excessive wear had occurred between the downstream cage rail and ring land. The wear on the rail and land was essentially uniform over the complete circumference indicating that the cage thermal growth was sufficient to cause the cage to bind. As in prior tests where this same problem occurred, it was attributed to insufficient lubrication between the surfaces at high speeds which increases the frictional heat generation rate causing the cage to expand until it binds. It was therefore

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concluded that the pumping grooves did not sufficiently increase the oil flow into the interface to provide adequate lubrication. A post test photograph of the bearing cage is presented in Figure 25.

The modified cage incorporating feed holes from the cage bore to the rail was the only one of the four designs tested which showed a significant improvement in reducing the wear at the cage-land interface. Although both runs performed with this cage utilized an average air temperature approximately 14°K (25°F) below that normally used, this difference is not considered to be significant in the improved perf rmance observed. This conclusion is based on the fact that cage wear problems were not encountered at lower speeds in Test 13 than in many tests where lower air temperatures were supplied. The average air temperature used in Test 13 was 33°K (60°F) above that normally used.

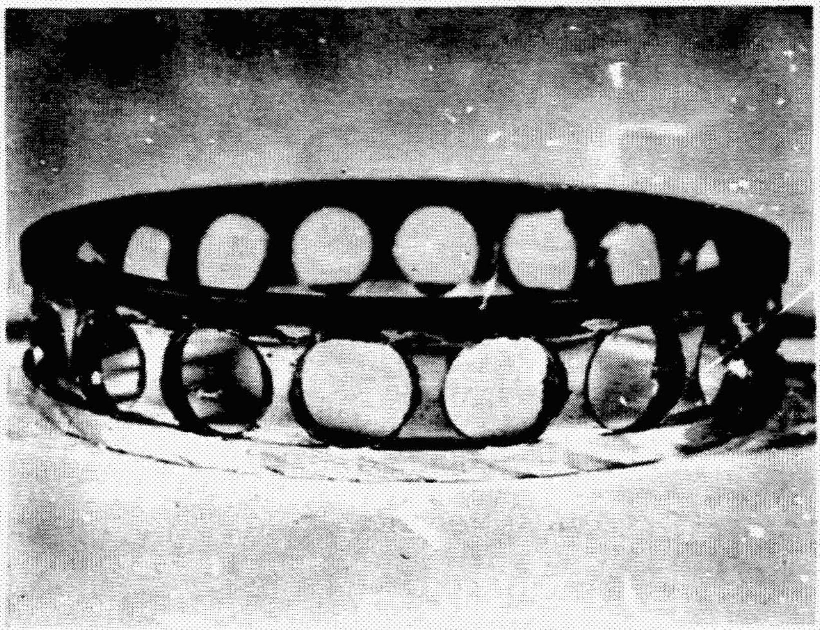
6.3 Extended Period Tests - Type III

Two extended period tests were performed (Test 21 and 22) utilizing the mist and air cooling system to lubricate and cool the test bearing. Both tests were performed with the lubricant meeting MIL-L-23699 specifications and with a thrust load of 1779 (400 lb.) applied. The purpose of the tests was to demonstrate extended periods of operation without thermal or lubrication problems occurring. The periods of 50 hours and 100 hours of operation at the selected operating speed in Test 21 and 22 respective were selected based on the conception that inadequate lubrication would manifest itself in the form of surface distress on the bearing race within this period if it existed. The originally conceived mist and cooling air system, and bearing configuration (through mist and cooling air with the increased chamfer on the upstream side of the inner ring and the cage bore tapered radially outward from both sides to the center) was used in both tests.

The first extended period test (Test 21) was performed at a speed of 55,000 rpm (2.5×10^6 DN) based on the maximum speed where cage instability and cage wear were considered to be unlikely to occur. The air flow rates and temperatures and the mist oil flow rate were selected to be quite conservative compared to those used, successfully in some of the more severe step-speed tests. Representative temperatures and flow rates used throughout the test are as follows:

<u>FIGURE 25</u>

POST TEST PHOTOGRAPH OF CAGE
WITH PUMPING GROOVES USED IN TEST 20



Mist Air Flow Rate (scmm-scfm)	Mist Air Temp. (°K-°F)	Cooling Air Flow Rate (scmm-scfm)	Cooling Air Temp. (°K-°F)	Average Air Temp. (°K-°F)	Mist Oil Flow Rate (cc/hr-in^3/hr)
0.39-13.6	371-210	0.19-6.8	353-175	365-198	623-38

The test was conducted in a total of seven runs of the following time durations at the test speed; 3.0, 0.5, 7.4, 10.2, 11.0, 11.2, and 7.5 hours for a total period of 50.8 hours. After each run, the test bearing and rig temperature was allowed to cool to room temperature before starting the next run. The average test bearing outer ring temperature and heat transfer rate to the air was 450°K (350°F) and 530 watts (1800 btu/hr) respectively. No cage instability was observed during the test.

The post test inspection of the bearing revealed the following conditions of the components:

Inner Ring - The ball track was visible basically due to a thin film of yellowish brown varnish deposited on the bearing lands and race outside of the track. Though slightly glazed, the original finish was still evident in the ball track indicating excellent lubrication. Several dents were present in the ball track indicating that dirt particles either introduced with the mist or cooling air, or generated in the bearing had been rolled over.

Outer Ring - Similar to inner ring except no deposition film on the cage riding lands which appeared to be equally polished and uniform for the full 360°.

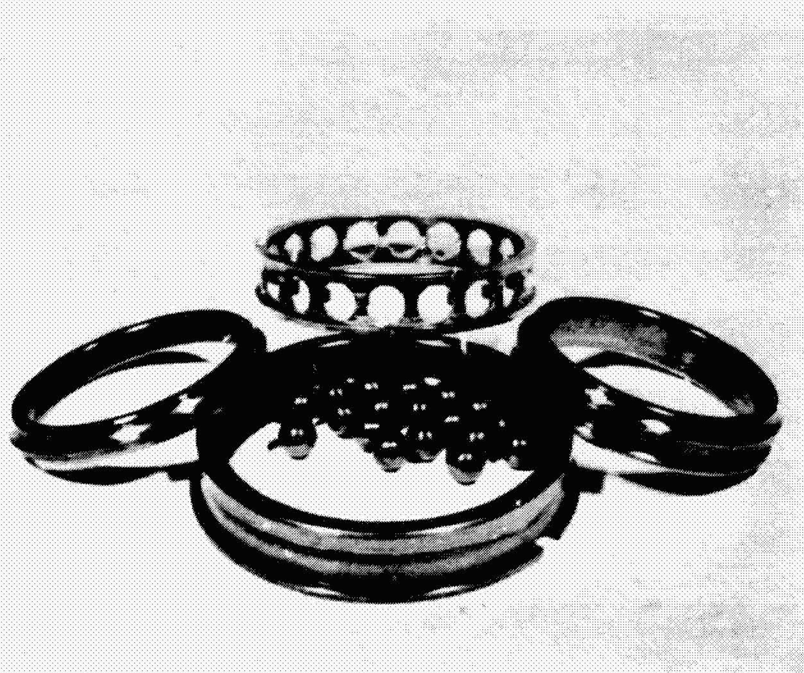
Cage - In good condition with minor polishing of the rails with one area on the upstream rail polished slightly more than the other areas. All pockets in good condition with minute spots in some of the pockets where normal microwear had occurred.

Balls - Good condition with a brownish discoloration indicating a slight varnish coating.

It was concluded that the bearing had performed exceptionally well and that it had been well lubricated. There was no indication that any failure mode was eminent. A post test photograph of the bearings is presented in Figure 26.

FIGURE 26

POST TEST PHOTOGRAPH OF BEARING
USED IN FIFTY HOUR TEST



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The second extended period test (Test 22) was performed at a speed between 43,600 to 45,000 rpm (2×10^6 DN) which is representative of the maximum condition experienced by mainshaft bearing in current helicopter engines. The average air temperature was increased to be more representative of that available in engines and the flow rate decreased approximately 15 percent. The mist oil flow was appreciably decreased to an average value of 293 cc/hr (17.9 in^3/hr) which was still considered conservative compared to that used in run 3 of Test 11. Representative temperature and flow rates used through the test are as follow:

Mist Air Flow Rate (scmm-scfm)	Mist Air Temp. (°K-°F)	Cooling Air Flow Rate (scmm-scfm)	Cooling Air Temp. (°K-°F)	Average Air Temp. (°K-°F)	Mist Oil Flow Rate (cc/hr-in^3/hr)
0.15-5.3	348-165	0.34-11.9	406-270	388-238	284-16.7

The test was performed in ten separate runs with operating periods at the test speed varying from 4 to 31 hours for a total period of 100 hours. The cooling air (combined mist and through bearing cooling air) varied from 380 to 392°K (225 to 245°F) with an average value for all runs of 390°K (241°F) and a flow rate of 0.49 scmm (17.2 scfm). The mist oil flow rate varied from 239 to 311 cc/hr (14.6 to 19 in^3/hr) during the ten runs with an average value of 292 cc/hr (17.8 in^3/hr). Under these conditions, the bearing temperature varied from 463 to 472°K (375 to 390°F) with a heat transfer rate to the air of approximately 380 watts (1300 Btu/hr).

The post test bearing inspection revealed the following condition of the components:

Inner Ring - Mild glazing of the ball track with most of the finish marks removed and a considerable number of minute dent marks. Thin layers of varnish over most surfaces except ball track.

Outer Ring - Similar to outer ring except more finish lines present in the ball track and less denting.

Cage - In good condition with minor polishing of silver plate on both cage rails. Uniform polishing of silver plate on leading and trailing sides of ball pockets. Layer of varnish on cage bore and downstream face.

Balls - In good condition with a few minute dents

The rig bearing used for this test run, which had seen recirculating lubrication was examined to obtain a relative evaluation between the two lubrication methods. the rig bearing was identical in design to the test bearing and had been lubricated with recirculating oil supplied at 367°K (200°F) and at a rate of 40,312 cc/hr (2460 in^3/min) or 138 times that supplied to the test bearing. This bearing operated at approximately 478°K (400°F) during the test or 10°K higher than the test bearing. The bearings were very similar in appearance with slightly less glazing present in the rig bearing. However, more wear had occurred in the ball pockets of the rig bearing. A post test photograph of the test bearing is presented in Figure 27. It was concluded from the relative appearances that the mist lubricated bearing had been as well lubricated as the recirculating oil lubricated bearing.

The results of these two tests demonstrated the feasibility of using a mist and cooling air system for lubrication and cooling of contemporary helicopter engine and transmission bearings.

6.4 General Discussion of Test Results

The results of the tests demonstrated the feasibility of using a once through mist lubrication system, in conjunction with auxiliary air cooling, to effectively lubricate and cool mainshaft engine and transmission bearings used in contemporary helicopters. In almost all of the tests, bearing speeds of 2 x 10^6 DN were obtained without any indications of bearing malfunctions precipitated by the method of lubrication and cooling. In several tests, bearing speeds as high as 3 x 10^6 DN were obtained successfully. However, at speeds between 2.5 x 10^6 DN and 3 x 10^6 DN cage land wear and/or cage instability problems were encountered.

The most serious of the two problems, with respect to producing early and possibly catastrophic failure, is the heavy wear and binding that occurs between the cage and guide land on the downstream size (opposite the side of mist and cooling air insertion) of the bearing. This problem, which was diagnosed from the wear pattern existing in the failed bearings, manifests itself by excessive wear on the guide land and cage rail on the downstream side of the bearing and wear on the leading face of the ball pocket where the ball was driving the cage. There was only slight or normal polishing of the silver plating on the upstream cage rail and on the trailing pocket face in the failed bearings.

FIGURE 27

POST TEST PHOTOGRAPH OF BEARING
USED IN ONE HUNDRED HOUR TEST

A similar problem was encountered in mist lubrication testing of a 125 mm bore angular contact aircraft engine ball bearing at speeds of 2.5×10^6 to 3.0×10^6 DN (4).

The cage land interface on the downstream side is the most vulnerable to this type of problem with the oil mist and cooling air system employed. With the outer land riding cage design, the gap is very small compared to the inner ring gap (0.13 to 0.20 mm outer, 1.1 mm inner). This difference in gap size, combined with intentionally directing the mist and cooling air at the inner ring gap because of the greater heat generation on the inner ring at high speeds, minimizes the oil and air reaching the cage land interface. In addition, the temperature increase of the air passing axially through the bearing results in greater downstream rail growth; thus, all guidance is performed at this location.

It was anticipated, during the design of the system, that the centrifugal pumping produced by the inner ring and cage rotation would force oil into the interface. However, the oil flow to this location was apparently inadequate at high speeds and the incorporation of two cage modification (increased gap size, axial pumping grooves on rail) did not improve lubrication sufficiently. The problem is not considered to result from an inadequate quantity of lubricant to the bearing, but poor distribution or flow in the bearing. This conclusion is based on experience with high speed bearings lubricated with a recirculating oil system where the same problem existed until one of the jets was directed at the outer ring gap. It is also noted that in Tests 14 and 15, the modification of the oil and air flow path in the bearing by incorporating a baffle plate appeared to reduce the wear appreciably.

The modified cage design, which incorporates pumping holes from the bore to rail OD, used in Test 18 eliminated the excessive wear problem in the two runs performed and indicate that the design may be reliable in eliminating the problem.

The same design was effective in reducing, but not totally eliminating the cage instability. This suggested that improved lubrication in the interface may have increased the damping and resulted in less vibration. Thus it appears that the incorporation of mist nozzles on the downstream side of the bearing to insert oil and air directly into this interface merits investigation. The increased lubrication and cooling could be sufficient to eliminate both the wear and the instability problems.

A mist oil flow rate of approximately 491 cc/hr (30 in^3/hr) was supplied in the majority of the step-speed tests. It was observed during these tests that considerable oil, not possible to measure, was emitted from the exhaust port as mist with the air. It is not known whether the oil had plated out and been remisted or passed directly through the bearing as mist. It seems reasonable, however, that at least a portion of the oil passed through the bearing without being utilized, thus indicating an excess supply.

The supply rate was based on the calculated oil replenishment rate of 557 cc/hr (34 in^3/hr) for operation at 65,000 rpm obtained in the design phase of the program (Section 4.2) using EHD lubrication theory. In applying the theory, it was assumed that the oil displaced out of the inner ring track as a ball passed must be replaced by the mist supply. In Test 11 it was demonstrated that considerably less oil, approximately 0.1 that calculated could be used to replenish the oil loss. This is in close agreement with information presented in Reference 4 where an oil replenishment rate of 13 percent of the calculated value was shown to be adequate in a short term test. Although the feasibility of operating with as little as 0.1 the calculated value was demonstrated, it is considered more reasonable to use a flow rate of approximately 50 percent of the theoretical value for bearing lubrication in high speed helicopter engines until reliability at lower oil flow rates is demonstrated by further testing.

In the first step-speed test it was demonstrated that appreciably less cooling air than anticipated was required to maintain the test bearing at a reasonable operating temperature. It was also shown that housing and shaft cooling air was not required to prevent thermal imbalance failure (removal of the bearing internal clearance due to unequal thermal growth between the rings) thus appreciably decreasing the complexity of the initial system design.

In the major portion of the tests, a total air flow rate of 0.48 to 0.62 scmm (17 to 22 scfm) was supplied compared to the calculated requirement of 2 scmm (72 scfm). The lower air flow rate required was basically the result of a lower bearing heat generation rate than anticipated with mist lubrication, see Figure 27. The heat transfer film coefficient used in the analysis were also lower than those determined from the test data at high speeds and air flow rate of 0.48 scmm (17 scfm). This also contributed to a lower air flow rate requirement. The film coefficient values of 322 to 362 $W/m^2 °K$ (57 to 64 Btu/hr $ft.^2 °F$) established during the Mobil study (4) were reasonably consistant with those determined from the Test 12 where 0.28 scmm (10 scfm) of air was used. The heat transfer film coefficients calculated from Test 16 and Test 12 are presented in Appendix I.

Based on these calculations the following set of heat transfer film coefficient values are considered to be reasonable for use in design analysis when the mist and cooling air is forced through the bearing:

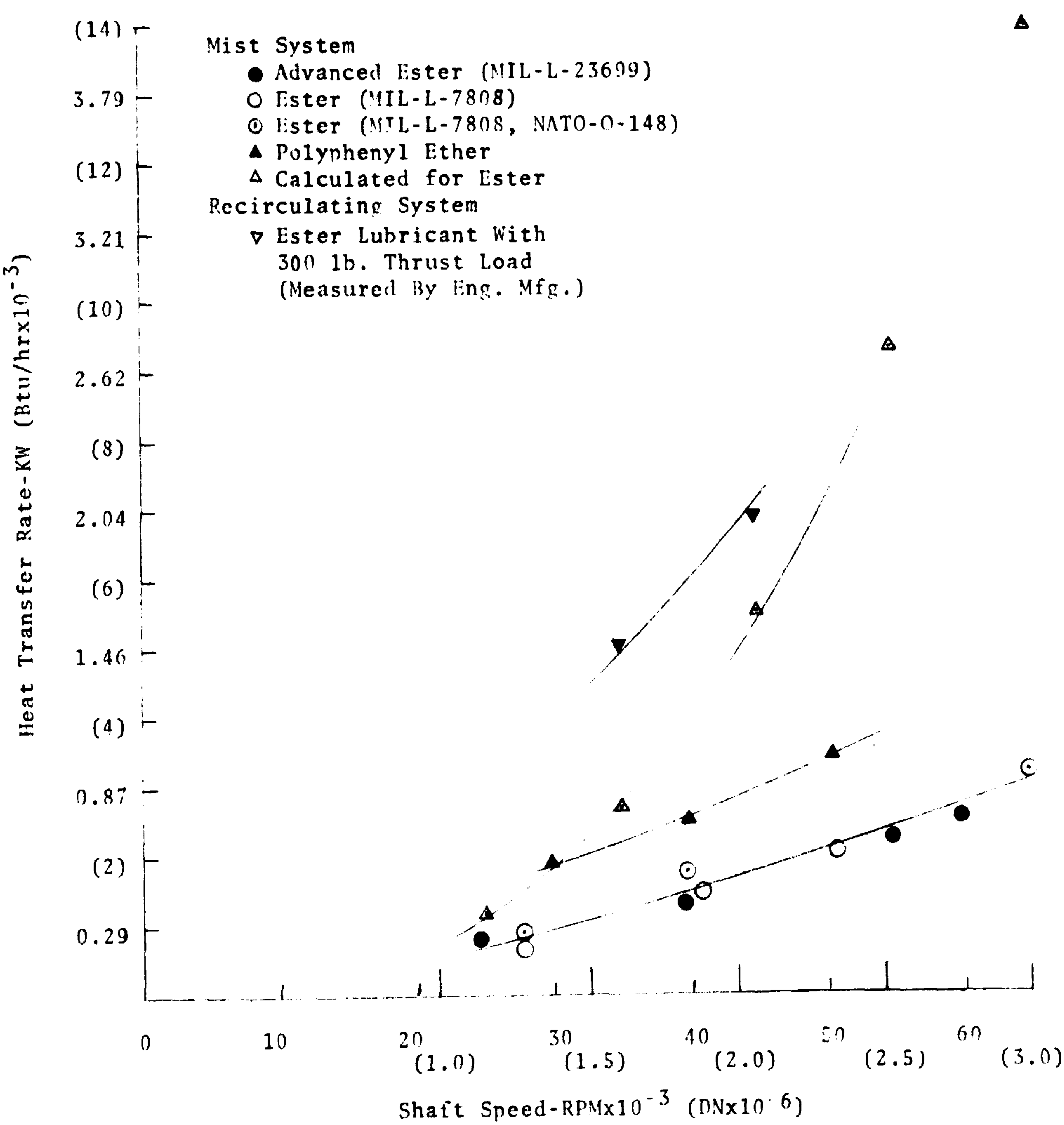
Bearing DN Value ($DN \times 10^6$)	Film Coefficient $W/m^2 °K$ - $Btu/hr.ft^2 °F$
1.5	289 - 51
2.0	340 - 60
2.5	391 - 69
3.0	442 - 78

The comparison of the bearing temperatures and heat generation rates in Test 11 and 13 show that an increase in the air temperature results in essentially the same increase in the bearing temperature when operating under similar conditions. This observation indicates that no major change occurs in the heat transfer film coefficient over the range of bearing and air temperatures investigated. Thus, by knowing the bearing and cooling air temperature at a set air flow rate and operating speed, the allowable increase in air temperature can be calculated which would result in the maximum permissible bearing temperature.

The graph presented in Figure 28 shows a plot of the experimentally determined heat generation rate of a bearing, identical to that used in this program, lubricated with a recirculating ester oil run in a developmental engine test by the turbine manufacturer. These values were established from the heat transferred to the oil when a thrust load of 1334 newtons (300 lb.) was applied. Also plotted on the graph are the bearing heat generation rates measured in the four lubricant evaluation tests (Tests 7-10) using mist lubrication conducted in the SKF test rig. The graph indicates that only one third to one fourth as much heat is generated in a mist lubricated bearing compared to a bearing lubricated with recirculated oil. Similar results were also obtained in tests performed with a 125 mm bore ball bearing (5). The reduced heat generation rate results from minimized oil churning and this reduced heat generation aids in making the operation of air cooled bearings feasible.

FIGURE 28

LUBRICATION SYSTEM HEAT REJECTION COMPARISON



7.0 CONCLUSIONS

A. Bearing Emergency Lubrication Systems Evaluation

1. The feasibility of an emergency aspirator lubrication system utilizing residual oil within the oil manifold was demonstrated as a viable survivability concept for helicopter mainshaft engine bearing.

2. The incorporation of shaft and bearing housing cooling air in combination with the emergency aspirator system was not necessary and in fact proved detrimental to the bearing operation.

3. It was demonstrated that emergency lubrication could not be provided by utilizing the residual oil retained on the shaft following oil cessation.

B. Bearing Mist Lubrication and Air Cooling System Evaluation

1. The feasibility of using oil mist and air to lubricate and cool helicopter engine bearings operating at current engine speeds (2×10^6 DN) was repeatedly demonstrated in numerous short term tests and one 100 hour test. Excellent bearing performance was also obtained while using the mist lubrication system in a 50 hour test at 2.5×10^6 DN.

2. The capability of achieving similarly successful bearing operation with a greatly simplified mist supply (drip/mist) system was demonstrated in one short term test.

3. The potential of applying the mist lubrication principle to bearings operating in the speed range of 2.5×10^6 to 3×10^6 DN was demonstrated, even though problems, ie. cage-land wear and/or cage instabilities, remain to be solved before long term successful operations can be achieved in this regime.

C. Parameters of Mist Lubrication/Air cooling System

1. Supplying cooling air through the shaft and bearing housing was shown to be unnecessary with the mist lubrication system. The flow of cooling air directly through the bearing was demonstrated to provide sufficiently uniform cooling of the inner rings to prevent thermal imbalance failures.

2. No detrimental effect was observed when straight, non-reclassifying mist nozzles were evaluated with a mist oil and air flow of 475 cc/hr. and 0.32 scmm (29 in^3/hr and 11.3 scfm). No evaluation was performed at lower oil flow rates where the efficiency of the plating out of the oil on the bearing could be more critical.

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3. The assumption, used in the mist oil flow rate requirement analysis, that all the oil displaced from the inner ring when a ball passes must be replaced was shown to be very conservative in several short term tests. In one short term test, as little as 0.1 (51 cc/hr) the theoretical value provided adequate lubrication.

4. Adequate bearing cooling was demonstrated with as little as 0.283 scmm (10 scfm) total air flow when supplied at a temperature of 359°K (185°F) for the operating conditions evaluated. The quantity of air required to cool the test bearing in all mist tests performed was appreciably less than the calculated requirement. This condition resulted from a lower bearing heat generation rate than calculated and somewhat higher heat transfer film coefficients than used in the air flow rate calculations.

5. Heat transfer film coefficients calculated from the test data indicated values increase with increasing air flow rates and bearing speeds. The increase is assumed to result from greater turbulence. The calculated values range from 181 to 499 watts/m^2°K which are in fair agreement with values (322 to 362 watts/m^2°K obtained by Mobil Research for mist impinging on a rotating disk.

6. Increasing the mist and cooling air temperature results in an essentially equal increase in the bearing operating temperature.

D. Corrective Actions for Operation Above 2.5 x 10^6DN

1. The incorporation of a baffle plate attached to the upstream side of the bearing appeared to modify the mist and air flow in the bearing and reduce cage-land wear at speeds between 2.5 x 10^6 and 3 x 10^6 DN. However, the reduction was not considered adequate to provide long term reliable performance at the high speeds.

2. One cage design incorporating radial holes in the cage rails showed potential in eliminating the cage wear and binding problems at high speeds and resulted in less cage instability in the one test performed. Three other modified cage designs (1. increased cage clearance, 2. inner ring guided cage, 3. axial pumping grooves) provided little or no improvement in cage-land wear. The increased cage clearance design aggravated cage instability.

8.0 RECOMMENDATIONS

1. This program demonstrated the feasibility of using a simplified mist (drip/mist) system to lubricate and cool helicopter engine bearings. Additional evaluations should be performed with the simplified system incorporating a constant oil flow rate or infusion pump to demonstrate reliable performance at bearing speeds encountered in current generation gas turbine engines, ie. in the speed range of 2×10^6 DN. Such a system could be activated to provide emergency lubrication and cooling or used as the primary system with an appreciably improved (reduced) vulnerability profile.

2. The compressor bleed air available for a cooling medium is quite hot in most engine configurations, ie. 300°F. This program demonstrated that more efficient bearing operation can result with cooler air flows. Vortex coolers, which are relatively light weight, and simple in design could be used to reduce the incoming air temperature. The feasibility of combining the vortex cooler with the simplified mist system to extend the range of possible applications, should be investigated.

3. This program demonstrated that additional effort is required in the area of mist deployment to eliminate failures in the speed range of 2.5×10^6 to 3.0×10^6 DN due to inadequate lubrication of the downstream cage-land interface. It is recommended that future work be directed at eliminating this condition by evaluation testing various mist application approaches.

4. It is also recommended that work be initiated in the area of cage design and analyses to eliminate cage instability observed at high speeds. This work should include the analytical evaluation with respect to stability of several cage designs using the recently developed stability criteria as established at Battelle Institute. The most promising designs should then be manufactured and evaluation tests performed.

9.0 LIST OF REFERENCES

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APPENDIX I
ESTIMATED BEARING COOLING AIR FLOW RATE REQUIREMENT

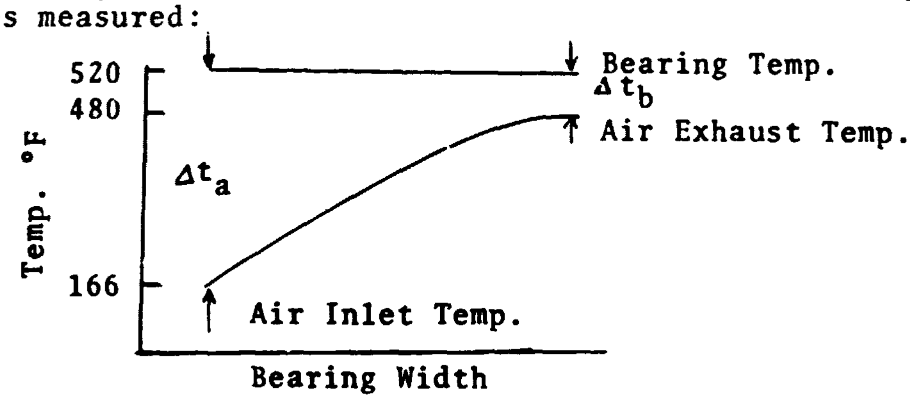
Based on theoretical calculations it was established that a 46 mm bore angular contact ball bearing would generate heat at a rate of 14,000 Btu/hr when operating at 65,000 rpm (3×10^6 DN) and lubricated with oil mist with a thrust load of 400 lbs. applied. Using this value of bearing heat generation rate the estimate of the cooling air required to remove the heat was performed in the following manner.

From test data obtained in mist lubrication testing reported in Reference 2 and considering the bearing as a heat exchanger where the bearing temperature is uniform on all surfaces the surface coefficient can be established using the equation

$$H = \frac{q}{A\Delta t_m}$$

where H = the surface heat transfer coefficient in Btu/hr-ft^2-°F
q = heat transfer rate in Btu/hr
A = surface area in ft^2
Δt_m = mean temperature difference between the bearing and air as the air passes axially through the bearing

For the particular case evaluated the bearing heat generation rate of 15,222 Btu/hr was established and the following temperatures measured:



The exposed surfaces of the 125 mm bore bearing (SKF 459981-G1) is 1.02 ft^2. The exposed surface includes all surfaces except the inner ring bore and the outer ring O.D. surfaces.

The mean temperature difference in a heat exchanger is obtained from the following equation from which Δt_m of 144°F was calculated.

$$\Delta t_m = \frac{\Delta t_a - \Delta t_b}{\ln \frac{\Delta t_a}{\Delta t_b}} = \frac{354-40}{\ln \frac{354}{40}} = 144°F$$

then $H = \frac{15,222}{(1.02)\ (144)} = 103$ Btu/hr-ft^2-°F

The efficiency of the bearing as a heat exchanger based on the ratio of the actual temperature rise of the air to the possible temperature rise of the air is 89 percent.

$$n = \frac{\text{Actual Temp. Rise}}{\text{Possible Temp. Rise}} \times 100 = \frac{314}{354} \times 100 = 89\%$$

Both of these values are considerably higher than published values of convective heat transfer coefficients and heat exchanger efficiencies. Surface heat transfer coefficients of mist oil impinging on a heated rotating disk were measured by Mobil Research and Development Corporation and reported in Reference 3 ranged from 57 to 64 Btu/hr-ft^2-°F. Since these values are in better agreement with other published data on convective heat transfer values and the surface area of the 46 mm bearing is considerably less than that of the 125 mm bearing, a bearing heat exchanger efficiency of 60 percent was selected for estimating the bearing cooling air requirement.

The bearing cooling air flow rate (Q) is then computed using the following formula:

$$Q = q/(\rho C_p\ t)$$

where

q = bearing heat generation rate at 65,000 rpm - 14,000 Btu/hr
ρ = air density at 70°F - 0.075 lb/ft^3
C_p = specific heat of air at constant pressure - 0.24 Btu/lb°F
Δt = air inlet and outlet temperature difference

For the expected air inlet temperature of 200°F and bearing temperature of 500°F, Δt equals 180°F (exit air temperature 320°F) when the bearing efficiency as a heat exchanger is 60 percent.

Then

$$Q = \frac{14,000}{0.075(0.24)(180)} = 4320 \text{ ft}^3/\text{hr} = 72 \text{ scfm}$$

Heat Transfer Film Coefficients Calculated From Data Recorded During Tests 12 and 16 Performed During the Study Reported In This Report

The following heat transfer film coefficients were calculated from Test 16 and Test 12.

Test No.	Total Air Flow (scmm - scfm)	Shaft Speed (rpm-DN x 10^6)	Film Coefficient (W/m^2°K-Btu/hr.ft.^{2}°F)
16	0.48-17	35,000-1.6	380 - 67
16	0.48-17	40,000-1.8	408 - 72
16	0.48-17	45,000-2.0	431 - 76
16	0.48-17	50,000-2.3	442 - 78
16	0.48-17	55,000-2.5	482 - 85
16	0.48-17	60,000-2.7	499 - 88
12	0.28-10	30,000-1.4	181 - 32
12	0.28-10	40,000-1.8	255 - 45
12	0.28-10	55,000-2.5	317 - 56
12	0.28-10	60,000-2.7	329 - 58

The calculations were based on the assumption that all surfaces of the bearing were at a uniform temperature equal to the measured outer ring temperature and the temperature of the air entering the bearing was equal to that measured value upstream of the nozzles. The coefficients were calculated using the equation used for a heat exchangers $H = \frac{q}{At_m}$.

The bearing surface area used in the calculations was 0.27 $ft.^2$ which includes all surfaces except the bore, outer ring O.D. and the two downstream faces.

The results of the calculations indicate that the film coefficient increases with both bearing speed and air flow rate. The increase is attributed at least in part to the increase in turbulance which increases with the two variables. It is also likely that the greater coefficient values calculated for the higher flow rate were the result of a higher Δt_m value actually existing due to the drop in the air temperature as is passed through the nozzles.

APPENDIX II

MIST AND COOLING AIR NOZZLE DESIGN

In determining the design of the mist nozzles the desired velocity of the mist impinging on the bearing components or passing through the bearing must be considered. In the analysis presented below it was considered desirable for the mist to have a velocity which would permit it to transverse a distance equal to a ball diameter in the time interval afforded by the distance between ball passes. This insures that mist is available for the leading surface of the balls to collide with as they pass by each nozzle. It can be argued that only half this velocity is necessary due to the shape of the rolling element; however, the retarding forces of the air currents generated by the bearing are not known and the higher velocity may be desirable. There will also be a natural deceleration of the mist particles once they leave the nozzle. In addition, higher mist velocities will produce a greater wetting of the surface on which the mist impinges. Therefore, the higher velocity seems desirable.

The maximum or design speed of the shaft is 65,000 rpm. At this speed the linear orbital velocity of the balls is 266 ft/sec. This velocity is based on the bearing pitch diameter of 2.305 in. and a cage to inner ring velocity ratio of 0.4 which is a reasonable approximation for most ball bearing designs. The ball diameter is 0.3125 in. and the gap between balls is approximately 0.1 in. Therefore, the mist velocity should be (0.3125/.1) 266 or 831 ft/sec.

With the air entering the nozzle at 200°F and zero velocity, and making the assumption that the flow through the nozzle is isentropic the flow rate at the nozzle exit can be calculated in the following manner:

First determine the temperature drop of the mist as it passes through the nozzle using the isentropic flow rate equation.

$$V_2 = \sqrt{2gCp(778)(T_0-T_2)}$$

where V_2 = velocity of mist exiting nozzle
Cp = specific heat at constant pressure
T_0 = stagnation temperature of gas entering nozzle
T_2 = temperature of gas exiting nozzle
g = acceleration constant

Rearranging and solving

$$\Delta t = \frac{V_2^2}{2gC_p(778)} = \frac{(831)^2}{2(32.2)(.24)(778)} = 57.4°R$$

Then the exit temperature $T_2 = T_o - \Delta t = 660 - 57 = 603°R$

The nozzle exit pressure is determined by calculating the pressure drop across the bearing. To perform this analysis the turbulent flow equation between parallel surfaces is used. This equation is used based on the assumption that the major restriction exists between the cage rails and the ring land. To calculate the pressure drop the equation is used in the following form:

$$P_1^2 - P_2^2 = \frac{0.133M^{7/4}RT\mu^{1/4}L}{gl^{7/4}h^3}$$

where P_1 = pressure upstream of bearing or at nozzle exit - lb/ft^2
P_2 = pressure downstream of bearing - lb/ft^2
M = combined mass flow rate of mist and cooling air - lb_m/sec
g = gravitational constant - $\frac{lb_m ft}{lb_f sec^2}$
R = gas constant $\frac{lb_f ft}{lb_m °R}$
T = absolute temperature of air - °R
u = dynamic viscosity of gas - $\frac{lb_f sec}{ft}$
L = flow length - ft
l = circumferential length of gap - ft
h = gap height - ft

Substituting values into this equation results in a pressure drop of less than .001 lb/in^2 and therefore could be neglected in establishing the density change in the gas at the nozzle exit.

Using a mist air flow rate (w) of 12.5 scfm, which is based on a theoretical mist oil flow rate requirement of 34 in^3/hr and a mist generator delivery rate of 2.7 in^3/hr per scfm, the mist air flow rate at the nozzle exit is:

$$w_2 = \frac{wT_2}{T_{std}} = 12.5 \frac{603}{530} = 14.2 \ ft^3/min$$

Using the flow rate equation

$$w_2 = V_2 A$$

where A = nozzle exit area

and rearranging to solve for A, the comlined 8 nozzle exit areas equal

$$A = \frac{14.2\ (144)}{831 \times 60} = 0.0410\ in^2$$

and the diameter of each nozzle equals

$$D = \sqrt{\frac{0.0051 \times 4}{\pi}} = 0.080\ in.$$

A convergent angle of 16.5° was selected for the nozzle design with a 100 mesh screen (50% opening) located at the inlet to reclassify (increase) the mist oil particle size. With this combination, the inlet area is approximately 3 times the outlet area.

For the eight cooling air nozzles, a straight opening .125 in. in diameter was selected to accommodate the theoretically required 23.5 scfm flow rate.

APPENDIX III

SUMMARY OF DATA FROM MIST LUBRICATION STEP-SPEED AND EXTENDED PERIOD TESTS

TABLE III-1

TEST NO. 7 (LUBRICANT EVALUATION)

Lubricant-Advanced Ester (MIL-L-23699)

Shaft Speed (rpm)	Brg. O.R. Temp. (°K-°F)	Mist Oil Flow Rate (cm^3/hr-in^3/hr)	Cooling Air Flow Rate (scmm-scfm)	Cooling Air In/Exhaust Temp. (°K-°F)
26,000	405-270	--	0.160-5.64	344/387-160/235
40,000	440-333	--	0.160-5.64	344/408-160/275
55,000	473-390	508-31	0.160-5.64	344/428-160/310
60,000	495-430	508-31	0.160-5.64	344/439-160/330
65,000	503-445	508-31	-- --	344/-- -160/--

Shaft Speed (rpm)	Mist Air Flow Rate (scmm-scfm)	Mist Air In/Exhaust Temp. (°K-°F)	Total Air Flow Rate (scmm-scfm)
26,000	0.153-5.42	350/387-170/235	0.313-11.06
40,000	0.153-5.42	345/408-160/275	0.313-11.06
55,000	0.273-9.66	356/428-180/310	0.439-15.3
60,000	0.273-9.66	356/439-180/330	0.439-15.3
65,000	-- --	356/-- -180/--	-- --

Heat Transfer Rate

Shaft Speed (rpm)	Cooling Air (Watts-Btu/hr)	Mist Air (Watts-Btu/hr)	Total (Watts-Btu/hr)
26,000	132-450	112-381	243-831
40,000	202-690	198-676	400-1366
55,000	264-900	397-1357	661-2257
60,000	299-1020	459-1566	757-2586
65,000	-- --	-- --	-- --

TABLE III-2

TEST NO. 8 - (LUBRICATION EVALUATION)

Lubricant - Ester (MIL-L-7807)

Shaft Speed (rpm)	Brg. O.R. Temp. (°K-°F)	Mist Oil Flow Rate (cm^3/hr-in^3hr)	Cooling Air Flow Rate (scmm-scfm)	Cooling Air In/Exhaust Temp. (°K-°F)
15,000	374-215	590-36	0.158-5.6	337/364-145/195
28,000	406-270	590-36	0.158-5.6	337/380-145/225
41,000	429-312	590-36	0.158-5.6	337/400-145/260
51,000	448-345	590-36	0.158-5.6	328/420-130/295

Shaft Speed (rpm)	Mist Air Flow Rate (scmm-scfm)	Mist Air In/Exhaust Temp (°K-°F)	Total Air Flow Rate (scmm-scfm)
15,000	0.385-13.6	374/364-215/195	0.543-19.2
28,000	0.385-13.6	374/380-215/225	0.543-19.2
41,000	0.385-13.6	371/400-210/260	0.543-19.2
51,000	0.385-13.6	371/420-210/290	0.543-19.2

Heat Transfer Rate

Shaft Speed (rpm)	Cooling Air (Watts-Btu/hr)	Mist Air (Watts-Btu/hr)	Total (Watts-Btu/hr)
15,000	[illegible]3-302	*(85)-(293)	184- 9
28,000	141-483	43 - 147	134- 630
41,000	203-693	215 - 734	419-1430
51,000	293-999	344 -1175	637-2174

* ( ) - Indicates minus value.

TABLE III-3

TEST NO. 9 - (LUBRICANT EVALUATION)

Lubricant - Ester (MIL-L-7808, NATO-O-148)

Shaft Speed (rpm)	Brg. O.R. Temp. (°K-°F)	Mist Oil Flow Rate ($cm^3/hr-in^3hr$)	Cooling Air Flow Rate (scmm-scfm)	Cooling Air In/Exhaust Temp. (°K-°F)
17,000	371-210	—	0.160-5.64	333/356-140/180
28,400	403-265	—	0.160-5.64	333/378-140/220
40,000	437-325	492-30	0.160-5.64	333/403-140/265
65,700	475-395	492-30	0.161-5.71	333/435-110/322

Shaft Speed (rpm)	Mist Air Flow Rate (scmm-scfm)	Mist Air In/Exhaust Temp (°K-°F)	Total Air Flow Rate (scmm-scfm)
17,000	0.273- 9.66	352/356-175/180	0.433-15.3
28,400	0.273- 9.66	352/378-175/220	0.433-15.3
40,000	0.385-13.6	363/403-195/265	0.543-19.2
65,700	0.425-15.0	369/435-205/322	0.586-20.7

Heat Transfer Rate

Shaft Speed (rpm)	Cooling Air (Watts-Btu/hr)	Mist Air (Watts-Btu/hr)	Total (Watts-Btu/hr)
17,000	71- 244	15- 52	86- 296
28,400	143- 488	138- 470	281- 958
40,000	223- 762	302-1030	525-1792
65,700	381-1302	571-1951	952-3253

TABLE III-4

TEST NO. 10 (LUBRICANT EVALUATION)

Lubricant - Polyphenyl Ether

Shaft Speed (rpm)	Brg. O.R. Temp. (°K-°F)	Mist Oil Flow Rate ($cm^3/hr-in^3/hr$)	Cooling Air Flow Rate (scmm-scfm)	Cooling Air In/Exhaust Temp. (°K-°F)
30,000	436-325	—	0.158-5.6	333/410-140/280
40,000	440-333	442-27	0.158-5.6	333/425-140/305
50,000	491-425	442-27	0.158-5.6	333/450-140/350

Shaft Speed (rpm)	Mist Air Flow Rate (scmm-scfm)	Mist Air In/Exhaust Temp (°K-°F)	Total Air Flow Rate (scmm-scfm)
30,000	0.373-13.2	369/410-205/280	0.532-18.8
40,000	0.425-15.0	371/425-210/305	0.583-20.6
50,000	0.425-15.0	375/450-215/350	0.583-20.6

Heat Transfer Rate

Shaft Speed (rpm)	Cooling Air (Watts-Btu/hr)	Mist Air (Watts-Btu/hr)	Total (Watts-Btu/hr)
30,000	246- 840	323-1103	5840-1943
40,000	290- 990	451-1539	741-2529
50,000	369-1260	640-2187	1009-3447

TABLE III-5

TEST NO. 11 (DECREASED MIST - RUN 3)

Lubricant - Advanced Ester (MIL-L-23699)

Shaft Speed (rpm)	Brg. O.R. Temp. (°K-°F)	Mist Oil Flow Rate ($cm^3/hr-in^3/hr$)	Cooling Air Flow Rate (scmm-scfm)	Cooling Air In/Exhaust Temp. (°K-°F)
20	370-208	51-3.1	0.198-7	350/364-170/195
30	387-234	51-3.1	0.198-7	344/375-160/215
40	397-252	51-3.1	0.198-7	340/390-152/243
55	433-320	51-3.1	0.198-7	342/418-157/290
60	445-340	51-3.1	0.198-7	342/428-157/310
65	451-352	51-3.1	0.198-7	341/431-155/318

Shaft Speed (rpm)	Mist Air Flow Rate (scmm-scfm)	Mist Air In/Exhaust Temp. (°K-°F)	Total Air Flow Rate (scmm-scfm)
20	0.261-9.22	369/364-205/195	0.459-16.22
30	0.261-9.22	367/375-200/215	0.459-16.22
40	0.261-9.22	364/390-195/243	0.459-16.22
55	0.261-9.22	363/418-193/290	0.459-16.22
60	0.261-9.22	363/428-192/310	0.459-16.22
65	0.261-9.22	364/431-195/318	0.459-16.22

Heat Transfer Rate

Shaft Speed (rpm)	Cooling Air (Watts-Btu/hr)	Mist Air (Watts-Btu/hr)	Total (Watts-Btu/hr)
20	55- 189	*(29)-(100)	26- 89
30	122- 416	44 - 149	165- 565
40	201- 688	140 - 478	341-1166
55	294-1005	283 - 966	577-1971
60	339-1157	344 -1175	683-2332
65	361-1232	359 -1225	719-2457

* ( ) - Indicates minus value.

TABLE III-6

TEST NO. 12 (DECREASED COOLING AIR)

Lubricant - Advanced Ester (MIL-L-23699)

Shaft Speed (rpm)	Brg. O.R. Temp. (°K-°F)	Mist Oil Flow Rate (cm^3/hr-in^3/hr)	Cooling Air Flow Rate (scmm-scfm)	Cooling Air In/Exhaust Temp. (°K-°F)
30	410-280	442-27	0	NA
40	436-325	442-27	0	
55	480-405	442-27	0	
60	500-440	442-27	0	

Shaft Speed (rpm)	Mist Air Flow Rate (scmm-scfm)	Mist Air In/Exhaust Temp. (°K-°F)	Total Air Flow Rate (scmm-scfm)
30	0.283-10	362/388-190/240	0.283-10
40	0.283-10	359/411-185/280	0.283-10
55	0.283-10	358/450-182/350	0.283-10
60	0.283-10	358/467-182/380	0.283-10

Heat Transfer Rate

Shaft Speed (rpm)	Cooling Air (Watts-Btu/hr)	Mist Air (Watts-Btu/hr)	Total (Watts-Btu/hr)
30	NA	158- 539	158- 539
40		300-1024	300-1024
55		530-1811	530-1811
60		606-2070	606-2070

TABLE III-7

TEST NO. 13 (INCREASED COOLING AIR TEMP.)
Lubricant - Advanced Ester (MIL-L-23699)

Shaft Speed (rpm)	Brg. O.R. Temp. (°K-°F)	Mist Oil Flow Rate ($cm^3/hr-in^3/hr$)	Cooling Air Flow Rate (scmm-scfm)	Cooling Air In/Exhaust Temp. (°K-°F)
20	394-250	236-14.4	0.359-12.7	394/387-250/235
25	408-275	236-14.4	0.351-12.4	406/398-270/255
30	417-290	236-14.4	0.345-12.2	409/402-278/265
35	424-303	236-14.4	0.351-12.4	406/408-271/275
40	431-318	236-14.4	0.351-12.4	405/417-269/290
45	441-335	236-14.4	0.351-12.4	406/425-270/305
50	452-355	236-14.4	0.345-12.2	400/435-260/321
55	465-378	236-14.4	0.351-12.4	395/440-250/334
60	480-405	236-14.4	0.351-12.4	401/450-262/350

Shaft Speed (rpm)	Mist Air Flow Rate (scmm-scfm)	Mist Air In/Exhaust Temp. (°K-°F)	Total Air Flow Rate (scmm-scfm)
20	0.142-5	352/387-175/235	0.501-17.7
25	0.142-5	352/398-175/255	0.492-17.4
30	0.142-5	350/402-170/265	0.488-17.2
35	0.142-5	350/408-171/275	0.492-17.4
40	0.142-5	350/417-171/290	0.492-17.4
45	0.142-5	350/425-170/305	0.492-17.4
50	0.142-5	350/435-170/321	0.488-17.2
55	0.142-5	349/440-168/334	0.492-17.4
60	0.142-5	347/450-162/350	0.492-17.4

Heat Transfer Rate

Shaft Speed (rpm)	Cooling Air (Watts-Btu/hr)	Mist Air (Watts-Btu/hr)	Total (Watts-Btu/hr)
20	*(60)-(205)	95-324	35-119
25	(59)-(200)	126-432	68-232
30	(50)-(170)	150-513	100-343
35	(16)-(53)	164-561	149-508
40	82-279	188-642	270-921
45	136-465	213-729	349-1191
50	234-800	239-815	473-1615
55	327-1117	254-869	581-1986
60	343-1170	297-1015	640-2185

* ( ) - Indicates minus value.

TABLE III-8

TEST NO. 14 (BAFFLE PLATE)

Lubricant - Advanced Ester (MIL-L-23699)

Shaft Speed (rpm)	Brg. O.R. Temp. (°K-°F)	Mist Oil Flow Rate ($cm^3/hr-in^3/hr$)	Cooling Air Flow Rate (scmm-scfm)	Cooling Air In/Exhaust Temp. (°K-°F)
26,000	383-230	451-27.5	0.184-6.5	350/372-170/210
40,000	414-285	451-27.5	0.187-6.6	339/391-150/245
55,000	444-340	451-27.5	0.190-6.7	333/419-140/295
60,000	461-370	451-27.5	0.190-6.7	333/430-140/315
65,000	467-380	451-27.5	0.190-6.7	333/433-140/320

Shaft Speed (rpm)	Mist Air Flow Rate (scmm-scfm)	Mist Air In/Exhaust Temp. (°K-°F)	Total Air Flow Rate (scmm-scfm)
26,000	0.321-11.34	364/372-197/210	0.505-17.84
40,000	0.321-11.34	364/391-197/245	0.508-17.94
55,000	0.321-11.34	362/419-192/295	0.511-18.04
60,000	0.321-11.34	360/430-190/315	0.511-18.04
65,000	0.321-11.34	359/433-188/320	0.511-18.04

Heat Transfer Rate

Shaft Speed (rpm)	Cooling Air (Watts-Btu/hr)	Mist Air (Watts-Btu/hr)	Total (Watts-Btu/hr)
26,000	82- 280	47- 159	129- 439
40,000	198- 677	172- 587	370-1264
55,000	328-1120	369-1261	697-2381
60,000	370-1264	448-1530	818-2794
65,000	381-1300	473-1616	853-2916

TABLE III-9

TEST NO. 15 (STRAIGHT NOZZLES, NO SCREEN)

Lubricant - Advanced Ester (MIL-L-23699)

Shaft Speed (rpm)	Brg. O.R. Temp. (°K-°F)	Mist Oil Flow Rate (cm^3/hr-in^3/hr)	Cooling Air Flow Rate (scmm-scfm)	Cooling Air In/Exhaust Temp. (°K-°F)
26,000	386-235	475-29	0.251-8.9	346/374-165/215
40,000	408-272	475-29	0.221-7.8	331/387-137/235
55,000	447-345	475-29	0.221-7.8	328/413-130/285
60,000	452-355	475-29	0.221-7.8	328/418-130/290
65,000	464-375	475-29	0.221-7.8	339/423-150/300

Shaft Speed (rpm)	Mist Air Flow Rate (scmm-scfm)	Mist Air In/Exhaust Temp. (°K-°F)	Total Air Flow Rate (scmm-scfm)
26,000	0.321-11.34	345/374-160/215	0.573-20.24
40,000	0.321-11.34	356/387-180/235	0.549-19.14
55,000	0.321-11.34	352/413-175/285	0.549-19.14
60,000	0.321-11.34	350/418-170/290	0.549-19.14
65,000	0.321-11.34	354/423-178/300	0.549-19.14

Heat Transfer Rate

Shaft Speed (rpm)	Cooling Air (Watts-Btu/hr)	Mist Air (Watts-Btu/hr)	Total (Watts-Btu/hr)
26,000	144- 493	197- 673	341-1166
40,000	242- 825	197- 673	439-1498
55,000	382-1305	394-1346	776-2651
60,000	395-1348	430-1469	825-2817
65,000	370-1264	437-1493	807-2757

TABLE III-10

TEST NO. 16 (DRIP MIST SYSTEM)

Lubricant - Advanced Ester (MIL-L-23699)

Shaft Speed (rpm)	Brg. O.R. Temp. (°K-°F)	Mist Oil Flow Rate ($cm^3/hr-in^3/hr$)	Cooling Air Flow Rate (scmm-scfm)	Cooling Air In/Exhaust Temp. (°K-°F)
26,000	380-225	295-18	0.458-16.2	338/362-150/192
35,000	401-262	295-18	0.458-16.2	338/379-150/222
40,000	414-287	360-22	0.458-16.2	336/389-145-240
45,000	427-310	442-27	0.495-17.5	338/398-150/258
50,000	441-336	475-29	0.495-17.5	338/408-150/275
55,000	451-352	524-32	0.495-17.5	336/416-145/290
60,000	462-372	475-29	0.495-17.5	333/424-140/306
65,000	478-400	360-22	0.495-17.5	336/440-145/335

Shaft Speed (rpm)	Mist Air Flow Rate (scmm-scfm)	Mist Air In/Exhaust Temp. (°K-°F)	Total Air Flow Rate (scmm - scfm)
26,000	Not Used	Not Applicable	0.458-16.2
35,000			0.458-16.2
40,000			0.458-16.2
45,000			0.495-17.5
50,000			0.495-17.5
55,000			0.495-17.5
60,000			0.495-17.5
65,000			0.495-17.5

Heat Transfer Rate

Shaft Speed (rpm)	Cooling Air (Watts-Btu/hr)	Mist Air (Watts-Btu/hr)	Total (Watts-Btu/hr)
26,000	216-737	Not Applicable	216-737
35,000	370-1263		370-1263
40,000	488-1667		488-1667
45,000	598-2041		598-2041
50,000	692-2363		692-2363
55,000	802-2740		802-2740
60,000	919-3137		919-3137
65,000	1051-3591		1051-3591

TABLE III-11

TEST NO. 17 (INCREASED CAGE CLEARANCE)

Lubricant - Advanced Ester (MIL-L-23699)

Shaft Speed (rpm)	Brg. O.R. Temp. (°K-°F)	Mist Oil Flow Rate (cm^3/hr-in^3/hr)	Cooling Air Flow Rate (scmm-scfm)	Cooling Air In/Exhaust Temp. (°K-°F)
15,000	410-280	426-26	0.252-8.9	Bad /402-Bad /263
20,000	430-305	426-26	0.252-8.9	T/C /408-T/C /275
35,000	452-355	426-26	0.252-8.9	T/C /428-T/C /310
40,000	451-353	426-26	0.252-8.9	T/C /441-T/C /335
45,000	478-400	426-26	0.252-8.9	T/C /456-T/C /360

Shaft Speed (rpm)	Mist Air Flow Rate (scmm-scfm)	Mist Air In/Exhaust Temp. (°K-°F)	Total Air Flow Rate (scmm-scfm)
15,000	0.275-9.7	358/402-185/263	0.526-18.6
20,000	0.275-9.7	357/408-183/275	0.526-18.6
35,000	0.275-9.7	355/428-180/310	0.526-18.6
40,000	0.275-9.7	354/441-178/335	0.526-18.6
45,000	0.275-9.7	354/456-178/360	0.526-18.6

Heat Transfer Rate

Shaft Speed (rpm)	Cooling Air (Watts-Btu/hr)	Mist Air (Watts-Btu/hr)	Total - (Calculated from Brg. Torque) (Watts-Btu/hr)
15,000	Not Applicable	(Housing Heaters On)	97- 331
20,000			136- 463
35,000			306-1040
40,000			401-1369
45,000			——

TABLE III-12

TEST NO. 18 (PUMPING HOLES THROUGH CAGE RAILS)

Lubricant - Advanced Ester (MIL-L-23699)

Shaft Speed (rpm)	Brg. O.R. Temp. (°K-°F)	Mist Oil Flow Rate ($cm^3/hr-in^3/hr$)	Cooling Air Flow Rate (scmm-scfm)	Cooling Air In/Exhaust Temp. (°K-°F)
15,000	350-170	359-28	0.289-10.2	302/344-85/160
25,000	372-210	359-28	0.289-10.2	302/359-85/185
35,000	389-240	359-28	0.289-10.2	302/370-85/208
45,000	407-272	359-28	0.289-10.2	302/380-85/225
55,000	428-310	359-28	0.289-10.2	302/389-85/240
65,000	452-356	359-28	0.289-10.2	302/412-85/285

Shaft Speed (rpm)	Mist Air Flow Rate (scmm-scfm)	Mist Air In/Exhaust Temp. (°K-°F)	Total Air Flow Rate (scmm-scfm)
15,000	0.385-13.6	371/344-210/160	0.674-23.8
25,000	0.385-13.6	369/359-205/185	0.674-23.8
35,000	0.385-13.6	369/370-205/208	0.674-23.8
45,000	0.385-13.6	366/380-200/225	0.674-23.8
55,000	0.385-13.6	363/389-195/240	0.674-23.8
65,000	0.246- 8.7	356/412-180/285	0.515-18.9

Heat Transfer Rate

Shaft Speed (rpm)	Cooling Air (Watts-Btu/hr)	Mist Air (Watts-Btu/hr)	Total (Watts-Btu/hr)
15,000	242- 825	*(193)-(661)	48- 164
25,000	322-1100	(83)-(284)	239- 816
35,000	396-1353	13 - 44	409-1397
45,000	450-1540	107 - 367	458-1907
55,000	499-1705	193 - 660	693-2368
65,000	628-2145	293 -1000	921-3145

* ( ) - Indicates minus value.

TABLE III-13

TEST NO. 19 (INNER RING GUIDED CAGE)

Lubricant - Advanced Ester (MIL-L-23699)

Shaft Speed (rpm)	Brg. O.R. Temp. (°K-°F)	Mist Oil Flow Rate ($cm^3/hr-in^3/hr$)	Cooling Air Flow Rate (scmm-scfm)	Cooling Air In/Exhaust Temp. (°K-°F)
20	394-250	524-32	0.254-9	341/389-155/240
30	414-285	524-32	0.254-9	344/411-160/280
35	428-308	524-32	0.254-9	344/422-160/300
40	430-315	524-32	0.254-9	344/428-160/310
45	434-321	524-32	0.254-9	352/430-176/315

Shaft Speed (rpm)	Mist Air Flow Rate (scmm-scfm)	Mist Air In/Exhaust Temp. (°K-°F)	Total Air Flow Rate (scmm-scfm)
20	0.385-13.6	363/389-195/240	0.640-22.6
30	0.387-13.6	363/411-195/280	0.640-22.6
35	0.387-13.6	366/422-200/300	0.640-22.6
40	0.387-13.6	366/428-200/310	0.640-22.6
45	0.387-13.6	366/430-200/315	0.640-22.6

Heat Transfer Rate

Shaft Speed (rpm)	Cooling Air (Watts-Btu/hr)	Mist Air (Watts-Btu/hr)	Total (Watts-Btu/hr)
20	242- 826	237- 808	478-1634
30	341-1166	409-1396	750-2562
35	398-1360	430-1470	829-2830
40	427-1458	473-1617	900-3075
45	396-1351	495-1690	891-3041

TABLE III-14

TEST NO. 20 (PUMPING GROOVES IN CAGE RAIL)

Lubricant - Advanced Ester (MIL-L-23699)

Shaft Speed (rpm)	Brg. O.R. Temp. (°K-°F)	Mist Oil Flow Rate ($cm^3/hr-in^3/hr$)	Cooling Air Flow Rate (scmm-scfm)	Cooling Air In/Exhaust Temp. (°K-°F)
25	386-235	459-28	0.28-10	333/373-140/212
30	395-250	459-28	0.28-10	333/380-140/225
35	403-265	459-28	0.28-10	333/387-140/235
40	407-272	459-28	0.28-10	333/397-140/252
45	439-330	459-28	0.28-10	333/409-140/278

Shaft Speed (rpm)	Mist Air Flow Rate (scmm-scfm)	Mist Air In/Exhaust Temp. (°K-°F)	Total Air Flow Rate (scmm-scfm)
25	0.38-13.6	359/373-185/212	0.67-23.8
30	0.38-13.6	359/380-185/225	0.67-23.8
35	0.38-13.6	359/387-185/235	0.67-23.8
40	0.38-13.6	359/397-185/252	0.67-23.8
45	0.38-13.6	359/406-185/278	0.67-23.8

Heat Transfer Rate

Shaft Speed (rpm)	Cooling Air (Watts-Btu/hr)	Mist Air (Watts-Btu/hr)	Total (Watts-Btu/hr)
——	Not Applicable (Housing Heater on During Part of Test).		

TABLE III-15

TEST NO. 21 (FIFTY HOUR TEST)

Lubricant - Advanced Ester (MIL-L-23699)

Shaft Speed (rpm)	Brg. O.R. Temp. (°K-°F)	Mist Oil Flow Rate ($cm^3/hr-in^3/hr$)	Cooling Air Flow Rate (scmm-scfm)	Cooling Air In/Exhaust Temp. (°K-°F)
55,000	450-350	623-38	0.194-6.8	353/411-175/280

Shaft Speed (rpm)	Mist Air Flow Rate (scmm-scfm)	Mist Air In/Exhaust Temp. (°K-°F)	Total Air Flow Rate (scmm-scfm)
55,000	0.385-13.6	371/411-210/280	0.577-20.4

Heat Transfer Rate

Shaft Speed (rpm)	Cooling Air (Watts-Btu/hr)	Mist Air (Watts-Btu/hr)	Total (Watts-Btu/hr)
55,000	228-778	302-1031	530-1809

TABLE III-16

TEST NO. 22 (100 HOUR TEST)
Lubricant - Mobil Jet II (MIL-L-23699)
Typical Conditions

Shaft Speed (rpm)	Brg. O.R. Temp. (°K-°F)	Mist Oil Flow Rate ($cm^3/hr-in^3/hr$)	Cooling Air Flow Rate (scmm-scfm)	Cooling Air In/Exhaust Temp. (°K-°F)
44,000	470-385	284-16.7	0.337-11.9	406/428-270/310

Shaft Speed (rpm)	Mist Air Flow Rate (scmm-scfm)	Mist Air In/Exhaust Temp. (°K-°F)	Total Air Flow Rate (scmm-scfm)
44,000	0.150-5.3	348/428-165/310	0.487-17.2

Heat Transfer Rate

Shaft Speed (rpm)	Cooling Air (Watts-Btu/hr)	Mist Air (Watts-Btu/hr)	Total (Watts-Btu/hr)
44,000	151-514	243-830	394-1344