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TEST RESULTS REPORT AND DESIGN TECHNOLOGY DEVELOPMENT REPORT - HLH/ATC HIGH-SPEED TAPERED ROLLER BEARING DEVELOPMENT PROGRAM

Joseph W. Lenski, Jr.

Boeing Vertol Company

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The high-speed and load capability of tapered roller bearings has been demonstrated by several research and development programs conducted jointly by the Boeing Vertol Company and the Timken Company. Tests were conducted on a 3.5-inch bore (6500 series) tapered roller bearing operating at 16,000 rpm (20,000 fpm) and supporting a thrust load of 5500 pounds and a radial

20 Continued.

load of 7500 pounds. Although the tests were successful and demonstrated the high-speed capability of tapered roller bearings, additional experimental testing appeared necessary to evaluate the tapered roller bearings selected for the HLH/ATC drive system.

The bearings designed to support various HLH transmission spiral bevel gears exceeded some of the design parameters tested on previous programs. Some of the parameters not tested were increased bearing size, increased loading, higher cone rib tangential velocity, higher contact angles and higher cone rib stresses. Additional testing and development was, therefore, considered necessary to evaluate the effects of these parameters on the performance of the HLH tapered roller bearings.

This report presents the results of efforts, conducted between September 1971 and August 1973, to define the design criteria and to design, fabricate, and test high-speed tapered roller bearings for the HLH/ATC combining transmission engine input pinion gear in accordance with Contract DAAJ01-71-C-0840(P40). Tests were performed on two sizes of tapered roller bearingss operating at 11,500 rpm and 14,000 rpm and loads equivalent to 8310 horsepower. Technical inspection and evaluation of the test results were used for selecting the initial end-play setting, oil-flow rates, and internal geometry of tapered roller bearings to be used in the HLH/ATC drive system.

Testing conducted during this development program has provided information necessary to successfully operate high-speed tapered roller bearings in the HLH/ATC applications. A limited endurance test of 150 hours was completed at the end of this program.

EUSTIS DIRECTORATE POSITION STATEMENT

The results of this program show that tapered roller bearings can operate at the high speeds and loads dictated by the HLH design; however, current technology and materials have not been developed to a point where it is possible to make a fail-safe (30 minutes operation without oil) bearing for operation at these speeds.

This directorate concurs with the conclusions presented herein.

The technical monitor for this effort was Mr. Wayne A. Hudgins, Heavy Lift Helicopter Project Office, Systems Support Division.

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PREFACE

The high-speed tapered roller bearing development work presented in this report has been completed in partial fulfillment of Contract DAAJ01-71-C-0840(P40). This effort was primarily a development program to define design criteria and to design, fabricate, and test HLH tapered roller bearings at the loads and speed of the HLH/ATC combining transmission input pinion gear.

This program was conducted at the Timken Company, Canton, Ohio, under the technical direction of Robert F. Cornish, Application Development Engineer, with Gary Dressler as principal investigator. Liaison with regard to program status and test results was maintained by Joseph W. Lenski, Jr., senior design engineer of the Boeing Vertol Company. Technical direction was provided by Wayne Hudgins, Project Engineer, Eustis Directorate, U.S. Army Air Mobility Research and Development Laboratory, Fort Eustis, Virginia.

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INTRODUCTION

This bearing program was directed toward the development of tapered roller bearings for the HLH/ATC aft and combining transmission by testing and evaluating an HM926700-series tapered roller bearing at the loads and speed of the HLH/ATC combining transmission engine input pinion gear.* The bearing was chosen for testing based on preliminary design of the combining transmission input pinion and its severe operating conditions. The operating conditions of this test program which simulated the HLH/ATC combining transmission input pinion were:

Shaft speed	11,500 rpm (normal) 14,000 rpm (autorotation)
Bearing mounting	Duplex indirect mounting
Applied load to thrust-carrying bearing (equivalent to 8310 hp)	Thrust 4394 lb Radial 15,619 lb
Lubricant	MIL-L-23699
Lubricant inlet temperature	195°F

Tests were conducted under these conditions to evaluate and establish criteria for the basic tapered roller design parameters for high-speed operation, oil-flow requirements, initial end-play settings, and emergency self-lubrication requirements. Upon completion of the development testing, a final verification (endurance) test of 150 hours was conducted on the final design of the HLH/ATC combining transmission input pinion bearing, P/N 301-10676-1.

To accomplish the previously mentioned objectives, a new high-speed test machine was designed and fabricated which closely simulated a transmission spiral bevel gear support bearing arrangement. In addition, several other test rigs were designed and fabricated during this program to provide needed data on oil distribution to the cone rib flange and to evaluate various fail-safe, wear-resistant materials. Forty individual tests were conducted on 67 HM926700-series and 301-10676-1 bearings. The results of these development tests are summarized in this test report.

*Boeing Vertol Report D301-10027-1, Rev. A., Test Plan - HLH/ATC Combining Transmission High Speed Tapered Roller Bearing Selection Test.

After completion of this program, additional testing of the 301-10676-1 bearing was authorized by Contract DAAJ01-73-A-0017 to establish fatigue or endurance data on high-speed tapered roller bearings. An additional 891 hours of testing was conducted on four bearings. The results of that program are reported in Boeing Vertol Report T301-10249-1 and are presented in Appendix IX of this report.

TECHNICAL APPROACH

BACKGROUND

In 1968, the Boeing Company's Vertol Division sponsored a tapered roller bearing research program with the Timken Company as subcontractor. The objective of this initial program was to develop high-speed tapered roller bearings to support spiral bevel gearing in advanced helicopter transmissions and drive systems. To provide a test specimen size for this development program, design studies were conducted using the CH-47C engine transmission. This transmission provided for a high-speed and high-load application with a potential growth from 2200 horsepower to 4500 horsepower within the same envelope size. The results of this study indicated that a 6500-series (3.50-inch bore) tapered roller bearing could be used to support the spiral bevel input pinion and provide improved operating characteristics over the present ball/roller bearing configuration.

Prior to this initial program, designers were aware that tapered roller bearings had the potential to carry heavier radial and thrust loads per pound of weight than any other bearing; however, the tapered roller bearings were restricted to relatively low speeds (less than 10,000 fpm cone rib velocity). The bearing proposed for the test program was to be operated at 16,000 rpm and have a cone rib velocity of 20,000 fpm. Under the Boeing Vertol/Timken IR&D program, a total of 134 test bearings of the 6500 series were tested and evaluated. The test program covered modifications to the bearing geometry, cage design, and lubrication concepts and provided assurance that the bearing could be operated at high speeds and loads. The test results also identified the need to optimize various design parameters and develop an understanding of the interactions of these parameters on bearing performance.

To continue this effort, the Boeing Company's Vertol Division was awarded a contract (DAAJ02-71-C-0025) from the Eustis Directorate in March of 1971 to design, fabricate, test, and evaluate spiral bevel support tapered roller bearings. The program consisted of a generalized analytical investigation and an experimental investigation which was based on the information attained in the analytical investigation. The experimental investigation was conducted on 53 tapered roller bearings of the 6500 series operating at 16,000 rpm with 5500 pounds of thrust and 7500 pounds of radial load. The results of this contract have been published in USAAMRDL Technical Report 73-16. The knowledge obtained from these various test programs indicated that tapered roller bearings could be used to support spiral bevel gears on the HLH/ATC aft and combining transmissions.

The preliminary designs of the HLH aft and combining transmissions showed that the tapered roller bearings selected for various locations resulted in design parameters that exceeded those tested on the 6500-series bearing. Some of the parameters which were not tested under the previous program are:

	HLH Condition	6500-Series Test Condition
Cone rib velocity	25,000 fpm (Autorotation)	20,000 fpm
Contact angle	26.2°	15°
Rib stress	64,147 psi	36,133 psi
Lubricant	MIL-L-23699	MIL-L-7808
Cage design	Z-type race guided centerline cage	Pressed-steel roller-riding cage
Oil inlet temperature	195°F	165°F
Equivalent loading	8310 hp	3600 hp
Maximum cone rib load	915 lb	236 lb

Because of the variations in the design parameters of the proposed HLH tapered roller bearings and those previously tested (6500-series), the need for additional testing was considered necessary to properly evaluate the performance and to establish adequate design criteria for the HLH transmission high-speed tapered roller bearings.

Additional testing was conducted on an HM926700-series tapered roller bearing which was originally selected to support the HLH combining transmission engine input pinion. This bearing, readily available with only minor modification, provided a means of evaluating all the untested parameters mentioned earlier. Prior to the completion of the program, tests were conducted on the actual transmission bearings, P/N 301-10676, which will be used in the HLH/ATC combiner transmission. The testing and fabrication of hardware was subcontracted to the Timken Company in September of 1971 and was completed in August 1973. The results of this effort are documented in this report.

STATEMENT OF PROBLEMS

An initial high-speed tapered roller bearing development program, undertaken by Boeing Vertol Company and Timken Company, indicated the feasibility of the use of tapered roller bearings in a high-speed helicopter transmission. This program proved that a 3.5-inch bore modified tapered roller bearing can be operated at approximately 20,000 fpm cone rib velocity (16,000 rpm). At the conclusion of this initial development program, it was determined that additional work was needed to optimize the bearing design and lubricant and lubrication system parameters. A government contract was obtained to further evaluate the performance of tapered roller bearings analytically and experimentally. This work was reported in USAAMRDL Technical Report 73-16.

Although the high-speed and load capability of tapered roller bearings has been demonstrated, additional test work was necessary to evaluate larger tapered roller bearings for application in the HLH drive system. The increase in bearing sizes, increase in loads, higher speeds, and the steep-angled, HM926700-series bearing were to be evaluated under HLH operating conditions.

The objective of this program was to develop a tapered roller bearing for an HLH application by testing and evaluating the HM926700-series tapered roller bearing at the loads and speed of the HLH combining transmission input pinion gear. The basic tapered roller bearing design parameters, oil-flow rates, initial test machine bearing adjustment, and emergency self-lubrication feasibility were to be determined.

The test program was organized to investigate and evaluate the following objectives:

- Design and build a high-speed tapered roller bearing test machine capable of testing 9-inch to 20-inch OD bearings at speeds up to 14,000 rpm with a maximum applied radial load of 25,000 pounds and a maximum applied thrust load of 10,000 pounds.

- Determine the number and size of radial lubrication oil holes required in the HM926700-series bearing.

- Develop the HM926700-series steep-angle (26° 9') bearing to run at 14,000 rpm. The 14,000-rpm speed is an overspeed condition and was changed during the program to 11,500 rpm, corresponding to the in-flight cruising speed.

- Evaluate selected materials that could be used in a fail-safe bearing design. The evaluation of these materials is to be performed in a Timken lubricant and wear test machine.

- Evaluate two fail-safe bearing designs and determine the wear characteristics of the better of these two designs when operating in oil.

- Run a repeat fail-safe test on four previously tested fail-safe bearings.

- Test additional modified HM926700-series bearings to determine an optimum bearing setting (end-play adjustment) and an optimum oil-flow rate both to the large and small end of the bearing.

- Conduct a tapered roller bearing cone-flange oil distribution spin test to evaluate the effect of the angle and length of the radial lube holes in the cone which will, in turn, provide adequate oil distribution at the cone rib/roller end contact area.

- Manufacture and test the XC11439-series bearing (Boeing Vertol P/N 301-10576) in the high-speed test machine. This bearing represented the actual bearing used in the HLH/ATC combining transmission.

- Test four XC11439-series bearings (Boeing Vertol P/N 301-10676) for 150 hours or until damage occurs.

The results of these tests were used to establish the final design and installation requirements to be used in the high-speed tapered roller bearings in the HLH/ATC aft and combining transmissions.

TEST METHOD

DESCRIPTION OF TEST SPECIMENS

Modified HM926700-Series Bearing

Forty-four bearings similar to the HM926700-series bearings were fabricated to Timken modified code 436. Table I gives some pertinent details of modified code 436. The basic dimensions of the HM926700-series tapered roller bearing are

Cone bore - 5.00 inches
Cup outside diameter - 9.00 inches
Overall width - 2.125 inches

These 44 bearings were modified as described below:

Eleven bearings were tested with cones having seventeen 0.052-inch-diameter holes through the large rib undercut to the back-face chamfer-type oil manifold plus four holes to lubricate the large end flange of the cage (see Figure 1). Four of these cones had an additional seventeen 0.052-inch-diameter holes Electrical Discharge Machined (EDM) between and in the same plane as the original 17 holes. This was necessary to provide a better distribution of oil to the cone rib/roller end contact area. These bearings were tested with machined Z-type race-guided, silver-plated centerline cage.

Eighteen of the 44 bearings were tested with cones having forty 0.040-inch-diameter holes through the large rib undercut to the backface chamfer oil manifold plus four holes to lubricate the large end flange of a Z-type silver-plated cage (see Figure 2).

Eleven of the 44 bearings were manufactured and tested with a modified oil manifold as shown in Figure 3. These cones have forty 0.040-inch-diameter radial lubrication oil holes. Standard stamped-steel, silver-plated cages were used in these bearings.

Four more bearings were manufactured and tested similar to the bearing in Figure 3, except the large rib undercut was reduced to allow more available contact between the rib face and roller spherical end radius. Standard stamped-steel silver-plated cages were used in these bearings.

The cages were silver-plated, according to Federal Specification QQ-S-365a. The plating was a type II, grade B with a matte finish. The cages were processed with a nickel strike and a silver strike before silver-plating to 0.001-inch to 0.002-inch thickness.

TABLE I. SUMMARY OF TIMKEN MODIFIED CODE 436

Cones

Material is standard air melt SAE 4320-steel.

Roller contact surface hardness 58 to 63 Rockwell "C".

100-pct standard double-etched to detect grinding burns.

The cone OD must be tape-honed to 8 microinches maximum.

Class and code to be etched on the part.

All cones checked for minimum hardness on either face.

The cone rib radius must be wrapped around ground; ground or polished tangent to the rib face.

The cone radii, both frontface and backface, must be polished. These radii must be blended into the cone bore.

Cone contacting surfaces must be free from nicks.

Cups

Material is standard air melt SAE 4320-steel.

Rolling contact surface hardness 58 to 63 Rockwell "C".

100-pct standard double-etched to detect grinding burns.

The cup ID must be tape-honed to 8 microinches maximum.

Class and code to be etched on the part.

All cups checked for minimum hardness on either face.

The cup radii, both frontface and backface, must be polished. These radii must blend into the cup OD.

Cup contacting surfaces must be free from nicks.

Rollers

Material is standard air melt SAE 4320-steel.

Rolling contact surface hardness 58 to 63 Rockwell "C".

TABLE I. Continued

The roller body must be honed to 8 microinches maximum.

The roller spherical end must be ground to 6 microinches maximum (not honed).

Sample lots to be standard double etched to detect grinding burns. Samples to be scrapped.

Roller contacting surfaces must be free from nicks.

Cages, Machine Z-Type Race-Guide Centerline

Material is SAE 4319-steel.

The guiding land at cage ID and OD to be ground to 60 microinches maximum.

Special handling to prevent distortion or nicking damage to the cage.

Cages, Standard Stamped Steel

Material is HRLC-steel deep drawing quality P and O.

Cage must not be shot-blasted or tumbled.

Cage must be vapor-blasted or glass-bead peened.

Special handling to prevent distortion or nicking damage to the cage.

Assembled Bearing

Roller spherical end contact with the cone large rib face flat must be as near as possible to the midway point of the distance from the roller recess diameter to the rib face undercut point.

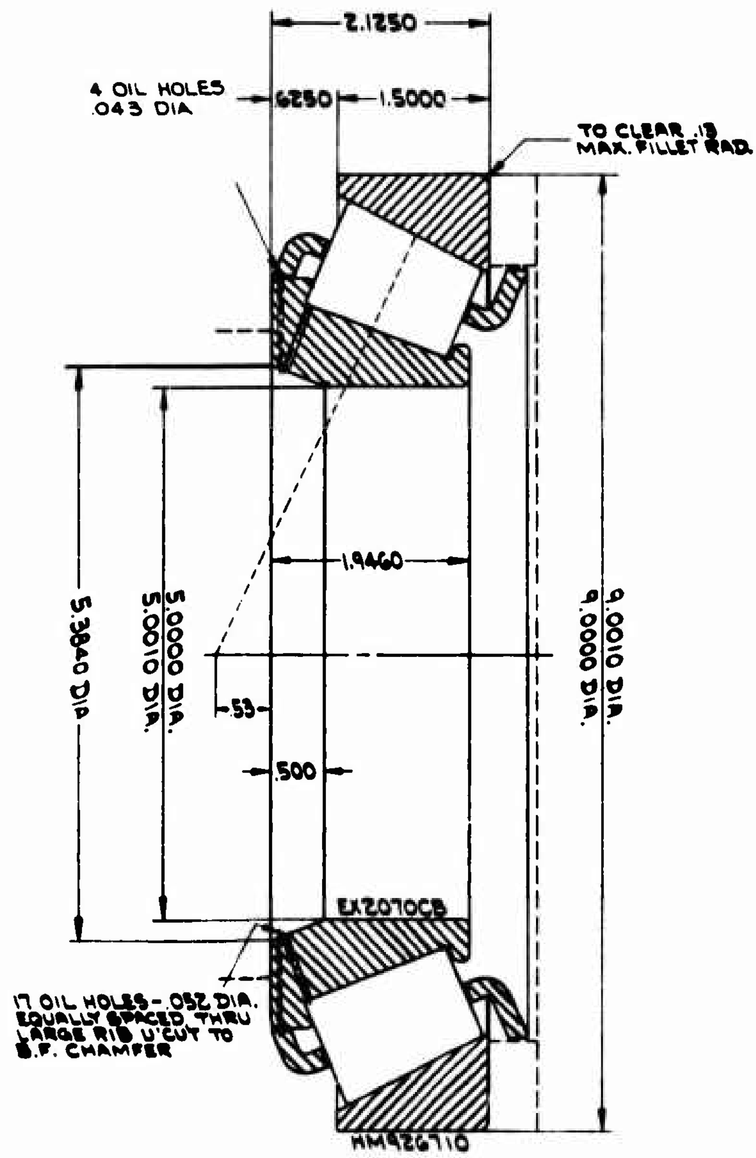

Figure 1. HM926700-Series Bearing Assembly - 17 Oil Holes.

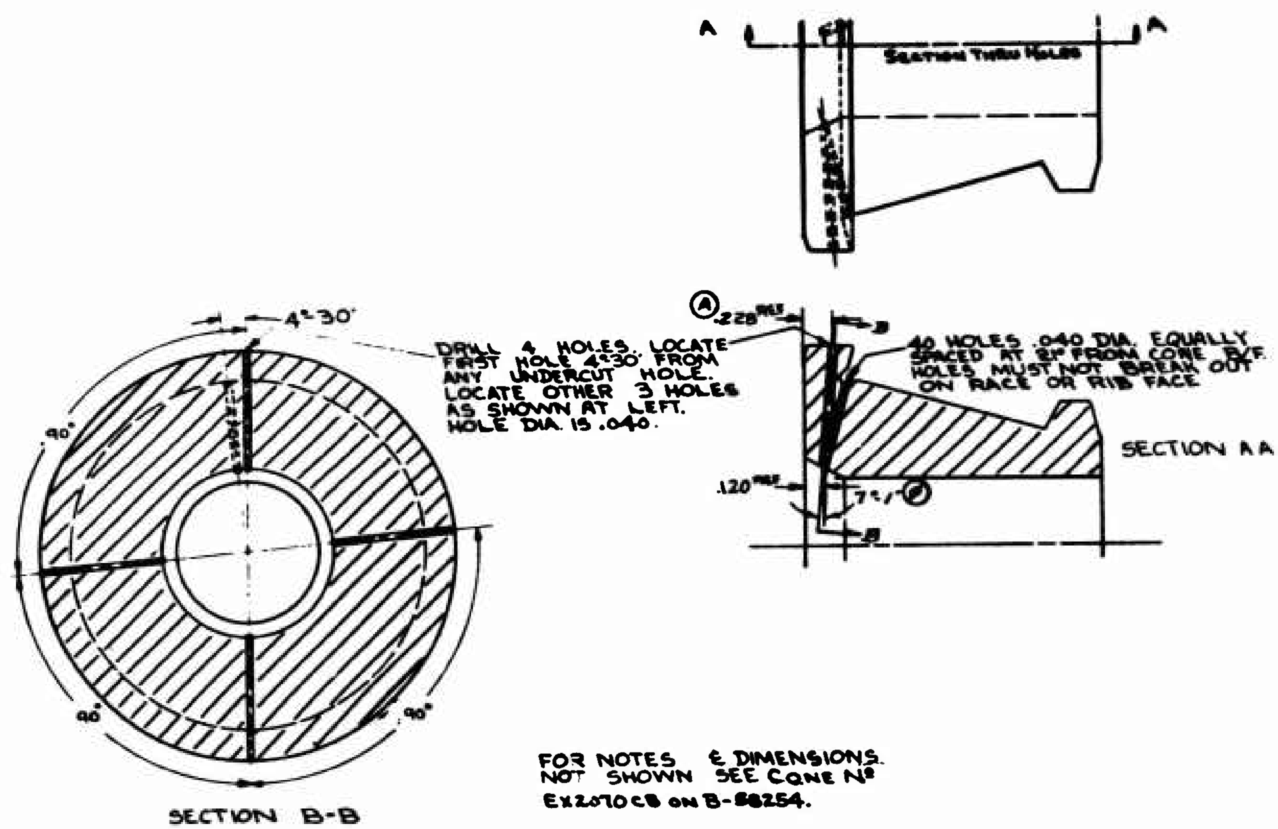

Figure 2. HM926700-Series Bearing - 40 Oil Holes.

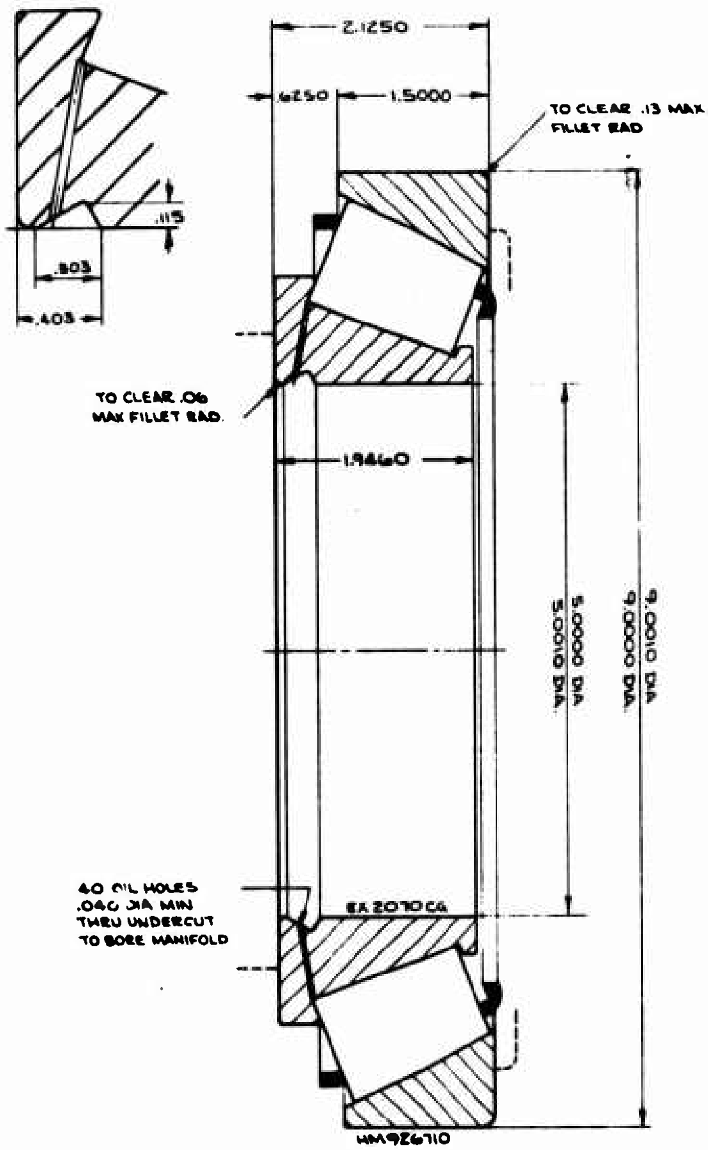

Figure 3. HM926700-Series Bearing Assembly New Manifold - 40 Oil Holes.

Fail-Safe Wear-Resistant Materials

Various wear-resistant materials were evaluated in the Timken lubricant and wear test machine. The machine requires test cages as shown in Figure 4 and test blocks as shown in Figure 5 to conduct these tests. The following material specimens were evaluated:

Sixty-three standard test cups for the Timken lubricant and wear test machine.

Three standard test blocks for the Timken lubricant and wear test machine.

Four test blocks made from DuPont Vespel SP-3 polyimide resin with 15-pct MOS_2 by weight.

Two test blocks made from DuPont Vespel SP-21 polyimide resin with 15-pct graphite by weight.

Two test blocks made from DuPont Vespel SP-22 polyimide resin with 40-pct graphite by weight.

Two copper-impregnated Marlin-Rockwell blocks submitted by Vertol.

One test block made from Kawecki Brylco No. 50 alloy beryllium copper.

Two test blocks made from DuPont LP404A alloy* with 20-pct nickel binder.

One test block made from DuPont LP404A alloy with 40-pct nickel binder.

One test block made from DuPont LP404A alloy with 20-pct 4600 steel binder.

One test block made from DuPont LP404A alloy with 40-pct 4600 steel binder.

One test block made from DuPont LP404A alloy with 20-pct binder composed of 80-pct nickel and 20-pct chrome.

*Metallurgically, LP alloys are hard grains of an intermetallic compound of a Laves Phase structure dispersed in a softer matrix. Superior anti-friction and anti-wear bearing performance is attributed to the combination of hard Laves Phase particles and the softer matrix. All of the DuPont LP alloy test blocks were processed by Wheeling-Pittsburgh Steel Corporation, Principio Alloys Division, Columbus, Ohio.

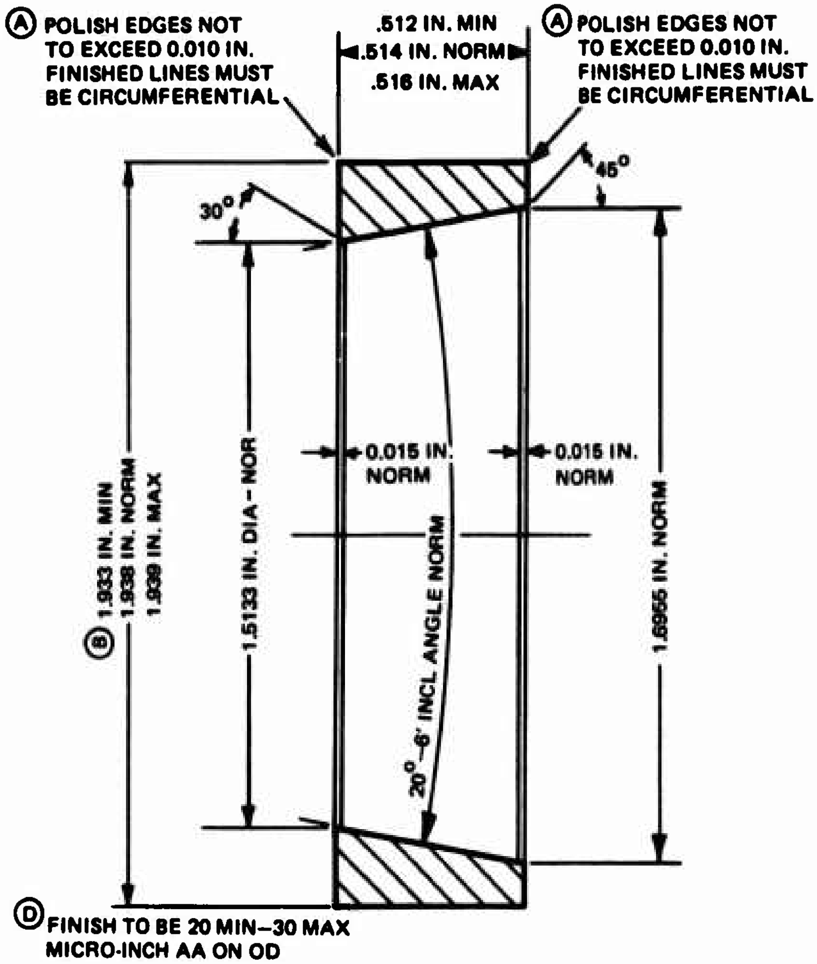

Figure 4. Test Cup for Timken Lubricant and Wear Test Machine.

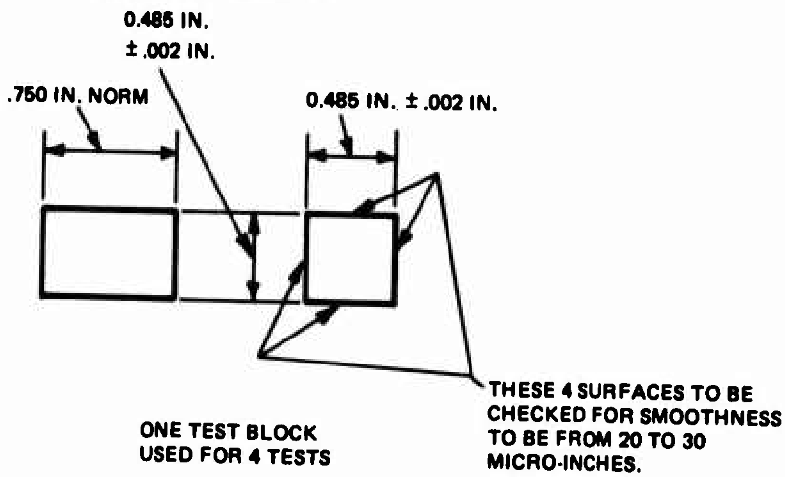

Figure 5. Test Block for Timken Lubricant and Wear Test Machine.

One test block made from DuPont LP404A alloy with 20-pct S-monel binder.

One test block made from DuPont LP404A alloy with 20-pct molybdenum.

Ten standard lubricant test blocks treated with Borkote* diffused to various depths.

Nine standard test cups treated with Borkote* diffused to various depths.

Eleven standard lubricant test blocks treated with Borkote* diffused to various depths.

Eleven standard test cups treated with Borkote* diffused to various depths.

Fail-Safe Bearing Designs

Twelve bearings, similar to the HM926700-series bearings, were fitted with separable large end cone ribs with DuPont LP404 + 40-percent, 4600-steel applied to the rib face (see Figure 6).

Four bearings, similar to the HM926700-series bearings, were fitted with separable large end ribs with Borkote applied to the rib face (see Figure 7).

Both of the designs had 36 radial oil lubrication holes, 0.047 inch in diameter. Both were tested with silver-plated Z-type cages.

Cone Flange Oil Distribution Specimens

One simulated cone, similar to Boeing-Vertol's HLH aft transmission sun gear bearing 301-10443-1 and Timken cone number XC11437, was made according to Figure 8. After testing, this cone was modified, as shown in Figure 9, and tested again.

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*Borkote is a patented boronizing process that is accomplished by enveloping the part to be treated in a proprietary material and heat treating at 1300° to 2100°F. Depth of diffusion of the boride layer is a function of time. This process was developed by Atlantic Advanced Metals, Incorporated, Woburn, Massachusetts.

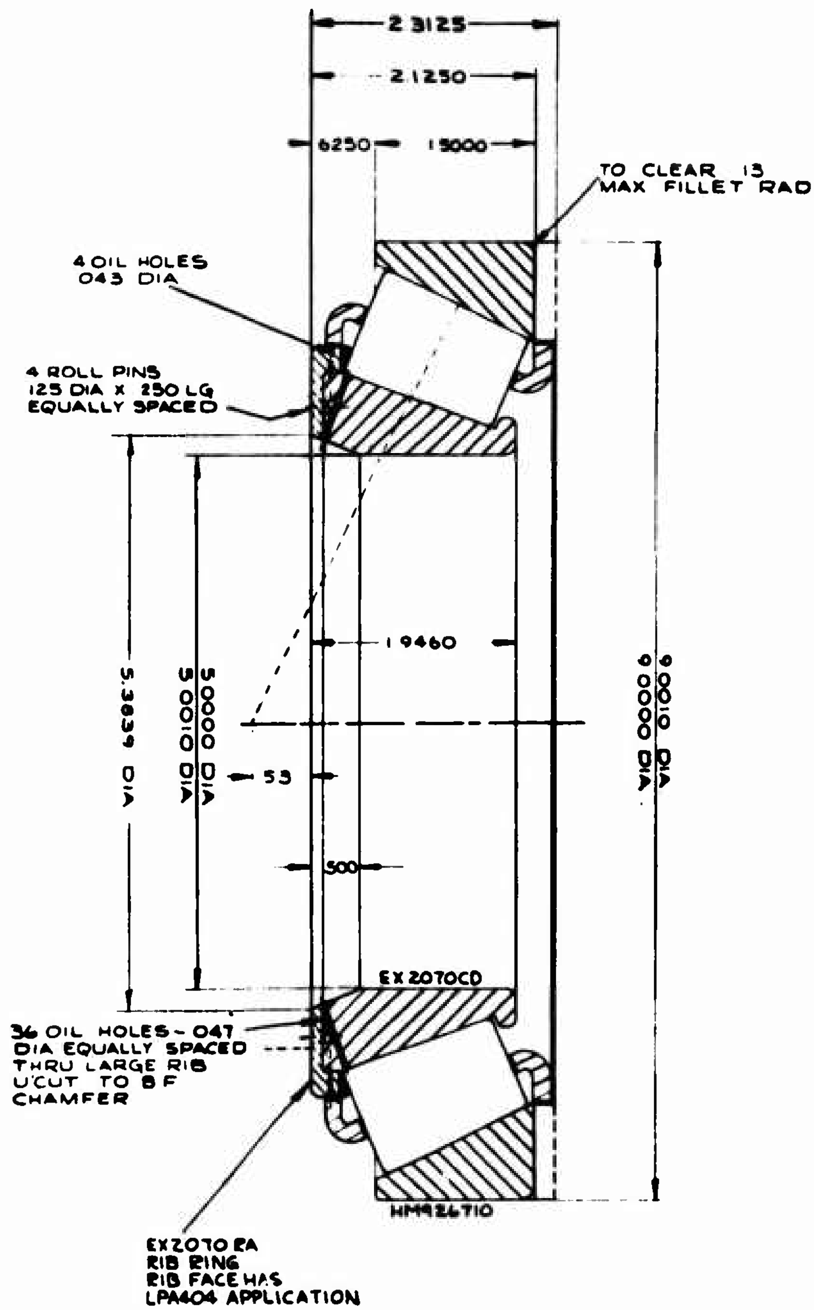

Figure 6. Fail-Safe Bearing With DuPont LP Alloy Separable Rib.

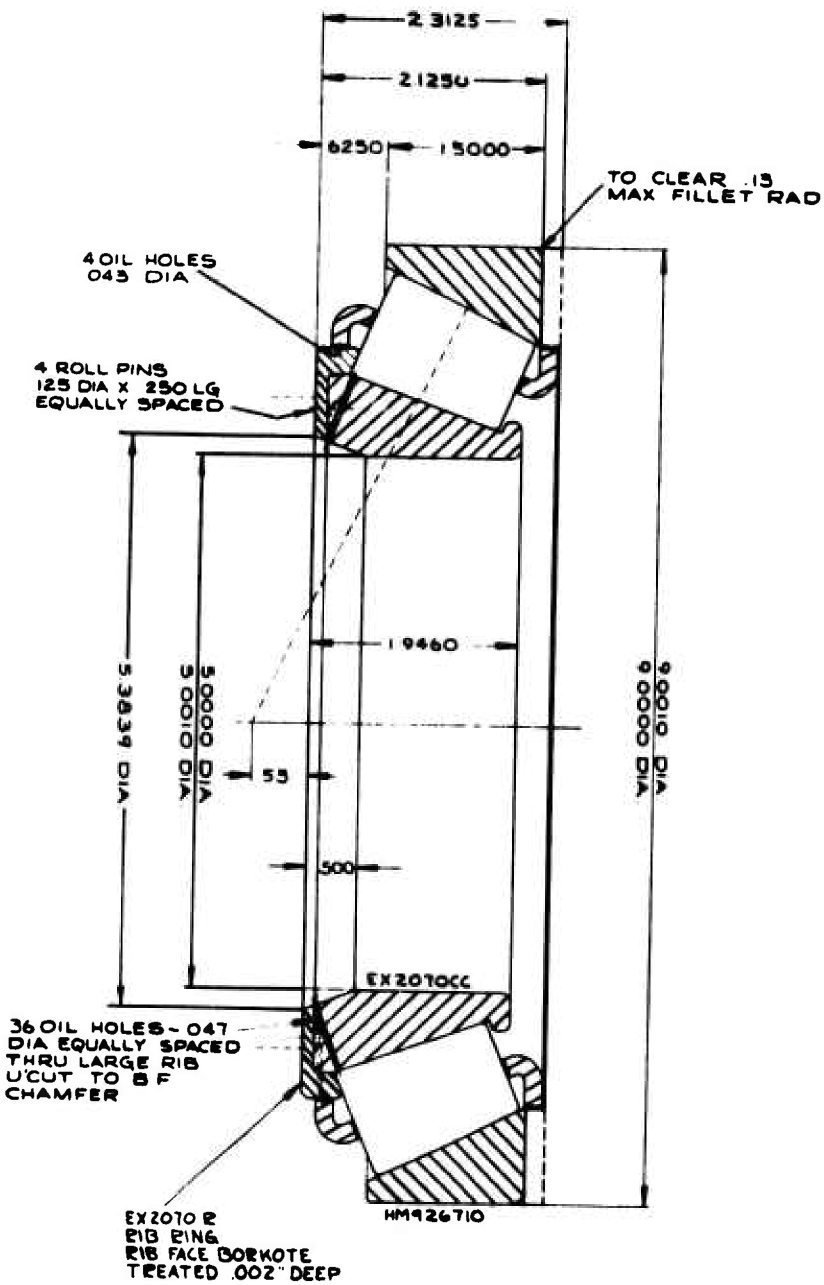

Figure 7. Fail-Safe Bearing With Borkote Separable Rib.

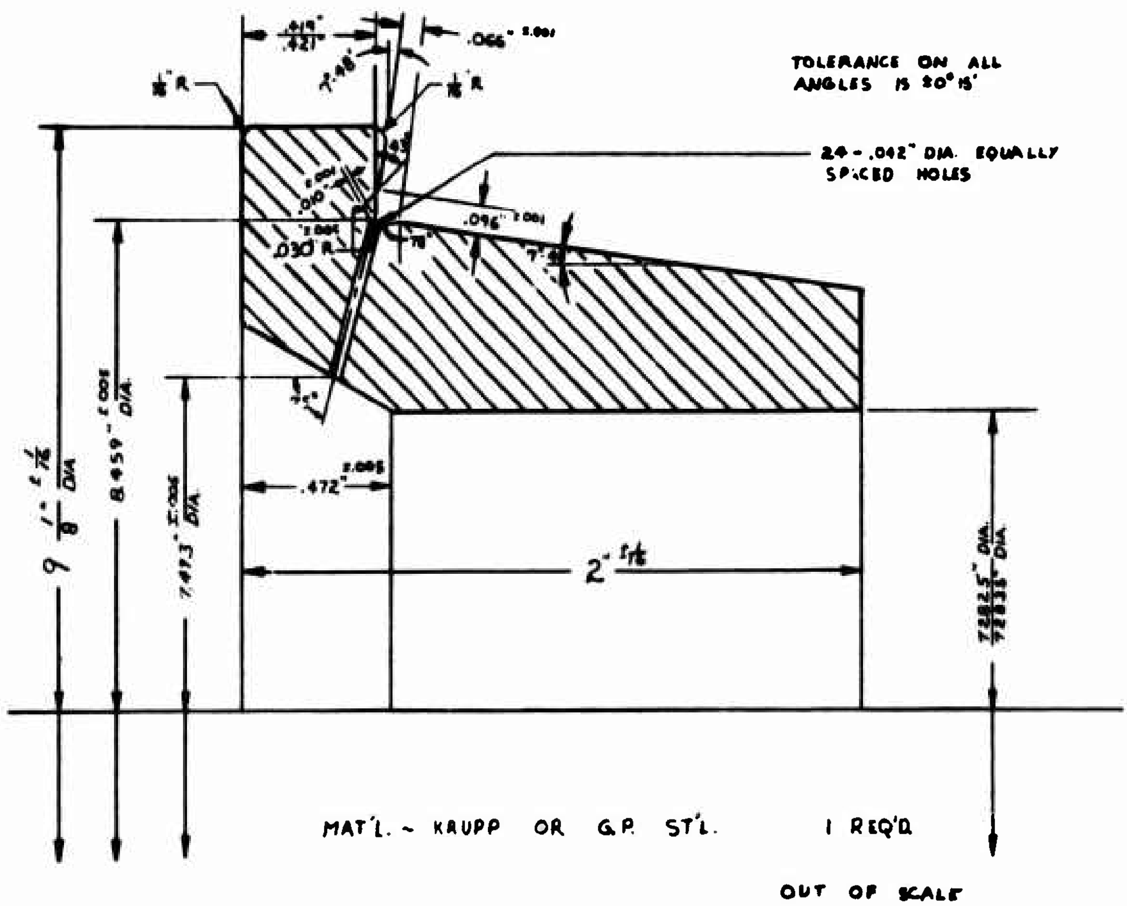

Figure 8. Simulated Cone (EX11437) With Test Number 2 Manifold.

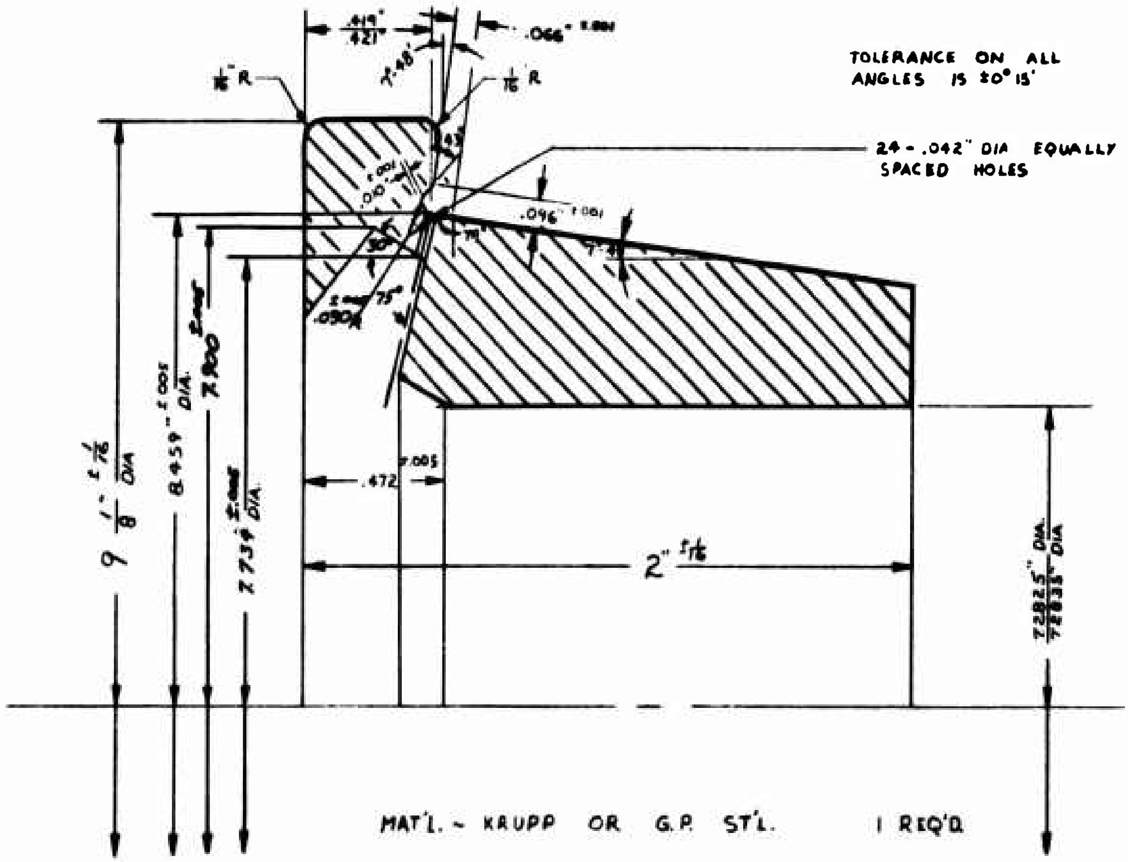

Figure 9. Simulated Cone (EX11437) With Test Number 2 Manifold Modified.

One simulated cone, similar to Boeing-Vertol's HLH combining transmission slant shaft pinion bearing 301-10612-1 and Timken cone number XC12088, was made according to Figure 10. After testing, this same cone was modified, as shown in Figure 11, and tested again.

A third simulated cone, shown in Figure 12, was tested. This cone is similar to Boeing-Vertol's part number 301-10612-1 and Timken part number XC12088. This cone was tested with the 75-degree hole angle and then with the 1-degree hole angle. The 75-degree holes were plugged (see Figure 13).

Bearing P/N 301-10676-1

Seventeen XC11439C- and XC11439D-series bearings (Boeing-Vertol P/N 301-10676) were manufactured and tested (see Figure 14). These bearings were manufactured according to Timken PS601, "Timken Control Requirements for Critical Bearings for Boeing-Vertol HLH Program."

All bearings were tested with machined Z-type, race-guided centerline cages. Nine of these cages were silver-plated by Canton Plating Company. The remaining cages were plated by Lancaster Electro Plating, Incorporated, Lancaster, Ohio--an approved Boeing-Vertol source.

Lubricating Oil

Mobil Jet II, which meets Specification MIL-L-23699D, was used in all tests. This lubricant appears under the formulation RM-1389-A and QPL-23699-6 list dated July 20, 1970, and carries the qualification number 01A.

DESCRIPTION OF TEST MACHINES

High-Speed Tapered Roller Bearing Test Machine

Originally, the test bearings were driven by a 100-horsepower AC constant-speed motor furnished by Boeing-Vertol. After several shakedown runs and the first development test run of the new machine, it was decided to move the test machine to make use of the Timken 700-horsepower dynamometer drive. This drive system permits running the bearings at various speeds to determine what is the speed limit for a particular bearing design modification. This 700-horsepower drive motor, belted to a 9:1 speed increaser, is capable of running the test bearings at a speed of 14,000 rpm. The test machine is shown in Figures 15 and 16.

Figure 17 is a subassembly section of the test machine housing.

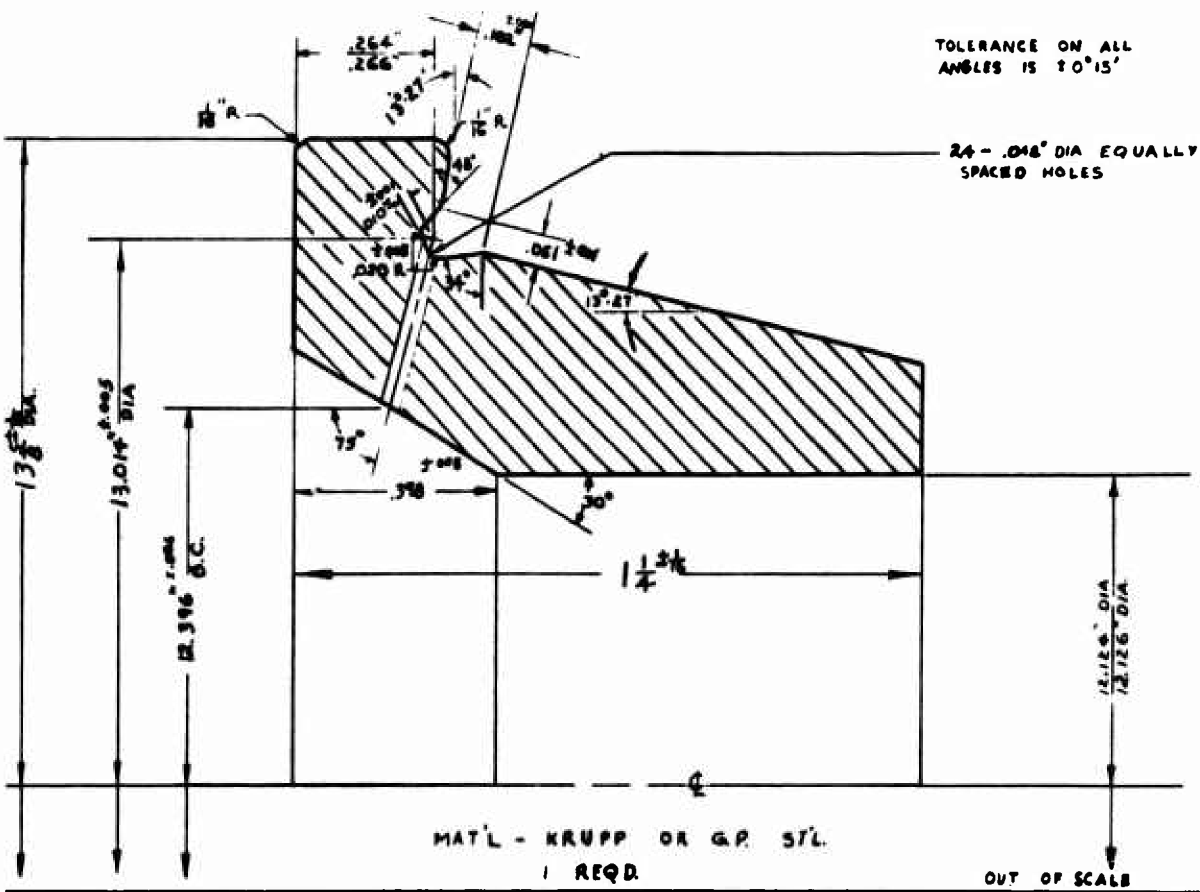

Figure 10. Simulated Cone (EX12088) With Test Number 2 Manifold.

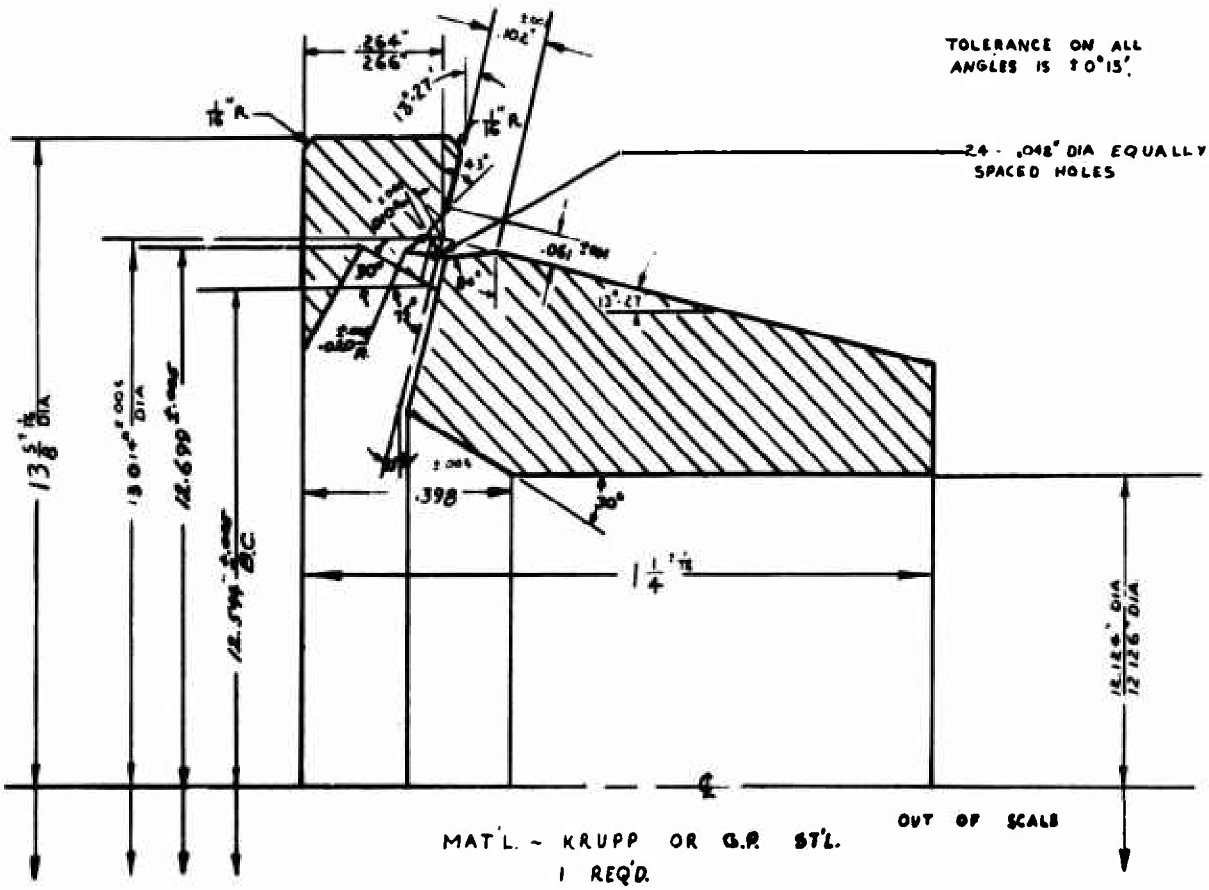

Figure 11. Simulated Cone (EX12088) With Test Number 2 Manifold Modified.

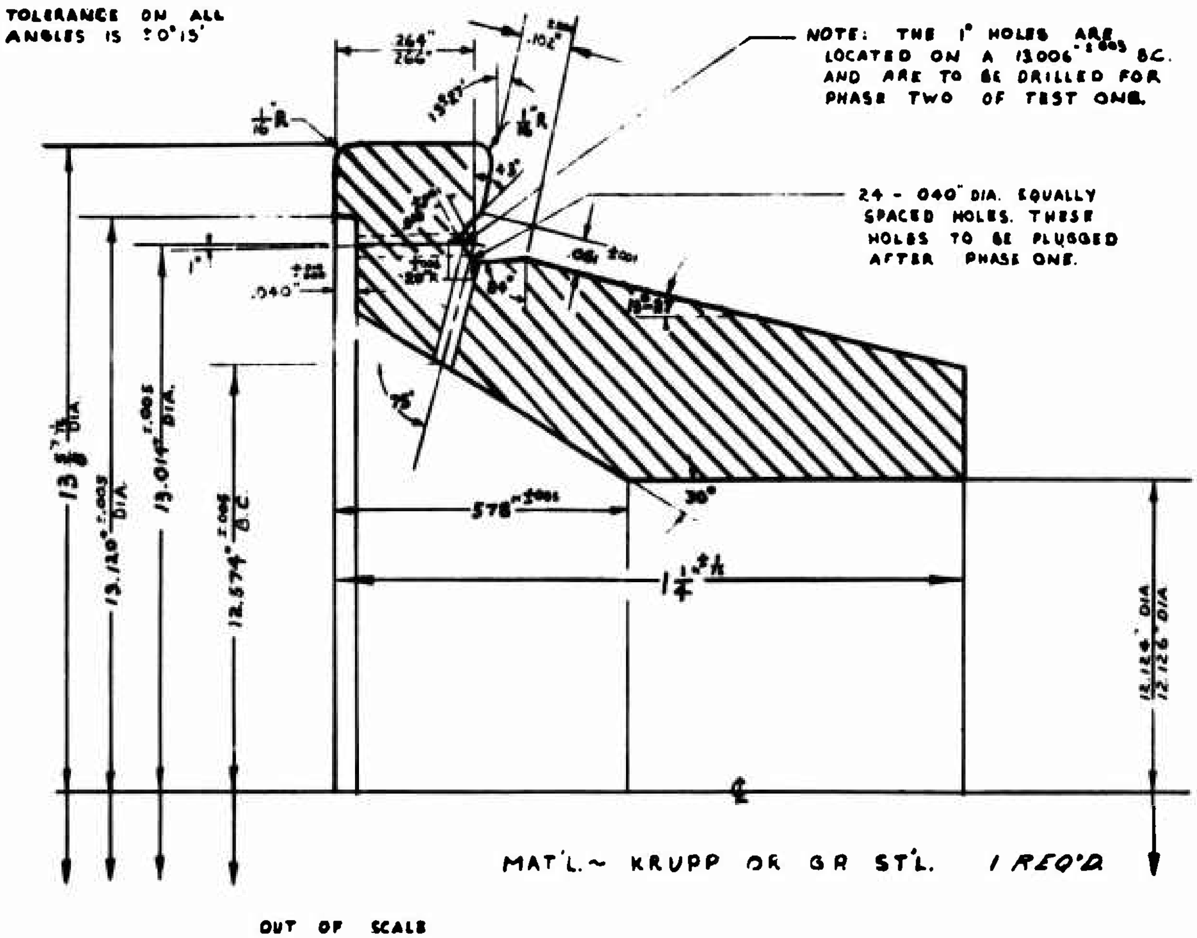

Figure 12. Simulated Cone (EX12088) With Test Number 1 Manifold.

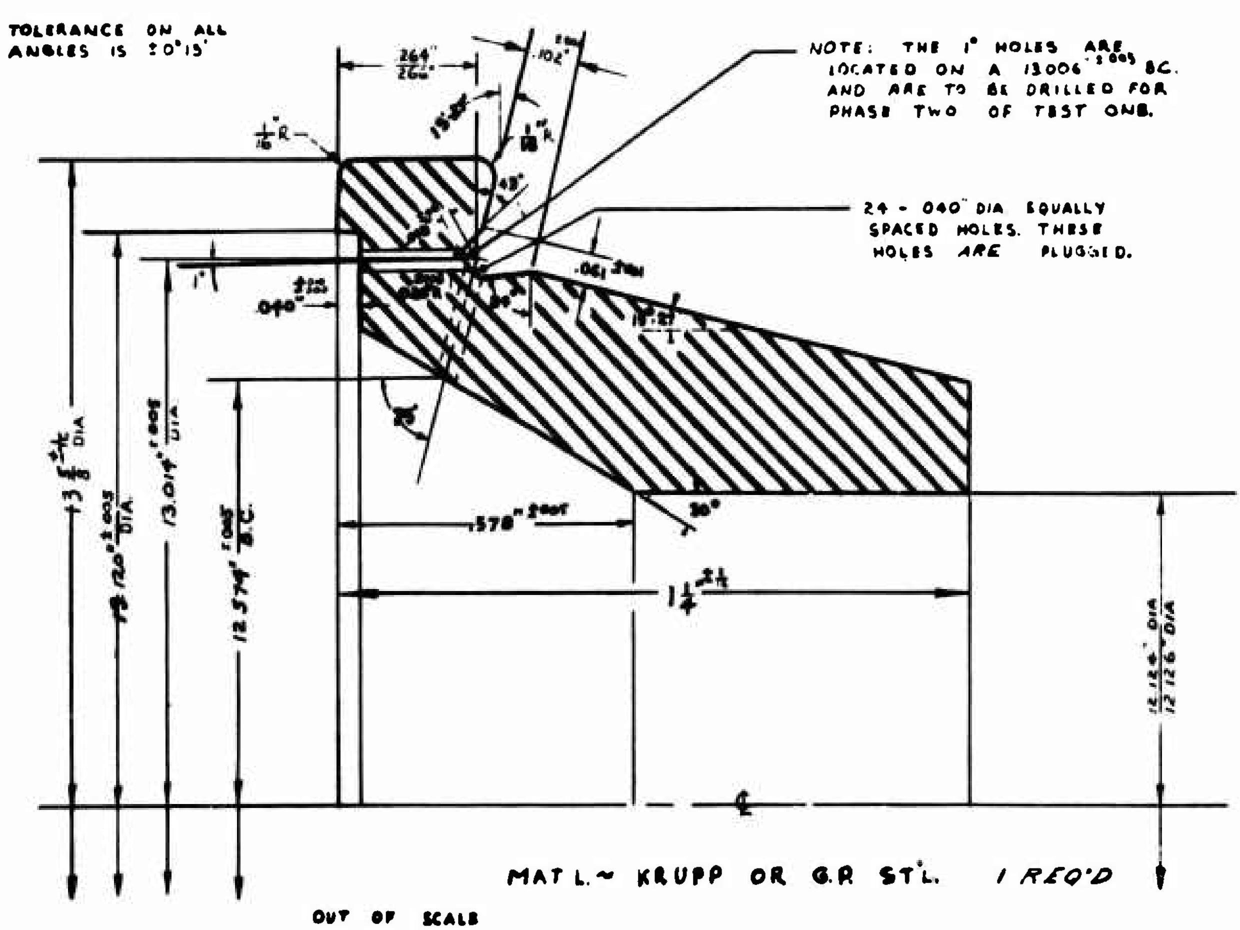

Figure 13. Simulated Cone (EX12088) With Test Number 1 Manifold Modified.

A

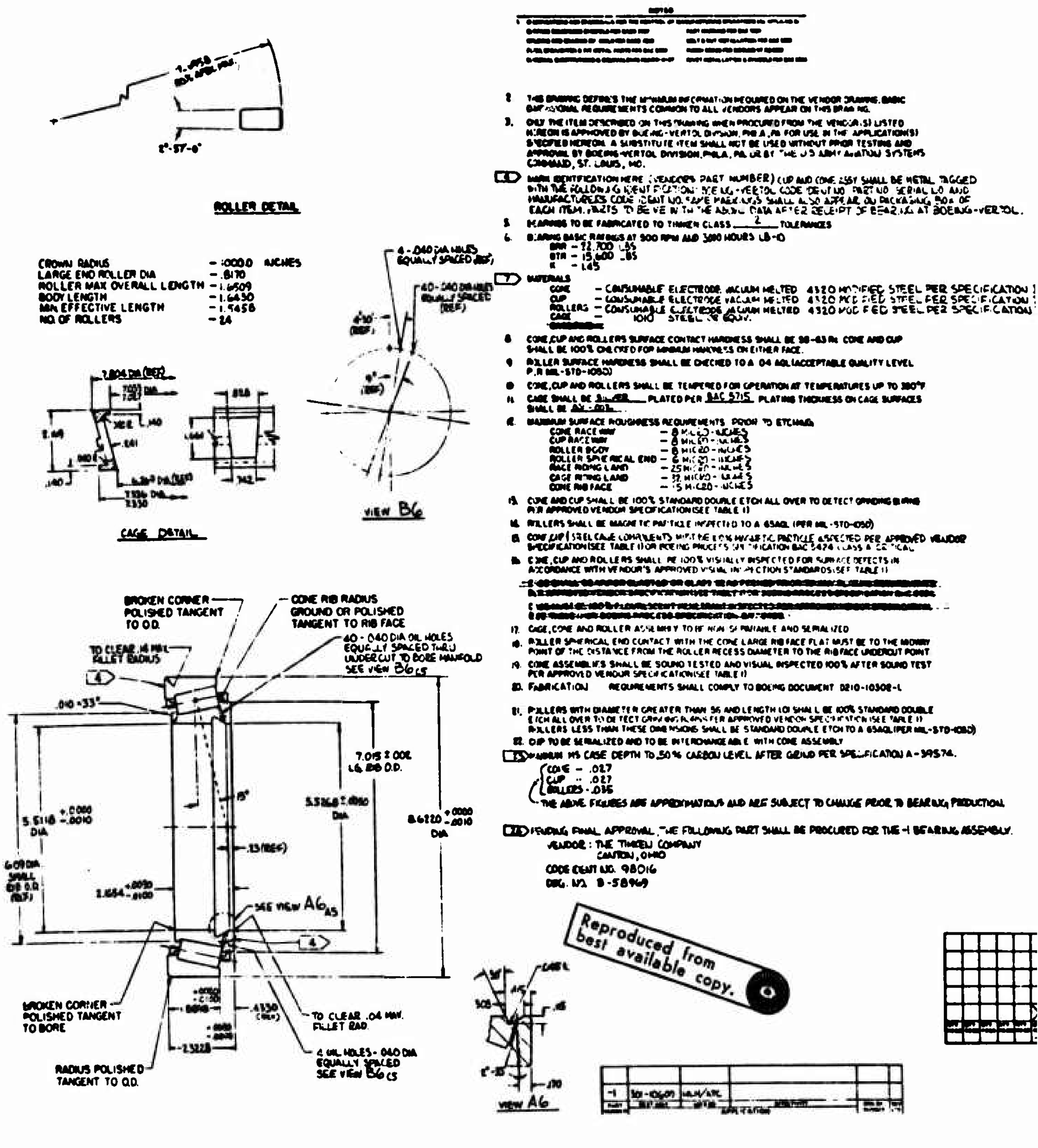

Figure 14. 301-10676 Bearing Assembly.

B

E VENDOR DRAWING, BASIC
ON THIS DRAWING.

I THE VENDOR(S) LISTED
USE IN THE APPLICATION(S)
HOUT PRIOR TESTING AND
U.S. ARMY AVIATION SYSTEMS

ND CONE ASSY SHALL BE METAL TAGGED
IDENT NO. PART NO. SERIAL NO. AND
LSO APPEAR ON PACKAGING BOX OF
R RECEIPT OF BEARING AT BOEING-VERTOL.

RANCES

4320 MODIFIED STEEL PER SPECIFICATION I.D. 20
4320 MODIFIED STEEL PER SPECIFICATION I.D. 20.
4320 MODIFIED STEEL PER SPECIFICATION I.D. 20

10-63 Rc CONE AND CUP

CCEPTABLE QUALITY LEVEL

TEMPERATURES UP TO 380°F
THICKNESS ON CAGE SURFACES

NG

R TO DETECT GRINDING BURNS

ER MIL-STD-105D)
BE INSPECTED PER APPROVED VENDOR
C 5424 CLASS A CRITICAL
OR SURFACE DEFECTS IN
NDARDS (SEE TABLE I)
[redacted]

SERIALIZED
E FLAT MUST BE TO THE BOUNDARY
I THE RIB FACE UNDERCUT POINT
CTED 100% AFTER SOUND TEST

CUMENT D210-10302-1

ALL BE 100% STANDARD DOUBLE
N SPECIFICATION (SEE TABLE I)
PLE ETCH TO A 65AQL (PER MIL-STD-105D)
ASSEMBLY
ND PER SPECIFICATION A-39574.

CT TO CHANGE PRIOR TO BEARING PRODUCTION

E PROCURED FOR THE -1 BEARING ASSEMBLY.

TABLE 1									
	24			[illegible]			[illegible]		
	SPEC	REV	DATE	SPEC	REV	DATE	SPEC	REV	DATE
MAGNETIC INSPECTION	* PS 407								
ETCH INSPECTION	* PS 505								
[redacted]									
VISUAL INSPECTION (NOTE 16)	* PS 402								
SOUND TEST AND VISUAL INSPECTION (NOTE 19)	* PS 402								
GLASS BEAD PEENED	[illegible]								
CASE DEPTH BY THE [illegible] TECHNIQUE	* PS 406								

TABLE 2 — SOURCES OF SUPPLY			
VERTOL PART NO.	24	[illegible]	[illegible]
301-10676-1	TIMKEN CO		

* NOTE: THESE PROCESS SPECIFICATIONS (PS & MPS) & DEVIATION CODE 470 WILL BE AVAILABLE FOR VERTOL INSPECTION ONLY ON VENDOR'S PREMISES

SOURCE CONTROL DRAWING

					1		301-10676-3	CUP		[illegible]	24		VE	
					1		301-10676-2	CONE ASSY		[illegible]	25		VE	
					X		301-10676-1	BEARING ASSY					VE	
					-1					PARTS LIST				

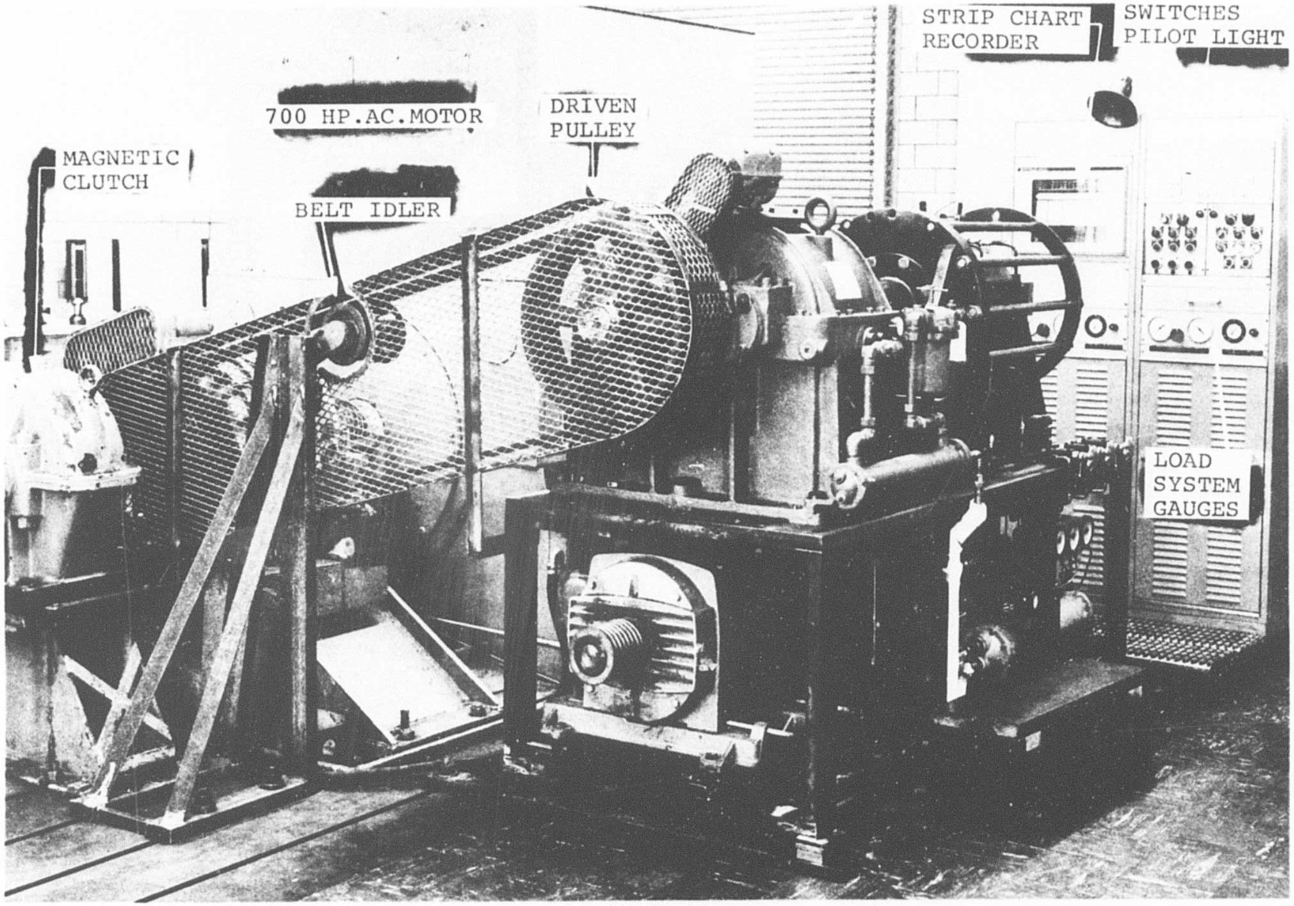

Figure 15. HLH High-Speed Tapered Roller Bearing Test Machine, Shown Driven by 700-HP Drive System.

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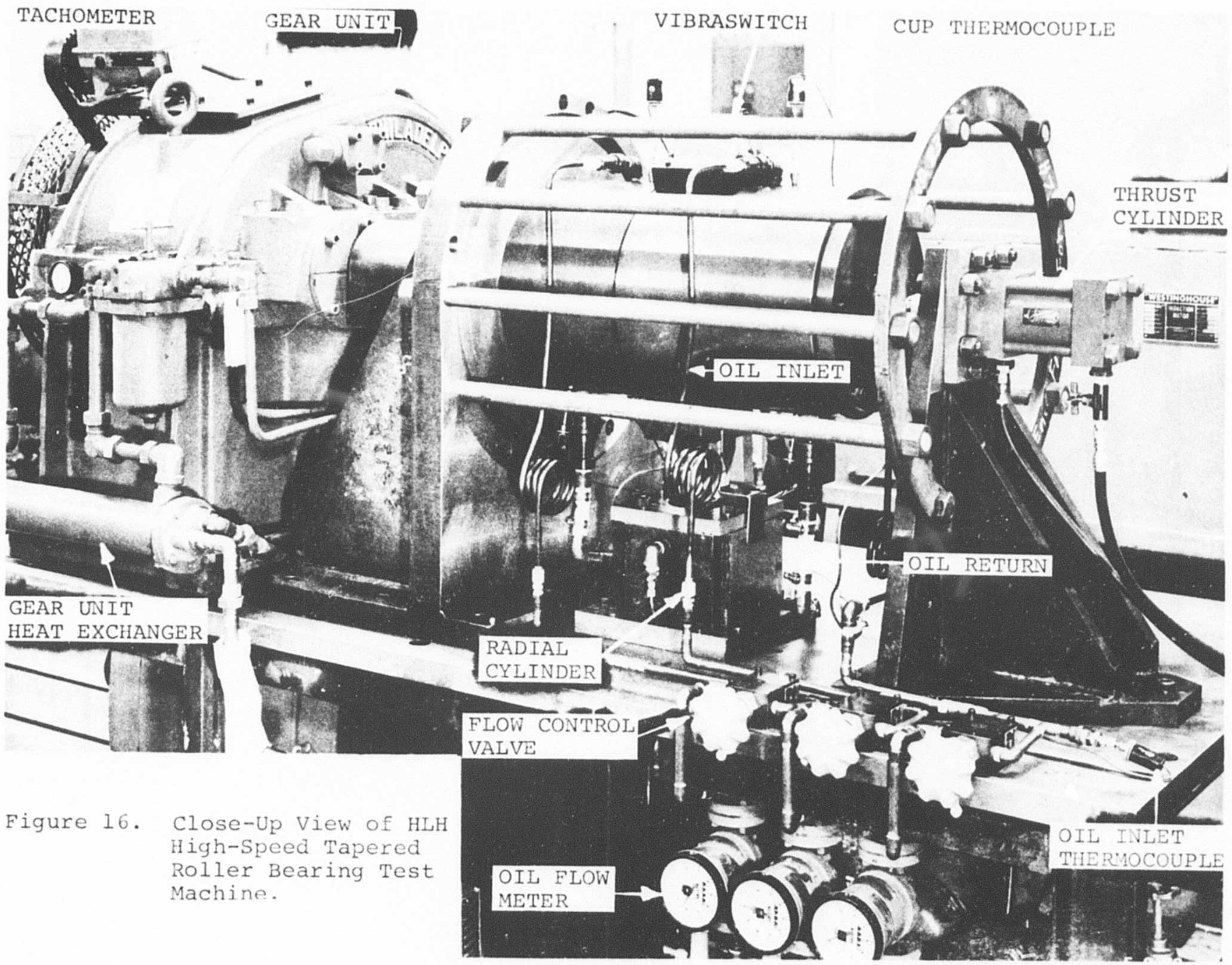

Figure 16. Close-Up View of HLH High-Speed Tapered Roller Bearing Test Machine.

A

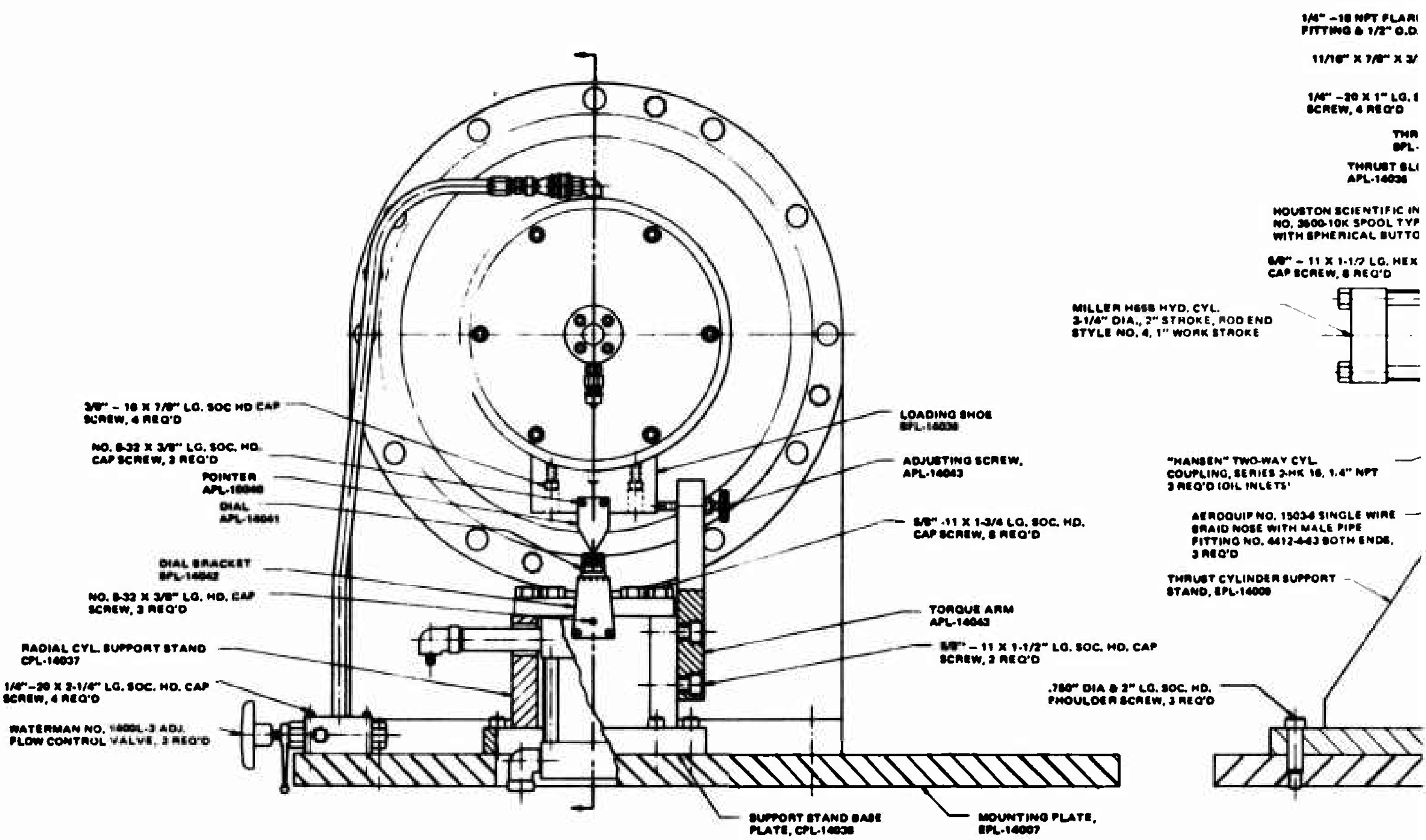

Figure 17. Subassembly Section Drawing of HLH Test Machine.

B

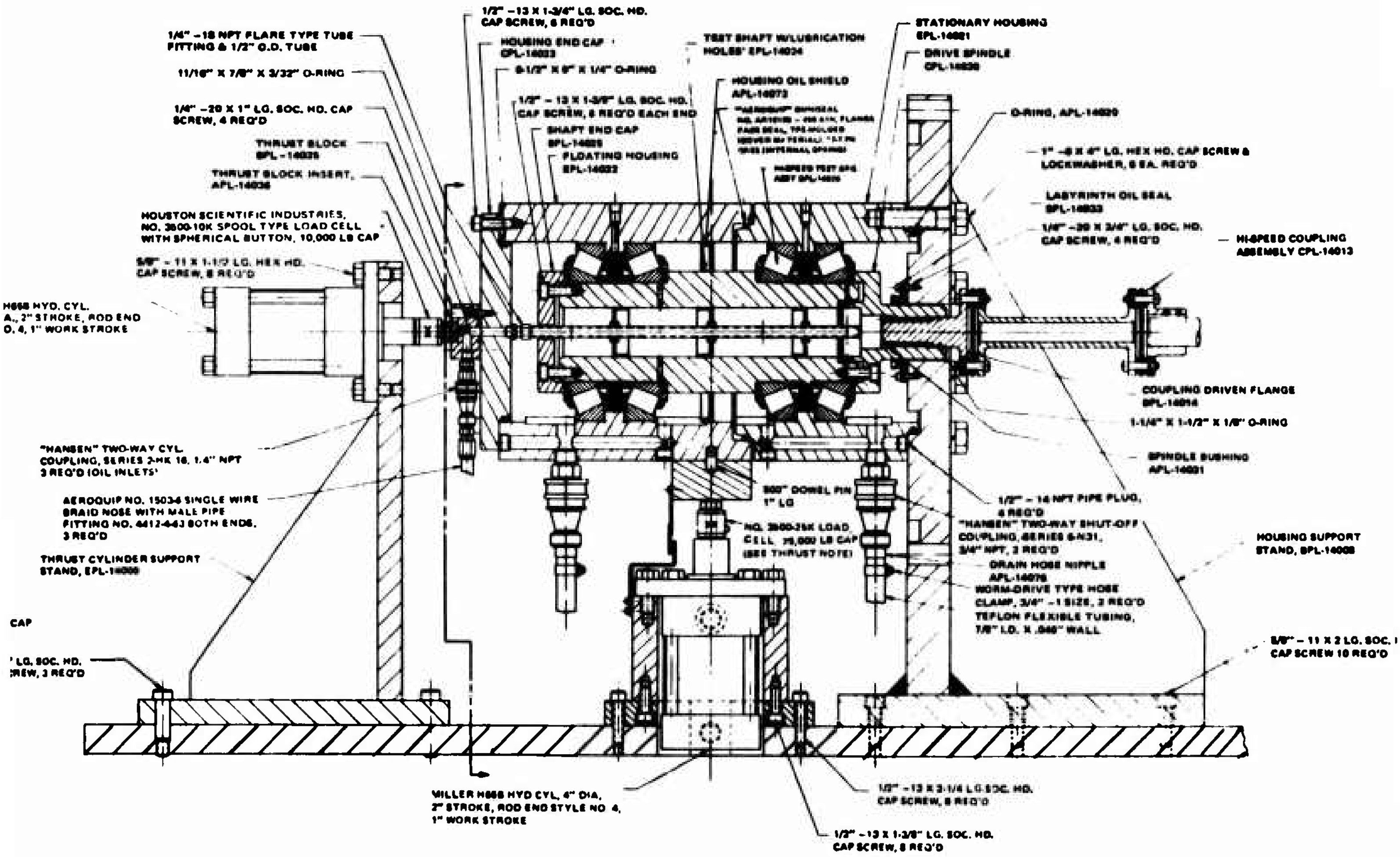
1/4" – 18 NPT FLARE TYPE TUBE FITTING & 1/2" O.D. TUBE
11/16" X 7/8" X 3/32" O-RING
1/4" –20 X 1" LG. SOC. HD. CAP SCREW, 4 REQ'D
THRUST BLOCK SPL – 14035
THRUST BLOCK INSERT, APL–14036
HOUSTON SCIENTIFIC INDUSTRIES, NO. 3500-10K SPOOL TYPE LOAD CELL WITH SPHERICAL BUTTON, 10,000 LB CAP
5/8" – 11 X 1-1/2 LG. HEX HD. CAP SCREW, 8 REQ'D
H668 HYD. CYL. A., 2" STROKE, ROD END O. 4, 1" WORK STROKE
"HANSEN" TWO-WAY CYL. COUPLING, SERIES 2-HK 16, 1/4" NPT 3 REQ'D (OIL INLETS)
AEROQUIP NO. 1503-6 SINGLE WIRE BRAID HOSE WITH MALE PIPE FITTING NO. 4412-4-63 BOTH ENDS, 3 REQ'D
THRUST CYLINDER SUPPORT STAND, SPL-14009
CAP
" LG. SOC. HD. CREW, 3 REQ'D
1/2" –13 X 1-3/4" LG. SOC. HD. CAP SCREW, 6 REQ'D
HOUSING END CAP CPL-14033
6-1/2" X 6" X 1/4" O-RING
1/2" – 13 X 1-3/8" LG. SOC. HD. CAP SCREW, 8 REQ'D EACH END
SHAFT END CAP SPL-14025
FLOATING HOUSING SPL-14022
TEST SHAFT W/LUBRICATION HOLES SPL-14024
HOUSING OIL SHIELD APL-14073
STATIONARY HOUSING SPL-14021
DRIVE SPINDLE CPL-14030
O-RING, APL-14020
1" –8 X 4" LG. HEX HD. CAP SCREW & LOCKWASHER, 8 EA. REQ'D
LABYRINTH OIL SEAL SPL-14033
1/4" –20 X 3/4" LG. SOC. HD. CAP SCREW, 4 REQ'D
HI-SPEED COUPLING ASSEMBLY CPL-14013
COUPLING DRIVEN FLANGE SPL-14014
1-1/4" X 1-1/2" X 1/8" O-RING
SPINDLE BUSHING APL-14031
.500" DOWEL PIN 1" LG
NO. 3500-25K LOAD CELL 25,000 LB CAP (SEE THRUST NOTE)
1/2" – 14 NPT PIPE PLUG, 4 REQ'D
"HANSEN" TWO-WAY SHUT-OFF COUPLING, SERIES 6-N31, 3/4" NPT, 3 REQ'D
DRAIN HOSE NIPPLE APL-14070
WORM-DRIVE TYPE HOSE CLAMP, 3/4" –1 SIZE, 3 REQ'D
TEFLON FLEXIBLE TUBING, 7/8" I.D. X .060" WALL
HOUSING SUPPORT STAND, SPL-14008
5/8" – 11 X 2 LG. SOC. HD. CAP SCREW 10 REQ'D
MILLER H668 HYD CYL, 4" DIA, 2" STROKE, ROD END STYLE NO 4, 1" WORK STROKE
1/2" –13 X 2-1/4 LG SOC. HD. CAP SCREW, 8 REQ'D
1/2" –13 X 1-3/8" LG. SOC. HD. CAP SCREW, 8 REQ'D

Figure 18 shows bearing mounting, and loading and lubrication systems. The bearing mounting arrangement simulates the mounting arrangement in the HLH combining transmission input bevel pinion gear.

The manifold projecting into the hollow shaft discharges oil through orifices located axially in line with the radial holes in the shaft. There are two 3/16-inch radial holes through the shaft at each bearing position. The dams located between each bearing position along the length of the shafts assure an equal distribution of lubricating oil to the four bearings. The cone bore manifold then collects the oil and distributes it to the radial holes in the cone to lubricate the cone rib and roller ends. Three 0.060-inch jets located at the small end of the bearing, 120 degrees apart, discharge the oil underneath the cage to lubricate the cage roller body contact and inner and outer race. The oil-flow rate supplied to each bearing was calibrated by collecting the oil with a graduated measuring cup, while being timed with a stop watch. The oil flow rate was recorded on an Esterline-Angus multipoint strip chart recorder. In addition, each orifice on the manifold projecting into the hollow shaft was sized and calibrated to assure an equal amount of oil supplied to each bearing.

The bearing cup OD, oil inlet, oil outlet, and ambient air temperatures were measured by strategically located thermocouples. Each thermocouple was calibrated with a precision laboratory thermometer. The temperatures are read and recorded on the multipoint strip chart recorder.

The thrust load applied to the bearings was measured by a Lockheed Model 3500-10 load cell. A Miller hydraulic cylinder, model number H65B, was used to apply the load to the end of the opposite drive-end floating housing. The thrust hydraulic load system and the load cell has a capability of applying 10,000 pounds thrust load.

The radial load applied to the bearings was measured by a Lockheed Model 3500-25 load cell. A Miller hydraulic cylinder was used to apply the load to the bottom of the outer floating housing. The radial hydraulic load system and load cell has a capability of applying 25,000 pounds radial cylinder load.

The pressure gauges used with both radial and thrust load hydraulic systems were calibrated with master pressure gauges. The load cells were calibrated in a 120,000-pound-capacity Baldwin BTE Universal Testing Machine. This calibration was done directly with the multipoint strip chart recorder.

The bearing lubrication system is equipped with a stainless steel heat exchanger to control the inlet oil temperature at a

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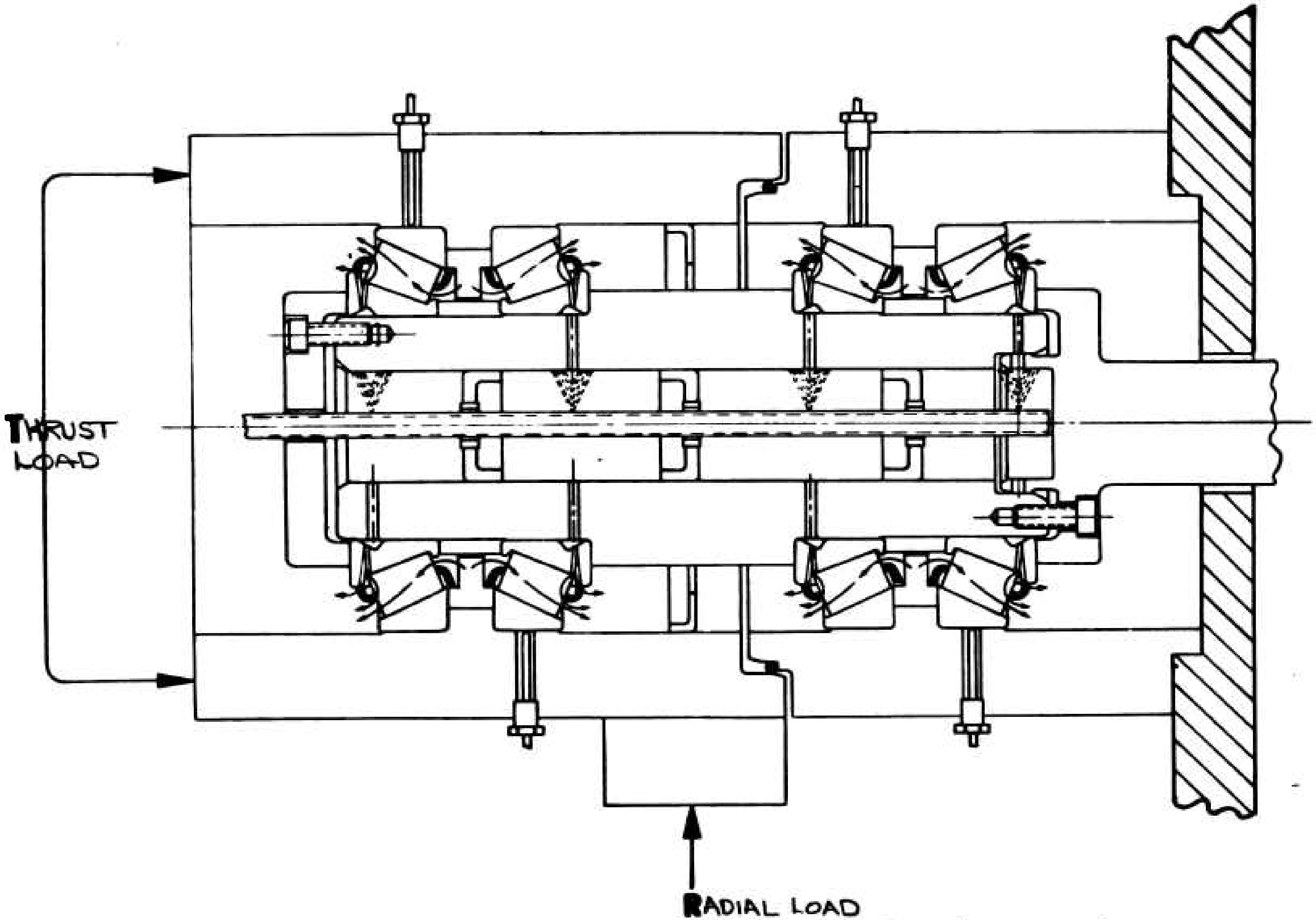


Figure 18. Schematic of HLH Test Machine Showing Bearing Mounting, Loading, and Lubrication System.

given value and with flow controls capable of maintaining any desired flow rate up to 16 pints per minute per bearing (oil inlet temperature at 190°F).

Timken Lubricant and Wear Test Machine

Various fail-safe wear-resistant materials were evaluated in the Timken Lubricant and Wear Test Machine. This test machine is used in ASTM Standard D2782-71, standard method for "Measurement of Extreme Pressure Properties of Lubricating Methods (Timken Method)."

Figure 19 shows the overall view of the test machine; Figure 20 shows a section drawing of the test machine. For these tests, a torque meter was installed between the drive motor and test spindle.

The test blocks and cups were weighed before testing. The lubricant in the machine was preheated to 100°F, and a heat exchanger was used to maintain this temperature throughout the test. The lubricating oil was MIL-L-23699, and the test specimens were flood-lubricated during the oil-on phase of the tests. The test cup was rotated against the block at 1165 rpm (591.1 feet per minute rubbing velocity). Figures 4 and 5 show the physical dimensions of the sample cup and test blocks.

Qualitative Comparison of Various Wear-Resistant Materials

A 1-pound weight was added to the lever arm every 10 minutes up to a total of 10 pounds, which gives a test load of 116.6 pounds and an initial stress of 36,000 psi for steel cups and blocks. The test was run an additional 2½ hours after full load was applied. At the end of the test, the test cups and blocks were reweighed, and the scar width on the test block was measured and tabulated.

Qualitative Analysis of Material's Ability to Survive Without Lubrication

A 1-pound weight was added to the lever arm every minute up to a total of 10 pounds, which gives a test load of 116.6 pounds and an initial stress of 36,000 psi for steel cups and blocks. After the load was fully applied, the test was run an additional 10 minutes. Then the lubricating oil was shut off and the residual oil was blown away with an air hose. The test is timed from when the residual oil is blown away until the cup speed is slowed by friction to 1000 rpm (507.4 feet per minute). The test cups and blocks were reweighed, and the scar width on the block was measured.

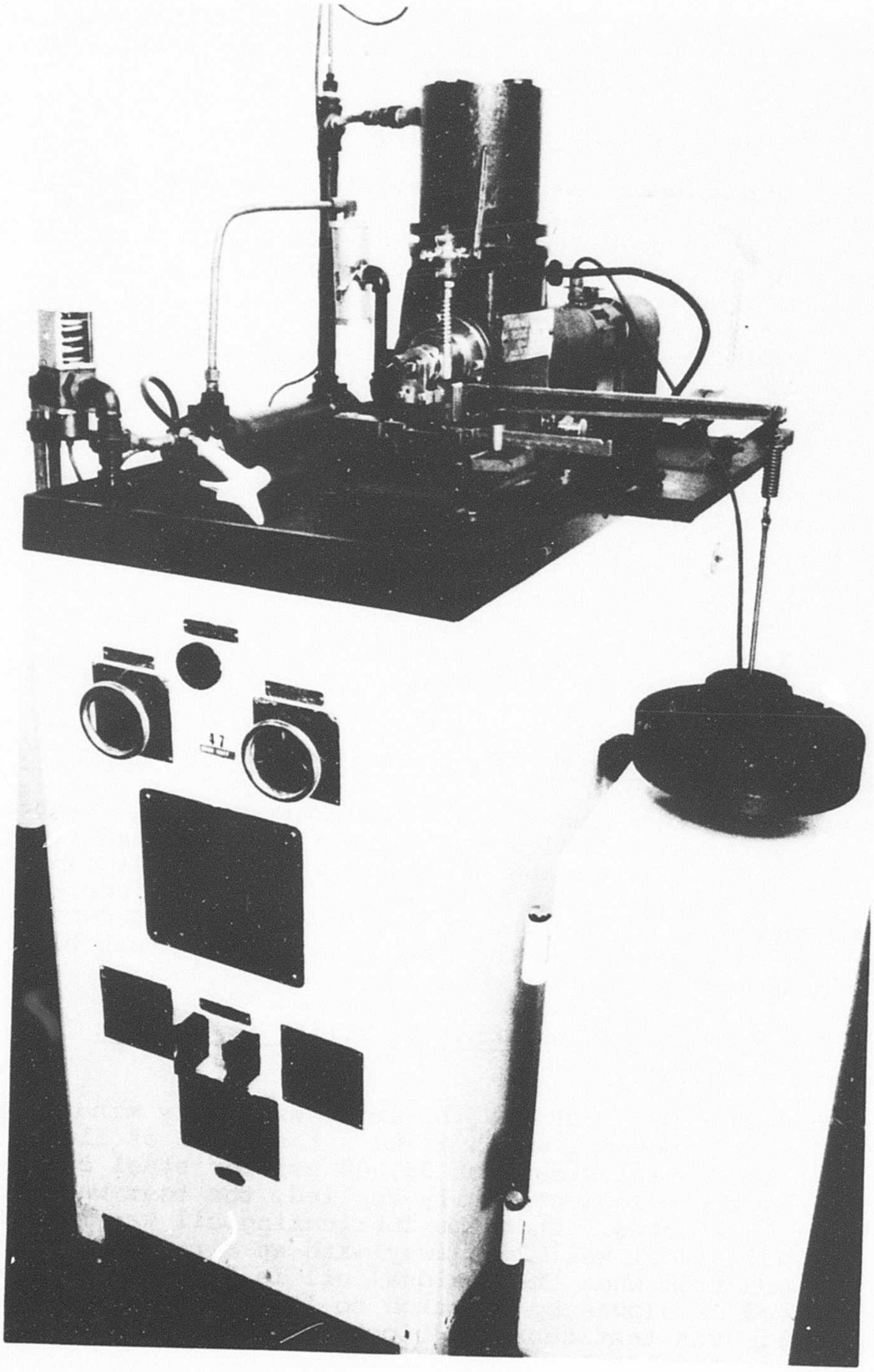


Figure 19. Timken Lubricant and Wear Test Machine.

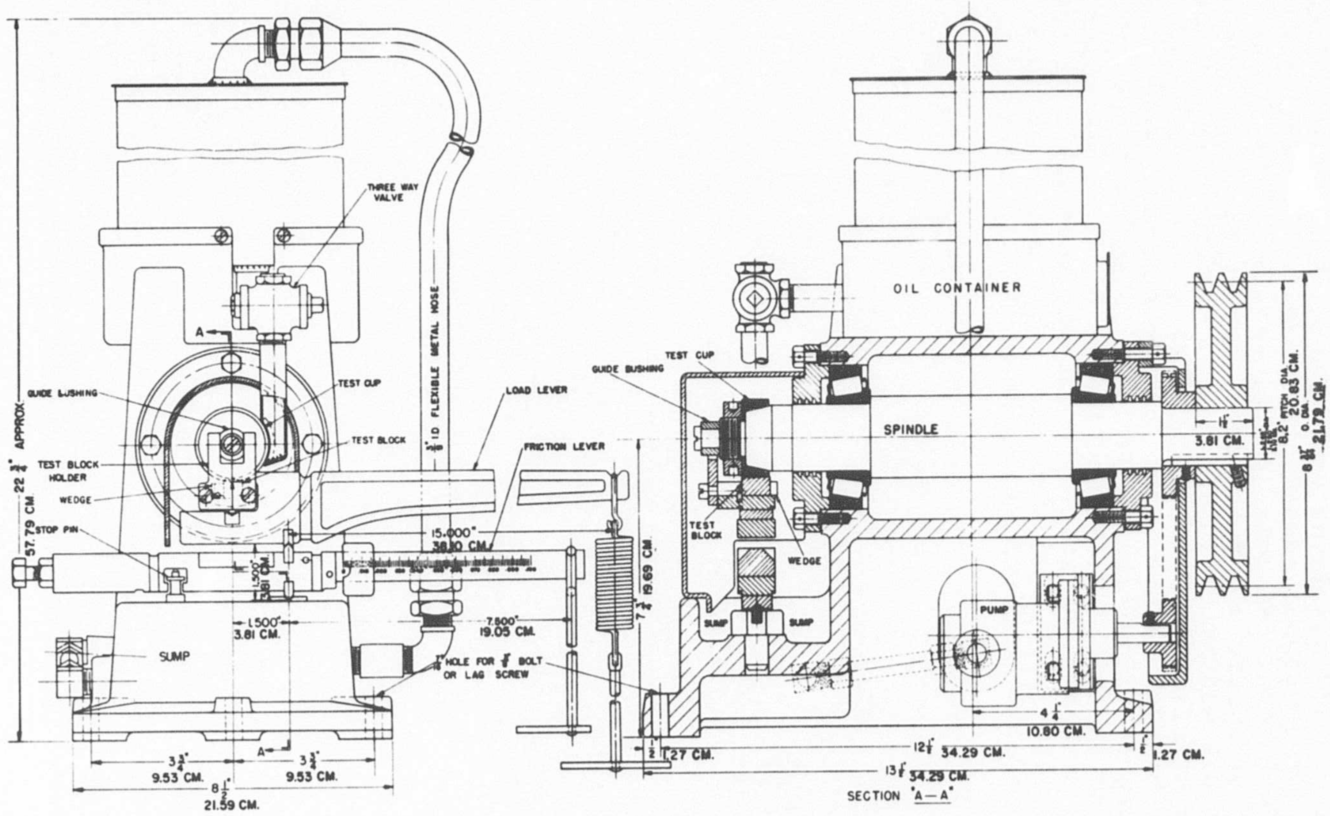


Figure 20. Section Drawing of Timken Lubricant and Wear Test Machine.

Cone Flange Oil Distribution Spin Test Machine

The smaller XC11437 simulated cone (Vertol P/N 301-10612) was run in the Timken company-owned, high-speed test machine. The larger XC12088 simulated cone (Vertol P/N 301-10443-1) was tested on a varidrive electric motor (see Figure 21). The lubricating oil was MIL-L-23699, heated to and maintained at 165°F. The oil-flow rate to the XC11437 simulated cone was 6 pints per minute; strobe light still photographs were taken at this oil-flow rate at 3000, 6000, and 8000 rpm. The oil-flow rate to the XC12088 simulated cone was 3 pints per minute, and strobe light still photographs were taken of this oil flow at 1000, 2000, and 3000 rpm. The conditions simulated the operating conditions in the HLH/ATC drive system.

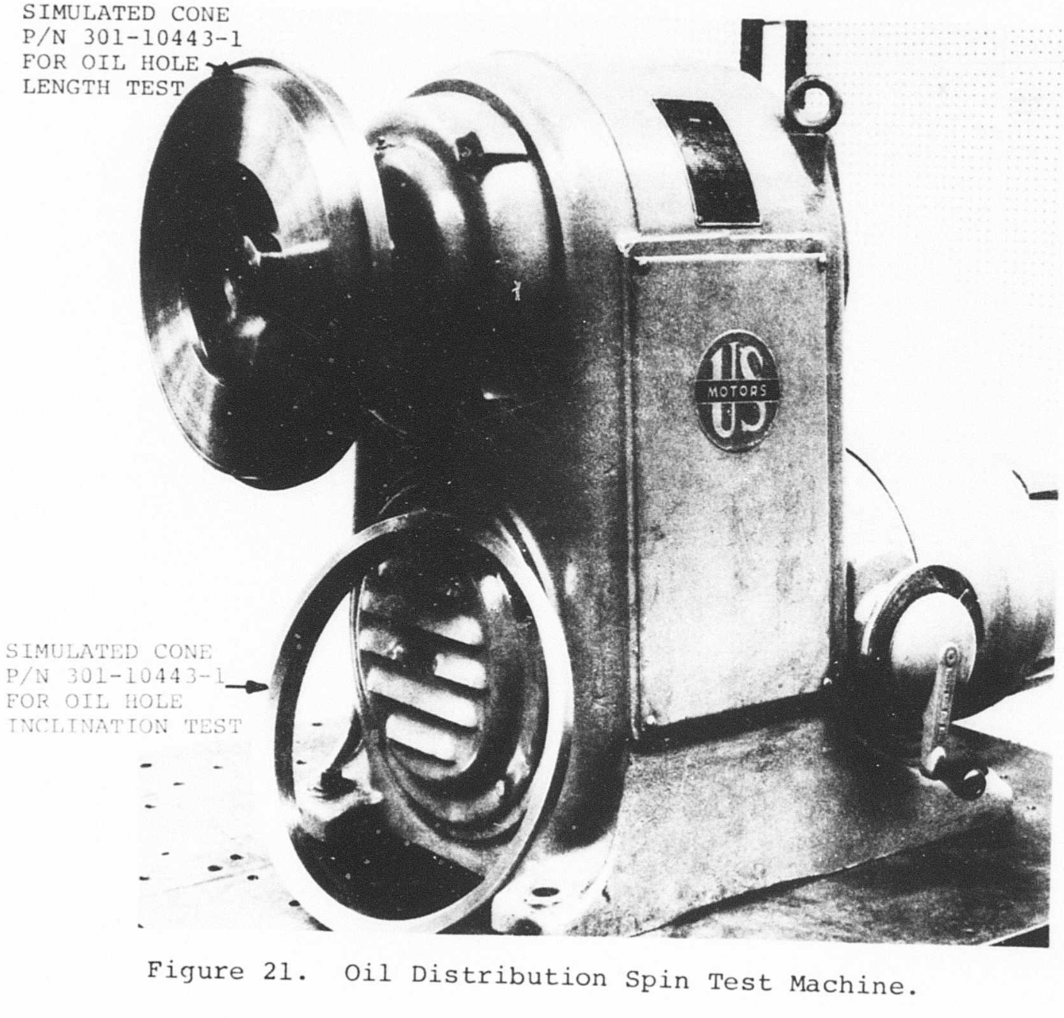


Figure 21. Oil Distribution Spin Test Machine.

TEST DATA AND RESULTS

DETERMINATION OF NUMBER AND SIZE OF RADIAL LUBRICATION HOLES IN HM926700-SERIES BEARING CONE

A standard HM926700-series bearing cone was modified as shown in Figure 22. Seventeen 0.052-inch-diameter holes were drilled from the backface oil manifold to the rib undercut. Figures 23 through 25 are strobe light still photographs taken at 5000, 10,000 and 14,000 rpm with 6 pints per minute oil flow. As can be seen on the photographs, it appears that all the radial lubrication holes receive an equal amount of lubricating oil. This could also be seen by visually observing the oil flow with a strobe tachometer.

DEVELOPMENT TESTS - MODIFIED HM926700-SERIES BEARING

The pretest inspections of all the modified HM926700-series bearings are included in Appendix II. This tabulation includes the cone rib face and race OD surface finishes, roller Spherical End Radius (SER) and roller body surface finishes, the cone race to rib OD angle, and the roller SER in percent of apex length.

Table II lists all modified HM926700-series bearings tested in the HLH high-speed test machine, including the fail-safe design test bearings. The table shows the test number, type of test performed, bearing number according to position in the machine, bearing condition after test, and maximum successful speed and time at that speed.

Figures 26 through 29 show typical new bearings before being tested. Appendix III includes photographs of all bearings after test.

Three shakedown runs were made with standard HM926700-series bearings modified for high speed to check out the operation of the new HLH high-speed bearing test machine. During the shakedown runs, two points became obvious that needed to be changed to continue to develop the HM926700-series bearings:

- The 17 radial oil lubrication holes were not adequate for the rapid acceleration and for high top-speed combination.
- Bearing development could not be effectively accomplished with the constant speed 100-horsepower motor with an acceleration rate of 0 to 14,000 rpm in 9 seconds.

A new pulley was purchased to limit the speed to 9300 rpm for the third shakedown run and test number 1. The acceleration

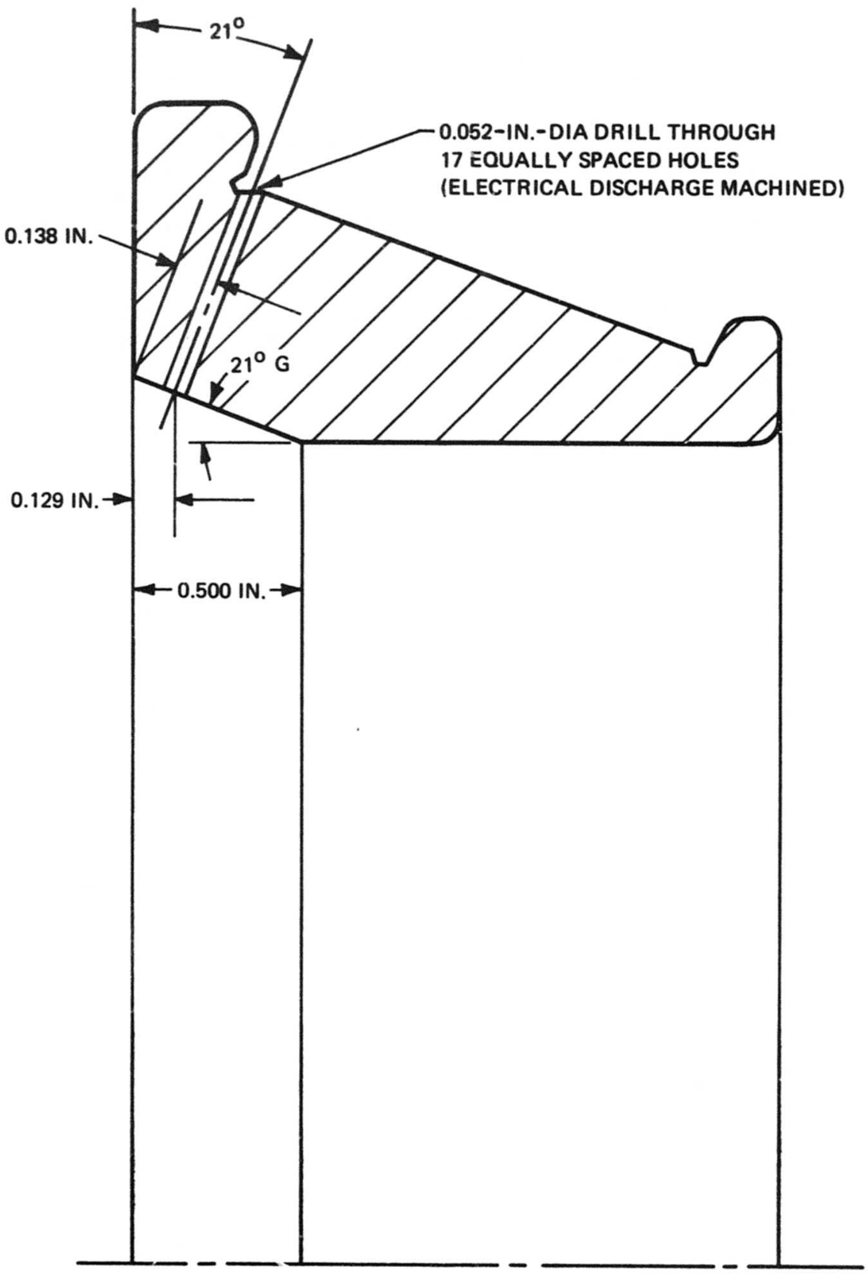
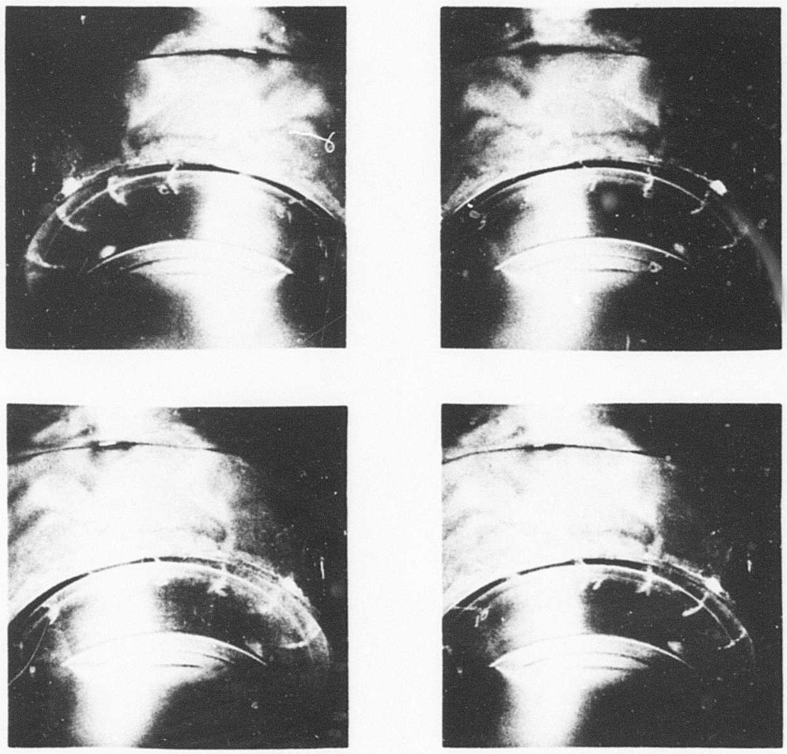
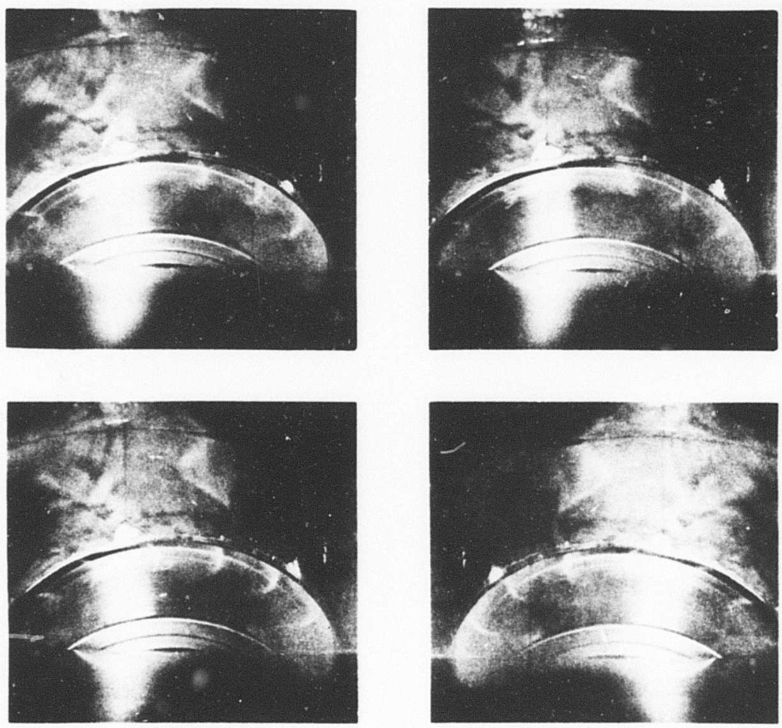


Figure 22. Standard HM926700-Series Bearing Cone Modified for Oil Distribution Spin Test.



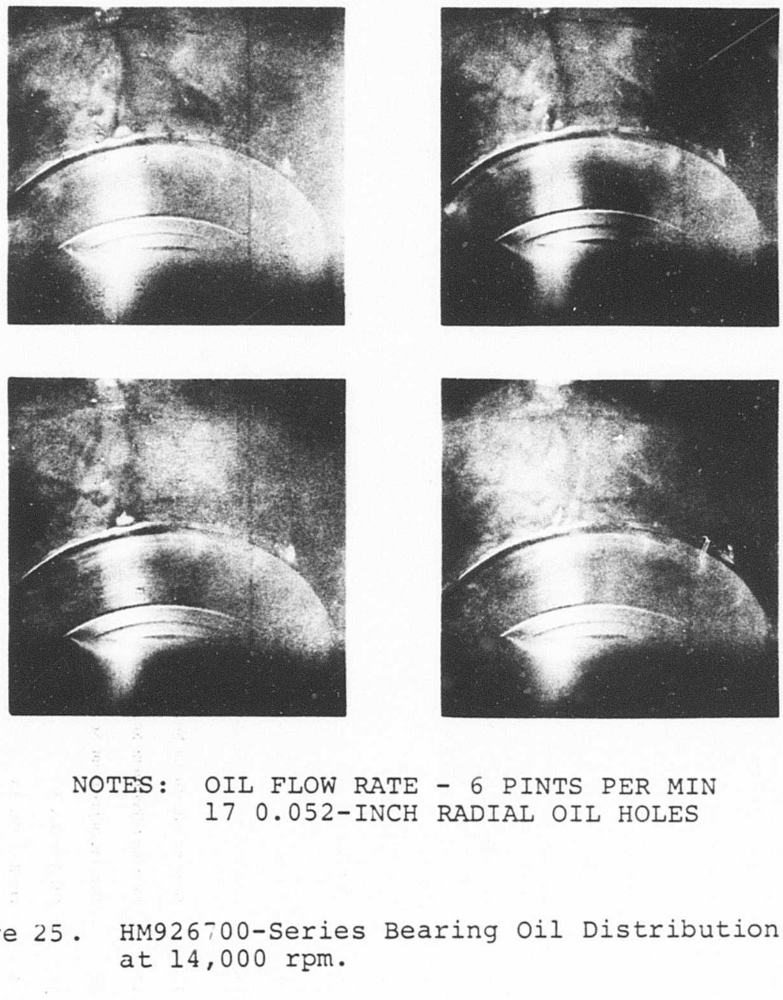
NOTES: OIL FLOW RATE - 6 PINTS PER MIN
17 0.052-INCH RADIAL OIL HOLES

Figure 23. HM926700-Series Bearing Oil Distribution Study at 5000 rpm.



NOTES: OIL FLOW RATE - 6 PINTS PER MIN
17 0.052-INCH RADIAL OIL HOLES

Figure 24. HM926700-Series Bearing Oil Distribution Study at 10,000 rpm.



NOTES: OIL FLOW RATE - 6 PINTS PER MIN
17 0.052-INCH RADIAL OIL HOLES

Figure 25. HM926700-Series Bearing Oil Distribution Study at 14,000 rpm.

TABLE II. SUMMARY OF HM926700-SERIES BEARING DEVELOPMENT AND FAIL-SAFE TESTS

Test No.	Type of Test	No. of Radial Holes in Cone	Bearing Number/Condition After Test				Remarks
			DE	DEC	ODEC	ODE	
	Constant-speed 100 hp drive						
1	Development	17-34	60-1/F	PL-7/OK	PL-6/OK	PL-8/OK	9300 rpm/1.5 min
	Variable-speed 700 hp drive						
2	Development	17	60-2/F	60-3/F	60-4/F	60-5/OK	14,000 rpm/45 min
3	Development	17	60-6/OK	60-7/OK	60-4A/F	60-5/OK	14,000 rpm/3 min
4	Development and acceleration tests	34	60-8/spall	PL-7/spall	PL-6/spall	PL-8/F	14,000 rpm/36 hr
5	Development and acceleration tests	40	60-9/spall	60-10/OK	60-11/OK	60-12/OK	14,000 rpm/27-1/2 hr
6	Fail safe	36	1-LP/F	2-LP/F	3-LP/OK	4-LP/OK	9500 rpm/2 hr
7	Fail safe	36	5-B/OK	6-B/crack	7-B/crack	8-B/OK	9500 rpm/2 hr
8	Fail safe	36	9-LP/OK	10-LP/OK	11-LP/F	12-LP/OK	11,500 rpm/24 hr oil off 8-1/2 sec
9	Fail safe	36	13-LP/OK	14-LP/OK	15-LP/F	16-LP/OK	11,500 rpm/50-1/2 hr
10	Fail safe standard cage	36	9a-LP/F	10a-LP/OK	13a-LP/OK	12a-LP/OK	11,500 rpm/3 hr oil off 10 sec
11	Bearing adjustment No. 1	40	1/OK	2/OK	3/F	4/F	9000 rpm/0 hr
12	Bearing adjustment No. 1A	40	5/OK	6/OK	7/OK	8/F	11,500 rpm/9 hr
13	Bearing adjustment No. 2	40	5/F	6/F	7/OK	9/OK	11,500 rpm/13 hr
14	Oil flow No. 1	40	10/OK	11/F	7/OK	9/OK	8000 rpm
15	Oil flow No. 1A	40	10/F	12/OK	7/OK	9/OK	--
16	Oil flow No. 1B	40	13/F	12/F	7/OK	9/OK	8000 rpm

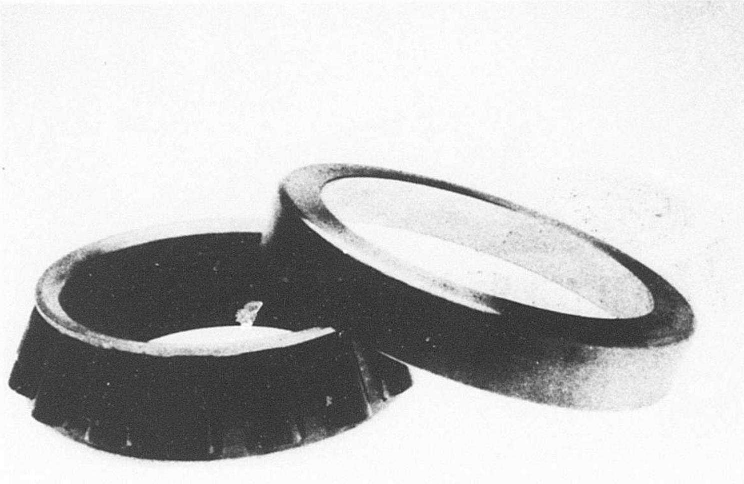
TABLE II - Continued

Test No.	Type of Test	No. of Radial Holes in Cone	Bearing Number/Condition After Test				Remarks
			DE	DEC	ODEC	ODE	
17	Oil flow No. 1C	40	1-S/OK	2-S/OK	7/F	9/OK	11,500 rpm/5 min
18	Oil flow No. 1D	40	X/OK	60-10/F	60-11/OK	60-12/OK	11,500 rpm/2 min
19	Oil flow No. 1E standard cage	40	X/OK	2-S/OK	60-11/F	60-12/OK	11,500 rpm/2 min
20	Additional development	40	X/F	2-S/OK	3-S/OK	60-12/OK	11,500 rpm/2 min
21	Additional development	40	9/OK	2/OK	14/F	60-12/F	11,500 rpm/2 min
22	Additional development (1,2)	40	T-5/OK (3,6)	T-6/OK (3,6)	T-7/OK (3,6)	T-8/F (3,6)	11,000 rpm/1 min
23	Additional development	40	T-5/OK (3,6)	T-6/F (3,6)	T-7/[illegible] (3,7)	T-9/OK (3,7)	10,000 rpm/0
24	Additional development	40	T-5/OK (3,7)	T-10/OK (3,7)	T-7/F (3,7)	T-9/F (3,7)	11,500 rpm/0
25	Additional development	40	T-11/F (5,6)	2-S/OK	3-S/OK	T-12/OK (5,6)	11,500 rpm/0 - 100 pct T-6.5 pct R
26	Additional development	40	T-13/F (4,9)	2-S/OK	3-S/OK	T-12/OK (9)	5000 rpm/16 hr damage at 8000 rpm
27	Additional development	40	T-15/OK (4,8)	2-S/OK	3-S/OK	T-14/F (5,6)	11,500 rpm/1 hr
28	Additional development	40	73-3/F (10)	2-S/OK	3-S/OK	73-5/OK (10)	8000 rpm/0
29	Additional development	40	1-S/OK	2-S/OK	3-S/OK	73-5/F (10)	0 to 10,000 rpm with 10 pct R and T damaged at 10,000 rpm
30	Additional development	40	T-15/OK (4,8)	2-S/OK	PL-7/OK (11)	60-8/OK (11)	100-pct load at 5000 rpm Then 11,500 rpm - 10-1/2 hr at 11,500 rpm
31	Additional development	40	T-15/OK (4,8)	2-S/OK	73-7/F (10,13)	73-13/OK (10,14)	Same load as above 24 min at 11,500 rpm

TABLE II. Continued.

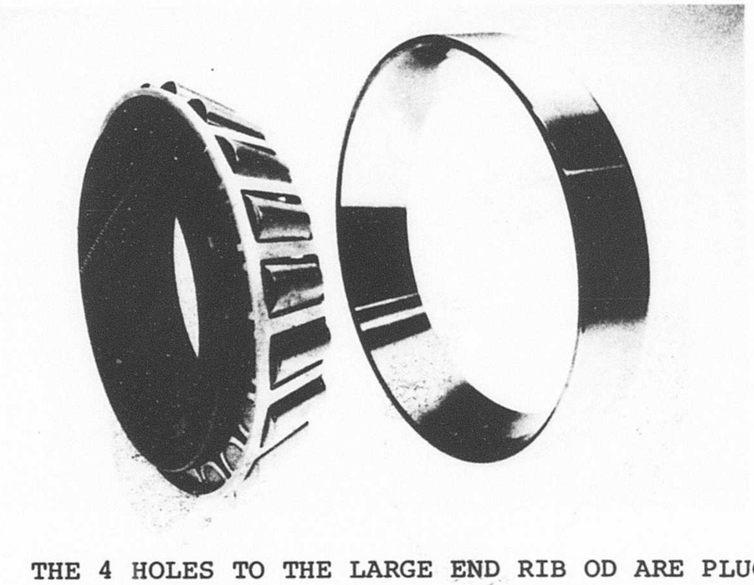
Test No.	Type of Test	No. of Radial Holes in Cone	Bearing Number/Condition After Test				Remarks
			DE	DEC	ODEC	ODE	
32	Additional development	40	T-15/OK (4, 8)	2-S/F	3-S/OK	73-13/OK (10, 14)	Same load as above 11,500 rpm/0
33	Additional development	40	T-15/OK (4, 8)	73-8/F (15)	3-S/OK	73-13/OK (10, 14)	9000 rpm/0

Key:
1. Tests No. 22 through 33 had bearings with standard stamped-steel, silver-plated cages
2. Testing started on cones with new manifold design
3. Cone hole--15°
4. Cone hole--10° by EDM
5. Holes enlarged in Lab
6. Cone rib--89°56'
7. Cone rib--89°41'
8. Cone rib--89°51'30"
9. SER equal to 85 pct apex length, rib--89°38'
10. Reduced undercut - rib--90°3'30", cone hole--10°
11. Bearings ran 36 hours at 14,000 rpm previously
12. Enlarged undercut
13. Run-in for 8 min with 40,000-pound load
14. Run-in for 7 min with 51,800-pound load
15. Reduced undercut - rib--89°54'



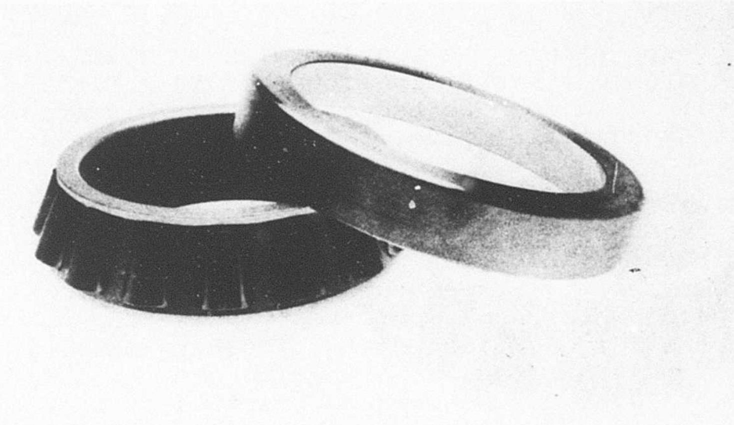
NOTE: THE 4 HOLES TO THE LARGE END RIB OD ARE PLUGGED

Figure 26. Typical New, Modified HM926700-Series Bearing With 17 Radial Holes to the Rib Face - Standard Stamped-Steel Silver-Plated Cage.



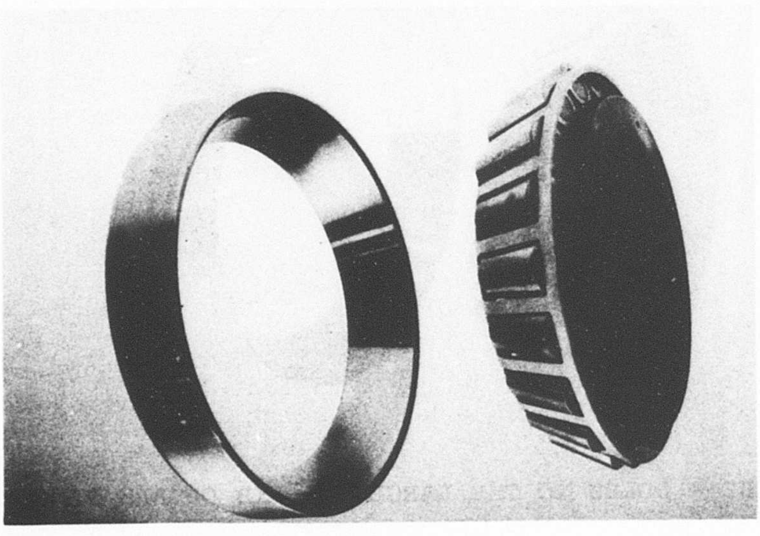
NOTE: THE 4 HOLES TO THE LARGE END RIB OD ARE PLUGGED.

Figure 27. Typical New, Modified HM926700-Series Bearing With 17 Radial Lubrication Holes to the Rib Face - Standard Stamped-Steel Silver-Plated Cage.



NOTE: THE 4 HOLES ARE OPEN TO LUBRICATE THE END FLANGE OF THE Z-TYPE SILVER-PLATED CAGE

Figure 28. Typical New, Modified HM926700-Series Bearing With 40 Radial Lubricating Holes to the Rib Face.



NOTE: THE 4 HOLES TO THE LARGE END RIB OD ARE PLUGGED

Figure 29. Typical New, Modified HM926700-Series Bearing With 40 Radial Lubrication Holes to the Rib Face - Standard Stamped-Steel Silver-Plated Cage.

rate was 0 to 9300 rpm in less than 9 seconds. Test number 1 was run with one bearing with 17 radial lubrication oil holes and three bearings with 34 holes. This test ran for 1.5 minutes at 9300 rpm before the bearing with 17 holes was damaged (roller end/cone rib scuffing). After this test, the test machine was moved to make use of the variable-speed Timken 700-horsepower dynamometer drive.

Tests 2 and 3 were run with bearings having 17 radial holes to lubricate the cone rib/roller end contact area. Test number 2 was accelerated from 4000 to 12,000 rpm in 2000 rpm increments. The test bearing temperatures were allowed to level off at each speed, which took approximately 1-1/2 to 2 hours. The bearings were then accelerated to 14,000 rpm and operated at that speed for 45 minutes until the lubrication pump shut down, causing three of the bearings to scuff at the cone rib/roller end contact.

Test 3 ran a total of 12 hours at 12,000 rpm. The test bearings were then accelerated to 14,000 rpm and ran for 3 minutes before cone rib/roller end scuffing damage occurred.

Tests 4 and 5 ran at 14,000 rpm for a total of 36 and 27-1/2 hours, respectively. Test 4 had bearings with 34 radial lubrication holes (17 holes electrical discharge machined between the original 17 holes). New bearing cones were made for Test 5 with forty 0.040-inch-diameter radial lubrication oil holes.

Figure 30 shows typical bearing cup OD temperatures versus shaft speed. The data for this graph was taken from test 4. Table XI in Appendix IV shows bearing cup OD temperatures for the remaining tests in which temperatures were allowed to level off at various speeds.

Tests 4 and 5 were subjected to rapid accelerations at convenient intervals during the running of these bearings. The acceleration run consisted of

- Accelerating from 0 to 10,000 rpm in less than 30 seconds and running for 5 minutes under 10-percent load (applied cylinder load of 1230 pounds radial and 440 pounds thrust).

- Accelerating from 10,000 to 11,500 rpm in 5 seconds, increasing load to 100 percent (12,300 pounds radial and 4394 pounds thrust applied cylinder load) and letting it run for 5 minutes.

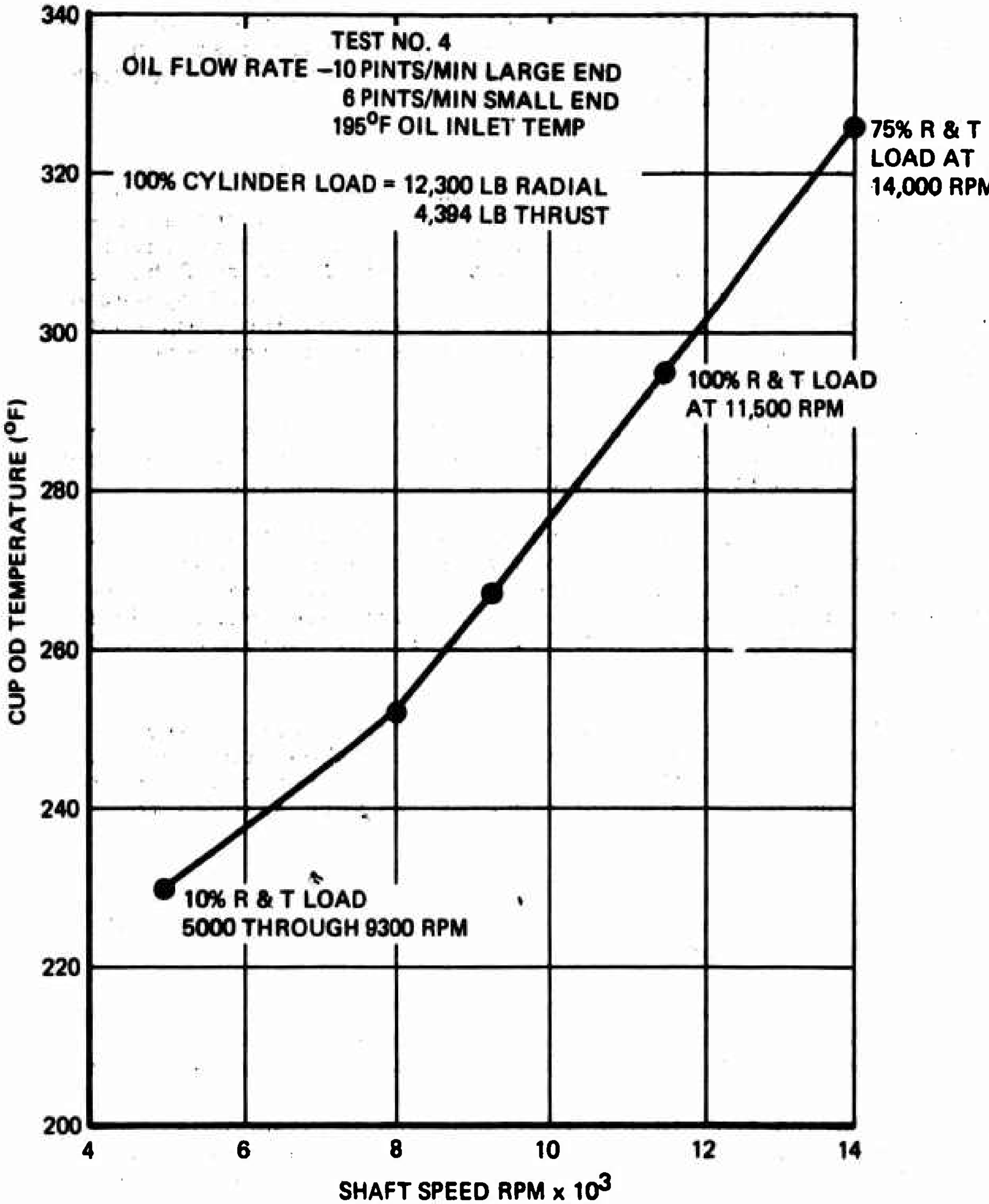


Figure 30. HM926700-Series Bearing Cup OD Temperature Versus RPM.

- Reducing the load to 75 percent of the applied radial and thrust loads and accelerating from 11,500 to 14,000 rpm in 5 seconds. The acceleration runs were started with both room temperature and preheated lubricating oil, and also with both oil pumps on first. The drive system and simultaneous oil pump-drive system then starts.

One of the bearings from test 4 was damaged (cone rib/roller end scuffing) during the third acceleration run. Test 5 was also subjected to three acceleration runs; however, no cone rib/roller end scuffing occurred.

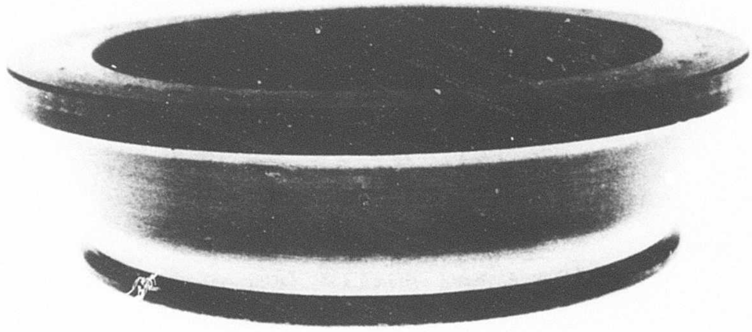
After-test inspections of tests 4 and 5 showed that three of the cones in test number 4 had spalls (Figure 31) and one of the cones of test 5 had a small spall. The three cones from test 4 were given to the Timken Company Metallurgical Department for investigation. There it was concluded that these spalls would not have occurred as early in the bearing life if metallurgical and geometry changes had not taken place because of operating temperatures in excess of 300°F. The bearing cup OD temperatures ranged from 325° to 335°F and, in all probability, the discolored portion of the cone race was near or above 360°F. Also, these spalls might not have occurred as early if CEVM steel had been used instead of the standard air-melt steel used in the test bearings. Visual examination showed the spalls to be of nonmetallic inclusion origin.

Blue checks before these bearings were tested showed that the cone races were flat. Traces taken on the Clevite 1200 Surfanalyzer System (Figure 32) after test showed the cone race to be from 0.00025-inch to 0.00035-inch crowned toward the large end (discolored area). Both x-ray analysis and metallographic examination showed the discolored area to be tempered, resulting in loss of hardness and an uneven transformation of the retained austenite over the cone race. The uneven transformation of the retained austenite due to uneven temperatures along the cone race is believed to have caused the crown to develop in this manner. The uneven temperature along the cone race can be explained by the longer heat flow path, due to the heavier section at the large end of the cone and the cooler oil striking the small end of the cone.

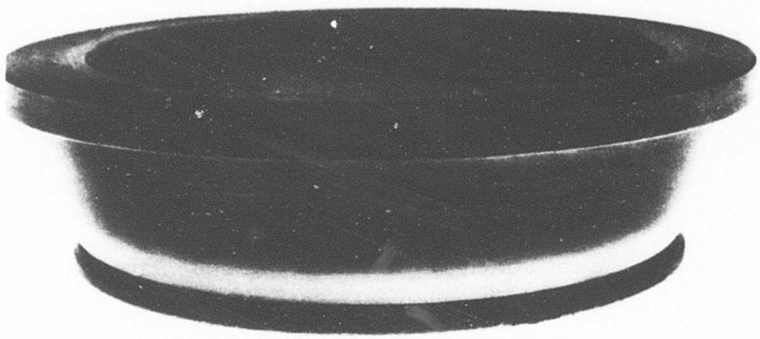
Retained austenite was examined by x-ray analysis in both the clear and discolored areas on one of the bearing cones. The results are:

	Retained Austenite	
Depth From Raceway	Clear Area	Discolored Area
Surface	21 percent	9.5 percent
0.001 inch	32 percent	7.7 percent
0.004 inch	32.6 percent	8.5 percent
0.008 inch	34.2 percent	12.7 percent

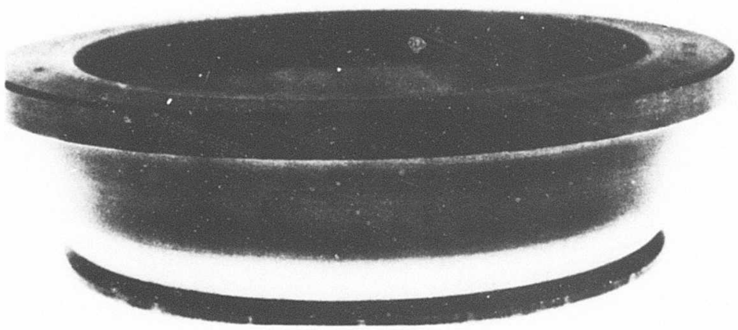
BEARING S/N 60-18



BEARING S/N PL-6



BEARING S/N PL-7



NOTE: SMALL SPALLS APPEARED AFTER 36 HOURS AT 14,000 rpm UNDER FULL RADIAL AND THRUST LOAD

Figure 31. HM926700-Series Bearing Cones From Test Number 4.

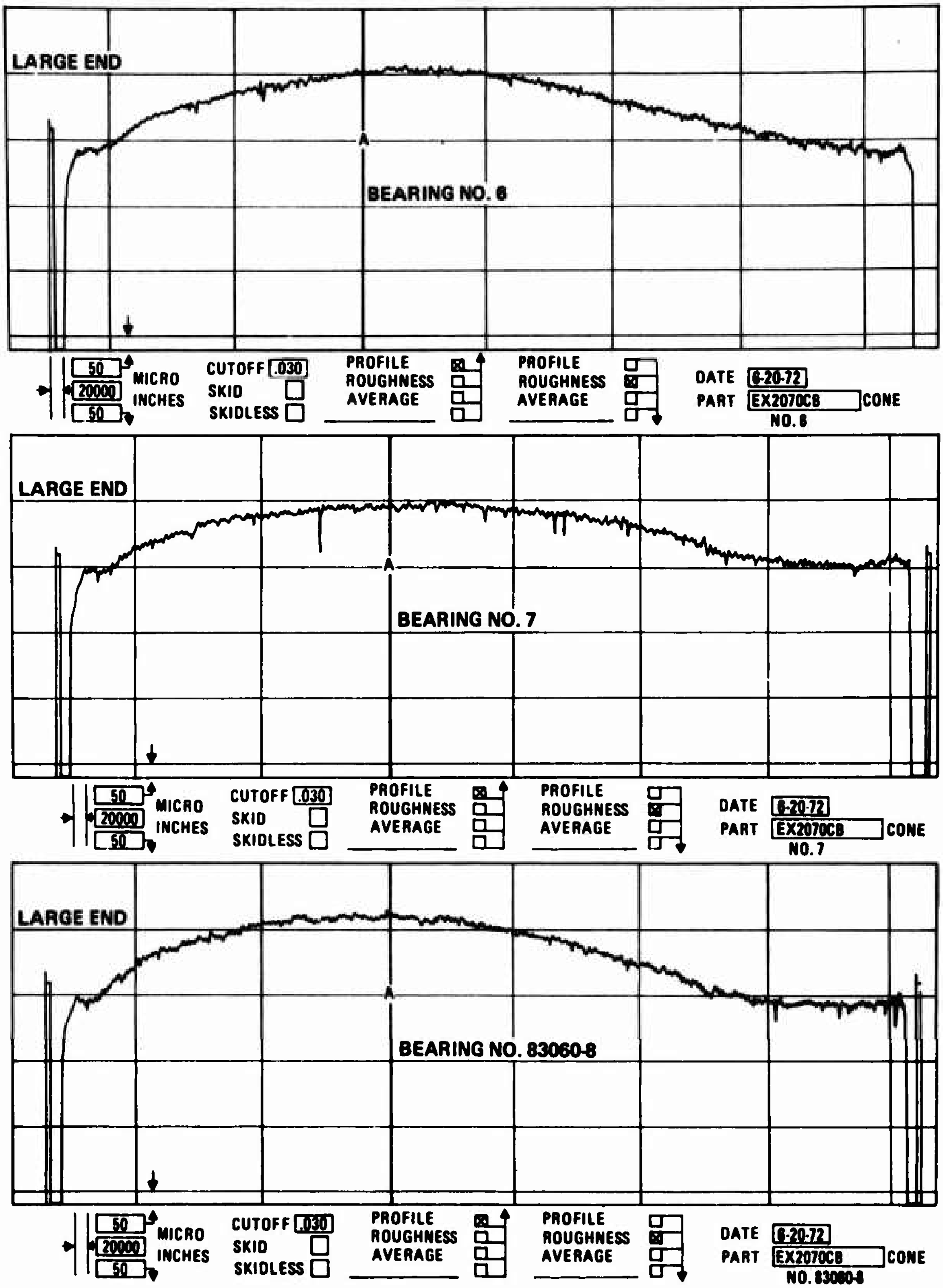


Figure 32. Rectilinear Traces of Bearing Cone Races From Test Number 4.

Five microhardness measurements between the surface and a depth of 0.038 inch below the surface were taken near the middle of the discolored area. They were 55.5 Rockwell "C", 55.0 Rockwell "C", 54.5 Rockwell "C", 56.0 Rockwell "C", and 55.0 Rockwell "C" with Knoop converted to Rockwell "C".

In the clear area, two microhardness measurements were taken near the surface and one measurement at 0.015 inch below the surface. They were 60.5, 60.0, and 58.0 Rockwell "C", respectively (Knoop converted to Rockwell "C").

The spalls are located on the highest point of crown on each cone. The localized overheating resulting in the crown apparently was responsible for the early appearance of these spalls.

Boeing-Vertol and Timken decided that testing these bearings at 11,500 rpm, helicopter flight speed, instead of the 14,000-rpm overspeed condition will lower the bearing operating temperatures below 300°F. This would substantially reduce the chance of the bearing cones' becoming tempered with resulting loss of hardness and transformation of retained austenite which had caused the premature spalling experienced in tests 4 and 5.

EVALUATION OF FAIL-SAFE WEAR-RESISTANT BEARING MATERIALS

The summary of results, Tables III and IV, lists all observed test work for this phase of the test program. Table III lists all materials tested in the longer 2½-hour oil-on-wear tests. Table IV lists all materials tested in the shorter 10-minute oil-on/oil-off tests.

Figure 33 compares running time after oil-off for all of the tested materials. Figure 34 shows typical wear scars on the materials. Timken standard steel test cups and blocks score immediately after the oil is shut off.

The Borkote materials show the least amount of wear during the oil-off tests. Borkote on one part performs as well as Borkote on both parts. The LP alloys, however, as a group show considerably more life than Borkote materials in the oil-off tests. The LP alloys gave the best overall life with a reasonable amount of wear. The LPA 404 alloy with 40-percent 4600-steel binder was the best of the LP alloys tested.

The DuPont Vespel and Marlin-Rockwell materials gave a good life in oil-off testing,but had a considerable amount of wear.

Beryllium copper material had a good life with fair wear.

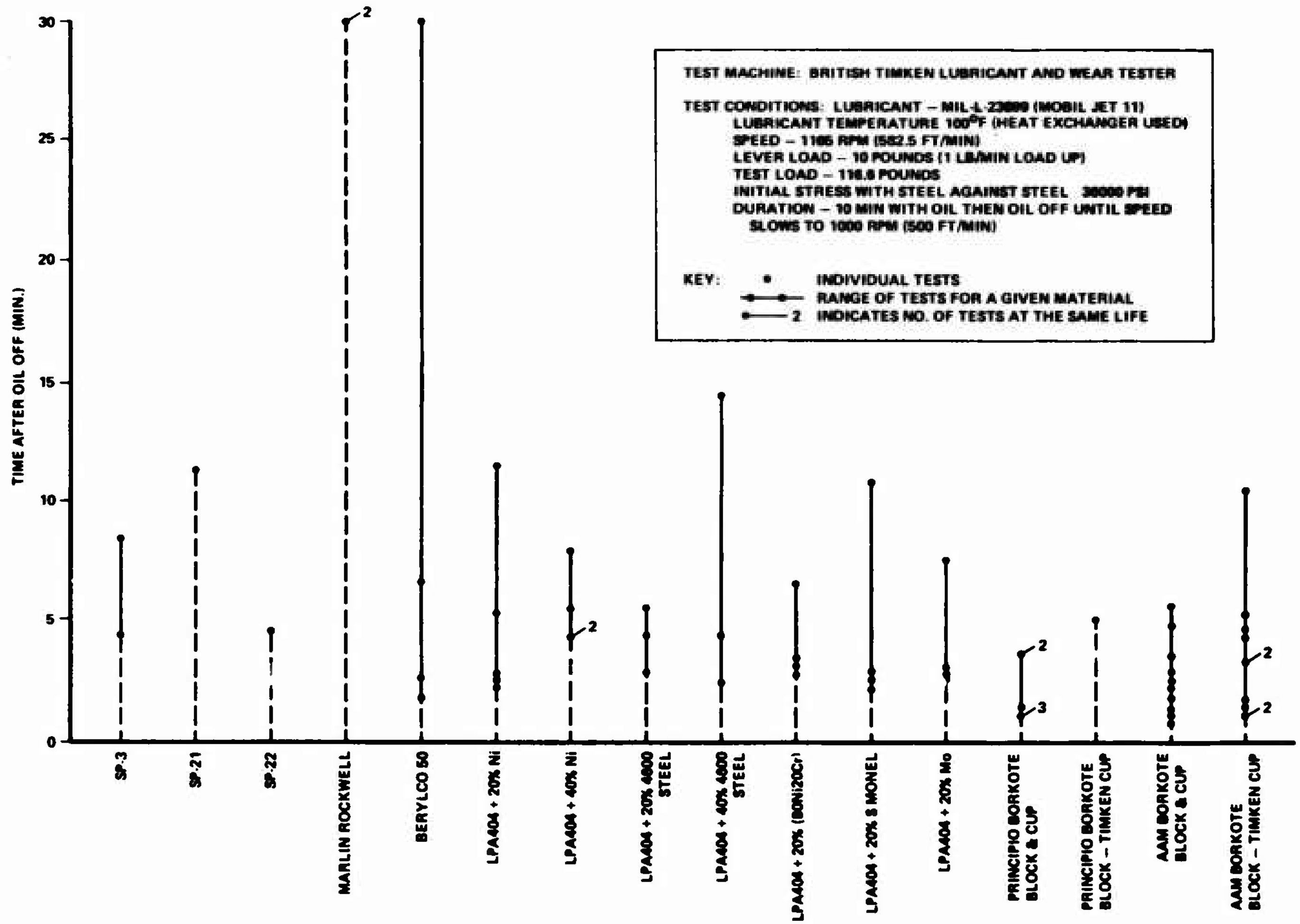
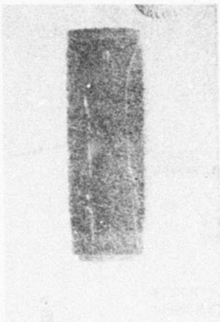
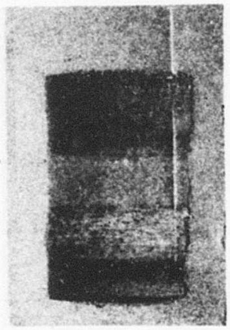


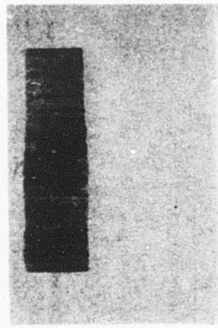
Figure 33. Comparison of Materials Considered for Use in the Fail-Safe Bearing Test Program.



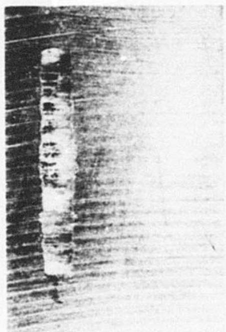
DUPONT VESPEL
SP-21



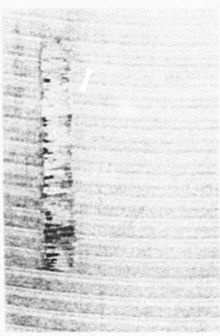
MARLIN ROCKWELL
CORP



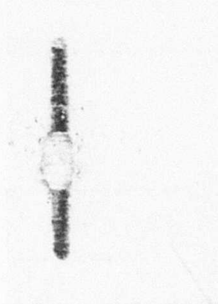
BERYLLIUM
COPPER



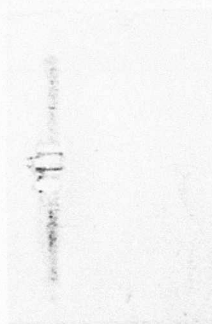
LP404 ALLOY WITH
40-PCT NICKEL BINDER



LP404 ALLOY WITH
40-PCT 4600-STEEL
BINDER



BORKOTE



BORKOTE WITH
BORKOTE RING

Figure 34. Typical Wear Scars on Fail-Safe Wear-Resistant Material Blocks.

TABLE III. SUMMARY OF RESULTS - QUALITATIVE COMPARISON OF VARIOUS WEAR-RESISTANT MATERIALS (2-1/2-HOUR DURATION)

Cup Material	Block Material	Block No.	Manufacturer	Description	Torque (lb in.) Start	Torque (lb in.) Finish	Weight Loss Ring (grams)	Weight Loss Block grams	Weight Loss Block pct	Scar Width (in.)	Final Stress (psi)
Standard Timken	Vespel SP-3	3	Dupont	Polyimide (15 pct by weight M_oS_2)	12	12	-0.0012	0.0111	0.230	0.2130	1095
Standard Timken	Vespel SP-3	4	Dupont	Polyimide (15 pct by weight M_oS_2)	12	12	-0.0008	0.0164	0.330	0.2030	1149
Standard Timken	Vespel SP-21	5	Dupont	Polyimide (15 pct by weight graphite)	6	7	-0.0014	0.0069	0.150	0.1610	1448
Standard Timken	Vespel SP-22	4	Dupont	Polyimide (40 pct by weight graphite)	15	16	+0.0005	0.0465	0.910	0.3400	686
Borkote	Borkote	1	Principio Division	Standard steel components with 0.001-in. Borkote	25	18	+0.0003	0.0001	0.000	0.0380	6137
Borkote	Borkote	2	Principio Division	Standard Timken components with 0.0015-in. Borkote	20	15	-0.0006	0.0001	0.000	0.0350	6663
Borkote	Borkote	3	Principio Division	Standard Timken components with 0.002-in. Borkote	20	12	+0.0022	+0.0016	+0.007	0.0375	6218
Standard Timken	Standard Timken	1	Timken	Standard Timken block and cup	22	15	-0.0024	0.0009	0.004	0.0260	8969
Standard Timken	Marlin Rockwell	1	Marlin Rockwell	Copper impregnated block	16	16	-0.0008	0.0014	0.016	0.1560	1495
Standard Timken	Borkote	1	Principio Division	Standard Timken cup with 0.0015-in. Borkote block	20	18	-0.0008	0.0031	0.014	0.0760	3068

NOTE: Test Machine: British Timken lubricant and wear tester

Test Conditions: Oil - MIL-L-23699 (Mobil Jet II)
Oil Temperature - 100°F heat exchanger used
Speed - 1165 rpm
Lever Load - 10 pounds (1 pound every 10 minutes)
Initial stress for steel cups and blocks - 36,000 psi

TABLE IV. SUMMARY OF RESULTS - QUALITATIVE ANALYSIS OF VARIOUS WEAR-RESISTANT MATERIALS (10-MINUTE DURATION)

Cup Material	Block Material	Block No.	Manufacturer	Description	Time Without Oil	Torque (lb in.)	Weight Loss: Ring (grams)	Weight Loss: Block grams	Weight Loss: Block pct	Scar Width (in.)	Final Stress (psi)
Standard Timken	Standard Timken	1-2	Timken	Standard Timken block and cup	Both scored at 7 lb load	--	--	--	--	--	--
Standard Timken	Standard Timken	2-2	Timken	Standard Timken block and cup		--	--	--	--	--	--
Standard Timken	Vespel SP-3	3-2	Dupont	Polyimide (15 pct by weight MoS_2)	8 min 45 sec	9	-0.0011	0.0074	0.150	0.2030	1149
Standard Timken	Vespel SP-3	4-2	Dupont	Polyimide (15 pct by weight MoS_2)	4 min 20 sec	16	-0.0013	0.0125	0.250	0.2050	1138
Standard Timken	Vespel SP-21	5-2	Dupont	Polyimide (15 pct by weight graphite)	11 min 20 sec	14	+0.0004	0.0040	0.080	0.1610	1448
Standard Timken	Vespel SP-22	4-2	Dupont	Polyimide (40 pct by weight graphite)	4 min 30 sec	16	+0.0001	0.0238	0.470	0.2670	873
Borkote	Borkote	1-2	Principio Division	Standard Timken components with 0.001-in. Borkote	1 min 10 sec	20	+0.0006	+0.0004	+0.002	0.0320	7288
Borkote	Borkote	1-3	Principio Division	Same as above with cup used in 2-1/2 hr run	1 min 20 sec	17	+0.0005	+0.0008	+0.004	0.0300	7773
Borkote	Borkote	2-2	Principio Division	Standard Timken components with 0.0015-in. Borkote	3 min 30 sec	21	+0.0003	+0.0013	+0.006	0.0320	7288
Borkote	Borkote	2-3	Principio Division	Same as above with cup used in 2-1/2 hr run	1 min 10 sec	19	-0.0002	0.0003	0.001	0.0300	7773
Borkote	Borkote	3-2	Principio Division	Standard Timken components with 0.002-in. Borkote	1 min 10 sec	22	-0.0012	0.0011	0.005	0.0345	6759
Borkote	Borkote	3-3	Principio Division	Same as above with cup used in 2-1/2 hr run	3 min 30 sec	18	-0.0009	+0.0006	+0.003	0.0295	7905
Standard Timken	Borkote	2-2-1	Principio Division	Standard Timken with 0.0015-in. Borkote block	8 min 30 sec	24	--	0.0001	0.001	0.0470	4962

TABLE IV. Continued.

Cup Material	Block Material	Block No.	Manufacturer	Description	Time Without Oil	Torque (lb in.)	Weight Loss Ring (grams)	Weight Loss Block grams	Weight Loss Block pct	Scar Width (in.)	Final Stress (psi)
Standard Timken	Borkote	1-2-1	Principio Division	Standard Timken with 0.001-in. Borkote block	3 min	23	+0.0013	0.0002	0.001	0.0340	6859
Standard Timken	Borkote	3-2-1	Principio Division	Standard Timken with 0.002-in. Borkote block	5 min	24	0.0000	0.0002	0.001	0.0470	4962
Borkote	Borkote	B-60-1	Atlantic Advanced Metals	Parts processed 60 min	5 min 30 sec	21	-0.0010	0.0010	0.005	0.0510	4573
Standard Timken	Borkote	B-60-1	Atlantic Advanced Metals	Standard Timken cup and block processed 60 min	3 min 11 sec	20	-0.0010	0.0009	0.004	0.0490	4759
Borkote	Borkote	B-60-2	Atlantic Advanced Metals	Parts processed 60 min	2 min 53 sec	22	-0.0007	0.0004	0.002	0.0510	4573
Borkote	Borkote	B-120-1	Atlantic Advanced Metals	Parts processed 120 min	2 min 18 sec	21	-0.0001	0.0005	0.002	0.0550	4240
Borkote	Borkote	B-120-2	Atlantic Advanced Metals	Parts processed 120 min	1 min 22 sec	22	-0.0003	0.0010	0.005	0.0630	3702
Borkote	Borkote	B-1	Atlantic Advanced Metals	Block and cup processed 15 min	1 min 50 sec	20	+0.0004	0.0016	0.007	0.0900	2591
Standard Timken	Borkote	B-1	Atlantic Advanced Metals	Block processed 15 min	Score at 9 lb	--	+0.0006	0.1570	--	0.1490	--
Borkote	Borkote	B-2	Atlantic Advanced Metals	Block and cup processed 20 min	2 min 20 sec	23	-0.0005	0.0008	0.004	0.0840	2776
Standard Timken	Borkote	B-2	Atlantic Advanced Metals	Block processed 20 min	1 min 15 sec	20	-0.0001	0.0017	0.008	0.0730	3195
Borkote	Borkote	B-3	Atlantic Advanced Metals	Block and cup processed 30 min	1 min 10 sec	23	-0.0010	+0.0003	+0.001	0.0370	6303
Standard Timken	Borkote	B-3	Atlantic Advanced Metals	Block processed 30 min	1 min 20 sec	22	-0.0002	0.0008	0.004	0.0420	5552
Borkote	Borkote	B-4	Atlantic Advanced Metals	Block and cup processed 45 min	2 min 5 sec	23	-0.0009	+0.0003	+0.001	0.0500	4664
Standard Timken	Borkote	B-4	Atlantic Advanced Metals	Block processed 45 min	5 min 5 sec	22	0.0000	0.0009	0.004	0.0330	7066
Borkote	Borkote	B-6	Atlantic Advanced Metals	Block and cup processed 90 min	3 min 35 sec	22	-0.0001	0.0004	0.002	0.0710	3285
Standard Timken	Borkote	B-6	Atlantic Advanced Metals	Block processed 90 min	3 min 10 sec	20	-0.0007	0.0008	0.004	0.0470	4962

TABLE IV. Continued.

Cup Material	Block Material	Block No.	Manufacturer	Description	Time Without Oil	Torque (lb in.)	Weight Loss Ring (grams)	Weight Loss Block grams	Weight Loss Block pct	Scar Width (in.)	Final Stress (psi)
Borkote	Borkote	B-8	Atlantic Advanced Metals	Block and cup processed 180 min	4 min 40 sec	22	-0.0012	0.0005	0.002	0.0770	3029
Standard Timken	Borkote	B-8	Atlantic Advanced Metals	Block processed 180 min	1 min 5 sec	17	0.0000	0.0004	0.003	0.0450	5182
Borkote	Borkote	B-10	Atlantic Advanced Metals	Block and cup processed 24 hr	45 sec	18	-0.0020	+0.0003	+0.001	0.0260	8969
Standard Timken	Borkote	B-10	Atlantic Advanced Metals	Block processed 24 hr	10 min 35 sec	18	+0.0011	0.0010	0.005	0.0800	2915
Standard Timken	Borkote	B-60-3	Atlantic Advanced Metals	Block processed 60 min	3 min 10 sec	20	--	--	--	0.0420	5552
Standard Timken	Borkote	B-60-4	Atlantic Advanced Metals	Block processed 60 min	1 min	20	--	--	--	0.0410	5688
Standard Timken	Borkote	B-120-3	Atlantic Advanced Metals	Block processed 120 min	3 min	21	--	--	--	0.0470	4962
Standard Timken	Borkote	B-120-4	Atlantic Advanced Metals	Block processed 120 min	4 min 30 sec	22	--	--	--	0.0500	4664
Standard Timken	LP 400 + 20 pct Nickel		Principio Division	Standard Timken cup and LP 400 + 20 pct Nickel block	11 min 30 sec	19	-0.0014	0.0008	0.003	0.0480	4858
Standard Timken	LPA 404	1076A	Principio Division	LP alloy + 40 pct 4600 steel binder	14 min 30 sec	18	+0.0006	0.0015	0.006	0.0790	2952
Standard Timken	LPA 404	1076B	Principio Division	LP alloy + 40 pct 4600 steel binder	5 min 10 sec	22	+0.0010	0.0018	0.007	0.0690	3380
Standard Timken	LPA 404	1076C	Principio Division	LP alloy + 40 pct 4600 steel binder	4 min 15 sec	14	-0.0001	0.0004	0.002	0.0670	3480
Standard Timken	LPA 404	1076D	Principio Division	LP alloy + 40 pct 4600 steel binder	2 min 25 sec	16	+0.0009	0.0012	0.005	0.0560	4164
Standard Timken	LPA 404	1073A	Principio Division	LP alloy + 40 pct Nickel binder	5 min 30 sec	20	+0.0005	0.0013	0.005	0.0760	3068
Standard Timken	LPA 404	1073B	Principio Division	LP alloy + 40 pct Nickel binder	4 min 20 sec	22	+0.0014	0.0020	0.008	0.0720	3239
Standard Timken	LPA 404	1073C	Principio Division	LP alloy + 40 pct Nickel binder	7 min 50 sec	16	+0.0013	0.0011	0.004	0.0710	3285
Standard Timken	LPA 404	1073D	Principio Division	LP alloy + 40 pct Nickel binder	4 min 20 sec	18	+0.0009	0.0018	0.007	0.0700	3331

TABLE IV. Continued.

Cup Material	Block Material	Block No.	Manufacturer	Description	Time Without Oil	Torque (lb in.)	Weight Loss: Ring (grams)	Weight Loss: Block grams	Weight Loss: Block pct	Scar Width (in.)	Final Stress (psi)
Standard Timken	LPA 404	1072A	Principio Division	LP alloy + 20 pct Nickel binder	2 min 15 sec	18	-0.0004	0.0017	0.006	0.0530	4400
Standard Timken	LPA 404	1072B	Principio Division	LP alloy + 20 pct Nickel binder	5 min 15 sec	17	+0.0010	0.0004	0.002	0.0540	4318
Standard Timken	LPA 404	1072C	Principio Division	LP alloy + 20 pct Nickel binder	2 min 10 sec	17	+0.0009	0.0012	0.005	0.0570	4091
Standard Timken	LPA 404	1072D	Principio Division	LP alloy + 20 pct Nickel binder	2 min 35 sec	17	+0.0011	0.0016	0.006	0.0600	3887
Standard Timken	LPA 404	1075A	Principio Division	LP alloy + 20 pct 4600 steel binder	5 min 30 sec	21	-0.0012	0.0019	0.008	0.0910	2563
Standard Timken	LPA 404	1075B	Principio Division	LP alloy + 20 pct 4600 steel binder	4 min	18	+0.0006	0.0003	0.001	0.0570	4091
Standard Timken	LPA 404	1075C	Principio Division	LP alloy + 20 pct 4600 steel binder	2 min 55 sec	18	+0.0011	0.0014	0.006	0.0630	3702
Standard Timken	LPA 404	1075D	Principio Division	LP alloy + 20 pct 4600 steel binder	4 min 10 sec	17	+0.0001	0.0014	0.006	0.0590	3953
Standard Timken	LPA 404	1077A	Principio Division	LP alloy + 20 pct (80 Nickel 20 Cr.) binder	3 min 25 sec	18	0.0000	0.0016	0.006	0.0560	4164
Standard Timken	LPA 404	1077B	Principio Division	LP alloy + 20 pct (80 Nickel 20 Cr.) binder	3 min	19	+0.0009	0.0012	0.005	0.0620	3761
Standard Timken	LPA 404	1077C	Principio Division	LP alloy + 20 pct (80 Nickel 20 Cr.) binder	2 min 40 sec	18	+0.0009	0.0006	0.002	0.0570	4091
Standard Timken	LPA 404	1077D	Principio Division	LP alloy + 20 pct (80 Nickel 20 Cr.) binder	6 min 38 sec	17	+0.0002	0.0013	0.005	0.0490	4759
Standard Timken	LPA 404	1078A	Principio Division	LP alloy + 20 pct S-monel binder	10 min 50 sec	18	-0.0003	0.0081	0.032	0.1310	1780
Standard Timken	LPA 404	1078B	Principio Division	LP alloy + 20 pct S-monel binder	2 min 30 sec	22	+0.0007	0.0011	0.004	0.0610	3823
Standard Timken	LPA 404	1078C	Principio Division	LP alloy + 20 pct S-monel binder	2 min 25 sec	18	+0.0015	0.0012	0.005	0.0670	3480
Standard Timken	LPA 404	1078D	Principio Division	LP alloy + 20 pct S-monel binder	2 min	20	0.0000	0.0020	0.008	0.0550	4240

TABLE IV. Continued.

Cup Material	Block Material	Block No.	Manufacturer	Description	Time Without Oil	Torque (lb in.)	Weight Loss Ring (grams)	Weight Loss Block grams	Weight Loss Block pct	Scar Width (in.)	Final Stress (psi)
Standard Timken	LPA 404	1079A	Principio Division	LP alloy + 20 pct Mo. binder	7 min 20 sec	17	+0.0002	0.0016	0.006	0.0610	3823
Standard Timken	LPA 404	1079B	Principio Division	LP alloy + 20 pct Mo. binder	2 min 40 sec	20	+0.0012	0.0019	0.007	0.0570	4091
Standard Timken	LPA 404	1079C	Principio Division	LP alloy + 20 pct Mo. binder	2 min 35 sec	20	-0.0007	0.0008	0.003	0.0590	3953
Standard Timken	LPA 404	1079D	Principio Division	LP alloy + 20 pct Mo. binder	2 min 30 sec	20	+0.0004	0.0014	0.006	0.0540	4319
Standard Timken	Marlin Rockwell	1-2	Marlin Rockwell Corp.	Copper impregnated block	30 min	14	+0.0007	0.0376	0.420	0.3200	729
Standard Timken	Marlin Rockwell	1-3	Marlin Rockwell Corp.	Copper impregnated block	30 min	17	+0.0007	0.0215	0.240	0.2900	804
Standard Timken	Be Cu	50A	Kawecki Berylco	Beryllium copper No. 50 alloy	6 min 20 sec	12	0.0002	0.0142	0.060	0.1350	1727
Standard Timken	Be Cu	50B	Kawecki Berylco	Beryllium copper No. 50 alloy	30 min	11	+0.0006	0.0052	0.020	0.0985	2368
Standard Timken	Be Cu	50C	Kawecki Berylco	Beryllium copper No. 50 alloy	2 min 30 sec	11	+0.0002	0.0082	0.030	0.1120	2082
Standard Timken	Be Cu	50D	Kawecki Berylco	Beryllium copper No. 50 alloy	1 min 40 sec	9	0.0003	0.0080	0.030	0.1065	2190

NOTE: Test Machine: British Timken lubricant and wear tester

Test Conditions: Oil - MIL-L-23699 (Mobil Jet II)
Oil Temperature - 100°F heat exchanger used
Speed - 1165 RPM
Lever Load - 10 pounds (1 pound every 10 minutes)
Initial stress for steel cups and blocks is 36,000 psi

EVALUATION OF FAIL-SAFE BEARING DESIGNS

DuPont LPA 404 alloy plus 40-percent 4600 steel and Borkote were selected as materials to evaluate in a fail-safe bearing design, based on the results from section, EVALUATION OF FAIL-SAFE WEAR-RESISTANT MATERIALS.

All fail-safe bearing test cones had 36 radial lubrication oil holes plus 4 holes to lubricate the large end guiding flange of the Z-type, silver-plated cage.

The first fail-safe bearing test was run with cones having LPA 404 plus 40-percent 4600-steel binder applied to the separable ribs. This test was stopped after running at 9500 rpm for 2 hours (with oil lubrication) due to a noise coming from the test housing. After-test inspection showed that a piece of the LPA 404 broke the separable rib (see Figure 35). The hot consolidated LPA 404 rib rings on these bearings were high-temperature brazed to the separable rib.

The second fail-safe test was run with bearing cones having Borkote applied to the separable ribs. This test ran for 2 hours at 9300 rpm with oil lubrication. Bearing damage occurred as the speed increased from 9300 to 11,500 rpm. The separable ribs of two of the bearings had radial cracks which caused the test to stop. One of the separable ribs had two cracks (Figure 36)--one crack occurring at one of the four pin holes at the backface and the other crack occurring at one of the four radial oil holes at the rib OD. The pins at the backface keep the separable rib from rotating on the cone, while oil from the radial oil holes on the rib OD lubricates the area where the cage ID guides on the rib OD.

The third fail-safe test was run with DuPont LPA 404 plus 40-percent SAE 4600-steel binder hot consolidated directly to the separable rib. This test ran successfully for 24 hours with oil lubrication. At the end of the 24 hours, the bearing end-play was checked. The end-play of the bearings in the drive end housing changed from 0.0025-inch initial to 0.003-inch; and the bearings in the opposite drive end housing changed from 0.0015-inch initial to 0.003 inch. These bearings were then reinstalled in the test machine and tested for 2 hours at 11,500 rpm with oil lubrication. After the 2-hour period, the oil was shut off and the bearings ran for 8.5 seconds before cone rib/roller end scuffing occurred on one of the bearings.

The fourth fail-safe test was run with bearings identical to the third test. This test was to be an extended 150-hour test with lubricating oil and measuring end-play at specified intervals. The end-play at the end of 24 hours did not change from the initial end-play of 0.0025-inch in the drive end housing and 0.005-inch in the opposite drive end housing. Cone

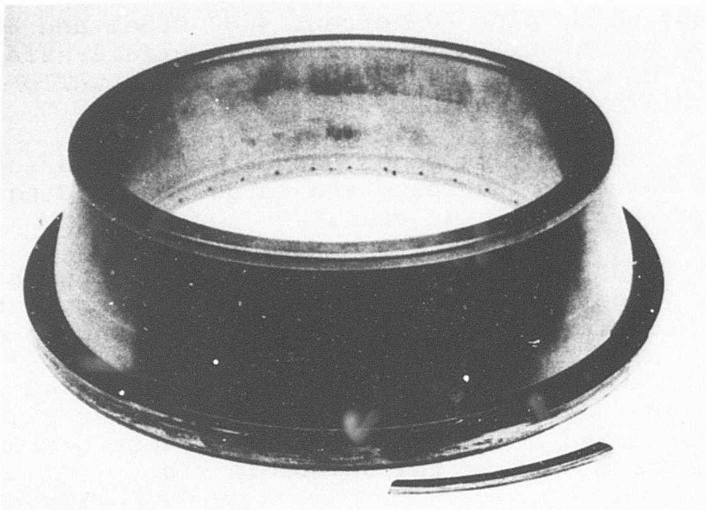


Figure 35. Failure of High-Temperature Brazed Bonding.

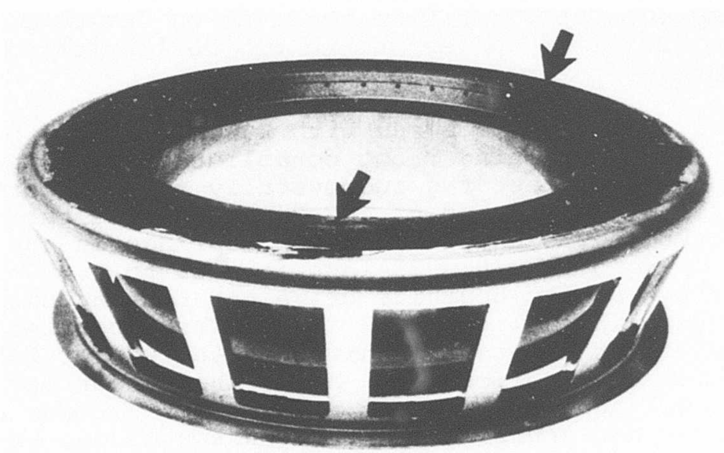


Figure 36. Cracking of Borkote Rib Flange.

rib/roller end scuffing damage occurred at 50.5 hours, running at 11,500 rpm with oil lubrication.

A tabulation in Appendix V shows level-off bearing temperatures at various speeds, the cylinder radial and thrust loads applied to the housings, and the bearing end-play adjustments for the fail-safe bearing tests.

A repeat fail-safe test was run with three bearings that were not damaged from the third fail-safe test and one undamaged bearing from the fourth fail-safe test. In this repeat test, the four bearings were tested with standard stamped-steel, silver-plated cages. This test was run at 11,500 rpm for 3 hours with oil lubrication to allow temperatures to level off. At the end of the 3-hour period, the oil was shut off and the test ran for 10 seconds before cone rib/roller end scuffing occurred on one of the bearings.

<u>End-Play Variation Tests - Modified HM926700-Series Bearing</u>

These tests were conducted with bearing cones having 40 radial lubrication oil holes from the cone manifold to the large end rib undercut.

The first additional HM926700-series bearing test (test number 11) was to be a bearing adjustment test. Bearing damage occurred to two of the bearings at 9000 rpm. The initial end-play adjustment was 0.0015-inch in the drive end housing and 0.003-inch in the opposite drive end housing. The bearing damage in this test could not be explained.

Test number 12 was set up with an initial end-play adjustment of 0.0011-inch drive end and 0.002-inch opposite drive end. This test ran for several hours at 11,500 rpm to allow temperatures to stabilize. This test was then reassembled with the end-play increased to 0.003-inch drive end and 0.0035-inch opposite drive end. After several hours running, the temperatures leveled off to the same temperatures of the original end-play adjustments. These bearings were again reassembled with the end-play reduced to 0.0001-inch drive end and 0.0001-inch opposite drive end. With this end-play adjustment, bearing damage occurred while accelerating to 11,500 rpm.

Test number 13 was run with two different end-play adjustments. Bearing damage occurred while accelerating from 8000 to 11,500 rpm with an end play adjustment of 0.0042 inch (same on both drive end and opposite drive end housings). Figure 37 shows a plot of bearing cup temperatures versus bearing end-play setting for several tests conducted during this phase.

Test numbers 14 and 15 had premature bearing damage due to test assembly problems. In test number 14, a Z-type cage was

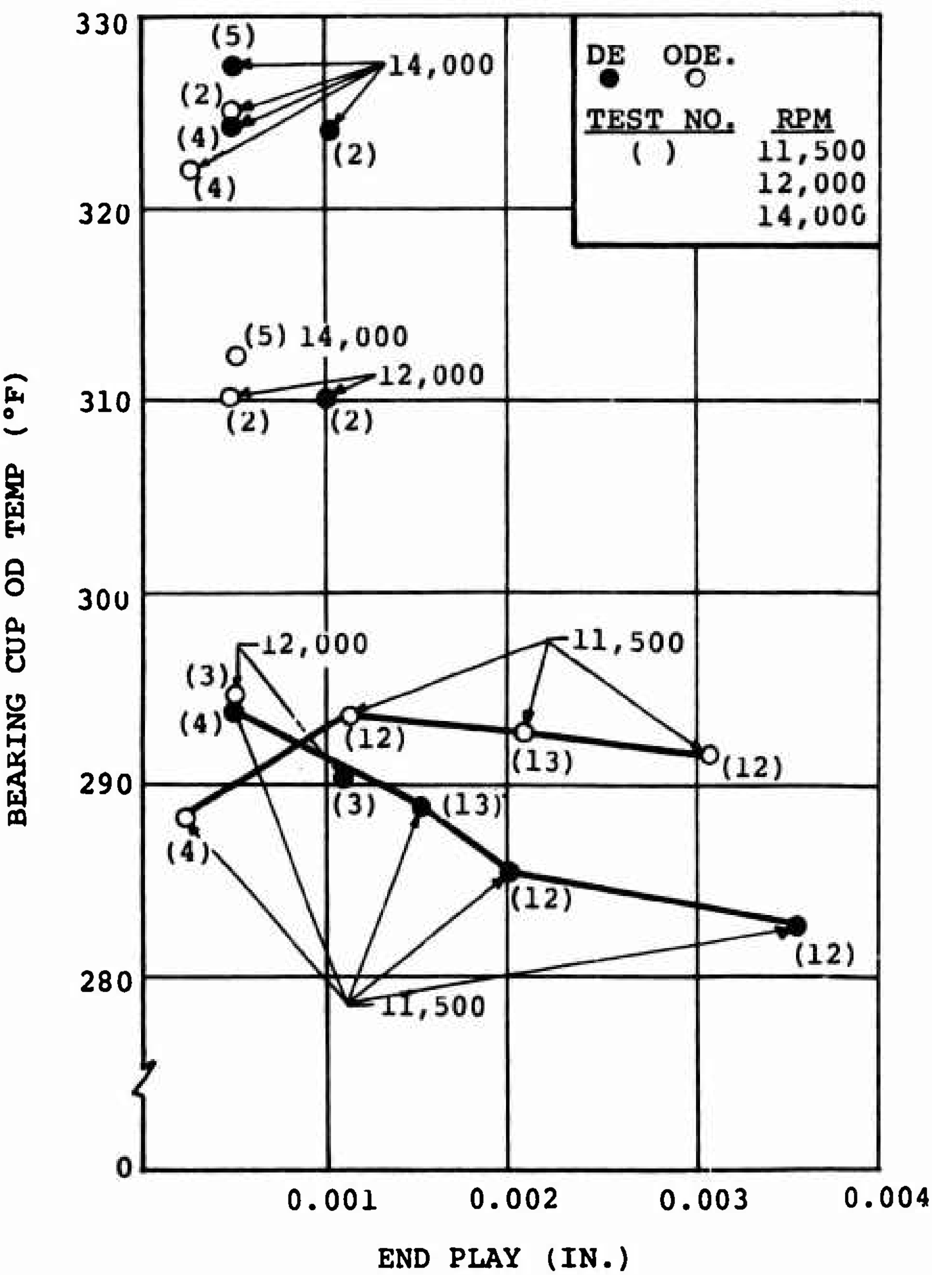


Figure 37. Bearing End-Play Versus Cup Outside Diameter Temperature.

unknowingly bent on assembly, causing early bearing damage. In test number 15, the drive spindle was assembled to the test shaft in such a way that it would not permit the lubricating oil from the center of the shaft to reach the drive end bearing.

In test numbers 16 through 33, attempts were made to reach 11,500 rpm. The cause for the bearing damage in these tests could not be determined at that time. Periodically during these tests, Timken-owned bearings were run successfully in the machine to determine if the unexplained bearing damage was a result of a machine malfunction. Figure 38 shows the scuffing damage that occurred during test number 7.

Several bearing design changes were made to find cause for unexplained scuffing damage and eliminate this bearing damage. Although the changes which follow did not eliminate the bearing damage, they were certainly logical steps to improve bearing performance.

The oil manifold backface chamfer of the cone was changed (Figure 39) to eliminate the possibility of the oil in the manifold leaking between the cone backface and shaft shoulder. The large end cone rib undercut was reduced to allow for more available contact area between the roller spherical end radius and rib face flat. Two of the bearings were run in a press for 7 and 8 minutes under 51,800 and 40,000 pounds thrust load, respectively,to determine if the running-in had an effect on bearing performance. The roller spherical end radius was increased from 80 percent of the apex length to 85 percent in order to reduce the unit stress on the rib face.

A correlation study made on Z-type cage flange guidance clearance, Appendix VI, did not show any correlation with the roller end/cone rib scuffing damage.

After the testing of the modified HM926700-series bearings was completed and the decision was made to test the XC11439 series bearing, it was discovered that there was a significant difference in the level of retained austenite from bearing lot to lot. The higher the level of retained austenite, the less the resistance to scuffing damage between the roller end and cone rib at the higher stress levels in the steep-angle, HM926700-series bearing.

Cone Flange Oil Distribution Spin Tests

Figures 8 through 13 are drawings of the simulated cones tested,and Appendix VIII contains high-speed strobe light photographs, showing the oil distribution at the cone rib interface for each test. A brief description of the test parameter is given on each photograph. The photographs are grouped

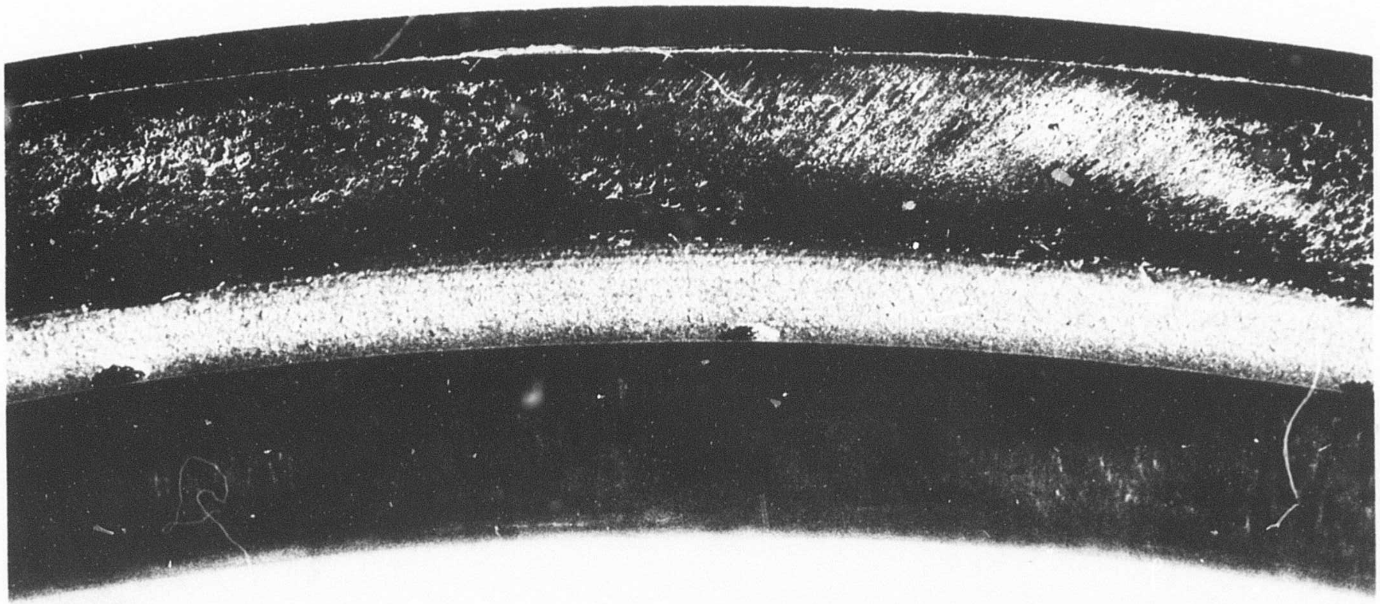


Figure 38. Cone Rib Scuffing Damage From Test Number 17.

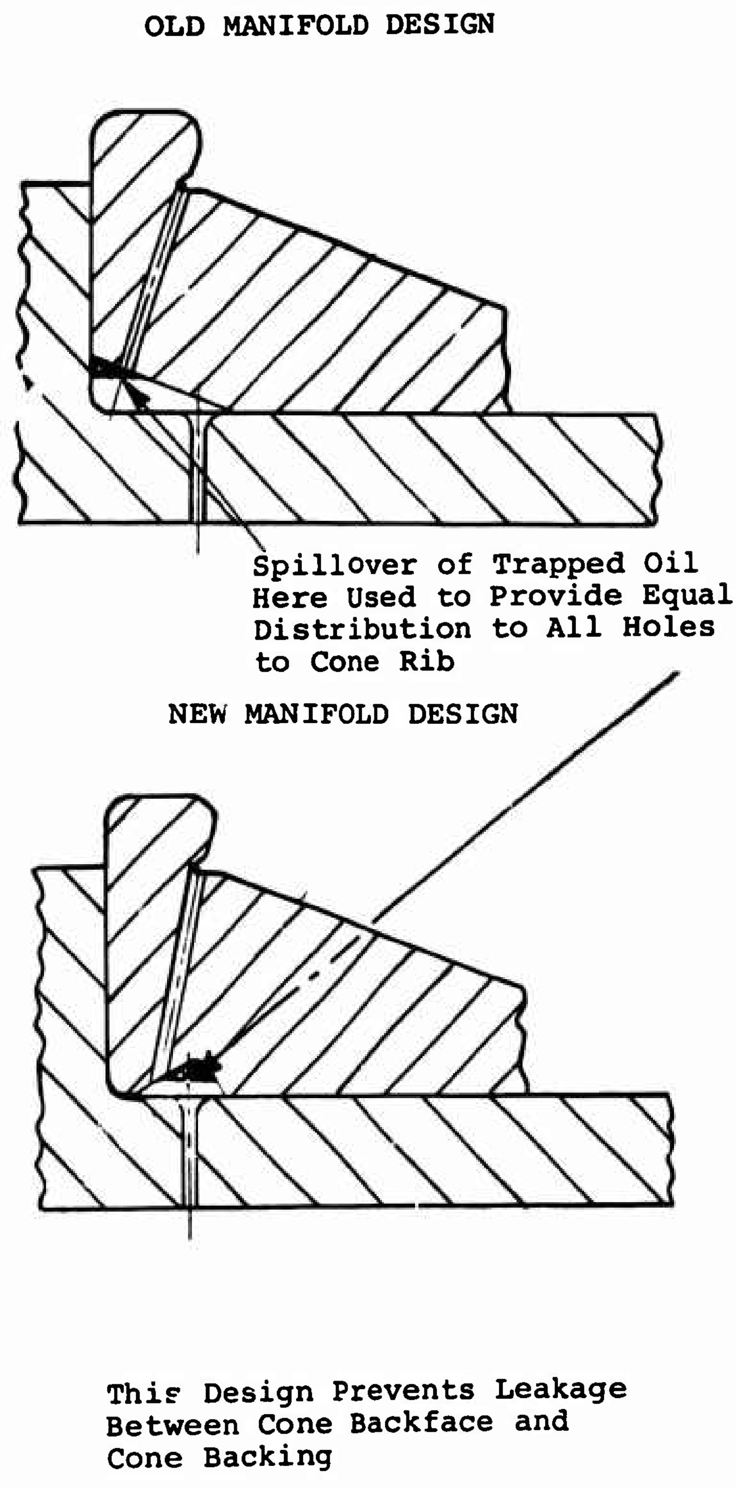


Figure 39. Old and New Type Cone Bore Manifold Design.

according to simulated cone part number, style of oil manifold, length of oil hole and angle of oil hole inclination.

Testing of 301-10676 Bearings

Table V is a summary of results of the XC11439-series (Boeing Vertol P/N 301-10676) bearing tests. The table lists the test number, bearing cone, cup and cage serial number according to position in the machine, total hours each component ran at 11,500 rpm, and the bearing end-play adjustments.

The HLH high-speed tapered roller bearing test machine was modified to test the XC11439-series bearing (Vertol P/N 301-10676). While a new test shaft was being made, the HM926700-series bearing test shaft was modified on one end to test two XC11439 with two HM926700 bearings. The first group of XC11439 bearing cones were ground to 0.030-inch on the large end rib OD. This is the portion of the cone where the machine Z-type cage guides on the cone. For this test the large end cage ID riding land was spray welded with Metcology No. 2 to a thickness of 0.015 inch on a side to make up for the cone rib being ground too small. This test ran for 24-1/2 hours at 11,500 rpm. All bearings were satisfactory after the test.

The next test was run with the new test shaft with all four XC11439-series bearings. Two of the bearings were tested with spray welded cages and two bearings were tested with cones with 0.015-inch-per-side chrome plating applied to the rib OD. This test was run for 1 hour at 11,500 rpm. All bearings were satisfactory after the test.

The third test was run with bearings with the large end cone rib OD ground to size. One of the bearings in this test was damaged upon reaching 11,500 rpm. This damage was attributed to insufficient bearing end-play setting. Settings of 0.0025-inch end-play in both the drive end and opposite drive end housings were used. The first two tests were run with 0.005-inch and 0.0035-inch end-play, respectively. It was decided to run the rest of the tests with at least 0.0035-inch end-play.

The fourth test was set up with bearing adjustments increased to 0.0035-inch in both the DE and ODE housings. These bearings ran for 24 hours at 11,500 rpm and were satisfactory after the test.

The fifth test was run with four new bearings. Only the cup tested with one of the bearings had been run before. One of the bearings was damaged at 11,500 rpm and two more had very slight cone rib/roller end scuffing. The cause for this bearing damage was due to the silver plating separating from the

TABLE V. SUMMARY OF RESULTS - 301-10676 BEARING

Test No.		Component Serial Numbers - ( ) Hours at 11,500 rpm				End Play DE Housing/ ODE Housing (in.)	Remarks
		DE	DEC	ODEC	ODE		
1	Cone	S-1	S-3	C73-34 (24.5)	C73-19 (24.5)	0.005/0.005	Ran 24.5 hr- bearings OK
	Cup	(Timken slotted bearings)		D73-8 (24.5)	D73-3 (24.5)		
	Cage			73-34 (24.5)	73-19 (24.5)		
				(Spray-welded cages)			
2	Cone	C73-19 (25.5)	C73-37 (1)	C73-25 (1)	C73-4 (1)	0.0035/0.0035	Ran 1 hr- bearings OK
	Cup	D73-23 (1)	D73-10 (1)	D73-8 (25.5)	D73-3 (25.5)		
	Cage	73-19 (25.5)	73-37 (1)	73-25 (1)	73-4 (1)		
		(Spray-welded cages)		(Chrome rib OD)			
3	Cone	C73-49 (0)	C73-59 (0)	C73-69 (0)	C73-73 (0)	0.0025/0.0025	73-49 damaged at 11,500 rpm
	Cup	D73-23 (1)	D73-10 (1)	D73-8 (25.5)	D73-3 (25.5)		
	Cage	73-49 (0)	73-59 (0)	73-69 (0)	73-73 (0)		
		(All cages Lancaster plated)					
4	Cone	C73-19 (49.5)	C73-59 (24)	C73-69 (24)	C73-73 (24)	0.0035/0.0035	Ran 24 hr OK
	Cup	D73-4 (24)	D73-10 (25)	D73-8 (49.5)	D73-3 (49.5)		
	Cage	73-19 (49.5)	73-59 (24)	73-69 (24)	73-73 (24)		
		(from test 2)	(---------------- from test 3 --------------)				
5	Cone	C73-80 (0)	C73-65 (0)	C73-56 (0)	C73-42 (0)	0.0035/0.0038	73-65 damaged at 11,500 rpm 73-80 and 73-42, light scuff
	Cup	D73-4 (24)	D73-11 (0)	D73-27 (0)	D73-22 (0)		
	Cage	73-80 (0)	73-65 (0)	73-56 (0)	73-42 (0)		
		(-------------------- All cages Canton plated -------------------)					
6	Cone	C73-50 (.75)	C73-60 (.75)	C73-64 (.75)	C73-71 (.75)	0.0035/0.0040	73-60 and 73-64 damaged due to pump shutdown
	Cup	D73-4 (24.75)	D73-11 (.75)	D73-27 (.75)	D72-22 (.75)		
	Cage	73-50 (.75)	73-59 (24.75)	73-69 (24.75)	73-73 (24.75)		45 min at 11,500 rpm. Remaining bearings - very light scuff
		(------------------ All cages Lancaster plated ----------------)					
7	Cone	C73-50 (126.75)	C73-59 (150)	C73-69 (150)	C73-73 (150)	0.0047/0.0042	Test ran 126 hr - bearings OK
	Cup	D73-4 (150.75)	D73-10 (151)	D73-27 (126.75)	D73-22 (126.75)		
	Cage	73-50 (126.75)	73-59A (126)	73-69 (150.75)	73-73 (150.75)		
		(73-50 Regrind rib and roller ends)					

Z-type cage in large sheets. These cages were plated locally at Canton Plating Company, Canton, Ohio.

The sixth test ran for 45 minutes at 11,500 rpm before the lubricating oil pump shut down and caused cone rib/roller end damage on all four bearings, two of them having very slight scuffing damage. Just prior to this test, all of the copper oil lubrication plumbing lines were replaced with steel tubing. The undersized steel tubing restricted the oil flow. To overcome this restriction, the pump pressure was increased to produce the required oil flow. This caused the pump motor to overload and shut down; therefore, larger steel tubing was installed before the next test.

Test number 7 was a 150-hour test. Three of the bearings had previously run 24 hours. The fourth bearing had run 3/4 hour prior to this test; however, the rib face and roller spherical end radii were reground to clean up the very slight scuffing damage from test number 6. Listed in Table VI is the visual observation of bearing components made after the 150-hour test.

The bearings used in test number 7 were tested for an additional 891 hours as part of an added contract (DAAJ01-73-A-0017) to evaluate endurance running on high-speed tapered roller bearings. The results of this program are included in Appendix IX of this report.

Figures 40 through 43 are rectilinear traces of the four cone races used in this test. The profiles of the cone races before test were from flat to a very slight crown. After the 150-hour test, traces were taken again. The cone races were found to be off taper and crowned toward the large end; that is, the uneven transformation of retained austenite due to uneven temperatures along the cone race caused the crown to develop in this manner. This phenomenon was discussed in a previous section.

At the conclusion of the test, the oil inlet temperature was lowered to 175°F and 165°F with oil flow rates of 8 pints per minute to the large end and 4 pints per minute to the small end, and the bearing continued to run until the cup temperatures leveled off. Table VII shows the level-off cup OD, oil outlet, and housing temperatures at various oil inlet temperatures. Appendix VII presents XC11349-series bearings after test.

Analysis of HLH/ATC Transmission Bearings

Until recently, the analysis of tapered roller bearings has been restricted to catalog rating methods or, at best, low-speed computer analysis techniques. In 1971, as part of a USAAMRDL contract, Boeing Vertol subcontracted Mr. A. B. Jones, Bearing Consultant, to develop an analysis for high-speed tapered roller bearings. This analysis provided an exact

TABLE VI. SUMMARY OF BEARING CONDITION AFTER 150-HOUR TEST

Test conditions: Speed - 11,500 rpm
Oil flow rate - 10 pts/min to LE (large end)
6 pts/min to SE (small end)
Inlet oil temperature - 190°F

Bearing Cone/Cup Number	Bearing Position	Hours Run	Cone	Cup	Rollers and Cage
73-50/ 73-4	DE	126.75	5 tiny surface pits	OK	OK
73-59/ 73-10	DEC	150	OK	OK	OK
73-69/ 73-27	ODEC	150	OK	OK	OK
73-73/ 73-22	ODE	150	OK	OK	OK

TABLE VII. REDUCED OIL INLET TEMPERATURE AND OIL FLOW RATE TEST						
Oil Flow Rate (Pts/Min)		Inlet Oil Temp °F	Level Off Temperature BE/DEC/ODEC/ODE			Ambient Temp °F
Large End	Small End		Cup OD	Oil Outlet	Housing	
10	6	190	299	265	265	86
			294	265	260	86
			309	300	275	86
			296	300	265	86
8	4	190	320	277	275	85
			315	277	275	85
			334	318	300	85
			316	318	275	85
8	4	175	304	261	255	85
			302	261	260	85
			318	303	265	85
			302	303	270	85
8	4	165	300	255	250	86
			298	255	255	86
			312	297	265	86
			297	297	255	86

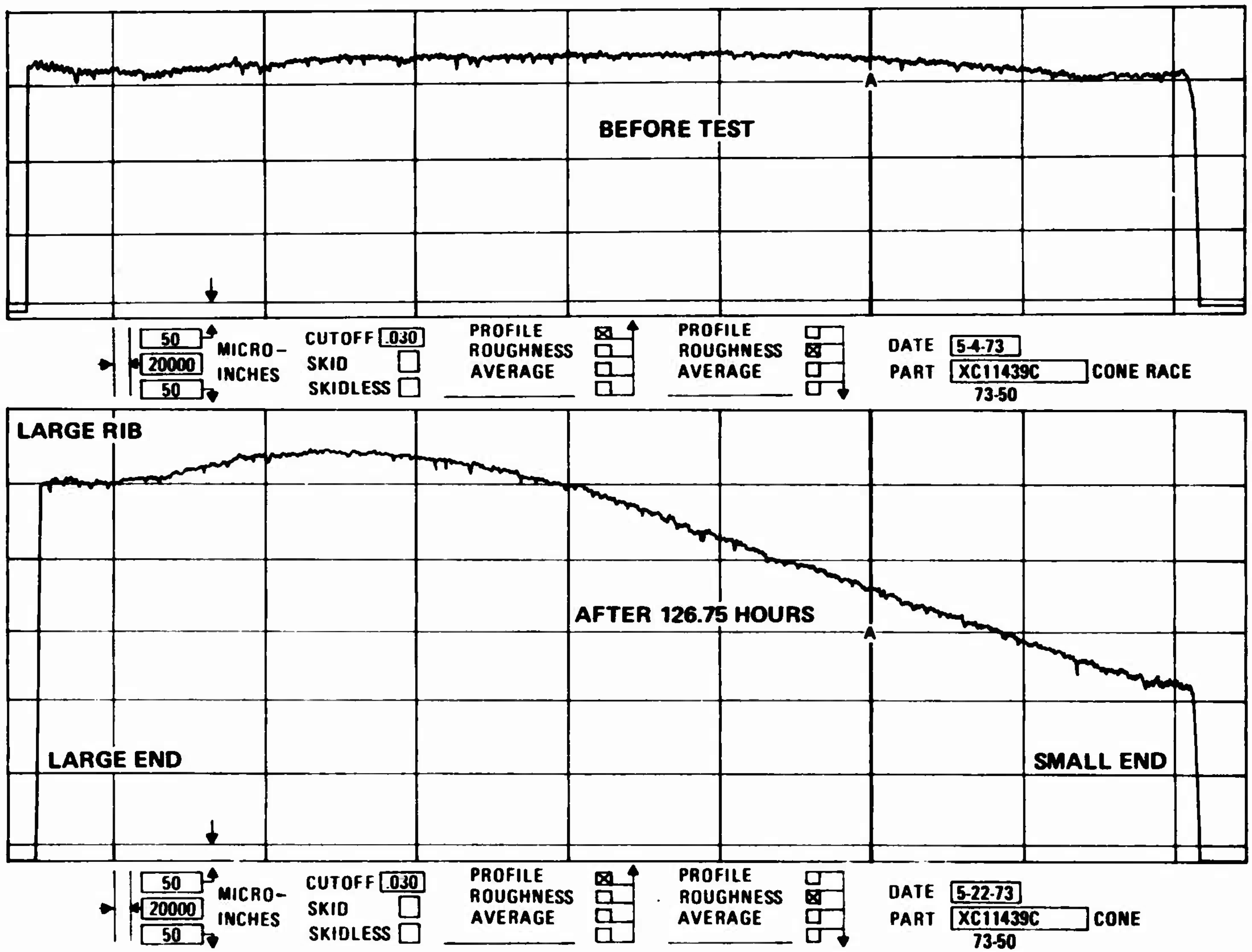


Figure 40. Traces of 301-10876 Cone Race S/N 73-50 Tested in the Drive End Position.

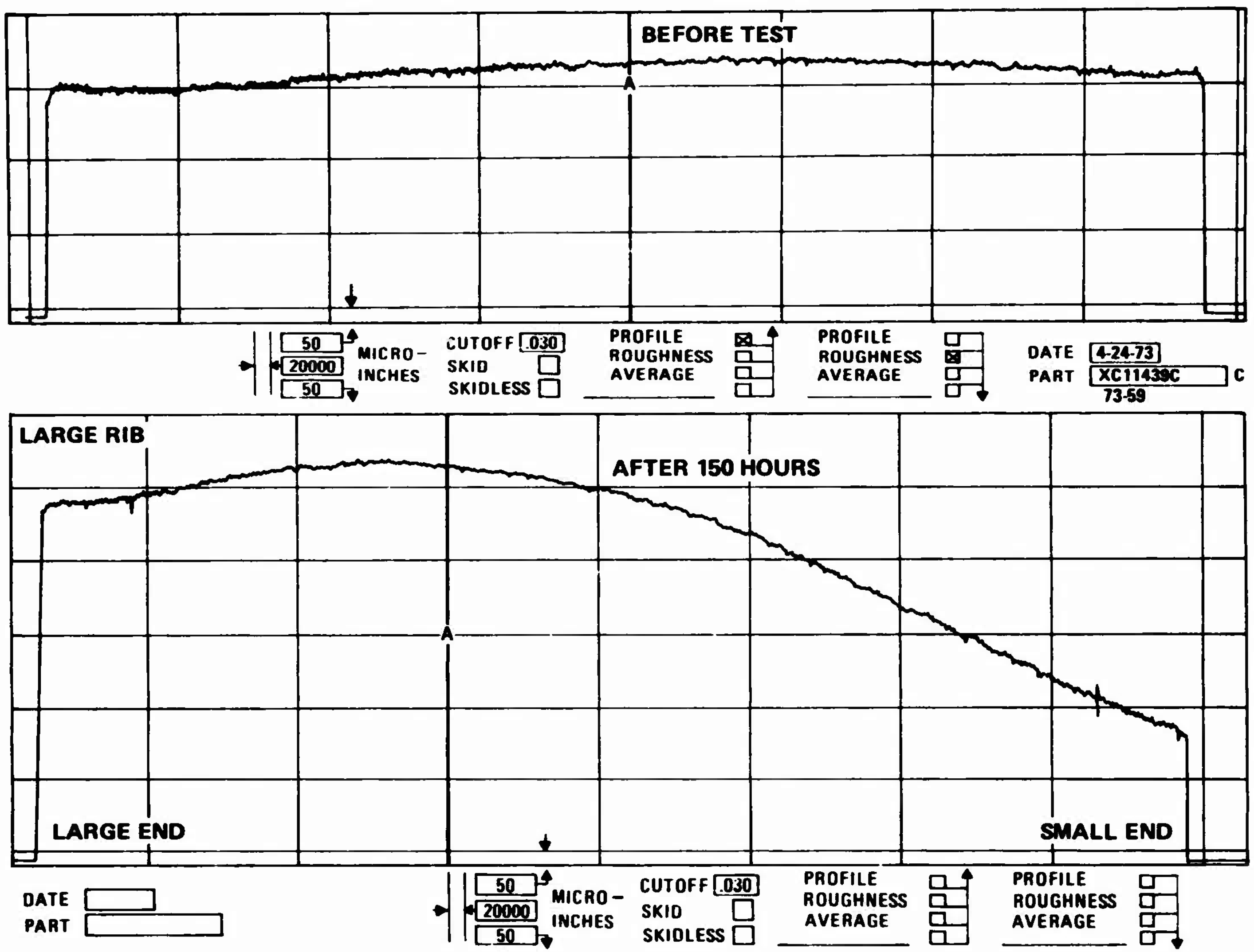


Figure 41. Traces of 301-10876 Cone Race S/N 73-59 Tested in the Drive End Center Position.

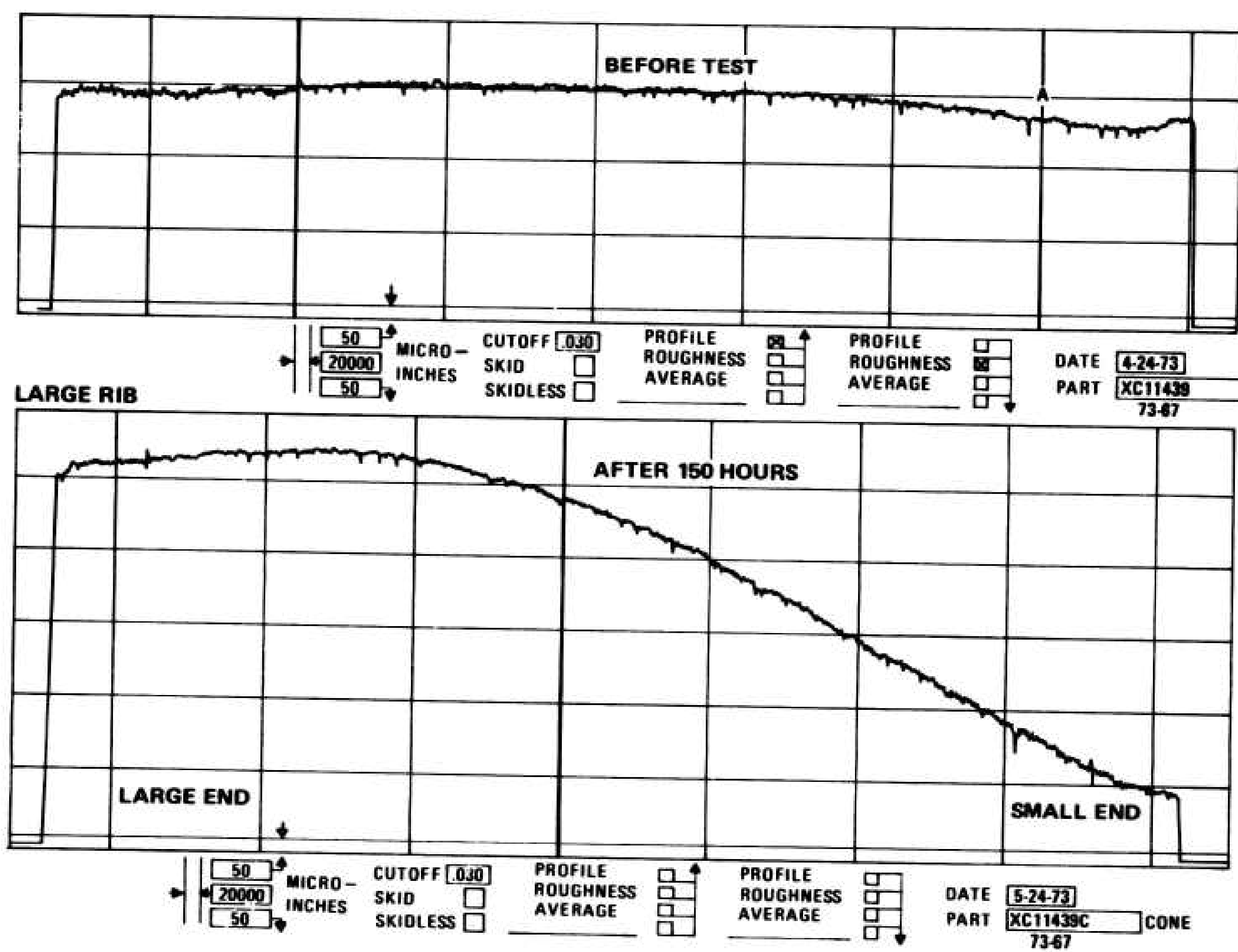


Figure 42. Traces of 301-10676 Cone Race S/N 73-59 Tested in the Opposite Drive Center Position.

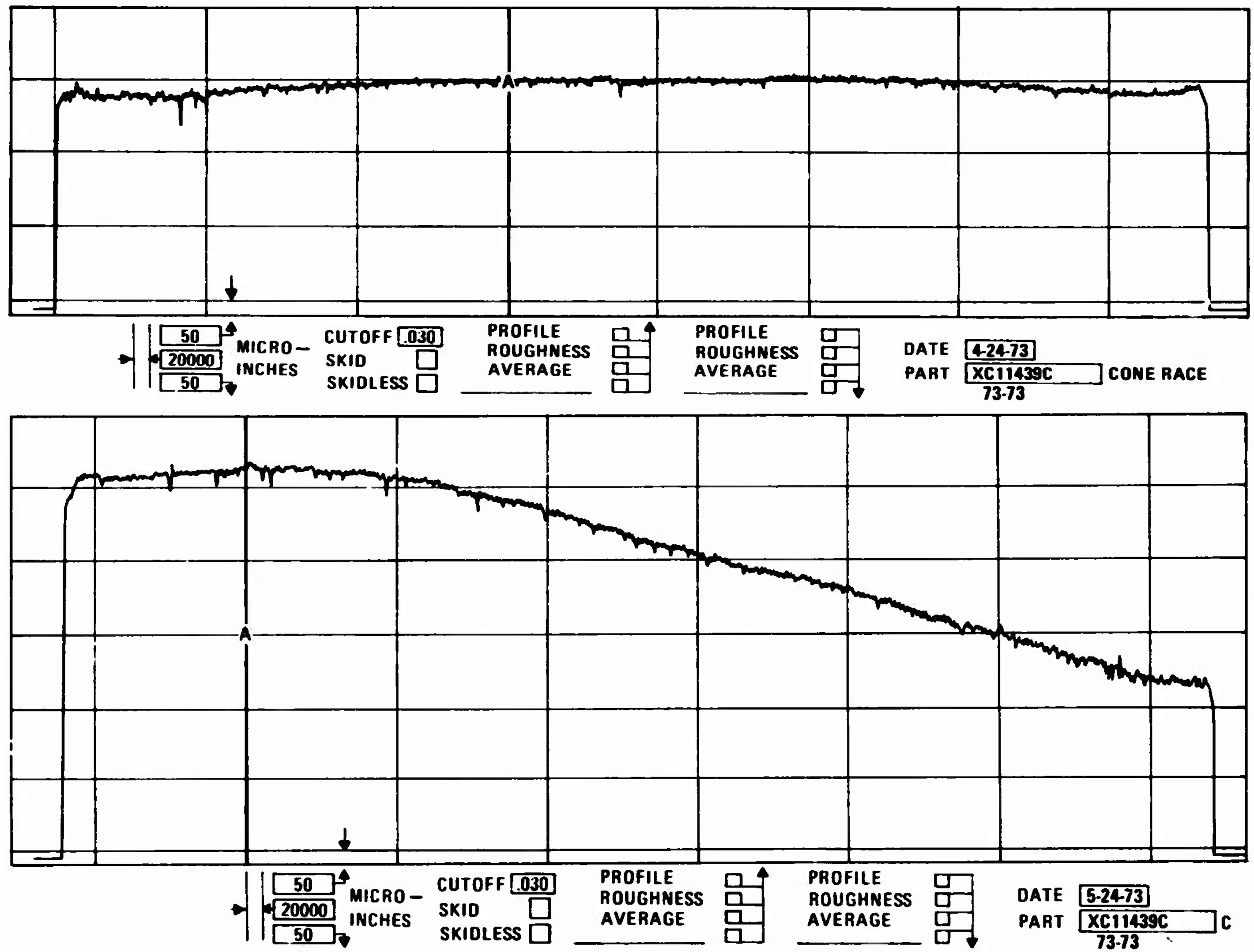


Figure 43. Traces of 301-10676 Cone Race S/N 73-73
Tested in the Opposite Drive End Position.

solution for internal loading, stress, life, deflections, cone rib loading, EHD oil film thickness and traction, flange sliding and spinning velocity, and the high-speed loading interactions between duplex-mounted bearings. Details of this analysis are included in USAAMRDL Technical Report 73-16. This work has been continued by Mr. Jones to develop a computer program, using these analysis techniques which can be used to analyze any configuration high-speed tapered roller bearing. This computer program has been obtained by Boeing Vertol and is designated as Computer Program S73.

Program S73 did not become fully operational until late in this test program and, therefore, was not used in the initial selection of the bearing for the HLH/ATC drive system or test program. The basic design parameters at the various high-speed tapered roller bearings used in both the test program and the HLH/ATC drive system are shown in Table VIII. This table provides a comparison between the bearings tested and those proposed for the HLH/ATC. This data, including gear tooth loading for each bevel gear supported by tapered roller bearings, was used to analyze the bearings. The results of the computer analysis of all bearings are summarized in Table IX. This table itemizes some of the critical operating parameters of the transmission and test bearings. All bearings are within the limits tested on the HM926700-series bearing and only one bearing, P/N 301-10420, has exceeded the test conditions of the 301-10676 test bearings.

A review of the test data from this program and previous test programs indicates that cone rib stress may be a critical factor that influences the probability of rib scuffing of high-speed tapered roller bearings. Bearings which have been operated at 20,000 fpm and rib stresses of 36,000 psi or less have experienced very few rib scuffing failures; however, bearings (HM926700) operating at the same speed, but with rib stress of approximatley 66,000 psi, have encountered severe rib scuffing. Although there appear to be other factors in addition to stress which determine that rib scuffing occurs at high speed, these other factors have not been fully established in this program. Rib stress will, therefore, be used as a guide to establish the potential risk of rib scuffing. The one bearing that appears to have a high risk of scuffing due to high stress is bearing P/N 301-10420. This bearing operates at only 17,000 fpm, which should help to reduce the risk of scuffing.

Since rib stress is a function of bearing loading and contact angle, the 301-10420 bearing could be redesigned or replaced with a lower contact angle bearing in order to reduce the rib scuffing if this becomes a serious problem in the testing of the HLH/ATC aft transmission. Present plans are to test the 301-10420 bearings in the HLH/ATC aft transmission to determine

TABLE VIII. BASIC DESIGN PARAMETERS OF HLH/ATC HIGH-SPEED TAPERED ROLLER TRANSMISSION BEARINGS SUMMARY

	Location	Part No. 301-	Shaft Speed (rpm)	Cone Rib Velocity (fpm)	Number of Roller	Roller Large End Diameter (in.)	Roller Length (in.)	Contact Angle (deg)	Pitch Diameter (in.)	Number of Lube Hole	Pct Flange Coverage
	HM926700 Test Bearing	--	11,500	20,472	16	1.062	1.500	26°9'	7.000	40	33.98
			14,000	24,923						34	25.51
Combiner transmission	Spiral bevel input pinion	10676	11,500	20,473	24	0.817	1.643	15°	7.067	40	31.43
		10671		19,293	24	0.740	1.215	12°30'	6.752	32	27.03
	Slant shaft	10612	7986	18,225	21	1.197	2.067	10°	9.350	28	27.44
		10616		17,951	33	0.764	1.256	22°30'	8.720	44	27.30
	Clutch shaft	10644	0 To 11,500	15,656	42	0.300	0.600	15°24'	5.260	36	20.57
		10651		10,116	34	0.250	0.540	17°5'	3.375	28	19.86
Aft transmission	Spiral bevel input pinion	10420	7986	16,966	21	1.175	2.052	25°	8.266	36	28.90
		10424		15,091	25	0.837	1.390	20°	7.441	28	21.14
	Sun gear shaft	10443	2788	9616	48	0.757	1.006	15°	13.504	32	18.51
		10440		9373	71	0.493	0.730	15°	13.032	44	17.73

TABLE IX. SUMMARY OF HLH/ATC HIGH-SPEED TAPERED ROLLER BEARING ANALYSIS, USING COMPUTER PROGRAM

Transmission	Location	Part No. 301-	Load Condition	Speed (rpm)	B-10 Life (No Material Factor) (1) (Hours)	Maximum Roller Load (lb)	Contact Stress (psi)	Maximum Rib Load (lb)	Rib Stress (psi)	Rib Film Thickness (in.)	Roller CF Force (lb)	Gyroscopic Moment (in.-lb)	Flange Torque (in.-lb)	Race Torque (in.-lb)	Depth of Maximum Shear (in.)	Inner Race Film Thick. (in.)
Combiner Transmission	SB engine input pinion	10676	100 pct	11,500	98	3448	154,111	328	33,504	49.0	577	29	8.9	94.6	0.00697	20.4
		10671			2793	841	98,373	111	19,671	55.5	350	13.5	2.1	36.4	0.00412	22.0
		10676	cubic mean	11,500	728	1940	121,200	250	27,748	50.7	577	29	4.3	57.4	0.00548	21.9
		10671			7244	523	80,984	97.7	18,855	56.1	350	13.5	16.2	35.9	0.00339	23.4
	Slant shaft	10612	100 pct	7986	416	4849	137,180	368	20,425	62.0	1032	52.7	10.0	178.7	0.00920	22.6
		10616			3607	933	99,419	151	26,986	43.5	249	17.3	4.2	44.8	0.00438	20.8
		10612	cubic mean	7986	1066	3694	122,720	323	19,565	62.6	1032	52.7	7.0	139.1	0.00820	23.4
		10616			11,726	594	82,788	131	25,727	43.9	249	17.3	3.2	38.0	0.00364	22.0
	Clutch shaft	10644	400 lb preload	11,500	96,515	19.7	43,410	7.4	11,061	43.8	25.8	0.5	3.0	3.5	0.00079	19.1
		10651			29,254	40.7	62,437	4.5	12,742	25.2	9.6	0.2	1.3	2.5	0.00091	11.2
Aft Transmission	Input pinion	10420	100 pct	7986	179	5870	156,100	904	52,150	36.9	761	87	44.6	150	0.00969	19.3
		10424			1901	1224	103,780	172	28,959	38.8	263	17.8	34.8	79.3	0.00483	18.7
		10420	cubic mean	7986	509	4383	138,000	757	49,142	37.4	761	8.7	30.6	115	0.00857	20.0
		10424			3249	1006	95,767	157	28,125	39.0	263	17.8	33.5	71.8	0.00446	19.2
	Sun gear shaft	10443	100 pct	2788	208	2105	146,950	68	12,224	38.7	41.2	2.0	16.6	178	0.00672	12.0
		10440			387	523	110,600	13	7352	43.2	13.0	0.4	26.1	206.6	0.00364	11.4
		10443	cubic mean	2788	711	1541	126,900	52	11,225	39.4	41.2	2.0	12.2	131	0.00580	12.5
		10440			1180	383	96,935	10.5	6829	43.9	13.0	0.4	21.8	152	0.00295	11.8
HM926700 Test Conditions		HM926700	100 pct	11,500	83	5069	173,050	915	66,235	40.0	809	88	35.7	90.3	0.00988	20.8
		HM926700			4195	1207	97,329	491	53,819	41.9	809	88	71.8	55.5	0.00555	25.1
301-10676-1 Test Conditions		301-10676	100 pct	11,500	131	2803	146,620	295	32,773	49.4	577	29	9.1	[illegible].4	0.00671	20.9
		301-10676			1536	1488	116,370	227	29,100	50.4	577	29	2.3	40.5	0.00535	22.7

Notes: (1) A material factor of 4 is recommended for CEUM carburized steel used in tapered roller bearings.

if the operating speed and rib stress of this bearing are critical for cone rib scuffing. If scuffing occurs, the bearing can be easily replaced with an H432600-series tapered roller bearing (14^{o} contact angle) which will reduce the rib stress to less than 32,000 psi with only a slight reduction in fatigue life.

CONCLUSIONS

The data derived from this program show that modified high-speed, steep-angle, HM926700-series bearings and shallow-angle, XC11439-series bearings can be successfully run at 14,000 rpm and 11,500 rpm, respectively, under a wide range of loading and can also withstand relatively high acceleration rates. Certain design, adjustment, and metallurgical considerations were found to be necessary and are covered in the detailed conclusions which follow.

Current technology and materials have not been developed to a point where it is possible to make a fail-safe (30-minute operation without oil) bearing for operation at these speeds (20,000 fpm tangential cone rib velocity). Length of operation without oil varies exponentially with speed dropping off sharply from 30 minutes above 6000 fpm cone rib velocity.

An alternate source of lubricating oil for fail-safe operation is the best approach to this problem. Previous work has shown that the tapered roller bearing can operate with small quantities of oil, thus making an alternate source practical.

DESIGN AND BUILD A HIGH-SPEED TAPERED ROLLER BEARING TEST MACHINE

A high-speed tapered roller bearing test machine was designed and built for this program to specifically test the HM926700-series bearing (9-inch OD). In the latter portion of the test program, the machine was modified to test the 8-5/8-inch-OD XC11439-series bearing (Boeing Vertol P/N 301-10676). This machine is also capable of testing up to a 20-inch-OD bearing with the proper modifications. Some minor machine shortcomings were discovered and corrected during testing. Basically, this machine performed satisfactorily throughout this test program.

DETERMINATION OF NUMBER AND SIZE OF RADIAL LUBRICATION OIL HOLES IN THE HM926700-SERIES BEARING

Forty 0.040-inch-diameter radial lubrication oil holes were needed to adequately lubricate the roller end/cone rib contact. The size of the holes could be substantially smaller than 0.040-inch diameter; however, the size was set at a minimum of 0.040-inch to prevent foreign matter from clogging the holes and restricting oil flow.

DEVELOPMENT TESTS - MODIFIED HM926700-SERIES BEARINGS

The modified HM926700-series bearing can be operated at 14,000 rpm (25,000 fpm) under 15,619 pounds radial load and 4394 pounds thrust load and 195°F oil inlet temperature. The bearing cup OD temperatures, when tested under these conditions, ranged from 325°F to 335°F. Some of the bearings had spalls on the cone races after the test as a result of geometry distortion due to the temperatures exceeding 300°F.

The modified HM926700-series bearings can be accelerated from 0 to 10,000 rpm in less than 30 seconds and from 10,000 to 11,500 rpm in less than 5 seconds under 10-percent load (1230 pounds radial and 440 pounds thrust cylinder load applied to housings) from a room temperature oil or preheated oil start. After running at 11,500 rpm under 100-percent load (12,300 pounds radial and 4394 pounds thrust load), the bearings can be accelerated to 14,000 rpm in less than 5 seconds under 75-percent thrust loads.

EVALUATION OF FAIL-SAFE WEAR-RESISTANT BEARING MATERIALS

Tests in the Timken lubricant and wear test machine showed that Borkote materials show the least amount of wear during the oil-off tests. Tests also showed that the LP alloys gave the best overall life with a reasonable amount of wear. The LPA 404 alloy with 40-percent 4600-steel binder was the best of the LP alloys tested.

EVALUATION OF TWO FAIL-SAFE BEARING DESIGNS

Bearings with LPA 404 with 40-percent 4600-steel binder applied to the cone large end rib performed better than Borkote applied to the rib under fully lubricated conditions. The bearings with LP alloy applied to the rib were run for 24 hours with no abnormal wear. At the end of 24 hours, the oil was shut off, and the bearings continued to run for 8½ seconds. The goal was 30 minutes operation without lubricating oil.

REPEAT FAIL-SAFE TEST

A repeat fail-safe test was run with four bearings with LP alloy that had been previously tested. This test was run for 3 hours before the oil was shut off. The bearings continued to rotate for 10 seconds. A suitable wear-resistant fail-safe material and a method of attaching this material to the bearing has not been reached with our present technology. An alternate source of lubricating oil should be considered as a fail-safe lubrication system.

ADDITIONAL HM926700-SERIES BEARING TESTS

These tests showed the best bearing adjustment to be between 0.001-inch and 0.0035-inch end-play. Bearing damage occurred when the end-play was reduced to less than 0.001-inch and increased above 0.0035-inch.

It was discovered that there is a difference in bearings from manufacturing lots that would not permit the bearings to operate at high speeds. The scheduled oil flow tests could not, therefore, be run.

The only significant difference from bearing sets appeared to be the level of retained austenite. Higher levels of retained austenite were less resistant to scuffing damage between the roller end and cone rib at the higher stress levels in steep-angle bearings. More work is needed to verify this conclusion, since the data obtained in these tests is very limited.

TAPERED ROLLER BEARING CONE FLANGE OIL DISTRIBUTION SPIN TEST

Results show that there is good correlation between the high-speed strobe light photographs showing the oil distribution from the radial oil lubrication holes and the computer program used to determine the percent of adequate lubrication at the cone rib interface. The computer printout is shown in Appendix I.

TESTING OF XC11439-SERIES BEARINGS

Testing of the XC11439-series bearing was successful. Seventeen bearings in eight test setups were tested during this phase of the program. The duration of testing ranged from 1 hour to 126 hours. Three of the bearings had accumulated 150 hours.

As was expected, the bearing end-play adjustment had to be increased slightly with this shallow-angle bearing over that for the steep-angle, HM926700-series bearing. Successful tests were run with bench bearing end-play adjustments ranging from 0.0035 inch to 0.005 inch. A bench bearing end-play adjustment of 0.0025 inch resulted in premature bearing damage in one of the tests.

A mathematical approach to predicting start-up bearing adjustment (running approximatley 5 minutes) and level-off condition bearing adjustment (running several hours) has been developed. It shows that under the successful bench end-play bearing adjustments that were tested, the bearings were actually slightly preloaded under level-off test conditions.

Based upon the data generated, the design HLH/ATC high-speed tapered roller bearing used in the aft and combiner transmission has finalized. A total of 10 different size tapered roller bearings will be tested by Boeing Vertol in both bench transmission and DSTR test stands prior to installation into the prototype HLH .

RECOMMENDATIONS

Based on the results and conclusions of this test program, it is recommended that additional testing be conducted on a large sample of bearings for an extended duration to determine fatigue life characteristics of high-speed tapered roller bearings versus calculated fatigue life. In addition, tests should be conducted on bearings fabricated from high-hot-hardness carburizing steels such as CBS-600 or CBS-1000M to determine improved cone rib scuffing resistance and to minimize raceway distortion due to high operating temperatures experienced under maximum operating conditions.

APPENDIX I
CONERIB LUBRICATION COMPUTER PRINTOUT

```
PHL  ENGRG PHL021-01          CONE RIB LUBRICATION             BF1       09/24/73
                                                               PAGE   1
     CONE NO. OIL DIS-1             CUP NO. 7.3" Bore Cone
                                       Cone Drawing Figure 8
     INPUT DATA

          SHAFT ID (IN)=    3.3000             SHAFT OD (IN)=    7.2850

          CONE INLET DIA. (IN)=  7.4930      CONE OUTLET DIA. (IN)=    8.4590

          CONE RIB-ROLLER END CONTACT DIA. (IN)=    8.7500

          CONE RIB OD (IN)=    9.1250

          CONE MANIFOLD TRAP LENGTH (IN)=    0.2910    MAX DIA=   7.72600

          CONE LUBRICATION DUCT ANGLE (D-M-S)=  15- 0- 0

          LUBRICANT VISCOSITY (CS) @ 100 F.=  32.00000
                                   @ 210 F.=   4.88000
                                   @ 165 F.=   8.89789

          SPECIFIC GRAVITY OF LUBRICANT @ 60 F.=  1.007199

          COEFFICIENT OF EXPANSION =    0.00041

          SHAFT LUBRICATION DUCTS
                            NO.=    2     ID (IN)=     0.1875    ID SURFACE-FIN. (IN)=    0.00014

          CONE LUBRICATION DUCTS
                            NO.=   24     ID (IN)=     0.0420    ID SURFACE-FIN. (IN)=    0.00014

          SHAFT SPEED (RPM)=  3000             INCREMENTS =    1
                              6000
                              8000
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PHL  ENGRG PHL021-01          CONE RIB LUBRICATION             BF1       09/24/73
                                                               PAGE   2
     CONE NO. OIL DIS-1             CUP NO. 7.3" Bore Cone

     CALCULATIONS

          SHAFT SPEED (RPM)= 3000.             RIB VELOCITY (FT/MIN)=    6872.

          DN =  555117.

          LUBRICANT FLOW CONDITIONS            SHAFT DUCTS          CONE DUCTS
               FRICTION FACTOR                   0.0348               0.0794
               HEADLOSS (IN/SEC)**2              49228.               34856.
               REYNOLDS NUMBER                    6760.                 848.
               EXIT VELOCITY (IN/SEC)          994.4626             557.2297

          MAXIMUM MANIFOLD PRESSURE (PSI)=    3.93

          TANGENTIAL VELOCITY OF LUBRICANT @ OUTLET (IN/SEC)=  1328.74

          ANGLE OF EXIT RELATIVE TO OUTLET (DEGREES)= 22.092

          EFFECTIVE EXPOSURE TIME (SEC)= 0.000247

          % COVERAGE OF RIB = 28.72

                           FLOW (PT/MIN)
          CONE               38.50
          MANIFOLD           78.75
          SHAFT             114.11
```

PHL ENGRG PHL021-01 CONE RIB LUBRICATION 8F1 09/24/73
PAGE 2

CONE NO. OIL DIS-1 CUP NO. 7.3" Bore Cone

CALCULATIONS

SHAFT SPEED (RPM)= 6000. RIB VELOCITY (FT/MIN)= 13744.

DN = 1110233.

LUBRICANT FLOW CONDITIONS	SHAFT DUCTS	CONE DUCTS
FRICTION FACTOR	0.0293	0.0399
HEADLOSS (IN/SEC)**2	167718.	73320.
REYNOLDS NUMBER	13619.	1785.
EXIT VELOCITY (IN/SEC)	2003.5498	1172.2673

MAXIMUM MANIFOLD PRESSURE (PSI)= 15.72

TANGENTIAL VELOCITY OF LUBRICANT @ OUTLET (IN/SEC)= 2657.47

ANGLE OF EXIT RELATIVE TO OUTLET (DEGREES)= 23.078

EFFECTIVE EXPOSURE TIME (SEC)= 0.000118

% COVERAGE OF RIB = 27.51

	FLOW (PT/MIN)
CONE	80.99
MANIFOLD	158.08
SHAFT	229.91

PHL ENGRG PHL021-01 CONE RIB LUBRICATION 8F2 09/24/73
PAGE 2

CONE NO. OIL DIS-1 CUP NO. 7.3" Bore Cone

CALCULATIONS

SHAFT SPEED (RPM)= 8000. RIB VELOCITY (FT/MIN)= 18326.

DN = 1480311.

LUBRICANT FLOW CONDITIONS	SHAFT DUCTS	CONE DUCTS
FRICTION FACTOR	0.0272	0.0454
HEADLOSS (IN/SEC)**2	278791.	160888.
REYNOLDS NUMBER	18208.	2350.
EXIT VELOCITY (IN/SEC)	2678.6440	1543.3704

MAXIMUM MANIFOLD PRESSURE (PSI)= 27.96

TANGENTIAL VELOCITY OF LUBRICANT @ OUTLET (IN/SEC)= 3543.30

ANGLE OF EXIT RELATIVE TO OUTLET (DEGREES)= 22.818

EFFECTIVE EXPOSURE TIME (SEC)= 0.000090

% COVERAGE OF RIB = 27.01

	FLOW (PT/MIN)
CONE	106.63
MANIFOLD	211.06
SHAFT	307.37

PHL ENGRG PHL021-01 CONE RIB LUBRICATION BF2 09/24/73
PAGE 1

CONE NO. OIL DIS-4 CUP NO. 7.3" Bore Cone

Cone Drawing Figure 47

INPUT DATA

SHAFT ID (IN)= 3.0000 SHAFT OD (IN)= 7.2850

CONE INLET DIA. (IN)= 7.7340 CONE OUTLET DIA. (IN)= 8.4590

CONE RIB-ROLLER END CONTACT DIA. (IN)= 8.7500

CONE RIB OD (IN)= 9.1250

CONE MANIFOLD TRAP LENGTH (IN)= 0.2910 MAX DIA= 7.90000

CONE LUBRICATION DUCT ANGLE (D-M-S)= 15- 0- 0

LUBRICANT VISCOSITY (CS) @ 100 F.= 32.00000
@ 210 F.= 4.88000
@ 165 F.= 8.89789

SPECIFIC GRAVITY OF LUBRICANT @ 60 F.= 1.007199

COEFFICIENT OF EXPANSION = 0.00041

SHAFT LUBRICATION DUCTS
NO.= 2 ID (IN)= 0.1875 ID SURFACE-FIN. (IN)= 0.00014

CONE LUBRICATION DUCTS
NO.= 24 ID (IN)= 0.0420 ID SURFACE-FIN. (IN)= 0.00014

SHAFT SPEED (RPM)= 3000 INCREMENTS = 1
6000
8000

PHL ENGRG PHL021-01 CONE RIB LUBRICATION BF2 09/24/73
PAGE 2

CONE NO. OIL DIS-4 CUP NO. 7.3" Bore Cone

CALCULATIONS

SHAFT SPEED (RPM)= 3000. RIB VELOCITY (FT/MIN)= 6672.

DN = 555117.

LUBRICANT FLOW CONDITIONS	SHAFT DUCTS	CONE DUCTS
FRICTION FACTOR	0.0348	0.0092
HEADLOSS (IN/SEC)**2	49220.	23159.
REYNOLDS NUMBER	6760.	751.
EXIT VELOCITY (IN/SEC)	994.4626	493.3376

MAXIMUM MANIFOLD PRESSURE (PSI)= 2.08

TANGENTIAL VELOCITY OF LUBRICANT @ OUTLET (IN/SEC)= 1320.74

ANGLE OF EXIT RELATIVE TO OUTLET (DEGREES)= 19.720

EFFECTIVE EXPOSURE TIME (SEC)= 0.000274

% COVERAGE OF RIB = 31.78

	FLOW (PT/MIN)
CONE	34.08
MANIFOLD	56.10
SHAFT	114.11

PML ENGRG PML021-01 CONE RIB LUBRICATION BF3 PAGE 2 09/24/73

CONE NO. OIL DIS-4 CUP NO. 7.3" Bore Cone

CALCULATIONS

SHAFT SPEED (RPM)= 6000. RIB VELOCITY (FT/MIN)= 13744.

DN = 1110233.

LUBRICANT FLOW CONDITIONS	SHAFT DUCTS	CONE DUCTS
FRICTION FACTOR	0.0293	0.0408
HEADLOSS (IN/SEC)**2	167718.	48378.
REYNOLDS NUMBER	13619.	1569.
EXIT VELOCITY (IN/SEC)	2003.5498	1030.4978

MAXIMUM MANIFOLD PRESSURE (PSI)= 11.51

TANGENTIAL VELOCITY OF LUBRICANT @ OUTLET (IN/SEC)= 2657.47

ANGLE OF EXIT RELATIVE TO OUTLET (DEGREES)= 20.534

EFFECTIVE EXPOSURE TIME (SEC)= 0.000132

% COVERAGE OF RIB = 30.66

	FLOW (FT/MIN)
CONE	71.20
MANIFOLD	112.62
SHAFT	229.91

PML ENGRG PML021-01 CONE RIB LUBRICATION BF4 PAGE 2 09/24/73

CONE NO. OIL DIS-4 CUP NO. 7.3" Bore Cone

CALCULATIONS

SHAFT SPEED (RPM)= 8000. RIB VELOCITY (FT/MIN)= 18326.

DN = 1480311.

LUBRICANT FLOW CONDITIONS	SHAFT DUCTS	CONE DUCTS
FRICTION FACTOR	0.0272	0.0303
HEADLOSS (IN/SEC)**2	278791.	65211.
REYNOLDS NUMBER	18208.	2115.
EXIT VELOCITY (IN/SEC)	2678.6440	1389.0496

MAXIMUM MANIFOLD PRESSURE (PSI)= 20.46

TANGENTIAL VELOCITY OF LUBRICANT @ OUTLET (IN/SEC)= 3543.30

ANGLE OF EXIT RELATIVE TO OUTLET (DEGREES)= 20.740

EFFECTIVE EXPOSURE TIME (SEC)= 0.000098

% COVERAGE OF RIB = 30.39

	FLOW (FT/MIN)
CONE	95.97
MANIFOLD	150.37
SHAFT	307.37

PML ENGRG PML021-01 CONE RIB LUBRICATION BF3 09/24/73
PAGE 1

CONE NO. OIL DIS-2 CUP NO. 12.1" Bore Cone

Cone Drawing Figure 29 (C)

INPUT DATA

SHAFT ID (IN)= 10.0000 SHAFT OD (IN)= 12.1270

CONE INLET DIA. (IN)= 12.3960 CONE OUTLET DIA. (IN)= 13.0000

CONE RIB-ROLLER END CONTACT DIA. (IN)= 13.2015

CONE RIB OD (IN)= 13.6250

CONE MANIFOLD TRAP LENGTH (IN)= 0.1625 MAX DIA= 12.99600

CONE LUBRICATION DUCT ANGLE (D-M-S)= 15- 0- 0

LUBRICANT VISCOSITY (CS) @ 100 F.= 32.00000
@ 210 F.= 4.00000
@ 165 F.= 8.09789

SPECIFIC GRAVITY OF LUBRICANT @ 60 F.= 1.007199

COEFFICIENT OF EXPANSION = 0.00041

SHAFT LUBRICATION DUCTS
NO.= 2 ID (IN)= 0.1875 ID SURFACE-FIN. (IN)= 0.00014

CONE LUBRICATION DUCTS
NO.= 24 ID (IN)= 0.0420 ID SURFACE-FIN. (IN)= 0.00014

SHAFT SPEED (RPM)= 3000 INCREMENTS = 3
3000
1000

PML ENGRG PML021-01 CONE RIB LUBRICATION BF3 09/24/73
PAGE 2

CONE NO. OIL DIS-2 CUP NO. 12.1" Bore Cone

CALCULATIONS

SHAFT SPEED (RPM)= 1000. RIB VELOCITY (FT/MIN)= 3456.

DN = 300026.

LUBRICANT FLOW CONDITIONS	SHAFT DUCTS	CONE DUCTS
FRICTION FACTOR	0.0453	0.2470
HEADLOSS (IN/SEC)**2	3094.	6635.
REYNOLDS NUMBER	2367.	258.
EXIT VELOCITY (IN/SEC)	340.1993	249.6539

MAXIMUM MANIFOLD PRESSURE (PSI)= 0.62

TANGENTIAL VELOCITY OF LUBRICANT @ OUTLET (IN/SEC)= 680.68

ANGLE OF EXIT RELATIVE TO OUTLET (DEGREES)= 13.536

EFFECTIVE EXPOSURE TIME (SEC)= 0.000550

% COVERAGE OF RIB = 21.67

	FLOW (PT/MIN)
CONE	11.72
MANIFOLD	12.00
SHAFT	39.96

PHL ENGRG PHL021-01 CONE RIB LUBRICATION BF3 09/24/73
PAGE 3

CONE NO. OIL DIS-2 CUP NO. 12.1" [illegible]

CALCULATIONS

SHAFT SPEED (RPM)= 2000. RIB VELOCITY (FT/MIN)= 6912.

DN = 616051.

LUBRICANT FLOW CONDITIONS	SHAFT DUCTS	CONE DUCTS
FRICTION FACTOR	0.0381	0.1127
HEADLOSS (IN/SEC)**2	13211.	14504.
REYNOLDS NUMBER	4757.	560.
EXIT VELOCITY (IN/SEC)	699.7813	372.8867

MAXIMUM MANIFOLD PRESSURE (PSI)= 2.46

TANGENTIAL VELOCITY OF LUBRICANT @ OUTLET (IN/SEC)= 1361.36

ANGLE OF EXIT RELATIVE TO OUTLET (DEGREES)= 14.819

EFFECTIVE EXPOSURE TIME (SEC)= 0.000254

% COVERAGE OF RIB = 20.06

	FLOW (FT/MIN)
CONE	25.76
MANIFOLD	25.05
SHAFT	80.30

PHL ENGRG PHL021-01 CONE RIB LUBRICATION BF3 09/24/73
PAGE 4

CONE NO. OIL DIS-2 CUP NO. 12.1" [illegible]

CALCULATIONS

SHAFT SPEED (RPM)= 3000. RIB VELOCITY (FT/MIN)= 10368.

DN = 924077.

LUBRICANT FLOW CONDITIONS	SHAFT DUCTS	CONE DUCTS
FRICTION FACTOR	0.0344	0.0720
HEADLOSS (IN/SEC)**2	26976.	22580.
REYNOLDS NUMBER	7193.	879.
EXIT VELOCITY (IN/SEC)	1052.2054	577.3369

MAXIMUM MANIFOLD PRESSURE (PSI)= 5.54

TANGENTIAL VELOCITY OF LUBRICANT @ OUTLET (IN/SEC)= 2042.03

ANGLE OF EXIT RELATIVE TO OUTLET (DEGREES)= 15.275

EFFECTIVE EXPOSURE TIME (SEC)= 0.000165

% COVERAGE OF RIB = 19.53

	FLOW (FT/MIN)
CONE	39.89
MANIFOLD	38.83
SHAFT	120.75

PHL ENGRG L021-01 CONE RIB LUBRICATION BF4 09/29/73
PAGE 2

CONE NO. OIL DIS-5 CUP NO. 12.1" [illegible]

CALCULATIONS

SHAFT SPEED (RPM)= 1000. RIB VELOCITY (FT/MIN)= 3456.

DN = 308026.

LUBRICANT FLOW CONDITIONS	SHAFT DUCTS	CONE DUCTS
FRICTION FACTOR	0.0453	0.2773
HEADLOSS (IN/SEC)**2	3894.	4476.
REYNOLDS NUMBER	2367.	231.
EXIT VELOCITY (IN/SEC)	340.1953	151.5958

MAXIMUM MANIFOLD PRESSURE (PSI)= 0.48

TANGENTIAL VELOCITY OF LUBRICANT @ OUTLET (IN/SEC)= 680.68

ANGLE OF EXIT RELATIVE TO OUTLET (DEGREES)= 12.141

EFFECTIVE EXPOSURE TIME (SEC)= 0.000601

% COVERAGE OF RIB = 23.70

	FLOW (FT/MIN)
CONE	10.47
MANIFOLD	10.75
SHAFT	39.96

PHL ENGRG L021-01 CONE RIB LUBRICATION BF4 09/29/73
PAGE 1

CONE NO. OIL DIS-5 CUP NO. 12.1" [illegible]

[illegible]

INPUT DATA

SHAFT ID (IN)= 10.0000 SHAFT OD (IN)= 12.1270

CONE INLET DIA. (IN)= 12.9440 CONE OUTLET DIA. (IN)= 13.0000

CONE RIB-ROLLER END CONTACT DIA. (IN)= 13.2015

CONE RIB OD (IN)= 13.6250

CONE MANIFOLD TRAP LENGTH (IN)= 0.1750 MAX DIA= 12.69900

CONE LUBRICATION DUCT ANGLE (D-M-S)= 15- 0- 0

LUBRICANT VISCOSITY (CS) @ 100 F.= 32.00000
@ 210 F.= 4.80000
@ 165 F.= 8.89789

SPECIFIC GRAVITY OF LUBRICANT @ 60 F.= 1.007199

COEFFICIENT OF EXPANSION = 0.00041

SHAFT LUBRICATION DUCTS
NO.= 2 ID (IN)= 0.1875 ID SURFACE-FIN. (IN)= 0.00014

CONE LUBRICATION DUCTS
NO.= 24 ID (IN)= 0.0420 ID SURFACE-FIN. (IN)= 0.00014

SHAFT SPEED (RPM)= 3000 INCREMENTS = 3
2000
1000

PHL ENGRG L021-01 CONE RIB LUBRICATION BF4 09/29/73
PAGE 3

CONE NO. OIL DIS-5 CUP NO. [illegible]

CALCULATIONS

SHAFT SPEED (RPM)= 2300. RIB VELOCITY (FT/MIN)= 6912.

DN = 616051.

LUBRICANT FLOW CONDITIONS	SHAFT DUCTS	CONE DUCTS
FRICTION FACTOR	0.0301	0.1277
HEADLOSS (IN/SEC)**2	13211.	9717.
REYNOLDS NUMBER	4757.	501.
EXIT VELOCITY (IN/SEC)	699.7013	329.0906

MAXIMUM MANIFOLD PRESSURE (PSI)= 1.93

TANGENTIAL VELOCITY OF LUBRICANT @ OUTLET (IN/SEC)= 1361.36

ANGLE OF EXIT RELATIVE TO OUTLET (DEGREES)= 13.143

EFFECTIVE EXPOSURE TIME (SEC)= 0.000282

% COVERAGE OF RIB = 22.21

	FLOW (FT/MIN)
CONE	22.74
MANIFOLD	21.58
SHAFT	80.30

PHL ENGRG L021-01 CONE RIB LUBRICATION BF4 09/20/73
PAGE 4

CONE NO. OIL DIS-5 CUP NO. 12.1" [illegible]

CALCULATIONS

SHAFT SPEED (RPM)= [illegible] RIB VELOCITY (FT/MIN)= 10368.

DN = 924077.

LUBRICANT FLOW CONDITIONS	SHAFT DUCTS	CONE DUCTS
FRICTION FACTOR	0.0344	0.0828
HEADLOSS (IN/SEC)**2	26976.	14902.
REYNOLDS NUMBER	7153.	773.
EXIT VELOCITY (IN/SEC)	1052.2094	507.3860

MAXIMUM MANIFOLD PRESSURE (PSI)= 4.34

TANGENTIAL VELOCITY OF LUBRICANT @ OUTLET (IN/SEC)= 2042.03

ANGLE OF EXIT RELATIVE TO OUTLET (DEGREES)= 13.496

EFFECTIVE EXPOSURE TIME (SEC)= 0.000184

% COVERAGE OF RIB = 21.73

	FLOW (FT/MIN)
CONE	35.06
MANIFOLD	32.41
SHAFT	120.75

PHL ENGRG PHL021-01 CONE RIB LUBRICATION BFS 09/24/73

PAGE 1

CONE NO. OIL DIS-3 CUP NO. 12.1" [illegible]

[illegible] Figure [illegible]

INPUT DATA

SHAFT ID (IN)= 10.0000 SHAFT OD (IN)= 12.1270

CONE INLET DIA. (IN)= 12.3740 CONE OUTLET DIA. (IN)= 13.0060

CONE RIB-ROLLER END CONTACT DIA. (IN)= 13.3150

CONE RIB OD (IN)= 13.6250

CONE MANIFOLD TRAP LENGTH (IN)= 0.1500 MAX DIA= 12.77400

CONE LUBRICATION DUCT ANGLE (D-M-S)= 15- 0- 0

LUBRICANT VISCOSITY (CS) @ 100 F.= 32.00000
@ 210 F.= 4.00000
@ 165 F.= 8.09789

SPECIFIC GRAVITY OF LUBRICANT @ 60 F.= 1.007199

COEFFICIENT OF EXPANSION = 0.00041

SHAFT LUBRICATION DUCTS
NO.= 2 ID (IN)= 0.1875 ID SURFACE-FIN. (IN)= 0.00014

CONE LUBRICATION DUCTS
NO.= 24 ID (IN)= 0.0420 ID SURFACE-FIN. (IN)= 0.00014

SHAFT SPEED (RPM)= 3000 INCREMENTS = 3
2000
1000

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CONE NO. OIL DIS-3 CUP NO. 12.1" [illegible]

CALCULATIONS

SHAFT SPEED (RPM)= 1000. RIB VELOCITY (FT/MIN)= 3486.

DN = 300026.

LUBRICANT FLOW CONDITIONS	SHAFT DUCTS	CONE DUCTS
FRICTION FACTOR	0.0453	0.2834
HEADLOSS (IN/SEC)**2	3894.	4149.
REYNOLDS NUMBER	2367.	226.
EXIT VELOCITY (IN/SEC)	340.1993	140.3166

MAXIMUM MANIFOLD PRESSURE (PSI)= 0.62

TANGENTIAL VELOCITY OF LUBRICANT @ OUTLET (IN/SEC)= 680.99

ANGLE OF EXIT RELATIVE TO OUTLET (DEGREES)= 11.808

EFFECTIVE EXPOSURE TIME (SEC)= 0.000087

% COVERAGE OF RIB = 34.78

	FLOW (PT/MIN)
CONE	10.25
MANIFOLD	11.90
SHAFT	39.96

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CONE NO. OIL DIS-3 CUP NO. 12.1" Bore Cone

CALCULATIONS

SHAFT SPEED (RPM)= 2000. RIB VELOCITY (FT/MIN)= 5972.

DN = 616051.

LUBRICANT FLOW CONDITIONS	SHAFT DUCTS	CONE DUCTS
FRICTION FACTOR	0.0381	0.1309
HEADLOSS (IN/SEC)**2	13211.	8987.
REYNOLDS NUMBER	4757.	489.
EXIT VELOCITY (IN/SEC)	699.7613	321.2615

MAXIMUM MANIFOLD PRESSURE (PSI)= 2.50

TANGENTIAL VELOCITY OF LUBRICANT @ OUTLET (IN/SEC)= 1361.99

ANGLE OF EXIT RELATIVE TO OUTLET (DEGREES)= 12.835

EFFECTIVE EXPOSURE TIME (SEC)= 0.000419

% COVERAGE OF RIB = 32.79

	FLOW (PT/MIN)
CONE	22.20
MANIFOLD	23.86
SHAFT	83.30

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CONE NO. OIL DIS-3 CUP NO. 12.1" Bore Cone

CALCULATIONS

SHAFT SPEED (RPM)= 3000. RIB VELOCITY (FT/MIN)= 10498.

DN = 924077.

LUBRICANT FLOW CONDITIONS	SHAFT DUCTS	CONE DUCTS
FRICTION FACTOR	0.0344	0.0849
HEADLOSS (IN/SEC)**2	26976.	13845.
REYNOLDS NUMBER	7153.	754.
EXIT VELOCITY (IN/SEC)	1052.2854	494.9448

MAXIMUM MANIFOLD PRESSURE (PSI)= 5.62

TANGENTIAL VELOCITY OF LUBRICANT @ OUTLET (IN/SEC)= 2042.98

ANGLE OF EXIT RELATIVE TO OUTLET (DEGREES)= 13.171

EFFECTIVE EXPOSURE TIME (SEC)= 0.000274

% COVERAGE OF RIB = 32.14

	FLOW (PT/MIN)
CONE	34.20
MANIFOLD	35.84
SHAFT	120.75

APPENDIX II

HM926700-SERIES BEARING PRETEST INSPECTION

TABLE X. SUMMARY OF MODIFIED HM926700-SERIES BEARING PRETEST INSPECTIONS

Test No.	Bearing No.	Cone Surface Finishes (in.) Rib Face	Race	Roller Surface Finishes (in.) SER	Bodies	Cone Race to Rib Angle	Roller SER Pct Apex
1	60-1	7-15	8-8-1/2	7-9	4-5	90°2'45"	78
	PL-7	From shakedown run					
	PL-6	From shakedown run					
	PL-8	From shakedown run					
2	60-2	5-6	9-10	7-9	4-5	89°59'15"	80
	60-3	6-7	9-10	7-8	4-5	89°59'15"	80
	60-4	5-6	9-10	7-8	4-5	89°59'15"	80
	60-5	5-7	7-1/2-8	7-8	4-5	89°59'15"	80
3	60-6	6-7	8-9	7-8	4-5	89°59'15"	78
	60-7	5-7	9-10	7-8	4-5	89°59'15"	80
	60-9A	5-7	9-10	4-5	4-5	89°56'	80
	60-5	Same as test 2					
4	60-8	8-10	9-10	6-7	4-5	89°59'15"	78
	PL-7	From shakedown run					
	PL-6	From shakedown run					
	PL-8	From shakedown run					
5	60-9	8-9	6-1/2-7	6-7	4-1/2-5-1/2	90°2'45"	78
	60-10	8-9	6-1/2-7	6-7	4-1/2-5	89°59'15"	78
	60-11	8-10	6-6-1/2	6-7	4-1/2-5	90°2'45"	78
	60-12	8-10	7-7-1/2	6-7	4-1/2-5	89°59'15"	78
6-10	Fail-safe tests						
11	1	9-13	6-7	4-5	5-6	89°56'	79
	2	10-14	6-7	4-5	5-6	89°53'	79
	3	10-13	6-7	4-5	5-6	89°56'	78
	4	10-14	5-6	4-5	5-6	89°56'	78
12	5	8-12	5-6	6-7	5-6	89°53'	78
	6	6-10	7-8	6-7	5-6	89°53'	78
	7	8-12	7-8	6-7	4-1/2-5-1/2	89°53'	78
	8	9-12	5-6	6-7	4-1/2-6	89°53'	78
13	5	Same as test 12					
	6	Same as test 12					
	7	Same as test 12					
	9	9-11	5-6	6-7	5-6	89°56'	78
14	10	10-16	4-8	5-7	6-9	89°56'	80
	11	10-16	4-8	5-7	5-8	89°56'	80
	7	Same as test 12					
	9	Same as test 13					
15	10	Same as test 13					
	12	8-10	4-8	5-7	6-9	89°56'	78
	7	Same as test 14					
	9	Same as test 14					
16	13	5-12	4-7	5-7	6-8	89°56'	78
	12	Same as test 15					
	7	Same as test 15					
	9	Same as test 15					

TABLE X. Continued.

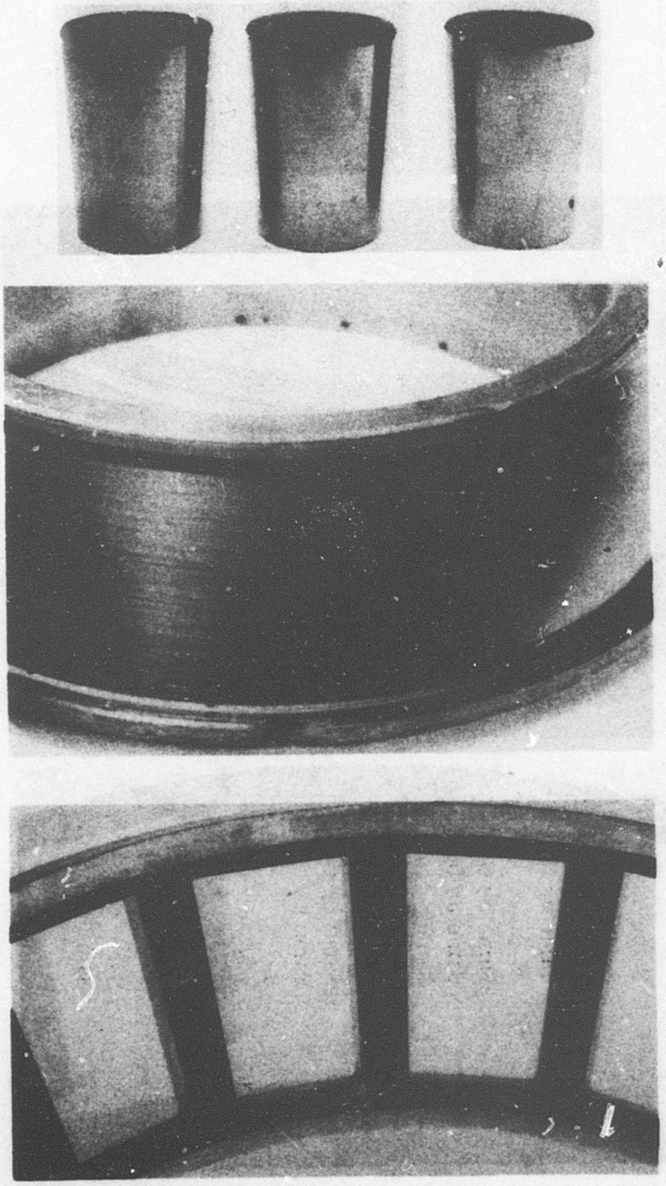
Test No.	Bearing No.	Cone Surface Finishes (in.) Rib Face	Race	Roller Surface Finishes (in.) SE	Bodies	Cone Race to Rib Angle	Roller SER Pct Apex	Cone Race Taper
17	1-S	Timken oil-slotted bearing						
	2-S	Timken oil-slotted bearing						
	7	Same as test 16						
	9	Same as test 16						
18	X	7-8	5-6	5-6	5-6	89°56'	80	
	60-10	9	5-6	5-6	5-6	89°59'15"	80	
	60-11	6	6-6-1/2	4-6	5-7	90°2'45"	80	
	60-12	7	6-6-1/2	4-7	4-1/2-5-1/2	89°59'15"	80	
19	X	Same as test 18						
	2-S	Same as test 17						
	60-11	Same as test 18						
	60-12	Same as test 18						
20	X	Same as test 18						
	2-S	Same as test 17						
	3-S	Same as test 17						
	60-12	Same as test 19						
21	9	Same as test 17						
	2	Same as test 11						
	14	8-9	6-6-1/2	4-6	5-7	89°59'15"	80	
	60-12	Same as test 20						
22	T-5	3-6	5-8	5-7	3-6	89°54'	80	+0.00005
	T-6	4-6	6-8	4-8	4-7	89°54'	80	+0.0002
	T-7	4-7	6-8	4-6	4-8	89°54'	80	+0.0001, +0.0002
	T-8	4-6	7-8	5-7	3-7	89°54'	80	-0.0006, +0.0008
23	T-5	Same as test 22				89°54'		+0.00005
	T-6	Same as test 22				89°54'		+0.0002
	T-7	Same as test 22				89°41'		+0.0001, +0.0002
	T-9	6-10	6-8	4-6	3-6	89°41'	80	+0.00005
24	T-5	4-7	6-9	6-7	5-6	89°41'	80	+0.00005
	T-10	6-10	6-9	3-6	4-8	89°41'	80	+0.00005
	T-7	Same as test 23				89°41'		+0.0001, 0.0002
	T-9	Same as test 23				89°41'		+0.00005
25	T-11	7-8	10-12	4-6	4-8	89°41'	80	+0.00015
	2-S	Same as test 17						
	3-S	Same as test 17						
	T-12	6-8	10-12	4-6	5-9	89°41'	80	+0.0001
26	T-13	7-8	10-12	1.5-2	14-15	89°38'	85	
	2-S	Same as test 17						
	3-S	Same as test 11						
	T-12	Same as test 25				89°38'	85	
27	T-15	3-7	10-12	2-3	10-14	89°51'30"	80	
	2-S	Same as test 17						
	3-S	Same as test 11						
	T-14	4-8	11-12	1.5-2	14-16	89°56'	85	
28	73-3	5-7		2-1/2-3-1/2	10-14	90°2'45"	80	
	2-S	Same as test 17						
	3-S	Same as test 17						
	73-5	6-8		2.5-3.5	10-14	90°2'45"	80	

TABLE X. Continued.

Test No.	Bearing No.	Cone Surface Finishes (in.) Rib Face	Cone Surface Finishes (in.) Race	Roller Surface Finishes (in.) SER	Roller Surface Finishes (in.) Bodies	Cone Race to Rib Angle	Roller SER Pct Apex
29	1-S	Same as test 17					
	2-S	Same as test 17					
	3-S	Same as test 11					
	73-5	Same as test 28					
30	T-15	Same as test 27					
	2-S	Same as test 17					
	PL-7	Same as test 4					
	60-8	Same as test 4					
31	T-15	Same as test 27					
	2-S	Same as test 17					
	73-7	5-7	6.8-7.2	2.6-2.8	4.2-7.0	90°2'45"	78
	73-13	5-8	6.6-7.6	2.2-3.8	3.8-5.4	90°2'45"	78
32	T-15	Same as test 27					
	2-S	Same as test 17					
	3-S	Same as test 17					
	73-13	Same as test 31					
33	T-15	Same as test 27					
	73-8			2.5-3.0	4.2-6.0	89°54'	78
	3-S	Same as test 17					
	73-13	Same as test 31					

APPENDIX III

<u>HM926700-SERIES BEARING AFTER TEST</u>



CONE HAS 17 RADIAL OIL LUBRICATION HOLES.

Figure 44. HM926700-Series Bearing After Test, Bearing No. 60-1, Test No. 1--Damaged After Running 1½ Minutes at 9300rpm.

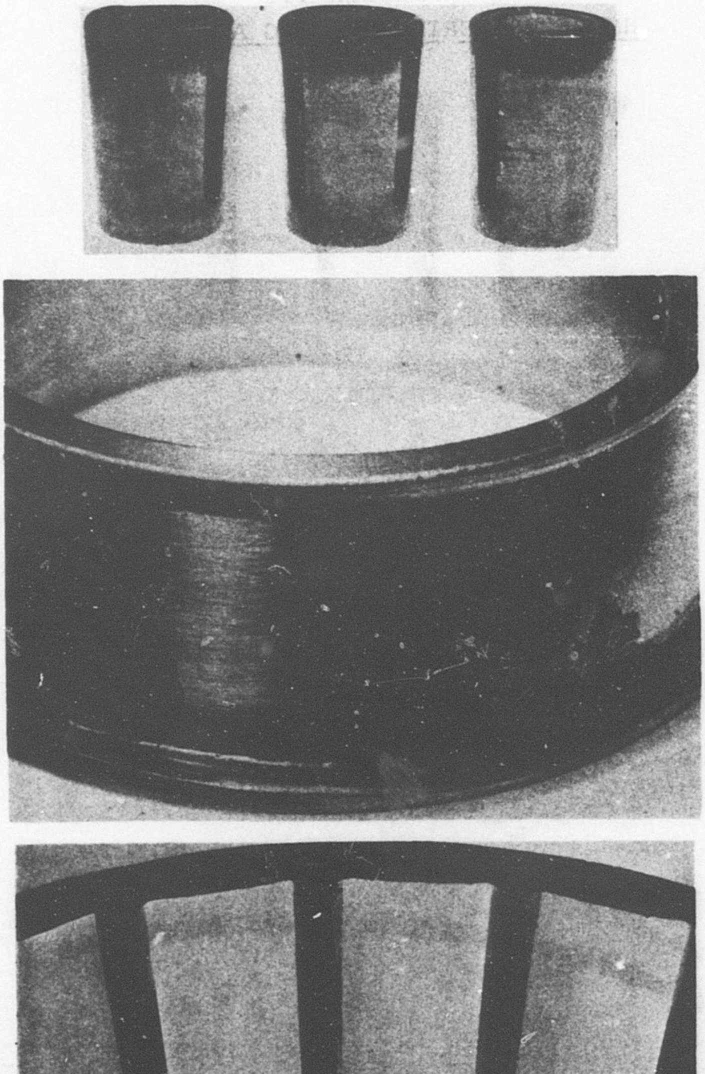
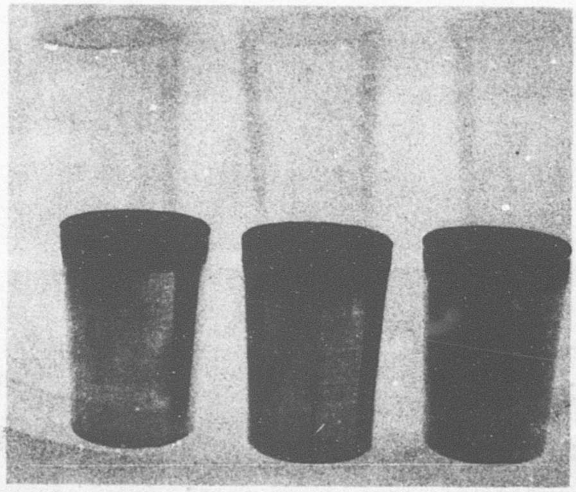
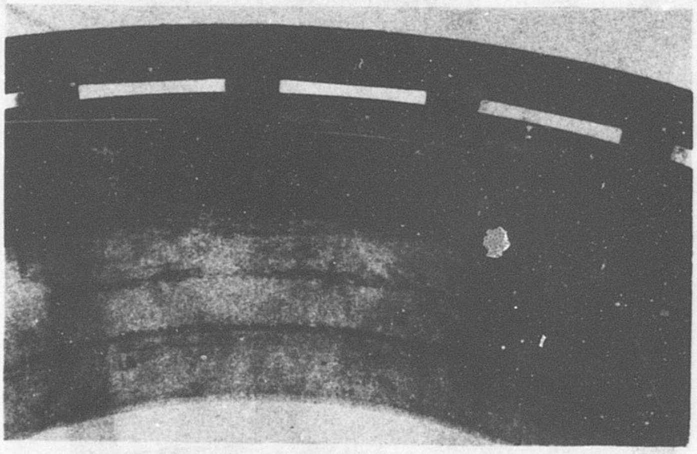


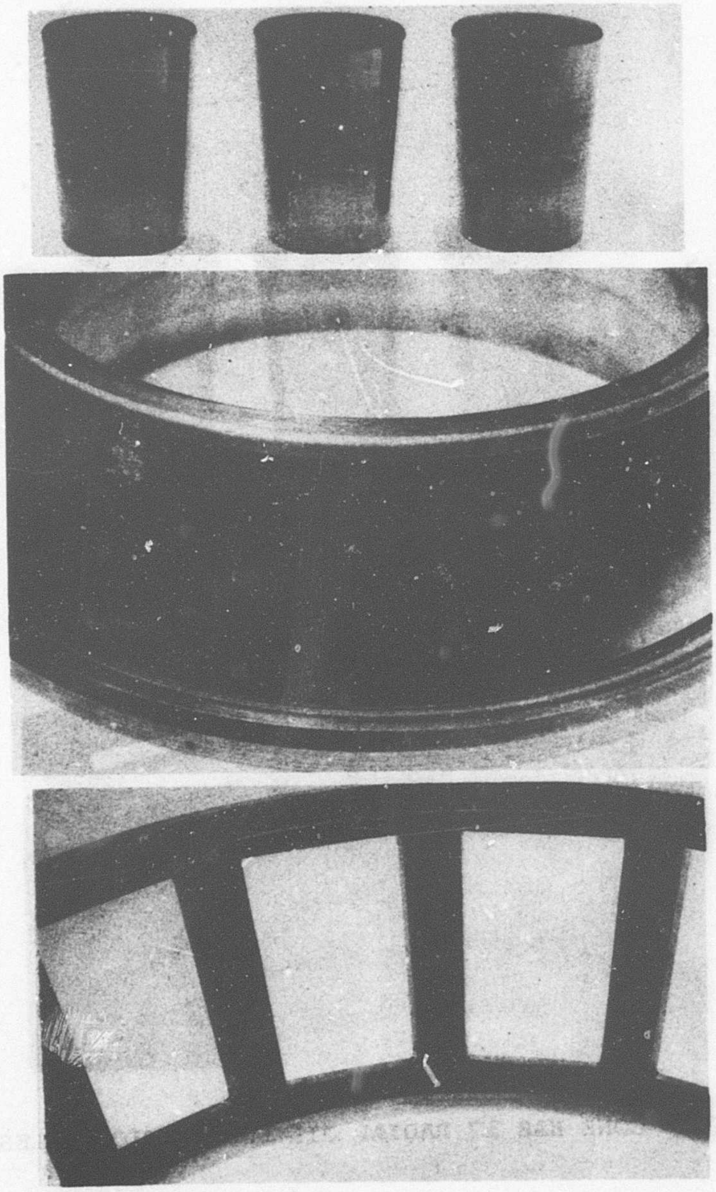
Figure 45. HM926700-Series Bearing After Test, Bearing No. 60-4, Test No. 2--Damaged After Running for 45 Minutes at 14,000 rpm.





NOTE: CONE HAS 17 RADIAL OIL LUBRICATION HOLES

Figure 46. HM926700-Series Bearing After Test, Bearing No. 60-4A, Test No. 3--Damaged After Running 3 Minutes at 14,000 rpm.



NOTE: CONE HAS 17 RADIAL OIL LUBE HOLES

Figure 47. HM926700-Series Bearing After Test, Bearing No. 60-7, Test No. 3--Okay After Running at 14,000 rpm for 3 Minutes.

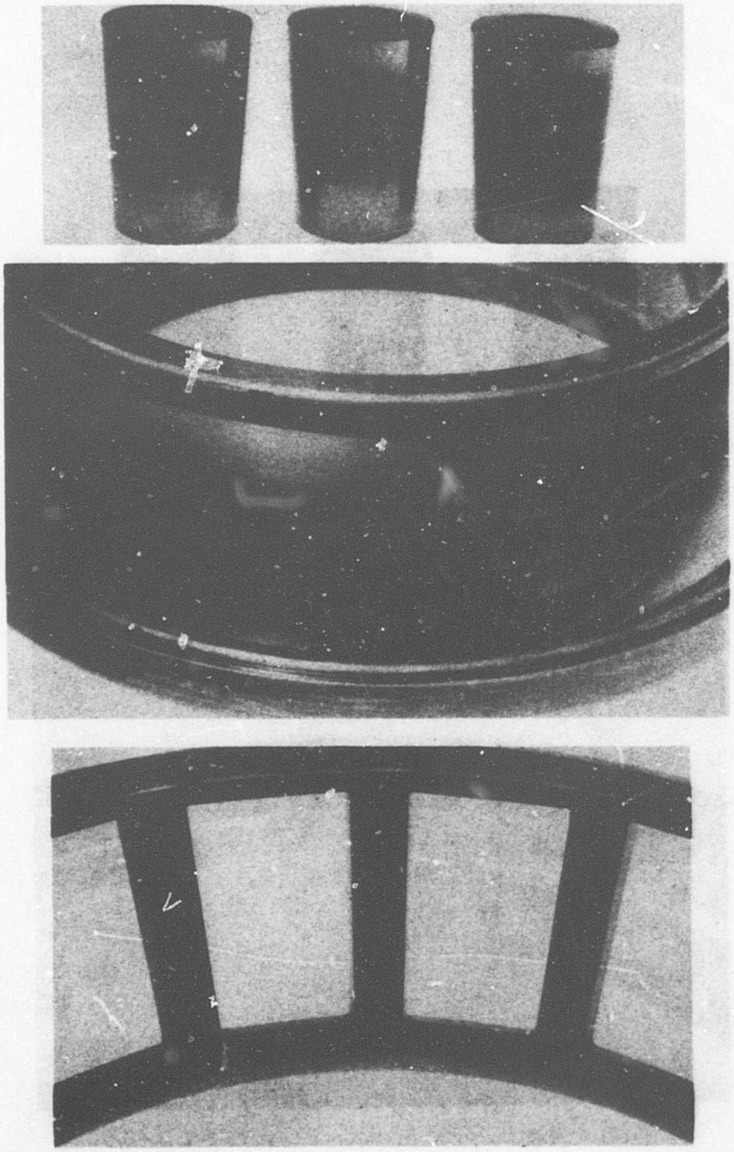
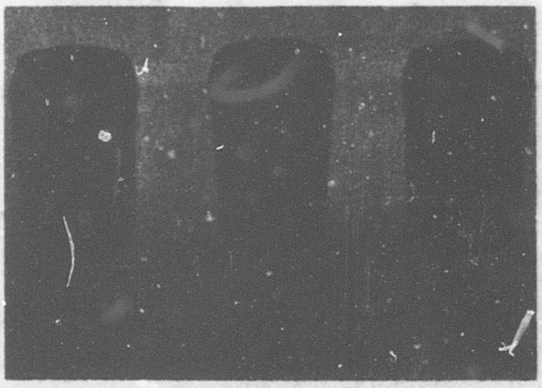


Figure 48. HM926700-Series Bearing After Test, Bearing No. 60-9, Test No. 5--Tested for 27.5 Hours at 14,000 rpm.



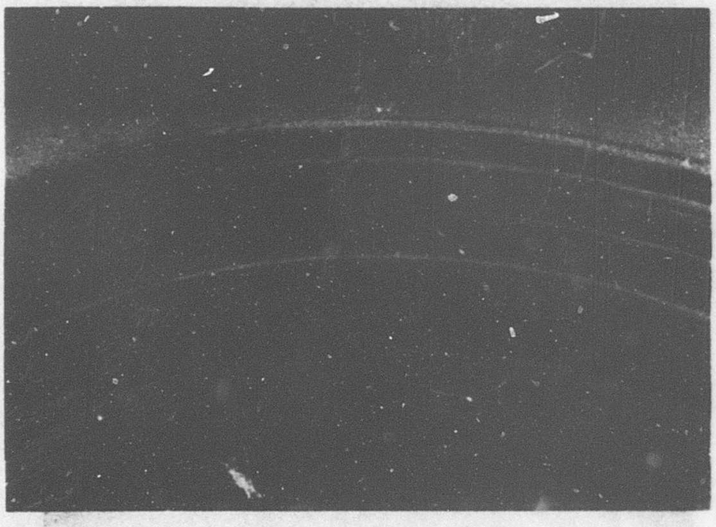


Figure 49. HM926700-Series Bearing After Test, Bearing No. 3, Test No. 11--Damaged While Accelerating to 9000 rpm.

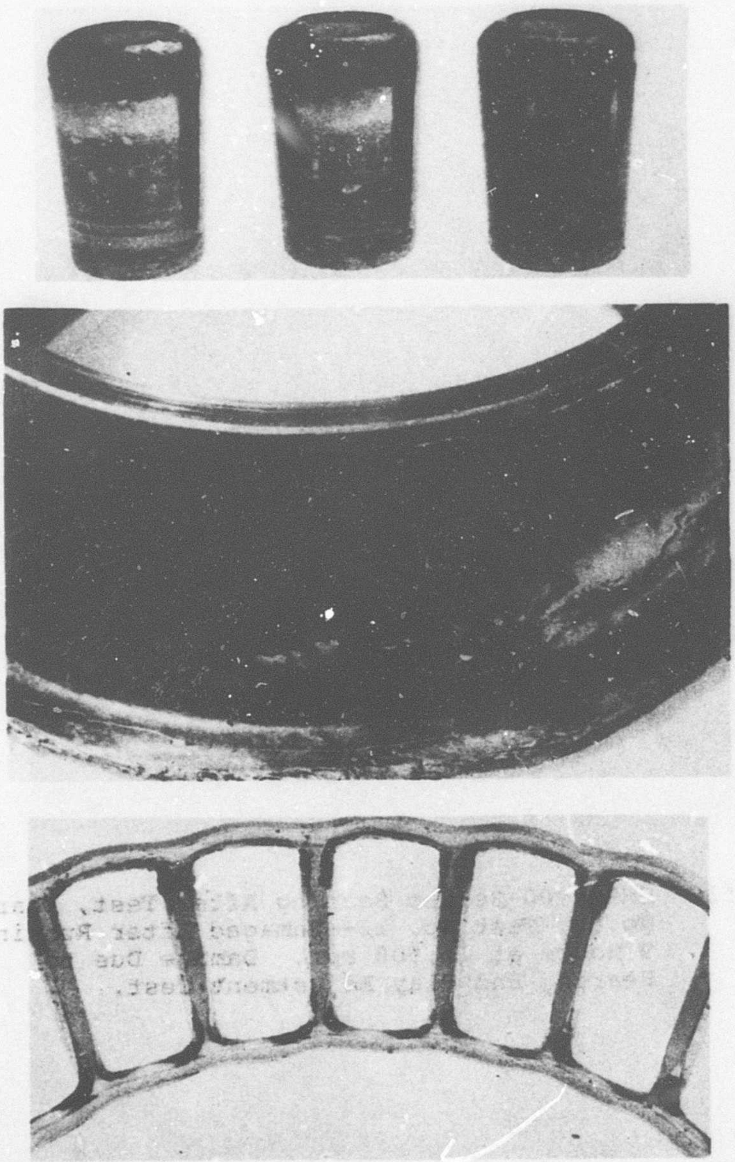


Figure 50. HM926700-Series Bearing After Test, Bearing No. 7, Test Nos. 12 through 17--Damaged After 22 hours at 11,500 rpm.

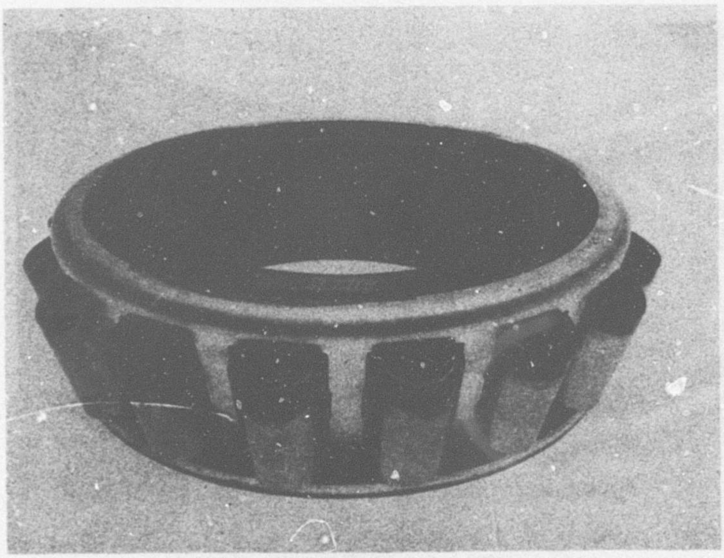


Figure 51. HM926700-Series Bearing After Test, Bearing No. 8, Test No. 12--Damaged After Running 9 Hours at 11,500 rpm. Damage Due to Bearing End-Play Adjustment Test.

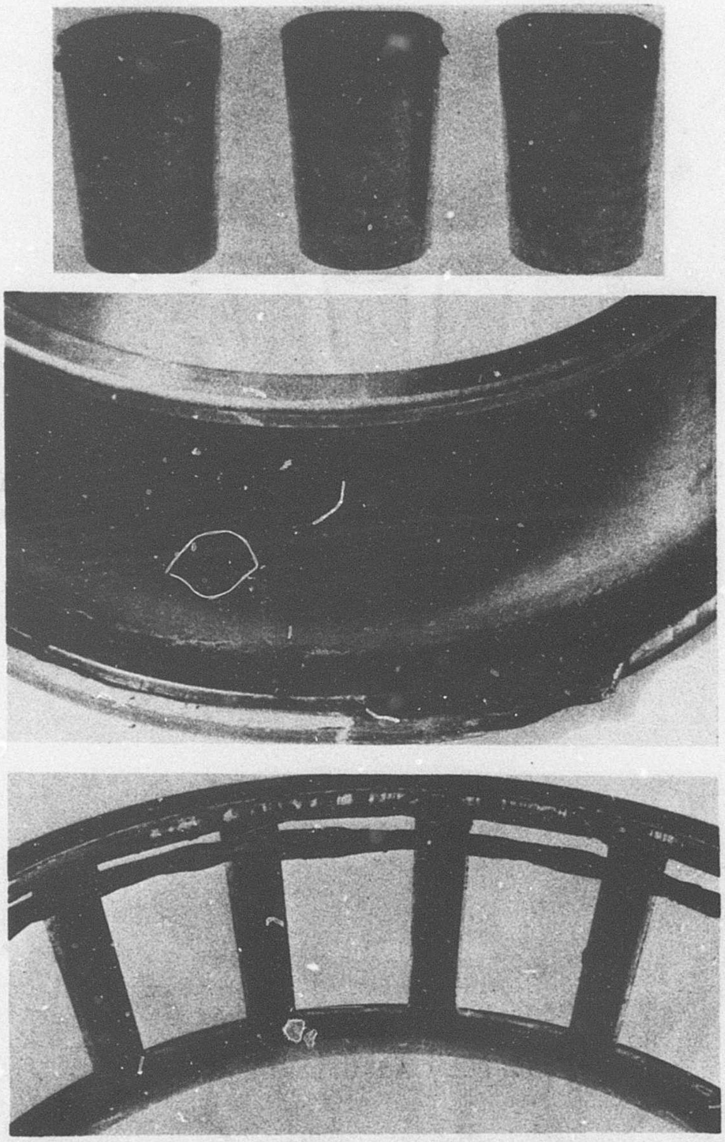


Figure 52. HM926700-Series Bearing After Test, Bearing No. 6, Test No. 13--Damaged After Running for 13 Hours at 11,500 rpm With Various Bearing End-Play Adjustments.

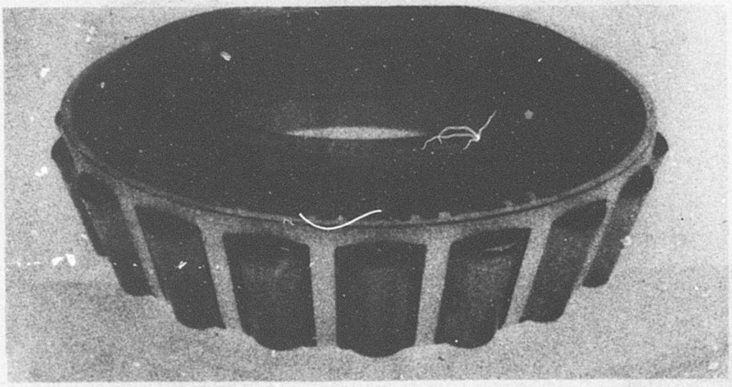


Figure 53. HM926700-Series Bearing After Test, Bearing No. 9-Okay After Running in Test Numbers 13 Through 17 and Test No. 21.

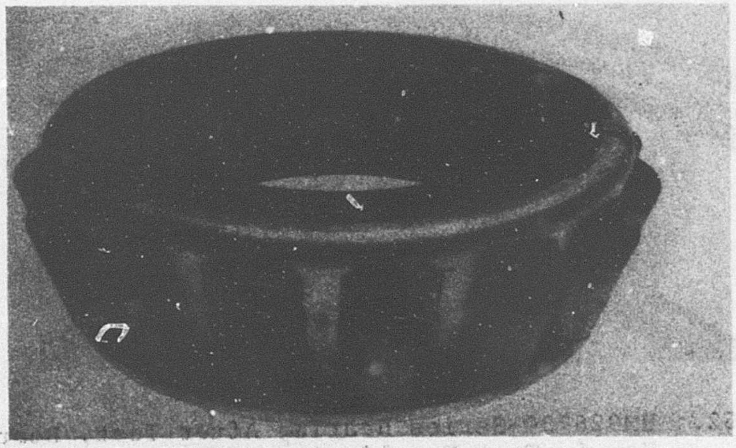


Figure 54. HM926700-Series Bearing After Test, Bearing No. 10, Test No. 15--Damaged Due to Test Assembly Problem.

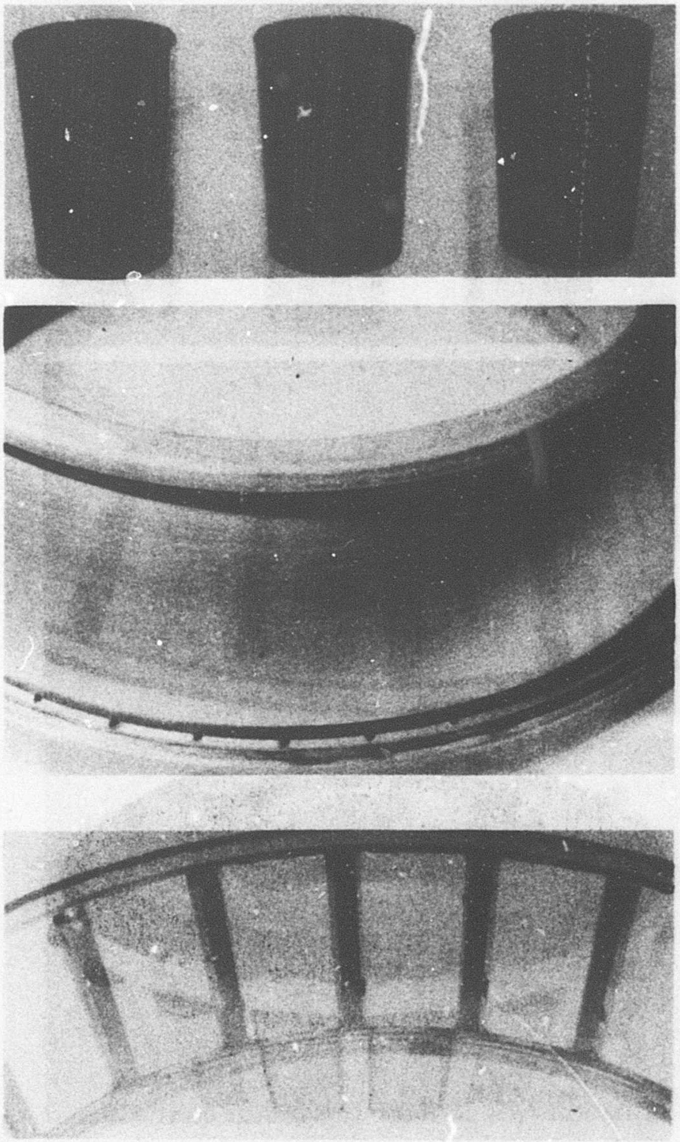


Figure 55. HM926700-Series Bearing After Test, Bearing No. T-8, Test No. 22--Damaged After Running 9 Hours at 11,500 rpm. Damage Due to Bearing End-Play Adjustment Test.

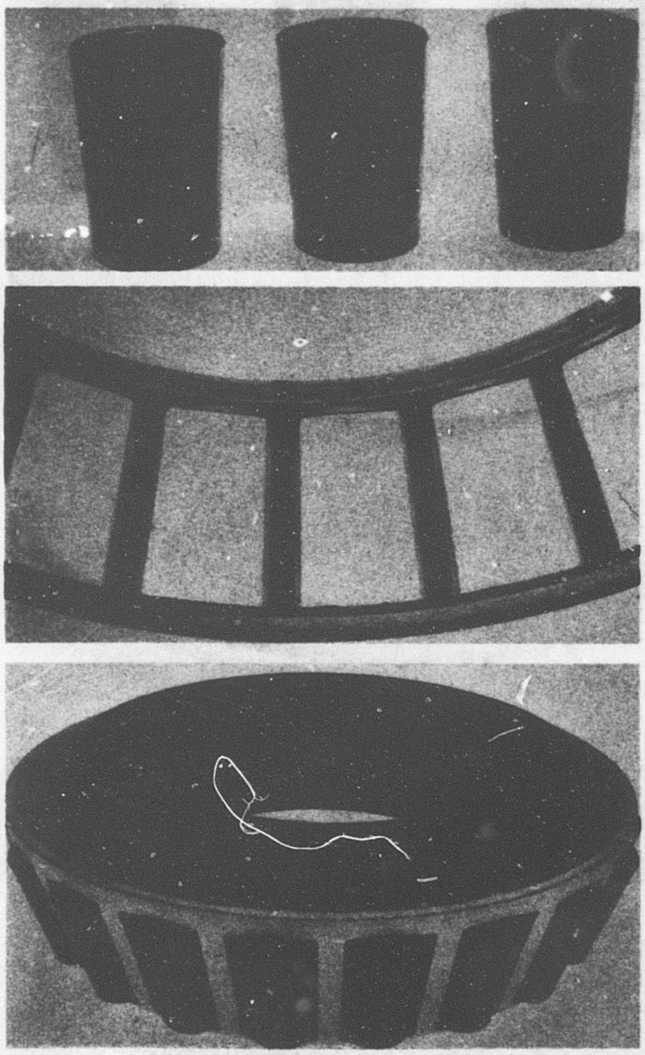
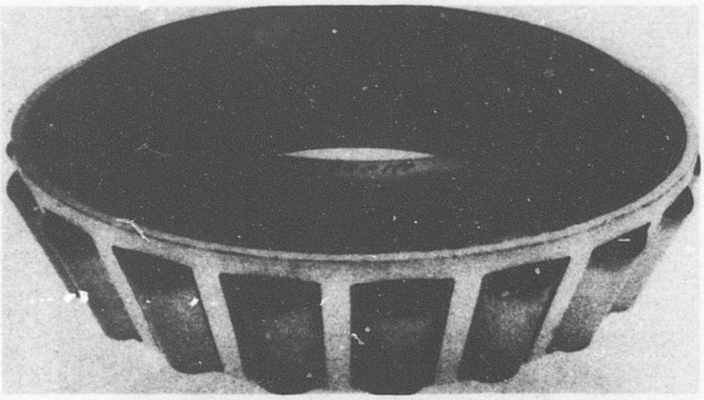


Figure 56. HM926700-Series Bearing After Test, Bearing No. T-6, Test No. 23--Damaged While Accelerating to 10,000 rpm.



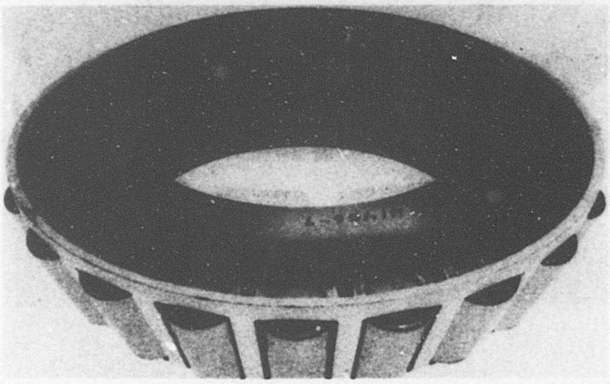


Figure 57. HM926700-Series Bearings After Test, Bearing Numbers T-7 and T-9, Test No. 24--Damaged at 11,500 rpm.

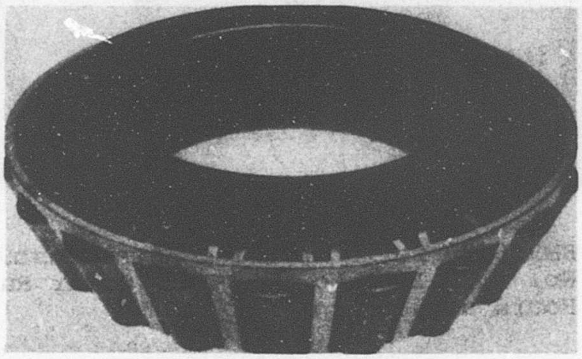
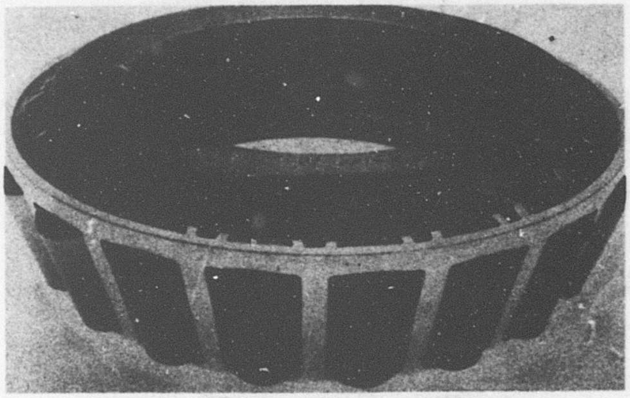


Figure 58. HM926700-Series Bearing After Test, Bearing No. T-10--Okay After Running in Test No. 24.



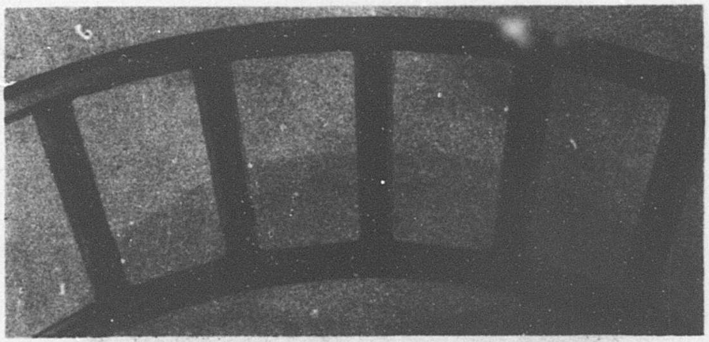


Figure 59. HM926700-Series Bearing After Test, Bearing No. T12, Test No. 26--Okay After Running 16 Hours at 5000 rpm.

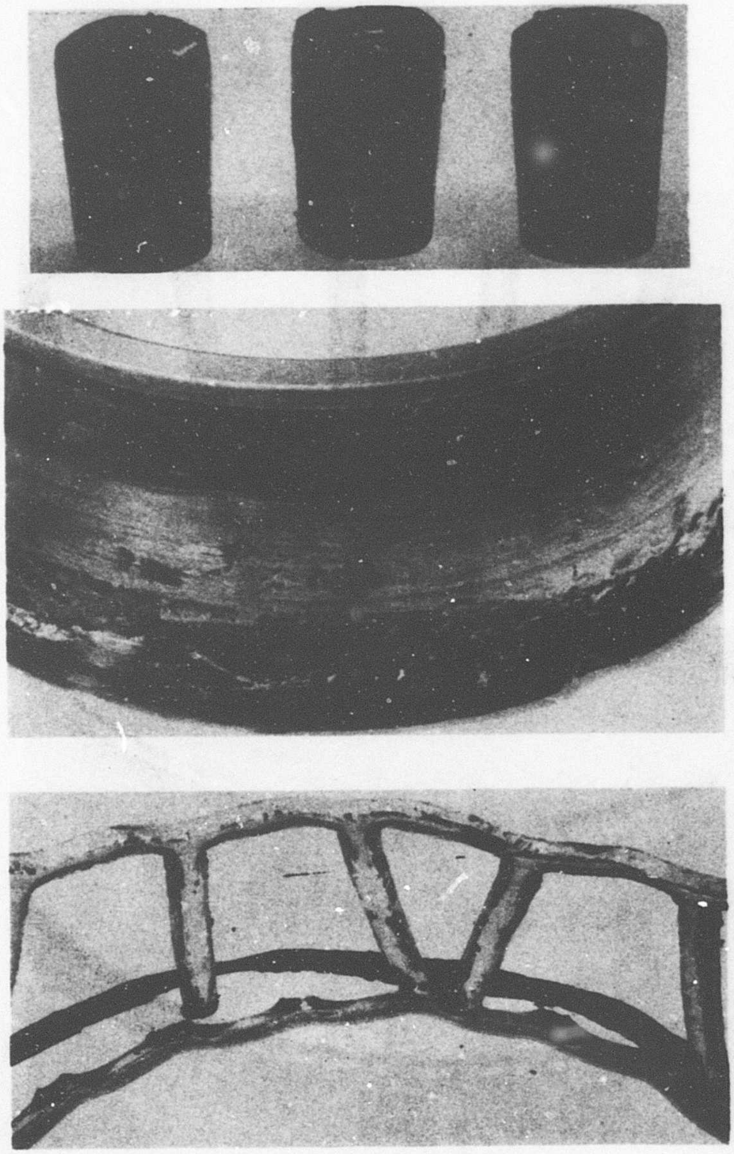


Figure 60. HM926700-Series Bearing After Test, Bearing No. T-14, Test No. 27--Damaged After Running 1 Hour at 11,500 rpm.

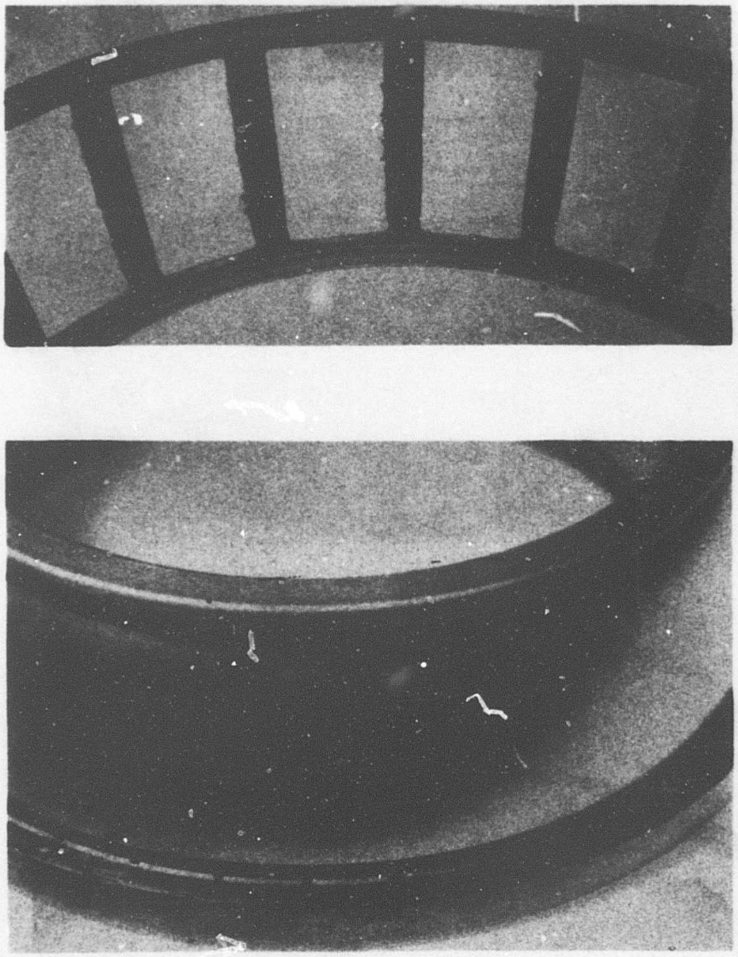


Figure 61. HM926700-Series Bearing After Test, Bearing No. 73-5, Test No. 29--Damaged at 10,000 rpm.

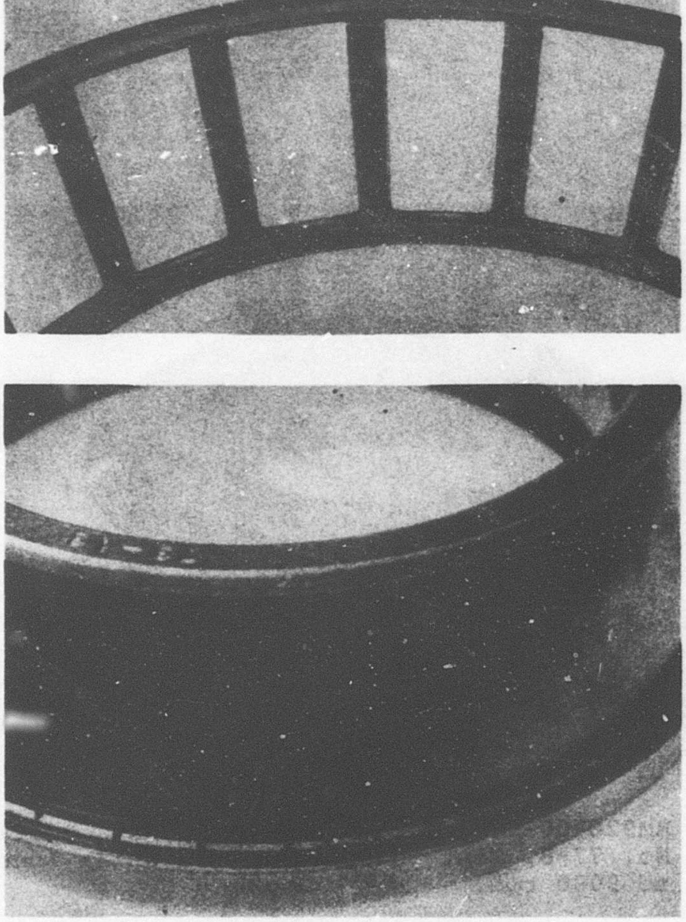


Figure 62. HM926700-Series Bearing After Test, Bearing No. 73-13, Test Numbers 31 Through 33--Okay After Running 24 Minutes at 11,500 rpm.

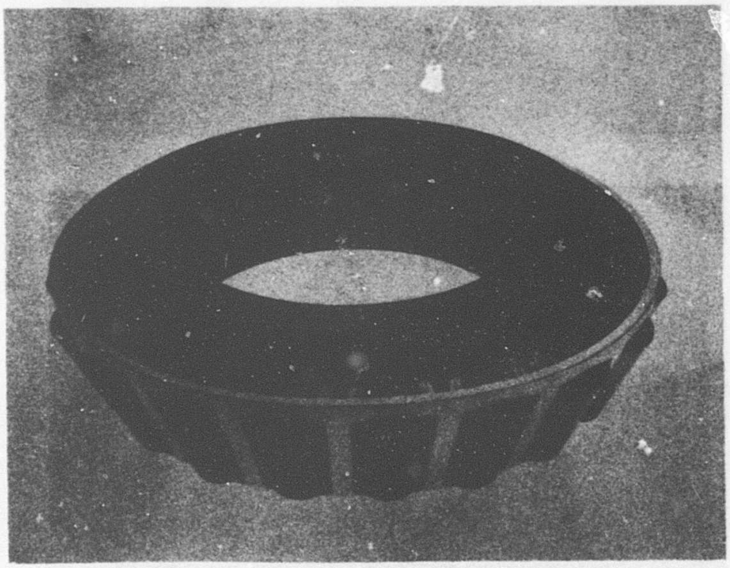


Figure 63. HM926700-Series Bearing After Test, Bearing No. 73-8, Test No. 33--Damaged While Accelerating to 9000 rpm.

APPENDIX IV
HM926700-SERIES BEARINGS
DEVELOPMENT TESTS LEVEL-OFF TEMPERATURES

TABLE XI. SUMMARY OF RESULTS - DEVELOPMENT TESTS

Test	Bearing No.	No. Lube Holes	Oil Flow Rate (Pints/min) *LE/**SE	End Play (in.)	Cylinder Load Applied to Housings (lb) Radial/thrust	Bearing Speed	Cup Temp (°F) at Speed (rpm) DE	DEC	ODEC	ODE	Outlet Oil Temp (°F) DE	ODE	Remarks
Constant-Speed 100-HP Drive													
Shakedown run	PL-4	17	6/6	0.0065	1230/440								Bearing PL-4 r-ller end/rib scuffing.
	PL-1	17			then								All bearings show signs of skidding.
	PL-2	17		0.0020	0/440								10,000 rpr max speed.
	PL-3	17											
Shakedown run	PL-7	17	6/6	0.0065	2460/900								Bearing PL-5 .oller end/rib scuffing.
	PL-5	17											13,000 rpm max speed.
	PL-6	17		0.0020									
	PL-8	17											
Shakedown run	PL-7	34	6/6	0.0017	2460/900								Bearing PL-9 roller end/rib scuffing.
	PL-9	34											Ran at 9300 rpm for 1 min.
	PL-6	34		0.0020									
	PL-8	34											
No. 1 Development	60-1	17	7.5/6	0.0008	1230/440								Bearing 60-1 roller end/rib scuffing.
	PL-7	34											Ran at 9300 rpm for 1.5 min.
	PL-6	34		0.0020									
	PL-8	34											
Variable-Speed 700-HP Drive													
No. 2 Development	60-2	17	10/6	0.0010	12,300/4394	4000	208	210	215	210	208	210	Bearings 60-2, 60-3 and 60-4 roller
	60-3	17				8000	246	244	248	246	238	244	end/rib scuffing after running 45
	60-4	17		0.0005		10,000	278	280	294	278	252	270	min at 14,000 rpm. Lube pump
	60-5	17				12,000	310	312	316	310	260	280	stopped causing damage.
						14,000	324	324	336	324	284	330	
No. 3 Development	60-6	17	10/6	0.0010	12,300/4394	4000	215	217	220	220	210	219	Ran for 12 hr at 12,000 rpm. Bearing
	60-7	17				8000	264	252	282	258	238	255	60-4A damaged after running 3 min at
	60-4A	17		0.0006		10,000	280	270	280	275	246	270	14,000 rpm.
						12,000	281	280	302	294	260	286	
No. 4 Development and acceleration test	60-8	34	10/6	0.0005	1230/440	5000	230	228	232	230	228	237	Ran 36 hr at 14,000 rpm. Bearing
	PL-7	34			to	8000	253	251	256	250	229	244	PL-8 roller end/rib scuffing after
	PL-6	34		0.00025	12,300/4394	9800	267	265	272	265	239	255	third acceleration test. Bearings
	PL-8	34				11,500	294	288	296	280	256	277	60-8, -6 and PL-7 had spall damage.
						14,000	324	319	334	322	280	306	
No. 5 Development and acceleration test	60-9	40	10/6	0.0005	1230/440	14,000	327	315	325	312	282	303	Ran 27.5 hr at 14,000 rpm. Only damage was small spall on bearing 60-9.
	60-10	40			to								
	60-11	40		0.0005	12,300/4394								
	60-12	40											

* Large End
** Small End

APPENDIX V
HM926700-SERIES BEARING FAIL-SAFE TEST LEVEL-OFF TEMPERATURES

TABLE XII SUMMARY OF FAIL-SAFE TEST RESULTS

Test	Bearing No.	No. Lube Holes	Oil Flow Rate (Pints/min) *LE/**SE	End Play DE/ODE (in.)	Cylinder Load Applied To Housings (lb) Radial/Thrust	Bearing Speed	Cup Temp (°F) at Speed (rpm) DE	DEC	ODEC	ODE	Outlet Oil Temp (°F) DE	ODE	Remarks
No. 6 Fail-safe LP-Alloy	LP-1	36	10/6	0.0005	6700/1900	5000	206	212	218	214	200	212	Piece of LP-Alloy broke off separable rib bearing LP-2. Bearing LP-1 roller end/rib scuff.
	LP-2	36				8000	234	230	240	242	218	226	
	LP-3	36		0.00075		9000	262	256	256	260	236	250	
	LP-4	36											
No. 7 Fail-safe Boriate	B-5	36	10/6	0.0005	6700/1900	5000	214	213	218	212	200	206	Separable rib crushed on two bearings, B-7, B-8.
	B-6	36				8000	246	246	256	230	220	200	
	B-7	36		0.0010		9000	260	266	276	266	240	256	
	B-8	36											
No. 8 Fail-safe LP-Alloy	LP-9	36	10/6	0.0005	6700/1900	5000	212	214	220	216	206	215	Test run for 26 hr at 11,000 rpm. Test run for 8.5 sec with oil off. Bearing LP-11 damaged.
	LP-10	36		(0.000)		8000	240	240	262	241	230	230	
	LP-11	36		0.0015		9500	240	260	263	260	236	247	
	LP-12	36		(0.000)		11,000	272	274	296	272	244	260	
No. 9 Fail-safe LP-Alloy	LP-13	36	10/6	0.0005	6700/1900	5000	214	218	228	223	206	218	Bearings OK at 24 hr end play check at 11,000 rpm. Bearing LP-15 is damaged at 30.5 total hr.
	LP-14	36		(0.0000)		8000	242	244	256	240	226	245	
	LP-15	36		0.0005		9000	258	268	270	250	236	257	
	LP-16	36		(0.0000)		11,000	279	276	290	266	240	272	
No. 10 Repeat Fail-safe LP-Alloy	LP-9A	36	10/6	0.000	6700/1900	11,000	[illegible]	266	296	306	290	265	Test run 3 hr with oil-on. Bearing 9A damaged after running 10 sec with oil off.
	LP-10A	36											
	LP-13A	36		0.000									
	LP-12A	36											

*Large End
**Small End

APPENDIX VI
CORRELATION STUDY OF Z-TYPE CAGE FLANGE GUIDANCE CLEARANCE AND ROLLER END/CONE RIB SCUFFING DAMAGE

TABLE XIII. TABULATION OF CONE RIB/CAGE CLEARANCES DUE TO CONE PRESS FIT ON SHAFT

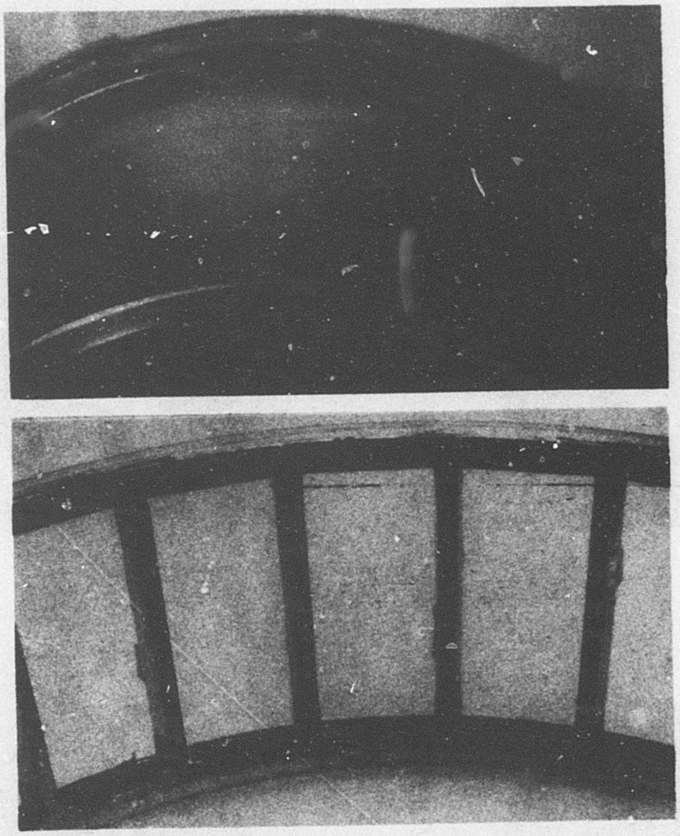
Test No.	Bearing No.	Shaft OD	Cone ID	Cone Fit	Cone Rib OD	Cone Rib OD due to Fit	Cage ID	Cage Clearances	Remarks
1 Development	60-1	5.0052	5.0004	0.0048	7.0264	7.0302	7.0341	0.0039	Damaged
	PL-7	5.0152	5.0108	0.0044	7.0271	7.0304	7.0325	0.0021	OK
	PL-6	5.0151	5.0101	0.0050	Stamped Cage		Stamped Cage		OK
	PL-8	5.0052	5.0007	0.0045	Stamped Cage		Stamped Cage		OK
2 Development	60-2	5.0052	5.0103	0.0051	7.0268	7.0306	7.0345	0.0039	Damaged
	60-3	5.0152	5.0102	0.0050	7.0265	7.0303	7.0335	0.0032	Damaged
	60-4	5.0151	5.0106	0.0045	7.0268	7.0302	7.0330	0.0028	Damaged
	60-5	5.0052	5.0004	0.0048	7.0253	7.0290	7.0350	0.0061	OK
3 Development	60-6	5.0052	5.0004	0.0048	7.0265	7.0301	7.0340	0.0039	OK
	60-7	5.0152	5.0101	0.0042	7.0268	7.0300	7.0360	0.0060	OK
	60-4A	5.0151	5.0106	0.0044	7.0268	7.0301	7.0330	0.0029	Damaged
	60-5	5.0052	5.0004	0.0048	7.0253	7.0290	7.0350	0.0061	OK
4 Development and acceleration tests	60-8	5.0052	5.0003	0.0049	7.0270	7.0307	7.0370	0.0063	OK (spall)
	PL-7	5.0152	5.0108	0.0044	7.0265	7.0298	7.0350	0.0052	OK (spall)
	PL-6	5.0151	5.0102	0.0049	7.0265	7.0302	7.0360	0.0058	OK (spall)
	PL-8	5.0052	5.0008	0.0044	7.0270	7.0303	7.0360	0.0057	Damaged
5 Development and acceleration tests	60-9	5.0052	5.0007	0.0045	7.0260	7.0294	7.0350	0.0053	OK (spall)
	60-10	5.0152	5.0097	0.0055	7.0252	7.0293	7.0380	0.0087	OK
	60-11	5.0151	5.0101	0.0050	7.0250	7.0288	7.0370	0.0082	OK
	60-12	5.0052	5.0006	0.0046	7.0260	7.0294	7.0360	0.0066	OK
6 Fail-safe	1-LP	5.0052	5.0004	0.0048	7.0275	7.0311	7.0335	0.0024	Damaged
	2-LP	5.0152	5.0090	0.0063	7.0280	7.0320	7.0340	0.0020	Damaged
	3-LP	5.0151	5.0103	0.0048	7.0275	7.0311	7.0330	0.0019	OK
	4-LP	5.0052	5.0003	0.0049	7.0215	7.0252	7.0315	0.0063	OK
7 Fail-safe	5-B	5.0052	5.0003	0.0049	7.0235	7.0272	7.0355	0.0083	OK
	6-B	5.0152	5.0107	0.0045	7.0230	7.0264	7.0340	0.0076	Damaged
	7-B	5.0151	5.0108	0.0043	7.0230	7.0262	7.0345	0.0083	Damaged
	8-B	5.0052	5.0002	0.0050	7.0230	7.0267	7.0340	0.0073	OK
8 Fail-safe	9-LP	5.0052	5.0005	0.0047	7.0240	7.0275	7.0360	0.0085	OK
	10-LP	5.0152	5.0105	0.0047	7.0238	7.0273	7.0365	0.0092	OK
	11-LP	5.0151	5.0105	0.0046	7.0240	7.0275	7.0358	0.0083	Damaged
	12-LP	5.0052	5.0003	0.0049	7.0230	7.0267	7.0371	0.0104	OK
9 Fail-safe	13-LP	5.0052	4.9998	0.0054	7.0235	7.0275	7.0345	0.0070	OK
	14-LP	5.0152	5.0102	0.0050	7.0240	7.0278	7.0350	0.0072	OK
	15-LR	5.0151	5.0101	0.0050	7.0240	7.0278	7.0360	0.0082	Damaged
	16-LP	5.0052	4.9994	0.0058	7.0240	7.0283	7.0355	0.0072	OK
10 Fail-safe standard cage	9a-LP	5.0052	5.0009	0.0043					
	10a-LP	5.0152	5.0101	0.0051					
	13a-LP	5.0151	5.0095	0.0056					
	12a-LP	5.0052	5.0009	0.0043					

TABLE XIII. Continued.

Test No.	Bearing No.	Shaft OD	Cone ID	Cone Fit	Cone Rib OD	Cone Rib OD due to Fit	Cage ID	Cage Clearances	Remarks
11 Bearing adjustment No. 1	1	5.0062	4.9998	0.0064	7.0250	7.0290	7.0360	0.0070	OK
	2	5.0152	5.0098	0.0054	7.0240	7.0281	7.0350	0.0069	OK
	3	5.0151	5.0097	0.0054	7.0240	7.0281	7.0350	0.0069	Damaged
	4	5.0052	5.0002	0.0050	7.0260	7.0297	7.0350	0.0053	Damaged
12 Bearing adjustment No. 1A	5	5.0052	5.0002	0.0050	7.0260	7.0297	7.0355	0.0058	OK
	6	5.0152	5.0093	0.0059	7.0250	7.0294	7.0340	0.0046	OK
	7	5.0151	5.0093	0.0058	7.0240	7.0284	7.0350	0.0066	OK
	8	5.0052	5.0002	0.0050	7.0240	7.0277	7.0340	0.0063	Damaged
13 Bearing adjustment No. 2	5	5.0052	5.0002	0.0050	7.0260	7.0297	7.0355	0.0058	Damaged
	6	5.0152	5.0093	0.0059	7.0250	7.0294	7.0340	0.0046	Damaged
	7	5.0151	5.0093	0.0058	7.0240	7.0284	7.0350	0.0066	OK
	9	5.0052	5.0004	0.0048	7.0250	7.0286	7.0370	0.0084	OK
14 Oil flow No. 1	10	5.0052	5.0001	0.0051	7.0260	7.0298	7.0355	0.0057	OK
	11	5.0152	5.0091	0.0061	7.0250	7.0296	7.0380	0.0084	Damaged
	7	5.0151	5.0093	0.0058	7.0240	7.0284	7.0350	0.0066	OK
	9	5.0052	5.0004	0.0048	7.0250	7.0286	7.0370	0.0084	OK
15 Oil flow No. 1A	10	5.0052	5.0001	0.0051	7.0260	7.0298	7.0355	0.0057	Damaged
	12	5.0152	5.0100	0.0052	7.0250	7.0289	7.0340	0.0051	OK
	7	5.0151	5.0093	0.0058	7.0240	7.0284	7.0350	0.0066	OK
	9	5.0052	5.0004	0.0048	7.0250	7.0286	7.0370	0.0084	OK
16 Oil flow No. 1B	13	5.0052	5.0005	0.0047	7.0240	7.0275	7.0315	0.0040	Damaged
	12	5.0152	5.0100	0.0052	7.0250	7.0289	7.0340	0.0051	Damaged
	7	5.0151	5.0093	0.0058	7.0240	7.0284	7.0350	0.0066	OK
	9	5.0052	5.0004	0.0048	7.0250	7.0286	7.0370	0.0084	OK
17 Oil flow No. 1C	1-S	5.0052							
	2-S	5.0152							
	7	5.0151	5.0093	0.0058	7.0240	7.0284	7.0350	0.0066	Damaged
	9	5.0052	5.0004	0.0048	7.0250	7.0286	7.0370	0.0084	OK
18 Oil flow No. 1D	X	5.0052							
	60-10	5.0152	5.0097	0.0055	7.0252	7.0293	7.0380	0.0087	Damaged
	60-11	5.0151	5.0101	0.0050	7.0250	7.0288	7.0370	0.0082	OK
	60-12	5.0052	5.0006	0.0046	7.0260	7.0294	7.0360	0.0066	OK
19 Oil flow No. 1E Standard cage	X	5.0052							
	2-S	5.0152							
	60-11	5.0151							
	60-12	5.0052							

APPENDIX VII

301-10676-SERIES BEARINGS AFTER TEST



CHIPPED CONE RIB OCCURRED WHEN CAGE WAS REMOVED FROM CONE

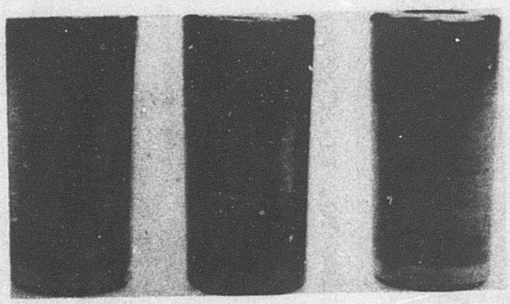
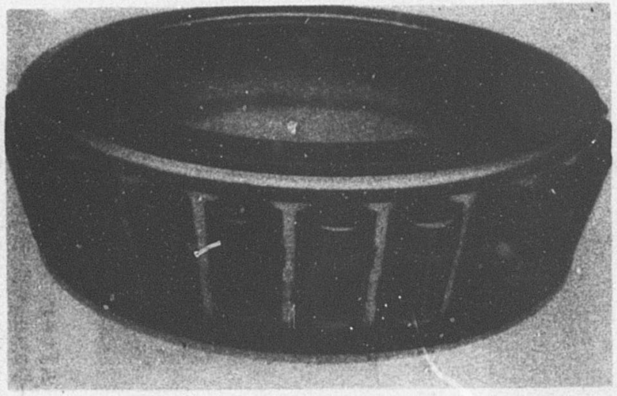
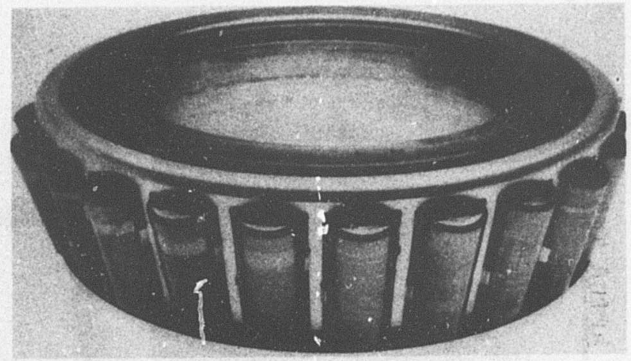


Figure 64. XC11439-Series Bearing After Test, Bearing No. 73-65, Test No. 5--Damaged Due to Large Sheets of Silver Plating Separating From the Cage.

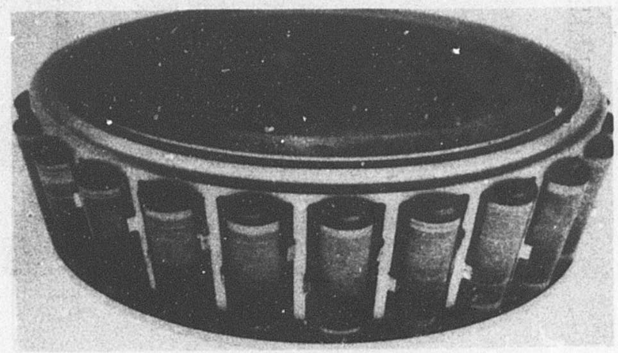


BEARING NO. 73-34 AFTER 24-1/2 HOURS



BEARING NO. 73-19 AFTER 49-1/2 HOURS

BEARING NO. 73-37 AFTER 1 HOUR



BEARING NO. 73-25 AFTER 1 HOUR

Figure 65. 301-10676-Series Bearings--Okay After Test.

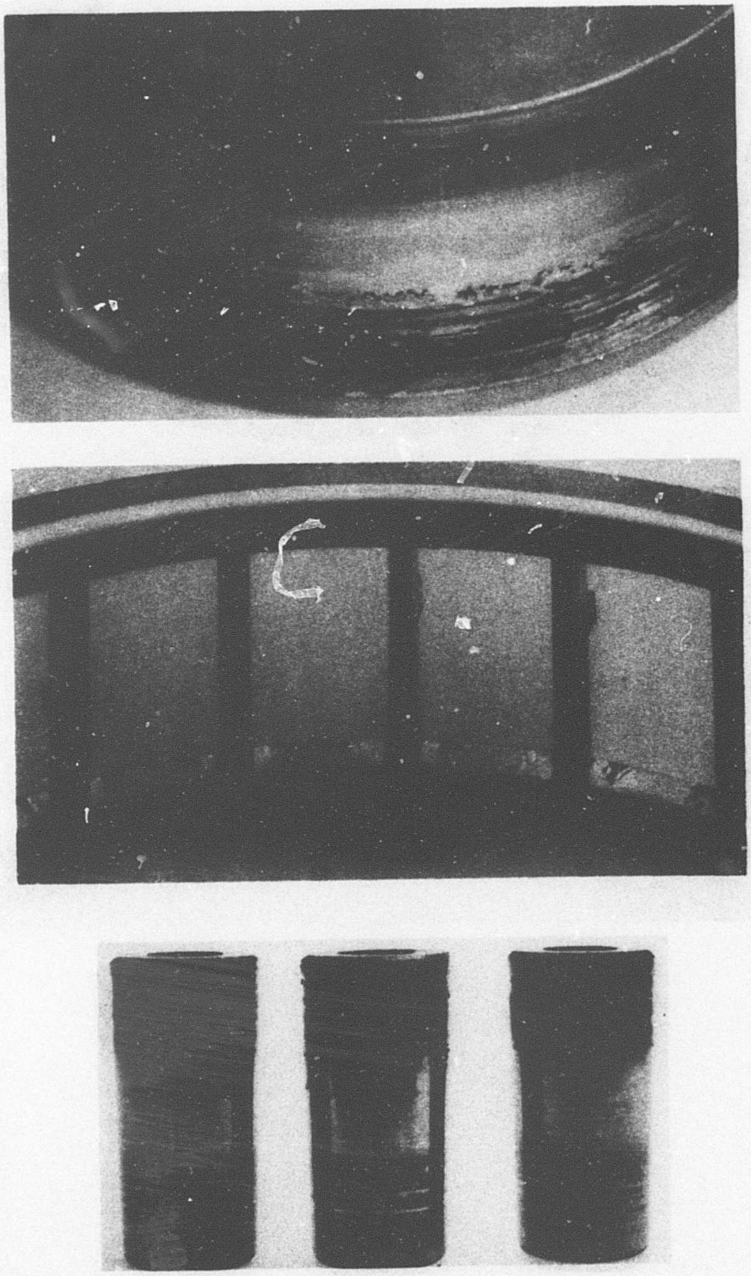


Figure 66. XC11439-Series Bearing After Test, Bearing No. 73-49, Test No. 3--Damaged Due to Insufficient Bearing End-Play Adjustment.

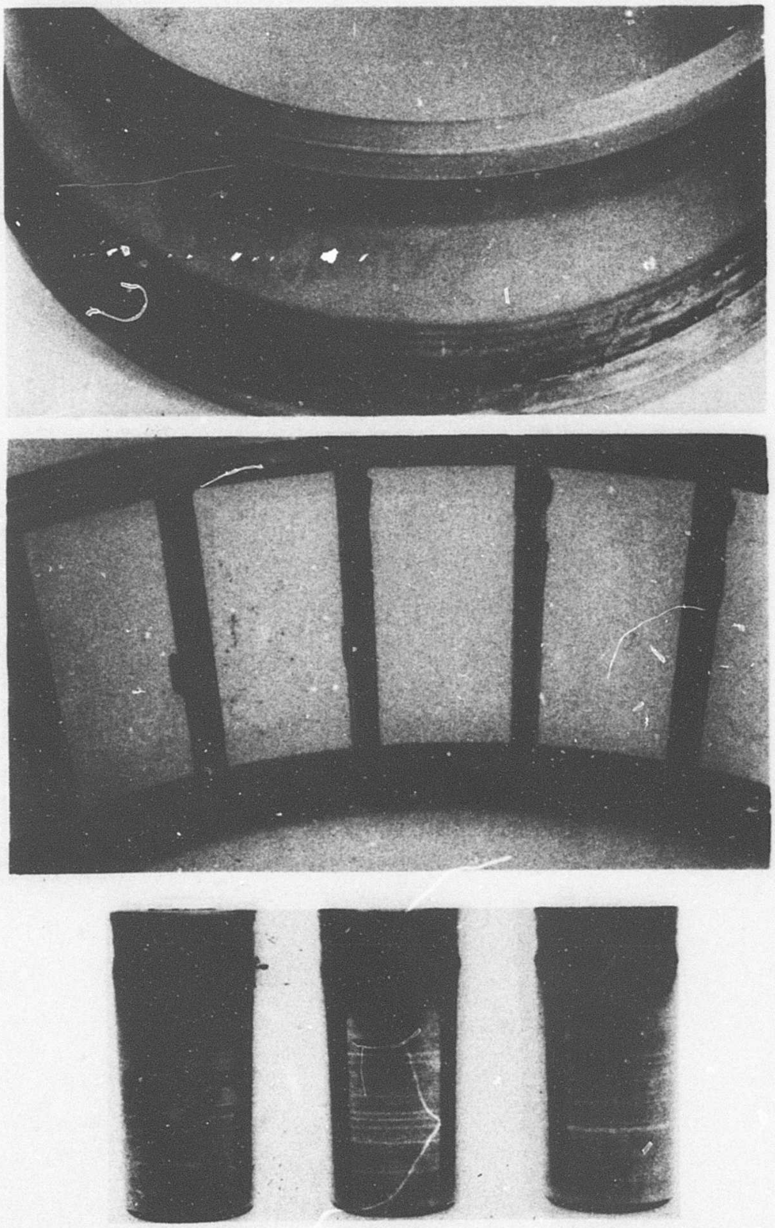


Figure 67. XC11439-Series Bearing After Test, Bearing No. 73, Test No. 6--Damaged Due to Lubricating Oil Pump Shutdown After Running 45 Minutes at 11,500 rpm.

APPENDIX VIII
FLANGE OIL DISTRIBUTION SPIN TESTS

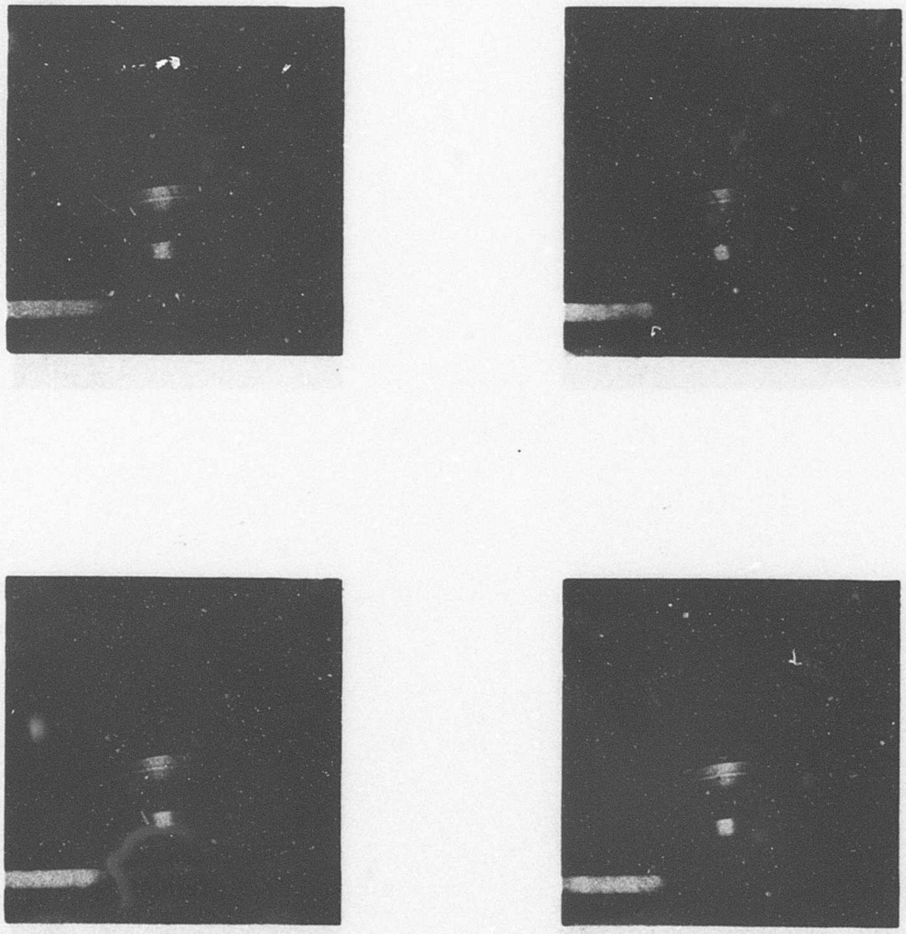


Figure 68. Simulated XC11437 Cone at 3000 rpm--Angle and Length of Oil Hole - 75°, 0.5 Inch; Oil Flow Rate - 6 Pints per Minute.

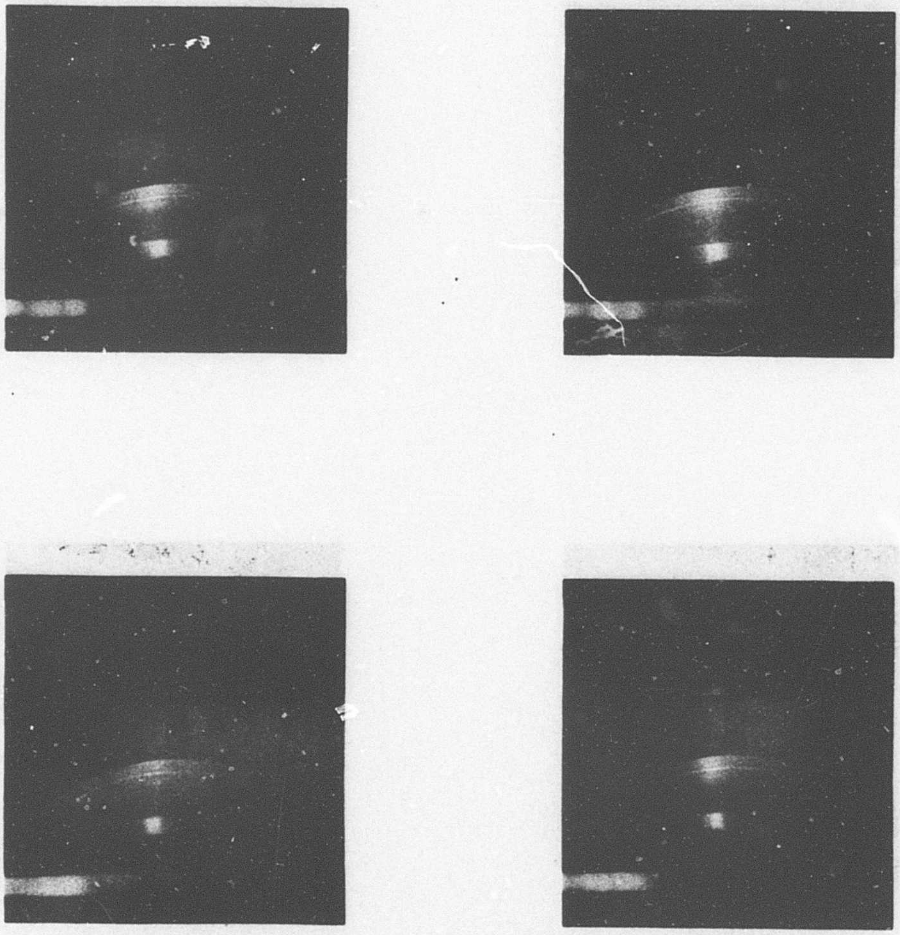


Figure 69. Simulated XC11437 Cone at 3000 rpm--Angle and Length of Oil Hole - 75°, 0.5 Inch ; Oil Flow Rate - 6 Pints per Minute.

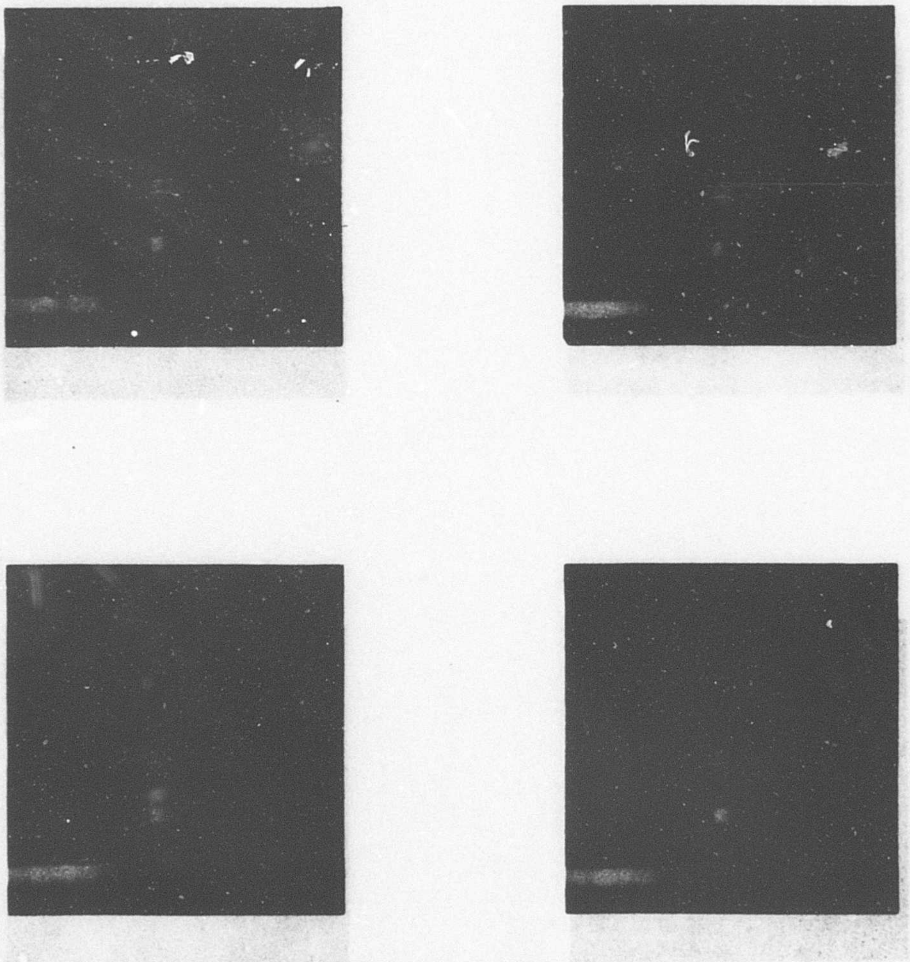


Figure 70. Simulated XC11437 Cone at 8000 rpm--Angle and Length of Oil Hole - 75°, 0.5 Inch ; Oil Flow Rate - 6 Pints per Minute.

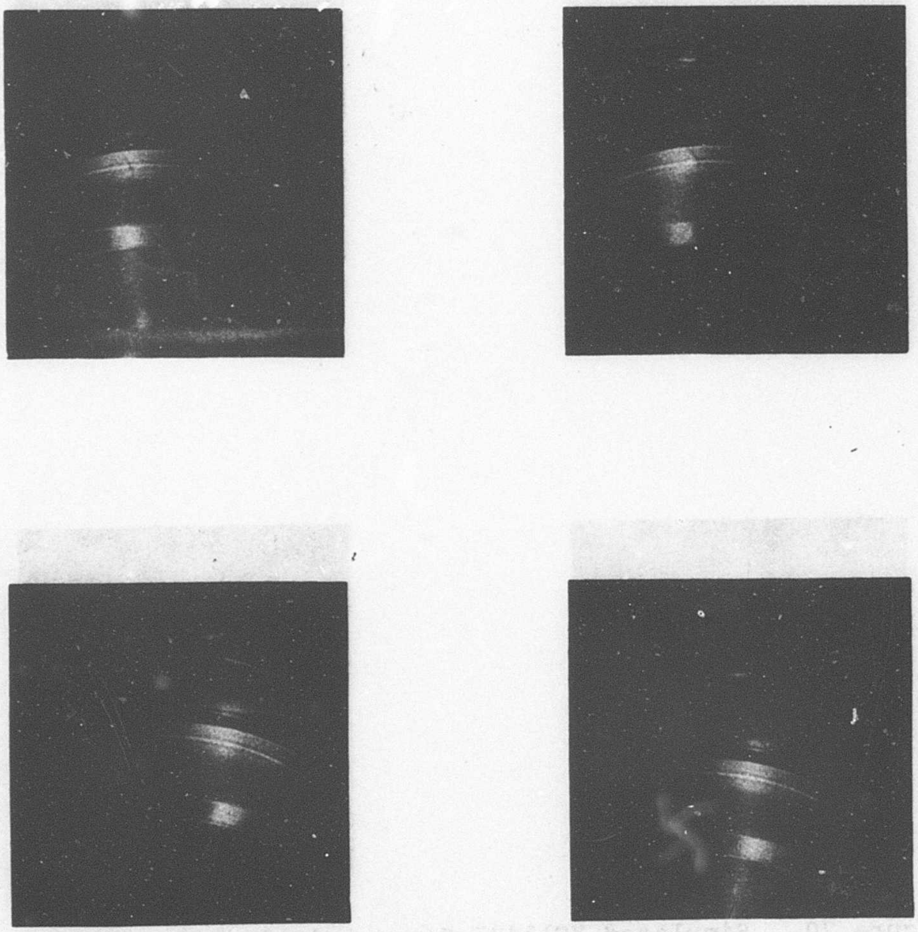


Figure 71. Simulated XC11437 Cone at 3000 rpm--Angle and Length of Oil Hole - 75°, 0.375 Inch; Oil Flow Rate - 6 Pints per Minute.

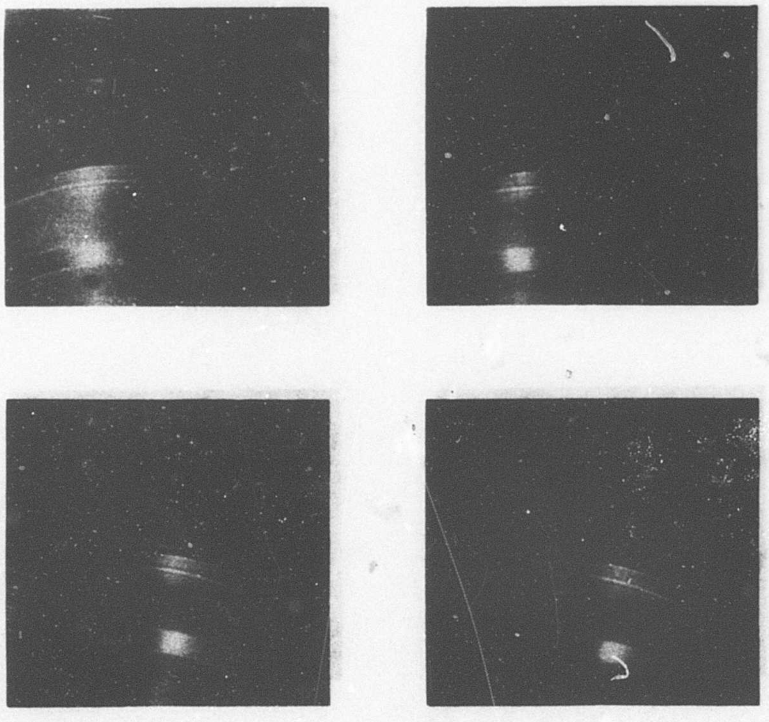


Figure 72. Simulated XC11437 Cone at 6000 rpm--Angle and Length of Oil Hole - 75°, 0.375 Inch; Oil Flow Rate - 6 Pints per Minute.

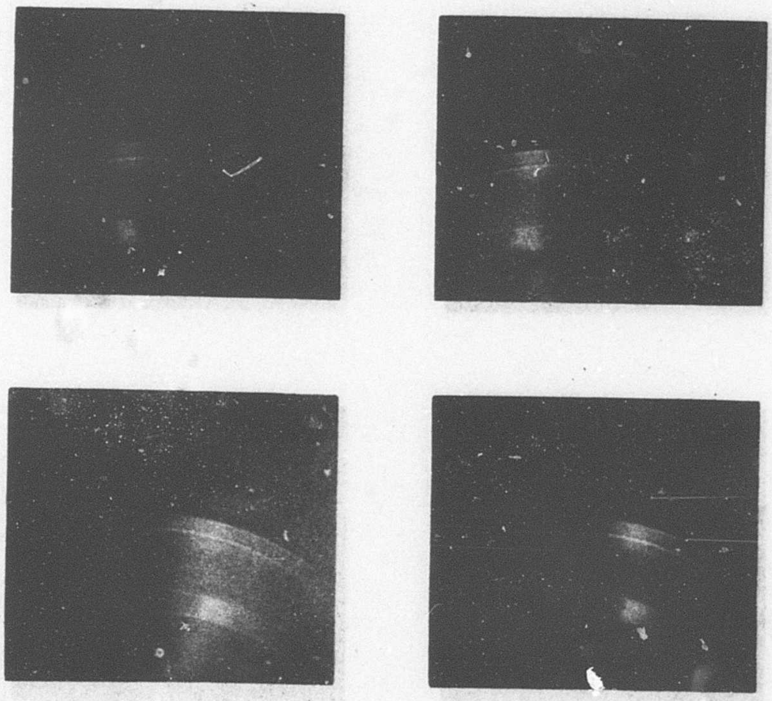
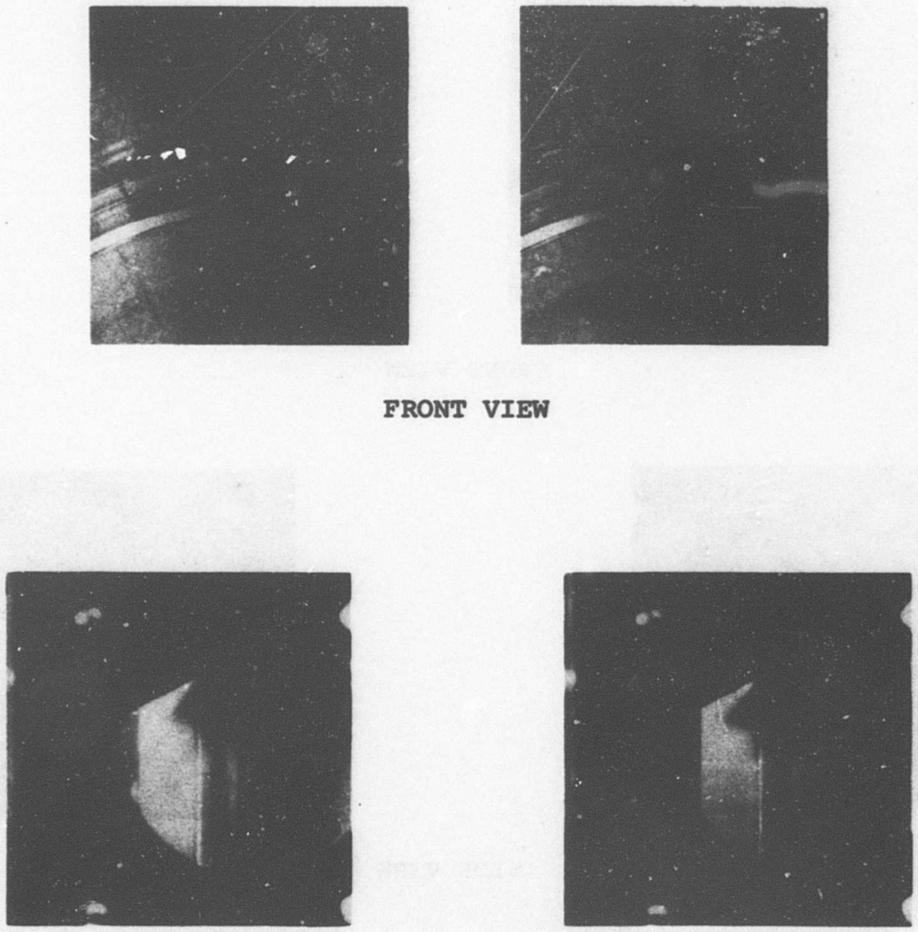


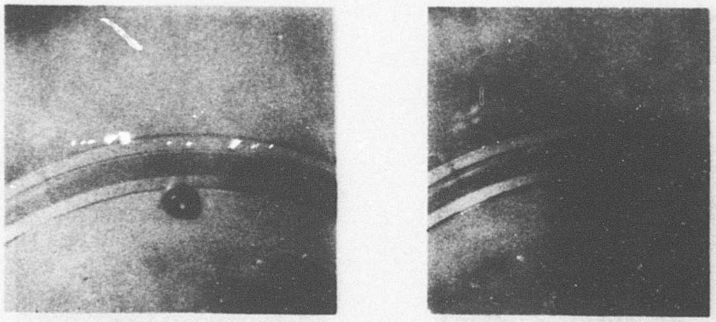
Figure 73. Simulated XC11437 Cone at 8000 rpm--Angle and Length of Oil Hole - 75°, 0.375 Inch; Oil Flow Rate - 6 Pints per Minute.



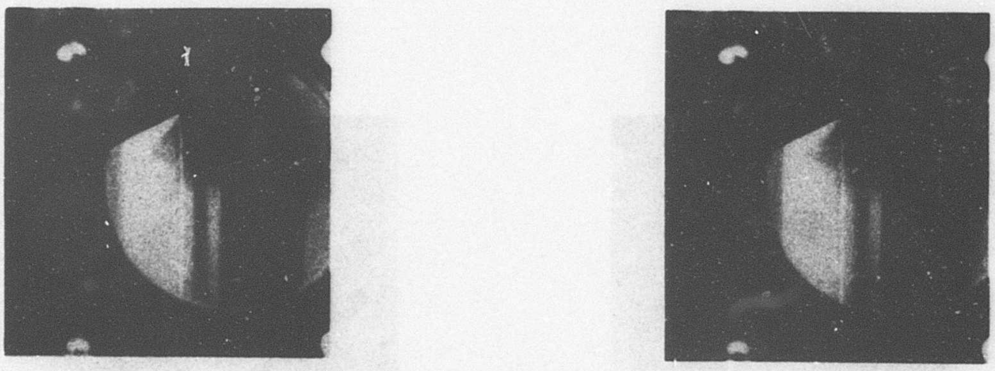
FRONT VIEW

SIDE VIEW

Figure 74. Simulated XC12088 Cone at 2000 rpm--Angle and Length of Oil Hole - 75°, 0.32 Inch; Oil Flow Rate - 3 Pints per Minute.

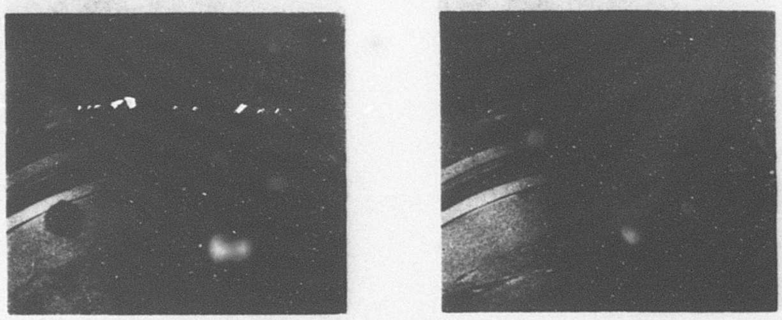


FRONT VIEW

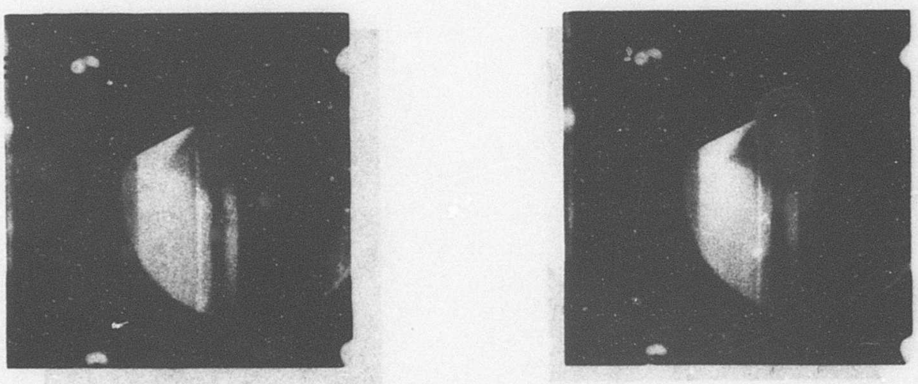


SIDE VIEW

Figure 75. Simulated XC12088 Cone at 2000 rpm--Angle and Length of Oil Hole - 75°, 0.32 Inch; Oil Flow Rate - 3 Pints per Minute.

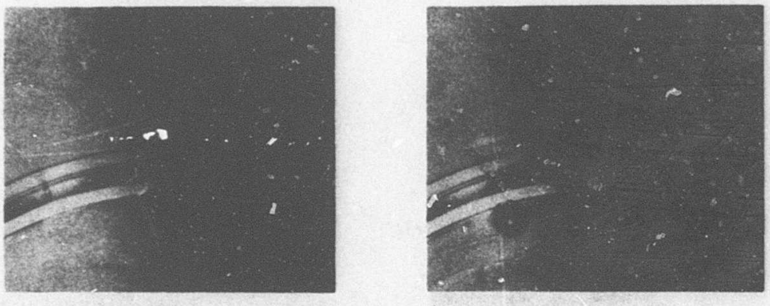


FRONT VIEW

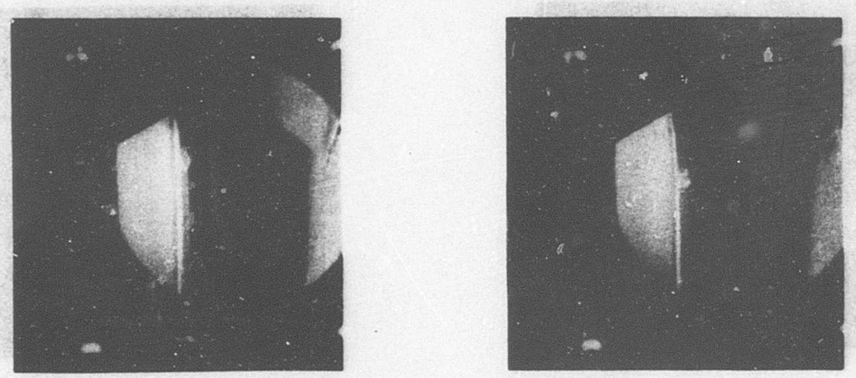


SIDE VIEW

Figure 76. Simulated XC12088 Cone at 3000 rpm--Angle and Length of Oil Hole - 75°, 0.32 Inch; Oil Flow Rate - 3 Pints per Minute.

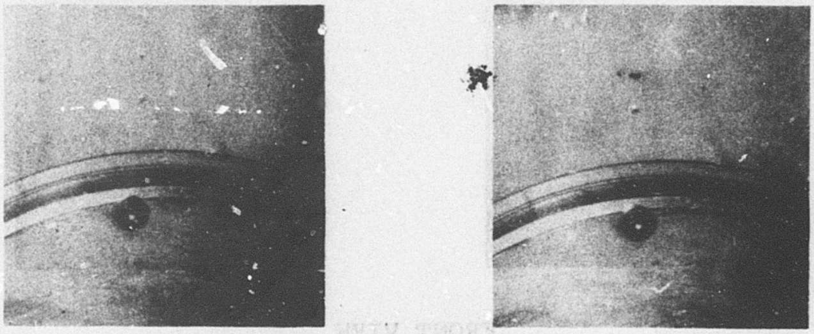


FRONT VIEW

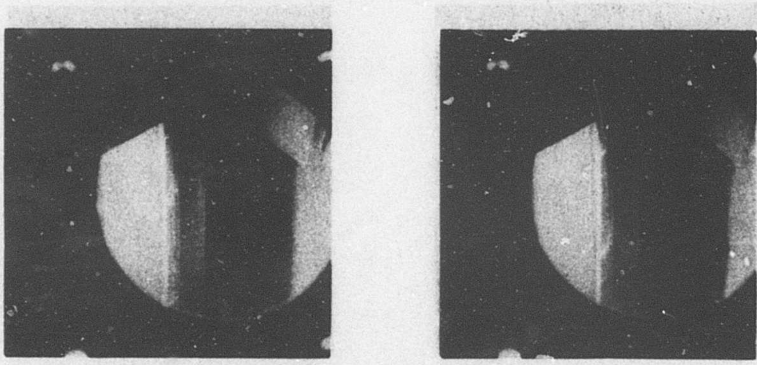


SIDE VIEW

Figure 77. Simulated XC12088 Cone at 1000 rpm--Angle and Length of Oil Hole - 75°, 0.243 Inch; Oil Flow Rate - 3 Pints per Minute.

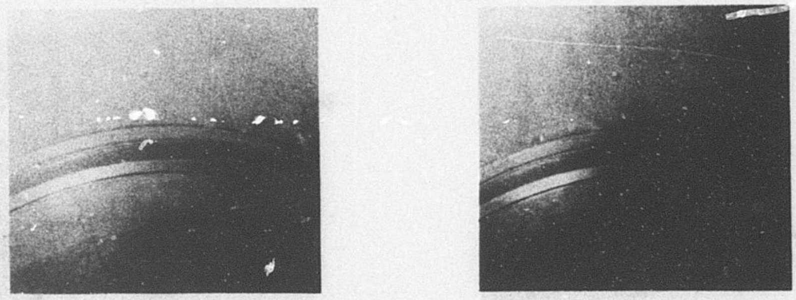


FRONT VIEW

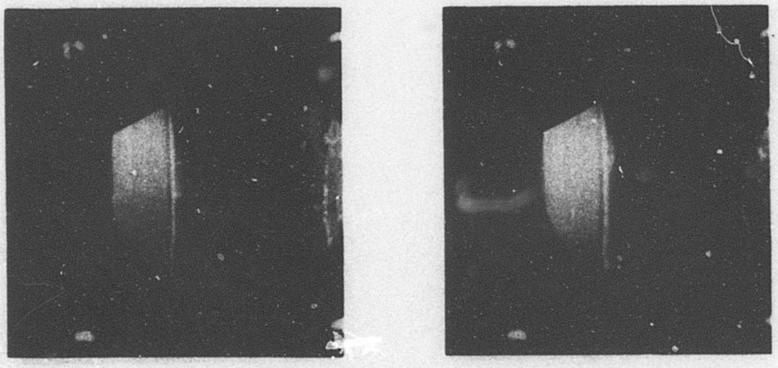


SIDE VIEW

Figure 78. Simulated XC12088 Cone at 2000 rpm--Angle and Length of Oil Hole - 75°, 0.243 Inch; Oil Flow Rate - 3 Pints per Minute.

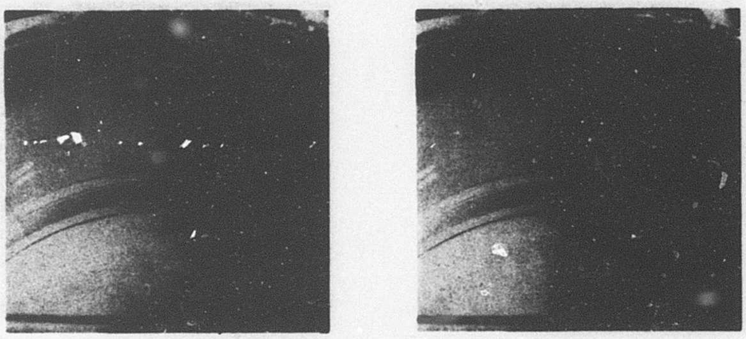


FRONT VIEW

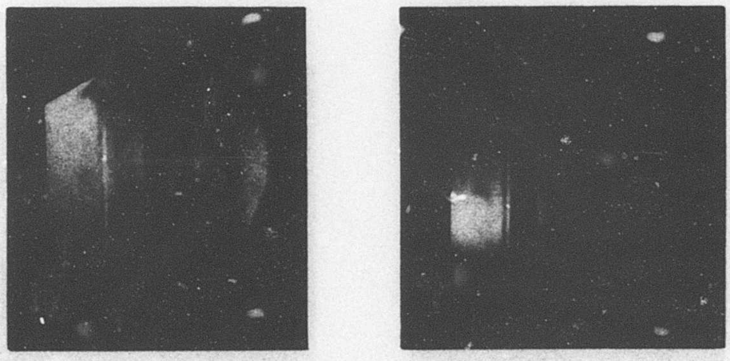


SIDE VIEW

Figure 79. Simulated XC12088 Cone at 3000 rpm--Angle and Length of Oil Hole - 75°, 0.243 Inch; Oil Flow Rate - 3 Pints per Minute.

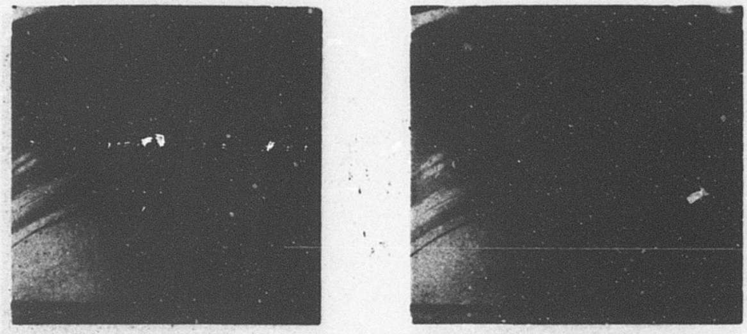


FRONT VIEW

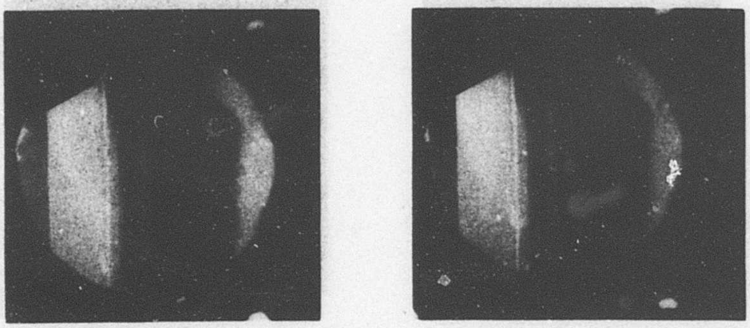


SIDE VIEW

Figure 80. Simulated XC12088 Cone at 1000 rpm--Angle and Length of Oil Hole - 75°, 0.225 Inch; Oil Flow Rate - 3 Pints per Minute.

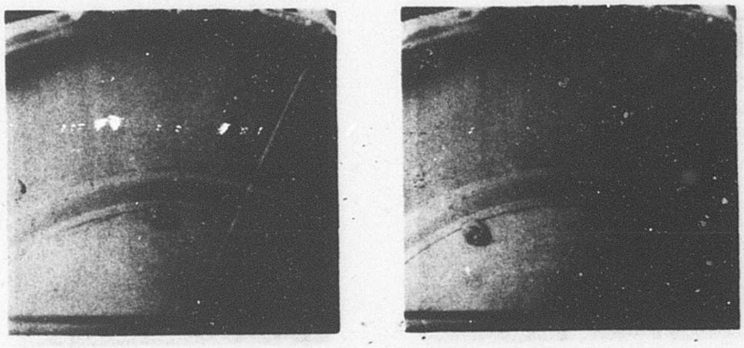


FRONT VIEW

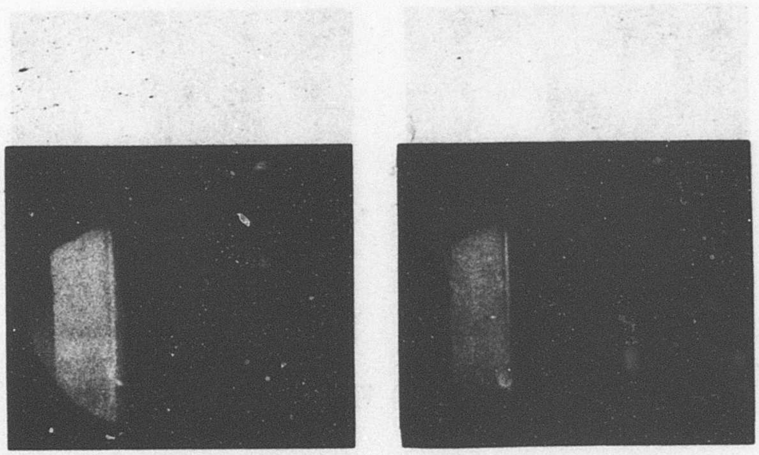


SIDE VIEW

Figure 81. Simulated XC12088 Cone at 2000 rpm--Angle and Length of Oil Hole - 75°, 0.225 Inch; Oil Flow Rate - 3 Pints per Minute.



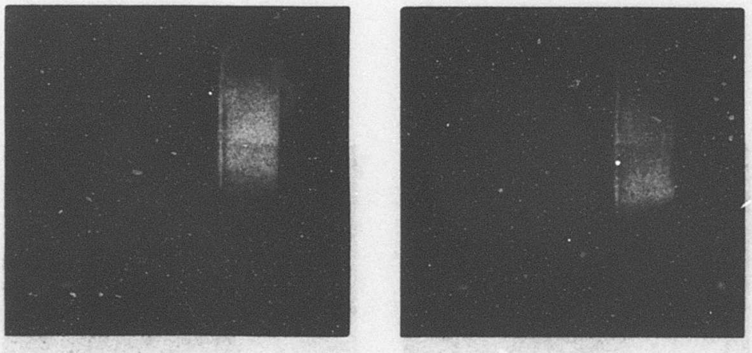
FRONT VIEW



SIDE VIEW

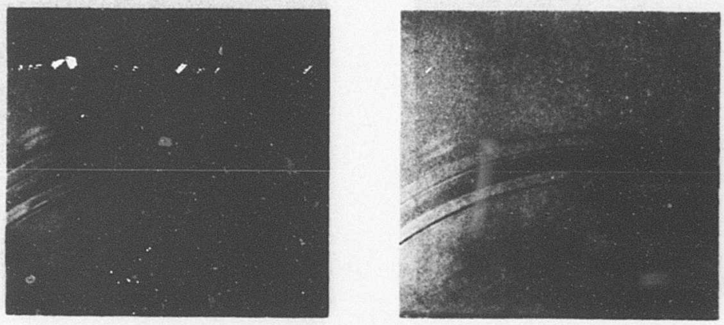
Figure 82. Simulated XC12088 Cone at 3000 rpm--Angle and Length of Oil Hole - 75°, 0.225 Inch; Oil Flow Rate - 3 Pints per Minute.

FRONT VIEW

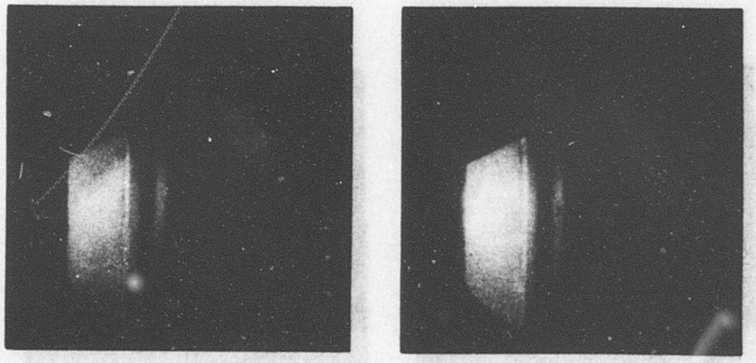


SIDE VIEW

Figure 83. Simulated XC12088 Cone at 1000 rpm--Angle and Length of Oil Hole - 1°, 0.225 Inch; Oil Flow Rate - 3 Pints per Minute.

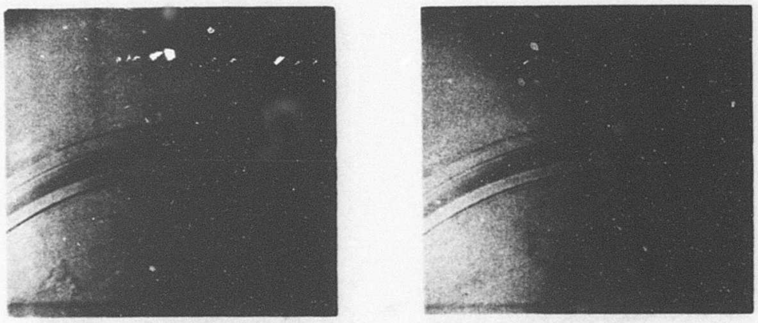


FRONT VIEW

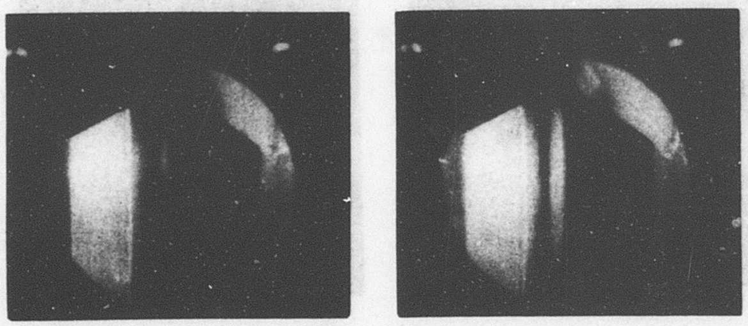


SIDE VIEW

Figure 84. Simulated XC12088 Cone at 2000 rpm--Angle and Length of Oil Hole - 1°, 0.225 Inch; Oil Flow Rate - 3 Pints per Minute.



FRONT VIEW



SIDE VIEW

Figure 85. Simulated XC12088 Cone at 3000 rpm--Angle and Length of Oil Hole - 1°, 0.225 Inch; Oil Flow Rate - 3 Pints per Minute.

APPENDIX IX
TEST RESULTS OF HIGH-SPEED TAPERED ROLLER BEARING ENDURANCE TEST PROGRAM

BACKGROUND

Four bearings which completed 150 hours of testing at 11,500 rpm and 100% load were inspected and determined to be acceptable for additional testing. An additional test was conducted for 850 hours at 11,500 rpm and 100% HLH engine input pinion loads in order to establish fatigue or endurance-type data on high-speed tapered roller bearings.

The bearings tested conformed to Figure 14. The component serial numbers, bearing position in the test rig, and actual HLH/ATC test time are shown on Table VI.

TEST EQUIPMENT AND PROCEDURE

The basic test equipment used is shown in Figures 15 and 16. The test rig is capable of testing two sets of tapered roller bearings (Figure 17) under conditions which simulate the speeds and loads of the HLH combiner transmission input pinion bevel gear. The operating conditions were as follows:

Speed	11,500 rpm
Radial Load	11,400 lb
Thrust Load	4200 lb
Lubricant	MIL-L-23699
Oil-In Temperature	190°F

TEST RESULTS

The test bearings identified in Table I were assembled into the test rig with the following initial end-play settings.

- Drive End Housing Bearings - .0047 inch
- Opposite Drive End Housing Bearings - .0042 inch

After completion of the test rig assembly, testing was started and was to continue for a maximum of 850 hours or until the bearing failed. If a bearing failure occurred before the completion of the 850 hours, the failed bearing(s) were to be replaced up to a maximum of four bearing failures.

During the 850-hour test, the test machine experienced a total of 20 shutdowns as a result of test machine related problems. Also during this same period, several large variations of test loads occurred for short periods of (less than 5 minutes) operation. At five different times, the radial and thrust load dropped to zero and on nineteen other times, the radial and thrust loads dropped off to approximately 60% to 70% of desired values.

All load variations occurred while the bearings were operating at full speed (11,500) and had no significant effect on bearing performance. One overload of approximately 125% also occurred during this test.

Oil inlet temperature was maintained at 190°F for all testing. Bearing outer race temperatures were recorded continuously during the program and showed very little variation. Table VII shows typical temperature readings of cup outer diameter, oil outlet, and housing outer diameter for each bearing. Slight variations in temperature were mainly the result in ambient temperature variations.

All testing was terminated at the end of 1041 hours (total HLH/ATC and HLH R&M test time) test time on the highest time bearing. The last 891 hours of testing were completed without any replacement of bearings due to fatigue or excessive wear. Prior to disassembly of the bearings for inspection, the final end-play settings were measured:

- Drive End Housing Bearings - .0025 inch
- Opposite Drive End Housing Bearings - .0015 inch

The bearings were disassembled and visually inspected for any evidence of fatigue or wear. Figures 86, 87, 88 and 89 show the overall condition of the bearings prior to disassembly of the cage and rollers from the cone. Table XIV provides a summary of the visual examination of all components of the four test bearings. In addition, Figures 90 through 105 show photographs of the condition of each component at the completion of the test. All cups of the test bearings had slight evidence of the initiation of surface damage in the form of small surface pits and some peeling. The pits were smaller than .01 square inch in area. The peeling or superficial surface distress is a mode of damage associated with marginal oil film thickness which usually occurs around bruises, grooves or ends of cup-roller or cone-roller contacts. This type of bearing damage has also been observed on CH-46 and CH-47 transmission bearings using synthetic lubricants without major influence on bearing removals. The peeling observed in the test bearings appeared to be localized in areas of high stress. Similar type of peeling damage was noted near the under cut groove and cone-roller contact area on three of the four cones. One roller from bearing serial number 73-69/73-27 also showed several tiny surface pits which were less than .01 square inch.

Table XV provides a summary of the machine loads and the resultant individual bearing radial and thrust loads. Also shown are the calculated test bearing B-10 lives, cone rib stresses, equivalent aircraft hours at cubic mean load, and the minimum oil film thickness for various component surfaces. The life of the center bearings exceeded the calculated life by approximately 2.5 times. The data generated during this program is insufficient to accurately evaluate the fatigue life properties of high-speed tapered roller bearings. The data appears to indicate that these bearings can be operated for long periods of time without major fatigue distress.

The conditions of all four bearings were very similar and mainly consisted of several small pits (less than .01 square inch in area), peeling, and light debris damage. This damage appears to be associated with thin oil films in areas of high load concentration.

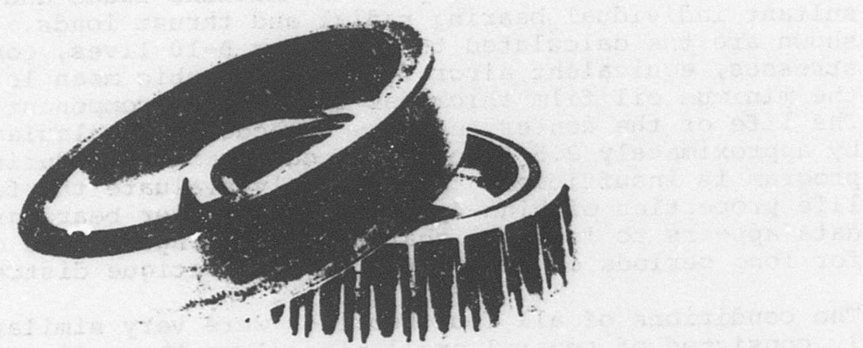


Figure 86. Drive End Bearing S/N 73-50 After Test.

Figure 87. Drive End Center Bearing S/N 73-59 After Test.

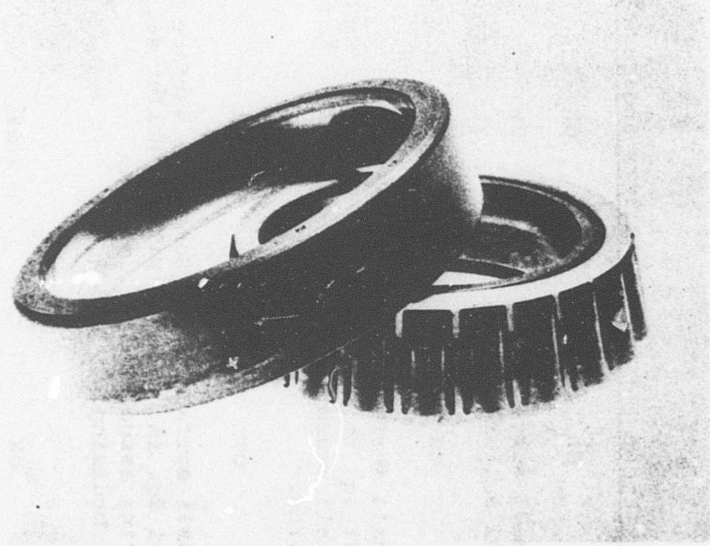


Figure 88. Opposite Drive End Center Bearing S/N 73-69 After Test.

Figure 89. Opposite Drive End Bearing S/N 73-73 After Test.

TABLE XIV. SUMMARY OF BEARING CONDITION AFTER ENDURANCE TEST

Test Conditions: Speed - 11,500 rpm
Oil Flow Rate - 10 pts/min to L.E. (large end)
6 pts/min to S.E. (small end)
Inlet Oil Temperature - 190°F

Bearing Cone/Cup Number	Bearing Position	Hours Run	Cone	Cup	Rollers and Cage
73-50/ 73-4	DE	126.75	5 tiny surface pits	OK	OK
	DE	1018	No change in size of pits plus a few more.	1 small surface pit $<$.01 sq. in.	OK
73-59/ 73-10	DEC	150	OK	OK	OK
	DEC	1041	Peeling at L.E. race. Cone turned on shaft.	1 small surface pit $<$.01 sq. in. 2 small areas of peeling.	OK
73-69/ 73-27	ODEC	150	OK	OK	OK
	ODEC	1041	Several tiny surface pits caused by pit on roller #6. Some peeling adjacent to pits.	2 small surface pits $<$.01 sq. in. Some peeling associated with bruising.	Some tiny surface pits on Roller #6. Cage OK.
73-73/ 73-22	ODE	150	OK	OK	CK
	ODE	1041	Very narrow band of peeling at L.E.	3 small surface pits $<$.010 sq. in.	OK

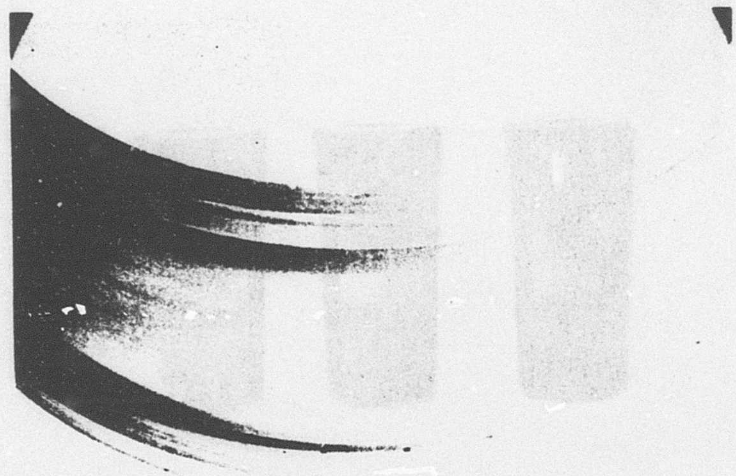


Figure 90. View of Cone Race Surface - Bearing S/N 73-50.

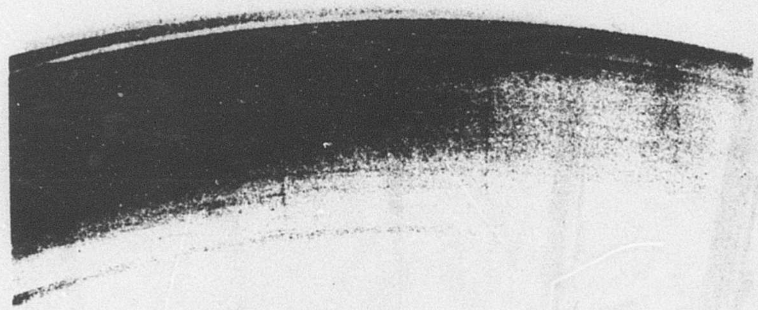


Figure 91. View of Cup Race Surface - Bearing S/N 73-50.

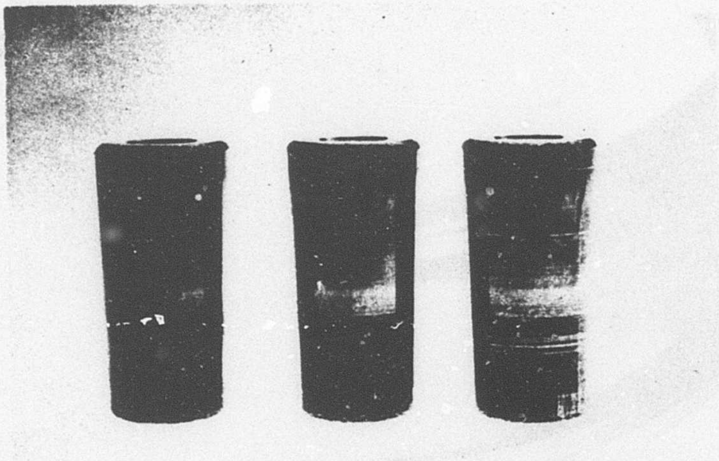


Figure 92. View of Rollers - Bearing S/N 73-50.

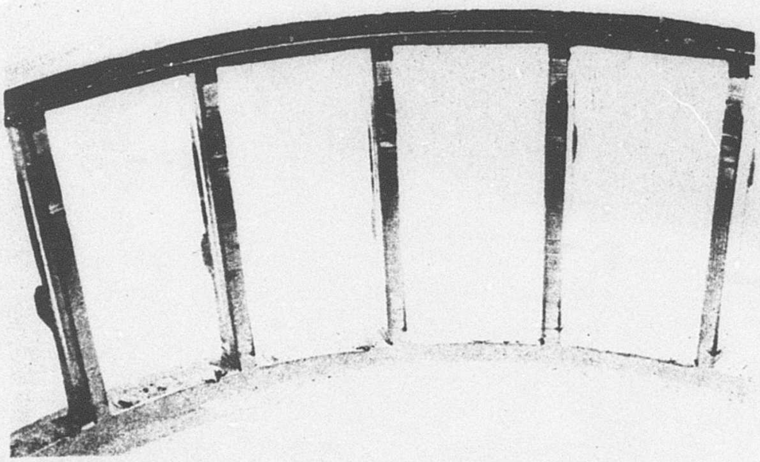


Figure 93. View of Cage Pockets - Bearing S/N 73-50.

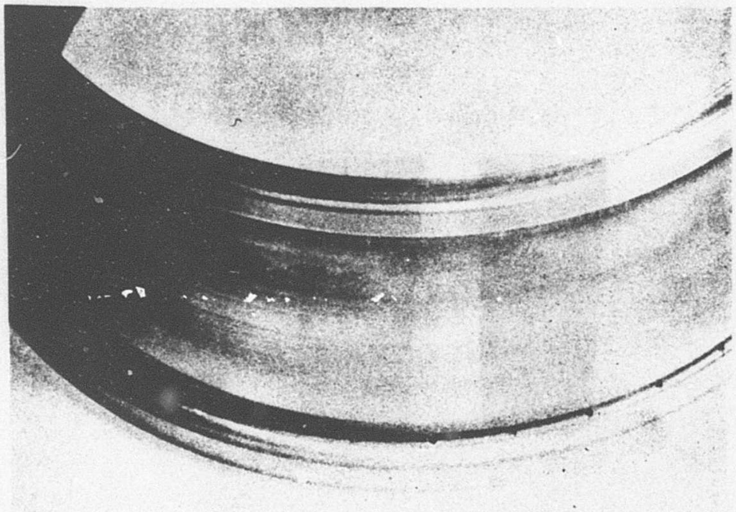


Figure 94. View of Cone Race Surface - Bearing S/N 73-59.

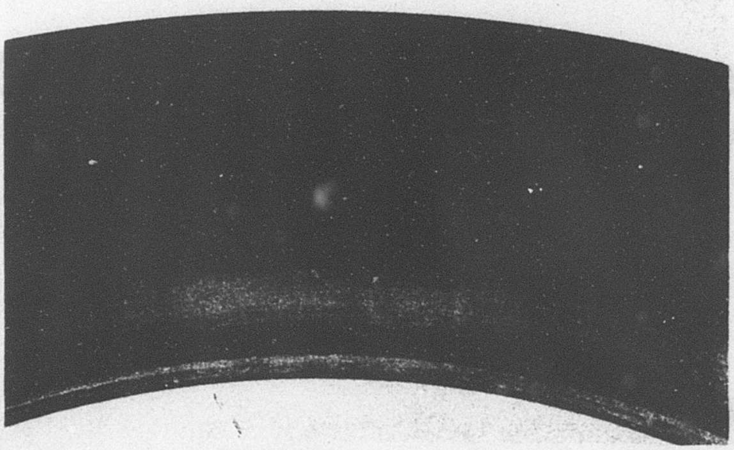


Figure 95. View of Cup Race Surface - Bearing S/N 73-59.

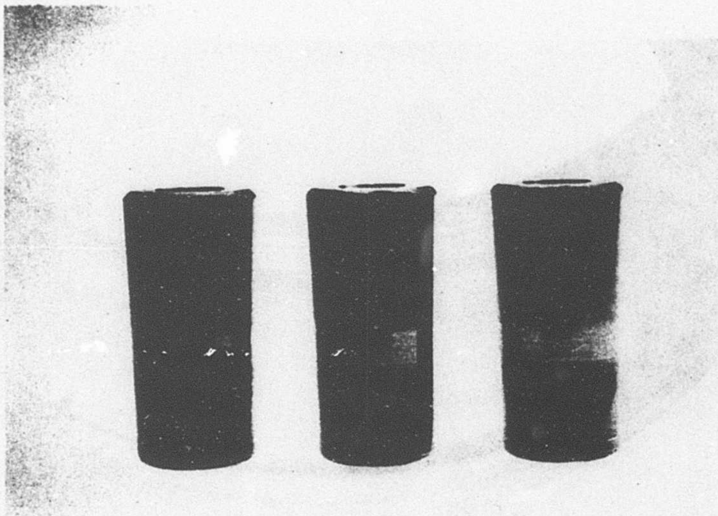


Figure 96. View of Rollers - Bearing S/N 73-59.

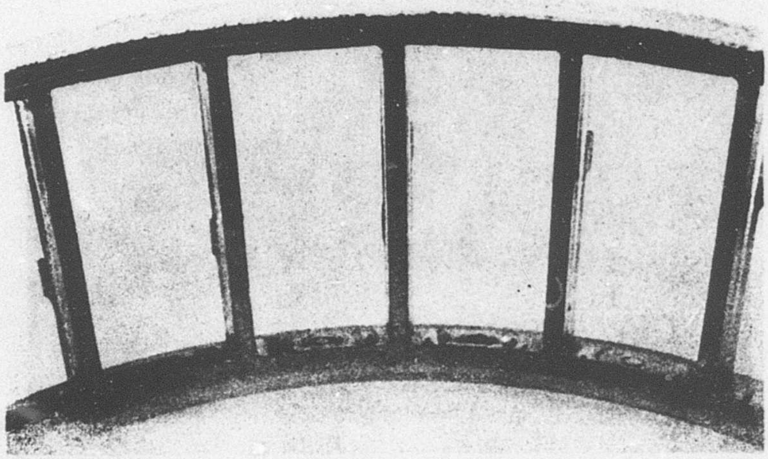


Figure 97. View of Cage Pockets - Bearing S/N 73-59.

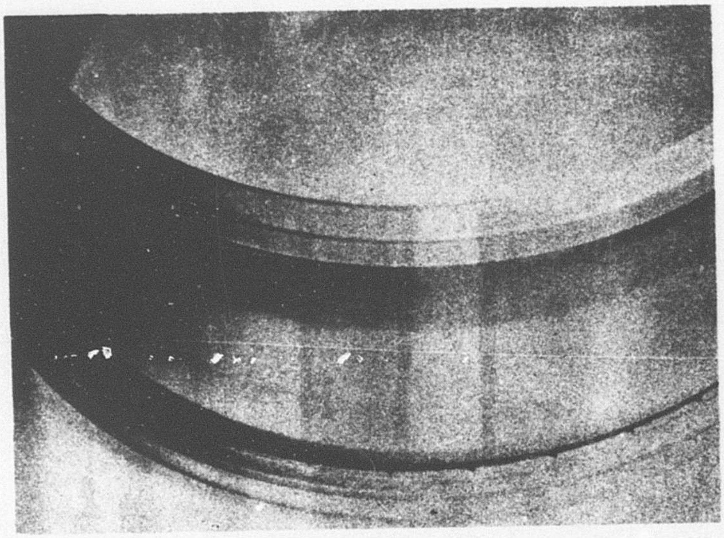


Figure 98. View of Cone Race Surface - Bearing S/N 73-69 (Note: Debris Damage on Race Surface).

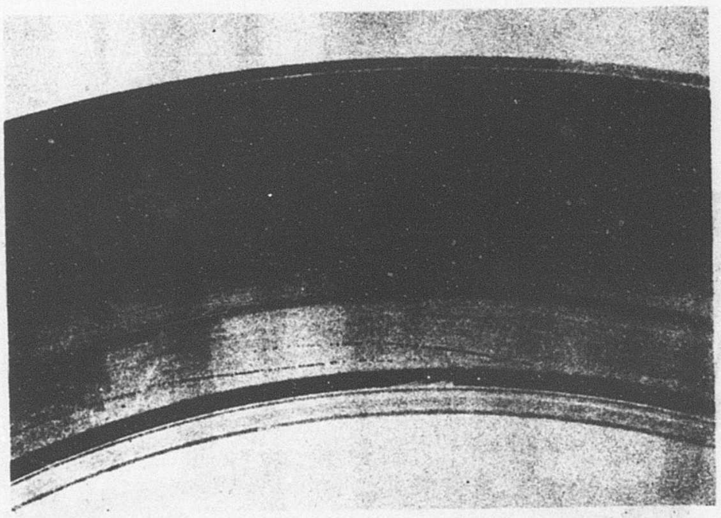


Figure 99. View of Cup Race Surface - Bearing S/N 73-69 (Note: Debris Damage on Race Surface).

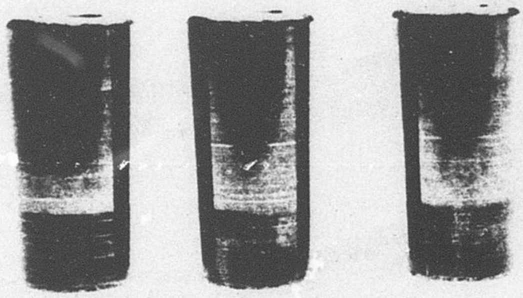


Figure 100. View of Rollers - Bearing S/N 73-69
(Note: Several Pits on Center Roller).

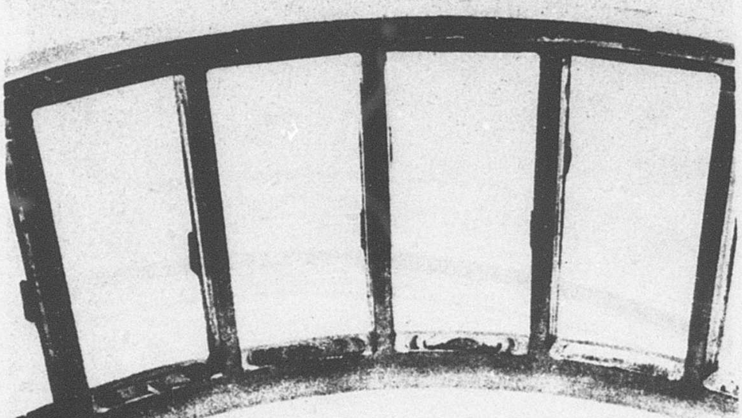


Figure 101. View of Cage Pockets - Bearing S/N 73-69.

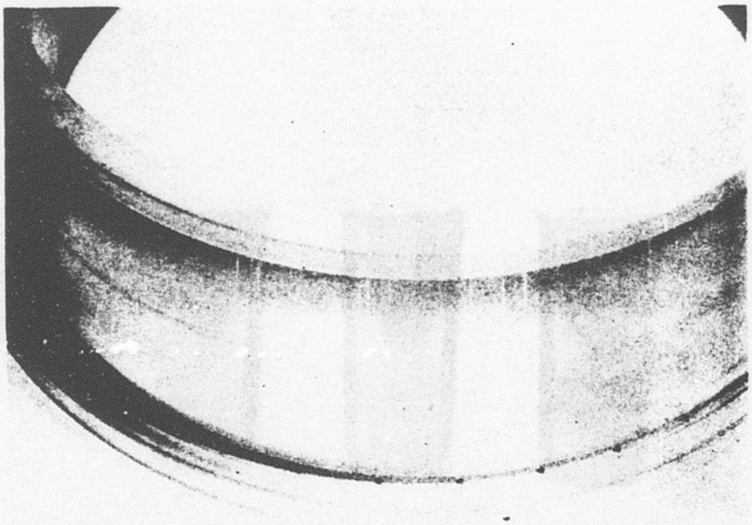


Figure 102. View of Cone Race Surface - Bearing S/N 73-73.

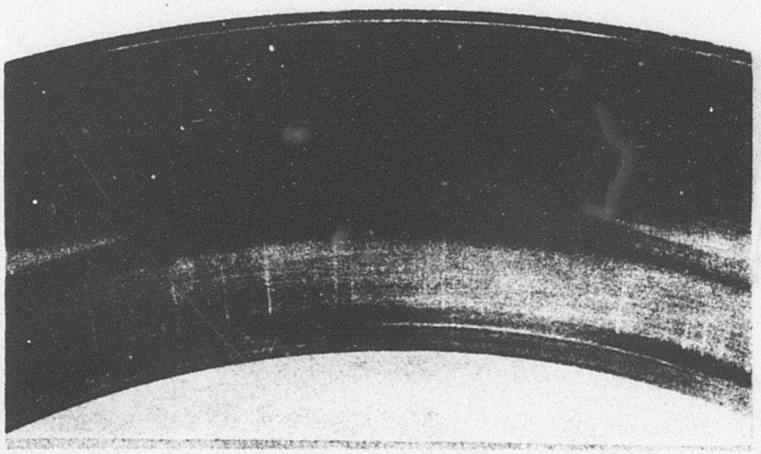


Figure 103. View of Cup Race Surface - Bearing S/N 73-73.

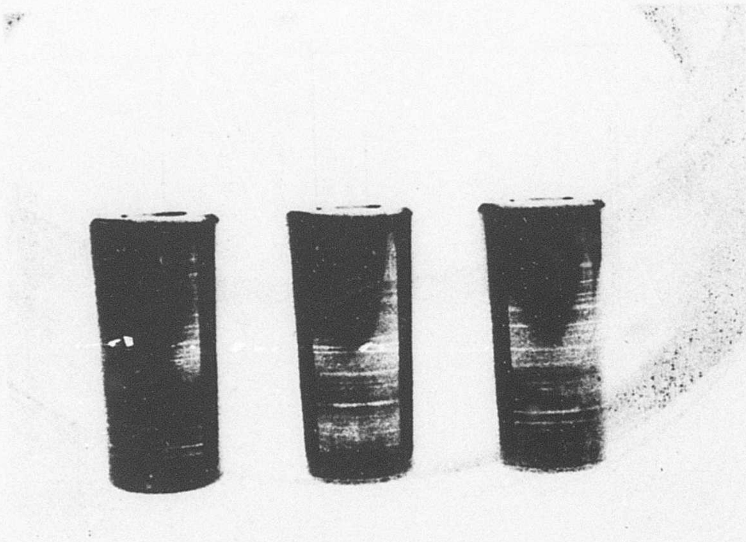


Figure 104. View of Rollers - Bearing S/N 73-73.

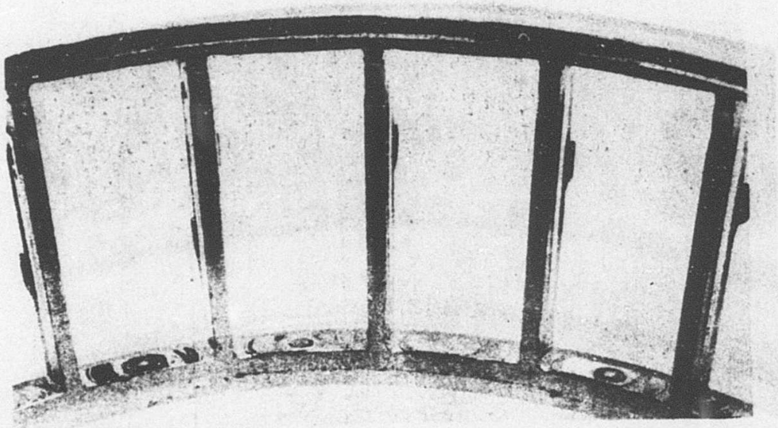


Figure 105. View of Cage Pockets - Bearing S/N 73-73.

TABLE XV. SUMMARY OF BEARING LOADS, B-10 LIFE AND OIL FILM THICKNESS FOR TEST BEARINGS

Bearing Position	Bearing Load Radial/Thrust (lb) *	Calculated Life** Hours/B-10	Cone Rib Stress PSI	Aircraft Hours Equivalent To Test Hours at Cubic Mean Power	Calculated Min. Film Thickness (275°F)		
					Cone/Roller	Cup/Roller	Roller/Cone Rib
DE & ODE	5053/4956	5547	29,100	307	14.6×10^{-6}	15.0×10^{-6}	21.4×10^{-6}
DEC & ODEC	16252/9156	420	32,773	5413	12.6×10^{-6}	13.5×10^{-6}	20.1×10^{-6}

* Machine loads are equivalent to 100% HLH single engine power of 8310 horsepower at 11,500 rpm, 11,400 lb radial load, and 4,200 lb thrust load.

** Considering centrifugal force and 3X for CEVM. B-10 life calculated using Boeing Vertol Computer Program S73.

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