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ADVANCED TRANSMISSION COMPONENTS INVESTIGATION PROGRAM — BEARING AND SEAL DEVELOPMENT

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August 1980

Final Report for Period June 1976 - December 1979

Approved for public release;
distribution unlimited.

Prepared for

APPLIED TECHNOLOGY LABORATORY
U. S. ARMY RESEARCH AND TECHNOLOGY LABORATORIES (AVRADCOM)
Fort Eustis, Va. 23604

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APPLIED TECHNOLOGY LABORATORY POSITION STATEMENT

The purpose of the program was to evaluate a modified tapered roller bearing component incorporating a VASCO-X2 integral inner race and ribbed cup for use on the spiral bevel input shaft of an advanced helicopter main transmission. The test results indicated that this bearing concept, with its through-shaft lubrication and magnetic shaft seal, is feasible for use in such an application and will aid in improving reliability and system weight in future transmissions. The limited oil-off survivability testing conducted did not produce expected results; however, it showed that this type of testing requires a more realistic test rig environment, to include mounting system. A follow-on report (to be published in FY 81) evaluates a composite material helicopter engine transmission housing.

Appropriate technical personnel of this Laboratory have reviewed this report and concur with the conclusions and recommendations contained herein.

Mr. James Gomez, Jr., of the Propulsion Technical Area, Aeronautical Technology Division, served as the Project Engineer for this effort.

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The objective of the Advanced Transmission Components Investigation Program was to conduct development testing on selected critical components for a helicopter advanced transmission which could enter engineering development in the 1980-90 timeframe. Under this contract, advanced transmission components have been designed, analyzed, and tested with the aim of reducing drive system weight and cost, increasing reliability, and improving other important attributes. The work reported in this volume includes the evaluation of VASCO-X2 steel as a bearing material, design and test of a ribbed-cup tapered-roller bearing with integrated inner race and shaft, test of a magnetic seal design, and development of an advanced finite-element analysis of complex bearing structure. The aims and objectives of each

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20. ABSTRACT (continued)

component development have been formulated and are compared against test results to develop a measure of improvement to be expected when these developments become operational.

PREFACE

This report summarizes the results of Task II and III of the "Advanced Transmission Components Investigation Program." Task II covers work done on bearing development and Task III covers work on seal development. The report describes the work accomplished during the 42-month period from June 28, 1976, to December 1979. Task I involves the advanced-composite engine transmission housing.*

The work outlined here has been performed under U.S. Army contract DAAJ02-76-C-0045 and under the technical cognizance of James Gomez, Applied Technology Laboratory, U.S. Army Research and Technology Laboratories, Fort Eustis, Virginia.

This program was conducted at the Boeing Vertol Company under the technical direction of Joseph W. Lenski, Jr. (Program Manager), of the Advanced Power Train Technology Department. Principal investigators for this program were John Mack (Project Engineer) and Fred Brown (Analysis).

Acknowledgment is made to Arthur Irwin and Harold Munson of TRW, Marlin Rockwell Division, Jamestown, New York, for their technical assistance in the fabrication and testing of VASCO-X2 ball bearings.

Acknowledgment is also made to Pete Orvos and Gary Dressler of the Timken Company, Canton, Ohio, for their technical assistance in the design, fabrication, and test of the tapered-roller bearings reported in this program.

*Lenski, Jr., Joseph W., ADVANCED TRANSMISSION COMPONENTS INVESTIGATION PROGRAM ADVANCED COMPOSITE ENGINE TRANSMISSION HOUSING, Boeing Vertol Co., USAAVRADCOM Technical Report, Applied Technology Laboratory, U. S. Army Research and Technology Laboratories (AVRADCOM), Fort Eustis, Virginia (to be published).

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INTRODUCTION

In 1975 a proposal was submitted to the Applied Technology Laboratory (ATL) entitled, "Helicopter Transmission Components Development and Test Program," by the Boeing Vertol Company's Advanced Power Train Technology and Drive System Design organizations pursuant to ATL contract DAAJ02-75-C-0022. The major objective of this proposed program was to provide improved helicopter transmission component technology for integration into an advanced-technology demonstrator helicopter drive system in the 1980-90 time frame.

The purpose of this work was to conduct development testing on selected critical components for a helicopter advanced transmission which could enter engineering development in the 1980-90 time frame. The design goals for the complete transmission system were established as follows:

- Weight 20-percent decrease
- Reliability and maintainability 3,000 hours MTBR minimum for scheduled and unscheduled removal
- Vulnerability Withstand 12.7-mm API impacts at 200 yards and fragments emanating from a functioned 23-mm HEI hit; decreased vulnerable area of the transmission shall also be a design target
- Survivability Operate at the gearbox torque limit without main gearbox lubricant for not less than 30 minutes and to design the transmission to MIL-STD-1290 (AV), paragraph 5.1.7.2
- Producibility 20-percent improvement in recurring production cost over contemporary transmissions

The individual transmission component design goals for the development work to be accomplished are as follows:

Housing 10-percent reduction in weight, 25-percent reduction in vulnerable area, 10-percent reduction in acquisition cost, and ability to provide improvements in MTBR and survivability

Bearings (input pinion) 5-percent reduction in weight, 65-percent improvement in MTBR, 4-percent reduction in vulnerable area, 5-percent reduction in acquisition cost, and ability to provide improvement in survivability

Seal 5-percent improvement in MTBR.

The Boeing Vertol Company's approach to the improvement of helicopter component technology included considerations of advanced design analysis, design, and fabrication techniques; advanced gear and bearing materials, new-concept composite housing materials and design, planet carrier rotor shaft, ring gear, rotor support bearing design; advanced ribbed-cup tapered-roller bearings for support of bevel pinion gears and shaft magnetic seals to achieve decreased weight and vulnerability, better integration characteristics, and increased efficiency, reliability, maintainability, and service life. Careful consideration was also given to the assessment of production and life-cycle costs for the component improvement approaches taken which will maximize the reliability, maintainability, weight, and performance in an integrated system.

In June of 1976 a 40-month contract (DAAJ02-76-C-0045) was awarded to Boeing Vertol entitled, "Advanced Transmission Components Investigation Program." At the same time, two additional contracts were awarded to the Bell Helicopter Company and Sikorsky Aircraft to conduct similar work.

To improve upon recently designed, highly efficient main helicopter transmissions, it was determined that minor improvements in component design would not meet the established design goals. Therefore, a review of the major transmission/rotor/airframe interfaces as well as the design of the internal components was conducted to ascertain their impact on achieving the desired goals. Based upon detailed trade studies, it was decided that the best way to reduce weight and cost and improve reliability would be to integrate parts, reduce interfaces, rearrange for maximum structural efficiency, use new materials, and shorten critical load paths.

Additional concept and design trade-off investigations were conducted to determine the best approach to attain the desired goals and objectives and to determine specific drive train technology improvements required to meet these goals.

The conceptual design was to be sized for a medium-power, twin-engine, single-rotor helicopter (approximately 15,000 pounds gross weight) with approximately 70:1 overall reduction ratio between engine speed and rotor speed. The Boeing Vertol YUH-61A main transmission and drive system design was used as the baseline contemporary helicopter drive system technology for all comparative assessments of the components relative to the design goals.

The overall reduction ratio of the YUH-61A main rotor transmission was 25.1 to 1. The engine bevel drive which is external to the main transmission accounts for the balance of the reduction ratio of 67.6 to 1. Power inputs to the main transmission are at the 90- and 270-degree positions. The main transmission also had provisions for a forward AGB drive at 0 degrees, and for a tail rotor and aft AGB drive at 180 degrees.

The loads criteria to which this main rotor transmission was designed are as follows:

Maximum single-engine input horsepower	1,521
Input rpm	7,419
Output horsepower	2,655
Output rpm	295
Output torque	562,730 ± 67,530 in.-lb
Lift load	17,004 ± 567 lb

The initial studies resulted in a redesign of the main rotor transmission of the YUH-61A helicopter as shown in Figure 1.

Based on this advanced conceptual design, a component development and test program was structured for each transmission component that must be considered to attain the goals set forth. The selection and nonselection rationale and the priority of each proposed component work task were defined in the proposal submitted to ATL.

To provide the needed technology to accomplish the proposed advanced design, a three-task program was

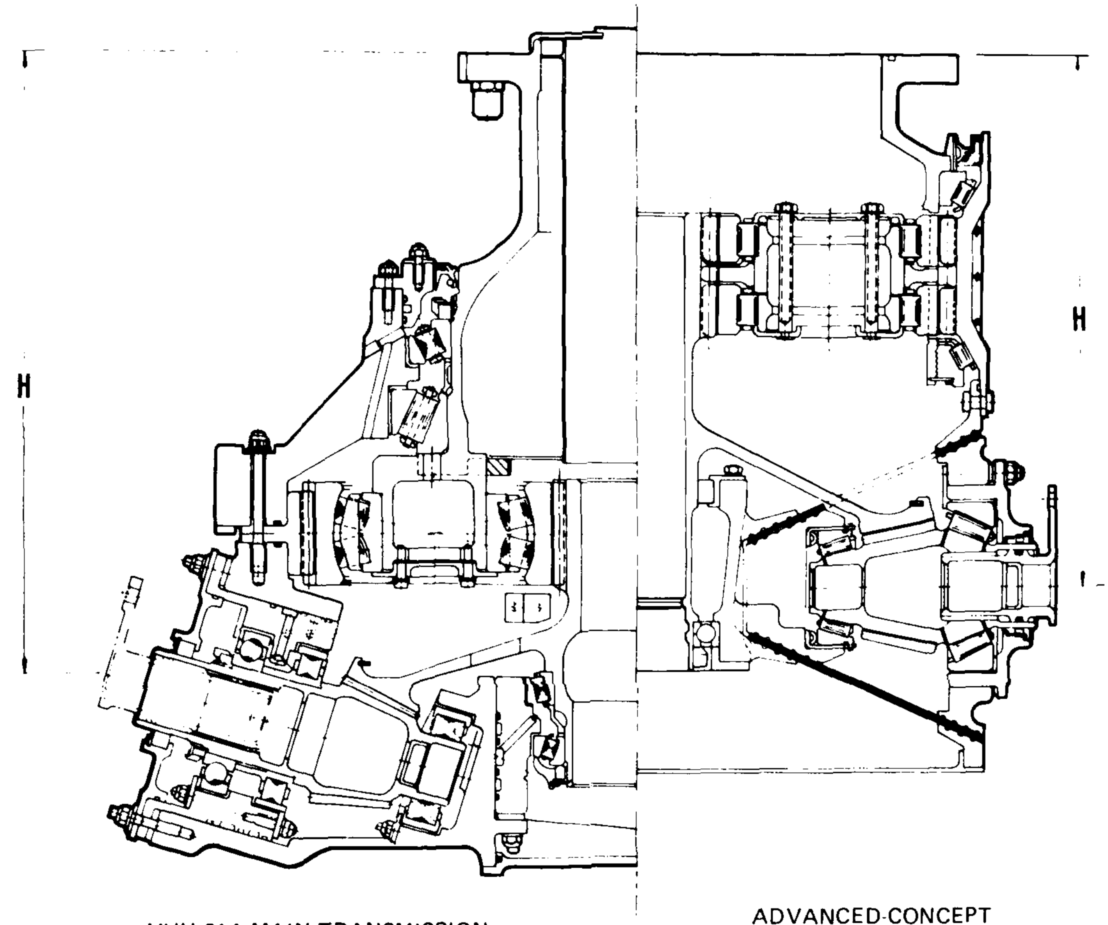


Figure 1. Main Transmission Configuration Comparison.

undertaken under Contract DAAJ02-76-C-0045, to investigate the high-risk, high-payoff areas. These tasks were classified as follows:

Task I	Advanced-Composite Housing
Task II	Bearing Development
	Evaluation of Hot-Hardness-Carburized Bearing Material
	Advanced Analysis of Complex Bearing Structures
	Advanced Ribbed-Cup Tapered-Roller Bearings
Task III	Oil Seal Development

Work was initiated in these areas in mid-1976. In 1978 the development effort on the composite transmission housing was modified and the overall completion date of the contract was extended to December 1980. The effort on Tasks II and III remained unchanged and proceeded on schedule. All bearing and seal work was completed in December 1979. It was therefore established that two technical reports would be issued. This report covers work completed on Tasks II and III and another report which will be released at a later date will cover work under Task I. A preliminary summary of the total work was presented in Reference 1.

This report is divided into three sections representing the three major development programs of Tasks II and III. The three programs are briefly summarized below.

- Evaluation of a high-hot-hardness carburizing bearing material

  The results of evaluation of VASCO-X2 steel as a bearing material are presented. The design, fabrication, and test of a 207S ball bearing demonstrated that this material can be used for integrated gear and bearing components. A Weibull plot of fatigue data indicates that a material life-improvement factor of more than 5 can be used for bearings fabricated from VASCO-X2 steel.

- Advanced ribbed-cup tapered-roller bearing and magnetic seal test program

  The results of seven development tests and one limited endurance test of a ribbed-cup tapered-roller bearing and magnetic seal are presented. All test objectives were achieved, including operation at loads equivalent to 1,500 hp at speeds as high as 14,000 rpm. The program was extended to conduct six oil-off survivability tests. The tests did not achieve the goal of 30-minute operation after loss of oil, but they did provide insight into critical operating parameters during oil-off operations.

- Advanced analysis of complex bearing structures

  The use of finite-element modeling (FEM) of complex bearing structures, such as rotor shaft support bearings, was investigated. The development of a spring-gap model of each bearing element node provided a means of achieving an accurate bearing internal load distribution due to structural stiffness. The effect on bearing life and performance can be evaluated in order to obtain an efficient structure to support relatively large bearings.

Complete details of each of these programs are discussed in the following sections.

---

1. Lenski, Joseph W., Jr., and Mack, John C., DRIVE SYSTEM DEVELOPMENT FOR THE 1980'S, Paper No. HPS-9, Presented at the Helicopter Propulsion System Specialists' Meeting of the American Helicopter Society, Williamsburg, Virginia, November 1979.

EVALUATION OF A HIGH-HOT-HARDNESS CARBURIZING BEARING MATERIAL

BACKGROUND

As part of the development of an advanced bearing concept, a test program was conducted to evaluate the performance of VASCO-X2 steel as a carburizing material suitable for high-temperature bearing applications. This material has been developed by Boeing Vertol for use as a gear material for improving the load capacity of highly stressed spur and spiral bevel gears. Performance as a gear material has been good and the steel shows potential as a good bearing material. The majority of rolling-element bearings presently manufactured for helicopter transmissions are made from through-hardened steels such as 52100 and M50. Only tapered-roller bearings or a few special bearings are manufactured from standard carburizing grades of steel.

To achieve the design goals of an advanced-concept transmission, there is a requirement that rolling-element-type bearings possess the following characteristics:

- Increased fatigue life and reliability
- Material stability during reduced-oil-flow operation
- Slow crack propagation
- Material compatible for both gears and bearings
- Economical to produce
- High hardness at elevated temperature (>300°F).

The above items have a common factor, which is the selection of an optimum bearing material that possesses all of these qualities. Present through-hardened bearing materials such as 52100 and M50 steel do not possess all the desired features mentioned.

In the majority of early helicopter applications, consumable-electrode vacuum-melted AISI 52100-type steels have satisfied bearing fatigue life requirements for normal operation up to 300°F. Above this temperature range, AISI 52100-type steels exhibit a loss of load-carrying capacity associated with a reduction in hot hardness at the higher temperatures. For applications requiring operation at temperatures exceeding 300°F or for increased fatigue-life requirements, consumable-electrode vacuum-melted M50 tool steel is used. While this material has shown excellent load-carrying capacity and stability for operation up to 650°F (Figure 2), the raw material procurement and manufacturing operations are considerably more expensive and the rate of crack propagation is such that M50 steel bearings may result in rapid failures due to race cracking from either fatigue or ballistic impact damage. In addition, M50 is not a suitable gear steel. Therefore M50 meets only some of the requirements of the advanced-concept transmission.

When comparing various case-carburized and through-hardened bearing materials, one must evaluate the material's ability to resist fracture during operation. In a failure due to fracture, a crack can occur either from a spall or from some other high-stress point in a bearing component and rapidly traverse the section by unstable crack extension. This type of failure in through-hardened steels can occur without warning and result in the loss of the structural integrity of the bearing (i.e., loss of fit and internal geometry), possible fragmentation of the bearing, and probable damage to the bearing and related transmission components. Experience has shown

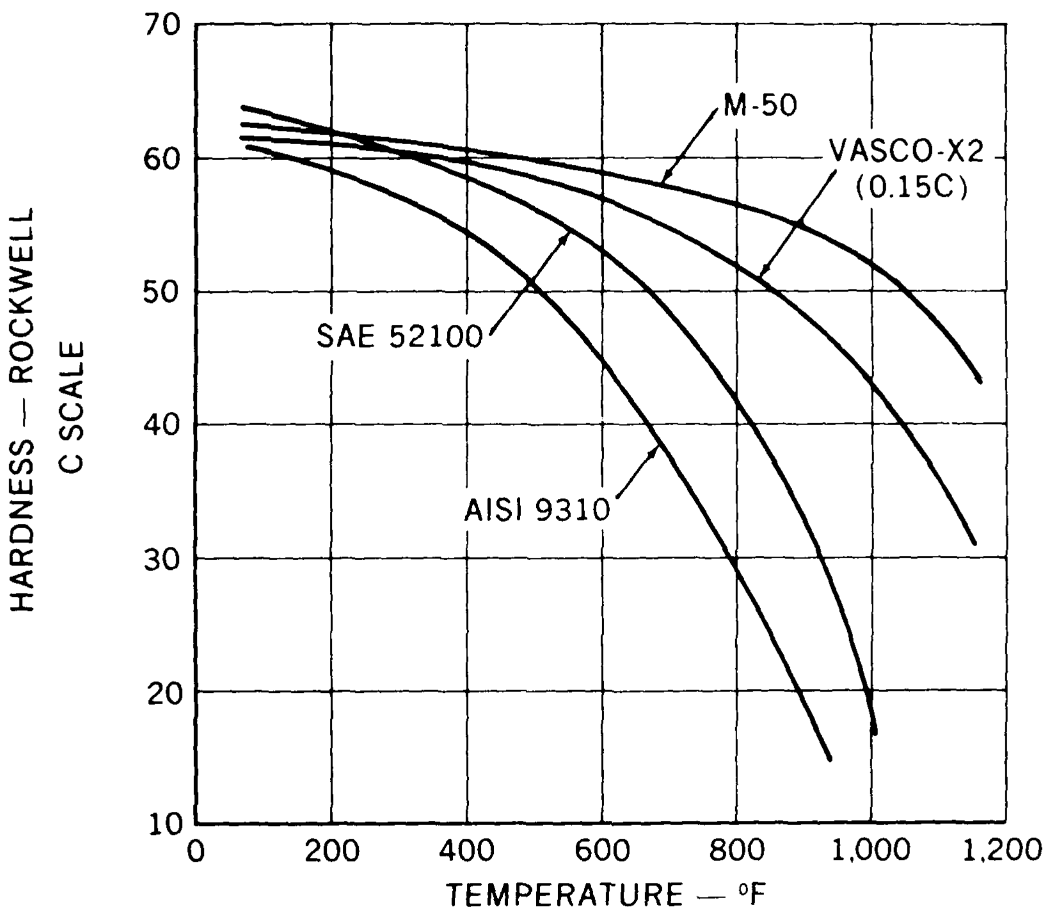


Figure 2. Hot Hardness Characteristics of Common Bearing Steels.

that bearings fabricated from high-carbon through-hardened steels such as 52100 and M50 are susceptible to fracture failures.*

Boeing Vertol has used AISI9310 case-carburized steel as a bearing material for special applications where the bearing races are integral with a gear with exceptionally good success. Both inner- and outer-race applications have been used successfully. Boeing Vertol has never experienced a fracture of any bearing race fabricated from this steel. It therefore appears that the VASCO-X2 material properties which improved the performance of helicopter gears at high temperatures should also provide similar improvements if used as a bearing material.

In addition to VASCO-X2 steel, there are other high-hot-hardness carburizing steels available which could provide high-temperature operation similar to that of VASCO-X2. These materials are CBS600 and CBS1000M. Table 1 provides the chemical composition of these materials.

CBS600 steel has been used by the Timken Company in bearing applications operating up to 600^{o}F, while CBS1000M can be used up to 1,000^{o}F and still retain the desired surface hardness required for good fatigue life properties at these higher operating temperatures. Timken has made and operated bearings of CBS600 and CBS1000M with good success. No bearings have been fabricated from VASCO-X2 and only limited test experience is available for race surfaces made from VASCO-X2.

To evaluate the use of VASCO-X2 steel as a bearing material, TRW's Marlin Rockwell Division was subcontracted to fabricate a lot of bearings meeting the dimensions of a standard 207S ball bearing, except that the inner races were fabricated from carburized VASCO-X2 steel and all other elements were made from M50 steel. These bearings were then subjected to fatigue endurance testing under an accelerated load schedule.

TEST BEARING DESIGN

TRW's Marlin Rockwell Division of Jamestown, New York, has conducted many bearing tests to evaluate new materials or improved processing methods This test data has been accumulated using a standard 207S-size deep-groove ball bearing as the test specimen. This bearing has been accepted as a good test specimen and therefore all future data is compared to this size bearing.

The design used in this program is based upon the standard MRC 207S deep-groove ball bearing; its basic dimensions are shown in Figure 3. The outer ring and balls were fabricated from consumable-electrode vacuum-melt M50 steel (AMS649) and the inner rings were made from a bar of consumable-electrode vacuum-melt VASCO-X2 (XBMS7-223) case-carburizing steel. This configuration was selected in order to minimize costs and to expedite the test program. The use of the inner rings as the test specimen is not a new concept. Because of the high contact stresses on the inner rings, most fatigue failures will occur on this ring. Therefore, to evaluate the material's rolling-contact fatigue properties, only the inner ring is fabricated from the material being evaluated and the remaining elements (balls, outer ring) are fabricated from a more readily available bearing material other than the test material.

Until this test program, the 207S ball bearing has been exclusively fabricated from a through-hardened bearing steel. Therefore, prior to the fabrication of these bearings from a case-carburizing grade of steel, a study was conducted to determine the case depth requirements for the test load conditions. In order to minimize the time to conduct fatigue testing of the material, an accelerated load condition was used. The load established

*Bearings fabricated from case-carburized steel are not susceptible to this type of failure because a crack in the case progresses until it reaches the relatively soft core and then stops.

TABLE 1. CHEMICAL COMPOSITION OF VARIOUS HIGH-TEMPERATURE BEARING STEELS

Steel Type	C	Mn	P	S	Si	Cr	Ni	Mo	Cu	V	W
CBS600*	0.16/0.22	0.50/0.70	0.025 max	0.025 max	0.90/1.25	1.25/1.65		0.90/1.10			
CBS1000*	0.18/0.23	0.40/0.60	0.025 max	0.025 max	0.40/0.60	0.90/1.20		4.75/5.25		0.75/1.00	
M50**	0.77/0.85	0.35 max	0.025 max	0.025 max	0.25 max	3.75/4.25	0.10 max	4.00/4.50		0.90/1.10	
VASCO-X2*	0.12/0.16	0.20/0.40			0.80/1.00	4.75/5.25		1.30/1.50		0.40/0.50	1.20/1.50
AISI52100**	0.95/1.10	0.25/0.45	0.025 max	0.025 max	0.20/0.35	1.30/1.60					
AISI9310*	0.07/0.13	0.40/0.70	0.015 max	0.015 max	0.20/0.35	1.06/1.40	3.00/3.50	0.08/0.18	0.35 max		

*Case-carburized Timken specification
**Through-hardened

TABLE 2. COMPUTER ANALYSIS OF TEST BEARING

Radial Load (lb)	Speed (rpm)	Mean Hertz Stress (psi)	B-10 Life (no factor) (hr)	Max Shear Depth (in.)	Max Orth Shear Depth (in.)	Required Case Depth to Rc 58 3 x Z	5 x Z
2,400	5,500	322,000	18.4	0.0082	0.0053	0.0246	0.0410
1,900	5,500	298,200	36.8	0.0076	0.0049	0.0228	0.0380
1,400	5,500	269,900	90.8	0.0069	0.0044	0.0207	0.0345
900	5,500	233,800	332.5	0.0060	0.0038	0.0180	0.0300

by previous testing was a 1,900-pound radial load at 5,500 rpm. Under this condition, an AFBMA B-10 life of approximately 38 hours is achieved without any material factors included.

Boeing Vertol's experience with case-carburizing grades of steel used as bearing races has established a criterion that the case depth to Rockwell hardness (Rc) 58 should be three to five times the depth of the maximum shear stress. A computer analysis study was conducted for various load levels for a 207S ball bearing and the results of this study are shown in Table 2. For the test condition of 1,900 pounds, the depth to maximum shear is 0.0076 inch and this results in a case depth requirement of 0.023 to 0.038 inch. Based upon this information, the effective case depth of 0.070 to 0.100 inch (Rc 50) was specified with a requirement of an Rc 58 depth of 0.030 inch minimum. Under normal loading, the case depth requirements would be much less.

BEARING FABRICATION

Because of the relatively thin bearing section and high case-depth requirements, a special heat-treatment procedure for these test bearings was specified to eliminate any problem of through-hardening the thin section under the raceway. The VASCO-X2 steel inner rings were initially turned to an oversize configuration as shown in Figure 4. Additional material remained on the bore of the inner ring to allow for ample section size to achieve an adequate case core ratio. These rings were fabricated by TRW and then sent to American Lohmann Corporation, Little Ferry, New Jersey, to be case-carburized to the specification shown in Table 3. Upon completion of the heat treatment, the inner rings were returned to TRW where they were finish-ground.

The material for the fabrication of the inner rings was obtained from Teledyne Vasco, Latrobe, Pennsylvania, per Boeing Vertol Specification BMS7-223 (VASCO-X2). A 3-inch-diameter bar was obtained; the analysis of the heat lot for this bar is shown in Table 4. Forty inner rings were machined in an oversized configuration from this stock and the inner rings were sent to American Lohmann Corporation for case-carburizing and heat-treatment. The inner rings in the as-received condition were carburized to the depth shown in Table 3. One ring was removed and sectioned to determine the case depth and microstructure of the inner ring. The results of this first check are shown in Figure 5 and indicated that an acceptable case/core was achieved; the remaining rings were then approved for tempering for 3 hours at approximately 1,200°F. After tempering, the bore of the inner rings was machined to a larger diameter in order to eliminate the case on the bore. If the bearing section was thicker or the case depth not as deep, this operation would not be required. A general rule is that a core of approximately 1/3 of the section thickness should be maintained to achieve the desired material properties of a case-carburized steel. For this test specimen, the bore would be at core hardness. After machining, the rings were hardened and heat-treatment was completed.

TABLE 3. HEAT-TREATMENT OF VASCO-X2 TEST SPECIMEN INNER (BALL-BEARING) RINGS

1.	Heat-treater shall carburize the inner rings all over to requirements. The effective case depth shall be 0.070 to 0.100 inch (Rc 50 depth). An Rc 58 depth of 0.030-inch minimum is required. Refer to Figure 5. Core hardness Rc 36-44.
2.	Temper all inner rings within 5 hours at 1,100° to 1,250°F for 3 hours minimum.
3.	Machine inner-ring bore to 1.368 $^{+0.000}_{-0.005}$ inches after temper.
4.	Nickel strike, copper plate, harden, and temper per requirements.
5.	Package and ship completed inner rings to Marlin-Rockwell Division of TRW, Jamestown, New York (zip code 14701).

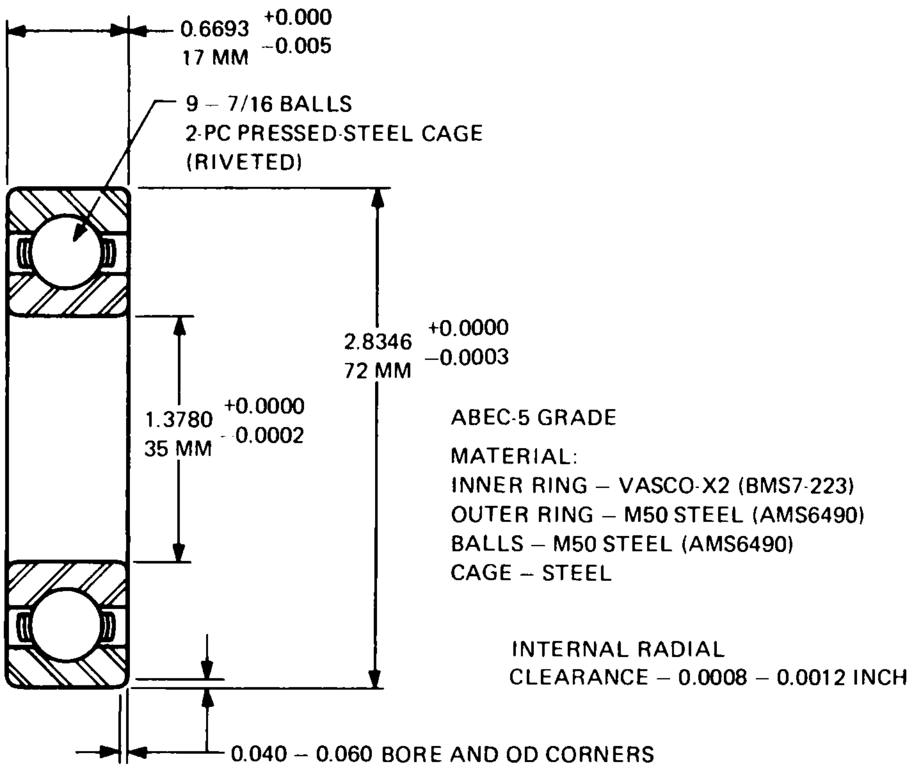


Figure 3. Design of Test Bearing.

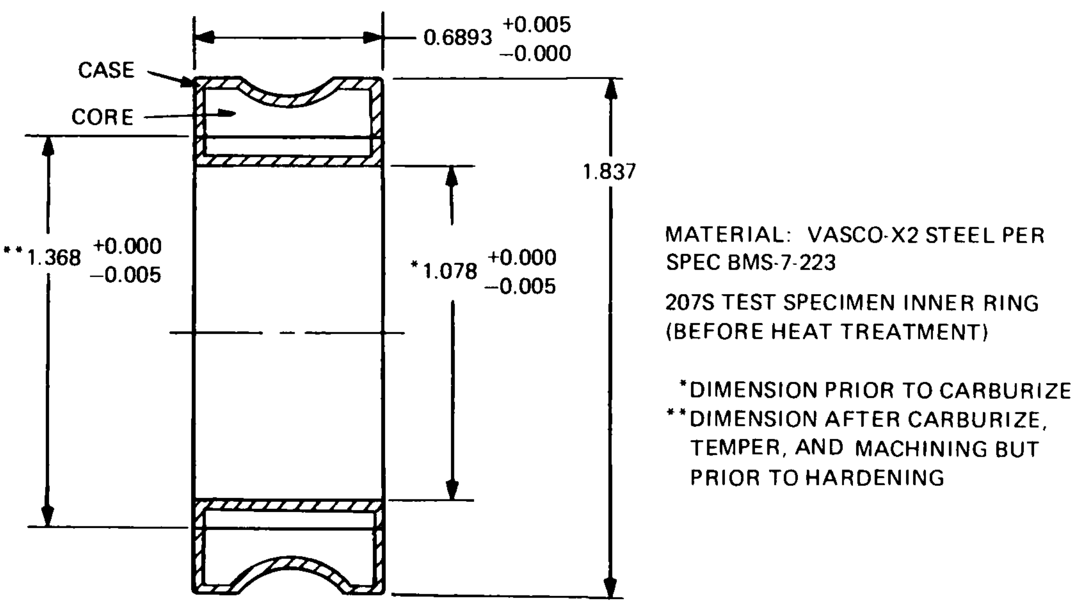
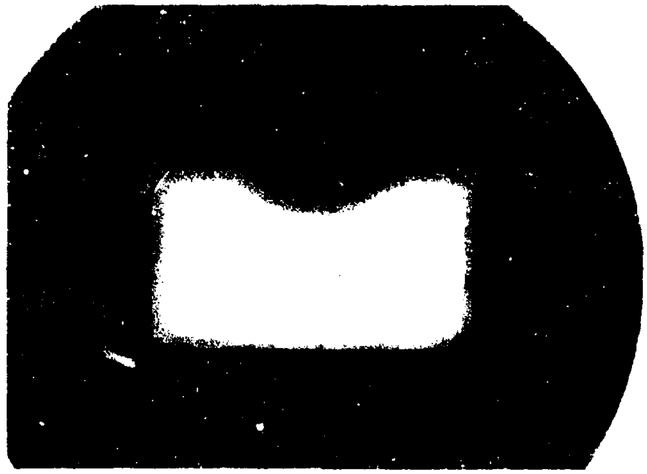
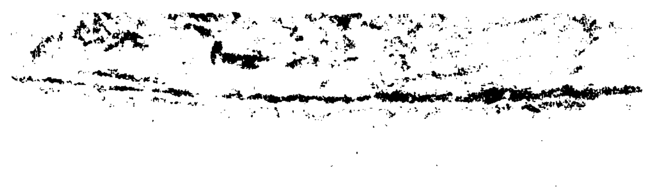


Figure 4. Original Oversize Configuration of Bearing Inner Race.



VILLELA'S ETCH 2.7X

CROSS SECTION THROUGH BEARING UNGROUND



VILLELA'S ETCH 100X

CROSS SECTION AS ABOVE AT BEARING RACE

Figure 5. Case Depth and Microstructure of Bearing Inner Race.

TABLE 4. ANALYSIS OF BEARING STEEL

Brand: CVM VASCO-X2 Mod Boeing Spec XBMS 7-223

Size	Pieces	Weight	Heat No.	Date Shipped
3 in. rd	1	122 lb	3432-A	10 26 76

Macrostructure Satisfactory

Grain Size 2T – 6
1B 6-1/2

Magnetic-Particle Inspection: 2T F/S 0/0
1B F/S 0/0

Jominy Hardenability:

	J1	J4	J8	J12	J16	J24	J32
2T	37.8	40.9	40.3	39.8	39.3	38.2	37.9
1B	39.0	39.4	39.0	38.5	37.8	37.0	36.4

J-K Rating

	A		B		C		D	
	Thin	Heavy	Thin	Heavy	Thin	Heavy	Thin	Heavy
T	0	0	½	0	0	0	1½	½
B	½	0	½	0	0	0	1½	½

Heat No.	C	Si	Mn	S	P	W	Cr	V	Mo
				Analysis					
3432-A	0.15	1.00	0.20	0.009	0.015	1.33	4.98	0.40	1.33
Middle	0.15								
B	0.15	1.00	0.20	0.007	0.015	1.35	4.98	0.40	1.33

The inner rings, upon return to TRW, were finish-ground and matched with the outer ring and balls fabricated from M50 steel. The material used for the outer ring and balls was supplied by Marlin Rockwell per specification AMS6490 and is typical of material used for aircraft-type bearings.

Upon completion of all operations, 33 bearing assemblies were available. During the inspection of the bearings after final grinding, several inner rings showed evidence of surface cracks on the side faces of the inner ring. All cracks appeared to be on the face or on the outside diameter, but none were recorded in the raceway of the bearing.

Prior to the start of testing, a destructive metallurgical examination of one bearing was scheduled. It was therefore determined that a bearing should be selected that appeared to have the largest surface indication for this examination. This bearing was returned to Boeing Vertol for a detailed destructive metallurgical examination to determine the cause of the surface cracks, to evaluate the microstructure, and to determine if the bearings were acceptable for fatigue testing.

The bearing selected for examination is shown in Figure 6 in the as-received condition, and Figure 7 shows the surface indications as they appeared on the inner ring side face. This inner ring contained two indications approximately 0.9 and 1.1 inches long. Metallographic sections through the bearing inner ring revealed a uniform carburized case as shown in Figure 8. Also shown in this figure was the indication of a case-core crack. An enlarged view of this area is shown in Figure 9, which shows a crescent-shaped case-core separation bisecting the outer-diameter corner of the bearing inner ring. Additional work was conducted to determine the origin of the cracks, and Figures 10, 11, and 12 show that the origin of the crack propagated from the intergranular zones centrally located within each fracture. These results indicated that the cracks observed on the faces of the inner rings were the result of case-core separation and that the most probable cause of this was the result of excessive case penetration from two sides of the face/outside-diameter intersection. This type of case-core separation has been experienced on case-carburizing thin-section gear teeth. After a complete review of this data, it was determined that the cracks were not in an area that would affect the fatigue life of the ball/raceway contact and that it was also very unlikely that the cracks would propagate during endurance testing, based upon Boeing Vertol experience with this type of cracking.

In addition to the investigation of the surface cracks, a detailed metallurgical examination was conducted to determine the case depth and microstructure. The carburized-case-hardness gradient is shown in Figure 13. This shows that the surface hardness in the race track is Rc 63 and a hardness of Rc 60 is maintained to a depth of 0.035 inch after final grind. An effective case depth of Rc 50 was maintained to approximately 0.070 inch as required by specification. A core hardness of Rc 41 was achieved with a discontinuous carbide network and retained austinite of less than 20 percent.

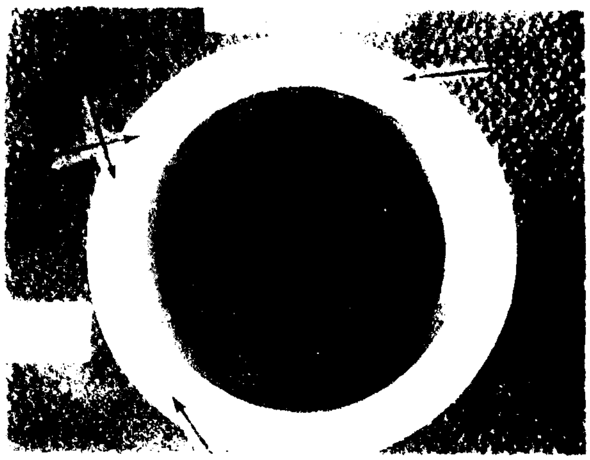
A case-carbon gradient was also recorded on this inner ring and is shown in Figure 14. The gradient is typical for VASCO-X2 and was acceptable. A metallographic section was prepared of the case and core section of the inner ring. The carburized-case microstructure showed an acceptable discontinuous carbide distribution as illustrated in Figure 15. An acceptable hardened and tempered core microstructure is also shown in Figure 16 and is considered typical of VASCO-X2 steel.

Based upon these findings, it was determined that these bearings could be used for fatigue testing. Except for the case-core separation which was considered due to an exceptionally high case depth requirement, all other factors indicated that the bearings were properly heat-treated and met all the requirements of properly heat-treated VASCO-X2 steel.

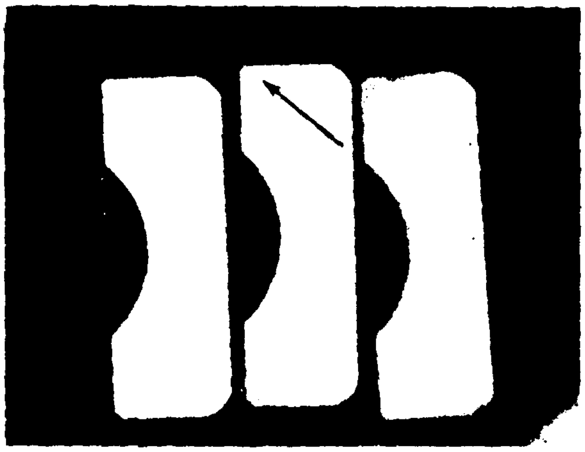
Upon completion of all inspections, 32 bearings were assembled and all met specifications except for three inner rings which were slightly oversized on bore and two bearings which had slightly larger internal clearances. Measurements of bore, outside diameter, internal clearance, and resultant shaft fit are shown in Table 5.

In addition to the test bearings, TRW provided 22 slave bearings which were fabricated from a single heat of vacuum-degassed 52100 steel forged ings. These bearings remained from three lots of bearings which were tested several years ago (see Appendix A) under the same load and speed but with mineral oil lubrication at somewhat lower oil inlet temperatures. These bearings met the same specifications as the test bearings except for material.

[illegible]



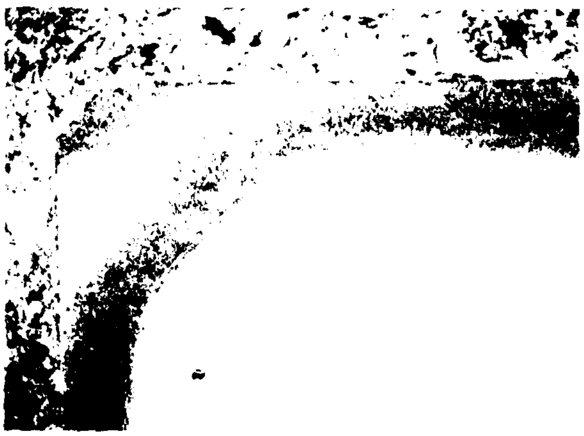
[illegible] End



3X

UNIFORM CARBURIZED CASE ALONG BALL TRACK.
PROFILE OF CASE-CORE CRACK SHOWN IN CENTER
SECTION (ARROW)

Figure 8. Metallographic Sections Through Bearing Inner Ring.



28X

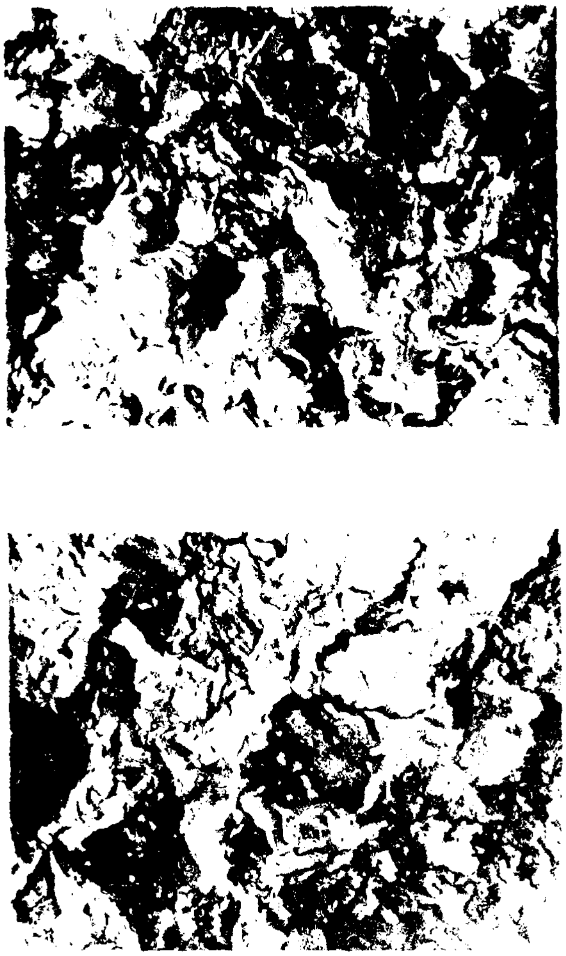
Figure 9. Enlarged View of Case-Core Separation at Outer-Diameter Corner of Bearing Inner Ring.

2.4X

Figure 10. Both Cracks Propagated From Intergranular Zones Centrally Located Within Each Fracture.

19X

Figure 11. Enlarged View of Intergranular Origin Areas



3,600X

Figure 12. Electron Fractographs Exhibited Intergranular Topography in Origin Areas.

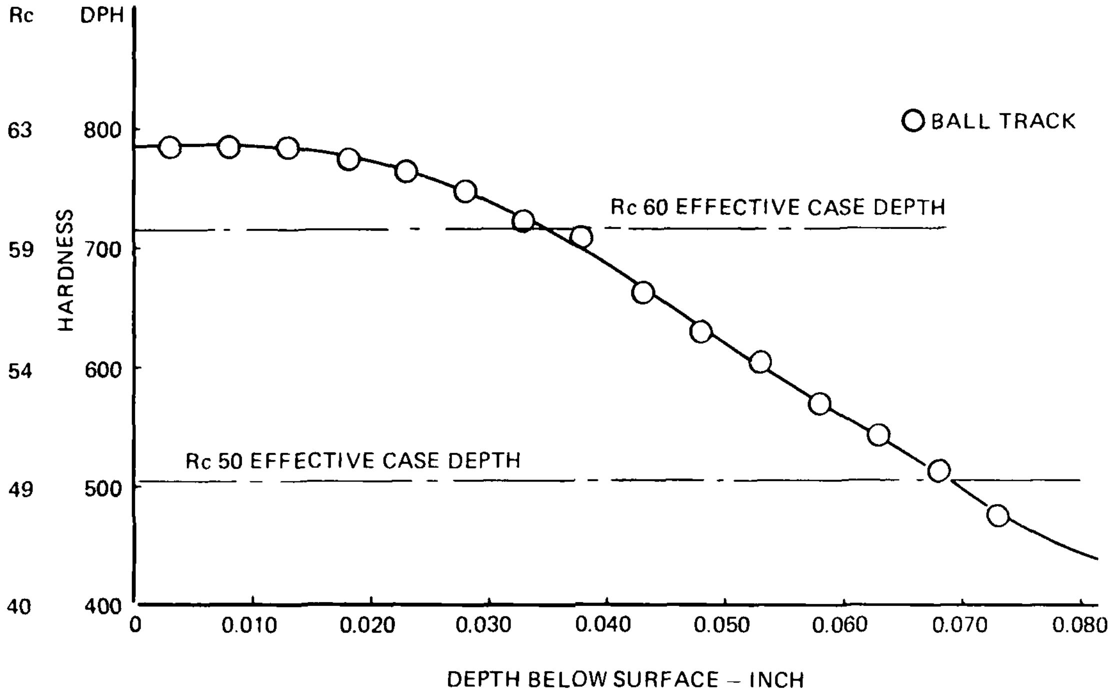


Figure 13. Carburized-Case-Hardness Gradient of VASCO-X2 Steel Inner Ring Specimens.

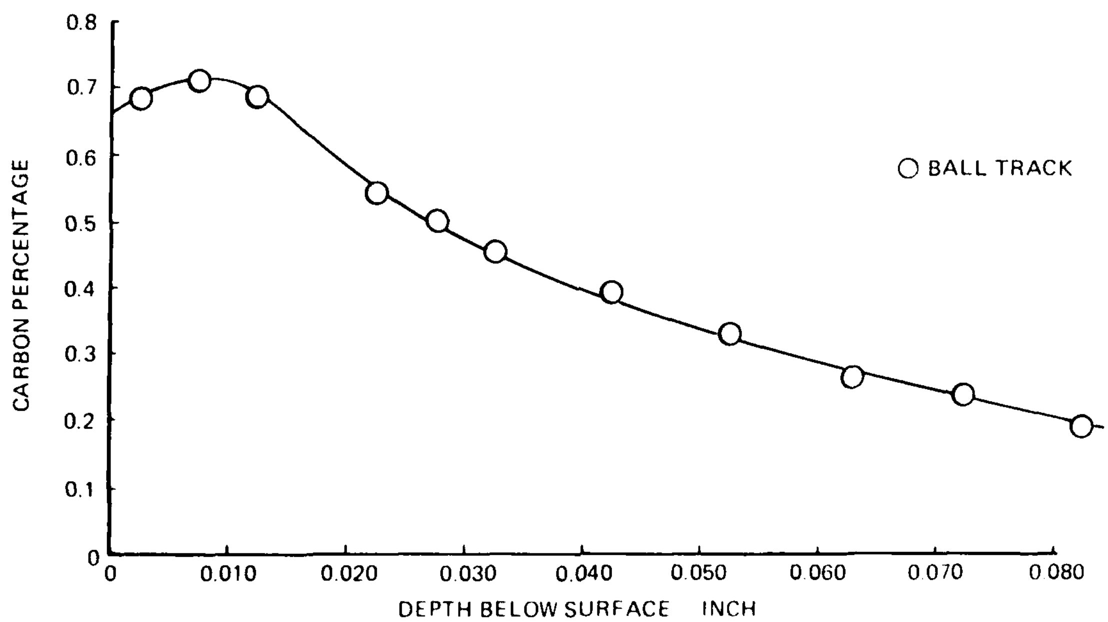
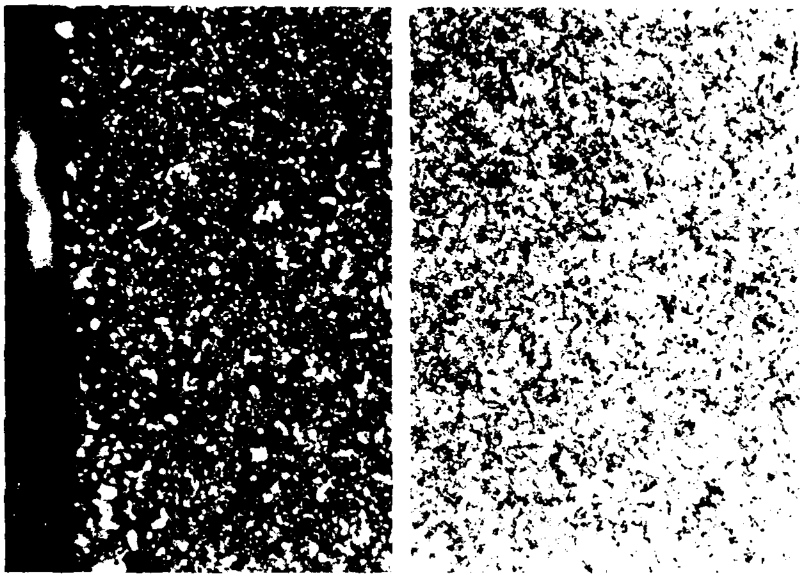


Figure 14. Carburized-Case-Carbon Gradient of VASCO-X2 Steel Inner Ring Specimens.



500X NITAL ETCH

Figure 15. Acceptable Discontinuous Carbide Distribution Exhibited in Carburized-Case Microstructure by VASCO-X2 Steel Bearing Ring.

500X NITAL ETCH

Figure 16. Acceptable Hardened and Tempered Core Microstructure Exhibited by VASCO-X2 Steel Bearing Ring.

TABLE 5. BEARING MEASUREMENT DATA

Inner Ring Serial No.	Bore (in.)	Press Fit on Shaft (in.)	Outside Diameter (in.)	Radial Clearance (in.)	Rockwell* Hardness Inner	Rockwell* Hardness Outer
1	1.3780	0.0007	2.8345	0.0010		
2	1.3780	0.0008	2.8345	0.0011		
3	1.3780	0.0004	2.8345	0.0010		
4	1.3780	0.0005	2.8346	0.0010		
5	1.37805	0.0007	2.8345	0.0008		
6	1.3779	0.0005	2.8346	0.0010		
7	1.3780	0.0007	2.8346	0.0009		
8	1.3780	0.0009	2.8346	0.0010		
9	1.3780	0.0008	2.8345	0.0011	62.5	63
10	1.3780	0.0008	2.8345	0.0012		
11	1.37802	0.0007	2.8346	0.0008		
12	1.3780	0.0009	2.8346	0.0010		
13	1.3779	0.0008	2.8346	0.0009		
14	1.3780	0.0007	2.8346	0.0009	62	63
15	1.3780	0.0009	2.8345	0.0009		
16	1.3780	0.0008	2.8346	0.0011		
17	1.3780	0.0009	2.8346	0.0010	60	63
18	1.37802	0.0006	2.8346	0.0009	62	63
19	1.3779	0.0006	2.8345	0.0009		
20	1.3779	0.0010	2.8346	0.0012		
21	1.3779	0.0009	2.8345	0.0013		
22	1.3780	0.0008	2.8345	0.0012	61	62
23	1.3779	0.0010	2.8345	0.0011		
24	1.3779	0.0005	2.8346	0.0010		
25	1.3779	0.0006	2.8345	0.0009	62	63
26	1.3780	0.0007	2.8345	0.0010		
27	1.3780	0.0007	2.8346	0.0009		
28	1.3779	0.0005	2.8346	0.0009		
29	1.3780	0.0007	2.8346	0.0009		
30	1.3779	0.0008	2.8346	0.0010		
31	1.3779	0.0009	2.8346	0.0014		
32	1.3780	0.0009	2.8345	0.0010		

*All bearings checked for hardness before final grind; hardness range Rc 60 to Rc 64. Readings shown are after finish-grind.

TEST PROGRAM

A battery of eight identical MRC Model A bearing test machines was used to subject the bearings to fatigue endurance running under an accelerated load schedule. A schematic of a typical test machine is shown in Figure 17 and several of the test machines used in this program are shown in Figure 18. Each machine had

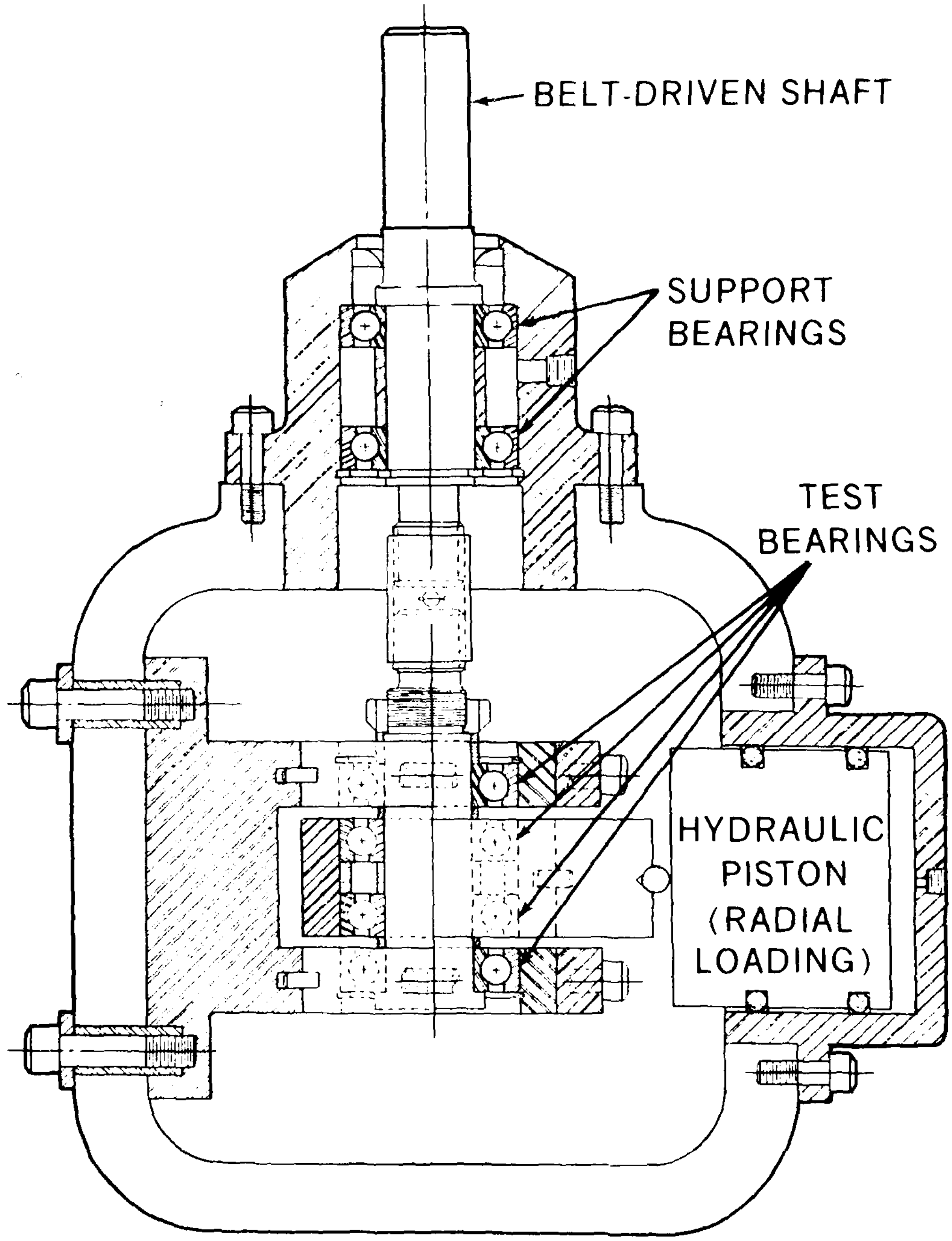


Figure 17. Schematic of a Ball-Bearing Fatigue-Test Machine.

Figure 18. Part of Battery of Test Machines Used in Program.

individual controls, but lubrication and hydraulic pressure (for radial load) were supplied by a single pumping system. The test spindle allowed four bearings to be tested at a time. The radial load was applied to the two inboard positions and reacted upon by the two outboard bearings. Because of the symmetrical location of the bearings on the arbor, each bearing experienced the same load. Test conditions were as follows:

- Speed — 5,500 rpm
- Load — 1,900 pounds radial, per bearing ($c/p \approx 2.3$)
- Lubricant — Mobil Jet II (MIL-L-23699)
- Temperature — $190^{\circ}F \pm 5^{\circ}$ outer ring
- Duration — failure or 1,000 hours
- B-10 Life — 38.7 hours (AFBMA)

Test rigs ran 24 hours a day, 7 days a week until automatic shutdown or completion of test. In case of a bearing failure, a printed-circuit grid under the bearings was shorted by metallic chips produced by the fatigue failure and the machine was shut down. Testing was also interrupted if hydraulic pressure deviated more than two percent from the preset level. Thermocouples were used to record the temperature of the outer races. A check of bearing temperatures showed that the outboard bearings ran in the 185° to $190^{\circ}F$ range while the inboard bearings ran at 190° to $195^{\circ}F$.

Thirty test bearings were started initially, with one machine running with two test bearings and two slave bearings. As a bearing failed, it was replaced by one of the two remaining test bearings or by a slave bearing. In addition to the 32 test bearings, 22 slave bearings fabricated from a baseline 52100 steel were tested. The slave bearings replaced the test bearings at failure and were also used to compare the life-improvement factor to VASCO-X2 steel.

TEST RESULTS

The fatigue endurance lives of the test bearings are shown in Table 6. Twelve of the 32 bearings achieved endurance lives in excess of 1,000 hours. Four bearings were removed from test due to outer-race failures; the remaining 16 bearings were removed from test due to fatigue damage on the inner race. Also shown in Table 6 is the serial number of the test rig on which each bearing was tested.

Table 7 provides a summary of the 22 slave bearing test times. Three bearings were removed due to inner-race failures, one with an outer-race failure, and one with a ball failure. Three bearings were suspended with more than 1,000 hours of testing without failure.

Twenty bearings out of the 32 test bearings experienced fatigue spalling, but four of these failures involved only the outer rings which were not made of the VASCO-X2 test material. In addition, a review of the test data indicated that five of the inner-ring failures may have been influenced by previous adjacent failures. Although all the bearings were visually inspected after a failure of one bearing on a test spindle, a detail inspection of the inner and outer raceways was not possible due to the riveted-cage-type construction of the test bearings. The five bearings which are suspected of being influenced by previous adjacent failures are as follows:

TABLE 6. FATIGUE ENDURANCE OF MRC 207S513 BALL BEARINGS

Inner Ring Serial No.	Hours	Test Machine Serial No.	Status
8	43.8	10	Spalled inner
17*	65.7	10	Spalled inner
12*	76.9	10	Spalled inner
11	87.5	5	Spalled inner
6	235.3	8	Spalled inner
14	246.0	5	Spalled inner
3**	264.2	8	Spalled outer
29	282.9	6	Spalled inner
27	314.5	6	Spalled inner
13**	474.0	6	Spalled outer
19	532.7	7	Spalled inner
7	623.2	5	Spalled inner
5*	628.7	5	Spalled inner
15*	669.0	6	Spalled inner
18	690.1	7	Spalled inner
30**	715.2	5 and 7	Spalled outer
24	759.8	7	Spalled inner
25*	768.3	7	Spalled inner
20	769.4	11	Spalled inner
2**	973.3	11	Spalled outer
16	1,000.3	11	Suspended
9	1,000.3	11	Suspended
10	1,007.1	12	Suspended
31	1,007.1	12	Suspended
22	1,007.1	12	Suspended
21	1,007.1	12	Suspended
32	1,008.8	10	Suspended
1	1,011.7	9	Suspended
26	1,011.7	9	Suspended
4	1,032.8	8	Suspended
28	1,032.8	8	Suspended
23	1,074.5	10	Suspended

*These failures have been considered as suspended data points. These failures influenced by previous adjacent failures due to debris damage.

**Outer race failures treated as suspended data. Material of outer races was CEVM M50 steel.

Inner Ring Serial No.	Hours	Remarks
17	65.7	A spall covering about 45 degrees of arc occurred on adjacent
12	76.9	inner ring serial no. 8 at 43.8 hours.
5	628.7	A spall covering about 45 degrees of arc occurred on adjacent inner ring serial no. 7 at 623.2 hours.
15	669.0	A spall covering about 90 degrees of arc occurred on adjacent slave bearing serial no. 1-34 at 658.7 hours.
25	768.3	A spall covering about 25 degrees of arc occurred on adjacent inner ring serial no. 24 at 759.8 hours.

Visible evidence that the five failures cited were initiated by debris from adjacent failures is lacking because, once started, failures progressed by a flaking process until shutdown was effected. When slave bearing no. 1-34 failed at 658.7 hours in machine 6, dents could be observed in test bearing no. 15. (An adjacent slave bearing which was more severely dented was suspended from test at the time.) The criteria for omitting the five failures were based upon the following conditions:

1. The size of the preceding adjacent failure
2. The time relationship between adjacent failures.

A Weibull plot was made based on this test data. Bearings removed due to outer-race failure, suspect inner-race failure, or nonfailure were considered as suspended data points; therefore the Weibull plot shown in Figure 19 was based upon 11 inner-race failures and 21 suspended failures. The plot shows that the B-10 life of the inner races fabricated from the VASCO-X2 test material was approximately 200 hours. The B-10 life of the complete bearing based upon an AFBMA calculated life is 38 hours. This shows a life-improvement factor of approximately 5.4 for the VASCO-X2 material.

A Weibull plot was also made for the slave-bearing failures (Figure 20). Five failures out of the 22 slave bearings were used to generate this plot. All the slave bearings in this test program were made from a single heat of vacuum-degassed 52100 steel and forged rings. Normally these bearings are fabricated from vacuum-degassed 52100 steel tubing. The only variation in these bearings is that they were heat-treated in three lots designated as 1-, 2-, and 3-. Ten 1- bearings, seven 2- bearings, and five 3- bearings were used to make up the 22 slave bearings. The endurance lives of these bearings when tested several years ago by TRW under the same load and speed but with mineral oil lubrication at somewhat lower operating temperatures were appreciably higher than that achieved in this test program. Test data from these previous tests is presented in Appendix A. Initial evaluation of these results would indicate that oil-film thickness due to different oils and temperatures may have caused the life reduction.

Photographs of the inner rings of the test bearings after test are shown in Figures 21 through 28 grouped as the bearings were assembled in each individual test machine. If a spall occurred on an inner ring, the spall was oriented so as to appear in the photograph. An initial review of the spalls indicates that all failures appear to be typical bearing fatigue failures. The ball path on each bearing also indicates that the load was distributed equally between each set of four bearings and that all loads were radial.

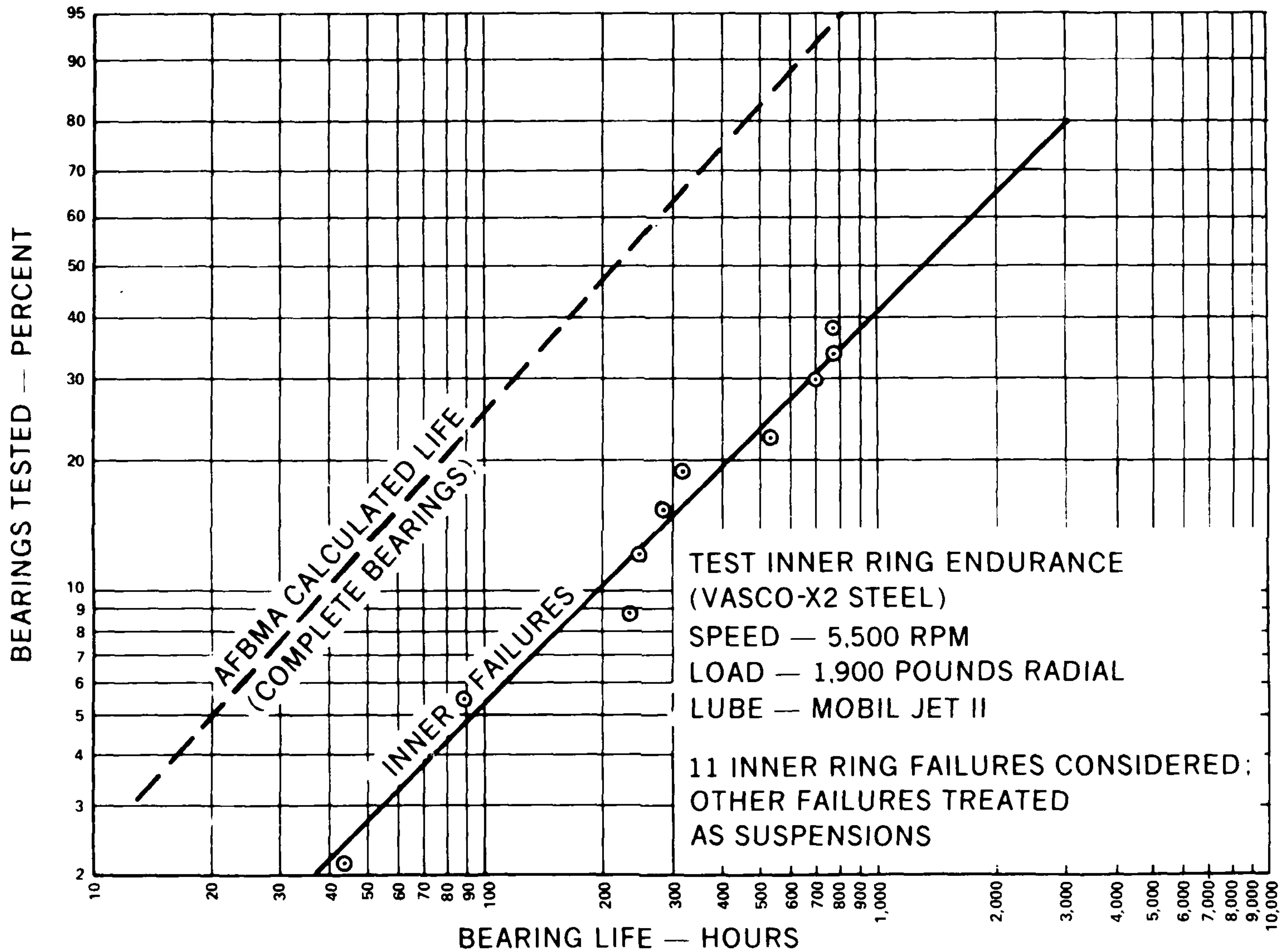


Figure 19. Weibull Plot of MRC 207S-513 Ball Bearing.

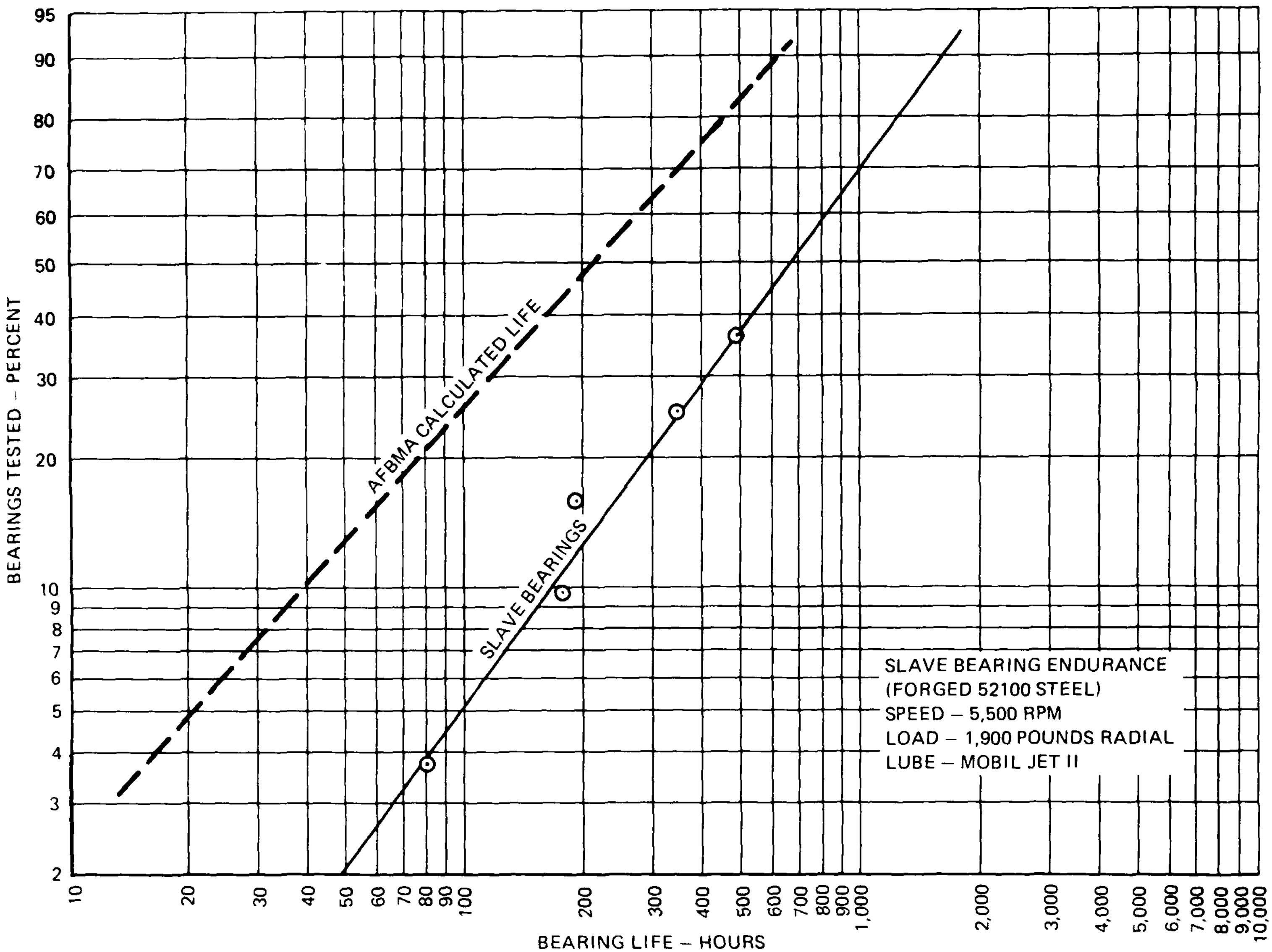


Figure 20. Evaluation of MRC 207S-513 Ball Bearing.

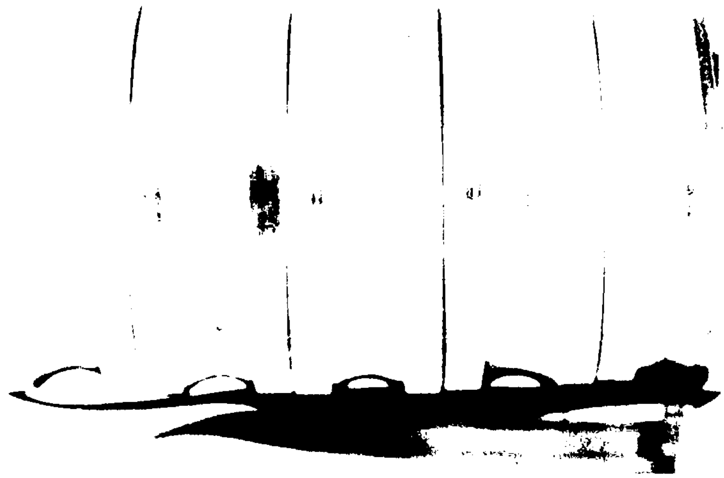


Figure 21. Inner Rings Serial No. 5, 7, 14, 11, and 30 After Test From Machine No. 5.

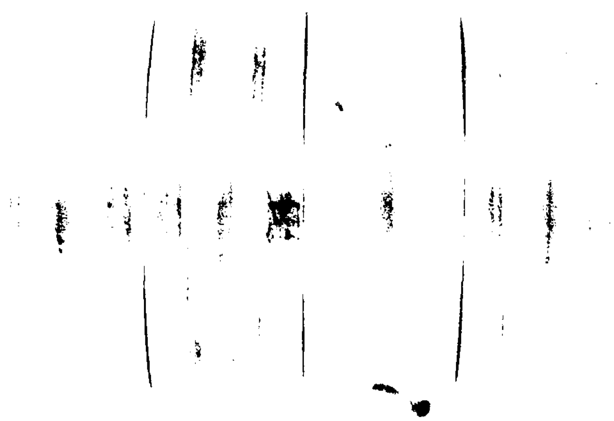


Figure 22. Inner Rings Serial No. 15, 27, 29, and 13 After Test From Machine No. 6.

Figure 23. Inner Rings Serial No. 24, 25, 19, and 18 After Test From Machine No. 7.

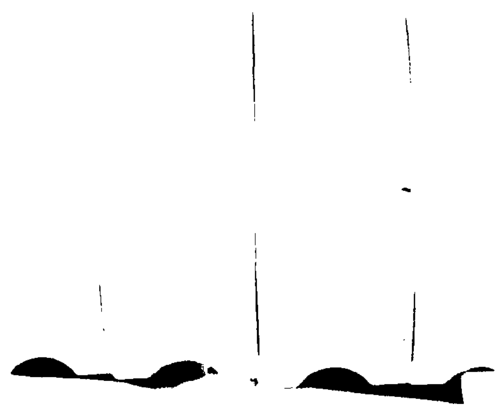


Figure 24. Inner Rings Serial No. 28, 6, 3, and 4 After Test From Machine No. 8.

Figure 25. Inner Rings Serial No. 1 and 26 After Test From Machine No. 9; Rings Overheated in Test.

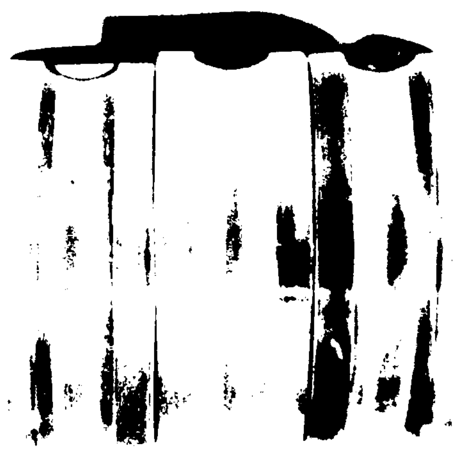


Figure 26. Inner Rings Serial No. 32, 12, and 23 After Test From Machine No. 10; Serial No. 8 and 17 Were Also Tested in Machine No. 10 But Are Not Shown.

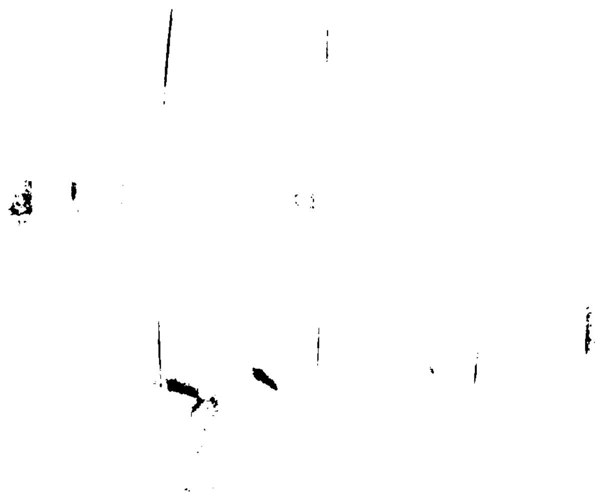


Figure 27. Inner Rings Serial No. 16, 9, 2, and 20 After Test From Machine No. 11.

Figure 28. Inner Rings Serial No. 21, 10, 23, and 31 After Test From Machine No. 12

TABLE 7. FATIGUE ENDURANCE OF MRC 207S SLAVE BEARINGS

Bearing No.	Hours	Test Machine Serial No.	Status
2-29	5.5	5	Suspended
2-20	10.3	6	Suspended
2-22	10.3	6	Suspended
2-19	33.1	7	Suspended
1-19	80.8	8	Spalled outer
3-13	128.8	7	Suspended
3-17	137.3	7	Suspended
1-20	174.1	6	Spalled ball
2-13	190.4	7	Spalled inner
2-10	195.0	6	Suspended
3-10	201.0	7	Suspended
2-31	201.7	6	Suspended
3-22	230.7	11	Suspended
3-28	283.5	8	Suspended
1-34	344.2	6	Spalled inner
1-23	382.7	5	Suspended
1-29	485.0	8	Spalled inner
1-35	716.7	8	Suspended
1-27	997.6	10	Suspended
1-1	1,011.7	9	Suspended
1-11	1,011.7	9	Suspended
1-24	1,030.7	10	Suspended

Figure 29 is a photograph of two slave bearing inner rings which failed in this test program. It appears that slave bearing no. 1-34 ran for some time after the initial spall failure, resulting in substantial damage over a wide area of the surface. Slave bearing no. 1-29 is a typical early spall failure and its appearance is essentially the same as the early failures observed in the test bearings.

Figure 30 shows two M50 steel outer rings which spalled in the test bearings. Both failures appear to be typical fatigue type spalling failure.

There was no correlation between the observed surface cracks on the inner ring face and outside diameter and location of the fatigue spalls. The first six inner ring failures of test bearings occurred in bearings which had no Magnaflux indications. Bearings which were recorded with cracks did not fail due to propagation of the crack. Magnaflux checks of the inner ring serial no. 29 before and after test shown in Figure 31 indicate very little or no extension of the surface crack. In no case did a crack influence the performance of the bearing or affect the fatigue life of the bearing. On several bearings, a piece of the corner broke free during the pressing of the bearing on the arbor during installation; this type of fracture is typical for case core separation and has been noted on gears. This removed material does not affect performance of the bearing, as noted by the fact that several bearings with surface cracks operated [illegible] than 1,000 hours without fatigue failure or extension of the surface crack.

Two bearings serial no. 1 and 26 experienced a short period of overheating at 149 hours into the test. At this time the test operator observed [illegible] from the test [illegible] down. Examination of the [illegible]

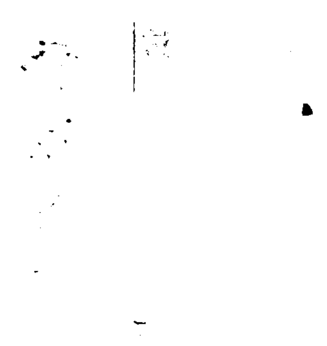


Figure 29. Slave Bearing Inner Rings Serial No. 1-34 and 1-29.

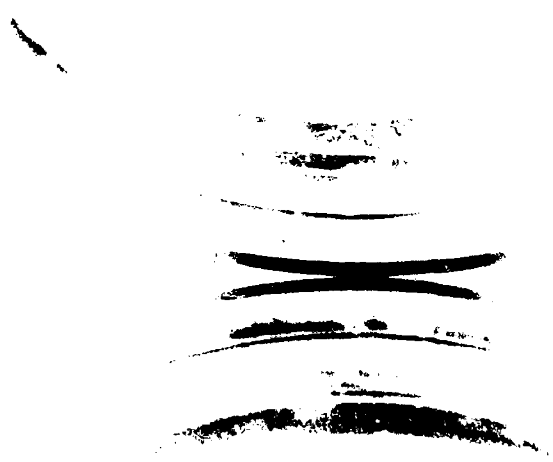


Figure 30. Outer Rings Run With Inner Rings Serial No. 13 and 3 Showing Fatigue Spalls.

BEFORE TEST

AFTER TEST

Figure 31. Face of Inner Ring Serial No. 29 Showing Surface Crack.

bearings indicated that the surface temperature of the bearings apparently reached 400°F or higher during this short period. Although the bearings were discolored, they were found to be operable and thus were returned to test. The cause of the overheating could not be ascertained, but it is believed that a spark was produced which ignited the oil and then, due to a shortage of available oxygen, the flames were extinguished. Both bearings continued testing and were suspended with more than 1,011 hours without failure. Therefore it appears that the short overheated condition did not affect the fatigue life of these bearings.

Failures of the case-carburized VASCO-X2 inner rings appeared to be typical in appearance of fatigue spalls in conventional through-hardened steel ball bearings. Two of the failed test bearing inner rings (no. 8 and no. 17) were returned to Boeing Vertol for evaluation of the cause of failure and condition of the material in the spalled areas. The condition of the two inner rings as received is shown in Figures 32, 33, and 34. Bearing no. 8 exhibited a greatly advanced surface spall, while bearing no. 17 contained a much smaller spall approximately 0.1 inch in length and 0.15 inch wide. A detailed metallurgical evaluation was conducted which indicated that the failures were due to subsurface fatigue with crack penetration to a depth of 0.007 inch as shown in Figure 35. This corresponds closely to the point of maximum shear which was used to establish case depth requirements. Figure 36 shows a closeup of the spall of bearing 17, and Figure 37 shows a 150X view of the ball track which indicates only light surface wear and debris dents. The wear has been light enough as to not remove the machined-surface finish marks. In addition, the microstructure of the case and core was examined to determine if heat treatment may have influenced the test results. Figure 38 shows an acceptable carburized-case microstructure in the spall area and also acceptable core microstructure. The checks of the initial heat-treated ring and the failed rings indicated that the VASCO-X2 material was properly heat-treated and that the material properties achieved during this test are based upon good material qualities.

CONCLUSIONS

The results of this initial 32-bearing lot test indicate that VASCO-X2 steel can be used as a bearing material for future applications. These results also indicate that a material factor of 5 or greater can be used This is within the range of factors used for CEVM M50 steel. The testing did show that proper design of the bearing geometry and adequate case depth are required to eliminate the possibility of case-core separation. It is anticipated that most conventional designs will not encounter this problem.

Based upon these results, VASCO-X2 steel is recommended as a bearing material. The use of VASCO-X2 steel as a material for the inner races of an advanced tapered-roller-bearing pinion concept is discussed in the next section. This combination allows for the use of a material that is suitable for both gears and bearings and also allows for an integration of components to simplify design and reduce parts count and fretting surfaces.

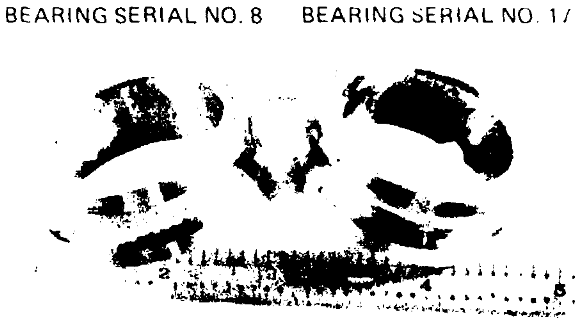


Figure 32. 207S Ball-Bearing Inner-Ring Fatigue-Test Specimens No. 8 and No. 17 Fabricated From VASCO-X2 (BMS 7-223) Alloy.

Figure 33. Test Bearing No. 8 Inner Ring Exhibiting Greatly Advanced Surface Spall 0.90 Inch in Length With 0.25-Inch Width.

Figure 34. Test Bearing No. 17 Inner Ring Exhibiting Advanced Surface Spall 0.10 Inch in Length With 0.15-Inch Width.

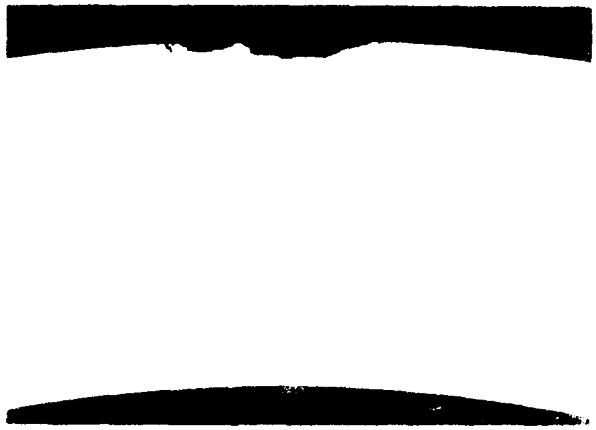
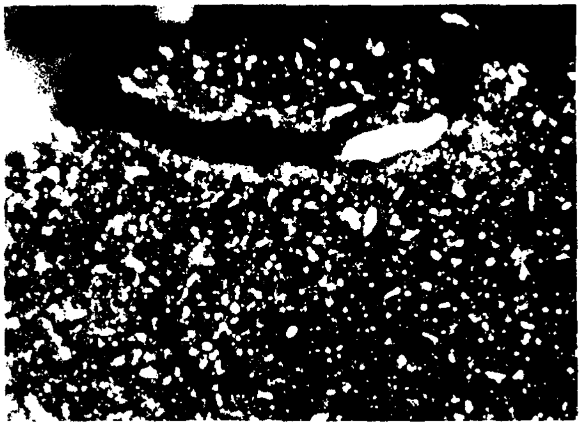


Figure 35. Circumferential Section Through Spalled Area Revealing [illegible]
Crack Penetration From Ball Track

Figure 36. Enlarged View Displaying Fractographic Characteristics [illegible]

Figure 37. Ball Track Enlarged [illegible]
[illegible] Surface Track



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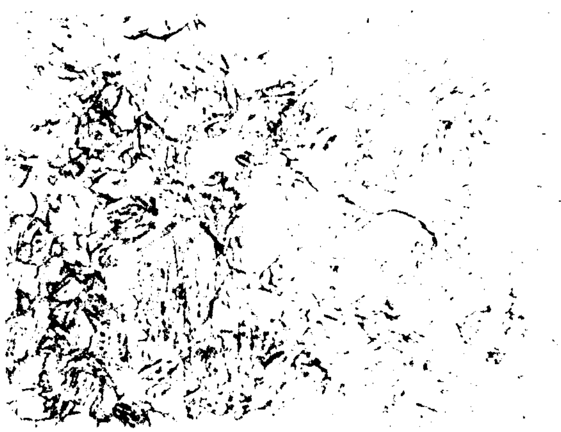

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Figure 38. Acceptable Carburized-Case Microstructure and Core Microstructure in Spalled Area of VASCO-X2 Steel Bearing Ring.

ADVANCED RIBBED-CUP TAPERED-ROLLER BEARING AND MAGNETIC SEAL TEST PROGRAM

BACKGROUND

In 1968 Boeing Vertol sponsored the first industry high-speed tapered-roller bearing research program with the Timken Company. The objective of this initial program was to develop tapered-roller bearings to support spiral bevel gearing in an advanced helicopter transmission and drive system. Tapered-roller bearings were selected in place of conventional ball and roller bearings because they offer the greatest potential for increased load capacity, increased fatigue life, and an appreciable reduction in bearing size and weight.

This development work on high-speed tapered-roller bearings was continued with a contract from the Eustis Directorate (ATL) in March of 1971 (DAAJ02-71-C-0025) to design, fabricate, test, and evaluate spiral-bevel-support tapered-roller bearings. This program consisted of a generalized analytical investigation and an experimental investigation. The results of this contract have been published in USAAMRDL Technical Report 73-16[2]. The knowledge obtained from this test program established that tapered-roller bearings provided a cost-effective means for supporting spiral bevel gears.

As a direct result of this program tapered-roller bearings were designed for the Heavy-Lift Helicopter drive system. Studies showed significant weight reduction and life improvement. Additional rig testing was conducted and was documented in USAAMRDL Technical Report 74-33[3]. In addition to rig testing, full-scale transmission tests were conducted on the HLH aft and combiner transmissions. These tests have shown that tapered-roller bearings can be used successfully to support spiral bevel gears in the actual transmission environment. These tests also provided insight into areas which could further improve the operating characteristics of tapered-roller bearings.

Several other contracts have been implemented by other organizations which have investigated higher speeds for gas turbine application (42,000 fpm), the use of high-temperature steels (M50, CBS1000M), and various bearing designs (cone rib, cup rib). Although significant advancements have been made in high-speed tapered-roller bearing technology, other areas of development needed to be pursued to achieve the desired design goals of the advanced-concept transmission.

In order to achieve the design goals of this program, several additional advanced design features in rolling-contact bearings were required. Weight reduction for bearings can be achieved in two ways: First, reduction in the number of components to achieve the same performance and life, such as making two bearings do the work of three. Second, by the integration of components to reduce the total parts count, such as making the bearing inner race an integral part of the shaft.

The advanced-concept transmission design proposed for this program illustrates how these design features can be incorporated. The input pinion design (Figure 39) shows that two tapered-roller bearings operating directly

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2. Lemanski, A.J., Lenski, J.W., Jr., and Drago, R.J., DESIGN, FABRICATION, TEST, AND EVALUATION OF SPIRAL BEVEL SUPPORT BEARINGS (TAPERED ROLLER), Boeing Vertol Company, USAAMRDL TR 73-16, Eustis Directorate, U.S. Army Air Mobility Research and Development Laboratory, Fort Eustis, Virginia, June 1973, AD769064.
3. Lenski, Joseph W., Jr., TEST RESULTS REPORT AND DESIGN TECHNOLOGY DEVELOPMENT REPORT – HLH/ATC HIGH-SPEED TAPERED ROLLER BEARING DEVELOPMENT PROGRAM, Boeing Vertol Company, USAAMRDL TR 74-33, Eustis Directorate, U.S. Army Air Mobility Research and Development Laboratory, Fort Eustis, Virginia, June 1974, AD786561.

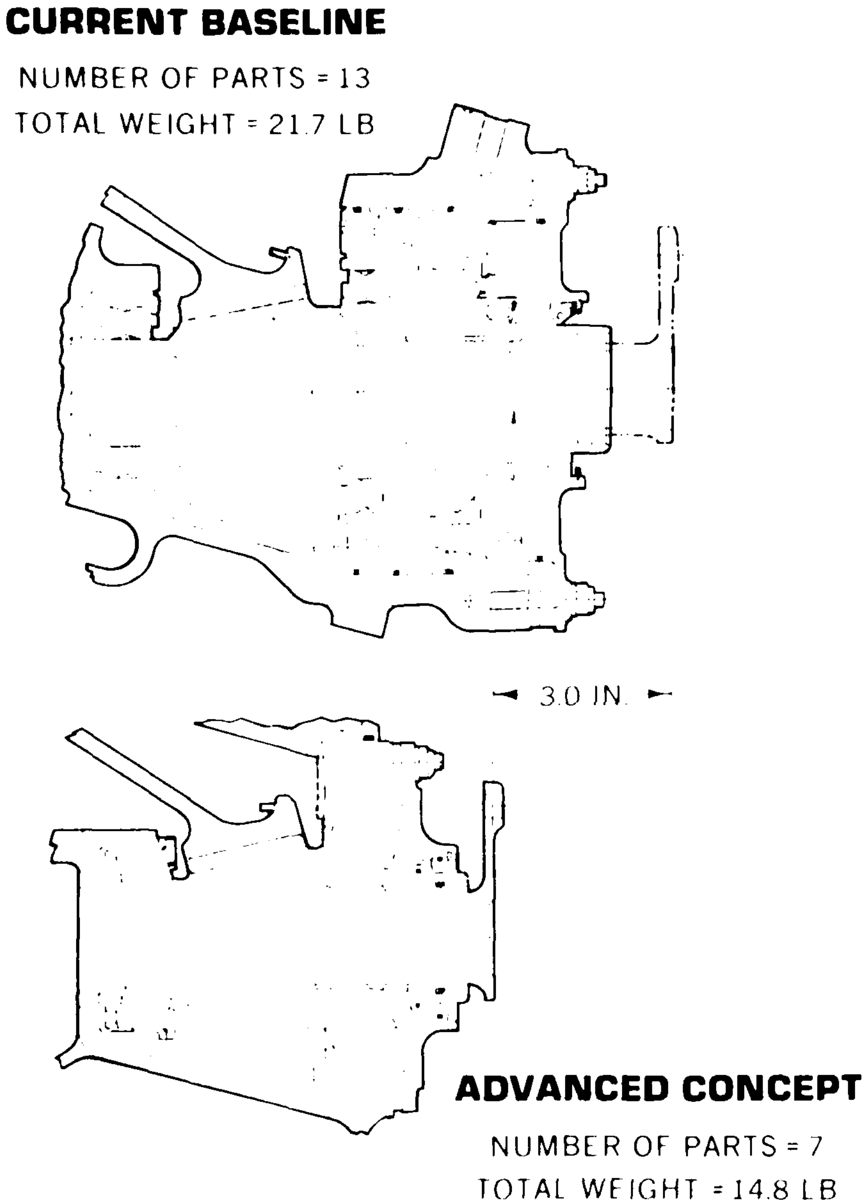

Figure 39. Design Comparisons of Input Pinion Bearing Support.

on the pinion shaft can be used to react the gear loads. Conventional design (also shown in Figure 39) required two roller bearings and a ball thrust bearing to perform the same function. In addition to the reduction of one bearing, the design also allows for simplicity of configuration, resulting in a total of six fewer major components. Preliminary weight calculations indicate a 6.9 pound weight saving for each pinion assembly and a reduction in shaft length of 3 inches. The producibility of the pinion assembly also improved because of the reduced number of components and simplicity of design.

Although the basic design concept appears simple, the concept had to be tested to evaluate several critical features that have not previously been investigated. The design used a spring loaded set of ribbed-cup tapered roller bearings which operate directly on the pinion shaft. The spring-loaded bearing floats in the housing. The proposed test program was intended to evaluate the following design features:

- Spring Preloading — The spring is used to maintain a constant preload on the set of tapered roller bearings and allows for thermal adjustment during operation. The design and establishment of the spring load were to be evaluated under simulated operating conditions to insure that proper bearing preload is maintained during all phases of operation. The spring feature should allow for increased operation after loss of oil because it will accommodate axial expansion due to heat buildup without adding to the internal preload. The buildup of internal preload results in rapid deterioration of a tapered-roller bearing.

- Floating Bearing — To insure that the spring load functions properly, one bearing in the set must be free to float axially. The advanced-concept design proposes the use of a pressurized oil annulus to maintain an oil film between the bearing cup outside diameter and the housing liner inside diameter. This oil film is intended to reduce the sliding friction and insure that the bearing cup will move freely in the axial direction under the spring load force. Proper design and operation of this feature are critical to the success of the spring-loaded tapered-roller bearings.

- Ribbed-Cup Tapered-Roller Bearing — Although ribbed-cup tapered roller bearings have been produced, there exists very little experience concerning their performance under relatively high speed operation (greater than 7,000 rpm). Work conducted by SKF under contract DAAJ02-70-C-0047 and documented in USAAMRDL Technical Report TR 73-46[4] indicated that this type of design is feasible for high-speed applications. These tests were conducted on four small bearings under the limited loading condition of thrust only.

  To provide adequate lubrication coverage of a cone rib designed tapered-roller bearing, many oil holes are required; the number of holes is a function of speed and bearing size. As many as 44 holes were required for HLH-type bearings. This results in increased costs and difficulty in manufacturing the bearing as an integral part of the shaft. The cup-rib design does not require a large number of oil supply holes because it provides a natural trap for all oil passing through the bearing from the small end (Figure 40). It is anticipated that only four oil holes will be required to adequately lubricate this type of bearing. In addition, this design eliminates the need for a critical flange to be machined as part of the shaft and reduces the possibility of damaging the shaft gear component due to a scuffed rib. An additional feature of this type of bearing design is that the natural trap for oil (Figure 40) in the cup should provide extended oil-off operational capability and always provide an oil flooded condition for rib roller contact.

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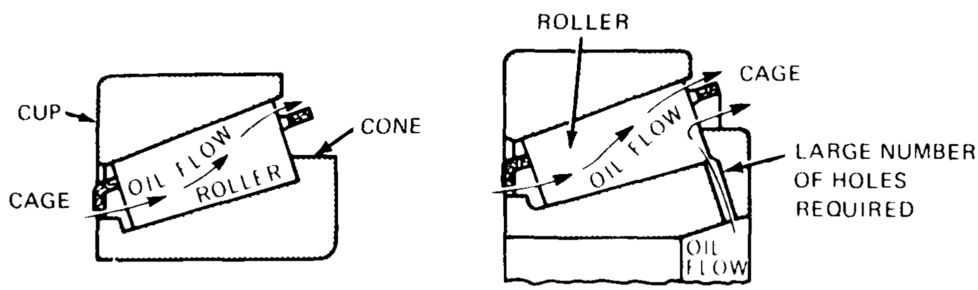
4. Conners, T.E., and Morrison, F.R., FEASIBILITY OF TAPERED ROLLER BEARINGS FOR MAIN SHAFT ENGINE APPLICATIONS, SKF Industries, Inc., USAAMRDL TR 73-46, Eustis Directorate, U.S. Army Air Mobility Research and Development Laboratory, Fort Eustis, Virginia, August 1973, AD771984.



CURRENT TECHNOLOGY

ADVANCED TECHNOLOGY*

- SPEED BREAKTHROUGH, OVER 42,000 FPM (PREVIOUS LIMIT 6,000 FPM)
- IMPROVED DESIGN AND LUBRICATION

*THIS DESIGN REQUIRES MANY SECONDARY LUBRICATION HOLES TO THE CONE RIB RACEWAY CONTACT TO SUPPLY THE NECESSARY OIL TO PREVENT RIB SCUFFING. THIS SYSTEM DOES NOT TAKE ADVANTAGE OF THE OIL FLOW PATH NOR DOES IT TRAP OIL FOR EMERGENCY OIL-OFF OPERATION.

PROPOSED RIBBED-CUP DESIGN

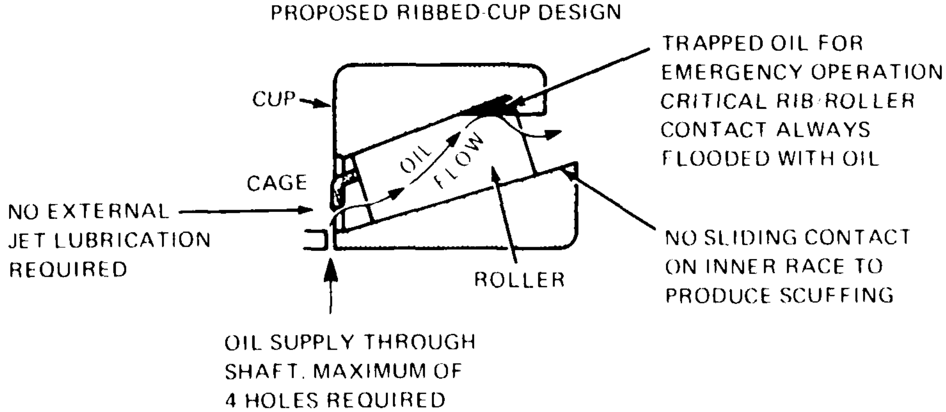


Figure 40. High-Capacity, High-Speed Tapered-Roller Bearings.

- Lubrication   Initial plans call for the lubrication of these bearings by oil supplied to the shaft bore and distributed by holes in the shaft to the bearings. No external jet lubrication is planned for these bearings. This method of lubrication should eliminate the risk of blocked jets due to foreign material contamination. Tests will be conducted to establish required oil flow and optimum location of the lubrication holes in the shaft. In addition, the failsafe operation of these bearings will be investigated using an auxiliary oil supply and loss of oil. Minimum oil flow to maintain satisfactory operation will be determined.

- High-Hot-Hardness Carburizing Steel   To enable the integration of components such as bearing races and gear shafts, a material which is suitable for both gears and bearings must be considered. Boeing Vertol has used VASCO as a gear material for many years with good success. To evaluate the use of VASCO-X2 steel as a bearing material, ball bearings were fabricated and tested to establish the rolling-contact fatigue properties of VASCO steel. This program was conducted before the start of the tapered-roller bearing program and the results are reported in the preceding section of this document. This test confirmed that a common steel can be used in an integrated gear and bearing system to achieve good performance for both the bearing and gear. The final evaluation of the integrated design was to be conducted under this test program.

- Magnetic Shaft Seal   The magnetic input shaft seal incorporates a face seal of carbon urged into contact with a lapped-steel runner by magnetic force. The expected benefit in this application is extended life compared to a lip seal, mechanical simplicity, and hence reliability compared to a spring-loaded face seal. The shaft speed is not considered excessive for standard seals; however, field and test experience indicated that seal leakage and wear are still a major problem. The use of a magnetic shaft seal should provide the needed improvement to achieve the objectives of this program. The seal will be evaluated for wear and leakage rates under the simulated environment of an input pinion. The effect of the pumping action of the tapered-roller bearing will be one of the determining factors for good performance.

Although the advanced transmission components investigation contract was awarded in June of 1976, the evaluation of the tapered-roller bearing phase was not initiated until August of 1977. This delay was required to complete the evaluation of the high-hot-hardness carburizing steel as a bearing material. The results of this program provided the confidence that the integration of bearing and gear components would result in the expected performance, as mentioned earlier. In August of 1977 the Timken Company of Canton, Ohio, contracted to fabricate and test an advanced-concept ribbed-cup tapered-roller bearing. This program duration was from August 1977 to September 1979.

The objective of this subcontract was to design, fabricate, and demonstrate a ribbed-cup tapered-roller bearing on a simplified input bevel pinion design. Nonstandard features of this high-speed tapered-roller bearing were a ribbed cup, the inner race or cone integral to the shaft, full through-shaft lubrication, and a completely machined outer-land-riding C-type cage.

Specific items for investigation were bearing performance, heat generation, lubrication requirements, mounting (spring preload, floating-cup looseness, and antirotation devices), and endurance. At the conclusion of this effort and considering the excellent condition of all test components, an extension of the program was awarded to the Timken Company in September of 1979. This add-on phase consisted of six oil-off survivability tests. These tests were incorporated into the initial effort and were completed by December of 1979. The complete results of this effort are documented in this section of this report.

BEARING DESIGN CRITERIA

Initial trade studies conducted to develop the advanced transmission resulted in the preliminary design of ribbed-cup tapered-roller bearings to support the bevel input pinion as shown in Figure 39. These design studies were conducted based upon the following design criteria.

- Maximum input pinion power — 13,513 in.-lb (1,543 hp)
- Input pinion speed — 7,196 rpm
- Bearing B-10 life — 900 hr*

*The 900-hour life is based upon no material factor and at a cubic mean load which is approximately 67 percent of maximum load.

During this preliminary design phase, both the inboard bearing (toe) and the outboard bearing (heel) were configured as shown in Figure 39. To minimize the cost of this test program, only the heel position was considered for full development. This approach enabled the Timken Company to tool up for one size bearing and did not greatly influence the intended results of this program. The main influence of this change in design was that the toe bearing was oversized for the imposed loads and the applied test loads had to be modified to maintain the desired loads on the heel bearing. The end results were that the heel bearing position experienced the same loads as expected in the actual advanced-concept transmission and the toe bearing loads varied as required to maintain this approach. The initial size of the heel bearing is shown in Table 8.

Based upon this initial design and the specified loads, an iterative technique was used to arrive at the final optimum design of the test bearing.

This approach consisted of the following steps:

1. Capacity and envelope dimensions supplied to the Timken bearing design group.
2. Bearing design group produced geometry details and sketches.

TABLE 8. VARIATIONS OF BEARING DESIGN

	Initial	Final
Outside Diameter (in.)	4.9606	5.0000
Cup Width (in.)	1.3780	1.7717
Roller I.E.D Inside Diameter (in.)	3.2000	3.4714
Cone Angle (deg, min)	18	16, 26
Fatigue Life (hr) at 7,177 rpm under cubic mean loading*	900	780
Roller Spherical End Radius (percent)	80	80
Cage	Machined steel, silver plated	Machined steel, silver plated

*Final design hours were calculated based on catalog approach with no material factor. Adjustments for positive lubrication and load zone effects produced an L-10 life of 1,693 hours. The results of the evaluation of VASCO-X2 steel indicate that a material factor of 5 could be used. This would result in an L-10 life of 8,465 hours.

TABLE 9. SUMMARY OF TEST COMPONENTS

Component Name	Drawing No.	Quantity	Manufacturer	Material	Comments
Test Bearing	Boeing Vertol SK27541	16	Timken	Cup CBS600 Steel Rollers CBS600 steel Cage 4320 steel	Timken part no. EX23926A/ EX23926D
Slave Bearing	Timken B-63421	5	Timken	Cone CBS600 steel Cup CBS600 steel Rollers CBS600 steel Cage 1010 steel	Timken part no. EX2456CC/ EX2456DD
Test Shaft	Timken E-32940	3	American Lohmann Corp and Timken	VASCO-X2 steel	American Lohmann to manufacture shaft except for final grind of bearing journals; Timken to complete final grind of part no. EX23926-S
Magnetic Seal	Magnetic Seal Corp 75046-1	8	Magnetic Seal Corp	Carbon graphite G5100, Alnico 5, 416 stainless steel	
Adapter	Magnetic Seal Corp 75046-1-5	6	Magnetic Seal Corp	303 stainless steel	Adapter required because of steel seal retainer used in the test rig
Belleville Spring	Timken A-43030	As required	Timken	Steel	Several springs manufactured to provide various preloads as established by test

3. Designs were analyzed for endurance life, normal loading, Hertzian stresses, EHD films, and heat generation.

4. New data was supplied to bearing design group, and steps 2 and 3 were repeated until the final design was approved.

Six iterations were required before the test bearing size was finalized.

The initial design and final design specifications for bearing envelope dimensions and fatigue life capacity are given in Table 8.

Based upon this final design, the remaining components (slave bearings, test shaft, and magnetic seal) were designed to simulate the input pinion bevel gear as closely as possible in order to properly evaluate this concept.

MATERIAL SELECTION AND COMPONENT FABRICATION

Eight complete test assemblies were fabricated during this program. Each test assembly consisted of two test bearings, one test shaft, one magnetic seal, and two slave bearings. For each assembly, new components were used except for the slave bearings. Table 9 provides a summary of all the components fabricated during this program. Further details concerning design, material selection, and fabrication of each component are provided in the following sections.

Test Bearings

Sixteen test bearings were manufactured to the Boeing Vertol drawing requirements shown in Figure 41. Each test bearing consists of an EX23926D ribbed cup, an EX23926A cage, and a set of 17 LX23926B rollers.

The material selected for the fabrication of these components was CEVM CBS600 high-temperature steel. The CBS600 steel was selected because of Timken's experience with this steel as a bearing material. Its properties are also very similar to the VASCO-X2 steel which was used as the material for the inner race which is an integral part of the gear shaft. The chemical composition of CBS600 steel is shown in Table 10.

TABLE 10. CHEMICAL COMPOSITION RANGES OF CUPS AND ROLLERS OF TEST BEARINGS

Material CEVM CBS600						
C	Mn	P	S	Si	Cr	Mo
0.16/0.22	0.40/0.70	0.025 max	0.025 max	0.90/1.25	1.25/1.65	0.90/1.10

The bearing cage is an outer-land-guided, C-type cage completely machined from AISI4320 steel. Prior to silver plating, the cages were dynamic-balance tested. All cages except serial no. 78-27 had a total imbalance of less than 3 grams per centimeter as measured at both pilots; number 27 had a total imbalance of 3.29 grams per centimeter. The bench diametrical clearances for the cage piloting surfaces were 0.006 to 0.016 inch after silver plating. The silver plating conformed to Federal Specification QQ-S-365b, Type II, Grade B, and was applied to a thickness of 0.001 to 0.002 inch.

Before assembly and test, each bearing component was closely inspected to insure uniform quality. The integral race of each test shaft was traced to check angle, contour, and surface finish. The inspections of the

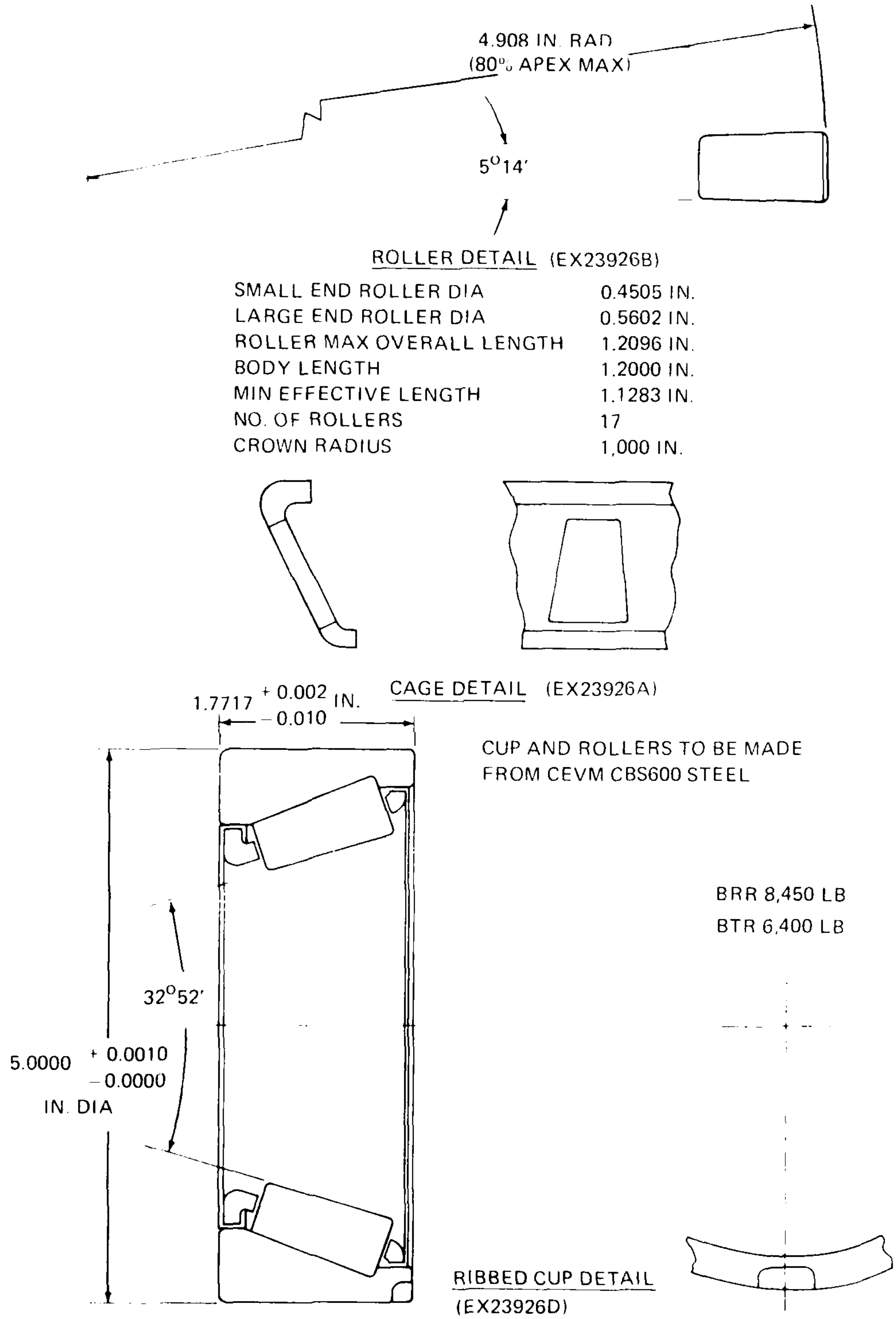


Figure 41. Design of Tapered-Roller Bearing for Test Series.

ribbed cups and rollers included angle checks, profile traces, roller size and spherical end geometry, surface finish measurements, etc. The cups, rollers, and cages were processed according to the Timken Company's "Critical Part" manufacturing and traceability plan. Typical profile traces of the roller spherical end radius (Figure 42), roller body (Figure 43), cup raceway (Figure 44), and cup rib face (Figure 45) are shown for bearing no. 79-5 which was used in assembly number 5.

Slave Bearings

The test shaft design for this program had provisions for two rows of slave bearings which were used to apply the equivalent gear reaction loads (radial, thrust, and moment) to the test bearings. The bearings selected for the slave bearing location were chosen from previously designed high-speed tapered-roller bearings with lube holes to the cone rib face. The bearing selected was substantially larger than the test bearings and therefore fewer bearings were required to complete this test program. Five EX2456CC-EX2456DD bearing assemblies similar to the HM624700-series bearing were fabricated to be used as slave bearings. Figure 46 is an assembly drawing of the slave bearing.

The slave bearing cone, cup, and rollers were also made from CEVM CBS600 steel. The bearing cage is an L-type, completely stamped cage made from AISI1010 hot-rolled steel. The cages were silver-plated according to Federal Specification QQ-S-365b, Type II, Grade B, to a thickness of 0.001 to 0.002 inch.

Test Shaft

The test shaft with integral bearing inner races was fabricated through a joint effort of the American Lohmann Corporation and the Timken Company. The Boeing Vertol Company supplied the VASCO-X2 steel to the American Lohmann Corporation to heat-treat and perform the finish-machining of the shafts except for the finish-grinding of the bearing races. The bearing races were finish-ground by the Timken Company. This sequence of operation is similar to what would be expected in a production run of gear shafts.

The material selected for fabrication of the test shaft is VASCO-X2 single-vacuum-melt steel per Boeing Vertol specification BMS 7-223. This is the same material used in the earlier testing of MRC 207S ball bearings and is also the material used for gear application at Boeing Vertol. The chemical and physical properties of the heat lot used to fabricate the test shafts are shown in Table 11.

The test shaft was designed by Timken and is shown in Figure 47. The shaft was then machined and heat-treated by the American Lohmann Corporation. The only areas not completed were the finish-grind of the two race journals for the test bearings. The heat treatment was conducted per Boeing Vertol manufacturing requirements and is the same procedure used for standard transmission gears fabricated from VASCO steel. The following specifications were to be met on the cone surfaces of the shaft after carburizing and final heat treatment.

a. Rc 58 minimum at 0.035 inch after a grind allowance of 0.018 inch minimum

b. Rc 50 minimum at 0.070 inch after a grind allowance of 0.018 inch minimum

c. Surface hardness of Rc 60 minimum after grind

Nine shaft specimens were processed in order to achieve the required eight test shafts. Prior to final test heat treatment, a sample slug of VASCO-X2 steel was checked to verify the heat-treated microstructure, hardness, and carburized-case depths.

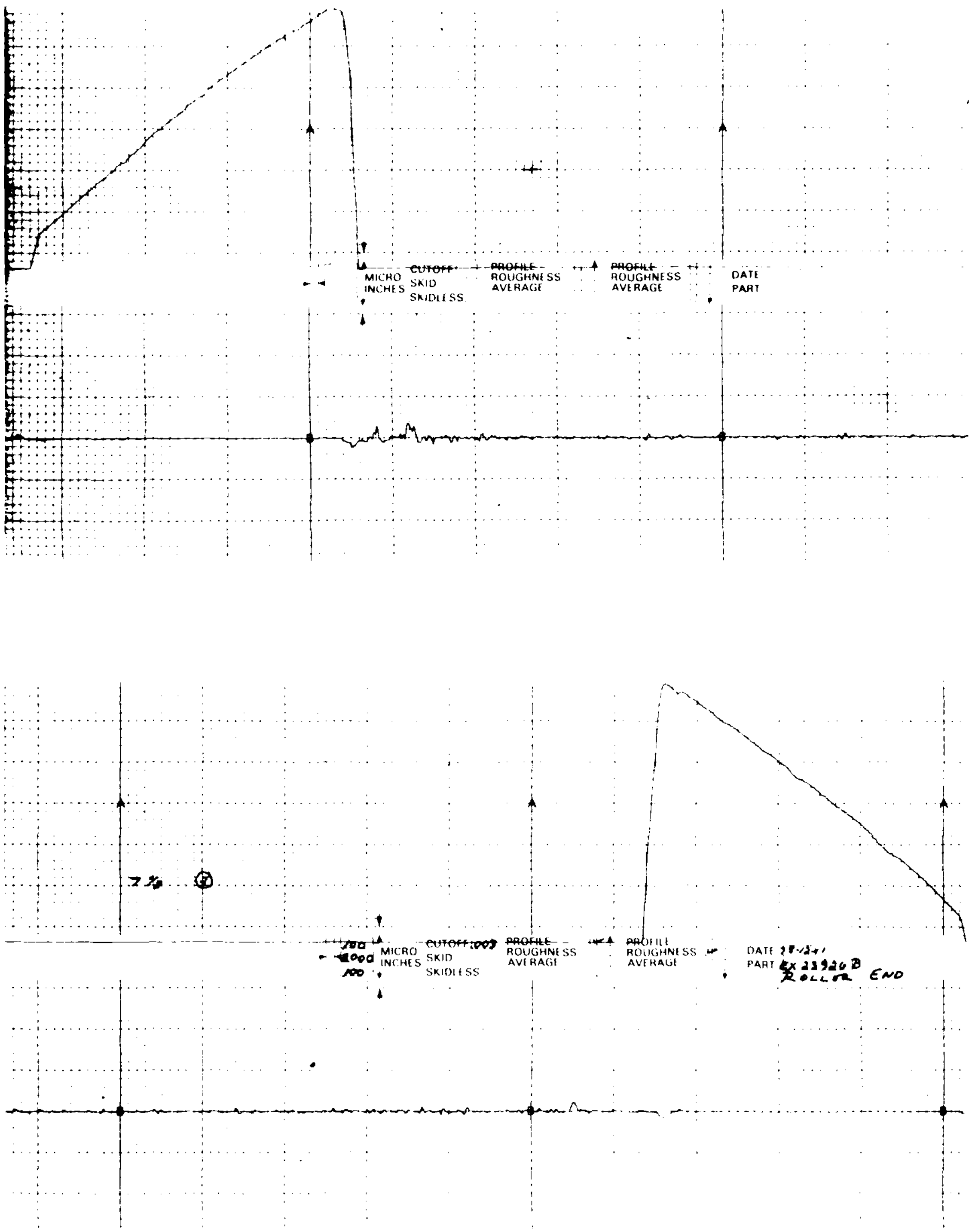


Figure 42. Typical Profile Trace of Roller Spherical End Radius Before Test.

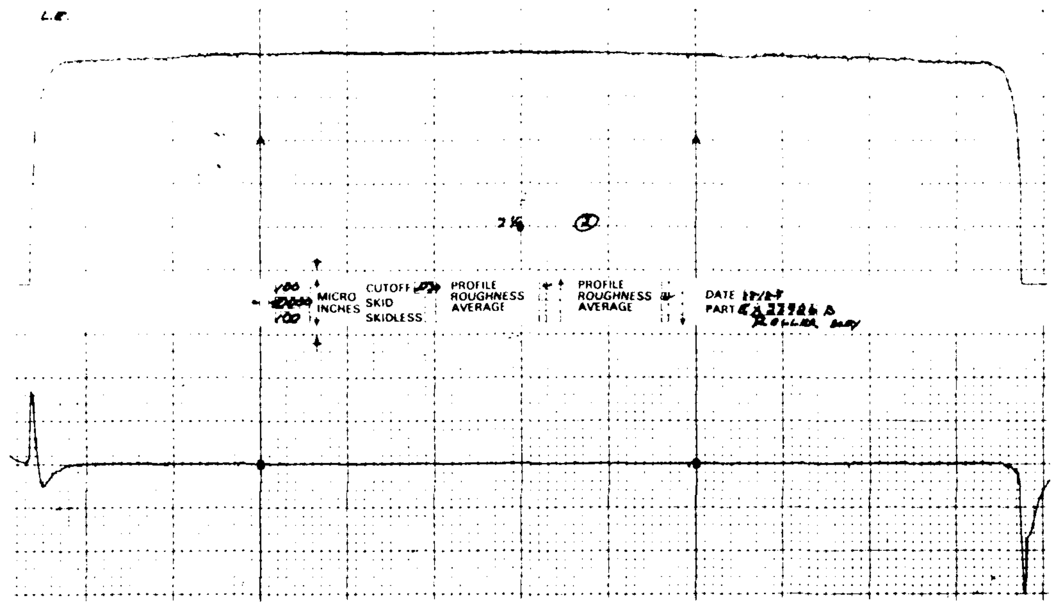


Figure 43. Typical Profile Trace of Roller Body Before Test.

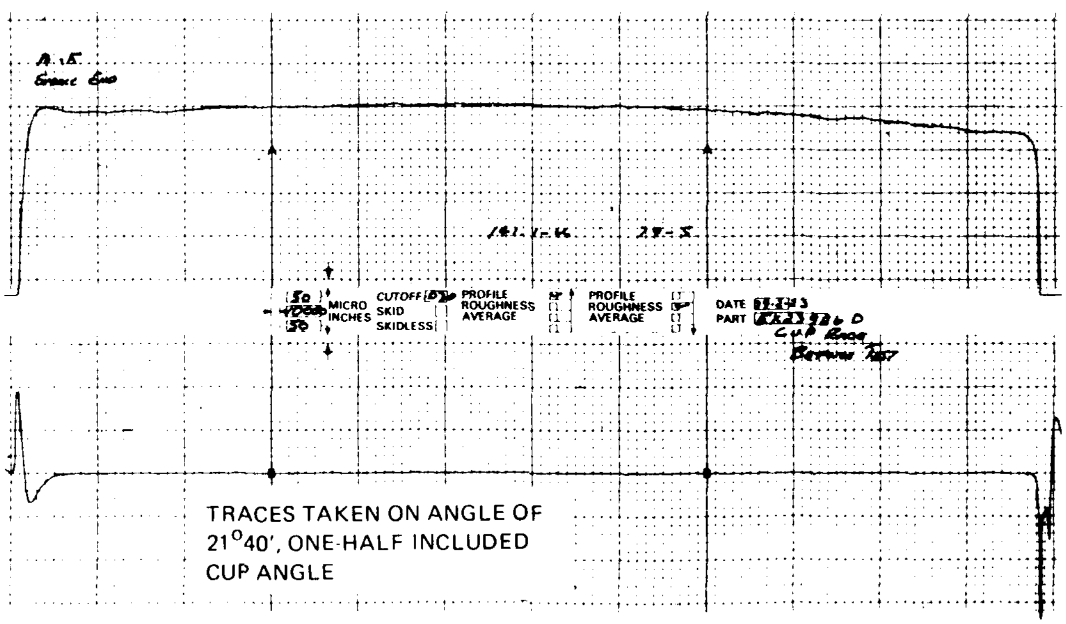


Figure 44. Typical Profile Trace of Cup Race Before Test.

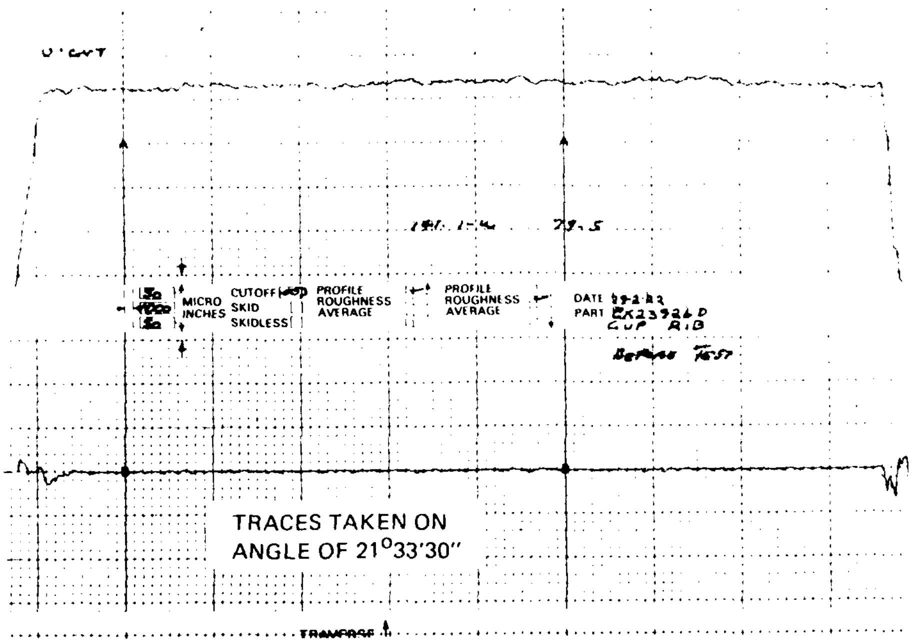


Figure 45. Typical Profile Trace of Cup Rib Face Before Test.

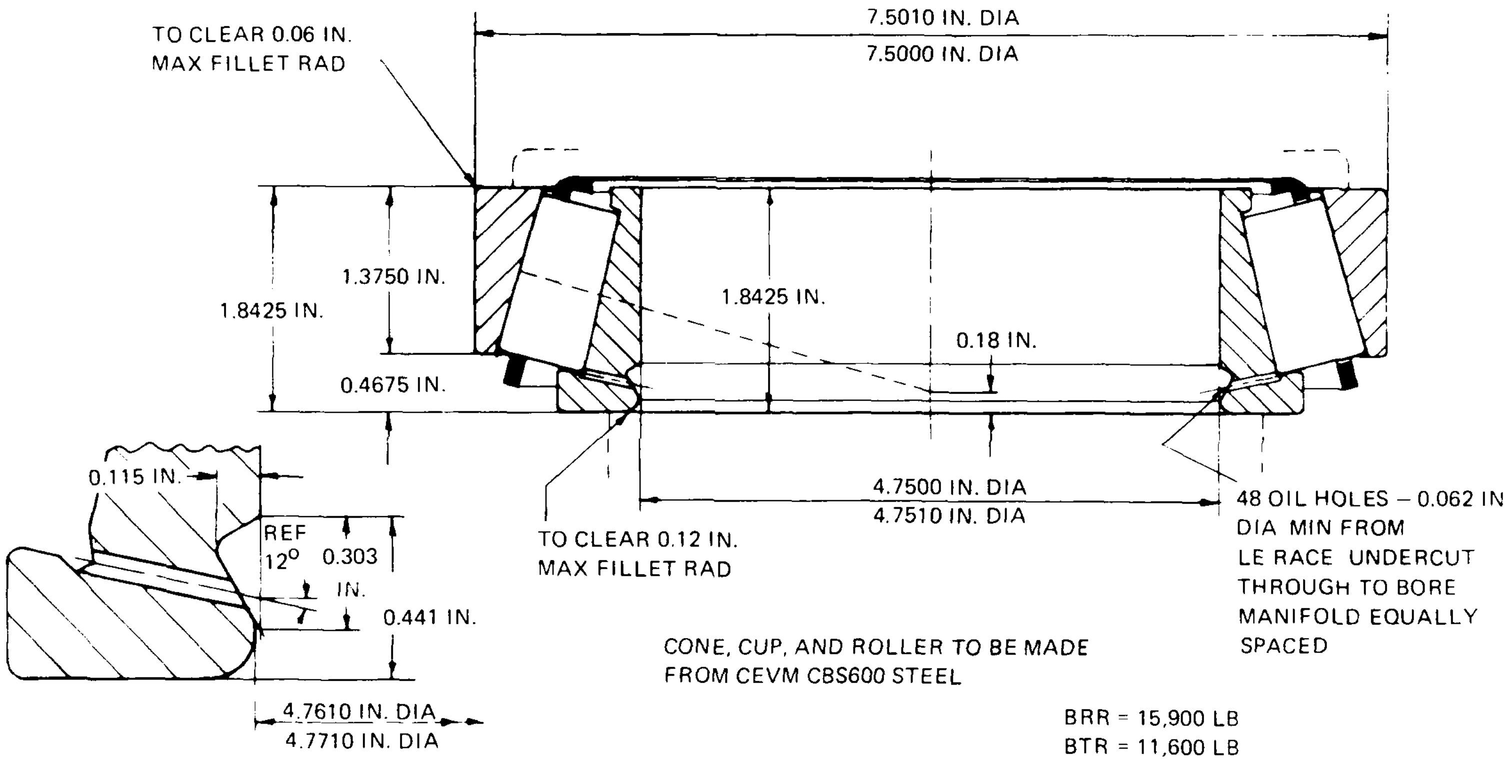


Figure 46. Assembly Drawing of Slave Bearing.

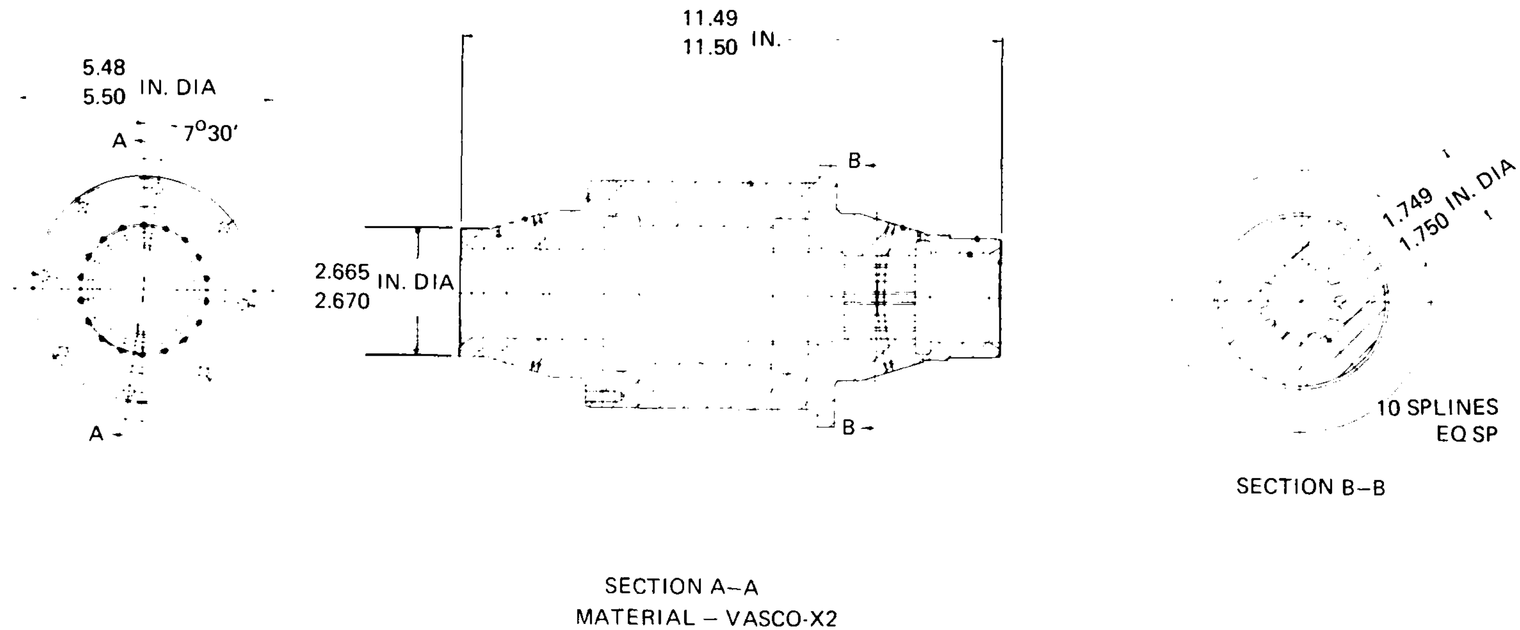


Figure 47. Test Shaft With Integral Bearing Inner Races.

TABLE 11. ANALYSIS OF MATERIAL FOR TEST SHAFTS

Brand: CVM VASCO-X2 MOD 0.13/0.16C BOEING SPEC BMS-7223

Size	Bars	Weight	Heat No.
6-1/8 in. rd	6	3,488 lb	3191-A

Macrostructure: Satisfactory

Grain Size - T -- 5-1/2
1B 6

Magnetic Particle Inspection: Top F/S 0/0
Bottom F/S 0/0

Jominy Hardenability:

	J1	J4	J8	J12	J16	J24	J32
Top	38.0	38.5	38.5	38.0	37.8	36.8	36.4
Bottom	41.5	42.0	41.8	41.2	41.2	41.2	40.8

J-K Rating:

	A		B		C		D	
	Thin	Heavy	Thin	Heavy	Thin	Heavy	Thin	Heavy
Top	1	0	½	0	0	0	1½	0
Bottom	1	0	0	0	0	0	1½	½

Heat No.	Analysis C	Si	Mn	S	P	W	Cr	V	Mo
3191-A									
Top	0.15	0.94	0.26	0.005	0.017	1.31	4.85	0.43	1.35
Middle	0.16								
Bottom	0.15	0.94	0.26	0.006	0.015	1.32	4.84	0.46	1.35

A 1/2-inch-thick slug was cut from a larger test specimen and polished to metallographic surface quality. The polished specimen contained two opposing carburized surfaces and two others containing essentially core carbon; that is, no carburized case.

Microhardness measurements were made at 0.005-inch intervals from the surface with a Tukon hardness tester using a 500-gram load and a Knoop diamond penetrator to assess case and core hardness along with hardness gradient in depth on both carburized surfaces. The results are listed in Table 12.

The specimen was then etched to establish case and core microstructures. A 4-percent nital solution was used to reveal the case microstructure (especially retained austenite) and a 10-percent Fe CL solution to better define the core structure which would not etch in nital. Microstructures were then rated visually at 500X and the [illegible] Table 12 also.

[illegible] were taken to show the essential features of the case structures at the surface after grind [illegible] Figures 48 and 49).

TABLE 12. CASE DEPTH, HARDNESS, AND MICROSTRUCTURE TEST DATA FOR TEST SHAFT

A. Case Depth and Hardness

	Rc Hardness, by Conversion from Knoop	
	As Heat Treated	After 0.018-in. Grind Allowance
Case Hardness	61/62	61/62
Core Hardness	43	43
Case Depth to Rc 60	0.045 in.	0.027 in.
Case Depth to Rc 58	0.057 in.	0.039 in.
Case Depth to Rc 50	0.097 in.	0.079 in.

B. Microstructure in Depth (observed visually at 500X)

Depth	Microstructure
Surface to 0.010 in.	Full-grain boundary carbide network, along with large spheroidized carbides in martensite plus 15% austenite
0.010 to 0.015 in.	Broken carbide network, with finer spheroidized carbides in martensite plus 15% austenite
0.015 to 0.060 in.	Fine spheroidized carbide in martensite with austenite decreasing from 15% to essentially zero at 0.060 in.
0.060 to 0.120 in.	Martensite with some fine carbides
0.120 in. on	Low-carbon martensite with 15 to 20% ferrite

C. X-Ray Retained-Austenite Results, Cr Radiation

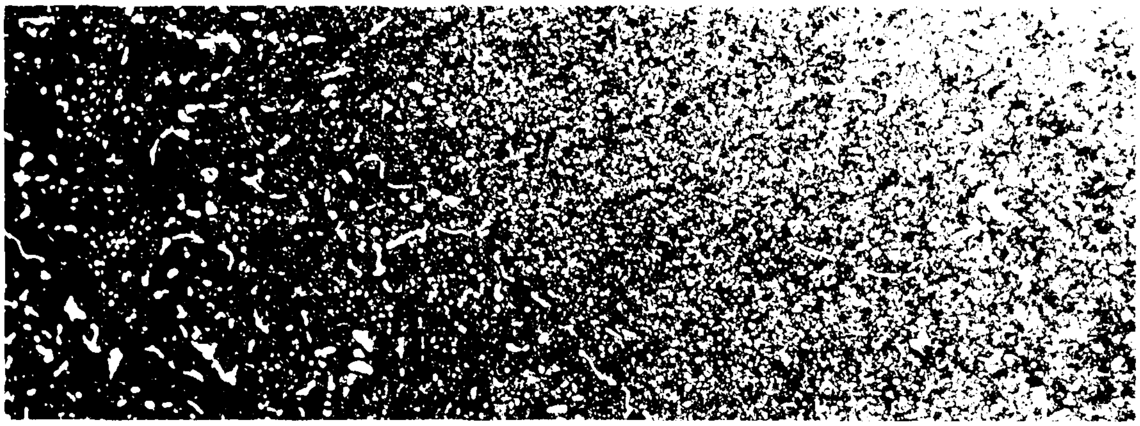
Depth (in.)	Percent	Other Constituents
0.005	19.9	$M_{23}C_6$ carbides
0.020	17.5	$M_{23}C_6$ carbides

Finally, X-ray retained-austenite measurements were made on both carburized surfaces at the 0.005- and 0.020-inch depths; the resulting data is also given in Table 12.

The data in Table 12 confirms that both the case depth aims noted earlier, as well as the desired case hardness and microstructure specifications, would be met in the cone shafts after the 0.018-inch grind allowance was applied. All undesirable case microstructure components, such as network carbides, would be removed at a depth of approximately 0.013 to 0.015 inch, which is within grinding stock limits.

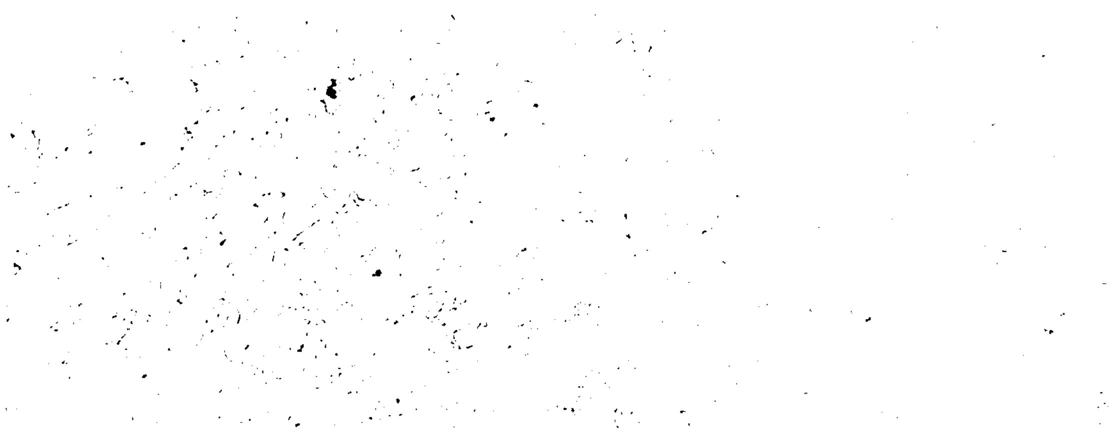
All nine test shafts were completed by the American Lohmann Company per the requirements of Figure 41. The nine shafts were then shipped to the Timken Company for the final grind operation on the two cone journals. After final grinding, the integral race of each test shaft was traced to check angle, contour, and surface finish. An example of the traces from shaft no. 78-4, which was used in test setup 5, is shown in Figure 50.

10% FECL ETCH 500X



0.010 INCH ——— DEPTH ———→ 0.020 INCH

Figure 48. Case Microstructure of VASCO-X2 Specimen Showing Typical Carbide Distribution at What May Be Working Surface of the Cone Shaft After Grind.



10% FECL ETCH 500X

Figure 49. Core Microstructure of VASCO-X2 Specimen at Depths Greater Than 0.120 Inch From the Surfaces

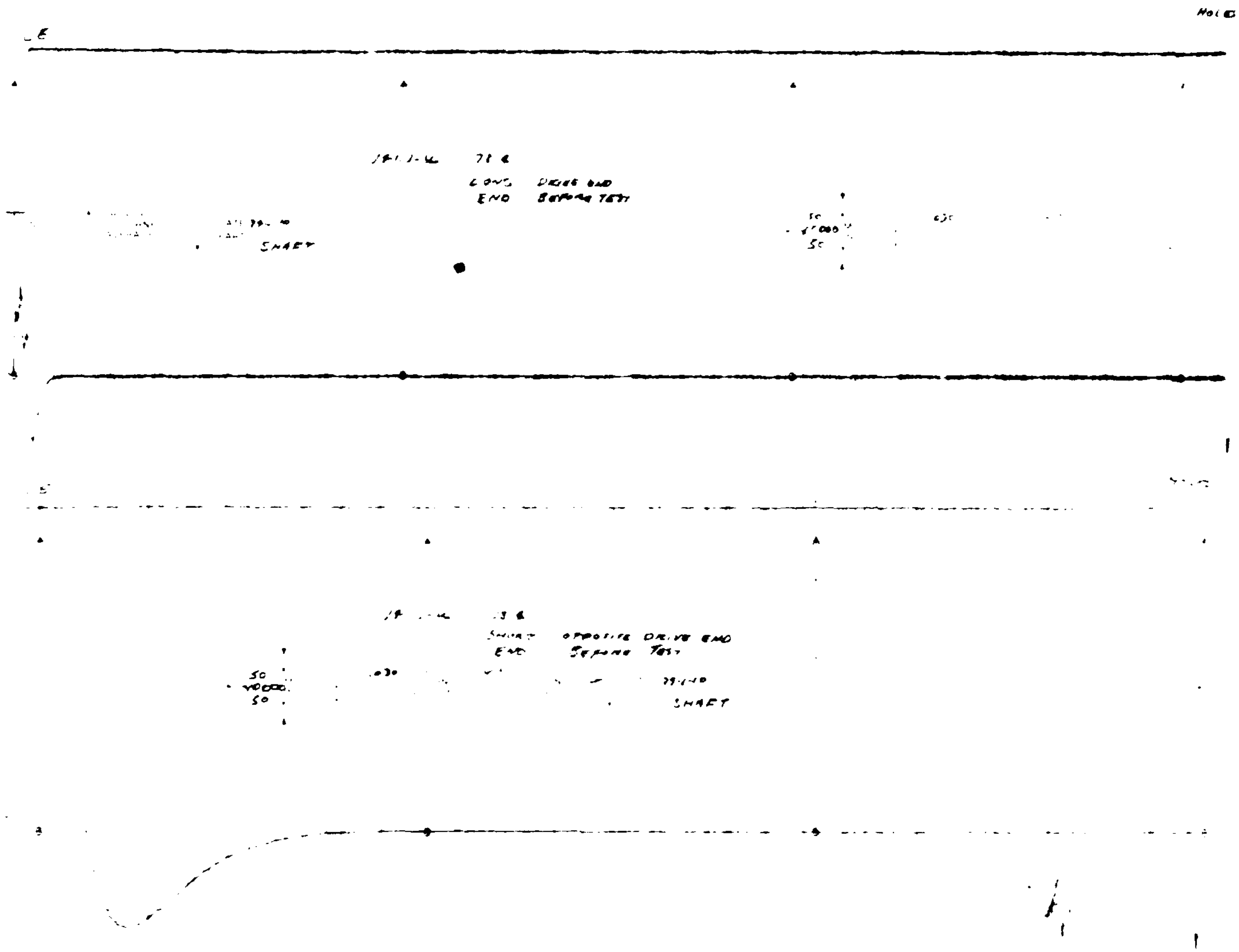


Figure 50. Race Traces From Both Ends of Test Shaft Before Test

Magnetic Seal

The magnetic seals used in this test program were furnished by the Boeing Vertol Company to the Timken Company for test. The magnetic seal design was conducted by the Boeing Vertol Company and the Magnetic Seal Corporation of West Barrington, Rhode Island. The seal design was based upon the requirements of the advanced-concept input pinion assembly which is shown in Figure 39. The design configuration of the magnetic seal is shown in Figure 51. The only difference between this magnetic seal and that of a production seal is the use of an adaptor between the seal and the test rig end cap. This adapter is an insulator which is required if the seal is placed in a magnetic housing. In production, transmission housings will be either magnesium or an advanced-composite material, neither of which will require this adaptor.

Provisions were made on the drive end (Figure 52) of the test shaft to evaluate the magnetic seal under conditions simulating an input pinion application. The magnetic seal assembly consists of two basic components:

1. A magnetized ring having an optically flat sealing surface fixed in a housing
2. A rotating ring with a carbon insert sealing surface.

The rotating member is fabricated from a magnetic stainless steel and floats axially along the shaft. The sealing surfaces of the stationary and rotating components are held together by a uniform magnetic force, creating a positive seal with minimum friction between sealing faces and proper alignment of surfaces through equal distribution of pressure.

Magnetic seals have the potential of operating for thousands of hours without excessive wear. The wear rate can vary widely with different operating conditions. The performance of a magnetic seal is largely dependent upon face load, surface speed, temperature at the seal interface, and the coefficient of friction; seal performance will be evaluated as part of the tapered-roller bearing test program.

In order to keep the two lapped sealing surfaces of a magnetic seal closed during the absence of hydraulic pressure, it is necessary to provide some form of mechanical load. Magnetic force is used which provides a reliable and uniform method of providing the specific face load necessary to insure a positive seal.

Unlike spring-loaded seals, magnetic seals operate at the specific face load for which they are designed with no variables under normal operating conditions. Manufacturing tolerances and stackup which cause variations in spring deflection and load no longer have to be considered. The magnetic seal is self-positioning on the shaft and its face load is unaffected by the shaft-housing relationship. For this reason the seal can be designed with the minimum amount of face load, usually about 0.5 to 0.75 of the value of spring-loaded seals.

The single most critical dimension of a face-type magnetic seal is the degree of flatness of its sealing surfaces or faces. Flatness is defined as the distance between two parallel planes which entirely contain the surface of a seal face. Magnetic Seal Corporation's standard manufacturing tolerances call for seal faces to be lapped flat within two helium light bands (23.2 millionths of an inch). This measurement is made by the use of a monochromatic light and an optical flat. Monochromatic light is light in which one wave length predominates. An optical flat is a flat, transparent test surface having no magnifying power.

When a series of bands occurs between two nearly flat surfaces, there is a wedge of air between them. The slope of the wedge is at right angles to the bands. The bands locate steps of 11.6 millionths of an inch vertical distance from the surface being tested to the optical flat when a helium monochromatic light is used.

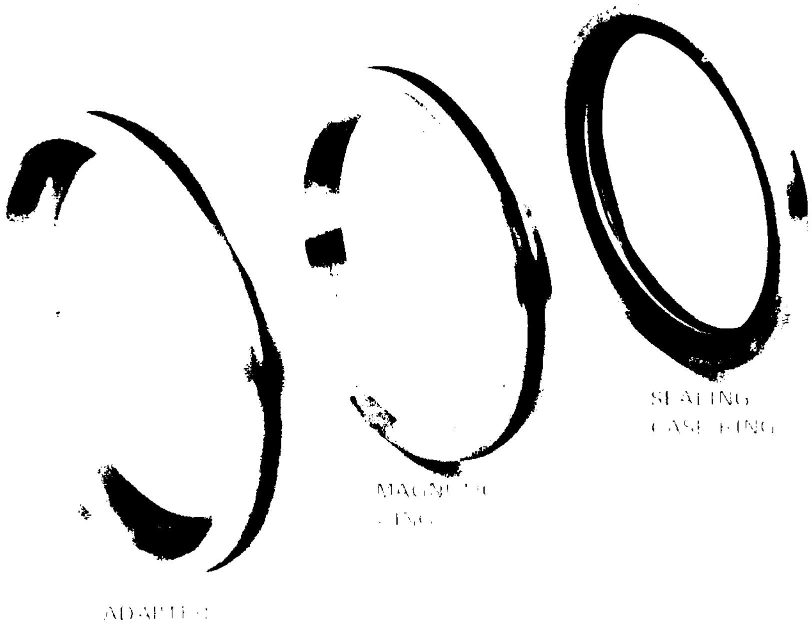


Figure 51. Bearing Test Rig Magnetic Seal.

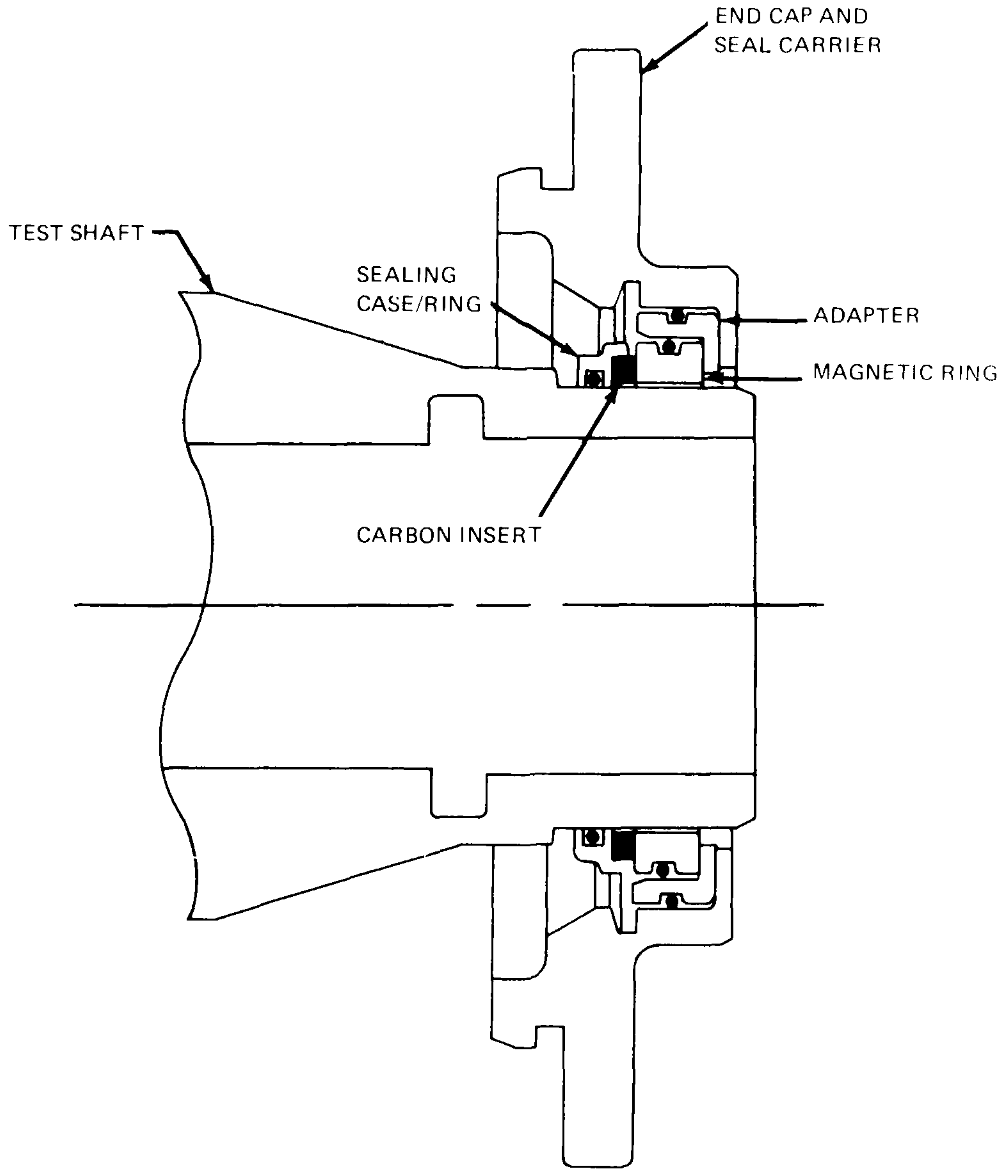


Figure 52. Magnetic Seal Tested in Conjunction With Advanced-Concept Transmission Tapered-Roller Bearings.

Bands occur because light reflections from the two surfaces which form the air wedge either interfere with or reinforce each other, according to the thickness of the air wedge. Interference of two reflections causes darkness and occurs where the air wedge thickness is exactly one-half the wave length for multiples of the light used. Parallel dark bands thus form up and down the slope of the air wedge at zones where the wedge thickness changes by one-half wave length. Halfway between each pair of dark bands a reinforcement produces a bright band. When viewed perpendicularly in helium light, the dark bands are located where the air wedge thickness changes by equal intervals of 0.0000116 inch.

Because of this high degree of flatness maintained in the seal faces, magnetic seals will create an effective positive seal upon installation either statically or rotating. No run-in time is needed to mate the sealing surfaces. It should also be pointed out that a magnetic seal is freely mounted on elastomers and is not subjected to any external forces, such as press fits or clamping. These external forces can distort and destroy the high degree of flatness mentioned previously.

The magnetic ring used in this seal is fabricated from cast Alnico V. Alnico V is a hard, crystalline, precipitation-hardened alloy which is made by conventional foundry techniques and specialized heat treatments. The heat treatment consists of heating the alloy to 1,300°C and holding that temperature until a homogenized structure is achieved. This is followed by a controlled cooling where a submicroscopic phase is precipitated. The alloy is then reheated to 600°C and held for a period of time. This stage is called aging and increases the coercive force and energy product by forming a second submicroscopic phase.

High magnetic energy is obtained by applying a magnetic field to the magnet ring during cooling. This causes the precipitate to align in the direction of the applied field, resulting in stronger magnetic properties in that direction. All Alnico V magnet rings are directionalized or magnetically oriented in this manner to obtain maximum uniform magnetic energy.

The success of any magnetic seal depends on the ability of the magnet to supply a constant amount of pull or flux through the air gap in any environment to which the seal is subjected. Once magnetized, the flux produced by the magnet will remain constant unless external energy is applied to change the balance of the internal energies.

Eight magnetic seal assemblies were manufactured by the Magnetic Seal Corporation to the specification defined in Figure 51. A new magnetic seal was used for each of the eight test setups. A passage was provided to the outside of the test housing to collect any lubricating oil leakage that might occur during test. Also, each seal was accurately weighed and dimensioned before and after test to determine wear rates.

Lubricant

The lubricant used throughout this test program was a qualified MIL-L-23699 specification. As the testing was being conducted, samples were monitored for changes in properties. Primary emphasis was given to the neutralization number (acid number). The rate of increase in the neutralization is an indication of the deterioration (oxidation) of the oil. Table 13 shows the properties of new oil.

The initial test rig oil fill was used for test numbers 1 through 6. The oil pump hour meter registered approximately 460 hours during this time. At the conclusion of test number 6, the entire lubrication system was thoroughly cleaned and refilled in preparation for test number 8, the endurance test. This second fill of oil was used for test number 7 and the endurance test for a total of 500 hours on the pump hour meter.

TABLE 13. LUBRICANT PROPERTIES

Lubricant	Viscosity (cSt) at 40°C	Viscosity (cSt) at 100°C	Viscosity Index	pH	Acid No.	Color
MIL-L-23699	25.38	4.99	124	8.1	0.08	3.5

The components required to make up each test assembly are shown in Figure 53. Each assembly requires two test bearings, one test shaft, one magnetic seal, and two slave bearings. Each of eight test assemblies used new components except for the slave bearings, which were replaced only as required.

TEST RIG DESIGN

To conduct these tests under simulated operating conditions in order to evaluate the performance of the ribbed-cup tapered-roller bearings and magnetic seal, a test rig was designed around a rig which was originally developed for the HLH ATC high-speed tapered-roller bearings. This work was originally conducted under USAAMRDL Contract DAAJ01-71-C-0840(P40) and documented in Reference 3.

Figure 54 is an overall view of the test rig and control panel. The test rig was driven by a 100-hp dc variable-speed electric motor belted to a 9:1 speed increaser. This system provided the capability of running the test bearings to a maximum speed of 14,000 rpm.

A new test head was designed which allowed the testing of two bearings on a simulated spiral bevel input pinion gear. A cross-sectional view of the test head is shown in Figure 55. The arrangement shown in the figure was used for most of the development tests and for the endurance test.

Bearing loading was applied hydraulically to the pins on the slave bearing cup housing as shown in Figure 56. The pins were located below the shaft centerline and the cylinder was inclined at an angle of 22 degrees to produce radial, thrust, and moment loading on the test shaft simulating loads produced by a bevel gear mesh. The 25,000-pound system included a spool-type load cell for monitoring applied loads. Figure 57 shows the test housing and hydraulic loading system.

For one of the development tests (number 6) and all of the oil-off survivability testing, a modified test head arrangement was used. This new arrangement involved the removal of the slave bearings and hydraulic loading device and installation of various Belleville springs for applying only thrust loads. This test arrangement is shown in Figure 58. This change was made in order to reduce the effects of the slave bearing heat generation on the final test results.

Lubricant for all tests was supplied to the test bearings entirely through the shaft center. The flow was metered by a stationary tube projecting into the shaft inside diameter. This tube also provided the slave bearing cone rib oil supply. The orifice diameters were sized analytically, then verified experimentally prior to assembly. The small-diameter ends of the slave bearings were lubricated by jets located between the bearing rows. The flow rates to both sources were monitored by turbine-type flow meters. The lube system was equipped with a heat exchanger and sump heater to control oil inlet temperatures.

Test parameters recorded included test bearing cup od, seal case od, oil inlet, oil outlet, and housing temperatures; oil flow rates; shaft speed; and hydraulic cylinder load.

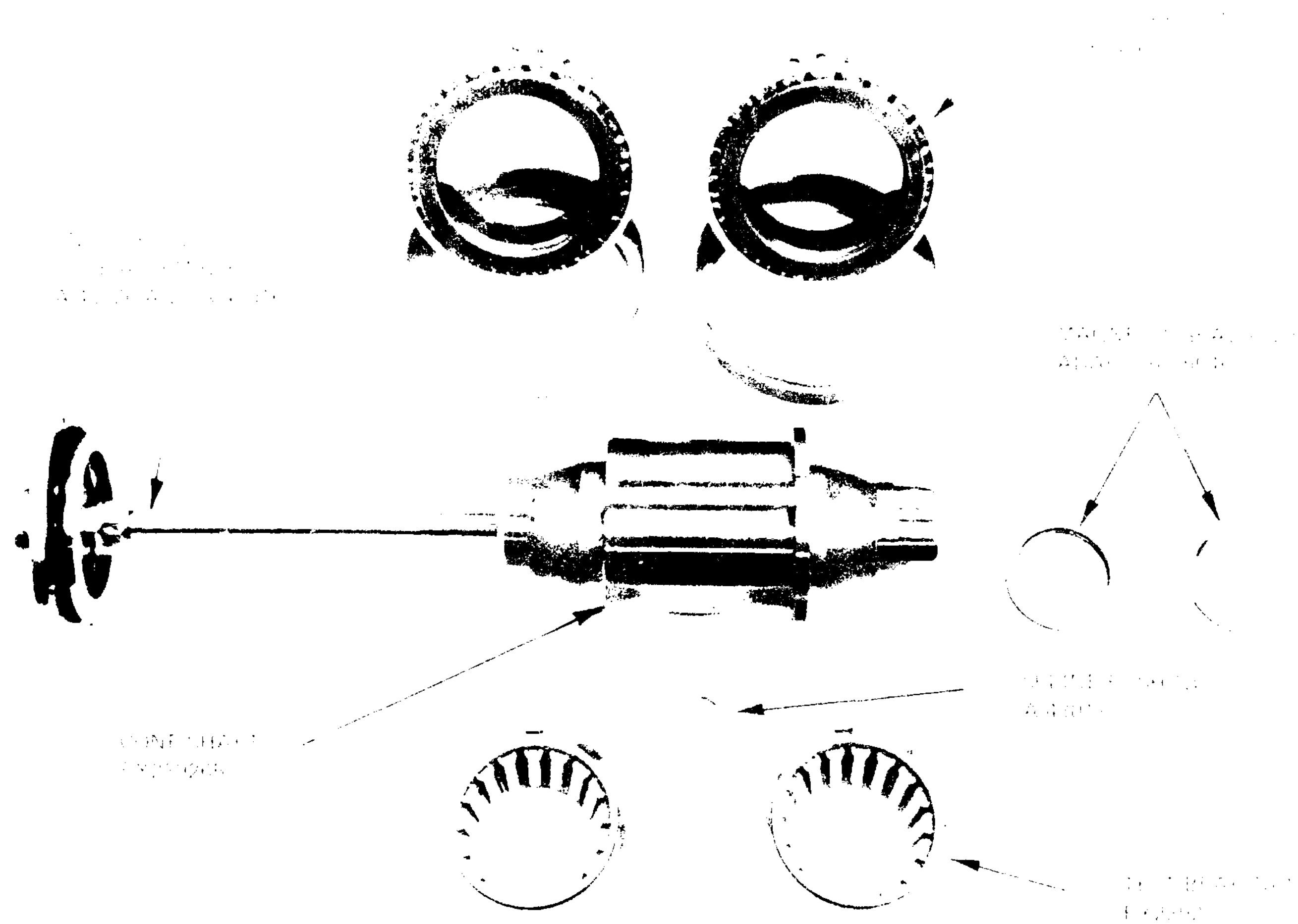


Figure 53. Test Components Prior to Final Assembly.

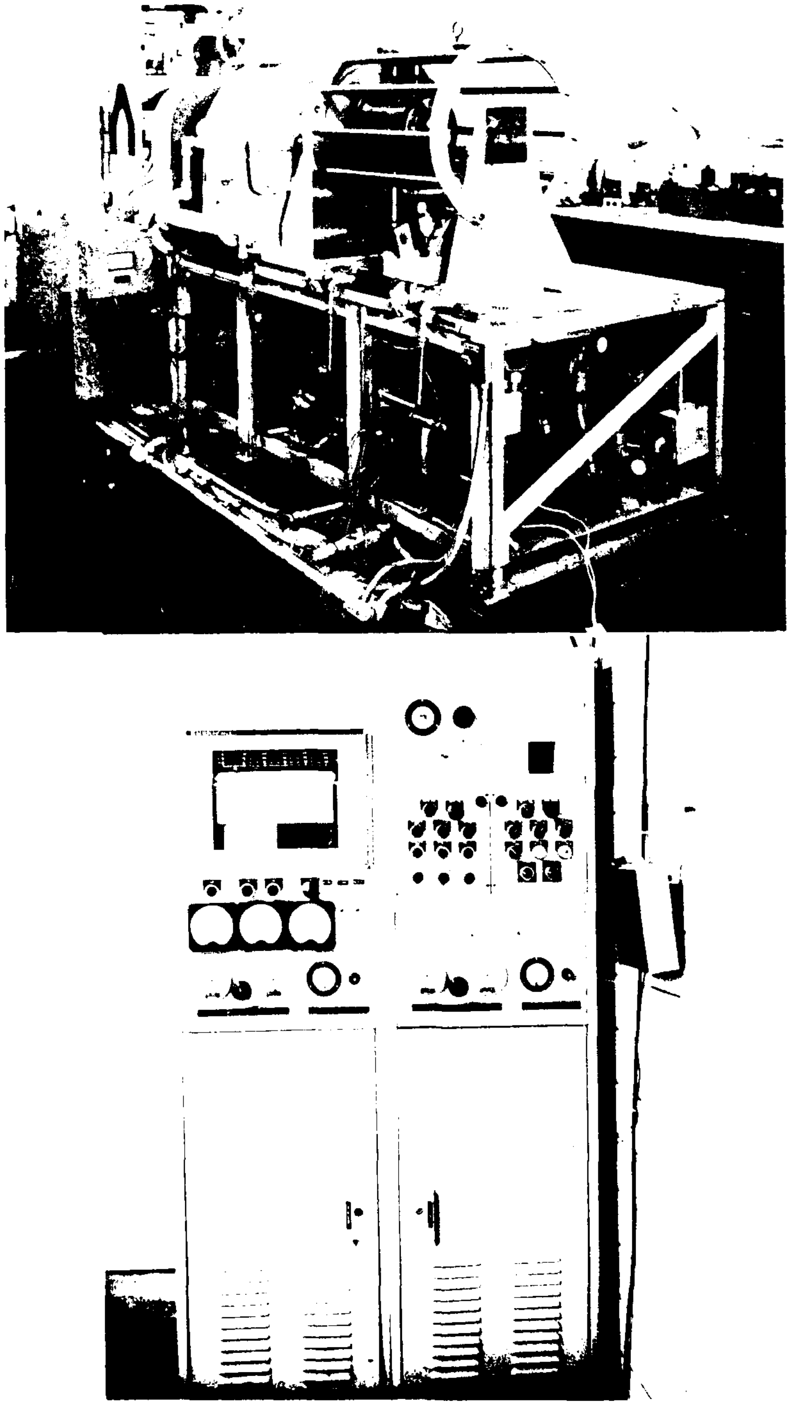


Figure 54. Overall View of Boeing Vertol Test Rig and Control Panel.

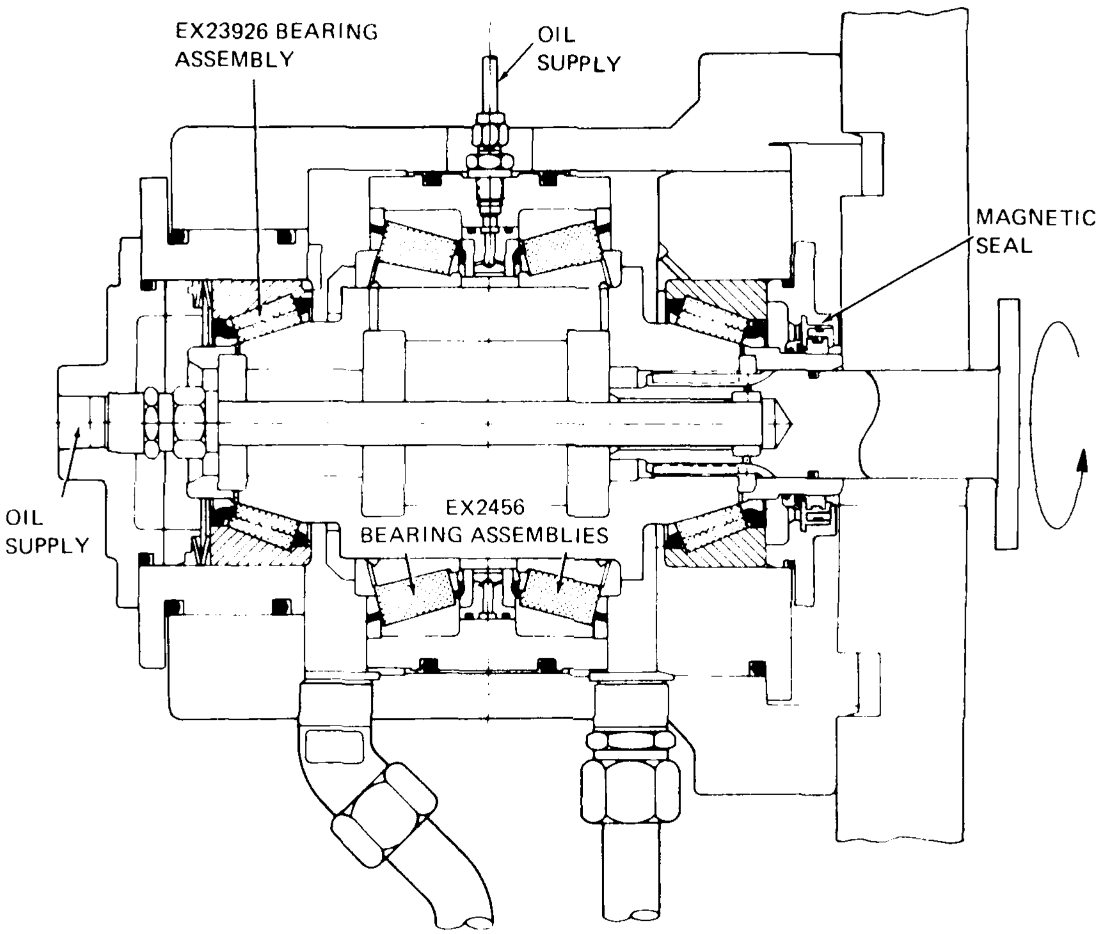


Figure 55. Test Arrangement for Applying Simulated Bevel Gear Loading.

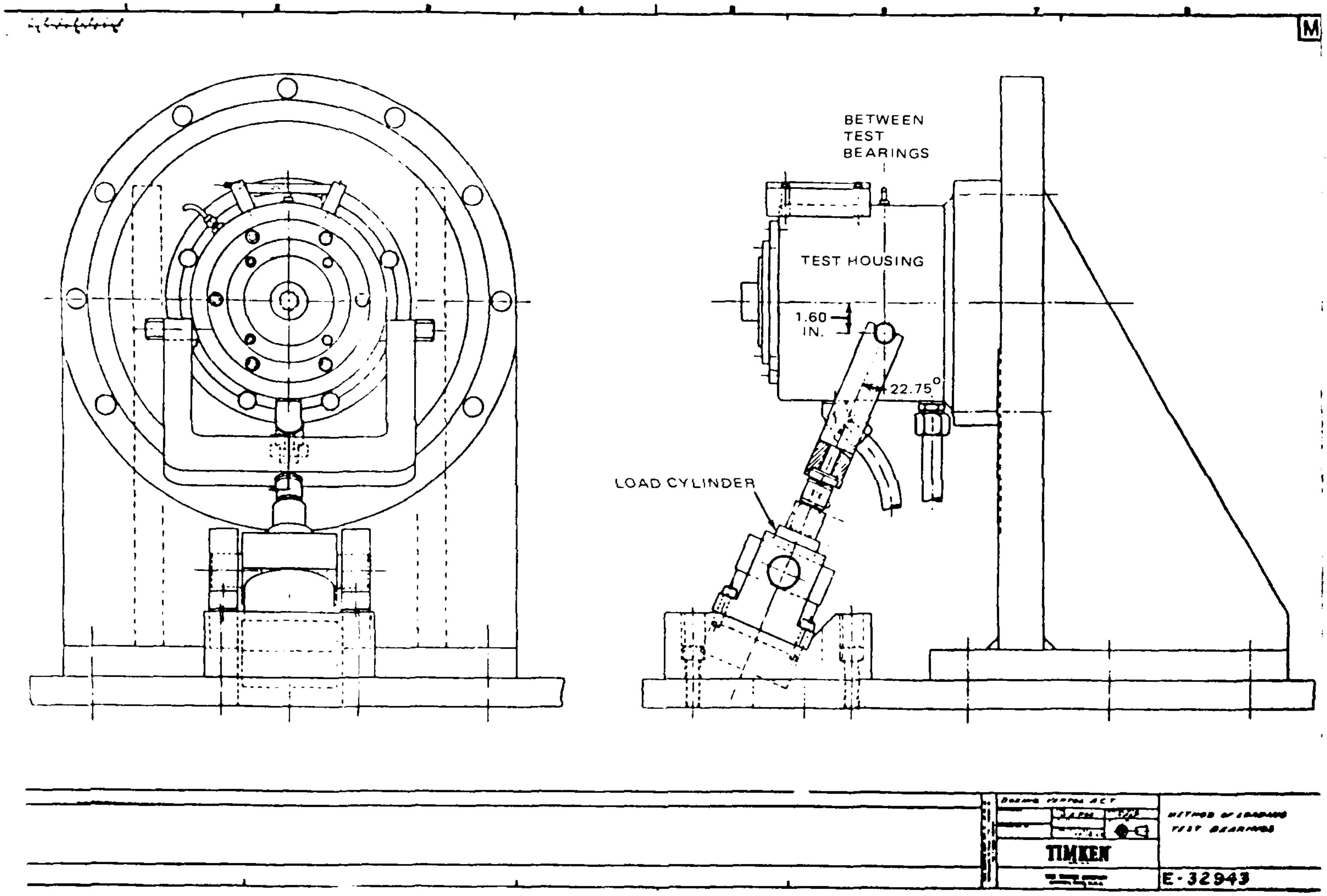


Figure 56. Test Loading.

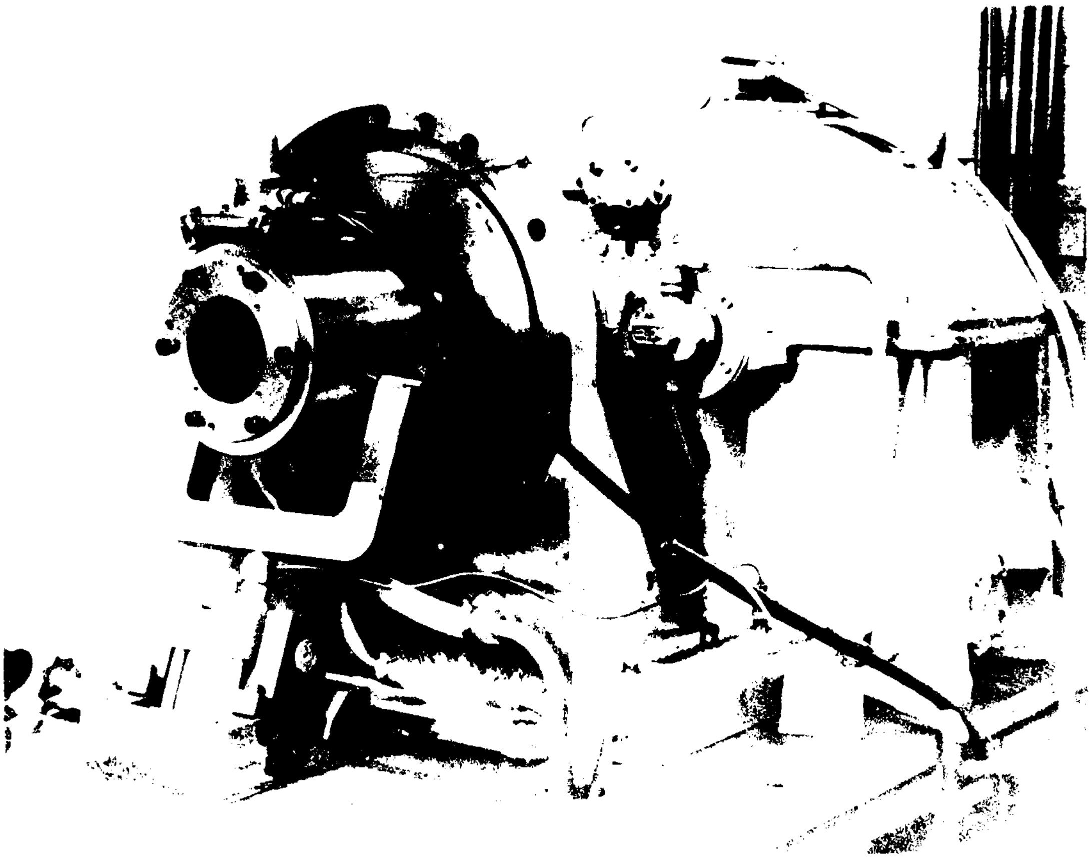


Figure 57. Tapered-Roller-Bearing Test Rig.

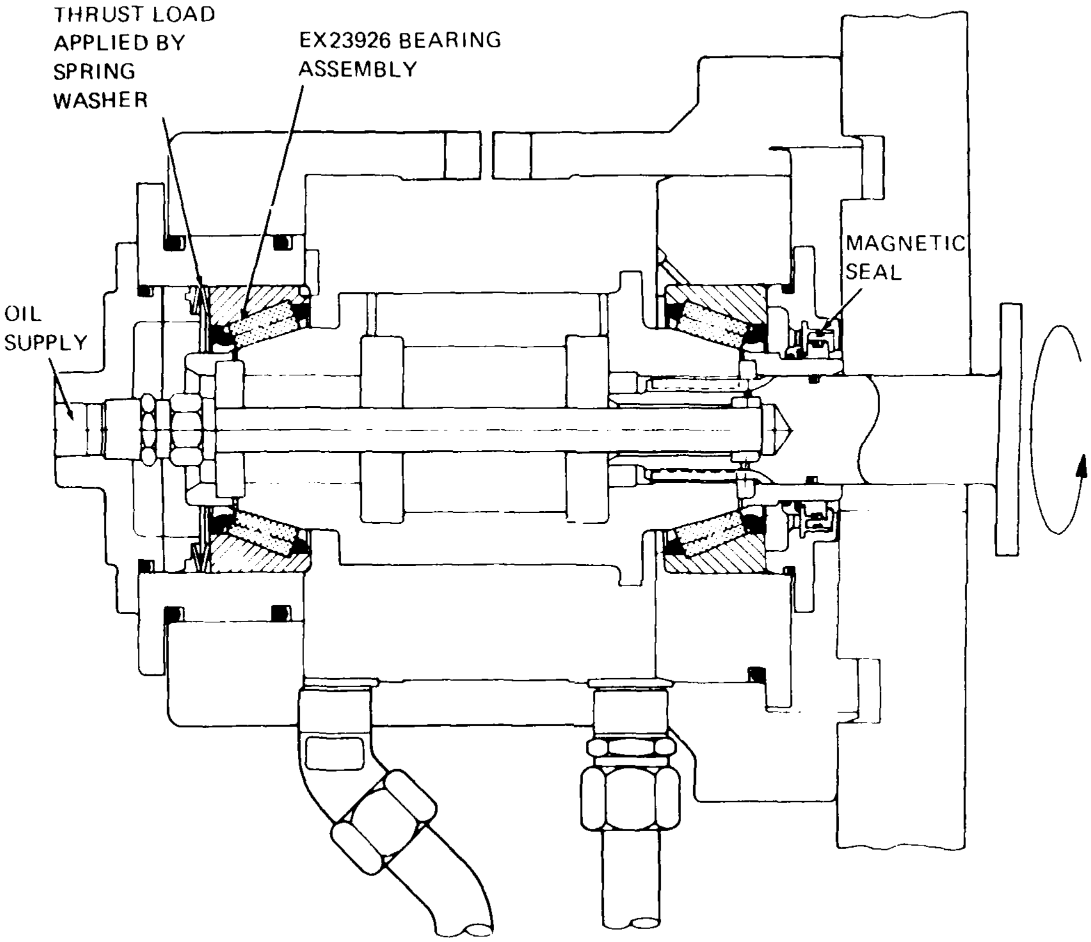


Figure 58. Test Arrangement for Thrust Loading Only.

TEST PROCEDURE

The following test procedure was used during most of this program. As testing progressed, several changes were made to the test procedure in order to obtain desired information or to evaluate other parameters. These changes are discussed in more detail in the next section concerning tests results. A test matrix indicating the various parameters to be investigated during the initial eight tests is shown in Table 14. The first seven tests were conducted to evaluate and optimize the operating parameters to be used in test number 8, which was a limited endurance test. The parameters investigated were oil flow rates, oil inlet temperature, speed, load, preload requirements and floating cup fits, and flow rate requirements.

A test assembly consisted of two new test bearings, two slave bearings, one new inner-race test shaft, and a new magnetic seal. The slave bearings were replaced only as needed. For test number 6 and all oil-off tests, slave bearings were not installed.

Prior to assembling and testing, each bearing component was closely inspected to insure uniform quality. The objective was to minimize bearing component influences on operating characteristics.

Before installation and test, both the magnetic ring and the seal case/carbon insert were ultrasonically cleaned and weighed. For the first two test setups the seal parts were weighed and measured each time the setup was disassembled for inspection. Since no appreciable weight or width loss was measured, the seal parts were weighed only at the beginning and end of tests 3 through 8.

The test bearings were designed for an L-10 life criterion of 900 hours catalog rating at 7,196 rpm at cubic mean loading with no material factors included. The cubic mean load for this application represented approximately 73.5 percent of the twin-engine (TE) rating of 1,401 hp per input pinion of an advanced-concept transmission. Because the input pinion arrangement of an advanced-concept transmission could not be exactly duplicated in this test program, it was decided that the parameters for the most critical bearing (heel location) should be maintained and the loads on the toe bearing would vary as required to maintain the proper loading on the heel bearing. Given the bearing spacing in the test rig application and given that both the heel and toe position would use identical size bearings, a load schedule was developed to achieve a radial load of 3,587 pounds on the heel bearing. This load was rounded off to 3,600 pounds and identified as the cubic mean load. By simplifying the radial loads, the following test scheme was used: 75 percent or 3,600 pounds (cubic mean ≈ 1,050 hp), 100 percent or 4,800 pounds (twin engine ≈ 1,400 hp), and 112.5 percent or 5,400 pounds (single engine ≈ 1,575 hp). A summary of loading conditions for the heel and toe bearings is shown in Table 15. The slave bearings had similar loads.

At the start of each test, the speed and load cycle shown in Table 16 was used; therefore all bearing assemblies were subjected to the same initial load cycle. Upon completion of this cycle, loads and speeds were varied as required to complete the planned test matrix shown in Table 14.

The first two tests were started at 1,000 rpm and 2,000 pounds cylinder load as a shakedown procedure. The tests that were run at higher speeds were increased in 2,200-rpm increments (9,600; 11,800; 14,000 rpm) with the same four load levels as shown in Table 14.

Test number 6 was conducted under thrust loading without the slave bearings. The two load levels used were equivalent axial thrust loads; that is, 2,964 pounds thrust would produce equivalent heat generation as 50 percent of twin engine loading and 6,418 pounds thrust would simulate 100 percent single-engine loading

TABLE 14. TEST MATRIX

Test Variable Test No.	Speed (rpm)	Load (lb)	Lube Flow (pt min)	Preload Spring	Floating Cup Fits	Comments	Min Run Time or Failure (hr)
1	3,700 → 7,400	Cubic mean Twin engine Single engine	4	110% or 150% of induced thrust	0.002 in. L to 0.005 in. L with oil flow to od	All combinations at each test will not be run for all tests. Various tests will be used to evaluate critical operating parameters prior to proceeding to the next test variable. All tests to be conducted with oil index temperature at 190°F or as noted.	24
2	3,700 → 11,800		8 and 4				24
3	3,700 → 7,400		2				24
4	3,700 → 7,400		1 and 0.5		0.0002 in. L to 0.0008 in. L No oil flow to od		24
5	3,700 → 14,000		8				24
6	3,700 → 14,000	Thrust load only	8 4 and 1	2,964 lb and 6,418 lb		Remove slave bearings;run with thrust load only	24
7	3,700 → 14,000	Cubic mean Twin engine Single engine	4	110% (2,115 lb)		Oil inlet temperature increased to 300°F	24
8	7,400	Maximum Single engine only	Optimum	Optimum	Optimum		150

TABLE 15. LOADING CONDITIONS

Engine Operation	Percent	Cylinder (lb)	Shaft Moment (in.-lb)	A Heel Radial (lb)	A Heel Thrust (lb)	B Toe Radial (lb)	B Toe Preload* (lb)
Cubic mean load	75	7,807	4,830	3,600	3,019	3,600	2,115
Twin engine	100	10,410	6,442	4,800	4,026	4,800	2,115
Single engine	112.5	11,710	7,246	5,400	4,529	5,400	2,115

*Preload equals 100 percent of induced thrust due to radial load at single engine.

TABLE 16. SPEEDS AND LOADS

Shaft Speed (rpm)	Cylinder Load (lb)	Approximate Duration (hr)
3,700	5,205	1.5
3,700	7,810	1.5
3,700	10,410	1.5
3,700	11,710	1.5
5,550	5,205	1.5
5,550	7,810	1.5
5,550	10,410	1.5
5,550	11,710	1.5
7,400	5,205	1.5
7,400	7,810	1.5
7,400	10,410	1.5
7,400	11,710	1.5

Following each test all bearing components were traced and photographed.

TEST RESULTS

A summary of test parameters for the development tests (number 1 through 7) and endurance test (number 8) is shown in Table 17. Appendix B includes a compilation of test data consisting of buildup sheets for each test assembly, a list of data points at each speed and load level at thermal stability, and photographs showing the after-test condition of each bearing component.

A summary of the results and test parameters of the six oil-off tests conducted with components from the development tests mentioned above is shown in Table 18. Data sets consisting of buildup sheets, computer printouts of measured and calculated test results, graphic presentations of results, and posttest photographs of bearing components used in this test program are included in Appendix C.

Following is a brief description of each of the tests conducted.

TABLE 17. SUMMARY OF DEVELOPMENT AND ENDURANCE TESTS

Test No.	Development							Endurance
	1	2	3	4	5	6	7	8
Max Speed (rpm)	7,400	11,800	7,400	7,400	14,000	14,000	14,000	7,400
Loading (lb)								
Cylinder	2,000 11,700	2,000 11,700	5,205 11,710	5,205 11,710	5,205 11,710		5,205 11,710	11,710
Spring	2,115	2,115	2,140	2,115	2,130	2,964 6,418	2,115	2,115
Oil Flow Setup No.*	1**	2, 21	3, 31	4, 41	5	6, 61, 62	7	8
Per Test Bearing (pt/min)	4	4, 8	2	1, 0.5	8	8, 4, 1	4	2
Per Slave Bearing Small End	4	4, 6	4	4, 4	4		8	4
Large End	4	4, 8	8	8, 8	8		8	4
Floating Cup Fit (in.)	0.0035 loose	0.0031 loose	3 0.0035 loose 31 0.0007 loose	0.0008 loose	0.0006 loose	0.0002 loose	0.0005 loose	0.0005 loose
Cup OD Flow (pt/min)	3.1 6.8	0 6.4	Trace to 7.1	0	0	0	0	0
Slave Bearing Setting (in.)	0.0017 end play	0.0023 end play	0	0	0 0.0004 preload		0.0005 preload	0.0005 preload
Total Hours	37.6	58	3 39.5 31 82.25	54.75	61	101.25	67.5	379

Total Test Time 880.85 hours

*Oil inlet temperature was 190 ± 5°F for all tests except no. 7. Oil inlet temperature increased to 300 ± 5°F for test no. 7.
**Identified in computer records included in Appendix B

TABLE 18. SUMMARY OF OIL-OFF SURVIVABILITY TESTS

Oil-Off Test No.	Components Previous Test No. (ref Table 17)	Modifications	Shaft Speed (rpm)	Thrust Loading at Oil-Off/Final*	Survival Time (min)
1	6	None	3,700	3,530/6,621	1.40
2	2	New preload springs	3,700	3,205/3,301	7.80
3	1	New preload springs opposite drive cup; fit 0.007 in. loose	3,700	3,209/3,256	2.48
4	3	New preload springs; both cups > 0.007 in. loose; cage pilot clearance > 0.0093 in.	3,700	3,192/3,311	8.88
5	4	New preload springs; both cups > 0.007 in. loose; cage pilot clearance 0.015/0.010 at large end/small end; cup lands phosphate-coated	7,400	3,208/3,365	4.37
6	5	Same as 5 above	3,700	3,195/3,332	3.67

*Final thrust load calculated based upon thermal expansion of test components at end of test

Development Test No. 1

Test 1 was initially started at 1,000 rpm and 2,000 pounds cylinder load to check the rig and operation of al components. After several hours of operation at these conditions, the test head was disassembled and all components were inspected. All components were satisfactory and they were reassembled and the test was continued.

The Belleville spring for this first test was set to provide 2,115 pounds preload on the opposite-drive-end floating cup. The oil flow rate to each test bearing was four pints per minute. The oil flow rate to each slave bearing was eight pints per minute, equally divided to both ends of the bearing. The opposite-drive-end floating cup od fit was 0.0035 inch loose and the oil flow rate to the floating cup od ranged from 3.1 to 6.8 pints per minute, increasing as the oil inlet temperature increased. The slave bearings were assembled with a 0.0017 inch end play setting. Data points were recorded at four shaft speeds from 1,000 to 7,400 rpm and five loads from 2,000 to 11,700 pounds.

After 14 hours of testing, inspection of test components revealed that one of the rollers from the drive-end test bearing developed a circumferential groove on the roller body toward the large end as shown in Figure 59. The groove was approximately 0.004-inch deep by 0.012-inch wide. Further inspection found a chip missing from the carbon insert of the magnetic seal as shown in Figure 60. The hardness of the carbon insert material was reported to be 58 to 65 Rc. It was thought that the chip became imbedded in the cage bridge and produced the groove. All components completed a total of 37.6 hours of testing with no additional damage or distress. All test data and photographs of components are contained in Appendix B.

Development Test No. 2

Test 2 was run at speeds of 1,000 rpm to 11,800 rpm and machine cylinder loads from 2,000 to 11,700 pounds. The opposite-drive-end floating cup od clearance was 0.0031 inch loose. The oil flow rate to the floating cup was from zero to 6.4 pints per minute. The slave bearings were assembled with 0.0023 inch end play. The load-up cycle was repeated for two oil flow rates.

There were three teardowns for inspection during the 58-hour test. The final run was to achieve maximum speed (14,000 rpm) operation in order to check the rig. The test was finally terminated during a run at 11,800 rpm due to slight scuffing damage on one slave bearing. Inspection of the slave bearings indicated that loss of shaft/bore interference fit of the bearing due to inertial loading and temperature allowed the slave bearing cone to move axially and break the capscrews that secure the cone clamping ring as shown in Figure 61. To correct this condition for the remainder of the tests, the bores of the slave bearings were chrome-plated to give a tighter interference fit. The opposite-drive-end slave bearing which was damaged was replaced. All other test components completed this series of tests without damage or distress.

Development Test Number 3

Test 3 was also started at 1,000 rpm and 2,000 pounds machine cylinder load. Within two hours, test conditions were increased to 11,800 rpm and 7,810 pounds cylinder load. After 15 minutes of operation at this condition, a noise indicated bearing damage and the test was terminated. Inspection revealed that the opposite-drive-end slave bearing again sustained skidding damage. The load zone on the damaged bearing cup was not continuous as shown in Figure 62. It was apparent that the two-point loading on the slave bearing cup adapter was deforming the outer race. The remainder of test 3 focused on solving the problems encountered with the slave bearings.

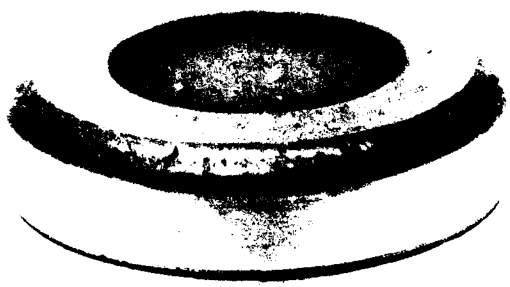
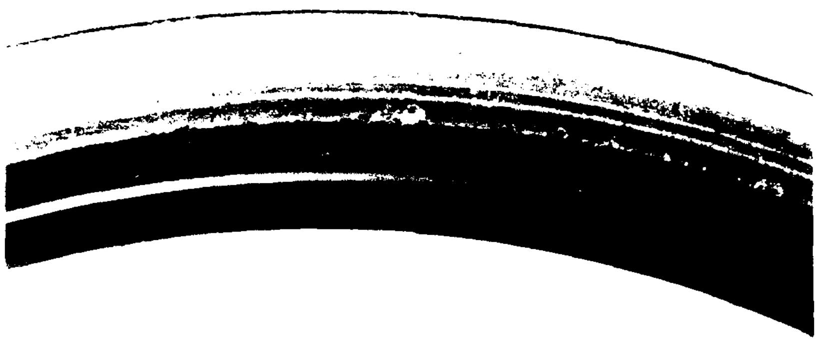


Figure 59. Roller From Test Setup No. 1 Showing Circumferential Groove.

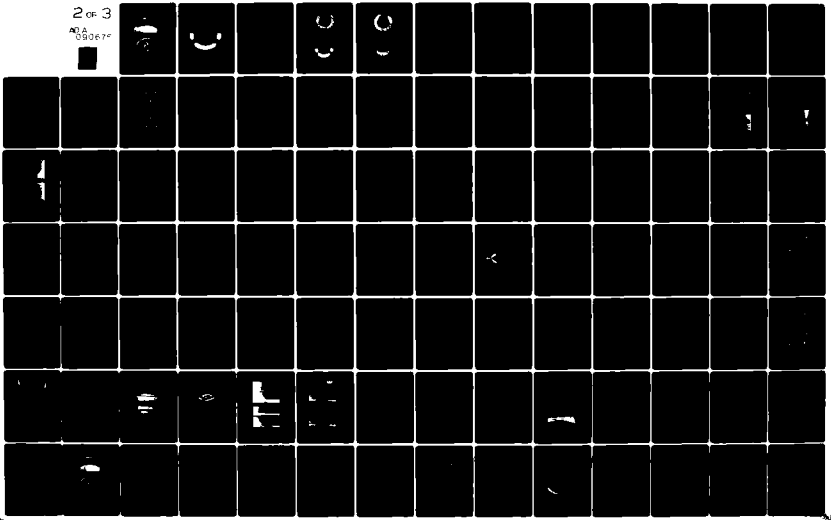


5X SIZE

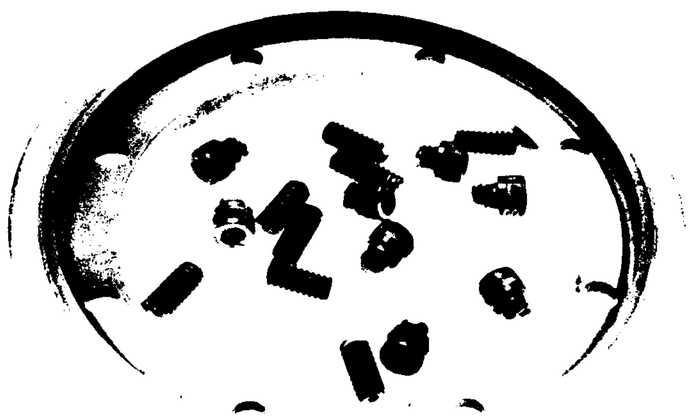
Figure 60. Magnetic Seal Ring From Test Setup No. 1 Showing Chipped Area.

AD-A090 675 BOEING VERTOL CO PHILADELPHIA PA F/G 1/3
ADVANCED TRANSMISSION COMPONENTS INVESTIGATION PROGRAM. BEARING--ETC(U)
AUG 80 J W LENSKI DAAJ02-76-C-0045
UNCLASSIFIED D210-11591-1 USAAVRADCOM-TR-80-D-19 NL

2 OF 3
AD A 090675



OPPOSITE-DRIVE-END SLAVE BEARING SEAT ON SHAFT



END CAP AND BROKEN BOLTS FROM SLAVE BEARING

HIGHEST SPEED = 11,800 RPM
HIGHEST LOAD = 7,800 LB

Figure 61. Results of Test Setup No. 2 With All Eight Bolts Breaking and Bearing Backing Off 0.125 Inch.

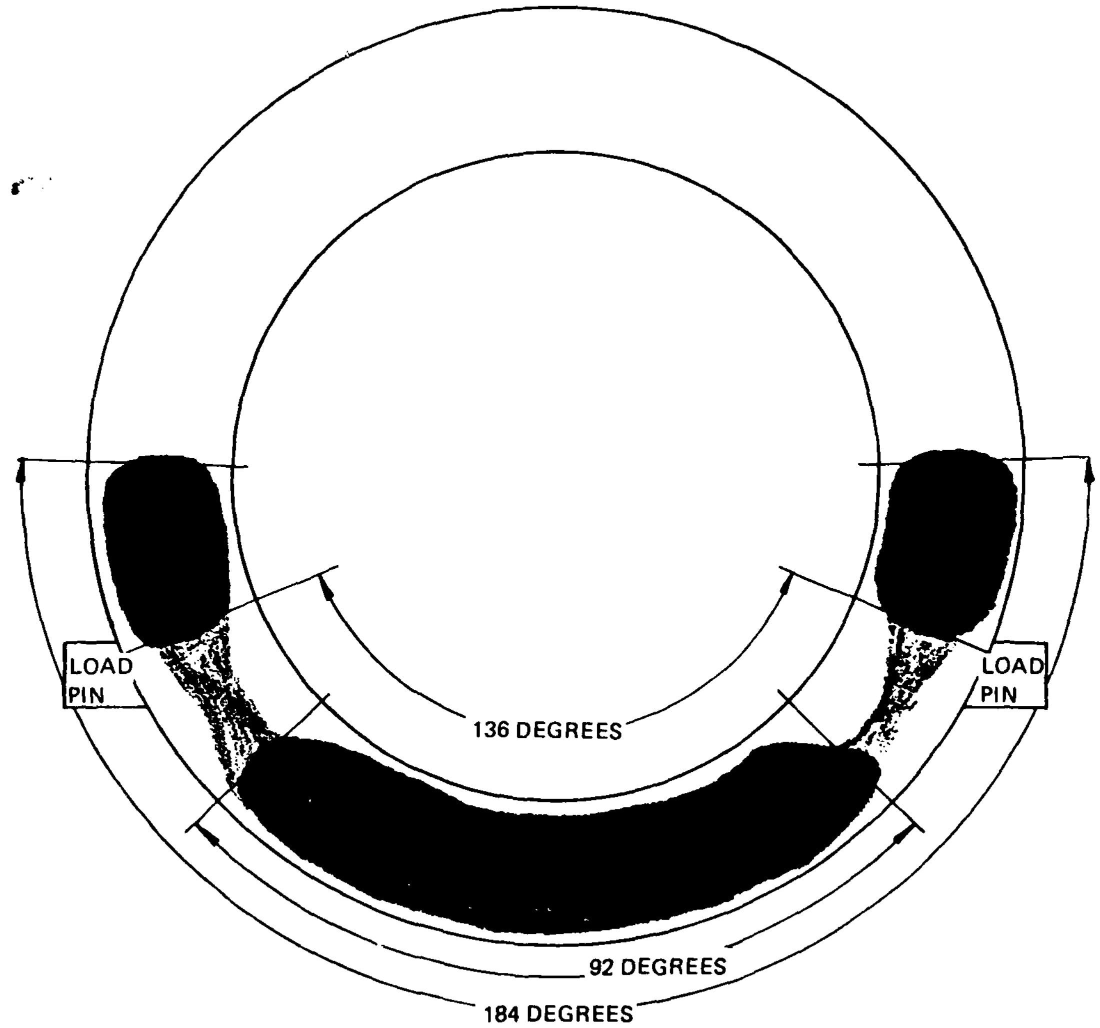


Figure 62. Load Zone of Opposite-Drive-End Slave Bearing Cup After Start of Test Setup No. 3.

Several tests were conducted to determine the effect of housing deformation on slave bearing load zones. The cups of the two slave bearings were blued and installed into the test rig. These bearings were operated for three hours under load. After running, the cups were inspected to determine their load zones as shown in Figure 63. These inspections revealed that local distortions at the load pin locations and partial loading have contributed to the slave bearing problems.

The proposed solutions consisted of fabricating a new center cup housing with smaller O-ring grooves and additional material and changing the bearing settings to zero to 0.0005-inch preload. These fixes were confirmed by a series of analytical, bench, and rig tests (Figure 64). The analytical study showed that at operating temperatures, bearing setting was the same as initial bench setting. This was experimentally confirmed by heating the shaft subassembly in an oven. The total test time accumulated under all conditions was 82.25 hours.

Inspection of all test components after test revealed no damage or distress. All objectives of the first three tests were achieved despite several problems experienced with the slave bearings. The modifications incorporated at the end of test number three resulted in an end to all slave bearing-originated problems.

Development Test Number 4

This test was conducted to evaluate the bearing operating characteristics under reduced oil flows. The test was run through two speed and load cycles. Oil flow rates were reduced to 1.0 and 0.5 pint per minute. Additionally, the slave bearing cups were nital-etched to confirm the fixes adopted in the previous test. Posttest inspection showed significant improvement in the contact patterns. Total elapsed time of this series of tests was 54.75 hours.

This test was successfully completed without any problems with the slave bearings. This test verified that the corrections made during test 3 eliminated scuffing damage to the slave bearings. All components inspected after test revealed no damage or distress.

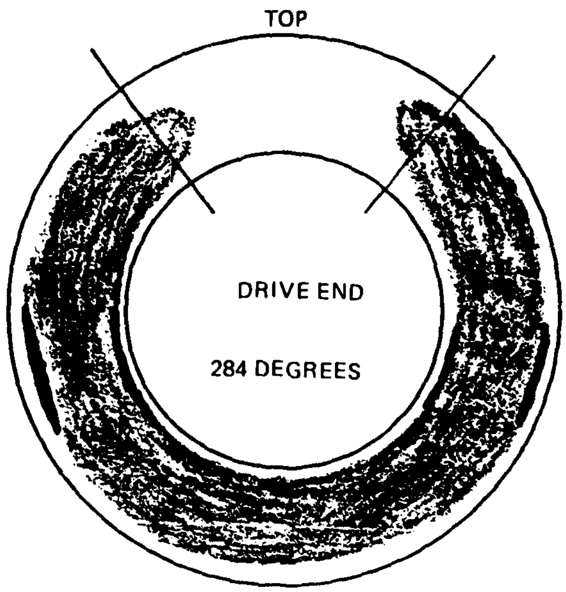
Development Test Number 5

Test 5 was the first successful run through the maximum speed and load. The first phase was run to single-engine load at 7,400 rpm. The test components and slave bearings were inspected and found to be in excellent condition. Slave bearing setting had changed from zero to 0.0006-inch end play after this test. The cone spacer was reground to yield 0.0004-inch preload. Subsequent inspections were made after reaching maximum load and speeds of 9,600, 11,800, and 14,000 rpm; all components were in excellent condition. Total test time was 61.5 hours.

After completion of each test, the test shaft races, cup races, and cup rib face were traced and the results compared to traces recorded prior to testing. Figures 44, 45, and 50 showed the traces of the shaft and test bearings used in test 5 before test. Figures 65 and 66 show the traces of the cup race and rib face of both test bearings after test and Figure 67 shows the traces of both races of the test shaft after test. Review of the before-and-after traces shows very little change on the operating surfaces of the test components.

Development Test Number 6

All testing prior to this test was conducted with both the test bearings and slave bearings. Due to the mixing of the outlet oil, it was impossible to determine the exact heat generation of the test bearings. A review of



NOTES:

1. TEST RUN AT 1,000 RPM
2. 7,810-LB LOAD FOR 3 HOURS
3. CUPS WERE PLUG-BLUED WITH AND WITHOUT LOAD PINS WITH NO SIGNIFICANT DIFFERENCE

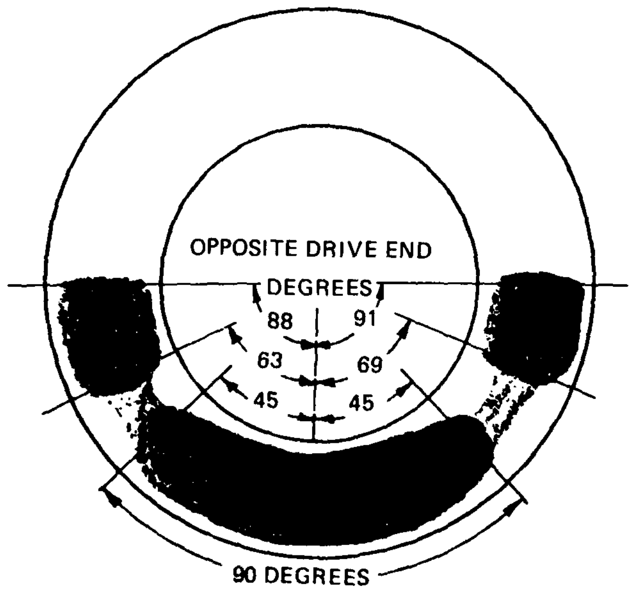
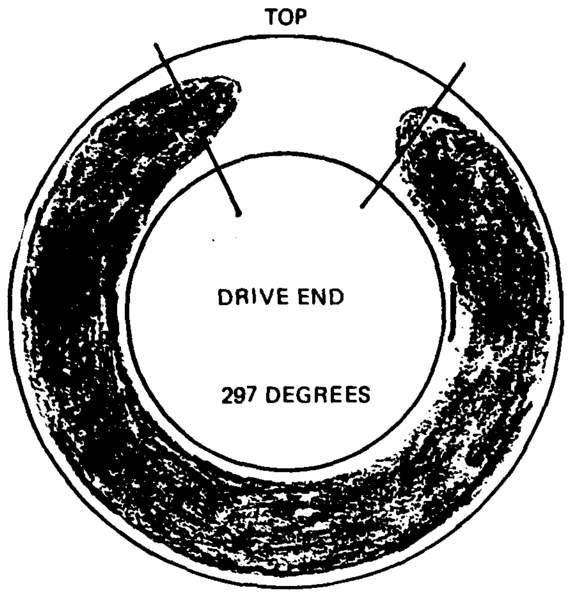


Figure 63. Load Zone Test on Slave Bearings With Initial Housing Design.



NOTES:
1. HEAVIER CUP ADAPTER
2. TEST RUN AT 1,000 RPM
3. 7,810-LB LOAD FOR 4 HOURS

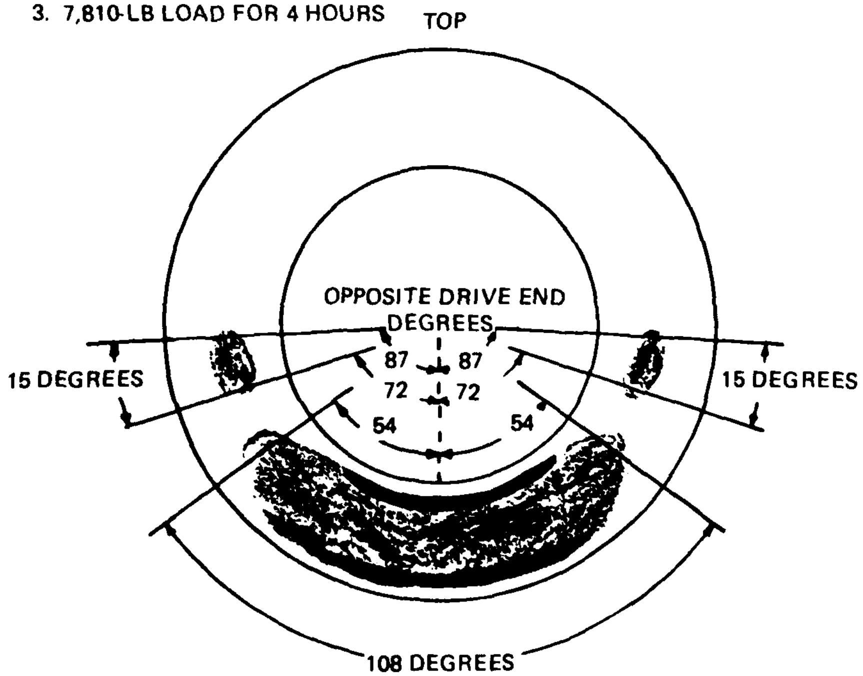


Figure 64. Load Zone Test on Slave Bearings After Housing Modification.

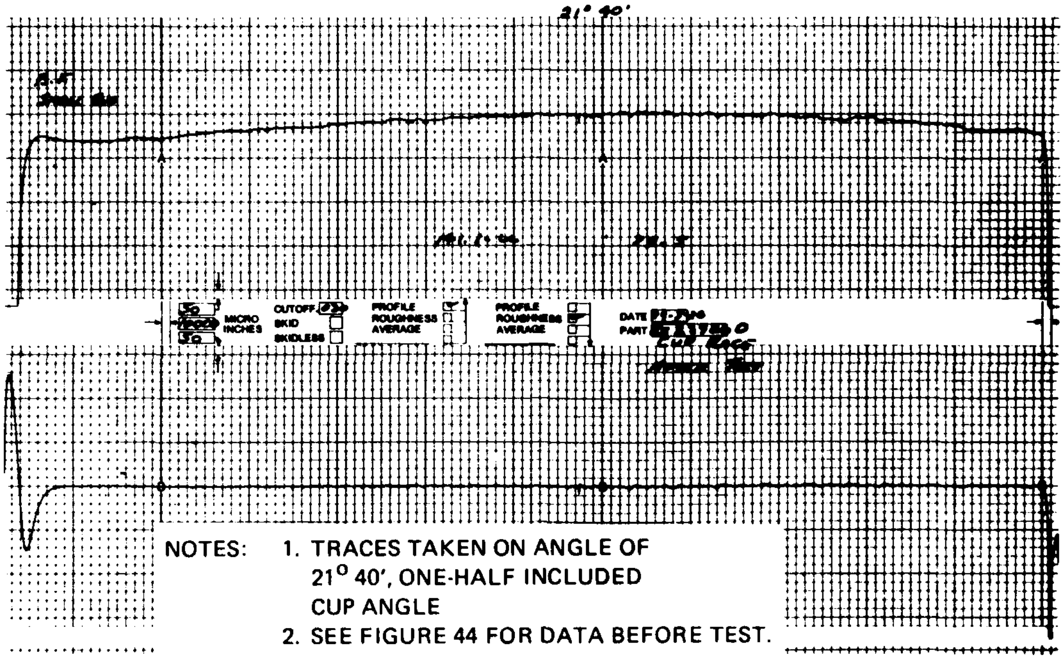


Figure 65. Profile Trace of Drive-End Cup Race After Test Setup No. 5.

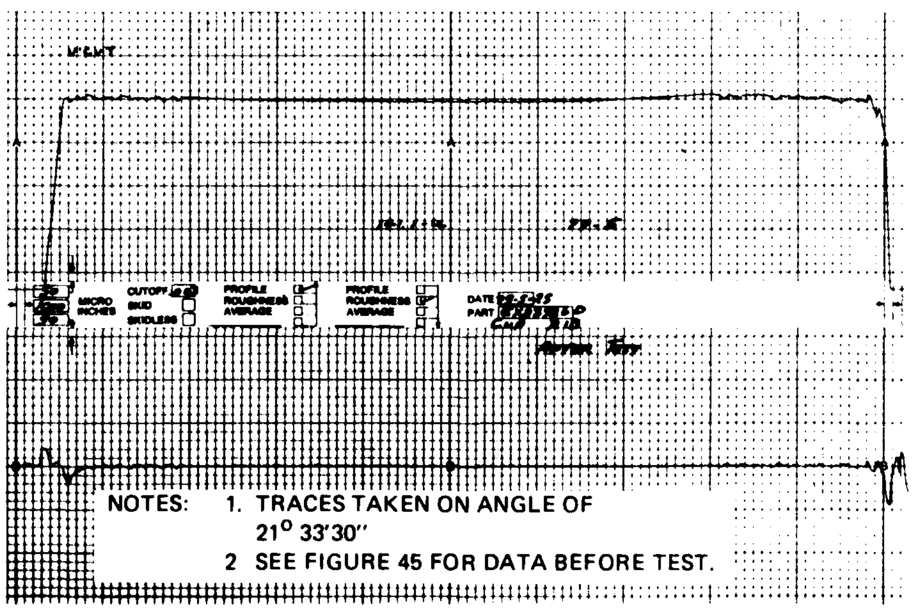


Figure 66. Profile Trace of Cup Rib Face After Test Setup No. 5.

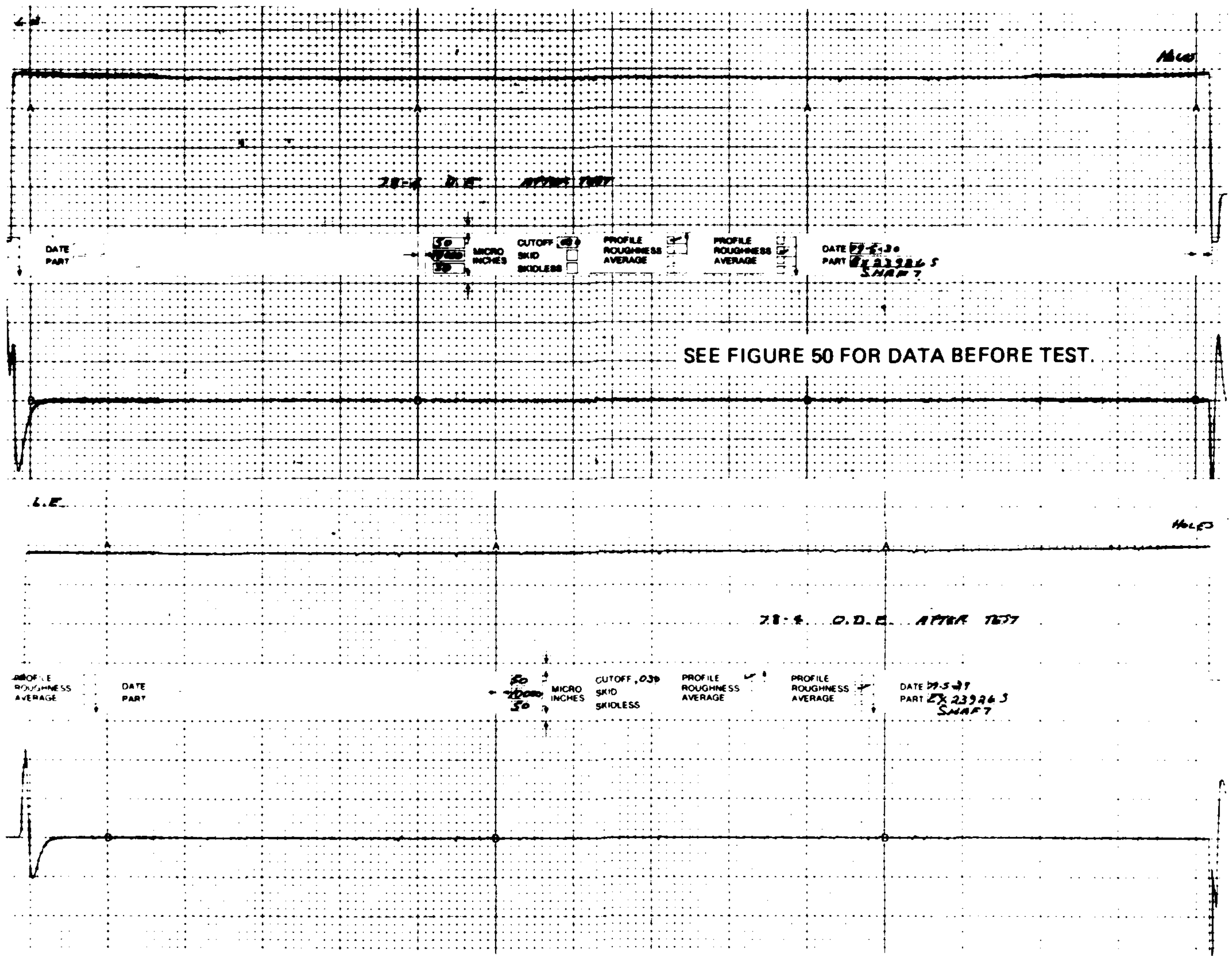


Figure 67. Race Traces From Both Ends of Test Shaft After Test Setup No. 5.

the test arrangement indicated that the slave bearings could be removed from the system and the test bearings operated under thrust load only. In order to achieve similar heat-generation characteristics, the combined load conditions were converted to an equivalent thrust load as shown in Table 19. A thrust of 6,418 pounds would be equivalent to single-engine loads. Therefore a new set of Belleville springs was fabricated to achieve equivalent single-engine and 50-percent twin-engine loads. After fabrication, a calibration check was made on the new set of springs to verify the applied loads. The load/deflection data for these springs is shown in Table 20.

TABLE 19. EQUIVALENT THRUST LOAD ON DRIVE-END TEST BEARING

Load Condition	Normal Loads: Thrust (lb)	Normal Loads: Radial (lb)	Normal Loads: Preload Spring (lb)	Equivalent Thrust Load,* F_{EQ} (lb)
Cubic Mean (CM)	3,019	3,600	2,115	4,993
Twin-Engine Rating (TE)	4,026	4,800	2,115	5,938
Single-Engine Rating (SE)	4,529	5,400	2,115	6,418

*Equivalent thrust load obtained to produce the same heat generation as the combined thrust and radial load; method used defined in Reference 5

5. TIMKEN ENGINEERING JOURNAL, Section 1, Timken Company, Canton, Ohio, 1973.

Test 6 was assembled to run under thrust load only, i.e., the slave bearings were removed for this test. The operating characteristics were monitored under three oil flow rates: 8, 4, and 1 pint per minute per bearing. The test was run at six speeds from 3,700 to 14,000 rpm under thrust loads of 2,964 and 6,418 pounds. At an oil flow rate of 1 pint per minute per bearing, the oil outlet temperature was 367°F. At this time, portions of the lubrication system were piped with copper tubing. Due to the possibility that the lubricant would react with it, no additional testing was attempted beyond this data point at 11,800 rpm.

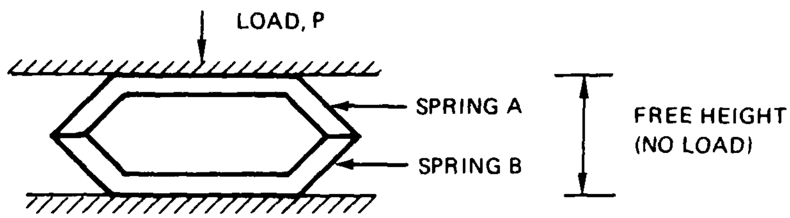
The results of the test at 6,418 pounds of thrust load are summarized in Figure 68. This curve provides information concerning the amount of heat removed from the bearing at various speed and oil flow rates. All components successfully completed this test program which accumulated a total of 101.25 hours.

The magnetic seals used in this and previous tests have not experienced any significant leakage, wear, or weight loss. For this test at an oil flow rate of 1 pint per minute per bearing, the maximum temperature of the seal reached 323°F at 11,800 rpm. Even at this condition, no weight loss or wear was noted upon completion of the test.

Development Test Number 7

This test was conducted to evaluate the effects of a 300°F oil inlet temperature. This test was operated for one speed and load cycle through 14,000 rpm. At speeds below 5,550 rpm and test loads below 10,410 pounds, the 300°F inlet oil temperature could not be generated. A total of 67.5 hours was accumulated on

TABLE 20. BELLEVILLE SPRING CALIBRATION



SPRING SET 1 FREE HEIGHT 0.5171 INCH
SPRING SET 2 FREE HEIGHT 0.504 INCH

Load, P (lb)	Spring	Avg Deflection (in.)	
		Set 1	Set 2
500	A	0.014	0.008
	B	0.014	0.008
1,000	A	0.019	0.013
	B	0.019	0.013
1,500	A	0.024	0.018
	B	0.024	0.018
2,000	A	0.029	0.024
	B	0.029	0.024
2,500	A	0.034	0.030
	B	0.034	0.030
3,000	A	0.040	0.036
	B	0.040	0.036
3,500	A	0.047	0.043
	B	0.047	0.043
4,000	A	0.053	0.050
	B	0.053	0.050
4,500	A	0.060	0.057
	B	0.060	0.057
5,000	A	0.067	0.065
	B	0.067	0.065
5,500	A	0.075	0.073
	B	0.075	0.073
6,000	A	0.083	0.081
	B	0.083	0.081
6,500	A	0.090	0.089
	B	0.090	0.089

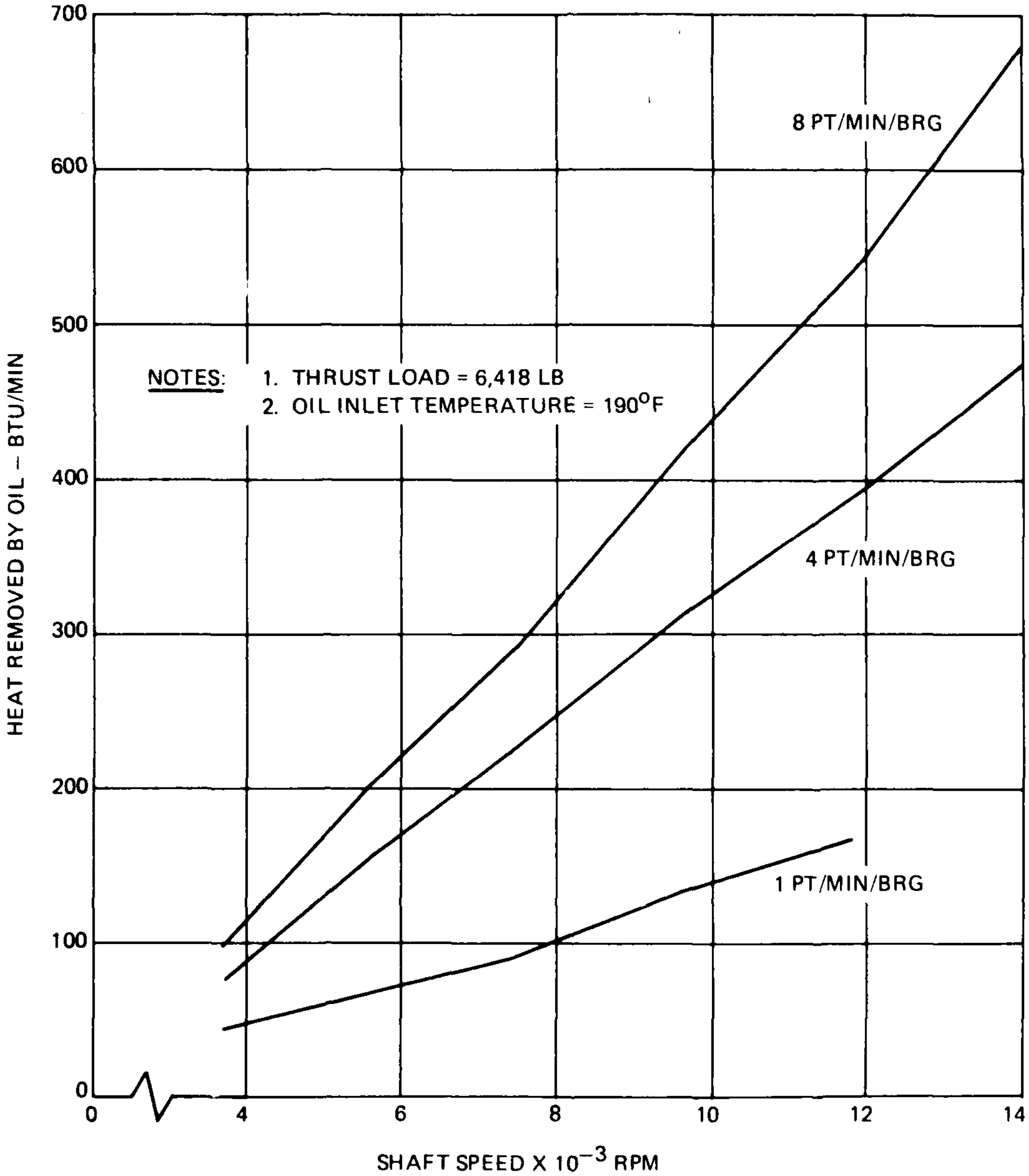


Figure 68. Results of Test Setup No. 6 With Thrust Load Only.

this test. All posttest components were in excellent condition, with the only notable difference being the lubricant staining on the test components.

Endurance Test Number 8

Upon completion of the scheduled seven development tests, an endurance test was planned in order to evaluate bearing and seal performance under extended operating conditions. This test was conducted under single-engine loading at 7,400 rpm and optimized parameters of oil flow and preload settings.

After operating a total of 24.75 hours through three load and speed cycles, the test components were inspected and found to be in excellent condition. After this initial inspection, the test was continued, stopping only for machine maintenance as required. At the end of 15 additional hours of testing, the test components were removed and visually inspected. All components were in satisfactory condition. Although this was the end of all scheduled testing for this program, the test rig was reassembled and operated at the endurance load condition for additional times until modifications were completed for conducting a series of oil-off tests. When testing was finally terminated, the accumulated test times were 379.0 hours under single-engine loading, 7,400 rpm, and 389.25 hours for all conditions. The calculated unadjusted catalog lives for this condition were 202 hours L-10 for the heel (DE) and 2,434 hours L-10 for the toe (ODE) positions. All components completed this test in excellent condition. No damage or distress was noted on the components.

Oil-Off Survivability Tests

Six oil-off tests were conducted with tested components from the original program. A summary of results and test parameters was presented in Table 18. A description of component preparation, test procedure, results, and brief discussion of each oil-off test follows. Details of test data and photographs of components after test are included in Appendix C.

Oil-Off Test No. 1

The test was conducted with the bearing components from development test 6. The slave bearings were not used in any oil-off tests. New Belleville springs were designed and fabricated to apply an axial load of 3,209 pounds (50-percent single-engine equivalent load). The assembly of the thrust spring is shown in Figure 69. The instrumentation used for this test was nearly the same as used in the primary program. Thermocouples were located on the cup and seal case outside diameters and were monitored by a multipoint strip chart recorder. The shaft speed sensor output was recorded by a continuous strip recorder.

The test was conducted in the following manner:

1. Bearing and test components were installed in the test rig, then run through a single 8-hour shakedown run. Speed and load were at test levels and oil was supplied at 4 pints per minute per bearing.
2. The rig was disassembled and components were visually inspected.
3. The rig was reassembled, then run until temperatures stabilized at the above-mentioned conditions.
4. The oil pump was shut off (flow meter output verified zero flow).
5. The test was terminated when either the drive belts between the dc motor and speed increaser slipped or audible noises were emitted from the test housing.

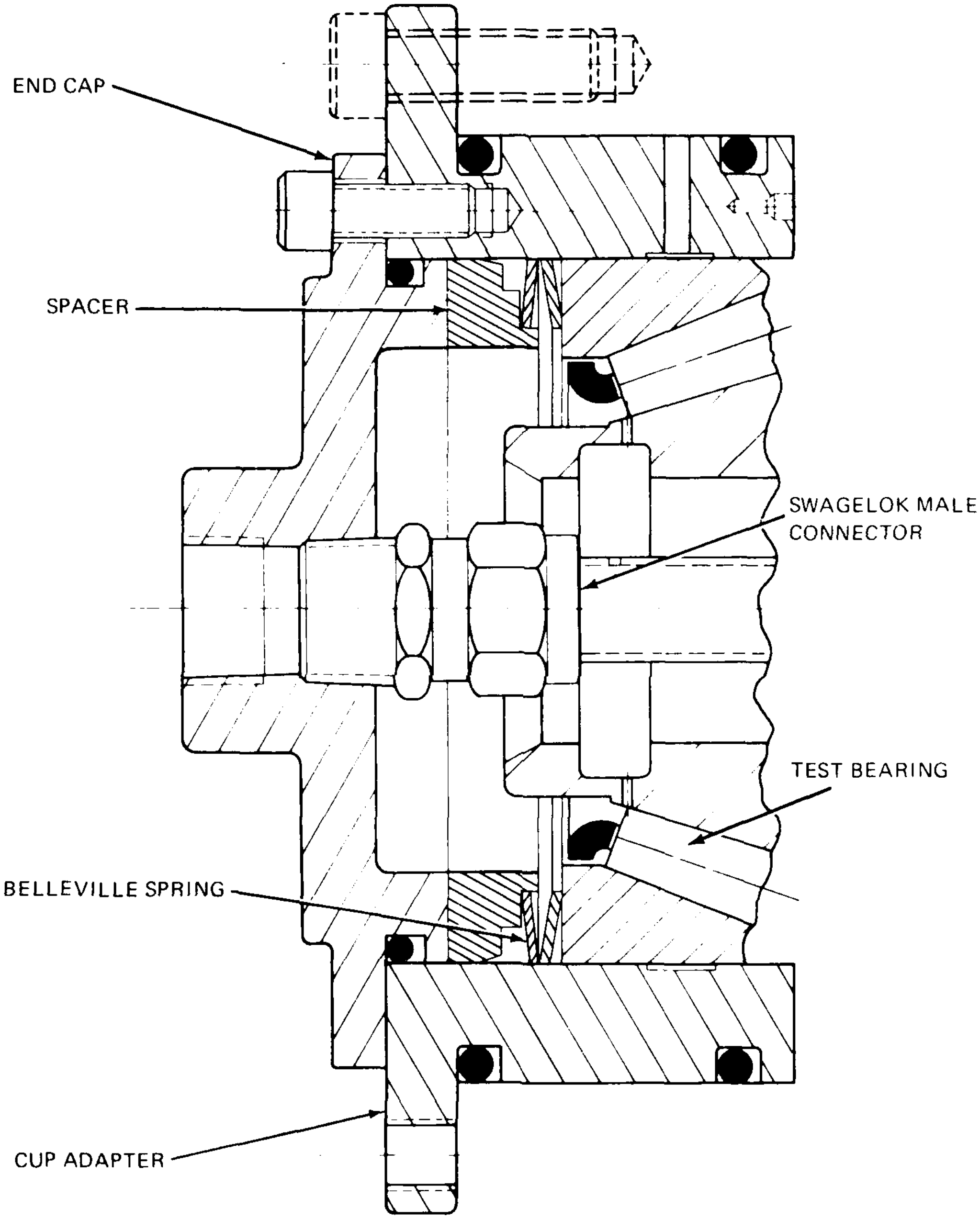


Figure 69. Thrust Load Subassembly for Oil-Off Test.

This first test ran for 1.4 minutes. After teardown, inspection of components revealed that scoring occurred at both rib-roller end conjunctions. Due to the unexpectedly short duration and slow recorder response, only three data points were recorded. During the final instrument cycle of 24 seconds, the temperature of the opposite-drive-end cup rose 80^{O}F. At the completion of this test the cup temperature was recorded as 314^{O}F.

From this limited data, computer analysis was used to formulate a thermal model. Textbook heat-transfer coefficients, considering the bearing as a semi-infinite slab and bearing geometry, produced the appended results. Assumptions were that the rollers were the heat source, cup and cone temperatures would be equal, and the housing temperature would lag cup temperatures as measured in the previous lubricated tests. Subsequent tests would show this last assumption to be completely in error. Regardless, this study as shown in Table 21 did show two significant facts. First, the floating cup (ODE) was tight prior to oil-off (0.0002 inch loose at assembly versus 0.0039 inch tight at oil-off), and even if the ODE cup floated, Belleville spring stiffness and axial thermal growth would apply bearing loads approximately 200 percent greater than intended (6,621 pounds versus 3,530 pounds thrust).

Based upon this thermal analysis, a change in the design of the Belleville springs was initiated in order to produce a spring assembly with a flat spring rate over the expected deflection change. The redesigned Belleville spring is shown in Figure 70. The load versus deflection for the spring used in test 1 and the springs to be used in all future oil-off tests is shown in Figure 71. This new spring design would prevent the buildup of excessive axial loads due to axial expansion of components during oil-off operation.

Oil-Off Test No. 2

Components from development test 2 were used. Revisions to the test hardware and instrumentation as a result of the first oil-off test included:

1. The Belleville springs and their mounting arrangement were redesigned and fabricated to produce a flat-spring-rate curve at the test loads (plotted on graph with data from spring set of test 1 as shown in Figure 71).

2. Three additional thermocouples were located on the test housing. These plus the cup od sensors were monitored by a rapid data logger capable of reading and printing within a 5-second interval.

3. The opposite-drive-end cup housing fit was increased to 0.0031 inch loose.

The same test procedure was followed as in the first test. The bearings operated for 7.8 minutes after loss of all oil supply.

Included in Appendix C for oil-off test 2 and subsequent are the buildup sheets and printout of a revised computer program that presents both measured and calculated test parameters. Table 22 provides a sample of the computer output for test 2.

Experimental data shown on this printout is "Sec", time in seconds with 0 being oil-off; "Hsg" temperature, the average housing od temperature measured at three locations; temperatures of "DE" and "ODE" cups. "Cup" is the measured cup temperature and "Rlr" and "C/S" were calculated roller and cone shaft temperatures. Fits "DE" and "ODE" were computed cup fits at measured temperatures. "Axl" displayed the thermal and inertial expansions across the cup backfaces and "Lb" presented this in terms of pounds preload.

TABLE 21. RESULTS OF THERMAL MODEL OF OIL-OFF TEST NO. 1

Speed (rpm)	Assumed Temperatures (^oF)					Cup Fits at Temp (Calculated, in.)		Change of Setting 0 = > 0.043 in.	
	Ref	Hsg	Cup	Rlr	C/S	DE	ODE	Deflect. (in.)	Load (lb)
Initial Cup Fits: DE 0.0014 in. ODE –0.0002 in.									
1	70	70	70	70	70	0.0014	–0.0002	0.0000	3,530
3,700	70	70	70	70	70	0.0014	0.0002	0.0001	3,534
3,700	70	199	220	220	190	0.0055	0.0039	0.0140	4,571
3,700	70	214	240	260	240	0.0064	0.0048	0.0212	5,082
3,700	70	228	260	280	260	0.0070	0.0054	0.0236	5,250
3,700	70	243	280	340	280	0.0081	0.0065	0.0288	5,609
3,700	70	257	300	380	300	0.0090	0.0074	0.0325	5,860
3,700	70	272	320	420	320	0.0099	0.0083	0.0363	6,116
3,700	70	286	340	460	340	0.0108	0.0092	0.0401	6,370
3,700	70	300	360	500	360	0.0116	0.0100	0.0439	6,621

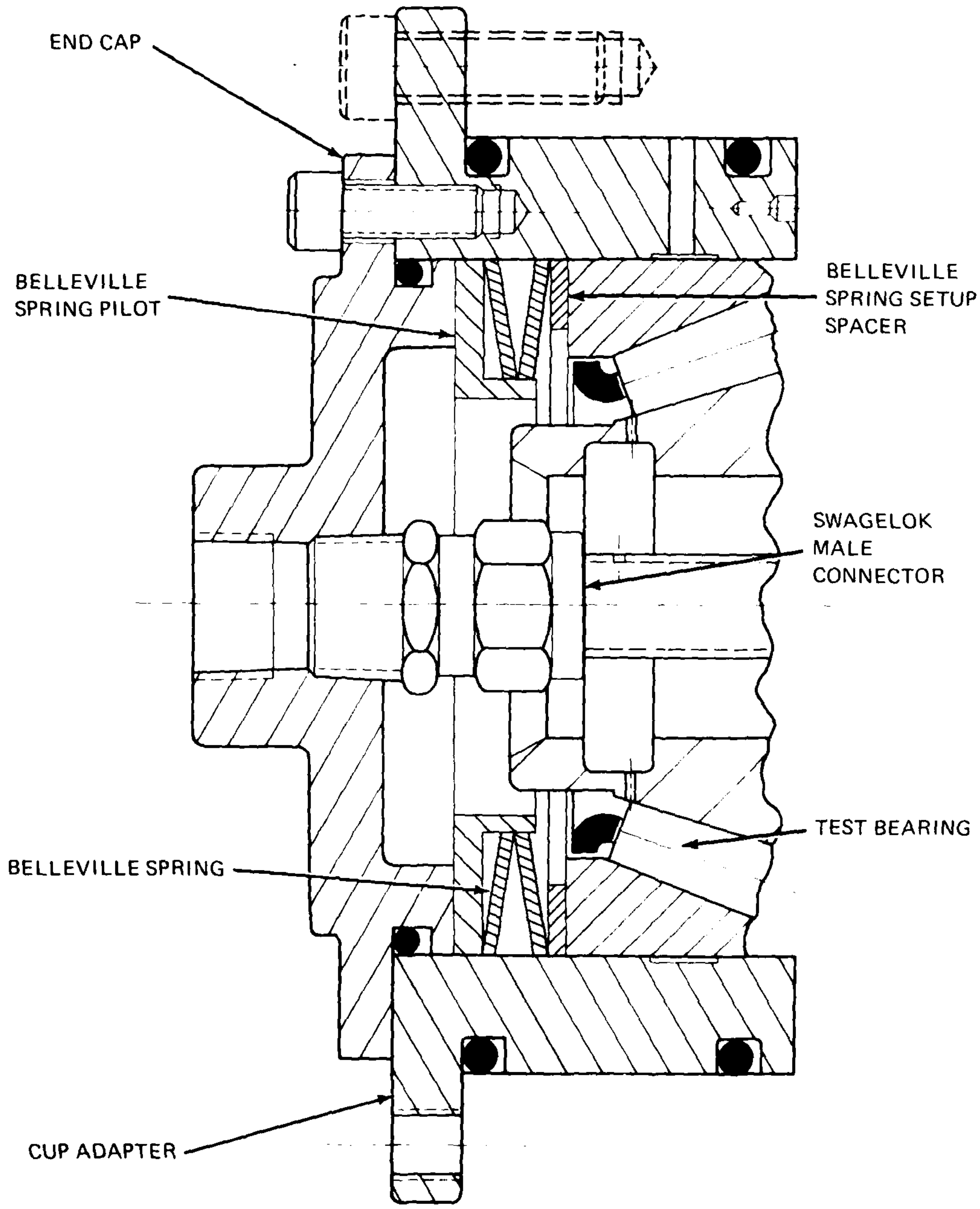


Figure 70. Revised Thrust Load Subassembly for Oil-Off Test.

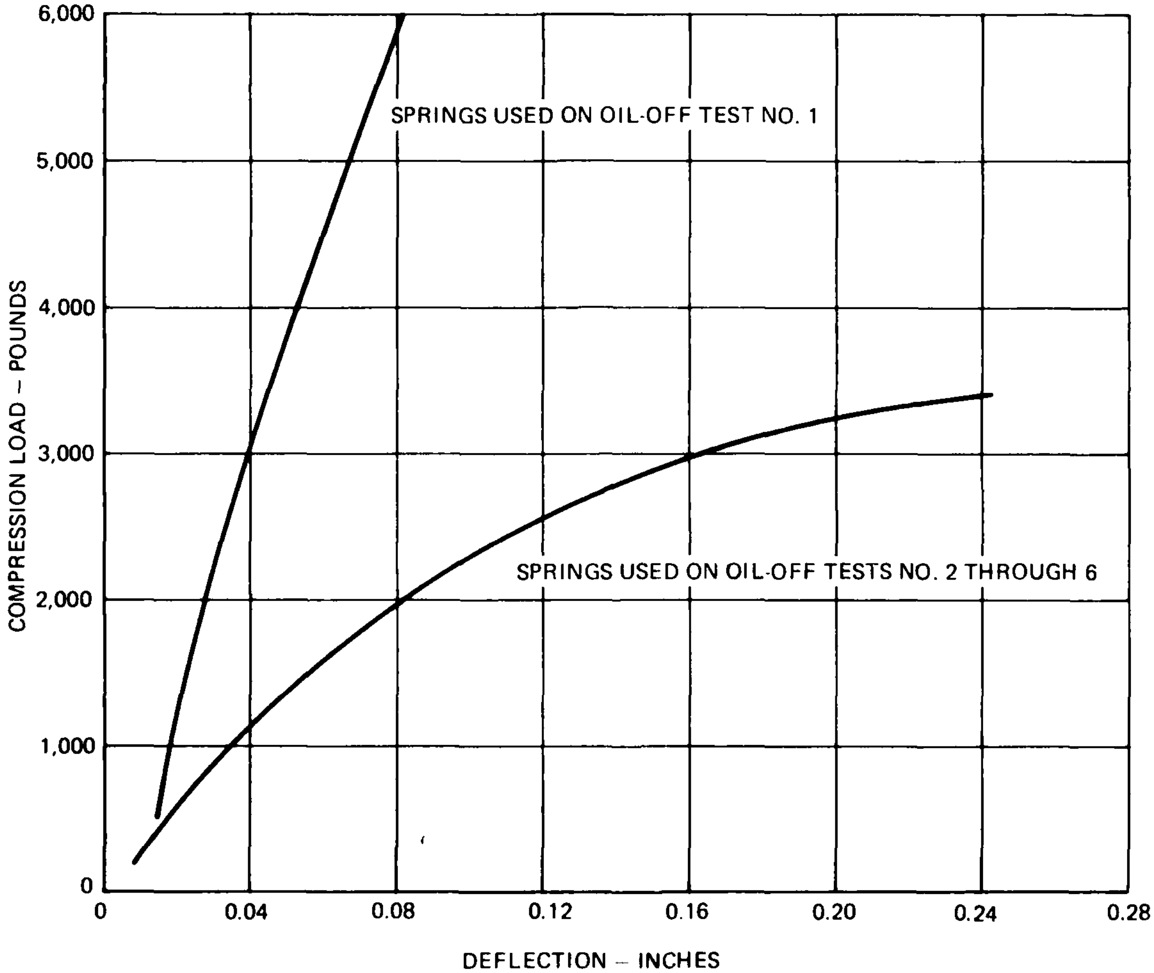


Figure 71. Deflection Curves for Springs Used in Oil-Off Tests.

TABLE 22. COMPUTER DATA OUTPUT FOR OIL-OFF SURVIVABILITY TEST NO. 2

Speed 3,700 rpm   Thrust Load 3,200 lb

Initial Cup Fits DE 0.0016 in. ODE −0.0031 in.

	Temperature (^oF)								Fits (in.)			
			DE			ODE						
Sec	Ref	Hsg	Cup	Rlr	C/S	Cup	Rlr	C/S	DE	ODE	Axl	Lb
0	70	181	196	196	197	198	198	197	0.0048	0.0001	0.0126	3,205
5	70	180	197	196	197	198	199	198	0.0048	0.0001	0.0126	3,205
11	70	181	197	196	197	198	200	198	0.0048	0.0001	0.0127	3,206
16	70	181	202	207	200	199	201	198	0.0048	0.0001	0.0130	3,208
21	70	181	211	226	207	200	203	199	0.0049	0.0001	0.0136	3,212
26	70	180	220	243	212	200	203	199	0.0050	0.0001	0.0141	3,215
435	70	181	300	403	265	247	297	230	0.0058	0.0006	0.0214	3,258
440	70	181	303	408	267	253	309	234	0.0058	0.0006	0.0219	3,260
445	70	181	305	414	269	271	345	246	0.0058	0.0008	0.0231	3,267
450	70	181	309	421	271	289	381	258	0.0059	0.0010	0.0243	3,274
456	70	181	313	428	273	309	421	271	0.0059	0.0012	0.0256	3,281
461	70	181	321	445	279	330	462	285	0.0060	0.0014	0.0273	3,289
466	70	183	339	481	291	352	507	299	0.0062	0.0016	0.0295	3,301

The results of the axial spring displacement and ODE cup fit for test 2 are plotted and shown in Figure 72. This plot shows that as the fit of the floating cup became tight, its ability to float axially diminished and the buildup of axial preload increased, resulting in bearing failure. This test appeared to verify the mechanism of failure and showed that the changes made in test 2 did increase the oil-off life of the test assembly.

The least-expected result of this test was that the housing temperatures actually cooled slightly at oil-off, then remained constant for the remainder of the run. This stable housing temperature with increasing cup temperature removed the floating capability of the ODE cup. This event coincided with a rapid increase in temperature at this position. This effect was not included in the initial thermal model. Therefore a review of the thermal model indicated that additional looseness of the floating cup fit would be required in order to prevent the cup fit from becoming tight under higher operating temperatures.

Oil-Off Test No. 3

Components from development test 1 were used. Speed and load levels were the same as the first two runs. The cup fit of the opposite-drive-end cup was ground for 0.007 inch looseness.

The test ran for 2.48 minutes. Posttest component inspection revealed the reason for this premature termination. The cage pilot-cup land at the large end of the drive end position had interfered, then welded. The welding broke at coastdown; however, it could be seen that the bearing had run with zero cage speed, imbedding the rollers into both inner and outer races. Posttest measurement of this pilot showed 0.014-inch wear on the pilot od. The ODE bearing sustained no damage. The test data showed that this position was cooling down at termination. The results of this test indicated that additional clearance would be required for the cage pilot-cup land in order to prevent this mode of failure.

Oil-Off Test No. 4

The initial plan for this test was to use the test parts from development test number 3; however, it was noted at assembly that the opposite-drive-end cup rib had sustained handling damage. The damaged cup was replaced by the undamaged cup from oil-off test 3, serial no. 78-1. To eliminate conditions that had terminated the earlier tests, the following modifications were performed:

1. New preload springs.
2. Both cup outside diameters ground for looser fits.
3. Assembled cage pilot clearances increased by grinding inside diameters of cups.

The load, speed, and procedure were repeated as before and the bearing survived for 8.88 minutes of oil-off operation.

Appendix C contains the buildup sheet, computer printouts of test data, graphs of same, and posttest component photographs.

In addition to the cup rib-roller spherical end damage, it appeared that the cage pilot also interfered at the ODE position. The condition was not as severe as in oil-off test no. 3, but the pilot at the large end had 0.005-inch wear.

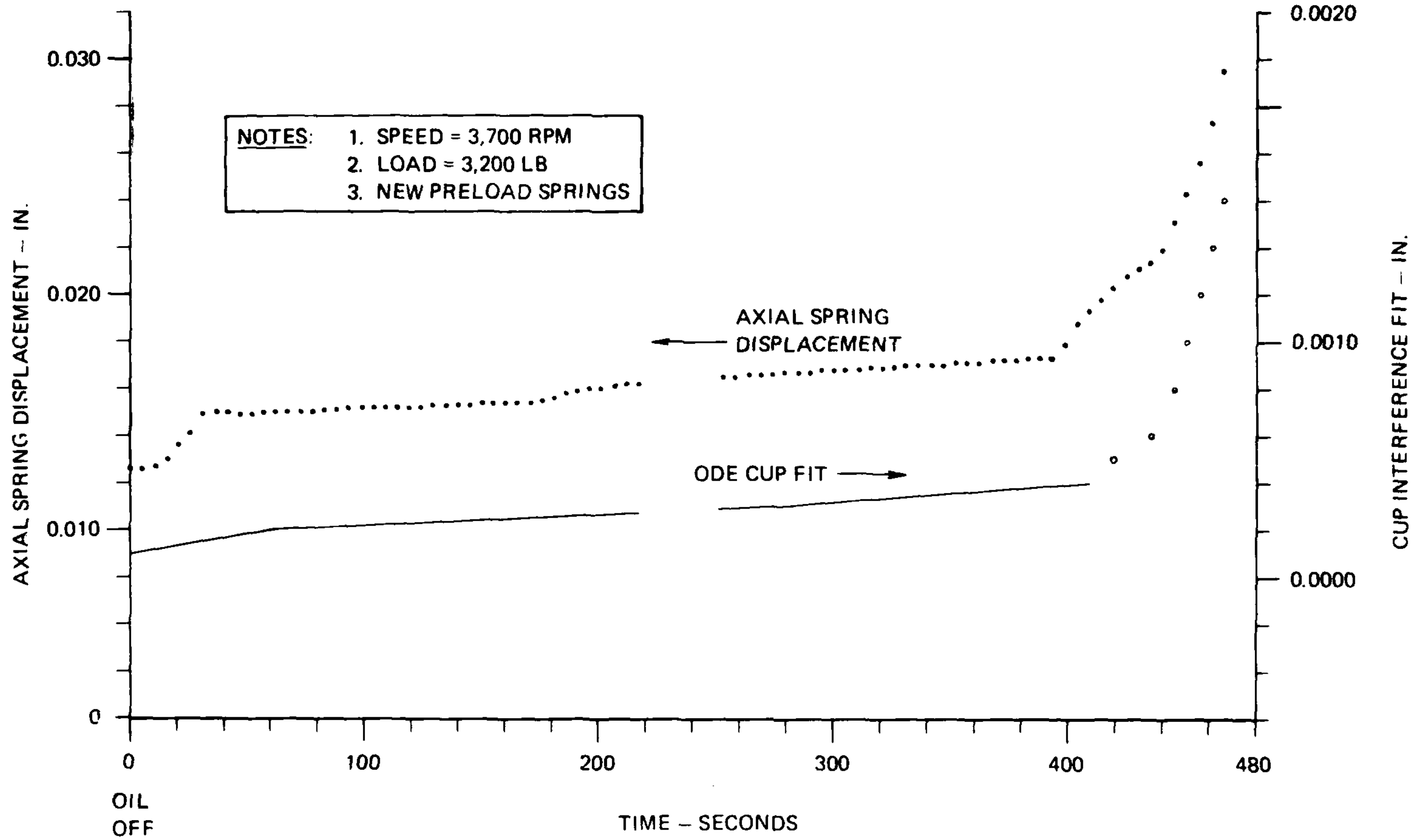


Figure 72. Spring Displacement and Cup Fit for Oil-Off Test No. 2.

Oil-Off Test No. 5

This test was run with the components tested under development buildup no. 4. The work statement pertaining to the oil-off contract extension had specified that a series of tests at speeds from 3,700 rpm to 14,000 rpm be conducted. Because of the unexpected problems, all the previous trials had been run at 3,700 rpm in order to understand the effects of each modification on life. To investigate the influence of speed, this test was performed at 7,400 rpm. This is the normal operating speed of the advanced-concept input pinion.

The three revisions specified for oil-off test no. 4 were incorporated. In addition, the cup lands at all positions were zinc-phosphate-coated. This was accomplished by chemically masking all bearing surfaces except the cup lands.

The test ran for 4.37 minutes. The buildup sheets, test data, graphs, and posttest component photographs are shown in Appendix C. The results of this test indicated that a longer operating life was achieved than expected with increased speed. It was speculated that the zinc-phosphate coating may have extended the life and therefore a sixth test was conducted at the lower speed (3,700 rpm) in order to verify this factor.

Oil-Off Test No. 6

The bearing parts for this test had been used in development test no. 5. All modifications developed in the previous five oil-off tests were incorpoated. The speed and load levels were 3,700 rpm at an equivalent thrust of 3,205 pounds (50 percent of single-engine loading).

The test ran for only 3.67 minutes. Appendix C includes buildup and data sheets, a graph of the cup and housing temperatures, and posttest component photographs.

As in the preceding tests, the origin of damage was the cup rib-roller spherical end. It was also apparent that the cage pilot-cup land contributed a significant amount of heat. Based on the results of these six oil-off tests, it was apparent that operating clearances under transient thermal conditions play an important part in the performance of these bearings during oil-off operation. A summary of the various housing cup fits and cup/cage clearances used in these six tests is shown in Table 23. These tests provided evidence that the modifications of increased clearance did extend life but were not sufficient to achieve our goal of 30 minutes operation without oil.

DISCUSSION OF TEST RESULTS

All test data presented in Appendixes B and C of this report was accumulated and stored in computer data files of the Timken Company's Physical Laboratory. Speed, load, temperatures, and oil flow rates were measured data, while the two columns under "HEAT" (generation) were calculated. The heat generation of the test and slave bearings (column headed "BRG") was computed considering the following:

1. Rolling torque equation as developed by Witte[6].

6. Witte, D.C., OPERATING TORQUE OF TAPERED ROLLER BEARINGS, presented at ASME-ASLE International Lubrication Conference, New York, New York, October 1972.

TABLE 23. SUMMARY OF HOUSING FITS AND CAGE CLEARANCES USED IN OIL-OFF TESTS

Oil-Off Test	Oil-Off Condition Speed (rpm)	Housing Cup Fit (in.)		Cup Pilot ID/Cage OD Clearance (in.)			
				Drive End		Opposite Drive End	
		DE	ODE	Lge End	Sm End	Lge End	Sm End
1	3,700	0.0014 tight	0.0002 loose	0.0063	0.0066	0.0063	0.0082
2	3,700	0.0017 tight	0.0031 loose	0.0077	0.0062	0.0065	0.0070
3	3,700	0.0016 tight	0.007 loose	0.0060	0.0065	0.0065	0.0070
4	3,700	0.0080 loose	0.007 loose	0.0103	0.0093	0.0098	0.0105
5	7,400	0.0081 loose	0.0098 loose	0.0152	0.0106	0.0145	0.0101
6	3,700	0.0079 loose	0.0078 loose	0.0154	0.0010	0.0152	0.0102

2. Viscous torque (required to develop boundary layer and propel lubrication through cup id) as derived by Leibensperger[7].

3. Hydrodynamic losses at test cage pilots computed using the equations of Fuller and Smith, with conditions given in Reference 8.

4. Lubricant dynamics   Torque to accelerate lubricant to cage speed in slave bearings and centrifugal pumping losses in both test and slave bearings.

The heat removed by the lubricant (column labeled "OIL") was computed using measured temperatures and flow rates. The mean specific heat of the lubricant was computed using the fluid properties from Reference 8 and integrated over the temperature range,

$$Q_{oil} = \dot{M}\,\bar{Cp}\,\Delta T,$$

where $\dot{M}$ = mass flow rate

$\bar{Cp} = \dfrac{\int_1^2 Cp\,(T)\,dT}{T_2 \quad T_1}$

$\Delta T = T_2 \quad T_1$

and T_2 = oil outlet temperature

T_1 = oil inlet temperature.

The temperatures used for the torque equations were outlet oil conditions. When lubricant was supplied to the floating cup seat at the ODE position, the measured oil outlet temperatures were compensated to consider this thermal dilution effect.

If we apply a generalized rule that 10 percent of the heat generated will be transferred through the housing by conduction and radiation, then excellent correlation is achieved between heat generated and heat carried away by the oil. Under high speed and temperature conditions, the somewhat greater than 10 percent value was likely as a result of the following:

1. Rib-roller spherical end effects neglected

2. Assumed constant spring preload (neglected axial thermal expansions) over all conditions.

The elastohydrodynamic film thicknesses were also computed for the test bearings using the equation of Dowson and Higginson for the roller body-race conjunctions and the Archard and Cowking formula for the roller spherical end-rib conjunctions. The application of these equations to tapered-roller bearings and assumptions are given in Reference 8.

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7. Leibensperger, R.L., AN ANALYSIS OF FLOW OF OIL THROUGH A TAPERED ROLLER BEARING, Journal of Lubrication Technology, American Society of Mechanical Engineers, New York, New York, April 1972.
8. Cornish, R.F., Orvos, P.S., and Dressler, G.J., DESIGN, DEVELOPMENT AND TESTING OF HIGH-SPEED TAPERED ROLLER BEARINGS FOR TURBINE ENGINES, Timken Company, Technical Report AFAPL-TR75-26, U.S. Air Force Aero Propulsion Laboratory, Wright-Patterson Air Force Base, Ohio, July 1975, ADA026908.

The results of these calculations are shown in Figure 73. The lower curves show the temperatures used for the calculations. The upper series of curves presents the film thickness at speeds and temperatures identified. The 190°F inlet oil conditions developed elastohydrodynamic film separations approximately twice those of 300°F oil (12-14 microinches versus 6-8, respectively).

Both of these conditions would be considered boundary lubrication and it was expected that visual evidence of surface contact would be recorded on tested components. Figures 74, 75, and 76 are scanning-electron-microscope photographs of inverted replicas of the outer races from a new bearing and those tested at 190° and 300°F oil inlet. The solid bars are reference scales for 100 micrometers (approximately 0.004 inch). The biased lines are honing scratches that are apparent on the new races. Comparison of the two tested races shows that almost all honing marks were worn from the 300°F tested cups. Although the bearing performed successfully under these conditions, it is apparent that the reduced elastohydrodynamic film thickness has resulted in a change to the surface condition of the tested bearings.

The magnetic seals used in this test program performed very satisfactorily. Eight different seals were used and tested under various conditions as previously noted. None of the seals had to be replaced or modified during this test program. A summary of the measured wear and weight losses recorded from measurements taken before and after test is shown in Table 24. The largest amount of wear recorded was during test 3 and measured 0.0003 inch. Weight losses were generally less than 0.033 gram. During the eight tests conducted, only two recorded any evidence of oil leakage. The largest amount of oil leakage occurred during test 8 and was recorded as only 5.2 grams of oil.

In addition, the seal case temperature at various speeds was plotted as shown in Figure 77. The temperatures shown on this plot were taken at stabilized conditions under maximum single-engine loads. As shown, seal temperature increased with speed and reduced oil flow rates to the test bearings. The highest seal temperature recorded with 190°F oil inlet temperature occurred during test 6; a temperature of 313°F occurred at full load and 11,800 rpm. When the oil inlet temperature was increased to 300°F, seal temperatures in excess of 330°F were measured and no distress was noted.

TABLE 24. SUMMARY OF MAGNETIC SEAL LEAKAGE AND WEAR

Test No.	Test Time (hr)	Measured Wear (in.)	Measured Weight Loss: Magnetic Ring (gm)	Measured Weight Loss: Seal Case/ Carbon Insert (gm)	Total Recorded Leakage (gm)
1	37.6	0	0.012	0.0126	0
2	58.0	0.0002	-0.006	0.0123	0
3	39.5	0.0003	+0.0012	0.0068	0
4	54.8	0	+0.002	+0.001	0
5	61.0	0.0001	0.008	+0.022	0
6	101.3	0	0.001	0.0004	0
7	67.5	0.0002	0.014	0.0327	3.017
8	379	0.0002	+0.001	0.023	5.20

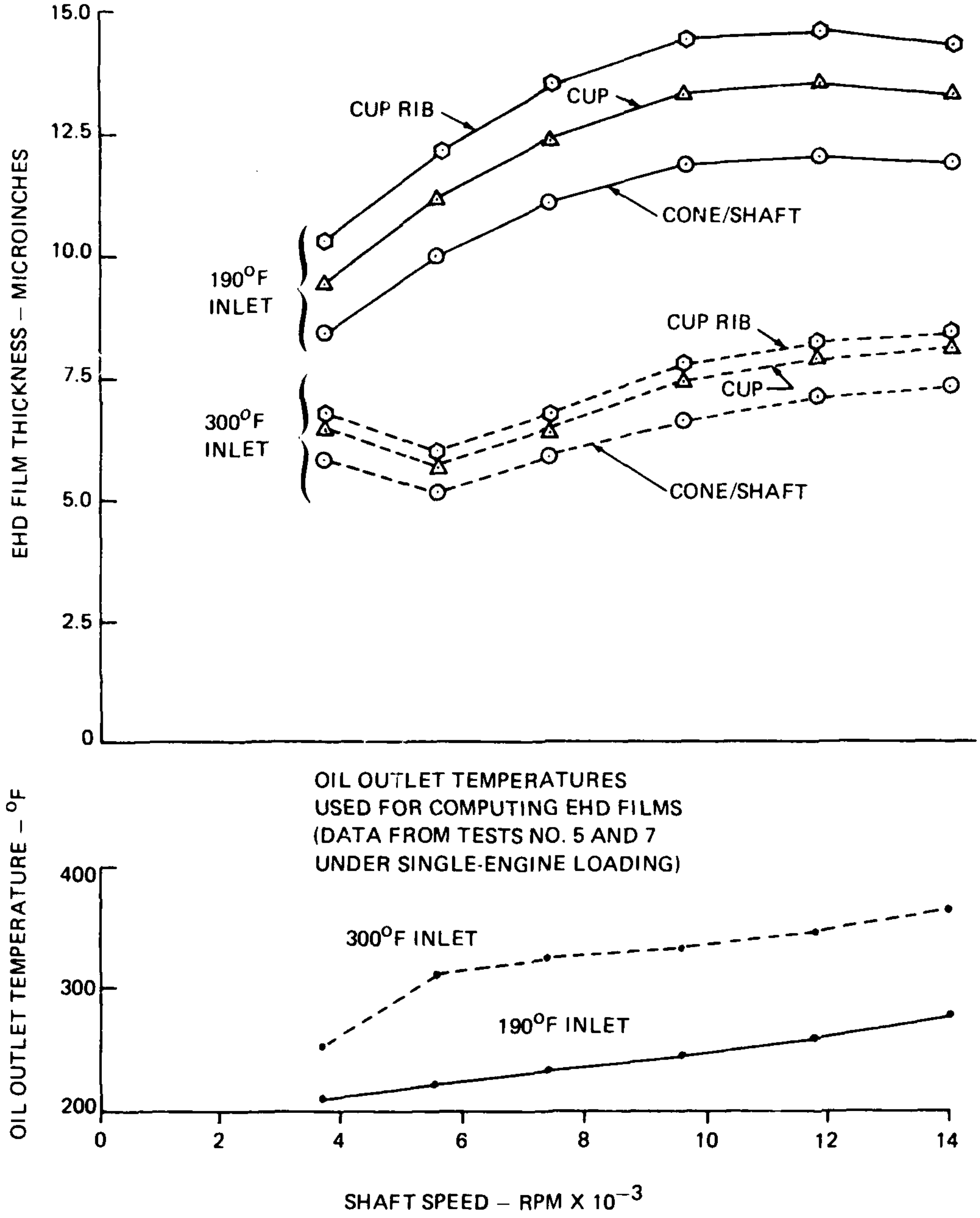


Figure 73. Elastohydrodynamic Film Thicknesses During Oil-Off Tests.

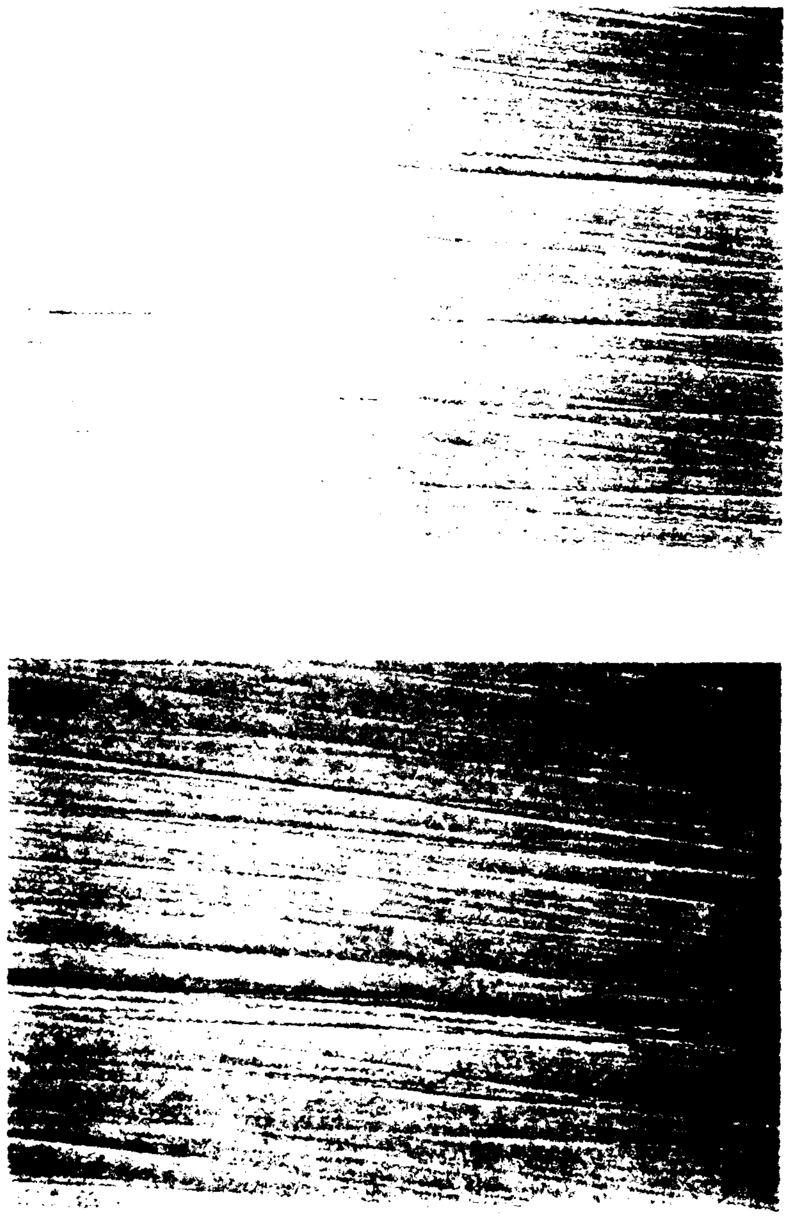


Figure 74. Scanning-Electron-Microscope Photographs of Inverted Replica of New Bearing Outer Race.

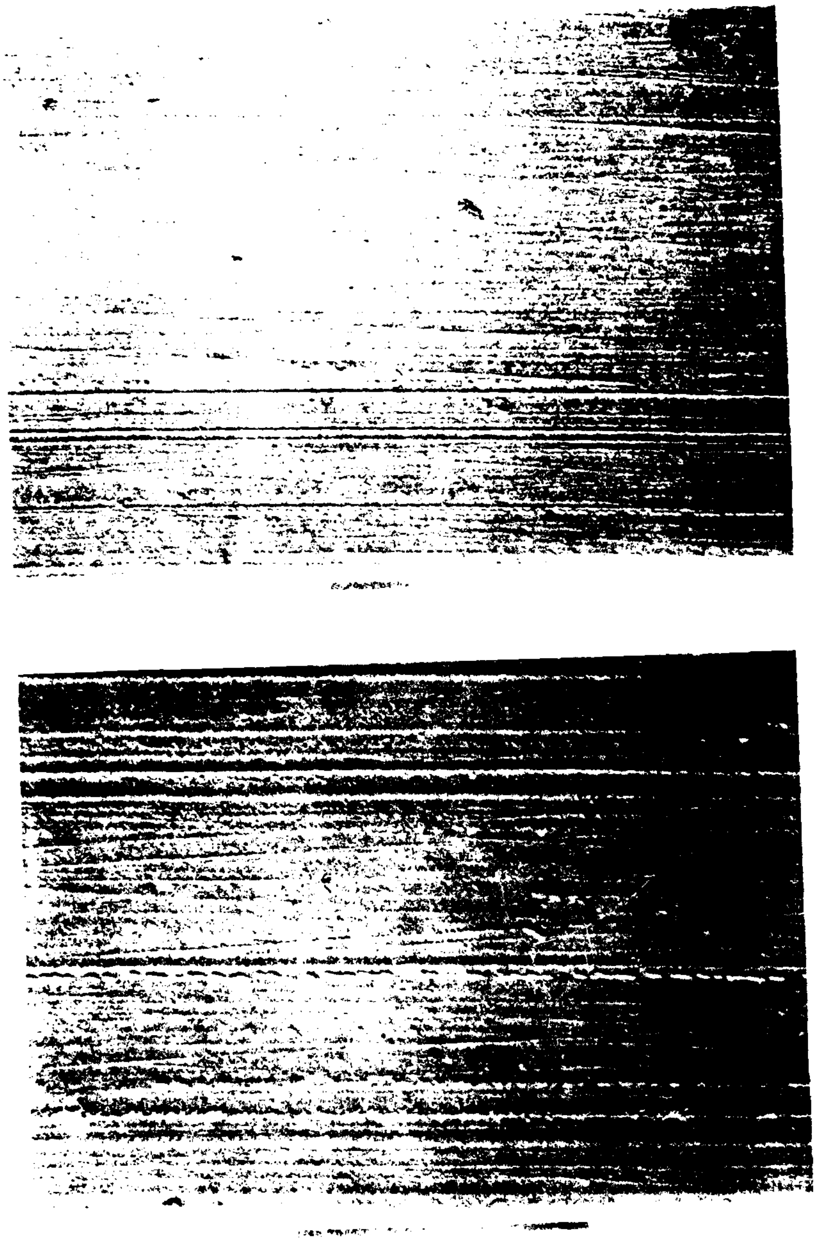


Figure 75. Scanning-Electron-Microscope Photographs of Inverted Replica of Bearing Outer Race From Test No. 6 at Oil Inlet Temperature of 190°F.

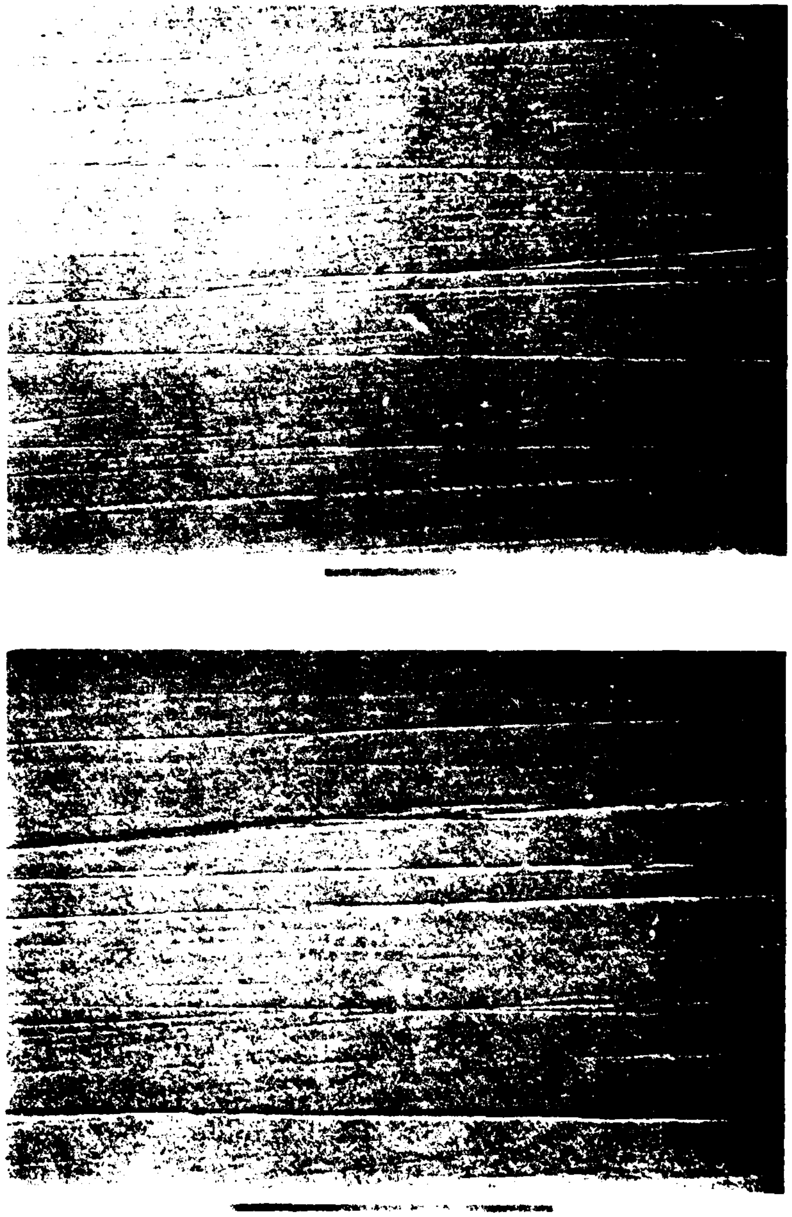


Figure 76. Scanning-Electron-Microscope Photographs of Inverted Replica of Bearing Outer Race From Test No. 7 at Oil Inlet Temperature of 300°F.

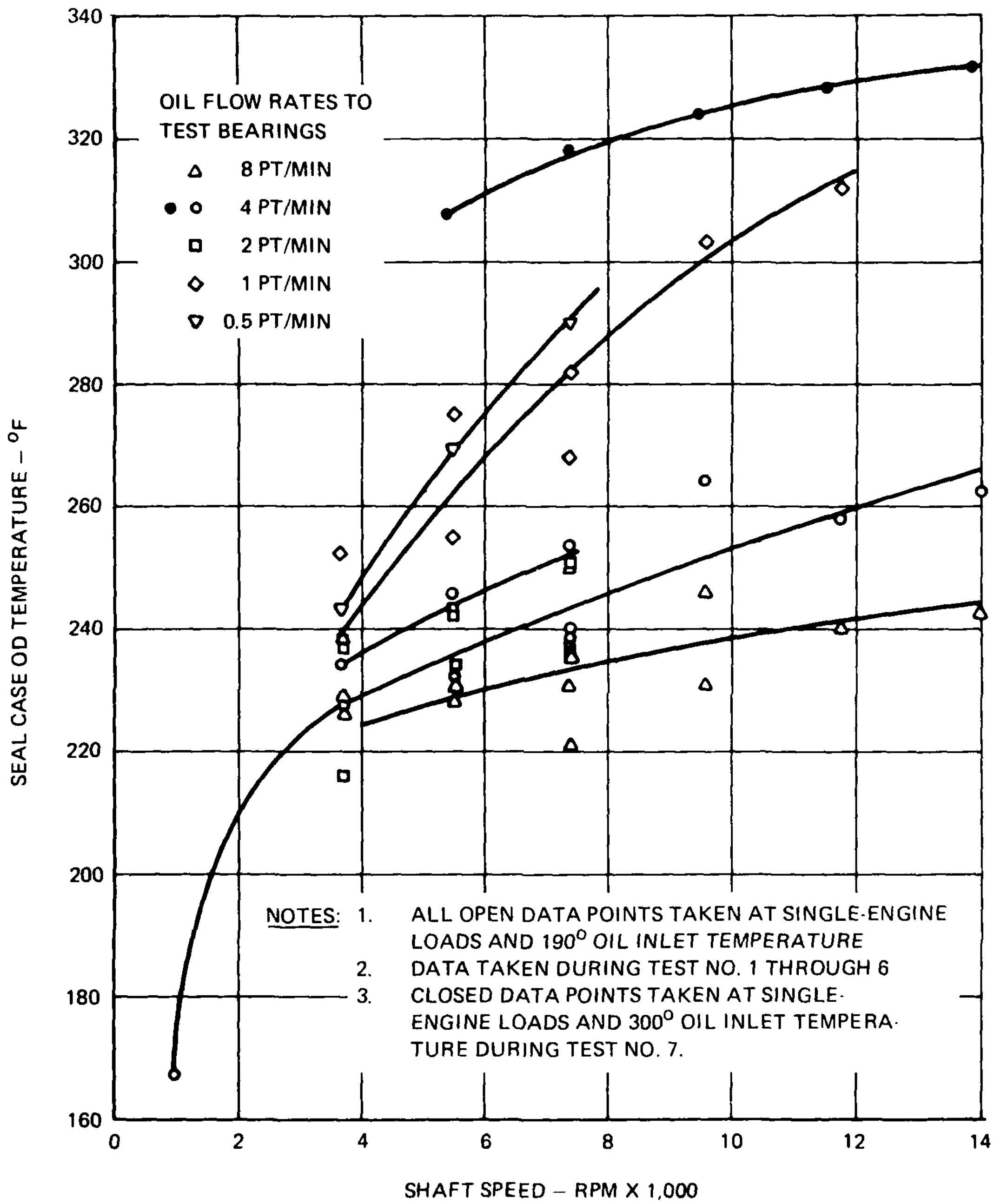


Figure 77. Temperatures at Outside Diameter of Magnetic Seal Case.

CONCLUSIONS

This program successfully demonstrated that a ribbed-cup, integral-inner-race shaft, tapered-roller bearing system and magnetic seal could be designed and fabricated to meet all requirements for an advanced helicopter input spiral bevel pinion application. All performance goals specified in the program statement of work were met or exceeded. These were the following:

1. Operate under single-engine loading (≈1,500 hp) at nearly twice design speed (14,000 rpm).
2. Demonstrate the feasibility of complete through-shaft lubrication.
3. Operate at minimum lubricant supply conditions.
4. Establish floating bearing requirements with regard to spring preload and lubrication.
5. Integrate bearing inner race with test shaft.
6. Fabricate test components from a high-hot-hardness carburizing steel (VASCO-X2 and CBS600).

Beyond the original scope of the program, it was shown that elevating the inlet oil temperature from 190°F to 300°F had no adverse impact on the bearings, and a single endurance test of 1.88 L-10 units duration did not produce bearing fatigue.

During the initial two tests, abrasive wear and debris denting were observed on the test components. At the conclusion of test number 6 an examination of the oil sump revealed heavy contamination with metallic debris. The source of this contamination appeared to be the new test rig; however, it was thought that initial flushing had cleaned the system. It was also postulated that a chip from the magnetic seal insert had become wedged in the cage pocket, producing the heavy roller grooving during test number 1. In any case, this did not adversely influence bearing performance and was eliminated prior to the endurance test.

The only bearing problems encountered were those relating to the slave bearings. These were solved by fitting and setting changes which occurred during tests number 2 and 3.

The magnetic seals were used in all tests and performed very satisfactorily, with no indications of leakage or wear.

The five oil-off survivability tests conducted at 3,600 rpm ran from 1.4 to 8.88 minutes. The single test at 7,400 rpm survived for 4.37 minutes. This latter test is within the range of thrust ball bearing designs.

The success of this program indicates that ribbed-cup tapered-roller bearings as tested and reported herein are ready for full-scale transmission tests.

The oil-off survivability tests did not reach the military goal of 30 minutes operation at maximum rated power. However, a significant advancement was achieved. It is possible that longer times could be achieved if testing were performed in an environment and mounting similar to those of a helicopter transmission. Additional work should be done on material development and cage design in order to extend the oil-off survivability of a ribbed-cup tapered-roller bearing.

ADVANCED ANALYSIS OF COMPLEX BEARING STRUCTURES

BACKGROUND

To achieve the major objectives of an advanced transmission, studies have shown that significant changes in the design of a planetary system are required. These initial design studies were based upon the Boeing Vertol UTTAS-class helicopter which used a single-stage planetary system with conventional spur gears supported by spherical roller bearings An advanced main rotor transmission developed under Contract DAAJ02-75-C-0022 incorporated many new features such as composite housing material, advanced ribbed-cup tapered-roller bearings, and a new concept planetary and rotor shaft support system. The selected design of the advanced-concept transmission is shown in Figure 78.

Some of the basic details used in this design study were as follows:

The overall reduction ratio is 25.1 to 1; the engine bevel drive accounts for the balance (67.6 to 1). Power inputs are at the 90- and 270-degree positions. Provision for a forward AGB drive is made at 0 degrees and for a tail rotor and aft AGB drive at 180 degrees.

The loads criteria to which this transmission has been designed are shown in Table 25.

TABLE 25. DESIGN LOADS OF THE ADVANCED-CONCEPT TRANSMISSION

Maximum Single-Engine Input Horsepower	1,521
Input RPM	7,419
Output Horsepower	2,655
Output RPM	295
Output Torque	562,730 ± 67,530 in.-lb
Lift Load	17,004 ± 567 lb
Drag Load	609 ± 586 lb
Rotor Hub Moment	262,000 in.-lb

These are the same as the Boeing Vertol YUH-61A criteria.

In the proposed advanced design (Figure 78), the planetary gear reduction and the rotor support functions are an integral design. Consequently, the development of a detailed analysis of the interrelationship of load and deflections is required to design the system. The planet gears used in this planetary system are high-contact-ratio noninvolute form (HCR/NIF), designed to approximately the same stresses as the conventional teeth of the baseline design. This advanced tooth form allows a 20-percent reduction in planetary volume.

The planet gears are supported on a carrier plate that runs between the upper and lower tier of planet gears (Figure 79). This effect is to balance the load on the plate and eliminate bending in this member. It also shortens the planet post to about half of the conventionally required length as shown in Figure 80. The cumulative effect is to significantly reduce the tendency of post deflection to end-load the gears and the bearings. In consequence of this reduced deflection, cylindrical roller bearings are used in place of self-aligning sphericals. Because of this planet carrier/bearing design, the life shows a marked improvement over conventional designs despite the reduction in planetary size.

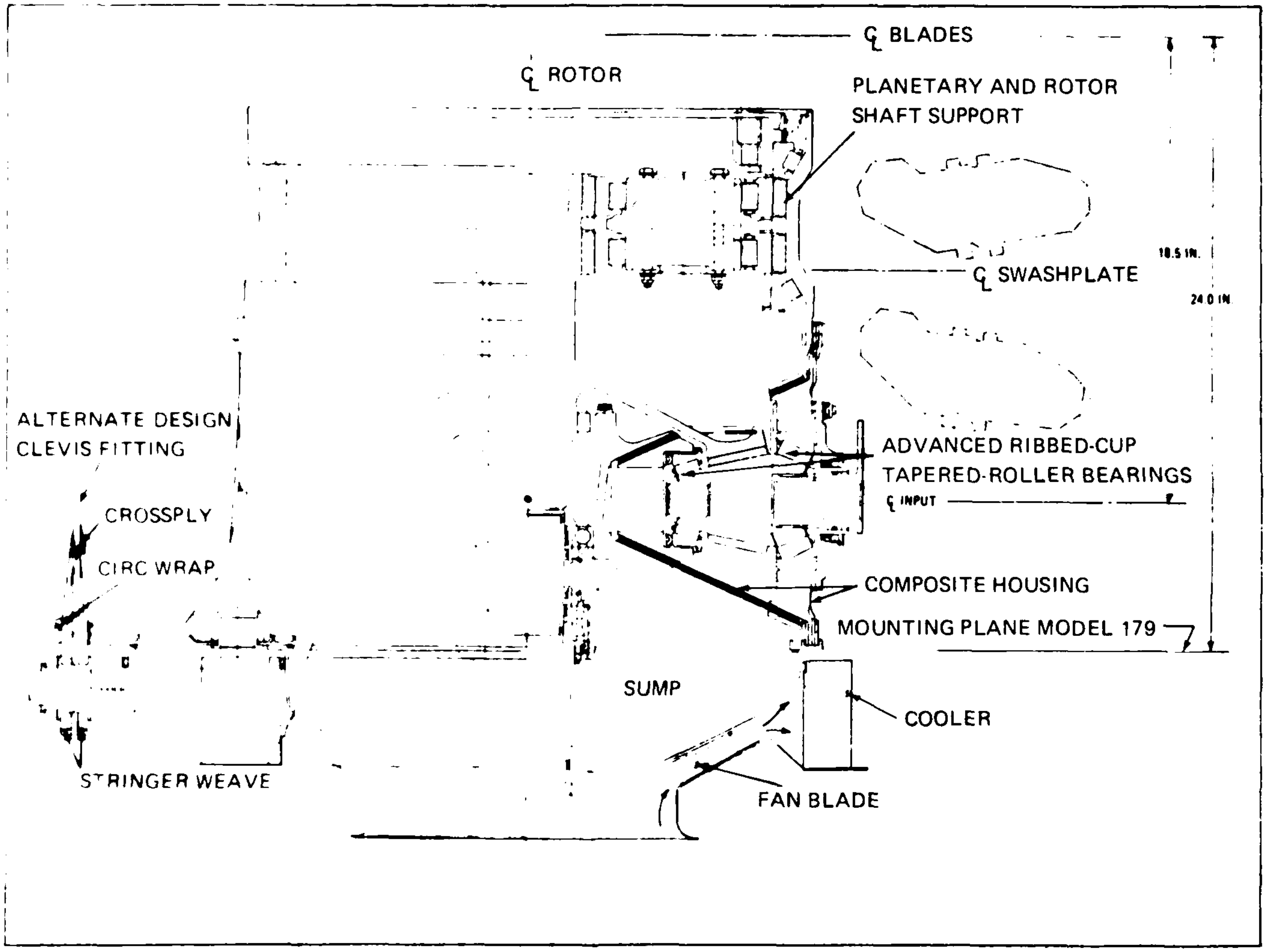


Figure 78. Advanced-Concept Transmission With Composite Housing.

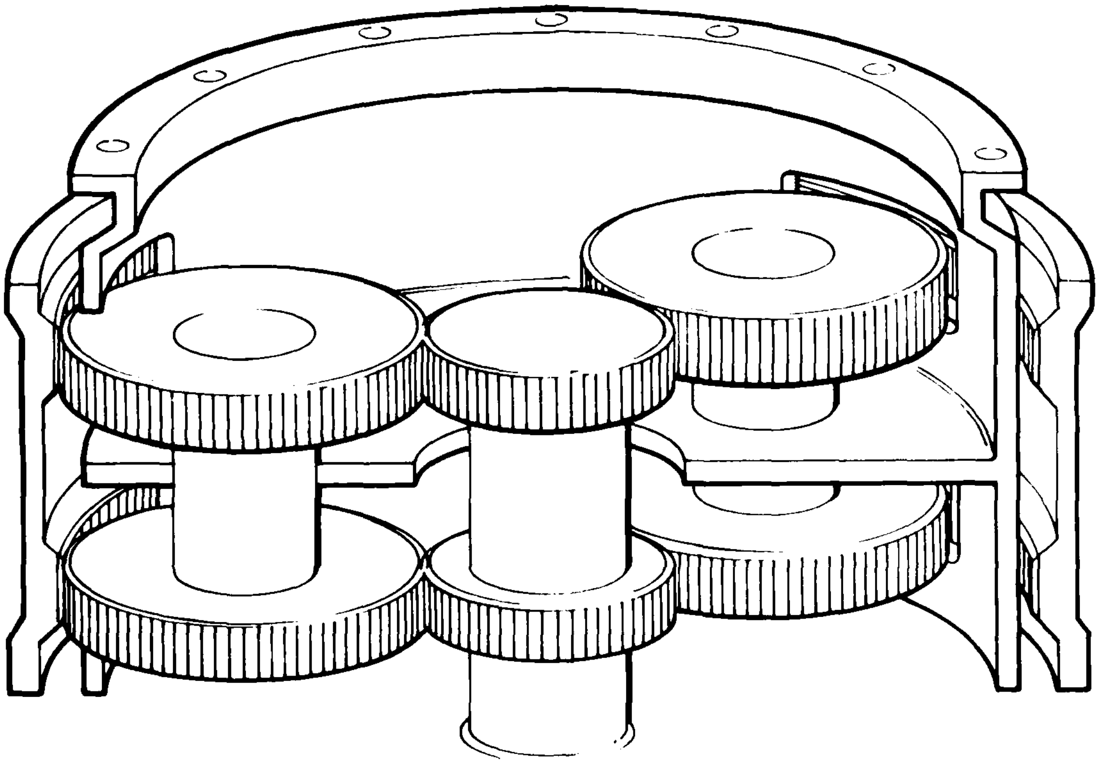


Figure 79. Advanced-Concept Planet Carrier.

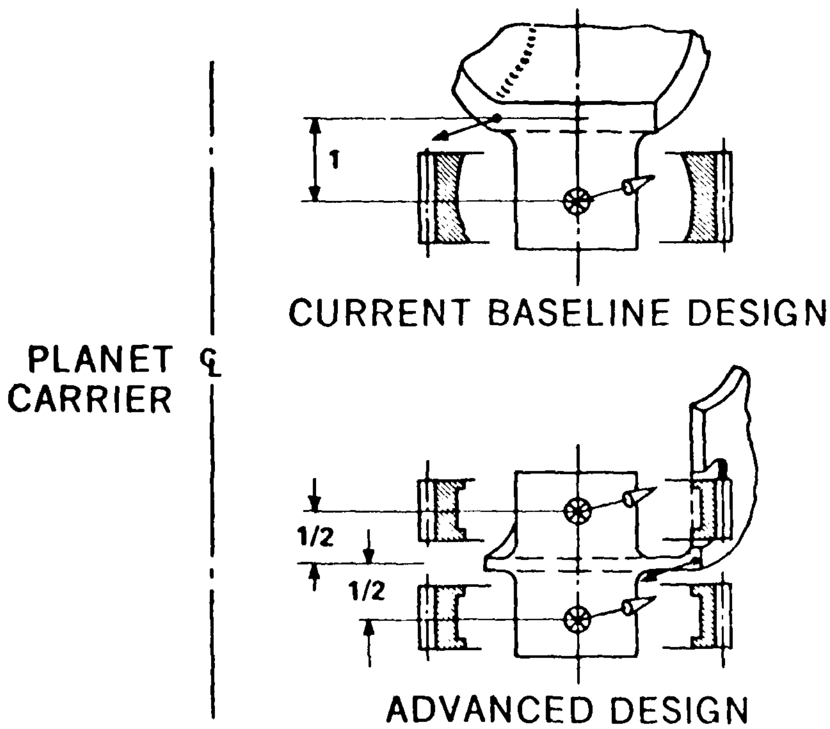


Figure 80. Balanced Planetary Post Loads.

At its periphery, the balanced planetary carrier connects to the rotor hub through a short tubular extension. On the outside of the carrier, the rotor loads are reacted through a set of large-diameter tapered-roller bearings. These, in turn, are retained in the planet ring gear which transfers the rotor loads to the housing and ultimately to the airframe. The design also incorporates the bearing outer races above and below as an integral part of the ring gear. Except for this, the bearing design itself does not represent a change in the state of the art, being similar to the YUH-61A.

The notable task in this area is the evaluation of the combined effects of gear, rotor, and bearing loads and deflections. Provisions were made in the ring gear to increase stiffness at low weight penalty by the addition of high-modulus material wound circumferentially on the outside.

To summarize the expected benefits of this planetary system, they are:

- Reduced diameter, height, and volume
- Increased gear and bearing life
- Minimum-weight and minimum-length rotor load path.

DESCRIPTION OF DESIGN

In a conventional main rotor transmission, the rotor loads are transferred to the airframe through mounting legs on the upper cover. Hence, the ring gear which is supported under the upper cover is not required to transmit these loads. Conversely, in the advanced-concept transmission design, as described earlier, the rotor loads are transmitted by the rotor shaft support bearings through the ring gear structure and transmission case and then into the helicopter frame. Since both the rotor loads and torque paths are through the ring gear structure, it is important to determine the deflections and stresses imposed upon the ring gear and rotor shaft support bearings due to the combined loads (rotor and gear) and to assess the effect of the deflections on the performance of the planetary gear/bearing system. Because of the complex nature of the loads, the development of a finite-element model (FEM) was required to determine the stress/deflection characteristics of the ring gear/rotor shaft support bearing system. A cross section of the bearing support/ring gear assembly is shown in Figure 81.

ANALYTICAL PROCEDURE AND FINITE-ELEMENT MODELING

Upon completion of the preliminary design analysis, two finite-element models were required in order to adequately develop an understanding of the interactions of all components and their resultant deflections and stresses. This information would be essential in the final design and development of the advanced-concept transmission.

The planetary ring gear was first analyzed using the NASTRAN finite-element analysis computer program. This finite-element model was designed to use a feature in NASTRAN Level 16.0 known as cyclic symmetry. The cyclic symmetry technique allows the NASTRAN user to model a small segment of an axisymmetric structure, with the remainder of the structure being mathematically simulated within the NASTRAN program. The benefits of using the cyclic symmetry approach are reduced modeling time and a reduction in computer run time. Figure 82 shows the single-segment model from which the entire ring gear model was simulated; Figure 83 shows the simulated full-configuration model. The model was analyzed for loading conditions, shown in Table 25, which were representative of operating conditions of the YUH-61A helicopter. For each loading

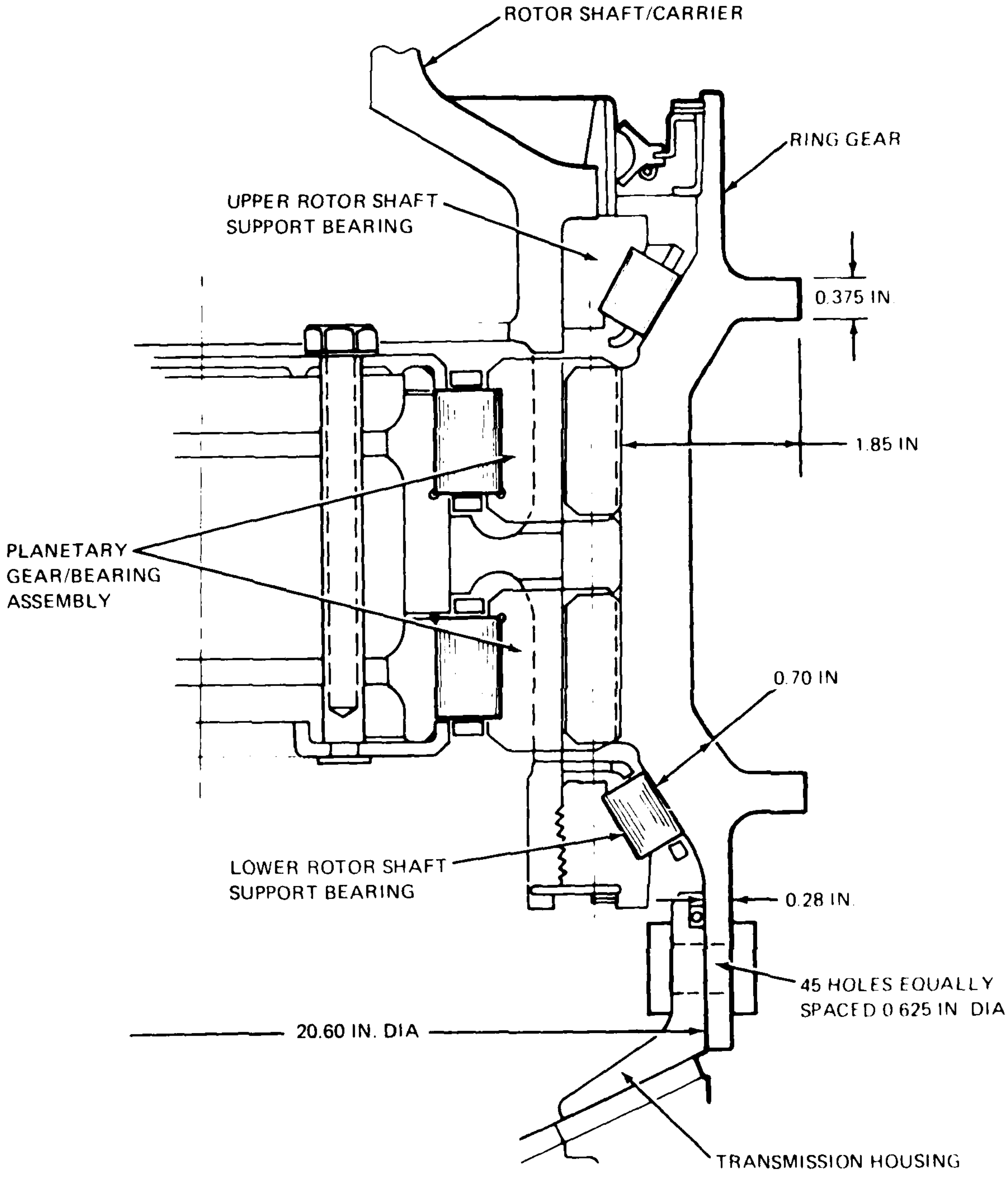


Figure 81. Rotor Shaft/Carrier Support Bearing and Planetary Ring Gear.

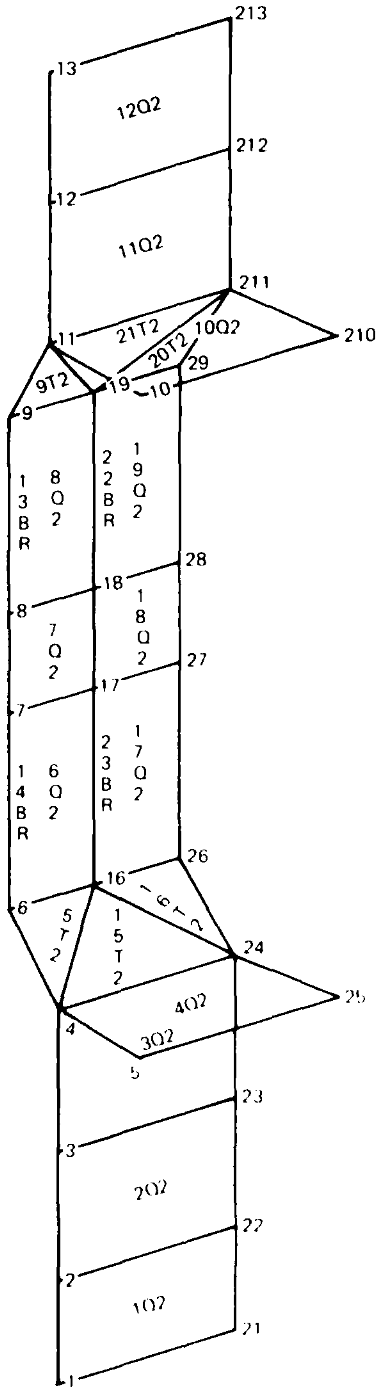


Figure 82. Single-Segment Cyclic-Symmetry Plot of Advanced-Concept Transmission Ring Gear.

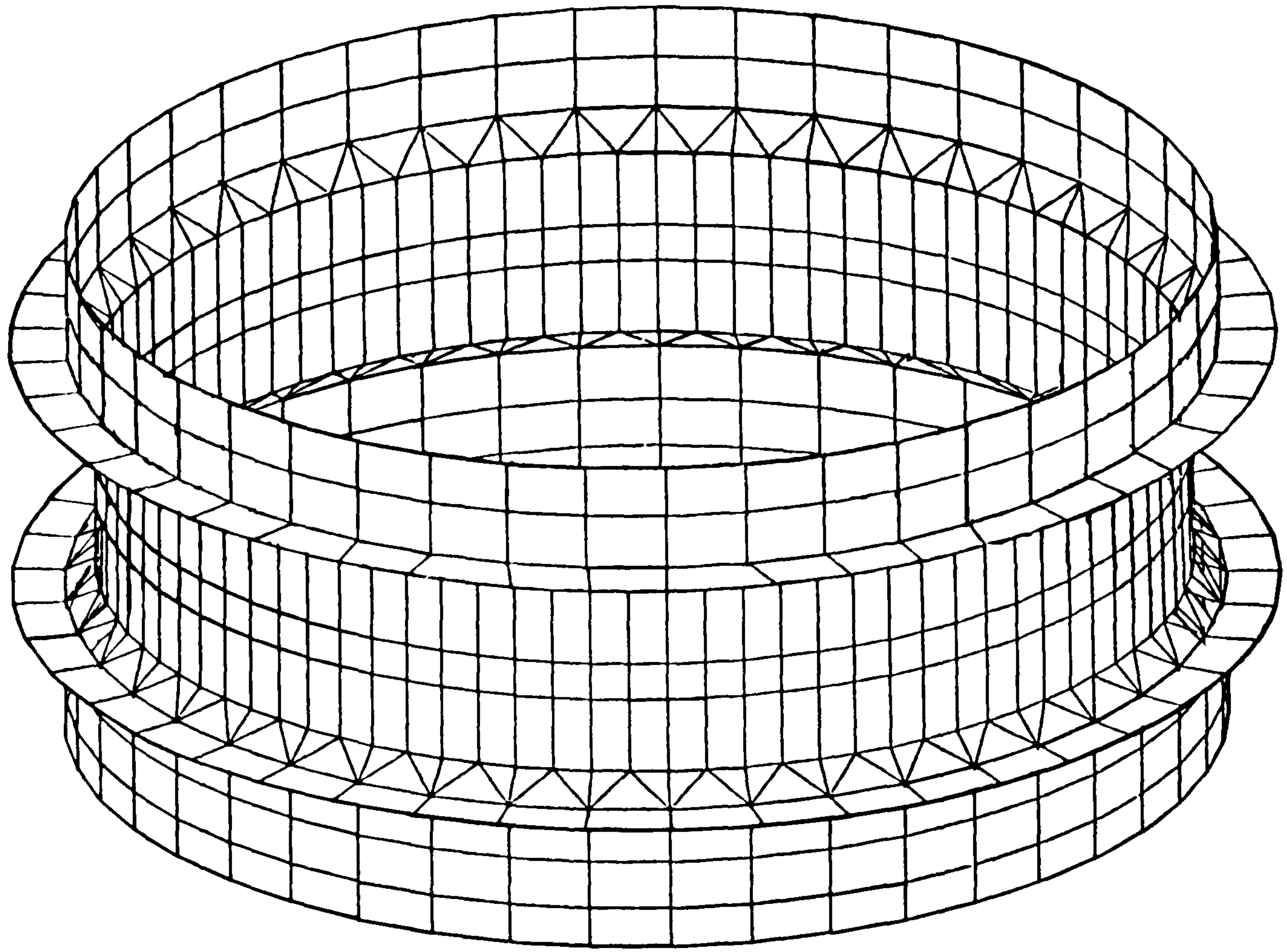


Figure 83. Finite-Element Model of Advanced-Concept Transmission Ring Gear.

condition, rotor loads were applied in the form of calculated rotor shaft bearing internal load distributions obtained from Boeing Vertol bearing computer programs. Rotor torque was applied in the form of planet gear contact loads at the planet gear azimuth locations.

Results from this preliminary computer run suggested that certain areas of the model would benefit from additional refinement. These refinements were aimed at selecting section thicknesses which would be efficient from a strength-to-weight standpoint, while maintaining the rigidity necessary for proper bearing support. These refinements also included modeling a more precise representation of the actual gear tooth configuration in order to assess the effect of the gear teeth on the bending stiffness of the ring gear wall.

Evaluation of results for the initial rotor shaft/carrier support bearing and ring gear NASTRAN finite-element analysis follows. A plot showing the final configuration of the finite-element model is shown in Figures 84 and 85. The ring gear/bearing support structure was analyzed for loadings representative of flight loading for both ultimate and fatigue conditions. Stress and deflection data was obtained for each element in the model structure. The results indicated that the bearing support/ring gear structure was deflection-critical rather than stress-critical. This was due to the rigidity requirements for the rotor shaft bearing races which were an integral part of the ring gear. Maximum deflections and stresses for the fatigue and ultimate conditions are shown in Figure 86. A sketch showing element and grid point locations is also included for reference.

The relative bearing race flexibility noted in the NASTRAN analysis led to efforts aimed at developing an accurate model of the rolling-element bearing load distribution behavior in the presence of bearing inner- and outer-race bending deformations. Conventional bearing computer analyses generally assume that the bearing races are rigid in bending and consider only contact deformations in their load distribution formulations. The initial NASTRAN finite-element analysis of the bearing support/ring gear was based on the bearing loads derived in this manner. This approach yields reasonable results provided there is adequate stiffness in the material surrounding the bearing races to prevent bending deflections of the race. However, in large-diameter bearings with relatively compliant outer and inner raceway backup material, bending deformation of the races can significantly influence the load distribution around the bearing. A change in the bearing rolling-element load distribution can affect bearing life and alter the state of stress in the supporting structure.

MODEL MODIFICATIONS

To investigate the effects of support structure compliance on bearing load distribution by the finite-element method, the analysis must consider both contact and structural stiffnesses. Structural stiffness is inherent in the finite-element mesh of the support structure. However, it was necessary to develop a somewhat simplified finite-element model of the rolling-element/raceway contact deformation as a function of load. Modeling this behavior is further complicated by the nonlinear dependence of contact deformation on load and the fact that bearing elements are capable of acting only in compression. Tension loads cannot be reacted at a rolling-element node point. To accomplish this feature in the rotor shaft bearing support analysis, the contact stiffness of each bearing roller was approximated by an assemblage of spring-gap finite elements connected in parallel as shown in Figure 87. With proper selection of spring stiffness and gap length on each element, a close approximation of the bearing roller raceway stiffness curve can be obtained from the finite-element assemblage. Figure 88 shows the calculated load-deflection curve for a bearing roller and the piecewise linear approximation of this curve obtained from the spring-gap finite-element assemblage. These spring-gap elements are available in the ANSYS program, a commercially available general purpose finite-element program.

To demonstrate the applicability of this approach for calculating bearing stiffness, an analysis of a radial roller bearing was performed. This analysis considered contact deformations only and therefore neglected race

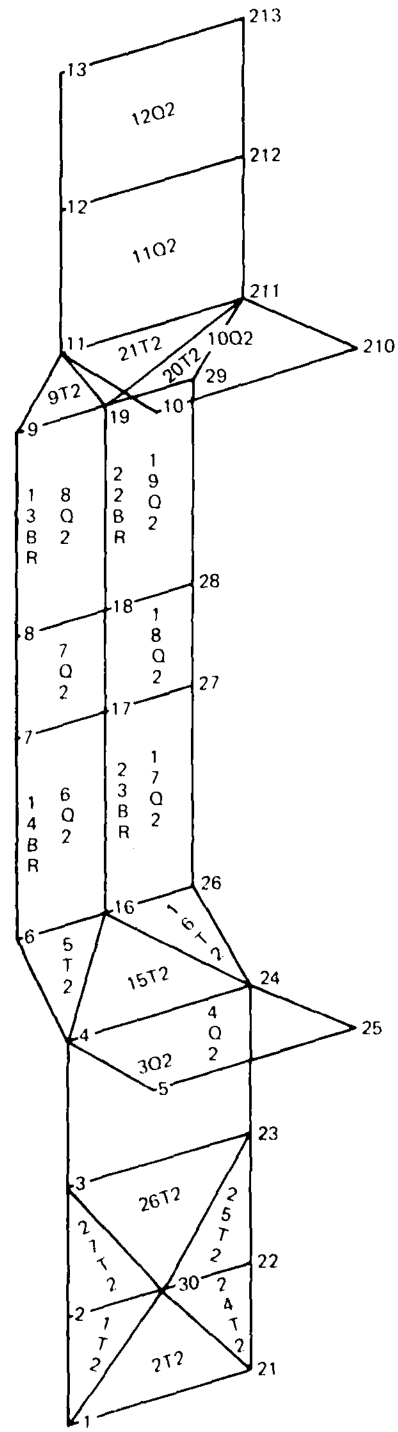


Figure 84. Final Finite-Element Model of Single Segment of Advanced-Concept Transmission Ring Gear.

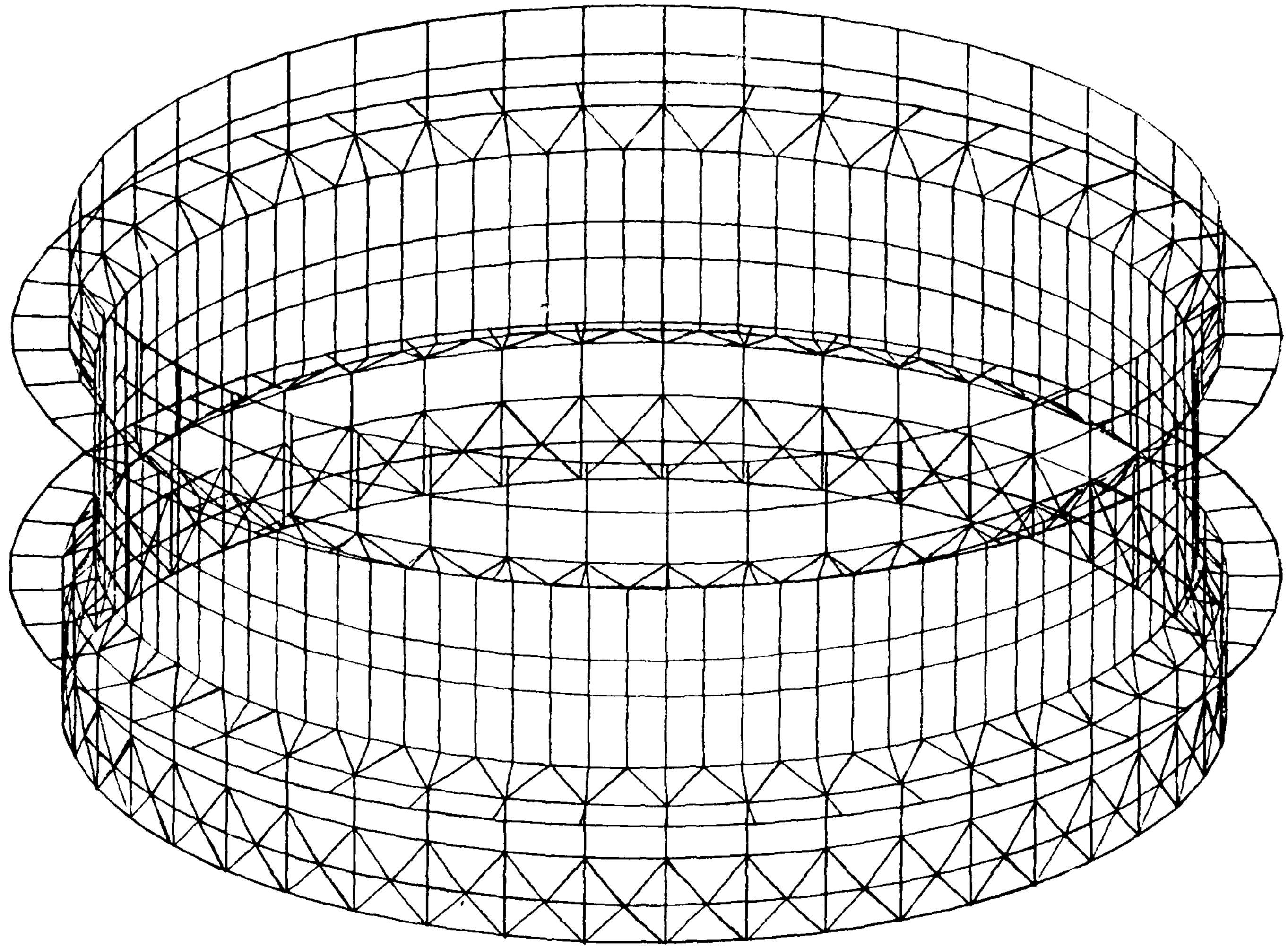
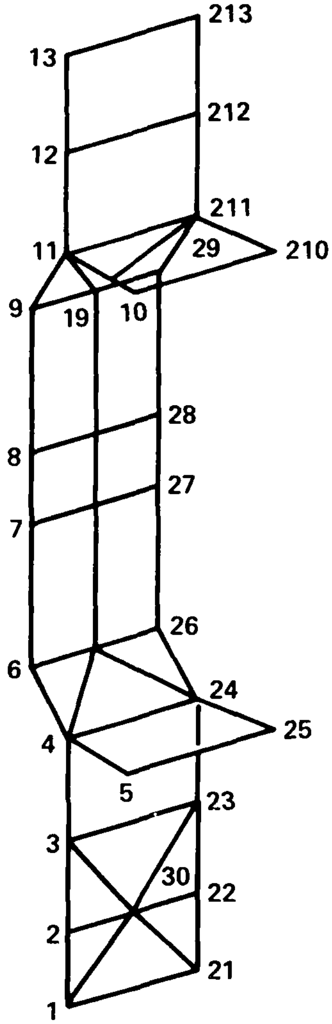
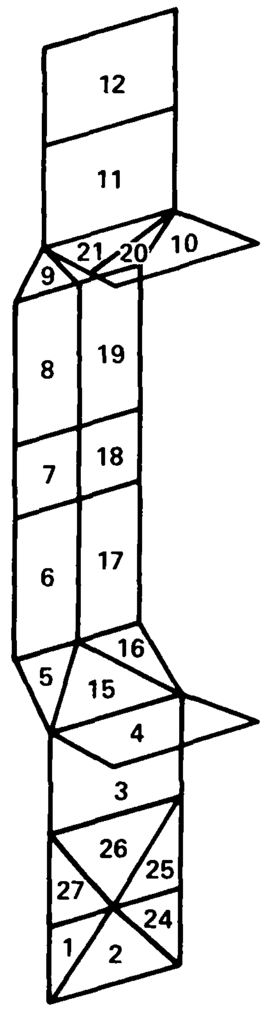


Figure 85. Final Finite-Element Model of Advanced-Concept Transmission Ring Gear.



GRID POINT LOCATIONS



ELEMENT LOCATIONS

LOADING CONDITION	MAXIMUM RADIAL DISPLACEMENT		MAXIMUM STRESS	
	DISPLACEMENT (IN.)	GRID POINT NUMBER	NORMAL STRESS (PSI)	ELEMENT NUMBER
ULTIMATE	0.0171	11	36,990	6
FATIGUE	0.0067	11	20,890	7

Figure 86. Element and Grid Point Locations on Single Segment of Ring Gear.

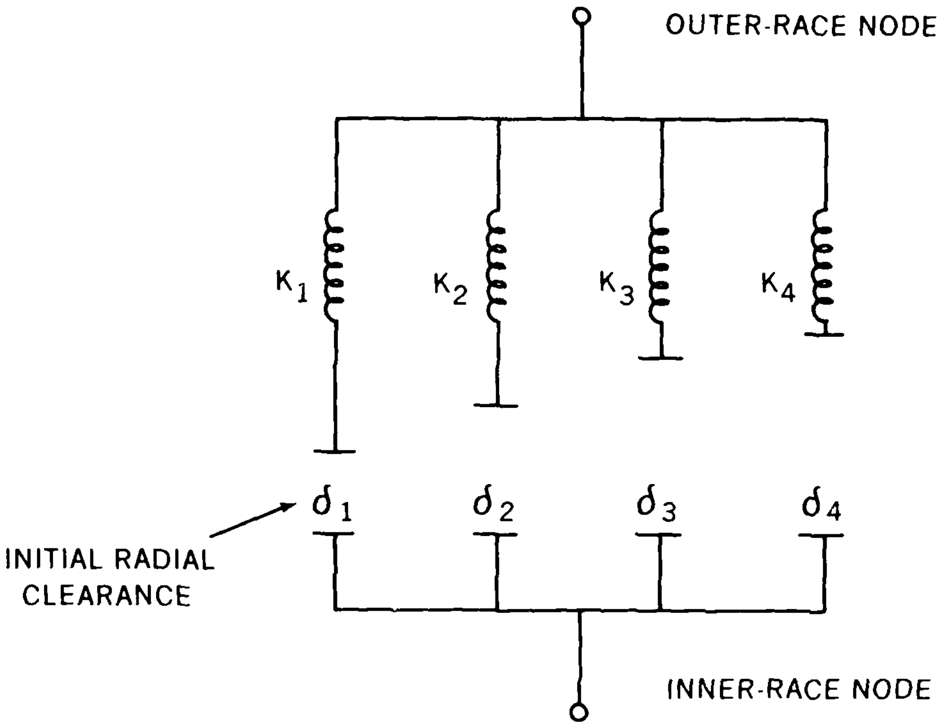


Figure 87. Model of a Bearing Rolling Element.

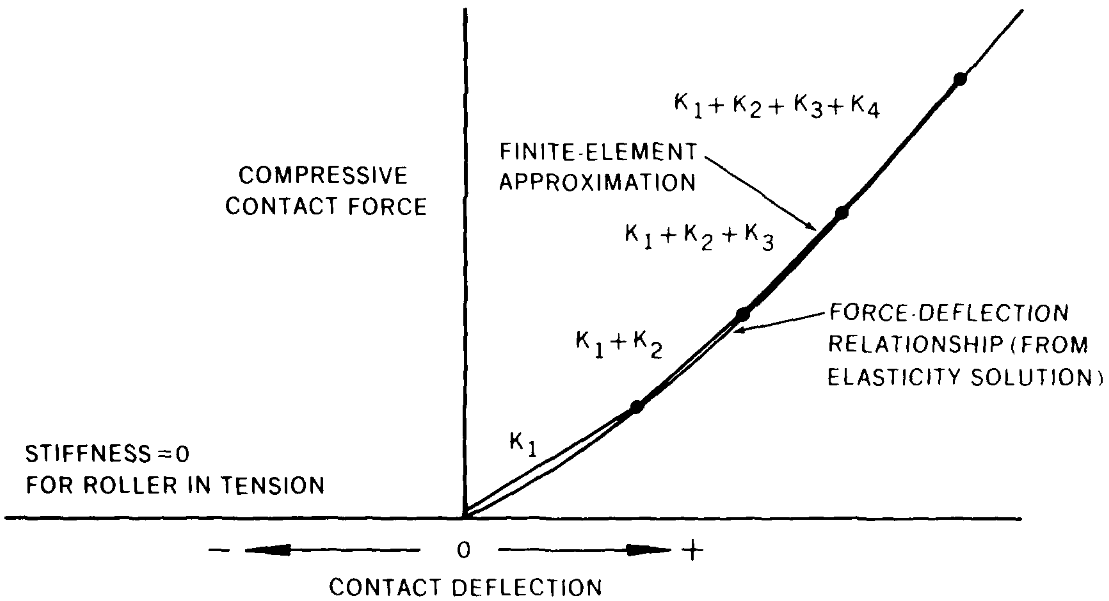


Figure 88. Approximation of the Stiffness Characteristics of a Bearing Rolling Element.

bending stiffness. The contact stiffness of each roller was modeled with an assemblage of four spring-gap elements as shown in Figure 87. Note that the spring stiffness does not become effective until the connecting nodes have displaced a distance toward each other equal to the specified gap distance. This model also has provisions to account for initial radial clearances in a bearing. Due to the stepwise engagement of the springs, an iterative solution is necessary to solve for an answer.

A sketch showing the roller bearing model is shown in Figure 89. Notice that the races do not deform, although due to contact deflections they displace relative to each other. Table 26 contains a comparison of contact load and contact deflection computed from the finite-element analysis and from a conventional bearing analysis computer program which calculates rigid-race contact forces and deflections using conventional techniques.

The results of the radial bearing analysis agree closely with contact loads and deformations calculated by the bearing computer program. This indicates that the spring-gap assembly is capable of modeling the contact behavior of rolling-element bearings to a degree of accuracy equal to that of more conventional rigid-race techniques.

TABLE 26. BEARING LOAD AND CONTACT DEFLECTION COMPARISON OF ANALYTICAL METHODS

Roller Contact Force Comparison			
Roller Location	Finite-Element Method (lb)	Bearing Program (lb)	Difference (%)
1	5,119	5,108	0.2
2	4,841	4,830	0.2
3	4,033	4,032	0.02
4	2,800	2,826	0.9
5	1,385	1,388	0.02

Contact Deflection Comparison			
Roller Location	Finite-Element Method (in.)	Bearing Program (in.)	Difference (%)
1	3.93×10^{-3}	3.93×10^{-3}	0
2	3.74×10^{-3}	3.74×10^{-3}	0
3	3.18×10^{-3}	3.18×10^{-3}	0
4	2.31×10^{-3}	2.31×10^{-3}	0
5	1.21×10^{-3}	1.21×10^{-3}	0

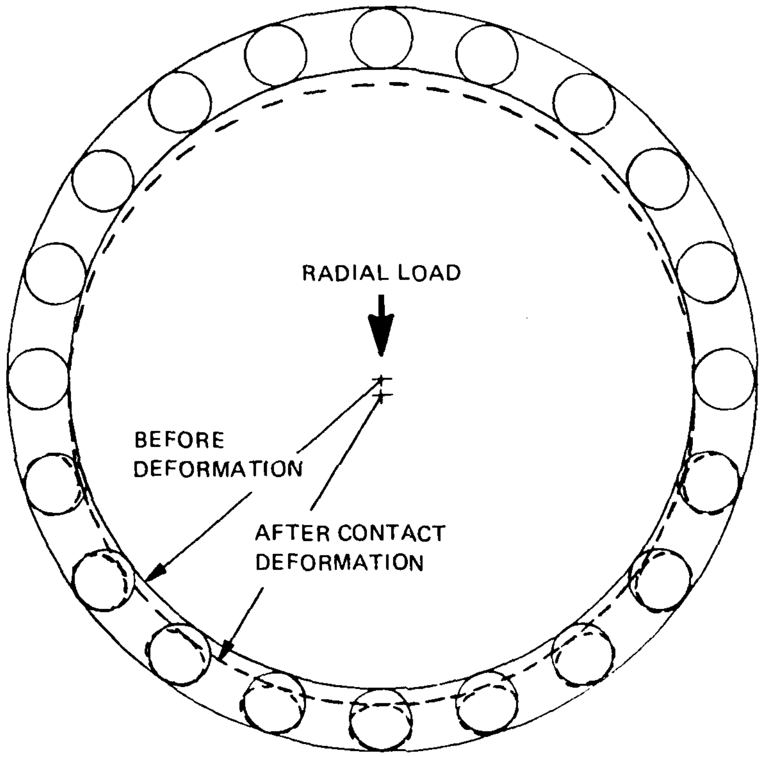


Figure 89. Finite-Element Model of a Roller Bearing.

FINITE-ELEMENT MODEL OF BEARING SUPPORT ASSEMBLY

Having verified an acceptable model for bearing roller/race contact stiffness, the rotor shaft/planet carrier and stationary ring gear assembly were modeled using the ANSYS finite-element program. The bearing support structure consisting of the stationary ring gear and rotor shaft/planet carrier components was modeled with quadrilateral plate elements. These elements are capable of both bending and in-plane deformation. The individual bearing rollers were modeled with an assemblage of spring-gap interface elements. The stiffness of the spring-gap element assemblages was chosen such that it closely approximated the nonlinear contact stiffness of the bearing rollers sized for this application. The rotor shaft bearing/support assembly model was then used to evaluate the distribution of load on the rotor shaft support bearing rollers. The internal roller load distribution calculated in this manner incorporated the effects of raceway/support structure compliance on the overall bearing load distribution.

To facilitate an efficient computer solution, a half-symmetry model was employed. This technique requires that only one half of a symmetrical structure need be modeled when the appropriate boundary conditions are input for the plane of symmetry. This feature considerably reduced computing time, especially since an iterative solution was required. A computer plot of the complete finite-element model is contained in Figure 90. The model geometry may be further clarified by referring to Figure 91, which shows a conceptual view of the finite-element idealization of a cross-section of this model. A sketch of the actual cross-sectional view of the rotor shaft bearing/bearing support structure is contained in Figure 92.

SUMMARY OF RESULTS

The finite-element model was analyzed for the loading conditions shown in Figure 93. These loads, applied at the rotor hub mounting flange, are representative of thrust and overturning moment ultimate condition loads of the YUH-61A main rotor. Calculated values of individual bearing roller loads and relative raceway displacement were output by the analysis. Roller loads and relative race displacements as a function of roller location azimuth are tabulated in Table 27 for both the upper and lower rotor shaft bearings. Negative relative displacement values indicate that the roller is in contact when at this azimuth location and conversely a positive relative displacement indicates that a gap exists between roller and race at that azimuth position, hence a value of zero load.

To ascertain the effect of raceway flexibility on bearing internal load distribution, the rigid-race internal load distribution was calculated by conventional analytical techniques. The rigid-race distribution along with the flexible-race distribution are plotted versus azimuth location and are shown in Figure 94. Notice that both the rigid- and flexible-race internal load distributions for the lower bearings are similar in shape, with the flexible-race loads slightly larger in magnitude than the rigid-race loads. The upper bearing internal load distributions show quite pronounced differences, however. The rigid-race internal load distribution is of greater maximum magnitude and is spread over a shorter arc than the calculated flexible-race bearing internal load distribution. From these results it appears as though the flexibility, or compliance, of the bearing raceways and supporting structure tend to allow the total load to distribute over more rollers, effectively reducing the maximum loading any one roller would experience.

The results of the flexible-race analysis can be used to modify the bearing internal load distribution and obtain an estimate of the B-10 life of the bearing. Using a conventional bearing analysis program and the modified internal load distribution showed that the life of the lower bearing decreased by approximately 13 percent while the life of the upper bearing increased by more than 50 percent. Additional work would be required to

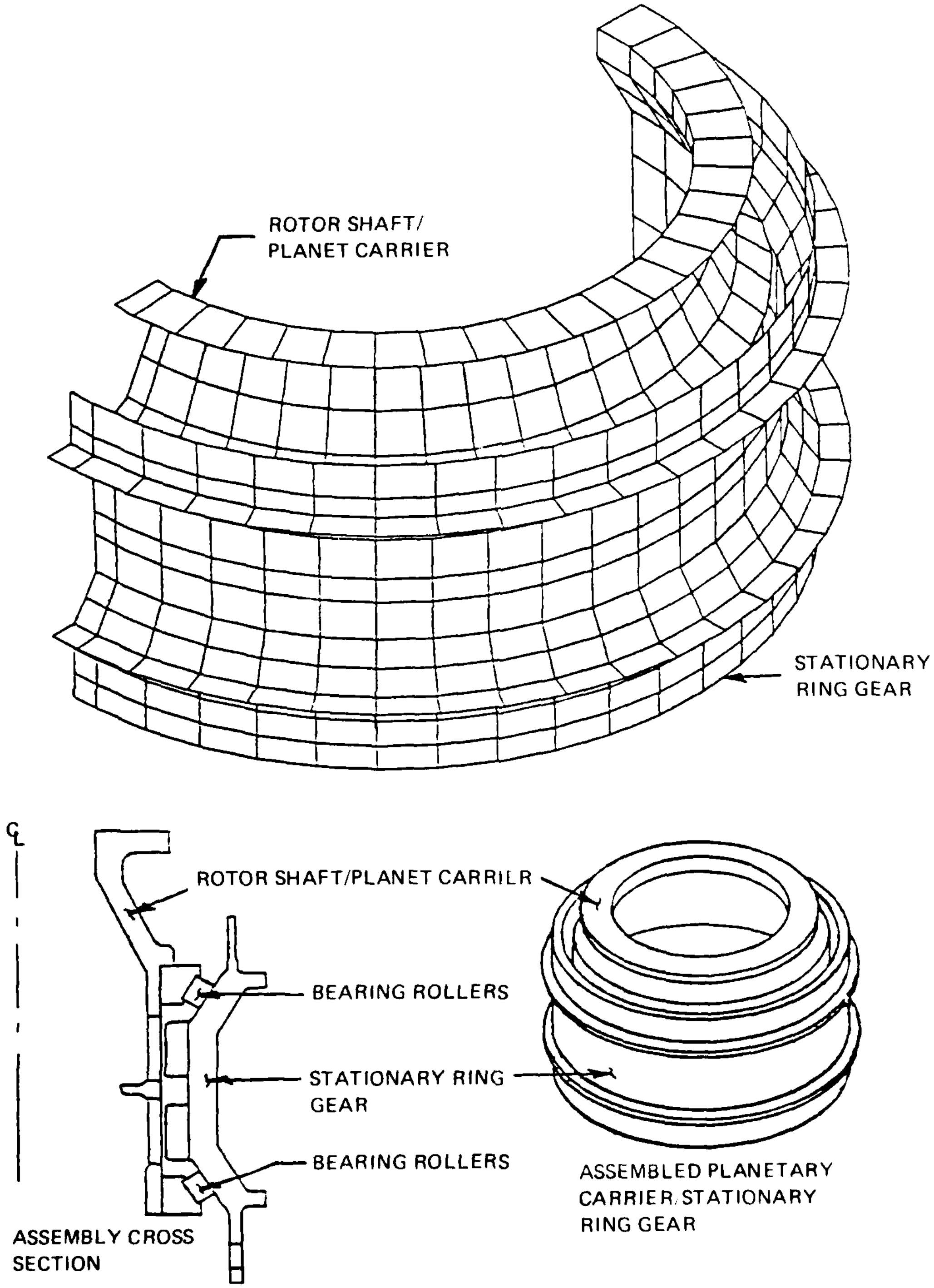


Figure 90. Finite-Element Model of a Planetary Carrier/Stationary Ring Gear.

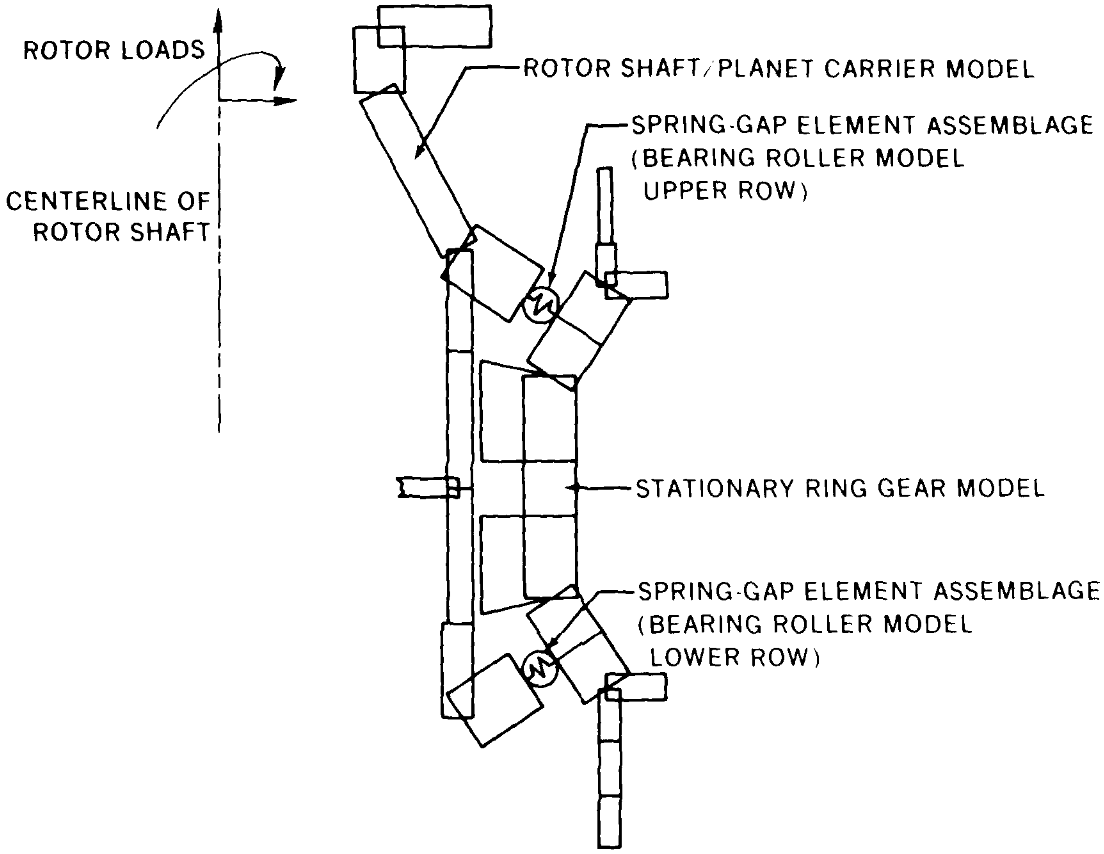


Figure 91. Finite-Element Idealization of Rotor Shaft Bearing and Bearing Support Structure.

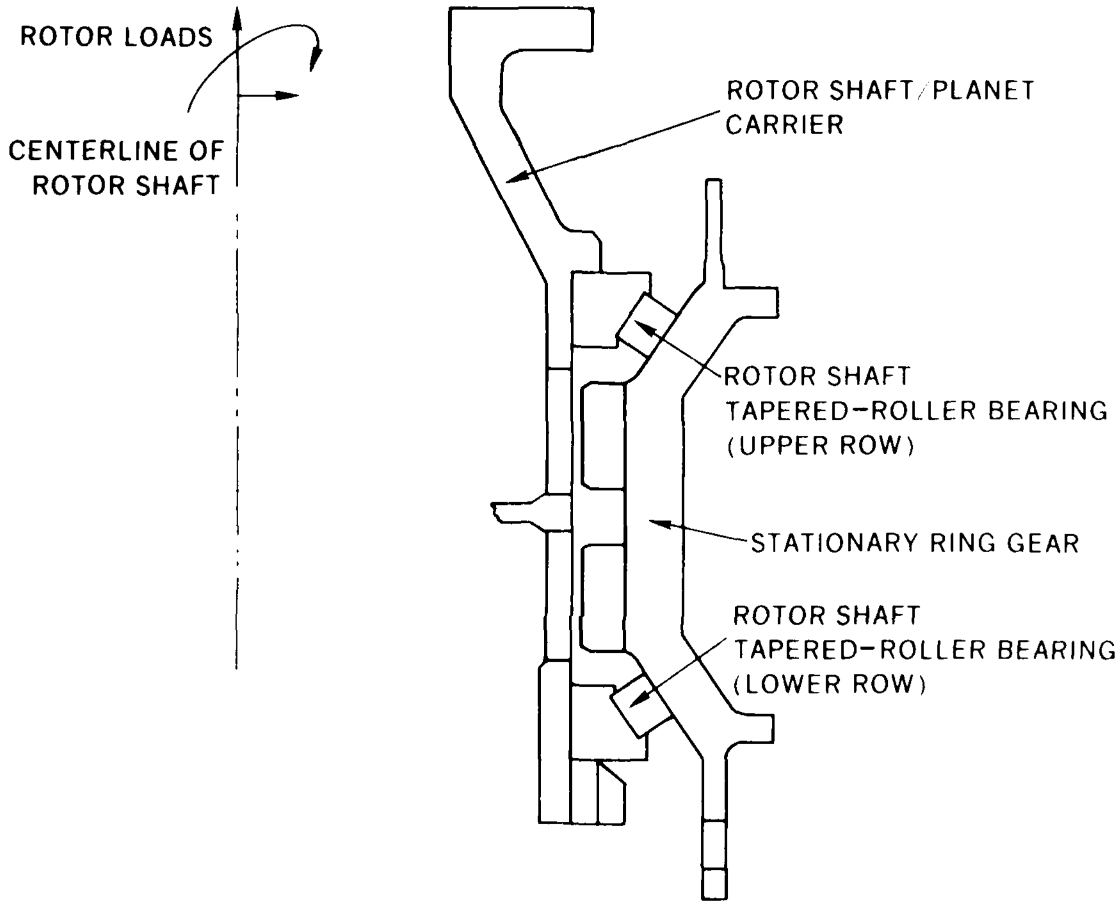


Figure 92. Cross-Sectional View of Rotor Shaft Bearing and Bearing Support Structure.

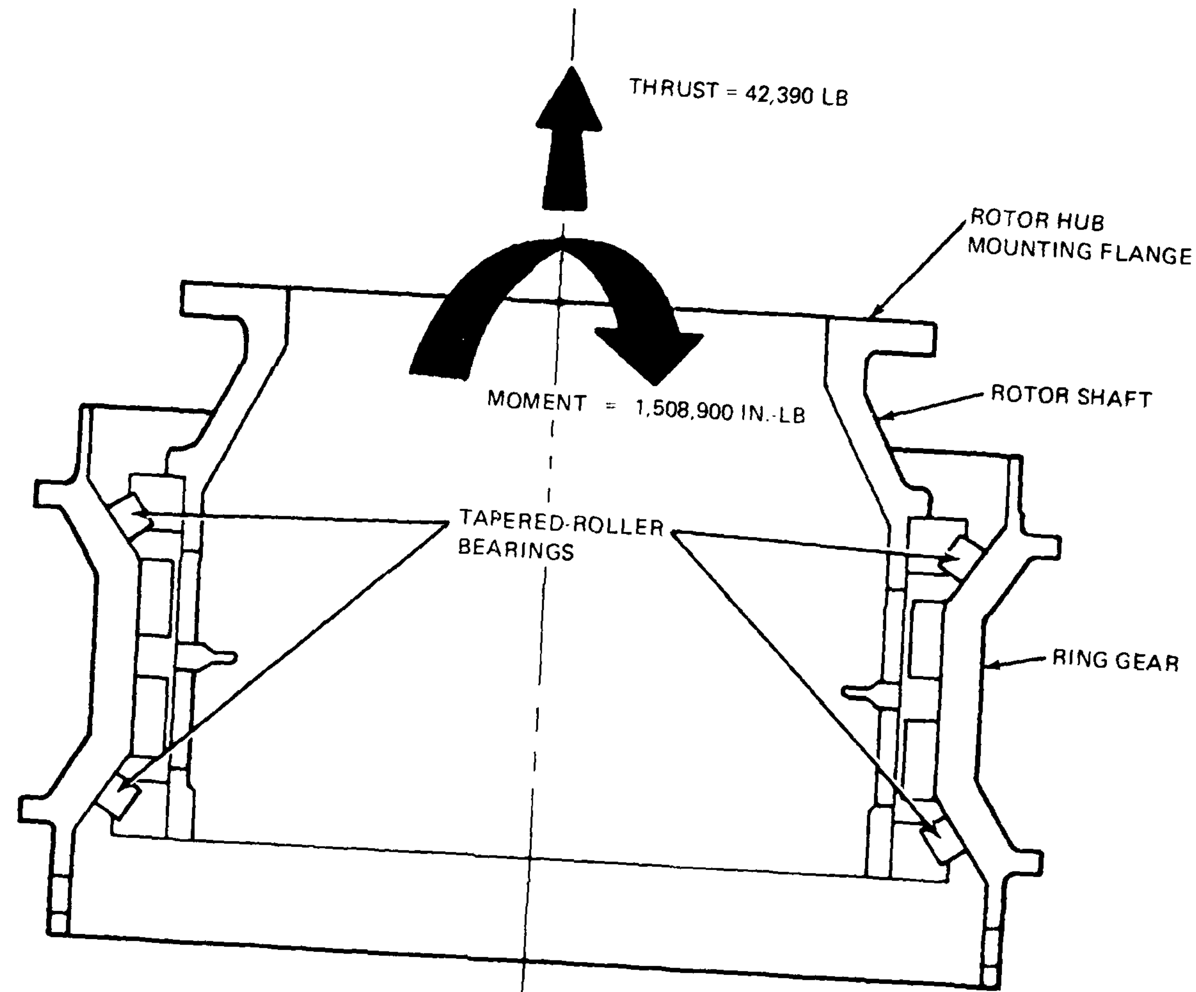


Figure 93. Cross-Sectional View of Ring Gear and Rotor Shaft Bearing Support Showing Equivalent Ultimate Loads Used in the Finite-Element Analysis.

TABLE 27. CALCULATED ROLLER LOADS AND RELATIVE RACE DEFLECTIONS AS A FUNCTION OF AZIMUTH POSITION

Azimuth Location (deg)	Upper Bearing Element Force (lb)	Upper Bearing Relative Displacement (in.)	Lower Bearing Element Force (lb)	Lower Bearing Relative Displacement (in.)	Azimuth Location (deg)	Upper Bearing Element Force (lb)	Upper Bearing Relative Displacement (in.)	Lower Bearing Element Force (lb)	Lower Bearing Relative Displacement (in.)
0	0	+ 0.0488	4,650	0.0034	96	923	0.0008	2,290	0.0018
8	0	+ 0.0481	4,783	0.0035	104	2,172	- 0.0017	1,919	0.0015
16	0	+ 0.0460	4,725	- 0.0034	112	2,783	0.0021	1,565	0.0013
24	0	+ 0.0427	4,615	- 0.0034	120	3,141	0.0024	1,220	0.0010
32	0	+ 0.0382	4,467	0.0033	128	3,420	0.0026	921	0.0008
40	0	+ 0.0329	4,290	- 0.0032	136	3,658	- 0.0027	664	0.0006
48	0	+ 0.0271	4,087	- 0.0030	144	3,858	- 0.0029	440	0.0004
56	0	+ 0.0209	3,858	- 0.0029	152	4,016	0.0030	252	0.00025
64	0	+ 0.0148	3,597	0.0027	160	4,126	- 0.0030	114	0.0001
72	0	+ 0.0093	3,306	0.0025	168	4,194	- 0.0031	7	0.000005
80	0	+ 0.0046	2,984	0.0023	176	4,228	- 0.0031	0	+ 0.000046
88	0	+ 0.0012	2,640	0.0020					

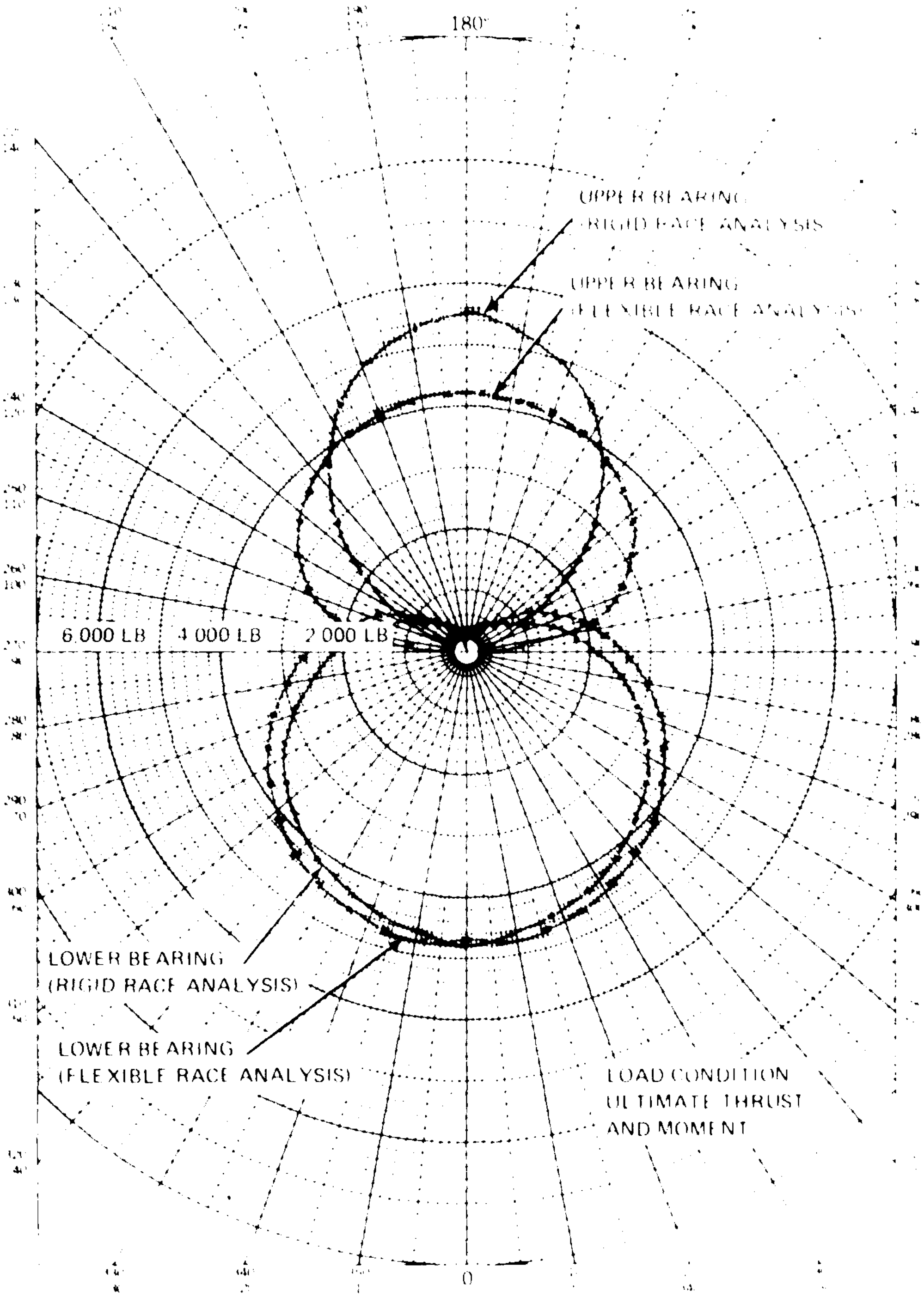


Figure 94 Rotor Shaft Support Bearing Internal Load Distribution

optimize this type of bearing system for life. Changes in the ring gear stiffness may be required to achieve an internal load distribution that will result in the maximum bearing fatigue life for the lowest weight system.

CONCLUSIONS

Based upon the results of this preliminary analysis of a complex bearing and bearing support structure using finite element analysis, the following conclusions resulted:

1. The advanced transmission ring gear rotor shaft bearing support structure was analyzed using the finite-element method and found to be deflection-critical.

2. A technique was devised and verified to employ a general-purpose finite-element program to model the contact behavior of rolling-element bearings.

3. This technique was used in conjunction with the structural modeling capabilities of the general-purpose finite element computer program to calculate the bearing internal load distribution for the advanced transmission rotor shaft bearings. This calculated distribution reflected the effects of the bearing race and support flexibility as well as the contact deflections of the bearing elements themselves.

4. Bearing fatigue life was calculated for the flexible system and was greatly influenced by the flexibility of its support structure.

5. The cited technique can be used to optimize the bearing system life for a minimum-weight support structure.

PROGRAM CONCLUSIONS

The overall objectives of the work conducted under this program were achieved. The elements of the bearing and seal development program: high-speed ribbed-cup tapered-roller bearings; magnetic seal; high-hot-hardness VASCO-X2 steel inner races; and improved methods of analysis of complex bearing structures can be combined to provide an advanced-concept drive system with significant advantages to the operator. The advantages will result in lighter weight, less complexity and cost, fewer components and faying surfaces, and potentially longer component life for an advanced drive system gearbox.

To illustrate the application of the advanced-component technology developed during this program, a design study was conducted by Boeing Vertol using the CH-47 helicopter engine transmission as a baseline specimen. The transmission was redesigned to incorporate the ribbed-cup tapered-roller bearings, magnetic seal, high-hot-hardness carburizing steel, and a composite housing which is being developed under a separate work task of this contract. The results of this design study are shown in Figure 95 which reveals that four ribbed-cup tapered-roller bearings support the input pinion and output gear, three of which operate with integral inner races, and that these replace six ball and cylindrical roller bearings in the conventional design. In addition, a reduction in the number of bearing-associated hardware such as locknuts, retainers, and lubrication rings further simplifies the new assembly and reduces fabrication costs. The calculated bearing fatigue life is increased and the spring rates of the tapered-roller bearings are also increased compared to conventional, which should decrease gear deflections under load.

The sum of the improvements of the advanced-concept transmission is shown in Table 28. This table compares the salient features of a current design and the same design incorporating the advanced-technology components in terms of weight, parts count, and design life. The only feature included in this summary which is not discussed in this report is the weight reduction due to an advanced-composite housing. The evaluation of the composite housing will be discussed in a follow-on report. The reduced weight configuration of the composite housing was 5 pounds. One of the significant features of this design is the reduction in the number of bearings, which has a direct impact on oil flow and cooling requirements. The improvements shown in Table 28 meet or exceed all of the design goals initially established for this program.

Based on this comparison, it is apparent that the design goals established for each component and for a total transmission design can be achieved through the use of the component technology developed during this program. No significant problems were experienced during this program which could affect the direct use of these components in an advanced-concept transmission which would enter the design phase in the 1980's.

TABLE 28. ADVANTAGES OF ADVANCED COMPONENT ASSEMBLY

	Baseline	Advanced	Improvement (%)
Weight (lb)	125	100.5	20
Main Bearings	6	4	33
Major Components	28	16	28
Faying Surfaces	18	9	50
B-10 Life (hr)	972	1,600	+ 65
Oil Flow (l/min)	22.7	18.0	20

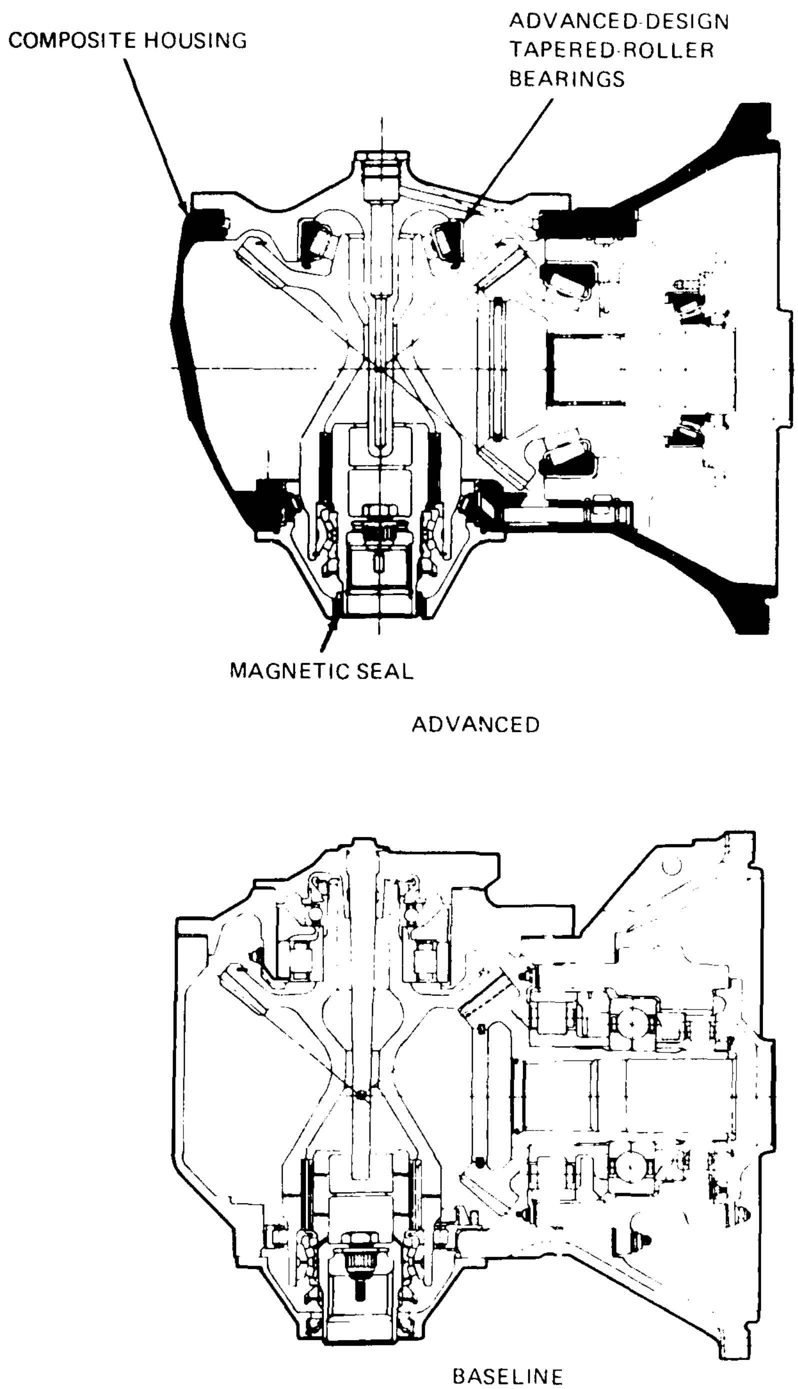


Figure 95. The Advanced Gearbox Assembly Is Lighter, More Reliable, and Quieter.

RECOMMENDATIONS

Although a significant amount of rig testing was conducted during this program, testing of each component in a complete transmission assembly was not planned or accomplished. It is recommended that the component technology developed under this program be incorporated into an existing helicopter transmission system and tested. One such proposed system was shown in Figure 95. This continuation of testing would permit a direct comparison of performance and life of the existing system versus the system modified with advanced-technology components. Testing would initially be conducted in a test rig which would eventually lead to a flight-test program. This type of program would provide additional confidence and verification of the advanced concept before its incorporation into a future advanced-concept helicopter drive system.

In addition, the oil-off survivability tests conducted on the ribbed-cup tapered-roller bearing design did not achieve the military goal of 30 minutes of operation at gearbox torque limit without lubrication. These tests did provide information concerning areas of additional testing and design modifications which could result in obtaining this goal. Therefore it is recommended that additional work be conducted on advanced material development for the cage and cup rib and modifications to the bearing geometry and cage design in order to extend or achieve the 30-minute oil-off requirement. The final evaluation of the optimum design should be performed in an environment and mounting arrangement similar to those of an actual helicopter transmission.

The marked advantages that can accrue from component development are evident in the results of this program. Continuing programs leading to integrated assembly and testing of improved components will provide these advantages for drive systems of the 1980's. Results to date indicate that the continuation of these component developments will provide significant and fruitful results.

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APPENDIX A

REFERENCE TEST DATA FOR AN MRC 207S BEARING IN A MODEL A TEST MACHINE

The test data and Weibull plots presented in Appendix A were obtained by MRC on a previous test program using the Model A test machines. This reference test data for an MRC 207S ball bearing fabricated from 52100 steel can be used to compare the results of the slave bearings. The slave bearings used in this program were obtained from the original three lots of bearings and were tested under the same conditions except for lubricant and operating temperatures.

TABLE A-1. MRC RESEARCH PROJECT NO. 1610, LOT NO. 1

(FORGED 52100 STEEL RINGS)
SPEED - 5500 RPM
LOAD - 1900# RADIAL
LUBE - SAE # 10

BEARING NO.	HOURS	REV. (MILLIONS)	REMARKS
8	150.8	49.7	Spalled Ball
11	435.8	143.8	Spalled Outer
2	460.0	151.8	Spalled Inner
22	525.3	173.3	Spalled Ball
30	741.8	244.8	Spalled Inner
16	1623.5	535.8	Spalled Inner
25	1695.9	---	Suspended, turned on arbor
26	1695.9	---	Suspended, turned on arbor
36	1723.7	568.8	Spalled Inner
10	2148.8	---	Suspended
33	2211.0	---	Suspended, turned on arbor
18	2782.5	918.2	Spalled Inner
5	2789.2	---	Suspended
21	2789.2	---	Suspended, turned on arbor
32	2899.4	956.8	Spalled Inner
28	2899.4	---	Suspended
31	2899.4	---	Suspended
3	3218.3	---	Suspended
7	3218.3	---	Suspended
4	3733.6	---	Suspended
6	3733.6	---	Suspended
15	3733.6	---	Suspended

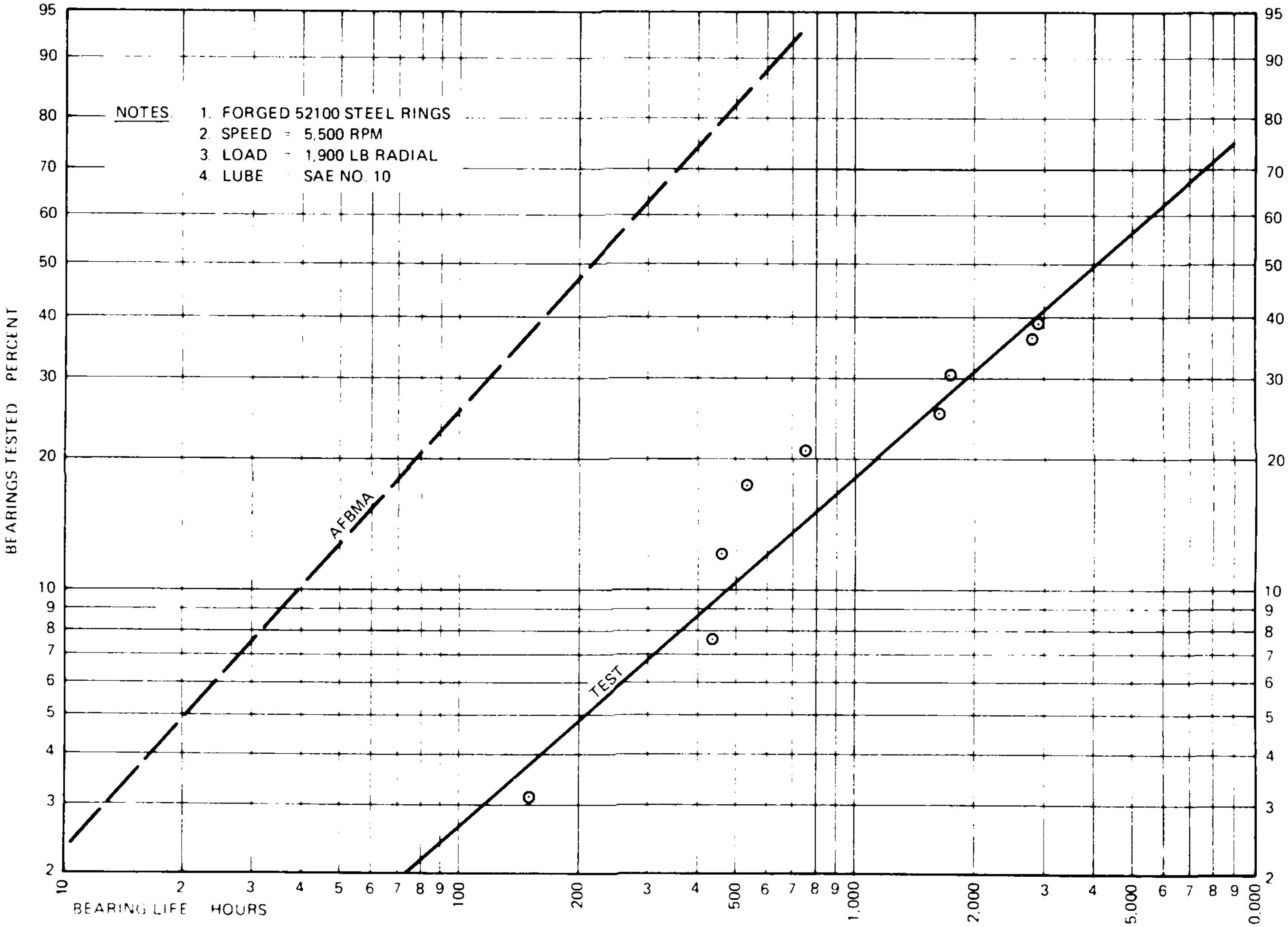


Figure A–1. Test of MRC 207S Bearings Lot No. 1.

TABLE A-2. MRC RESEARCH PROJECT NO. 1610, LOT NO. 2

(FORGED 52100 STEEL RINGS)
SPEED - 5500 RPM
LOAD - 1900 # RADIAL
LUBE - SAE # 10

BEARING NO.	HOURS	REV. (MILLIONS)	REMARKS
9	377.8	124.7	Spalled Ball
14	454.4	150.0	Spalled Inner
7	493.3	162.8	Spalled Outer
5	787.8	---	Suspended, lube failure
21	1015.8	---	Suspended, lube failure
23	1547.2	510.6	Spalled Inner
30	1821.3	601.0	Spalled Ball
8	1885.2	---	Suspended, lube failure
35	2278.1	751.8	Spalled Inner
16	2637.6	---	Suspended
15	2671.7	881.7	Spalled Ball
17	2891.4	954.2	Spalled Ball
24	2891.4	---	Suspended
33	2891.4	---	Suspended
32	2924.2	---	Suspended
3	3089.6	---	Suspended
18	3089.6	---	Suspended
1	3130.9	---	Suspended
4	3130.9	---	Suspended
34	3232.3	1066.7	Spalled Outer
25	3232.3	---	Suspended
11	3273.6	---	Suspended

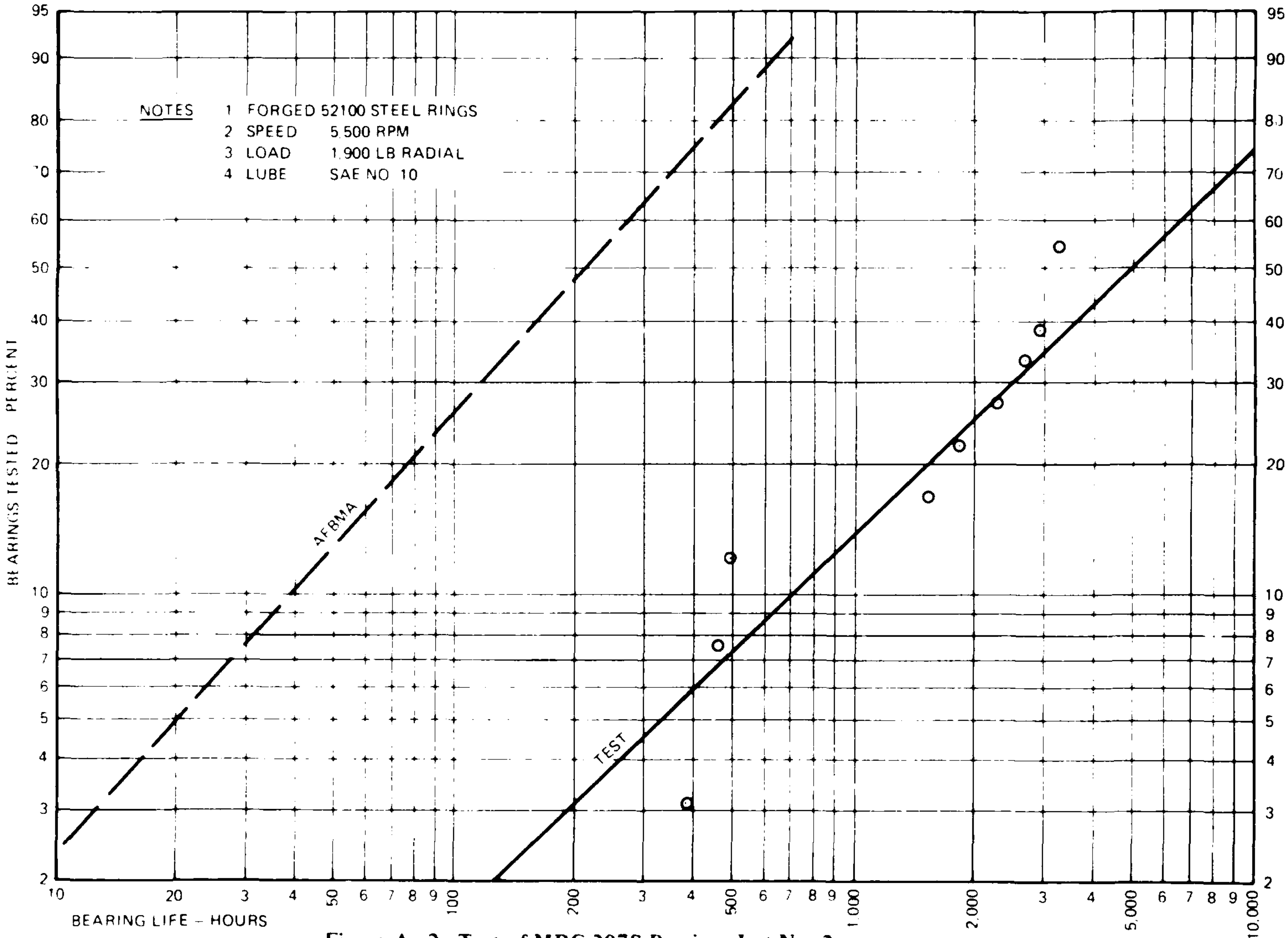


Figure A-2. Test of MRC 207S Bearings Lot No. 2.

TABLE A-3. MRC RESEARCH PROJECT NO. 1610, LOT NO. 3

(FORGED 52100 STEEL RINGS)
SPEED - 5500 RPM
LOAD - 1900 # RADIAL
LUBE - SAE # 10

BEARING NO.	HOURS	REV. (MILLIONS)	REMARKS
11	71.2	23.5	Spalled Outer
30	424.0	139.9	Spalled Outer
7	470.0	155.1	Spalled Inner
24	829.9	273.9	Spalled Inner
20	1258.5	---	Suspended
32	1722.6	568.5	Spalled Inner
12	2358.4	---	Suspended, overheated
5	2609.4	861.1	Spalled Ball
9	2609.4	861.1	Spalled Ball
3	2609.4	---	Suspended
4	2609.4	---	Suspended
18	2637.0	---	Suspended
25	2637.0	---	Suspended
1	2741.0	---	Suspended
2	2741.0	---	Suspended
23	2741.0	---	Suspended
26	2741.0	---	Suspended
19	2891.4	---	Suspended
14	2955.1	---	Suspended
15	3609.0	---	Suspended

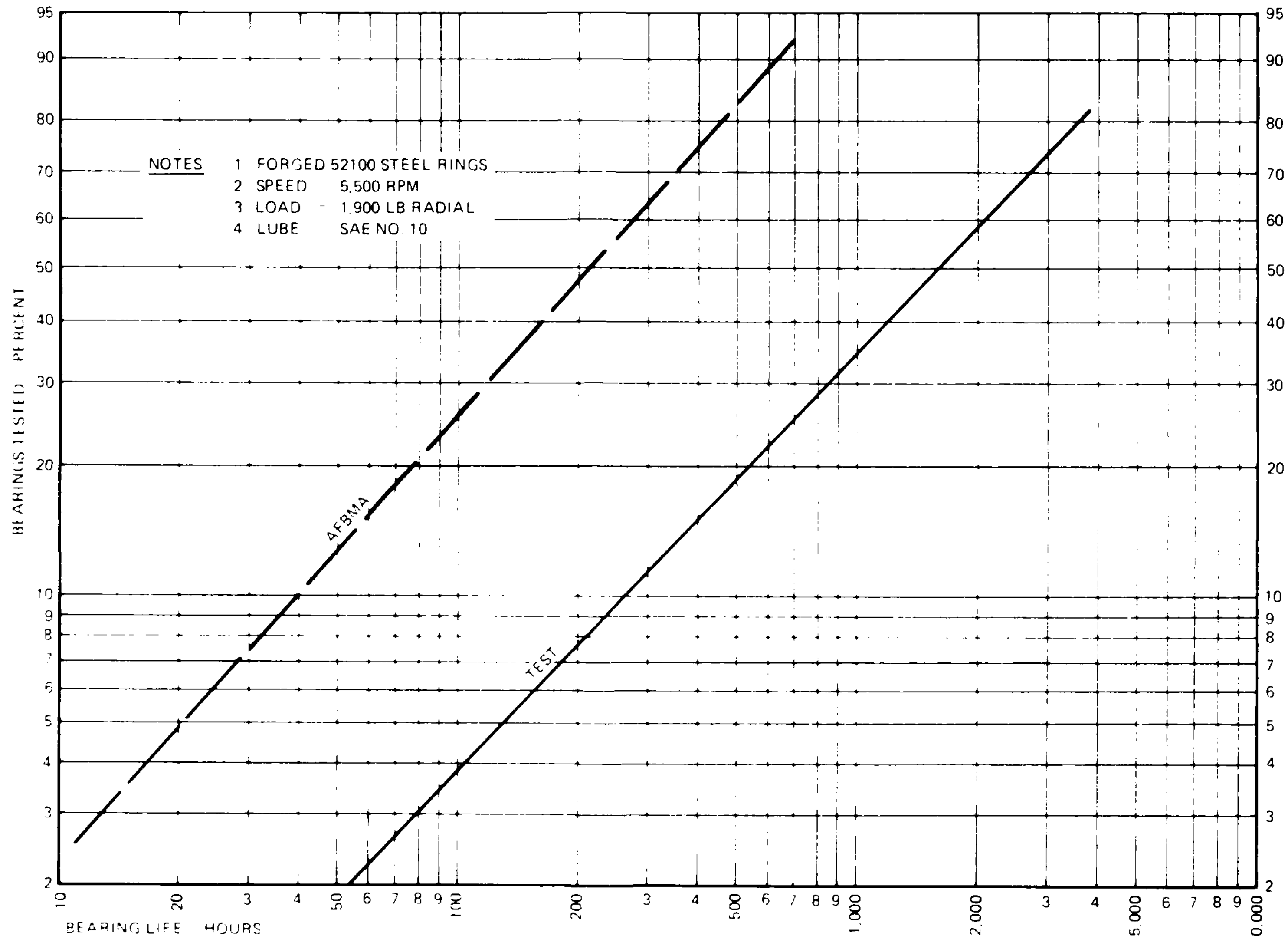


Figure A 3. Test of MRC 207S Bearings Lot No. 3.

APPENDIX B

RIBBED-CUP TAPERED-ROLLER BEARING AND MAGNETIC SEAL TEST DATA

Appendix B contains the ribbed-cup tapered-roller bearing and magnetic seal test data. Included are the build-up sheets for each test, test data recorded at each data point, and photographs of the condition of the bearings after completion of test.

TABLE B-1. TEST NO. 141.1-U - BUILDUP SHEET, SETUP NO. 1

TEST BEARINGS

	DRIVE END	OPPOSITE DRIVE END
SHAFT SERIAL NO.	78-8	78-8
CAGE SERIAL NO.	78-24	78-16
CUP SERIAL NO.	78-21	79-1
ROLLER SIZE	7	7 4
CUP O.D.	5.0005	5.0003
HOUSING I.D.	4.9989	5.0038
CUP/HOUSING FIT	.0016 TIGHT	.0035 LOOSE

	DRIVE END LARGE END	DRIVE END SMALL END	OPPOSITE DRIVE END LARGE END	OPPOSITE DRIVE END SMALL END
CUP PILOT I.D.	4.2777	3.6198	4.2779	3.6191
CAGE O.D.	4.2705	3.6123	4.2705	3.6118
CUP/CAGE CLEARANCE	.0072	.0075	.0074	.0073
RADIAL CAGE GROWTH DUE TO ROTATION (7400 RPM) x 2	.000045	.000022	.000045	.000022
RADIAL CUP DEFORMATION DUE TO FIT x 2	.0012	.0010	—	—
RUNNING CUP/CAGE CLEARANCE	.006	.0065	.0074	.0073

BELLEVILLE LOADING SPRINGS USED - SET NO. 2

FREE HEIGHT .407

SPRING DEFLECTION .112

PRELOAD 2115 POUNDS

SLAVE BEARINGS

	DRIVE END CENTER	OPPOSITE DRIVE END CENTER
CONE SERIAL NO.	1	2
CUP SERIAL NO.	1	2
ROLLER SIZE	7	6 8
SHAFT O.D.	4.7523	4.7523
CONE I.D.	4.7505	4.7507
CONE/SHAFT FIT	.0018 TIGHT	.0016 TIGHT
CUP O.D.	7.5001	7.5004
HOUSING I.D.	7.4976	7.4974
CUP/HOUSING FIT	.0025	.0030
CAGE SHAKE	.0096"	.0106

BEARING ADJUSTMENT AIM .002"-.003" EP ACTUAL .0017" EP

MAGNETIC SEAL NO. 1

	MAGNET RING	SEAL CASE/CARBON INSERT
WEIGHT	95.919 GM 95.907 .012 GM LOSS	95.3944 GM 95.3318 .0126 GM LOSS
WIDTH	.3792" .3792" 0	.3361" .3361" 0

TOTAL TEST TIME 37.6 HOURS

TABLE B-2. ADVANCED-TRANSMISSION COMPONENTS INVESTIGATION, BEARING SHAFT SET 1

SPRING PRELOAD 2115 LBF

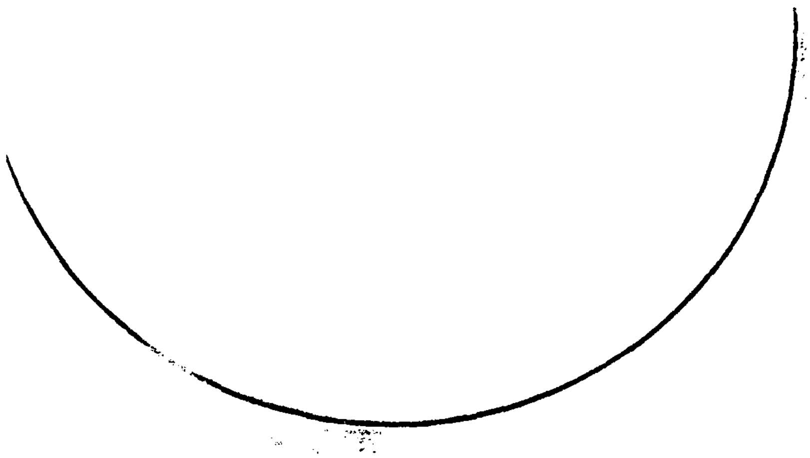
SHAFT FLOW DISTRIBUTION TEST 4.00 SLAVE 4

FLOATING CUP DIAMETRICAL FIT .0035 LOOSE

SLAVE BEARING SETTING .0017

		-----TEMPERATURES (F)-----						--OIL FLOWS--- PT/MIN			---HEAT-- BTU/MIN	
RPM	LOAD	OIL IN	CUP DE	OD ODE	OIL DE	OUT ODE	OD SEAL	SLV	OD	SHFT	BRG	OIL
1000	2000	145	144	144	143	145	166	8.2	3.1	16.0	44	-11
1000	5200	144	145	145	145	145	165	8.2	3.7	16.0	50	14
1000	7800	143	145	144	144	144	167	8.3	4.0	16.0	55	14
1000	10400	143	147	144	146	145	170	8.3	4.3	16.0	56	34
1000	11650	143	146	143	144	144	167	8.1	4.4	15.9	58	14
3700	2000	190	206	198	203	202	230	8.4	5.8	16.8	246	194
3700	5200	192	211	202	209	205	237	8.0	5.9	16.0	260	220
3700	7750	191	208	200	210	205	232	8.3	6.5	16.0	274	248
3700	10400	190	210	199	210	202	236	8.3	6.8	16.0	288	237
3700	11700	191	212	200	211	204	238	8.3	6.7	16.0	288	246
5550	2000	191	220	210	221	215	229	8.5	5.4	16.2	453	403
5550	5200	191	224	210	226	214	230	8.1	6.0	16.1	481	425
5550	7810	191	226	210	230	214	231	8.2	6.5	16.1	499	458
5550	10400	188	226	207	229	211	229	8.1	6.8	16.0	521	470
5550	11700	190	227	208	231	213	232	8.1	6.7	16.0	519	469
7400	2000	191	235	220	237	225	230	8.0	5.0	16.0	710	574
7400	5200	193	241	221	244	227	236	8.0	5.8	16.0	728	620
7400	7800	192	243	221	246	226	237	8.0	6.2	16.0	759	645
7400	10400	193	246	220	249	225	241	8.0	6.7	16.0	776	647
7400	11700	192	246	219	248	224	240	8.0	6.7	16.0	787	647

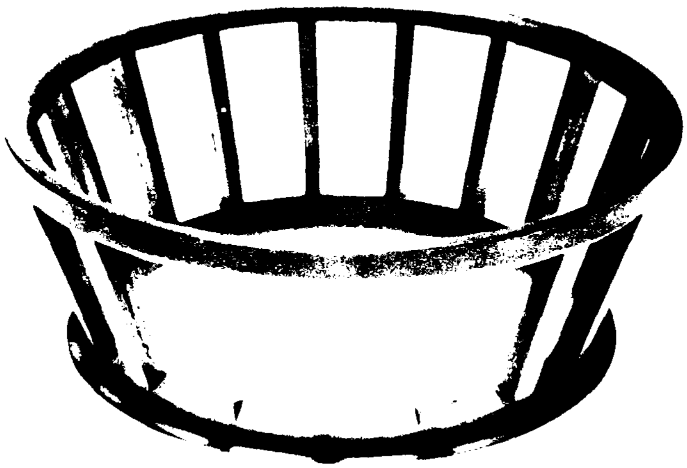
CUP 78-21 VIEWED FROM SMALL END



CUP 78-21 VIEWED FROM LARGE END

Figure B-1. Cup, Rollers, and Cage Used at Drive End of Shaft 78-8 in Test Setup No. 1 (Sheet 1 of 2).

ROLLERS



CAGE 78-24

Figure B-1. Cup, Rollers, and Cage Used at Drive End of Shaft 78-8 in Test Setup No. 1 (Sheet 2 of 2).

CUP 78-1 VIEWED FROM SMALL END

CUP 78-1 VIEWED FROM LARGE END

Figure B-2. Cup, Rollers, and Cage Used Opposite Drive End of Shaft 78-8 in Test Setup No. 1 (Sheet 1 of 2).

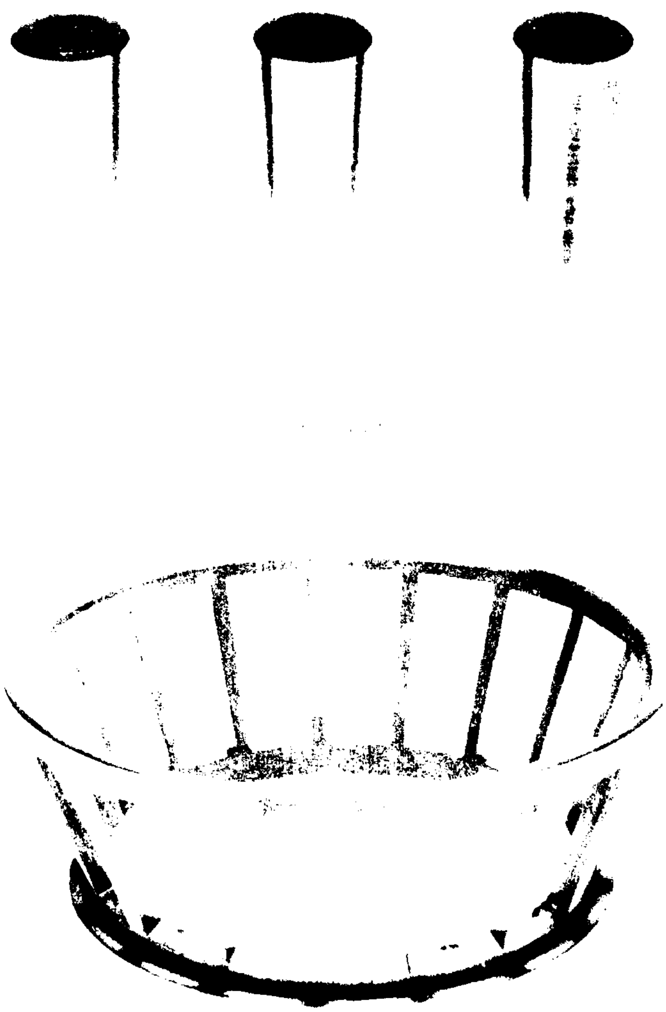


Figure B-2. Cup, Rollers, and Cage Used Opposite Drive End of *Shaft 78-8 in* Test Setup No. 1 (Sheet 2 of 2)

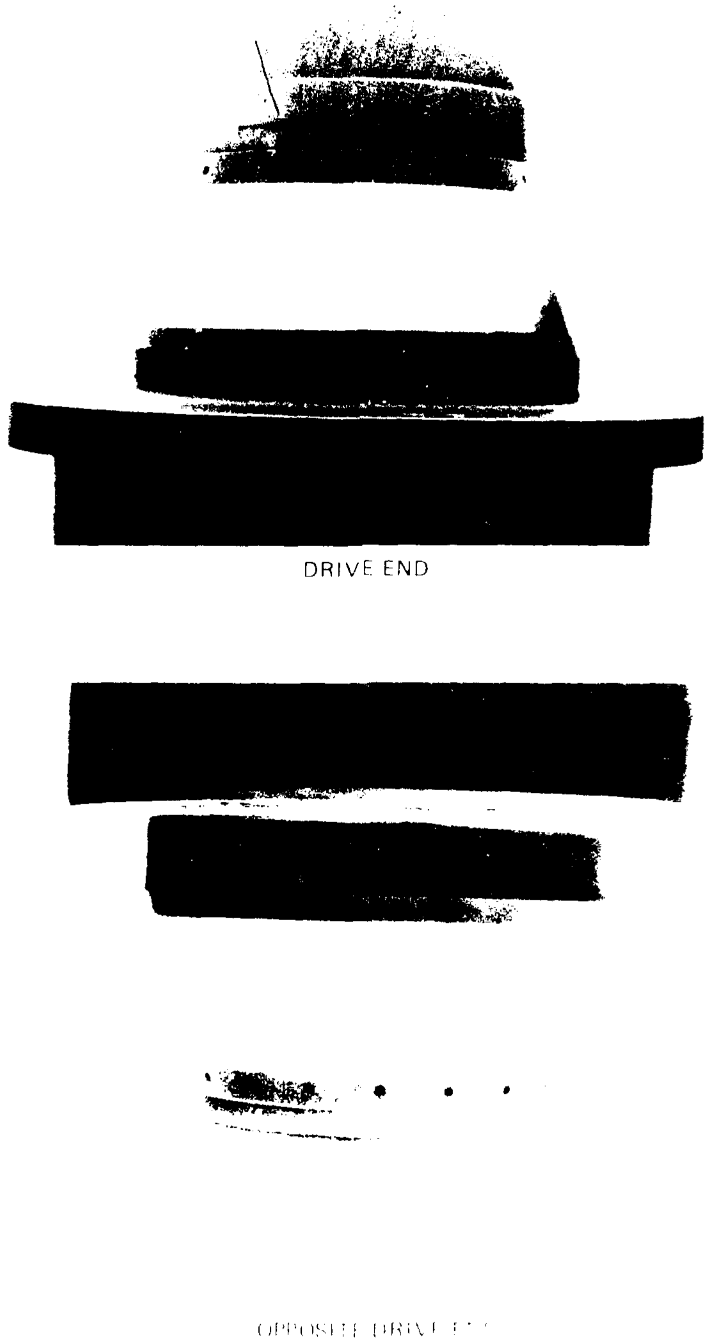
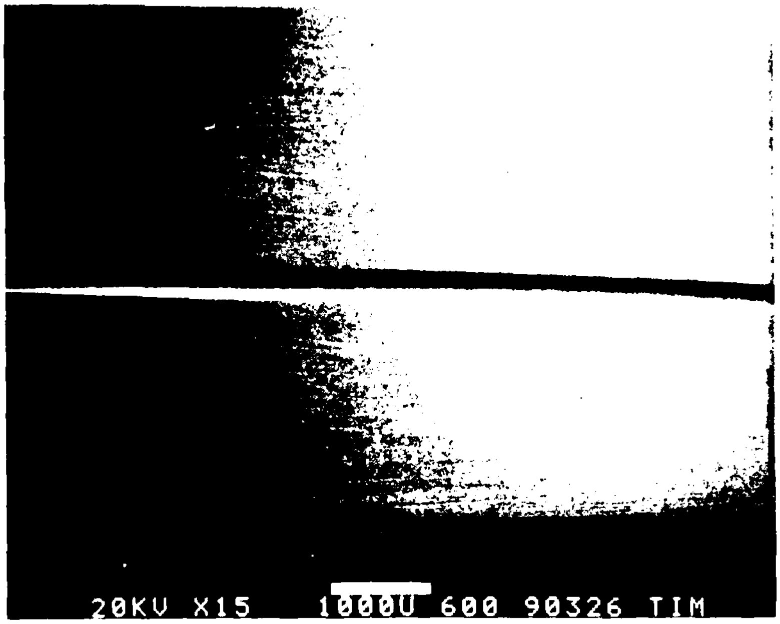


Figure B-3. Shaft No. 78-8 From Test Setup No. 1.

Figure B-4. Roller From Drive End of Cup 78-21 From Test Setup No. 1 With Groove 0.004 Inch Deep by 0.012 Inch Wide.



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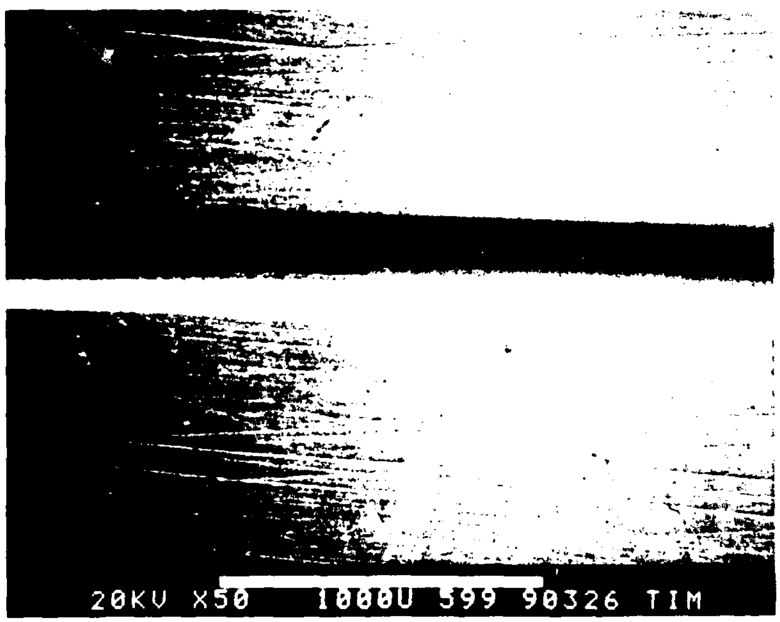


50X

Figure B 5. Scanning-Electron-Microscope Photographs of Roller From Drive End of Cup 78-21 From Test Setup No. 1.

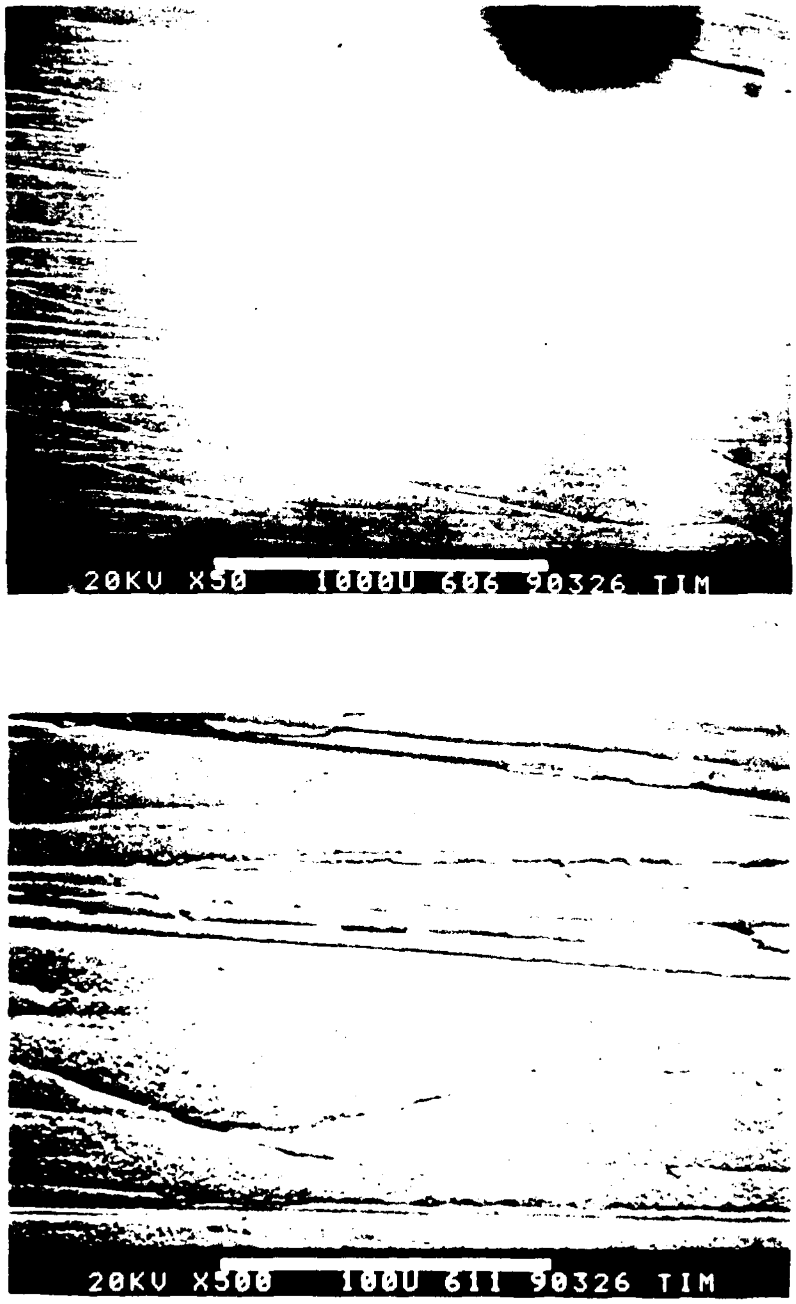


Figure B-6. Scanning Electron Microscope Photographs Showing Bright Circumferential Band on Roller From Cup 78-21.

TABLE B-3. TEST NO. 141.1-U - BUILDUP SHEET,

SETUP NO. 2

TEST BEARINGS

	DRIVE END	OPPOSITE DRIVE END
SHAFT SERIAL NO.	78-1	78-1
CAGE SERIAL NO.	15-36	78-15
CUP SERIAL NO.	78-3	78-2
ROLLER SIZE	(3) $7\frac{3}{8}$	(4) 7 [illegible]
CUP O.D.	5.0005	5.0007
HOUSING I.D.	4.9984	5.0038
CUP/HOUSING FIT	.0017	[illegible]3

	DRIVE END LARGE END	DRIVE END SMALL END	OPPOSITE DRIVE END LARGE END	OPPOSITE DRIVE END SMALL END
CUP PILOT I.D.	4.2772	3.6191	4.2780	3.6190
CAGE O.D.	4.2682	3.6119	4.2715	3.6120
CUP/CAGE CLEARANCE	.0090	.0072	.0065	.0070
RADIAL CAGE GROWTH DUE TO ROTATION (2400 RPM) 400 speed 3210 RPM x2	.000045	.000022	.000045	.000022
RADIAL CUP DEFORMATION DUE TO FIT	[illegible]	[illegible]		—
RUNNING CUP/CAGE CLEARANCE	.0077	.0062	.0065	.007

BELLEVILLE LOADING SPRINGS USED - SET NO. 2

FREE HEIGHT .467

SPRING DEFLECTION .112

PRELOAD 2115 POUNDS

SLAVE BEARINGS

	DRIVE END CENTER	OPPOSITE DRIVE END CENTER
CONE SERIAL NO.	1	2
CUP SERIAL NO.	1	2
ROLLER SIZE	7	$6\frac{5}{8}$
SHAFT O.D.	4.7521	4.7521
CONE I.D.	4.7505	4.7507
CONE/SHAFT FIT	.0016 -T	.0014 -T
CUP O.D.	7.5001	7.5004
HOUSING I.D.	7.4976	7.4974
CUP/HOUSING FIT	.0025 -T	.0030 -T
CAGE SHAKE	B: .010" A: .009"	B: .0108" A: .0?48

BEARING ADJUSTMENT AIM .002" - .003" E.P. ACTUAL .0023

MAGNETIC SEAL NO. 2

	MAGNET RING	SEAL CASE/CARBON INSERT
WEIGHT	95.167 GM 95.161 .006 LOSS	95.3782 GM 95.3659 .0123 LOSS
WIDTH	.3791" .3790 .0001 LOSS	.3364" .3353 .0011 LOSS

O.D.E. SLAVE BRG. BACKED OFF .125"
TOTAL TEST TIME 58 HOURS

TABLE B-4. ADVANCED-TRANSMISSION COMPONENTS INVESTIGATION, BEARING/SHAFT SET 2

SPRING PRELOAD 2115 LBF

SHAFT FLOW DISTRIBUTION TEST 8.00 SLAVE 8

FLOATING CUP DIAMETRICAL FIT .0031 LOOSE

SLAVE BEARING SETTING .0023

		-----TEMPERATURES (F)-----						--OIL FLOWS--- PT/MIN			---HEAT-- BTU/MIN	
RPM	LOAD	OIL IN	CUP DE	OD ODE	OIL DE	OUT ODE	OD SEAL	SLV	OD	SHFT	BRG	OIL
3700	5200	189	197	197	203	200	227	12.0	0.0	32.0	281	277
3700	7800	191	201	199	205	201	231	12.0	0.0	32.0	295	266
3700	10400	189	200	198	205	200	229	12.0	0.0	32.0	307	299
3700	11700	189	201	198	207	201	229	12.0	0.0	32.0	306	333
5550	5200	187	207	204	212	208	215	12.0	0.0	32.0	547	511
5550	7800	187	209	205	213	209	216	12.0	0.0	32.0	572	534
5550	10410	187	210	207	215	210	215	12.0	0.0	32.0	585	567
5550	11700	190	214	207	218	211	231	12.0	0.0	32.0	581	545
7400	5200	192	222	220	227	225	221	12.0	0.0	32.0	834	759
7400	7800	186	220	215	224	222	220	12.0	0.0	32.0	891	826
7400	10400	187	222	217	227	223	221	12.0	0.0	32.0	905	849
7400	11700	189	223	216	229	223	221	12.0	0.0	32.0	906	827
9600	5200	187	233	229	239	237	225	12.0	0.0	32.0	1263	1144
9600	7800	187	234	229	240	237	225	12.0	0.0	32.0	1312	1155
9600	10400	190	240	231	245	240	230	12.0	0.0	32.0	1315	1179
9600	11700	191	242	232	247	240	231	12.0	0.0	32.0	1316	1179
11800	5200	191	251	249	259	257	229	12.0	0.0	32.0	1691	1511
11800	7800	191	255	248	261	256	231	12.0	0.0	32.0	1738	1523

TABLE B-5. ADVANCED-TRANSMISSION COMPONENTS INVESTIGATION, BEARING/SHAFT SET 21

SPRING PRELOAD 2115 LBF

SHAFT FLOW DISTRIBUTION TEST 4.00 SLAVE 4

FLOATING CUP DIAMETRICAL FIT .0031 LOOSE

SLAVE BEARING SETTING .0023

		-----TEMPERATURES	(F)-----					--OIL	FLOWS---		---HEAT--	
		OIL	CUP	OD	OIL	OUT	OD		PT/MIN		BTU/MIN	
RPM	LOAD	IN	DE	ODE	DE	ODE	SEAL	SLV	OD	SHFT	BRG	OIL
1000	2000	158	159	155	159	157	174	8.0	3.1	16.0	38	-2
3700	5200	191	207	199	213	205	222	8.0	5.5	16.0	255	257
5550	7800	192	226	208	233	216	226	8.0	6.0	16.0	491	469
5550	10410	191	227	207	233	213	229	8.0	6.4	16.0	512	461
7400	11700	190	246	235	252	225	237	8.0	6.4	16.0	773	708
3700	5205	191	205	203	211	209	225	8.0	0.0	16.0	259	230
3700	7800	187	205	200	209	205	225	8.0	0.0	16.0	281	242
3700	10400	189	207	201	213	209	225	8.0	0.0	16.0	283	267
3700	11700	190	209	201	214	209	227	8.0	0.0	16.0	285	261
5550	5200	188	221	215	226	220	223	8.0	0.0	16.0	489	426
5550	7800	191	225	218	230	222	226	8.0	0.0	16.0	504	426
5550	10400	192	227	219	233	225	226	8.0	0.0	16.0	511	451
5550	11700	190	229	219	234	225	229	8.0	0.0	16.0	514	482
7400	5200	187	239	231	243	237	232	8.0	0.0	16.0	743	649
7400	7800	192	246	236	251	241	238	8.0	0.0	16.0	752	662
7400	10400	193	246	237	252	242	239	8.0	0.0	16.0	770	662
7400	11700	190	247	233	251	240	239	8.0	0.0	16.0	782	680

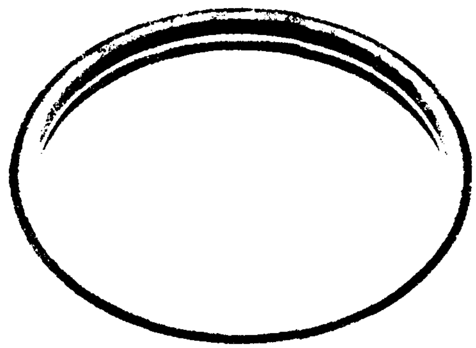


Figure B-7. Magnetic Seal Ring From Test Setup No. 2.

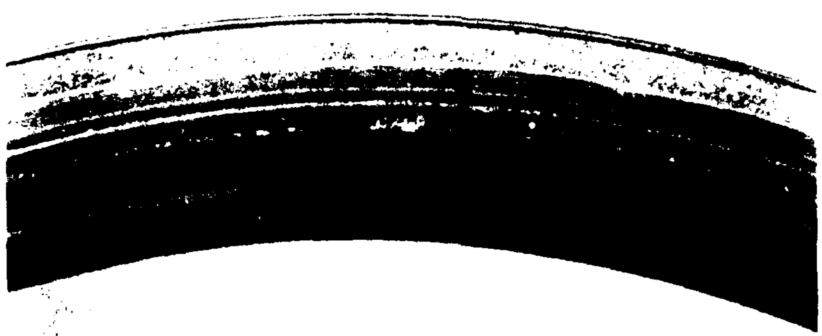


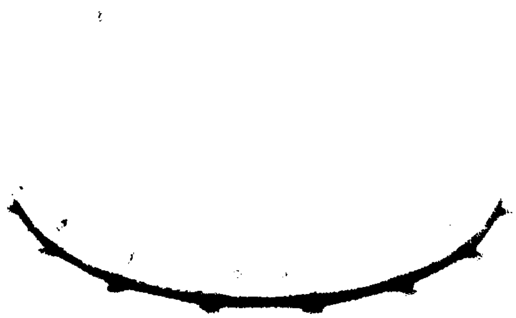
Figure B-8. Magnetic Seal Ring From Test Setup No. 4 Showing the Carbon Insert on the Inside Diameter and Front Face

CUP 78 3 VIEWED FROM SMALL END

CUP 78 3 VIEWED FROM LARGE END

Figure B 9. Cup, Rollers, and Cage Used at Drive End of Shaft 78-1 in Test Setup No. 2 (Sheet 1 of 2)

ROLLERS



CAGE 78-36

Figure B-9. Cup, Rollers, and Cage Used at Drive End of Shaft 78-1 in Test Setup No. 2 (Sheet 2 of 2).

CUP 78 2 VIEWED FROM SMALL END

CUP 78 2 VIEWED FROM LARGE END

Figure B 10. Cup, Rollers, and Cage Used Opposite Drive End of Shaft 78-1 in Test Setup No. 2 (Sheet 1 of 2).

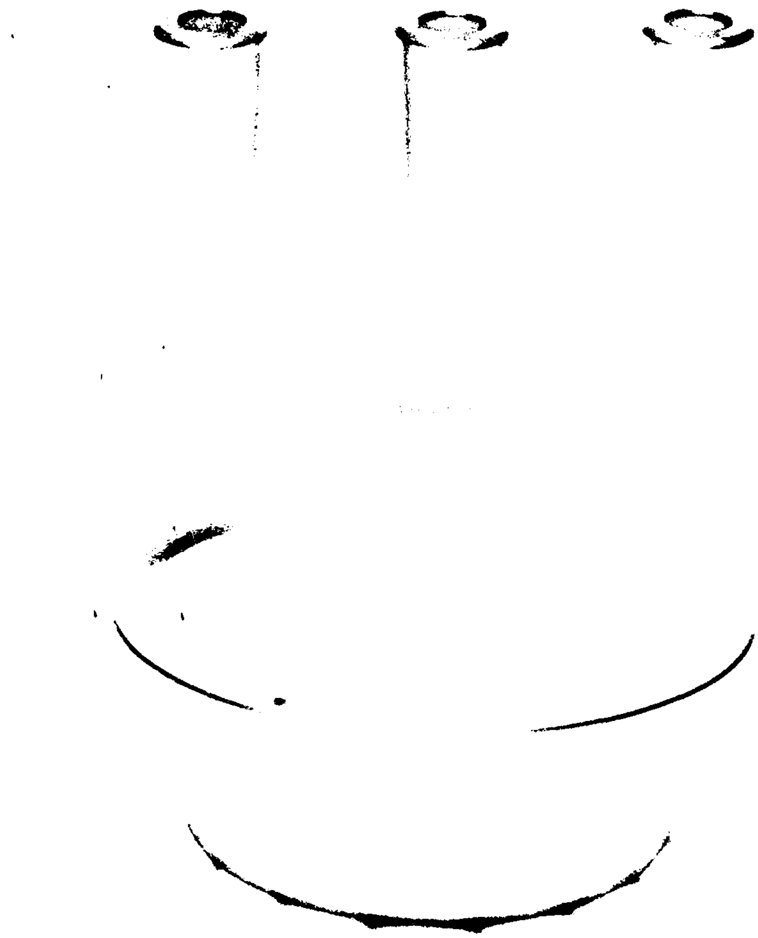


Figure B-10. Cup, Rollers, and Cage Used Opposite Drive End of Shaft 78-1 in Test Setup No. 2 (Sheet 2 of 2).

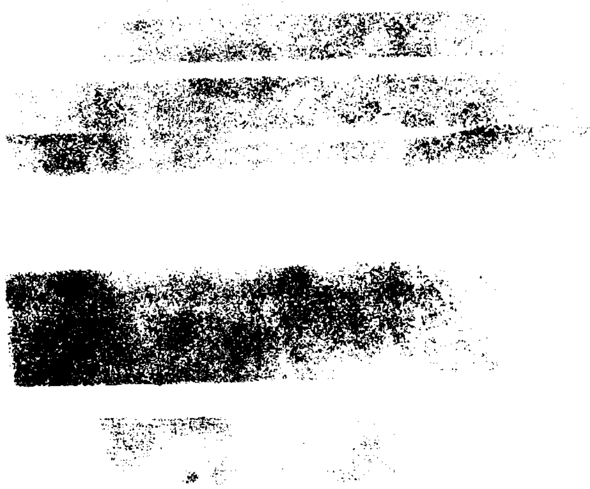


Figure B-[illegible] [illegible] No. [illegible] From Test Setup No. [illegible]

Figure B-12. Results of Test Setup No. 2 With All Eight Bolts Breaking and Bearing Backing Off 0.125 Inch.

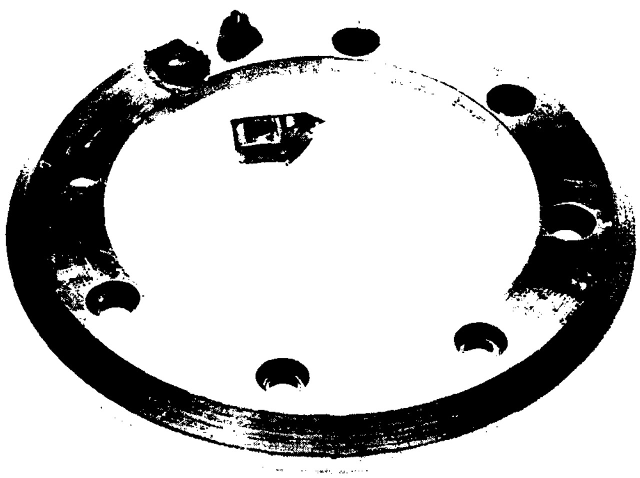


Figure B-13. End Cap for Opposite-Drive-End Slave Bearing Found After 7.5 Hours of Test Setup No. 2; Bolt Believed to Have Broken After 3.75 Hours of Test.

TABLE B-6. TEST NO. 141.1-1 – [illegible] SHEET,

SET NO. 3

TEST BEARINGS

	DRIVE END		OPPOSITE DRIVE END	
SHAFT SERIAL NO.	78-3		78-3	
CAGE SERIAL NO.	78-13		78-30	
CUP SERIAL NO.	78-11		78-[illegible]	
ROLLER SIZE	7 7/8		7 7/8	
CUP O.D.	5.0005		5.0003	
HOUSING I.D.	4.9989		5.0033	
CUP/HOUSING FIT	.0016 TIGHT		.0030 LOOSE	
	LARGE END	SMALL END	LARGE END	SMALL END
CUP PILOT I.D.	4.2783	3.6191	4.2782	3.6190
CAGE O.D.	4.2700	3.6093	4.2700	3.6093
CUP/CAGE CLEARANCE	.0083	.0098	.0082	.0097
RADIAL CAGE GROWTH x 2 DUE TO ROTATION (4000 RPM)	.00016	.000079	.00016	.000079
RADIAL CUP DEFORMATION DUE TO FIT	[illegible]	[illegible]	–	–
RUNNING CUP/CAGE CLEARANCE	.008	.0082	.008	.0096

BELLEVILLE LOADING SPRINGS USED - SET NO. 2

FREE HEIGHT .427

SPRING DEFLECTION .1145 (3-A with New Cup Adapter .35 used .018' Shim)

PRELOAD 2140 POUNDS

ODE Cup has .003 END MOVEMENT

DE Cup ADAPTOR TO Cu[illegible] .102

SLAVE BEARINGS

	DRIVE END CENTER	OPPOSITE DRIVE END CENTER
CONE SERIAL NO.	1	3
CUP SERIAL NO.		2
ROLLER SIZE	7	5 7/8 (3)
SHAFT O.D.	4.25215	4.25215
CONE I.D.	4.24770	[illegible]
CONE/SHAFT FIT	.00445 T	.00475 T
CUP O.D.	7.2001	
HOUSING I.D.	7.1975	7.1974
CUP/HOUSING FIT		
CAGE SHAKE	.005"-.006"-.007" – .0060	.007"-.006".001 – .0066"

BEARING ADJUSTMENT AIM [illegible] ACTUAL .0030" AT START OF TEST

MAGNETIC SEAL NO. 3

	MAGNET RING	SEAL FACE/CARBON INSERT
	[illegible]	AFTER 190
WEIGHT	15.0957	[illegible]
	.3077	33-7
WIDTH	[illegible]	.3260

39.5 Hours

TABLE B-7. ADVANCED-TRANSMISSION COMPONENTS INVESTIGATION, BEARING/SHAFT SET 3

SPRING PRELOAD 2140 LBF

SHAFT FLOW DISTRIBUTION TEST 2.00 SLAVE 8

FLOATING CUP DIAMETRICAL FIT .0035 LOOSE

SLAVE BEARING SETTING 0.0000

		-----TEMPERATURES (F)-----						--OIL FLOWS--- PT/MIN			---HEAT-- BTU/MIN	
RPM	LOAD	OIL IN	CUP, DE	OD ODE	OIL DE	OUT ODE	OD SEAL	SLV	OD	SHFT	BRG	OIL
3700	5200	187	207	193	207	197	211	8.0	6.1	20.0	273	242
3700	7800	187	208	193	207	197	215	8.0	6.6	20.0	289	245
3700	10400	187	209	193	208	197	215	8.0	6.9	20.0	299	254
3700	11710	186	209	192	208	195	216	8.0	7.0	20.0	304	251
5550	5200	191	227	204	221	209	223	8.0	6.1	20.0	509	396
5550	7810	191	230	205	225	209	225	8.0	6.7	20.0	527	430
5550	10410	191	232	206	227	209	231	8.0	7.0	20.0	540	447
5550	11710	190	231	206	227	209	234	8.0	7.1	20.0	544	465
7400	5200	189	243	214	232	220	235	8.0	5.8	20.0	792	617
7400	7800	185	245	210	232	219	236	8.0	6.2	20.0	829	683
7400	10400	187	247	210	234	218	237	8.0	6.6	20.0	852	658
7400	11700	186	249	211	235	219	237	8.0	6.7	20.0	852	696
3700	5205	188	207	201	207	201	0	8.0	0.0	20.0	273	226
3700	7810	188	209	201	209	201	0	8.0	0.0	20.0	287	240
3700	10410	191	210	204	211	204	0	8.0	0.0	20.0	293	233
3700	11710	189	211	204	211	203	0	8.0	0.0	20.0	297	254
5550	5205	190	227	219	222	216	0	8.0	0.0	20.0	508	411
5550	7810	187	229	217	222	214	0	8.0	0.0	20.0	539	440
5550	10410	189	232	220	226	217	0	8.0	0.0	20.0	544	461
5550	11710	187	230	217	225	215	0	8.0	0.0	20.0	554	468
7400	5205	190	246	236	234	231	0	8.0	0.0	20.0	791	605
7400	7810	189	249	237	236	232	0	8.0	0.0	20.0	821	641
7400	10410	190	251	235	239	234	0	8.0	0.0	20.0	834	663
7400	11710	187	250	235	237	232	0	8.0	0.0	20.0	851	677

TABLE B-8. TEST NO. 141.1-U - BUILDUP SHEET,
SETUP NO. 3A

TEST BEARINGS

	DRIVE END		OPPOSITE DRIVE END	
SHAFT SERIAL NO.				
CAGE SERIAL NO.	SAME TEST BRG.		SAME TEST BRG.	
CUP SERIAL NO.	AS #3		AS #3	
ROLLER SIZE				
CUP O.D.			5.0003	
HOUSING I.D.			5.001	
CUP/HOUSING FIT			.0007 LOOSE	
	LARGE END	SMALL END	LARGE END	SMALL END
CUP PILOT I.D.				
CAGE O.D.				
CUP/CAGE CLEARANCE				
RADIAL CAGE GROWTH x 2 DUE TO ROTATION (________ RPM)				
RADIAL CUP DEFORMATION DUE TO FIT				
RUNNING CUP/CAGE CLEARANCE				

BELLEVILLE LOADING SPRINGS USED - SET NO.	2	
FREE HEIGHT	.407	
SPRING DEFLECTION	.1135	
PRELOAD	2130	POUNDS

SLAVE BEARINGS

	DRIVE END CENTER	OPPOSITE DRIVE END CENTER
CONE SERIAL NO.		4
CUP SERIAL NO.		3
ROLLER SIZE		6 4/8
SHAFT O.D.		4.25215
CONE I.D.		4.2476
CONE/SHAFT FIT		.00455
CUP O.D.		7.5003
HOUSING I.D.		7.4974
CUP/HOUSING FIT		.0029 T
CAGE SHAKE		.006-.008-.006 = .0067"

BEARING ADJUSTMENT AIM .000"   ACTUAL .000"

TABLE B-9. ADVANCED-TRANSMISSION COMPONENTS INVESTIGATION, BEARING/SHAFT SET 31

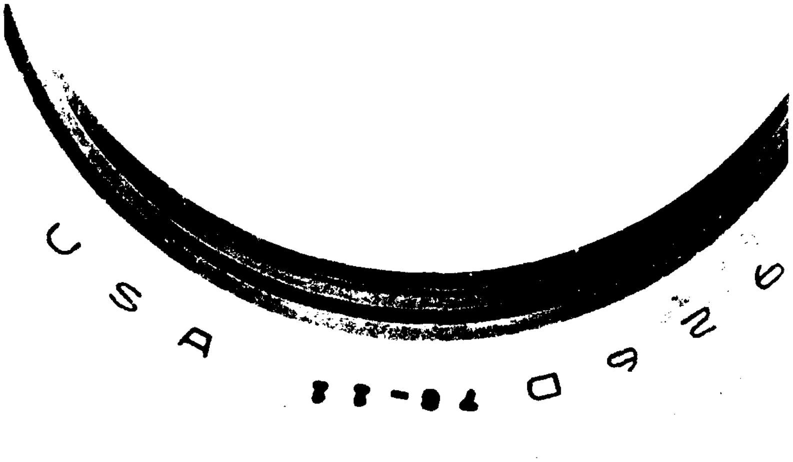
SPRING PRELØAD 2130 LBF

SHAFT FLØW DISTRIBUTIØN TEST 2.00 SLAVE 8

FLØATING CUP DIAMETRICAL FIT .0007 LØØSE

SLAVE BEARING SETTING 0.0000

		TEMPERATURES (F)						OIL FLØWS PT/MIN			HEAT BTU/MIN	
RPM	LØAD	ØIL IN	CUP DE	ØD ØDE	ØIL DE	ØUT ØDE	ØD SEAL	SLV	ØD	SHFT	BRG	ØIL
3700	5200	189	209	203	207	202	234	8.0	0.0	20.0	272	219
3700	7810	190	213	203	211	203	237	8.0	0.0	20.0	283	240
3700	10410	188	212	200	210	201	237	8.0	0.0	20.0	296	247
3700	11710	190	215	201	211	203	237	8.0	0.0	20.0	297	240
5550	5205	189	233	220	222	216	240	8.0	0.0	20.0	508	426
5550	7810	190	234	221	225	218	241	8.0	0.0	20.0	526	447
5550	10410	191	236	221	227	218	242	8.0	0.0	20.0	540	447
5550	11710	191	238	221	228	219	242	8.0	0.0	20.0	541	462
7400	5205	191	253	242	235	235	249	8.0	0.0	20.0	779	627
7400	7810	191	254	242	239	237	249	8.0	0.0	20.0	802	671
7400	10410	189	255	240	240	236	249	8.0	0.0	20.0	826	699
7400	11710	191	257	241	241	237	250	8.0	0.0	20.0	828	685
3700	5205	189	210	204	207	202	236	8.0	0.0	20.0	272	219
3700	7800	190	214	205	210	205	239	8.0	0.0	20.0	282	247
3700	10410	189	216	205	212	204	239	8.0	0.0	20.0	291	269
3700	11710	189	215	204	212	204	239	8.0	0.0	20.0	295	269
5550	5205	190	234	223	225	218	243	8.0	0.0	20.0	500	447
5550	7810	190	234	222	226	218	241	8.0	0.0	20.0	524	454
5550	10410	190	237	221	227	219	244	8.0	0.0	20.0	538	469
5550	11710	190	237	220	227	218	243	8.0	0.0	20.0	545	462
7400	5205	192	253	242	237	236	251	8.0	0.0	20.0	772	634
7400	7810	190	254	241	239	235	250	8.0	0.0	20.0	806	670
7400	10410	190	253	238	239	235	248	8.0	0.0	20.0	831	670
7400	11710	191	256	239	241	237	251	8.0	0.0	20.0	828	685



CUP 78-11 VIEWED FROM SMALL END

CUP 78-11 VIEWED FROM LARGE END

Figure B-14. Cup, Rollers, and Cage Used at Drive End of Shaft 78-3 in Test Setup No. 3 (Sheet 1 of 2).

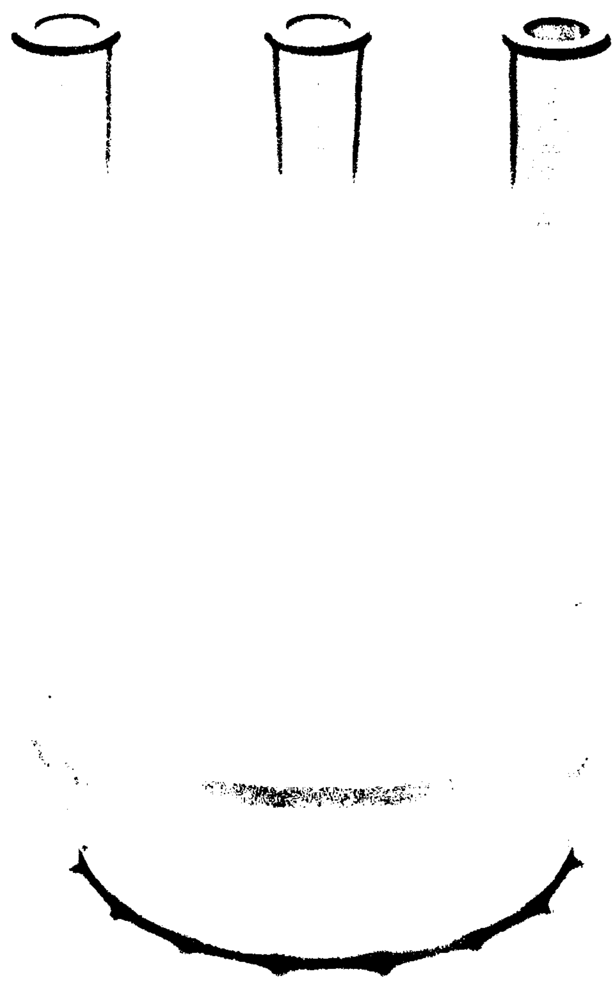
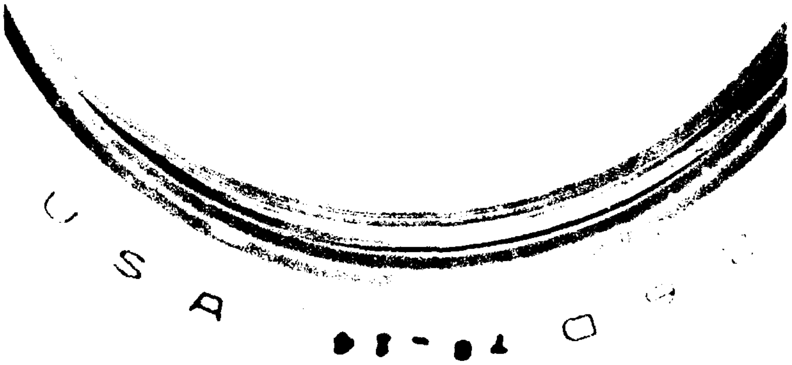


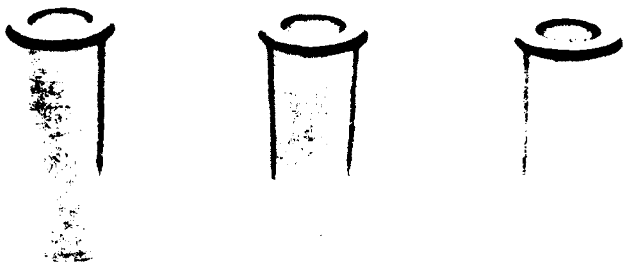
Figure B-14. Cup, Rollers, and Cage Used at Drive End of Shaft 78-3 in Test Setup No. 3
(Sheet 2 of 2).



CUP 78-14 VIEWED FROM SMALL END

CUP 78-14 VIEWED FROM LARGE END

Figure B-15. Cup, Rollers, and Cage Used Opposite Drive End of Shaft 78-3 in Test Setup No. 3 (Sheet 1 of 2).



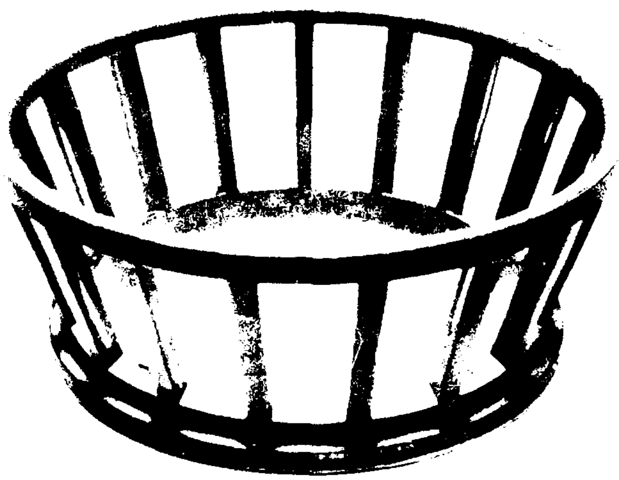


Figure B-15. Cup, Rollers, and Cage Used Opposite Drive End of Shaft 78-3 in Test Setup No. 3 (Sheet 2 of 2).

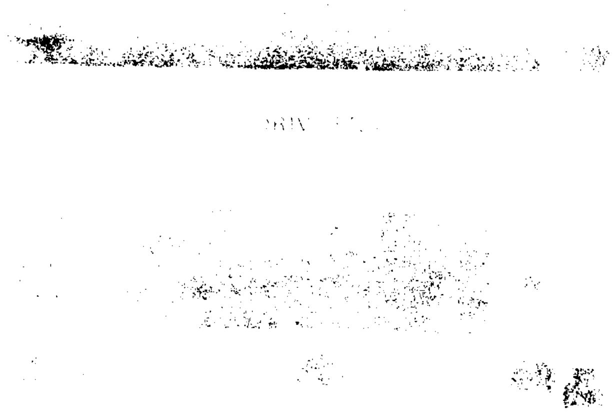


Figure B-[illegible] Shaft No. [illegible]S-3 From Test Setup No. [illegible]

TABLE B-10. TEST NO. 141.1-U - BUILDUP SHEET, SETUP NO. 4

TEST BEARINGS

	DRIVE END		OPPOSITE DRIVE END	
SHAFT SERIAL NO.	78-5		78-5	
CAGE SERIAL NO.	78-17		78-21	
CUP SERIAL NO.	79-1		78-5	
ROLLER SIZE	7¾		7¼	
CUP O.D.	5.0048		5.0042	
HOUSING I.D.	4.9929		[illegible]	
CUP/HOUSING FIT	.0019-T		.0018-T	
	LARGE END	SMALL END	LARGE END	SMALL END
CUP PILOT I.D.	[illegible]	[illegible]	[illegible]	[illegible]
CAGE O.D.	[illegible]	3.5724	[illegible]	3.5137
CUP/CAGE CLEARANCE	.0068	[illegible]	[illegible]	[illegible]
RADIAL CAGE GROWTH DUE TO ROTATION ([illegible] RPM)	.000045	.000022	.000045	.000022
RADIAL CUP DEFORMATION DUE TO FIT	.0014	.0011	—	—
RUNNING CUP/CAGE CLEARANCE	.0046	[illegible]4	.0074	[illegible]3

BELLEVILLE LOADING SPRINGS USED - SET NO. 2
FREE HEIGHT .407
SPRING DEFLECTION .0226
PRELOAD 2115 POUNDS

SLAVE BEARINGS

	DRIVE END CENTER	OPPOSITE DRIVE END CENTER
CONE SERIAL NO.		4
CUP SERIAL NO.		3
ROLLER SIZE		
SHAFT O.D.	4.75225	4.75225
CONE I.D.	[illegible]	4.75770
CONE/SHAFT FIT	.00432-T	.00455-T
CUP O.D.	7.5000	7.5003
HOUSING I.D.	7.4969	[illegible]
CUP/HOUSING FIT	.0031-T	.0030-T
CAGE SHAKE	.006	.006

BEARING ADJUSTMENT AIM .0005 ACTUAL .005

MAGNETIC SEAL NO. 4

	MAGNET RING	SEAL CASE/CARBON INSERT
WEIGHT	87.951.123 Gram / 87.95.135 / ±.007 Gram	87.95.225 Gram / 87.45.226 / ±.001 Gram
WIDTH	.2790 NO CHANGE AFTER TEST	.3361 NO CHANGE AFTER TEST

TABLE B-11. ADVANCED-TRANSMISSION COMPONENTS INVESTIGATION, BEARING/SHAFT SET 4

SPRING PRELØAD 2115 LBF

SHAFT FLØW DISTRIBUTIØN TEST 1.00 SLAVE 8

FLØATING CUP DIAMETRICAL FIT .0008 LØØSE

SLAVE BEARING SETTING 0.0000

		-----TEMPERATURES (F)-----						--ØIL FLØWS---			---HEAT--	
		ØIL	CUP	ØD	ØIL	ØUT	ØD	PT/MIN			BTU/MIN	
RPM	LØAD	IN	DE	ØDE	DE	ØDE	SEAL	SLV	ØD	SHFT	BRG	ØIL
3700	5205	189	219	213	209	202	247	8.0	0.0	18.0	270	216
3700	7810	187	221	210	208	201	249	8.0	0.0	18.0	289	229
3700	10410	189	225	210	211	203	252	8.0	0.0	18.0	294	236
3700	11710	189	226	209	212	204	253	8.0	0.0	18.0	295	249
5550	5205	187	240	233	221	217	251	8.0	0.0	18.0	510	422
5550	7810	190	244	234	227	220	253	8.0	0.0	18.0	520	442
5550	10410	188	245	232	226	219	253	8.0	0.0	18.0	541	455
5550	11710	189	247	231	228	219	255	8.0	0.0	18.0	541	455
7400	5205	191	265	258	238	239	268	8.0	0.0	18.0	764	629
7400	7810	190	268	259	240	240	268	8.0	0.0	18.0	794	663
7400	10410	189	269	257	241	240	268	8.0	0.0	18.0	815	683
7400	11710	189	271	256	241	240	269	8.0	0.0	18.0	822	683

AD-A090 675 BOEING VERTOL CO PHILADELPHIA PA F/G 1/3
ADVANCED TRANSMISSION COMPONENTS INVESTIGATION PROGRAM. BEARING--ETC(U)
AUG 80 J W LENSKI DAAJ02-76-C-0045
UNCLASSIFIED D210-11591-1 USAAVRADCOM-TR-80-D-19 NL

3 OF 3
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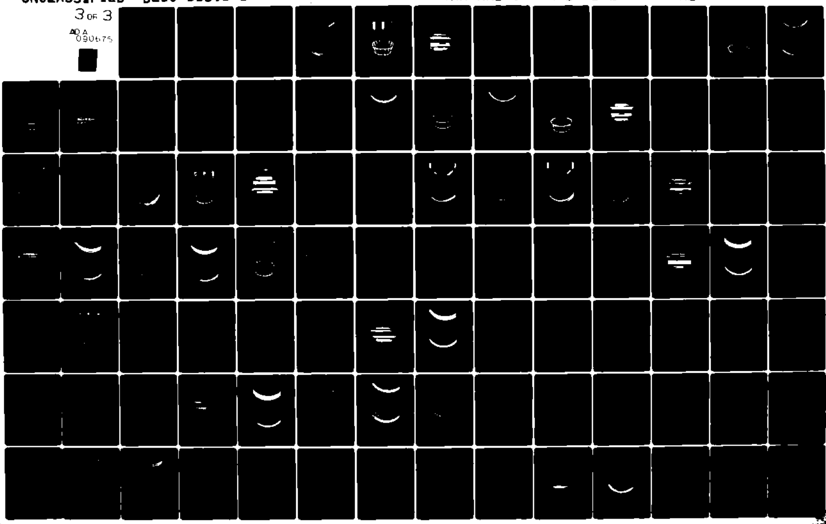


TABLE B-12. ADVANCED-TRANSMISSION COMPONENTS INVESTIGATION, BEARING/SHAFT SET 41

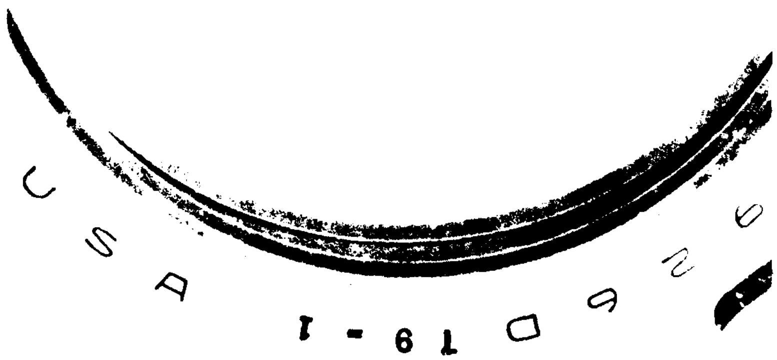
SPRING PRELOAD 2115 LBF

SHAFT FLOW DISTRIBUTION TEST .50 SLAVE 8

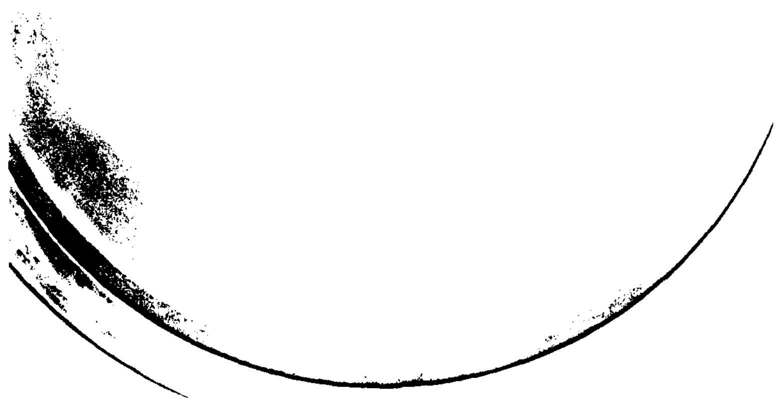
FLOATING CUP DIAMETRICAL FIT .0008 LOOSE

SLAVE BEARING SETTING 0.0000

		-----TEMPERATURES (F)-----						--OIL FLOWS--- PT/MIN			---HEAT-- BTU/MIN	
RPM	LOAD	OIL IN	CUP OE	OD ODE	OIL OE	OUT ODE	OD SEAL	SLV	OD	SHFT	BRG	OIL
3700	5205	191	219	220	210	205	241	8.0	0.0	17.0	266	208
3700	7810	190	220	219	210	204	243	8.0	0.0	17.0	283	214
3700	10410	190	222	217	212	203	244	8.0	0.0	17.0	292	221
3700	11710	189	223	216	212	203	243	8.0	0.0	17.0	295	234
5550	5205	190	248	244	226	222	265	8.0	0.0	17.0	493	431
5550	7810	189	251	254	228	222	267	8.0	0.0	17.0	515	457
5550	10410	189	248	240	227	220	264	8.0	0.0	17.0	537	438
5550	11710	190	251	240	231	221	269	8.0	0.0	17.0	532	457
7400	5205	190	270	264	237	241	286	8.0	0.0	17.0	762	624
7400	7810	191	273	264	240	243	285	8.0	0.0	17.0	787	644
7400	10410	191	276	265	242	244	291	8.0	0.0	17.0	804	663
7400	11710	190	275	265	241	242	290	8.0	0.0	17.0	818	657

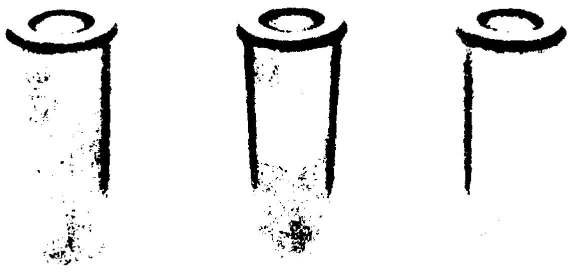


CUP 79 1 VIEWED FROM SMALL END

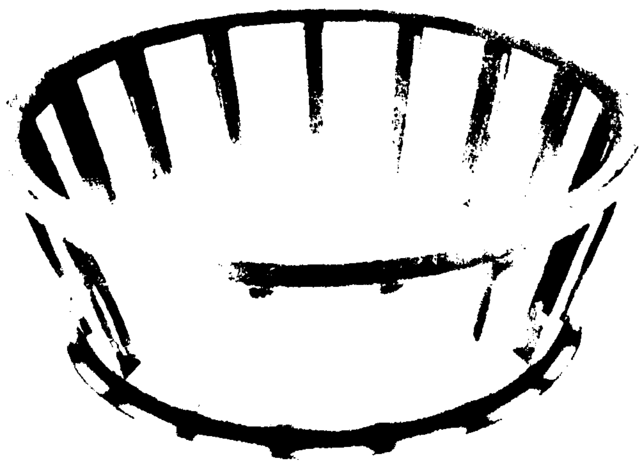


CUP 79 1 VIEWED FROM LARGE END

Figure B – 17. Cup, Rollers, and Cage Used at Drive End of Shaft 78-5 in Test Setup No. 4 (Sheet 1 of 2).

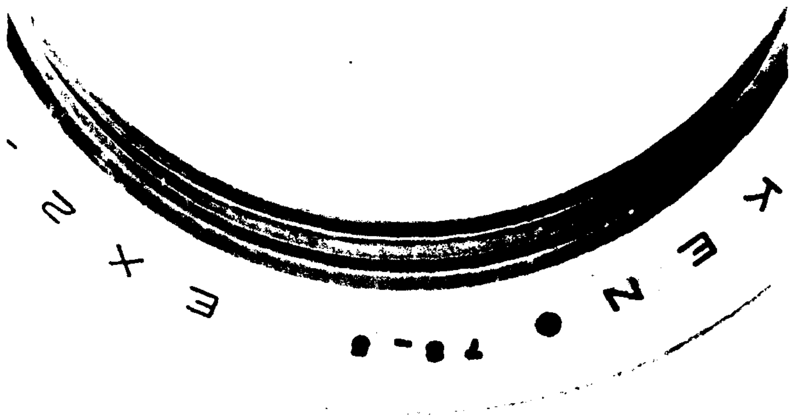


ROLLERS

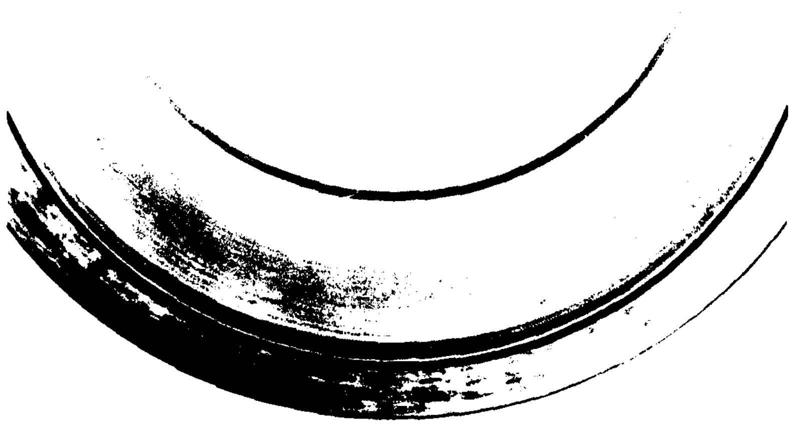


CAGE 78-17

Figure B - 17. Cup, Rollers, and Cage Used at Drive End of Shaft 78-5 in Test Setup No. 4 (Sheet 2 of 2).

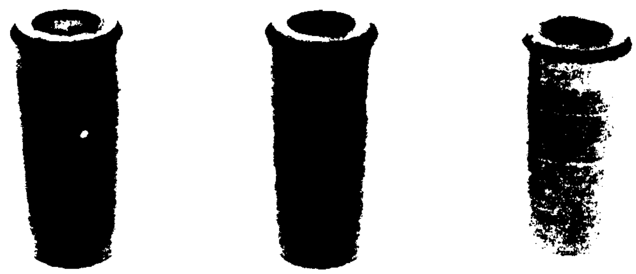


CUP 78-5 VIEWED FROM SMALL END

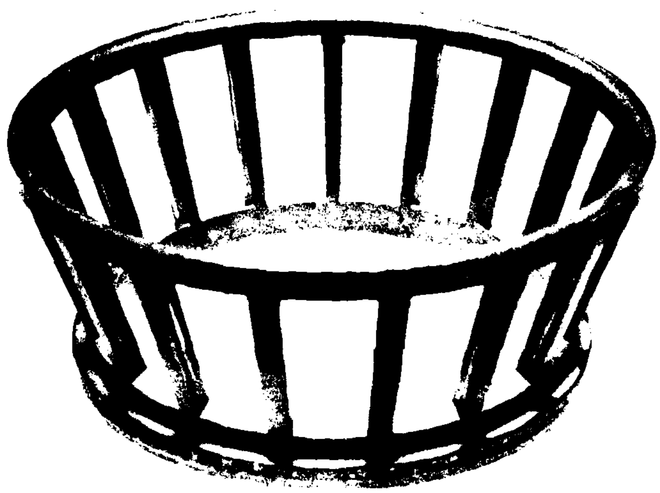


CUP 78-5 VIEWED FROM LARGE END

Figure B–18. Cup, Rollers, and Cage Used Opposite Drive End of Shaft 78-5 in Test Setup No. 4 (Sheet 1 of 2).



ROLLERS



CAGE 78-21

Figure B–18. Cup, Rollers, and Cage Used Opposite Drive End of Shaft 78-5 in Test Setup No. 4 (Sheet 2 of 2).

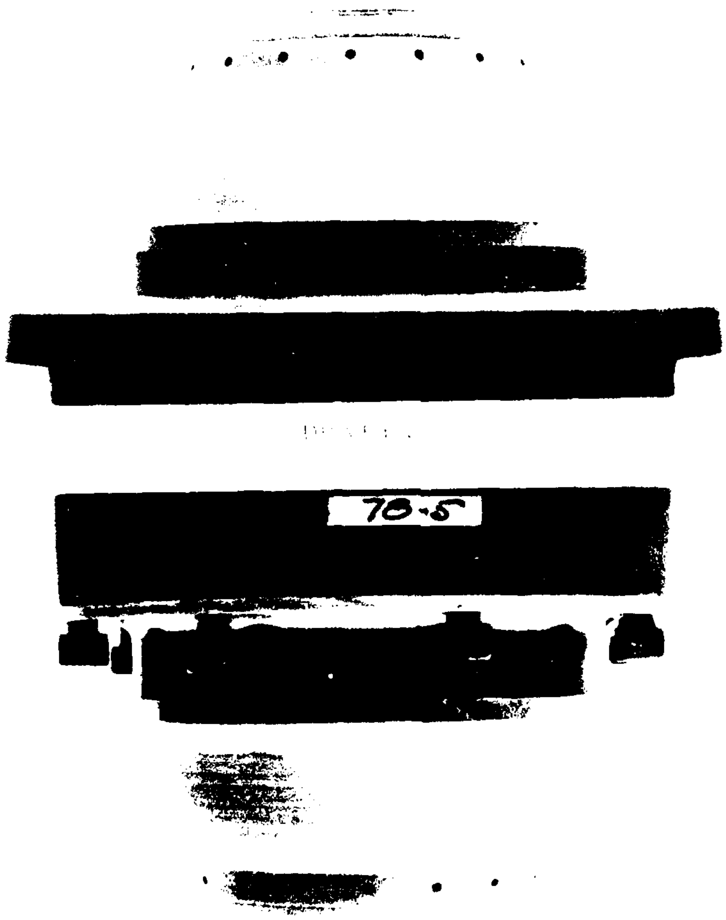


Figure B-19. Shaft No. 78-5 From Test Setup No. 4.

TABLE B-13. TEST NO. 141.1-U - BUILDUP SHEET, SETUP NO. 5

TEST BEARINGS

	DRIVE END		OPPOSITE DRIVE END	
SHAFT SERIAL NO.	78-4		78-4	
CAGE SERIAL NO.	78-23		78-25	
CUP SERIAL NO.	79-5		78-9	
ROLLER SIZE	7 3/8		7 4/8	
CUP O.D.	5.0009		5.0004	
HOUSING I.D.	4.9989 -T		5.0010	
CUP/HOUSING FIT	.0020 -T		.0006 -L	
	LARGE END	SMALL END	LARGE END	SMALL END
CUP PILOT I.D.	4.2777	3.6188	4.2781	3.6193
CAGE O.D.	4.2734	3.6122	4.2708	3.6110
CUP/CAGE CLEARANCE	.0043	.0066	.0073	.0083
RADIAL CAGE GROWTH DUE TO ROTATION (14000 RPM)	.00016	.000079	.00016	.000079
RADIAL CUP DEFORMATION DUE TO FIT	.0015	.0014	—	—
RUNNING CUP/CAGE CLEARANCE	.0026	.0051	.0071	.0082

BELLEVILLE LOADING SPRINGS USED - SET NO. 2
FREE HEIGHT .407
SPRING DEFLECTION .1120 actual .1155
PRELOAD 2115 POUNDS 2130

SLAVE BEARINGS

	DRIVE END CENTER	OPPOSITE DRIVE END CENTER
CONE SERIAL NO.	1	4
CUP SERIAL NO.	1	3
ROLLER SIZE		
SHAFT O.D.	4.7523	4.7523
CONE I.D.	4.7479	4.7477
CONE/SHAFT FIT	.0044 -T	.0046 -T
CUP O.D.	7.5000	7.5003
HOUSING I.D.	7.4968	7.4973
CUP/HOUSING FIT	.0032 -T	.0030
CAGE SHAKE	.006"	.006"

BEARING ADJUSTMENT AIM 0 ACTUAL 0

MAGNETIC SEAL NO. 5

	MAGNET RING		SEAL CASE/CARBON INSERT	
after test WEIGHT	94.372		45.324	
WEIGHT	94.380	.008 GM LOSS	45.302	.022 GM GAIN
after WIDTH	.3774		.3360	
WIDTH	.3773"		.3362"	

61 HOURS 3 MINUTES

TABLE B-14. ADVANCED-TRANSMISSION COMPONENTS INVESTIGATION, BEARING/SHAFT SET 5

SPRING PRELØAD 2130 LBF

SHAFT FLØW DISTRIBUTIØN TEST 8.00 SLAVE 8

FLØATING CUP DIAMETRICAL FIT .0006 LØØSE

SLAVE BEARING SETTING 0.0000

		-----TEMPERATURES (F)-----						--ØIL FLØWS---			---HEAT--	
		ØIL	CUP	ØD	ØIL	ØUT	ØD	PT/MIN			BTU/MIN	
RPM	LØAD	IN	DE	ØDE	DE	ØDE	SEAL	SLV	ØD	SHFT	BRG	ØIL
3700	5205	189	206	199	207	200	224	8.0	0.0	32.0	276	292
3700	7810	190	207	199	207	200	227	8.0	0.0	32.0	293	272
3700	10410	189	209	198	209	200	225	8.0	0.0	32.0	301	313
3700	11710	189	209	198	209	199	227	8.0	0.0	32.0	305	303
5550	5205	191	220	210	219	211	230	8.0	0.0	32.0	528	486
5550	7810	190	220	209	219	210	228	8.0	0.0	32.0	557	496
5550	10410	188	221	208	220	209	228	8.0	0.0	32.0	575	536
5550	11710	189	223	209	221	210	228	8.0	0.0	32.0	576	537
7400	5205	191	231	220	230	223	233	8.0	0.0	32.0	829	721
7400	7810	189	231	219	229	222	231	8.0	0.0	32.0	874	741
7400	10410	190	232	220	232	222	232	8.0	0.0	32.0	891	752
7400	11710	188	231	219	231	222	230	8.0	0.0	32.0	901	782

TABLE B-15. ADVANCED-TRANSMISSION COMPONENTS INVESTIGATION, BEARING/SHAFT SET 51

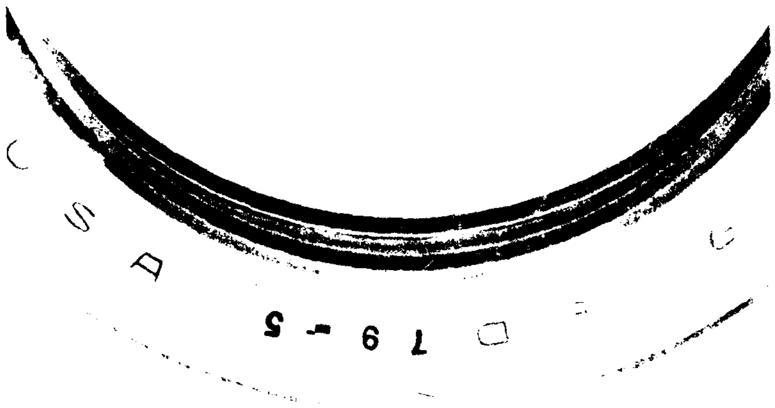
SPRING PRELØAD 2130 LBF

SHAFT FLØW DISTRIBUTIØN TEST 8.00 SLAVE 8

FLØATING CUP DIAMETRICAL FIT .0006 LØØSE

SLAVE BEARING SETTING -.0004

		-----TEMPERATURES (F)-----						--ØIL FLØWS--- PT/MIN			--HEAT-- BTU/MIN	
RPM	LØAD	ØIL IN	CUP DE	ØD ØDE	ØIL ØUT DE	ØDE	ØD SEAL	SLV	ØD	SHFT	BRG	ØIL
9600	5205	189	241	232	241	238	245	8.0	0.0	32.0	1249	1030
9600	7810	191	245	232	245	239	249	8.0	0.0	32.0	1281	1041
9600	10410	190	245	231	245	238	248	8.0	0.0	32.0	1319	1051
9600	11710	189	245	234	245	239	246	8.0	0.0	32.0	1325	1082
11800	5205	187	255	247	257	255	245	8.0	0.0	32.0	1700	1415
11800	7810	190	259	248	260	256	235	8.0	0.0	32.0	1734	1394
11800	10410	191	261	250	263	257	237	8.0	0.0	32.0	1755	1416
11800	11710	191	260	249	262	256	239	8.0	0.0	32.0	1775	1395
14000	5205	189	272	273	277	271	238	8.0	0.0	32.0	2203	1751
14000	7810	188	273	273	278	270	242	8.0	0.0	32.0	2229	1772
14000	10410	190	273	262	281	272	240	8.0	0.0	32.0	2239	1783
14000	11710	190	274	262	283	272	241	8.0	0.0	32.0	2238	1804



CUP 79-5 VIEWED FROM SMALL END

CUP 79-5 VIEWED FROM LARGE END

Figure B-20. Cup, Rollers, and Cage Used at Drive End of Shaft 78-4 in Test Setup No. 5 (Sheet 1 of 2).

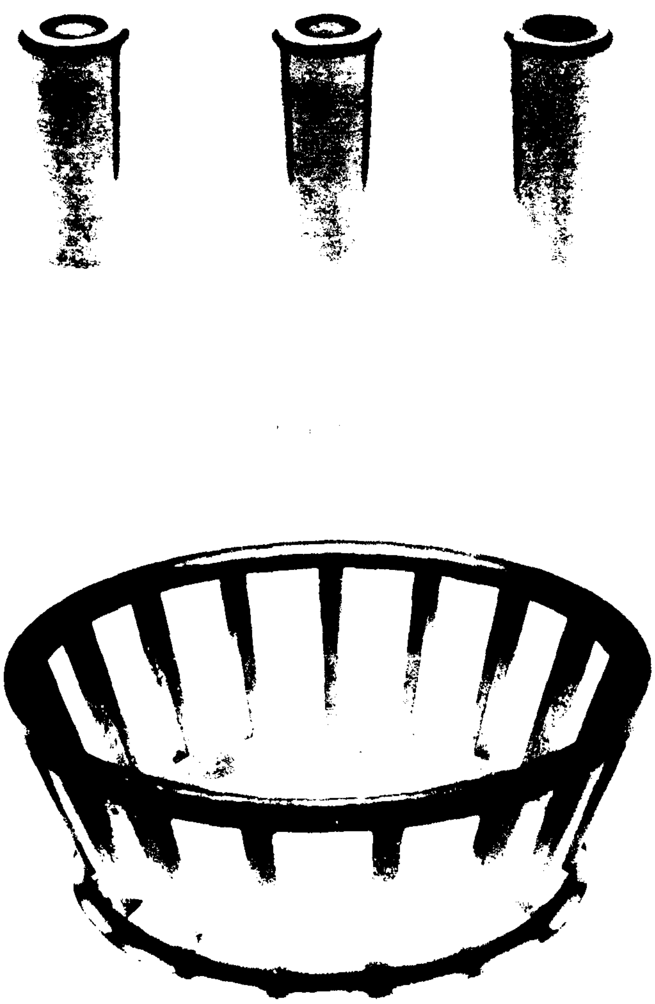
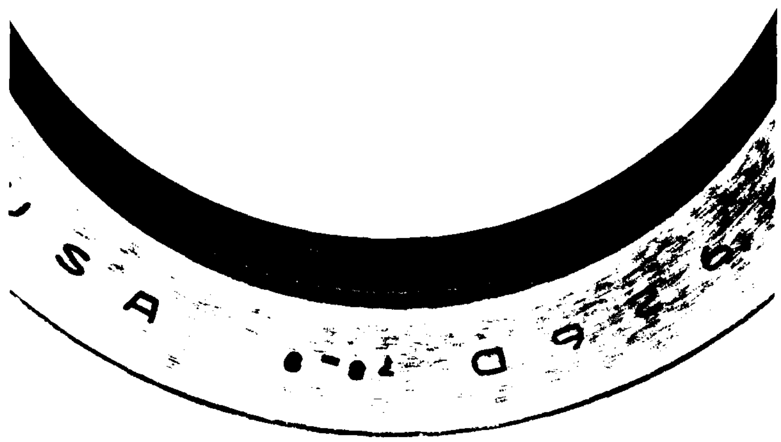


Figure B-20. Cup, Rollers, and Cage Used at Drive End of Shaft 78-4 in Test Setup No. 5 (Sheet 2 of 2).

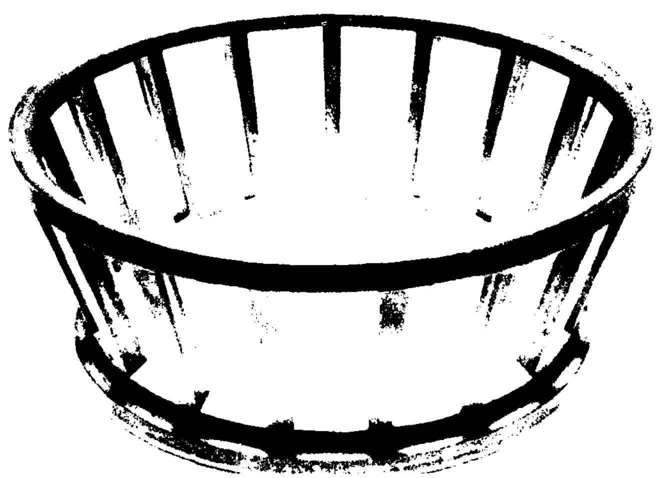


CUP 78-9 VIEWED FROM SMALL END

CUP 78-9 VIEWED FROM LARGE END

Figure B–21. Cup, Rollers, and Cage Used Opposite Drive End of Shaft 78-4 in Test Setup No. 5 (Sheet 1 of 2).

ROLLERS



CAGE 78-25

Figure B–21. Cup, Rollers, and Cage Used Opposite Drive End of Shaft 78-4 in Test Setup No. 5 (Sheet 2 of 2).

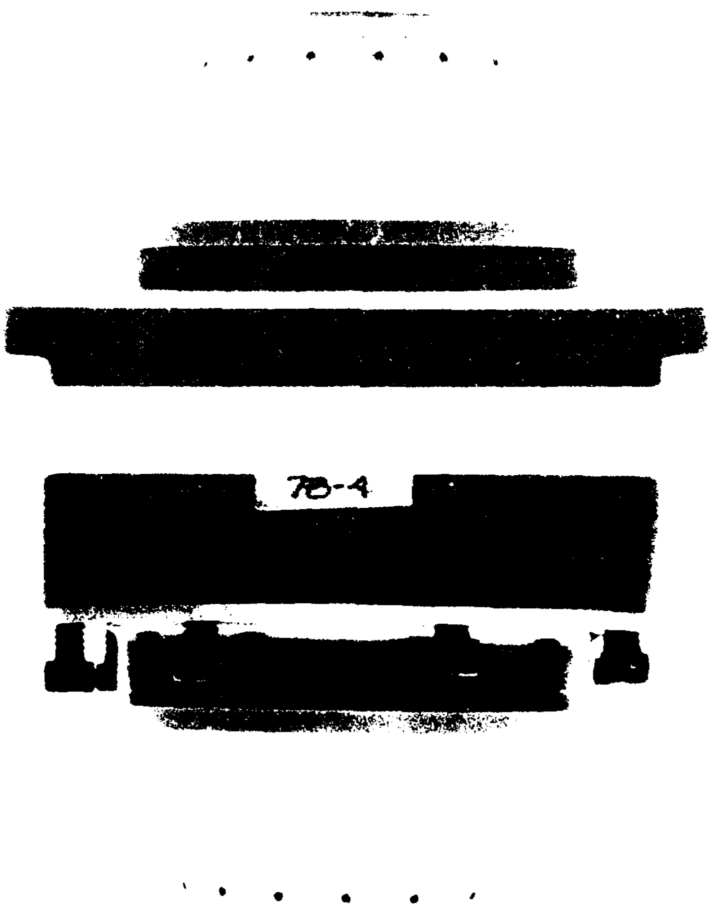


Figure B-22. Shaft No. 78-4 From Test Setup No. 5

TABLE B-16. TEST NO. 141.1-U - BUILDUP SHEET, SETUP NO. 6

TEST BEARINGS

	DRIVE END		OPPOSITE DRIVE END	
SHAFT SERIAL NO.	78-7		78-7	
CAGE SERIAL NO.	78-27		78-28	
CUP SERIAL NO.	79-8		79-2	
ROLLER SIZE	7		7 5/8	
CUP O.D.	5.0003		5.0008	
HOUSING I.D.	4.9989		5.0010	
CUP/HOUSING FIT	.0014" TIGHT		.0002" LOOSE	
	LARGE END	SMALL END	LARGE END	SMALL END
CUP PILOT I.D.	4.2774	3.6192	4.2761	3.6190
CAGE O.D.	4.2700	3.6118	4.2698	3.6108
CUP/CAGE CLEARANCE	.0074	.0074	.0063	.0082
RADIAL CAGE GROWTH DUE TO ROTATION (19000 RPM)	.00016	.000079	.00016	.000079
RADIAL CUP DEFORMATION DUE TO FIT	.00106	.00053	—	—
RUNNING CUP/CAGE CLEARANCE	.00618	.0065	.0061	.0081

BELLEVILLE LOADING SPRINGS USED - SET NO. 1 A-43677
FREE HEIGHT .5171
SPRING DEFLECTION .0411
PRELOAD 2964 POUNDS

SLAVE BEARINGS

	DRIVE END CENTER	OPPOSITE DRIVE END CENTER
CONE SERIAL NO.		
CUP SERIAL NO.		
ROLLER SIZE		
SHAFT O.D.		
CONE I.D.		
CONE/SHAFT FIT		
CUP O.D.		
HOUSING I.D.		
CUP/HOUSING FIT		
CAGE SHAKE		
BEARING ADJUSTMENT AIM ______	ACTUAL ______	

MAGNETIC SEAL NO. 6

	MAGNET RING	SEAL CASE/CARBON INSERT
WEIGHT	BEFORE 99.993 GM AFTER 99.992 .001GM	95925 GM 959254 -.0004 GM
WIDTH	.3784" .3784 .0000"	.3362" .3362 .0000"

TOTAL HOURS
101 HOURS 15 MINUTES

TABLE B-17. ADVANCED-TRANSMISSION COMPONENTS INVESTIGATION, BEARING/SHAFT SET 6

SHAFT FLØW DISTRIBUTIØN TEST 8.00 SLAVE 0

FLØATING CUP DIAMETRICAL FIT .0002 LØØSE

		-----TEMPERATURES (F)-----						--ØIL FLØWS--- PT/MIN			---HEAT-- BTU/MIN	
RPM	SPRING LØAD	ØIL IN	CUP DE	ØD ØDE	ØIL DE	ØUT ØDE	ØD SEAL	SLV	ØD	SHFT	BRG	ØIL
3700	2964	184	196	196	196	196	220			16.0	84	97
5550	2964	189	211	211	211	211	237			16.0	145	178
7400	2964	189	221	221	221	223	227			16.0	216	268
9600	2964	191	240	238	238	240	238			16.0	300	391
11800	2964	187	250	250	250	252	232			16.0	404	524
14000	2964	187	261	261	263	266	234			16.0	505	637
3700	6418	190	207	203	201	205	235			16.0	100	97
5550	6418	190	221	213	215	214	233			16.0	177	198
7400	6418	190	233	224	228	223	237			16.0	263	288
9600	6418	190	250	240	244	239	247			16.0	367	420
11800	6418	190	263	251	258	253	240			16.0	486	537
14000	6418	188	280	266	274	269	243			16.0	598	688

TABLE B-18. ADVANCED-TRANSMISSION COMPONENTS INVESTIGATION, BEARING SHAFT SET 61

SHAFT FLØW DISTRIBUTIØN TEST 4.00 SLAVE 0

FLØATING CUP DIAMETRICAL FIT .0002 LØØSE

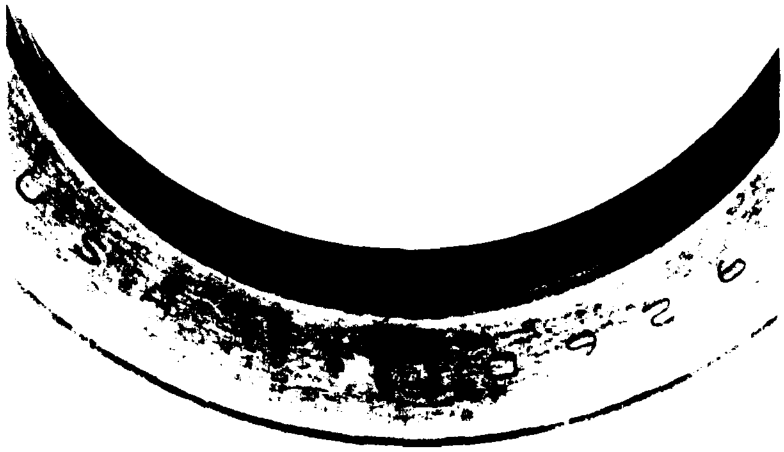
		-----TEMPERATURES (F)-----						--ØIL FLØWS--- PT/MIN			---HEAT-- BTU/MIN	
RPM	SPRING LØAD	ØIL IN	CUP DE	ØD ØDE	ØIL DE	ØUT ØDE	ØD SEAL	SLV	ØD	SHFT	BRG	ØIL
3700	2964	185	205	201	201	201	228			8.0	79	64
5550	2964	188	226	221	221	222	231			8.0	132	136
7400	2964	187	243	236	236	238	241			8.0	192	204
9600	2964	189	267	258	258	259	255			8.0	264	285
11800	2964	188	287	275	275	275	258			8.0	347	359
14000	2964	190	312	298	298	294	253			8.0	421	439
3700	6418	189	211	209	209	208	234			8.0	94	79
5550	6418	189	234	225	228	226	245			8.0	160	154
7400	6418	189	250	242	245	243	254			8.0	230	225
9600	6418	189	275	265	266	265	265			8.0	317	314
11800	6418	190	294	285	285	284	258			8.0	409	390
14000	6418	191	319	307	307	305	262			8.0	496	478

TABLE B-19. ADVANCED-TRANSMISSION COMPONENTS INVESTIGATION, BEARING/SHAFT SET 62

SHAFT FLOW DISTRIBUTION TEST 1.00 SLAVE 0

FLOATING CUP DIAMETRICAL FIT .0002 LOOSE

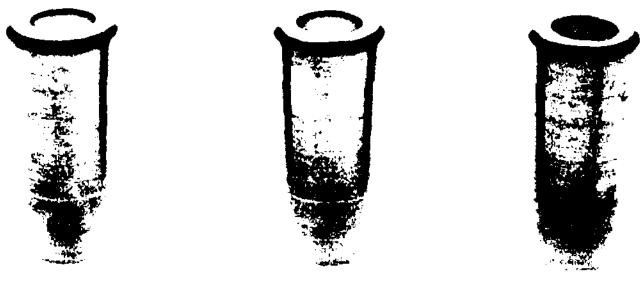
		-----TEMPERATURES (F)-----						--OIL FLOWS---			---HEAT--	
	SPRING	OIL	CUP	OD	OIL	OUT	OD	PT/MIN			BTU/MIN	
RPM	LOAD	IN	DE	ODE	DE	ODE	SEAL	SLV	OD	SHFT	BRG	OIL
3700	2964	178	218	227	207	212	235			2.0	73	32
5550	2964	179	255	263	226	245	258			2.0	119	58
7400	2964	180	290	299	255	278	281			2.0	160	89
9600	2964	188	322	339	288	317	301			2.0	211	119
11800	2964	192	367	385	331	365	323			2.0	249	164
3700	6418	174	223	231	211	217	239			2.0	89	41
5505	6418	185	269	271	245	257	276			2.0	134	68
7400	6418	186	297	308	263	285	283			2.0	193	91
9600	6418	187	337	356	302	333	305			2.0	246	136
11800	6418	189	368	386	333	367	313			2.0	307	169



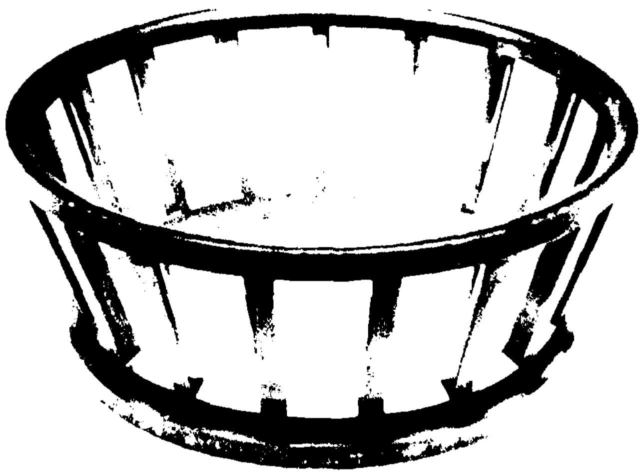
CUP 79-8 VIEWED FROM SMALL END

CUP 79-8 VIEWED FROM LARGE END

Figure B–23. Cup, Rollers, and Cage Used at Drive End of Shaft 78-7 in Test Setup No. 6 (Sheet 1 of 2).



ROLLERS



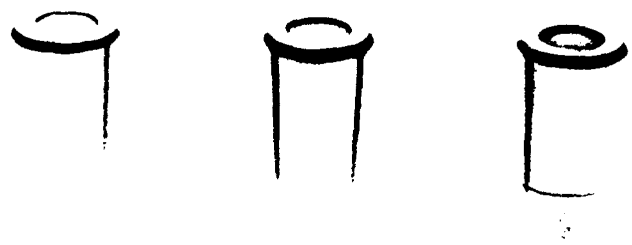
CAGE 78-27

Figure B 23. Cup, Rollers, and Cage Used at Drive End of Shafts 78-7 in Test Setup No. 6 (Sheet 2 of 2).

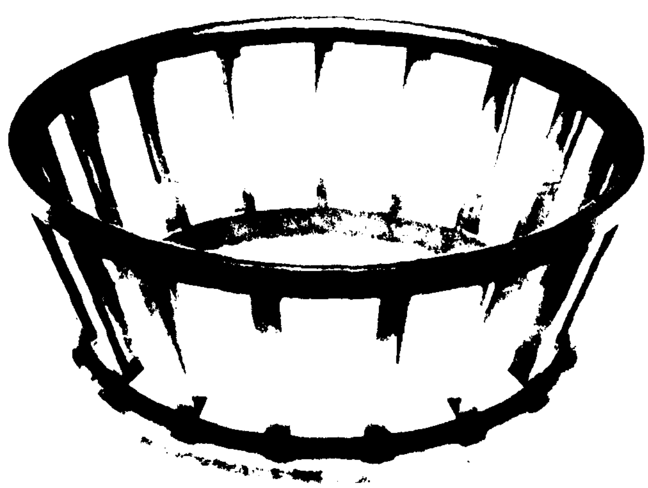
CUP 79-2 VIEWED FROM SMALL END

CUP 79-2 VIEWED FROM LARGE END

Figure B–24. Cup, Rollers, and Cage Used Opposite Drive End of Shaft 78-7 in Test Setup No. 6 (Sheet 1 of 2).



ROLLERS



CAGE 78 28

Figure B 24. Cup, Rollers, and Cage Used Opposite Drive End of *Shaft* 78-7 *in Test* Setup No. 6 (Sheet 2 of 2).

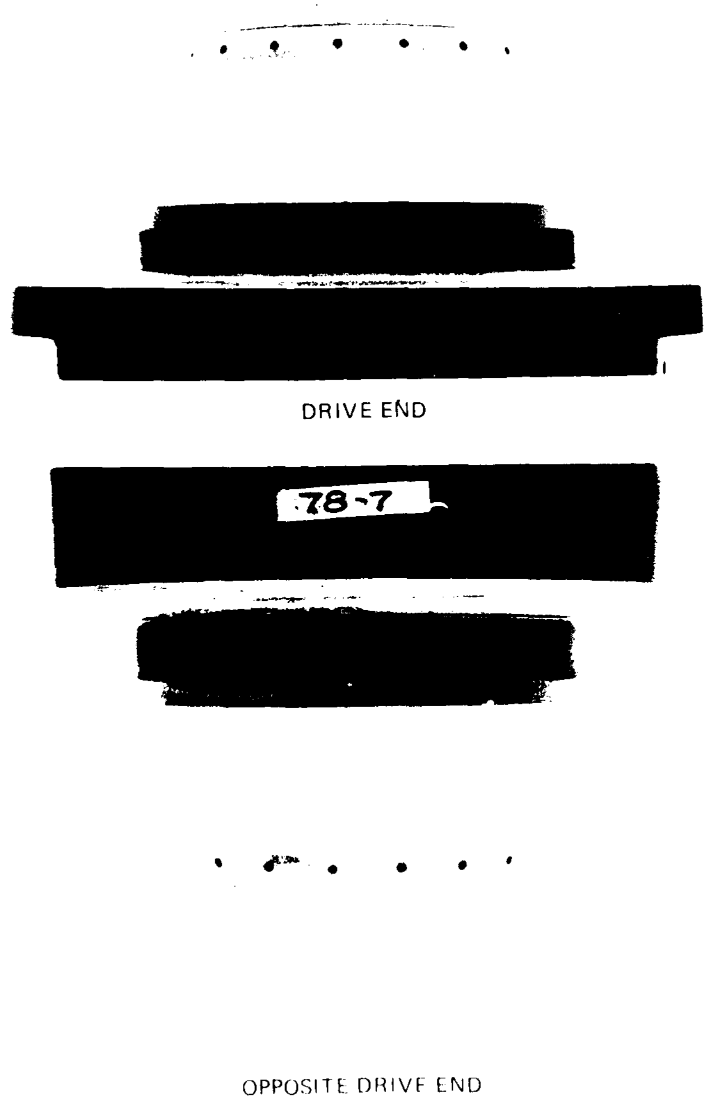
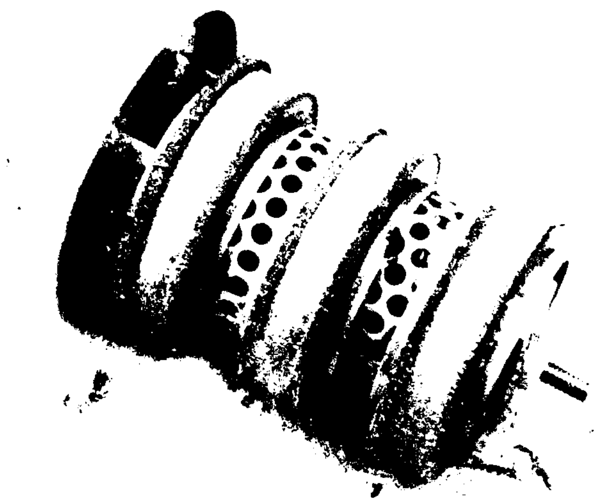


Figure B-25. Shaft No. 78-7 From Test Setup No. 6.

BOTTOM OF OIL RESERVOIR



MAGNETIC STRAINER FROM OIL RESERVOIR

Figure B-26. Condition of Components After Running Test Setups No. 1 Through 6 and Before Running the 150-Hour Endurance Test Setup No. 8.

TABLE B-20. TEST NO. 141.1-U - BUILDUP SHEET,

SETUP NO. 7

TEST BEARINGS

	DRIVE END		OPPOSITE DRIVE END	
SHAFT SERIAL NO.	[illegible]		[illegible]	
CAGE SERIAL NO.	78-31		78-32	
CUP SERIAL NO.	79-10		79-[illegible]	
ROLLER SIZE	[illegible]8		[illegible]	
CUP O.D.	5.0007		5.0005	
HOUSING I.D.	4.9989		[illegible]	
CUP/HOUSING FIT	.0018 T		.0005 -L	
	LARGE END	SMALL END	LARGE END	SMALL END
CUP PILOT I.D.	4.2764	3.6187	4.2782	3.6[illegible]
CAGE O.D.	4.27[illegible]	3.6[illegible]5	4.2725	3.6[illegible]
CUP/CAGE CLEARANCE	.0047	.0072	.0057	[illegible]
RADIAL CAGE GROWTH DUE TO ROTATION (19000 RPM)	.00015	.000079	.00016	.000079
RADIAL CUP DEFORMATION DUE TO FIT	.00136	.00106	—	—
RUNNING CUP/CAGE CLEARANCE	.0032	.0061	.0055	.0079

BELLEVILLE LOADING SPRINGS USED - SET NO. 2

FREE HEIGHT .477

SPRING DEFLECTION .112

PRELOAD 2115 POUNDS

SLAVE BEARINGS

	DRIVE END CENTER	OPPOSITE DRIVE END CENTER
CONE SERIAL NO.	1	4
CUP SERIAL NO.		
ROLLER SIZE	[illegible]8	[illegible]8
SHAFT O.D.	[illegible]3	[illegible]
CONE I.D.	4.7479	[illegible]
CONE/SHAFT FIT	[illegible]	[illegible]
CUP O.D.	7.5000	7.5003
HOUSING I.D.	[illegible]	[illegible]3
CUP/HOUSING FIT	.0032	[illegible]
CAGE SHAKE	006 007 008	005 006 006
BEARING ADJUSTMENT AIM .0005" PRELOAD ACTUAL .0006		

MAGNETIC SEAL NO. ________

	MAGNET RING	SEAL CASE/CARBON INSERT
WEIGHT	94.386 94.372	45.0648 45.032[illegible]
WIDTH	.3761 .3760	.335[illegible] .337[illegible]

TOTAL 7.5 HOURS

3.017 GRAM SEAL LEAKAGE

TABLE B-21. ADVANCED-TRANSMISSION COMPONENTS INVESTIGATION, BEARING/SHAFT SET 7

SHAFT FLØW DISTRIBUTIØN TEST 4.00 SLAVE 8

FLØATING CUP DIAMETRICAL FIT .0005 LØØSE

SLAVE BEARING SETTING -.0005

		-----TEMPERATURES (F)-----						--ØIL FLØWS---			---HEAT--	
		ØIL	CUP	ØD	ØIL	ØUT	ØD	PT/MIN			BTU/MIN	
RPM	LØAD	IN	DE	ØDE	DE	ØDE	SEAL	SLV	ØD	SHFT	BRG	ØIL
3700	5202	231	241	239	241	239	256	15.2	0.0	24.0	218	179
3700	7810	242	253	250	253	249	265	15.4	0.0	24.0	217	180
3700	10410	245	256	252	255	252	266	15.4	0.0	24.0	222	170
3700	11710	242	253	249	253	249	262	15.4	0.0	24.0	227	180
5550	5205	295	308	308	308	307	311	15.6	0.0	24.0	325	252
5550	7810	293	307	305	307	305	309	15.5	0.0	24.0	343	262
5550	10410	299	313	310	313	310	313	15.5	0.0	24.0	347	252
5550	11710	300	315	311	315	311	314	15.5	0.0	24.0	347	262
7400	5205	299	323	322	322	319	321	15.5	0.0	24.0	524	434
7400	7810	298	324	321	321	318	322	15.6	0.0	24.0	550	435
7400	10410	300	324	320	324	320	324	15.7	0.0	24.0	561	447
7400	11710	299	325	320	324	319	303	15.7	0.0	24.0	566	457
9600	5205	298	333	335	331	332	330	15.7	0.0	24.0	822	682
9600	7810	297	331	333	333	333	324	15.7	0.0	24.0	850	733
9600	10410	298	334	334	334	333	326	15.7	0.0	24.0	871	723
9600	11710	298	334	333	335	333	325	15.7	0.0	24.0	875	733
11800	5205	299	347	351	345	349	326	15.7	0.0	24.0	1165	980
11800	7810	300	349	351	348	350	327	15.7	0.0	24.0	1192	1001
11800	10410	299	351	351	351	351	329	15.7	0.0	24.0	1210	1063
11800	11710	300	354	354	352	352	333	15.7	0.0	24.0	1214	1063
14000	5205	302	369	370	366	366	331	15.7	0.0	24.0	1554	1312
14000	7810	300	370	369	368	367	341	15.7	0.0	24.0	1565	1384
14000	10410	300	371	368	368	366	331	15.7	0.0	24.0	1593	1374
14000	11710	299	373	366	369	376	333	15.7	0.0	24.0	1577	1510

CUP 79-10 VIEWED FROM SMALL END

CUP 79-10 VIEWED FROM LARGE END

Figure B–27. Cup, Rollers, and Cage Used at Drive End of Shaft 78-9 in Test Setup No. 7 (Sheet 1 of 2).

ROLLERS

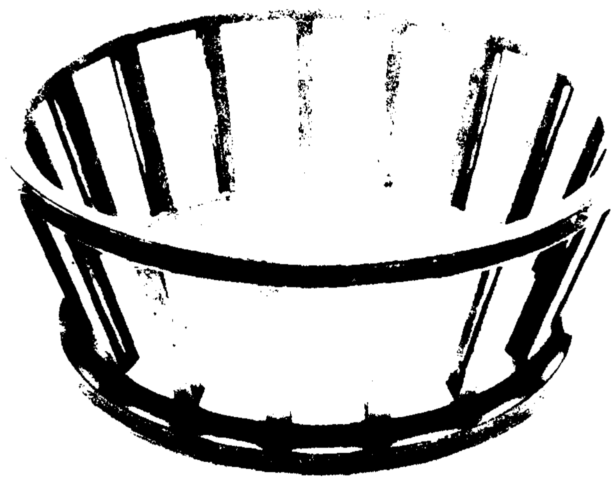
CAGE 78-31

Figure B-27. Cup, Rollers, and Cage Used at Drive End of Shaft 78-9 in Test Setup No. 7 (Sheet 2 of 2).

CUP 78-16 VIEWED FROM LARGE END

Figure B-28. Cup, Rollers, and Cage Used Opposite Drive End of Shaft 78-9 in Test Setup No. 7 (Sheet 1 of 2).

ROLLERS



CAGE 78-32

Figure B–28. Cup, Rollers, and Cage Used Opposite Drive End of Shaft 78-9 in Test Setup No. 7 (Sheet 2 of 2).

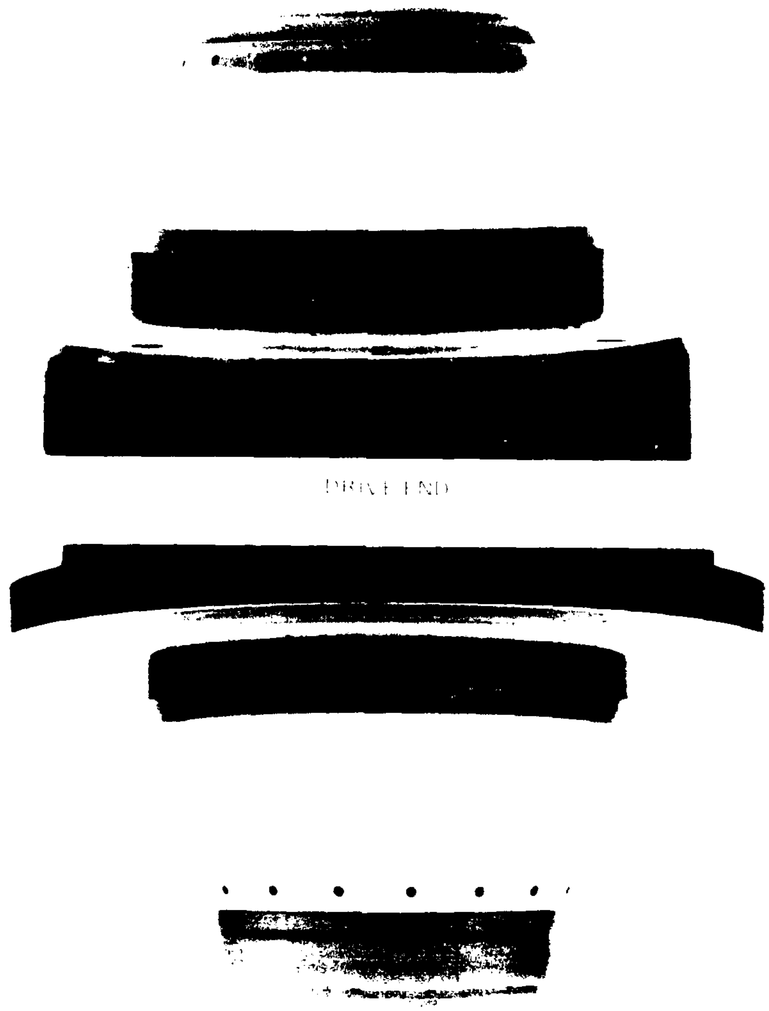


Figure B-29. Shaft No. 78-9 From Test Setup No. 7

TABLE B-22. TEST NO. 141.1-U - BUILDUP SHEET, SETUP NO. 8

TEST BEARINGS 150 HOUR ENDURANCE TEST

	DRIVE END		OPPOSITE DRIVE END	
SHAFT SERIAL NO.	78-6		78-6	
CAGE SERIAL NO	78-26		78-29	
CUP SERIAL NO.	79-9		79-14	
ROLLER SIZE	7-18		7-38	
CUP O.D.	5.0001"		5.0005"	
HOUSING I.D.	4.9989"		5.0010"	
CUP/HOUSING FIT	.0012-T		.0005-L	
	LARGE END	SMALL END	LARGE END	SMALL END
CUP PILOT I.D.	4.2761	3.6187	4.2773	3.6191
CAGE O.D.	4.2697	3.6074	4.2730	3.6116
CUP/CAGE CLEARANCE	.0064	.0113	.0043	.0075
RADIAL CAGE GROWTH DUE TO ROTATION (18,000 RPM)	.00016	.000079	.00016	.000079
RADIAL CUP DEFORMATION DUE TO FIT	.0004	.0009	—	—
RUNNING CUP/CAGE CLEARANCE	.0058	.0103	.0041	.0074

BELLEVILLE LOADING SPRINGS USED - SET NO. 2 A-43573

FREE HEIGHT .407

SPRING DEFLECTION .112 .123

PRELOAD 2115 POUNDS

SLAVE BEARINGS

	DRIVE END CENTER	OPPOSITE DRIVE END CENTER
CONE SERIAL NO.	1	4
CUP SERIAL NO.		
ROLLER SIZE		
SHAFT O.D.	4.75235"	4.75235
CONE I.D.	4.74790	4.74770
CONE/SHAFT FIT	.00445	.00465
CUP O.D.	7.5000	7.5003
HOUSING I.D.	7.4968"	7.4973"
CUP/HOUSING FIT	.0032"	.0030"
CAGE SHAKE	005-005-006	009-007-004

BEARING ADJUSTMENT AIM .0005 PRELOAD ACTUAL .0005 PRELOAD

MAGNETIC SEAL NO. 8

	MAGNET RING	SEAL CASE/CARBON INSERT
WEIGHT	94.832 Gram	452.766 Gram
WIDTH	.3791	.3362
	94.833	452.7[illegible]
	.3788	

TOTAL TEST TIME 379 HOURS

5.2 GRAM SEAL LEAKAGE

TABLE B-23. ADVANCED-TRANSMISSION COMPONENTS INVESTIGATION, BEARING/SHAFT SET 8

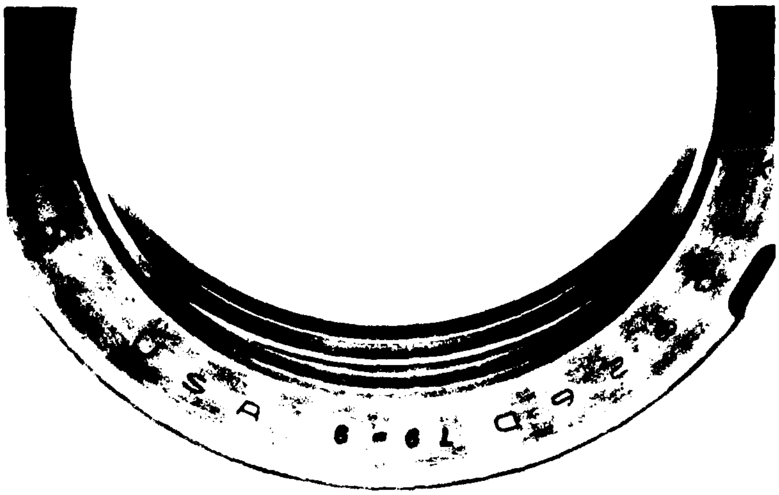
SPRING PRELØAD 2115 LBF

SHAFT FLØW DISTRIBUTIØN TEST 2.00 SLAVE 4

FLØATING CUP DIAMETRICAL FIT .0005 LØØSE

SLAVE BEARING SETTING -.0005

		-----TEMPERATURES	(F)-----					--ØIL	FLØWS---		---HEAT--		
		ØIL	CUP	ØD	ØIL	ØUT	ØD		PT/MIN		BTU/MIN		
RPM	LØAD	IN	DE	ØDE	DE	ØDE	SEAL	SLV	ØD	SHFT	BRG	ØIL	HOURS
7400	11710	190	270	250	261	245	256	8.0	0.0	12.0	746	645	5
7400	11710	189	267	252	260	241	251	8.0	0.0	12.0	756	629	16
7400	11710	192	269	253	265	247	257	8.0	0.0	12.0	734	656	27
7400	11710	191	269	250	264	245	258	8.0	0.0	12.0	739	651	49
7400	11710	193	271	257	266	248	261	8.0	0.0	12.0	730	656	59
7400	11710	191	270	255	265	247	266	8.0	0.0	12.0	734	666	80
7400	11710	191	270	255	264	247	269	8.0	0.0	12.0	736	661	99
7400	11710	191	270	255	264	247	267	8.0	0.0	12.0	736	661	123
7400	11710	193	271	257	267	249	265	8.0	0.0	12.0	726	666	147
7400	11710	193	271	257	266	249	266	8.0	0.0	12.0	728	661	150
7400	11710	192	267	252	266	248	266	8.0	0.0	12.0	730	666	170
7400	11710	191	265	250	263	245	268	8.0	0.0	12.0	742	645	188
7400	11710	193	266	251	265	246	260	8.0	0.0	12.0	735	640	209
7400	11710	187	262	246	261	244	254	8.0	0.0	12.0	748	671	224
7400	11710	189	263	247	262	245	251	8.0	0.0	12.0	744	661	240
7400	11710	192	265	249	264	246	252	8.0	0.0	12.0	738	645	264
7400	11710	191	263	249	263	245	252	8.0	0.0	12.0	742	645	272
7400	11710	191	264	248	264	245	251	8.0	0.0	12.0	739	651	296
7400	11710	193	267	250	267	249	259	8.0	0.0	12.0	726	666	308
7400	11710	188	265	247	264	245	257	8.0	0.0	12.0	739	682	324
7400	11710	191	265	250	265	247	252	8.0	0.0	12.0	734	666	348
7400	11710	191	265	249	265	247	251	8.0	0.0	12.0	734	666	356
7400	11710	190	264	249	264	245	249	8.0	0.0	12.0	739	661	372
7400	11710	191	263	249	264	246	249	8.0	0.0	12.0	738	656	379



CUP 79-9 VIEWED FROM SMALL END

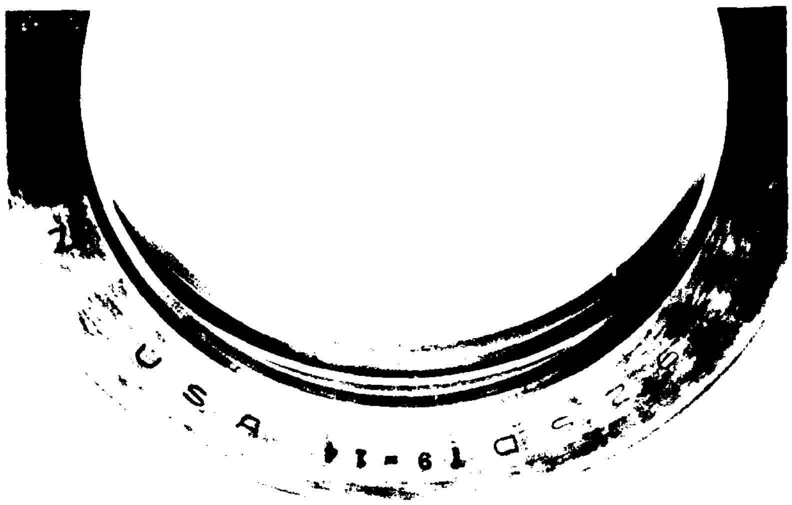
CUP 79-9 VIEWED FROM LARGE END

Figure B–30. Cup, Rollers, and Cage Used at Drive End of Shaft 78-6 in Test Setup No. 8 (Sheet 1 of 2).

ROLLERS

CAGE 78-26

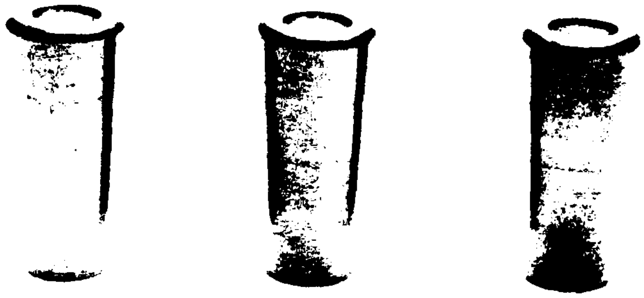
Figure B-30. Cup, Rollers, and Cage Used at Drive End of Shaft 78-6 in Test Setup No. 8 (Sheet 2 of 2).



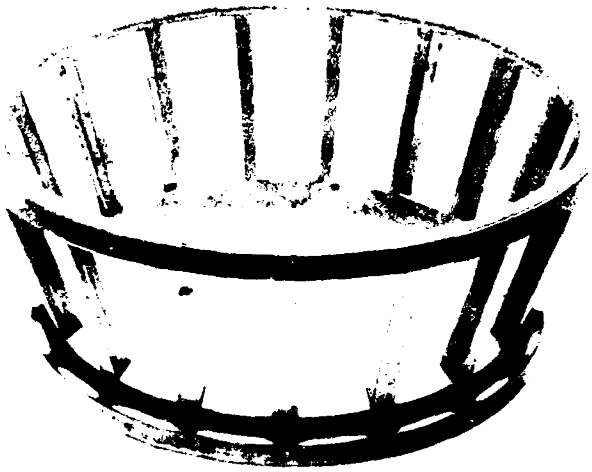
CUP 79-14 VIEWED FROM SMALL END

CUP 79-14 VIEWED FROM LARGE END

Figure B-31. Cup, Rollers, and Cage Used Opposite Drive End of Shaft 78-6 in Test Setup No. 8 (Sheet 1 of 2).



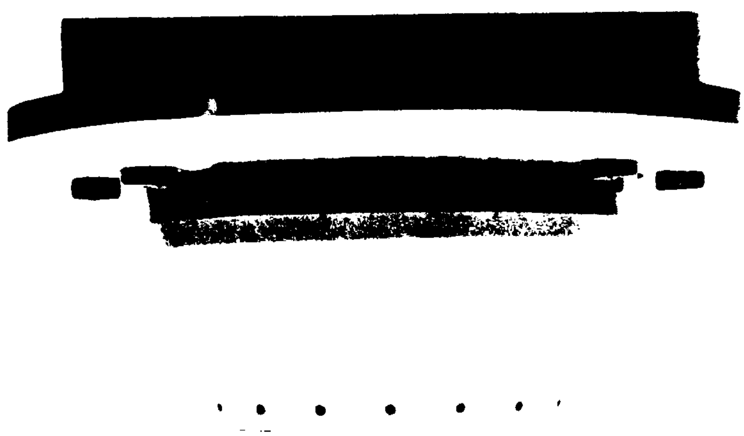
ROLLERS



CAGE 78-29

Figure B-31. Cup, Rollers, and Cage Used Opposite Drive End of Shaft 78-6 in Test Setup No. 8 (Sheet 2 of 2).

DRIVE END



OPPOSITE DRIVE END

Figure B-32. Shaft No. 78-6 From Test Setup No. 8

APPENDIX C

RIBBED-CUP TAPERED-ROLLER BEARING AND MAGNETIC SEAL OIL-OFF SURVIVABILITY TEST DATA

Appendix C contains the ribbed-cup tapered-roller bearing and magnetic seal oil-off survivability test data. Included in this appendix are the buildup sheets for each test, test data recorded at each data point, plots of temperature versus time, and photographs of the components after test.

TABLE C-1. OIL-OFF TEST NO. 1

TEST NO. 141.1-U - BUILD-UP SHEET

SET-UP NO. 6

TEST BEARINGS

	DRIVE END		OPPOSITE DRIVE END	
SHAFT SERIAL NO.	78-7		78-7	
CAGE SERIAL NO.	78-27		78-28	
CUP SERIAL NO.	79-8		79-2	
ROLLER SIZE	7		7 3/8	
CUP O.D.	5.0003		5.0008	
HOUSING I.D.	4.9989		5.0010	
CUP/HOUSING FIT	.0014" TIGHT		.0002" LOOSE	
	LARGE END	SMALL END	LARGE END	SMALL END
CUP PILOT I.D.	4.2774	3.6192	4.2761	3.6190
CAGE O.D.	4.2700	3.6118	4.2698	3.6108
CUP/CAGE CLEARANCE	.0074	.0074	.0063	.0082
RADIAL CAGE GROWTH DUE TO ROTATION (14000 RPM)	.00016	.000079	.00016	.000079
RADIAL CUP DEFORMATION DUE TO FIT	.00106	.00083	—	—
RUNNING CUP/CAGE CLEARANCE	.00618	.0065	.0061	.0081

BELLEVILLE LOADING SPRINGS USED - SET NO. 1 A-43677

FREE HEIGHT .5171

SPRING DEFLECTION ~~.040~~ ~~.0345~~ - ACTUAL .043"

PRELOAD ~~2964~~ **POUNDS** 3209

SLAVE BEARINGS

	DRIVE END CENTER	OPPOSITE DRIVE END CENTER
CONE SERIAL NO.		
CUP SERIAL NO.		
ROLLER SIZE		
SHAFT O.D.		
CONE I.D.		
CONE/SHAFT FIT		
CUP O.D.		
HOUSING I.D.		
CUP/HOUSING FIT		
CAGE SHAKE		

BEARING ADJUSTMENT AIM ______ **ACTUAL** ______

MAGNETIC SEAL NO. 6

	MAGNET RING	SEAL CASE/CARBON INSERT
WEIGHT		
WIDTH		

OIL-OFF = 1 MIN. 24 SEC. @ 3700 RPM

MAX TEMPS. DE CUP OD = 248°F
ODE CUP OD = 314°F
SEAL = 248°F

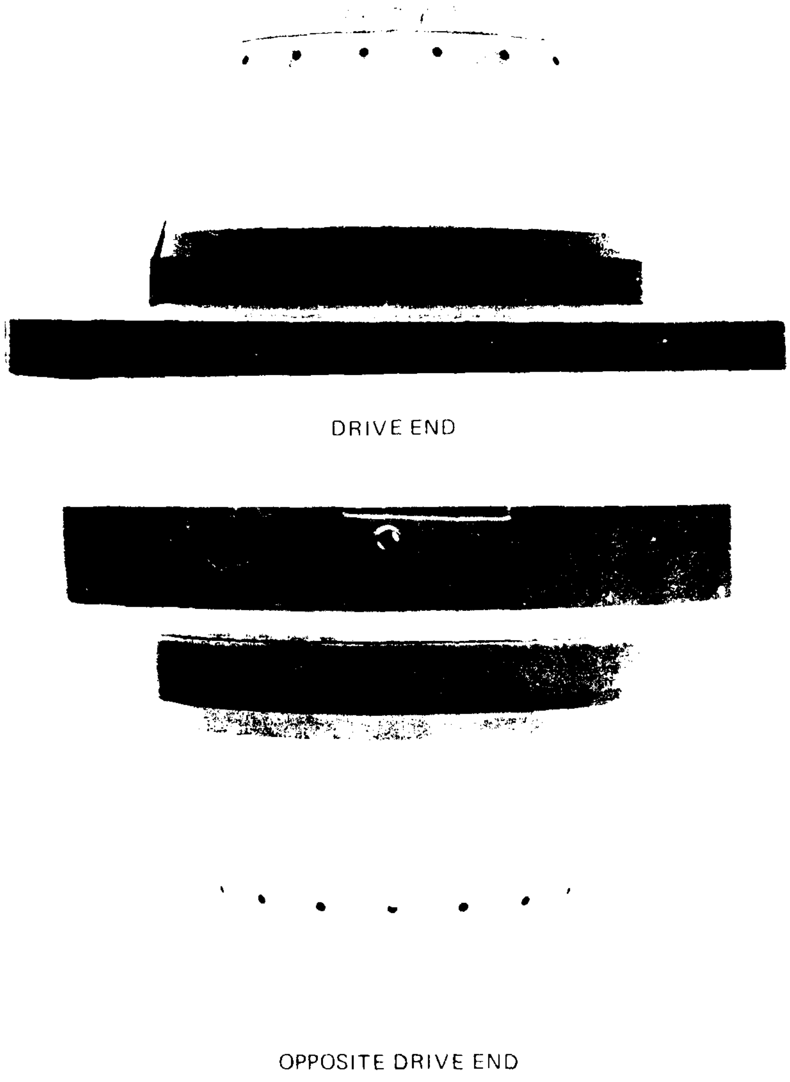


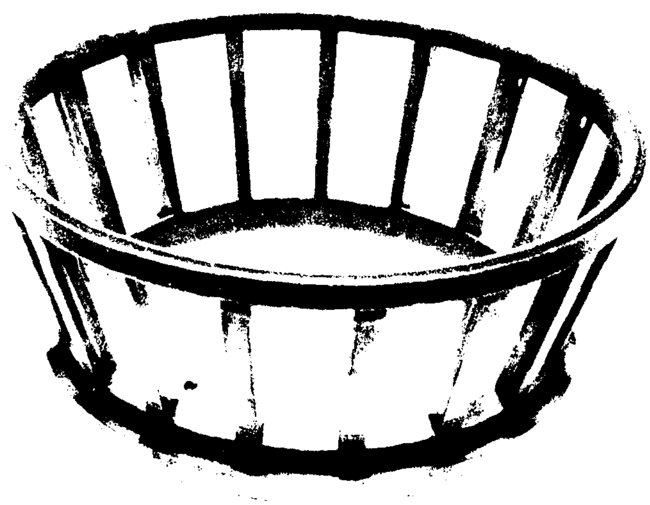
Figure C-1. Shaft No. 78-7 From Oil-Off Test No. 1

CUP 79-8 VIEWED FROM SMALL END

CUP 79-8 VIEWED FROM LARGE END

Figure C-2. Cup, Rollers, and Cage Used at Drive End of Shaft 78-7 in Oil-Off Test No. 1 (Sheet 1 of 2).

ROLLERS



CAGE 78-27

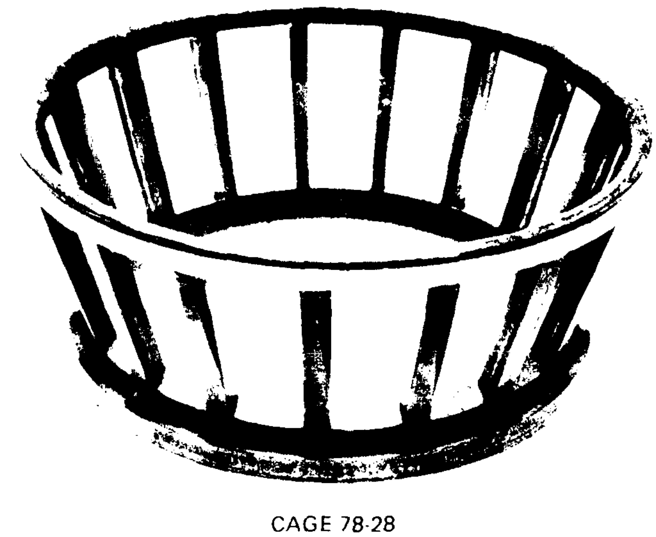
Figure C-2. Cup, Rollers, and Cage Used at Drive End of Shaft 78-7 in Oil-Off Test No. 1 (Sheet 2 of 2).

CUP 79 2 VIEWED FROM SMALL END

CUP 79 2 VIEWED FROM LARGE END

Figure C 3. Cup, Rollers, and Cage Used Opposite Drive End of Shaft 78-7 in Oil-Off Test No. 1 (Sheet 1 of 2).

ROLLERS



CAGE 78-28

Figure C 3. Cup, Rollers, and Cage Used Opposite Drive End of Shaft 78-7 in Oil-Off Test No. 1 (Sheet 2 of 2).

TABLE C-2. OIL-OFF TEST NO. 2

TEST NO. 141.1-U - BUILD-UP SHEET

SET-UP NO. 2

TEST BEARINGS

	DRIVE END		OPPOSITE DRIVE END	
SHAFT SERIAL NO.	78-1		78-1	
CAGE SERIAL NO.	78-36		78-15	
CUP SERIAL NO.	78-3		78-2	
ROLLER SIZE	7 3/8		7 4/8	
CUP O.D.	5.0006		5.0007	
HOUSING I.D.	4.9989		5.0038	
CUP/HOUSING FIT	.0017		.0031	
	LARGE END	SMALL END	LARGE END	SMALL END
CUP PILOT I.D.	4.2772	3.6191	4.2780	3.6190
CAGE O.D.	4.2682	3.6119	4.2715	3.6120
CUP/CAGE CLEARANCE	.0090	.0072	.0065	.0070
RADIAL CAGE GROWTH DUE TO ROTATION (7400 RPM) ×2	.000045	.000022	.000045	.000022
RADIAL CUP DEFORMATION DUE TO FIT	.0013	.0010	—	—
RUNNING CUP/CAGE CLEARANCE	.0077	.0062	.0065	.007

BELLEVILLE LOADING SPRINGS USED - SET NO. 4
FREE HEIGHT .5335
SPRING DEFLECTION .1953
PRELOAD 3200 POUNDS

SLAVE BEARINGS

	DRIVE END CENTER	OPPOSITE DRIVE END CENTER
CONE SERIAL NO.		
CUP SERIAL NO.		
ROLLER SIZE		
SHAFT O.D.		
CONE I.D.		
CONE/SHAFT FIT		
CUP O.D.		
HOUSING I.D.		
CUP/HOUSING FIT		
CAGE SHAKE		

BEARING ADJUSTMENT AIM ______ ACTUAL ______

MAGNETIC SEAL NO. 1

	MAGNET RING	SEAL CASE/CARBON INSERT
WEIGHT		
WIDTH		

OIL-OFF 7 MIN. 48 SEC.
@ 3700 RPM

TABLE C-3. TEST NO. 2, OIL-OFF SURVIVABILITY

OIL-OFF SURVIVABILITY

SPEED 3700 RPM THRUST LOAD 3200 LBF

INITIAL CUP FITS DE .0016 ODE -.0031

			****** TEMPERATURE (F) *****									
			------DE-----			-----ODE-----			-----FITS-----			
SEC	REF	HSG	CUP	RLR	C/S	CUP	RLR	C/S	DE	ODE	AXL	LBF
0	70	181	196	196	197	198	198	197	.0048	.0001	.0126	3205
5	70	180	197	196	197	198	199	198	.0048	.0001	.0126	3205
11	70	181	197	196	197	198	200	198	.0048	.0001	.0127	3206
16	70	181	202	207	200	199	201	198	.0048	.0001	.0130	3208
21	70	181	211	226	207	200	203	199	.0049	.0001	.0136	3212
26	70	180	220	243	212	200	203	199	.0050	.0001	.0141	3215
31	70	180	232	268	220	201	205	200	.0051	.0001	.0149	3219
37	70	180	234	271	221	202	206	200	.0051	.0001	.0150	3220
42	70	180	232	267	220	202	208	200	.0051	.0001	.0150	3220
47	70	180	231	265	219	203	209	201	.0051	.0001	.0149	3220
52	70	180	231	264	219	204	210	201	.0051	.0001	.0149	3220
57	70	180	230	263	219	204	212	202	.0051	.0001	.0150	3220
62	70	180	230	263	219	205	213	202	.0051	.0002	.0150	3220
67	70	179	230	263	219	205	213	202	.0051	.0002	.0150	3220
73	70	179	230	263	219	206	214	203	.0051	.0002	.0150	3220
78	70	179	229	262	218	206	215	203	.0051	.0002	.0150	3220
83	70	179	229	262	218	207	217	203	.0051	.0002	.0151	3221
88	70	179	230	263	219	207	217	204	.0051	.0002	.0151	3221
93	70	179	230	263	219	208	218	204	.0051	.0002	.0152	3221
99	70	179	230	263	219	208	219	204	.0051	.0002	.0152	3221
104	70	179	230	262	219	208	220	205	.0051	.0002	.0152	3221
109	70	179	229	262	218	209	221	205	.0051	.0002	.0152	3221

TABLE C-3 - Continued

114	70	179	229	261	218	209	221	205	.0051	.0002	.0152	3221
119	70	179	229	261	218	209	222	205	.0051	.0002	.0152	3222
124	70	179	229	261	218	210	223	205	.0051	.0002	.0152	3222
129	70	179	229	261	218	210	223	206	.0051	.0002	.0153	3222
135	70	179	229	261	218	211	224	206	.0051	.0002	.0153	3222
140	70	179	229	261	218	211	224	206	.0051	.0002	.0153	3222
145	70	179	229	262	218	211	225	206	.0051	.0002	.0153	3222
150	70	179	229	262	218	211	226	207	.0051	.0002	.0154	3222
155	70	179	230	262	219	212	227	207	.0051	.0002	.0154	3223
160	70	179	230	262	219	212	227	207	.0051	.0002	.0154	3223
165	70	179	230	262	219	212	227	207	.0051	.0002	.0154	3223
171	70	179	230	263	219	212	228	207	.0051	.0002	.0154	3223
176	70	179	231	264	219	213	228	207	.0051	.0002	.0155	3223
181	70	179	233	268	220	213	229	207	.0051	.0002	.0156	3224
186	70	179	235	273	222	213	230	208	.0051	.0002	.0158	3225
191	70	179	236	276	223	213	230	208	.0051	.0002	.0159	3226
196	70	179	238	279	224	213	230	208	.0052	.0002	.0160	3226
201	70	179	239	281	225	214	231	208	.0052	.0002	.0160	3227
207	70	179	240	283	225	214	231	208	.0052	.0002	.0161	3227
212	70	179	241	285	226	214	231	208	.0052	.0002	.0162	3227
217	70	179	242	286	226	214	232	208	.0052	.0002	.0162	3228
253	70	179	245	293	229	216	234	209	.0052	.0002	.0165	3229

TABLE C-3 - Continued

258	70	179	246	294	229	216	235	209	.0052	.0002	.0165	3229
264	70	179	246	295	229	216	235	210	.0052	.0002	.0166	3230
269	70	179	246	296	230	216	235	210	.0052	.0002	.0166	3230
274	70	179	247	297	230	216	236	210	.0052	.0003	.0166	3230
279	70	179	247	298	230	217	236	210	.0052	.0003	.0167	3230
285	70	179	248	298	230	217	236	210	.0052	.0003	.0167	3230
290	70	179	248	299	231	217	236	210	.0053	.0003	.0167	3231
295	70	179	249	300	231	217	237	210	.0053	.0003	.0168	3231
300	70	179	249	301	231	217	237	210	.0053	.0003	.0168	3231
305	70	179	249	302	232	217	237	210	.0053	.0003	.0168	3231
311	70	179	250	302	232	217	237	210	.0053	.0003	.0168	3231
316	70	180	250	304	232	218	238	211	.0053	.0003	.0169	3232
321	70	180	251	304	232	218	239	211	.0053	.0003	.0169	3232
326	70	180	251	305	233	218	239	211	.0053	.0003	.0169	3232
331	70	180	251	306	233	218	239	211	.0053	.0003	.0170	3232
336	70	180	252	307	233	218	239	211	.0053	.0003	.0170	3232
342	70	180	252	307	233	218	239	211	.0053	.0003	.0170	3232
347	70	180	253	308	234	218	240	211	.0053	.0003	.0170	3233
352	70	180	253	309	234	218	240	211	.0053	.0003	.0171	3233
357	70	180	253	310	234	219	240	211	.0053	.0003	.0171	3233
362	70	180	254	311	235	219	240	211	.0053	.0003	.0171	3233
368	70	180	254	312	235	219	241	211	.0053	.0003	.0172	3233

TABLE C-3 - Continued

373	70	180	255	313	235	219	241	211	.0053	.0003	.0172	3233
378	70	180	255	314	236	219	241	212	.0053	.0003	.0172	3234
383	70	180	256	315	236	219	241	212	.0053	.0003	.0173	3234
388	70	180	256	315	236	219	241	212	.0053	.0003	.0173	3234
393	70	180	257	317	236	219	242	212	.0054	.0003	.0173	3234
399	70	180	266	336	243	219	242	212	.0054	.0003	.0179	3237
404	70	181	280	363	252	221	245	213	.0056	.0003	.0188	3243
409	70	181	284	371	254	226	255	216	.0056	.0004	.0193	3246
414	70	181	289	380	257	231	264	219	.0057	.0004	.0198	3249
419	70	181	292	388	260	236	274	222	.0057	.0005	.0203	3251
425	70	181	295	394	262	241	285	226	.0057	.0005	.0208	3254
430	70	181	298	400	264	244	290	228	.0058	.0005	.0211	3256
435	70	181	300	403	265	247	297	230	.0058	.0006	.0213	3258
440	70	181	303	408	267	253	309	234	.0058	.0006	.0219	3260
445	70	181	305	414	269	271	345	246	.0058	.0008	.0231	3267
450	70	181	309	421	271	289	381	258	.0059	.0010	.0243	3274
456	70	181	313	428	273	309	421	271	.0059	.0012	.0256	3281
461	70	181	321	445	279	330	462	285	.0060	.0014	.0273	3290
466	70	183	330	481	291	352	507	299	.0062	.0016	.0295	3301

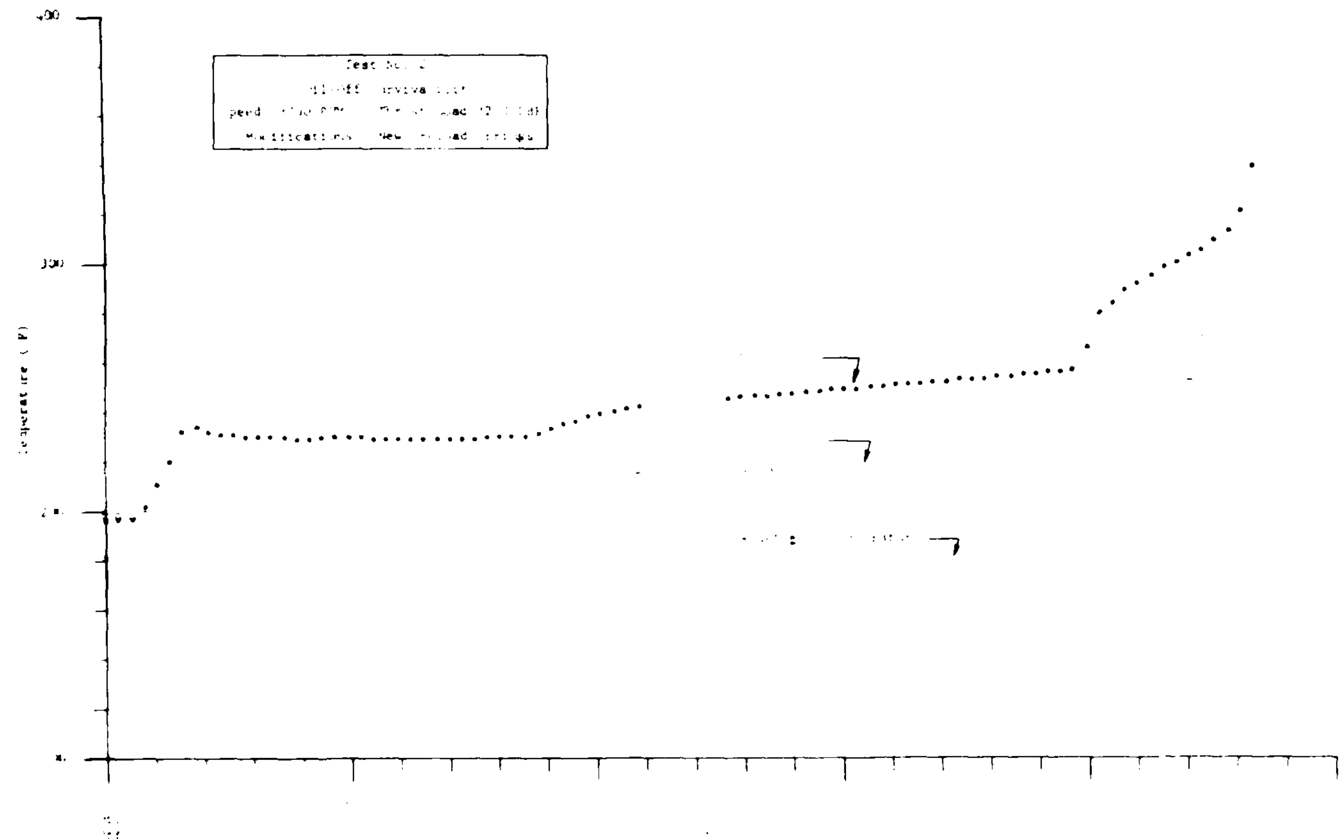


Figure C-4. Bearing and Housing Temperatures in Oil-Off Test No. 2.

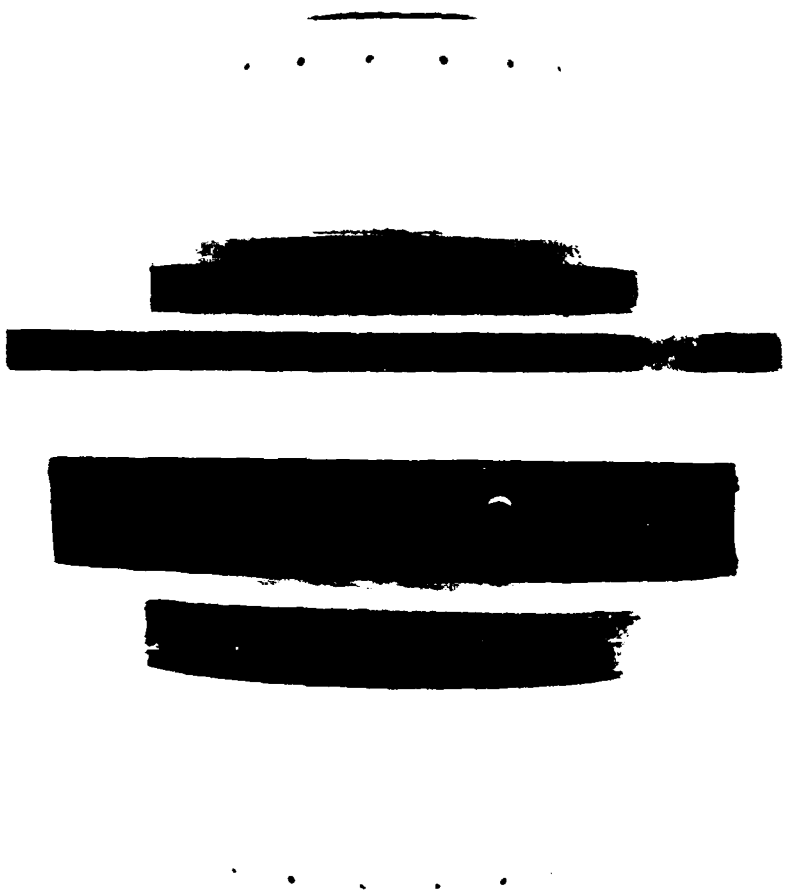


Figure C-8. Shaft No. 78-? From Oil-Off Test No. 2

CUP 78-3 VIEWED FROM SMALL END

CUP 78-3 VIEWED FROM LARGE END

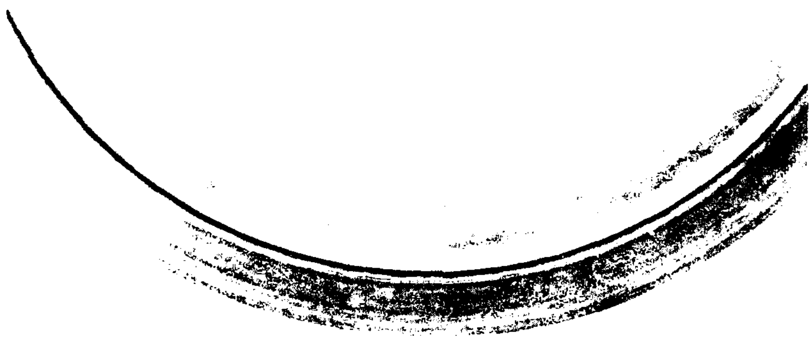
Figure C-6. Cup, Rollers, and Cage Used at Drive End of Shaft 78-1 in Oil-Off Test No. 2 (Sheet 1 of 2).

ROLLERS

CAGE 78-36

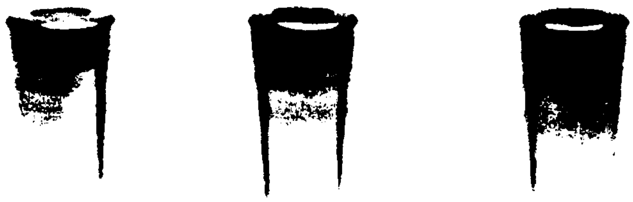
Figure C-6. Cup, Rollers, and Cage Used at Drive End of Shaft 78-1 in Oil-Off Test No. 2 (Sheet 2 of 2)

CUP 78-2 VIEWED FROM SMALL END

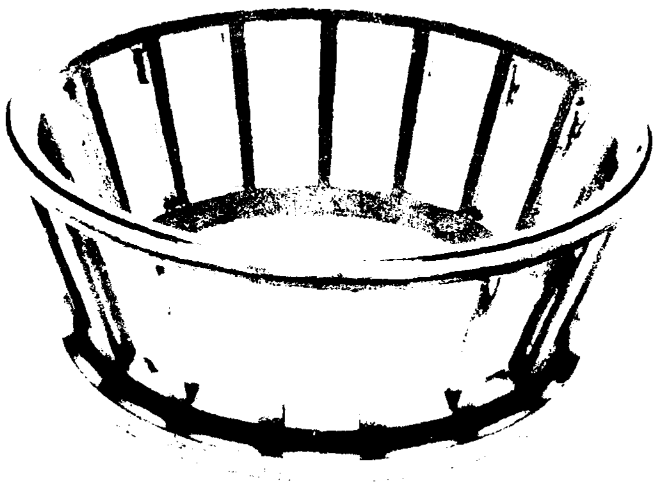


CUP 78-2 VIEWED FROM LARGE END

Figure C–7. Cup, Rollers, and Cage Used Opposite Drive End of Shaft 78-1 in Oil-Off Test No. 2 (Sheet 1 of 2).



ROLLERS



CAGE 78-15

Figure C–7. Cup, Rollers, and Cage Used Opposite Drive End of Shaft 78-1 in Oil-Off Test No. 2 (Sheet 2 of 2).

TABLE C-4. OIL-OFF TEST NO. 3

TEST NO. 141.1-U - BUILD-UP SHEET

SET-UP NO. 1

TEST BEARINGS

	DRIVE END		OPPOSITE DRIVE END	
SHAFT SERIAL NO.	78-8		78-8	
CAGE SERIAL NO.	78-24		78-16	
CUP SERIAL NO.	78-21		78-1	
ROLLER SIZE	7		7 1/8	
CUP O.D.	5.0005		4.9968	
HOUSING I.D.	4.9989		5.0038	
CUP/HOUSING FIT	.0016 TIGHT		.007 LOOSE	
	LARGE END	SMALL END	LARGE END	SMALL END
CUP PILOT I.D.	4.2777	3.6198	4.2779	3.6191
CAGE O.D.	4.2705	3.6123	4.2705	3.6118
CUP/CAGE CLEARANCE	.0072	.0075	.0074	.0073
RADIAL CAGE GROWTH DUE TO ROTATION (RPM)	.000045	.000022	.000045	.000022
RADIAL CUP DEFORMATION DUE TO FIT	.0012	.0010	—	—
RUNNING CUP/CAGE CLEARANCE	.006	.0065	.0065	.007

BELLEVILLE LOADING SPRINGS USED - SET NO. 4

FREE HEIGHT .5335

SPRING DEFLECTION .1827 .182" ACTUAL

PRELOAD 3179 POUNDS

SLAVE BEARINGS

	DRIVE END CENTER	OPPOSITE DRIVE END CENTER
CONE SERIAL NO.		
CUP SERIAL NO.		
ROLLER SIZE		
SHAFT O.D.		
CONE I.D.		
CONE/SHAFT FIT		
CUP O.D.		
HOUSING I.D.		
CUP/HOUSING FIT		
CAGE SHAKE		

BEARING ADJUSTMENT AIM ______ ACTUAL ______

MAGNETIC SEAL NO. 2

	MAGNET RING	SEAL CASE/CARBON INSERT
WEIGHT		
WIDTH		

OIL-OFF TEST

2 MIN. 29 SEC. @ 3700 RPM

(149 SEC.)

TABLE C-5. TEST NO. 3, OIL-OFF SURVIVABILITY

SPEED 3700 RPM THRUST LOAD 3200 LBF

INITIAL CUP FITS DE .0016 ODE -.0070

			* * * * * * TEMPERATURE (F) * * * * *									
			DE			ODE			FITS			
SEC	REF	BRG	CUP	BLR	C/S	CUP	BLR	C/S	DE	ODE	AXL	LBF
0	70	191	209	218	206	205	209	203	.0051	-.0036	.0132	3209
10	70	191	209	218	206	205	210	203	.0051	-.0036	.0132	3209
15	70	191	209	211	[illegible]	205	[illegible]	[illegible]	.0051	-.0036	.0132	3209
20	70	191	209	219	206	206	211	204	.0051	-.0036	.0133	3209
25	70	191	210	220	206	207	[illegible]	[illegible]	[illegible]	-.0035	.0134	3210
30	70	191	212	220	207	[illegible]	[illegible]	[illegible]	[illegible]	-.0035	.0135	3211
40	70	191	211	221	207	215	231	[illegible]	.0051	-.0035	.0138	3213
45	70	191	211	222	207	[illegible]	[illegible]	[illegible]	.0051	[illegible]	.0140	3214
50	70	191	211	223	207	221	[illegible]	[illegible]	[illegible]	-.0034	.0142	3215
55	70	191	212	223	208	222	245	215	.0051	-.0034	.0143	3216
60	70	190	212	224	209	222	[illegible]	[illegible]	.0051	-.0034	.0143	3216
65	70	190	212	224	208	224	248	[illegible]	[illegible]	-.0034	.0144	3216
70	70	190	213	225	208	226	252	217	.0051	-.0034	.0145	3217
80	70	190	222	245	215	229	257	219	.0052	-.0034	.0152	3221
85	70	190	227	254	218	230	261	[illegible]	.0052	-.0033	.0155	3223
90	70	190	231	262	220	231	262	220	.0053	-.0033	.0158	3225
95	70	190	236	273	224	232	264	221	.0053	-.0033	.0162	3228
100	70	190	242	285	228	233	267	222	.0054	-.0033	.0166	3230
105	70	190	247	295	231	235	270	223	.0054	-.0033	.0170	3232
110	70	190	251	302	234	237	273	224	.0055	-.0033	.0173	3234
120	70	190	257	314	238	239	277	225	.0055	-.0033	.0177	3237
125	70	190	259	316	238	240	280	226	.0055	-.0033	.0179	3237

TABLE C-5 - Continued

130	70	190	263	325	241	242	283	227	.0056	-.0032	.0182	3239
135	70	189	270	341	246	242	285	228	.0056	-.0032	.0187	3242
140	70	190	282	354	254	244	288	229	.0057	-.0032	.0195	3247
145	70	191	299	397	265	245	290	230	.0059	-.0032	.0204	3252
155	70	190	313	426	274	241	283	227	.0060	-.0032	[illegible]	3256

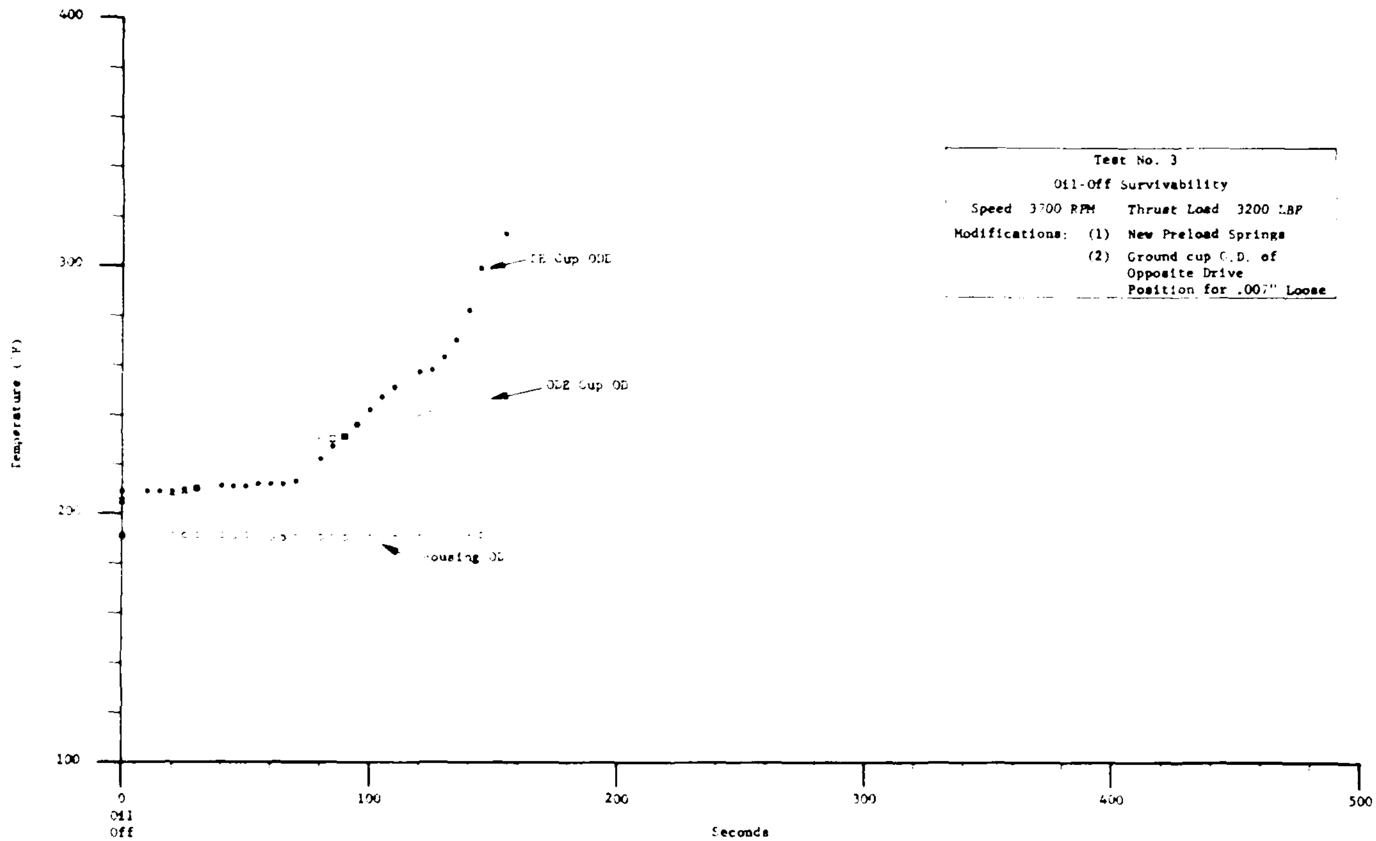


Figure C–8. Bearing and Housing Temperatures in Oil-Off Test No. 3.

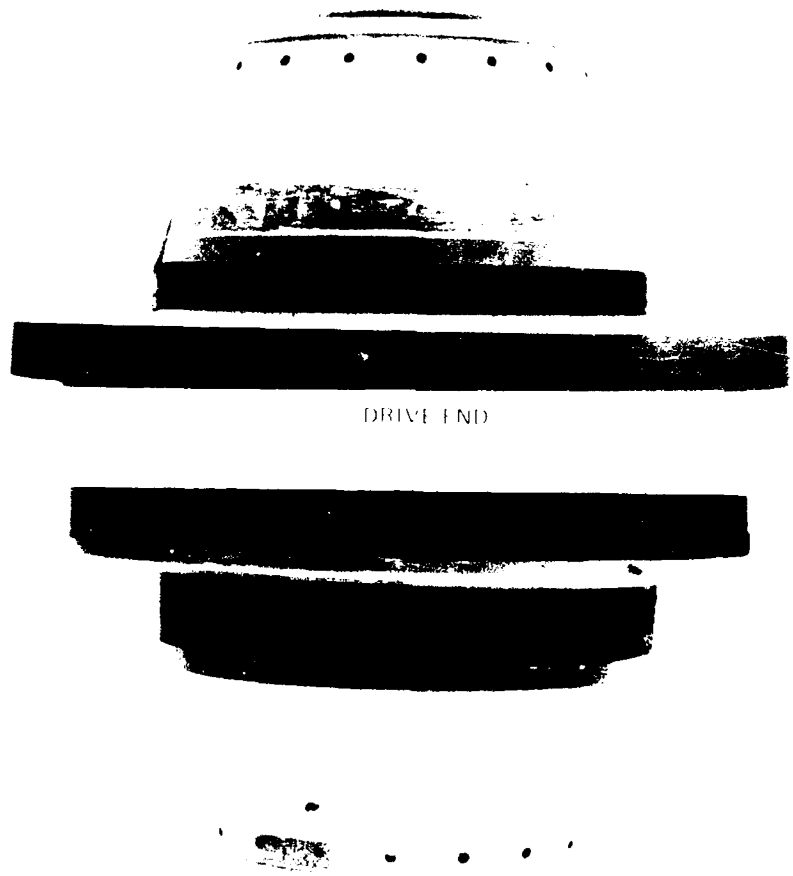


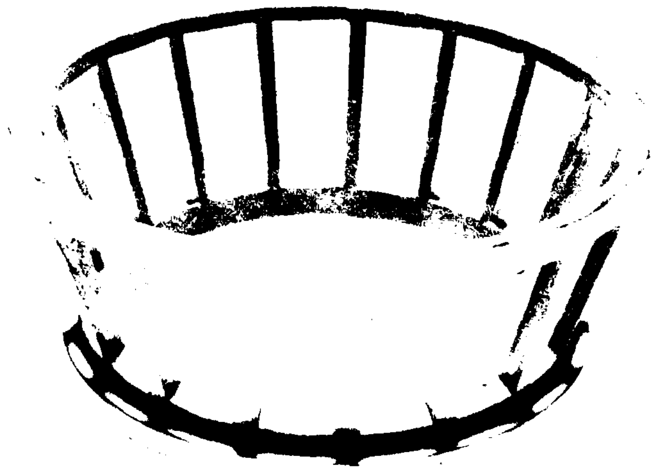
Figure C-9. Shaft No. 78-S From Oil Off Test No. 3

CUP 78-21 VIEWED FROM SMALL END

CUP 78-21 VIEWED FROM LARGE END

Figure C-10. Cup, Rollers, and Cage Used at Drive End of Shaft 78-8 in Oil-Off Test No. 3 (Sheet 1 of 2)

ROLLERS



CAGE 78-24

Figure C 10. Cup, Rollers, and Cage Used at Drive End of Shaft 78-8 in Oil-Off Test No. 3 (Sheet 2 of 2).

TABLE C-6.

OIL-OFF TEST NO. 4

TEST BEARINGS

	DRIVE END		OPPOSITE DRIVE END	
SHAFT SERIAL NO.	78-3		78-3	
CAGE SERIAL NO.	78-18		78-30	
CUP SERIAL NO.	78-11		~~[illegible]~~ 78-1	
ROLLER SIZE .	7 3/4		7 3/8	
CUP O.D.	4.9909		4.9968	
HOUSING I.D.	4.9989		5.0038	
CUP/HOUSING FIT	.0080 L		.0070 L	
	LARGE END	SMALL END	LARGE END	SMALL END
CUP PILOT I.D.	4.2803	3.6198	4.2798	3.6198
CAGE O.D.	4.2700	3.6098	4.2700	3.6093
CUP/CAGE CLEARANCE	.0103L	.0093	.0098L	.0105
RADIAL CAGE GROWTH DUE TO ROTATION (RPM)				
RADIAL CUP DEFORMATION DUE TO FIT	—	—	—	—
RUNNING CUP/CAGE CLEARANCE				

BELLEVILLE LOADING SPRINGS USED - SET NO. 4

FREE HEIGHT .5335

SPRING DEFLECTION .1827 ACTUAL .1826"

PRELOAD 3119 POUNDS

SLAVE BEARINGS

	DRIVE END CENTER	OPPOSITE DRIVE END CENTER
CONE SERIAL NO.		
CUP SERIAL NO.		
ROLLER SIZE		
SHAFT O.D.		
CONE I.D.		
CONE/SHAFT FIT		
CUP O.D.		
HOUSING I.D.		
CUP/HOUSING FIT		
CAGE SHAKE		

BEARING ADJUSTMENT AIM ______ ACTUAL ______

MAGNETIC SEAL NO. 3

	MAGNET RING	SEAL CASE/CARBON INSERT
WEIGHT		
WIDTH		

OIL OFF - 8 MIN 53 SEC.
@ 3700 RPM

TABLE C-7. TEST NO. 4, OIL-OFF SURVIVABILITY

SPEED 3700 RPM THRUST LOAD 3200 LBF

INITIAL CUP FITS DE -.0070 ODE -.0070

			****** TEMPERATURE (F)	*****								
			DE			ODE			FITS			
SEC	REF	MSC	CUP	RLR	C/S	CUP	RLR	C/S	DE	ODE	AVE	LBF
-301	70	187	204	205	203	202	201	202	-.0036	-.0037	.0105	3192
0	70	187	204	205	203	202	201	202	-.0036	-.0037	.0105	3192
6	70	187	204	205	203	202	201	202	-.0036	-.0037	.0105	3192
11	70	187	204	205	203	202	201	202	-.0036	-.0037	.0105	3192
15	70	187	204	206	203	202	203	202	-.0036	-.0036	.0106	3192
19	70	187	204	207	203	204	207	204	-.0036	-.0036	.0107	3193
24	70	187	205	207	204	208	213	206	-.0036	-.0036	.0109	3193
28	70	187	205	208	204	210	218	207	-.0036	-.0036	.0110	3195
32	70	187	205	208	204	213	224	209	-.0036	-.0036	.0112	3196
37	70	187	205	209	204	217	231	212	-.0036	-.0035	.0114	3197
41	70	186	208	214	206	220	238	214	-.0036	-.0035	.0117	3200
45	70	186	213	225	209	223	244	216	-.0036	-.0035	.0121	3202
49	70	186	218	234	213	225	248	217	-.0035	-.0034	.0125	3204
54	70	186	223	243	216	226	250	218	-.0035	-.0034	.0128	3206
58	70	185	228	253	219	228	253	219	-.0034	-.0034	.0131	3209
62	70	185	232	261	222	229	256	220	-.0034	-.0034	.0134	3210
67	70	185	234	266	223	230	259	221	-.0034	-.0034	.0136	3211
71	70	185	235	268	224	231	260	221	-.0034	-.0034	.0137	3212
75	70	185	236	271	225	232	262	222	-.0034	-.0034	.0138	3213
80	70	185	238	273	226	233	264	222	-.0033	-.0034	.0139	3214
84	70	185	239	277	227	233	265	223	-.0033	-.0034	.0140	3214
88	70	185	242	281	228	234	266	223	-.0033	-.0034	.0142	3215

TABLE C-7 - Continued

93	70	185	243	284	229	234	267	223	-.0033	-.0034	[illegible]	3216
97	70	185	244	287	230	235	268	224	-.0033	[illegible]	[illegible]	3216
101	70	185	247	291	231	235	26[illegible]	22[illegible]	-.0033	-.0034	[illegible]	3217
106	70	185	248	294	232	236	271	225	-.0032	-.0034	.0146	3218
110	70	185	249	297	233	239	275	226	-.0032	-.0033	[illegible]	3219
114	70	185	250	298	234	240	277	227	-.0032	-.0033	.0148	3220
118	70	185	251	299	234	241	280	228	-.0032	-.0033	.0150	3220
123	70	185	252	302	235	242	282	228	-.0032	-.0033	.0151	3221
127	70	185	253	304	236	243	284	229	-.0032	-.0033	.0152	3222
131	70	185	254	306	236	244	285	229	-.0032	-.0033	.0153	3222
136	70	185	255	308	237	244	286	230	-.0032	-.0033	.0153	3222
140	70	185	256	310	238	244	286	230	-.0032	-.0033	.0154	3223
144	70	185	257	312	238	244	287	230	-.0032	-.0033	.0155	3223
149	70	185	258	315	239	245	288	230	-.0031	-.0033	.0156	3224
153	70	185	259	317	240	245	288	230	-.0031	-.0033	.0156	3224
157	70	185	261	320	241	245	288	230	-.0031	-.0033	.0157	3225
162	70	185	263	324	242	245	289	231	-.0031	-.0033	.0158	3225
166	70	185	264	326	243	246	289	231	-.0031	-.0033	.0159	3225
170	70	185	265	327	243	245	289	231	-.0031	-.0033	.0159	3226
175	70	186	265	328	244	245	289	231	-.0031	-.0033	.0159	3226
179	70	186	266	329	244	245	289	231	-.0031	-.0033	.0159	3226
183	70	186	266	329	244	245	289	230	-.0031	-.0033	.0159	3226

TABLE C-7 - Continued

188	70	186	266	330	244	245	288	230	-.0031	-.0033	.0159	3226
192	70	186	267	332	245	245	288	230	-.0031	-.0033	.0160	3226
196	70	186	268	334	245	245	288	230	-.0030	-.0033	.0160	3226
200	70	186	269	335	246	245	288	230	-.0030	-.0033	.0160	3227
205	70	186	269	337	246	245	289	231	-.0030	-.0033	.0161	3227
209	70	186	270	338	247	245	289	231	-.0030	-.0033	.0161	3227
213	70	186	271	339	247	245	289	231	-.0030	-.0033	.0161	3227
218	70	186	271	339	247	245	289	231	-.0030	-.0033	.0161	3227
222	70	187	271	340	248	245	289	231	-.0030	-.0032	.0162	3227
226	70	187	272	342	248	245	289	231	-.0030	-.0032	.0162	3227
231	70	187	273	343	249	245	289	231	-.0030	-.0032	.0162	3228
235	70	187	273	344	249	245	289	231	-.0030	-.0032	.0162	3228
239	70	187	273	345	249	245	289	231	-.0030	-.0032	.0162	3228
244	70	187	274	345	249	245	289	231	-.0030	-.0032	.0162	3228
248	70	187	275	348	250	245	289	231	-.0030	-.0032	.0163	3228
252	70	187	280	357	253	245	289	231	-.0029	-.0032	.0165	3229
257	70	187	285	367	257	245	289	231	-.0029	-.0032	.0168	3231
261	70	187	281	360	254	245	289	231	-.0029	-.0032	.0166	323[illegible]
265	70	188	278	354	252	245	289	231	-.0029	-.0032	.0164	3229
270	70	188	276	350	251	245	289	231	-.0029	-.0032	.0163	3228
274	70	188	274	347	250	245	289	231	-.0030	-.0032	.0162	3228
278	70	188	273	344	249	245	289	231	-.0030	-.0032	.0162	3227

TABLE C-7 - Continued

283	70	188	272	342	248	245	288	230	-.0030	-.0032	.0161	3227
287	70	188	271	340	247	245	288	230	-.0030	-.0032	.0160	3226
291	70	188	270	338	247	245	288	230	-.0030	-.0032	.0160	3226
296	70	188	269	336	246	245	288	230	-.0030	-.0032	.0159	3226
300	70	188	269	335	246	245	288	230	-.0030	-.0032	.0159	3226
304	70	188	268	334	245	245	288	230	-.0030	-.0032	.0159	3225
308	70	188	267	332	245	245	288	230	-.0030	-.0032	.0158	3225
313	70	199	266	331	244	245	289	231	-.0030	-.0032	.0158	3225
317	70	189	266	330	244	245	289	231	-.0030	-.0032	.0157	3225
321	70	189	265	328	244	245	288	230	-.0030	-.0032	.0157	3224
326	70	189	265	327	243	245	289	231	-.0030	-.0032	.0157	3224
330	70	189	264	326	243	245	289	231	-.0030	-.0032	.0156	3224
334	70	199	264	325	243	245	289	231	-.0030	-.0032	.0156	3224
339	70	189	263	325	242	245	289	231	-.0030	-.0032	.0156	3224
343	70	189	263	324	242	245	289	231	-.0030	-.0032	.0156	3224
347	70	190	263	323	242	245	289	231	-.0030	-.0032	.0155	3223
352	70	190	262	322	242	247	293	232	-.0030	-.0032	.0156	3224
356	70	190	262	322	241	249	295	233	-.0030	-.0032	.0157	3224
360	70	190	262	321	241	249	297	233	-.0030	-.0032	.0157	3224
365	70	190	262	321	241	249	297	233	-.0030	-.0032	.0157	3224
369	70	190	261	321	241	250	297	233	-.0030	-.0032	.0157	3224
373	70	190	261	320	241	249	297	233	-.0030	-.0031	.0156	3224

TABLE C-7 - Continued

378	70	190	261	320	241	249	296	233	-.0030	-.0031	.0156	3224
382	70	190	261	319	241	249	295	233	-.0030	-.0032	.0155	3224
386	70	191	261	319	241	249	295	233	-.0030	-.0031	.0155	3223
391	70	191	261	319	241	248	294	232	-.0030	-.0032	.0155	3223
395	70	191	261	319	241	248	294	232	-.0030	-.0032	.0155	3223
399	70	191	260	319	241	248	294	232	-.0030	-.0031	.0155	3223
404	70	191	260	319	241	249	295	233	-.0030	-.0031	.0155	3223
408	70	191	260	319	241	249	297	233	-.0030	-.0031	.0155	3223
412	70	191	260	319	241	250	298	234	-.0030	-.0031	.0156	3224
417	70	191	260	318	240	249	297	233	-.0030	-.0031	.0155	3223
421	70	191	260	318	240	248	294	232	-.0030	-.0031	.0154	3223
425	70	191	260	318	240	247	292	232	-.0030	-.0032	.0154	3223
429	70	191	260	318	240	249	295	233	-.0030	-.0031	.0155	3223
434	70	191	260	318	240	250	298	234	-.0030	-.0031	.0155	3223
438	70	191	260	318	240	252	302	235	-.0030	-.0031	.0156	3224
442	70	192	260	318	240	253	305	236	-.0030	-.0031	.0157	3224
447	70	192	260	318	240	257	312	238	-.0030	-.0031	.0158	3225
451	70	192	260	318	240	264	318	240	-.0030	-.0030	.0160	3226
455	70	192	260	318	240	276	349	251	-.0030	-.0029	.0168	3231
460	70	192	260	317	240	290	379	260	-.0030	-.0027	.0175	3235
464	70	192	260	317	240	304	406	269	-.0030	-.0026	.0182	3239
468	70	192	260	317	240	316	429	277	-.0030	-.0025	.0187	3243

TABLE C-7 - Continued

473	70	192	259	317	240	326	451	284	-.0030	-.0024	.0193	3246
477	70	192	260	317	240	336	470	290	-.0030	-.0023	.0198	3249
481	70	192	260	318	240	346	490	297	-.0030	-.0022	.0203	3252
486	70	192	261	320	241	355	507	303	-.0030	-.0021	.0208	3254
490	70	193	262	321	241	363	525	308	-.0030	-.0020	.0212	3257
494	70	193	263	324	242	371	541	314	-.0030	-.0020	.0217	3259
499	70	193	264	326	243	380	558	319	-.0030	-.0019	.0222	3262
503	70	193	265	328	244	389	575	325	-.0030	-.0018	.0226	3265
507	70	193	266	330	244	397	591	330	-.0029	-.0017	.0231	3267
512	70	193	267	333	245	404	606	335	-.0029	-.0016	.0235	3269
516	70	193	268	334	246	411	621	340	-.0029	-.0016	.0239	3272
520	70	194	269	336	246	420	639	346	-.0029	-.0015	.0244	3274
525	70	194	273	344	249	434	665	355	-.0029	-.0013	.0253	3279
529	70	194	317	431	278	452	702	367	-.0025	-.0012	.0234	3295
533	70	194	361	520	307	474	746	381	-.0020	-.0010	.0317	3311

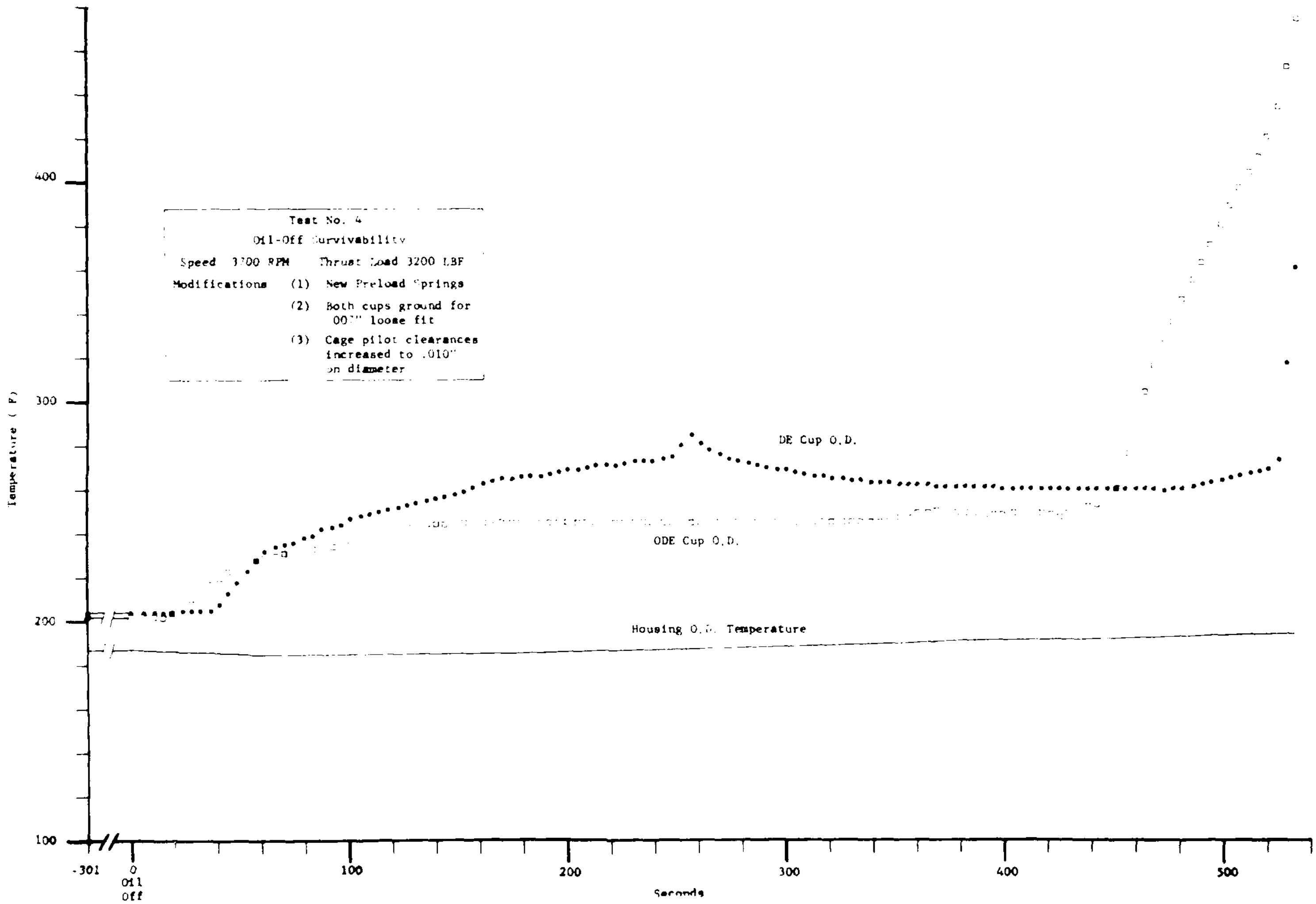


Figure C-11. Bearing and Housing Temperatures in Oil-Off Test No. 4.

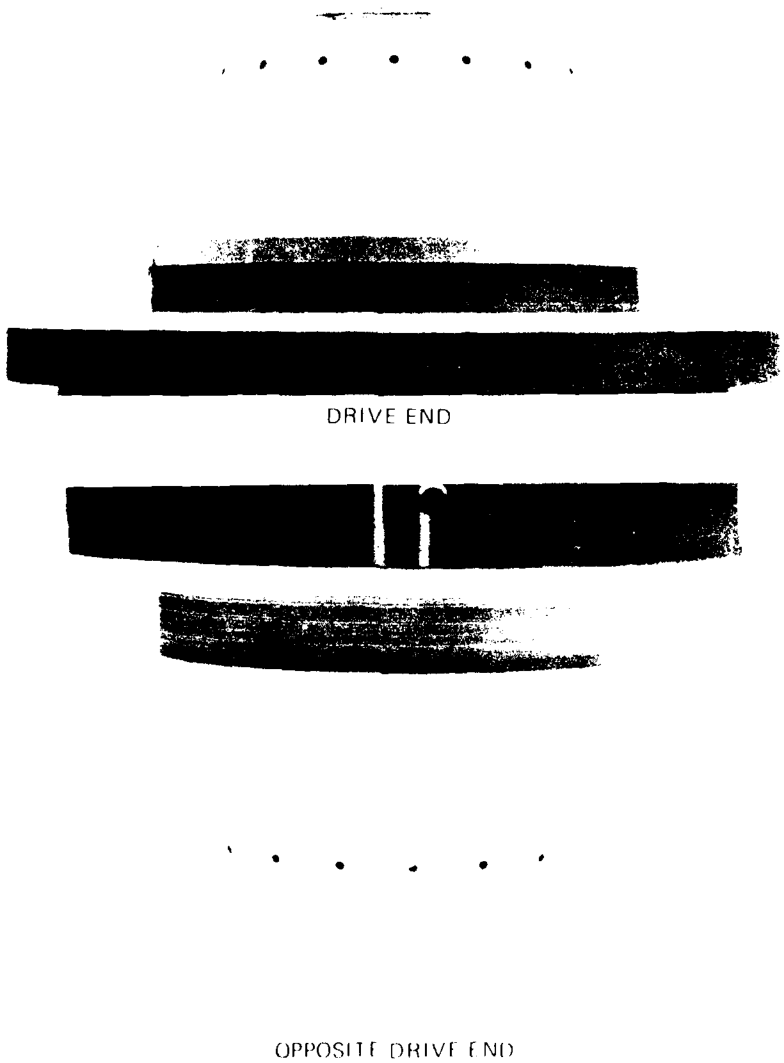


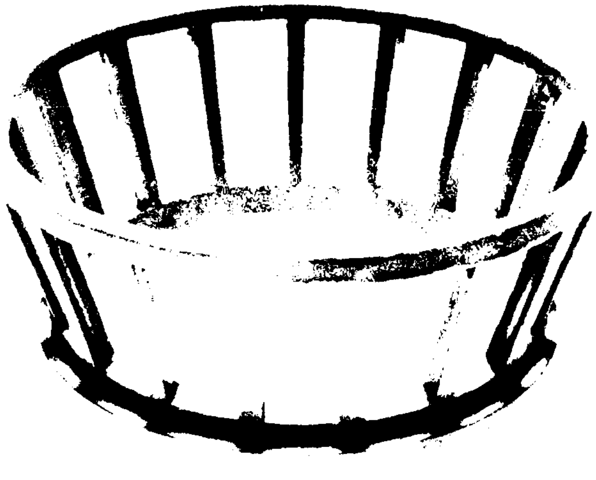
Figure C-12. Shaft No. 78-3 From Oil-Off Test No. 4

CUP 78 11 VIEWED FROM SMALL END

CUP 78 11 VIEWED FROM LARGE END

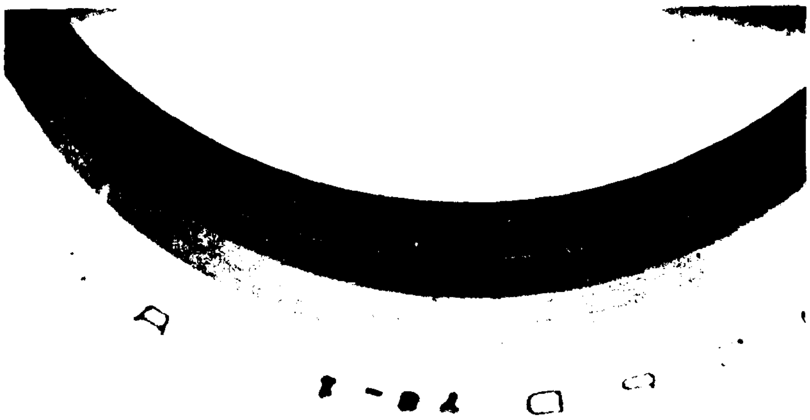
Figure C 13. Cup, Rollers, and Cage Used at Drive End of Shaft 78-3 in Oil-Off Test No. 4 (Sheet 1 of 2).

ROLLERS



CAGE 78-18

Figure C-13. Cup, Rollers, and Cage Used at Drive End of Shaft 78-3 in Oil-Off Test No. 4 (Sheet 2 of 2).

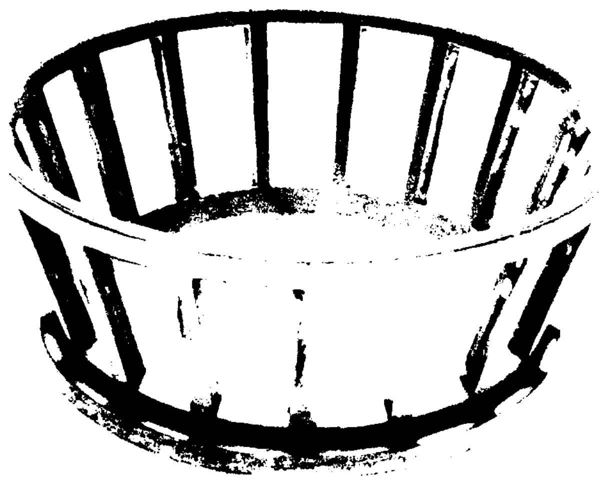


CUP 78-1 VIEWED FROM SMALL END

CUP 78-1 VIEWED FROM LARGE END

Figure C 14. Cup, Rollers, and Cage Used Opposite Drive End of Shaft 78-8 in Oil-Off Test No. 4 (Sheet 1 of 2).

ROLLERS



CAGE 78-16

Figure C-14. Cup, Rollers, and Cage Used Opposite Drive End of Shaft 78-8 in Oil-Off Test No. 4 (Sheet 2 of 2).

TABLE C-8. OIL-OFF TEST NO. 5

TEST NO. 141.1-U - BUILD-UP SHEET

SET-UP NO. 4

TEST BEARINGS

	DRIVE END		OPPOSITE DRIVE END	
SHAFT SERIAL NO.	78-5		78-5	
CAGE SERIAL NO.	78-17		78-21	
CUP SERIAL NO.	79-1		78-5	
ROLLER SIZE	7 2/8		7 1/8	
CUP O.D.	4.9908		4.9912	
HOUSING I.D.	4.9989		5.0010	
CUP/HOUSING FIT	.0081 -L		.0098 -L	
	LARGE END	SMALL END	LARGE END	SMALL END
CUP PILOT I.D.	4.2852	3.6230	4.2861	3.6238
CAGE O.D.	4.2700	3.6124	4.2716	3.6137
CUP/CAGE CLEARANCE	[illegible]	[illegible]	.0145	[illegible]
RADIAL CAGE GROWTH DUE TO ROTATION (________ RPM)				
RADIAL CUP DEFORMATION DUE TO FIT	—	—	—	—
RUNNING CUP/CAGE CLEARANCE				

BELLEVILLE LOADING SPRINGS USED - SET NO. 4

FREE HEIGHT .5235

SPRING DEFLECTION .180 Actual .1303

PRELOAD 3100 POUNDS

SLAVE BEARINGS

	DRIVE END CENTER	OPPOSITE DRIVE END CENTER
CONE SERIAL NO.		
CUP SERIAL NO.		
ROLLER SIZE		
SHAFT O.D.		
CONE I.D.		
CONE/SHAFT FIT		
CUP O.D.		
HOUSING I.D.		
CUP/HOUSING FIT		
CAGE SHAKE		

BEARING ADJUSTMENT AIM ____________ ACTUAL ____________

MAGNETIC SEAL NO. 4

	MAGNET RING	SEAL CASE/CARBON INSERT
WEIGHT		
WIDTH		

Oil-Off
4 min 22 sec
@ 1400 RPM

TABLE C-9. TEST NO. 5, OIL-OFF SURVIVABILITY

SPEED 7400 RPM   THRUST LOAD 3200 LBF

INITIAL CUP FITS   DE -.0081   ODE -.0098

	* * * * * * TEMPERATURE (F) * * * * *											
			------DE-----			-----ODE-----			-----FITS-----			
SEC	REF	USG	CUP	BLR	C/S	CUP	BLR	C/S	DE	ODE	AXL	LBF
-13	70	214	233	226	235	228	216	232	-.0040	-.0057	.0131	3208
-9	70	214	233	226	235	228	216	232	-.0040	-.0057	.0131	3208
-5	70	215	233	226	235	228	216	232	-.0039	-.0057	.0131	3208
0	70	214	233	226	235	228	216	232	-.0040	-.0057	.0131	3208
4	70	214	233	226	236	228	216	232	-.0040	-.0057	.0131	3208
8	70	214	233	227	236	228	216	232	-.0040	-.0057	.0131	3208
12	70	214	234	227	236	229	217	232	-.0040	-.0057	.0131	3209
17	70	214	234	228	236	229	218	233	-.0040	-.0057	.0132	3209
21	70	214	234	228	236	230	219	233	-.0040	-.0057	.0132	3209
25	70	214	234	229	236	230	220	234	-.0039	-.0057	.0133	3210
30	70	214	235	230	237	231	222	234	-.0039	-.0057	.0134	3210
34	70	214	235	231	237	232	223	234	-.0039	-.0057	.0134	3210
38	70	213	237	233	238	232	224	235	-.0039	-.0057	.0135	3211
43	70	213	240	239	240	233	226	235	-.0039	-.0057	.0137	3212
47	70	213	242	243	241	233	227	236	-.0039	-.0057	.0139	3213
51	70	213	242	245	242	234	228	236	-.0039	-.0057	.0139	3214
56	70	213	243	247	242	235	230	237	-.0039	-.0057	.0140	3214
60	70	213	246	251	244	236	231	237	-.0039	-.0057	.0142	3215
64	70	212	264	288	256	236	232	237	-.0037	-.0057	.0152	3221
69	70	212	296	352	277	237	234	238	-.0034	-.0057	.0168	3231
73	70	212	327	415	298	238	235	238	-.0031	-.0056	.0184	3241
77	70	212	348	457	312	238	237	239	-.0029	-.0056	.0195	3247

TABLE C-9 - Continued

82	70	212	357	473	317	239	238	239	-.0028	-.0056	.0200	3250
86	70	212	356	472	317	240	239	240	-.0028	-.0056	.0200	3250
90	70	212	356	472	317	240	241	240	-.0028	-.0056	.0200	3250
95	70	212	358	475	318	241	242	241	-.0028	-.0056	.0202	3251
99	70	211	360	479	319	242	243	241	-.0028	-.0056	.0203	3252
103	70	211	362	484	320	242	244	241	-.0028	-.0056	.0205	3252
108	70	211	365	490	323	243	246	242	-.0027	-.0056	.0207	3254
112	70	211	368	496	325	243	246	242	-.0027	-.0056	.0208	3254
116	70	211	371	502	326	244	247	242	-.0027	-.0056	.0210	3255
120	70	211	373	507	328	244	247	242	-.0027	-.0056	.0211	3256
125	70	211	376	511	329	243	247	242	-.0027	-.0056	.0212	3257
129	70	211	377	515	331	243	247	242	-.0026	-.0056	.0213	3257
133	70	211	379	518	332	243	246	242	-.0026	-.0056	.0214	3258
138	70	211	381	521	333	243	246	242	-.0026	-.0056	.0215	3258
142	70	211	383	525	334	243	246	242	-.0026	-.0056	.0216	3259
146	70	211	385	529	335	243	246	242	-.0026	-.0056	.0217	3259
151	70	211	386	531	336	243	247	242	-.0026	-.0056	.0217	3260
155	70	211	386	531	336	244	247	242	-.0026	-.0056	.0217	3260
159	70	211	386	531	336	244	247	242	-.0026	-.0056	.0217	3260
164	70	211	385	530	336	244	247	242	-.0026	-.0056	.0217	3260
168	70	211	385	529	335	244	247	242	-.0026	-.0056	.0217	3259
172	70	211	384	527	335	244	248	243	-.0026	-.0056	.0217	3259

TABLE C-9 - Continued

177	70	211	382	523	333	244	249	243	-.0026	-.0056	.0216	3259
181	70	211	380	520	332	245	250	243	-.0026	-.0056	.0215	3258
185	70	211	380	520	332	245	250	243	-.0026	-.0056	.0215	3259
190	70	211	379	519	332	245	251	243	-.0026	-.0056	.0215	3258
194	70	212	378	515	331	246	251	244	-.0026	-.0056	.0214	3258
198	70	212	376	512	330	246	252	244	-.0026	-.0056	.0213	3257
203	70	212	374	508	329	246	252	244	-.0026	-.0056	.0212	3257
207	70	212	373	506	328	246	252	244	-.0027	-.0056	.0212	3256
211	70	212	372	504	327	246	253	244	-.0027	-.0056	.0211	3256
216	70	212	371	502	326	247	253	244	-.0027	-.0056	.0211	3256
220	70	212	370	500	326	247	255	245	-.0027	-.0056	.0211	3256
224	70	212	370	499	325	256	271	250	-.0027	-.0055	.0215	3258
229	70	213	371	502	326	277	314	265	-.0027	-.0053	.0226	3264
233	70	213	374	508	328	303	366	281	-.0026	-.0050	.0240	3272
237	70	213	377	515	331	332	424	301	-.0026	-.0047	.0257	3281
242	70	213	381	523	333	362	484	320	-.0026	-.0045	.0274	3290
246	70	213	387	533	337	387	535	332	-.0025	-.0042	.0289	3298
250	70	213	393	547	341	410	580	352	-.0025	-.0040	.0304	3305
255	70	213	410	581	352	444	648	375	-.0023	-.0037	.0330	3317
259	70	213	476	711	395	497	755	410	-.0017	-.0032	.0389	3344
263	70	213	535	831	435	542	845	440	-.0011	-.0027	.0442	3365

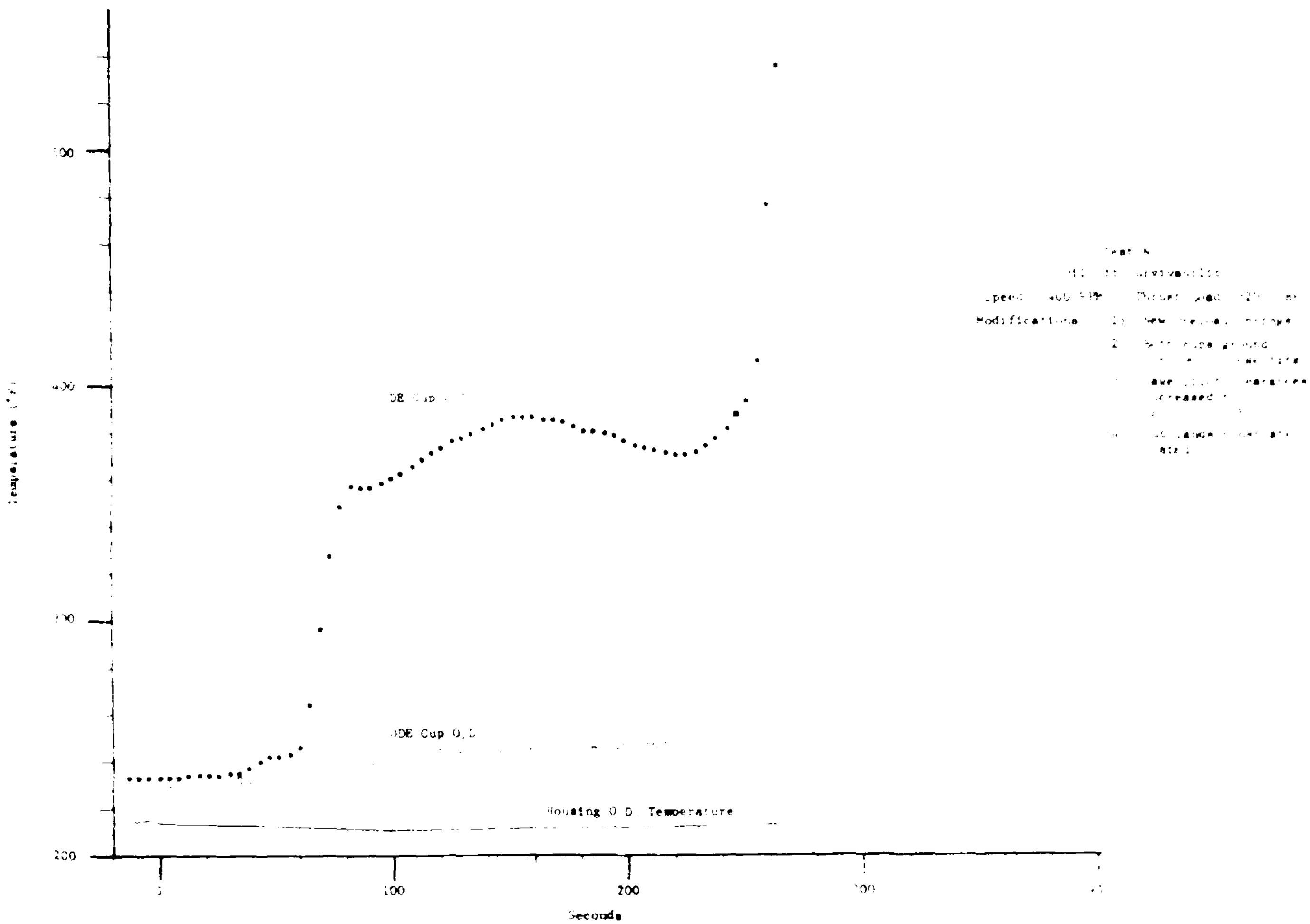
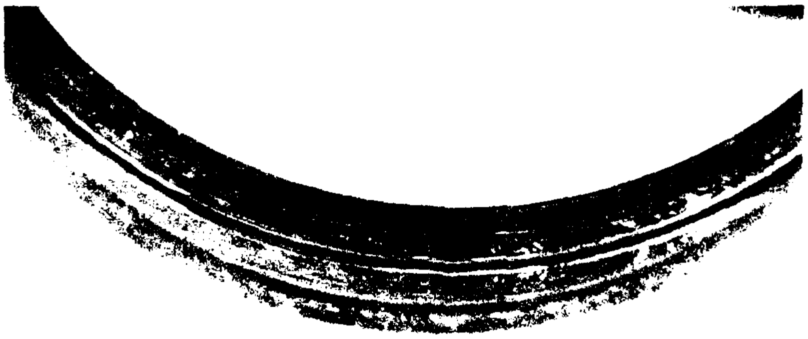


Figure … 5. Bearing and Housing Temperatures in Oil-Off Test No. 5

OPPOSITE DRIVE END

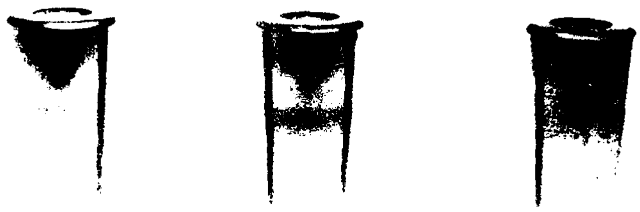
Figure C-16. Shaft No. 78-5 From Oil-Off Test No. 5



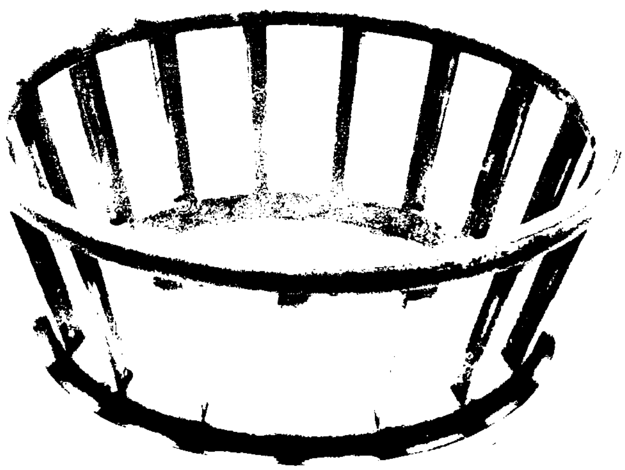
CUP 79-1 VIEWED FROM SMALL END

CUP 79-1 VIEWED FROM LARGE END

Figure C 17. Cup, Rollers, and Cage Used at Drive End of Shaft 78-5 in Oil-Off Test No. 5 (Sheet 1 of 2).



ROLLERS



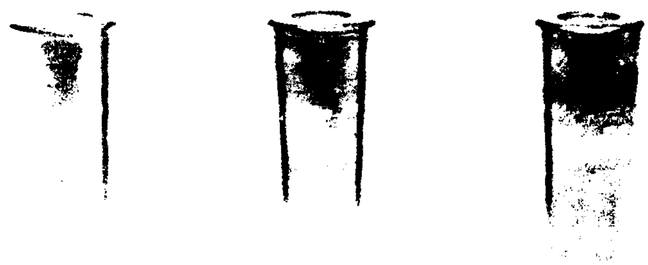
CAGE 78-17

Figure C 17. Cup, Rollers, and Cage Used at Drive End of Shaft 78-5 in Oil-Off Test No. 5 (Sheet 2 of 2).

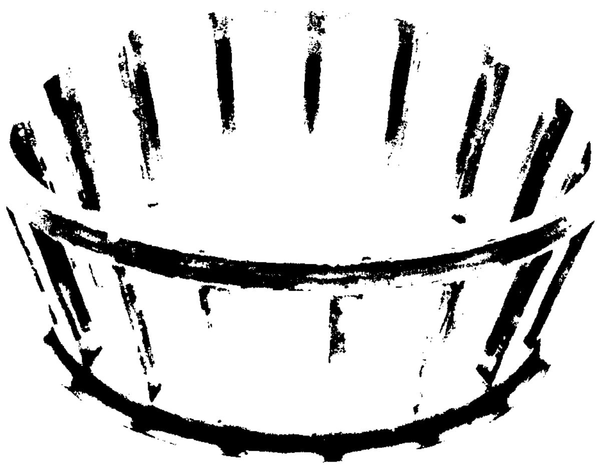
CUP 78-5 VIEWED FROM SMALL END

CUP 78-5 VIEWED FROM LARGE END

Figure C-18. Cup, Rollers, and Cage Used Opposite Drive End of Shaft 78-5 in Oil-Off Test No. 5 (Sheet 1 of 2).



ROLLERS



CAGE 78-21

Figure C 18. Cup, Rollers, and Cage Used Opposite Drive End of Shaft 78-5 in Oil-Off Test No. 5 (Sheet 2 of 2).

TABLE C-10. Oil-Off Test No. 6

TEST NO. 141.1-U - BUILD-UP SHEET

SET-UP NO. 5

TEST BEARINGS

	DRIVE END		OPPOSITE DRIVE END	
SHAFT SERIAL NO.	78-4		78-4	
CAGE SERIAL NO.	78-23		78-25	
CUP SERIAL NO.	79-5		78-9	
ROLLER SIZE	7 2/8		7 6/8	
CUP O.D.	4.9910		4.7960	
HOUSING I.D.	4.9989		5.0038	
CUP/HOUSING FIT	.0079 L		.0078 L	
	LARGE END	SMALL END	LARGE END	SMALL END
CUP PILOT I.D.	4.2888	3.6222	4.2860	3.6212
CAGE O.D.	4.2734	3.6122	4.2708	3.6110
CUP/CAGE CLEARANCE	.0154 L	.0100 L	4.0052 .0152 L	.0102 L
RADIAL CAGE GROWTH DUE TO ROTATION (RPM)				
RADIAL CUP DEFORMATION DUE TO FIT				
RUNNING CUP/CAGE CLEARANCE				

BELLEVILLE LOADING SPRINGS USED - SET NO. 4

FREE HEIGHT .5335

SPRING DEFLECTION .1827 .1820"

PRELOAD 3119 POUNDS

SLAVE BEARINGS

	DRIVE END CENTER	OPPOSITE DRIVE END CENTER
CONE SERIAL NO.		
CUP SERIAL NO.		
ROLLER SIZE		
SHAFT O.D.		
CONE I.D.		
CONE/SHAFT FIT		
CUP O.D.		
HOUSING I.D.		
CUP/HOUSING FIT		
CAGE SHAKE		

BEARING ADJUSTMENT AIM ________ ACTUAL ________

MAGNETIC SEAL NO. 5

	MAGNET RING	SEAL CASE/CARBON INSERT
WEIGHT		
WIDTH		

OIL OFF 3 MIN 40 SEC.
@ 3700 RPM

TABLE C-11. TEST NO. 6, OIL-OFF SURVIVABILITY

TABLE C-11. TEST NO. 6, OIL-OFF SURVIVABILITY
SPEED 3700 RPM THRUST LOAD 3200 LBF

INITIAL CUP FITS DE -.0079 ODE -.0078

☆ ☆ ☆ ☆ ☆ TEMPERATURE (F) ☆ ☆ ☆ ☆ ☆

			DE			ODE			FITS			
SEC	REF	[illegible]	CUP	RLR	C/S	CUP	RLR	C/S	DE	ODE	AXL	LBF
-9	70	185	204	203	204	206	206	205	-.0046	-.0044	.0109	3195
-4	70	185	204	204	205	206	206	205	-.0046	-.0044	.0110	3195
0	70	185	204	204	205	206	206	205	-.0046	-.0044	.0110	3195
4	70	185	204	204	205	206	206	205	-.0046	-.0044	.0110	3195
9	70	185	204	204	205	206	206	205	-.0045	-.0044	.0110	3195
13	70	185	205	204	205	206	206	205	-.0045	-.0044	.0110	3195
17	70	185	204	204	205	206	207	206	-.0046	-.0044	.0110	3195
22	70	185	205	205	205	206	208	206	-.0045	-.0044	.0110	3195
26	70	185	205	205	205	207	209	206	-.0045	-.0044	.0111	3196
30	70	185	205	204	205	207	210	207	-.0046	-.0044	.0111	3196
35	70	185	205	205	205	208	210	207	-.0045	-.0044	.0111	3196
39	70	185	205	205	205	208	211	207	-.0045	-.0045	.0111	3196
43	70	185	206	206	205	208	212	207	-.0045	-.0045	.0112	3196
47	70	185	206	206	205	209	213	207	-.0045	-.0044	.0112	3196
52	70	185	206	207	206	209	214	208	-.0045	-.0044	.0113	3197
56	70	184	206	207	206	210	214	208	-.0045	-.0044	.0113	3197
60	70	184	206	208	206	210	215	208	-.0045	-.0044	.0113	3197
65	70	184	207	209	206	210	216	209	-.0045	-.0044	.0114	3197
69	70	184	207	209	206	211	216	209	-.0045	-.0044	.0114	3198
73	70	184	207	209	206	211	218	209	-.0045	-.0044	.0114	3198
78	70	184	207	210	207	211	218	209	-.0045	-.0044	.0115	3198
82	70	184	208	210	207	212	219	209	-.0045	-.0044	.0115	3198

TABLE C-11 - Continued

86	70	184	208	210	207	213	220	210	-.0045	-.0044	.0116	3199
91	70	184	208	210	207	215	225	212	-.0045	-.0044	.0117	3200
95	70	184	208	211	207	219	233	214	-.0045	-.0043	.0119	3201
99	70	183	208	210	207	222	240	217	-.0045	-.0043	.0121	3202
104	70	183	208	210	207	226	247	219	-.0045	-.0043	.0123	3203
108	70	183	208	211	207	228	251	220	-.0045	-.0043	.0124	3204
112	70	183	208	211	207	230	254	221	-.0045	-.0042	.0125	3204
116	70	183	208	212	207	231	257	222	-.0045	-.0042	.0126	3205
121	70	183	209	212	207	238	271	227	-.0045	-.0042	.0129	3207
125	70	183	209	212	207	253	301	237	-.0045	-.0040	.0137	3212
129	70	183	209	212	207	269	332	247	-.0045	-.0039	.0145	3217
134	70	183	209	213	207	282	350	256	-.0045	-.0037	.0152	3221
138	70	183	209	213	208	295	385	264	-.0045	-.0036	.0158	3225
142	70	183	209	213	208	308	411	273	-.0045	-.0035	.0165	3229
147	70	183	209	214	208	319	432	280	-.0045	-.0034	.0171	3233
151	70	183	210	214	208	328	451	286	-.0045	-.0033	.0176	3236
155	70	183	210	215	208	336	467	292	-.0045	-.0032	.0180	3238
160	70	183	210	215	208	344	484	297	-.0045	-.0032	.0184	3240
164	70	183	211	216	209	352	499	302	-.0045	-.0031	.0188	3243
168	70	182	211	217	209	360	516	307	-.0045	-.0030	.0192	3245
173	70	182	211	218	209	371	537	314	-.0045	-.0029	.0198	3240
177	70	182	213	222	210	380	554	320	-.0045	-.0028	.0203	3252
181	70	182	216	227	212	387	569	325	-.0045	-.0028	.0208	3254
186	70	183	218	231	214	400	594	333	-.0045	-.0026	.0216	3259
190	70	183	220	235	215	412	619	342	-.0044	-.0025	.0223	3263
194	70	183	222	238	216	424	642	349	-.0044	-.0024	.0230	3266
198	70	183	226	248	219	434	662	356	-.0044	-.0023	.0237	3270

TABLE C-11 - Continued

TABLE C-11 - Continued

203	70	183	232	259	223	442	679	361	-.0043	-.0022	.0244	3274
207	70	183	253	301	237	451	696	367	-.0041	-.0021	.0259	3282
211	70	183	284	363	257	465	726	377	-.0038	-.0020	.0282	3294
216	70	183	343	481	296	483	761	388	-.0033	-.0018	.0320	3313
220	70	183	399	593	333	512	818	407	-.0027	-.0016	.0363	3332

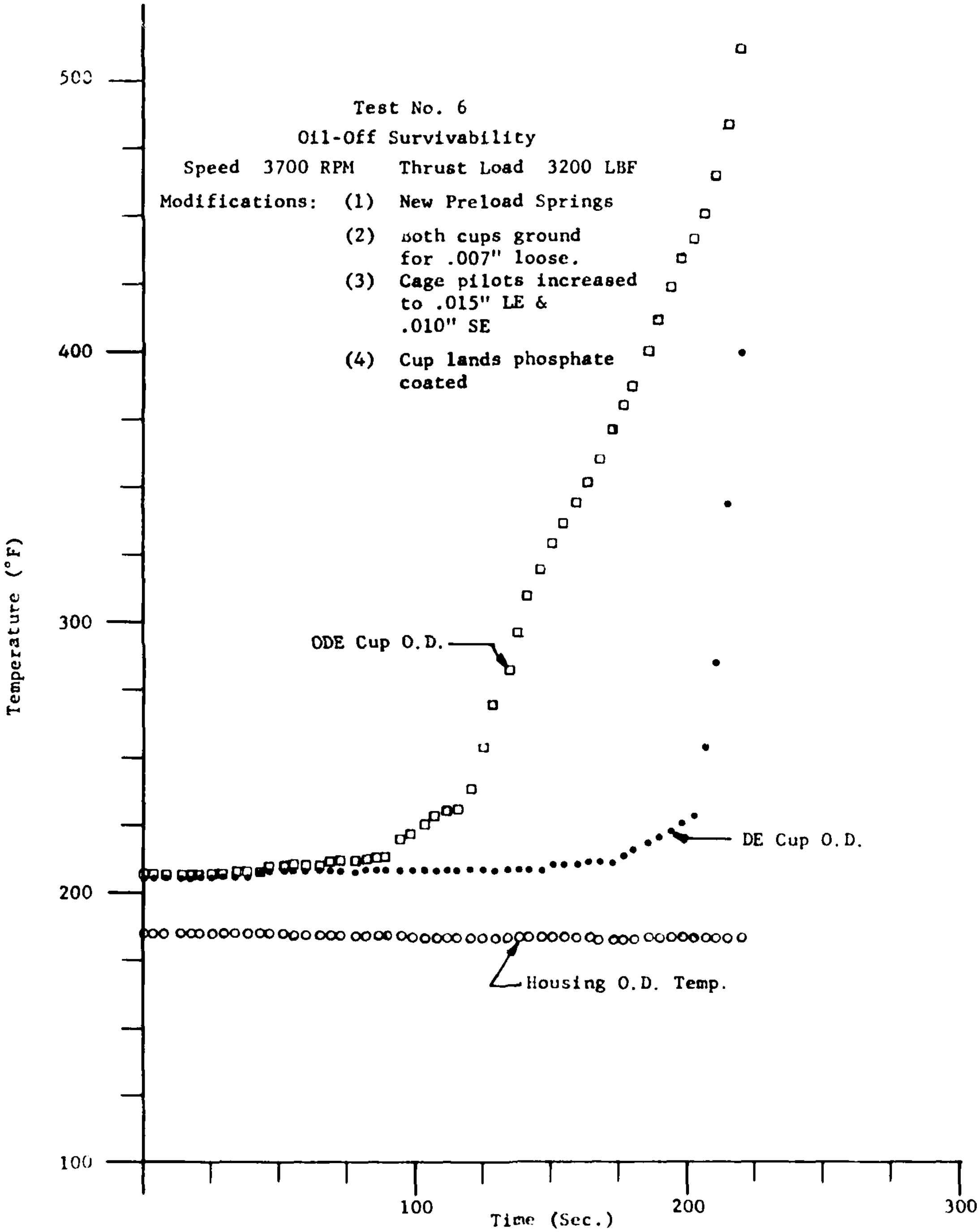


Figure C 19. Bearing and Housing Temperatures in Oil-Off Test No. 6.

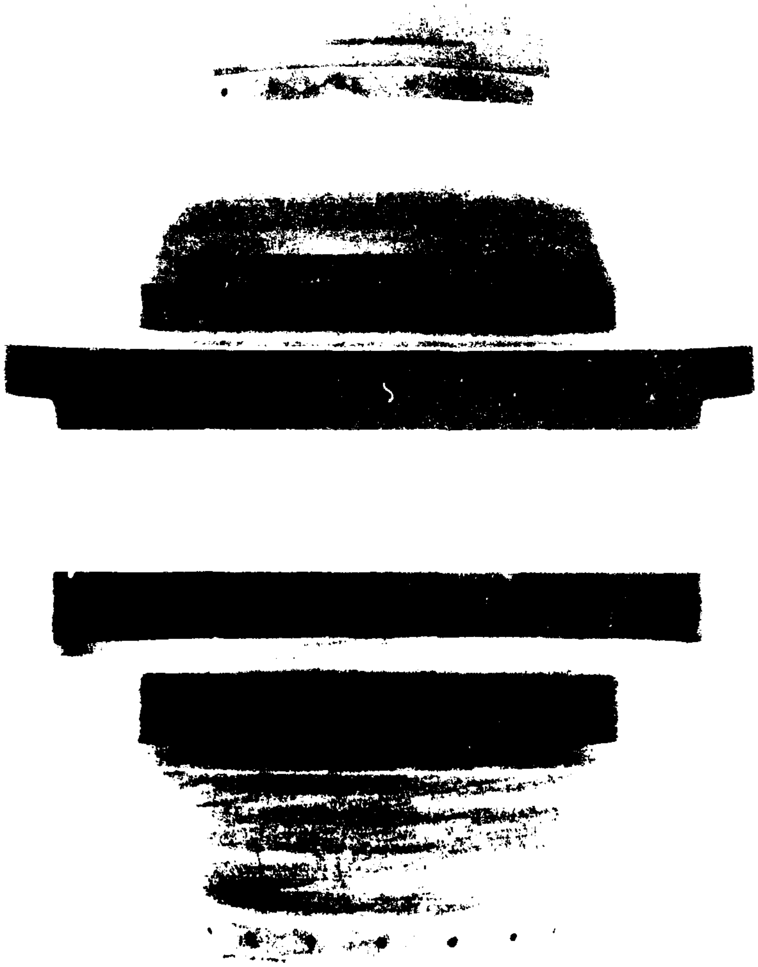


Figure C-20. Shaft No. [illegible] Oil [illegible] Test No. [illegible]

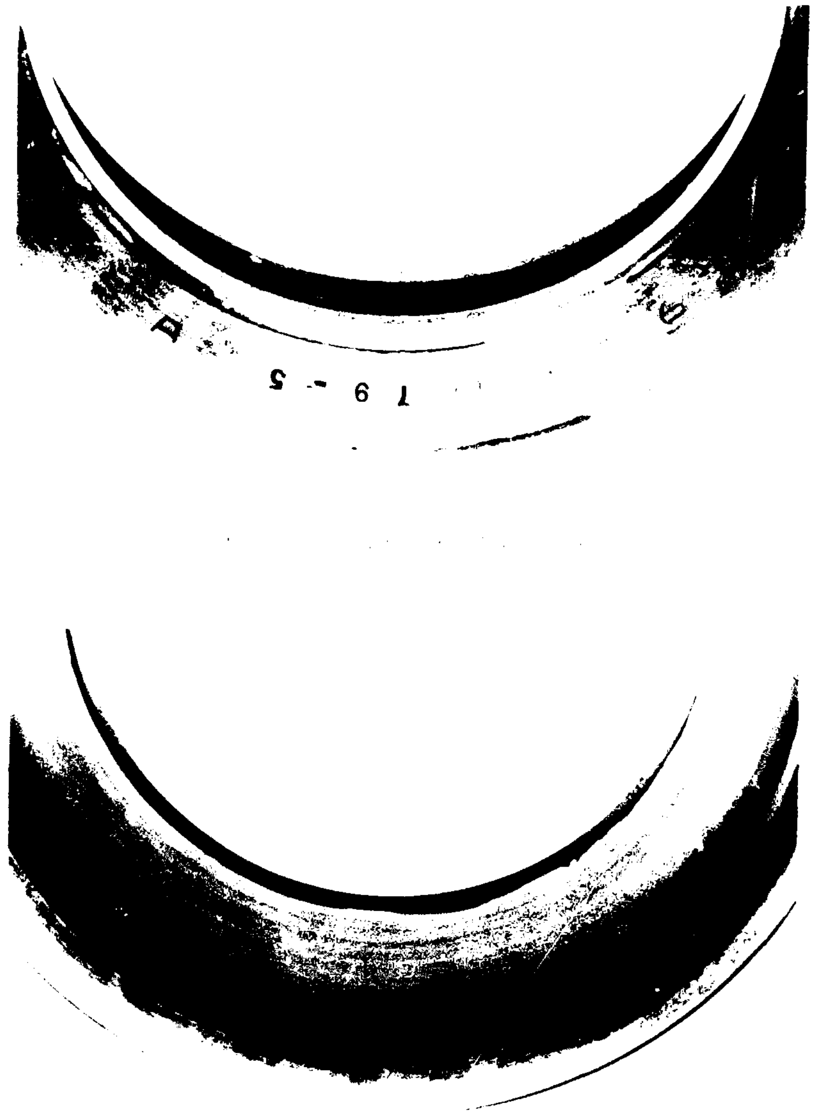
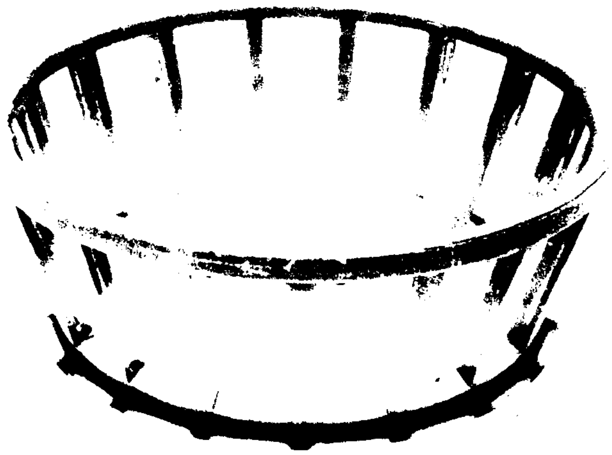


Figure C-21. Cup, Rollers, and Cage Used at Drive End of Shaft 78-4 in Oil-Off Test No. 6 (Sheet 1 of 2).

ROLLERS



CAGE 78-23

Figure C–21. Cup, Rollers, and Cage Used at Drive End of Shaft 78-4 in Oil-Off Test No. 6 (Sheet 2 of 2).

CUP 78-9 VIEWED FROM SMALL END

CUP 78-9 VIEWED FROM LARGE END

Figure C–22. Cup, Rollers, and Cage Used Opposite Drive End of Shaft 78-4 in Oil-Off Test No. 6 (Sheet 1 of 2).

ROLLERS

CAGE 78-25

Figure C-22. Cup, Rollers, and Cage Used Opposite Drive End of Shaft 78-4 in Oil Off Test No. 6 (Sheet 2 of 2).