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ADVANCED TURBINE ENGINE MAINSHAFT LUBRICATION SYSTEM INVESTIGATION

PHASE II

Part I - Background, Test Elements and Results, and Conclusions for System Performance

by

W. L. Rhoads and L. A. Peacock

prepared for

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

NASA Lewis Research Center
Contract NAS3-6267
William R. Loomis, Project Manager

RESEARCH LABORATORY
SKF INDUSTRIES, INC.
ENGINEERING AND RESEARCH CENTER
KING OF PRUSSIA, PA.

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FINAL REPORT

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Fluid System Components Division

RESEARCH LABORATORY
SKF INDUSTRIES, INC.
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FOREWORD

The research described herein, which was conducted by the SKF Industries, Inc. Research Laboratory, was performed under NASA Contract NAS3-6267. The work was completed under the management of the NASA Project Manager, Mr. William R. Loomis, Fluid Systems Components Division, NASA Lewis Research Center.

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ABSTRACT

Ball bearings and face seals for use on Mach 3 aircraft gas turbine engine mainshafts have been evaluated in this program with several selected lubricants in a recirculating oil system having provisions for inert gas blanketing. Testing has been conducted at typical advanced engine load and speed conditions with the seals exposed to 1200°F hot air and a pressure differential of 100 psi.

Using M-50 and WB-49 tool steel ball bearings of current design with two of the most promising fluid lubricants (two synthetic paraffinic hydrocarbon base stock fluids, one with a heavy paraffinic resin additive and the other with a proprietary anti-wear additive) found in the screening test tasks of this program and newly developed hydrodynamic lift design oil seals, it was possible to run 250-hour tests at 650°F bearing outer ring temperatures with no signs of distress. Reliable longer-term inerted operation at representative advanced engine conditions appears to be feasible.

In a separate task of this work, 125 mm-bore bearings and face seals were run successfully for short periods at speeds to 20,000 rpm corresponding to a DN value (product of bearing bore in millimeters and shaft speed in rpm) of 2.5 million and a seal face speed of 550 feet per second.

Part I of this report contains a presentation and discussion of the system performance test results while Part 2 (CR-72873) contains a detailed writeup of face seal performance, a mass spectroscopic study of a test fluid, and supplementary test data tabulations.

FINAL SUMMARY REPORT ON PHASE II
ADVANCED TURBINE ENGINE MAINSHAFT LUBRICATION SYSTEM INVESTIGATION

SUMMARY

This is the Phase II Final Summary Report under NASA Contract NAS3-6267 and covers the work done from June 23, 1967 through September 7, 1970.

The performance of aircraft gas-turbine mainshaft ball bearings, seals, and lubricants under simulated high-performance engine conditions (Mach 3) has been studied under NASA Contract NAS3-6267 (Phase II), using the most advanced materials, designs, and manufacturing techniques available. A circulating oil system was used for this investigation. Both inert gas blanketing and open atmosphere were utilized at high temperatures.

One test machine and related systems with recirculating oil was used for the evaluation of 125-mm bore ball bearings, with both bellows type and hydrodynamic lift design face seals (both supplied under subcontract by Koppers Company, Inc.) and five candidate lubricants. All systems and components were designed to operate at 14,000 rpm shaft speed, 3280 lbs. thrust load on the test bearing (AFBMA computed L_{10} life = 500 hours), 600°F to 800°F bearing temperature, 100 psi differential pressure across the test seals, and 1200°F air at the outboard test seal.

Test bearings and seals were provided under Task I of this project. Test bearings made of M-50 steel rings and balls and WB-49 steel rings with M-1 steel balls were used. Test oil seals having AM350 steel bellows with sealing rings of USG2777 and CDJ83 carbon-graphite in conjunction with hard chromium and chromium carbide plated shoulders were used as well as hydrodynamic lift design seals (piston ring secondary) with CDJ83 carbon and a tungsten carbide plated shoulder. Test air seals had bellows of AM350 and Inco 718 with 56HT and CDJ83 carbon sealing rings.

Under Task II, screening tests of three hours' duration at various specifically defined conditions have been run with Mobil Jet II, an ester meeting the Mil-L-23699 specification; Mobil

XRM-154D, a modified silicone oil, and Mobil XRM-109F, a synthetic paraffinic hydrocarbon with the addition of 10% by weight of Kendall Heavy Resin 0839 (highly refined, high molecular weight paraffinic resin). These screening tests were a continuation of those run in Phase I of this contract, using inert gas blanketing. The fluids tested represent promising candidates which came to light after the Phase I tests were conducted. It was found that the ester ran successfully at a bearing outer-ring temperature of 550°F. Although some surface distress was present, the silicone ran successfully at 650°F and the blend of resin and synthetic hydrocarbon ran successfully at 680-695°F, the highest test temperature required in this Phase. Under Task III one baseline and two qualifying tests up to 50 hours duration and two endurance tests of 250 hours duration were run at specified test conditions, all in increments of no more than 10 hours continuous running. The baseline test was run without inert gas blanketing using Mobil Jet II and ran 50 hours at 500-530°F. Mobil XRM-109F blended with 10% by weight Kendall Heavy Resin 0839 and a blend of Mobil XRM-109F, Mobil XRM 127B (less viscous version of XRM-109F) and 10% Kendall Heavy Resin 0839, in such quantities to give the blend the same bulk viscosity as Mobil XRM-109F, were used in 50-hour qualifying tests at 650°F with inert gas blanketing.

Mobil XRM-177F and Mobil XRM-109F plus 10% Kendall Heavy Resin 0839 were used in the endurance tests. Both successfully reached the specified 250 hours at 650°F bearing outer ring temperature.

In Task IV a conventional 125 mm-bore bearing was run successfull at speeds to 20,000 rpm (2.5 million DN). Testing was conducted at four speed increments, of approximately 2 hours duration each, of 14,000, 16,000, 18,000 and 20,000 rpm respectively using Mobil Jet II lubricant under open atmosphere conditions. Bearing outer ring temperature varied from 500°F at the lowest speed to 665°F at the highest speed. Results of this Task indicate that operation for short periods at 20,000 rpm is feasible using conventional bearings together with seals similar to those used under Task III. Results also indicate that operation at even higher DN values might be possible, perhaps up to the 3.0 million DN for which some advanced engines are being designed. At speeds of 2.5 million DN and higher, howerver, the high centrifugal force on the balls in these test bearings could result in early classical fatigue spalling at the outer race contacts.

Testing in Phase II has shown that extended operation (at least 250 hours) under the inerted extreme temperature conditions specified for Mach 3 aircraft turbine engines, is possible. While carbon face oil seals with a bellows secondary have proven to be troublesome at the test conditions utilized, short-term (30 hours) satisfactory operation was realized by design and material optimization. The carbon-face air seal with bellows secondary proved trouble-free giving lives of over 570 hours. The most encouraging seal development of the program was a carbon face oil seal with a piston ring secondary and hydrodynamic lift shoulder which gave trouble-free operation at test conditions for over 500 hours at 14,000 rpm. A similar seal also operated successfully during much shorter tests which were run at speeds up to 20,000 rpm. From the appearance of the tested bearing cages, the ion deposited silver cage plating seemed to perform about as satisfactorily as the standard electro-silver plating.

INTRODUCTION

A. Background

In the gas turbine engine, the turbine wheel and compressor are required to rotate at high speed to operate efficiently. By virtue of their design, each exerts a high axial load, the difference between these loads being taken by the bearings supporting the shaft connecting these two components. As greater performance is called for from manned flight vehicles, such as the supersonic transport, the engines must provide more thrust and achieve higher overall efficiency. The main shaft bearings are thus called upon to operate satisfactorily at higher speeds under greater thrust loads and in high-temperature environment created partly by the frictional heating of the external airframe surfaces and partly by the increased turbine operating temperatures.

Due to the need for a small frontal area of the aircraft and minimum weight, the space for the bearing and its cooling and lubrication equipment is minimal. The bearing itself must, therefore, be capable of operating at high temperatures (600°F and above). As a consequence, the fluids used to lubricate it must be satisfactory at these temperatures. To prevent the combustion gases from reacting with the bearing surfaces or the lubricating fluid, adequate seals are required. Also substantial improvements in the performance and satisfactory operating life of most lubricants at these extreme temperatures can be expected if the bearing chamber and lubricating system is blanketed with an inert gas. This calls for seals that can maintain low leakage rates for long periods if inert gas blanketing to reduce lubricant degradation is to be practical.

Two basic methods of lubrication have been investigated in this program which are compatible with inert blanketing. These are, firstly, oil-jet lubrication of the bearing, simply replacing the air in the bearing chamber and oil sumps by nitrogen, and secondly, using the nitrogen as a carrier for the oil, in mist form, to the bearing and allowing a large portion of the heat generated by the bearing to be removed by gas cooling. In the second system, the oil is not recovered and thus the system is of the once-through type, dispensing with the need for oil coolers and scavenge pumps and also with many of the thermal stability properties necessary in the recirculating oils. Phase II of this program, reported herein, continued Phase I screening tests of promising fluids and obtained extended duration testing (50 to 250 hours) at representative test conditions. All testing on Phase II was conducted with the recirculating oil system.

B. Research Objectives

It has been the purpose of this program of research to investigate the limits of operational feasibility of using the best available bearings, seals, and lubricants in high-temperature lubrication systems under conditions simulating those expected in the main propulsion power units of an advanced Mach 3 aircraft. Phase II of this program, reported herein, continued and extended the work performed in Phase I. The Phase II program was divided into four separate tasks.

Briefly these tasks encompass the following:

Task I - Procure sufficient bearings, seals, lubricants, and other hardware to conduct the test program.

Task II - Perform a series of screening tests (3 hours at specified test conditions) with three candidate fluids.

Task III - Extended duration tests (baseline, qualifying, and endurance) of 50 to 250 hours each at specified conditions.

Task IV - High speed testing (up to 20,000 rpm, 2.5 x 10^6 DN).

TEST FACILITY

A general schematic drawing of the test rig is shown in Enclosure 1. A photograph of the recirculating test rig is shown in Enclosure 2. Detailed descriptions of the apparatus and its capabilities are given in (1)* This equipment was used in testing in Phase I of the contract.

TEST ELEMENTS (Task I)

A. Bearings

The test bearing design selected for this program is a 125-mm bore split-inner-ring, angular-contact ball bearing; the type most widely used in aircraft propulsion turbines. This design, which permits a maximum ball complement by virtue of the separable inner-ring halves, can support high thrust loads in either direction. The high race lands permit the acceptance of large thrust loading without overriding the available groove area. The separable ring feature also permits the use of a precision-machined one-piece cage which is required for high-speed high-temperature operation. These cages normally are designed so that the balls are retained both radially inward and outward for ease of assembly in engines. Cages for test bearings on this program, however, are made so that the balls are retained radially inward only in order to facilitate changing the balls to produce different material combinations.

The initial bearing design (designated Series I) was based on an analysis presented in Appendix III of the Phase I First Semiannual Report (1). Further studies of the kinematics and elastohydrodynamics of high-speed thrust bearings resulted in a design modification (designated Series II), which increases the ratio of ball spin torque on the outer ring to that on the inner ring and reduces the frictional heat generation and thus the power consumption when compared with the original design. A summary of the design parameters is included in (2).

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*Numbers in parentheses designate references as listed on pages 38 and 39.

1. High-Speed Thrust Bearing Design and Lubrication

The design of the mainshaft angular contact thrust bearings for aircraft gas turbine power plants is based on a computation of the mean effective load (cubic mean load) to which the bearing will be subjected during engine operation between overhauls. The bearing life usually is computed using AFBMA standards for industrial applications and standard bearing steel and then a life-increase factor is applied to allow for the life improvement due to the use of vacuum-melted steel and favorable high-speed lubrication conditions based on bearing tests. This procedure has evolved to the point where it is now common practice to design engine mainshaft thrust bearings operating at 1.0 to 1.5 x 10^6 DN (bore diameter in mm times shaft speed in rpm) for an AFBMA L_{10} = 500 hrs. (10% probability of failure) and to use a life-increase factor of 6 for an engine design life between overhauls of 3000 hrs.

It is the current experience among airline operators that jet-engine overhaul times as long as 20,000 hrs. are being realized without large numbers of bearing failures in service. This favorable experience is due partly to the fact that the modern bearing steels used in aircraft bearings have a life of more than six times the AFBMA-rated life, and that the cubic mean loads for which these bearings were originally designed were estimated conservatively high and are rarely, if ever, realized in airline service.

The AFBMA rating method, of course, does not take into account the detailed design characteristics of aircraft bearings, such as special ball-groove conformities (an industry standard conformity is assumed), nor the effects of the extreme operating conditions of speed and temperature to which jet engine bearings are exposed. Special methods have had to be developed to handle these effects.

2. Ball Motion and Force Balance at High Speed

As the operating speed on an angular-contact ball bearing (one that carries thrust load) is increased, the centrifugal force on the balls becomes equal to or greater than that portion of the applied bearing thrust load carried by each ball. This large centrifugal body force on the balls increases the ball load on the outer ring and forces the ball contact angle to decrease on the outer ring and to increase on the inner ring. Since the axis of rotation of each ball forms an angle with the bearing axis, the ball axis must change direction as the ball orbits around the bearing, thus generating a gyroscopic moment as an additional body force on the balls requiring further adjustments in the inner and outer ring contact angles to obtain an equilibrium of forces on the balls.

A theoretical analysis of the centrifugal and gyroscopic ball forces in high-speed angular-contact bearings has been programmed for a computer by Jones (3), Harris (4) and others, for use in the design of jet-engine bearings. In order to obtain computer results, it is necessary to assume the type of friction that exists at the ball-race contacts. The assumption normally used is that of Coulomb friction, from which it was deduced that the balls must roll without sliding or spinning (about an axis running through the ball center and the center of the contact with a ring) on one of the rings. Spinning then must occur on the other ring, as well as rolling, but gross sliding, or skidding, is not allowed in any currently used ball bearing analyses. Instead, since the ball gyroscopic moments must be absorbed by tangential forces at the ball-race contacts, the current computer analyses predict the minimum coefficient of friction required to prevent skidding due to the gyroscopic moment.

3. Elastohydrodynamic Effects

Several aspects of recent elastohydrodynamic research are considered to be related to the problem of high-speed bearing design and lubrication. It is shown in elastohydrodynamic (EHD) theory that a lubricant film considerably thicker than the surface roughness can exist even at a heavily loaded rolling contact. For small amounts of sliding, the film thickness based on an isothermal approach is found to be negligibly different from that which is obtained when sliding and thermal effects are considered. The minimum film thickness of the elliptical contacts in a ball bearing have been computed by SKF using a modified line-contact formula . Design charts for the calculation of minimum oil-film thickness at the ball (or roller) raceway contact of all types of rolling bearings have been prepared by SKF. These charts present film thickness as a function of shaft speed, load, lubricant properties and a geometry factor determined by the dimensions of the particular bearing.

Surface microgeometry (roughness) analyses have been conducted at SKF on practical bearing surfaces so that the oil film thickness can be compared to the critical roughness parameter for bearing contacts. The ratio of the thickness to the roughness has been shown to be an important guide for predicting the degree of asperity contact and resulting interactions and surface damage affecting bearing performance.

The frictional traction of the surfaces in rolling and sliding can be estimated by elastohydrodynamic theory. It has been calculated that the friction traction of cylinders in pure rolling is small, the computed value of the coefficient of friction in a pure rolling bearing being of the order of 10^{-4}. The frictional traction due to sliding is known to be caused by the shearing of the oil film in which the viscosity is dependent not only upon the pressure in the contact, but also upon the heat generation due to viscous shearing. Therefore, the determination of frictional traction under rolling and sliding conditions is much more complex than that of minimum film thickness and an isothermal approach is inadequate. It is now known that for lubricated cylinders the friction increases nearly linearly with the sliding rate only at very low sliding rates. Due to the thermal effect, the traction decreases with increasing sliding rate after passing through a peak.

4. Centrifugal Expansion of Bearing Parts

At bearing operating speeds higher than about 1.5×10^6 DN, the centrifugal forces on the rotating ring and cage become large enough to result in centrifugal expansions that are significant fractions of the design clearances between these parts and other parts of the bearing. Such expansions alter the operating characteristics of the bearing and therefore, must be accounted for in bearing design for operation over a wide range of speeds in the same way that the shaft and housing fits and thermal expansions must be accounted for to permit the maintenance of the intended bearing design clearances over a wide range of operating temperatures.

The Series I bearing tested in this program is designed with a nominal static mounted contact angle of 26° at the design temperature, for which the internal radial looseness of the bearing is 0.0048". Centrifugal expansion of the inner-ring ball path at 14,000 rpm shaft speed (1.75×10^6 DN) is almost 0.002", as determined by an elasticity analysis (Appendix VII of (1)), resulting in a reduction of the nominal contact angle to 21.7°, which is not intolerable. At 20,000 rpm shaft speed (2.5×10^6 DN), however, the inner-ring expansion is 0.004" and the nominal contact angle decreases to 16.3°, causing drastic changes in bearing operating characteristics.

The centrifugal expansion of bearing cages differs from that of the rings in that the centrifugal force produced by the entire cage mass must be carried by the thin cage rails, which therefore stretch more elastically than the bearing rings, even for high-elasticity steel cages. The effect on the 125-mm bore bearing cage tested in this program at 14,000 rpm shaft speed is twice as pronounced at 20,000 rpm. No exact elasticity analysis exists for predicting the centrifugal expansion of a bearing cage; however, a conservative estimate (upper limit) can be made by assuming that all of the mass of the cage is concentrated in the rails which are unrestrained elastically by the cage bars. This assumption, of course, neglects the effects of the centrifugal warping of the cage rails, but nevertheless predicts a diametral expansion of the cage guiding surface of about 0.008" at a shaft speed of 14,000 rpm and 0.016" at 20,000 rpm.

High-speed bearing steel cages as presently designed, therefore, probably have centrifugal expansions greater than the centrifugal inner-ring expansions, even though their rotational speeds are slower. It is possible that modifications to the cage geometry would result in a reduced centrifugal cage expansion and reduced warping of the cage guide surfaces. (Such modifications would also increase the cage surface area which may be desirable for the additional cooling required during high-speed operation.) If the cage centrifugal expansion could be reduced to the same magnitude as the inner-ring expansion, there would be some advantage of inner-ring guiding of cages over outer-ring guiding, where the clearance is reduced drastically at high speed. Otherwise, outer-ring guiding allows the design of a tighter operating clearance at high speed when accurate cage guiding is required to center the cage and maintain balance.

(There is also some field experience indicating that outer-ring cage guiding provides better lubrication to the cage guide surfaces under meager lubrication conditions, since all lubricant flung centrifugally outward within the bearing must go out through the cage guide clearance.)

Enclosures 3 and 4 show the test bearings used in this program. Both WB-49 steel bearings (459980H) and M-50 steel bearings (459981G) were utilized. Reasons for the selection of these two steels has been discussed in (1). A few M-1 steel electro-silver-plated cages shown in Enclosure 5 were used; however, most cages were ion-silver-plated 4340 steel (one of the purposes of these tests was to evaluate the relative adherence properties of ion-deposited silver).

B. Seals

The dual test seal arrangement shown in Enclosure 6 has been used in all testing to date. This arrangement consists of an oil face seal (either with a bellows or piston ring secondary) with a 105 psi pressure drop across it and an air bellows face seal with a 5 psi pressure drop across it.

Several variations of materials have been used in both the air and oil seal locations. The particular variations utilized in each test are specified in Enclosure 7. The oil seal and shoulder designs are shown in Enclosures 8, 9, and 10 and the basic air seal and shoulder are shown in Enclosure 11. Both type bellows seals have alloy or steel bellows and the carbon face is machined with a dam so located that the bellows is essentially pressure balanced. Carbon pads or lands are provided (inboard of the sealing dam on the oil seals and both inboard and outboard of the sealing dam on the air seal) to distribute the bellows spring face load and residual gas pressure unbalance over a large contact area. The carbon contacts the shoulder, made of steel and plated with wear resistant material, which rotates with the shaft. The other type of oil seal has a carbon face with piston ring secondary and a shoulder provided with hydro-dynamic lift pads. This type of seal has proven most satisfactory and has solved the recurring problems observed in this location when bellows oil seals having standard carbon face design are used.

Tested seals requiring the replacement of carbon rings or replating of the shoulder are reworked by Koppers Company on a subcontract basis. The lapping of both carbon and shoulder surfaces when necessary is done by SKF Laboratory personnel using lapping plates and techniques supplied by Koppers. A complete discussion of seals and seal test results throughout both Phases I and II of the program is included as Appendix I of Part 2 (5).

C. Lubricants

Temperature-visoosity data for the lubricants evaluated in Phase II testing are presented in Enclosure 12. These fluids are:

1. Mobil Jet II- an ester-base formulation meeting the Mil-L-23699 specification. This oil is a second generation of a type of oil previously supplied to Mil-L-7808 specifications. In its present formulation more stable base stock components have been used which were not previously available.

2. Mobil XRM-154D - a modified silicone oil which according to the supplier has had long chain alkyl groups added to its structure either in addition to or in place of the normal methyl or phenyl groups which one finds in the siloxane structure of the current silicone oils.

3. Mobil XRM-109F plus 10% (by weight) of Kendall Heavy Resin 0839 - XRM 109F is a paraffinic hydrocarbon material synthesized by the polymerization of a fairly pure mono-olefin so that it can be a single chemical species composed of molecules of a chain length distribution depending on the type, method and degree of polymerization. It has good resistance to thermal degradation and is susceptible to additive improvement. The Kendall Heavy Resin is a very high molecular weight paraffinic residual (4500CS at 100°F)* of a Pennsylvania crude oil which has been subjected to a rigid super refining process, the details of which are proprietary. Properties data are given in (6). In bench scale experiments it has shown properties of an additive capable of enhancing the lubricating qualities of other hydrocarbon oils and in this project was used to test the possibility of replacing conventional synthetic anti-wear additives.

* This represents typical manufacturer's data (6). The actual samples used for blending were all around 6000CS at 100°F.

4. Mobil XRM-109F plus Kendall Heavy Resin plus Mobil XRM-127B. This is a blend of XRM-109F plus 10% Kendall resin with enough (approximately 10%) lower viscosity hydrocarbon designated as Mobil XRM-127B added to bring the bulk viscosity back to that of XRM-109F (44.26 cs @ 100°F). This mixture was tested to verify that the good performance of the blend tested in Task II was due to the additive effect of the Kendall resin and not to the (relatively small) bulk viscosity increase of the test fluid.

Mobil XRM-127B is a synthetic paraffinic hydrocarbon of the same chemical family as XRM-109F, but with a viscosity of 62.7 cs @ 100°F.

5. Mobil XRM-177F - is the synthetic hydrocarbon base XRM-109F with a proprietary boundary lubrication additive added.

TEST RESULTS

A compilation of test results is presented in Enclosure 13. This compilation includes results from Phase I testing as well. (Task IV results are also given with additional details in Enclosures 47 and 52). Measurements of lubricant viscosity and acid number before and after each recirculating oil test are given in Enclosure 14.

A. Task II - Screening Tests

1. Mobil Jet Oil II

In the first two tests with Mobil Jet II oil, a new WB-49 steel test bearing was used along with an AM350 steel bellows oil seal (USG2777 carbon) and an Inco 718 bellows air seal (CDJ83 Carbon). Each of the shoulders was chromium plated. All seal components were reused in the second test after only minor relapping of their mating surfaces. Both tests were nitrogen gas blanketed.

The first test ran 2.2 hours at a bearing outer-ring temperature of 540°F and an oil inlet temperature of 450°F when loss of oil through the test chamber vent, caused by excessive oil-seal leakage and subsequent bearing failure, caused termination of the test. The oil seal performed well at the beginning, leaking only about 3 scfm, which increased during the test to 10 scfm.

It was found at disassembly that the test bearing cage had seized on the outer ring lands. There was moderate cage pocket wear and debris of silver on all parts of the bearing; however, the ball tracks appeared in reasonably good condition. There was no evidence of oil coking on any of the bearing cavity parts. A sample of the oil showed that the viscosity increased from 27.8 cs (new oil) to 28.1 cs at 100°F and the acid number also increased from 0.1 to 0.2. No measureable wear was observed on either the oil or air test seals. The oil seal and shoulder were slightly scored. Photographs documenting the test are included in Enclosure 15 and 16. Data sheet III-1 for the test is included in Appendix III of Part 2 (5).

A second test with Mobil Jet II oil was run for one hour and was terminated due to high oil seal leakage before full test conditions were reached. To avoid the type of cage seizure observed in the first test, the outer-ring lands were ground to increase the new cage clearance to 0.030". The same seals used in the first screening test were reused after reconditioning. The oil seal mounting flange of this seal was cut back by 0.020" to give an increase of approximately 15% over the net face load used in the previous test in an attempt to prevent seal lift-off.

The test bearing was found to be in good condition with slight cage wiping in the outer ring lands and rather heavy pocket wear. The oil seal and shoulder were in poor condition with the seal carbon completely worn down and the shoulder face badly scored. The air seal and shoulder appeared in good condition. The oil viscosity showed a decrease from 27.8 cs (new oil) to 23.9 cs while the acid number increased from 0.1 to 0.2. Photographs documenting this test are included in Enclosure 17 and 18. Data sheets III-2 for this test are in Appendix III of Part 2 (5).

Because this was not considered a valid test of the lubricant in the 600-700°F temperature range a third test of the same bearing was run. For this test a new oil seal with AM350 bellows (CDJ83 carbon) and a new shoulder with a chromium carbide plated surface were used. Total running time of this test was 1.4 hours of which 0.2 hours were at 600°F test conditions. The oil seal leakage was consistently in the neighborhood of 1 scfm while air seal leakage varied between 3 and 4 scfm. At an outer-ring temperature slightly above 540°F the test bearing began to lose thermal stability (as evidenced by several inner-ring temperature excursions) and required considerable manipulation of the I-housing heaters, shaft cooling, and oil flow to control both the rise of inner and outer ring temperatures as well as the differential between the two rings. Test termination was caused by test shaft seizure.

Upon disassembly it was found that the test bearing was badly smeared on both halves of the inner as well as the outer ring. The "unloaded" half of the inner-ring was found to have a radial crack, considered to be due to the sudden seizure. The cage rails had wiped the outer-ring, as would be expected under the circumstances, and heavy ball pocket wear was found. No evidence of coking was seen on any rig part except for the test bearing, which was heavily coked. The oil seal and air seal appeared in rather good condition with about .001" of wear on each carbon face. Both seal shoulders had "chatter" marks.

Photographs documenting this test are included as Enclosure 19 and 20. Data sheets III-3 for the test are included in Appendix III of Part 2 (5). A more detailed description of all tests in this section can be found in (7, 8).

2. Mobil XRM-154D (Silicone Fluid)

The first of two test runs with Mobil XRM-154D oil ran a total of 2.3 hours using a new WB-49 steel bearing, an AM350 steel bellows oil seal (CDJ83 carbon), and a chromium plated oil seal shoulder. The air seal bellows was Inco 718 (CDJ83 carbon) with a chromium carbide plated shoulder. The test ran for 1.2 hours at 470° to 490°F bearing outer-ring temperature with an oil inlet temperature of 460°F. While attempting to increase outer-ring temperature, an outer-ring temperature excursion and subsequent rig seizure forced shutdown. During the test the oil seal leakage held constant at 1 scfm except at the end of the test when a seal lift-off occurred with a resultant sudden increase in bearing chamber pressure (above 25 psi) which blew the oil charge out through the venting system.

Upon disassembly it was found that heavy oil deposits (some "gummy", some dry white powder) were on all the rig parts exposed to the oil. It is possible that these deposits contributed to the oil seal lift-off and the bearing thermal excursion. It should be noted that a high oxygen content (1.0 vs. 0.5% permitted) was observed in the bearing chamber at one point in the test.

Both air and oil seal carbons were in reasonably good condition with negligible wear. Some chatter was evident on the oil seal shoulder but not as much as in the previous Mobil Jet II oil test. The test bearing had heavy oil deposits. Evidence of thermal take-up of internal clearances (running on the unloaded half of the inner-ring), moderate surface distress in both ball tracks, heavy cage pocket wear and contact between the cage O.D. and bearing outer ring were found. Enclosure 21, 22 and 23 present photographs of this test run. Data sheet III-4 for this test are included in Appendix III of Part 2 (5).

Because of the high oxygen content present which may have contributed to the heavy build-up of deposits in the test bearing chamber a second test was run. This test was run for a total of

3 hours at 630-650°F bearing outer-ring temperature, using an AM350 steel bellows oil seal (CDJ83 carbon) with a chromium carbide plated shoulder and a AM350 steel bellows air seal (56 HT carbon) with chromium - carbide plated shoulder. The total seal leakage was 1.4 to 1.7 scfm with the majority going through the oil seal for the first 2 hours then splitting evenly in the final hour between the oil and air seals. At this time, an attempt was made to increase the outer-ring temperature from 600° to 700°F but an oil-seal lift-off occurred and the bearing cavity pressure increased from 6 to 20-25 psig. In an effort to get the oil seal to reseat the test was allowed to continue for 15 minutes in this condition during which time 2 gallons of makeup oil was added to the sump. The seal did not reseat however and the rig was shut-down.

Upon disassembly, the test bearing was found in good condition. The air seal and shoulder were in good condition with negligible wear. The oil seal carbon had worn a negligible amount but its mating shoulder was heavily worn, with up to 0.005" deep grooves at the location of the carbon contact. It was noted that during this test the oil seal shoulder ran 50° to 100°F hotter than it did during a previous test run at similar test bearing and oil-in temperatures using Mobil XRM-109F plus 10% Kendall Heavy Resin . This may indicate that the silicone fluid is a poorer heat transfer medium or that its heat generation or deposit forming characteristics are different in the rubbing seal interface.

In spite of the low oxygen content in this test (0.009 to 0.02%) some oil deposits (less than previous test) were still visible in the bearing housing after the test. This fluid seems to provide adequate lubrication but decomposes at the test temperature and produces deposit which are considered unacceptable for long-term operation. Therefore no further testing was undertaken with this fluid.

Enclosures 24 and 25 presents photographs of the test elements from this run. Data sheets III-5 for this test are included in Appendix III of Part 2 (5).

3. <u>Mobil XRM-109F + 10% (by weight) Kendall Heavy Resin 0839</u>

One test was run with this fluid using a WB-49 steel bearing, a new (reconditioned) AM350 steel bellows oil seal (CDJ83 carbon) with a chromium plated shoulder and an Inco 718 bellows air seal (CDJ83 carbon) with a chromium carbide plated shoulder. It was originally scheduled to run the test for two 3-hour periods, the first at an outer ring temperature of 550°F and an oil inlet temperature of 450°F. If no failure occurred the second period was to be run at an outer ring temperature of 600°F and an oil inlet temperature of 500°F. It was found, however, that with the Kendall additive, it was not possible to run at a bearing outer-ring temperature below approximately 620°F with 450°F oil inlet temperature. It was accordingly decided that the rig would be run at as low a temperature as was practical for 3 hours and then a 3 hour test at 700°F would be attempted.

It was possible by lowering the oil inlet temperature from 450° to 430°F and the rig-housing from 550°F to 420°F, to run the first three hours at outer-ring temperature between 580°F and 620°F.

During the test the total seal leakage stabilized at between 0.3 and 0.4 scfm. Because of the very low leakage it was impossible to determine the leakage path.

The outer ring temperature was then increased to 700°F. Approximately 0.4 hours after reaching 700°F test condition the oil seal lifted and did not seal for the remaining 2.6 hours of the run. The oil seal leakage ranged from 5 to 10 scfm during this time. Because of this high leakage the outer-ring temperature fluctuated between 670°F and 710°F until the test was terminated at the time-up of 3 hours.

Upon disassembly the test bearing was found to be very slightly glazed in the ball paths with light cage pocket wear. There was approximately 0.007" of air seal carbon wear and evidence of some carbon build-up on the shoulder. The oil seal carbon had worn 0.010" and had worn a groove about 0.002" deep in the chromium plate of the shoulder. Due to the oil seal lift-off which caused oil to be blown out of the system, it was not possible to circulate oil through the system during cool down. Therefore, a considerable amount of coking was formed on the test elements.

Enclosures 26 and 27 present photographs of test elements from this run. Data sheet III-6 for this test run is included in Appendix III of Part 2 (5)

B. Task III

1. Baseline Test

An open atmosphere baseline test was run using Mobil Jet II oil and an M-50 steel bearing with an ion silver plated 4340 steel cage. A piston-ring secondary oil seal with CDJ83 carbon and tungsten carbide plated shoulder employing a new hydrodynamic lift design were used along with an AM350 steel bellows air seal (56HT carbon) and a chromium carbide plated shoulder.

The test bearing was run for a total of 50 hours in three 10-hour increments, one 9.6-hour, one 7.6-hour, one 2.5-hour and one 1-hour increments for a total of 50.7 hours at an outer ring temperatu of 500° to 535°F with an oil inlet temperature of 390°-420°F. Odd hour increments are due to shutdowns caused by shear pin breakage and other similar occurrences.

During the first 3-hours of testing the total seal leakage was about 10 scfm. During the remainder of the test the total seal leakage was 3.8 to 6.8 scfm.

Upon disassembly the test bearing was found to have some slight surface distress on the inner and outer ring. There was also evidence of heavy ball contact in the cage pockets as well as evidence of cage land contact on about 90° of the circumference. There were some oil deposits on the bore of the test bearing housing. All sealing elements were found to be in excellent condition with 0.003" carbon wear on each seal. Enclosure 28 through 30 present photographs of test elements from this run. Data sheet III-7 through III-12 for this test run are included in Appendix III of Part 2 (5). More detailed description of this test can be found in (8).

2. Qualification Tests

a. Mobil XRM-109 Plus 10% Kendall Heavy Resin 0839

The first qualifying test was run using a WB49 steel bearing with an ion silver plated 4340 steel cage. An AM350 steel bellows oil seal (CDJ83 carbon) and a chromium-carbide plated shoulder were used along with an INCO 718 bellows air seal (CDJ83 carbon) and a chromium-carbide plated shoulder except during the last 1.3 hours of the test when the air seal was replaced with one having an AM350 steel bellows and 56 HT carbon.

The test ran for a total of 32.3 hours, at 640°F to 660°F bearing outer ring temperature which was accumulated in three 10-hour, one 1-hour, and one 1.3-hour increments with an oil inlet temperature of 490°F to 525°F.

After running 31.0 hours it was necessary to replace the air seal carbon which had eroded to a point where it was considered unserviceable. The rig was assembled and restarted several times (due to initial oil seal leakage) before an additional 1.3 hours at test conditions was accumulated. At this time an oil seal lift-off occurred with the resultant loss of sealing necessitating shut down of the rig.

Upon disasembly it was noted that the oil seal shoulder showed signs of chatter in the groove (0.002 to 0.003 inches deep worn by the carbon). The air seal shoulder was in good condition. The test bearing was found to be in very good condition with very light glazing in the ball paths and light cage pocket wear.

Enclosures 31 to 33 present photographs of the test elements from this test. Data sheets III-13 through III-16 for this test run are included in Appendix III of Part 2 (5).

b. Mobil XRM 109F; XRM 127B, plus 10% Kendall Heavy Resin 0839

The second qualifying test was run using a WB49 steel bearing and an ion silver plated 4340 steel cage. A piston ring secondary oil seal with CDJ83 carbon and tungsten carbide plated shoulder employing a new hydrodynamic lift design were used along with an AM350 steel bellows air seal (56 HT carbon) and a chromium carbide plated shoulder.

The test ran for a total of 50 hours in four 10-hour, one 6-hour, and one 4-hour increment at a bearing outer-ring temperature of 640° to 650°F. During the first 36 hours of testing, (prior to the breaking of the nitrogen supply line to the inter seal cavity) the total seal leakage was 0.4 to 13.5 scfm. Upon restart after repairing the broken lines, leakage increased to 19.5 to 29.8 scfm. The increased leakage is thought to be the result of either seal "hanging" in a partially open position as a result of the sudden depressurization of the inter seal cavity.

When the test rig was disassembled the test bearing and both the air and oil seals were found to be in very good condition. The carbon wear on the oil seal was negligible and the air seal carbon was worn 0.005". Both seal shoulders were in good condition with some chatter marks on the air seal shoulder. Some oil deposits were found in the bearing cavity and on the unloaded half of the bearing inner-ring. Photographs documenting this test are presented in Enclosures 34 through 36. Data sheets III-17 through III-21 for this test are included in Appendix III of Part 2 (5).

3. Endurance Tests

a. Mobil XRM 177F

The first 250-hour endurance test was run in two parts (permission received from project manager). Twenty hours were run at an earlier date as a checkout test using a WB49 steel bearing of proven design with an ion-silver plated 4340 steel cage at an outer ring temperature of 600°F. An AM350 steel bellows oil seal with CDJ83 carbon and a chromium plated shoulder were used along with an Inco 718 bellows air seal with CDJ83 carbon and a chromium carbide plated shoulder.

After running the first 7.1 hours in this test, a failure of a motor pulley component caused the test rig to be shut down. After this was corrected, it was found that the air and oil seals were hung open. Total seal leakage during this period was 1.7 to 2.4 scfm with a majority going through the air seal. The rig was disassembled to free the seals at which time they were visually examined. The oil seal and shoulder were in good condition with no evidence of chromium wear. Only the outer wear pad of the air seal carbon was making contact with the mating shoulder which explains the leakage across the seal.

After reassembly the rig was re-started and brought to test conditions. It ran at the previously stated conditions for an additional 12.9 hours after which time the rig was shut down due to failure of the test shaft support (roller) bearing which evidently resulted from an oil seal lift off and resultant oil loss through the vent system. For the first (approximately) 7 hours of this second test segment, total seal leakage was again very low. However, intermittent oil seal lift-off, lasting approximately 15 seconds at a time, occurred at this point and continued through the remainder of the twenty hour run. Leakage rates during the lift-off periods were as high as 20 scfm.

Upon disassembly, the working surfaces of the test bearing were found to be in excellent condition. The air seal and shoulder were in very good condition with negligible carbon wear. The contact area on the carbon had widened so that the sealing dam was making full contact with the shoulder. The oil seal shoulder was heavily worn, with grooves at the location of the carbon contact up to 0.009" deep, penetrating (going completely through) the chromium plating. The carbon had worn 0.010" and the edges of the carbon were rounded. No evidence of oil coking was found in the rig. Photographs documenting this test are included as Enclosures 37 and 38.

The remaining 230 hours of this endurance test was begun at a later date using a new M-50 steel bearing, as contractually required in the endurance tests with an ion silver plated 4340 steel cage. (M-50 steel is more commonly used in aerospace bearings than WB49 steel and apparently is capable of satisfactory operation at temperatures higher than 600°F (9).) An oil seal with a carbon piston-ring secondary, CDJ83 primary carbon face, and a tungsten carbide plated shoulder incorporating a hydrodynamic lift pad design was used along with an AM350 steel bellows air seal with 56HT carbon and a chromium carbide plated shoulder.

The test was run at a bearing outer-ring temperature from 640° to 670°F and an oil inlet temperature of 490° to 530°F.

During the first 3.2 hours of testing (specified conditions not reached) a total of 5 shutdowns occurred due to various rig malfunctions. After this series of problems was resolved the test was restarted and run for ten hours at conditions without incident and was then shutdown manually to cool. During the ten hour test period the total seal leakage ranged from 19.5 to 4.7 scfm in the first 4 hours and 8.0 to 9.0 scfm for the remaining 6 hours. The rig was then restarted and after an additional 2.8 hours of running at an outer-ring temperature ranging from 600° to 620°F, it became necessary to shut down the rig due to excessive seal leakage on the order of 24 to 49 scfm. Upon disassembly the piston-ring seal carbon was found to have worn 0.048" the hydrodynamic lift shoulder was grooved to a depth of 0.005-0.010" and the lift pads were worn away. The used seal and shoulder assembly from this test were then replaced with a more serviceable seal shoulder assembly of the same design but which had been used previously in two 50-hour tests (for a total of 100 hours).

The rig was restarted and the remaining test hours were run in fifteen 10-hour, one 10.2-hour, one 9.1-hour, one 8.5-hour, one 7.8-hour, one 6.5-hour, one 6.0-hour, one 4.7-hour, one 4.4-hour, one 2.1-hour, one 1-hour and one 0.8-hour increments.

During this time the total seal leakage ranged from a low of 3.0 scfm to an occasional high of 15.3 scfm.

Upon disassembly the oil-seal and shoulder were found to be in excellent condition. The oil seal carbon wear was 0.002" with negligible shoulder wear. The air-seal was also found in very good condition with carbon wear of 0.006 to 0.007". The shoulder showed negligible wear.

The test bearing was in good condition, however the cage silver plating (ion-deposited) had flaked off on all sides exposing the copper flash binder as shown in Enclosure 39. Also, moderate to heavy ball pocket wear (through the silver in some areas) was noted along with very light contact of the cage guide rails with the bearing outer-ring lands. All surfaces wetted by the oil were found heavily varnished.

Photographs documenting this test are included as Enclosures 40 through 42. Data sheets III-22 through III-49 of this test are included in Appendix III of Part 2 (5).

b. Mobil XRM 109F and 10% by Weight of Kendall Heavy Resin 0839

This second 250-hour endurance test was run using an M-50 steel bearing as delineated in the Scope of Work along with an ion-silver-plated 4340 steel cage. The oil seal shoulder assembly consisted of an oil seal having a carbon piston ring secondar and a CDJ83 carbon face together with a tungsten carbide plated should incorporating hydrodynamic lift pads. The air seal/shoulder assembly consisted of an air seal having an AM350 steel bellows with 56HT carbo face together with a chromium carbide plated shoulder.

In this test the bearing outer-ring temperature ranged from 640° to (occassionally) 670°F while the oil inlet temperature ranged from 490° to (occasionally) 525°F. During the first 77.5 hours of testing, which were accumulated in seven 10-hour increments and one 7.5 hour increment, the total seal leakage varied from 4.6 to 8.5 scfm. For the remaining 172.5 test hours, which were accumulated in fifteen 10-hour, one 8.5-hour, one 6.8-hour, one 4.3-hour, and one 2.9-hour increments, the total seal leakage varied from 4.2 to 12.8 scfm with leakage during the final 79.7 hours not exceeding 8.5 scfm.

Upon disassembly the oil seal and shoulder were found to be in excellent condition. The oil seal carbon had worn 0.003" while the mating shoulder showed very light wear ($\approx$ 0.0005"). The air seal was also in good condition with only 0.001" carbon wear. The shoulder showed negligible wear.

The test bearing inner and outer rings were found in good condition. However the cage silver plating (ion-deposited) had flaked off in places exposing the copper flash as shown in Enclosure 43. Also, moderate ball pocket wear and light contact of the cage guide rails with the bearing outer-ring lands was noted.

All surfaces wetted by the oil were varnished. Photographs documenting the test are included as Enclosures 44 through 46. Data Sheets III-50 through III-74 of this test are included in Appendix III of Part 2 (5).

C. Task IV - High Speed Tests

Testing on this task was conducted at four speed increments of 14,000, 16,000, 18,000 and 20,000 rpm respectively. A new M-50 steel bearing of 459981G design having an ion-silver plated 4340 steel cage was used. Test lubricant was Mobil Jet II under open atmosphere conditions, supplied at flow rates of 2.0, 1.5, 1.0, and 0.75 gpm for each speed increment. Seals for this task were similar to those used successfully in Task III endurance tests and were designed originally for 14,000 rpm operation. Test rig input power (current) was monitored at the various oil flow rates along with bearing inner and outer-ring temperature, oil-inlet and outlet temperature and total seal leakage. Test results obtained at the four speed increments are given in Enclosure 47.

The intended limits of operation for all speed phases were an oil inlet temperature of 380° to 400°F and bearing outer ring temperature of 475° to 525°F. However, increased bearing heat generation at the higher speeds plus occasional boiling of the cooling water in the oil cooler at high temperatures and possibly also some oil foaming to be discussed, resulted in higher oil and bearing temperatures as shown in Enclosure 47.

1. Testing at 14,000 rpm

Testing at the 14,000 rpm baseline condition was successfully completed, at oil flows of 2.0, 1.5, and 1.0 gpm with results given in Enclosure 47. Bearing temperature at this speed, for the three oil flows, stabilized at various points between 500° and 560°F including both rings, and oil inlet temperature between 380° and 410°F. An attempt to run at 0.75 gpm resulted in temperature excursions of both bearing rings which could only be controlled by increasing the flow rate above 1 gpm. For each oil flow setting the bearing temperature was allowed to stabilize for 1/2 hour prior to recording data.

The seal assemblies used to complete this speed increment were the same ones used successfully in the Task III endurance tests and had accrued lives, prior to this testing, of 570.1 hours for the oil seal/shoulder assembly and 480.1 hours for the air seal/shoulder assembly. These seals are described as follows with design numbers and serial numbers given in Enclosure 47. The oil seal/shoulder assembly consisted of a seal having a carbon piston ring secondary and CDJ83 carbon face along with a tungsten carbide plated shoulder incorporating hydrodynamic lift pads. The air seal/shoulder assembly consisted of an AM 350 steel bellows seal with a 56 HT carbon face along with a chromium carbide plated shoulder.

2. Testing at 16,000 rpm

After completing the 14,000 rpm increment, the speed was immediately increased to 16,000 rpm. After 15 minutes at this speed the bearing cavity pressure increased from 6 to 20 psig and the total seal leakage increased from 15 to 38 scfm indicating possible seal failure. The test rig was shut down manually for seal inspection.

Upon disassembly the oil seal shoulder was found to have three grooves (corresponding to the carbon rings of the seal) worn in it. The depth of these grooves varied from 0.001" to 0.008" indicating some misalignment may have occurred during operation, although standard indicator checks of the rotating parts during disassembly did not show any excessive run-outs. The oil seal carbon was worn 0.016". The air seal carbon required some light lapping to make it suitable for further testing. The air seal shoulder was found to have a small spall on its surface in a carbon wear track. The complete oil seal/shoulder assembly and the air seal shoulder were replaced. It was, however, later necessary to also replace the air seal with a similar design new seal due to high static seal leakage apparently caused by excessive friction on the damping springs which tended to hold the seal open. Photographs of these tested seal components are shown in Enclosure 48. The replacement seal assemblies (set #2), which were also used in the 18,000 and 20,000 rpm increments, are identical to the original seal assemblies (set #1), with the exception that the base material of the replacement air seal shoulder is Inco X alloy steel while that of the original air seal shoulder is AMS-6322 alloy (see Enclosures 7 & 47). Testing at 16,000 rpm was then successfully completed with results given in Enclosure 47. The highest bearing temperature for this speed phase was 580°F measured at the bearing inner-ring with 1.0 gpm oil flow. The corresponding oil temperatures were 430°F in, and 500°F out.

3. Testing at 18,000 rpm

In the 18,000 rpm tests, it became quickly apparent that operation at this speed with 2 gpm oil flow rate could not be maintained due to excessively high rig seal leakage rates and automatic "low oil level" alarms. The oil flow was then decreased to 1.5 gpm resulting in reseating of the rig seal and no more low oil level alarms were triggered. It was possible to run and collect data at oil flow rates of 1.5 and 1.0 gpm. At 1.0 gpm oil flow and 400°F inlet temperature, bearing inner and outer ring temperatures were 635°F and 615°F respectively. Some data was acquired at 0.75 gpm; however, bearing inner and outer ring temperatures could not be stabilized and were increasing above 640 °F and 620°F respectively. Results obtained at this speed increment are given in Enclosure 47.

The exact cause of the rig seal leaking excessively during this run and its connection with the low oil alarm is not fully known, however, it is conjectured that excessive "churning" of the oil by the bearing resulted in heavy oil foaming which completely filled the bearing cavity and caused the rig seal to lift-off. Since the oil was foaming and not returning to the sump (indeed the presence of foam may have obstructed the oil return lines) the automatic low oil level alarm went off. Lowering the flow rate to the bearing decreased oil foaming and operation became normal.

4. Testing at 20,000 rpm

In the 20,000 rpm tests, stable operation at 2.0 and 1.5 gpm flow rates was not possible for the same reasons described above; however, some data was obtained at the latter flow rate. Bearing temperatures reached 675°F and 665°F on the inner and outer rings respectively, with an oil inlet temperature of 500°F, and an outlet temperature of 565°F. Data obtained at 1 gpm flow rate are given in Enclosure 47, however these also were recorded with temperatures increasing since the test did not run long enough for the bearing temperatures to stabilize, as described below.

After running for 15 minutes at 20,000 rpm and 1 gpm oil flow rate and before temperature stabilization had occurred, with bearing inner and outer ring temperatures at 670°F and 660°F respectively, a sudden speed decrease of 2,000 rpm occurred causing the operators to believe the drive system shear-pins had failed. The rig was shutdown and the drive system inspected. The shear-pins were found intact. It was then observed that with the rig not rotating, the bearing temperature was increasing at a very rapid rate above 600°F. Total seal leakage at this time was observed to be in excess of 40 scfm. Shortly thereafter, the exterior of the test rig burst into flames which were extinguished by activation of the test cell overhead CO_2 fire prevention system and flooding the rig interior with nitrogen gas.

Evidently, during the rig shut-down process a fire started within the bearing cavity (which was filled with air) which then spread to the exterior of the rig. Upon disassembly, definite signs of fire having occurred within the bearing cavity were observed (see Enclosures 49 through 51). The cause of ignition may lie with a test shaft heat-shield sleeve which was found to have suffered a "permanent-set" of approximately 1/16" diametrial expansion causing it to become loosened and to rub against the bore of the air seal and also the front face of the air seal shoulder most likely resulting in sparks in the hot-air cavity. Since there was leakage across both seals (8.1 scfm total) prior to rig shutdown, it is conceivable that the sparks may have been swept through the inter seal cavity (also pressurized with air in these tests) and into the bearing chamber where ignition took place. It is also possible that a source of ignition could have existed within the bearing chamber itself, however, none could be isolated after a thorough inspection.

Another possible explanation for ignition involves seal leakage. During the shut-down process or while stopped, hot-air (1100°F) could enter the bearing chamber under conditions where both air and oil seals were leaking so heavily that sufficient gas flow to the inter seal cavity to maintain the normal pressure therein, was not available. (It was not noted if this occurred just prior to the observance of fire, however, shortly after the bearing temperatures were observed to be increasing rapidly, it was noticed that the total seal leakage was very high (above 40 scfm)). The bearing chamber is probably always filled with an air-oil vapor mist, or mixture and possibly this mixture was ignited upon contact with the hot air.

The rubbing described above helps explain the 2,000 rpm speed decrease which led to rig shutdown. Mechanical failure of the heat-shield sleeve may have occurred due to a combination of high centrifugal forces resulting from the high speed and the high temperature (1100°F) existing in that part of the rig. The performance of the seals (designed for operation at 14,000 rpm) can be considered good at the more severe conditions.

The fire in the interior of the rig appears to have been isolated within the test bearing chamber, i.e., the portion of the rig extending from the oil-side of the oil seal, including the bearing itself, to the rig seal at the aft end of the rig.

(see Enclosure 6). The inter seal cavity (space between oil seal and air seal) and the hot-air cavity (space in front of air seal) showed no signs of fire. The oil seal carbon wear was 0.003" to 0.007"; shoulder wear was 0.0005". Heavy oil coking was noted on the oil seal housing and on the outer-most carbon ring but not on, or inside of, the sealing dam indicating that no oil leakage had occurred across this seal. The air seal and shoulder rubbing surfaces were in good condition except for some erosion of the inner wear pad carbon which indicates considerable leakage across this seal. The wear pad carbon is circumferentially segmented and erosion occurred at the leading edge of each segment. Total carbon face wear was 0.0005". The face side of the air seal shoulder was scored from contact with the loosened sleeve described above. Enclosure 50 shows photographs of the tested seal components.

Aside from the one sleeve, the test rig did not suffer irreparable damage. All other rotating parts were measured and found free of distortion. Silver solder was found melted in some places. The silver plating (ion-deposited) on the bearing cage appeared "blistered" in some places and totally removed from most areas. The bearing rings and balls were covered with a "flaky" material, ostensibly oil char and coking as shown in Enclosure 49. The test bearing raceways, as well as could be determined, were free of surface distress.

The final test on this contract was run at a speed of 20,000 rpm (and slightly above) for a duration of 2.4 hours using an M-50 steel bearing with Mobil Jet II lubricant in open atmosphere. A hydrodynamic oil seal was used along with a bellows secondary air seal. The test was terminated due to seizure of the test bearing. Data for this test is tabulated in Enclosure 52.

The test was started and the rig brought up to 20,000 rpm over a period of 2.1 hours. During this time the total seal leakage ranged from a static reading of 21.2 scfm to 9.8 scfm at 20,000 rpm. Oil inlet temperature was set at 190° - 210°F to prevent excessive bearing temperatures above 600°F. Approximately 0.1 hours after reaching 20,000 rpm and with bearing temperatures at 520°F and 500°F on the inner and outer rings respectively, the test rig shut down automatically, due to an inadvertent shutdown of one of the cooling pumps for the magnetic clutch drive.

The test rig was restarted and brought up to 20,000 rpm within 0.6 hours and allowed to run there for two hours. During this time the total seal leakage ranged from 7.6 to 8.5 scfm with the bearing outer and inner-ring temperatures stabilized at 570°F and 610°F respectively. Oil flow was 0.75 to 1.0 gpm.

At this time it was decided to increase the speed to the maximum limit (with the present drive system) which turned out to be 21,000 rpm. Shortly after reaching 21,000 rpm a slight temperature excursion occurred in the test bearing and the speed was decreased back to 20,000 rpm. The bearing temperatures dropped somewhat but a gradual increase in temperature continued in spite of an increase in oil flow (to 1.5 gpm) until a major temperature excursion occurred causing seizure of the test bearing. Rig shutdown procedures were initiated and, as usual, the oil was left circulating. Approximately three minutes after bearing seizure a stainless steel pipe which connects the oil sump to the oil vapor condenser was seen to be glowing cherry red, indicating a fire in the sump. The oil flow was immediately shutdown and the fire was quenched by purging the entire rig-sump system with nitrogen gas.

Upon disassembly the test bearing was found smeared on all parts. The oil seal and air seal carbons had worn 0.008 and 0.002 inches respectively while the oil seal and air seal shoulder wear was negligible.

There was evidence that a fire existed in the bearing chamber. However it was of short duration or low intensity since there was no indication of a temperature rise in the bearing chamber, other than at the time of the bearing failure, and there was very little carbon residue which was quite evident in the previous fire.

Evidently, upon seizure of the test bearing, some oil was heated above the flash point (485°F) and upon returning to the sump, started a fire therein which was supported by and propagated by the oil vapor as evidenced by the rapidly spreading fire in the vapor condensate line. The inner oil-sump walls were heavily coked and charred.

Sump fires appear to be a definite problem associated with high temperature bearing lubricant systems in an open atmosphere especially at high bearing speeds which tend to generate more oil mist. . Photographs documenting the condition of test elements are presented as Enclosures 53 through 55.

In general, from Enclosures 47 and 52, it can be seen that reducing oil flow reduces power consumption. It also appears that operation at increasing speeds tends to reduce input power required. This is probably due to increased hydrodynamic lift on the seals plus lower oil viscosity due to higher operating temperatures at the higher speeds.

DISCUSSION

The results of the investigation reported herein demonstrate clearly the advantages of the systems-development approach in the pursuit of reliable bearing-lubricant systems for advanced turbine engine mainshafts. The interrelationships between bearings, lubricants, ancillary equipment and system operating parameters have been investigated in equipment which closely simulates the operating conditions expected in advanced engines. In this way, the effects on performance of various component designs and materials selections, and the benefits derived from inert gas blanketing have been most thoroughly investigated. The systems approach has identified problems with components, particularly with seals, which might not have been observed from component tests alone.

However, the basic problem of the systems-development approach has also been clearly demonstrated in this program; i.e. the difficulty of generating and maintaining a "total" simulated environment. The interrelationships between the various components precipitate unanticipated failure modes and interactions with the test equipment which sometimes prevent the complete fulfillment of the original test objectives or which obscure the primary parameters being explored. Nonetheless, the investigation has produced the desired information on both component and system operational parameters.

A. Systems Performance

In phase I of this program, the benefits of operating high temperature bearing-lubricant systems under inert gas blanketing were demonstrated through a reduction in oil decomposition and coking. However, in that investigation, tests were limited to short duration and were frequently interrupted by component failures. In Phase II, where some of the component problems were overcome and long-term operational tests were scheduled and conducted, the benefits of inert gas blanketing were more dramatically shown. In particular, the inerted atmosphere minimized the effects of the most serious limitation on the use of hydrocarbon lubricants; oxidation instability. In the inerted atmosphere the more faborable heat removal characteristics of the hydrocarbon fluid and its susceptibility to viscosity improvement through additives allowed it to perform better than the more oxidation-stable candidates. The lack of oxygen in the bearing cavity showed no detrimental effects on any of the system components.

Even through the inerted system shows significant merit for bearing-lubricant performance, it does not obviate all problems of reliability in the maintenance of the system integrity. In particular, the oil seals represent a critical area of concern. Two aspects of seal failure were demonstrated in this program; loss of lubricant through entrainment in escaping gas, and fire hazard through admission of oxygen. Significant improvements in seal reliability were effected in the latter part of Phase II, although seal wear rates and leakage rates still need further improvement. These are discussed below in the section dealing with components.

Observed interactions between the bearing and lubricants have demonstrated the relative importance of various lubricant properties and bearing design parameters. The most important failure mode appears to be thermal instability, in which the lubricant fails to remove the heat generated by the bearing. The subsequent temperature increase reduces bearing internal clearances which results in increased heat generation. A thermal runaway occurs and the bearing ultimately seizes. It was shown that heat generation can be reduced through bearing design modification, but for all designs, optimum performance requires an adequate supply of a lubricant with good ability to remove heat (high specific heat, good wetting properties) and good high-temperature viscosity characteristics. Anti-wear or boundary lubricating properties of the lubricant did not seem to be especially critical in controlling thermal instability in the large bearing tests reported here. However, previous testing with small bearings has demonstrated improved resistance to thermal imbalance failures when an anti-wear additive is used (10). Oxidation resistance is not important in the inerted system since the amount of oxygen system since the amount of oxygen in the system is kept low, but the lubricant must possess sufficient resistance to thermal degradation or cracking to maintain its original viscosity throughout the required operational life at the operating temperatures of the bearing. In these tests, only the synthetic hydrocarbon oils, which have inherently high specific heat and viscosity at high temperatures, have shown adequate performance capability.

A second failure mode, related to cage performance, has been observed in these tests. Cage performance is affected by the thermal problems discussed above and places some additional requirements on the lubricant. The cage is subject to both thermal and centrifugal expansion, which can lead to heavy loading on the outer ring land (where it is guided). Design modifications to reduce this loading were partially successful, and further design refinements to reduce centrifugal warping and improve cage cooling should be explored. In addition, cage wiping is reduced by operating with a lubricant of high viscosity and good anti-wear or boundary lubricating properties. Wear in the cage pockets, where hydrodynamic film generation is marginal, is also reduced by these same lubricant properties. The effects of boundary lubrication properties on oil seal performance can not be deduced from these tests. It appears that there is little chemical interaction between the seal and the lubricant, and seal performance improvements must be derived through either design changes to improve lubrication (hydrodynamic seal) or through design and material changes to improve their unlubricated operation.

B. Component Performance

1. Bearings

Throughout the program, even at speeds up to and above 20,000 rpm (2.5 million DN), the basic bearing design has been shown to be entirely workable with only slight adjustments in cage-to-bearing operating clearance required due to centrifugal cage expansion. Providing adequate lubrication/heat transfer at extreme conditions continues to be the major problem area. The bearing materials used were shown to be adequate for operation above 700°F.

Two different bearing designs, designated Series I and Series II, were utilized during this program. Briefly, they differ in that Series II bearings have more open conformity which theoretically reduces heat generation at some expense in calculated bearing life. A complete discussion of the two bearing designs is given in the section on Test Elements.

From the tests run on this program in both phases, there are three sets of data with the same oil at essentially the same conditions from which heat generation comparisons can be made. Two of these show that Series I bearings generate more heat than the Series II design (with Mobil Jet II about 20% more, with Mobil XRM 109F plus Kendall Resin, about 7 times more, based on calculations of heat rejection to the oil, using data given in Appendix III of Part 2 (5). The third test, with Mobil XRM 177F, shows that Series II bearings generated about 40% more heat than Series I. Thus, any effect of groove conformity on heat generation that may exist is confounded by interactions with lubricant chemistry effects.

The bearings used in these tests have limited applicability for aircraft service at the extreme speed conditions encountered in Task IV tests. Operation at speeds of 2.5 million DN and higher produces large centrifugal forces of the balls against the outer ring which not only causes high heat generation, but also causes a drastic reduction in the fatigue life of the bearing. The high speed operation in combination with aircraft maneuvering produces momentary unloading and gyroscopic effects which induce severe sliding and spinning of the balls at the race contacts. They may induce lubrication-related surface distress and skid damage which may place greater emphasis on the anti-wear and boundary lubricating capabilities of the lubricants than were demonstrated in these tests. In order to limit the bearing life reduction and ball/race sliding and spinning consequences of ultra high operating speeds, it is highly desirable that a bearing used at such ultra high speeds have lower density or reduced weight balls to relieve the centrifugal loads.

2. Cages

Two different cages were utilized during this program; electro-silver plated M-1 steel and ion-silver plated 4340 steel. No discernible difference was noted in cage material performance and in no case was bearing failure related to cage failure. It also appears that the ion plating offered no advantage over the standard electro-plating, based on such observations as pocket wear and plating adherence after test. In short, both types of cages and plating were found to be adequate for advanced engine usage.

3. Seals

Considerable progress was made with the aid of the Koppers Company, during this program in developing seals having acceptable low leakages for long periods of time at these severe test conditions. A complete discussion of all seal experience and results to date is presented in Appendix I of Part 2 (5). Seal performance during this program may be summarized briefly as follows:

By utilizing a face load heavy enough to prevent lift-off (0.4 lbs/in of circumference) in conjunction with wear-resistant carbon and shoulder plating, leakage rates on the order of 1 scfm were realized for up to 32 hours with a bellows secondary oil seal. This is a decided improvement over previous results and can probably be improved upon further by the use of harder shoulder plating, such as carbides or ceramics, since the failure mode is excessive wear of the shoulder plating.

The most promising oil seal is the piston ring secondary design with a shoulder employing hydrodynamic lift pads. One of these seals has been run for 573.1 hours at test conditions. While this type seal was found in these tests to have minimum leakage rates 3 to 5 times that of the best bellows seals, it is reliable and not prone to lift-offs. It can be conjectured that certain desirable features of the two types of oil seals, namely the low leakage, simpler design, bellows and the wear resistance and stability of hydrodynamic shoulder lift pads, can be combined for reliable long-term low leakage performance.

The bellows air seal remains essentially trouble-free and one has been run for 483.1 hours at conditions in this program.

4. Lubricants

The test program has shown that a relatively viscous synthetic paraffinic hydrocarbon with either an antiwear additive or a high viscosity paraffinic resin as a non-reactive additive is capable of serving as a lubricant for advanced turbine-engine operating conditions in an inert atmosphere environment. Most interesting is the finding that a very high molecular weight (viscous) mineral oil is an effective additive, possibly because it increases the film thickness in the Hertzian contact areas more than the overall increase in bulk oil viscosity would indicate (11).

It has also been shown that a high heat transfer capacity of the lubricant at operating temperatures is necessary to remove the heat generated by the bearing. Both the synthetic hydrocarbon and ester lubricants have this property. However, the other major parameter ,good high temperature viscosity, is lacking in the ester lubricant tested in this program.

Of special interest for application to actual engine operation are the results of the 250-hour tests with respect to oil consumption, degradation, etc. These tests were run with both Mobil XRM-177F and XRM109F (plus Kendall resin) lubricants at 650°F bearing temperature. Typical current commercial engine oil consumption at about 300°F is approximately 0.05 to 0.10 gal./hr. per bearing compartment. Increased oil consumption was found during the long-term tests at 650°F (0.2 to 0.3 gals/hr) probably mostly due to increased lubricant volatility at this temperature. Acid numbers for both tests were generally in the 0.05 range, well within the current specifications for hydrocarbon fluids (which are more stringent than the specifications for ester-based fluids). Viscosity increases were on the order of 25% for the XRM-177F and 25 to 50% for the XRM-109F plus Kendall resin fluids, which are on the upper end of typical current specifications. Degradation mechanisms for the latter fluid were deduced from mass spectrometric analyses, as discussed in Appendix IV of Part 2 (5).

Specific comments on the lubricants tested in Phase II are given below:

<u>a. Mobil Jet II</u> - ester is apparently satisfactory for short-term use up to about 550°F bearing outer-race temperature with inert gas blanketing. At higher temperatures thermal instabilities of the system arise resulting in bearing thermal imbalance failures. This lubricant performed adequately for 50 hours at bearing temperatures of 500 to 530°F in an open atmosphere although some bearing surface distress was evident which suggests that the oil may have marginal performance capabilities for long-term operation in this temperature region.

This lubricant was also used in the high speed running in Task IV of this program in an open atmosphere. The fact that two fires occurred is not surprising since the flash point of this fluid is 485°F, considerably below the bearing temperature which reached 600 to 650°F during the running at 20,000 to 21,000 rpm.

The bearing failure experienced in the last test (at speeds up to about 21,000 rpm) is most likely associated with changes in heat transfer ability or with instability of the fluid.

b. Mobil XRM-154D - silicone lubricant provided adequate bearing lubrication at 650°F in inerted short term tests, but extensive decomposition products were produced even in the presence of less than 0.02% oxygen, so that it is considered unsuitable for long-term operation. In addition, severe oil seal wear was observed with this fluid. It is not clear that this is necessarily attributable to the fluid properties, since heavy wear was observed sporadically with other fluids.

c. Mobil XRM-109F with 10% Kendall Heavy Resin - was found suitable at temperatures to 700°F in an inert atmosphere for at least short-term use. This fluid was successful in a 250 hour test at 650°F conditions with an inert gas blanket. This result clearly domonstrates that a high molecular weight paraffinic resin oil can act as an effective additive since the base stock was not found to be an effective lubricant in previous tests. Under NASA contracts NASw-492 (10) and NAS3-11171 (12), Mobil XRM-109F provided only marginal lubrication of 25 mm-bore bearings operating at 600°F and 1.1 million DN.

d. A Blend of Mobil XRM-109F, Mobil XRM-127B and Kendall Heavy Resin - performed well for 50 hours under an inert blanket at 650°F conditions. The purpose of this blend being tested was to determine if the increase in bulk oil viscosity of XRM-109F plus 10% Kendall Heavy Resin was responsible for its good performance. To test this hypothesis, enough XRM-127B, a fluid of the same homologous family as XRM-109F but with a lower viscosity, was added to make the blend viscosity equal to that of XRM-109F. The short term performance was satisfactory which suggests that Kendall Heavy Resin does not improve performance by increasing the bulk viscosity. However, the suitability of this fluid for longer-term operation at these conditions is doubtful since the viscosity of the blend was increased by 87% during the test, probably due to distillation of the lighter XRM-127B constituents.

e. Mobil XRM-177F - the most successful lubricant from Phase I testing, was successfully employed for 250 hours at 650°F inerted test conditions. The lubricating performance of this oil is about equivalent to that of Mobil XRM-109F plus Kendall Heavy Resin. Slightly greater cage wear was observed with XRM-177F during the 250-hour endurance tests, but this may have resulted from the greater number of start-ups [32] than with XRM-109F plus Kendall Resin [27].

CONCLUSIONS AND RECOMMENDATIONS

1. In this program a system comprising bearings, seals and lubricants has been developed which can provide several hundred hours of reliable system operation at representative advanced turbine engine conditions (bearing temperature of 600 to 650°F, DN value of 1.75 million, and in an inerted atmosphere). In addition, results indicate that operation is possible for short periods at speeds to 20,000 rpm (2.5 million DN) using conventional 125-mm bearings having sufficient internal clearance for operation under the high centrifugal loading of the higher DN levels. The experimental seals developed in this program also appear to have such high-speed capabilities, at least for several hours of operation.

2. Oil consumption, degradation, etc. observed during the two long-term (250 hour) tests conducted during this program are of the same order as current commercial aircraft gas turbine usage which indicates the feasibility of using the bearing-seal-lubricant/lubrication system package in advanced applications. Developmental refinements in the system should provide overall performance favorably comparable to current engines.

3. In order to permit economical engine compartment inerting with the advantages of higher operating temperatures and less fire danger, a reliable oil seal is needed with sustained leakages considerably below 1 scfm. Face seals employing zero leakage bellows secondaries and hydrodynamic lift shoulders should be tried in combination, since these were found, individually, to be the best features of the seals tested in this program.

4. In order to keep up with the advancement of turbine state-of-the-art, attention should be given to operation in the 3 million DN range at temperatures on the order of 600°F.

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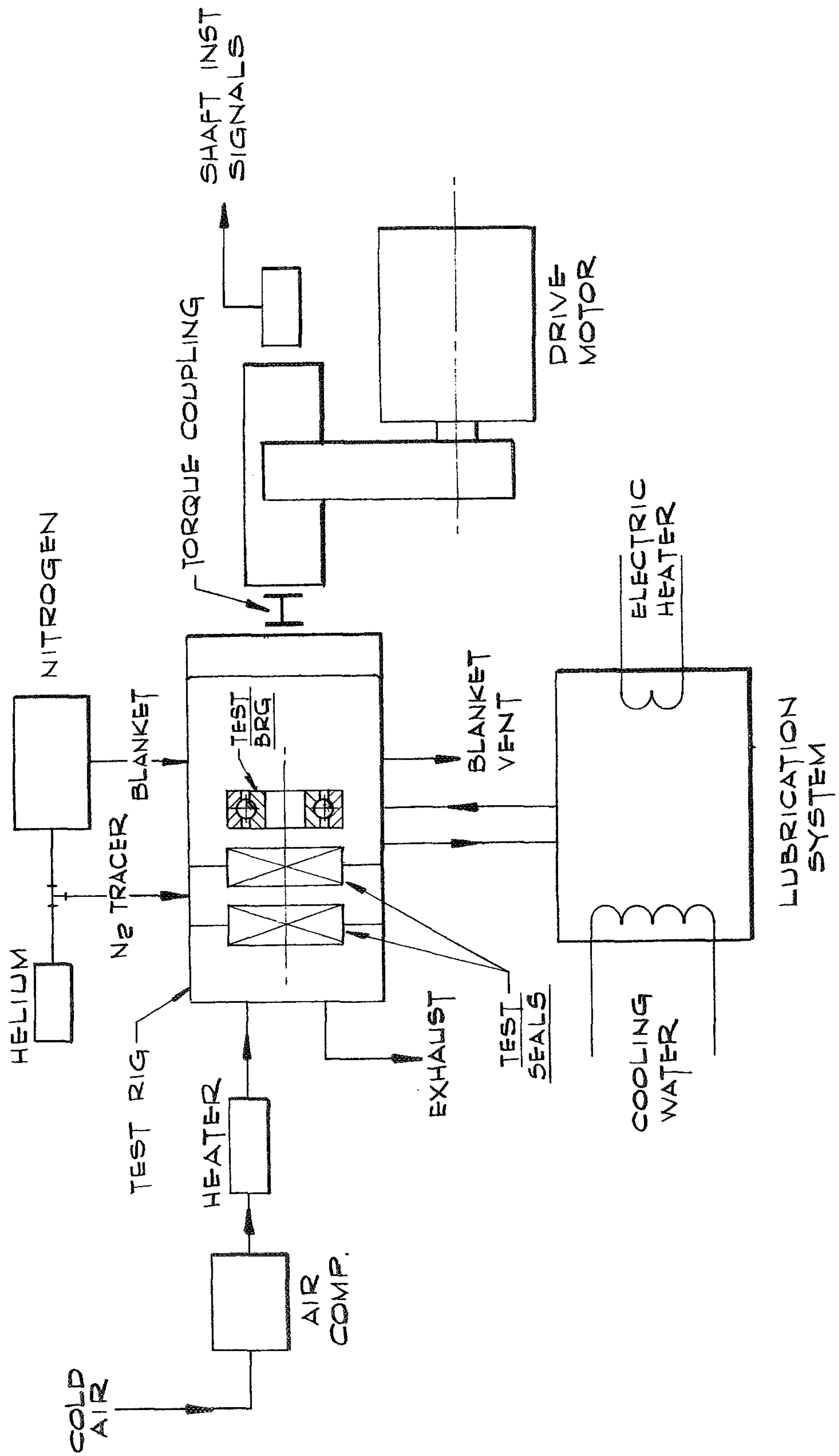
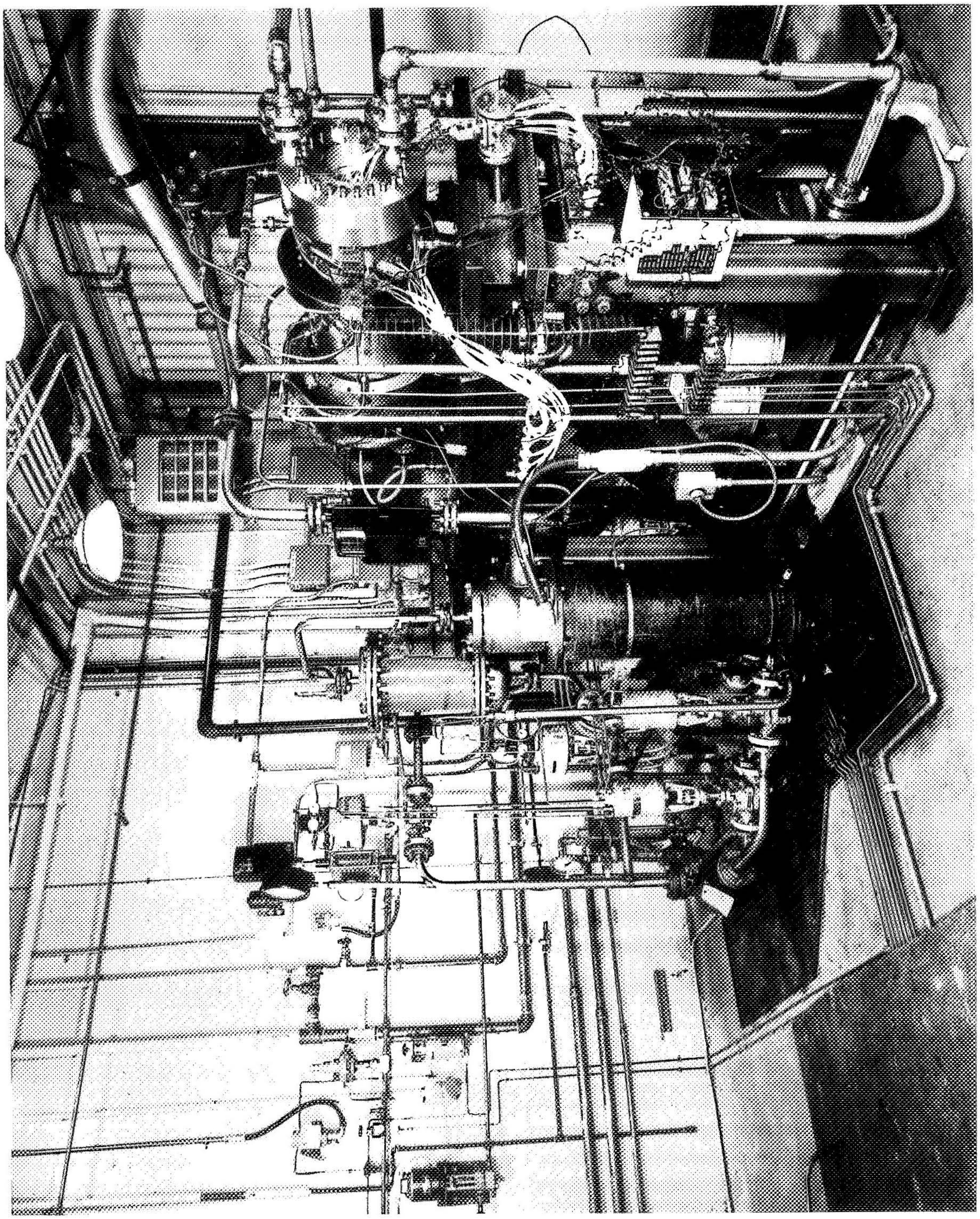
ENCLOSURE 1

GENERAL TEST RIG LAYOUT SCHEMATIC



ENCLOSURE 2

GENERAL VIEW OF RECIRCULATING-OIL TEST RIG CELL



ENCLOSURE 3

459980H BEARING DESIGN DATA

76272-2

SKF

	BEARING DATA						
1	VENDORS PART NO.	459980 I I	459981 D	459980 J ①	459980 K ①	459981 E ①	459981 F ②
2	VENDORS DRAWING NO.	76272-2	76272-2	76272-2	76272-2	76272-2	76272-2
3	BEARING TOLERANCE	ABEC 7	ABEC 7	ABEC 7	ABEC 7	ABEC 7	ABEC 7
4	NO & SIZE OF BALLS	21-13/16 DIA.	21-13/16 DIA.	21-13/16" DIA.	21-13/16" DIA.	21-13/16" DIA.	21-13/16" DIA.
5	INTERNAL RADIAL LOOSENESS (UNMOUNTED)	.0097"-.0109"	.0078"-.0090"	.0085"-.0097"	.0087"-.0099"	.0068"-.0080"	.0068"-.0080"
6	MAX END PLAY (UNMOUNTED)	.035"	.031"	.032"	.032"	.029"	.029"
7	DESIGN CONTACT ANGLE	31°30'-35°30'	28°30'-32°30'	32°30'-36°30'	32°30'-36°30'	29°30'-33°30'	29°30'-33°30'
8	OUTER GROOVE RADIUS AS PERCENT OF BALL DIA.	51.50-51.75	51.50-51.75	52.00-52.25	52.00-52.25	52.00-52.25	52.00-52.25
9	INNER GROOVE RADIUS AS PERCENT OF BALL DIA.	52.75-53.00	52.75-53.00	51.50-51.75	51.50-51.75	51.50-51.75	51.50-51.75
10	SHIM WIDTH	.013"	.013"	.009"	.009"	.009"	.009"
11	PITCH DIA.	6.280"	6.280"	6.280"	6.280"	6.280"	6.280"
12	CAGE WIDTH (MAX)	1.024"	1.024"	1.024"	1.024"	1.024"	1.024"
13	CAGE SPEC. (FOR OUT-OF-ROUNDNESS & SQUARENESS)	470939	470939	470939	470939	470939	470939
14	CAGE POCKET CLEARANCE	.009"-.017"	.009"-.017"	.014"-.022"	.009"-.017"	.014"-.022"	.009"-.017"
15	BALL GRADE (AFBMA)	15	15	15	15	15	15
16	INNER RING MATERIAL	WB49	M-50	WB 49	WB 49	M50	M50
17	OUTER RING MATERIAL	WB49	M-50	WB 49	WB 49	M50	M50
18	BALL MATERIAL	M-1	M-1	WB 49	M1	M50	M50
19	CAGE MATERIAL	M-1	M-1	M1	M1	M1	M1
20	CAGE TYPE	ONE PIECE MACHINED OUTER LAND RIDING	ONE PIECE MACHINED OUTER LAND RIDING	ONE PIECE MACHINED OUTER LAND RIDING	ONE PIECE MACHINED OUTER LAND RIDING	ONE PIECE MACHINED OUTER LAND RIDING	ONE PIECE MACHINED OUTER LAND RIDING
21	BEARING DESIGNED TO OPERATE AT (MAX.)	850°	650°	850°	850°	650°	650°
22	INNER RING HARDNESS	67 MIN	61 MIN	67 MIN.	67 MIN.	61 MIN.	61 MIN.
23	OUTER RING HARDNESS	67 MIN	61 MIN	67 MIN.	67 MIN.	61 MIN.	61 MIN.
24	BALL HARDNESS	62 MIN	62 MIN	62 MIN.	62 MIN.	62 MIN.	62 MIN.
25	CAGE HARDNESS	53 MIN	53 MIN	53 MIN.	53 MIN.	53 MIN.	53 MIN.
26	CAGE RUNNING CLEARANCE ON DIA.	.010"-.020"	.010"-.020"	.010"-.020"	.010"-.020"	.010"-.020"	.010"-.020"
27	CAGE O.DIA "L"	6.681"-6.673"	6.681"-6.673"	6.681"-6.673"	6.681"-6.673"	6.681"-6.673"	6.681"-6.673"
28	CAGE I.DIA. "K"	6.111"-6.121"	6.111"-6.121"	6.111"-6.121"	6.111"-6.121"	6.111"-6.121"	6.111"-6.121"
29	CAGE PLATING — TYPE	SILVER AMS 2410	SILVER AMS 2410	SILVER AMS 2410	SILVER AMS 2410	SILVER AMS 2410	SILVER AMS 2410
	— THICKNESS	.001"-.002"	.001"-.002"	.001"-.002"	.001"-.002"	.001"-.002"	.001"-.002"

CAGE TO BE 100% FLUORESCENT PENETRANT INSPECTED (470908)
100% VISUAL INSPECTION (470653)
100% MAGNAFLUX INSPECTION (471682)
100% ETCH INSPECTION (471631)
MATERIAL AND IDENTIFICATION CONTROL AS PER SPEC. 471085
INNER RING, OUTER RING AND BALLS TO BE BLACK OXIDE TREATED
AS PER AMS 2485

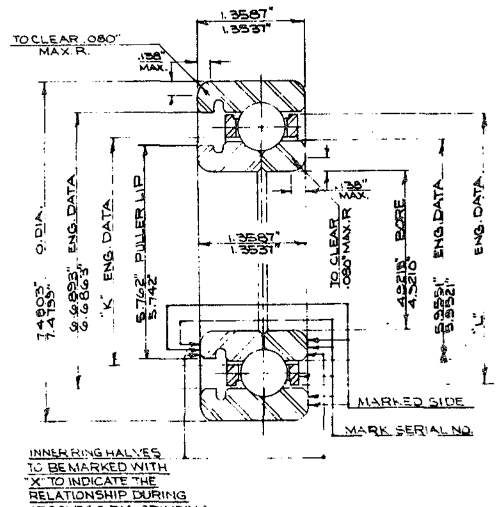
SURFACE FINISH REQUIREMENTS

	RMS	MAX
INNER RING BORE	6	"
INNER RING GROOVE	4	"
OUTER RING O.DIA.	6	"
OUTER RING GROOVE	4	"
CAGE SUPPORT SURFACE	20	"
BALL	2	"
OUTER RING BORE	4	"
INNER RING O.DIA.	4	"

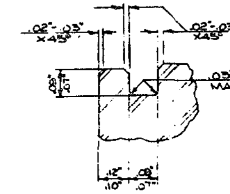
MAX CAGE UNBALANCE 3GM-CM. BALANCE TO BE CHECKED AT 500 RPM (MIN)

INNER & OUTER RING LAND TOLERANCES
A. 3 POINT OUT-OF-ROUND WITHIN .0004" F.I.R.
B. ECCENTRICITY TO O.DIA. WITHIN .0004" F.I.R.
C. SECTION HEIGHT OF LAND TO O.DIA. EQUAL WITHIN .0004" SAME AXIAL PLANE
D. 80% CONTACT WHEN ROLLED ON A BLUED SUPPORT

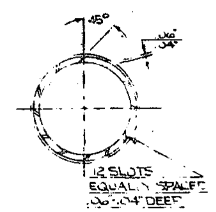
CAGE SUPPORT
A. 3 POINT OUT-OF-ROUND WITHIN .003" F.I.R.
B. 3 POINT DIAMETERS EQUAL WITHIN .002" IN SAME AXIAL PLANE
C. SQUARENESS TO COMMON MACHINING REF FACE WITHIN .002"
D. 80% CONTACT WHEN ROLLED ON BLUED FLAT PLATE



INNER RING HALVES TO BE MARKED WITH "X" TO INDICATE THE RELATIONSHIP DURING GROOVE & O.DIA. GRINDING



ENLARGED VIEW OF PULLER GROOVE INNER & OUTER RINGS

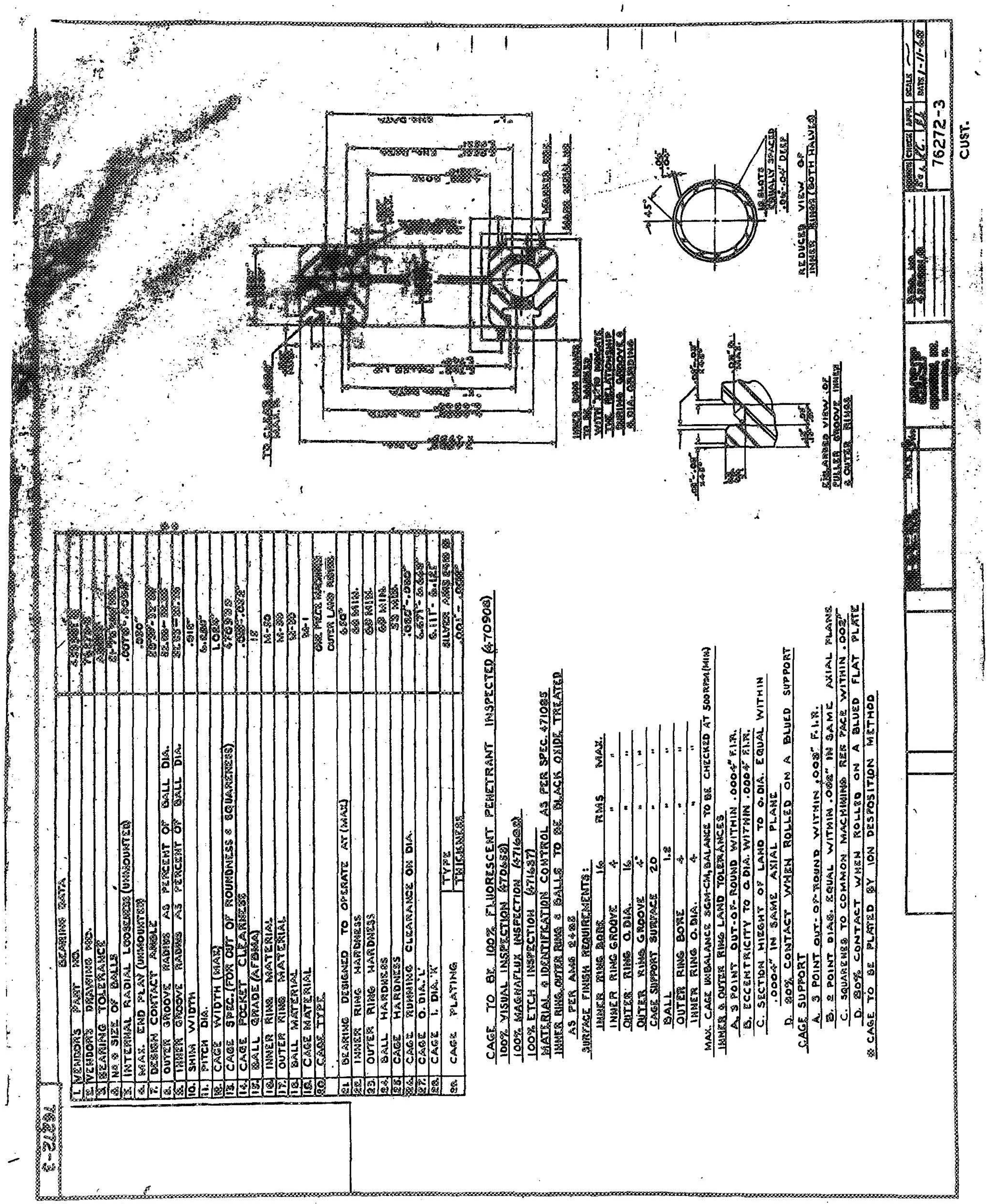


REDUCED VIEW OF INNER RINGS (BOTH HALVES)

ADDED SIZES	SKF INDUSTRIES, INC. PHILADELPHIA, PA.	BRG NO 459980 H, J, K 459981 D, E, F	DRAWN / CHECK / APPR. / SCALE / DATE	76272-2

ENCLOSURE 4

459981G BEARING DESIGN DATA



BEARING DATA	
1. VENDORS PART NO.	[illegible]
2. VENDORS DRAWING NO.	[illegible]
3. BEARING TOLERANCE	[illegible]
4. NO & SIZE OF BALLS	[illegible]
5. INTERNAL RADIAL LOOSENESS (UNMOUNTED)	.0070"-.00[illegible]
6. MAX. END PLAY (UNMOUNTED)	.030"
7. DESIGN CONTACT ANGLE	[illegible]
8. OUTER GROOVE RADIUS AS PERCENT OF BALL DIA.	[illegible]
9. INNER GROOVE RADIUS AS PERCENT OF BALL DIA.	[illegible]
10. SHIM WIDTH	.918"
11. PITCH DIA.	6.[illegible]"
12. CAGE WIDTH (MAX)	1.025"
13. CAGE SPEC. (FOR OUT OF ROUNDNESS & SQUARENESS)	470939
14. CAGE POCKET CLEARANCE	[illegible]
15. BALL GRADE (AFBMA)	10
16. INNER RING MATERIAL	M-50
17. OUTER RING MATERIAL	M-50
18. BALL MATERIAL	M-50
19. CAGE MATERIAL	[illegible]-1
20. CAGE TYPE	ONE PIECE MACHINED OUTER LAND RIDING
21. BEARING DESIGNED TO OPERATE AT (MAX)	650°
22. INNER RING HARDNESS	60 MIN.
23. OUTER RING HARDNESS	60 MIN.
24. BALL HARDNESS	60 MIN.
25. CAGE HARDNESS	[illegible] MIN.
26. CAGE RUNNING CLEARANCE ON DIA.	.0[illegible]"-.0[illegible]"
27. CAGE O. DIA. 'L'	6.671"-6.6[illegible]"
28. CAGE I. DIA. 'K'	6.111"-6.12[illegible]"
29. CAGE PLATING — TYPE	SILVER AMS 2410 [illegible]
29. CAGE PLATING — THICKNESS	.001"-.00[illegible]"

CAGE TO BE 100% FLUORESCENT PENETRANT INSPECTED (470908)
100% VISUAL INSPECTION (470658)
100% MAGNAFLUX INSPECTION (471602)
100% ETCH INSPECTION (471637)
MATERIAL & IDENTIFICATION CONTROL AS PER SPEC. 471085
INNER RING, OUTER RING & BALLS TO BE BLACK OXIDE TREATED AS PER AMS 2485

SURFACE FINISH REQUIREMENTS:

		RMS	MAX.
INNER RING BORE	16	RMS	MAX.
INNER RING GROOVE	4	"	"
OUTER RING O. DIA.	16	"	"
OUTER RING GROOVE	4	"	"
CAGE SUPPORT SURFACE	20	"	"
BALL	1.2	"	"
OUTER RING BORE	4	"	"
INNER RING O. DIA.	4	"	"

MAX. CAGE UNBALANCE 3GM-CM, BALANCE TO BE CHECKED AT 500 RPM (MIN)

INNER & OUTER RING LAND TOLERANCES
A. 3 POINT OUT-OF-ROUND WITHIN .0004" F.I.R.
B. ECCENTRICITY TO O. DIA. WITHIN .0004" F.I.R.
C. SECTION HEIGHT OF LAND TO O. DIA. EQUAL WITHIN .0004" IN SAME AXIAL PLANE
D. 80% CONTACT WHEN ROLLED ON A BLUED SUPPORT

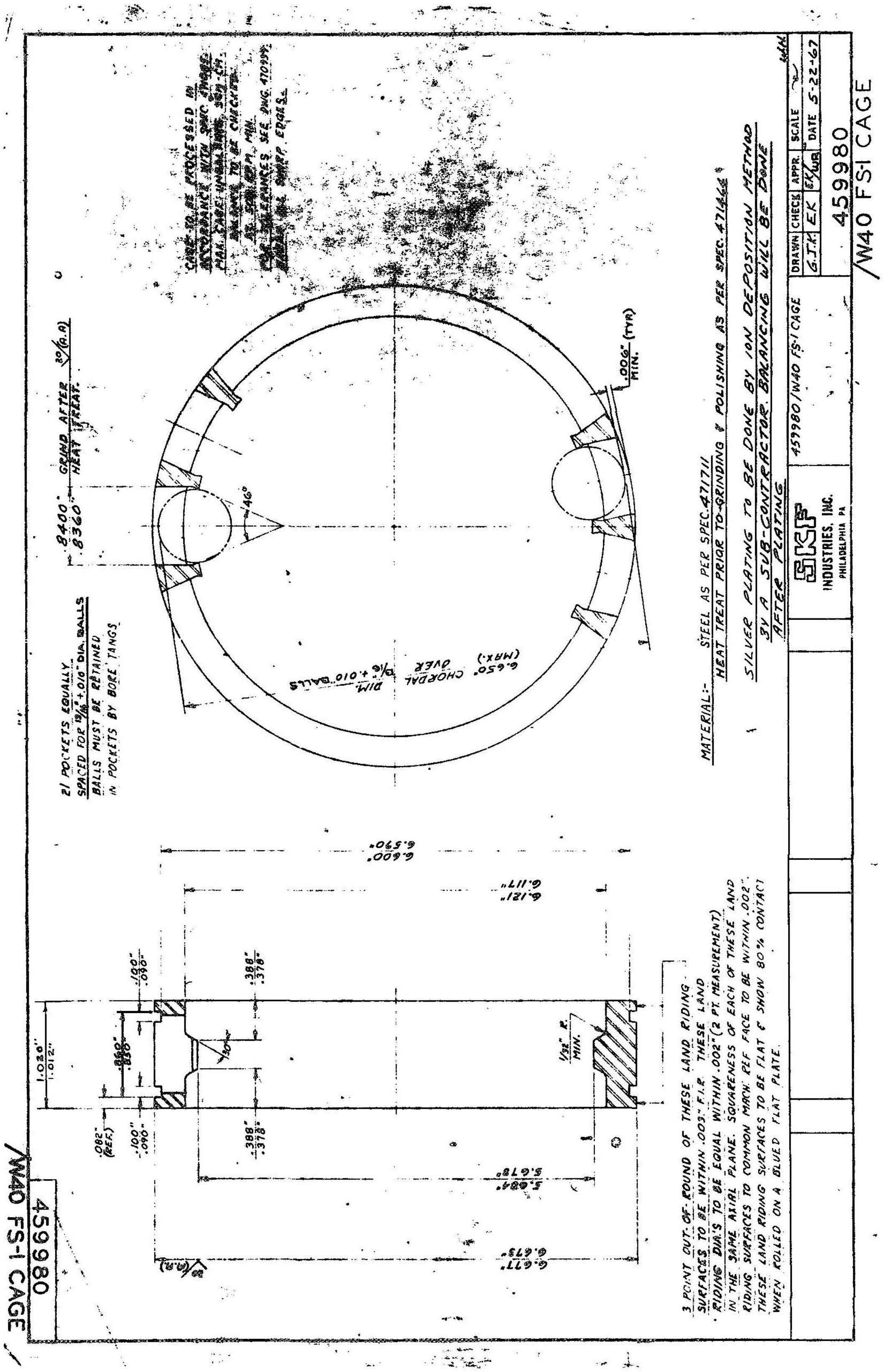
CAGE SUPPORT
A. 3 POINT OUT-OF-ROUND WITHIN .003" F.I.R.
B. 2 POINT DIAS. EQUAL WITHIN .002" IN SAME AXIAL PLANE
C. SQUARENESS TO COMMON MACHINING REF FACE WITHIN .002"
D. 80% CONTACT WHEN ROLLED ON A BLUED FLAT PLATE

⊗ CAGE TO BE PLATED BY ION DESPOSITION METHOD

F4186A - R80 - 10.62 X 13.75

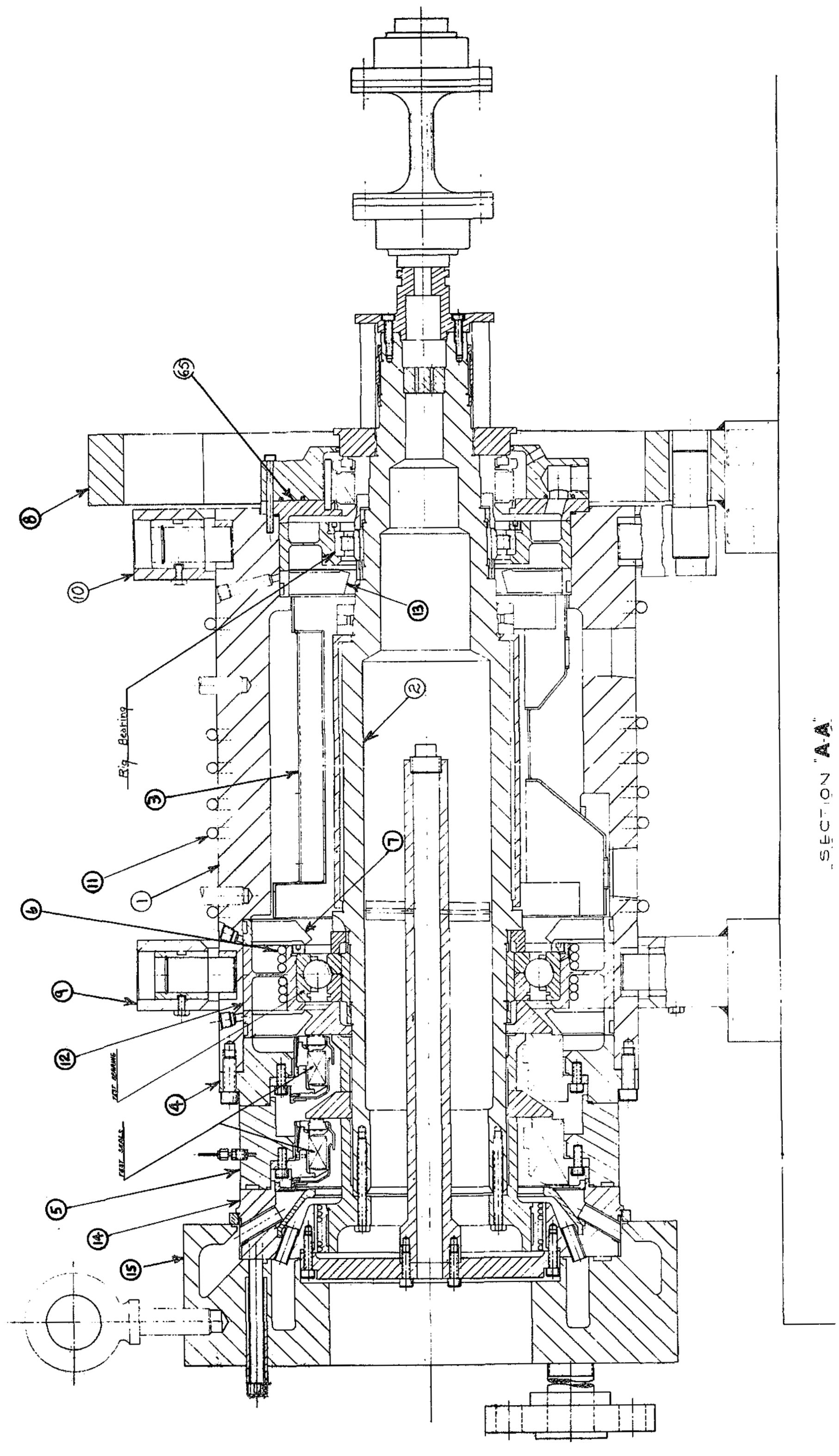
ENCLOSURE 5

459980 FS-1 CAGE



ENCLOSURE 6

TEST RIG ASSEMBLY



1. Test Housing
2. Test Shaft
3. Baffle
4. Inner Housing Ring (Oil Seal Mount)
5. Middle Housing Ring (Air Seal Mount)
6. Test Bearing Heaters
7. Test Bearing Lubrication Ring
8. Rig Mounting Ring
9. Rig Mount (Front)
10. Rig Mount (Back)
11. Test Housing Heaters
12. Test Bearing I-Housing Mount
13. Rig Bearing Lubrication Ring
14. Outer Housing Ring
15. Hot Air Manifold

F-4186A - R50 - 17.00 X 22.00

TABULATION OF TEST ELEMENTS USED IN TASK II, III, AND IV TESTS

TEST NO.	LUBRICANT	BEARING USED (1)	(2) CAGE	OIL SEAL (3) CARBON MAT'L	OIL SEAL (3) DESIGN	OIL SEAL SHOULDERS (4) PLATING MAT'L	OIL SEAL SHOULDERS (4) DESIGN	AIR SEAL (5) CARBON MAT'L	AIR SEAL (5) DESIGN	AIR SEAL SHOULDERS (6) PLATING MAT'L	AIR SEAL SHOULDERS (6) DESIGN
1*	MOBIL JET II	267101	#2	USG2777	700488SN/1	CHROME	700488-10SN/1	CDJ83	700405SN/2	CHROME	700397SN/A-1
2*(7)	MOBIL JET II	267102	#2	USG2777	700488SN/1	CHROME	700488-10SN/1	CDJ83	700405SN/2	CHROME	700397SN/3
3*	MOBIL JET II	267103	#3	CDJ83	700489SN/4	CHROME	700488SN/1B	CDJ83	700405SN/2	CHROME	700397SN/2
4*	MOBIL XRM 154D	267104	#7	CDJ83	700489SN/4	CHROME	700488SN/2	CDJ83	700405SN/2	CHROME CARBIDE	700405-10SN/2
5**	MOBIL XRM 177F	267105	#5	CDJ83	700489SN/4	CHROME	700488SN/2	CDJ83	700405SN/2	CHROME CARBIDE	700405-10SN/2
6*	MOBIL XRM 109F + 10% BY WGT. OF KENDALL HEAVY RESIN 0839	267106	#9	CDJ83	700489SN/10	CHROME	700488SN/1A	CDJ83	700405SN/2	CHROME CARBIDE	700405-10SN/2
7**	MOBIL XRM 109F + 10% WGT. OF KENDALL HEAVY RESIN 0839	267107	#10	CDJ83	700489SN/10	CHROME CARBIDE	700488-10SN/1	CDJ83	700405SN/1	CHROME CARBIDE	700405-10SN/1
8*	MOBIL XRM 154D	267108	#11	CDJ83	700489SN/4	CHROME CARBIDE	700488 SN/1BCC	56HT	700397SN/7	CHROME CARBIDE	700405-10SN/1
9**	MOBIL JET II	267109	#17	CDJ83	101056BSN/1	TUNGSTEN CARBIDE	101056SN/1	56HT	700397SN/2	CHROME CARBIDE	700405-10SN/1
10**	MOBIL XRM-109F MOBIL 127B + 10% BY WGT KENDALL HEAVY RESIN 0839	267110	#6	CDJ83	101056BSN/1	TUNGSTEN CARBIDE	101056SN/1	56HT	700397SN/2	CHROME CARBIDE	700405-10SN/1
11**(8)	MOBIL XRM 177F	267111	#1	CDJ83	101056BSN/2	TUNGSTEN CARBIDE	101056SN/2	56HT	700397SN/6	CHROME CARBIDE	700397SN/1
12**	MOBIL XRM-109F + 10% BY WGT. KENDALL HEAVY RESIN 0839	267112	#13	CDJ83	101056BSN/1	TUNGSTEN CARBIDE	101056SN/1	56HT	700397SN/6	CHROME CARBIDE	700397SN/1
13(9)***	MOBIL JET II	267113	#16	CDJ83	101056BSN/1	TUNGSTEN CARBIDE	101056BSN/1	56HT	70039SN/6	CHROME CARBIDE	700397SN/1
				CDJ83	101056BSN/2	TUNGSTEN CARBIDE	101056BSN/2	56HT	700397SN/3	CHROME CARBIDE	700405SN/1
14***	MOBIL JET II	267114	#14	CDJ83	101056BAN/1	TUNGSTEN CARBIDE	101056BSN/2	56HT	700495SN/1	CHROME	700397SN/3

* TASK II TESTS
** TASK III TESTS
*** TASK IV TESTS

(1) ALL BEARINGS WERE WB49 STEEL (459980H) EXCEPT THOSE USED IN TESTS 9, 11, 12, AND 13 WHICH WERE M-50 STEEL (459981G).
(2) ALL CAGES WERE ION-SILVER PLATED AND WERE MADE OF 4340 STEEL EXCEPT THE ONE USED IN TEST #1 WHICH WAS MADE OF M-1 STEEL.

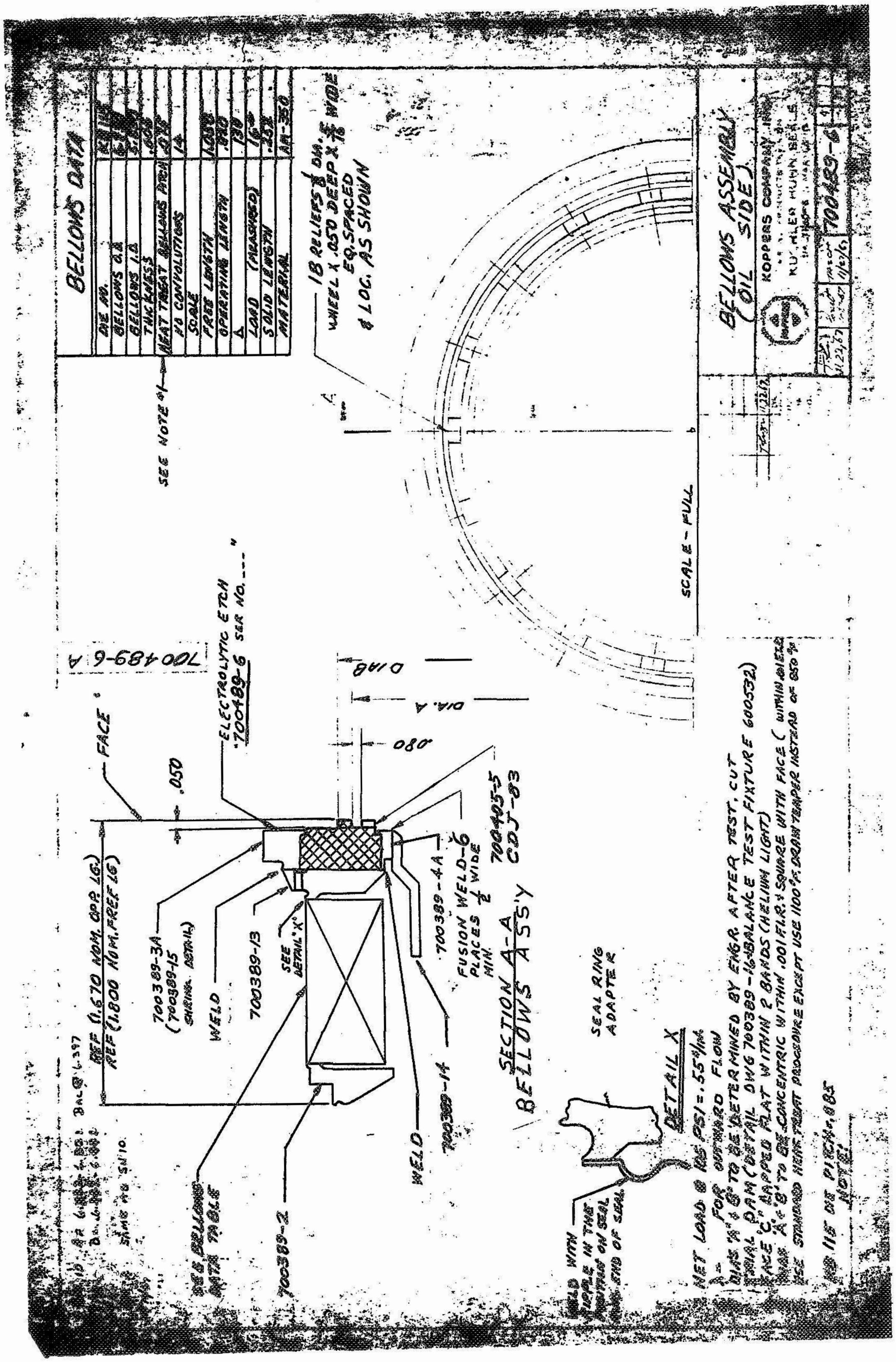
(3) ALL OIL SEAL BELLOWS ARE AM350 EXCEPT TEST #9, #10, #11, #12, AND #13 WHERE PISTON TYPE OIL SEALS WERE USED.
(4) ALL OIL SEAL SHOULDERS WERE MADE OF AMS6322.

(5) AIR SEAL BELLOWS IN TEST #1, #2, #3, #4, #5, #6, AND THE FIRST 31.3 HOURS OF #7 WERE MADE OF INCO 718 WHILE THE REMAINING 18.7 HOURS OF THE TEST #7 AND ALL OF TEST #8, #9, #10, #11, #12, AND #13 USED AN AIR SEAL BELLOWS MADE OF AM350.
(6) THE AIR SEAL SHOULDERS USED IN TEST #1, #2, #3, #11, #12, AND THE FIRST 2.9 HOURS OF #13 WERE MADE OF AMS6322 AND THE SHOULDERS USED IN #4 THROUGH #10 INCLUSIVE AND THE REMAINING 5.2 HOURS OF TEST #13 WERE MADE OF INCONEL X.

(7) IN ORDER TO INCREASE THE SEATING FORCE ON THE OIL SEAL, THE MOUNTING FLANGE OF THIS SEAL WAS CUT BACK BY .020" PRIOR TO ASSEMBLY (AN INCREASE OF APPROXIMATELY 15% OVER THE NET FACE LOAD USED IN PREVIOUS TEST).
(8) THE OIL SEAL/SHOULDER ASSEMBLY WAS REPLACED AFTER THE FIRST 12.8 HOURS OF RUNNING DUE TO EXCESSIVE GROOVING OF THE SHOULDER AND WEAR OF THE CARBON FACE. IT WAS LATER FOUND THAT THE WRONG CARBON MATERIAL WAS USED.
(9) THE OIL SEAL/SHOULDER ASSEMBLY WAS REPLACED AFTER THE FIRST 2.9 HOURS OF RUNNING DUE TO EXCESSIVE GROOVING OF THE SHOULDER AND WEAR OF THE CARBON FACE. THE AIR SEAL/SHOULDER ASSEMBLY WAS REPLACED ALSO.

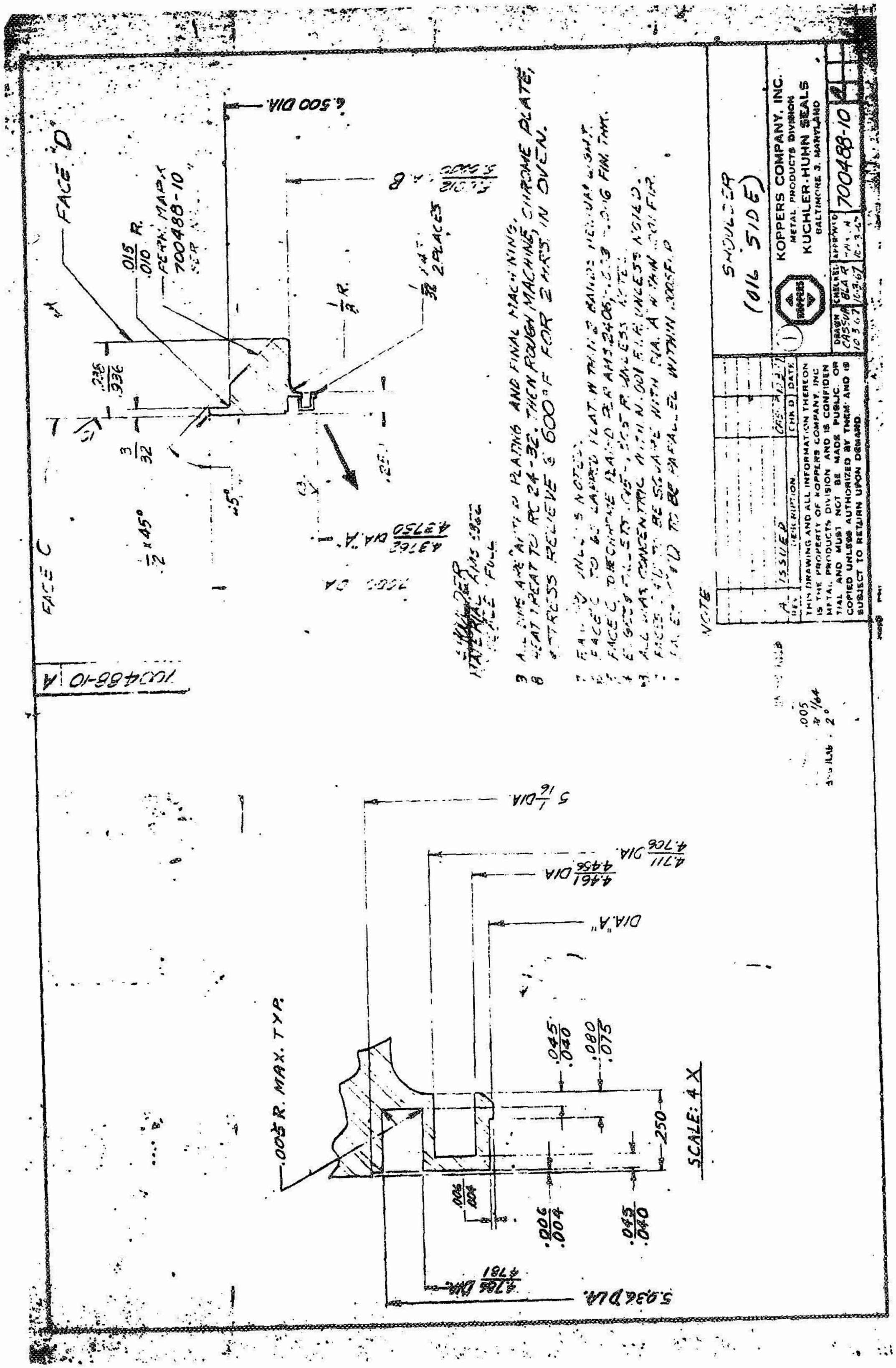
ENCLOSURE 8

BASIC BELLOWS ~~OIL SEAL~~ DESIGN

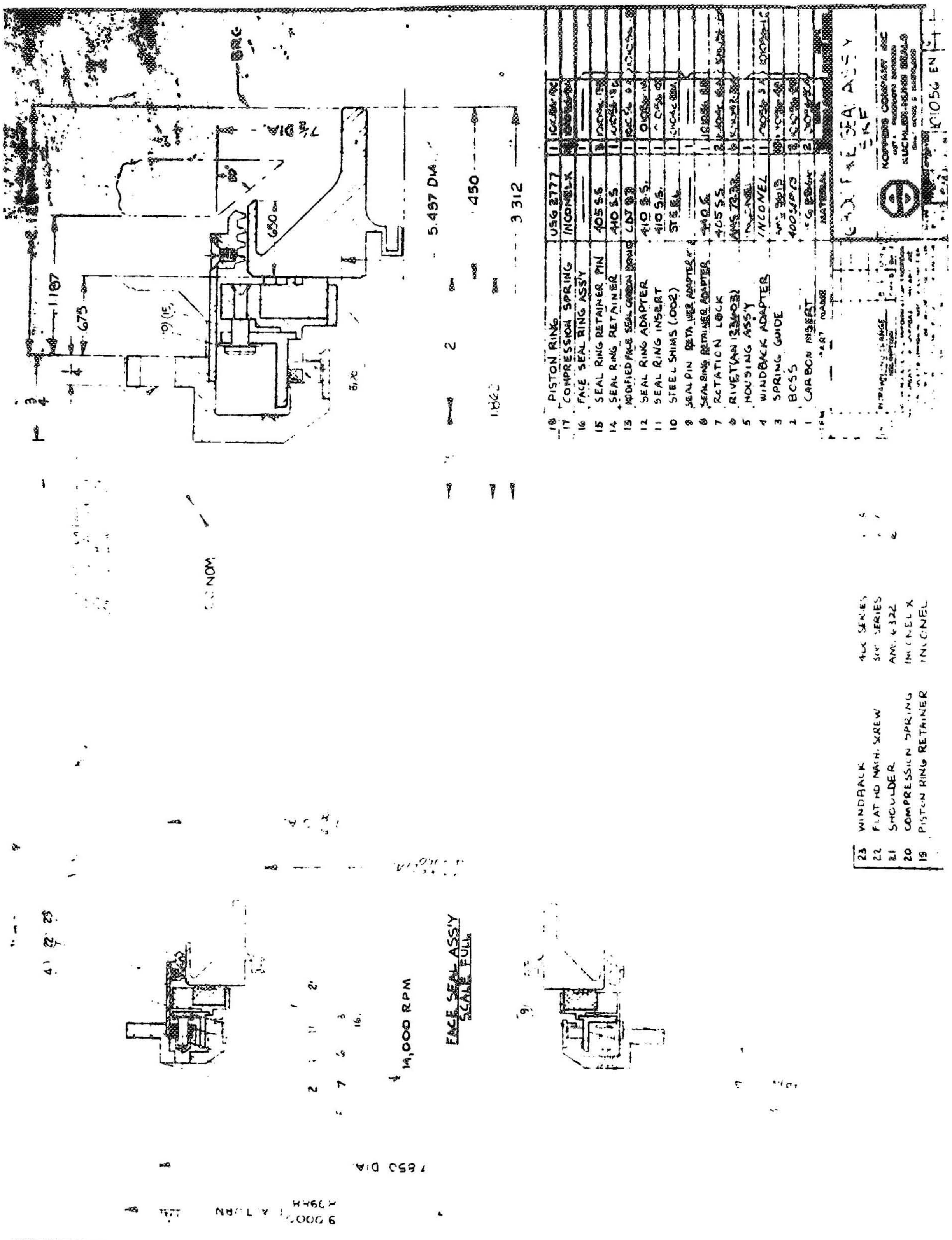


ENCLOSURE 9

BASIC BELLOWS OIL SEAL SHOULDER DESIGN

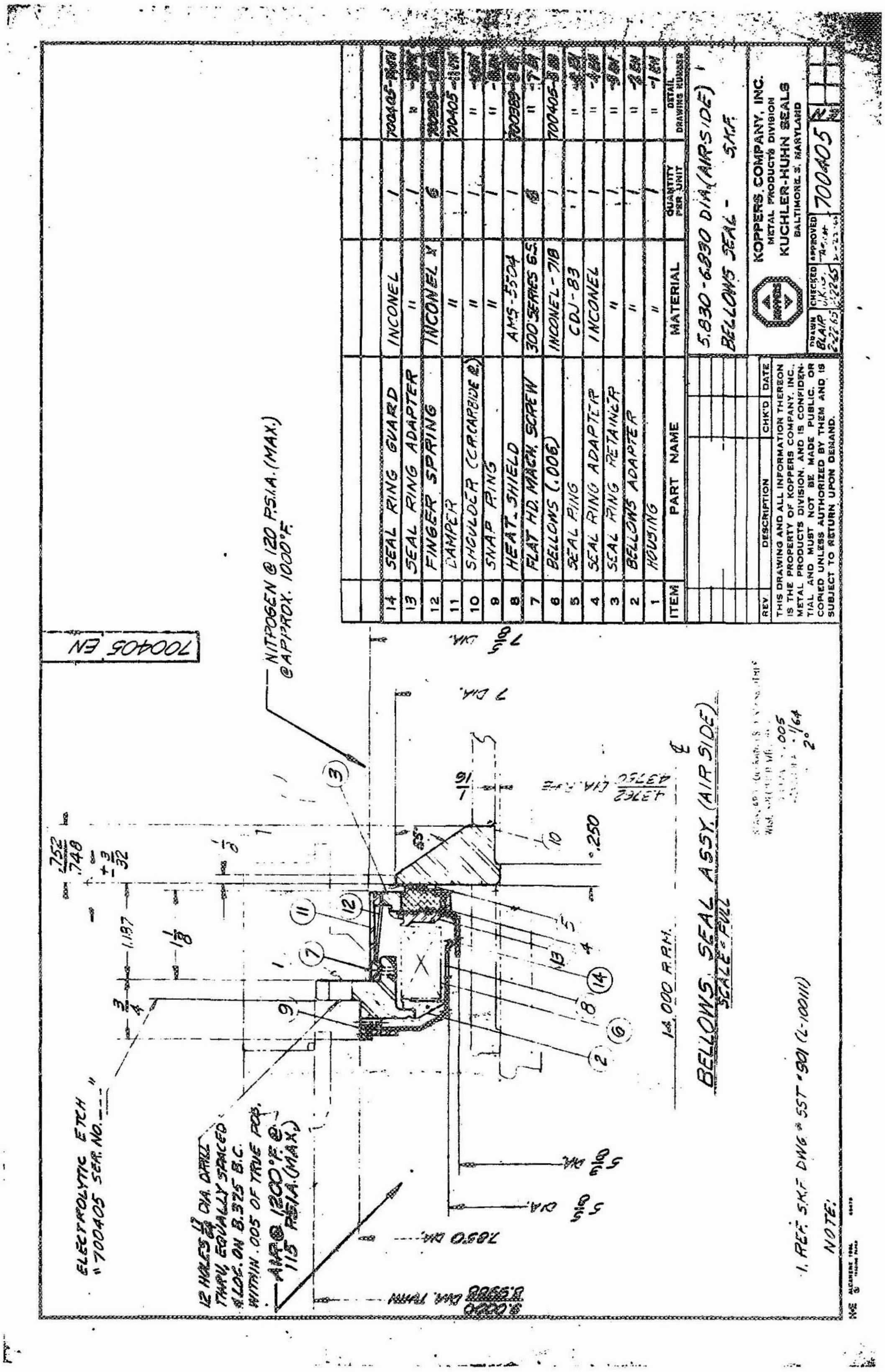


PISTON TYPE OIL SEAL DESIGN



ENCLOSURE 11

TYPICAL BELLOWS AIR SEAL AND SHOULDER DESIGN

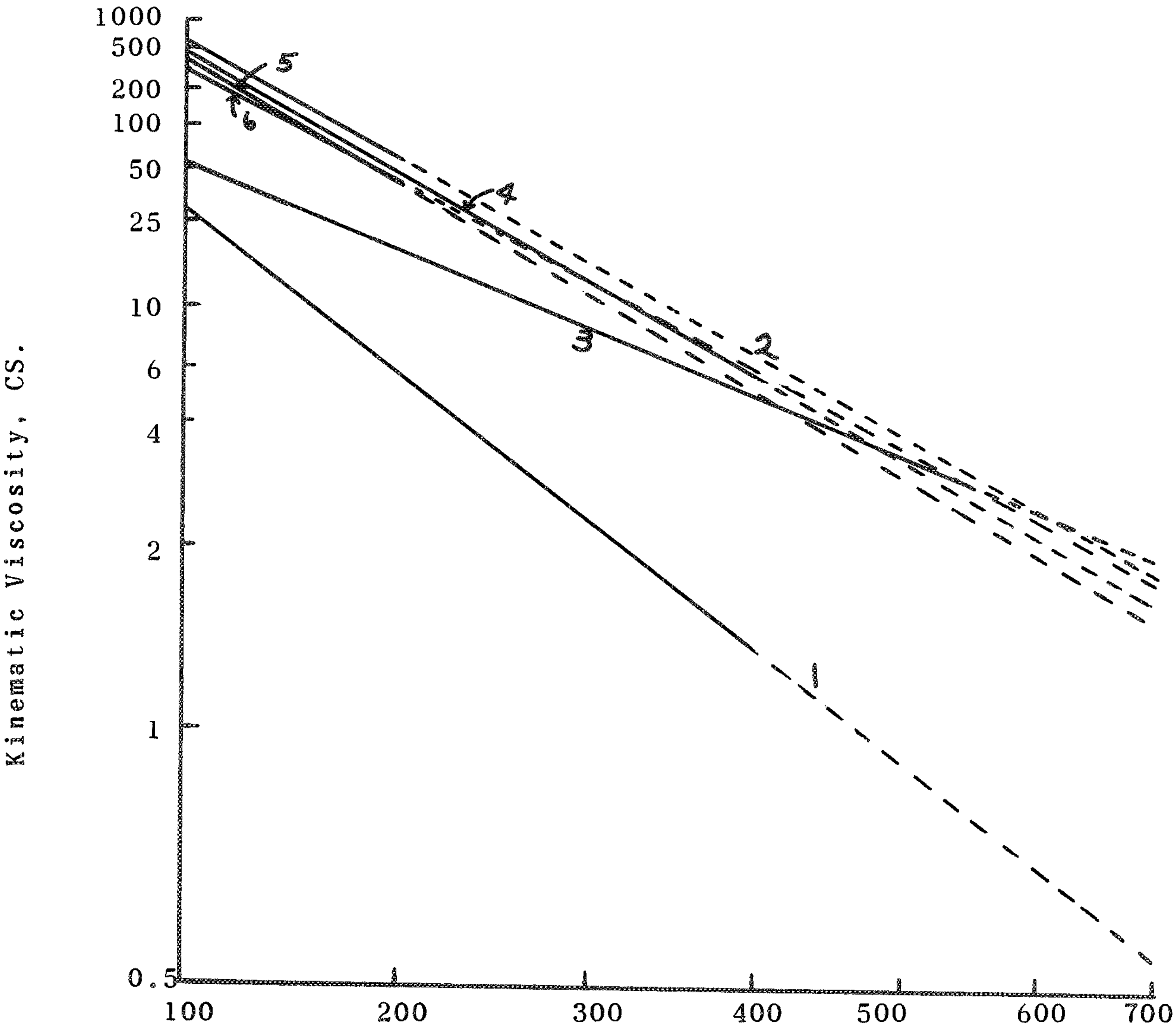


ENCLOSURE 12

VISCOSITY TEMPERATURE RELATION FOR CIRCULATING OIL

From Manufacturer's Data, Except 2, 5, and 6

1. Mobil Jet II
2. Blended Mobil XRM-109F and 10% Kendall Heavy Resin 0839
3. Mobil XRM-154D
4. Mobil XRM-109F (shown as reference)
5. Blended Mobil XRM-109F, Mobil XRM-127B and 10% Kendall Heavy Resin (
6. Mobil XRM-177F



Temperature, °F
(extrapolated values shown as dashed lines)

ENCLOSURE 13

SUMMARY OF TASKS II AND III TEST RESULTS IN RECIRCULATING-OIL RIG
N2 BLANKET EXCEPT WHERE NOTED
(PHASE I)

	Oil Flow Rate, gpm	Time at Test Conditions, Hours	Oil Inlet Temp., °F	Test Brg. Outer Ring / Inner Ring Temp. °F	Total Seal Leakage Rate, scfm	Conditions at End of Test Period: Oil	Test Bearing	Test Seals	Reason for Test Termination	Test Conclusion and Recommendation
4040	4 to 5	1.8*	500 to 555	590 to 600	1.5 to 10	Darkened, acid number very high	Considerable glazing, heavy cage pocket wear	Air Seal scored by particles from heater; oil seal OK	Test Brg. Failure	Lubricant not suitable for 600°F bearing operation; do not test @ higher temperature
77F	1 to 4	3*	460 to 550	575 to 625	3 to 21	Acid number unchanged, viscosity increased somewhat	Excellent	Oil seal coked and worn from previous use; air seal OK	High Seal Leakage	Lubricant acceptable for 600°F bearing operation; test @ next higher temperature
air S-	0.5 to 1	3.3*	480 to 500	590 to 625	6 to 8	Somewhat darker, acid number unchanged	Excellent	Oil seal leakage due to incorrect dam position in manufacture; air seal OK	Time-Up*	Lubricant acceptable for 600°F bearing operation-test @ next higher temperature
nto 93	0.5 to 1		480 to 500	580 to 600	1	Dark particles; properties unchanged	Outer ring badly flaked cage pockets worn	Excellent	Test Brg. Failure	Insufficient oil for adequate cooling flow; test @ next higher temperature with higher oil flow
nto 93 Atmos-	1	3.2	500 to 525	590 to 620	2 to 12	Dark particles, viscosity increased somewhat; acid number unchanged	Excellent	Oil seal somewhat scored; air seal OK	Time-Up	Suitable for testing @ next higher temperature
nto 93 Freon	1.25 to 2	3	490 to 515	600 to 620 / 595 to 630	7 to 11	Viscosity increased, acid number very high	Very good; a few small dirt dents in ball path	Relatively good; some scoring of carbon on oil seal. Air seal leaked excessively	Time-Up	Lubricant suitable at 600°F test at higher temperature
t 3 AC	2.5		500 to 505	610 to 615 / approx. 30°F above outer	10 to 13	Changed from transparent to milky color	Very good; some discoloration caused by fluid	Reasonably good. Oil seal runner showed signs of surface distress	Time-Up	Lubricant suitable at 600°F test at higher temperature
t 3 AC	2	3.7	500 to 510	690 to 780 (700 for .9 hrs.)	15 to 23	Viscosity increased acid number unchanged	Excellent; slight cage pocket wear brg. discolored	Good Condition	Time-Up	Lubricant suitable at 700°F test at higher Temperature
t 3 AC pt gs.)	0.5 to 2	0.7	490 to 510	600 to 740	5 to 10	Viscosity increased acid number unchanged	One brg. smeared at 600°F (0.5 gpm oil flow) other brg. smeared at 740°F	Some oil seal wear and scoring	Test Brg. Failures	Lubricant suitable at 700°F, endurance test at 650°F, if selected
air -S	0.8 to 1	3.3	450 to 500	600 to 700 (670 for 1.8 hrs.) / 630 to 650	10 to 11	Somewhat darker viscosity increased acid number unchanged	Balls and rings discolored and etched	Good Condition	Time-Up	Lubricant suitable at 700°F; test at higher temperature
air -S	1	3.4	495 to 500	730 to 755 (750 for .4 hrs) / 760 to 780	1 to 3	Dark Viscosity and acid number unchanged	Good: discoloration, slight cage wiping and pocket wear	Good Condition	Time-Up	Lubricant suitable at 750°F test at higher temperature
air -S	1	0	500	720	6-10	Dark Viscosity and acid number slightly increased	Mild Surface distress on balls & rings heavy cage pocket wear and wiping	Good Condition	Test Brg.	Lubricant somewhat thermally unstable in the 700-800°F failure range
77F	1 to 1.5	7.2	480 to 500	700 to 765 (1.8 hrs at 750) / 705 to 765	2-3 except at end of test	Viscosity decreased acid number unchanged	Good; moderate cage pocket wear noticable cage wiping	Good Condition	Oil Seal lift off	Lubricant suitable up to 750°F
air -S	1 to 1.5	9.7	475 to 520	610 to 640 / 640 to 670	6 to 26	Viscosity increased. Acid no. increased.	Good; moderate cage pocket wear	Oil seal coked	Oil seal lift off	Lubricant suitable for further endurance testing
177F	1.5	0	500	600 max.	8-10		Good Condition	Good Condition	High Oil Seal Leakage high bearing cavity pressure	Need improved oil seals

denotes tests where there were minor deviations of temperatures other the test bearing from full test conditions (i.e., test bearing housing, air, etc.).
mpted long term test; conditions never reached.

ENCLOSURE 13 (CONT'D)

SUMMARY OF TASKS II AND III TEST RESULTS IN RECIRCULATING OIL RIG

ENDURANCE TEST PHASE II

	Oil Flow Rate gpm	Time at Test Conditions hrs.	Oil Inlet Temp °F	Test Brg. Outer Ring / Inner Ring Temp. °F	Total Seal Leakage rate scfm	CONDITIONS AT END OF TEST PERIOD Oil	Test Bearing	Test Seals	Reason for Test Termination	Test Conclusion and Recommendati
	1.5	2.1	510	640-660 -	5.1-13.8					
	2.0	9.1	500-510	640-650 640-660	3.7- 7.6	Visc. down slightly				
	2.0	10.0	500-515	640-650 660-695	3.5- 4.9	Visc. down				
	2.0	10.0	500-520	645-665 670-695	4.8- 6.0	Visc. & acid no. up slightly				
	1.5-2.0	10.0	500-510	640-660 670-690	12.3-15.3					
	2.0	6.5	495-500	645-655 670-690	4.7- 7.3	Visc. up				
	2.0	6.0	500-505	645 665-670	7.0-10.6	Visc. up slightly	Good. Silver flaking off of cage showing copper flash.	Oil seal carbon wear 0.002". Air seal carbon wear 0.006" to 0.007"	Time-up	Suitable for fur long term testin
M50 I (112) MOBIL XRM	2	10.0	500-510	640-660 660-685	4.6- 7.9	Visc. & acid no. up				
109F + 10%	2	10.0	510	650-670 640-685	4.9- 6.4	Visc. up acid no down				
KENDALL	1	10.0	510	640-650 650-675	4.6- 6.0	Visc. & acid no. up				
HEAVY	1.5-1.75	10.0	510-525	650-665 625-640	4.6- 6.1	Visc. down				
RESIN	1.5	10.0	500-520	650-660 640-665	5.7- 6.9	Visc. & acid no. up				
0839	1.5-1.75	10.0	510-520	650-660 640-660	7.0- 8.5	Visc. up slightly acid no. down				
	1.5	10.0	510-520	650-670 650-680	5.5- 8.0	Visc. up slightly				
	1.5	7.5	510-520	640-660 640-660	5.6- 7.5	Visc. up slightly acid no down			Oil Heater fuse blown	
	1.75	8.5	500-510	645-675 645-680	9.0-17.4	Visc. up			Torque control malfunction	
	1.5	10.0	490-510	640-670 640-675	9.0-11.0	Visc. up & acid no. up				
	1.75	10.0	505-520	650-665 650-665	5.5- 7.0	Visc. up acid no. down				
	1.5	4.3	490-510	640-650 650-670	5.3-12.7	Visc. down slightly acid no. up			Rig seal lift off roller brg replaced	
	1.75	10.0	495-520	640-650 640-660	5.8- 7.6	Visc. up acid no down				
	1.5	10.0	510-515	640-650 645-660	7.5-11.1	Visc. down				
	1.5	10.0	500-505	640-660 640-665	6.1- 8.5	Visc. up & acid no. up				
	1.75	10.0	510	645-650 650-670	6.8- 7.7	Visc. up & acid no. up				
	1.5	10.0	500-520	640-660 650-670	6.2- 7.6	Visc. down				
	1.7-2.0	10.0	500	650-660 655-670	6.8- 8.8	Visc. and acid no. down				

ENCLOSURE 13 (CONT'D)

SUMMARY OF TASKS II AND III TEST RESULTS IN RECIRCULATING OIL RIG

BASELINE AND QUALIFYING TESTS, PHASE II

	Oil Flow Rate gpm	Time at Test Condit-ions hrs	Oil Inlet Temp °F	Test Brg. Outer Ring / Inner Ring Temp. °F	Total Seal Leakage Rate scfm	Conditions at End of Test Period: Oil	Conditions at End of Test Period: Test Brg.	Conditions at End of Test Period: Test Seals	Reason for Test Termi-nation.	Test Conclusion & Recommendation
WB49 II (107)										
Mobil XRM-109F	1	10	490-500	640-645 / 650-665	2	Viscosity &	Slight	Oil Seal	Oil Seal	Suitable for
plus	1.25	10	490-500	640-650 / 655-690	2.5- 2.8	acid no. up	glazing	shoulder	lift-off	further long term
Kendall Heavy	1.25	10	500-515	650-660 / 670-680	1.7- 2.5	slightly		grooved	Oil Seal	testing
Resin 0839	1.25	2.3	500	655 / 655	9			.002-.003	shoulder	
									grooved	
M-50 I (109)										
Mobil Jet II*	2	1	405	520 / 505	9.3-10.2	Viscosity &	Slight	All seal-	Time-up	Conditions con-
	2	9.6	390-435	518-545 / 500-545	3.8- 5.9	acid no. up	surface	ing elements		sidered too
	2	10	390-420	500 / 500	4.7- 6.8	slightly	distress	in good		severe for long
	2	7.6	390-410	510-525 / 500-525	4.9- 6.6		& heavy	condition		term operations
	2	2.4	390-405	500-520 / 485-520	4.5- 5.8		cage poc-			in this system.
	2	10	400-420	510-530 / 500-525	5.1- 6.8		ket wear			
	2	10	385-405	500-515 / 495-520	4.9- 6.4					
WB49 II (110)										
Mobil XRM-109F,	2	10	500-530	630-650 / 630-650	9.3-11.7	Viscosity	Good	Oil seal	Time-up	Suitable for
plus	2	10	520-530	630-650 / 640-650	8.9-10.6	increased		& shoulder		further long
Mobil XRM-127B &	2	10	515-530	640-660 / 640-650	10.6-13.6			good. Air		term testing.
plus	2	6	520	645-650 / 645-650	10.0-11.9			seal .005"		
Kendall Heavy	1.75	10	500-525	650-660 / 635-660	23.0-30.0			wear and		
Resin 0839	1.75	4	505	650 / 650	19.2-22.5			signs of		
								chattering		
								on the air		
								shoulder.		

* Open Atmosphere

ENCLOSURE 13 (CONT'D)

SUMMARY OF TASKS II AND III TEST RESULTS IN RECIRCULATING OIL RIG

SCREENING TESTS

(PHASE II)

***	Oil Flow Rate, gpm	Time at Test Conditions Hours	Oil Inlet Temp. °F	Test Brg. Outer Ring / Inner Ring Temp. °F	Total Seal Leakage Rate scfm	Conditions at End of Test Period: Oil	Test Bearing	Test Seals	Reason for Test Termination	Test Conclusion and Recommendati
WB49 II (101) Mobil Jet II	0.9 to 1.2	2.2	450 to 460	540	13.5 to 20	Viscosity & acid no. up slightly	Cage seized on the O.R. lands	Oil seal and oil seal shoulder were scorrd	Test brg. failed due to loss of oil	Test at next highe temperature
WB49 II (102) Mobil Jet II	1.0 to 1.75	0.30	500	500	2.5 to 15.0	Viscosity decreased. Moderate increase in acid number	Slight cage wiping on O.R. lands	Oil seal carbon completely worn down	Oil seal failure	Test was not run long enough to determine 600°F operation potentia Another test will run
WB49 II (103) Mobil Jet II	0.8 to 1.5	0.2	500	600	4 to 5	Viscosity up acid no. up sharply	Test brg. smeared	Oil seal & Air seal shoulders show signs of chatter	Test brg. failure	Max. short term te 500°F.
WB49 II (104) Mobil XRM-154D	3/4 to 3	2.3**	460	470 to 490	8 to 12	Viscosity increased. Acid No. unchanged. Heavy Oil Deposits in rig.	Moderate Surface Distress & Heavy Cage Pocket Wear	Good Condition. Oil seal shoulder showed signs of chattering.	Oil Seal Lift-off	Due to high oxygen content another test should be rur before definite conclusions are made.
WB49 II (108) Mobil XRM-154D	1.5 to 2.0	3*	480 to 495	630 to 650	2 to 4.7	Very light oil deposits in rig	Slightly glazed	Oil Seal scored severely	Oil Seal lift off	Suitable as a lubricant, however because of deposit formation not suitable for further running.
WB49 II (106) Mobil	1.25 to	3*	430	515 to 618 / 520 to 618	.0.3 to 0.4	Viscosity	Very slight	Oil seal	Time-up	Suitable for furtl
XRM-109F	1.5	3	490 to 520	680 to 725 / 670 to 700	1.8 to 11.4	increased	glazing	shoulder		long term testing
plus Kendall Heavy						high acid		severely		
Resin 0839						No.		scored		

* This denotes tests where there were minor deviations of temperatures other than the test bearing from full test conditions (i.e. test bearing housing, hot air, etc.)

** Attempted long term test; conditions never reached.

*** Bearing material (WB49 or M-50), bearing design series (I or II), bearing serial no. (101, 102, etc), and test lubricant.

ENCLOSURE 13 (CONT'D)

SUMMARY OF TASKS II AND III TEST RESULTS IN RECIRCULATING OIL RIG

ENDURANCE TESTS PHASE II

	Oil Flow Rate gpm	Time at Test conditions hours	Oil Inlet Temp °F	Test Brg. Outer Ring / Inner Ring Temp. °F	Total Seal Leakage rate scfm	CONDITIONS AT END OF TEST PERIOD: Oil	Test Bearing	Test Seals	Reason for Test Termination	Test Conclusion and Recommendation
9 II (105)										
il XRM	2.0	6.0	520-530	588-600 / 600-615	1.7- 2.4	Visc. up slightly	Good	Air seal & shoulder good oil seal .010" wear shoulder grooved .009"	Roller brg. failure & oil seal shoulder grooved	Suitable for further long term testing. Brg. puller groove has a 1 gm. chunk of metal pulled out of it.
F	2.0	12.9	515-545	580-620 / 605-625	1.3-20.4					
I (111)										
il XRM	1.0-1.5	10.0	490-520	640-670 / 640-655	4.7-19.5	Visc. up, acid no. up slightly	Good	Air seal & shoulder good oil seal carbon worn .048" shoulder grooved 0.005-0.010"		This oil seal failure was found by Koppers to be due to the inadvertant use of USG 2925 carbon.
F										
	1.0-2.0	8.9	510-530	640-660 / 650-680	8.5-12.7					
	2.0	10.0	510-530	640-655 / 645-670	9.8-11.9	Visc. up slightly				
	2.0	10.0	490-525	650-660 / 650-660	3.8- 4.6	Visc. up, acid no. down				
	2.0	7.8	500	655-670 / 655-670	3.6- 6.1				Oil sump heater blown	
	1.5-1.75	10.0	490-505	650-606 / 665-680	3.1- 4.8	Visc. down				
	1.75	1.0	500	650 / 670	4.4				Oil return line broken	
	1.25	10.2	500-515	640-660 / 645-675	3.0- 4.6	Visc. down				
	1.5	10.0	510-520	650-660 / 660-670	6.0- 9.3	Visc. up and acid no. up				
	2.0	10.0	500-520	640-650 / 650	4.8- 6.8	Visc. up and acid no. up slightly				
	1.5-2.0	10.0	510-515	640-650 / 640-650	9.7-12.3	Visc. and acid no. down slightly				
	1.0-1.5	10.0	510-515	645-660 / 650-670	9.5-12.6	Visc. and acid no. about the same				
	1.5	10.0	510	645-650 / 660-665	10.0-12.7	Visc. up acid no. down				
	1.5	10.0	510	645-660 / 655-670	12.0-14.7	Visc. & acid no. down				
	1.5	0.8	515	650 / 650	8.9				Broken oil line fitting	
	2.0	10.0	500-520	645-655 / 655-660	11.0-13.6	Visc. down				
	2.0	4.4	490-510	650-660 / 660-675	7.6- 9.8				Shear pin broke	
	2.0	10.0	510-515	640-665 / 650-670	4.0- 5.9	Visc. & acid no. down slightly				
	2.0	8.5	510	655-660 / 670-675	4.1- 6.8	Visc. down slightly			Shear pin broke	
	2.0	10.0	490-500	650 / -	4.2- 5.3					
	2.0	4.7	490-500	630-650 / 630-660	6.1- 9.3	Visc. up slightly				

ENCLOSURE 13 (CONT'D)

SUMMARY OF TASK IV TEST RESULTS IN RECIRCULATING OIL RIG

HIGH SPEED TESTS PHASE II

	Oil Flow Rate gpm	Time at Test Conditions hrs.	Oil Inlet Temp. °F	Test Brg. Outer Ring / Inner Ring Temp. °F	Total Seal Leakage rate scfm	CONDITIONS AT END OF TEST PERIOD: Oil	Test Bearing	Test Seals	Reason for Test Termination	Test Conclusion and Recommendation
M50 I (113) MOBIL *	2 [1]	0.5	395-410	490-500 505-515	12.7-14.4					
JET II	1.5 [1]	0.5	390	505-510 520-530	15.3					
	1.0 [1]	0.5	380	510-545 530-560	15.3-38			Oil Seal Carbon wear .016" Oil seal runner grooved. Air seal shoulder spalled. Air seal replaced.	Excessive seal leakage	
	2.0 [3]	0.5	-	510-560 510-570						
	1.5 [3]	0.5	450	560-600 570-575	7.2					
	1.0 [3]	0.5	430	590-600 600-580	6.8				Belt slipped off while going to 18,000 rpm	
	2.0 [4]	0.1	-	600-610 605-620					Rig seal lift off	
	2.0 [4]	0.4	450	560-645 560-650	7.6				Rig seal lift off	
	1.5 [4]	0.5	430	575-640 575-645	6.8- 7.2					
	1.0 [4]	0.5	430-400	650-620 680-635	5.9- 6.8					
	0.75 [4]	0.5	400-355	610-620 630-645	6.8- 7.6					
	1.5 [5]	0.4		610-620 620-630					Rig seal lift off	
	1.0 [5]	0.3		610-650 675-610		Visc. Increased Acid No. very high	IR, OR, balls bluish color scaling of metal. Cage silver plating melted away exposing copper.	Oil seal carbon wear .003-.007". Air seal carbon wear .0005".	Speed change Fire in rig	Long term high-spe tests are feasible.
M50 I (114) MOBIL*	1.25 [2]	0.1	190	390 420	12.3					
JET II	1.0 [3]	0.2	190	410 440	11.5					
	0.75 [4]	0.2	205	440 460	10.2					
	0.75 [5]	0.1	210	500 520	9.8				Low water pass to magnetic clutch caused automatic shutdown	
	0.75-1.0 [5]	3.1	175-210	510-575 535-620	6.4-7.6	Visc. and acid no. increased	All part seized	Oil seal carbon wear .008 air seal carbon wear .002	Bearing seizure	Long term high-spe tests are feasible

* Open Atmosphere
1 14,000 RPM
2 15,000 RPM
3 16,000 RPM
4 18,000 RPM
5 20,000 RPM

ENCLOSURE 13 (CONT'D)

SUMMARY OF TASKS II AND III TEST RESULTS IN RECIRCULATING OIL RIG

ENDURANCE TESTS PHASE II

Oil Flow Rate gpm	Time at Test Conditions hrs.	Oil Inlet Temp. °F	Test Brg. Outer Ring / Inner Ring Temp. °F	Total Seal Leakage rate scfm	Conditions at End of Test Period: Oil	Conditions at End of Test Period: Test Bearing	Conditions at End of Test Period: Test Seals	Reason for Test Termination	Test Conclusion and Recommendation
1.75	10.0	500	650-660 / 640-665	6.6- 8.6	Visc. down acid no up				
1.5	10.0	500	650-665 / 655-670	6.2- 8.3	Visc. up acid no. down				
1.75	10.0	505-515	645-650 / 650-665	4.2- 8.6	Visc. and acid no up slightly				
1.5	10.0	490-515	640-650 / 650-660	4.4- 5.8	Visc. down				
1.5	2.9	495-500	640-650 / 650-660	4.4- 5.1				Motor-clutch gear coupling failed	
2	10.0	500	645-660 / 655-665	4.2- 5.1					
1.75 to 2.0	10.0	490-500	645-660 / 645-670	4.2- 5.9					
1.5	10.0	490-515	640-660 / 640-670	4.4- 6.5					
1.5	6.8	510-520	645-660 / 660-675	4.7- 7.0	Visc. increased	IR & OR good signs of silve flaking on cage exposing copper	Oil seal carbon wear .003" air seal carbon wear .001"	Time-up	Long term testing is possible using present brgs, seal & oils. Investigate bellows seals further with hydrodynamic shoulder to further decrease seal leakage.

ENCLOSURE 14

SUMMARY OF TEST OIL VISCOSITY AND ACID NO. BEFORE AND AFTER TEST

Test Number	Oil	NEW DEGASSED Visc. @ 100°F Cs/Acid No.	USED Visc. @ 100°F Cs/Acid No.	Condition	Oil Added (Gallons)
1	Mobil Jet II	27.8/0.1	28.1/0.2	2.2 hrs. @ 550°F	
2	Mobil Jet II	27.8/0.1	23.9/0.2	0.7 hrs. @ 550°F	
3	Mobil Jet II	27.8/0.1	28.7/0.4	0.2 hrs. @ 600°F	
4	Mobil XRM 154D	55.3/0.2	60.6/0.2	1.1 hrs. from 400 to 500°F	
5	Mobil XRM 177F	442.3/0.01	462.8/0.05	20 hrs. from 590 to 620°F	
6	Mobil XRM 109F + 10% by wgt. Kendall Heavy Resin 0839	550.4/0.05	588.8/0.11	3 hr. @ 600°F	
				3 hr. @ 700°F	
7	Mobil 109F + 10% by wgt Kendall Heavy Resin 0839	550/0.05	648/0.16	20 hrs. @ 630-650°F	1.5
			609/0.16	30 hrs. @ 630-660°F	
8	Mobil XRM 154D	55.4/0.05	66.5/0.2	3 hrs. @ 650°F	
9	Mobil Jet II	28/0.1	32/0.2	10 hrs. @ 510-545°F	
			31/0.2	20 hrs. @ 470-500°F	
			31/0.1	30 hrs. @ 500-515°F	
			33/0.3	40 hrs. @ 510-530°F	
			33/0.3	50 hrs. @ 500-530°F	
10	Mobil 109F, Mobil 127B + 10% by weight Kendall Heavy Resin 0839	465/0.1	685/0.1	10 hrs. @ 630-660°F	
			742/0.1	20 hrs. @ 630-650°F	
			776/0.1	30 hrs. @ 640-660°F	2.5
			675/0.1	35.9 hrs. @ 640-650°F	
			829/0.1	45.9 hrs. @ 640-660°F	
			872/0.1	50 hrs. @ 650°F	
11	Mobil XRM-177F	442.3/0.01	557.4/0.05	10 hrs. @ 640-650°F	
			559.5/0.05	28.9 hrs. @ 635-660°F	
			603.2/0.03	38.9 hrs. @ 650-660°F	
			497.6/0.04	56.7 hrs. @ 650-670°F	
			482.2/0.04	67.9 hrs. @ 630-660°F	5
			570.5/0.06	77.9 hrs. @ 650-660°F	7.5
			580.3/0.07	87.9 hrs. @ 630-650°F	1.5
			575.0/0.06	97.9 hrs. @ 640-650°F	4.0
			576.5/0.05	107.9 hrs. @ 645-655°F	3.0
			605.6/0.04	117.9 hrs. @ 640-650°F	1.0
			589.8/0.03	127.9 hrs. @ 635-655°F	2.0
			559.0/0.03	138.7 hrs. @ 650-655°F	11.0
			544.4/0.02	153.1 hrs. @ 630-665°F	8.0
			511.2/0.02	161.6 hrs. @ 630-660°F	
			551.3/0.02	176.3 hrs. @ 640-650°F	
			544.4/0.02	187.5 hrs. @ 630-660°F	2.0
			496.1/0.02	197.5 hrs. @ 640-650°F	
			503.0/0.03	207.5 hrs. @ 645-650°F	5.0
			559.7/0.03	224.0 hrs. @ 640-655°F	
			569.1/0.03	230.0 hrs. @ 630-645°F	10.0
12	Mobil XRM-109F plus 10% by wgt Kendall Heavy Resin 0839	550.4/0.05	595.2/0.09	10 hrs. @ 640-660°F	4.3
			676.2/0.07	20 hrs. @ 650-670°F	
			718.1/0.08	30 hrs. @ 640-650°F	1.1
			657.4/0.08	40 hrs. @ 650-665°F	3.4
			721.6/0.10	50 hrs. @ 650-660°F	2.4

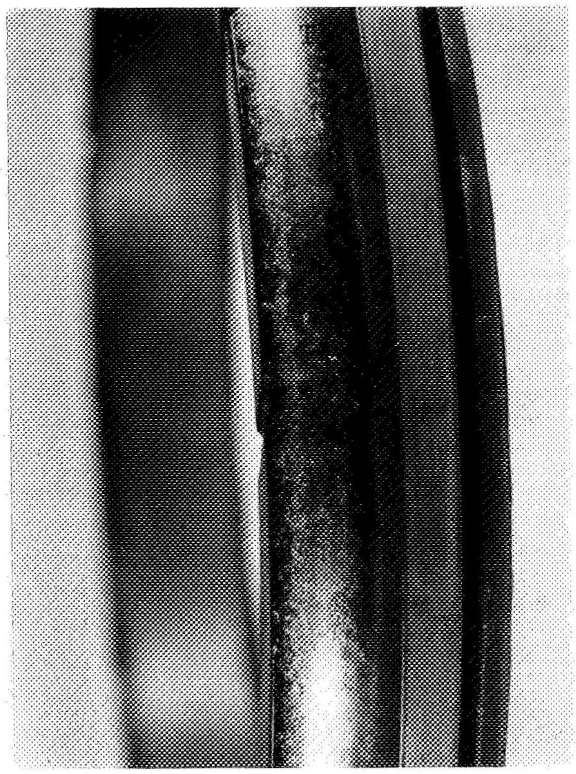
ENCLOSURE 14 (CONT'D)

SUMMARY OF TEST OIL VISCOSITY AND ACID NO. BEFORE AND AFTER TEST

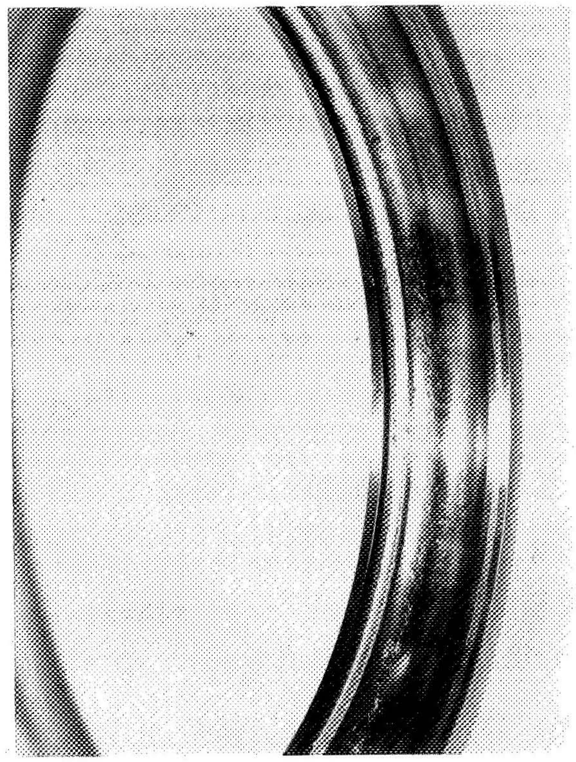
Test Number	Oil	NEW DEGASSED Visc. @ 100°F Cs/ Acid No.	USED Visc. @ 100°F Cs/Acid No.	Condition	Oil Added (Gallons)
12	Mobil XRM-109F plus 10% by wgt. Kendall Heavy Resin 0839 (continued)	550.4/0.05	731.2/0.09	60 hrs. @ 640-665°F	1.0
			742.4/0.09	70 hrs. @ 650-670°F	1.0
			748.9/0.07	77.5 hrs. @ 640-660°F	1.3
			773.0/0.07	86 hrs. @ 650-675°F	2.0
			808.5/0.08	96 hrs. @ 650-670°F	
			817.9/0.07	106 hrs. @ 650-665°F	3.6
			803.1/0.10	110.3 hrs. @ 640-650°F	
			841.2/0.07	120.3 hrs. @ 640-650°F	2.6
			667.8/0.07	130.3 hrs. @ 640-650°F	1.4
			846.3/0.08	140.3 hrs. @ 655-660°F	1.7
			920.2/0.12	150.3 hrs. @ 645-655°F	2.5
			852.0/0.12	160.3 hrs. @ 650-660°F	
			814.6/0.09	170.3 hrs. @ 650-660°F	5.0
			740.2/0.14	180.3 hrs. @ 650-660°F	2.4
			817.5/0.07	190.3 hrs. @ 650-665°F	
			824.7/0.08	200.3 hrs. @ 645-650°F	4.0
			773.9/0.08	210.3 hrs. @ 650-655°F	
			793.3/0.06	223.2 hrs. @ 650-660°F	3.1
			780.9/0.06	233.2 hrs. @ 655-660°F	1.4
			776.9/0.06	243.2 hrs. @ 655-660°F	
			774.4/0.09	250 hrs. @ 640-660°F	
13	Mobil Jet II	28/0.1	65.4/3.8	13.3 hrs. @ speeds from 14,000 to 20,000 rpm and @ 410°-660°F	
14	Mobil Jet II	28/0.1	29.5/0.8	4.5 hrs. @ speeds from 14,000 to 20,000 rpm and @ 390° - 575°F	2.5

ENCLOSURE 15

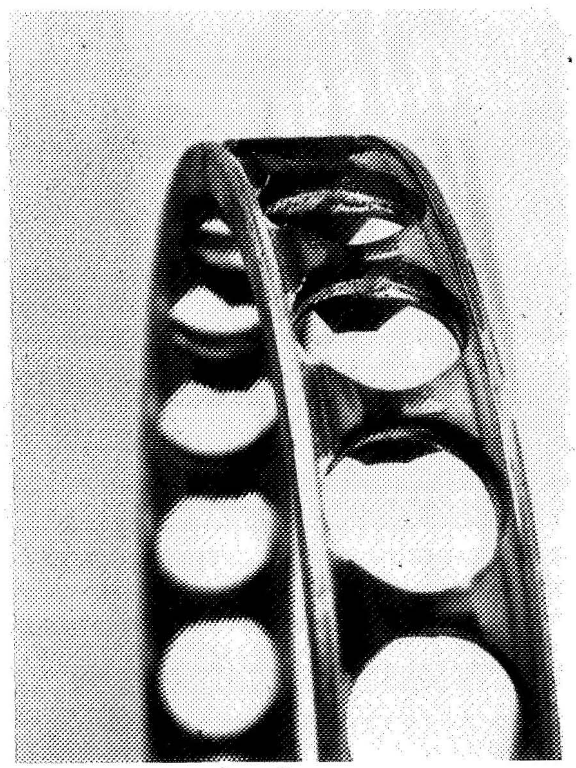
Test Bearing Parts After 550°F Mobil Jet Oil II Screening Run for 2.2 Hours



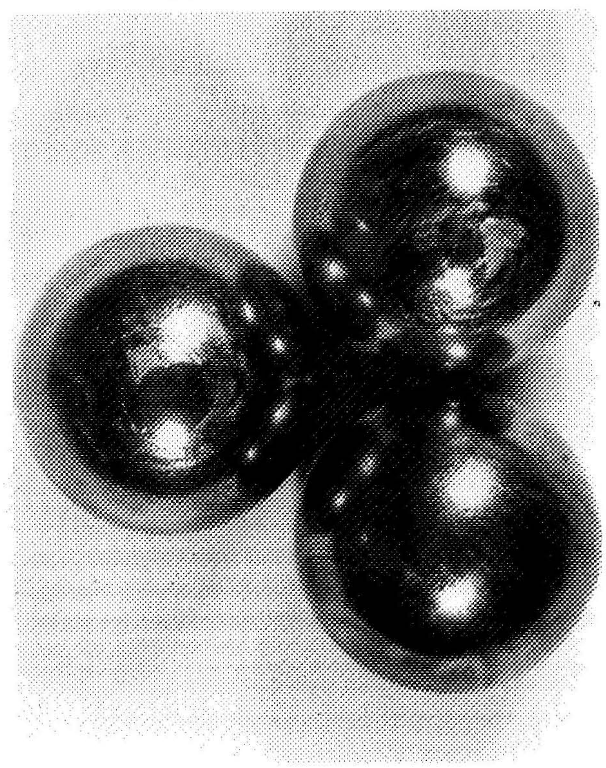
a) Inner Race



b) Outer Race

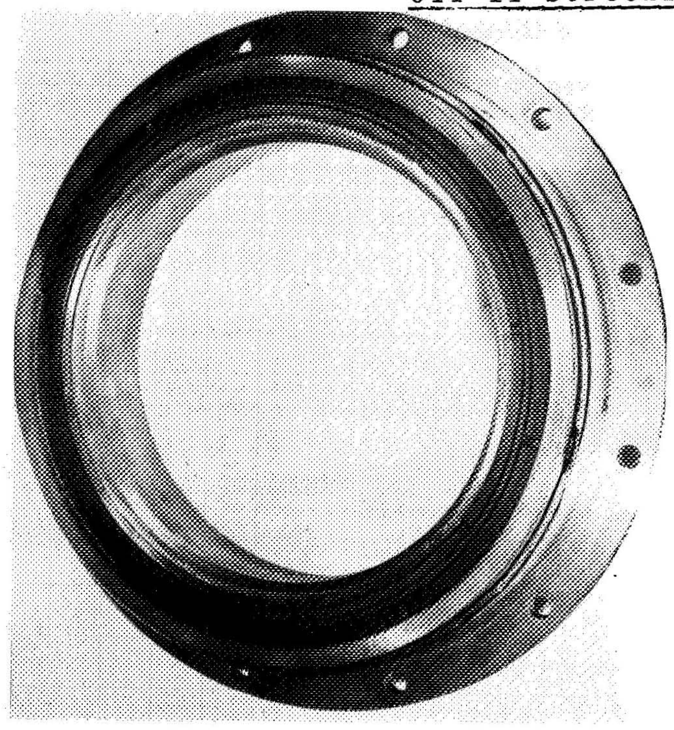


c) Cage

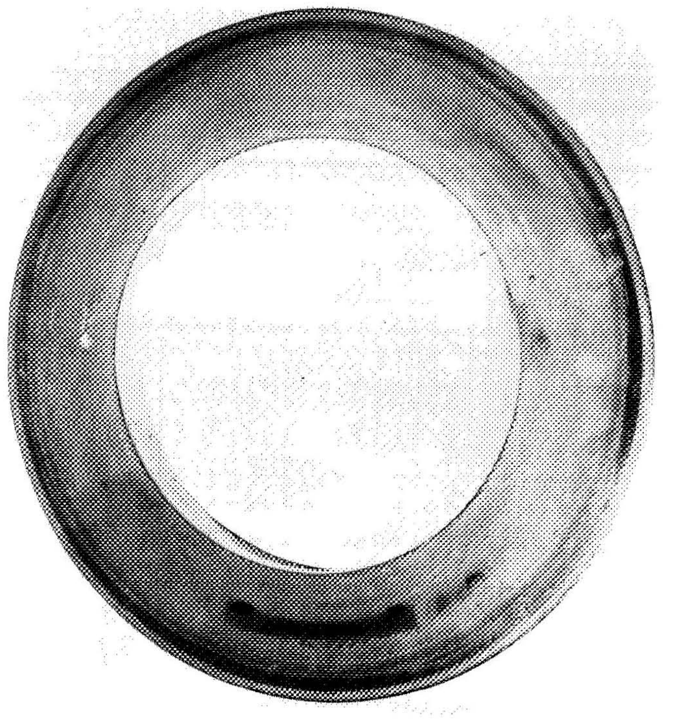


ENCLOSURE 16

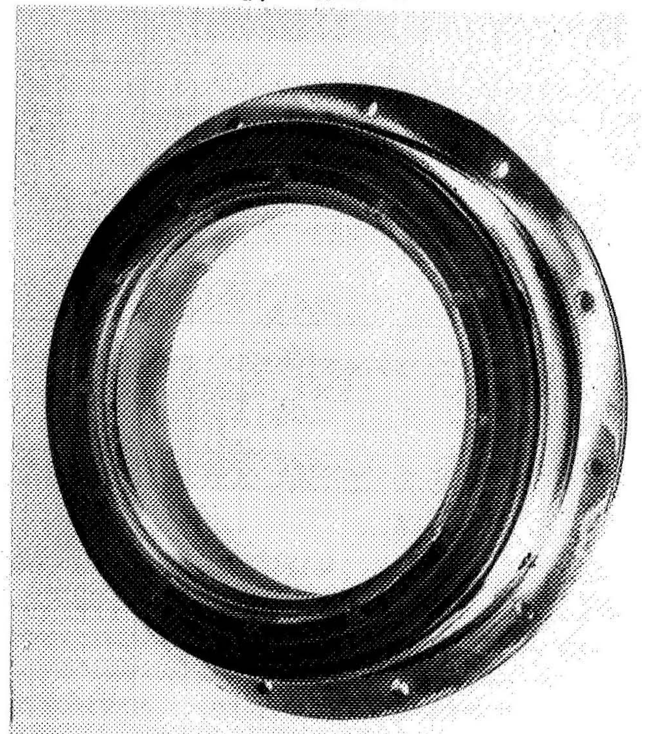
Test Seal Parts After 550°F Mobil Jet Oil II Screening Run for 2.2 Hours



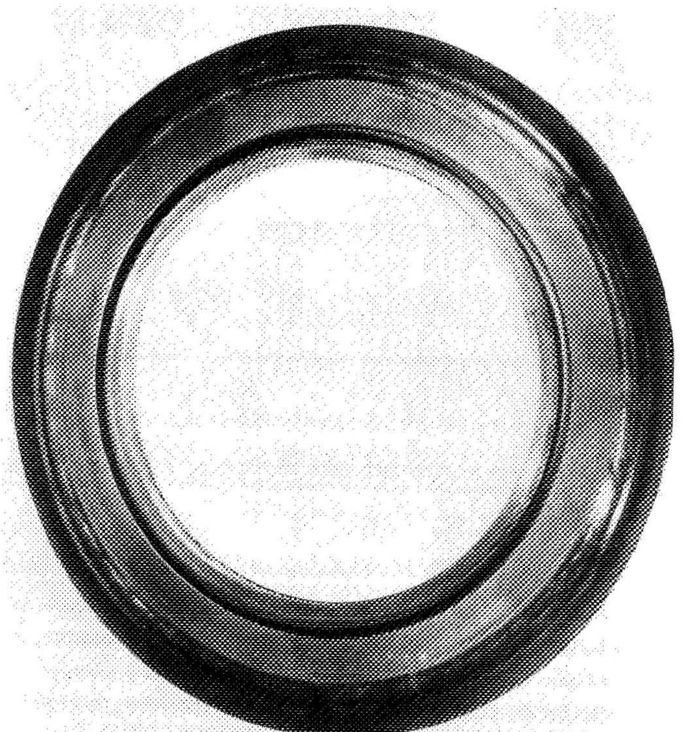
a) Air Seal



b) Air Seal Runner



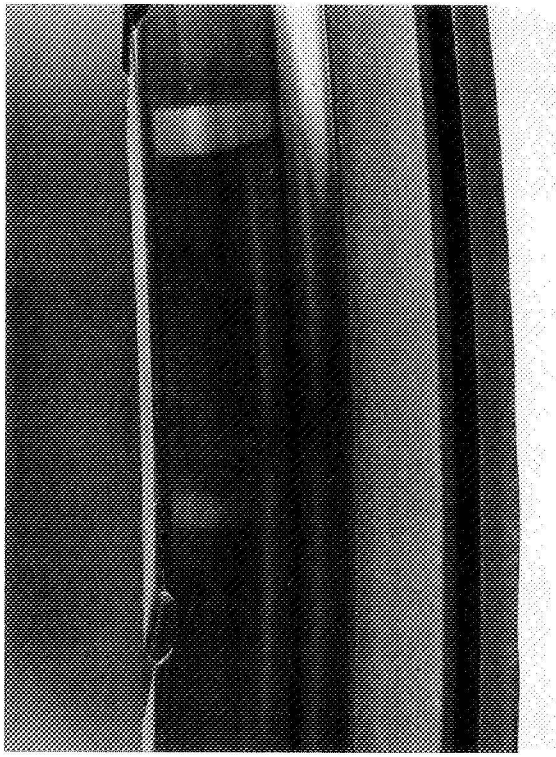
c) Oil Seal



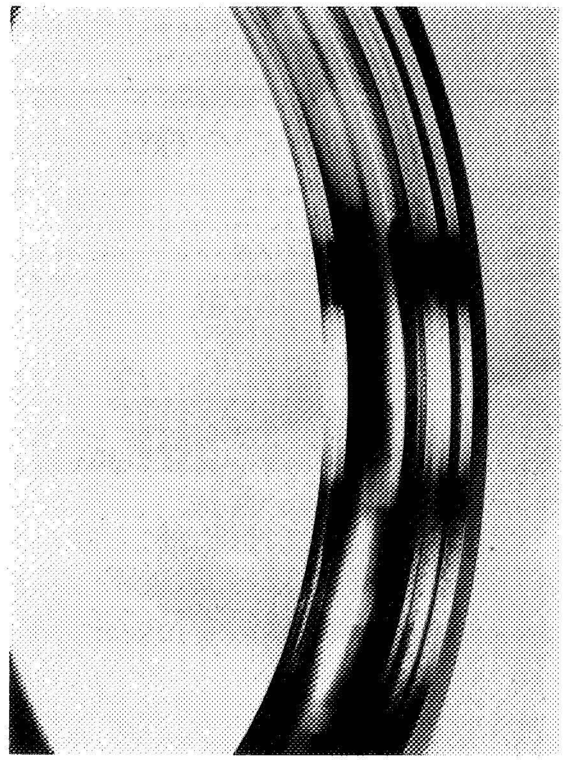
d) Oil Seal Runner

ENCLOSURE 17

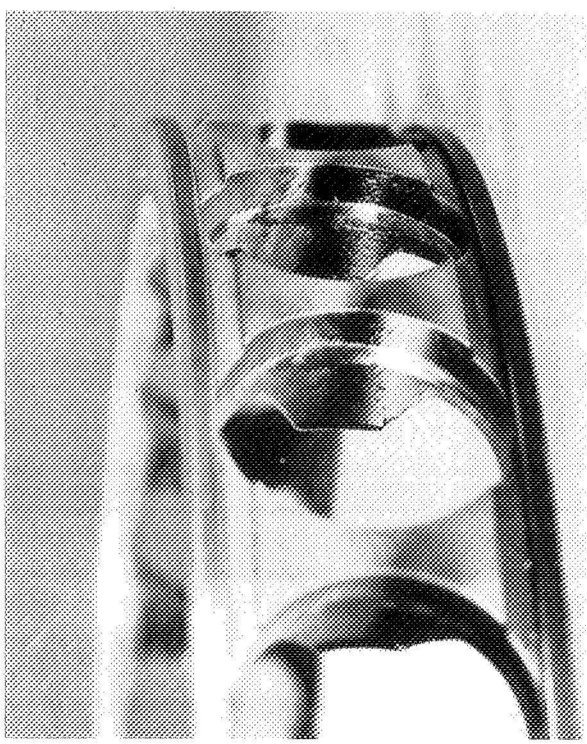
TEST BEARING PARTS AFTER 560°F MOBIL JET OIL II
SCREENING RUN FOR 1.0 HOUR



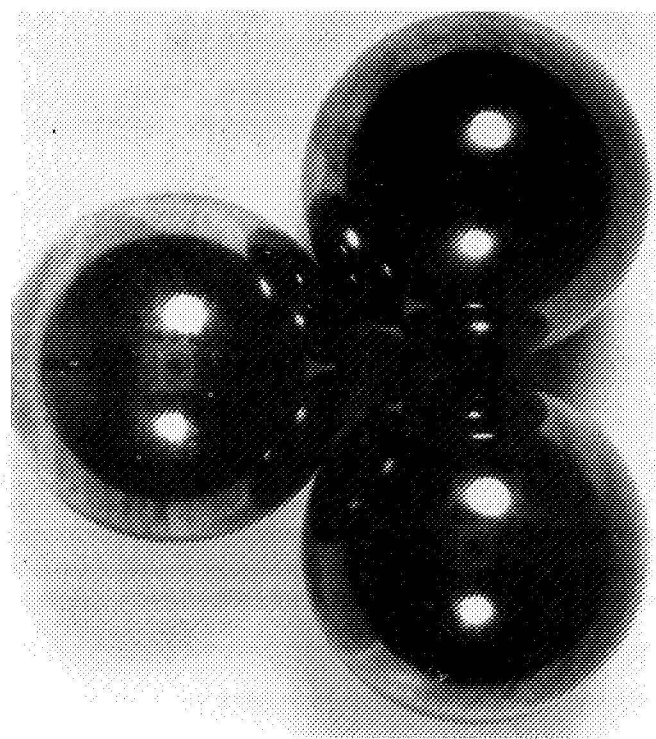
a) Inner Race



b) Outer Race



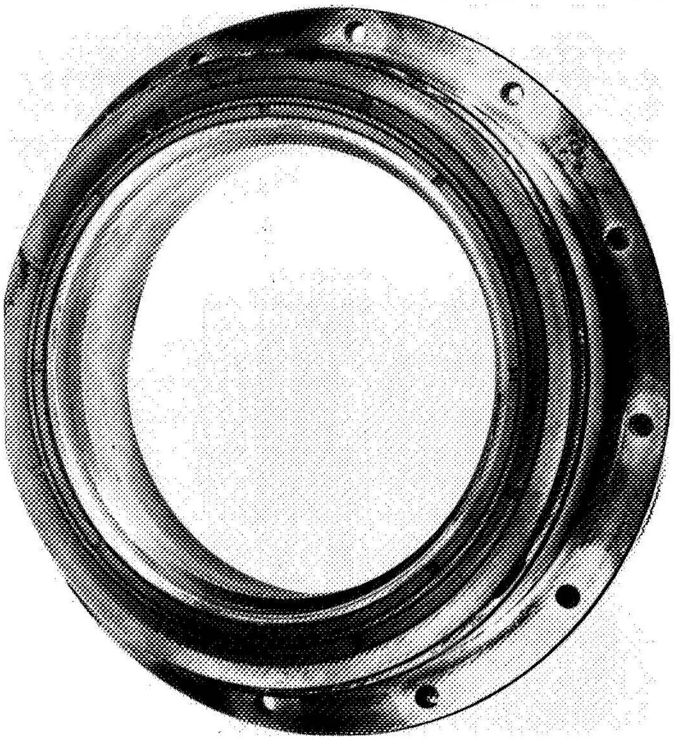
c) Cage



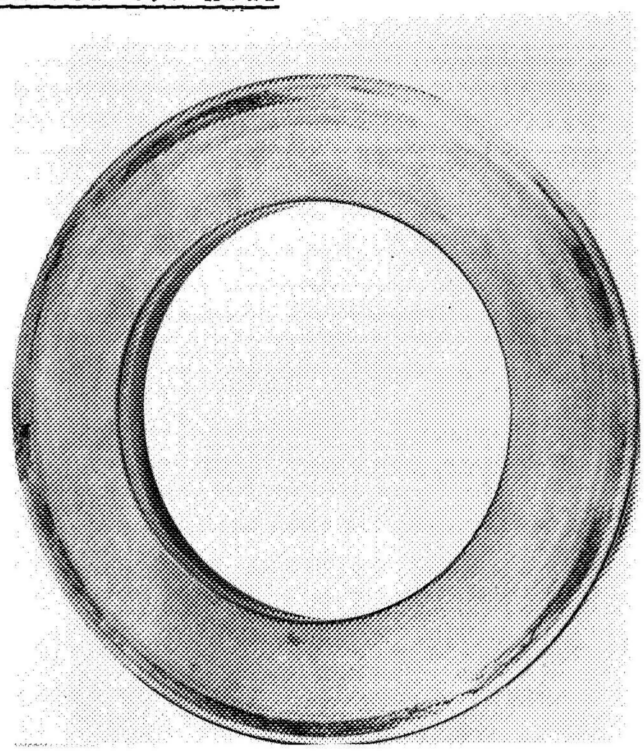
d) Balls

ENCLOSURE 18

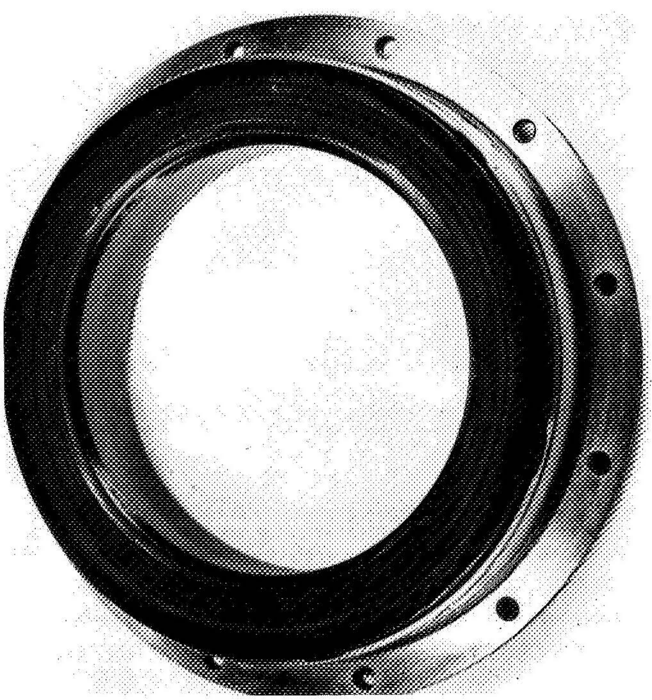
Test Seal Parts After 560°F Mobil Jet Oil II Screening Run for 1.0 Hour



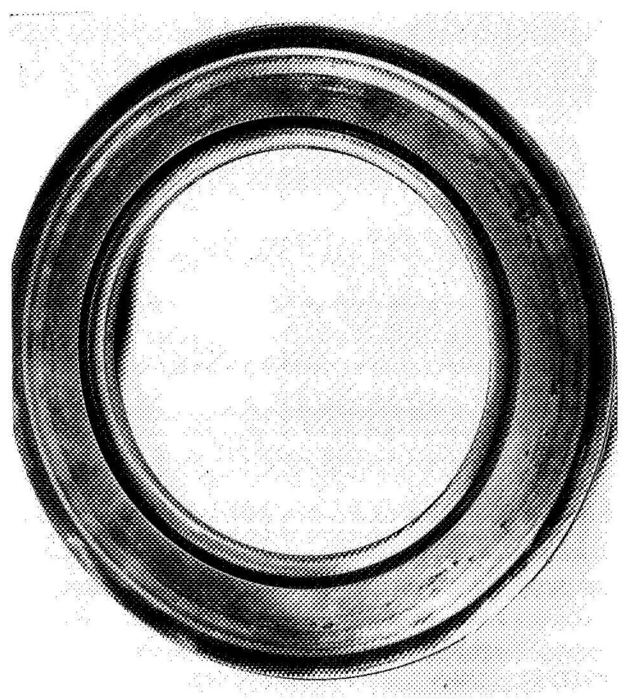
a) Air Seal



b) Air Seal Runner



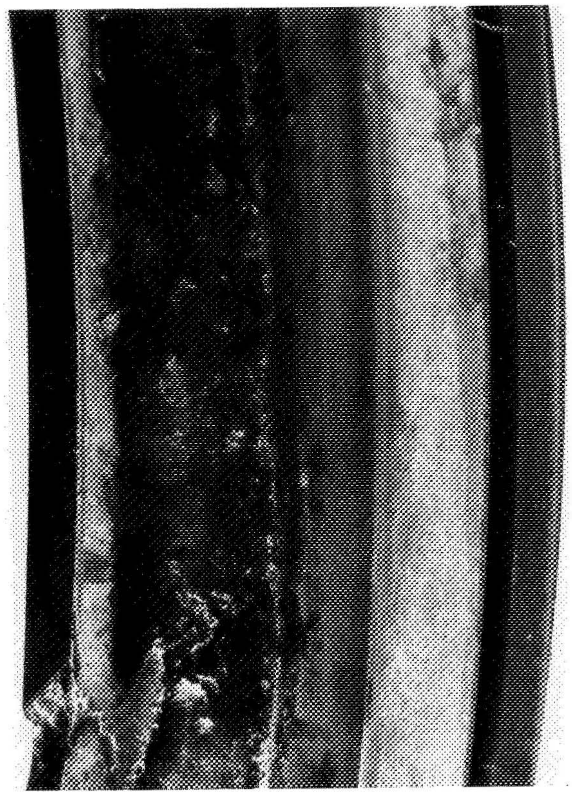
c) Oil Seal



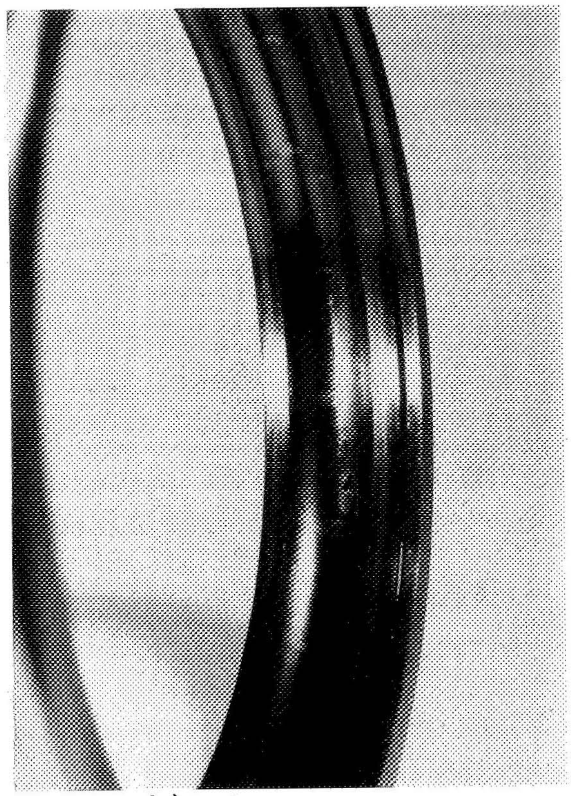
d) Oil Seal Runner

ENCLOSURE 19

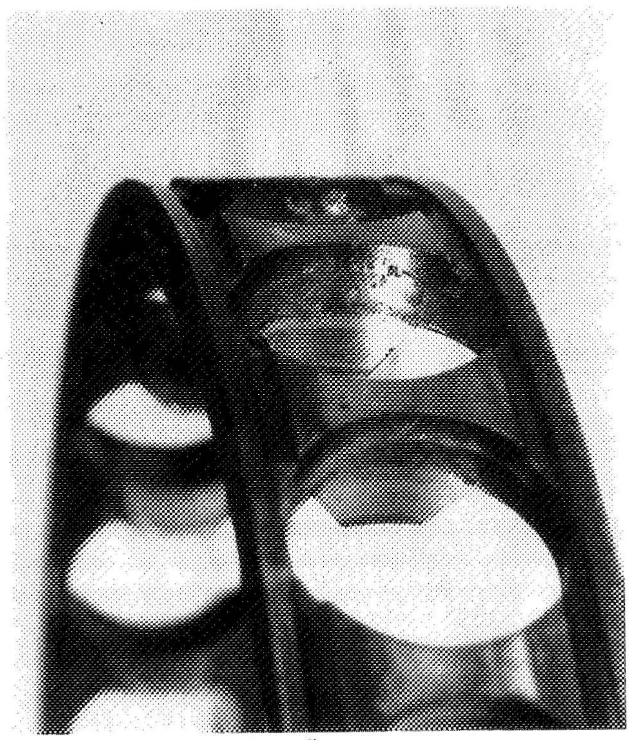
Test Bearing Parts After 600°F Mobil Jet Oil II Screening Run for 1.4 Hours



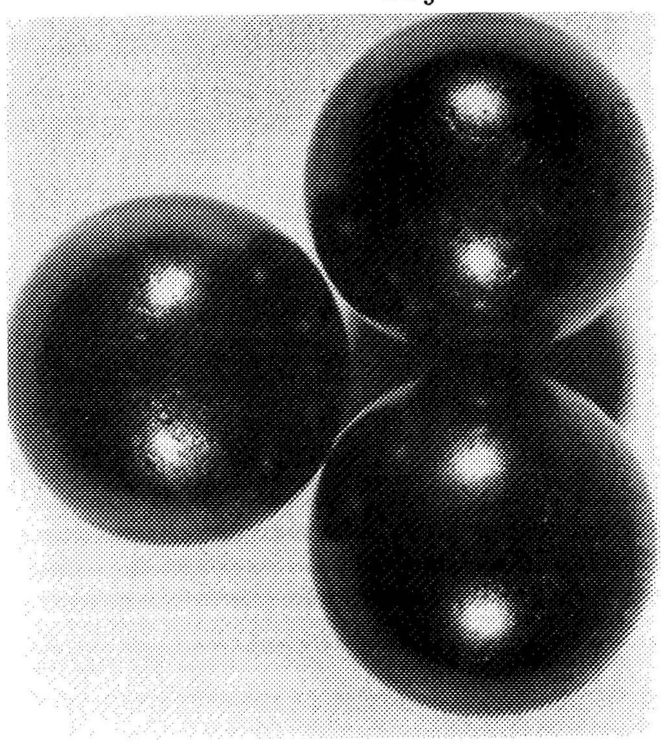
a) Inner Ring



b) Outer Ring



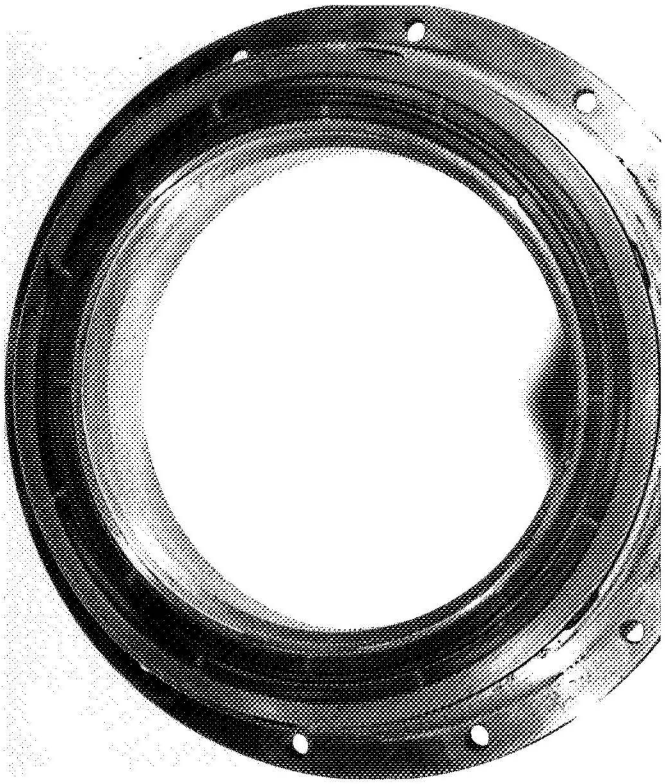
c) Cage



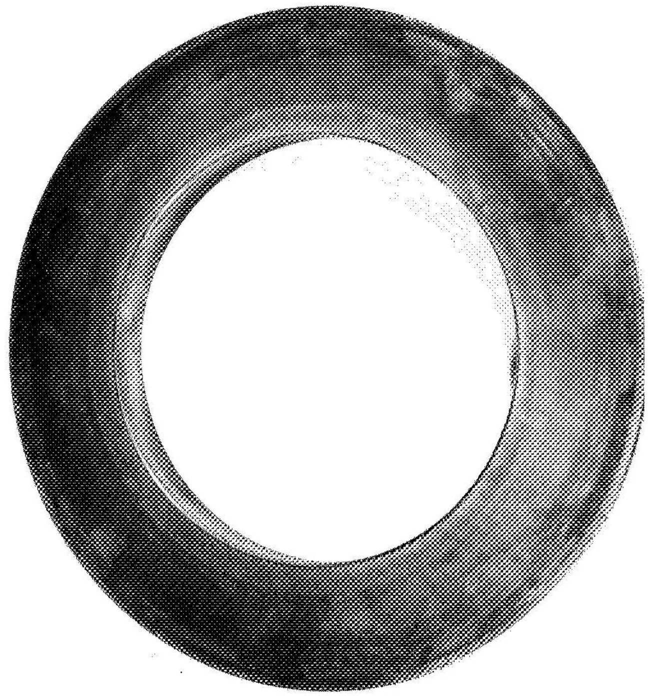
d) Balls

ENCLOSURE 20

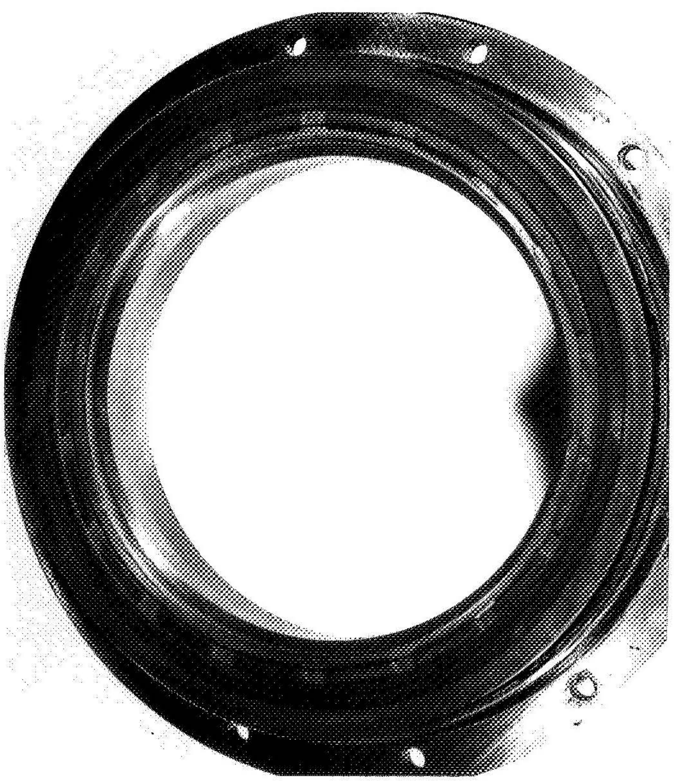
<u>Test Seal Parts After 600°F Mobil Jet Oil II Screening Run for 1.4 Hours</u>



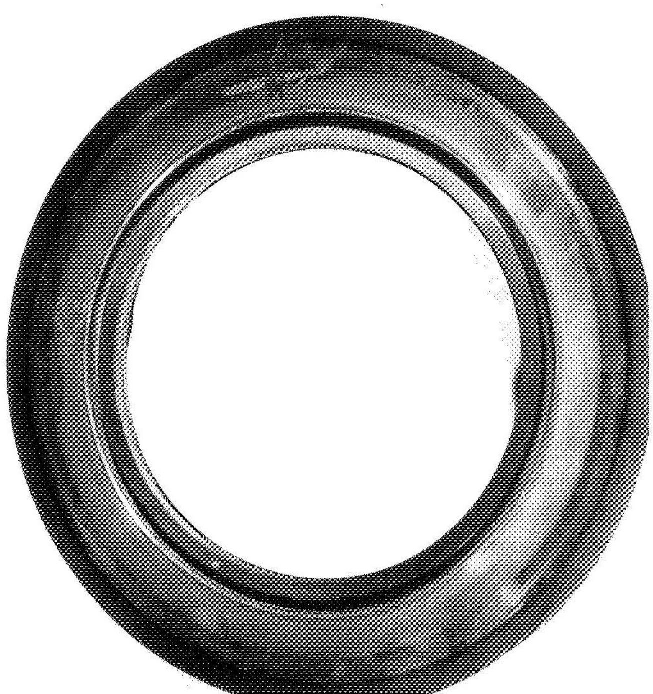
a) Air Seal



b) Air Seal Runner



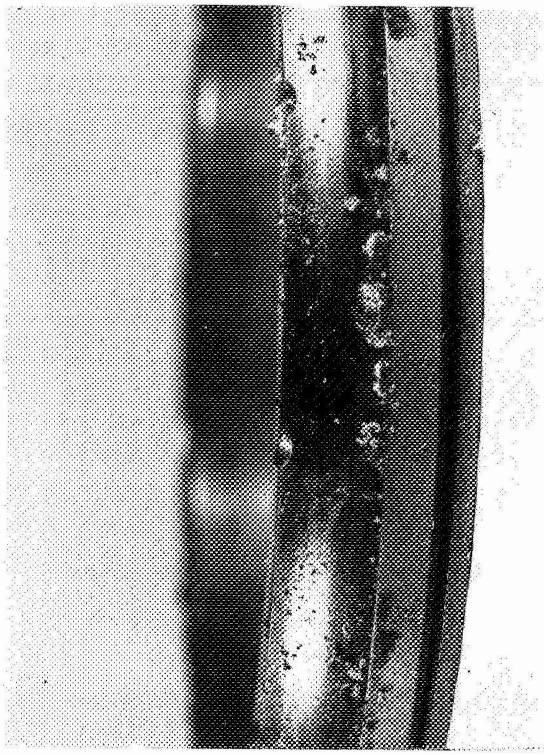
c) Oil Seal



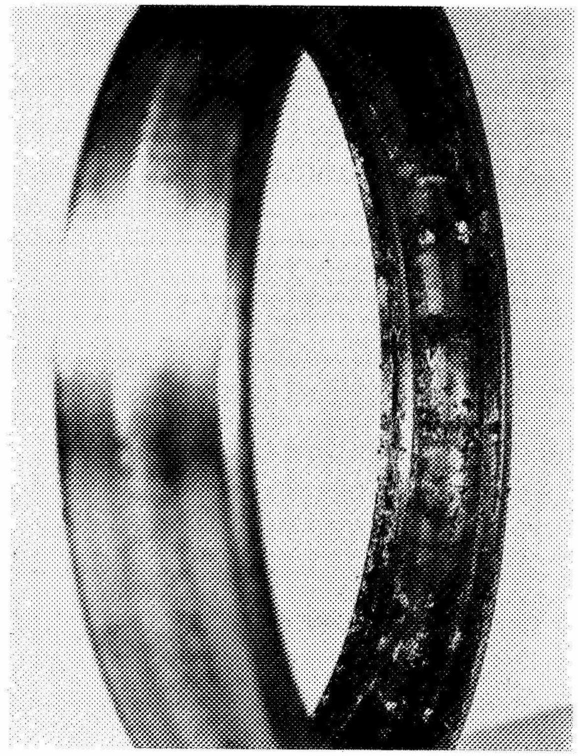
d) Oil Seal Runner

ENCLOSURE 21

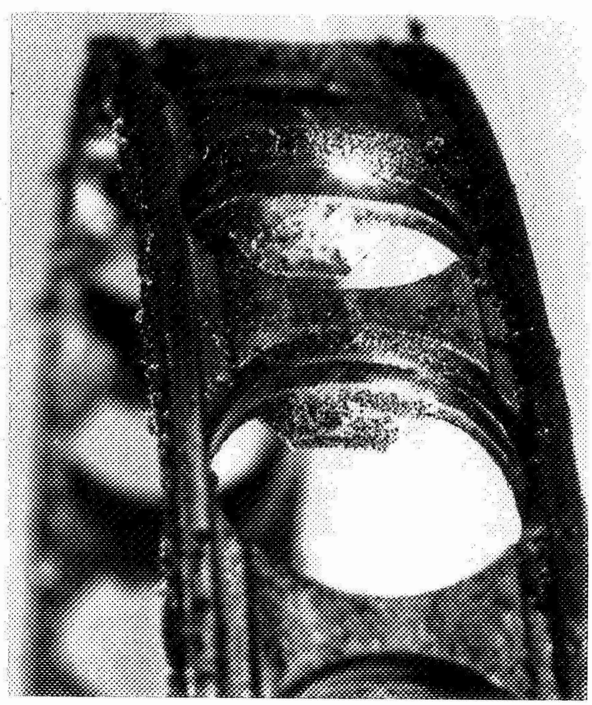
Test Bearing Parts After 500°F Mobil XRM 154 D
Screening Run for 2.3 Hours
Note: Heavy Oil Decomposition Products on Parts



a) Inner Race



b) Outer Race



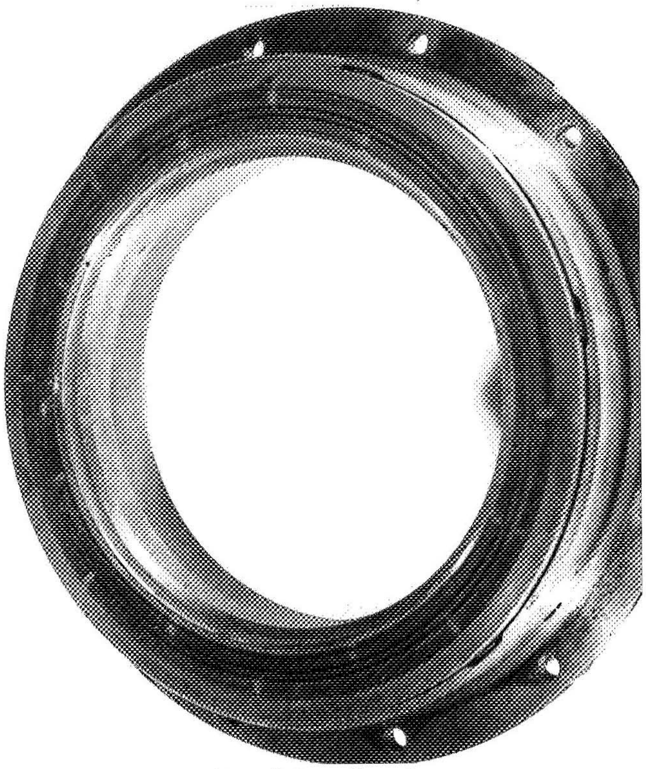
c) Cage

d) Balls

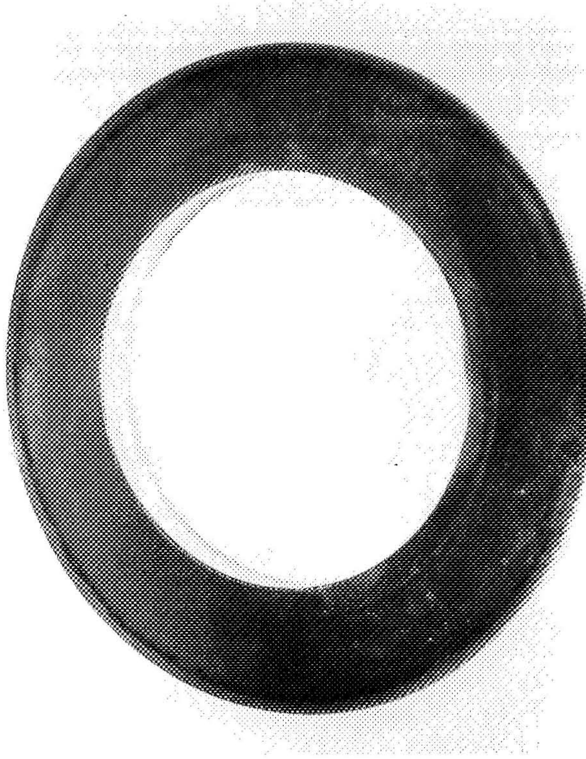
ENCLOSURE 22

Test Seal Parts After 500°F

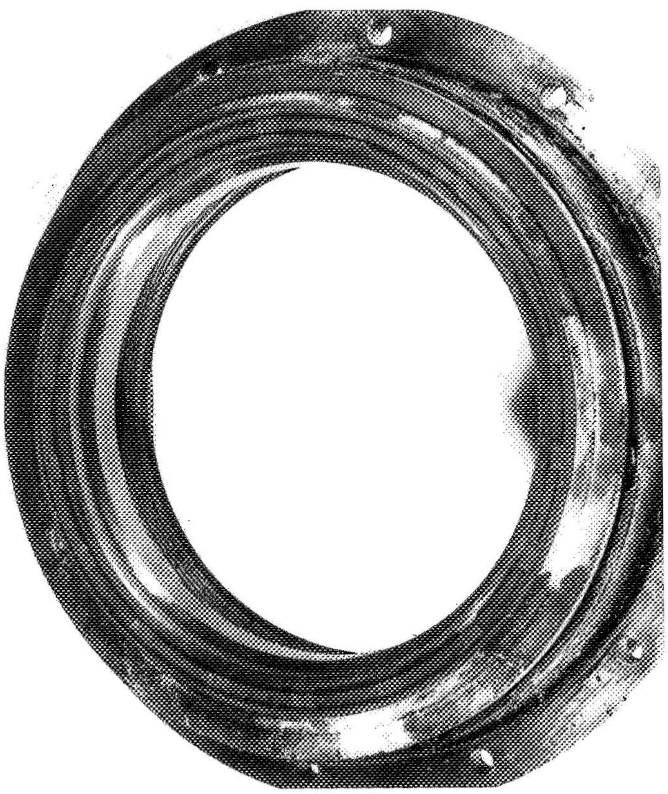
Mobil XRM 154D Oil Screening Run For 2.3 Hours



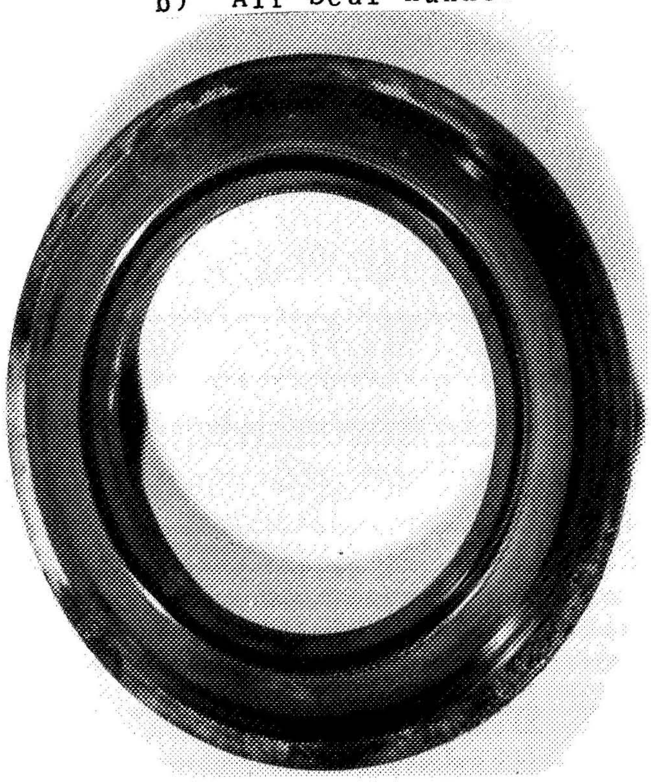
a) Air Seal



b) Air Seal Runner



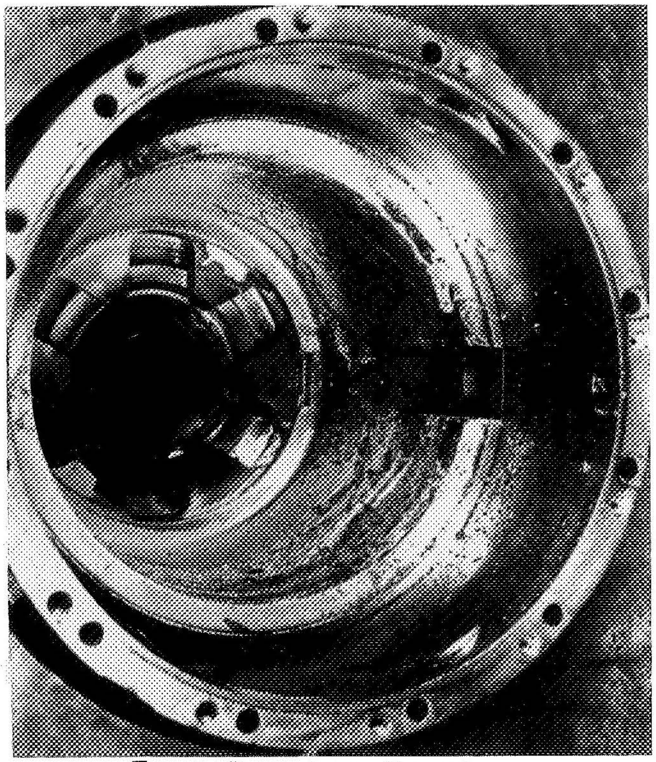
c) Oil Seal



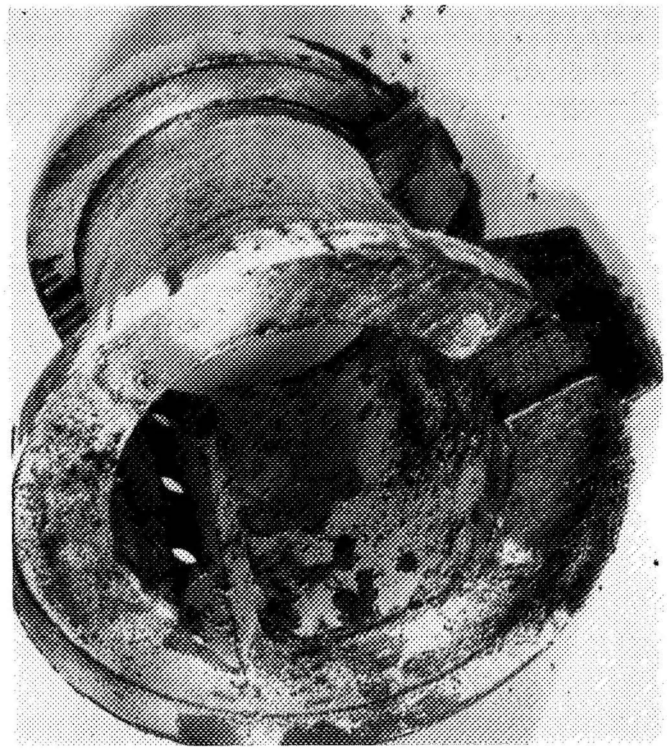
d) Oil Seal Runner

ENCLOSURE 23

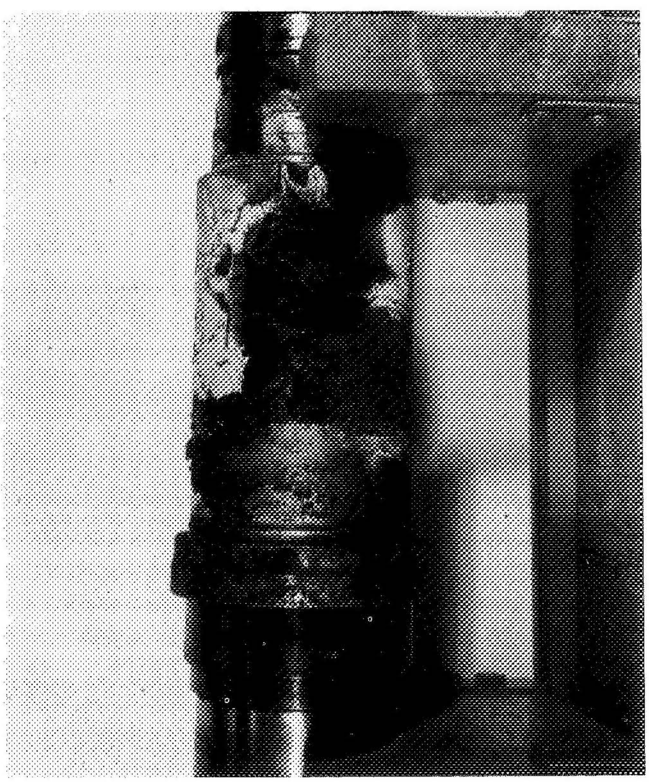
TEST RIG PARTS AFTER 500°F MOBIL XRM 154D OIL SCREENING RUN FOR 2.3 HRS.
NOTE: Heavy Oil Decomposition Products on All Parts



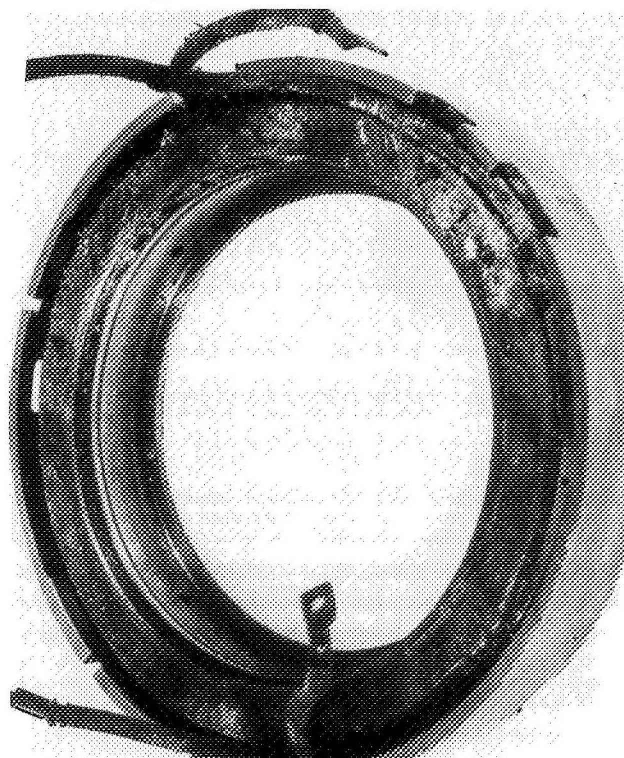
Test Bearing Housing



b) Heat Shield



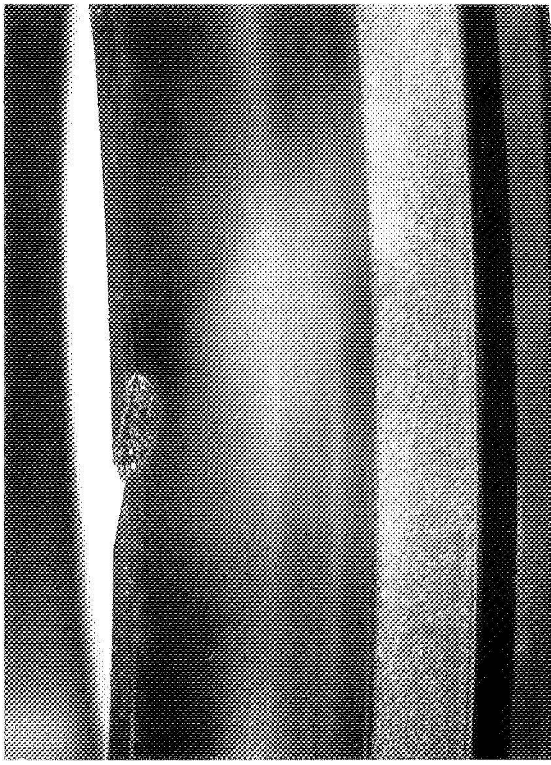
c) Shaft



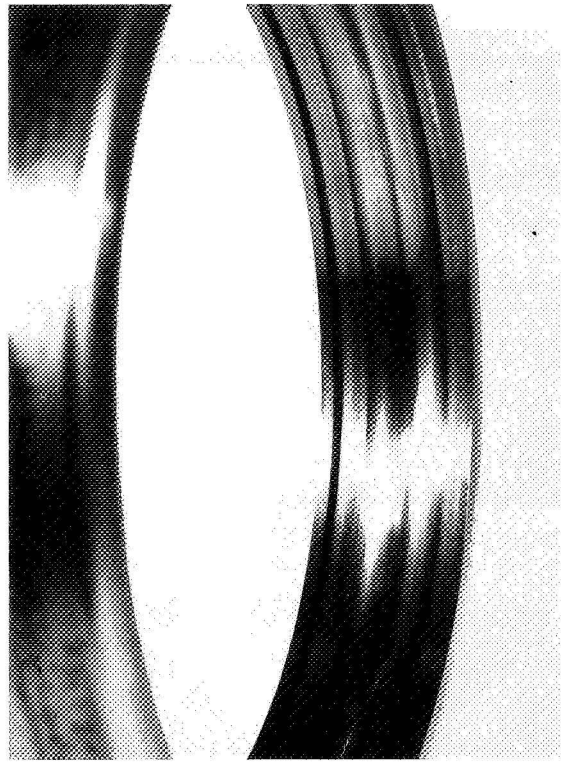
d) Housing Heater

ENCLOSURE 24

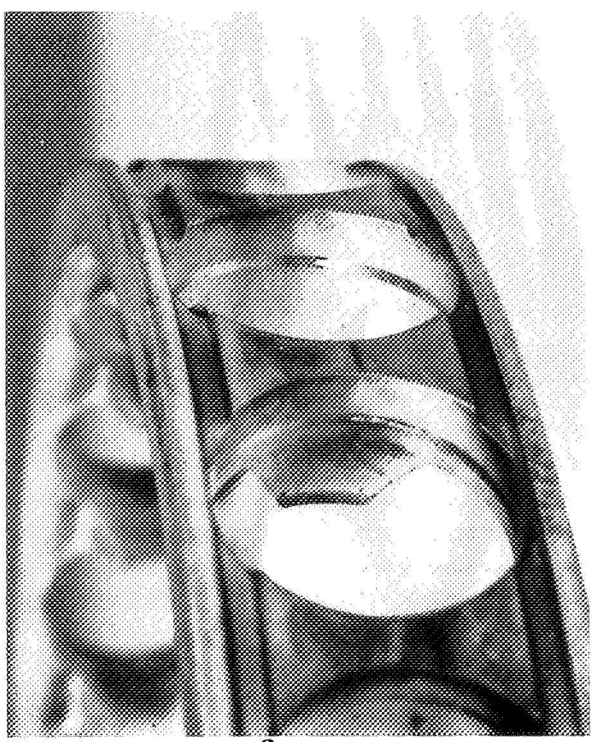
Test Bearing Parts After 650°F
Mobil XRM 154D Oil Screening Run for 3.0 Hours



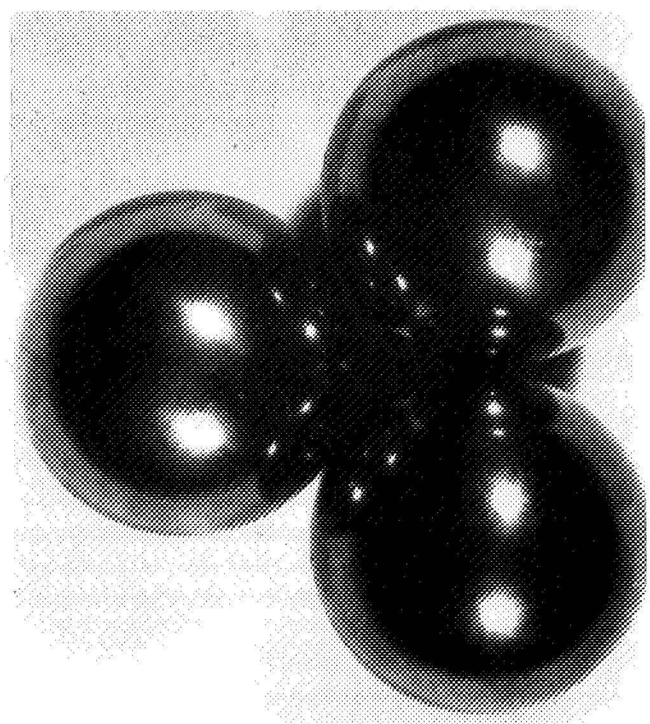
a) Inner Race



b) Outer Race



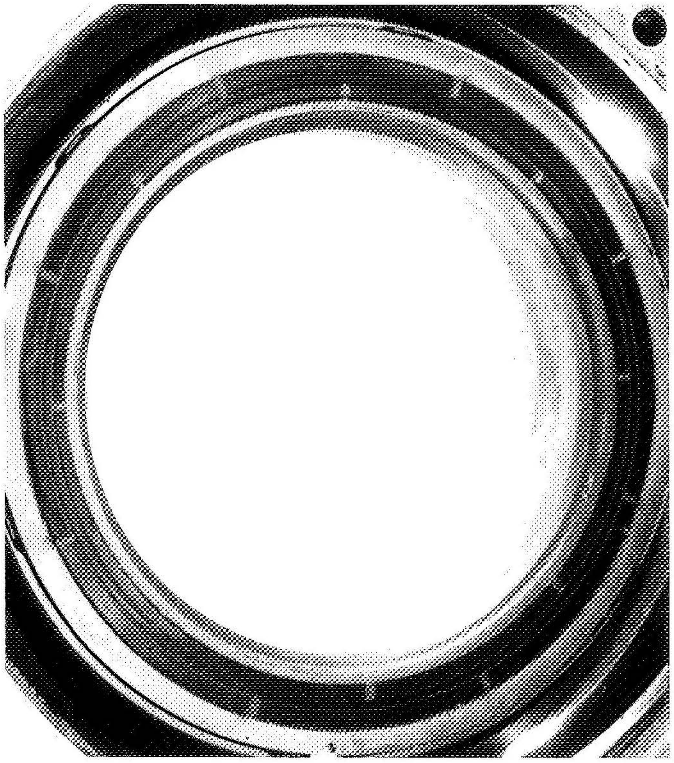
c) Cage



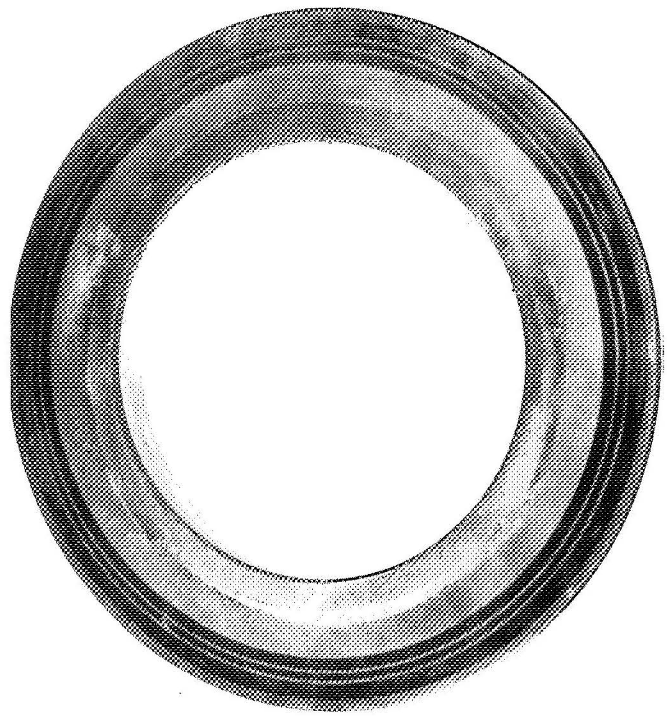
d) Balls

ENCLOSURE 25

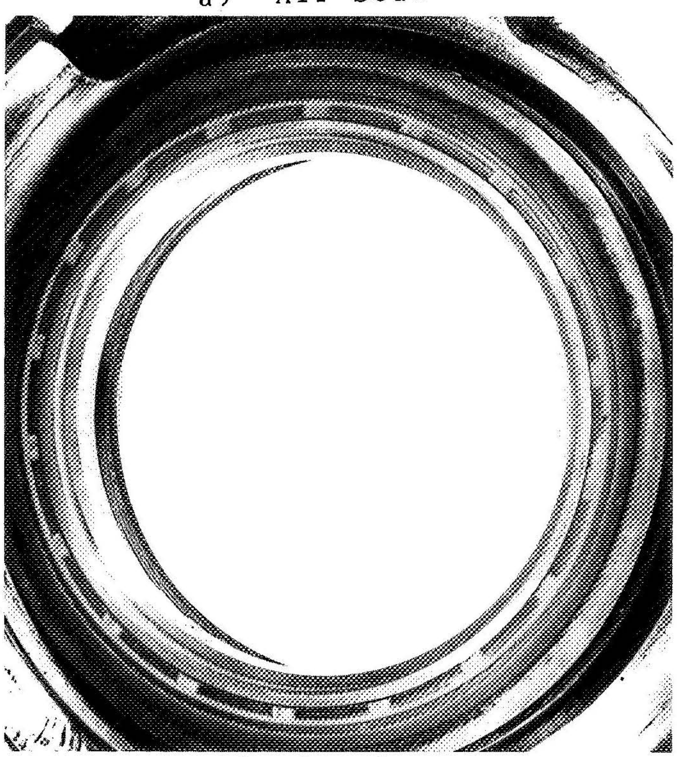
Test Seal Parts After 650°F
Mobil XRM 154D Oil Screening Run For 3.0 Hours



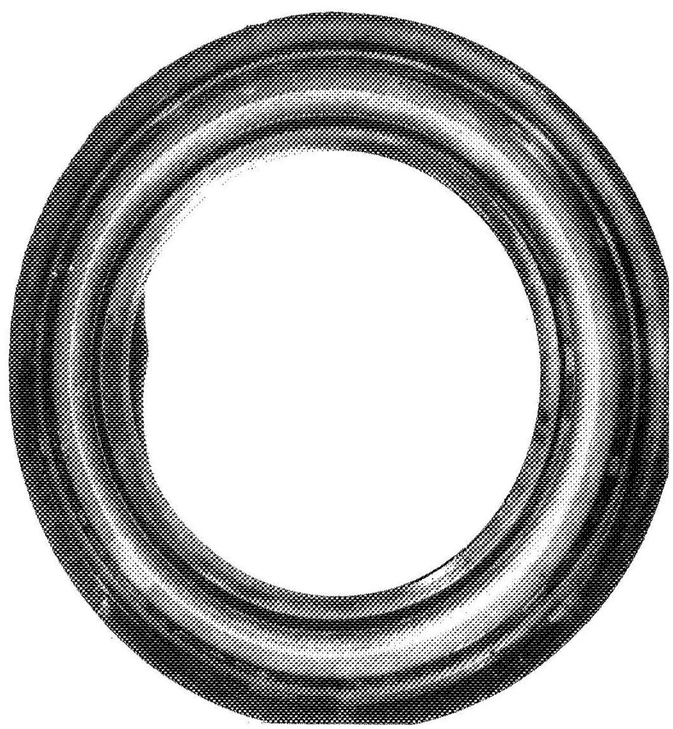
a) Air Seal



b) Air Seal Runner



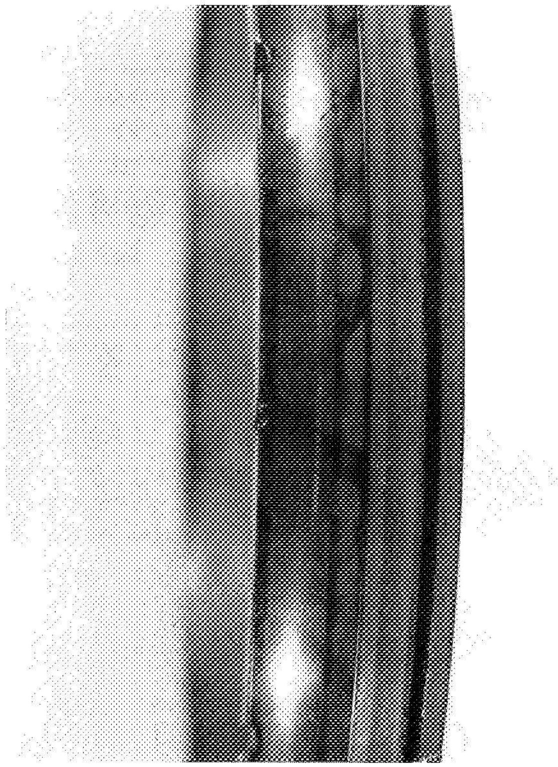
c) Oil Seal



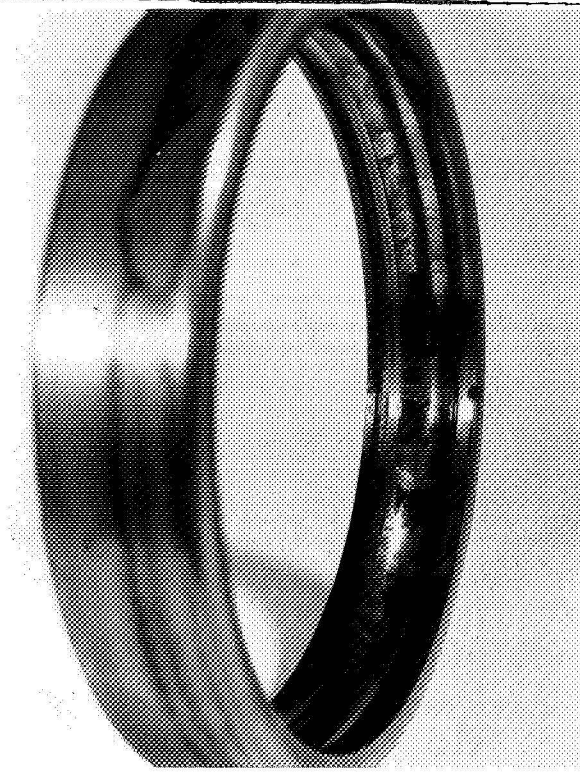
d) Oil Seal Runner

ENCLOSURE 26

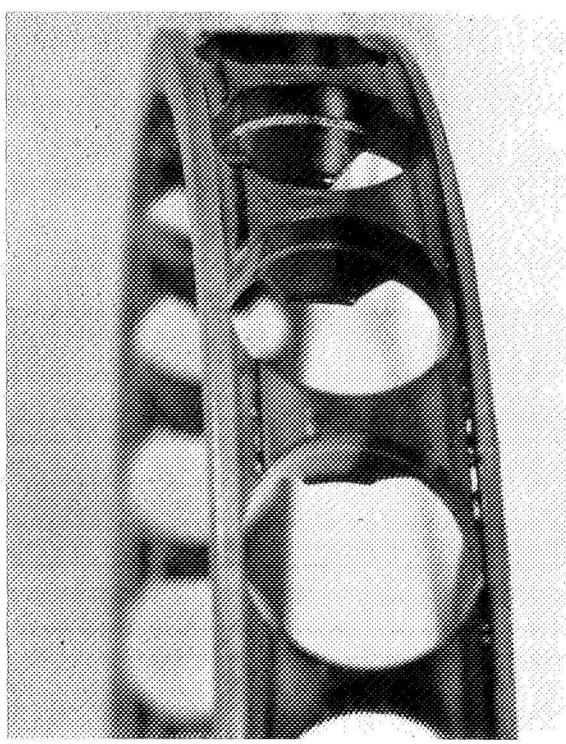
Test Bearing Parts After 700°F Mobil XRM 109F and 10% by Weight of Kendall Heavy Resin 0839 Oil Screening Run for 6.0 Hours



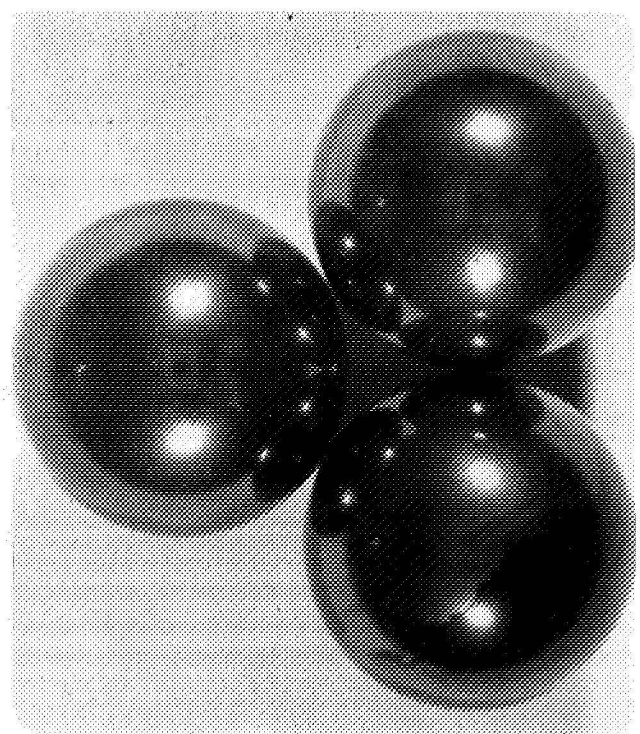
a) Inner Race



b) Outer Race



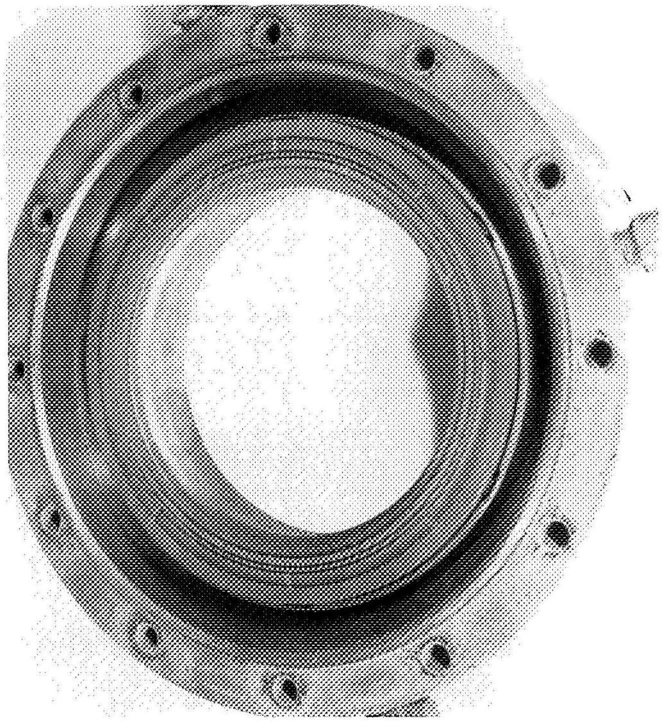
c) Cage



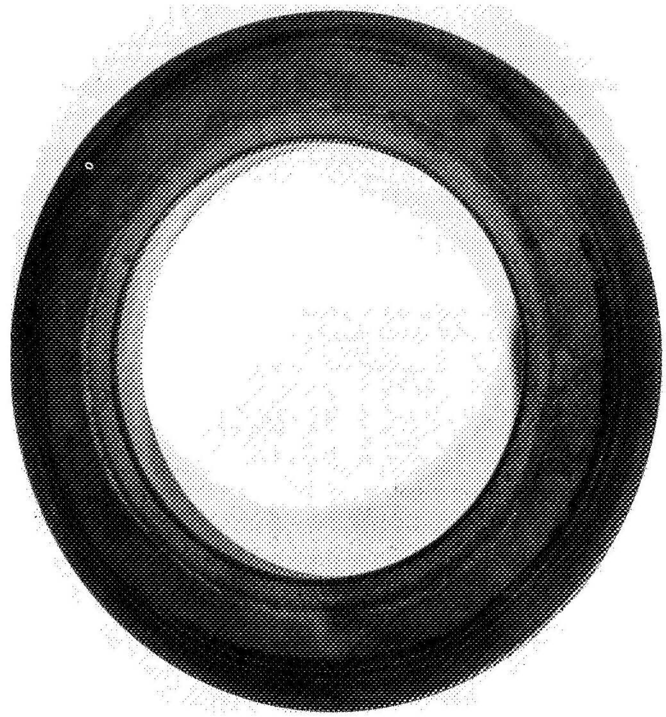
d) Balls

ENCLOSURE 27

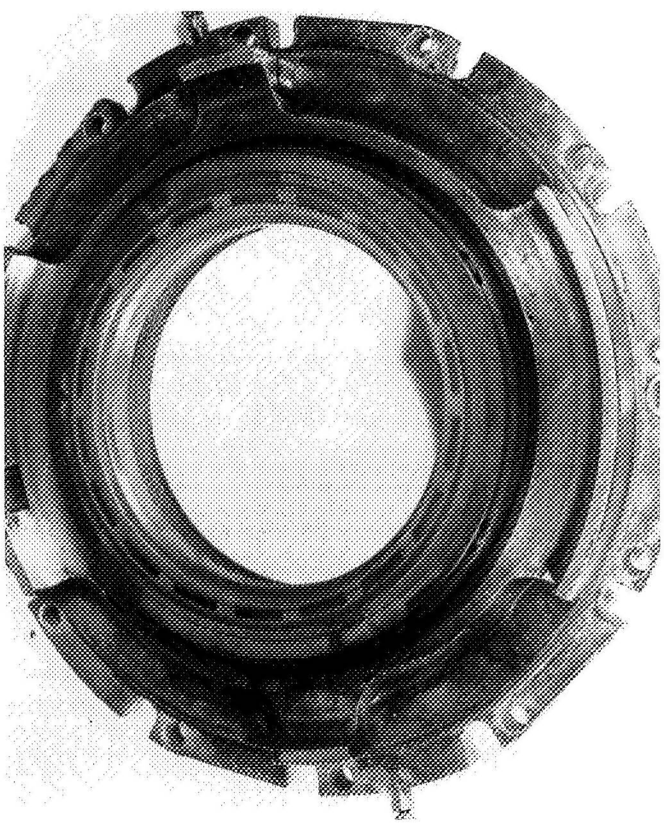
Test Seal Parts After 700°F Mobil XRM 109F and 10% by Weight of Kendall Heavy Resin 0839 Oil Screening Run for 6.0 Hours



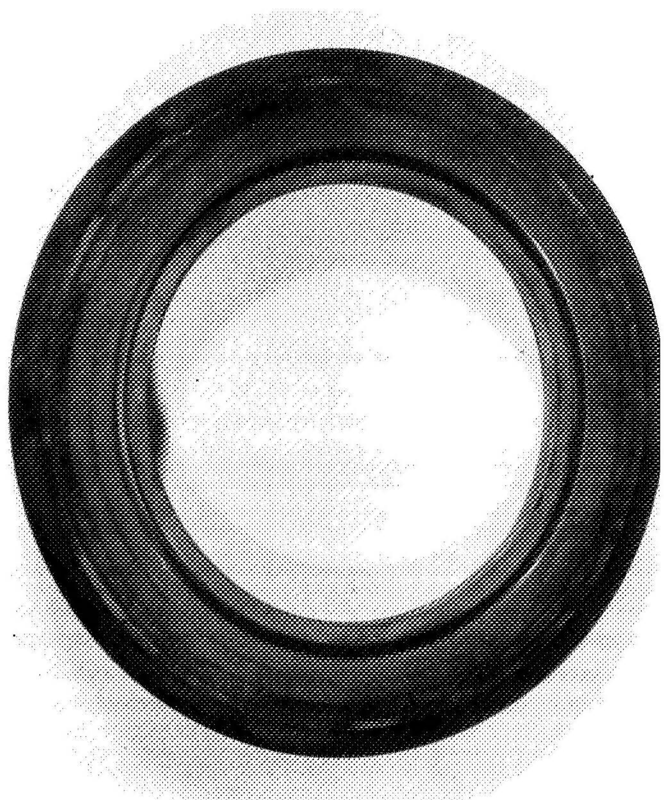
a) Air Seal



b) Air Seal Runner



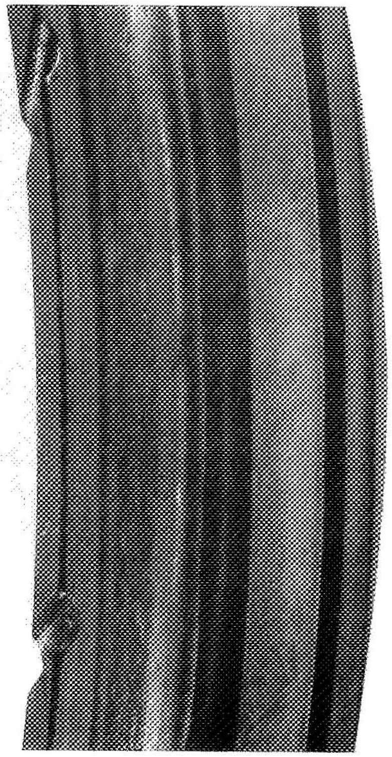
c) Oil Seal



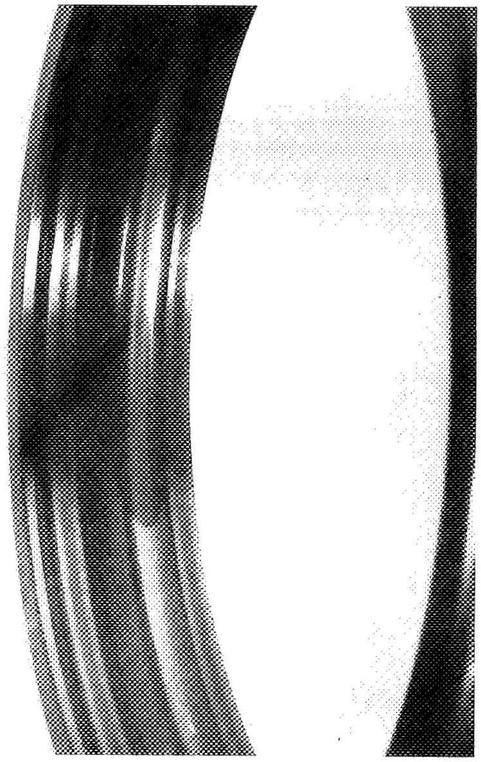
d) Oil Seal Runner

ENCLOSURE 28

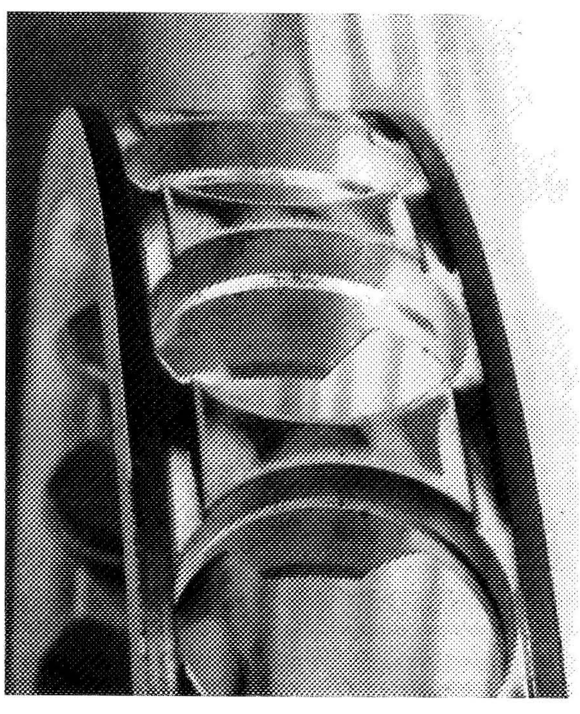
TEST BEARING PARTS AFTER 500°F MOBIL JET OIL II OPEN ATMOSPHERE BASELINE TEST FOR 50 HOURS



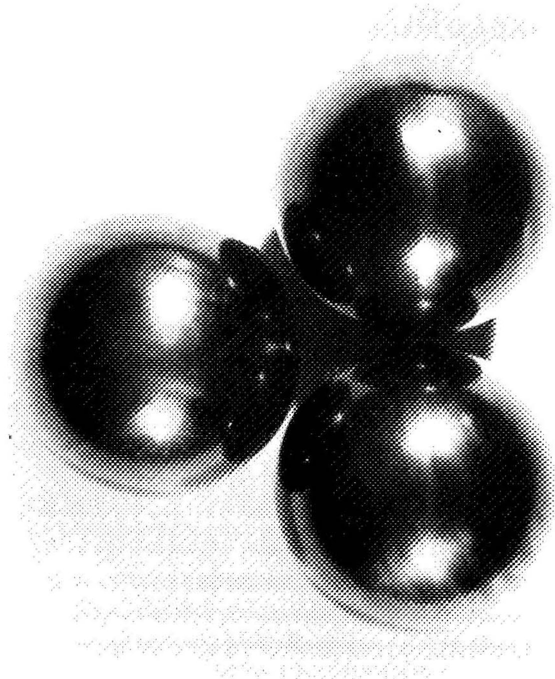
a) Inner Race



b) Outer Race



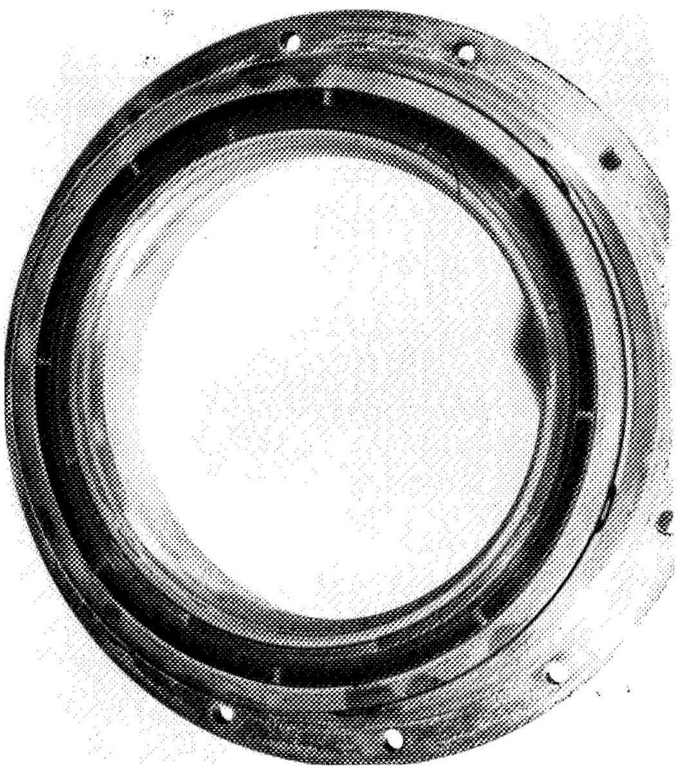
c) Cage



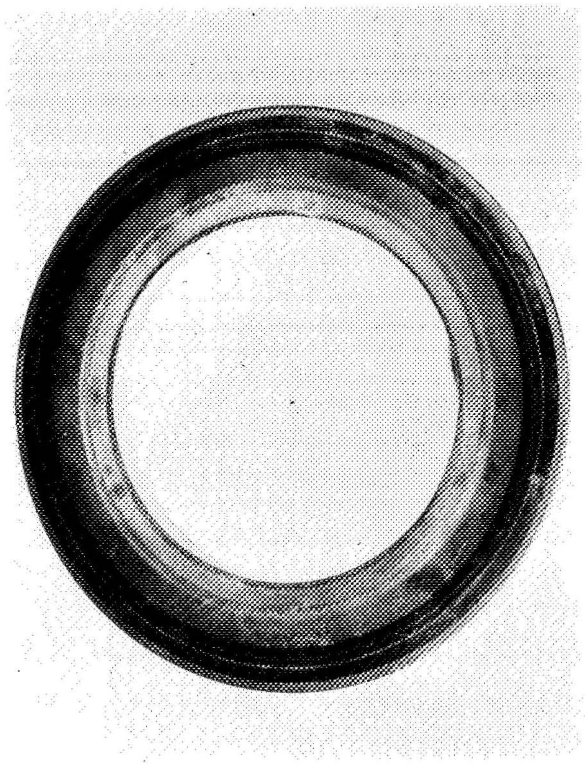
d) Balls

ENCLOSURE 29

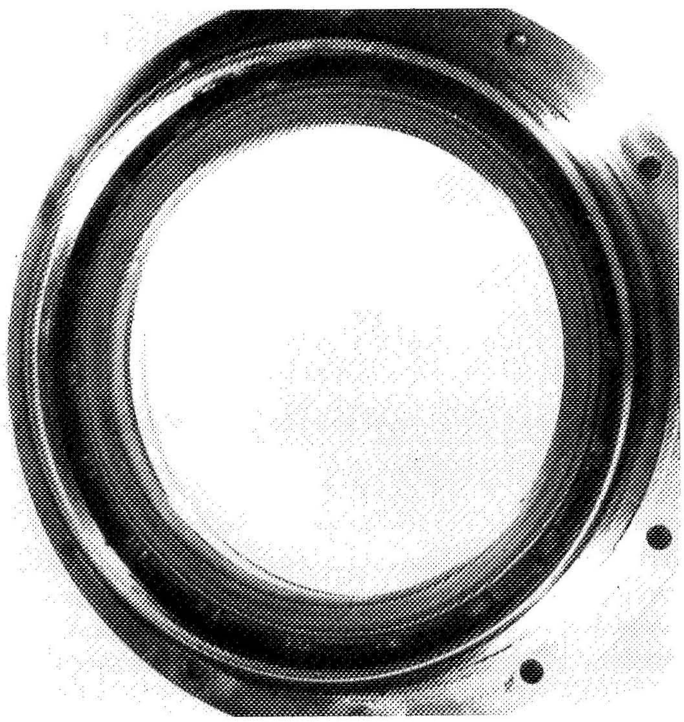
TEST SEAL PARTS AFTER 500°F MOBIL JET OIL II OPEN ATMOSPHERE BASELINE TEST FOR 50 HOURS



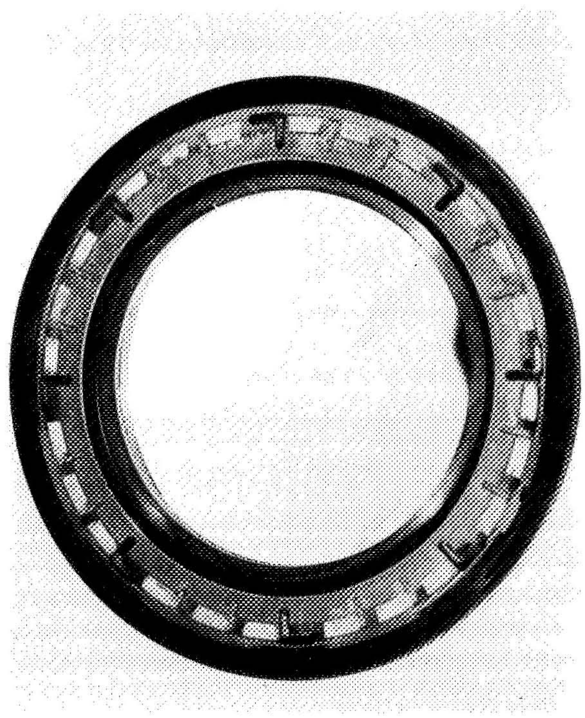
a) Air Seal



b) Air Seal Runner



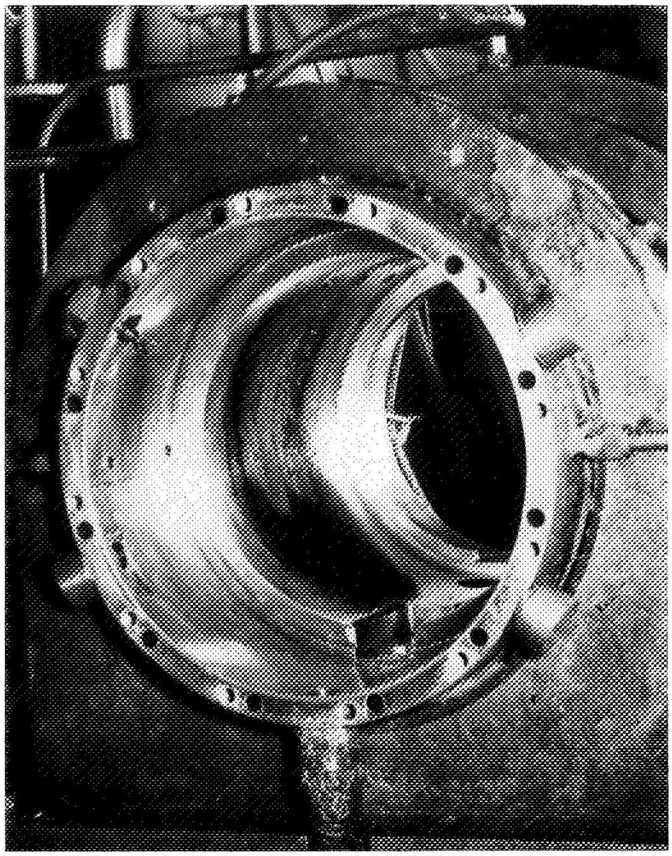
c) Oil Seal

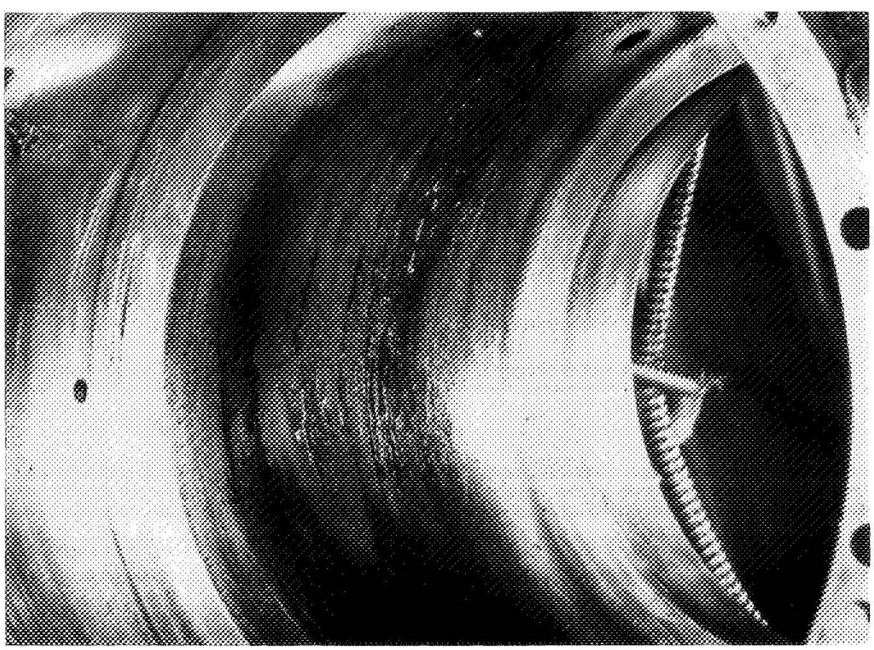


d) Oil Seal Runner

ENCLOSURE 30

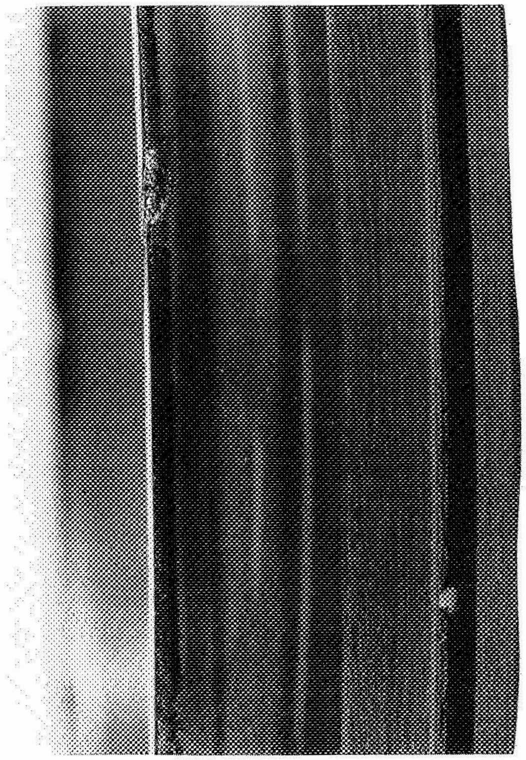
BEARING HOUSING BORE AFTER 500°F MOBIL JET OIL II OPEN ATMOSPHERE BASELINE TEST FOR 50 HOURS



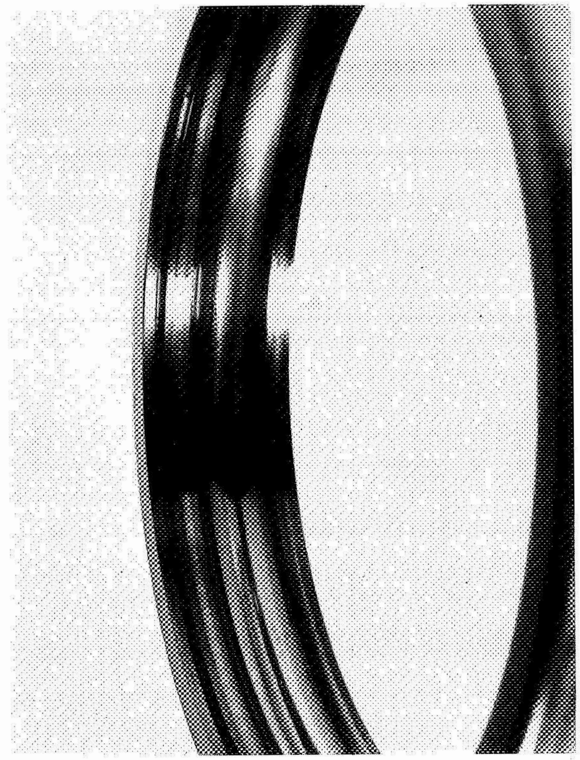


ENCLOSURE 31

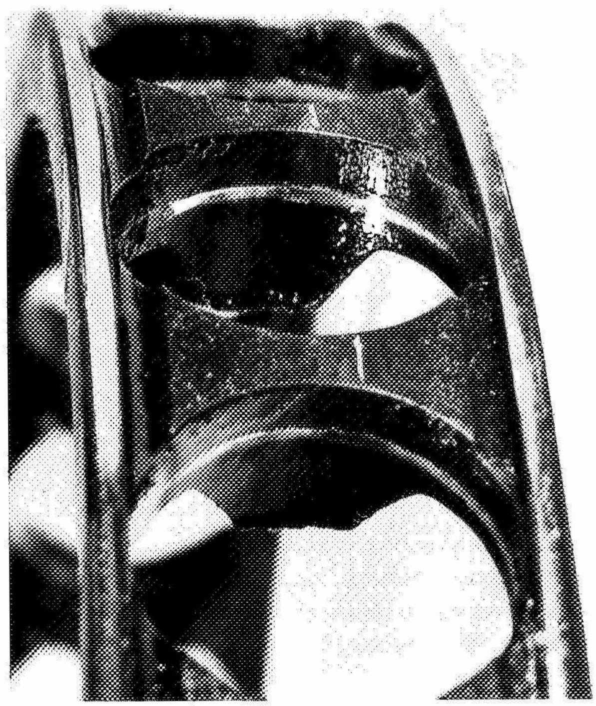
TEST BEARING PARTS AFTER 650°F MOBIL XRM-109F AND 10% BY WEIGHT OF KENDALL HEAVY RESIN 0839 QUALIFICATION TEST FOR 32 HOURS



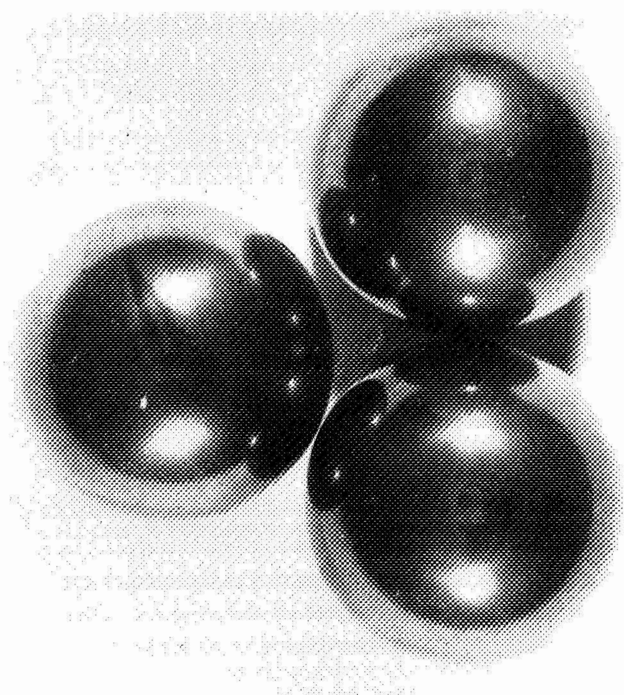
a) Inner Race



b) Outer Race



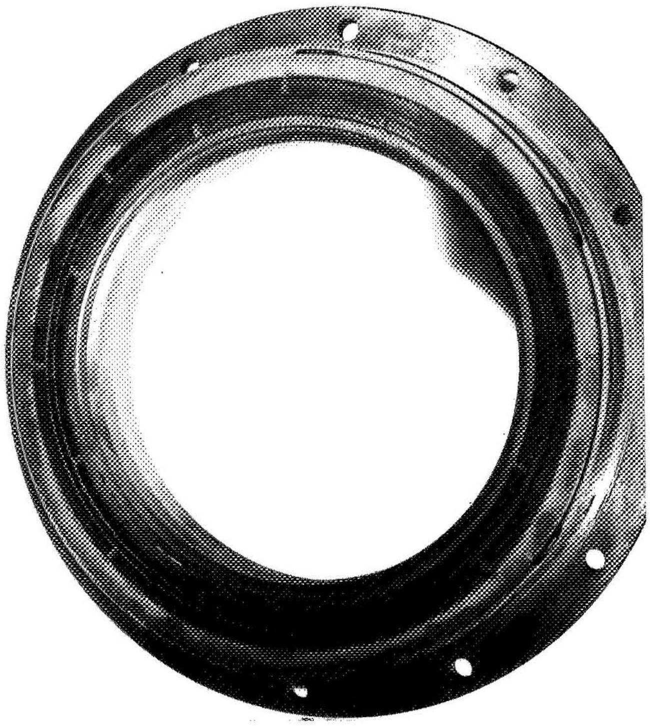
c) Cage



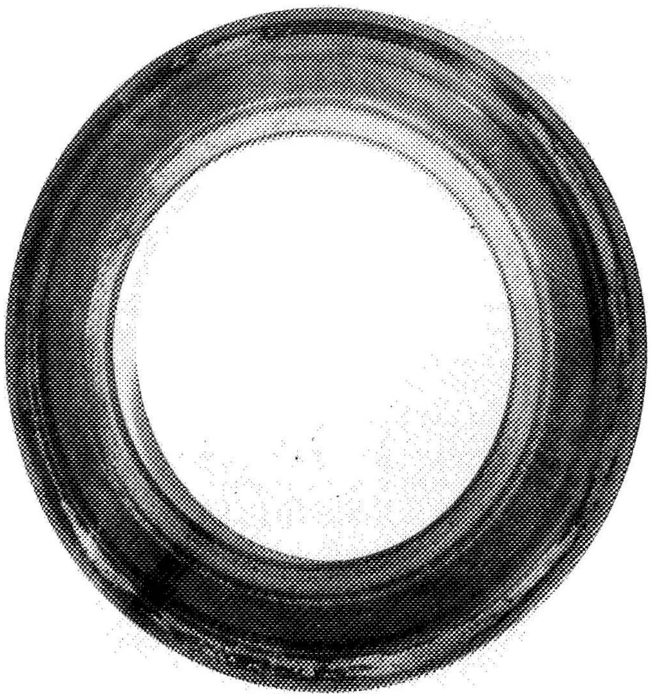
d) Balls

ENCLOSURE 32

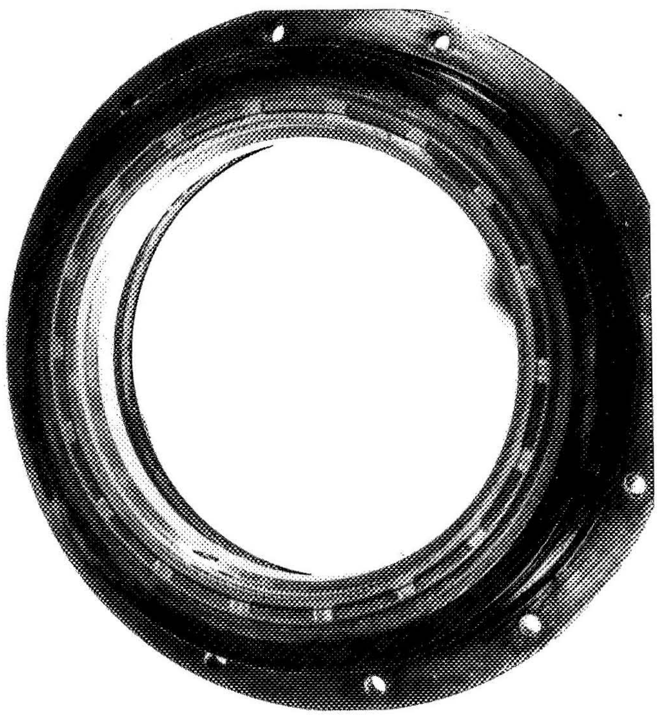
TEST SEALS PARTS AFTER 650°F MOBIL XRM-109F AND 10% BY WEIGHT OF KENDALL HEAVY RESIN 0839 QUALIFICATION TEST FOR 32 HOURS



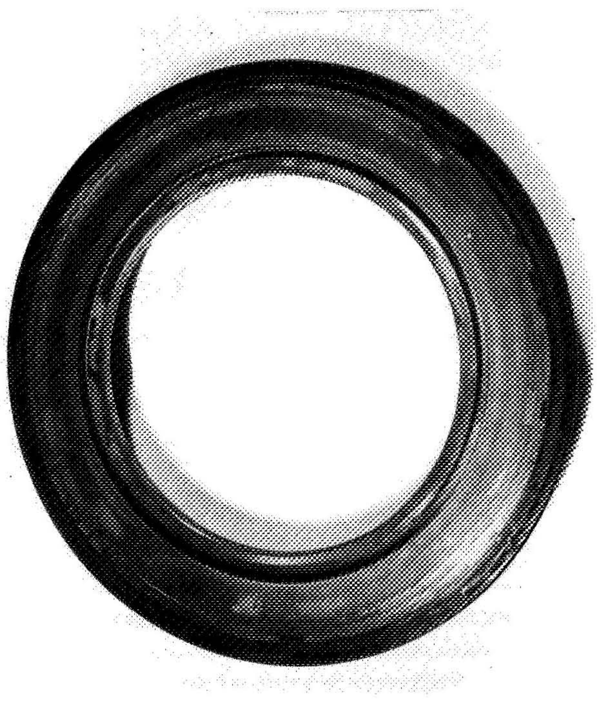
a) Air Seal



b) Air Seal Runner



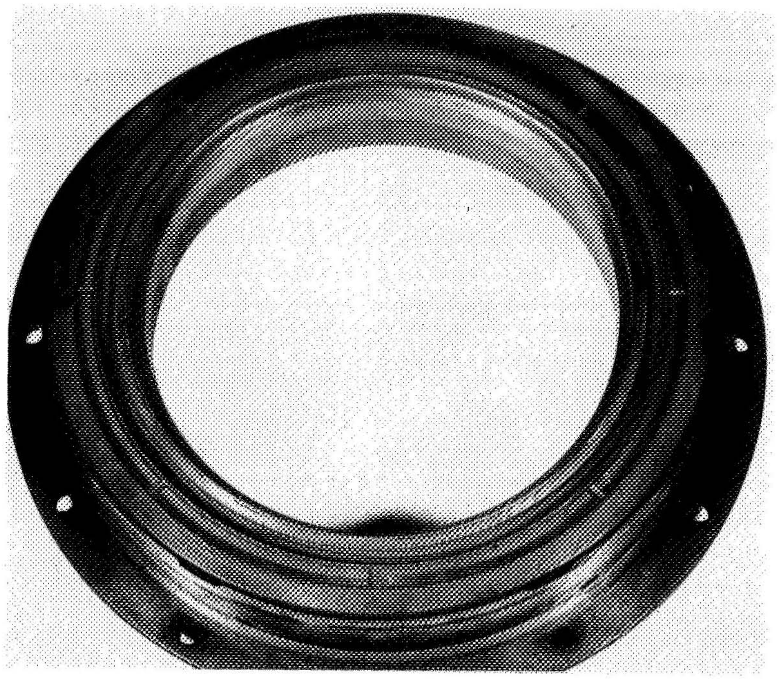
c) Oil Seal

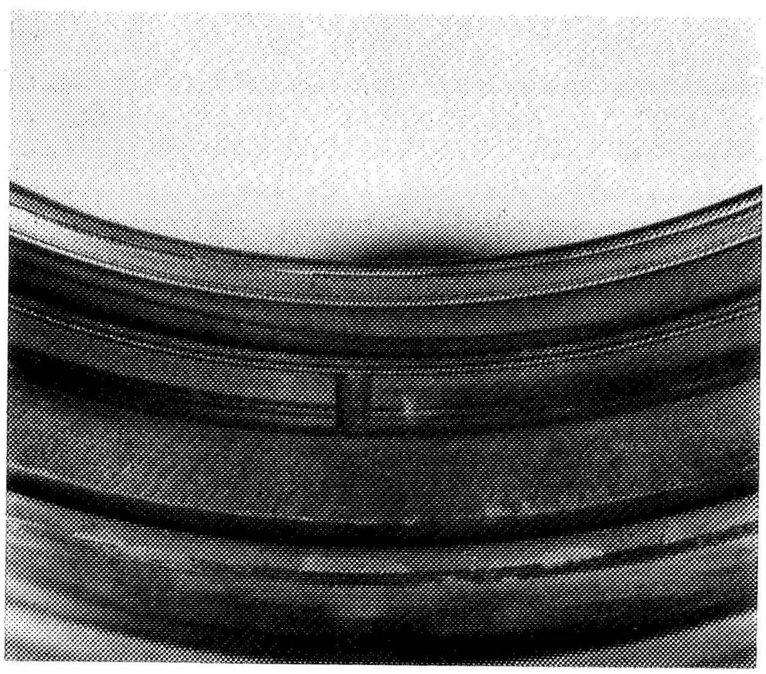


d) Oil Seal Runner

ENCLOSURE 33

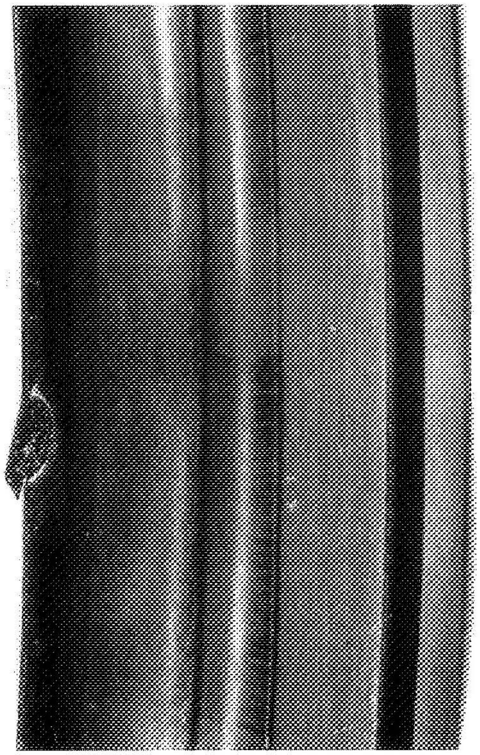
ERODED AIR SEAL AFTER 650°F MOBIL XRM-109F AND 10% BY WEIGHT OF KENDALL HEAVY RESIN 0839 QUALIFICATION TEST FOR 32 HOURS



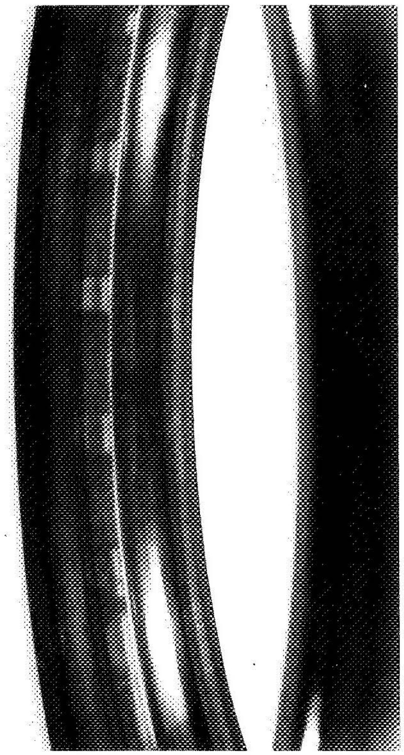


ENCLOSURE 34

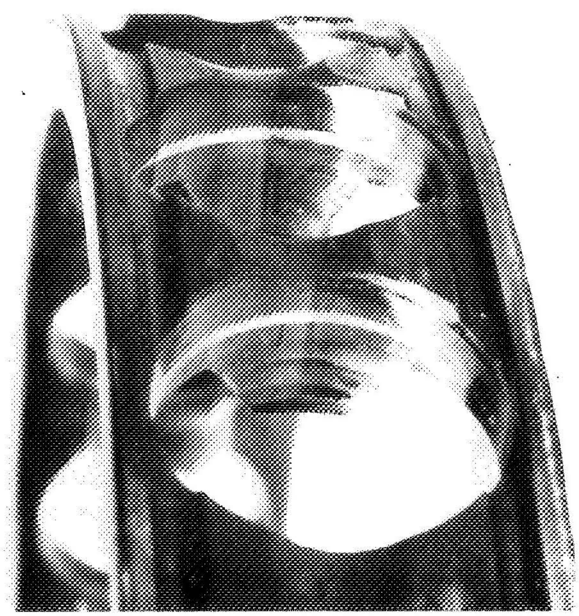
TEST BEARING PARTS AFTER 650°F MOBIL XRM-109F, MOBIL XRM-127B AND BY WEIGHT KENDALL HEAVY RESIN 0839 QUALIFICATION TEST FOR 50 HOURS



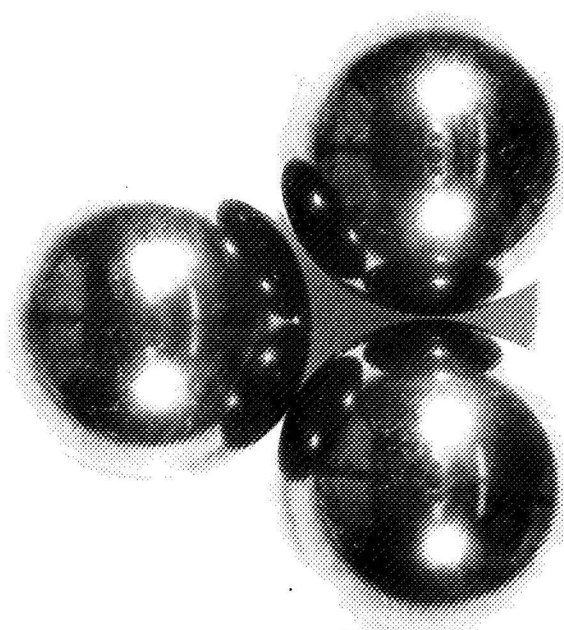
a) Inner Race



b) Outer Race



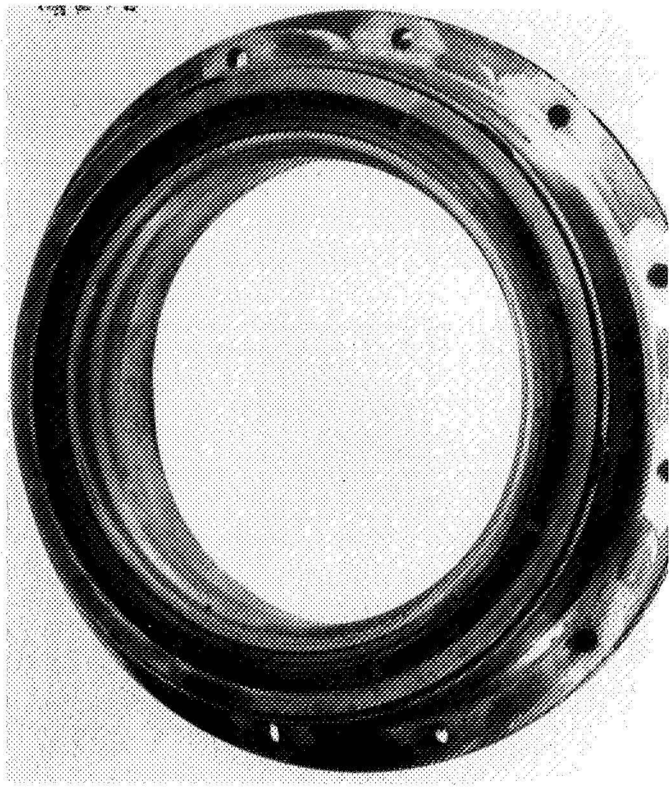
c) Cage



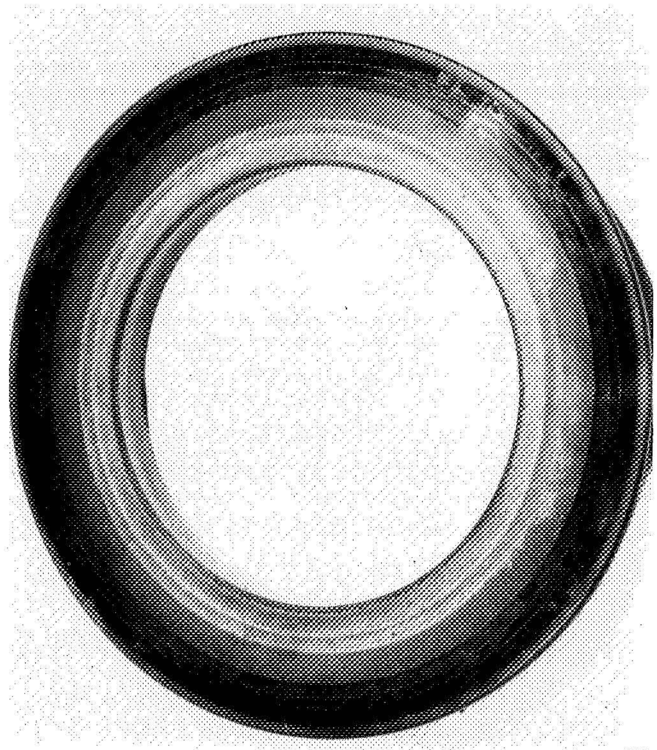
d) Balls

ENCLOSURE 35

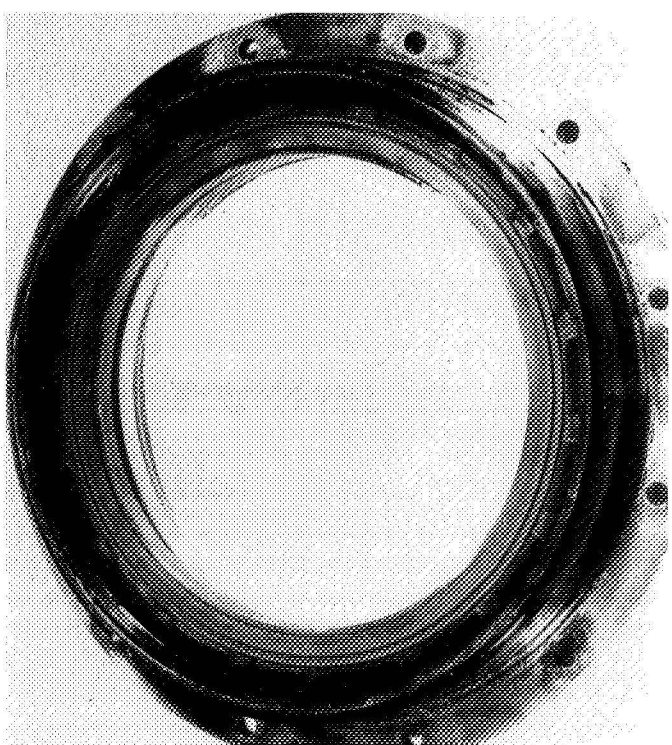
TEST SEAL PARTS AFTER 650°F MOBIL XRM-109F, MOBIL XRM-127B AND 10% BY WEIGHT KENDALL HEAVY RESIN 0839 QUALIFICATION TEST FOR 50 HOURS



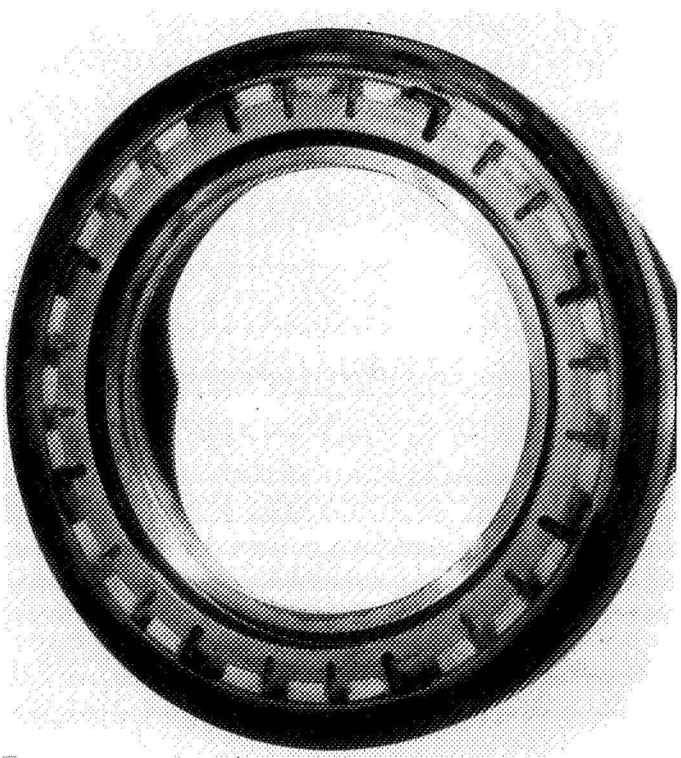
a) Air Seal



b) Air Seal Runner



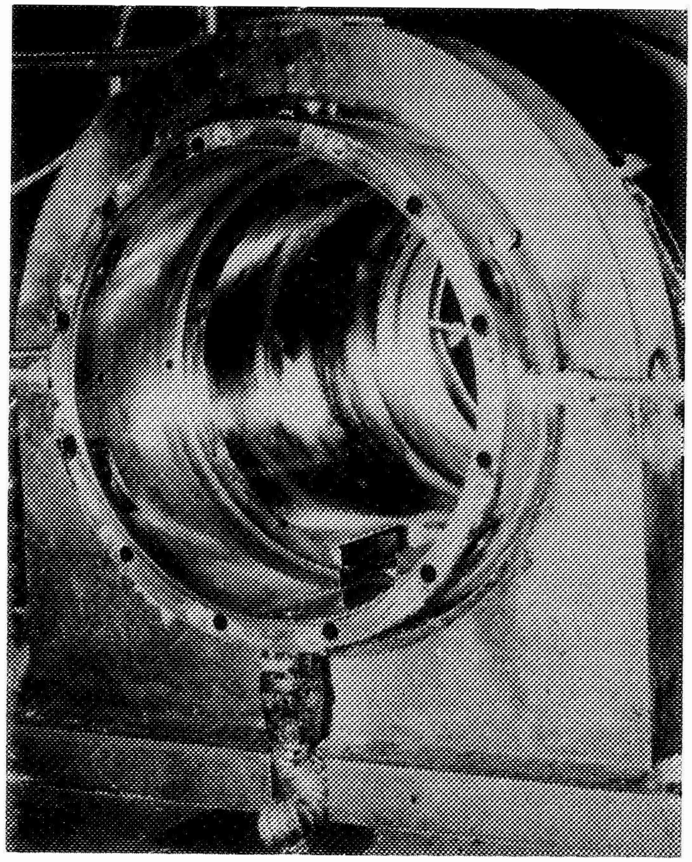
c) Oil Seal



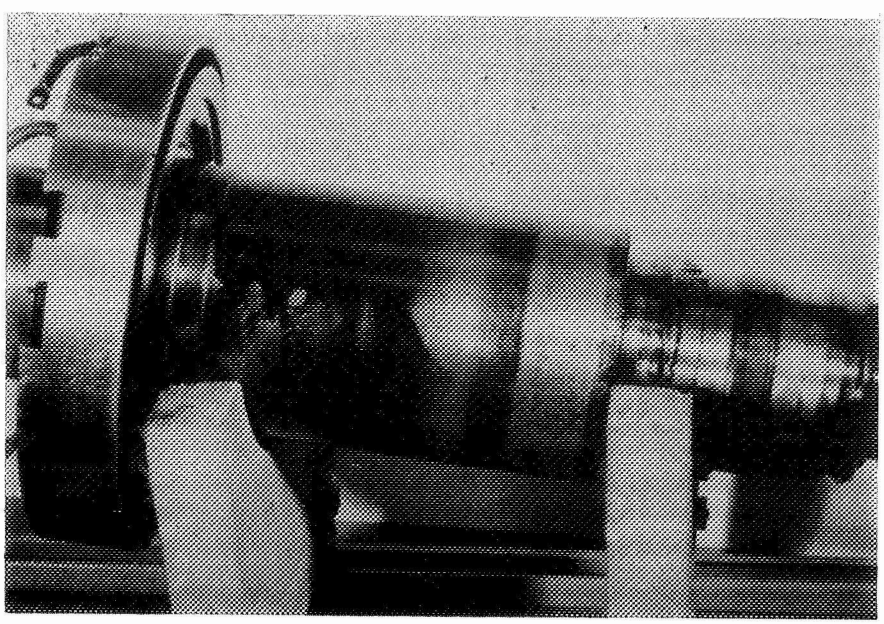
d) Oil Seal Runner

ENCLOSURE 36

TEST RIG PARTS AFTER 650°F MOBIL XRM-109F, MOBIL XRM-127B AND 10% BY WEIGHT KENDALL HEAVY RESIN 0839 QUALIFICATION TEST FOR 50 HOURS



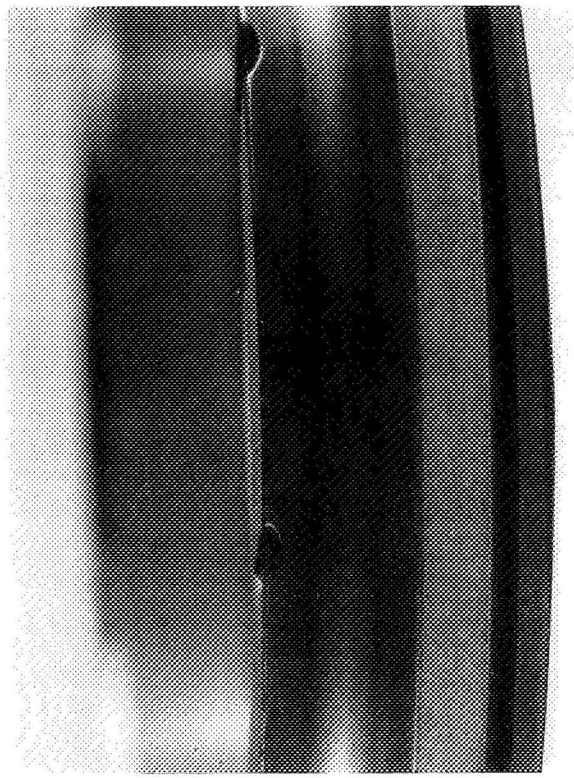
TEST BEARING HOUSING



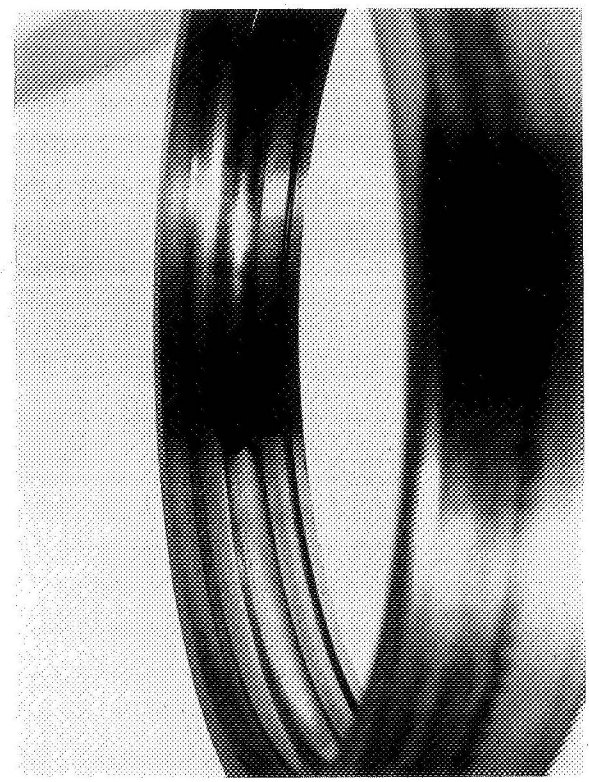
TEST SHAFT

ENCLOSURE 37

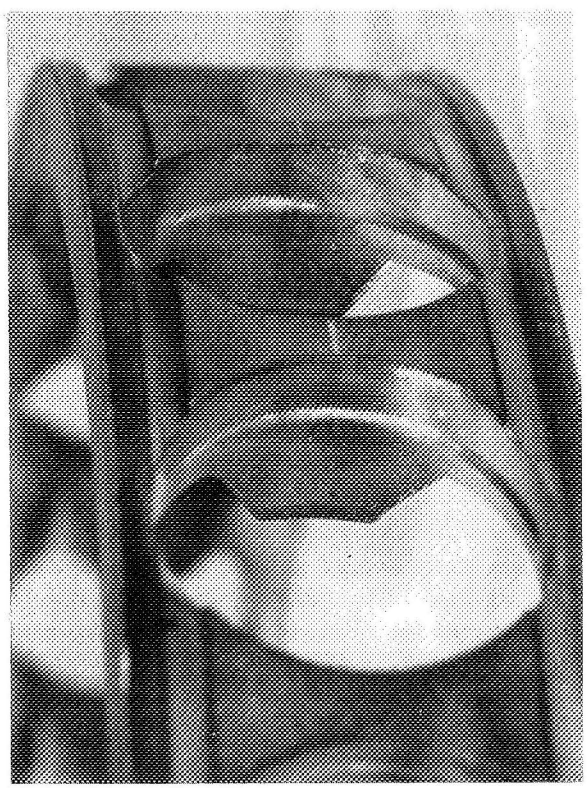
TEST BEARING PARTS AFTER FIRST 20 HOUR PORTION OF THE FIRST 250 HOUR ENDURANCE RUN (MOBIL XRM 177F OIL AT 600°F)



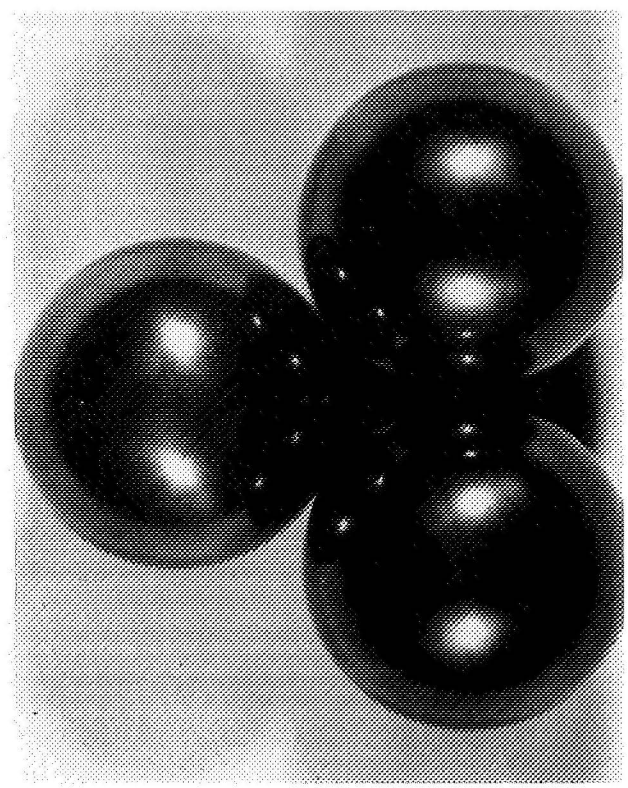
a) Inner Race



b) Outer Race



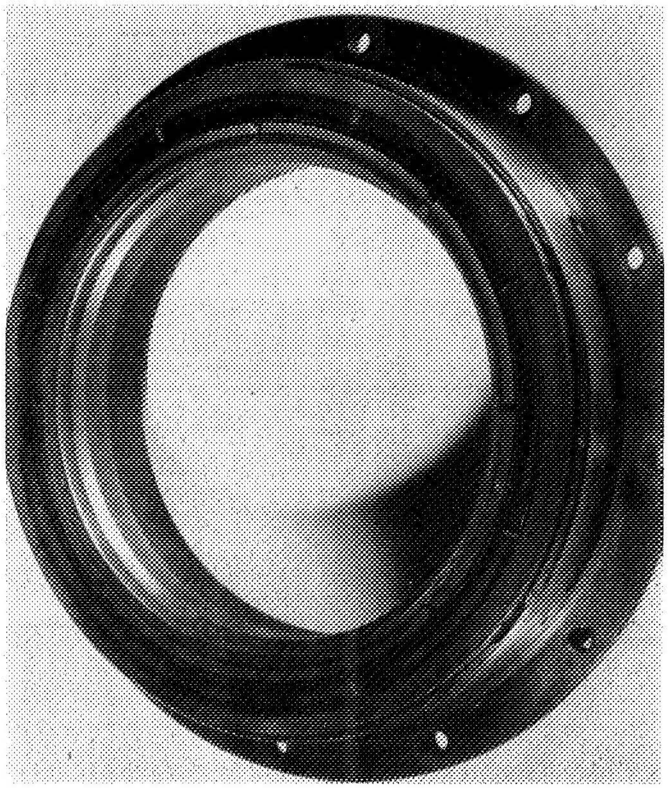
c) Cage



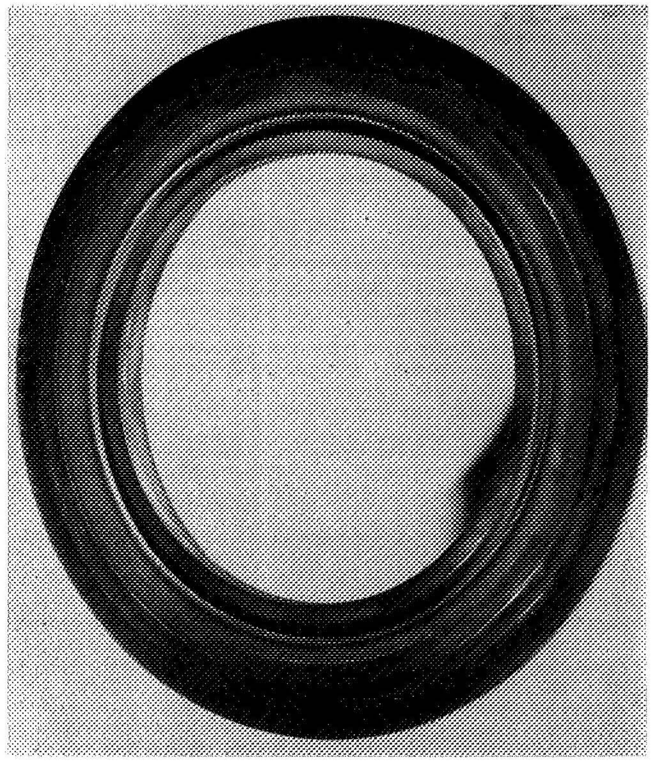
d) Balls

ENCLOSURE 38

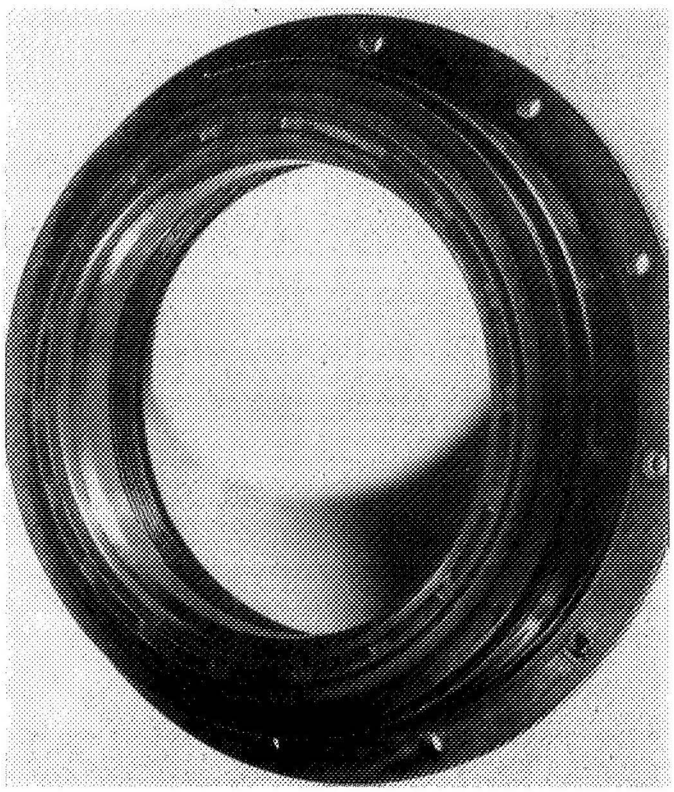
TEST SEAL PARTS AFTER FIRST 20 HOUR PORTION OF THE FIRST 250 HOUR ENDURANCE RUN (MOBIL XRM-177F OIL AT 600°F)



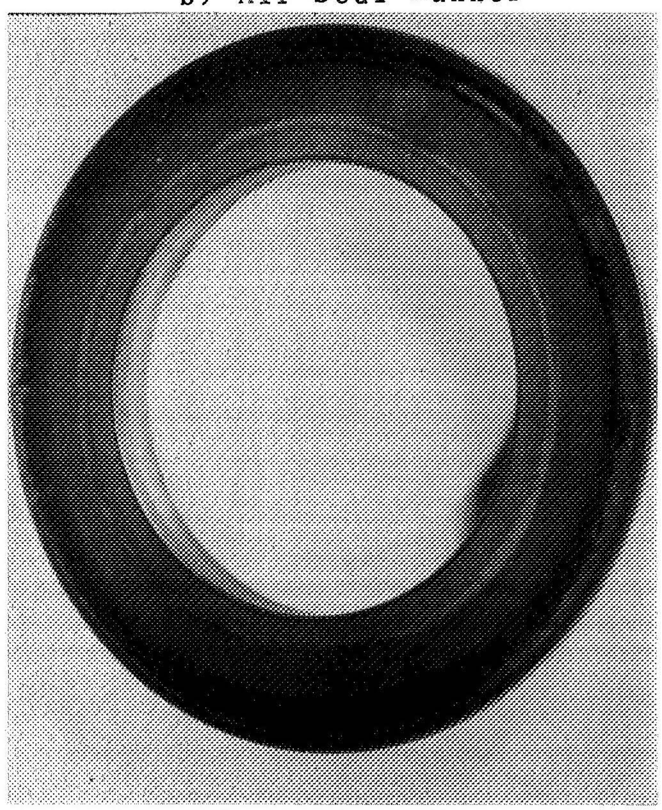
a) Air Seal



b) Air Seal Runner



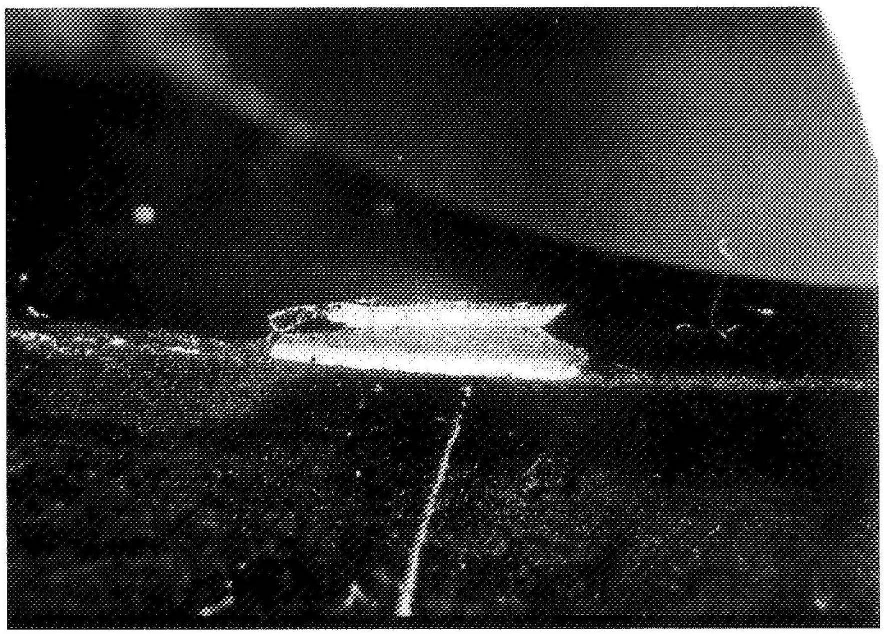
c) Oil Seal



d) Oil Seal Runner

ENCLOSURE 39

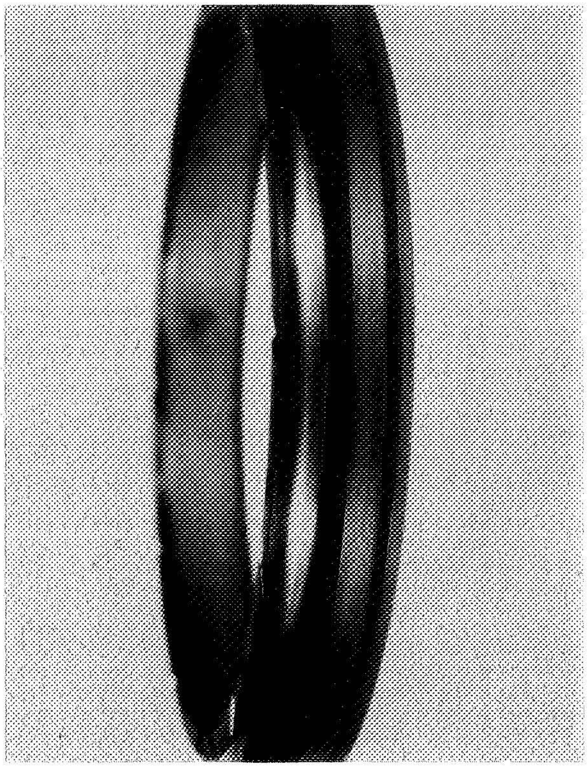
Flaking of the Ion-Deposited Silver Plating on a 4340 Steel Cage after 650°F Mobil XRM 177F Oil Endurance Run for 230 Hours



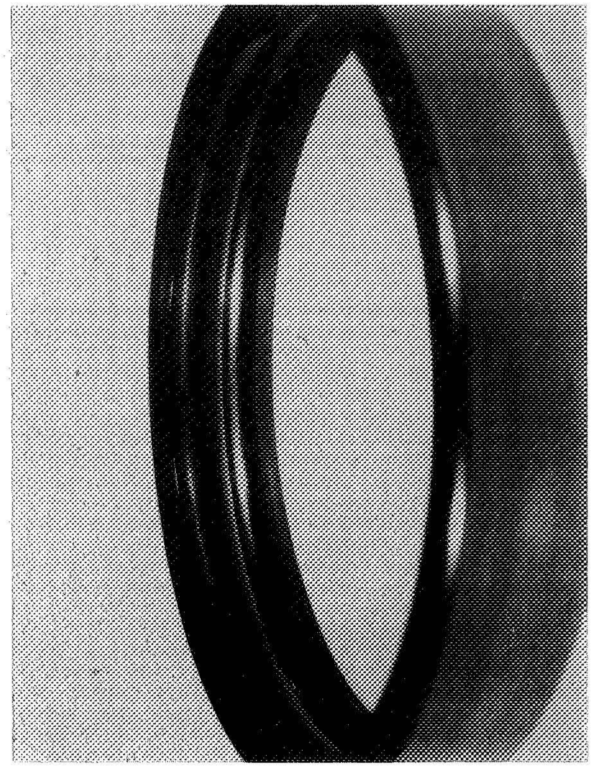
Magnification 10X

ENCLOSURE 40

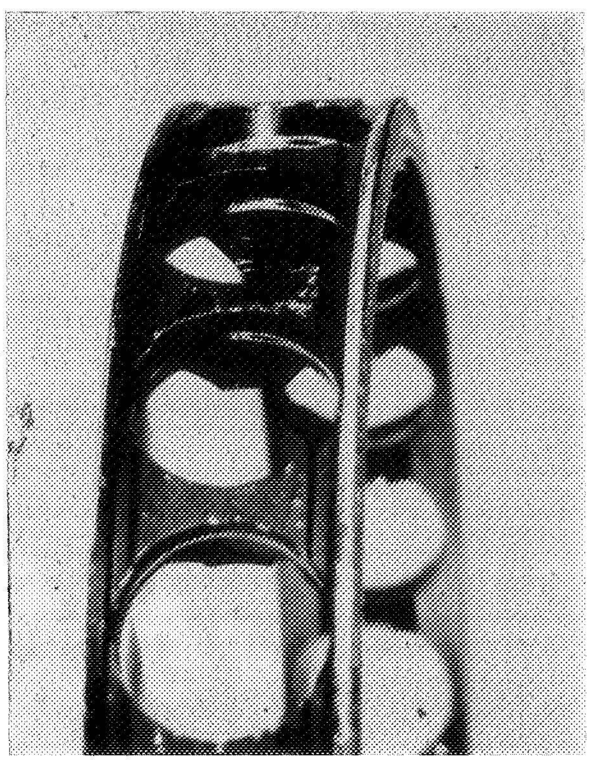
TEST BEARING PARTS AFTER 650°F MOBIL XRM 177F OIL ENDURANCE RUN FOR 230 HOURS



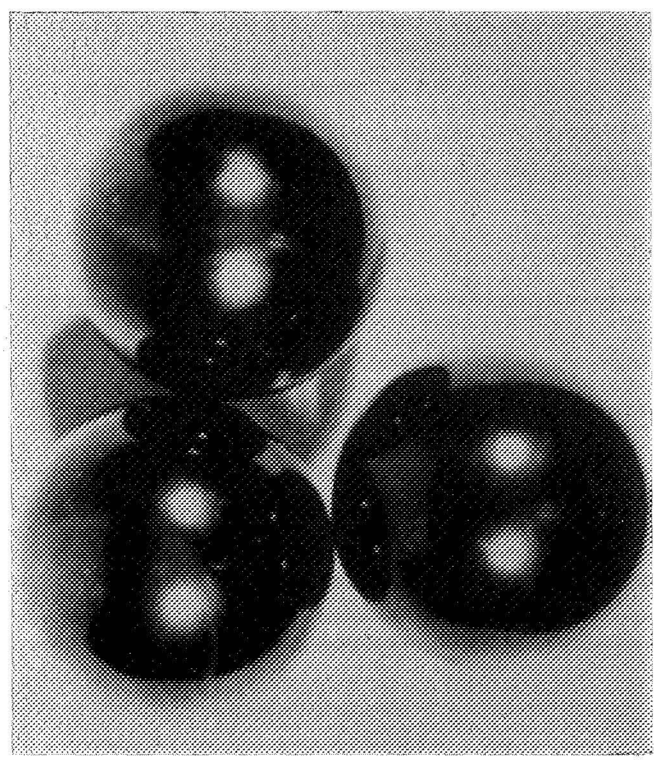
a) Inner Race



b) Outer Race



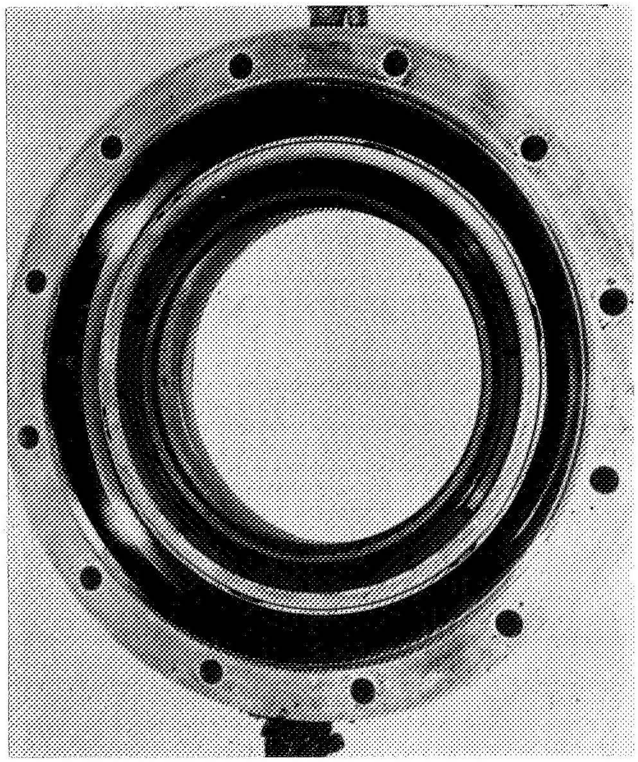
c) Cage



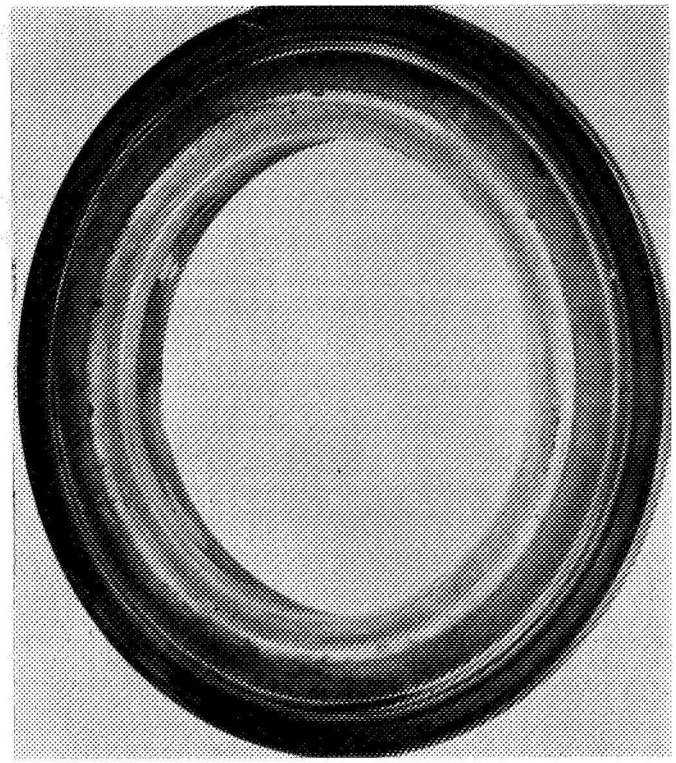
d) Balls

ENCLOSURE 41

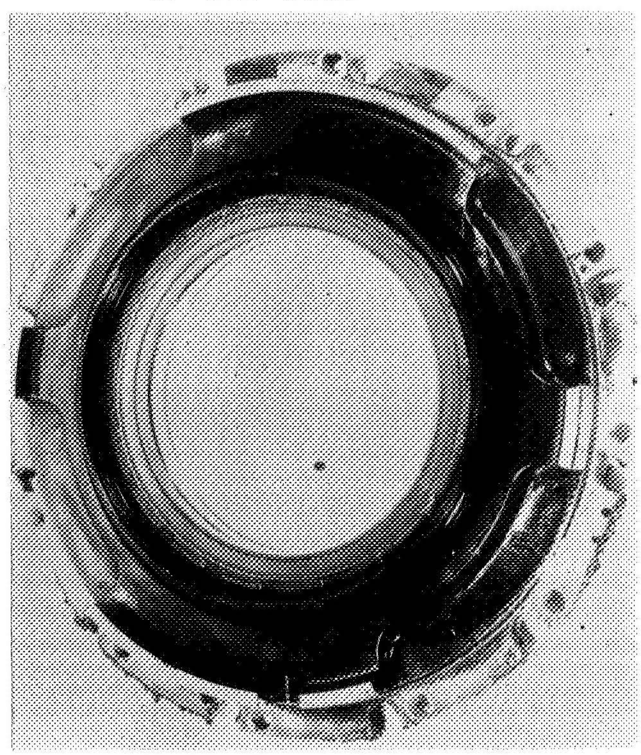
TEST SEAL PARTS AFTER 650°F MOBIL XRM 177F OIL ENDURANCE RUN FOR 230 HOURS



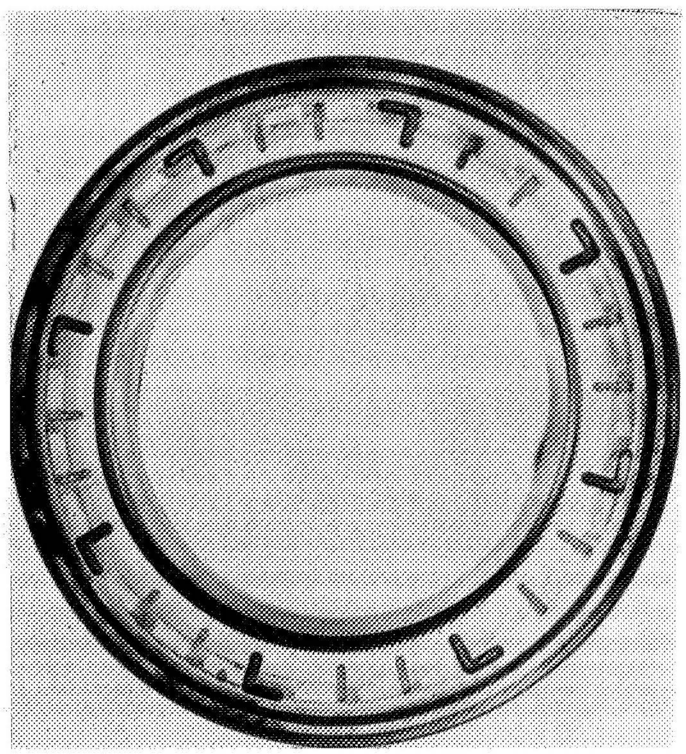
a) Air Seal



b) Air Seal Runner



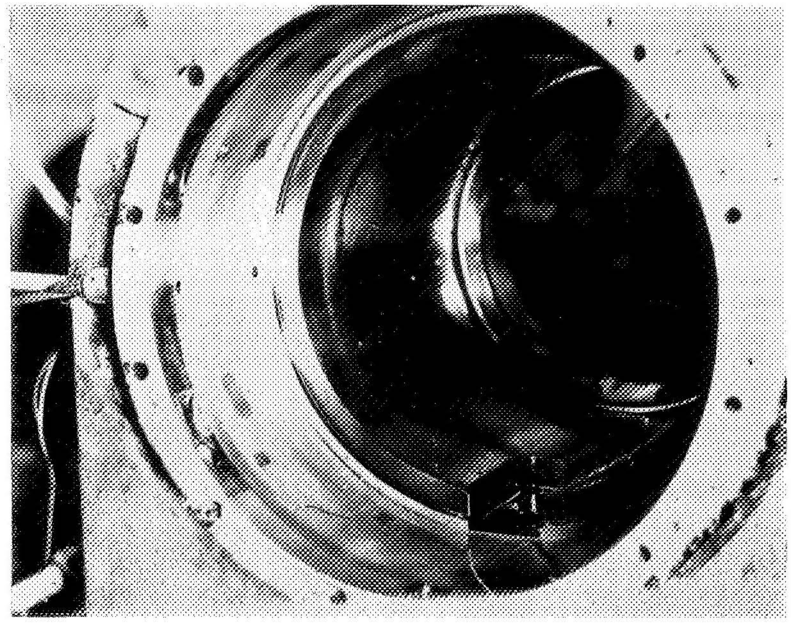
c) Oil Seal



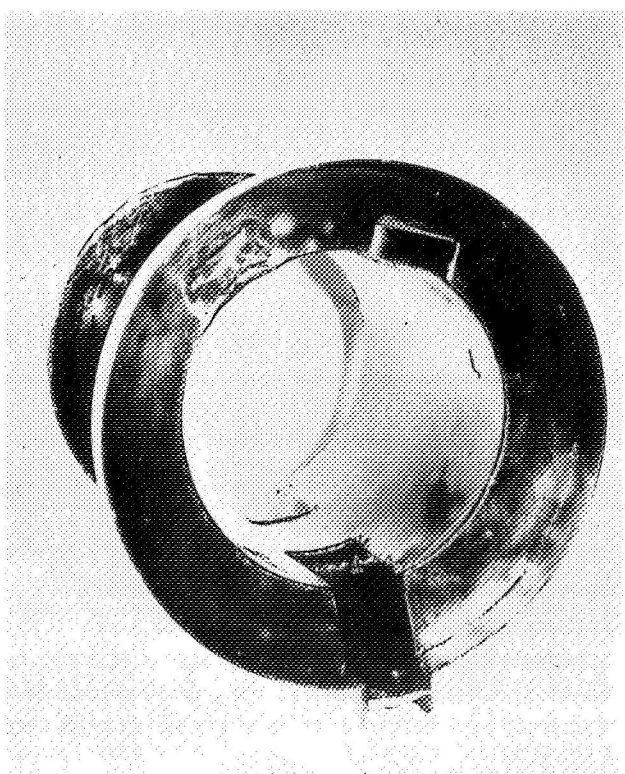
d) Oil Seal Runner

ENCLOSURE 42

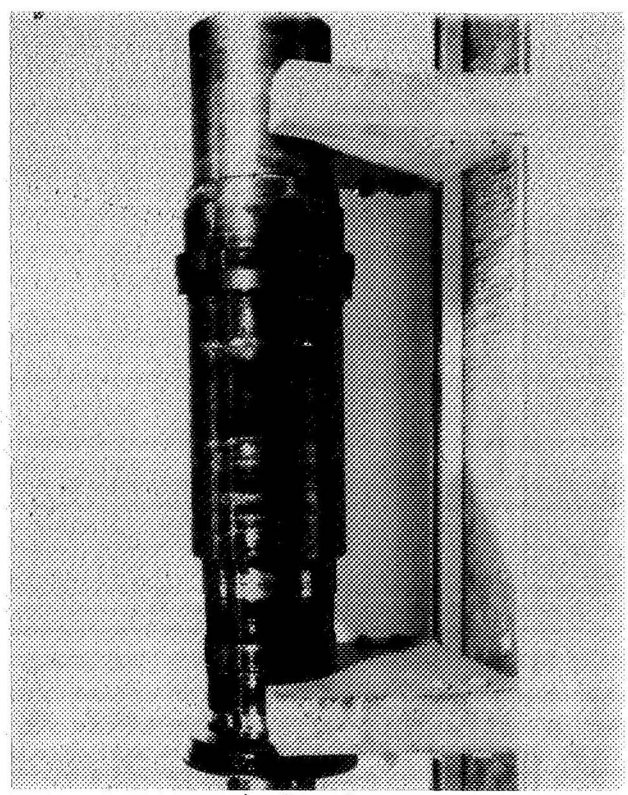
TEST RIG PARTS AFTER 650°F MOBIL XRM 177F OIL ENDURANCE RUN FOR 230 HOURS



a) Test Bearing Housing



b) Heat Shield



c) Shaft

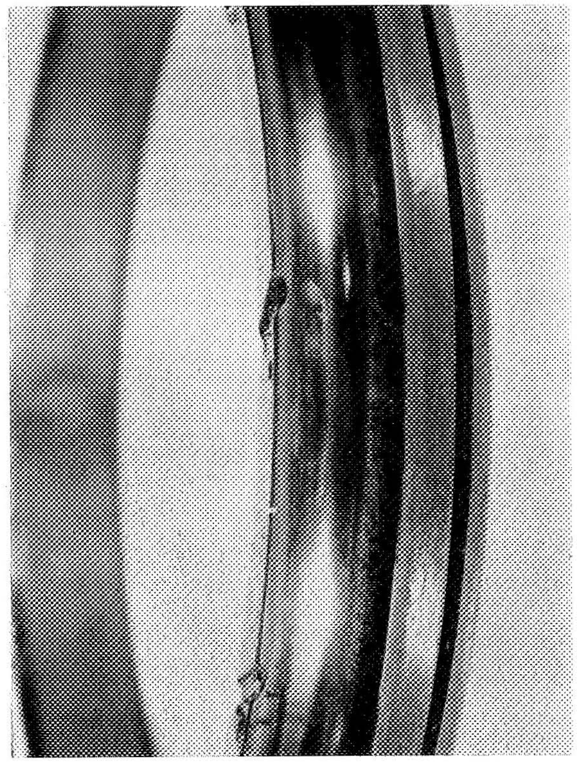
ENCLOSURE 43

Flaking of the Ion-Deposited Silver Plating on a 4340 Steel Cage after 650°F Mobil XRM-109F and 10% by Weight Kendall Heavy Resin 0839 Endurance Run for 250 Hours

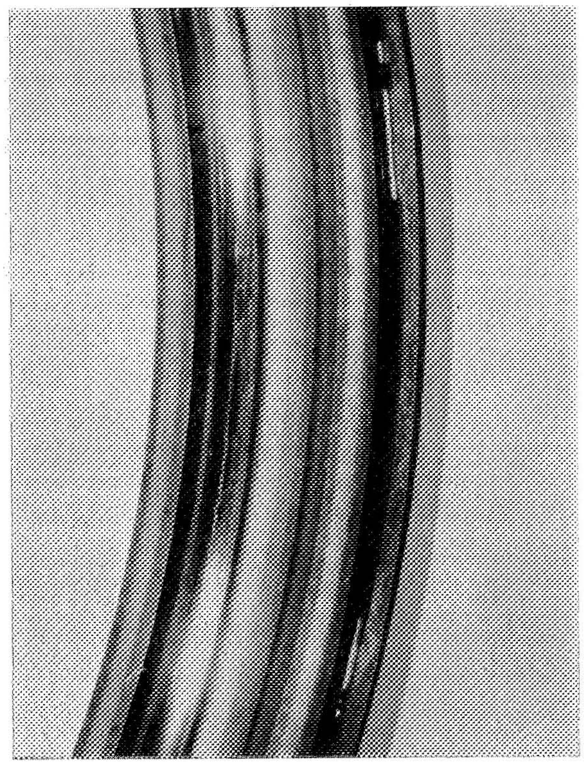
Magnification 10X

ENCLOSURE 44

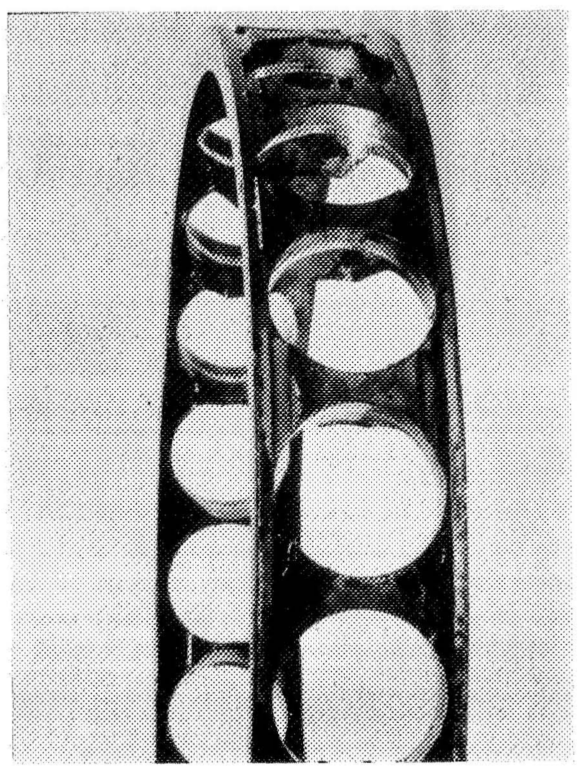
TEST BEARING PARTS AFTER 650°F MOBIL XRM 109F AND 10% BY WEIGHT KENDALL HEAVY RESIN 0839 ENDURANCE TEST FOR 250 HOURS



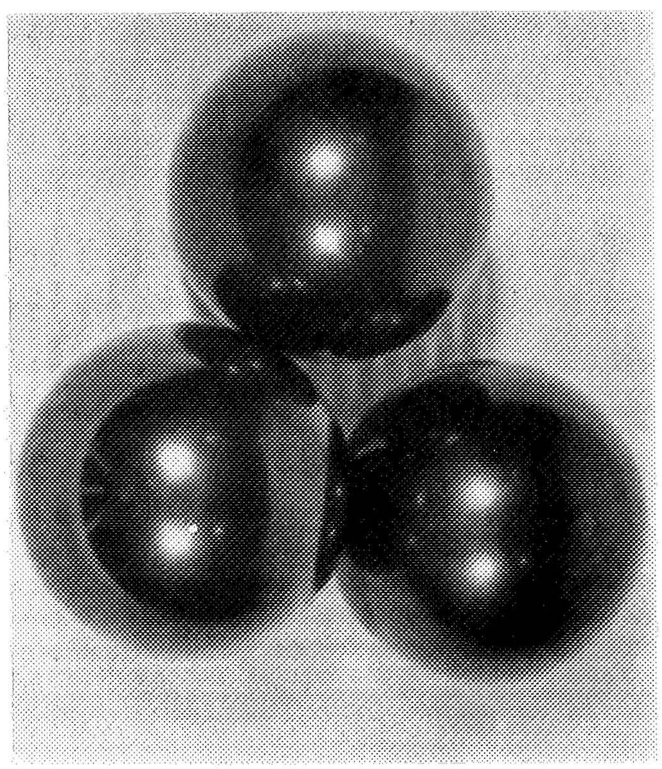
a) Inner Race



b) Outer Race



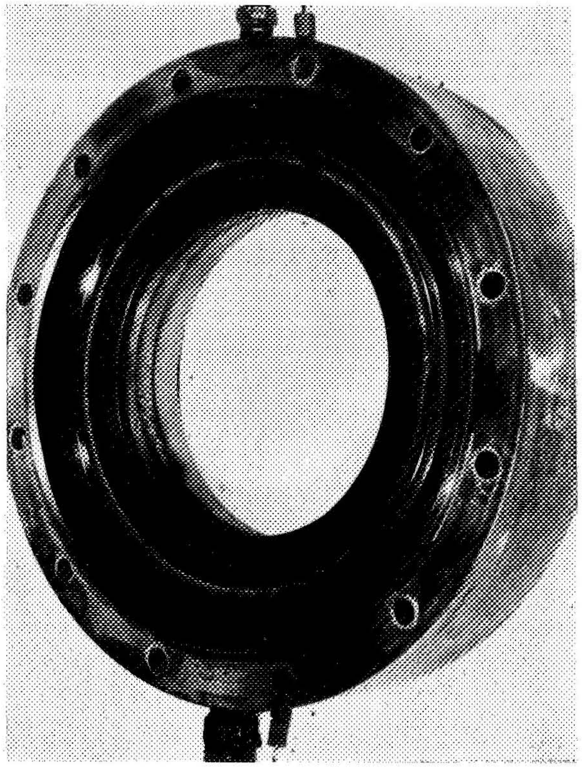
c) Cage



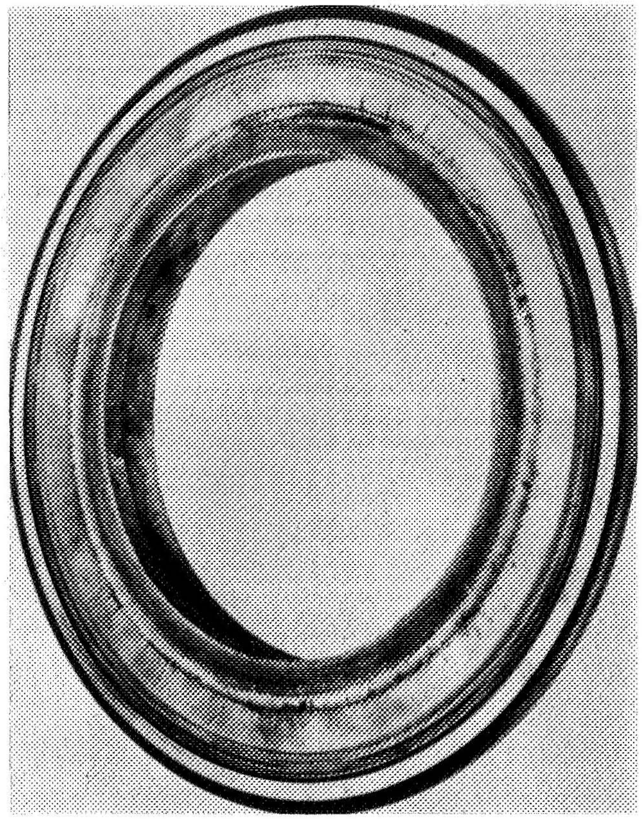
d) Balls

ENCLOSURE 45

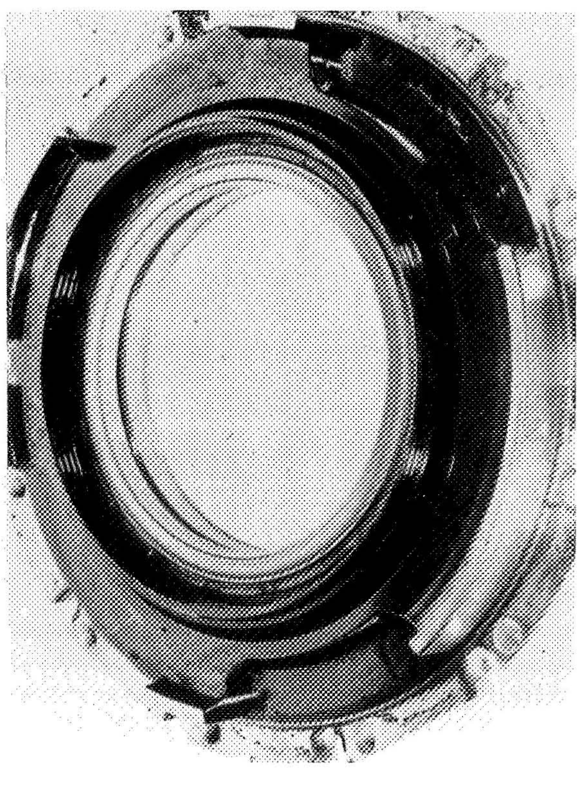
TEST SEAL PARTS AFTER 650°F MOBIL XRM 109F AND 10% BY WEIGHT KENDALL HEAVY RESIN 0839 ENDURANCE RUN FOR 50 HOURS



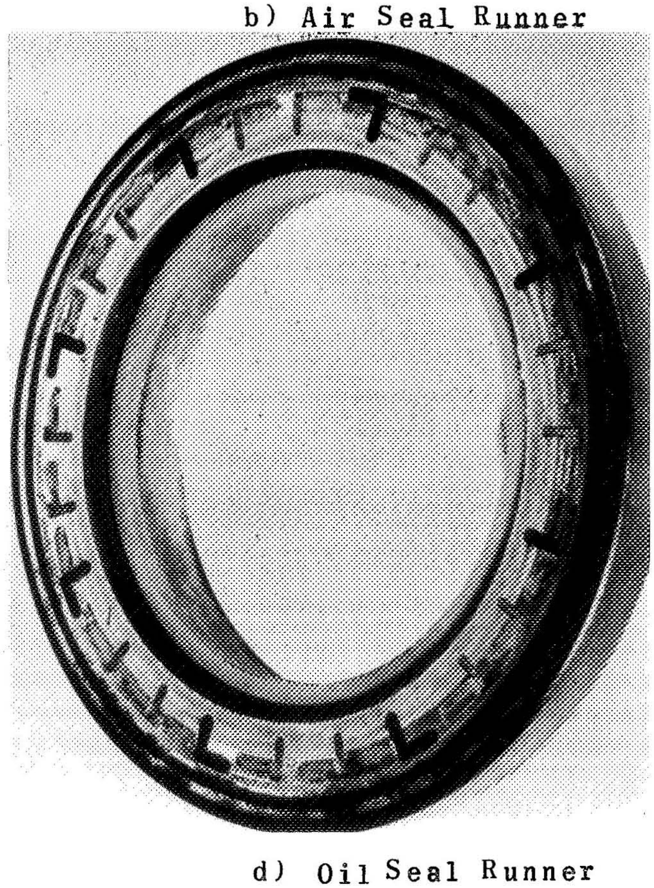
a) Air Seal



b) Air Seal Runner



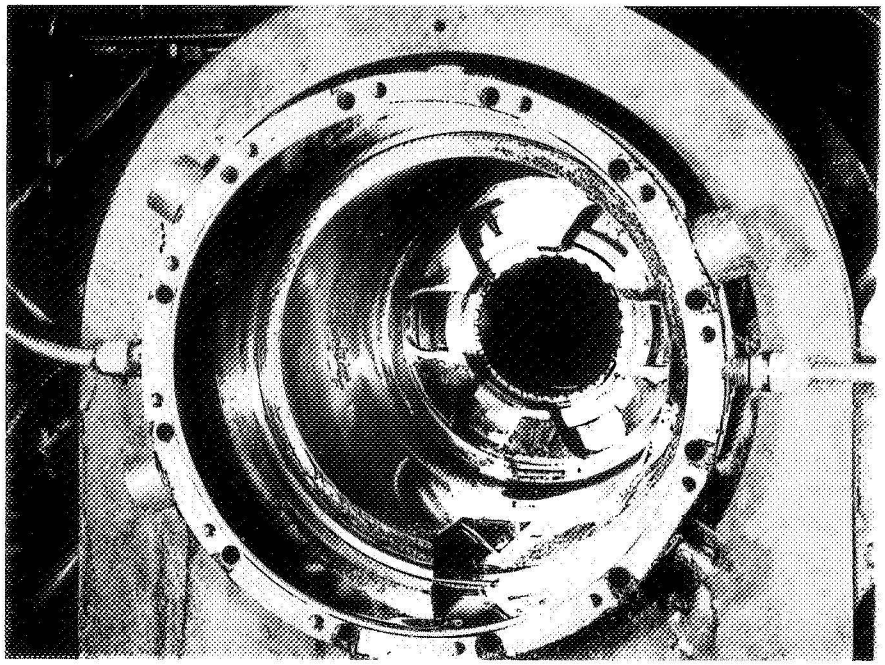
c) Oil Seal



d) Oil Seal Runner

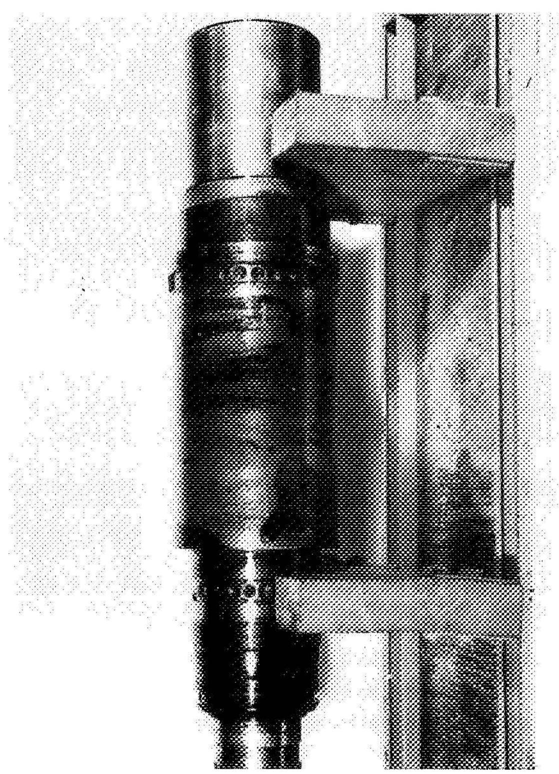
ENCLOSURE 46

TEST RIG PARTS AFTER 650°F MOBIL XRM 109F AND 10% BY WEIGHT KENDALL HEAVY RESIN 0839 ENDURANCE RUN FOR 250 HOURS



a) Test Bearing Housing

b) Heat Shield



c) Shaft

ENCLOSURE 47

SUMMARY OF EARLY TASK IV HIGH SPEED RESULTS*

Test 13

	Bearing Speed RPM	Oil Flow GPM	OR Temp. °F	IR Temp. °F	Oil Inlet Temp. °F	Oil Out Temp.	Total Seal Leakage SCFM	AMPS**	Volts	Time at Speed Hrs.	Seals***
	14,000	2.0	500	518	410	450	12.7	40-44	440		Set #1
	14,000	1.5	510	530	390	450	15.3	39-41	440	(3.1)	Set #1
	14,000	1.0	545	560	380	450	15.3	24-26	440		Set #1
	16,000	2.0	555	558	450	495	11.0	50-53	440		Set #2
	16,000	1.5	560	575	450	525	7.2	54-56	440	(2.0)	Set #2
	16,000	1.0	565	580	430	500	6.8	44-46	440		Set #2
	18,000	1.5	650	650	440	570	6.8	49-51	440		Set #2
	18,000	1.0	615	635	400	520	5.9	37-39	440	(2.5)	Set #2
a)	18,000	0.75	620	640	355	475	7.6	36-38	440		Set #2
b)	20,000	1.5	665	675	500	565		51-53	440		Set #2
c)	20,000	1.0	660	670			8.1	40	440	(0.7)	Set #2

a) *Bearing temperature still rising.*
b) Speed fluctuating (± 300 rpm); bearing temperature rising.
c) Taken just after reducing oil flow from 1.5 to 1.0 gpm with temperatures not stabilized. The total seal leakage was measured after 9 minutes at 1 gpm oil flow rate.

* All high-speed data obtained using bearing no. 267113 (459981G design).

** Drive motor is a 3-phase synchronous device; input current was measured for a single phase only. Variable speed obtained by water-cooled eddy-current clutch transmitting essentially constant torque.

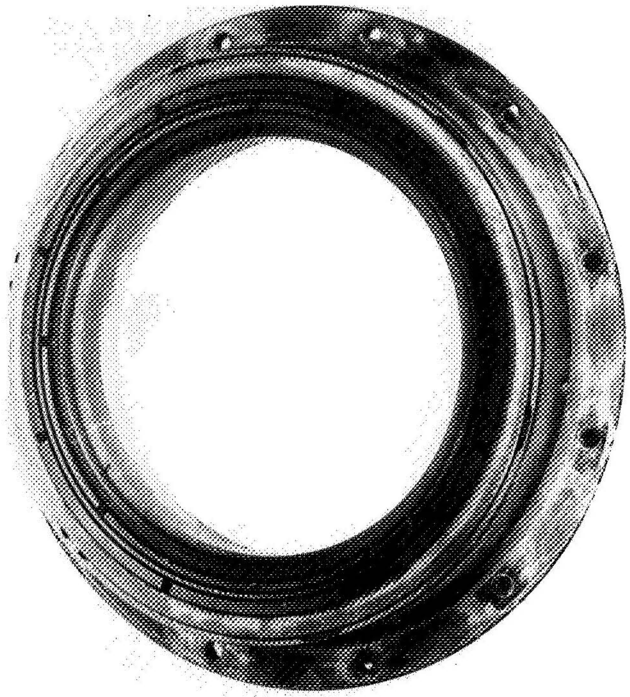
*** Seal Set #1 - (Original set) air seal 700397, ser. no. 6; shoulder 700397, ser. no. 1.
oil seal 101056B, ser. no. 1; shoulder 101056B, ser. no. 1.
Seal Set #2 - (Replacement set) air seal 700397, ser. no. 3; shoulder 700405, ser. no. 1.
oil seal 101056B, ser. no. 2; shoulder 101056B, ser. no. 2.

ENCLOSURE 48

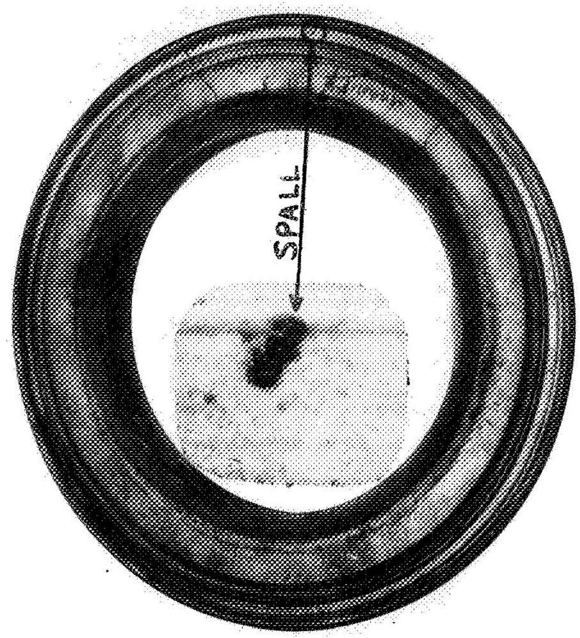
Test Seal Parts used in Mobil Jet II Open Atmosphere High Speed Test Replaced after 3.1 Hours at 14,000 rpm

Total life on oil seal/shoulder assembly 573.1 hours.

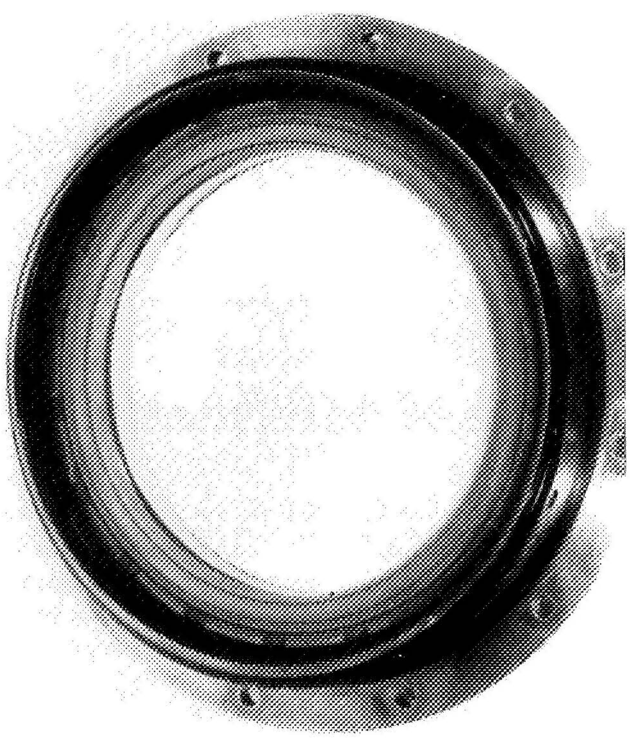
Total life on air seal/shoulder assembly 483.1 hours.



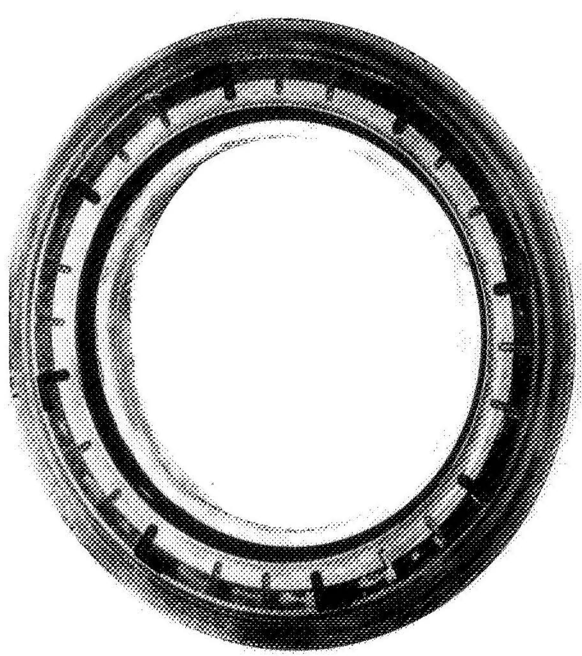
a) Air Seal



b) Air Seal Runner



c) Oil Seal

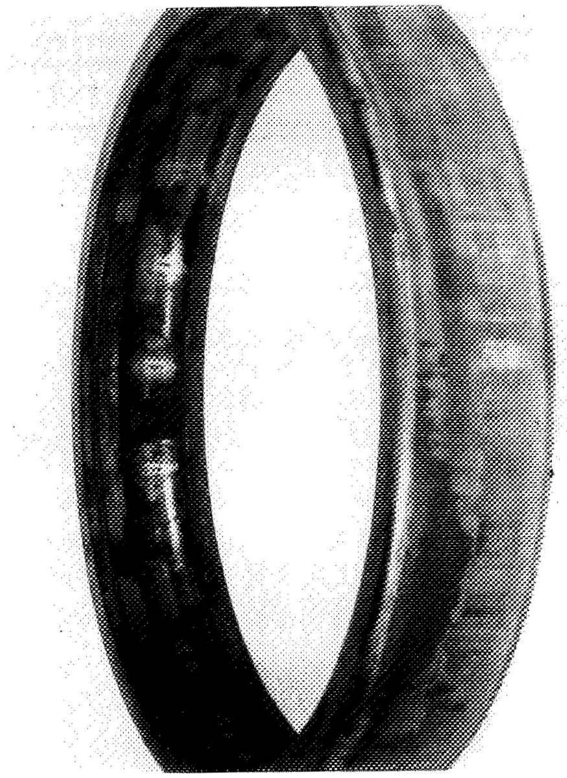


d) Oil Seal Runner

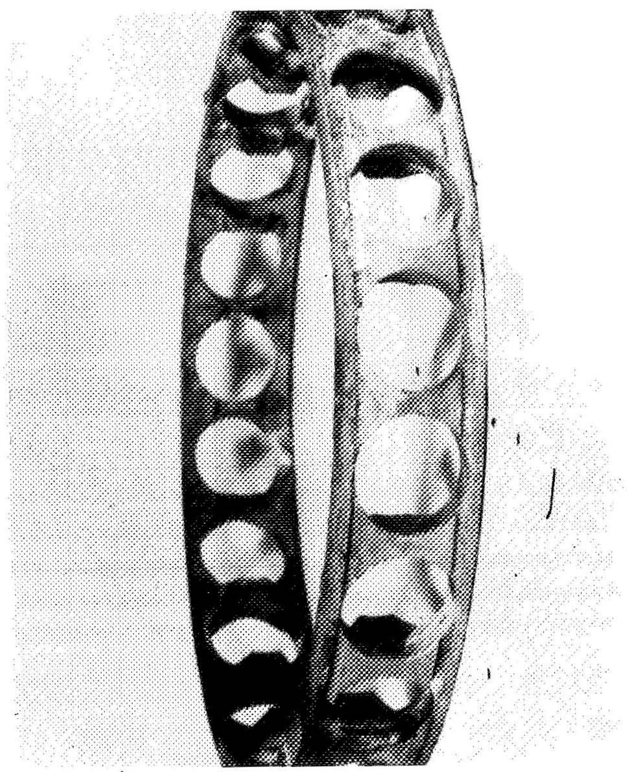
ENCLOSURE 49

TEST BEARING PARTS AFTER MOBIL JET II OPEN ATMOSPHERE TESTS AT 490° TO 660°F AND SPEEDS FROM 14,000 TO 20,000 RPM

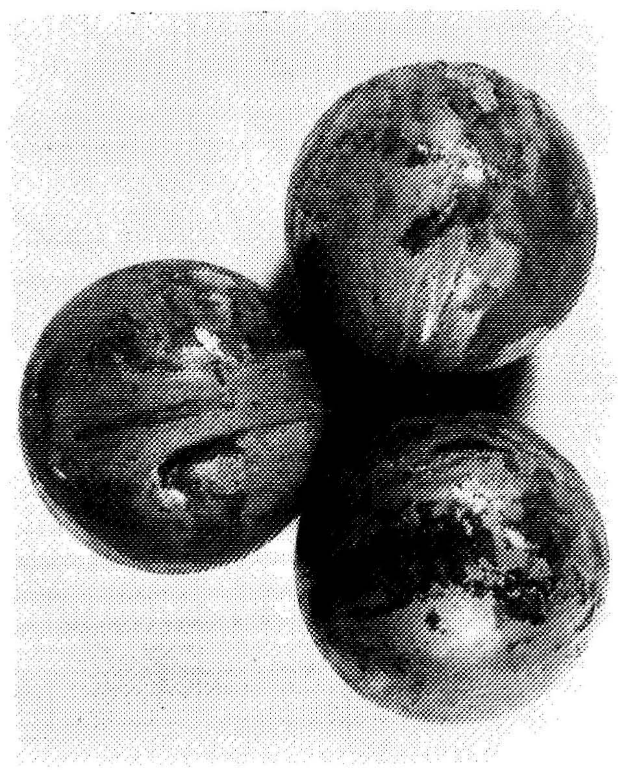
a) Inner Race



b) Outer Race



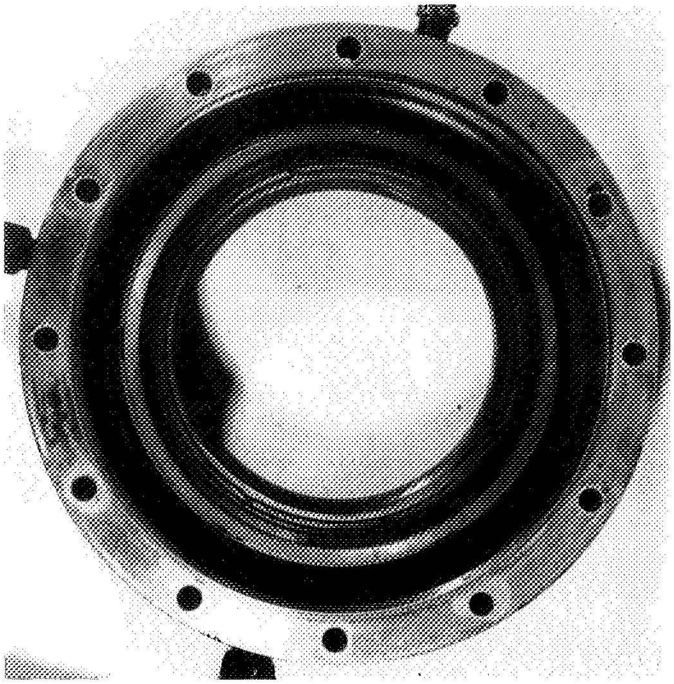
c) Cage



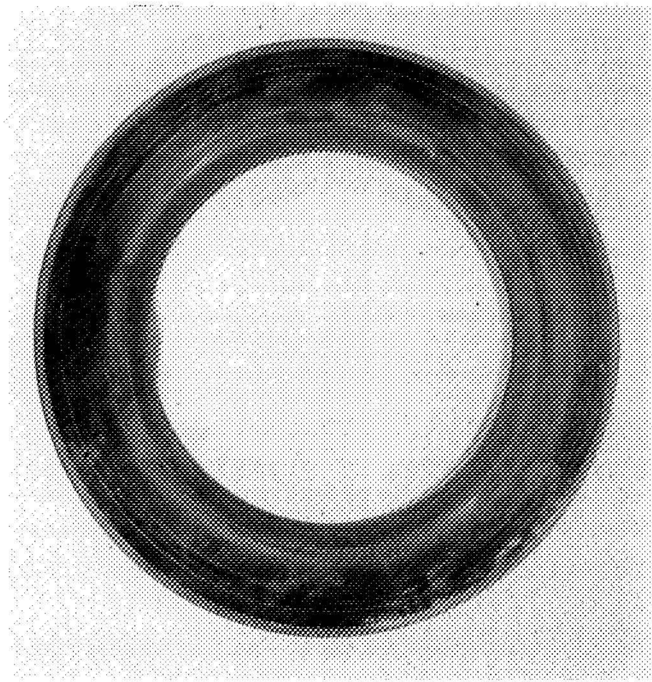
d) Balls

ENCLOSURE 50

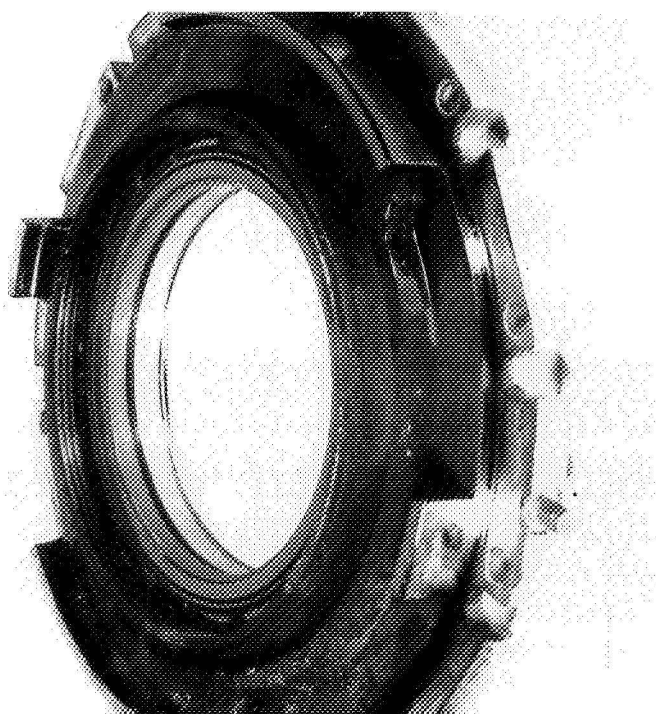
TEST SEAL PARTS AFTER MOBIL JET II OPEN ATMOSPHERE TEST AT 490° TO 660°F AND SPEEDS FROM 16,000 TO 20,000 RPM



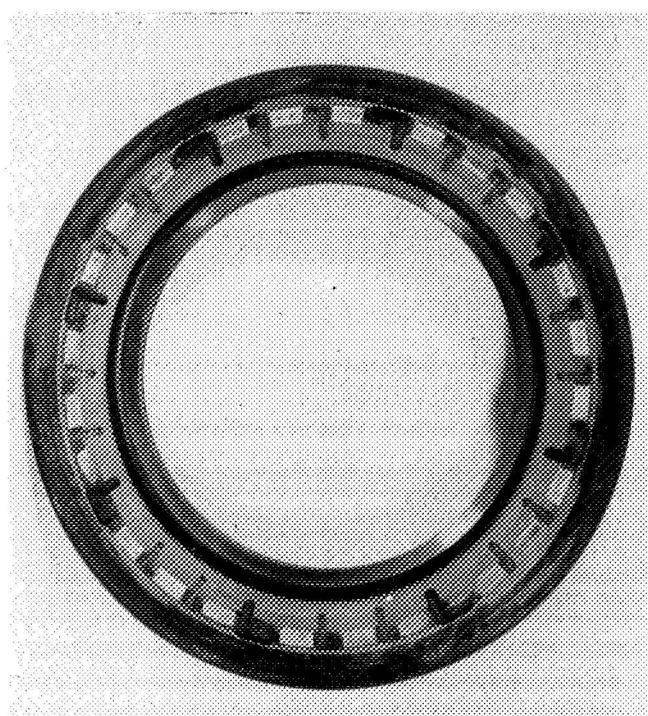
a) Air Seal



b) Air Seal Runner



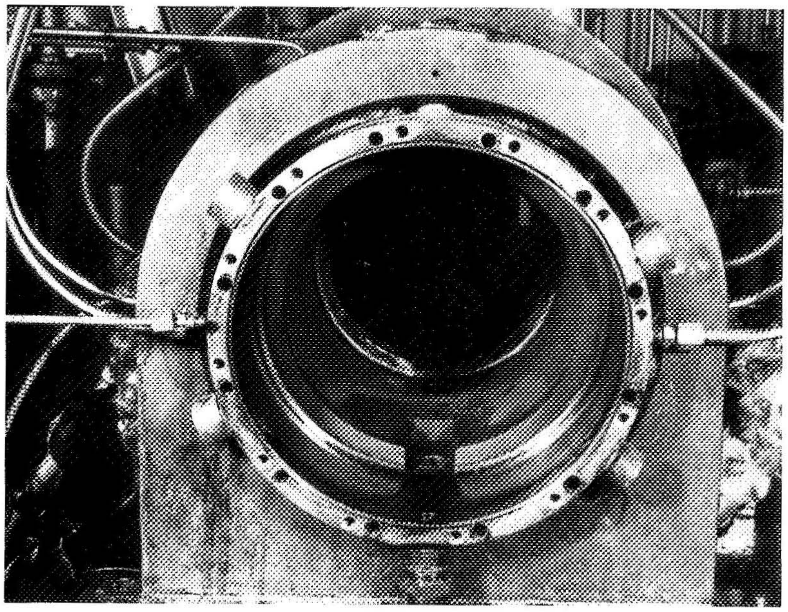
c) Oil Seal



d) Oil Seal Runner

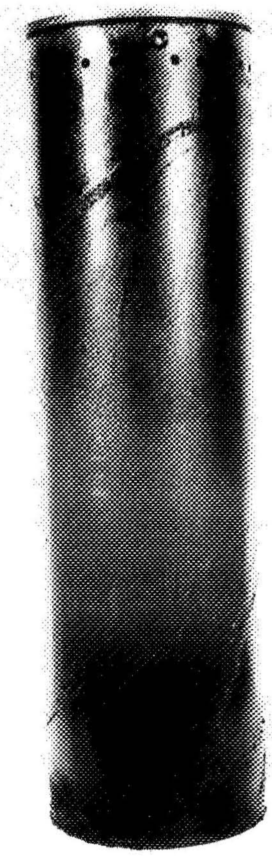
ENCLOSURE 51

TEST RIG PARTS AFTER MOBIL JET II OPEN ATMOSPHERE TESTS AT 490° TO 660°F AND SPEEDS FROM 14,000 TO 20,000 RPM



a) Bearing Housing

b) Shaft



c) Oil Sump Insert

ENCLOSURE 52

SUMMARY OF LATER TASK IV HIGH SPEED RESULTS*

TEST 14

Bearing Speed RPM	Oil Flow GPM	OR Temp °F	IR Temp °F	Oil Inlet Temp. °F	Oil Out Temp. °F	Total Seal Leakage SCFM	AMPS**	Volts	Time at Speed Hrs.	Seals***
15,000	1.25	390	420	190	390	12.3	45	440	0.1	
16,000	1.0	410	440	190	345	11.5	44	440	0.2	
18,000	0.75	440	460	205	370	10.2	47	440	0.2	
20,000	0.75	500	520	210	400	9.8	48	440	0.1	
	LOW WATER PRESSURE TO MOTOR CLUTCH CAUSED AUTOMATIC SHUT DOWN.									
20,000	0.75	510	535	175	405	7.6	46	440		
20,000	1.0	570	600	210	435	8.1	46	440	0.4	
20,000	1.0	560	600	200	430	7.6	46	440	1.0	
20,000	1.0	570	610	210	430	8.5	48	440	2.0	
20,000	1.0	575	620	210	440	6.4	50	440	2.4	

* All high speed data obtained using bearing no. 267114 (459981G design).

** Drive motor is a three-phase synchronous device; input current was measured for a single phase only. Variable speed obtained by water-cooled eddy-current clutch transmitting essentially constant torque.

*** Air Seal 700495 ser No. 1 shoulder 700397 ser. no. 3
Oil Seal 101056B, ser. No. 1 shoulder 101056B, ser. no. 2

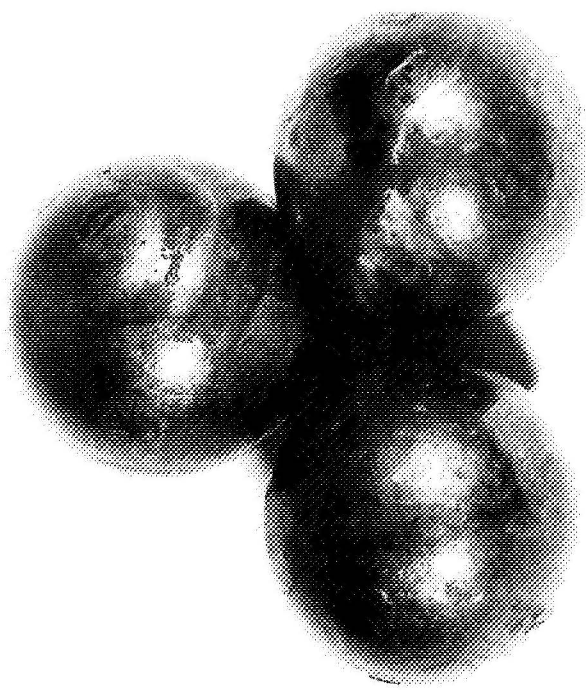
ENCLOSURE 53

TEST BEARING PARTS AFTER MOBIL JET II OPEN ATMOSPHERE TEST AT 390° TO 575°F AND SPEEDS FROM 14,000 TO 20,000 RPM

a) Inner Race

b) Outer Race

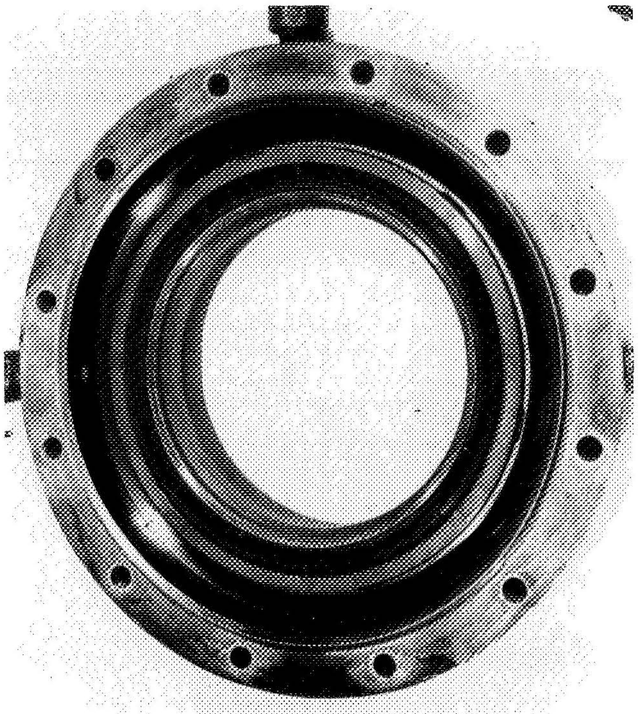
c) Cage



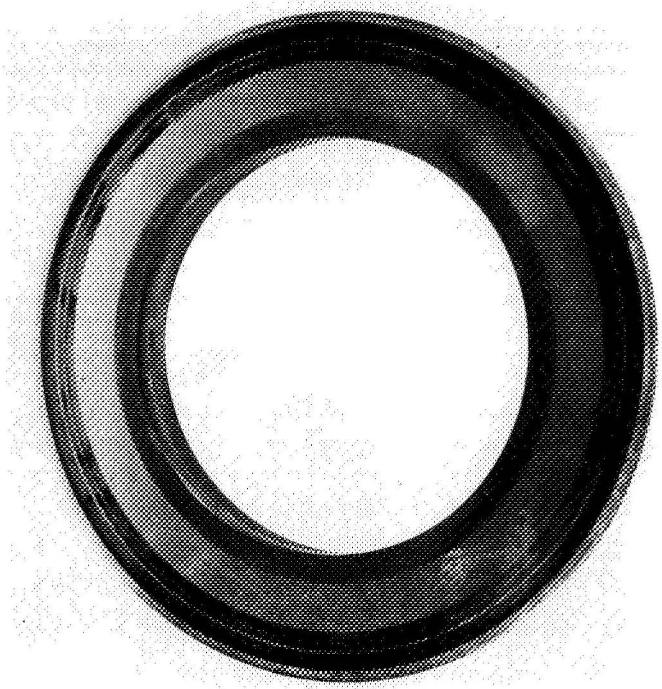
d) Balls

ENCLOSURE 54

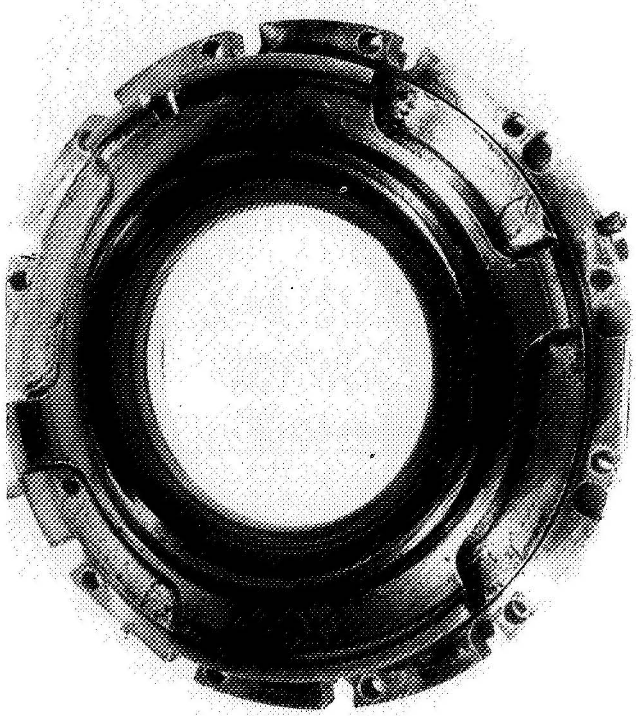
TEST SEAL PARTS AFTER MOBIL JET II OPEN ATMOSPHERE TEST AT 390° TO 575°F AND SPEEDS FROM 14,000 TO 20,000 RPM



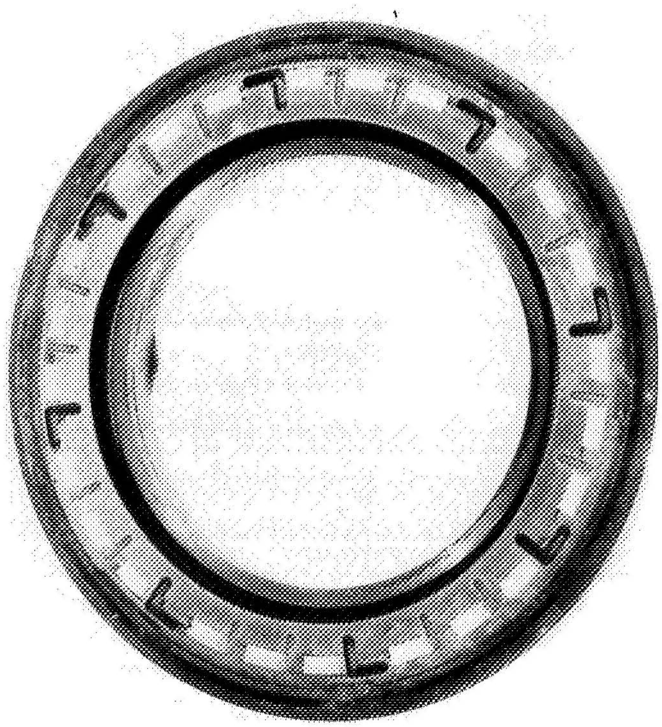
a) Air Seal



b) Air Seal Runner



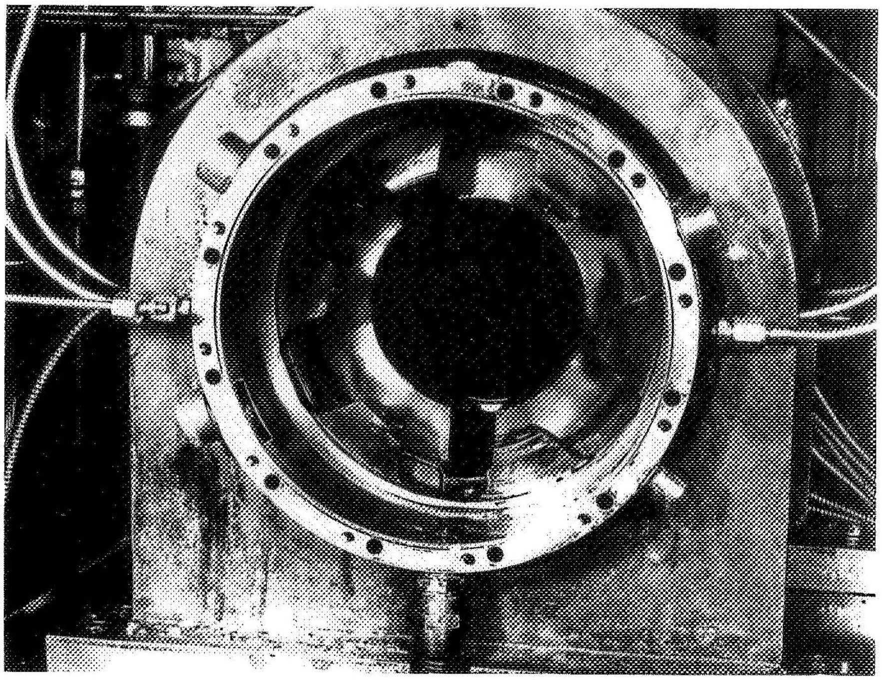
c) Oil Seal



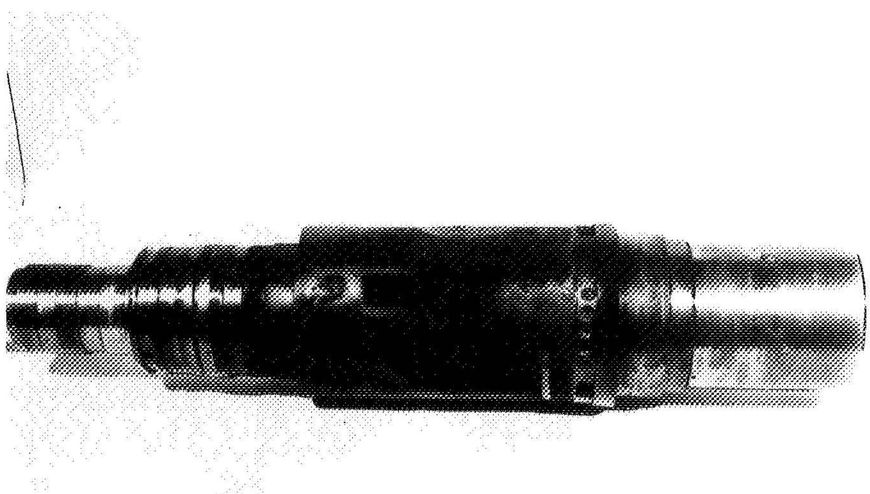
d) Oil Seal Runner

ENCLOSURE 55

TEST RIG PARTS AFTER MOBIL JET II OPEN ATMOSPHERE TEST. AT 390° to 575°F AND SPEEDS FROM 14,000 TO 20,000 RPM



a) Bearing Housing



b) Test Shaft

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