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HIGH TEMPERATURE LUBRICANT SCREENING AND SYSTEMS STUDIES

by

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16. Abstract

Four candidate lubricants for next generation aircraft gas turbine application were tested under open atmosphere conditions in a rig simulating an advanced engine 125 mm bore mainshaft thrust bearing position. Testing was conducted at speeds to 24,000 rpm (3×10^6 bearing DN), bearing ring temperature of 500°F, and with 1200°F air and 100 psi differential pressure across the seals installed in a dual tandem arrangement. Test bearing was a 125 mm bore split inner ring, outer race riding angular contact ball bearing under a 3280 lb. thrust load. One lubricant, a type II ester, performed extremely well. The main-shaft seal limited the performance. Numerous design improvements for this seal were indicated.

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FORWARD

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SUMMARY

Four candidate lubricants were tested to evaluate their capability to function in the severe operating environment imposed by the upcoming generation of ultra high speed aircraft gas turbine engines.

Testing was conducted in a test rig simulating a single advanced design aircraft turbine engine mainshaft bearing position. A 125 mm bore split inner race M50 tool steel test bearing (outer race riding angular contact under a 3280 lb thrust load) utilizing the latest available high speed high temperature design features was used in conjunction with a specialized dual tandem shaft seal (called the "air" and "oil" seals) arrangement.

The lubricants tested were FN 3158, a super-refined naphthenic hydrocarbon supplied by the Esso Oil Co. (This was blended with 5% by weight of Kendall 0839 heavy paraffinic resin); MCS 2931, an improved modified polyphenylether supplied by the Monsanto Company; DN 600 type 2, a synthetic hydrocarbon supplied by the Continental Oil Co; and Aeroshell Turbine Oil 555, a type II ester utilizing a proprietary additive package, supplied by the Shell Oil Co.

The shaft seal used in the air seal position was a sliding type carbon nosepiece face seal with bellows secondary sealing designed and manufactured by the Koppers Company. In the oil seal position two "self-acting" or hydrodynamic lift face seals, both using piston ring secondary sealing, were used. The first of these was also designed and manufactured by the Koppers Company. The second was a new high speed seal which had been designed by NASA, developed to its current state by Pratt and Whitney Aircraft Company on NASA funded research programs, and manufactured for use in this program by the Stein Seal Co.

The objective of the program was to evaluate the performance of these lubricants in open atmosphere (i.e., not inert blanketed) at speeds up to 24,000 rpm (3 x 10^6 DN) or as close to that as found attainable, 500°F bearing outer ring temperature, and with 1200°F air at 100 psi differential pressure applied to the seals.

Instrumentation used in the program included a seal leak measurement system using mass spectrometer analysis of a helium tracer flow introduced into the inter-seal cavity air supply system and a new search coil system to monitor the movements of a magnetized ball in the bearing.

The test plan consisted of a short duration screening test with each of the four lubricants to establish the limits of their operating capabilities. These were followed by two extended duration tests at 20,000 rpm using Aeroshell 555 and the Exxon FN 3158/Kendall 0839 mixture, the two most promising lubricants.

Testing showed the Aeroshell Turbine Oil 555 to be capable of successful operation under the test conditions in all of the areas investigated. This fluid showed excellent chemical stability and gave evidence of good EHD film performance at the ball to race contact and minimal cage wear. The other fluids gave marginal or unacceptable performance in one or more of these areas although these tests may have been somewhat biased by factors unassociated with the lubricants and further testing is indicated.

Testing on the program was limited in the maximum speed attained by failures of the seals in the "oil" position. (The Koppers sliding face type "air" operated successfully at all speeds without failure.) The seals may have failed due to shaft dynamics as a result of operation near the "whipping" mode critical speed. Numerous design changes were indicated for both seals.

Testing did show an ability on the part of the test bearing to run at the 3×10^6 DN level without problems, at least for the short duration tested, given acceptable lubrication. The program also demonstrated the ability of the search coil system to identify the occurrence of changes in ball rotational speed or rotational axis orientation. The coil additionally gave indications, in one case, of an ability to identify a bearing smearing failure in its early microsmearing stage.

INTRODUCTION

Current state-of-the-art aircraft jet engines, operating in the 1.8 x 106 bearing DN range (bearing bore diameter in mm times speed in rpm), are already close to the operational capability limits of major components in their shaft support systems, particularly the bearings, seals and lubricant. Requirements foreseen for future engine designs include increases in bearing DN values to the 2.5 to 3.0×10^6 DN range and bearing operating temperatures to 600°F. The need for an intensive R & D effort to meet these goals is clear. The purpose of the effort discussed in this report was to conduct evaluative testing of candidate lubricants, seals, and bearings toward attainment of this level of capability.

The present program was conceived to extend high-temperature lubricant screening studies that had been conducted by SKF Industries, primarily at 14,000 rpm (1.75×10^6 DN on the 125 mm bore test bearing used), under NASA contract NAS3-6267, into the 3×10^6 DN range.

The report will be of interest to personnel involved in gas turbine engine system level design and development work as well as technical specialists associated with the bearings, seals or lubricants used with these systems.

DETAILS

A. Background

The performance requirements of modern gas turbine engines, particularly those slated for use in the upcoming generation of commercial and military aircraft engines, have generated increasingly severe demands for improved mainshaft bearing performance. Higher thrust to weight ratios and lower fuel consumption dictate increased bearing speeds and higher operating temperatures. As pointed out in Reference 1 mainshaft bearing DN levels (the bearing bore in mm times its speed in rpm) have risen from the 1.0-1.2 x 10^6 level for the first generation of turbojet engines operational after World War II to over 1.8 x 10^6 for current state-of-the-art second generation designs. It is expected that engines utilizing bearings running in the 2.5 to 3.0 x 10^6 DN range will be operational by 1980. Bearing operating temperatures have increased accordingly and are expected to exceed 600°F in the more severe upcoming applications.

These requirements impose severe demands on the entire engine shaft support system, particularly the bearings themselves, the seals which separate the sump from the high pressure high temperature engine gas system, and the bearing lubricant and the lubricant itself. SKF Industries has conducted extensive research in these areas over the last twelve years. These programs have covered four different bearing sizes and test rig configurations. Much of this effort was conducted on NASA contracts, among them NASW-492, NAS3-7912, NAS3-11171, NAS3-6267, and presently NAS3-14320 which is the subject of this report.

Contract NAS3-14320 is itself an extension of work conducted on the previous contract NAS3-6267 in which the performance of aircraft gas turbine mainshaft ball bearings, shaft seals, and lubricants was studied under Mach 3 supersonic transport engine conditions. The results of contract NAS3-6267 were reported in references 2 and 3.

A.1 Summary of Previous Testing on Contract NAS3-6267

The larger of the two rigs used for the NAS3-6267 program was designed to simulate a single 125 mm bore mainshaft thrust bearing installation utilizing a double shaft seal arrangement which allowed both inert gas blanketing and open air tests to be run. The rig used in the present program NAS3-14320 is a moderately modified version of the original NAS3-6267 rig. This rig, shown in Enclosures 1 through 4, incorporated the most advanced design, materials and manufacturing techniques available. Design details on the rig in the NAS3-14320 configuration are given in the Test Equipment section of this report.

Two basic methods of lubrication were investigated on the NAS3-6267 program, both of which were compatible with inert blanketing. These were, first, oil-jet lubrication of the bearing, simply replacing the air in the bearing chamber and oil sumps by nitrogen, and second, mist lubrication using the nitrogen to carry the oil to the bearing and to remove a large portion of the heat generated. In the second of these two systems, the oil was not recovered. The system was of the once-through type, dispensing with the need for oil coolers and scavenge pumps and also with many of the thermal stability problems associated with recirculating oil systems.

Testing on the 125 mm bore rig in the NAS3-6267 program utilized face seals of the more conventional sliding face primary - bellows secondary design as well as hydrodynamic lift primary-piston ring secondary designs (both supplied under sub-contract by Koppers Company, Inc.) with a number of advanced 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 L10 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 made of either M-50 steel rings and balls or WB49 steel rings with M1 steel balls were used.

In a separate high speed phase of the NAS3-6267 program, a 125 mm-bore bearing was run successfully at speeds to 20,000 rpm (2.5 million DN). Testing was conducted at four speed levels, of approximately 2 hours duration each, of 14,000, 16,000, 18,000 and 20,000 rpm respectively using a Type II ester lubricant under open atmosphere conditions. Results of this test indicated that endurance testing at 20,000 rpm was feasible using conventional bearings together with specially designed seals. Results also indicated that it was likely that successful operation at even higher DN values might have been achieved, perhaps in the 3.0 million DN range.

B. Research Objectives

The objective of the NAS3-14320 program was basically to continue the investigation begun under earlier contracts NAS3-11171 and NAS3-6267, although with a shift in the direction of investigation. The two earlier programs included testing on both 25 mm and 125 mm bearing rigs. Testing in the larger rig was, for the bulk of the testing, limited to a maximum speed of 14,000 rpm (1.76×10^6 DN) at temperatures up to 800°F although a small amount of open air testing was conducted at up to 20,000 rpm (2.5×10^6 DN) at bearing ring temperatures of 500°F. As previously discussed, this testing at 2.5×10^6 DN, as far as it went, successfully demonstrated the feasibility of system operation in the speed ranges approaching those expected for future aircraft engines. More work in this area was, however, needed.

The present program, NAS3-14320, was thus conceived to extend testing on the 125 mm bore rig up to the 3.0×10^6 DN range or as close to it as possible. Four new candidate lubricants were to be tested under open atmosphere conditions at bearing outer ring temperatures of 500°F. Lubrication would be by circulating oil only; no mist testing would be conducted. A short duration screening test would be run with each of the four lubricants and extended duration testing would be conducted on the best two of the four. As with the previous program the dual shaft seal arrangement would be retained.

Also to be added for the program were an infra-red pyrometer system and a view port for it in the rig housing to allow its use in measuring test bearing ball and cage temperature, an under-race cooling system for the test bearing inner ring, "search coil" instrumentation to monitor ball motion in the test bearing, and a system to supply air at a constant 50% ± 10% relative humidity to the test bearing chamber.

The details of the program are given in the Contract Work Statement included as Appendix I to this report.

The program was broken down into three tasks as follows:

Task I - Redesign and modify the rig, and procure bearings, seals and lubricants as required by the test program.

Task II - Conduct screening tests on each of four candidate lubricants at 24,000 rpm or at the maximum speed obtainable below this. Conduct extended duration tests on the two best performing lubricants indicated in the screening tests. Extended tests to be run at the most severe feasible speeds indicated during the screening tests and to be of a duration of 50 hours or until the first indication of approaching failure.

Task III - Reporting.

C. Test Facilities

The equipment shown diagrammatically in Enclosure 1 is described in the following section. To facilitate this the system is subdivided into the following parts:

C.1 Test Rig
C.2 Drive System
C.3 Hot Air System
C.4 Inter-seal Cavity Tracered Air Supply
C.5 Bearing Chamber Air Supply
C.6 Test Oil Recirculating and Conditioning System
C.7 Under-race Lubrication and Cooling System
C.8 Pneumatic Loading System.

C.1 Test Rig

The test rig is designed to simulate aircraft engine mainshaft designs by avoiding thick sections in the shaft and bearing housings and by introducing flexible sections between the main rig outer housing and the bearing outer rings. This flexibility is intended to simulate the self-aligning ability of current aircraft engine bearing mounts. Enclosures 2 and 3 show photographs of the rig and a section view is shown in the assembly drawing in Enclosure 4. The numbers in brackets in the following text refer to the part numbers shown on the rig assembly drawing in Enclosure 4.

The test rig consists of a 12" diameter cylindrical housing [1] in which a hollow shaft of approximately 5" maximum diameter [2] is supported by the test bearing at one end and a cylindrical roller bearing at the other. The housing itself is mounted in a horizontal position on a table by means of a special support system which maintains centerline height and parallelism with the table while freely permitting both radial and axial thermal expansion of the rig. This arrangement is best shown by the isometric sketch in Enclosure 5.

Pedestal [11] is positioned in line with the roller bearing. Pedestals [11] and [12] are bolted securely to the rig table and dowelled to maintain alignment.

The environment inside the bearing test chamber is maintained by a test seal (referred to in this report as the "oil seal") adjacent to the test bearing at one end and a rig seal [65] at the other. The high-temperature air-side seal (referred to in this report as the "air seal") is mounted on an extension ring [6] attached to ring [5] . Another ring component [61] directs the hot air onto the back of the air seal and also provides static sealing.

The rig therefore contains a total of four seals defining three distinct chambers. These are the test bearing chamber, the inter-seal cavity, and the hot air chamber. These are discussed separately below.

The test bearing chamber is supplied with air at a controlled pressure of 6 psi and a relative humidity of 50% ± 10%. Within this space are accommodated three lubricant feed rings, two for the test bearing [9] and one for the roller bearing [36] . The number and positions of the drain holes have been selected to prevent a head of oil from accumulating and being churned in the recirculating rig.

The inter-seal cavity is of small volume and is supplied with air which may be tracered with helium for leak detection as described elsewhere in this report. The air pressure, at 111 psig, is maintained above the pressure in the adjoining chambers and thus provides a positive separation between the hot air and the bearing chamber.

The hot-air chamber, which is essentially the space enclosed by [62] , is supplied with air at 100 psi above the test bearing chamber pressure and 1200°F. This simulates engine operating environmental conditions for the air-side test seal. The thrust load on the test bearing is applied by the 100 psi difference in pressure between the hot-air chamber and the test bearing chamber as described in section C.8.

The bearing inner ring mounting on the test rig shaft must be designed to restrain the ring from rotation on the shaft and at the same time not induce excessive mounting stresses or deflections affecting the internal clearance in the bearing. The ring materials selected for the test and rig bearings, M-50 and WB49 respectively, both have lower coefficients of thermal expansion than the shaft material, Inconel X-750. If the normal practice of fitting the bearing inner rings directly onto the shaft with a slight interference fit were employed, excessive tensile hoop stress would be generated at operating temperature due to the unequal thermal expansions of the two materials. This could lead to ring fracture or a significant reduction in internal bearing clearance. Sleeves of M-1 steel were therefore interposed between the shaft and bearing inner rings. These are of the form shown in Enclosure 6 and have a gap at room temperature between the shaft and the inside surface of the sleeve under the bearing. Rotation of the sleeve on the shaft is prevented by an interference fit between the shaft and the thinned-down ends of the sleeve. Radial location of the bearings at low temperatures is achieved by the bending stiffness of the sleeves.

As the temperature rises during the heat-up period and the shaft expands it progressively closes the gap under the sleeve until contact occurs at a temperature of 500°F. It should be noted that the under-race cooling and lubrication flow holes shown in the sleeve in Enclosure 6 are present only in the sleeve used with the split inner race test bearing. The rig roller bearing on the other end of the shaft utilizes a similar mounting sleeve without these holes.

A rig roller bearing is used to support the mainshaft of the test rig at the end opposite the test bearing. This cylindrical roller bearing (Enclosure 7), using WB49 steel rings and M-1 steel rollers, has an out-of-round outer ring which provides radial preloading to prevent skidding.

C.2 Drive System

The rig drive system is shown diagrammatically in Enclosure 8. A constant speed 75 HP motor drives the test rig through an eddy-current clutch to provide variable speed.

The motor and clutch combination, mounted on an adjustable base bolted to the rig table, drives a jackshaft through a flat belt drive. The jackshaft unit consists of a hollow shaft mounted in matched pairs of preloaded angular-contact bearings with a 3" diameter removable, slightly crowned pulley at its center. The bearings are supported in steel pillow blocks welded to a rigid base and are lubricated by a separate circulating cold mineral oil supply fed to the top cap of each bearing. An oil drain running horizontally across the width of the pillow block returns the oil to the scavenge lines.

The rig shaft is connected to the jackshaft by a Koppers gear coupling. The other end of the jackshaft drives the slip ring and tachometer assemblies through a rotating electrical connector and a small flexible coupling. The jackshaft is hollow to carry the wiring from transducers in the test rig to the slip rings. Shear pins are provided in the boss of the motor pulley to permit rapid stoppage of the rig in the event of seizure of a rotating component within the rig. A plain bearing is fitted to the motor shaft to prevent damage to the surfaces in the event of shear pin breakage.

C.3 Hot Air System

A schematic diagram of the hot-air system is shown in Enclosure 9. The air flow commences with the air compressor which has a rated output of 91 scfm at 200 psig. Air feeds directly to a dryer and filter column which reduces its moisture content to a -50°F dew point and entrained hydrocarbons to 13 parts per million. The clean dry air then passes to a 20 cu. ft. receiver and thence through a shut-off valve to a pressure regulator. A pneumatic servo control on this regulator maintains the desired pressure in the rig air chamber as indicated by the dashed line in Enclosure 9. The regulated air then passes through an indicating flowmeter to a 45 kw electrical heater, also shown in Enclosure 9, in which the air passes through approximately 22 feet of type 316 stainless steel tube which is radiantly heated by electric elements, and thence to the rig hot air inlet manifold.

The manifold distributes the incoming air to a ring of nozzles which guide it directly to the back of the air-side test seal. The output of a thermocouple in the air stream at the outlet of the heater is fed to the electrical input controls of the 45 kw heater so that the desired temperature can be maintained.

The air passes beneath the baffle and exits by way of more nozzles and the exhaust manifold. It then passes to a remotely controlled back pressure valve situated beneath the rig table from where it is exhausted to atmosphere via a large diameter pipe passing through the cell wall to a stack.

C.4 Inter-seal Cavity Tracered Air Supply

Placing two seals in tandem creates a problem in measuring the leakage rate of the individual seals. This is resolved by introducing a small known quantity of helium into the air supplied to the inter-seal cavity. A mass spectrometer unit then draws samples from both the inter-seal and bearing cavities and analyzes them for helium content. The helium content of the bearing cavity sample is used to compute the leakage rate past the oil seal. Since the total air flow rate supplied to the inter-seal cavity equals the sum of the leakage rates of the individual leakage paths out of the cavity, leakage past the air-side seal is found by subtraction.

To provide an air-helium mixture, a flow mixing system was developed. This is shown diagrammatically on the right hand side of Enclosure 10. Air drawn from the compressor receiver tank is passed through a pressure regulator set to the desired inter-seal chamber pressure and then through a transmitting flowmeter to a receiving tank. This tank is fitted with a constant leak device set at 0.5 scfm so that the flow does not fall below the operating range of the flowmeter.

Gaseous helium from a bank of bottles is fed, after suitable pressure regulation, to two additional valves. The first maintains a constant pressure differential of 50 psi across the second by means of the sensing lines and the controller shown in Enclosure 10. The second valve, proportionally actuated from the flowmeter signal, serves as a flow controller under the conditions of constant pressure drop. The helium then mixes in preset concentrations of 1% - 5% at an abrupt junction with the air and flows to the receiver. This arrangement provides a constant air-helium mixture over a variable flow range.

The tracered air then passes through a dry positive displacement gas meter to the seal chamber. This meter serves to measure very low leakage levels at which the accuracy of the transmitting flowmeter reading is inadequate. When leakage rates are greater than 5 scfm, exceeding the capacity of the transmitting flowmeter, readings are again taken from the positive displacement meter. In this flow range, of course, the helium concentration is not maintained constant so that accurate leakage rates are not obtainable from the tracer analysis.

C.5 Bearing Chamber Air Supply

The air supply system for the bearing chamber is also shown in Enclosure 10. Air drawn from the compressor receiver tank at a point in front of the test cell passes through a humidity control cabinet which is capable of maintaining the relative humidity of the incoming air at 50 ± 5%. The humidified air then passes through a shut-off valve to a flowmeter. The output from this transmitting flowmeter is used to control a valve up-stream from it. The air is then admitted directly to the test bearing chamber. Air exits the test bearing chamber by the oil drain system described elsewhere in this report.

A certain amount of oil vapor is carried away by the departing air. The air is passed through a water-cooled condenser to remove as much of this entrained oil as possible. The cooled air at the condenser outlet exhausts through a pressure regulating valve servoed to test bearing chamber pressure so as to maintain the pressure in the chamber.

C.6 Test Oil Recirculating and Conditioning System

This system is constructed of Monel 400 alloy for corrosion resistance.

Oil circulation proceeds from an internal-gear-type Monel pump, and a Monel filter unit accepting fiberglass elements, through a flowmeter to the rig. The fiberglass elements have a specified pore size of 40 microns and deliberately have excess flow capacity, by about two and a half times, in order to secure low pressure drops and long life, even with an oil that is undergoing some thermal degradation.

The combined oil tank and oil heater unit shown in Enclosure 11 is also designed to act as a defoamer. Oil from the rig drain manifold enters the top of the tank, which is below rig level, in a direction tangential to the cylindrical inside surface of the tank. The swirling motion set up in this way allows time for the entrained air to separate from the oil. The oil then collects in the bottom of the tank where it is drawn out by the pump, either through an oil cooler or through a by-pass line, and the air passes out through the oil condenser mentioned previously.

Both the condenser and the oil cooler shown in Enclosure 11 are cooled with water from a central recirculating water system. The oil-in temperature to the rig is maintained at its specified limit by controlling the water flow to the oil cooler. The oil tank and defoamer unit has been sized to accommodate up to six gallons of lubricant which, together with the capacity of the pipes, pump, filter and inlet manifold, gives a maximum capacity of approximately 6 1/2 gallons.

To keep the system oil capacity to a minimum, heat is supplied to the oil through the large cylindrical inside surface of the oil tank by surrounding it with an oil heater. To preclude the possible occurrence of local "hot-spots" in the heater, which could lead to premature coking of the oil on the tank surfaces, liquid metal has been chosen as a heat transfer medium. The metal selected for this purpose is a lead-bismuth-tin eutectic which has a melting point of 158°F. This is directly heated by a 230V 12kw electric immersion heater. Corrosion problems with this liquid were minimized by selecting carbon steel for the outer container and heater sheath and by using hard chromium plating on the surfaces of the Monel oil tank which are exposed to the liquid metal. The toxicity of the liquid metal is considered tolerable under the operating conditions.

All components of the system, including the oil tank, the heat exchanger, and the condenser, are designed for ready disassembly and cleaning. This meant the inclusion of a number of extra flange joints and the avoidance of threaded connections wherever possible.

C.7 Under-race Lubrication and Cooling System

The test bearing under-race lubrication system shown in Enclosure 12 is supplied with oil by two tubes extending through the rig housing. These tubes jet oil into an annulus mounted adjacent to the bearing inner ring mounting sleeve (shown in Section view in Enclosure 6). The annulus acts as a reservoir in which oil collects as a result of centrifugal force and from which it flows into 12 axial grooves in the bore of the inner race sleeve. These 12 grooves break into 12 radial holes in the center of the sleeve. These holes are drilled in line with the test bearing inner ring split and they thereby feed oil into the bearing through the inner ring split face slots. Six of these 12 holes additionally intersect six axial grooves on the O.D. of the sleeve which direct some of the oil from the radial holes across the O.D. of the sleeve under the loaded half of the inner ring thereby providing direct cooling oil to this critical part of the bearing. Oil exits from the six sleeve O.D. grooves into one of two areas depending on which of the two oil seals used in the program happens to be in use. In the case of Koppers seal 101056B (Enclosure 13) the oil exits into an annular area in the bore of the mating ring from which in turn it passes through slots directly into the bearing cavity without performing much mating ring cooling. In the case of NASA seal CC850685 (Enclosure 14) the oil exits into an annular area in the bore of the part number CD850688 mating ring mounting sleeve from which it is directed successively by centrifugal force through holes in the sleeve and in the mating ring itself thus providing cooling for the mating ring.

In this program the first screening test and the first half of the second screening test were run with the Koppers seal but without the slots cuts in the mating ring to allow oil flow in the under-race cooling system. This configuration allowed flow to the test bearing split face slots but not across the bore of the loaded half of the inner ring. The mating ring slots were added prior to the second half of the second test making the system fully functional at that point.

Since the oil supply for the under-race cooling system taps off the lube pump outlet line upstream of the test bearing oil supply flowmeter, flow in the URC system is not included in the flowmeter reading. Flow tests, however, indicate a total flowrate of .25 gpm in the URC oil supply tubes when flowrate to the test bearing is 1.0 gpm. This 4 to 1 flowrate proportion may be used as well at the other test bearing flow levels used in the program.

C.8 Pneumatic Loading System

Thrust load is applied to the test bearing by means of the pneumatic pressure differences between the hot air chamber, the test bearing chamber, and the outside atmosphere acting on the balance diameters of the test seals and rig seal respectively. Appendix II presents a calculation of the load applied to the test bearing as a function of the rig cavity pressures.

D. Instrumentation

D.1 Data Collected

The output of temperature transducers located at various points in the test rig are recorded on two temperature recorders. One of these is a continuous recording Honeywell two-pen strip chart recorder and the other is a commutated multipoint Esterline Angus recorder with a forty-eight point capacity. The use of the two-pen strip chart recorder permits bearing inner and outer ring temperatures to be continuously monitored during testing.

This allows evaluation of any temperature excursions in these parameters which might occur. Any of the parameters recorded discretely on the multipoint recorder may also be plugged into the two-pen recorder at any time for continuous monitor.

The multipoint recorder records the temperature of a point every 70 seconds. The following parameters are recorded, in duplicate in most cases:

air seal housing temperature
oil outlet temperature
hot air temperature
oil tank heater temperature
air seal bellows temperature
test bearing chamber temperature
rig housing temperature, test bearing end
rig housing temperature, roller bearing end
drive system jackshaft pillow block temperature
roller bearing outer ring temperature
air seal mating ring temperature
oil seal mating ring temperature

In addition the following data is collected manually:

rig running time
shaft speed
motor current
motor voltage
current to all rig heaters
total air leakage flow rate
bearing chamber pressure
inter-seal cavity pressure
hot air cavity pressure
hot air flow rate
test bearing oil flow rate

D.2 Slip Rings and Connector Assembly

To achieve reliable monitoring of the test bearing inner ring temperature, and the temperatures of the two test seal mating rings, three pairs of thermocouples are embedded in the test shaft. In order to avoid any spurious voltages or cross-talk, the neutral lines, Constantan in this case, should not be commoned and thus twelve separate contacts to the roating shaft are required.

The slip ring assembly is mounted at the free end of the jackshaft and is driven from it. A tachometer is, in turn, driven from the slip ring assembly. This layout necessitates the passage of the thermocouple leads through the coupling and on through the hollow jackshaft. To enable the test shaft to be removed from the rig for either a bearing change or inspection, the wires must be disconnected. This capability has been achieved by supporting the wires in a special 1/2" diameter probe extending from the slip ring assembly through the jackshaft and coupling which has a twelve pin connector mounted on its forward end.

The mating component of the twelve-pin connector is carried on a short extension tube, itself supported in thermal insulating material. The extension tube keeps the connector well clear of the high-temperature shaft to minimize oxidation problems or deterioration of the insulating material. This configuration is shown in Enclosure 15.

The probe is constrained only at its two ends so that its own flexure will accommodate the small degree of axial misalignment tolerated by the gear coupling.

D.3 Mass Spectrometer

A model 12-101A spectrometer operating on the time-of-flight principle and supplied by the Bendix Corporation of Cincinnati is used to measure the concentration of helium in the gas sample taken. This is done by manual readout of the selected peaks by the electrometer circuit. Enclosure 16 shows a schematic of the reading procedure.

D.4 Infrared Pyrometer

For this program an infrared pyrometer is used to monitor the temperature of the test bearing cage. Optical access to the cage is provided by a hole drilled through the test rig housing at such an angle as to make observation of a section of the cage face possible. A nine inch long half inch O.D. stainless steel tube is inserted in this hole. The outboard end of the tube is capped with a windowed plug through which the infra-red pryometer is sighted. The plug is manufactured from borosilicate glass (Pyrex) which is partially transparent to infra-red light.

D.5 Search Coil

The search coil is a unique bearing instrumentation concept used for the first time in high speed testing on this program. The concept was originally tested in simplified form by Hirano and Tanoue (References 4 and 5) as discussed in detail in Appendix III.

Two different search coil systems were tested on this program. The first, used in tests 1 through 5, is referred to in this report as a single coil or "single element" type. The second, a more sophisticated system used only during test 7, is referred to as a "dual differential coil system". No coil was used during test 6 due to a malfunction of the system.

The simpler single element coil device consists basically of a single continuously wound induction coil, of diameter approximately the same as the cage, placed adjacent to the side of the bearing and co-axial with it. A single ball in the bearing is permanently magnetized (saturated). As the ball rolls in the bearing with its dipolar axis in any orientation other than coincident with the ball's rotational axis, a periodic voltage is induced in the coil. The frequency of the induced voltage is proportional to the ball's rotational velocity. The signal amplitude is a function of the rate of change of the magnetic flux cutting the coil and, in the single element coil, this rate of change is a complex function of the relative attitudes of the ball's rotational and magnetic axes and the coil attitude as well as the ball's rotational velocity.

The coil output voltage may be monitored on an oscilloscope during the test and recorded on various devices. In this program the coil signal was recorded on magnetic tape during most of the tests and was run off on a writing oscillograph from the tape after the test. Analysis of these traces, as taken from the single element coil, will show the occurrence of changes in the ball's rotational velocity or in the orientation of the rotational axis and thus identify the occurrence of such phenomena as abrupt ball skidding.

The single element coil, however, is not sufficiently sophisticated to allow a determination of the relationship between changes in the ball's rotational velocity and changes in its rotational axis in any particular event within the bearing which generates a change in amplitude of the signal (although the appearance of the trace envelope can be surprisingly revealing in this regard). This problem is resolved by the dual differential coil system described in Appendix III.

The dual differential coil system consists of two concentric differential coils. Each differential coil consists of two thin cylindrical "half-coils" with an equal number of windings wound in opposite senses, and placed concentrically within each other, leaving a ring-shaped gap between them. When such a "differential" coil is placed in a moving magnetic field a voltage is induced in the coil only by that portion of the magnetic flux which penetrates the space between the two half coils since the magnetic flux inboard of the inner half coil and outboard of the outer half will induce opposite voltages which cancel each other. When two such differential coils are placed adjacent to a bearing, with each coil in a different known position with respect to the path traveled by the ball set, the position of the axis of rotation of a magnetized ball in the bearing may be determined by a comparison of the output signals from the two coils. SKF has generated a mathematical model of the dual differential coil system which enables the computation of ball rotational axis position and rotational velocity from the two differential coil output signals recorded during testing. Attendant to this analysis was a static mapping of the magnetic flux field surrounding the bearing for given ball magnetic axis orientations.

As noted earlier the dual differential coil system was used only during test 7 of this program. Detailed analysis of the dual coil data is not part of the present program. It is expected, however, that these results will become available at a later date in conunction with work presently in process for another Government sponsor.

E. Test Elements

E.1 Test Bearing

The bearing used for all of the testing on the NAS3-14320 program is SKF part number 459981G-1 and is shown in Enclosure 17. Its design features make it typical of the mainshaft thrust bearings currently in use, either singly or in tandem, in modern turbojet aircraft engines.

The bearing is basically a 125 mm bore, 190 mm O.D. split inner ring ball bearing with "black oxide" coated CVM M50 rings and balls, 21-.8225" ball complement, ABEC7 tolerances, an AISI 4340 steel one piece fully machined and balanced outer land riding cage silver plated .001" to .002" thick on all surfaces, inner ring split face lubricant feed slots, and puller lips on both the inner and outer rings. The bearing incorporates geometry and surface finish controls on functional surfaces that are consistent with the best available aircraft practice.

The split inner race design avoids the ball complement limitations of a conrad assembly and permits a greater number of balls to be used thereby increasing the thrust carrying ability of the bearing. The split inner design also permits the use of a one piece cage giving increased strength and land riding surface geometry control.

The cage for the 459981G-1 bearing, shown in Enclosure 18, utilizes ball retention only at the cage bore. This feature allows removal of the balls from the cage pockets for inspection of replacement during a test series. Cage pocket clearance, at 0.019" - .032", is consistent with current aircraft practice in applications with predominant thrust loading such as this one. Diametral cage running clearance with the outer ring land, at .030" - .040", was increased over that used with the 125 mm bearings in the previous program NAS3-6267. This was necessary to allow for the increased closure of the cage to outer ring land gap due to cage growth as a result of centrifugal force at the 24,000 rpm targeted test speed. The cage silver plating used in these bearings was applied by the electro-plate method since the ion deposition plating used in the previous program had not shown any significant advantages.

The consumable electrode vacuum melted M50 tool steel material used for the balls and rings of the test bearing was chosen as being consistent with the contract requirement that the bearing be designed for a maximum outer race temperature of 700°F in view of the targeted maximum operating ring temperature of approximately 515°F. M50 maintains adequate hot hardness to 700°F and furthermore gives excellent life. The fatigue life capability of the material in use was a matter of some importance to the program, although not a direct test item in itself, because of the concern that a test not be "lost" through a classical fatigue failure of the bearing rather than through lubricant related effects. M50 additionally presented the advantage of being the best known of the high temperature bearing steels as far as its surface degradation effects under various running conditions go, thereby potentially aiding in lubricant performance analysis. Finally, M50 steel is the current material of choice for the most severe modern aircraft engine applications and thus its use in the test program creates gains in regard to the relevancy of program results to current applications.

E.2 Test Seals

As with its predecessor program, the test rig used for program NAS3-14320 contains a shaft seal system designed to isolate the test bearing cavity, at 6 psig and roughly 500°F, from a 1200°F/106 psig hot air environment. Because of the severity of this combination of high pressure and temperature a dual seal arrangement was used with a first or "air" seal acting as a barrier against the high temperature, but operating at a small pressure differential, and a second or "oil" seal (actually both an oil and gas seal) providing sealing against high pressure but operating at a much lower temperature differential. This seal arrangement is shown in Enclosure 4.

The two seals define three "chambers" within the rig. These are referred to in this report as the "hot air chamber", the "inter-seal cavity" and the "test bearing chamber". Nominal targeted pressurization levels for these chambers during the NAS3-14320 program were 106, 111, 6 psig respectively. Holding the inter-seal cavity pressure at 5 psi above that of the hot air chamber provided an effective method of keeping hot air from leaking into the inter-seal cavity where the oil seal would be exposed to it.

E.2.a Koppers "Air" Seal

The "air" seal and mating ring used for NAS3-14320 are essentially the same basic designs as were used (in various configurations) successfully during the previous NAS3-6267 program. They were designed and manufactured by Koppers Company, Inc. of Baltimore, Maryland. Three "air" seal assembly (includes the mating ring) part numbers, representing slight configuration differences, were associated with the present program. These are part numbers 700397, 700405 and 700495. The seal is basically a rubbing carbon nose face seal with bellows type secondary sealing. It utilizes an Inconel-X housing, Inconel-718 bellows and 56HT carbon nosepiece. The mating ring used with the "air" seal is of Inconel-X with chromium or chromium carbide (both were tested) flame sprayed plating on the operating face. The carbon nosepiee face is machined with the sealing dam located such that the bellows is pressure balanced, that is such that no moment exists on the axial cross-section of the nosepiece due to the pressure differential across it. Carbon wear pads are provided both board and outboard of the seal dam to distribute the bellows in-face load and residual gas pressure unbalance over a large contact area. Both the inboard and outboard wear pads are grooved radially to prevent any pressure buildup between them and the seal dam. Typical photographs of the carbon nosepiece face and of the mating ring are shown in Enclosures 32 and 36, and an assembly drawing of a typical "air" seal is shown in Enclosure 19.

One of the Koppers "air" seal subassemblies, 700495 serial number 1, was used on the program. This had previously been refurbished by the Koppers Company and then had seen approximately 7 hours of service in the previous program NAS3-6267 before its use in NAS3-14320. It was in excellent condition, however, for the start of NAS3-14320. Three Koppers "air" seal mating rings were used in this program. All of these were newly replated prior to use.

E.2.b "Oil" Seals

Two types of "oil" seals were used. The first seal used in the oil side position for the first five of the seven tests run on NAS3-14320 was again a Koppers face seal design, this time, however, incorporating hydrodynamic life "pads" in the mating ring face and utilizing a piston ring secondary seal rather than a bellows. It was designed and manufactured by Koppers Company under their part number 101056B and is shown in Enclosure 13 and 45. This particular seal was chosen because it had given better long term performance in the oil-side position (through at generally lower speeds than those of NAS3-14320) on the previous program than had the bellows type seal.

Features of the part number 101056B seal include a 405 stainless steel housing, a USG 2777 carbon secondary piston ring, and a CDJ-83 carbon nosepiece. The mating ring is manufactured from AMS 6322 and was flame spray plated with tungsten carbide for the early tests of the program and chromium carbide for the later ones. The carbon nosepiece has a sealing dam and inboard and outboard wear pads machined into its face. Only the outboard pad, however, is grooved for pressure relief. The inboard wear pad is ungrooved and serves to react the pressure generated by the lift pads which are machined into the mating ring face.

Two P/N 101056B oil-side seals were used in this program, S/N 1 and S/N 2. S/N 1 had been newly refurbished by the Koppers Company. S/N 2 had also been refurbished by Kopper but had subsequently seen 7 hours of service in an unrealted NASA test program (NAS3-14310) prior to use on NAS3-14320.

The second "oil" seal used was the NASA self-acting lift pad seal. This seal, in the configuration in which it was tested on NAS3-14320, is designated as NASA part number CC 850685. An installation drawing showing the seal assembly and its associated mounting components installed in the rig as used in the present program is shown in Enclosure 14.

The seal utilizes a carbon (CDJ-83) nosepiece with twenty Rayleigh step lift pads machined into its operating face immediately inboard of a narrow seal dam. The nosepiece is spring loaded against an AISI 4340H mating ring which is flame spray plated with .004" - .008" thick chromium carbide. The mating ring is supported radially at the center (only) of its bore on a AISI 4340 mounting sleeve to which it is keyed to prevent rotation. The central support point for the mating ring serves to decrease its distortion due to thermal gradients in the sleeve. The mating ring is held in position axially against a shoulder in the sleeve by a machined Inconel X-750 bellows spacer which provides a known clamping force on the mating ring. The mating ring is cooled by the test bearing under-race cooling oil flow which, after exiting the under-race cooling sleeve under the bearing bore, flows by centrifugal force successively through holes in the mating ring mounting sleeve to and through holes in the mating ring itself and exits into the test bearing chamber as shown in Enclosure 14. The seal is provided with a threaded "windback" which is supported by the nosepiece retainer and is held adjacent to the mating ring O.D. in order to prevent the migration of oil onto the nosepiece operating face. Secondary sealing is provided by a piston ring operating in a circumferential groove in the main seal housing. A narrow nose on the piston ring O.D. provides sealing against the bore of the seal carrier.

With reference to Enclosure 14, the following seal parts were manufactured for this program by the Stein Seal Company of Philadelphia: 2 mating rings under NASA part number CD849674, 1 seal carrier CF848839, 1 seal housing CF850687, 1 set of springs CB850686, 3 lugs 850701. The remaining seal parts were supplied directly by the NASA Lewis Research Center. Stein Seal Company additionally manufactured the following parts required to mount the seal in the SKF rig: 1 spacer CC850689, 1 mating ring mounting sleeve CD850688 and two keys under PWA part number SKZ-71962. The bellows, PWA part number SKZ-71964, was supplied by NASA.

The basic design for this seal was generated by NASA based on the results of Phase I of contract NAS3-7609 conducted by Pratt and Whitney Aircraft Company of East Hartford, Connecticut. In the subsequent Phases II and III of NAS3-7609 (reported respectively in references 6 and 7) Pratt and Whitney undertook to evaluate the capability of various versions of the seal (along with other candidate designs) to perform in environmental conditions whose severity exceeded those established as the limits for the positive-contact or sliding type mainshaft seals currently in use. These limits are generally taken to be a pressure differential of 130 psi, a gas temperature of 800°F, and a sliding speed of 350 ft/sec (or roughly 12,000 rpm with this seal which is equivalent to 1.5 x 10^6 DN on a 125 mm bore bearing).

In the Phase II and III testing Pratt and Whitney demonstrated successful operation of certain versions of the seal for respectable durations and at low leakage levels at operating conditions far exceeding those mentioned above. Specifically they were able to reach pressure differentials up to 300 psi, gas temperatures of up to 1200°F and seal sliding speeds up to 500 ft/sec (17,300 rpm or 2.13 x 10^6 DN on the 125 mm bore bearing).

The existence of contract NAS3-14320 at SKF, a contract conceived primarily to test lubricants but utilizing a rig simulating mainshaft conditions and requiring ultra-high speed mainshaft seals. Thus a single seal and two mating rings were provided for use on this program with the plan that they would be used as back-up test hardware and would be tested only if needed due to failure of the two existing Koppers seals which had been allocated to the program for use in the oil-side position. As it occurred, and as related under the test results section of this report, failure of both of the Koppers "oil" seals necessitated use of the NASA seal in the last two tests (Nos. 6 and 7) of the program.

The seal used in the present program differed in certain critical areas from that which had been successfully run by PWA. In particular a mating ring of 3/4" width was used. PWA testing with the 3/4" mating ring had been unsuccessful due to ring face waviness induced by installation forces, temperature differentials and other factors. These problems were resolved in the PWA program by, among other things, going to 1" and 1 1/8" wide mating rings.

E.3 Lubricants

The selection of test lubricants for this program was made jointly by SKF and the NASA Lewis Research Center on the basis of properties data from the suppliers, published performance if such was available and experience under related conditions with analogous materials. Temperature vs. viscosity and vapor pressure data for the selected fluids are given in Enclosures 20 and 21 respectively. Such other properties data as were available from the suppliers may be found in Enclosure 22.

1. FN 3158

This is supplied by the Esso Company, USA. It is a super-refined naphthenic hydrocarbon, has a very high degree of thermal stability and contains an additive package to provide oxidation inhibition and improved anti-wear capabilities.

2. MCS 2931

This material is supplied by the Monsanto Chemical Co. and is an improved modification of their MCS 293 which in turn is described by the supplier as in "a chemical class related to the polyphenyl ethers". Its use is recommended in applications where greater oxidative, thermal or radiation stability is required than can be offered by existing conventional lubricants and where better low-temperature properties are needed than those provided by the polyphenyl ethers." It has the lowest vapor pressure of any of the four fluids examined in this project. This is apparent from the curve of vapor pressure vs. temperature presented in Enclosure 21. The material has been designed for jet engine lubrication and in testing to date was shown to be a capable gear lubricant.

3. DN 600, Type 2

This is a synthetic hydrocarbon provided by the Continental Oil Company. It is made by polymerizing alkyl hydrocarbons onto an aromatic nucleus and its base stock again possesses a high degree of thermal stability, a good viscosity index, and good susceptibility to additives. It may be noted here that the type used has an additive package intended for use as a hydraulic fluid (e.g. for automatic transmissions).

4. Turbine Oil 555

This fluid, a type II ester utilizing a new proprietary additive package, is supplied by the Shell Oil Co. and is essentially an improved candidate for MIL-L-23699. It has the highest viscosity index of the four materials examined, but also has the highest vapor pressure in the 300 to 500°F range, which could be a cause of excessive evaporation during operation. The plot of vapor pressure data presented for this fluid in Enclosure 21 shows a discontinuity in the vapor pressure curve which corresponds to an isoteniscope decomposition temperature of about 570°. Thus, its thermal stability measured in this fashion will be some 50 to 100°F lower than the other candidates in the project.

F. Test Results

A detailed description of the test procedures used is given in the contract work statement presented in Appendix I. A tabulation of the test elements (the lubricant, test bearing and seals) used in each test is given in Enclosure 23, and a presentation of the test results is given in Enclosure 24. Lubricant chemical analysis results for all of the testing are presented in Enclosure 25. Finally, the test data recorded during each test are presented in Enclosure 26. A recording of these latter data was made roughly every half hour during the screening tests and every hour during the extended duration tests as well as at any point during testing that any significant change occurred in the test conditions.

The rig and sump heaters were used for partial rig warm up prior to starting each test. Full test bearing and lubricant temperatures (targeted 500° ± 15°F outer ring temperature and 400°F ± 10°F lubricant inlet temperature) were generally reached, however, by running the rig at lower speeds until test temperatures were reached. This procedure plus low speed leak testing and checkout of other systems resulted in a good deal of time spent in the lower speed ranges prior to reaching full test conditions.

It should be noted that this "Test Results" section deals primarily with the conditions under which the tests were run, the events before, during, and after them and their results. A minimum of conclusions and cause/effect relationship evaluations are generated in this section since these are covered in the subsequent section "Discussion of Results".

F.1 Screening Test Results

F.1.a. Screening Test Results Using Humble FN3158 Blended with 5% Kendall 0839 Resin (Test No. 1)

In this first test of the program a total of 13.6 hours of running time were attained. The test was terminated normally after completing all of the test objectives. Rig speed reached 23,600 rpm (2.95×10^6 DN), this being the maximum speed obtainable with the rig drive motor speed control turned wide open.

Pre-Test Setup:

As noted in Enclosure 23 both the Koppers air and oil seals in use for this test had been used previously on another program, though for only seven hours. Both appeared to be in excellent condition at the start of the test. Both mating rings had been newly replated prior to the test with chromium on the air seal ring and tungsten carbide on the oil seal ring.

A new test bearing was installed for the test. The under-race cooling system was not fully functional since the Koppers oil seal mating ring had not been slotted to allow full flow in the system. With the system in this configuration flow was provided only to the test bearing split face slots (the mating ring was later slotted after phase a of Test No. 2).

Test Summary:

During the initial stages of the test a problem was encountered with the rig flat drive belt attempting to ride off the drive pulleys. This was eventually resolved by "crowning" the pulley but resulted in extended running at speeds below 20,000 rpm before the repair.

As inspection of the data given in Enclosure 26 shows: total seal leakage increased from about 8.0 scfm to as high as 27.2 scfm in the final hours of the test ; and fluctuations in test bearing cavity pressure to as high as 20 psig (nominal is 6 psig) were noted indicating seal degradation which was verified on subsequent disassembly as discussed below.

As might be expected, the differential temperature between the oil inlet and bearing outer ring temperatures increased with speed (at any given constant set of conditions of oil flow rate, heat exchanger water flowrate and rig heater settings). In order to maintain the 100°F differential temperature (implicit in the individual limits) between these two parameters as speed was increased it was necessary to make adjustements in the oil flow rate, heat exchanger water flowrate and/or the rig heater settings. Eventually, however, these parameters reach their adjustement limits and at the highest speeds it was therefore found necessary to live with a differential temperature higher than the implicitly specified 100°F nominal. In this event the procedure of using a lower oil inlet temperature was chosen rather than to allow ring temperature to rise above its maximum limit. Thus Enclosure 26 indicates some values of oil inlet temperature below the required 400°F $\pm$ 10°F band. As later testing would show this situation also existed with all of the other oils.

The single element search coil was in use during this test for functional checkout and calibration. Its output signal was monitored on the oscilloscope, however no permanent data recordings were made and no conclusion was drawn as to bearing or lubricant performance during the test. Coil operation proved to be satisfactory.

Bearing Inspection:

The test bearing was found to be in generally good condition with no evidence of lubrication related surface distress on the raceway or ball surfaces. Some debris denting was noted in the load paths on the raceways and on the balls. The cage showed normal operational wear markings in the silver plate in the pockets and on the O.D. land riding surfaces.

Seal Inspection:

Measurements taken on the Koppers "air" seal, serial number 1, indicated an average wear of the primary carbon nosepiece face of .013". Inspection of the air seal mating ring, serial number 3, showed evidence of a slight amount of wear on the chromium plated operating face. A very slight degree of plate chipping was noted in a few isolated spots in the worn areas. The heavy wear noted on the carbon nosepiece face was attributed to its riding over the relatively rough worn areas of the mating ring. Other than for the nosepiece and mating ring wear the air seal was in good condition.

Inspection of the Koppers "oil" seal, serial number 2, showed very heavy carbon nosepiece wear (material CDJ-83), deep scoring of the operating face of the serial number 1 mating ring, and a broken secondary carbon piston ring (material USG 277). The nosepiece was worn an average of .020". The mating ring wear was predominant on one side of the ring for a run of approximately 180° around it suggesting that the ring had been cocked on the shaft. The wear measured a maximum of .013" deep in the outside wear pad rub area, .006" in the seal dam area and .005" in the inside wear pad area, of course completely removing the tungsten carbide plating, originally .0025" thick, in these areas.

Investigation of the "oil" seal failure, including fit checks of the shaft stack-up hardware, was conducted at this point but failed to reveal the cause of the eccentric wear pattern on the oil seal mating ring face. The actual cause of this condition was eventually traced, after test 2a, to a condition of insufficient radial piloting of one end of the main seal clamping sleeve which axially clamps the shaft stack-up including both mating rings. This is discussed in more detail under test 2 (Results section), however it is pertinant to point out that the failure of the Koppers "oil" seal and the degradation of the Koppers "air" seal during test 1 are both felt to be attributable to this problem.

<u>Oil Analysis:</u>

Chemical analysis of the lubricant from test 1 showed a roughly average degree of chemical change (compared to the other oils tested on this program). Viscosity increased by 28%, acid number by 50%, and dirt content by 150% during this time. The dirt content increase, however, is believed to be due in part to the heavy wear experienced by the oil seal. Debris from the nosepiece and mating ring could migrate, with the help of centrifugal force, over the mating ring O.D., past the windback, and into the test bearing chamber. This also would explain the relatively heavy debris denting noted in the test bearing.

Rig disassembly after the test indicated very heavy coking of the rig interior. This was rated "severe" as noted in Enclosure 25 for comparison with the other oils run on the program.

Photographs illustrating the post test condition of the test elements and rig components are given in Enclosures 32 through 34.

F.1.b. Screening Test Results using Monsanto MCS-2931 (Test No. 2)

This test was conducted in two phases. Rig modifications and hardware changes were made at the 12 hour point. To facilitate data reference, the parts of this test conducted before and after the 12 hour point are referred to, here and in the data presentations, as phases 2a and 2b respectively. Phase 2a was terminated at 21,900 rpm with an oil seal failure; phase 2b ended at 18,000 rpm with a smearing failure of the test bearing.

Pre-Test Setup:

Phase 2a was conducted using the same air seal as had been used in test 1 but with a different air seal mating ring, serial number 1, replacing the moderately worn test 1 mating ring. The new mating ring utilized chromium carbide flame sprayed plating .0035" thick on the operating face as opposed to the chromium used in test 1.

The damaged oil seal and mating ring used in test 1 were both replaced for phase 2a with newly refurbished parts of the same design. These were oil seal S/N 1 and mating ring S/N 2. As in test 1 the mating ring was plated with tungsten carbide.

A new test bearing was installed for phase a. The single coil search coil was used during the test and, as in test 1, was monitored only on the oscilloscope. As in test 1, the under-race cooling system was not fully functional for phase a of test 2 and provided oil flow only to the test bearing split face slots.

Test Summary (Phase 2a):

As shown by the Enclosure 26 data, speed was successively increased through 20,000 rpm, over the first 12 hours of testing with an attendant general rise in total seal leakage from 8 scfm to nearly 20 scfm. After one hour at 20,000 rpm leakage rose to approximately 25 scfm. On a subsequent attempt to increase speed to 22,000 rpm an oil seal failure occurred at 21,900 rpm and initiated an automatic rig shutdown due to depletion of air supply pressure to the rig. Total leakage subsequent to the failure was in excess of 40 scfm with test bearing chamber pressure rising from its normal 6 psig to about 11 psig. No problems related to the test bearing or lubricant were indicated at any point in the testing.

Seal Inspection and Seal and Rig Modification Pre-Phase 2b:

Tear-down showed the "air" seal to be in good condition. Measurement showed that no significant wear had taken place on the air seal nosepiece. The "air" seal mating ring also showed evidence of very slight wear.

The "oil" seal and its mating ring, however, showed evidence of a rubbing failure very similar to that which had occurred during test 1. As in test 1 the mating ring face showed an eccentric wear pattern, that is with the wear heavier on one side of the ring than on the other. The mating ring face was scored over approximately a 270° arc to a maximum depth of .0095" in the outer wear pad rub area and .0080" in the inner wear pad rub area. Measurement showed the carbon nosepiece to have worn .041" from its pre-test condition. The secondary carbon piston ring was found to be broken in one place.

During the investigation of the seal failure, re-evaluation of the mating ring clamping sleeve showed that it had insufficient radial piloting on the shaft on one end. This sleeve bolts to the end of the shaft and provides axial clamping of the components involved in the shaft stackup, including the two shaft seal mating rings. Because of this condition uneven tightening of the six bolts which secure the sleeve to the shaft was found to result in cocking of the sleeve and both the air and oil seal mating rings. The clamping sleeve was therefore modified as shown in Enclosure 27 to give the needed radial piloting. The six clamping sleeve holes were also redrilled and tapped to eliminate an uneven thread drag condition with the original holes which could contribute to uneven tightening.

The failed "oil" seal and mating ring were replaced with Koppers seal serial number 2 and mating ring serial number 1, both of which had failed during test 1 and had since been refurbished by the Stein Seal Company. During refurbishment a number of changes were made in these parts at NASA's request. The mating ring plating was changed to chromium carbide from the original tungsten carbide due to evidence of a tungsten carbide to carbon incompatibility found on another NASA program. Also the depth of the lift pads in the mating ring face was increased to .0075" from the original .0035" nominal as run during test no. 1.

Also at this time the Koppers serial number 1 "oil" seal mating ring was modified to make the under-race oil cooling system fully functional. Six slots were added to the face of the mating ring which bears on the test bearing inner ring side face. These holes opened this previously dead-ended leg of the under race cooling system to allow oil from the under-race sleeve to flow, aided by centrifugal force, into the test bearing chamber and on to the oil sump.

Also prior to phase 2b a tape recorder was added to the search coil system in order to allow the recording of the coil signal on magnetic tape during testing and thereby permit post-test oscillogram production and analysis.

During installation of the "oil" seal mating ring a special procedure was developed to allow an approximation of a runout check on the mating ring face. This was undertaken after the previous seal failures were attributed to excessive runout. The word approximation is used here because with the air seal installed it is impossible to gain access to the oil seal mating ring in order to directly measure its axial runout. Experimentation showed, however, that. although axial runout levels as high as 0.002" could still be developed (in spite of the modification just completed) by varying the sequence used in tightening the main shaft champing sleeve bolts, this runout was always nearly equal and in the same angular orientation on both the air and oil seal shoulders. Good correlation was thus established between the air and oil seal mating rings, and it was decided that the air mating ring runout might be used as a guide to the runout of the "oil" seal after final assembly.

Through a trial and error process in tightening the mainshaft clamping sleeve bolts, "air"(and therefore "oil") seal runout was brought down to 0.0005" prior to phase 2b testing.

Test Summary (Phase 2b):

After initial warm-up for phase 2b, rig speed was taken directly up to 14,000 rpm and from there to 18,000 rpm (2.25×10^6 DN) in 2,000 rpm increments. No problems were encountered through the 18,000 rpm point and data was obtained at 14,000, 16,000 and 18,000 rpm at test bearing oil flow rates between 1.0 and 2.0 gpm as shown in Enclosure 26. After 0.7 hours at 18,000 rpm a test bearing smearing failure occurred approximately 6 minutes after a reduction in test bearing oil flow rate from 1.5 gpm to 1.0 gpm (although the bearing had run successfully under these conditions in phase 2a). No unusual bearing temperature excursions had been noted prior to the failure. A total of 4.3 hours of additional testing were attained in phase 2b prior to the failure for a total 16.3 hours time for test 2.

As in phase 2a total seal leakage rose slightly with speed during the test (a normal characteristic for the seal). However, the maximum total leakage rate was 9.4 scfm, less than half that of phase 2a, indicating a likelihood that the shaft clamp modifications and the oil seal mating ring plating and lift pad depth changes had improved the seal's ability to survive at high speed.

After initiation of the bearing failure a fire was detected in the test bearing chamber. This was extinguished by depressurizing the chamber and purging it with nitrogen gas. On subsequent disassembly the rig shaft and chamber walls were found coated with heavy carbonaceous deposits as a result of the fire. Ignition of the fire was attributed to the high temperatures generated by the bearing failure.

Search Coil Results:

Oscillograms prepared from the search coil tape records, presented in Enclosures 2, 4, 5 and 6 of Appendix IV, show that a number of independent high amplitude excursions occurred in the search coil signal prior to the actual bearing failure. As discussed in more detail in Appendix III, a very brief and violent change in the location of the magnetic axis of the ball took place at these points. It is considered that these high frequency perturbations are due to the occurrence of micro-smearing (skid-damage) between the ball and raceway. These contacts generate forces on the ball which result in drastic short-term changes in ball velocity. It can be conjectured that this condition proceeds characteristically to macrosmearing failure such as was noted in this test.

As shown on trace 23 presented in Enclosure IV-5 of Appendix IV a much lower frequency, nearly periodic modulation of the envelope of the search coil output signal commenced at 0.88 sec. prior to the bearing failure and continued through to the end of the test. It is believed that this phenomenon was due to a relatively small new component of ball angular velocity induced by the repeated ball roll-over of the skid damage initiated by the earlier high frequency perturbations.

Analysis of the search coil data thus supports the conclusion that the failure was lubrication-related with attendant negative implications for the Monsanto MCS-2931 lubricant.

Bearing Inspection:

Rig disassembly confirmed the existence of a smearing type failure of the bearing with the raceway and ball surfaces showing evidence of gross skidding.

Seal Inspection:

Inspection of both the "air" and "oil" seals and both mating rings after phase 2b showed them all to be in excellent condition. Neither mating ring showed evidence of wear. Measurements indicated an average 0.001" wear on the "air" seal carbon nosepiece and 0.002" wear on the "oil" seal nosepiece over their pre-phase b conditions. Since both mating rings had been newly replated prior to phase b and the oil seal nosepiece itself had been new, these wear figures are not felt to be excessive and seal performance was judged to be acceptable.

Oil Analysis:

Chemical analysis of the Monsanto MCS-2931 oil used through both phases of test 2 is presented in Enclosure 25. Increases of 10% in viscosity and 46% in dirt content, were noted with no noticeable change in acid number. These levels of chemical change are relatively small, particularly in view of the fact that the entire rig oil supply made roughly one full cycle through the test bearing while temperatures were abnormally high due to the bearing failure and the subsequent fire. In this regard bearing outer ring temperature peaked at 665°F and was above normal limits for roughly 8 minutes with oil flow held at 1.0 gpm. It is believed that the relatively large increase of 46% in dirt content is due primarily to particle contamination from the failed bearing and from the products of combustion generated by the fire rather than to chemical change in the oil. In general, therefore, this oil performed extremely well as far as chemical degradation alone goes.

Inspection of the rig interior for coking after test 2 also confirmed the minimal apparent level of chemical degradation of the MCS-2931. Although the soot from the fire masked the condition of the rig interior, close inspection showed coking to be about average for the oils tested in this program in spite of the high temperatures generated by the failure.

Photographs illustrating the post test condition of the test elements and rig components are given in Enclosures 35 through 37.

F.1.c Screening Test Results using Conoco DN-600 Fluid, Type 2, (Test No. 3)

This test was terminated after reaching 16,000 rpm due to premature chemical degradation of the lubricant.

Pre-Test Setup:

A new test bearing was installed for this test. However, the air and oil shaft seals and the mating rings were the same as had been used in the previous test 2. A new single element type search coil was installed to replace the one destroyed in the fire of the previous test. The coil output signal was recorded on magnetic tape as in phase b of test 2.

Test Summary:

After 4.2 hours of running time, during which speed was limited to a maximum of 16,000 rpm by drive belt problems, a coupling in the rig drive system failed and the rig was shut down. Speed at the time of the failure was 16,000 rpm. Total seal leakage had not exceeded 10.6 scfm. Subsequent routine inspection of the Conoco DN-600 lubricant revealed that it had darkened considerably. Operational data is presented in Enclosure 26.

Oil Analysis:

Chemical analysis of both used and unused samples of the fluid showed that it met specification requirements prior to the test and that it had undergone a significant degree of chemical change during the test. Results of these analyses are presented in Enclosure 25 and show a 40% increase in viscosity and a change from basic to an acid number of 0.3. Dirt content (not necessarily solid particles) was such that the fluid would not pass through the standard test filter. This large magnitude of chemical degradation under the not overly stringent conditions seen by the oil (4.2 hours, bulk oil out temperature 490°F maximum and inner ring temperature 530°F maximum) indicates severe chemical breakdown of the Conoco DN-600 Type 2 fluid and the screening test program of this oil was terminated accordingly with the agreement of the sponsor.

Conoco DN-600 Type 2 fluid was designed as a multi-purpose oil usable as an automatic transmission fluid. Testing at SKF with various brands of hydraulic fluid has generally shown chemical breakdown of the additives in this type of fluid at temperatures well below those experienced in this test and as low as 230-250°F. Discussions with Continental Oil Company indicated that the base stock for the oil should be capable of operating in the 500°F range. The present testing indicates, however, that the additive package impairs this capability.

Bearing Inspection:

Inspection of the test bearing subsequent to test 3 showed it to be in very good condition in spite of the oil degradation. No evidence of lubrication distress, skidding or other abnormal operating conditions were found. Some debris denting was present on the rolling contact surfaces and, as might be expected, the bearing and the interior of the rig were heavily "coked" by the oil.

Seal Inspection:

Inspection of the shaft seals after the test showed both still in good condition. A small amount of carbon chipping was evident at the outer edge of the seal dam on the "oil" seal. This was not considered severe enough to retire the seal from further service. Wear on the "oil" seal carbon nosepiece face during the 4.2 hours of running was measured at .0035" average. The "oil" seal mating ring face was in good condition with no wear evident.

The "air" seal nosepiece was in excellent condition with a total measured wear during test 3 of .0015" average. The nosepiece face showed a minimal amount of shallow circumferential scratch marks, a normal condition for a rubbing face seal of this type. The "air" seal mating ring remained in excellent condition with no noticeable wear.

Search Coil Results:

A minimum amount of data was obtained from the search coil during this test since the coil failed early in the test when insulation burn-through initiated loop-to-loop shorting. As pointed out in Appendix IV, none of the high frequency perturbations seen during test 2 were noted in this test prior to the coil failure. The fact that no evidence of skidding was found in the bearing post-test correlates well with this observation.

Photographs illustrating the post test condition of the test elements and rig components are given in Enclosures 38 through 40.

F.1.d Screening Test Results Using Aeroshell Turbine Oil 555 (Test No. 4)

This test reached 21,400 rpm and was terminated, without malfunction, due to rig cooling system limitations.

Pre-Test Setup:

A new test bearing was installed for this test. The air and oil shaft seals used in the previous test were used again without change. Installation was conducted, as in the previous test, using the air seal mating ring face runout as an indication of oil mating ring runout. This was again held to .0005". The single element search coil was used and the signal was recorded on magnetic tape.

Test Summary:

Speed was increased, as shown in the Enclosure 26 data sheets, through 20,000 rpm with no problems indicated other than a relatively high differential between oil inlet and outer ring temperatures.

After obtaining data at the 20,000 rpm point, an attempt was made to increase speed to 22,000 rpm while holding the test bearing oil flow rate at 2.0 gpm. It was found impossible, however, to exceed approximately 21,400 rpm without the outer ring temperature exceeding its specified upper limit of 515°F, in spite of maximum oil cooling. Test No. 4 was therefore terminated after obtaining data at the 21,400 rpm point. Total running time was 8.4 hours.

It should be noted that the ring temperature excursion which was found to originate at 21,400 rpm was not an indication of temperature instability of the bearing lubrication system but rather was merely a result of insufficient cooling capacity in the oil sump heat exchanger. If added oil cooling had been available the bearing might hawe been run to higher speeds.

The shaft seals performed well throughout the test. As reference to the data of Enclosure 26 shows, total leakage varied between 7.2 and 13.6 scfm. The oil shaft seal component of total leakage was monitored during the test by mass spectrometer analysis of the helium trace flow introduced into the inter-seal cavity. Six different checks of "oil seal" leakage were taken with this system at speeds between 14,000 and 21,500 rpm. These showed oil seal (only) leakage to run between 1.2 and 2.1 scfm with no discernible relationship to speed evident.

Bearing Inspection:

Inspection of the test bearing post-test showed no evidence of skid damage, lubrication distress or any other type of lubrication related problems on the rolling contact surfaces. A small amount of debris damage was present.

Seal Inspection:

Inspection post test showed the "air" seal to be in good condition and reuseable. Measurement showed negligible additional wear on the air seal carbon nosepiece to have occurred during this test.

"Oil" seal inspection, however, showed that the breakdown at the outboard edge of the seal dam (noted after test 3) had progressed somewhat. Wear of the "oil" seal nosepiece face during this test was measured as 0.001" average. Cummulated wear in the 16.9 hours since this "oil" seal was installed during test 2 totaled .0065" at this point. Due to this relatively high total wear rate (for the self-acting hydrodynamic design) and the seal dam breakdown, it was decided at this point to conduct a more detailed inspection of the nosepiece face in order to determine its contour.

The SKF Gsip machine was used to produce a contour map of the "oil" seal nosepiece surface. A sketched plot of these data is presented in Enclosure 28. The data show a roughly "saddle" shaped contour, a common type of nosepiece face wear pattern. The two "low points" of the saddle dip a maximum of approximately 0.0018" below the level of the two high points. The two high points were found to align with the two anti-rotation pins in the retainer adapter assembly. This was judged to be severe enough to preclude further use of the seal without rework.

The "oil" seal mating ring was also checked, in this case using an optical flat device. This showed the mating ring face to be slightly dished (concave) to a depth of approximately .000120". Additionally a small depression, approximately 0.000050" deep was noted immediately outboard of each of the "L-shaped" air feed grooves directly in the seal dam running zone. Although it did exhibit some operational degradation this was not judged severe enough to retire the mating ring from further use.

Oil Analysis:

Chemical analysis of the Aeroshell Turbine Oil 555 showed that very little chemical change had taken place. As shown by Enclosure 25 viscosity increased by only 7% while acid number decreased by 55%. No change at all was found in dirt content.

Rig disassembly also showed that coking of the test bearing and the rig interior was nearly non-existant, by far the least seen with any of the oils tested.

The noted minimal level of chemical change and coking places Aeroshell 555 at the head of the list in this regard among the other oils in the program.

Photographs illustrating the post test condition of the test elements and rig components are given in Enclosures 41 through 43.

Search Coil Results:

While at 21,400 rpm, just prior to termination of the test, a continuous long-term non-periodic oscillation had developed in the envelope of the search coil signal. This is shown in Enclosure IV-9 of Appendix IV. Inspection indicates that this modulation was different in appearance from that noted immediately prior to bearing failure during Test No. 2b. Whereas the Test No. 2 modulation was notably periodic with a distinct continuously repeating form, the test No. 4 voltage envelope shape was highly irregular and made up of random excursions in voltage. Furthermore the Test No. 4 voltage trace did not show the very rapid high amplitude spikes in voltage ("perturbations" in ball angular velocity) which were evident in the Test No. 2 data prior to the start of the periodic modulation. In line with the theory of search coil analysis related in the discussion of the search coil results for Test No.2, therefore, it is not believed that the modulation noted during Test No. 4 was caused by microsmearing or that lubrication "problems" were a factor in it. This conclusion was borne out by inspection of the raceway surface which, as previously related, showed no evidence of lubrication-related damage. The coil signal modulation may have been induced by the ball rolling over the noted debris denting but this remains unsubstantiated.

F.2 Extended Duration Testing

According to the plan of action for the NAS3-14320 program, the two oils giving the best performance during screening testing were to be given extended duration testing for up to 50 hours. Actually three rather than two such tests were run, the final one being a rerun of test 6 due to a seal failure during the latter which resulted in early termination with limited data.

F.2.a Considerations in the Choice of Lubricants for Extended Duration Testing

The two lubricants chosen for extended testing were Humble FN3158 blended with 5% Kendall 0839 resin and Aeroshell Turbine Oil 555. This choice was based on evaluation of the screening test results in the following three areas:

1. Bearing condition after testing.

2. Level of chemical degradation of the oil as indicated by chemical analysis and observed "coking" of the rig and bearing.

3. Severity of the conditions under which it was possible to make the bearing run without failure, specifically the speed and temperature reached.

In consideration of the first area above, very little evidence of lubrication distress or damage other than debris denting was found in any of the bearings except for that tested with the Monsanto MCS-2931. The latter ended with a smearing failure at 18,000 rpm. MCS-2931 was eliminated primarily because of this failure and the EHD film stability problems it indicates.

In regard to chemical degradation, the second area of consideration, chemical analysis and rig coking results are presented in Enclosure 25. Conoco DN600 fluid was eliminated based on its high level of chemical degradation in the relatively short time period noted.

In consideration of the third area above, the maximum speeds and temperatures reached are shown in Enclosure 24. The significance of some of these figures is limited by the fact that some of the tests were terminated for reasons other than legitimate bearing failure in response to operating conditions. In spite of this, however, the data supports the lubricant choices already indicated by the other two areas of consideration in that the two chosen lubricants ran very well at high speeds.

F.2.b Test Conditions Used in Extended Duration Testing

Unlike the screening tests, for which the primary objective had been to establish the operational conditions limits of each of the oils, the extended duration tests targeted a single set of fixed operating conditions at the highest speed found to be feasible during the screening tests. Both oils were tested under conditions as nearly identical as possible to facilitate comparison of the results.

Based on the performance of the Humble and Aeroshell oils during the screening tests, a speed of 20,000 rpm and a test bearing oil flow rate of 2.0 gpm were chosen as within the capability of both oils. Outer ring temperature was targeted at 500°F ± 15°F as in the screening tests and oil inlet temperature was allowed to stabilize at whatever temperature was necessary in order to maintain the outer ring temperature within its limits.

As required by the Contract Work Statement (Appendix I) an attempt was made to run the tests in 5 to 10 hour segments (although hardware failures modified this) with a rig cooldown and an oil chamical analysis conducted between each segment. A check of the "oil" seal component leakage rate using the helium tracer and mass spectrometer analysis was also conducted at least once during each segment. A full data recording was made roughly every hour.

The search coil was used during the testing. Its output was monitored continuously on the oscilloscope and the signal was recorded briefly on magnetic tape every hour as well as at any time that abnormal ball movement effects were noted.

F.2.c Results of Extended Duration Test using Aeroshell Turbine Oil 555 (Test No. 5)

This test was terminated after 26.7 hours of running time (17.4 hours of which were at the 20,000 rpm targeted test speed) due to an "oil" seal failure. A special series of oil seal leak checks were conducted during the initial start-up phase of the test in order to monitor break-in characteristics of the newly refurbished "oil" seal.

Pre-test Setup:

The test was conducted using the same test bearing, "air" seal and mating ring, and charge of oil as had been used in the preceding screening test in order to gain increased total running time on these components and on the oil.

The same Koppers "oil" seal, serial number 2, was also used. In view of the nosepiece chipping and wear noted subsequent to test 4 it was reworked by substituting the entire nosepiece retainer adapter assembly from Koppers seal serial number 1 for the damaged serial number 2 parts. The seal was cleaned and the nosepiece relapped during assembly. Inspection of the serial number 2 carbon secondary piston ring at this time showed it to be in "nearly-new" condition. Optical flat inspection of the re-lapped carbon nosepiece face showed it to be flat within 0.000012".

Axial runout of the "air" seal mating ring face during final assembly of the rig was measured to be 0.0003".

Seal Checks:

In order to establish the "break-in" characteristics of the newly refurbished Koppers "oil" seal, a special series of leak tests was conducted during the first 18 hours of this test. These included both total leakage checks and "oil" seal component leakage checks, the latter using the mass spectrometer analysis previously described. Total leakage rate was monitored on a nearly continuous basis over this period and was seen to drop rather continuously from approximately 17 scfm to the 9 scfm range in the first 5 hours. It then remained at the latter level until the end of the test. Mass spectrometer analysis commenced at about 3 hours into the test and these data are presented in Enclosure 29. This shows "oil" seal component leakage to vary from over 5 scfm early in the test to the 1.5 scfm range at 18 hours. Enclosure 29 also shows the corresponding total leakage readings taken at the same times.

During a shutdown at the 10.1 hours point (to repair a static rig seal leak) the opportunity was taken to check the total spring force of the oil seal. This indicated a total force required to depress the nosepiece to its operating position of 8 lbs. 15 1/2 oz. The "spring constant" was found to be 29.3 lbs per inch. These figures were the same as upon initial installation and indicated essentially no change due to operation of the seal up to that point. A check was also made of carbon nosepiece wear at this point and this showed an average wear of 0.0035" for the 10.1 hours since refurbishment of the seal. This level of wear was considered quite heavy but had been expected as part of the nosepiece wear-in process. The appearance of the face of the nosepiece was good with no indications of seal dam or wear pad breakdown. The mating ring showed no wear at this point.

Results indicate that non-contacting operation of the seal was not achieved.

Test Summary:

Testing proceeded at 20,000 rpm as noted in Enclosure 26 after approximately 6 hours at lower speeds during seal break-in and leak testing. No particular problems were noted relating to the performance of the bearing, shaft seals or the lubricant.

During a normal end-of-the-day shutdown, after 26.7 hours, an "oil" shaft seal failure was experienced. This was evidenced by total leakage readings in the 30 scfm range and an interseal cavity pressure of 9 psig observed after the shutdown. No indication of the leak had existed during a routine data recording taken 8 minutes prior to shutdown and it is believed that the failure was initiated by nosepiece dynamics occurring as a result of the shutdown transient itself.

Seal Inspection:

Inspection of the "oil" seal post test showed the outboard wear pad to be severely worn. This wear had taken the form of superficial pitting or surface spalling of the carbon face of the wear pad. Microscopic examination of the surface showed minute pieces of carbon broken out of the surface. Inspection of the inboard wear pad and the seal dam, however, showed these areas of the nosepiece face to be in relatively good condition and worn no further than they had been at the 10.1 hour point. The wear on the outer wear pad was minimal in the two areas of the face lying immediately above the two anti-rotation pins (lying 180° apart), and was at a maximum in the areas between the pins.

Inspection of the "oil" seal matng ring face after the test showed a definite but shallow wear pattern in the chromium carbide surface. The ring face also exhibited numerous minute hairline radial cracks in the plating. The density of these cracks was found to increase along a radial path traveling outboard and they were most dense outboard of the outer wear pad riding area. Both the wear on the plating and the density of the cracks were somewhat greater on one side of the mating ring than on the other.

It is believed that the shutdown transient may have set up a rocking motion of the nosepiece retainer adapter assembly on the anti-rotation pins. This could have resulted in repeated carbon to mating ring contact along the periphery of the seal (that is on the outer wear pad) in the zones between the anti-rotation pins.

An out of square condition of the face of the mating ring with respect to the shaft centerline may have played a role in setting up the above rocking motion. The fact that the mating ring plating wear and cracking was heaviest on one side of the ring indicates the probable existence of such an out of square condition although no post-disassembly evidence was found that such a condition did exist and the axial runout of the air seal mating ring face during pre-test assembly had been a relatively low 0.0003".

The "oil" seal secondary piston ring was found to be in excellent condition with its sealing nose intact.

"Air" seal inspection after the test showed both the seal and the mating ring to be in excellent condition and unchanged from their pre-test condition. Measurement showed no "air" seal nosepiece wear during this test.

<u>Bearing Inspection:</u>

Inspection of the test bearing after the test showed no evidence of lubrication related problems. Cage land riding surface and pocket wear were found to be at a minimum.

Oil Analysis:

Chemical analysis of the Aeroshell Turbine Oil 555 lubricant was conducted at intervals during the test and at its completion. These data are shown in Enclosure 25. The analysis data show a very minimal level of overall chemical change and indicate a significant degree of chemical "staying power". For the total 35.1 hours of running time, viscosity rose 11%, dirt content 47% and acid number decreased 47%.

The rig interior also showed only a small degree of coking except on the Koppers oil seal where it was heavier due to the extreme temperatures generated by the seal rubbing failure. What coking did exist in the rig exhibited an unusual characteristic in that it had formed a relatively poor bond with those surfaces on which it had accumulated and could be removed more easily than in the usual case. In general Aeroshell 555 maintained its previous rating as the oil least likely to coke.

Search Coil Results:

The search coil was monitored by oscilloscope during the test with no abnormal coil output voltage modulations noted (although the significance of this is minimal since only the lower frequency modulations can be noted visually in real time; high frequency perturbations of the type noted during test 2b are too rapid for visual monitor). No oscillograms of the search coil data were prepared for this test.

Photographs illustrating the post test condition of the test elements and rig components are given in Enclosures 44 through 46.

F.2.d Results of First Extended Duration Test using Exxon FN3158 Blended with 5% Kendall 0839 Resin (Test No. 6)

This was the first test in which the NASA part number CC850685 self acting "oil" seal was used. Although unknown at the time of its installation the seal's primary lift geometry and windback thread direction had been made for rotation in the opposite direction from that of the rig. For this reason a rubbing type failure of the seal occurred after only 2.15 hours of operation and the test was terminated at that point. A maximum speed of 10,800 rpm was reached.

Pre-test Setup:

During installation of the NASA seal the mating ring O.D. cooling jets were deleted by replacing the forward (seal side) test bearing lubricant jet ring. These jets, six in number, were an integral part of the original lube jet ring on the seal side of the test bearing and had been in use for all of the previous tests using the Koppers "oil" seal. Their use with the Koppers seal was necessary since its mating ring had been designed for use with jetted O.D. cooling of this type. The NASA seal mating ring, however, utilizes integral oil cooling holes making jetted cooling unnecessary.

The same Koppers "air" seal and mating ring were used in this test as had been used in the previous test 5. As with the Koppers "oil" seal, very good correlation was found for both the axial runout magnitude and angular location of its high point between the "air" and "oil" seal mating ring faces. As in previous tests this allowed the use of "air" seal mating ring runout as an indication of oil seal mating ring runout after final assembly of the rig when it was impossible to gain access to the oil seal mating ring face.

Initial attempts at installation of the NASA "oil" seal mating ring resulted in high levels of runout. For this test an interference fit was used between the bore of the mating ring and its mounting sleeve (part number CD850688 in Enclosure 14). Simple heating of the ring, installation and cool-down was found to result in the mating ring not being fully seated in the axial direction against the sleeve's shoulder causing the high runout. This was rectified by holding the ring against the sleeve shoulder with an 18 lbs. weight (vertical assembly) during cool-down. With this procedure final assembly runout readings of 0.0005" on the air seal mating ring were attained.

A new test bearing was installed and a new charge of Exxo FN3158 blended with 5% Kendall 0839 was used. This lubricant was mixed using the same lots of Exxon and Kendall fluids as had been used in test 1.

Test Summary:

In initial testing, speed was increased up to 10,800 rpm in incremental steps over a period of approximately one hour.

Total seal leakage readings varied between 8.1 and 16.6 scfm, a rather wide range. Leakage was found to be fairly stable at constant rig speeds. However, seal leakage increased drastically to over 30 scfm during each speed change.

At 10,800 rpm, after 1.05 hours of operation, lube pump out pressure became erratic, and the rig was shut down to check for an apparent lube system blockage. A check of the oil filter showed it to be contaminated with coked oil and sump disassembly showed the sump to be coked. Investigation showed that the lube pump motor circuit breaker had shut the motor off during the warm-up phase prior to testing due to the excessive current draw associated with pumping the higher than normal viscosity oil in use. This had allowed the stagnant oil to overheat and coke in the heated sump during the remainder of the warm up period. The pump was restarted prior to the test. As a result of this problem the rig was flushed with solvent and refilled with a new charge of lubricant at this point.

During preparations for restart a very high level of static total leakage was noted, approximately 30 scfm. It was concluded that the primary "oil" seal was most likely not fully seated on the mating ring and an attempt was made to reseat it by turning the rig over. This was successful and speed was increased to 7,700 rpm. Although total seal leakage was again high during the speed transient it stabilized at 8.5 scfm at 7,700 rpm at which point a data reading was taken, the last one obtained during the test. Upon initiation of a subsequent speed change total leakage increased, as before, to the 25-30 scfm area. The speed increase was stopped at 10,300 rpm. Seal leakage did not drop back to normal but rather increased drastically to the 90 scfm range while test bearing chamber pressure rose to 20 psig. The test was terminated at this point, after 2.15 hours, due to the apparent "oil" seal failure.

Seal Inspection:

As previously mentioned, investigation after the test revealed that the "oil" seal lift pads and windback thread had inadvertently been designed for shaft rotation in the opposite direction from that of the SKF rig. Inspection of the seal showed that the seal had undergone a gross rubbing type primary face failure. With the lift pads reversed the seal had acted as a rubbing face seal with abnormally high face loading. The failure was the predictable result.

Average nosepiece carbon wear of the failed "oil" seal was measured as 0.020". Mating ring plating wear averaged approximately 0.005". Additionally it was found that the mating ring was a loose fit on its mounting sleeve and measurements indicated that this was due to mating ring growth. The reason for this latter condition remains undetermined. The heat treatment used with the mating ring yields minimal levels of retained austenite ruling out growth of the part due to austenitic transformation. Additionally the bore of the mating ring showed no wear which would explain the condition.

Inspection of the "air" seal showed minimal carbon face wear averaging only 0.002". The "air" seal mating ring remained in excellent condition with no change due to test 6.

Bearing Inspection:

Post test inspection of the test bearing showed its raceway and rolling element surfaces to be intact with no evidence of spalling, skidding or lubrication distress. Cage wear was also found to be extremely mild.

Oil Analysis:

The results of chemical analysis of the oil sample after the test (1.10 hours on this charge of oil) are shown in Enclosure 25. These show an 11% increase in viscosity, no change in acid number and a 185% increase in dirt content weight. As with most of the previous tests, the high level of change in dirt content is believed to be the result of the seal failure which, in spite of the reversed windback thread sense, would probably contaminate the lubrication system with wear particles which were thrown out of the nosepiece rubbing area by centrifugal force and migrated over the mating ring O.D. and into the lubrication system.

The interior of the rig was found to be severely coked on post-test disassembly with evidence of a fire in the test bearing cavity. The fire explains a sudden rise in the bearing ring temperatures which had been noted immediately prior to the shutdown, and there is little doubt that the fire was caused by the heat generated by the seal failure. Much of the internal contamination of the rig, however, appeared to be coked oil rather than soot from the fire, and the extent of coking in the short time period run is believed to be primarily a function of the high temperature level of the rig components resulting from the continuous abnormally high frictional heat generated by the seal. (This is to be compared to the results of test 7 where use of the same oil with a seal with effective hydrodynamic lift resulted in minimal coking over a comparably short time period.)

Search Coil Results:

The search coil was not functional during this test. The coil signal was lost immediately prior to testing and was not repaired since continuity checks showed it to be open-circuited inside the rig.

Photographs illustrating the post test condition of the test elements and rig components are given in Enclosures 47 through 49.

F.2.e Results of Repeated Extended Duration Test Using Exxon FN3158 Blended with 5% Kendall 0839 Resin (Test No. 7)

Because of the early termination of test 6 which yielded a minimal amount of usable data it was decided to run a second extended test using this oil and the NASA oil seal. The seal was refurbished with a new nosepiece and windback designed for rotation consistent with the SKF rig. The newly designed dual differential type search coil system was used for the first time during this test. During preparation for the test a more sophisticated seal installation procedure, patterned after that used by Pratt and Whitney Aircraft in their work with the seal, was used. Basically, seal face waviness during the test run was minimized by assembling the seal on the shaft outside of the rig, observing the resulting waviness with an optical flat, minimizing this value by improving rig component face finishes and through to optimal angular alignment of these components, establishing those portions of the assembly procedure critical to seal face waviness, and reassembling the seal in the tester taking all these factors into account. The test ended after 6.3 hours at speeds up to 20,100 rpm, with a failure of the "oil" seal.

Pre-Test Setup:

Refurbishment of the NASA oil seal was conducted by the Stein Seal Company. Stein produced the new carbon nosepiece with reversed lift geometry and installed a new NASA supplied windback with reversed thread sense. The primary seal face load was also reduced at this time from the previous 25 lbs. to 20 lbs. total force by deleting springs symmetrically. This latter change was consistent with the results of a NASA Lewis computer study which indicated this as the best overall compromise for both high and low speed running. Also at this time Stein eliminated the interference fit between the bore of the mating ring and its mounting sleeve. A line to line fit was used in this area to eliminate any induced mating ring face waviness.

All axial bearing faces in the shaft stack-up were lapped and inspected by optical flat. Inspection of the remaining unused mating ring showed it to be flat within from 1/2 to 1 helium lightband on one side and from 1 to 1 1/2 on the other side in the free state.

Since limited access to the "oil" seal mating ring after final assembly precluded taking mating ring waviness checks after shaft installation, shaft buildup was conducted first on a trial basis outside of the rig using a special optical flat with a center hole to monitor mating ring waviness. During this process changes were made in the angular orientation of each of the shaft components in order to establish the combination of angular positions which would yield the minimum obtainable level of ring face waviness. Enclosure 30 presents the results of this study and shows that in all cases a "saddle" was found to exist on the mating ring face with its two high points in line with the two anti-rotation keyways, 180° apart, in the mating ring bore. Rotation of the components in the stack-up changed the depth of this saddle but did not eliminate it. The final relative orientation reached gave saddle low point depths of 9 and 7 light bands (.000104" and 0.000081") respectively. The shaft components were then scribed in order that their relative orientation might be maintained during final assembly. Disassembly and reassembly established that the waviness readings were repeatably obtainable.

The dual differential search coil, described in the test equipment section of this report, was installed for the first time for this test along with an oscilloscope for direct monitoring during the test and a tape recorder for permanent record. Installation of the coil involved deleting the "aft" (drive end of the rig) test bearing lube jet ring and mounting the coil on a support ring in its place. In order to maintain the same total test bearing oil flow rate as with the original dual ring system used in previous tests the jets of the remaining forward ring were drilled out from their original 0.055" diameter of 0.074".

Additional stock of Exxon FN3158 oil was contributed for the test by the Exxon Company. This oil was not of the same lot as that tested in tests number 1 and 6 since none of the earlier lot remained. It is considered, however, that this change in lot would have a negligible effect on the test results. The lot number of the Kendall 0839 resin was the same as that previously used. The same air seal and air seal mating ring as had been used in the previous test were again used in test 7. A new test bearing was installed.

Test Summary:

Speed was taken up to over 3,000 rpm immediately on all starts made during the test in order to assure "oil" seal hydrodynamic lift-off. Although seal lift-off theoretically occurs well below this point this initial test speed was chosen with an eye toward a large factor of safety.

After 4.7 hours of running time a failure of the rig roller bearing (on the drive end of the shaft) occurred at 19,000 rpm, the highest speed reached at that point. The rig was shut down for replacement of the bearing. Loss of the bearing was attributed to routine fatigue failure. No evidence was found to indicate that the lubricant played any part in the failure.

Because of the potential detrimental effect of the bearing failure on the "oil" seal through vibration and/or shaft misalignment due to increased radial looseness in the failed roller bearing the seal was inspected under low power magnification at this point. No wear or chipping was evident on either the nosepiece face or the face of the mating ring. The mating ring exhibited a small number of black streaks on its face, a normal phenomenon. The nosepiece showed only a very small number of circumferential scratches, also a normal occurrence due to minor contamination entrained in the air supply.

The rig roller bearing was replaced, the rig oil system was cleaned by circulation, filter change and oil drainage, and the oil replaced with 4 gallons of the same lot. Testing resumed and proceeded normally through 20,000 rpm.

Total seal leakage was monitored closely throughout the test, both before and after the rig roller bearing failure. Data recorded during these checks is presented in Enclosure 31. Additionally, checks of the oil seal component level leakage were conducted at 4,800 rpm and 10,100 rpm at 4.9 and 5.3 hours respectively. These checks showed 2.3 scfm and 2.7 scfm for these two test points. Total leakage readings taken at the same time gave 7.3 scfm at both speed settings. Both the "oil" seal component leakage and the total leakage readings of Enclosure 31 indicate that the seal was performing well through 20,000 rpm.

After reaching 20,100 rpm, however, total leakage was observed to slowly increase in a fairly linear manner with time. After 13 minutes at this speed total seal leakage rose abruptly to the 100 scfm area while bearing cavity pressure commenced fluctuating rapidly between 10 and 33 psig. At this point the test was terminated due to the apparent "oil" seal failure. Total test time was 6.3 hours.

<u>Seal Inspection:</u>

Inspection of the "oil" seal after the test showed that the lift pads had been completely abraded from the face of the carbon nosepiece. The carbon was not, however, broken or cracked. A comparison of measurements taken before and after testing showed an average wear of the carbon face of 0.004".

The seal mating ring face showed four highly worn areas spaced at approximately equal angular intervals around the face with narrow unworn areas between them. The keyways in the bore of the mating ring, which had been aligned with high spots of waviness during set-up, were more closely aligned with the unworn than the worn areas post test. Discussions with the Stein Seal Company indicated they had noted the occurrence of similar four-point wear patterns in the past and that this was perhaps due to vibration modes set up after the failure started. The flame sprayed chromium carbide plating on the face of the mating ring showed numerous hairline radial cracks and heat discoloration in each of the four worn areas. The cracking was most likely due to thermal transients experienced in these areas as the wear progressed. The mating ring was also, as in test 6, slightly looser on its mounting sleeve than it had been before the test.

The mating ring O. D. showed evidence of contact with the windback threads in the areas adjacent to each of the worn areas of the mating ring face. Depth measurements showed that the nosepiece had slipped axially in the nosepiece retainer (part CF848833 in Enclosure 14) in which it is held by an interference fit. This movement resulted in tilting the retainer and windback by 0.022" with respect to the nosepiece (and therefore the mating ring). The combination of the tilted windback and the expansion of the mating ring O.D. due to high temperatures explains the noted windback contact. It is significant that the diametral interference fit of the retainer on the nosepiece O.D. was decreased from 0.040" to 0.012" during manufacturing of the seal to resolve assembly problems.

It should also be noted that it is conceivable that slippage of this type could occur prior to testing through inadvertent physical impact, assembly procedures, etc., although there is no evidence that this happened in this case. With the retainer cocked on the nosepiece O.D. the result would be a complex combination of forces and moments on the nosepiece resulting undoubtedly in severe distortion of its operating face. Under these conditions, the trouble-free seal operation attained for over 6 hours is very difficult to visualize. Thus it is quite unlikely that the test ran for any length of time with the slipped nosepiece.

Disassembly of the seal also revealed the presence of a small piece of contamination of unknown origin in the area between the carbon nosepiece and the nosepiece carrier (part CF848839 in Enclosure 14). The thickness of this was estimated to be 0.001". If this foreign particle was present prior to the failure it may have caused or contributed to it by inducing nosepiece face distortion.

In regard to contamination between the nosepiece and the carrier it is noted that the existing seal design lacks shielding to prevent oil spray and/or drain-off from adjacent hardware from migrating into the open area between the nosepiece retainer and the carrier face. In the event that oil were to coke or to deposit solid foreign contamination in this area, subsequent momentary lift-off of the back of the nosepiece from the carrier due to vibration could allow the coke or other contamination to lodge in this critical seating area for the nosepiece. In the case of this particular test, however, as shown in the photograph of Enclosure 56, no coking was found in this area.

Inspection of the secondary piston ring and its bearing surfaces in the housing groove and carrier cylinder bore showed these components to be in good condition with no problems indicated.

In consultation on the failure of the seal Stein Seal Company expressed the opinion that the pre-test mating ring face waviness level of 7-9 lightbands was excessive. In their judgement 3 lightbands is considered to be the maximum safe value for this parameter. Stein's analysis was that mating ring face waviness probably initiated the failure by gradually wearing the lift pads off the nosepiece face. They noted that the axial slippage of the nosepiece retainer was most likely a secondary effect resulting from hammering of the nosepiece on the mating ring face initiated after the wear process had progressed appreciably. In regard to the Stein analysis it should be pointed out that a computer study by NASA Lewis indicates a nominal operating film thickness of 0.0003" for the seal under the conditions of this test. By comparison, 9 bands of waviness, equating to just over 0.0001" appeared to be adequate.

Bearing Inspection:

Inspection of the test bearing after the test showed all components to be in extremely good condition. The raceway and balls showed no evidence of marginal lubrication conditions. The cage showed normal operating markings.

Oil Analysis:

Chemical analysis of the Exxon FN3158 and 5% Kendall 0839 resin mixture was conducted prior to use, after the rig roller bearing failure at 4.7 hours (after which essentially all of the oil was changed), and at 6.3 hours (1.6 hours since the oil change) after test termination with the seal failure. The results of these checks are given in Enclosure 25.

Although presented for reference, the 4.7 hour sample data represents oil which was subjected to an abnormal heat input of unknown magnitude during the roller bearing failure and is not, therefore, realistically comparable to the chemical performance data of the other oils tested in the program.

Since essentially all of the oil was replaced subsequent to the roller bearing failure the 6.3 hours data actually represents oil which experienced only the final 1.6 hours of testing. It is believed that this replacement charge of oil mixed with a slight amount of residual oil from the original charge which remained in the lines after drainage at the 4.7 hours point. This may account for the somewhat large chemical change noted in this charge of oil for the time run (viscosity increased 20%, dirt content increased 150% and acid number increased 100%) compared to the results of test 1.

Unlike tests 1 and 6 with this oil, the rig interior showed only mild coking. This is believed attributable to the short time run with somewhat lower than usual average oil temperature.

Search Coil Results:

Performance of the dual differential search coil was excellent from the functional standpoint and a full set of data was recorded on tape during the test. The individual differential signals from the two coil sets were recorded during speed transients between successive speed settings of 0, 4000, 8000, 12000, 16000, 18000 and 20,000 rpm. Constant speed data was also recorded at each of these speed settings and at 10,000 rpm.

Analysis of these data is not presented in this report since this effort is not a part of the present contract, the objective of the dual differential coil work on this contract being only to demonstrate the operation of the coil hardware. The analysis results are, however, expected to become available at a later date as a result of work currently in process for the U.S. Air Force Aero Propulsion Laboratories under Contract Number F33615-72-C-1467.

Photographs illustrating the post test condition of the test elements and rig components are given in Enclosures 50 through 58.

G. Discussion of Results

G.1 Discussion of Lubricant Performance

As reference to the percentage change in viscosity, acid number and dirt content data of Enclosure 24 shows, when viewed with due allowance for test duration and the relatively small test to test variations in operating temperatures, Aeroshell Turbine Oil 555 experienced a smaller degree of chemical change than any of the other test fluids. Monsanto MCS-2931 would appear to be a very close second with the Exxon FN3158/Kendall 0839 mixture in third place but well behind the previous two, and with Conoco DN600, Type 2 well behind all of the others.

In regard only to chemical decomposition it is believed that Aeroshell Turbine Oil 555 and Monsanto MCS-2931 show promise for extended operation in the 500°F range although further testing at longer durations than were reached in this program are necessary to establish the long term behavior of these fluids. The Exxon FN3158 and 5% Kendall 0839 mixture appears, based on the limited testing to date, to be marginal in regard to chemical decomposition in this temperature range and the Type 2 Conoco DN600 definitely unacceptable.

As previously noted the dirt content readings taken during this program were in most cases, as noted in Enclosure 25, influenced strongly by heavy oil seal primary face wear (tests 1, 2, 5, 6 and 7) or by bearing failure (test 7). These data should therefore not be given undue emphasis in judgement of the oil's chemical stability performance.

As discussed in the Test Results section of this report the very poor chemical stability performance of the Conoco DN600 Type 2 is probably attributable to the additive package included to adapt it to automatic transmission use. According to Continental Oil Company the base stock for this fluid would exhibit much improved high temperature chemical stability.

The four lubricants were also evaluated in regard to their ability to maintain an adequate lubricant film at the ball to raceway contact and at the cage to land and cage to ball riding surfaces. In this area only one of the fluids was associated with a lubricant-related failure; this was the Monsanto MCS-2931. The second screening test, using this fluid, ended after 16.3 hours with a smearing failure of the test bearing. Inspection of the raceway surfaces and analysis of the search coil data established that the failure was lubrication related. None of the other three lubricants generated any evidence of surface distress on the raceways.

Cage wear investigation, however, did show a variation between all of the lubricants. The Aeroshell Turbine Oil 555 performed best in this regard and showed very little pocket or land riding surface wear, in spite of the fact that a total of 35.1 hours of running time (both test 4 and 5) was logged with this oil and bearing combination.

The Conoco DN 600 Type 2 generated a medium amount of cage wear (somewhat more than the Aeroshell 555 but less than the Exxon FN3158/Kendall 0839 mixture); however, it accumulated this degree of wear in only 4.2 hours of running time. The excessive degree of chemical change observed in the Conoco fluid may also have been a factor in the degree of cage wear noted with it.

The Exxon FN3158/Kendall 0839 mixture generated fairly heavy cage wear in tests 1 and 7 in which it logged 13.6 and 6.3 hours respectively. In test 6 only very light cage wear was noted with this fluid. However, only 2.1 hours of running time were logged in the latter test. The very high degree of dirt content observed in the oil used in all three of these tests may be a major factor in the degree of wear with the Exxon 3158/Kendall 0839 mixture. This is especially true since it is believed that the high dirt content levels in all three tests, was, as previously mentioned, due to shaft seal failure involving the probable release of carbide particals, worn from the mating ring face, into the oil system. This is, of course, also true of test 2 with the Monsanto MCS 2931 and of test 5 with the Aeroshell 555, the latter described above as having very little cage wear. However, it should be noted that although the oil seal was retired during test 5, the mating ring wear during that test was nowhere near as great as in the other test (1, 2, 6, and 7).

Finally, the failure of the test bearing during test number 2 Phase b) with the Monsanto MCS-2931 made it impossible to gain a worthwhile rating of cage wear with that lubricant.

In summary of the cage wear results, it can be said that the Aeroshell Turbine Oil 555 outpaced the other fluids in this regard with the Conoco DN 600 producing only mediocre performance and the Exxon FN3158/Kendall 0839 mixture giving poor performance with recognition being given to the fact that foreign contamination may have influenced the results in the latter two cases.

In overall summary of lubricant performance, then, the conclusion is drawn that the Aeroshell Turbine Oil 555 showed excellent performance in all three areas of chemical stability, lube film maintenance at the ball to race contacts, and avoidance of cage wear. This lubricant clearly deserves further study. The Exxon FN3158/ Kendall 0839 gave only marginal performance in the chemical stability and wear areas although lubricant film maintenance in the rolling contacts appeared acceptable. Its marginal cage wear performance may be due to foreign contamination of the fluid. Although the Monsanto MCS-2931 showed good chemical stability (though perhaps not quite up that of the Aeroshell 555), it appears unacceptable since it failed to maintain an EHD film capable of preventing smearing and thereby resulted in a failure of the test bearing. Finally, although the Conoco DN 600 Type 2 fluid gave no evidence of poor EHD film performance in the rolling contacts and gave acceptable cage wear performance, its extremely poor chemical stability makes it unacceptable in the present formulation.

G.2 Discussion of Shaft Seal Performance

G.2.a Discussion of Koppers "Air" Seal Performance

As shown by Enclosure 23 a single Koppers "air" seal was used throughout the test program and performed extremely well. Three mating rings differing only in plating were used, the first in test 1, the second in test 2a and the third from test 2b through the end of the program. Heavy nosepiece wear and moderate mating ring plating wear and chipping during test 1, and slight mating ring plating wear during test 2 are believed to be attributable to the condition of inadequate radial piloting of the mainshaft clamping sleeve discussed in section F.1.b of this report. The use of chromium plating on the mating ring during test 1 may have been the cause of the somewhat greater plating wear observed during that test (and the only "air" seal plating checks or cracks noted during the program) than was experienced in test 2a where chromium carbide was used under the same poor radial piloting conditions for nearly the same test duration.

The poor radial piloting condition was repaired between phases a and b of test 2 and from that point through the end of the program no further air seal changes were required and no problems of any type developed. In fact the seal rode out such potentially damaging situations as a test bearing smearing failure and a rig roller bearing spalling failure with no noticeable adverse effects.

G.2.b Discussion of Koppers "Oil" Seal Performance

As with the Koppers "air" seal, failures involving very heavy carbon nosepiece and mating ring face wear were experienced with the Koppers "oil" seal during test 1 and phase a of test 2. The primary cause of these early failures was eliminated after test 2a by improving the radial piloting of the mainshaft clamping sleeve thereby eliminating a source of mating ring face runout. Also after test 2a the "oil" seal mating ring face plating was changed from tungsten carbide to chromium carbide.

Unlike the "air" seal, however, the "oil" seal continued to experience severe primary face wear, involving both the nosepiece and the mating ring, after the clamping sleeve modification had been completed. This continuing wear led to refurbishment of the seal again after test 4 and a subsequent failure during test 5. The details of the inspection results on these seals have already been given in section F and are not repeated here. These inspection results and observation of the seal's performance led to the following conclusions in regard to the existing seal design:

1) The alignment of the high points of the saddle shaped wear pattern on the seal's nosepiece face with the two retainer-adapter assembly anti-rotation pins indicates that the retainer-adapter assembly was rocking on these pins during operation. Mating ring face runout would act as a driving force for a rocking motion of this type. Presumably, at some level of sliding speed the inertia of the retainer-adapter assembly would generate enough resisting force to cause the nosepiece to break through the supporting hydrodynamic film and initiate wear.

Similarly, rapid acceleration or deceleration of the shaft could create a change in the required rocking speed too rapid for the retainer adapter assembly to follow, again resulting in breakthrough of the hydrodynamic film and nosepiece wear. Significantly in this regard the oil seal failures during tests 2a and 5 occurred, respectively, during phases of acceleration and deceleration. It is considered that the seal should be redesigned to eliminate the two-pin method of rotation control.

2) Optical flat inspection of the "oil" seal mating ring face showed shallow depressions to exist immediately outboard of each of the L-shaped air feed grooves in the face of the mating ring. Analysis suggests that these are caused by gas errosion (compounded by solid contamination entrained in the air). Redesign to decrease this effect is indicated.

3) Chipping of the nosepiece face during operation may be due to "bouncing" of the nosepiece (possibly but not necessarily related to item 1 above). This is presumably a function of the spring force, the mass of the retainer-adapter assembly, the lift geometry and the damping inherent in the secondary seal and the anti-rotation pins. Re-evaluation of the complex relationships between these is required to eliminate this problem.

4) Finally, although no direct test evidence exists other than the high "oil" seal failure rate, it is suggested that the non-symmetrical design of the mating ring is not best for high speed operation since it is particularly subject to thermal gradient and centrifugal force effects capable of inducing distortion in the mating ring face.

5) Neither the tungsten carbide nor the chromium carbide platings on the "oil" seal mating ring performed without problems although the failure of the former in tests 1 and 2a was initiated by extraneous causes. Heavy nosepiece wear in the other tests of the program was also accompanied by wear and the development of radial cracks in the chromium carbide plating in use for those tests.

G.2.c Discussion of NASA "Oil" Seal Performance

The NASA "oil" seal was used in tests 6 and 7, both of which were terminated prematurely due to primary face rubbing type seal failures. The test 6 failure was not attributable to the seal's design but was caused by an inadvertant reversal of the rotational direction of the seal's lift geometry and windback. Test 7, however, was terminated at 20,000 rpm due to a rubbing failure of the seal.

The details of the test setup, seal performance and post test inspection during tests 6 and 7 are given in sections F.2.d and F.2.e. Evaluation of the seal's performance in these tests leads to the following conclusions:

1) A study conducted by NASA to determine the critical speed(s) of the rig is presented in Appendix V. This showed that the first critical speed occurs at about 24,000 rpm. This was a "whip" mode. Since both the Koppers and NASA "oil" seal designs failed in the 20,000 rpm speed range, it is possible that the "whipping" motion caused the seal to rub and resulted in seal failure.

2) The two anti-rotation keyways in the mating ring bore and on the mating ring mounting sleeve O.D. are responsible for an induced saddle shape on the operating face of the mating ring.

3) Mating ring growth was noted during both tests. This resulted in a loose fit of the mating ring on its mounting sleeve. The cause of this problem has not been established and it is not known if it is associated directly with the failures.

4) Some question remains as to the maximum acceptable level of mating ring face waviness. Arguments exist both for and against the adequacy of the 7-9 lightbands figure attained in the test 7 setup. This figure would appear to be acceptable in view of the NASA computer program's prediction of a 0.0003" nominal film thickness while the failure of the seal under these conditions during test 7 and the experience of Pratt and Whitney Aircraft and Stein Seal Company give the opposite indication.

In this regard it should also be pointed out, however, that the lack of seal wear noted during inspection at the 4.7 hour point while the rig was disassembled to replace a failed rig bearing does militate against the position that the seal failed through gradual removal of the lift pads by wear due to excessive mating ring face waviness. Further study is needed in this area to determine acceptable limits on as-installed mating ring face waviness.

5) In the configuration tested the seal utilizes no shielding to prevent oil and/or foreign contamination entrained in the oil from migrating either into the secondary sealing area or into the open space between the nosepiece retainer ring and the carrier. Contamination or coked oil in the former area could conceivably create secondary sealing problems (coking in the secondary area was found in testing on this program however no indication of gross secondary leakage was noted in these particular cases). Contamination or coked oil in the latter area could further migrate into the nosepiece seating zone on the carrier face and induce nosepiece distortion.

6) The existing design relies on a heavy press fit (0.040" nominal on dia.) to maintain the nosepiece retainer (and windback which bolts to the retainer) in position on the nosepiece O.D. Reduction of this fit to 0.012" in test 7 was associated with slippage (including relative tipping) between these parts. Although the test 7 slippage may have originated during the seal failure it is to be noted that slippage (for whatever reason) occurring before or during a test could induce gross nosepiece face distortion, a potential failure mode. It is suggested that a more positive means of relative positioning of these parts is required.

7) As with the Koppers seal, radial cracks were noted in the chromium carbide plating after failure. Although this does not indicate a failure mode existing during normal operation it could compound a primary face rub which might otherwise heal itself.

8) As a positve feature it was noted that the seal survived the heavy vibration of the rig roller bearing failure which occurred during test 7 with no indication of any detrimental running conditions at all. As noted previously the opportunity was taken to conduct a thorough inspection of the seal after the bearing failure at 4.7 hours and this revealed no nosepiece chipping and virtually no wear at that point.

G.3 Discussion of Test Bearing Performance

Performance of the test bearing during the program can generally be described as excellent. One test bearing failure occurred by smearing during test 2b. No fatigue failures occurred and, other than during the smearing failure, no lubrication related degradation of the raceways was noted.

Although the bulk of the running time on the bearing was conducted at 20,000 rpm (2.5 x 10^6 DN) or below, speed was taken as high as 23,600 rpm (nearly 3.0 x 10^6 DN) without any indication of problems. Even at the highest speeds run no thermal imbalance situations developed (i.e., inner ring temperature exceeding outer ring temperature to the point of loss of radial looseness leading to even higher temperature in a compounding situation progressing irreversibly to failure of the bearing).

Cage performance was good throughout the program. Performance of the electro-plated silver plate bond was excellent with no evidence of plate separation from the base material. Land and pocket wear were judged to be generally nominal as compared to other silver plated steel aircraft mainshaft bearings of this type, confirming the acceptability of the clearances chosen for use in these areas.

Although bearing internal geometry might be contributory to the test 2b smearing failure, it is believed that the failure's primary cause lies in the characteristics of the EHD film generated with the Monsanto MCS 2931 lubricant in use for that test. In its current level of development, rolling contact EHD film theory does not fully describe the events leading to smearing failure. It is believed, however, that changes in the various parameters affecting the film (i.e., load, speed, bearing internal geometry, surface finish and waviness, lubricant temperature, solid or gaseous foreign matter entrained in the lubricant, chemical composition and physical properties of the lubricant, etc.) may, in certain cases, result in a loss of traction and film thickness and generate a skidding/smearing failure of the bearing. This phenomena has been shown empirically to be dependent upon, among other things, the type of lubricant in use, and its occurrence during test 2b implicates the Monsanto MCS 2931. The absence of this phenomena during the other tests, using identical test bearings under very similar conditions tends to exonerate the bearing as a primary cause of the failure. Additionally, analysis of the search coil data showed discrete high frequency perturbations in the ball rotational velocity profile prior to the test 2b failure indicating, in theory, microsmearing occurrences at the raceway contacts. These perturbations were not seen to occur with the other oils used in the program, in spite of the fact that they were run under nearly identical conditions. This also implicates the lubricant in the failure and tends to vindicate the bearing.

H. Conclusions

The basic ability of the existing bearing design, some of the lubricants, and the rubbing carbon nose face seal used in "air" seal position to run in the 2.5 to 3.0 x 10^6 DN range with bearing ring temperatures of 500°F was demonstrated. Although the maximum program speed objective of 24,000 rpm (3 x 10^6 DN) was essentially attained, the bulk of the testing was limited to 20,000 rpm and below by "oil" shaft seal failures and other problems.

One lubricant, Aeroshell Turbine Oil 555, a type 2 ester, gave exceptional performance and showed minimal chemical degradation and coking, adequate lubricant film maintenance at the ball to race contacts, and a minimum amount of cage wear. The Exxon FN3158 super-refined naphthenic hydrocarbon blended with 5% Kendall 0839 paraffinic resin gave only marginal performance in the chemical stability and wear areas. However, its ability to maintain a lubricant film in the rolling contacts was acceptable. The Monsanto MCS-2931 improved modified polyphenyl ether showed good chemical stability but appears unacceptable since it failed to maintain an EHD film capable of preventing smearing and its use resulted in the only test bearing failure of the present program. Although the Conoco DN 600 Type 2 formulated synthetic hydrocarbon gave no evidence of poor EHD film performance and gave an acceptably small degree of cage wear, it showed extremely poor chemical stability making it unacceptable in the present formulation.

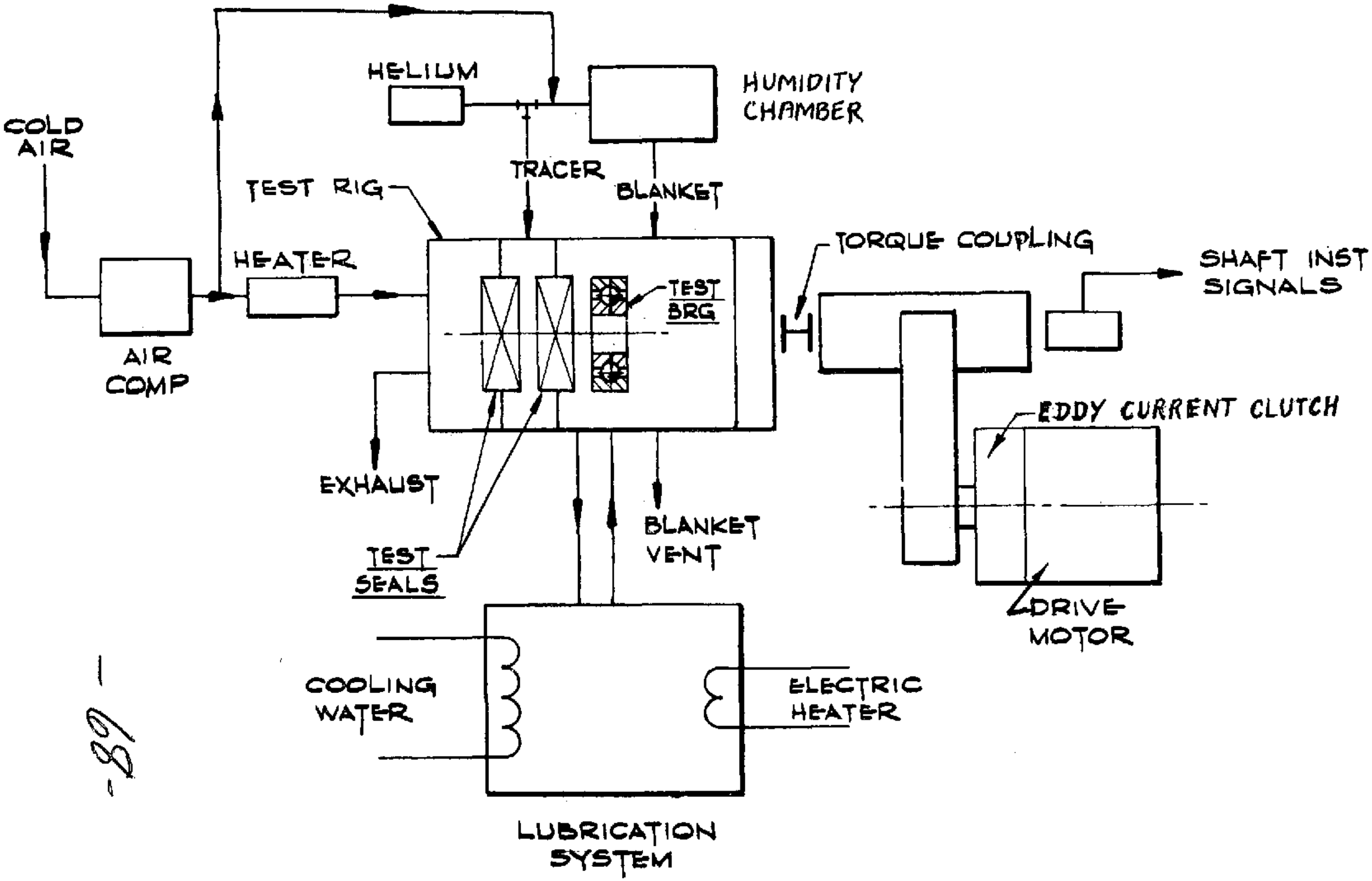
Performance of the Koppers shaft seal used in the "air" position, of relatively conventional sliding face and bellows secondary design, was excellent. This seal showed an ability to run without problems in the 600-700 fps sliding speed range under the low pressure drop conditions used.

The Koppers and NASA shaft seals used in the "oil" position, both using self acting primary face lift geometry and piston ring secondary sealing, gave less acceptable performance and a number of primary face rubbing failures were experienced. It is believed that these failures were, however, not indicative of any basic inadequacy of the self-acting design concept but were, rather, a result of the need for further design refinements and the establishment of proper installation requirements.

The split inner ring test bearing gave excellent performance throughout the program. The one failure experienced was of the smearing type and was attributed to the Monsanto MCS-2931 oil rather than to the bearing.

ENCLOSURE 1

GENERAL TEST RIG LAYOUT SCHEMATIC



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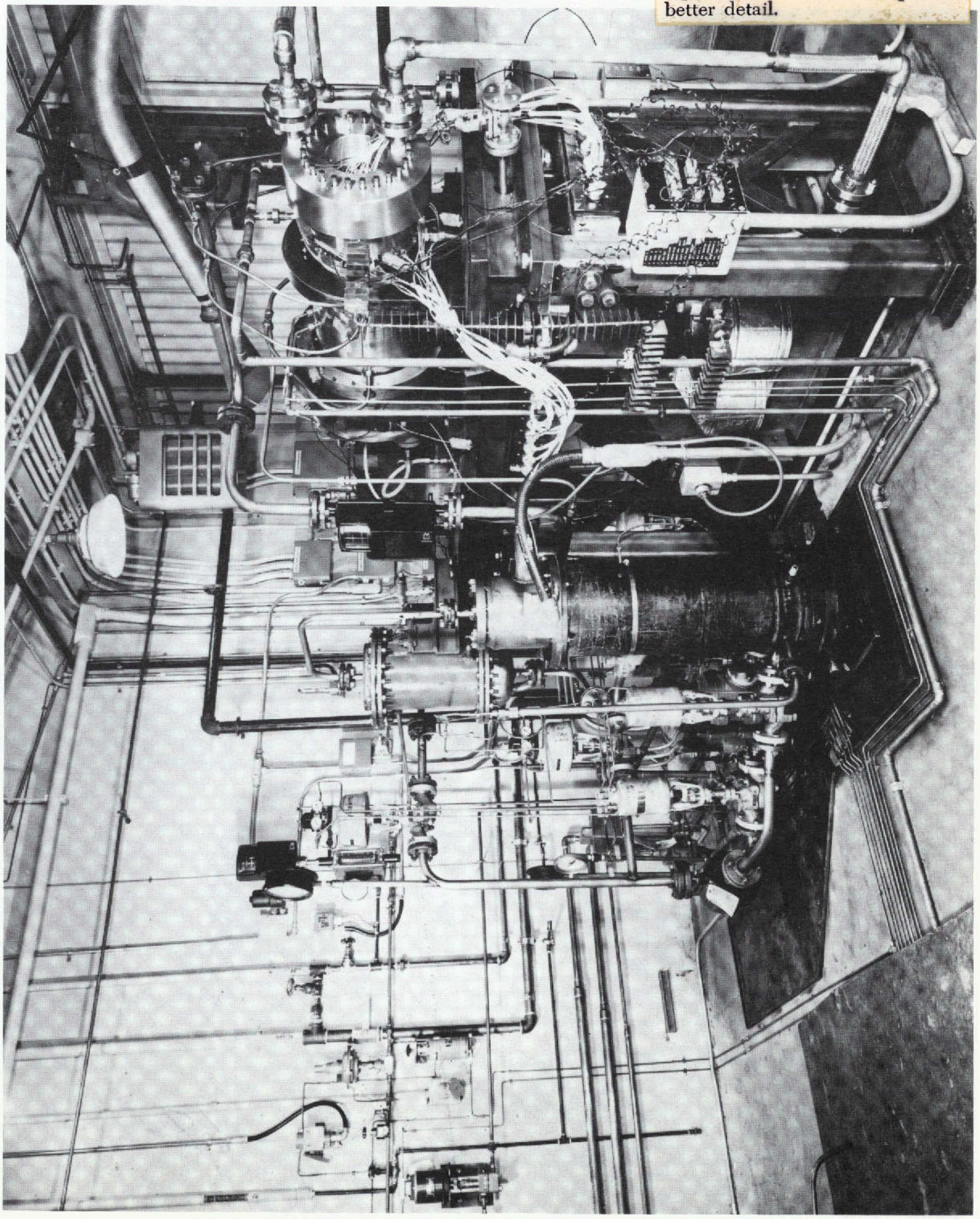
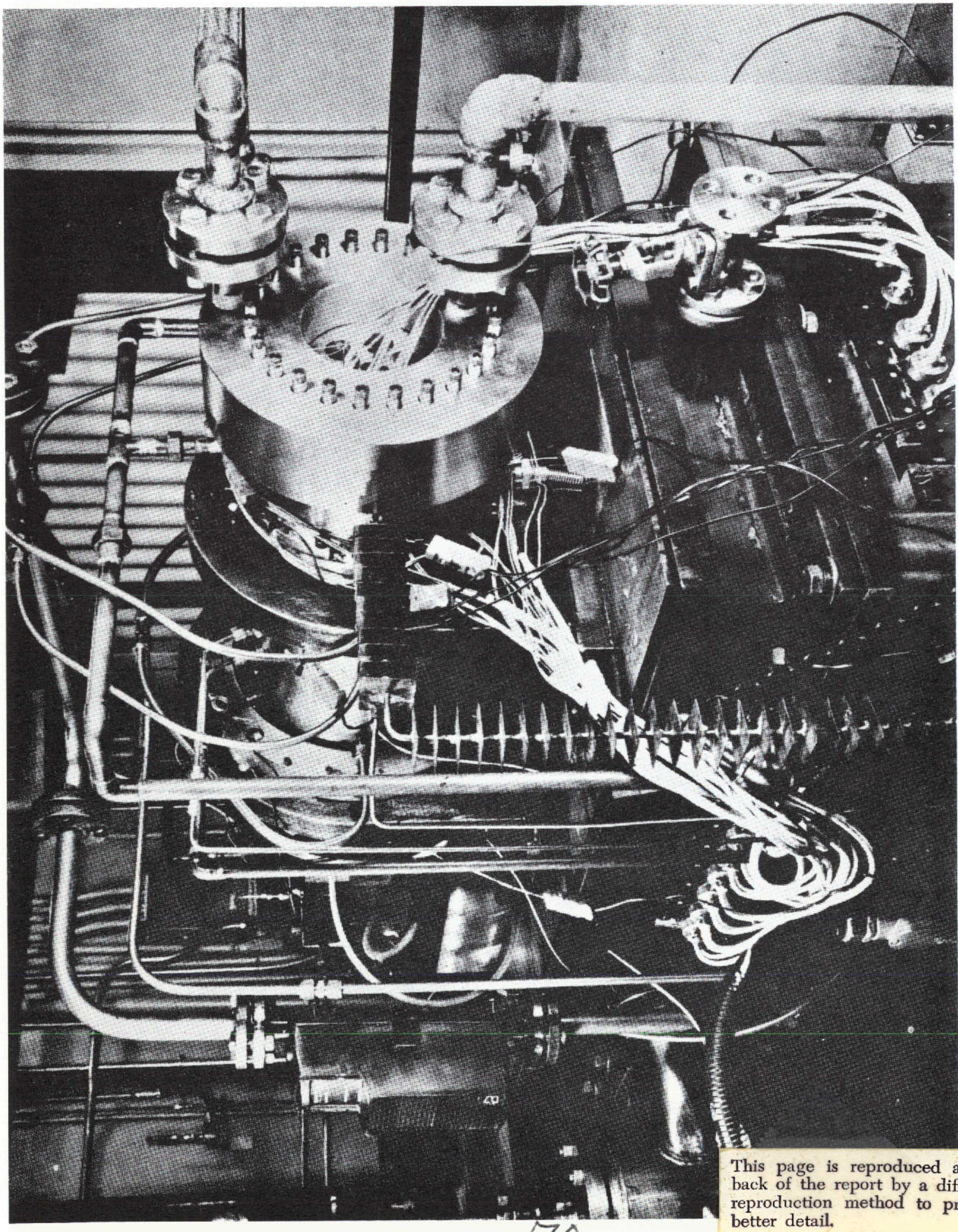
ENCLOSURE 2

GENERAL VIEW OF RECIRCULATING-OIL TEST RIG CELL



ENCLOSURE 3

CLOSE UP VIEW OF RECIRCULATING OIL TEST RIG
SHOWING HOT AIR SUPPLY TO THE BACK OF THE TEST SEAL PAIR



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ENCLOSURE IV-1

SEARCH COIL OSCILLOGRAMS

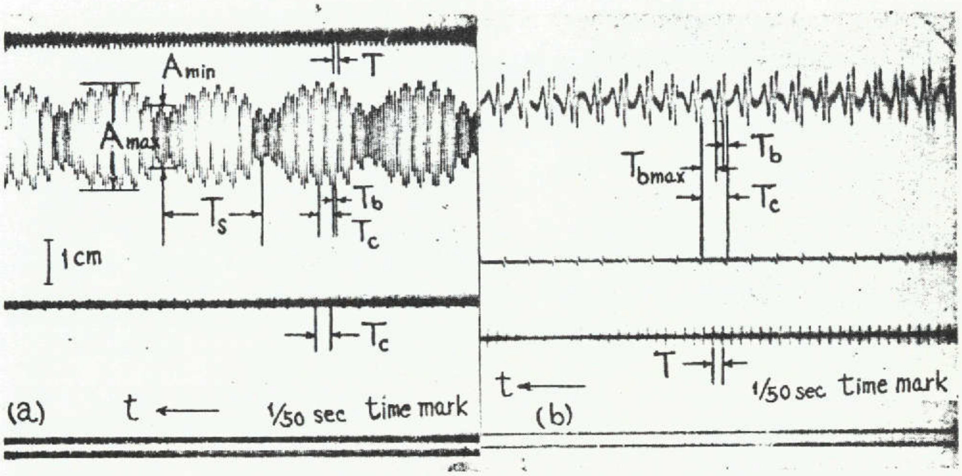
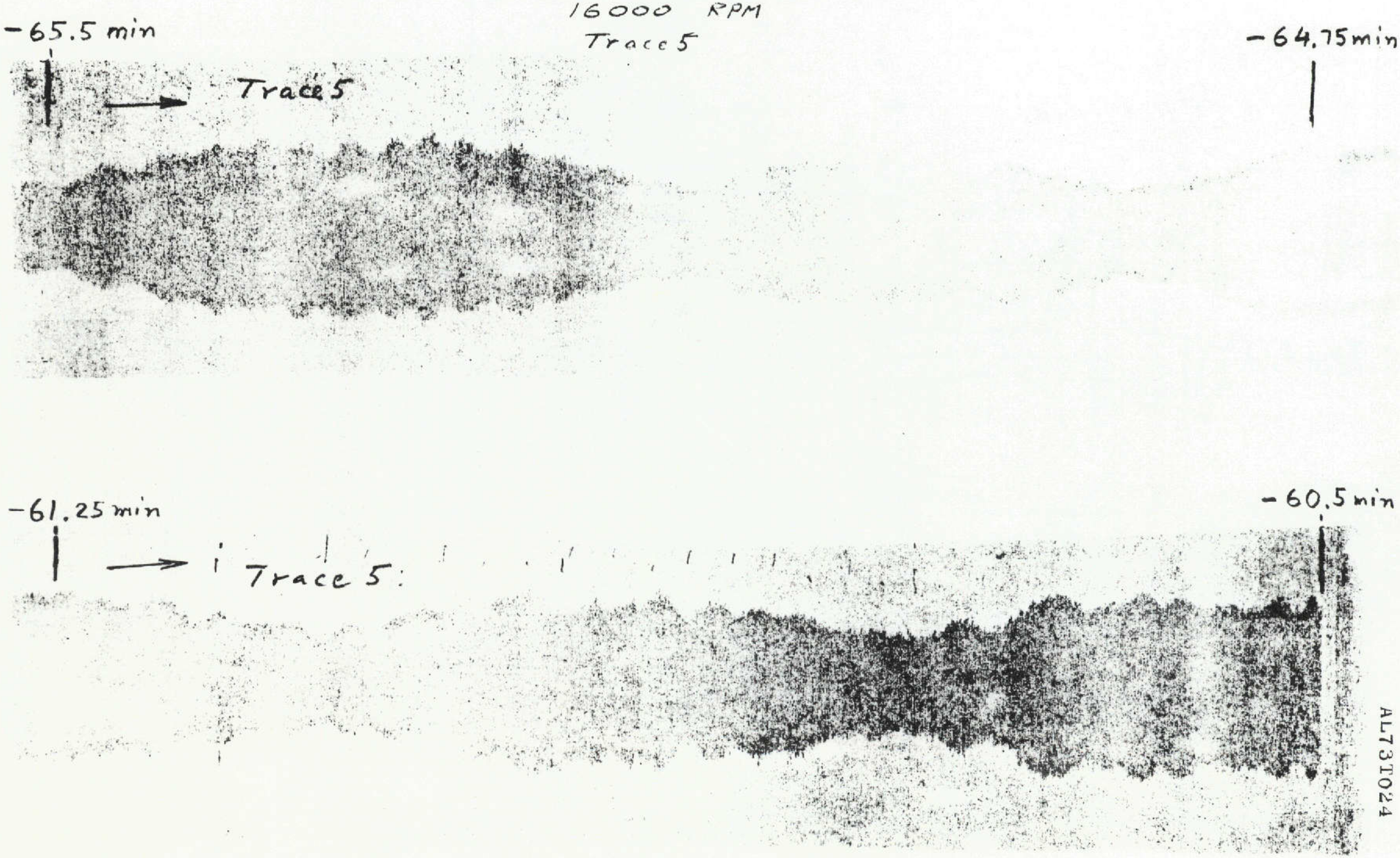


Fig. 6. Explanation of oscillogram. (a) No. 6307, $P = 100$ kg, $N = 1290$ rev/min. (b) No. 7307 $P = 100$ kg, $N = 750$ rev/min. T: Period of revolution of inner ring or shaft; T_b: Period of rolling ball, T_{bmax}: Maximum of T_b; T_c: Period of revolution of cage or ball centre; T_s: Period of fluctuation of amplitude.

ENCLOSURE IV-3

VISICORDER TRACES, 0.2 in/sec.

MONSANTO MCS-2931 OIL
16000 RPM
Trace 5



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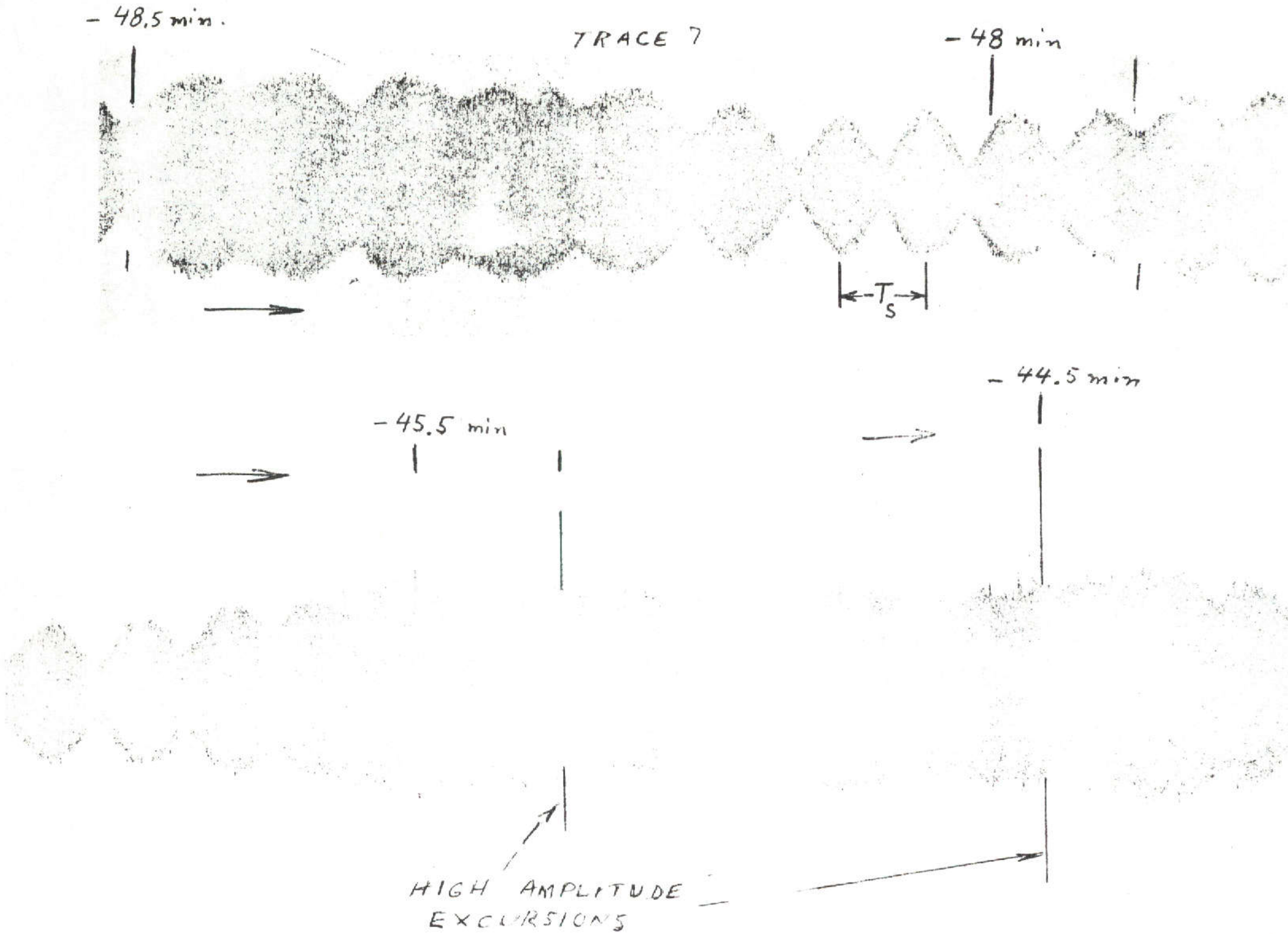
ENCLOSURE IV-4

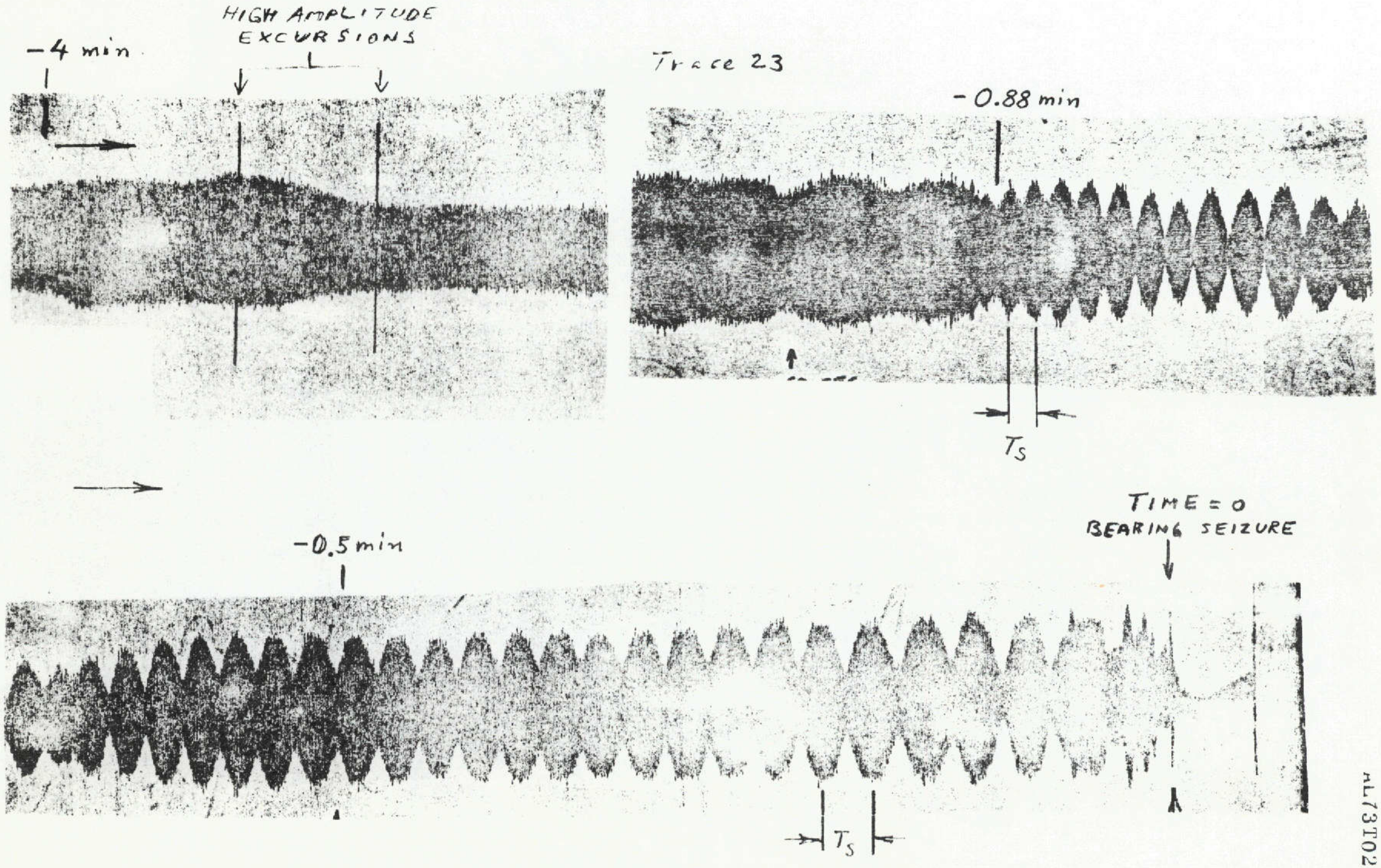
VISICORDER TRACES, 0.2 in/Sec
MONSANTO MCS - 2931 OIL
16000 — 18000 RPM



AL73T024

ENCLOSURE IV-5

VISICORDER TRACES, 0.2 in/sec.
MONSANTO MCS-2931 OIL
18000 RPM



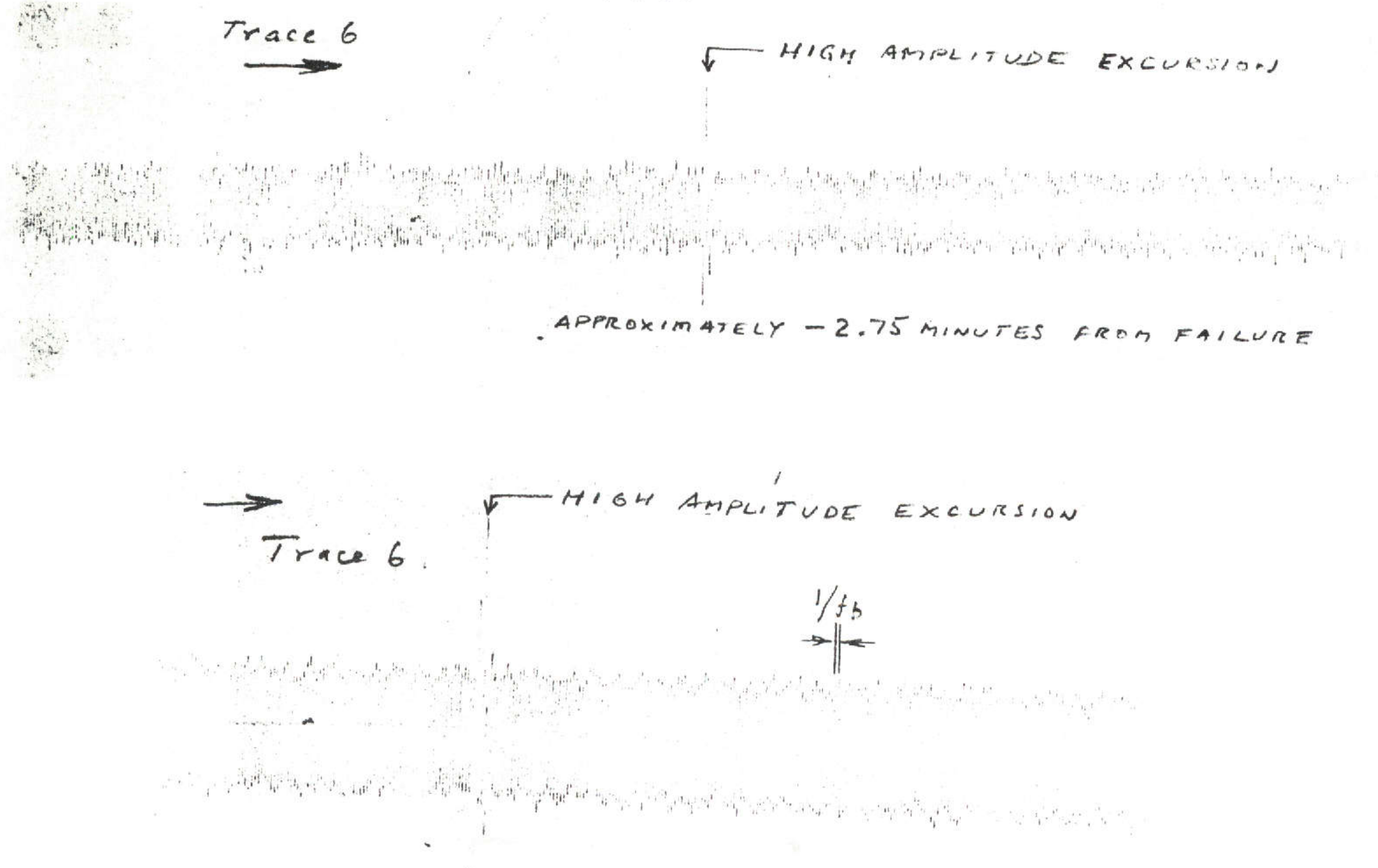
AL73T024

ENCLOSURE IV-6

VISICORDER TRACES, 25 inches/sec.

MONSANTO MCS-2931 OIL

Trace 6.



APPROXIMATELY -3.85 minutes from failure

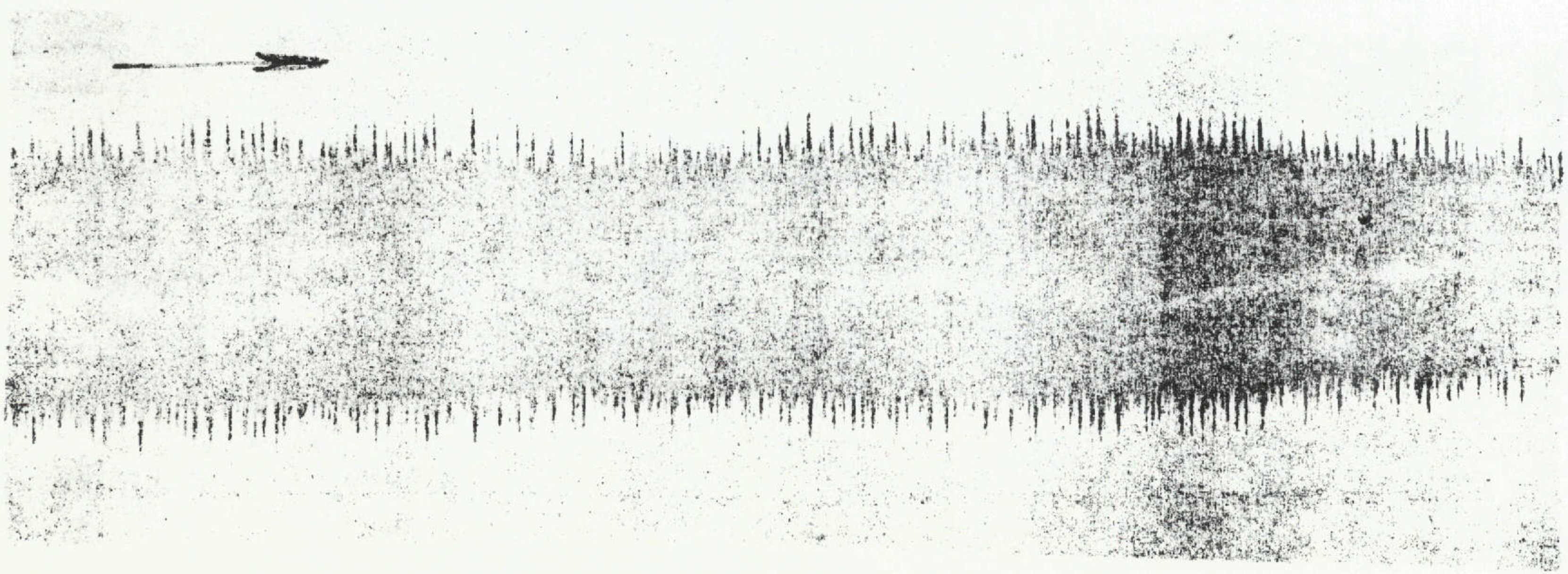
ENCLOSURE IV-7

VISICORDER TRACE, 0.2 inches/sec.

CONOCO DN 600 OIL

16000 RPM

TRACE 1.



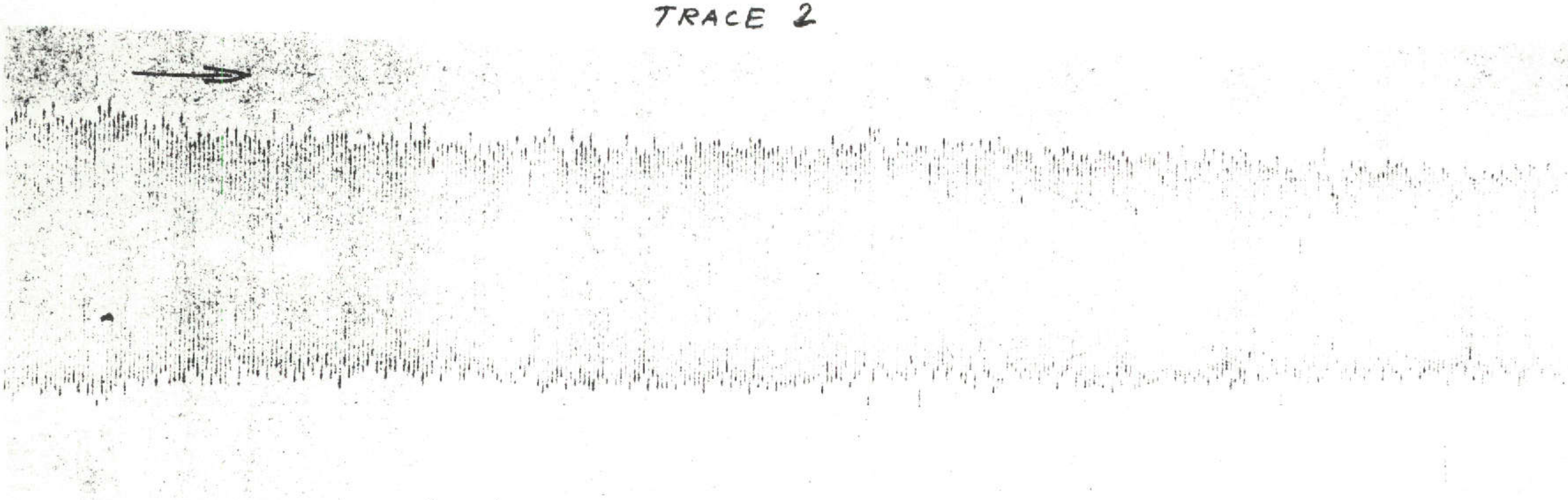
ENCLOSURE IV-8

VISICORDER TRACE, 25 inches/sec.

CONDEO DN 600 OIL

16000 RPM

TRACE 2



AL73T024

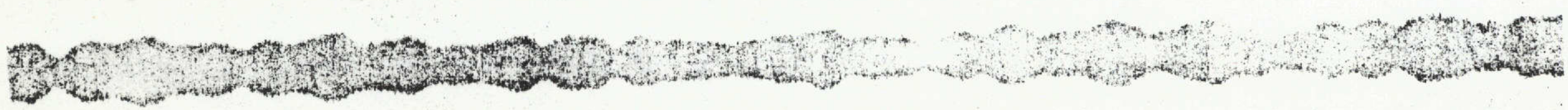
ENCLOSURE IV-9

VISICORDER TRACE, 0.2 inches/second

AEROSHELL TURBO OIL 555

21400 RPM

Trace 2

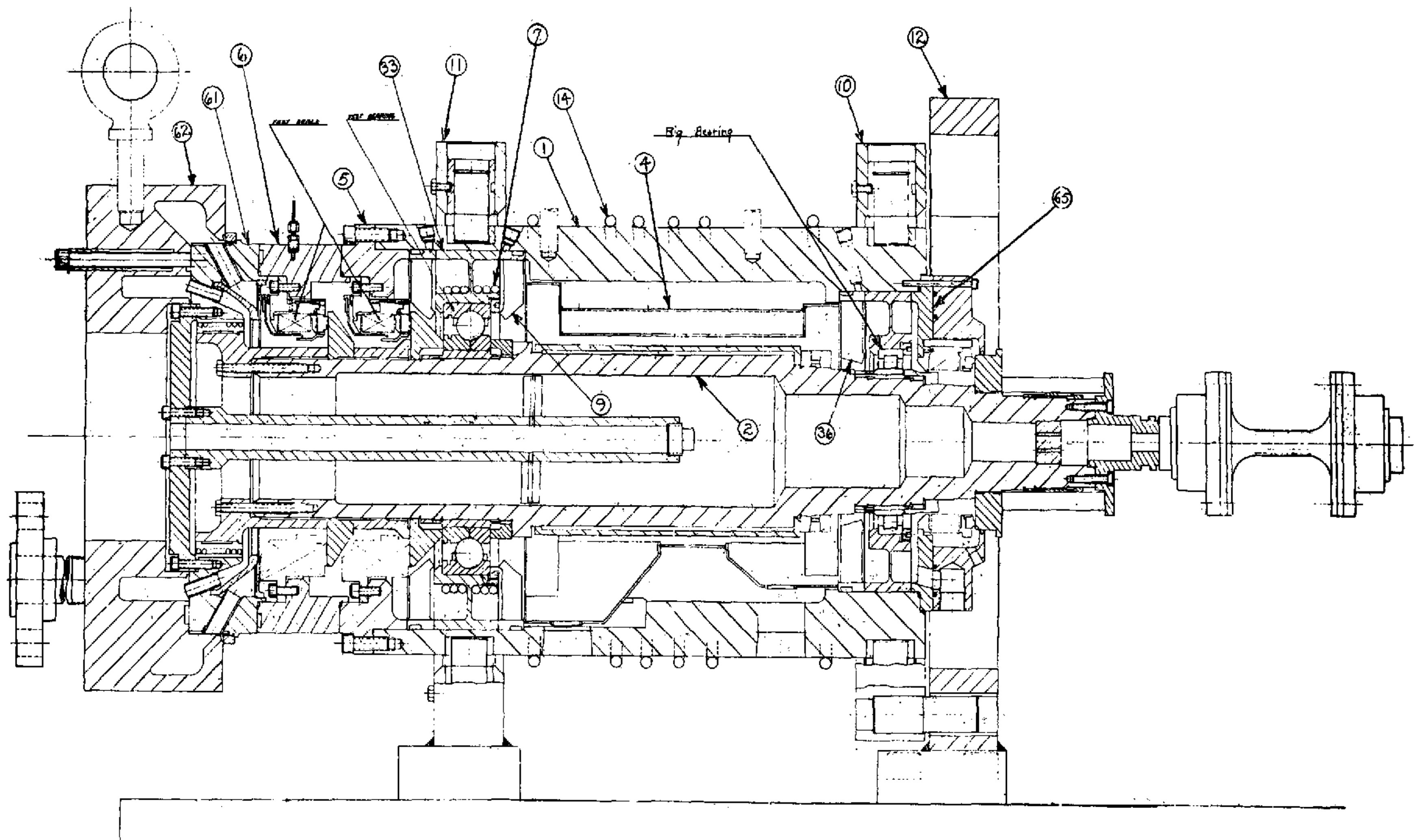


AL73T024

AL73T024

ENCLOSURE 4

TEST RIG ASSEMBLY

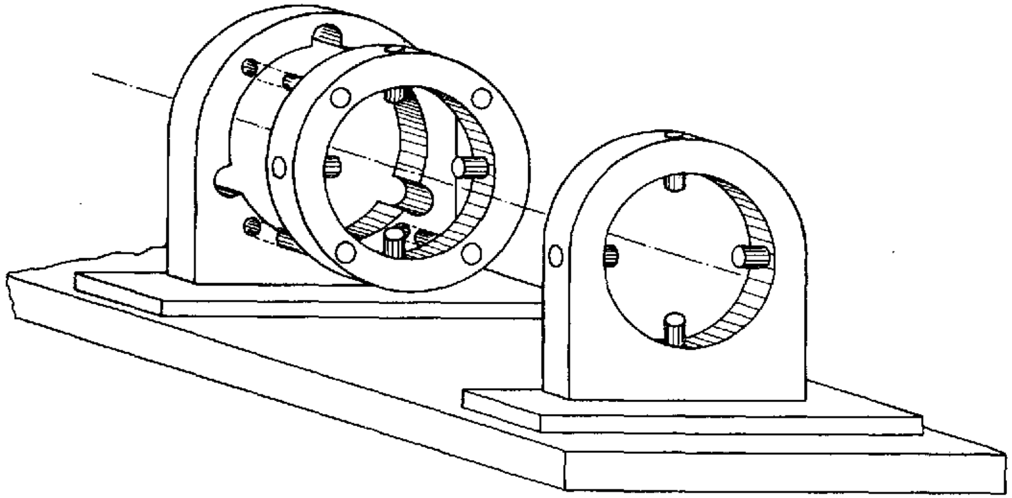


RESEARCH LABORATORY SKF INDUSTRIES, INC.

ENCLOSURE 5

ISOMETRIC VIEW OF MOUNTING ARRANGEMENT

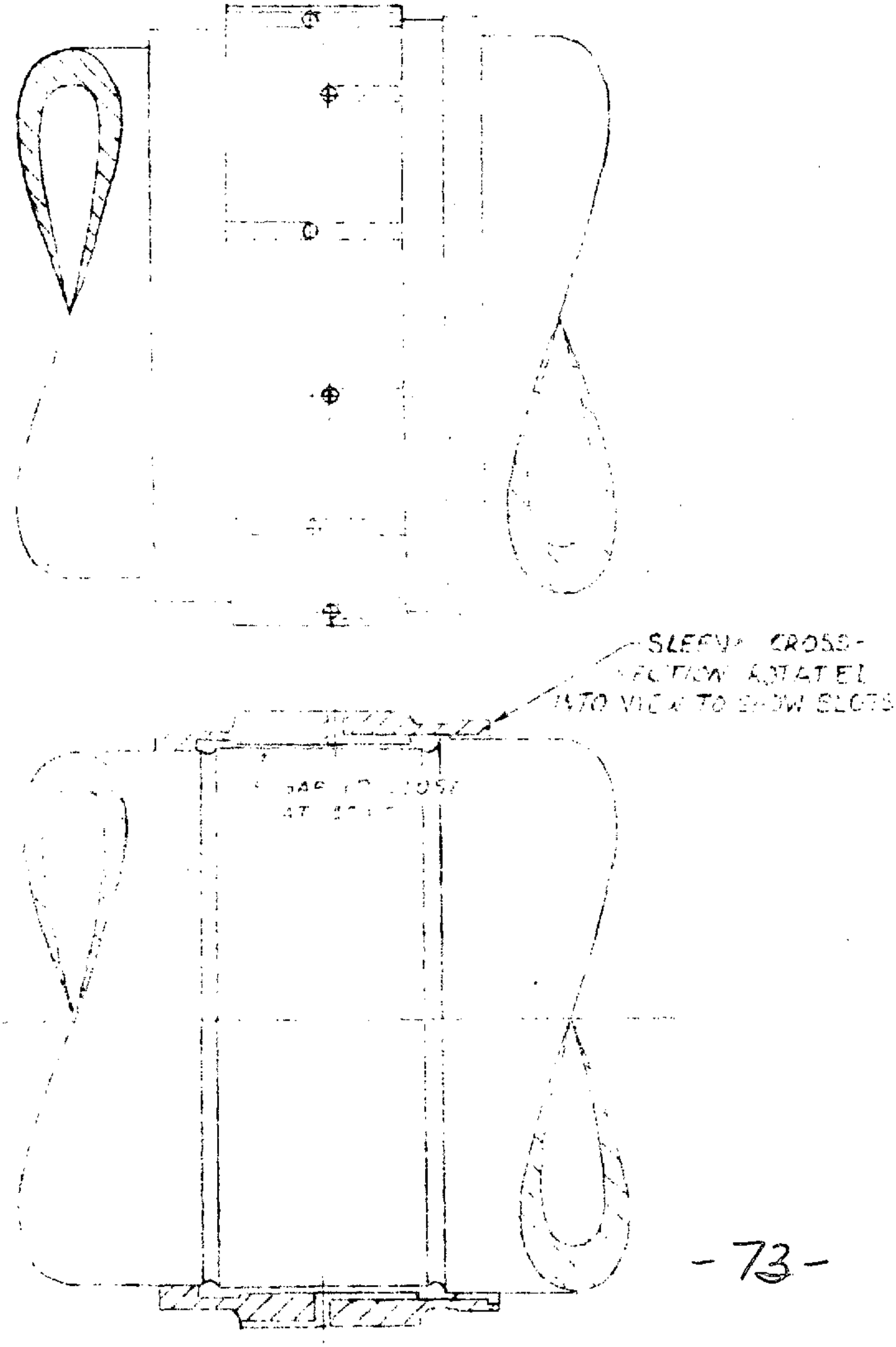
L - 41041



SKF INDUSTRIES, INC. PHILADELPHIA, PA.	TEST RIG MOUNT 3/4 RIGHT SIDE FRONT VIEW SST-801	DRAWN: JDC CHECK: APPR.:	SCALE — DATE 6-14-1964
		L - 41041	

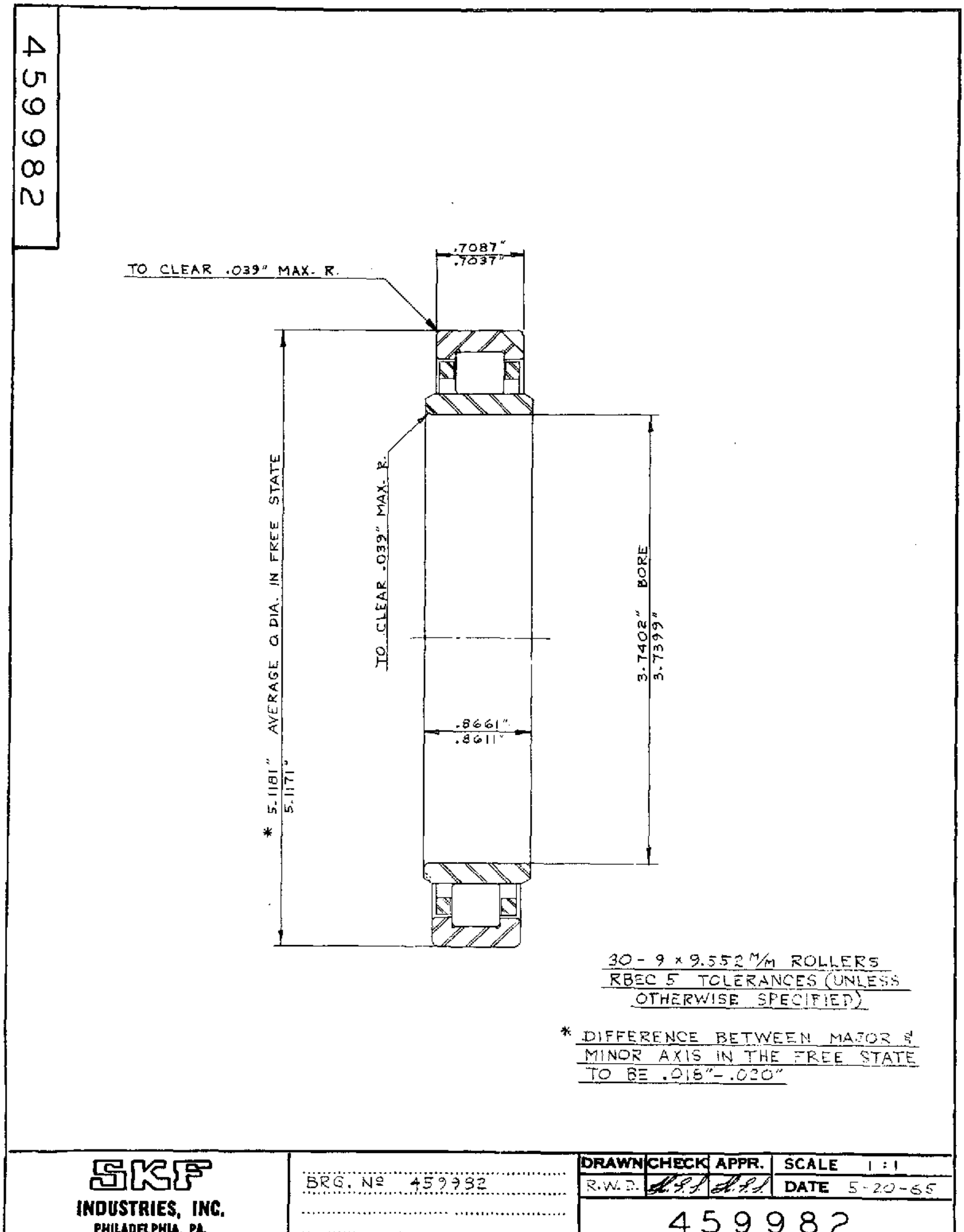
ENCLOSURE 6

BEARING INNER RACE MOUNTING SLEEVE



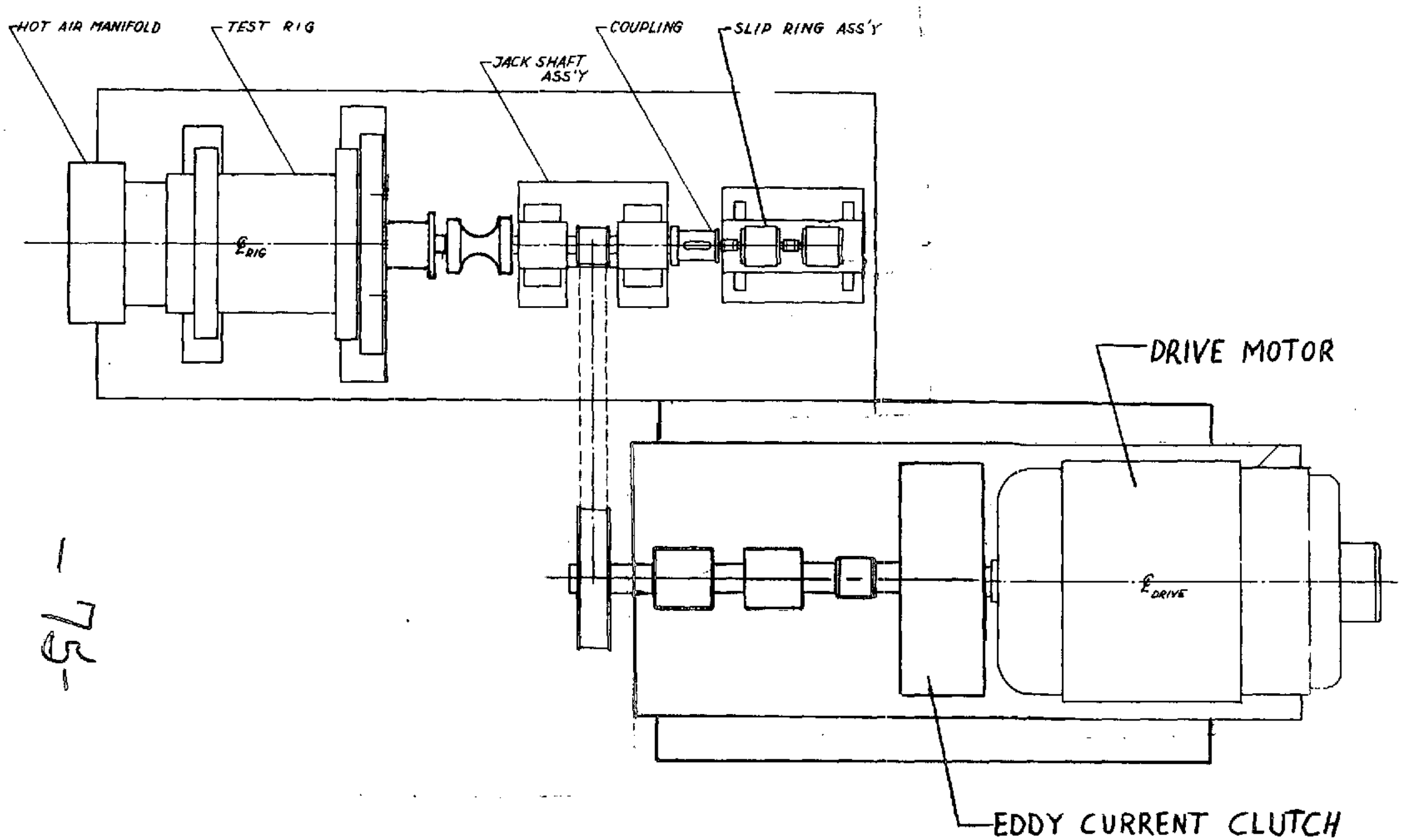
ENCLOSURE 7

TEST RIG ROLLER BEARING



ENCLOSURE 8

TEST RIG - GENERAL PLAN VIEW



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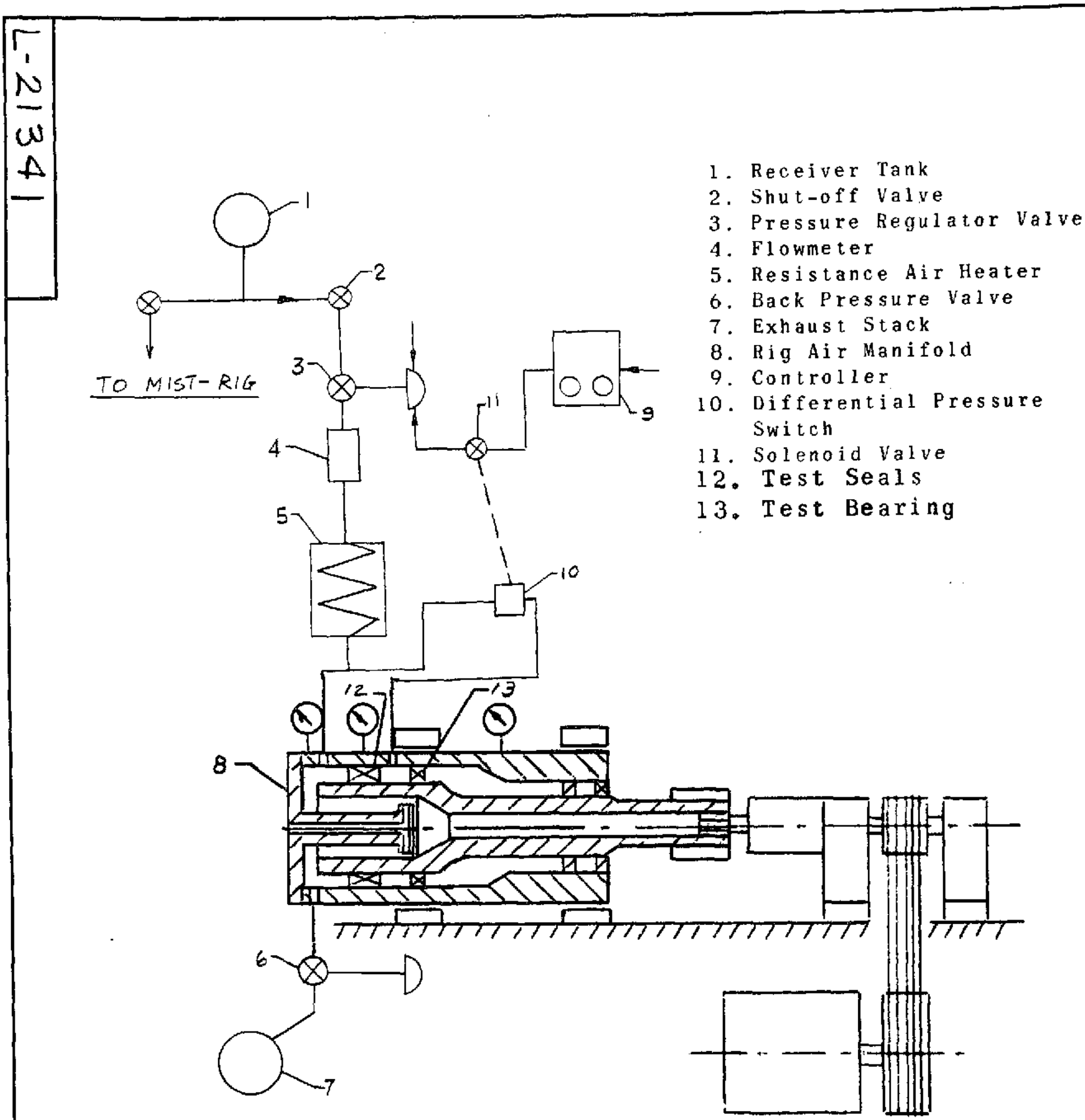
AL73T024

ENCLOSURE 9

SCHEMATIC OF HOT AIR SYSTEM



SKF INDUSTRIES, INC. PHILADELPHIA, PA.	BLOCK FLOW DIAGRAM FOR RIG HOT AIR SYSTEM	DRAWN	CHECK	APPR.	SCALE —
		GK	GS		DATE 5-14-65
		L-21341			

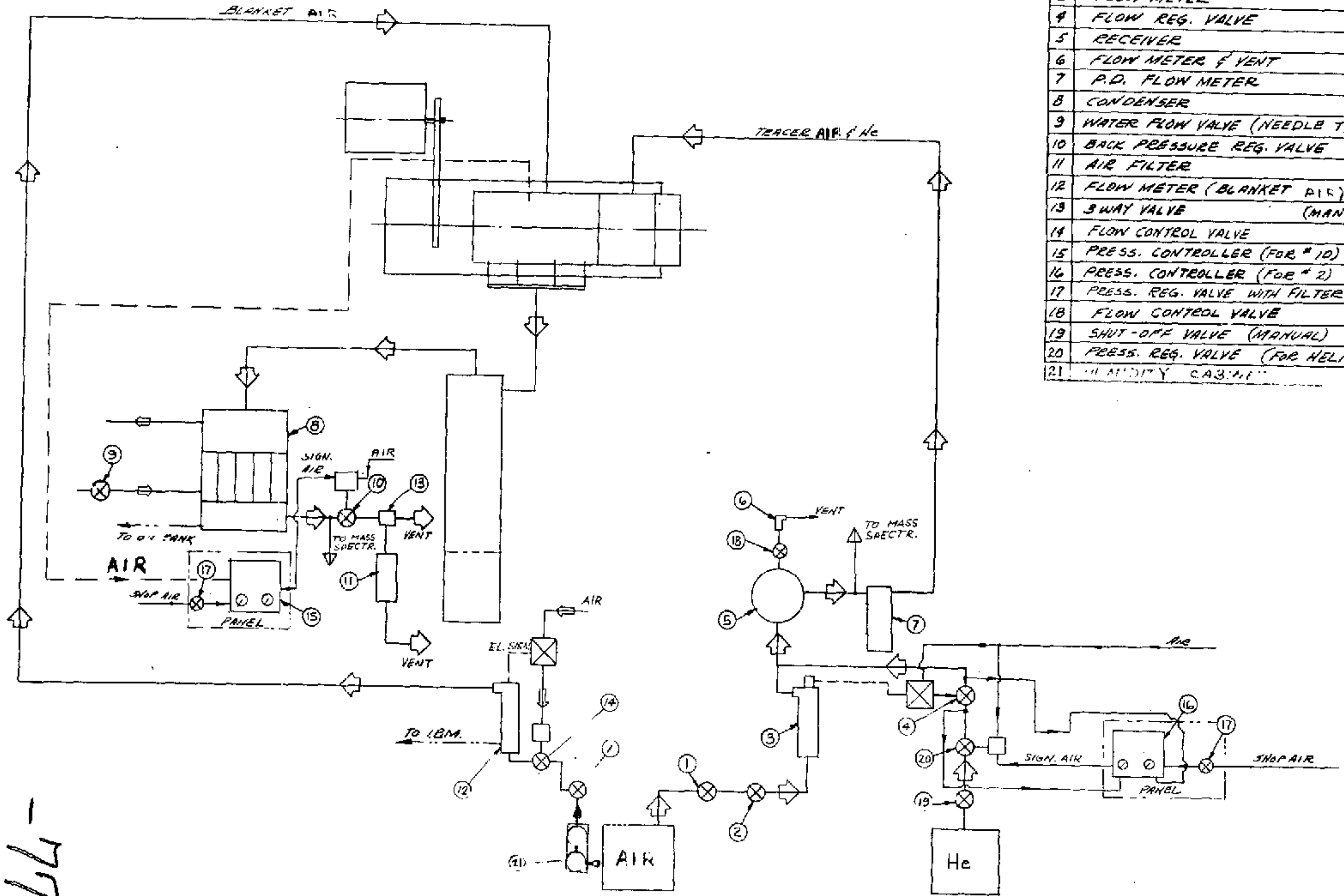
RESEARCH LABORATORY SKF INDUSTRIES, INC.

ENCLOSURE 10

HELIUM TRACER FLOW SCHEMATIC

L-61125

NO	DESCRIPTION
1	SHUT-OFF VALVE (MANUAL)
2	PRESSURE REG. VALVE
3	FLOW METER
4	FLOW REG. VALVE
5	RECEIVER
6	FLOW METER & VENT
7	P.D. FLOW METER
8	CONDENSER
9	WATER FLOW VALVE (NEEDLE TYPE)
10	BACK PRESSURE REG. VALVE
11	AIR FILTER
12	FLOW METER (BLANKET AIR)
13	3 WAY VALVE (MANUAL CONTROL)
14	FLOW CONTROL VALVE
15	PRESS. CONTROLLER (FOR # 10)
16	PRESS. CONTROLLER (FOR # 2)
17	PRESS. REG. VALVE WITH FILTER
18	FLOW CONTROL VALVE
19	SHUT-OFF VALVE (MANUAL)
20	PRESS. REG. VALVE (FOR HELIUM)
21	HUMIDITY CABINET



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FOR BAIL SEE

SKF INDUSTRIES, INC. PHILADELPHIA, PA.

HELIUM TRACER CIRCUIT

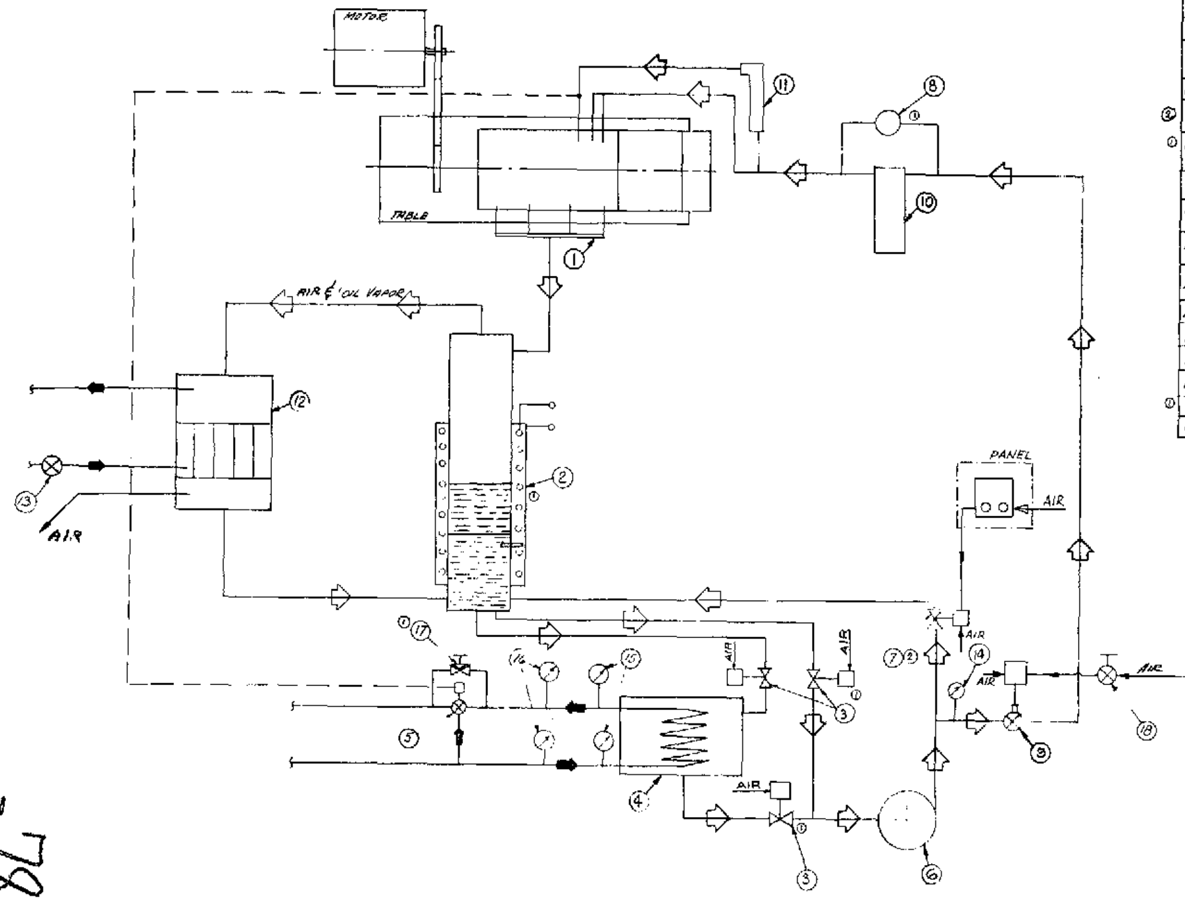
DRAWN	CHECK	APPR.	SCALE
GK			DATE 2-17-65

L-61125

AL73T024

RESEARCH LABORATORY SKF INDUSTRIES, INC.

L-61124



NO.	DESCRIPTION
1.	OUTLET-MANIFOLD
2.	OIL TANK & OIL HEATER
3.	2 WAY VALVE (REMOTE CONTROLLED) (ON-OFF) (3 REQ'D)
4.	HEAT EXCHANGER
5.	PROPORTIONING 3 WAY VALVE WITH TEMP. CONTROL
6.	PUMP
7.	RELIEF VALVE (REMOTE CONTROLLED)
8.	DIFF. PRESSURE CONTROL
9.	NEEDLE VALVE (REMOTE CONTROL)
10.	FILTER (OIL)
11.	FLOW METER
12.	CONDENSER
13.	NEEDLE VALVE (MANUAL)
14.	PRESS. GAUGE
15.	PRESS. GAUGE (2 REQ.)
16.	TEMP. GAUGE (2 REQ.)
17.	1/4 NEEDLE VALVE (MANUAL)
18	PRESS. REGULATOR

ENCLOSURE 11

CIRCULATING OIL SYSTEM SCHEMATIC

AL73T024

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② ITEM 7 WAS MANUAL ... NOW IS REMOTE CONTROLLED

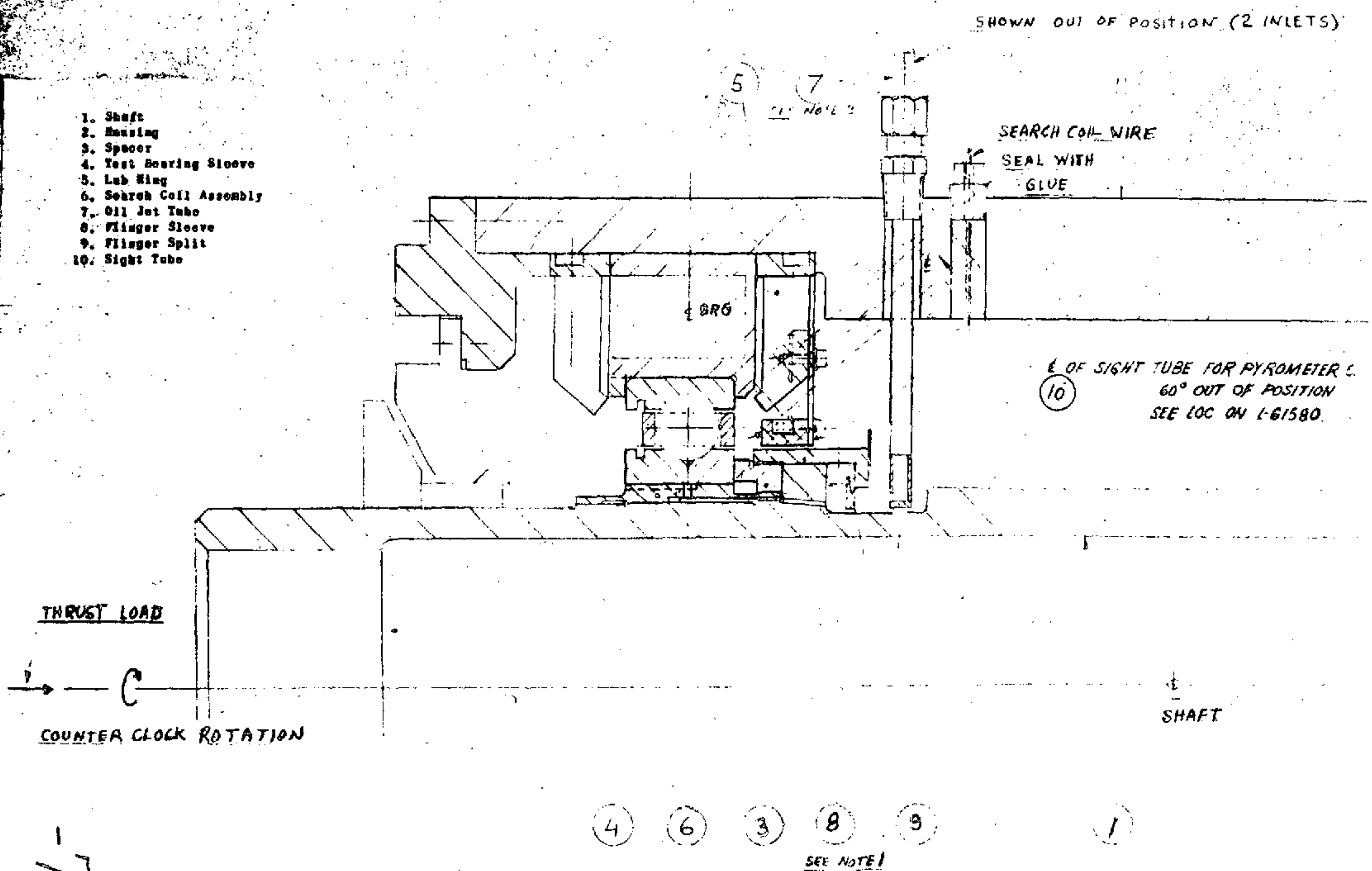
① IT. 2 WAS OIL TANK; IT. 3 WAS 3 WAY VA.; QUAN. WAS ... WAS OIL HEATER; ADDED IT. 17

SKF INDUSTRIES, INC. PHILADELPHIA, PA.

CIRCULATING OIL RIG HOT OIL CIRCUIT

DRAWN	CHECK	APPR.	SCALE —
			DATE 2-16-65

L - 61124



AL73T024

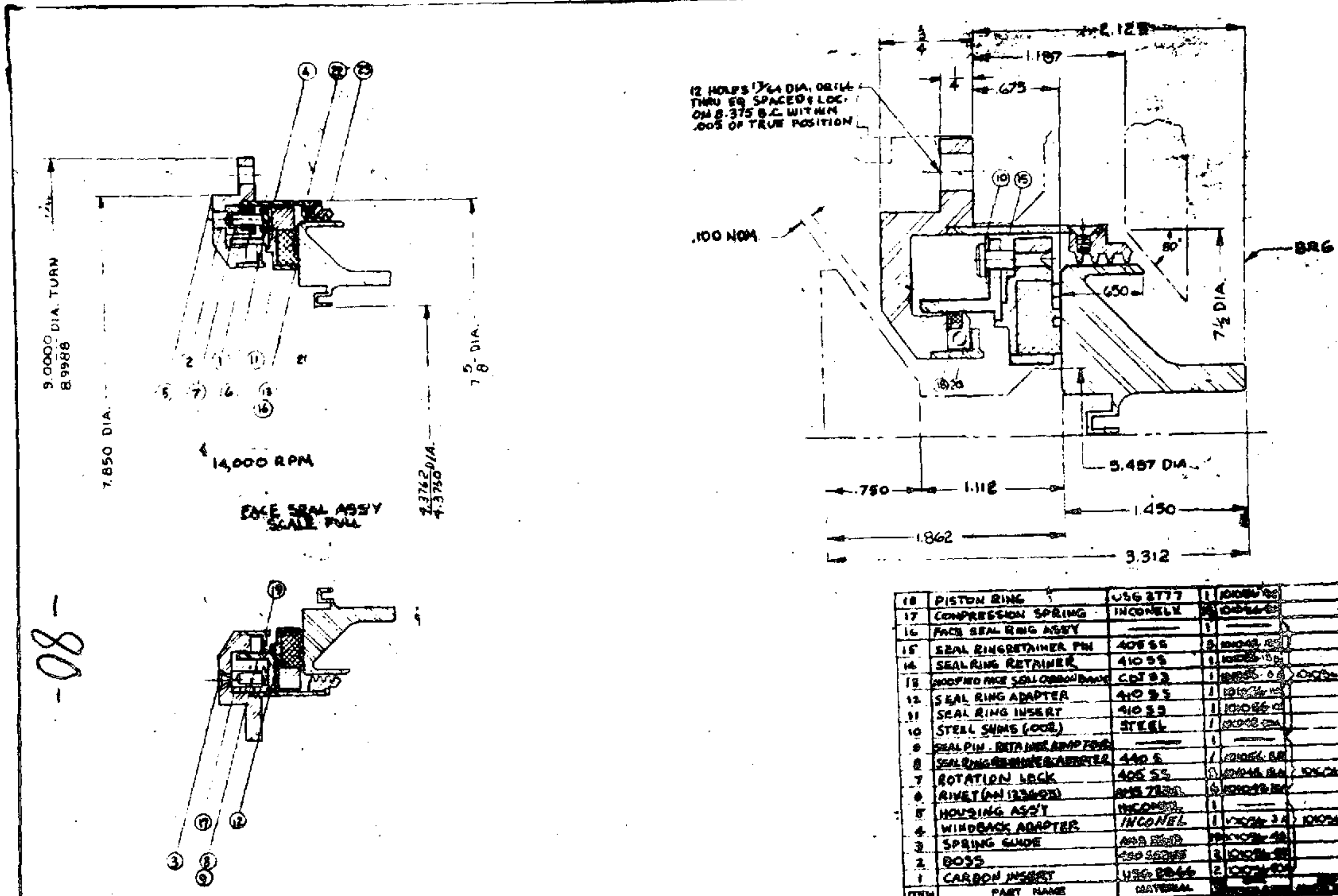
TEST BEARING UNDER RACE COOLING AND LUBRICATION SYSTEM

ENCLOSURE 12

ENCLOSURE 13

KOPPERS HYDRODYNAMIC LIFT SHAFT SEAL USED IN THE OIL POSITION

AL73T024



ITEM	PART NAME	MATERIAL		
18	PISTON RING	USG 2777	1	[illegible]
17	COMPRESSION SPRING	INCONEL X	[illegible]	[illegible]
16	FACE SEAL RING ASSY	—	1	—
15	SEAL RING RETAINER PIN	405 SS	3	[illegible]
14	SEAL RING RETAINER	410 SS	1	[illegible]
13	MODIFIED FACE SEAL CARBON BAND	CDT 83	1	[illegible]
12	SEAL RING ADAPTER	410 SS	1	[illegible]
11	SEAL RING INSERT	410 SS	1	[illegible]
10	STEEL SHIMS (.002)	STEEL	1	[illegible]
9	SEAL PIN - RETAINER ADAPTER	—	1	—
8	SEAL RING RETAINER ADAPTER	440 C	1	[illegible]
7	ROTATION LOCK	405 SS	[illegible]	[illegible]
6	RIVET (AN 123603)	[illegible]	6	[illegible]
5	HOUSING ASS'Y	INCONEL	1	—
4	WINDBACK ADAPTER	INCONEL	1	[illegible]
3	SPRING GUIDE	[illegible]	[illegible]	[illegible]
2	BOSS	[illegible]	3	[illegible]
1	CARBON INSERT	USG [illegible]	2	[illegible]

ITEM	PART NAME	MATERIAL		
23	WINDBACK	400 SERIES		101056-7B
22	FLAT HD MACH SCREW	300 SERIES	6	101056-5A
21	SHOULDER	AMS 6322	1	101056-8A
20	COMPRESSION SPRING	INCONEL X	1	101056-7B
19	PISTON RING RETAINER	INCONEL	1	101056-AC

6300 FACE SEAL ASS'Y SKF

KOPPERS COMPANY, INC.
KUCHLER-HURSH SEALS

101056 EN

NOTE

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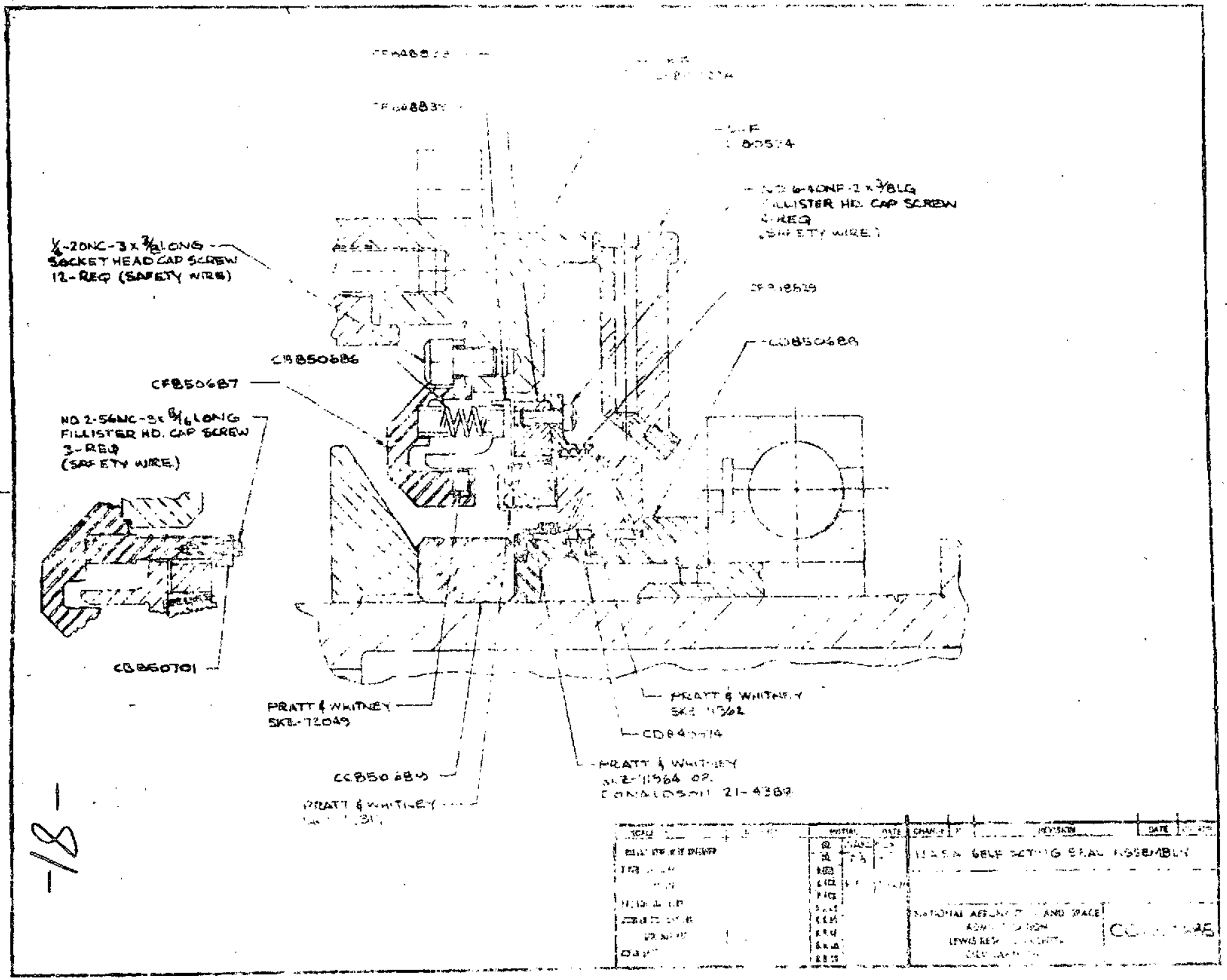
AL73T024

ENCLOSURE 14

NASA HYDRODYNAMIC LIFT SHAFT SEAL
USED IN OIL POSITION



RESEARCH LABORATORY SKF INDUSTRIES, INC.

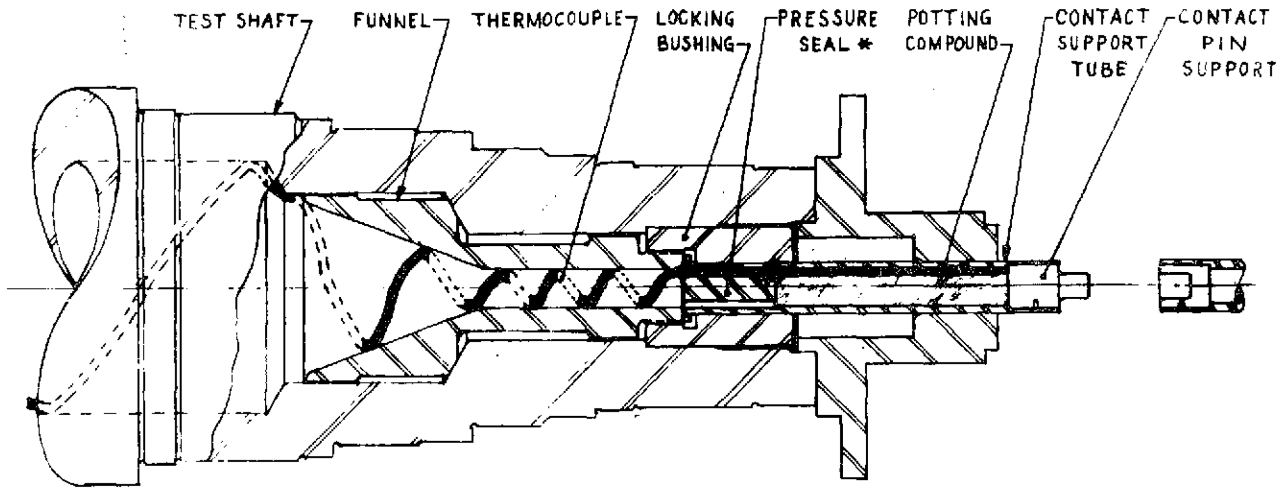
ENCLOSURE 15

TEST SHAFT ELECTRICAL EXTENSION TUBE

L 41127

TOLERANCES UNLESS OTHERWISE SPECIFIED

FRACTIONS	±1/64
DECIMALS	±.005
ANGLES	±1/2°
SURFACE FINISH	125



* THERMOCOUPLE SILVER SOLDERED IN POSITION

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L 41176
USED ON
ASSEMBLY

SKF INDUSTRIES, INC. PHILADELPHIA, PA.

PRESSURE SEAL ASSY
SST 901 TEST RIG
(ENCLOSURE)

SCALE FULL
DATE 17 NOV 65

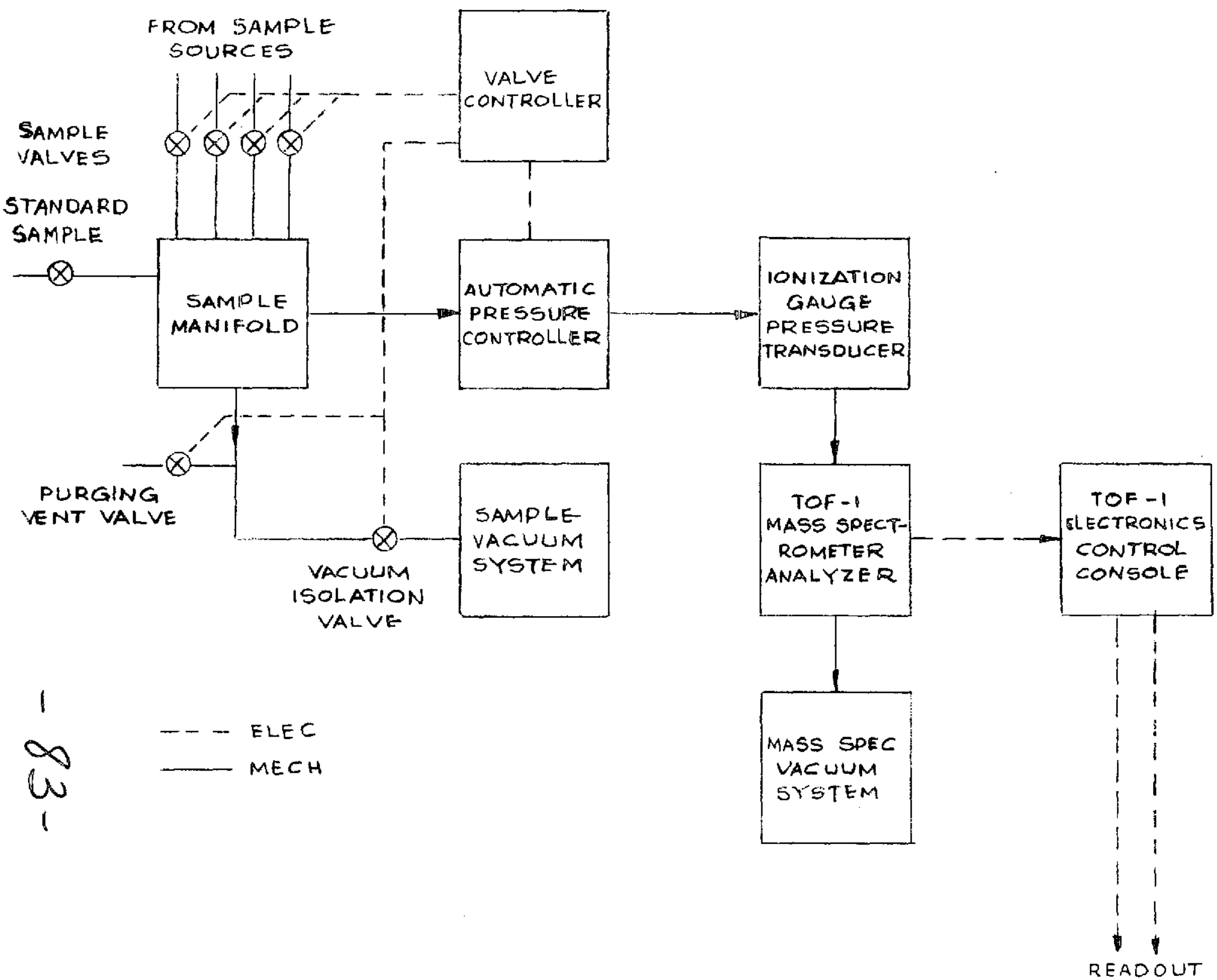
L 41127

AL73T024

ENCLOSURE 16

AL73T024

MASS SPECTROMETER GAS SAMPLING SYSTEM SCHEMATIC



76272-4

BEARING DATA		
1.	VENDORS PART NO	439981 G-1
2.	VENDOR'S DRAWING NO.	76272-3
3	BEARING TOLERANCE	ABEC7
4	NO & SIZE OF BALLS	21-13/16 ±.010 DIA.
5	INTERNAL RADIAL LOOSENESS (UNMOUNTED)	.0078"-.0084"
6.	MAX. END PLAY (UNMOUNTED)	.030"
7.	DESIGN CONTACT ANGLE	28°30'-32°30'
8.	OUTER GROOVE RADIUS AS PERCENT OF BALL DIA	52.00-52.24
9.	INNER GROOVE RADIUS AS PERCENT OF BALL DIA.	51.63-51.75
10.	SHIM WIDTH	.012
11.	PITCH DIA.	6.280"
12	CAGE WIDTH (MAX)	1.024"
13	CAGE SPEC. (FOR OUT OF ROUNDNESS & SQUARENESS)	470939
14	CAGE POCKET CLEARNESS	.019-.032
15	BALL GRADE (AFBMA)	15
16	INNER RING MATERIAL	M-50
17	OUTER RING MATERIAL	M 50
18	BALL MATERIAL	M 50
19	CAGE MATERIAL	4340
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	35 MIN
26	CAGE RUNNING CLEARANCE ON DIA	.030"-.040"
27	CAGE O. DIA 'L'	6.671"-6.663
28	CAGE I. DIA 'K'	6.113"-6.121
29	CAGE PLATING — TYPE	SILVER AMS 2410 #
	CAGE PLATING — THICKNESS	.001"-.002"

CAGE TO BE 100% FLUORESCENT PENETRANT INSPECTED (470908)
100% VISUAL INSPECTION (470659).
100% MAGNAFLUX INSPECTION (471682)
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:

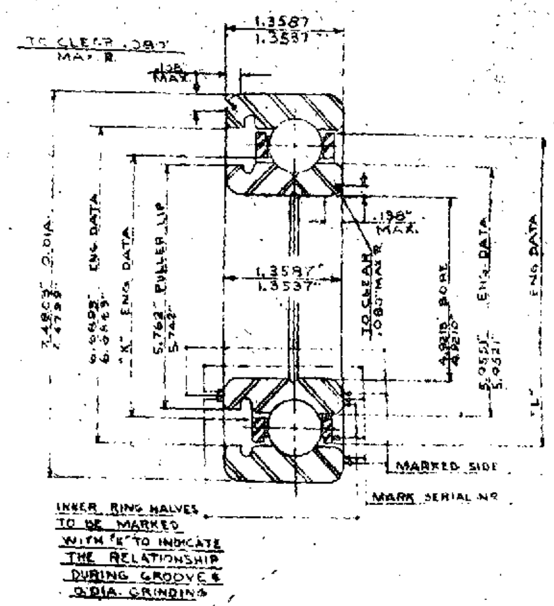
INNER RING BORE	16	RMS	MAX
INNER RING GROOVE	4	"	"
OUTER RING O. DIA.	16	"	"
OUTER RING GROOVE	4	"	"
CAGE SUPPORT SURFACE	20	"	"
BALL	1.2	"	"
OUTER RING BORE	4	"	"
INNER RING O. DIA	4	"	"

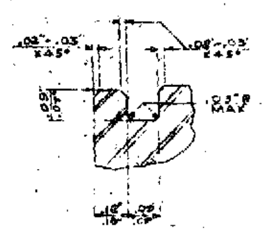
MAX. CAGE UNBALANCE 3GM-CM, BALANCE TO BE CHECKED AT 500RPM (MIN)

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 HIEGHT OF LAND TO O. DIA. EQUAL WITHIN .0004" IN SAME AXIAL PLANE
D. 80% CONTACT WHEN ROLLED ON A BLUED SUPPORT

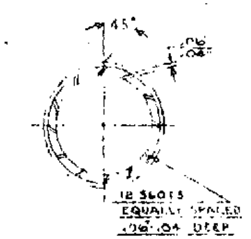
CAGE SUPPORT
A. 3 POINT OUT-OF-ROUND WITHIN .003" F.I.R.
B. 2 POINT DIAS. EQUAL WITHIN .002" IN SAME AXIAL PLANE
C. SQUARENESS TO COMMON MACHINING REF FACE WITHIN .002"
D. 80% CONTACT WHEN ROLLED ON A BLUED FLAT PLATE

CAGE TO BE PLATED BY ELECTRO-SILVER PLATED METHOD





ENLARGED VIEW OF PULLER GROOVE INNER & OUTER RINGS



REDUCED VIEW OF INNER RINGS (BOTH HALVES)

ENCLOSURE 17
TEST BEARING

AL73T024

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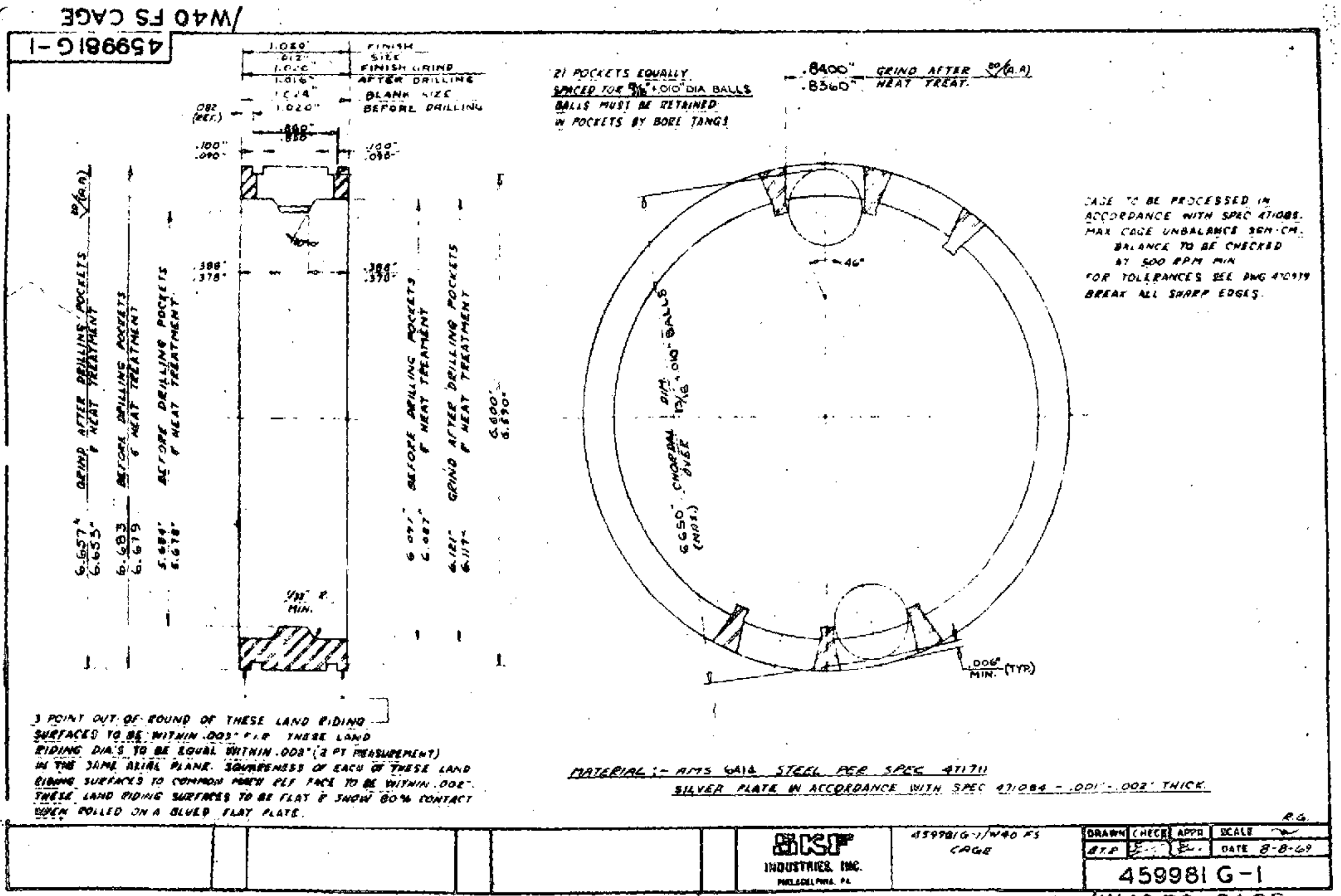
RESEARCH LABORATORY SKF INDUSTRIES, INC.

BRG. NO 439981 G-1
76272-4
CUST.

RESEARCH LABORATORY SKF INDUSTRIES, INC.



/W40 FS CAGE

ENCLOSURE 18

TEST BEARING CAGE

AL73T024

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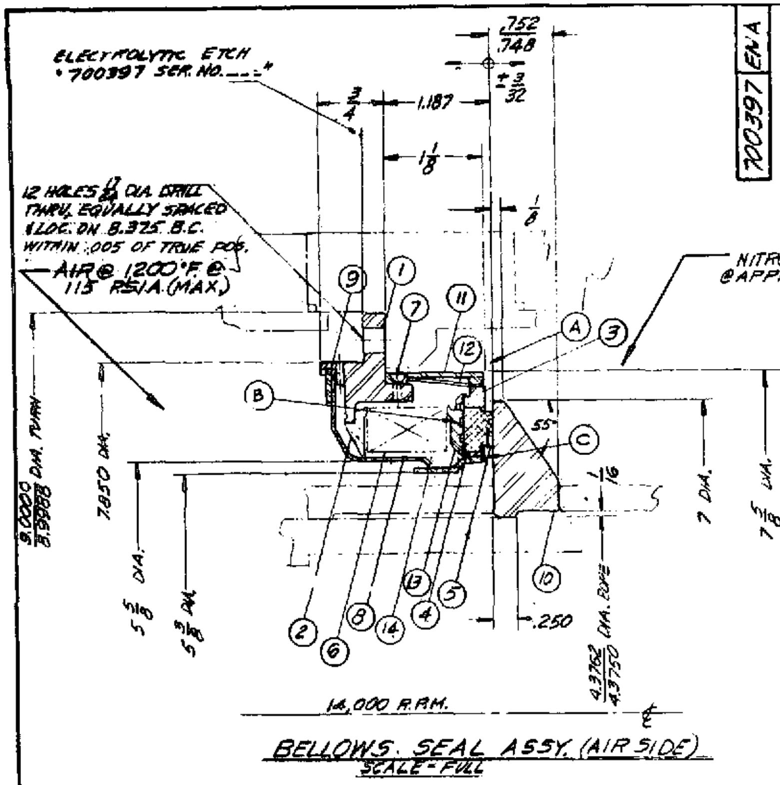
ENCLOSURE 19

TYPICAL KOPPERS SLIDING FACE TYPE SHAFT SEAL USED IN THE AIR SEAL POSITION



BELLOWS SEAL ASSY. (AIR SIDE)
SCALE - FULL

ITEM	PART NAME	MATERIAL	QUANTITY PER UNIT	DETAIL DRAWING NUMBER
14	SEAL RING GUARD	17-4 PH	1	700389-14EN
13	SEAL RING ADAPTER	"	1	" -13EN
12	FINGER SPRING	INCONEL X	6	" -12EN
11	DAMPER	17-4 PH	1	" -11EN
10	SHOULDER (CR.R.)	AMS 6322	1	700397-10EN
9	SNAP RING	410 OR 416 S.S.	1	700389-9EN
8	HEAT SHIELD	AMS-5504	1	" -8EN
7	FLAT HD. MACH. SCREW	300 SERIES C.S.	6	" -7EN
6	BELLOWS ASSEMBLY (.006)	AM 350	1	700397-6EN
5	SEAL RING	PURE P 56 HT	1	" -5EN
4	SEAL RING ADAPTER	17-4 PH	1	700389-4EN
3	SEAL RING RETAINER	"	1	" -3EN
2	BELLOWS ADAPTER	"	1	" -2EN
1	HOUSING	"	1	" -1EN

5.830-6.830 DIA. (AIR SIDE) BELLOWS SEAL - S.K.F.

REV.	DESCRIPTION	CHK'D	DATE
EN A	REVISED TO DETAIL	BLAIR	1-20-65

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KOPPERS COMPANY, INC.
METAL PRODUCTS DIVISION
KUCHLER-HUHN SEALS
BALTIMORE, 3, MARYLAND

DRAWN	CHECKED	APPROVED		
BLAIR 1-20-65	T.K.W. 1-20-65	TASCH 1-20-65	700397	N A

STANDARD TOLERANCES UNLESS OTHERWISE SPECIFIED WILL BE: DECIMAL ± .005, FRACTIONAL ± 1/64, ANGULAR ± 2°

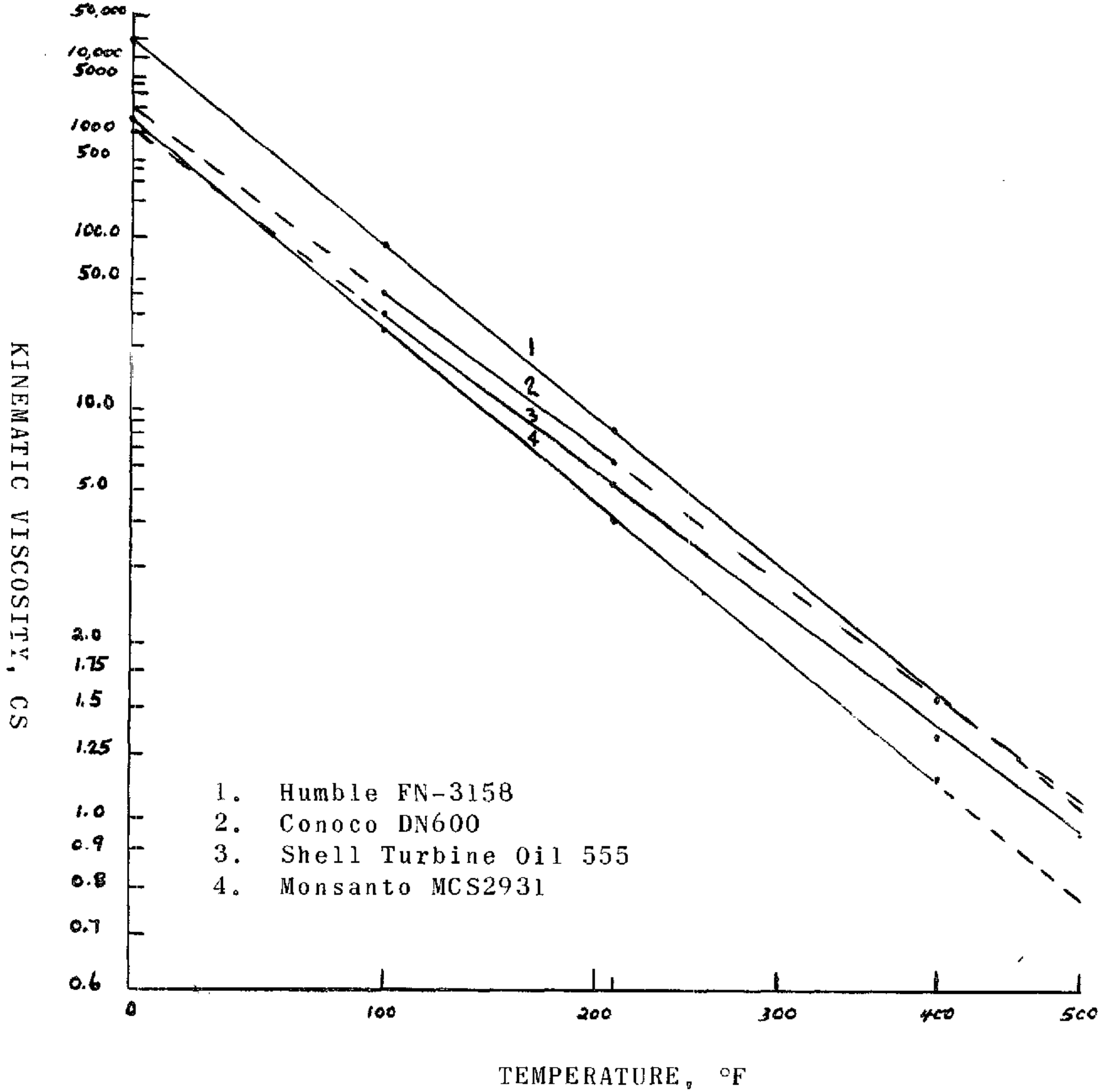
NOTE:
1. REF. S.K.F. DWG. # SST #901 (L-100111)

AL73T024

ENCLOSURE 20

KINEMATIC VISCOSITY VS. TEMPERATURE
FOR LUBRICANTS TESTED

AL73T024



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RESEARCH LABORATORY SKF INDUSTRIES, INC.

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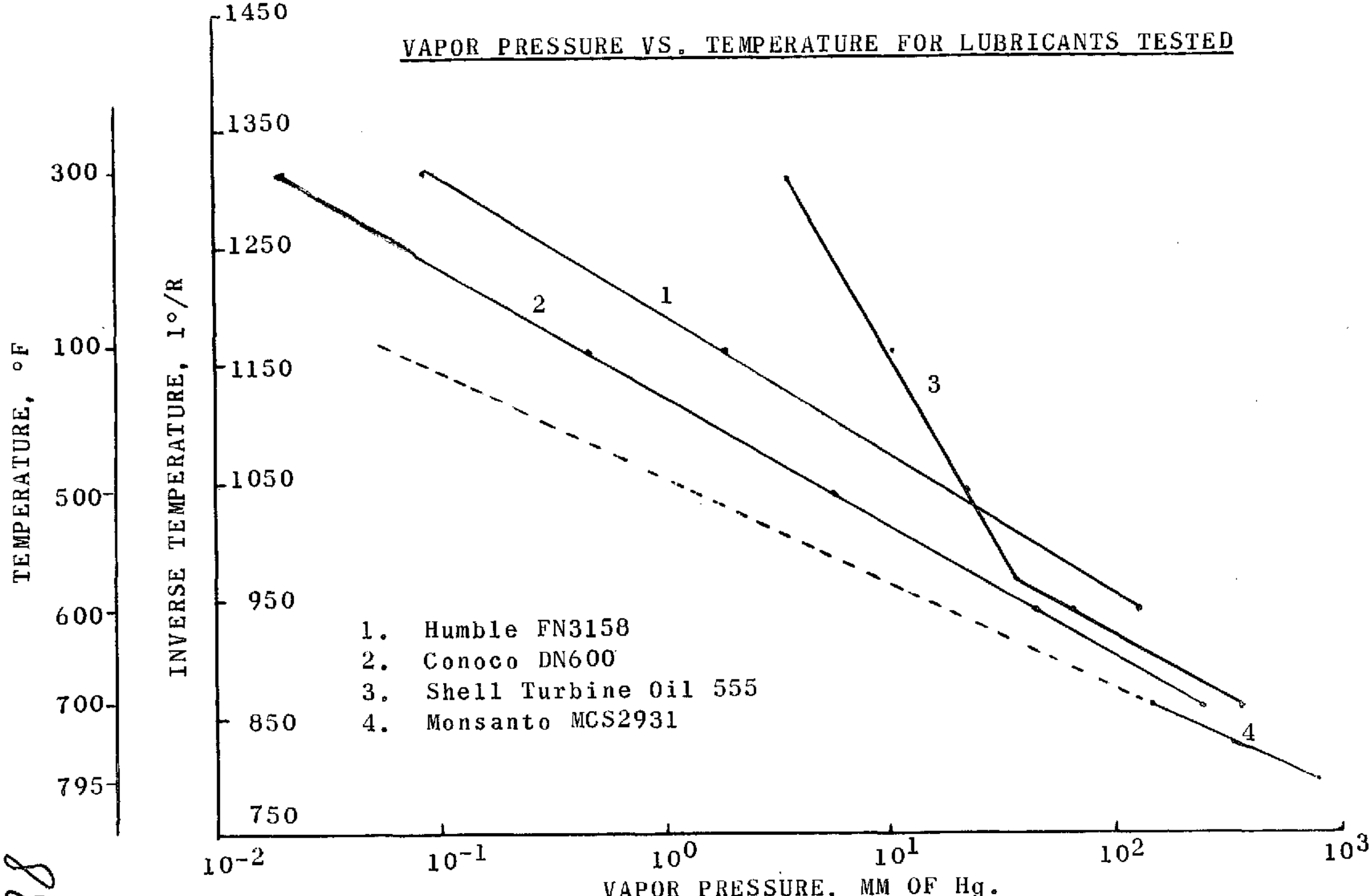
ENCLOSURE 21

VAPOR PRESSURE VS. TEMPERATURE FOR LUBRICANTS TESTED



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RESEARCH LABORATORY SKF INDUSTRIES, INC.

ENCLOSURE 22

LUBRICANT PROPERTIES DATA

Fluid Source Type	FN 3158 Exxon Oil Super-refined Naphthenic Formulated Mineral Oil	MCS 2931 Monsanto Chem. Improved Modified Polyphenyl Ether	DN-600 Type II Continental Oil Formulated Polyalkyl Aromatic Hydrocarbon	Turbine Oil 555 Shell Oil Formulated Advanced Ester
Properties				
Viscosity (cs.)°F				
-30			8,945	10,750
0	10,289	10,690		
100	78.1	24.1	39.3	29.3
210	8.3	4.1	6.25	5.35
400	1.65	1.15		1.38
500				.95
Pour Point°F	-30	-20	-65	-75
Density gms/ml°F				
0	0.908	1.225		
77			0.8756 @ 60°F	0.9937 @ 60°F
100		1.183		0.940 @ 200°F
300		1.100		
400	0.768			
Surface Tension Dynes/cm. at 77°F	30.9	44.9	25	
Thermal Stability Isoten. Decomp. Temp.°F	ca 620	ca 675	ca 640	
Aut. Ig. Temp. °F	735	910		790
Flash Point °F	450	465	430	500
Fire Point °F	495	530		
Volatility, %wt. loss after 6.5 hrs. @400°F	14.3		6.5-7	2-3
@500°F		50.7		
Vapor Pressure, mm of Hg.				
300°F	0.085		0.0189	3.5
400	1.85		0.456	10.0
500	21		5.66	22.0
600	120		43.8	63.0
700		440	237.6	100.0
Neutralization No.	0.2	0.23	1.65	0.14
Thermal Cond. BTU/ft.^{2}/hr./°F/ft.				
100°F				0.098
300°F				0.078
500°F				0.075
Specific Heat BTU/lb./°F				
100°F				0.466
200°F				0.501
300°F				0.542
400°F				0.580

ENCLOSURE 23

NAS3-14320, TABULATION OF TEST ELEMENTS USED

Test No.	Lubricant	Test Bearing S/N (1.)	Air Seal P/N	Air Seal S/N	Air Seal Mating Ring P/N	Air Seal Mating Ring S/N	Air Seal Mating Ring Plating	Oil Seal P/N	Oil Seal S/N	Oil Seal Mating Ring P/N	Oil Seal Mating Ring S/N	Oil Seal Mating Ring Plating
Screening Tests												
1.	Humble FN3158 & 5% Kendall 0839 (new)	4 (new)	Koppers 700495	$1^{(2)}$ (used)	Koppers 700397	$3^{(3)}$	chromium	Koppers 1010568	$2^{(2)}$ (used)	Koppers 1010558	$1^{(3)}$	tungsten carbide
2a.	Monsanto MCS-2931 (new)	1 (new)	"	"	"	$1^{(3)}$	chromium carbide	"	$1^{(3)}$	"	$2^{(3)}$	tungsten carbide
2b.	Monsanto MCS-2931	1 (not changed)	"	"	Koppers 700405	$1^{(3)}$	"	"	$2^{(3)}$	"	$1^{(3)}$	chromium carbide
3.	Conoco DN-600 Type 2 (new)	5 (new)	"	"	"	"	"	"	"	"	"	"
4.	Aeroshell Turbine Oil 555 (new)	7 (new)	"	"	"	"	"	"	"	"	"	"
Extended Duration Tests												
5.	Aeroshell Turbine Oil 555 (Test 4 oil rerun w/o chg.)	7 (not changed)	"	"	"	"	"	"	"$^{(3)}$	"	"	"
6.	Humble FN3158 & 5% Kendall 0839 (new)	11 (new)	"	"	"	"	"	NASA CC850685	02721 (new)	NASA CD849671	1 (new)	
7.	Humble FN3158 & 5% Kendall 0839 (new)	12 (new)	"	"	"	"	"	"	"$^{(3)}$	"	2 (new)	

Notes:
(1.) SKF bearing P/N 459931G-1 was used without change in all testing.
(2.) These seals previously used on NAS3-14310.
(3.) Refurbished prior to this test.

ENCLOSURE 24

NAS3-14320, SUMMARY OF TEST RESULTS

Test No.	Lubricant	Total Running Time (hrs.)	Max. Speed Reached (rpm)	Oil Flow Rate Range (gpm)	Oil[(1)] Inlet Temp. Range (°F)	Outer[(1)] Ring Temp. Range (°F)	Total Seal Leakage Range (scfm)	Reason For Test Termination	Post Test Condition: Oil		Post Test Condition: Test Bearing	Post Test Condition: Air Seal	Post Test Condition: Oil Seal
Screening Tests:													
1.	Humble FN3158 & 5% Kendall 0839 (new)	13.6	23,600 (wide open)	1.0- 2.0	200- 435	415- 580	5.5- 27.2	Max. rig motor capability	viscosity acid no. dirt content	+28% +50% +150%	good except heavy debris denting	heavy face wear	eccentric mating ring wear, broken secondary (failure)
2a.	Monsanto MCS-2931 (new)	12.0	21,900	1.0- 2.5	200- 425	450- 555	6.4- 26.0	oil seal rubbing failure	not checked		not inspected	good	eccentric mating ring wear, broken secondary (failure)
2b.	Monsanto MCS-2931 (test 2a oil rerun w/o change)	4.3	18,000	1.0- 2.0	205- 360	460- 545	5.5- 9.4	test bearing smearing failure	viscosity acid no. dirt content	+10% 0% +46%	gross smearing (failed)	good	good condition
3.	Conoco DN-600, Type 2 (new)	4.2	16,000	1.5- 2.0	300- 385	420- 510	8.1- 10.6	chemical degradation of oil	viscosity acid no. dirt content	+40% high high	good except light debris denting	good	good ex. slight seal dam chipping
4.	Aeroshell Turbine Oil 555 (new)	8.4	21,400	1.0- 2.0	255- 390	420- 540	7.2- 13.6	brg. temp. past rig's control capability	viscosity acid no. dirt content	+7% -55% N.C.	good except light debris denting	good	moderate seal dam chipping and saddle shaped nosepiece wear
Extended Duration Tests													
5.	Aeroshell Turbine Oil 555 (test 4 oil rerun w/o change)	26.7	20,000	2.0	220- 400	470- 515	8.1- 16.0	oil seal rubbing failure	viscosity acid no. dirt content	+11% -47% +47%	good except grain boundary corrosion (not lube related)	good	nosepiece and mating ring wear; plating cracks (failure)
6.	Humble FN-3158 & 5% Kendall 0839 (new)	2.1	10,800	1.0- 1.5	200- 355	360- 460(2)	8.1- 16.6	oil seal (NASA) rubbing failure	viscosity acid no. dirt content	+11% N.C. +183%	good	good	severe nosepiece and mating ring wear (pads reversed, failure)
7.	Humble FN-3158 & 5% Kendall 0839 (new)	6.3	20,100	2.0	220- 400			oil seal (NASA) rubbing failure	viscosity(3) acid no. dirt content	+23% N.C. +125%	good	good	nosepiece and mating ring wear; nosepiece dislocation on retainer (failure)

Notes:

(1) Excluding warm-up phase.

(2) Bearing never reached full warm-up. Severe coking was the result of high temperatures generated by oil seal with lift geometry reversed.

(3) Represents 1.6 hours running time only. Oil was changed at 4.7 hours.

RESEARCH LABORATORY SKF INDUSTRIES, INC.

AL73T024

ENCLOSURE 23

NAS3-14320, LUBRICANT CHEMICAL ANALYSIS DATA AND RIG COKING EXTENT

Test No.	Lubricant	Accumulated Time on Oil (hrs.)	Viscosity @ 100°F (cs.)	Acid No.	Dirt Content (gm/100 ml.)	Oil Added at Time (gals.)	Extent of Rig Coking
Screening Tests							
1	Humble FN3158 &	0	94.0	0.20	0.020(5.)	0.4 at 2.3 hrs.	severe
	5% Kendall 0839	13.6	119.9	0.30	0.050	0.5 at 3.2 hrs.	
2	Monsanto MCS2931	0	24.8	0.06	0.039(5.)	1.0 at 2.4 hrs.	average
		16.3	27.2	0.06	0.057	0.5 at 4.0 hrs.	
3	Conoco DN-600,	0	38.0	basic	0.042	none	severe
	Type 2	4.2	53.0	0.30	unfilterable(2)		
4	Aeroshell Turbine	0	28.6	0.38	0.024	none	none
	Oil 555	8.4	30.6	0.17	0.024		
Extended Duration Tests							
5	Aeroshell Turbine	18.5	31.3	0.20	0.023(5.)	1.2 at 15.8 hrs.	mild
	Oil 555	21.7	30.9	0.20	0.035		
		35.1	31.7	0.20	0.035		
6	Humble FN3158 &	0	94.0	0.20	0.020(5.)	none (oil changed	severe
	5% Kendall 0839	1.1(4.)	103.9	0.20	0.057	at 1.05 hrs.)	
7	Humble FN3158(8.)	0	96.4	0.10	0.018(5.)	none (oil changed	mild
	& 5% Kendall 0839	4.7(6.)	167.2	0.30	0.036	at 4.7 hrs.)	
		1.6(7.)	115.8	0.20	0.045		

Notes:

(1) The 5 possible rig coking ratings are, in order; extreme, severe, average, mild, none. They are relative only to each other.

(2) Fluid would not pass through filter.

(3) This and the following times on test 5 include the 8.4 hrs. run on this oil in the previous test 4.

(4) This actually represents 2.15 hrs. in test time however oil was changed at 1.05 hours after discovery of a pre-test overheating condition.

(5) Oil shaft seal failures during these tests could have contaminated the lubrication system and therefore cast suspicion on the validity of final dirt content levels as an indication of oil performance.

(6) This oil was subjected to heat of unknown magnitude by rig roller bearing failure.

(7) This actually represents 6.3 hours in test time however oil was changed at 4.7 hours after roller bearing failure.

(8) Humble FN3158 came from a different lot than that used in tests 1 and 6, therefore had different initial readings.

AL73T024

RESEARCH LABORATORY SKF INDUSTRIES, INC.

TEST BEARING # 368101

OIL USED FN 3158 + 5% KENDALL RESIN

DATE 9/28 - 10/5/71

RUNNING TIME, HOURS	1.1	2.5	2.55	3.1	3.2	3.2	3.2	3.8	4.1	4.3	4.7	4.7
SPEED, RPM X 10^3	14	14	16	18	18			14	16		16	
AIR MANIFOLD PRESS. (PSI)	106	106	106	106	106			106	—		106	
BEARING CAVITY PRESS. (PSI)	6	6	6	6	6			6	—		6	
SEAL CAVITY PRESS. (PSI)	111	111	111	111	111			111	—		111	
HOT AIR FLOW (SCFM)	52	52	-	53				48	—		—	
TEST OIL FLOW (GPM)	1.5	—	—	2.0	2.0			1.5	1.5		—	
TOTAL SEAL LEAKAGE (SCFM)	8.3	—	—	—				9.1	—		—	
TEST BEARING OUTER RING (°F)	506	500	515	550	680			500	525		500	
TEST BEARING INNER RING (°F)	540	—	520	560	600			520	540		—	
ROLLER BEARING OUTER RING (°F)	470	485	485	520	505	STOP: FLAT BELT FAILED	RESTART NEW BELT	480	450	STOP: BELT CAME OFF PULLEY RESTART: RESEATED BELT	465	STOP: BELT CAME OFF PULLEY
OIL SEAL HOUSING (°F)	—	—	—	—	—			—	—		—	
AIR SEAL HOUSING (°F)	815	865	855	870	865			860	860		885	
TEST BEARING HOUSING (°F)	450	435	435	460	465			450	430		390	
ROLLER BEARING HOUSING (°F)	400	375	375	380	445			400	380		340	
AIR SEAL BELLOWS (°F)	—	—	—	—	—			—	—		—	
HOT AIR IN MANIFOLD (°F)	970	1010	1005	1005	1005			980	980		1000	
OIL INLET (°F)	410	—	415	435	4			410	—		—	
OIL OUTLET (°F)	460	465	465	495	480			480	460		475	

Test Data

Test No. 1, page 1 of 4

Enclosure 26

AL73T024

TEST BEARING # 368101

OIL USED FN 3158 + 5% KENDALL RESIN

DATE 10/5 - 10/8/71

RUNNING TIME, HOURS	4.7	5.3	5.5	5.6	5.6	6.1	6.3	6.7	7.1	7.4	7.8	8.2
SPEED, RPM $\times 10^3$		16	16			14	16	18	18	20	20	22
AIR MANIFOLD PRESS. (PSI)		106	106			106	106	106		106	—	—
BEARING CAVITY PRESS. (PSI)		6	6			6	2.22	—		6-8	—	—
SEAL CAVITY PRESS. (PSI)		111	111			111	111	111		111	—	—
HOT AIR FLOW (SCFM)		—	—			—	—	—		—	—	—
TEST OIL FLOW (GPM)		1.75	1.0			1.5	1.5	1.5		1.5	1.7	2.0
TOTAL SEAL LEAKAGE (SCFM)		5.5				7.6	7.2	—		—	11.0	—
TEST BEARING OUTER RING (°F)		510	565			450	490	500		510	560	570
TEST BEARING INNER RING (°F)		—	565			—	505	555		525	—	—
ROLLER BEARING OUTER RING (°F)		480	480	BELT SLIPPING CAN'T GET ABOVE 12,600 RPM		400	455	495	OIL COOLING ON	490	490	530
OIL SEAL HOUSING (°F)		—	—			—	—	—		—	—	—
AIR SEAL HOUSING (°F)		830	855			720	815	875		885	845	845
TEST BEARING HOUSING (°F)	RESTART	430	425		RESTART	350	370	420		395	440	440
ROLLER BEARING HOUSING (°F)		380	365			320	320	350		330	370	370
AIR SEAL BELLOWS (°F)		—	—			—	—			—	—	
HOT AIR IN MANIFOLD (°F)		940	9.5			890	940	1000		1010	1020	1020
OIL INLET (°F)		—	—			—	380	400		330	410	390
OIL OUTLET (°F)		495	495			410	450	490		480	500	500

BELT FAILURE

Test Data

Test No. 1, page 2 of 4

Enclosure 26

RESEARCH LABORATORY SKF INDUSTRIES, INC.

TEST BEARING # 368101

OIL USED FN 3158 + 5% KENDALL RESIN DATE 10/8 - 10/26/71

RUNNING TIME, HOURS	8.2	8.4	8.4	9.0	9.2	9.5	9.6	9.7	10.3	10.7	10.7	11.5
SPEED, RPM	RESTARTED AFTER RESEATING BELT	22.5 STOP BELT FLYING APART RESTART	14	16	16	18	BELT SLIPPED SPEED DOWN TO 16,000 RPM	STOP: COULD NOT GET TO 18,000 RPM RESTART	16	STOP TRIED TO GO TO 20,000 RPM BELT SLIPPED OFF PULLEY	RESTART	14
AIR MANIFOLD PRESS. (PSI)			106	106	106	—			106			106
BEARING CAVITY PRESS. (PSI)			5-7	5-7	6	—			6			6
SEAL CAVITY PRESS. (PSI)			111	111	111	—			111			111
HOT AIR FLOW (SCFM)			48	—	48	—			48			46
TEST OIL FLOW (GPM)			1.5	1.5	1.5	—			1.5			1.5
TOTAL SEAL LEAKAGE (SCFM)			14.9	—	14.0	13.1			18.3			25.5
TEST BEARING OUTER RING (°F)			440	490	500	520			460			415
TEST BEARING INNER RING (°F)			450	490	505	—			470			430
ROLLER BEARING OUTER RING (°F)			430	455	480	480			450			415
OIL SEAL HOUSING (°F)			—	—	—	—			—			—
AIR SEAL HOUSING (°F)			740	820	855	880			840			800
TEST BEARING HOUSING (°F)			350	365	390	400			395			310
ROLLER BEARING HOUSING (°F)			315	320	340	355			360			270
AIR SEAL BELLOWS (°F)			—	—	—	—			560			—
HOT AIR IN MANIFOLD (°F)			840	935	965	1015			940			935
OIL INLET (°F)			370	—	400	—			375			320
OIL OUTLET (°F)			425	450	475	415			435			390

Test Data

Test No. 1, page 3 of 4

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AL73T024

RESEARCH LABORATORY SKF INDUSTRIES, INC.

TEST BEARING # 368101

OIL USED FN3158 + 5% KENDALL RESIN DATE 10/26/71

RUNNING TIME, HOURS	11.6	11.8	12.3	12.4	12.6	12.9		13.2	13.4	13.6		
SPEED, RPM	16		18	20	20	22		22	23.6			
AIR MANIFOLD PRESS. (PSI)	106		106	106	106	106		106	106			
BEARING CAVITY PRESS. (PSI)	6		6	6	6	6		6	6			
SEAL CAVITY PRESS. (PSI)	111		111	111	111	111		111	111			
HOT AIR FLOW (SCFM)	43		52	—	—	—		—	—			
TEST OIL FLOW (GPM)	1.5		1.5	1.5	2.0	2.0		1.5	2.0			
TOTAL SEAL LEAKAGE (SCFM)	27.2		23.0	24.6	24.6	24.6		—	—			
TEST BEARING OUTER RING (°F)	450		510	485	470	510		560	540			
TEST BEARING INNER RING (°F)	440		510	495	470	—		—	—			
ROLLER BEARING OUTER RING (°F)	430		505	455	440	500		445	460			
OIL SEAL HOUSING (°F)	—		—	—	—	—		—	—			
AIR SEAL HOUSING (°F)	—	OIL COOLING ON	865	870	870	890	STOP. LOW AIR PRESSURE RESTART	820	840	STOP HEATER BECAUSE UNABLE TO SEE FLOW OF RIG SEAL LIFT OFF		
TEST BEARING HOUSING (°F)	330		420	380	340	375		320	350			
ROLLER BEARING HOUSING (°F)	285		305	360	320	325		295	310			
AIR SEAL BELLOWS (°F)	—		—	—	—	—		—	—			
HOT AIR IN MANIFOLD (°F)	975		1005	1000	1020	1015		990	990			
OIL INLET (°F)	345		385	310	—	360		280	320			
OIL OUTLET (°F)	410		490	425	400	470		405	420			

Test Data

Test No. 1, page 4 of 4

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AL73T024

RESEARCH LABORATORY SKF INDUSTRIES, INC.

TEST BEARING # 368102

OIL USED MONSANTO MCS 2931

DATE 12/14

RUNNING TIME, HOURS	1.4	1.5	1.5	2.0	3 0	4.0	5.0	6.0	6.5	6.5	7.3	7.5
SPEED, RPM	14	STOP: SPEED DROPPED CHECK BELT AND PULLEY	RESTART	14	14	14	14	14	14	STOP: END OF DAY: RESTART	14	14
AIR MANIFOLD PRESS. (PSI)	54			54	54	54	54	54	54		106	106
BEARING CAVITY PRESS. (PSI)	6			6	6	6	6	6	6		6	6
SEAL CAVITY PRESS. (PSI)	58			58	58	58	58	58	58		111	111
HOT AIR FLOW (SCFM)	49			49	48	47	47	47	47		49	—
TEST OIL FLOW (GPM)	2.0			1.75	1.75	2.0	2.0	2.0	2.0		2.0	2.0
TOTAL SEAL LEAKAGE (SCFM)	—			6.8	6.8	6.4	8.5	8.1	8.5		11.05	15.0
TEST BEARING OUTER RING (°F)	500			460	510	510	515	515	515		450	475
TEST BEARING INNER RING (°F)	—			—	520	520	520	520	520		—	500
ROLLER BEARING OUTER RING (°F)	480			450	485	495	490	490	445		460	475
OIL SEAL HOUSING (°F)	—			—	—	—	—	—	—		—	—
AIR SEAL HOUSING (°F)	725			695	725	730	735	735	735		690	710
TEST BEARING HOUSING (°F)	405			390	425	440	430	440	440		400	410
ROLLER BEARING HOUSING (°F)	350			340	350	365	360	360	365		360	380
AIR SEAL BELLOWS (°F)	—			—	—	—	—	—	—		—	—
HOT AIR IN MANIFOLD (°F)	860			860	870	875	865	875	870		920	925
OIL INLET (°F)	405			365	410	425	420	415	415		350	380
OIL OUTLET (°F)	500			455	500	500	490	500	500		450	470

Test Data

Test No. 2, page 1 of 5

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AL73T024

TEST BEARING # 368102

OIL USED MONSANTO MCS 2931

DATE 12/17-20/71

RUNNING TIME, HOURS	7.8	8.1	8.15	9.1	9.35	9.5	9.7	10.0	10.1	10.2	10.5	10.8
SPEED, RPM	14	14		16	16	16	16	18	18	18	18	18
AIR MANIFOLD PRESS. (PSI)	106	106	STOP: LOW PRESSURE COMPRESSOR, RESTART	106	106	106	106	106	106	106	106	106
BEARING CAVITY PRESS. (PSI)	6	6		6	6	6	6	6	6	6	6	6
SEAL CAVITY PRESS. (PSI)	111	111		111	111	111	111	111	111	111	111	111
HOT AIR FLOW (SCFM)	—	—		—	—	—	—	—	—	—	—	—
TEST OIL FLOW (GPM)	1.5	1.0		2.0	1.5	1.25	2.5	2.0	2.0	1.5	1.0	1.0
TOTAL SEAL LEAKAGE (SCFM)	13.0	16.0		15	12	12	11	16	13	14	14	17
TEST BEARING OUTER RING (°F)	500	515		525	545	555	540	490	480	500	495	490
TEST BEARING INNER RING (°F)	—	—		—	—	—	—	—	—	—	—	—
ROLLER BEARING OUTER RING (°F)	510	510		500	510	530	485	455	450	440	445	440
OIL SEAL HOUSING (°F)	—	—		—	—	—	—	—	—	—	—	—
AIR SEAL HOUSING (°F)	730	730		730	740	765	780	775	765	765	750	750
TEST BEARING HOUSING (°F)	435	435		420	450	455	430	400	365	355	330	290
ROLLER BEARING HOUSING (°F)	415	410		400	400	405	385	350	315	305	280	265
AIR SEAL BELLOWS (°F)	—	—		—	—	—	—	—	—	—	—	—
HOT AIR IN MANIFOLD (°F)	940	940		960	985	990	995	1010	1020	1015	1000	1000
OIL INLET (°F)	370	330		380	400	380	350	280	280	270	220	200
OIL OUTLET (°F)	480	465		500	495	510	470	440	420	415	395	370

Test Data

Test No. 2, page 2 of 5

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AL73T024

TEST BEARING # 368102

OIL USED MONSANTO MCS 2931

DATE 12/20/71 - 3/20/72

RUNNING TIME, HOURS	10.9	11.1	11.15	11.5	11.6	11.9	12.0	13.1	13.4	13.5	14.0	14.2
SPEED, RPM	20	20	20	20	20	20	STOP; LOW PRESSURE, COMPRESSOR OIL SEAL REPLACED	14	14	14	14	14
AIR MANIFOLD PRESS. (PSI)	—	—	—	—	—	—		106	106	106	106	106
BEARING CAVITY PRESS. (PSI)	—	—	—	—	—	—		6	6	6	6	6
SEAL CAVITY PRESS. (PSI)	—	—	—	—	—	—		111	111	111	111	111
HOT AIR FLOW (SCFM)	—							—	—	—	—	—
TEST OIL FLOW (GPM)	2	2	1.5	1.5	1.25	1.25		2.0	2.0	1.5	1.5	1.0
TOTAL SEAL LEAKAGE (SCFM)	14	14	13	13	16	26		6.4	5.5	6.0	6.8	6.8
TEST BEARING OUTER RING (°F)	500	520	520	515	525	515		470	470	520	460	535
TEST BEARING INNER RING (°F)	—	—	—	—	—	—		490	500	555	500	565
ROLLER BEARING OUTER RING (°F)	440	460	460	450	460	425		430	450	455	420	425
OIL SEAL HOUSING (°F)	—	—	—	—	—	—		—	—	—	—	—
AIR SEAL HOUSING (°F)	755	780	780	775	770	770		800	800	785	770	795
TEST BEARING HOUSING (°F)	330	360	360	345	330	305		335	350	355	320	370
ROLLER BEARING HOUSING (°F)	270	290	290	290	285	275		280	295	300	295	330
AIR SEAL BELLOWS (°F)	—	—	—	—	—	—		660	670	670	640	660
HOT AIR IN MANIFOLD (°F)	1020	1020	1010	1010	1005	1005		945	945	950	940	940
OIL INLET (°F)	250	270	270	250	220	200		320	320	360	300	345
OIL OUTLET (°F)	410	430	430	420	410	390		420	425	460	410	455

end phase 2a, start phase 2b

Test Data

Test No. 2, page 3 of 5

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AL73T024

RESEARCH LABORATORY SKF INDUSTRIES, INC.

TEST BEARING # 368102

OIL USED MONSANTO MCS 2931

DATE 3/20/72

RUNNING TIME, HOURS	14.3	14.5	14.7	14.9	15.0	15.2	15.3	15.5	15.6	15.8	15.9	16.0
SPEED, RPM	14	16	16	16	16	16	16	16	18	18	18	18
AIR MANIFOLD PRESS. (PSI)	106	106	106	106	106	106	106	106	106	106	106	106
BEARING CAVITY PRESS. (PSI)	6	6	6	6	6	6	6	6	6	6	6	6
SEAL CAVITY PRESS. (PSI)	111	111	111	111	111	111	111	111	111	111	111	111
HOT AIR FLOW (SCFM)	—	—	—	—	—	—	—	—	—	—	—	—
TEST OIL FLOW (GPM)	1.0	2.0	2.0	2.0	1.5	1.5	1.0	1.0	2.0	2.0	2.0	1.5
TOTAL SEAL LEAKAGE (SCFM)	6.0	6.8	7.2	7.2	7.2	7.6	8.1	8.1	9.4	7.6	8.9	9.4
TEST BEARING OUTER RING (°F)	545	480	480	470	480	485	500	500	480	500	520	510
TEST BEARING INNER RING (°F)	575	505	505	500	510	510	545	550	550	555	570	545
ROLLER BEARING OUTER RING (°F)	500	465	450	465	460	460	445	490	455	465	470	465
OIL SEAL HOUSING (°F)	—	—	—	—	—	—	—	—	—	—	—	—
AIR SEAL HOUSING (°F)	800	795	820	825	825	825	830	835	775	780	785	780
TEST BEARING HOUSING (°F)	400	390	365	360	360	360	375	385	370	375	385	385
ROLLER BEARING HOUSING (°F)	350	350	330	335	340	340	345	350	340	350	350	350
AIR SEAL BELLOWS (°F)	670	660	655	660	660	660	660	665	650	655	660	660
HOT AIR IN MANIFOLD (°F)	940	960	965	970	970	970	970	970	970	970	970	970
OIL INLET (°F)	360	285	280	270	260	255	220	210	260	270	270	205
OIL OUTLET (°F)	475	440	430	425	425	420	410	405	430	445	450	440

Test Data

Test No. 2, page 4 of 5

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RESEARCH LABORATORY SKF INDUSTRIES, INC.

TEST BEARING # 368102

OIL USED MONSANTO MCS-2931

DATE 3/20/72

RUNNING TIME, HOURS	16.2	16.25	16.3									
SPEED, RPM	18	18										
AIR MANIFOLD PRESS. (PSI)	106	106										
BEARING CAVITY PRESS. (PSI)	6	6										
SEAL CAVITY PRESS. (PSI)	111	111										
HOT AIR FLOW (SCFM)	—	—										
TEST OIL FLOW (GPM)	1.5	1.0										
TOTAL SEAL LEAKAGE (SCFM)	8.5	—	FAILURE									
TEST BEARING OUTER RING (°F)	500	540										
TEST BEARING INNER RING (°F)	550	—										
ROLLER BEARING OUTER RING (°F)	465	485										
OIL SEAL HOUSING (°F)	—	—										
AIR SEAL HOUSING (°F)	790	790										
TEST BEARING HOUSING (°F)	385	385	BEARING									
ROLLER BEARING HOUSING (°F)	345	345										
AIR SEAL BELLOWS (°F)	660	665										
HOT AIR IN MANIFOLD (°F)	975	975										
OIL INLET (°F)	230	—										
OIL OUTLET (°F)	415	405										

Test Data

Test No. 2, page 5 of 5

Enclosure 26

AL73T024

RESEARCH LABORATORY SKF INDUSTRIES, INC.

TEST BEARING # 368103

OIL USED CONOCO DN600 FLUID

DATE 4/13-17/72

Running Time, Hours	0.6	1.1	1.3	1.5	2.2	2.5	2.6	2.9	3.1	3.6	3.9	4.2
Speed, RPM		14	14		14	14	14	14	16	16	16	
Air Manifold Press. (psi)		106	106		106	106	106	106	106	106	106	
Bearing Cavity Press. (psi)		6	6		6	6	6	6	6	6	6	
Seal Cavity Press. (psi)		111	111		111	111	111	111	111	111	111	
Hot Air Flow (SCFM)		—	—		—	—	—	—	—	—	—	
Test Oil Flow (GPM)		—	2.0		2.0	2.0	1.5	1.5	2.0	2.0	1.5	
Total Seal Leakage (SCFM)		—	10.6		8.1	10.2	—	9.4	9.4	8.5	8.1	
Test Bearing Outer Ring (°F)	EXCESSIVE OIL LEAKAGE SHUT DOWN TO REPAIR	420	450	STOP REALIGN BELT	460	470	490	510	470	500	440	STOP: COUPLING BETWEEN PILLOW BLOCKS MAGNETIC CLUTCH AND FAILED
Test Bearing Inner Ring (°F)		—	460		475	480	500	530	480	500	460	
Roller Bearing Outer Ring (°F)		420	450		455	465	475	490	450	470	430	
Oil Seal Housing (°F)		—	—		—	—	—	—	—	—	—	
Air Seal Housing (°F)		770	800		710	740	740	750	750	755	745	
Test Bearing Housing (°F)		360	390		405	420	430	445	405	430	380	
Roller Bearing Housing (°F)		340	360		350	380	380	390	370	385	350	
Air Seal Bellows (°F)		—	—		—	—	—	—	—	—	—	
Hot Air in Manifold (°F)		870	905		870	890	880	890	890	890	890	
Oil Inlet (°F)		335	355		380	—	385	—	330	300	—	
Oil Outlet (°F)		420	440		460	465	475	490	440	475	415	

Test Data

Test 3, page 1 of 1

Enclosure 26

AL73T024

TEST BEARING # 368104

OIL USED AEROSHELL TURBINE OIL 555

DATE 5/10/72

RUNNING TIME, HOURS	1.6	2.3	2.7	3.0	3.1	3.3	4.0	4.1	4.3	4.5	4.8	5.1
SPEED, RPM	14	14	14	14	14	14	14	14	14	16	16	16
AIR MANIFOLD PRESS. (PSI)	106	106	106	106	106	106	106	106	106	106	106	106
BEARING CAVITY PRESS. (PSI)	6	6	6	6	6	6	6	6	6	6	6	6
SEAL CAVITY PRESS. (PSI)	111	111	111	111	111	111	111	111	111	111	111	111
HOT AIR FLOW (SCFM)	—	48	—	48	47	—	—	48	48	50	50	50
TEST OIL FLOW (GPM)	2.0	1.5	1.5	1.5	1.0	1.0	1.0	2.0	2.0	2.0	2.0	1.5
TOTAL SEAL LEAKAGE (SCFM)	8.9	13.6	8.1	8.1	8.0	—	8.9	7.2	—	9.8	9.3	9.3
TEST BEARING OUTER RING (°F)	420	480	500	510	515	500	520	490	490	505	510	530
TEST BEARING INNER RING (°F)	440	500	530	530	545	540	550	520	520	530	540	560
ROLLER BEARING OUTER RING (°F)	360	410	440	435	430	410	430	430	425	430	440	450
OIL SEAL HOUSING (°F)	—	—	—	—	—	—	—	—	—	—	—	—
AIR SEAL HOUSING (°F)	—	—	—	—	—	—	—	—	—	—	—	—
TEST BEARING HOUSING (°F)	360	400	420	430	425	410	425	425	425	425	440	440
ROLLER BEARING HOUSING (°F)	330	365	380	390	385	375	380	385	385	385	390	395
AIR SEAL BELLOWS (°F)	575	595	630	635	635	630	635	635	635	635	640	650
HOT AIR IN MANIFOLD (°F)	875	905	905	910	915	915	910	910	915	920	930	935
OIL INLET (°F)	310	350	385	390	360	350	365	390	390	380	385	390
OIL OUTLET (°F)	420	505	510	515	520	505	520	490	490	505	515	530

Test Data

Test No. 4, page 1 of 2

Enclosure 26

TEST BEARING # 368104

OIL USED AEROSHELL TURBINE OIL 555

DATE 5/10-11/72

RUNNING TIME, HOURS	5.3	6.4	6.7	7.0	7.2	7.3	7.5	7.6	7.8	8.1	8.4	
SPEED, RPM		16	16	18	18	18	20	20	21	21.4		
AIR MANIFOLD PRESS. (PSI)		106	106	106	106	106	106	106	106	106		
BEARING CAVITY PRESS. (PSI)		6	6	6	6	6	6	6	6	6		
SEAL CAVITY PRESS. (PSI)		111	111	111	111	111	111	111	111	111		
HOT AIR FLOW (SCFM)		50	—	—	—	—	—	—	—	—		
TEST OIL FLOW (GPM)		2.0	2.0	2.0	1.5	1.5	2.0	1.5	2.0	2.0		
TOTAL SEAL LEAKAGE (SCFM)		8.1	11.9	7.2	7.5	—	9.3	8.1	7.4	8.9		
TEST BEARING OUTER RING (°F)		490	495	535	520	540	510	526	530	540	END OF TEST	
TEST BEARING INNER RING (°F)	END OF DAY	—	525	550	560	570	540	545	—	—		
ROLLER BEARING OUTER RING (°F)		395	460	490	460	470	450	440	440	450		
OIL SEAL HOUSING (°F)		—	—	—	—	—	—	—	—	—		
AIR SEAL HOUSING (°F)		—	—	—	—	—	—	—	—	—		
TEST BEARING HOUSING (°F)		345	385	405	375	370	355	340	340	350		
ROLLER BEARING HOUSING (°F)		285	320	345	330	320	310	300	295	305	STOP	
AIR SEAL BELLOWS (°F)	OFF	640	645	665	670	670	660	660	660	665		
HOT AIR IN MANIFOLD (°F)		955	960	970	970	970	975	980	980	980		
OIL INLET (°F)		—	375	380	300	345	275	260	255	255		
OIL OUTLET (°F)		425	505	535	530	535	520	530	530	525		

Test Data

Test No. 4, page 2 of 2

Enclosure 26

TEST BEARING # 368104

OIL USED AEROSHELL TURBINE OIL 555 DATE 6/12-14/72

RUNNING TIME, HOURS	0.7	1.6	2.7	3.6	4.7	5.5	6.7	7.7	8.1	8.7	9.7	10.1
SPEED, RPM	14	14	18	18	18		20	20		20	20	
AIR MANIFOLD PRESS. (PSI)	106	106	106	106	106		106	106		106	106	
BEARING CAVITY PRESS. (PSI)	6	6	6	6	6		6	6		6	6	
SEAL CAVITY PRESS. (PSI)	111	111	111	111	111		111	111		111	111	
HOT AIR FLOW (SCFM)	—	51	51	51	58		50	50		46	45	
TEST OIL FLOW (GPM)	2.0	2.0	2.0	2.0	2.0		2.0	2.0		2.0	2.0	
TOTAL SEAL LEAKAGE (SCFM)	16	15.3	12.7	12.3	11.1		—	9.6		8.5	8.1	
TEST BEARING OUTER RING (°F)	480	490	510	470	480		—	510		500	510	
TEST BEARING INNER RING (°F)	—	515	535	500	500		—	—		—	560	
ROLLER BEARING OUTER RING (°F)	470	510	495	480	470	P	—	—	P	—	—	P
OIL SEAL HOUSING (°F)	—	—	—	—	—	O	—	—	O	—	—	O
AIR SEAL HOUSING (°F)	760	785	810	820	815	T	—	820	T	780	800	T
TEST BEARING HOUSING (°F)	390	440	420	395	395	S	385	355	S	330	325	S
ROLLER BEARING HOUSING (°F)	355	430	390	380	375		345	325		270	290	
AIR SEAL BELLOWS (°F)	615	650	655	665	660		650	675		625	645	
HOT AIR IN MANIFOLD (°F)	920	960	975	980	910		920	955		905	940	
OIL INLET (°F)	—	400	365	310	310		—	250		220	225	
OIL OUTLET (°F)	440	490	470	440	440		435	410		400	390	

HOT AIR FLANGE REPAIRED

OIL LEAK

REPLACE BELT

Test Data

Test No. 5, page 1 of 3

Enclosure 26

AL73T024

RESEARCH LABORATORY SKF INDUSTRIES, INC.

TEST BEARING # 368104

OIL USED AEROSHELL TURBINE OIL 555

DATE 6/19-27/72

Running Time, Hours	11.3	12.3	13.3	14.3	15.3	16.3		16.7	17.7	18.7	19.7	20.7
Speed, RPM x 10^3	20	20	20	20	20	20		20	20	20	20	20
Air Manifold Press. (psi)	106	106	106	106	106	106		106	106	106	106	106
Bearing Cavity Press. (psi)	6	6	6	6	6	6		6	6	6	6	6
Seal Cavity Press. (psi)	111	111	111	111	111	111		111	111	111	111	111
Hot Air Flow (SCFM)	52	52	52	52	52	51		42	51	49	49	48
Test Oil Flow (GPM)	2.0	2.0	2.0	2.0	2.0	2.0		2.0	2.0	2.0	2.0	2.0
Total Seal Leakage (SCFM)	10.6	9.8	8.5	10.6	9.8	10.2		9.3	9.1	8.9	10.2	8.9
Test Bearing Outer Ring (°F)	510	515	515	510	510	510		510	510	500	500	500
Test Bearing Inner Ring (°F)	510	540	540	540	540	540		—	535	520	520	525
Roller Bearing Outer Ring (°F)	—	—	—	—	—	—	STOP	—	—	—	—	—
Oil Seal Housing (°F)	—	—	—	—	—	—		—	—	—	—	—
Air Seal Housing (°F)	845	870	870	860	875	880		810	875	870	870	880
Test Bearing Housing (°F)	320	330	330	335	335	330		320	340	330	340	335
Roller Bearing Housing (°F)	280	300	300	300	300	295		275	300	290	300	295
Air Seal Bellows (°F)	620	640	640	640	650	650		620	690	680	680	680
Hot Air in Manifold (°F)	915	955	955	955	970	965		910	995	990	990	990
Oil Inlet (°F)	225	220	220	225	225	225		—	245	240	240	240
Oil Outlet (°F)	385	390	395	395	400	395		385	405	390	395	395

REPLACE BELT

Test Data

Test No. 5, page 2 of 3

Enclosure 26

AL73T026

RESEARCH LABORATORY SKF INDUSTRIES, INC.

TEST BEARING # 368104

OIL USED AEROSHELL TURBINE OIL 555

DATE 6/27/72

Running Time, Hours	21.7	22.7	23.7	24.7	25.7	26.7						
Speed, RPM x 10^3	20	20	20	20	20	20						
Air Manifold Press. (psi)	106	106	106	106	106	106	STOP, END OF TEST PERIOD					
Bearing Cavity Press. (psi)	6	6	6	6	6	6						
Seal Cavity Press. (psi)	111	111	111	111	111	111						
Hot Air Flow (scfm)	49	49	49	49	49	49						
Test Oil Flow (gpm)	2.0	2.0	2.0	2.0	2.0	2.0						
Total Seal Leakage (scfm)	10.6	10.6	10.2	9.8	11.3	10.6						
Test Bearing Outer Ring (°F)	500	500	500	500	500	500						
Test Bearing Inner Ring (°F)	525	535	–	520	535							
Roller Bearing Outer Ring (°F)	–	–	–	–	–	–						
Oil Seal Housing (°F)	–	–	–	–	–	–						
Air Seal Housing (°F)	870	880	875	875	870	880						
Test Bearing Housing (°F)	335	335	330	325	330	330						
Roller Bearing Housing (°F)	290	295	290	290	295	295						
Air Seal Bellows (°F)	675	680	680	670	675	675						
Hot Air in Manifold (°F)	985	990	985	985	995	990						
Oil Inlet (°F)	240	235	235	230	230	230						
Oil Outlet (°F)	390	395	390	385	390	390						

Test Data

Test No. 5, page 3 of 3

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AL73T024

RESEARCH LABORATORY SKF INDUSTRIES, INC.

TEST BEARING # 368105

OIL USED FN-3158 + 5% KENDALL RESIN

DATE 9/6-12/72

RUNNING TIME, HOURS	0.2	0.4	STOP: CHANGE OIL FILTER	0.5	0.65	0.45	1.05	STOP: FLUSH AND REFILL OIL SYSTEM	1.55	2.0	2.1	STOP: BEARING TEMPERATURE EXCESSIVE
SPEED, RPM	2.0	—		5.3	7.7	10.8	10.8		7.7	7.7	10.3	
AIR MANIFOLD PRESS. (PSI)	106	106		106	106	106	106		106	106	106	
BEARING CAVITY PRESS. (PSI)	6	6		6	6	6	6		6	6	6	
SEAL CAVITY PRESS. (PSI)	111	111		111	111	111	111		111	111	111	
HOT AIR FLOW (SCFM)	—	—		—	—	—	—		—	—	—	
TEST OIL FLOW (GPM)	—	—		2.0	—	—	—		1.0	1.0	—	
TOTAL SEAL LEAKAGE (SCFM)	10.6	—		12.1	8.1	—	—		8.5	8.5	—	
TEST BEARING OUTER RING (°F)	420	440		330	410	420	460		360	400	500	
TEST BEARING INNER RING (°F)	450	470		370	430	440	460		420	460	560	
ROLLER BEARING OUTER RING (°F)	425	455		—	—	390	420		295	330	340	
OIL SEAL HOUSING (°F)	—	—		—	—	—	—		—	—	—	
AIR SEAL HOUSING (°F)	570	595		—	—	705	730		595	650	650	
TEST BEARING HOUSING (°F)	495	510		—	—	395	415		255	255	250	
ROLLER BEARING HOUSING (°F)	380	390		—	—	360	365		100	240	220	
AIR SEAL BELLOWS (°F)	—	—		—	—	—	—		435	490	490	
HOT AIR IN MANIFOLD (°F)	630	645		—	—	790	790		670	705	705	
OIL INLET (°F)	—	260		315	355	—	—		200	240	—	
OIL OUTLET (°F)	455	425		—	—	430	440		295	315	355	

Test Data

Test No. 6, page 1 of 1

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AL73T024

RESEARCH LABORATORY SKF INDUSTRIES, INC.

TEST BEARING # 368106

OIL USED FN-3158 + 5% KENDALL RESIN

DATE 12/4-12/72

RUNNING TIME, HOURS		0.3	0.4	0.7	1.7	3.1	3.7	3.8	4.2	4.45	4.6	4.7
SPEED, RPM $\times 10^3$		3.3		8.1	11.1	11.1	11.1	14	15	18	19	
AIR MANIFOLD PRESS. (PSI)		106		106	106	106	106	106	106	106	106	
BEARING CAVITY PRESS. (PSI)		6.0		6.0	6	6	6	6	6	6	6	
SEAL CAVITY PRESS. (PSI)		111		111	111	111	111	111	111	111	111	
HOT AIR FLOW (SCFM)		38		49	44	45	45	—	—	—	—	
TEST OIL FLOW (GPM)		1.5		2.0	2.0	2.0	2.0	2.0	2.0	2.0	2.0	
TOTAL SEAL LEAKAGE (SCFM)		6.8		6.8	13.2	14.4	14					SPEED DROPPED
TEST BEARING OUTER RING (°F)	SPEED TO 3300 RPM IMMEDIATELY	210		360	460	460	450	480	510	480	480	
TEST BEARING INNER RING (°F)		—		380	450	460	450	—	—	—	—	
ROLLER BEARING OUTER RING (°F)		260	STOP	390	490	500	500	515	550	540	595	
OIL SEAL HOUSING (°F)		—		—	—	—	—	—	—	—	—	
AIR SEAL HOUSING (°F)		315		530	800	815	815	825	840	830	835	
TEST BEARING HOUSING (°F)		245		365	430	415	410	410	440	380	360	
ROLLER BEARING HOUSING (°F)		235		340	390	385	380	380	390	350	330	
AIR SEAL BELLOWS (°F)		240		365	590	590	590	600	630	610	600	
HOT AIR IN MANIFOLD (°F)		350		545	760	755	760	765	780	820	825	STOP
OIL INLET (°F)		180		320	385	400	400	—	310	275	270	
OIL OUTLET (°F)		210		375	425	450	450	440	490	360	330	

Test Data

Test No. 7, page 1 of 2

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AL73T024

RESEARCH LABORATORY SKF INDUSTRIES, INC.

TEST BEARING # 368106

OIL USED FN-3158 + 5% KENDALL RESIN

DATE 12/12-19/72

RUNNING TIME, HOURS	4.7	4.8	4.9	5.3	5.55	5.65	5.8	5.95	6.05	6.3		
SPEED, RPM $\times 10^3$	RESTART HEAVY VIBRATION ROLLER BRG FAILURE	STOP AND STARTED THREE TIME SPEED UP TO 3500 RPM	4.8	10.1	STOP END OF DAY	8.1	12.1	18.1	20	STOP SEAL FAILURE		
AIR MANIFOLD PRESS. (PSI)			106	106		106	106	106	106			
BEARING CAVITY PRESS. (PSI)			6	6		6	6	6	6			
SEAL CAVITY PRESS. (PSI)			111	111		111	111	111	111			
HOT AIR FLOW (SCFM)			28			41	41	41	41			
TEST OIL FLOW (GPM)			2.0			2.0	2.0	2.0	2.0			
TOTAL SEAL LEAKAGE (SCFM)			1.3									
TEST BEARING OUTER RING (°F)			250	320		250	300	400	490			
TEST BEARING INNER RING (°F)			—	—		—	—	—	480			
ROLLER BEARING OUTER RING (°F)			265	—		—	—	—	—			
OIL SEAL HOUSING (°F)			—	—		—	—	—	—			
AIR SEAL HOUSING (°F)			330	465		475	510	660	720			
TEST BEARING HOUSING (°F)			330	315		285	300	340	380			
ROLLER BEARING HOUSING (°F)			310	305		280	285	305	330			
AIR SEAL BELLOWS (°F)			285	405		400	450	510	570			
HOT AIR IN MANIFOLD (°F)			510	650		890	930	925	945			
OIL INLET (°F)			230	280		220	230	310	280			
OIL OUTLET (°F)			250	320		275	305	375	390			

Test Data

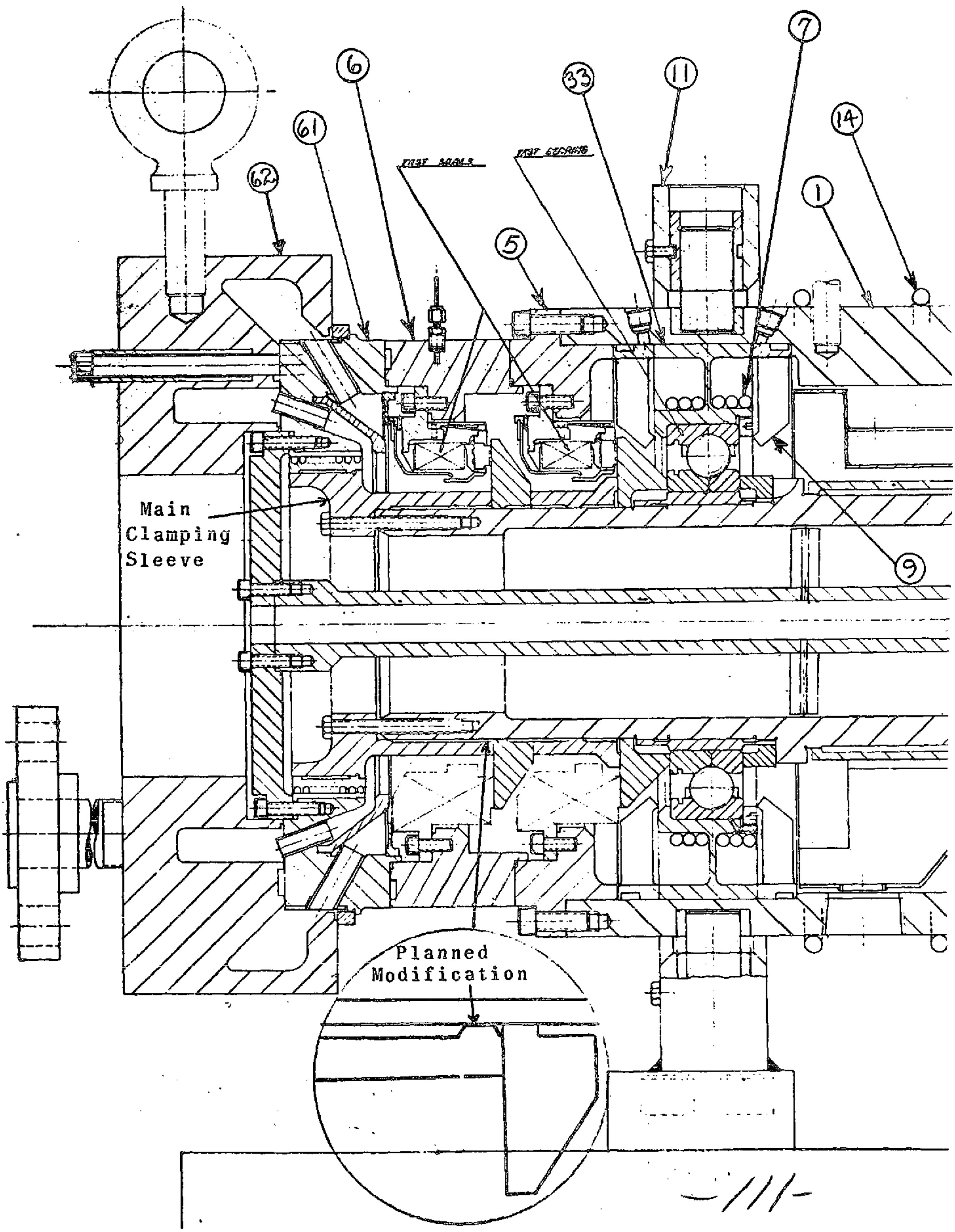
Test 7, page 2 of 2

Enclosure 26

AL73T024

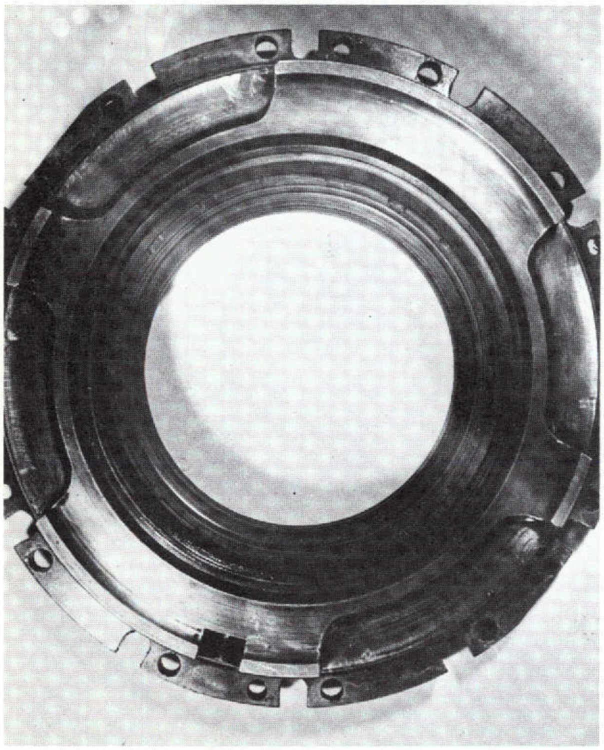
ENCLOSURE 27

MODIFICATION MADE TO MAIN SHAFT CLAMPING SLEEVE

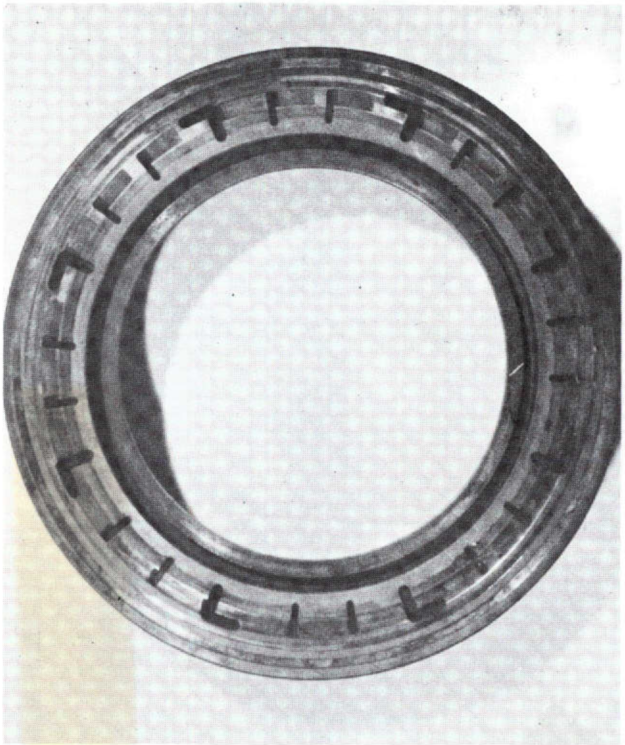


ENCLOSURE 32

TEST SEALS AND MATING RINGS AFTER SCREENING TEST USING HUMBLE FN-3158 OIL BLENDED WITH 5% KENDALL 0839 RESIN AT SPEEDS TO 23,600 RPM



KOPPERS OIL SEAL



OIL SEAL MATING RING

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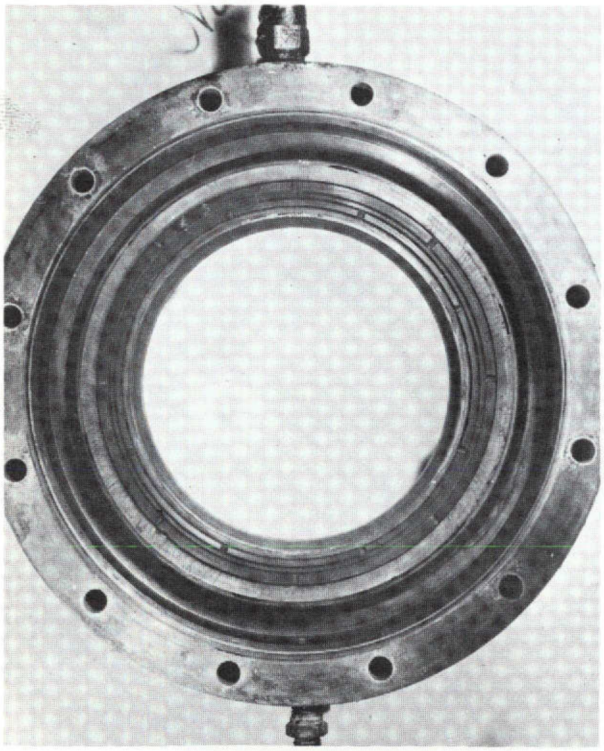
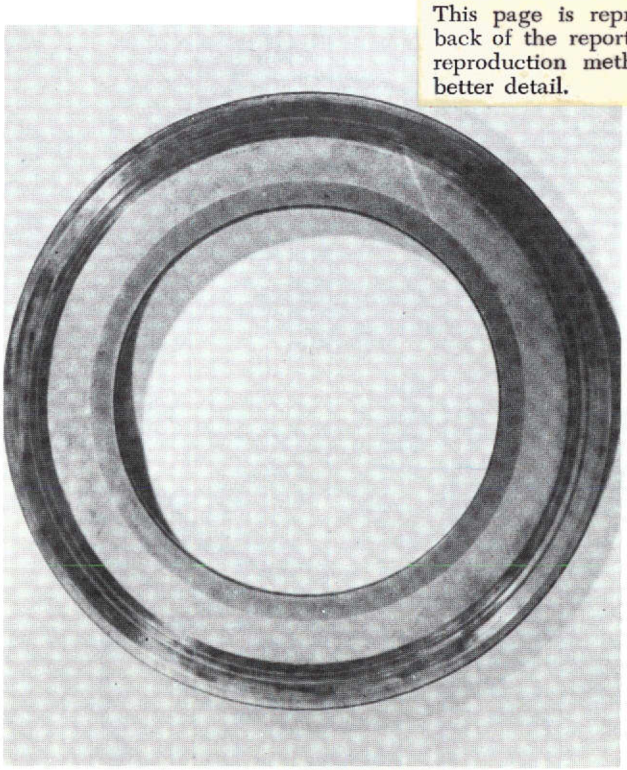


KOPPERS AIR SEAL



AIR SEAL MATING RING

ENCLOSURE 33

TEST BEARING COMPONENTS AFTER SCREENING TEST USING HUMBLE FN-3158 OIL BLENDED WITH 5% KENDALL 0839 RESIN AT SPEEDS TO 23,600 RPM

INNER RING

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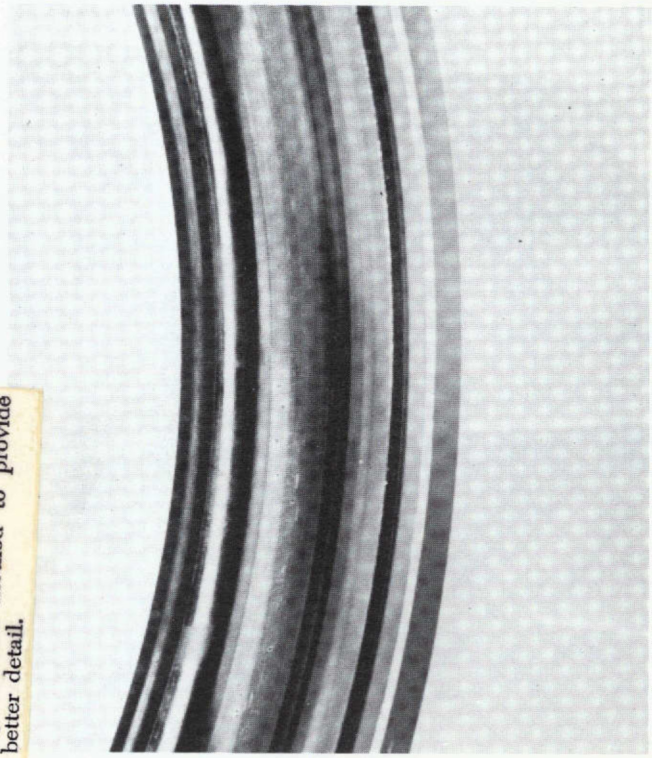
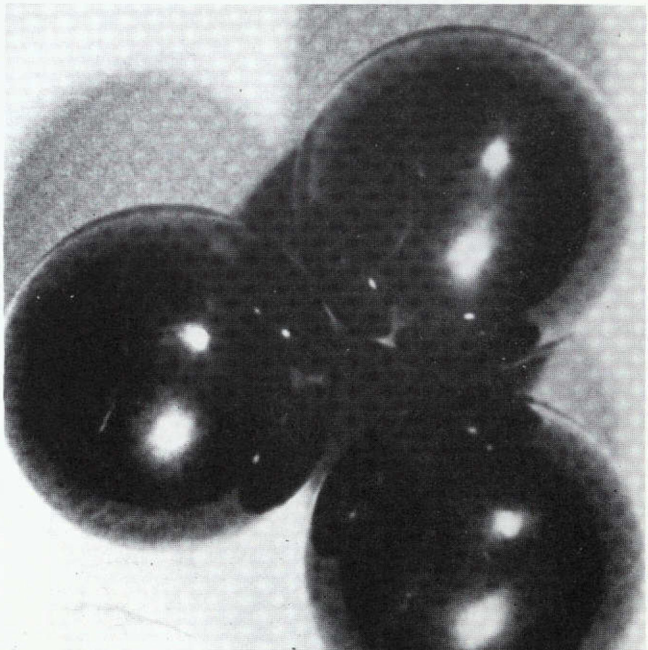


OUTER RING

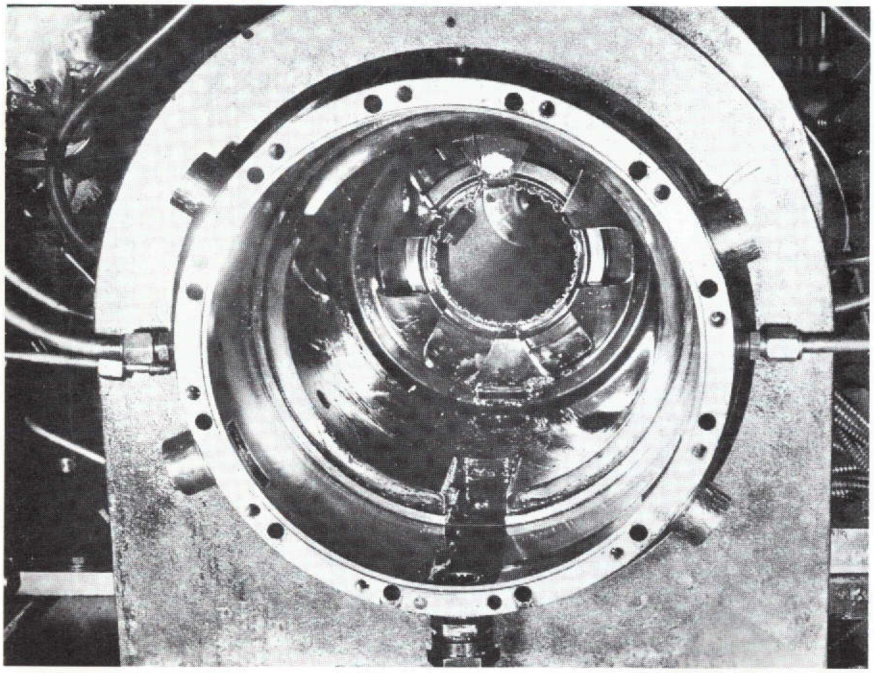
CAGE



BALLS

ENCLOSURE 34

TEST RIG PARTS AFTER SCREENING TEST USING HUMBLE FN-3158 OIL BLENDED WITH 5% KENDALL 0839 RESIN AT SPEEDS TO 23,600 RPM



TEST BEARING HOUSING

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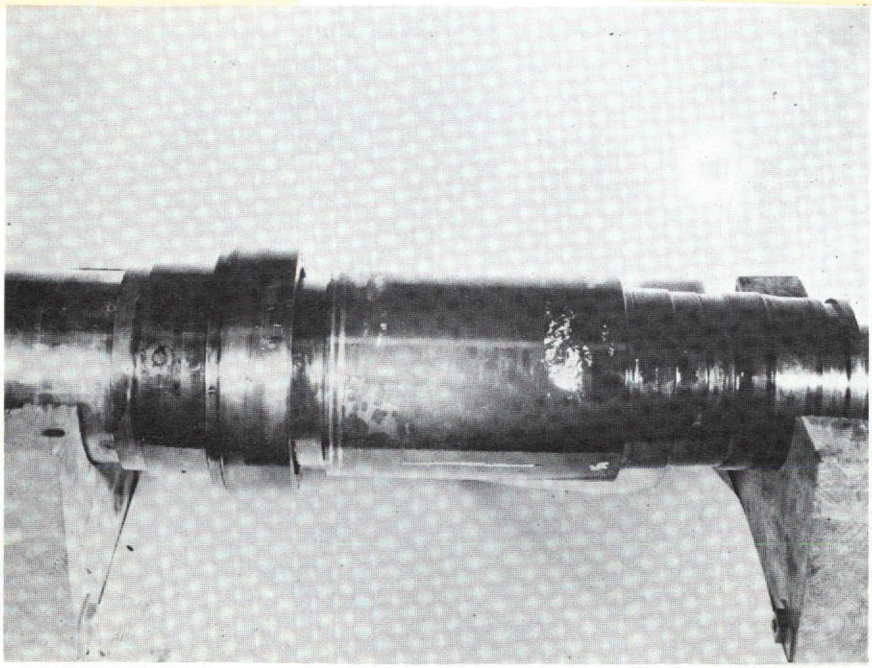

TEST SHAFT

ENCLOSURE 35

TEST BEARING COMPONENTS AFTER SCREENING TEST USING MONSANTO MCS-2931 OIL AT SPEEDS TO 21,900 RPM

INNER RING

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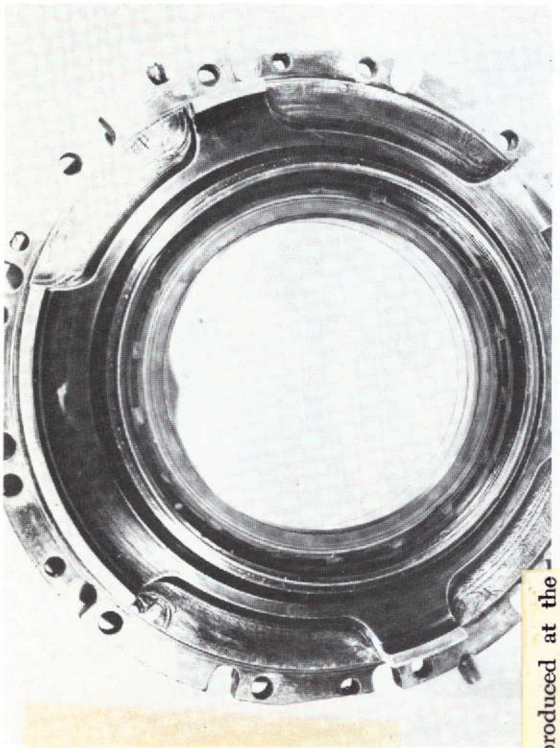
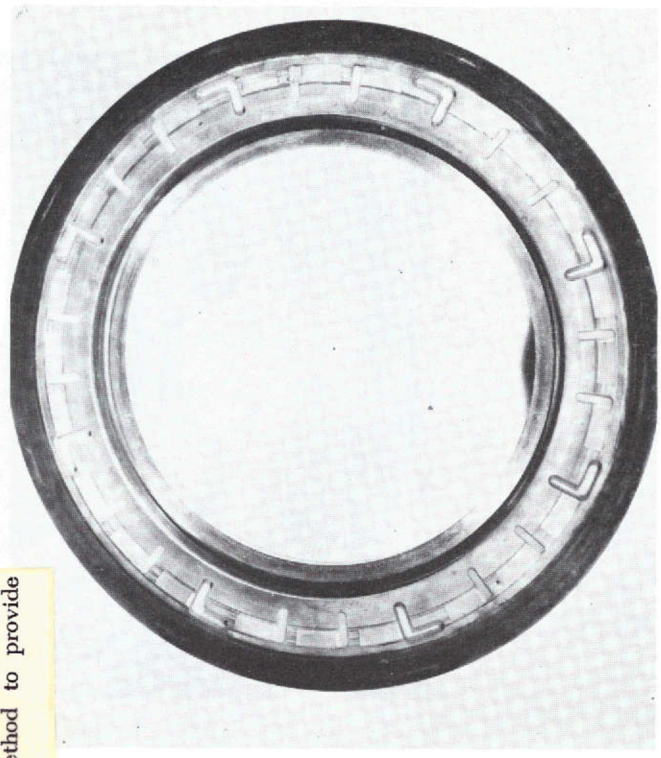
OUTER RING, CAGE, AND BALLS

ENCLOSURE 36

TEST SEALS AND MATING RINGS AFTER SCREENING TEST USING MONSANTO MCS-2931 OIL AT SPEEDS TO 21,900 RPM



KOPPERS OIL SEAL



OIL SEAL MATING RING

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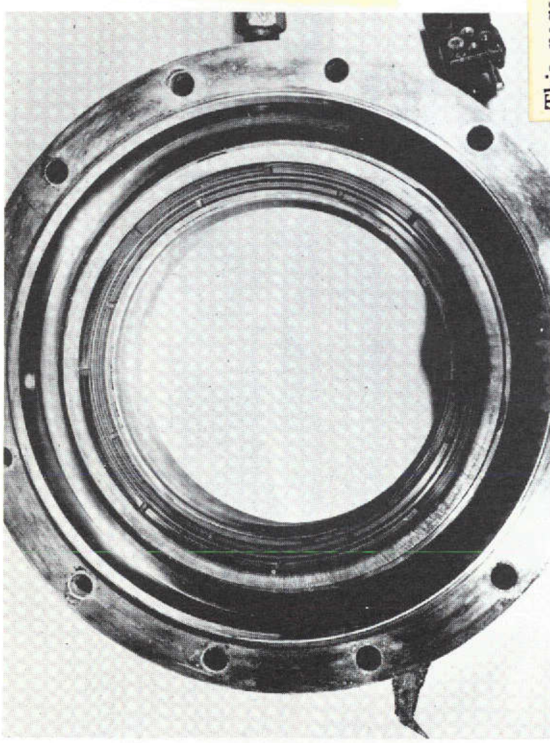
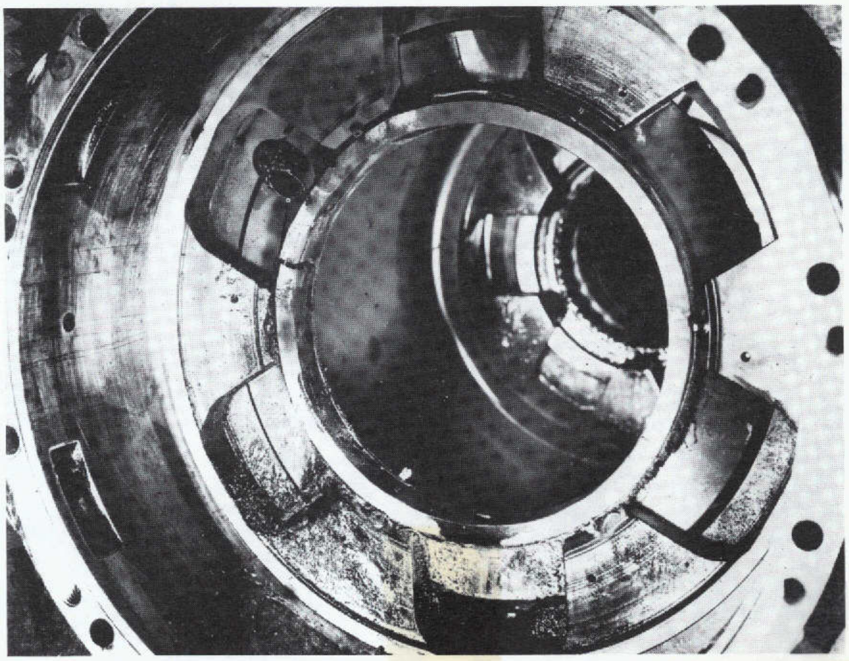


KOPPERS AIR SEAL

AIR SEAL MATING RING

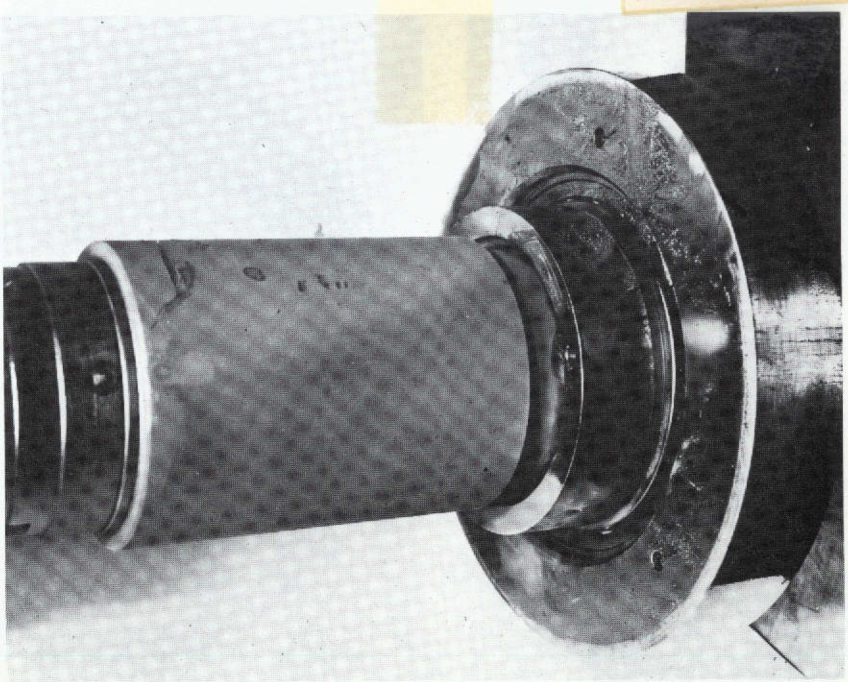
ENCLOSURE 37

TEST RIG PARTS AFTER SCREENING TEST USING MONSANTO MCS-2931 OIL AT SPEEDS TO 21,900 RPM



TEST BEARING HOUSING

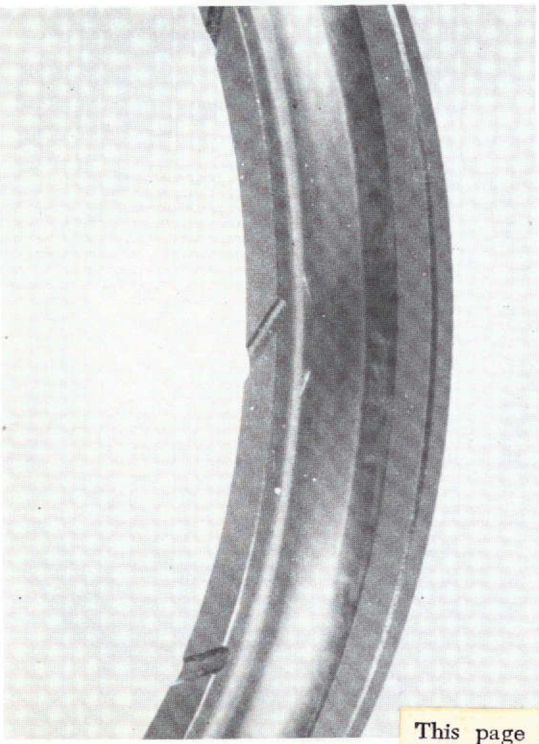
This page is reproduced at the back of the report by a different reproduction method to provide better detail.



TEST SHAFT

ENCLOSURE 38

TEST BEARING COMPONENTS AFTER SCREENING TEST USING CONOCO DN 600 FLUID AT SPEEDS TO 16,000 RPM



INNER RING

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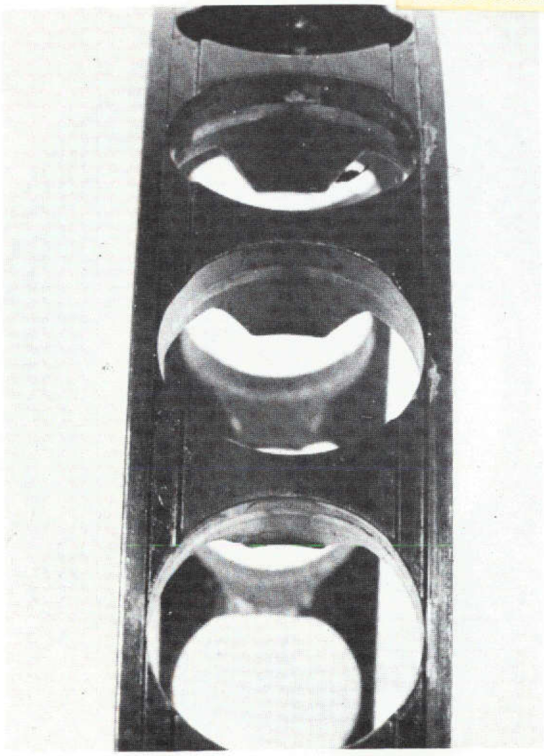
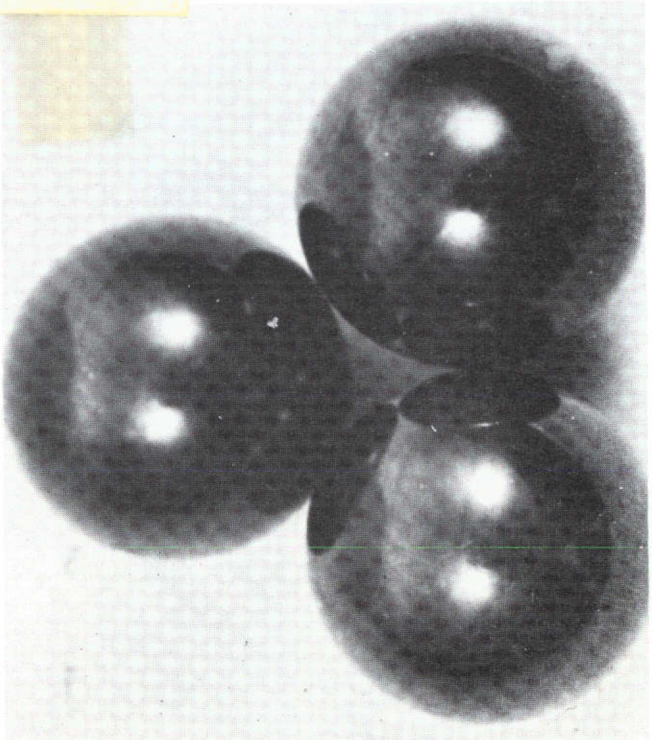
OUTER RING



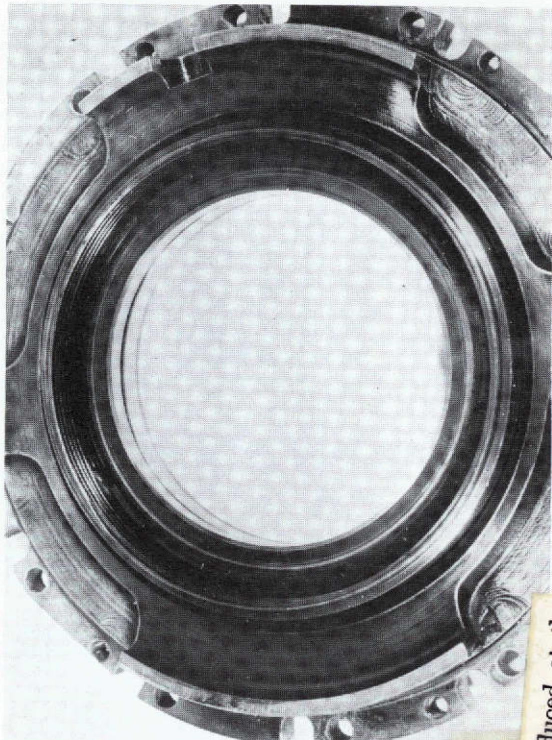
CAGE



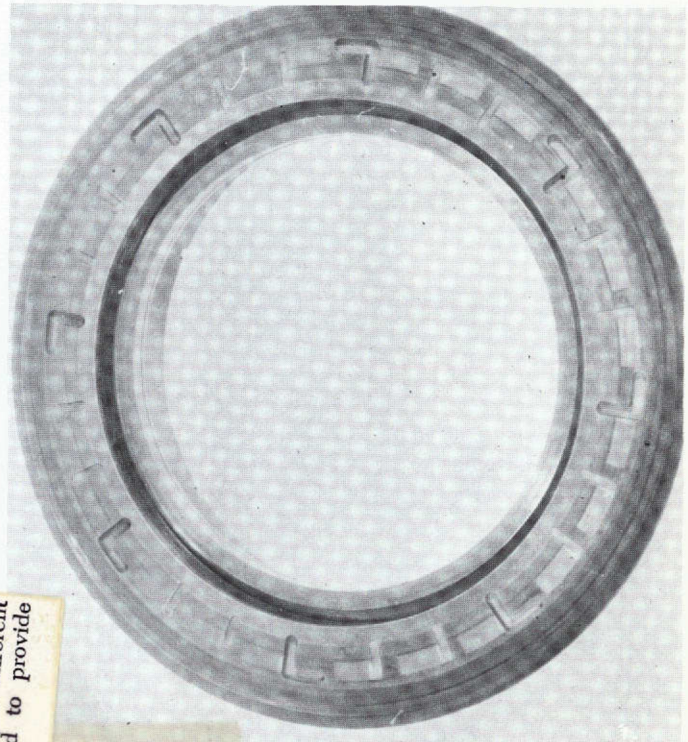
BALLS

ENCLOSURE 39

TEST SEALS AND MATING RINGS AFTER SCREENING TEST USING CONOCO DN 600 FLUID AT SPEEDS TO 16,000 RPM



KOPPERS OIL SEAL



OIL SEAL MATING RING

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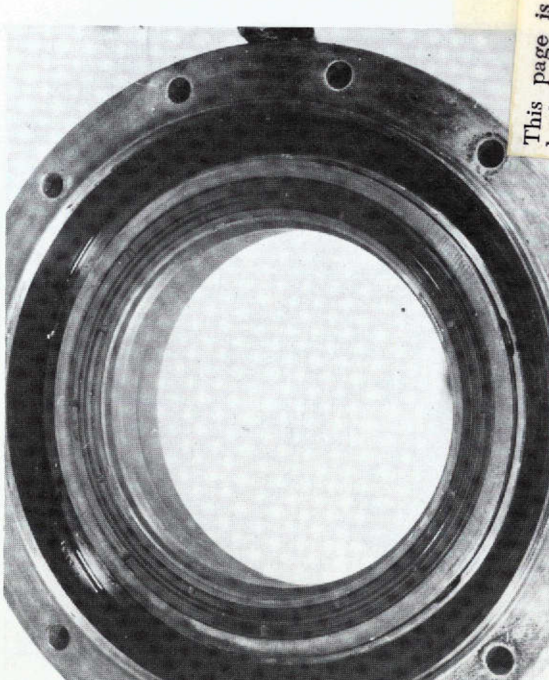
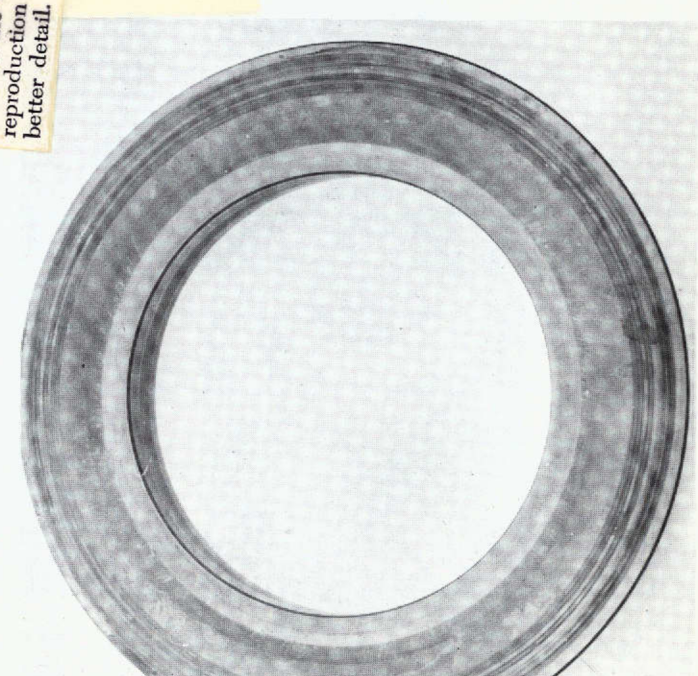


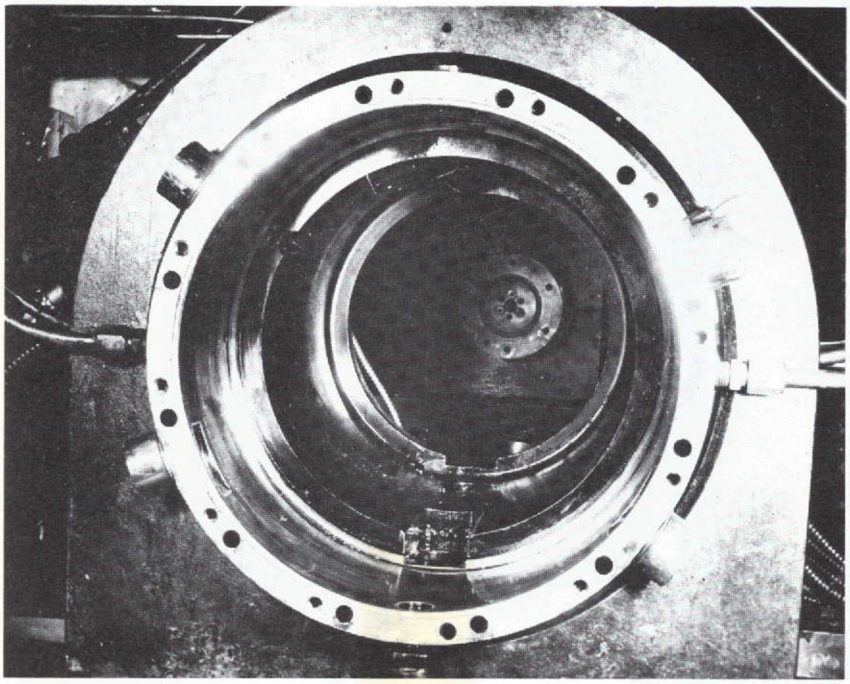
KOPPERS AIR SEAL



AIR SEAL MATING RING

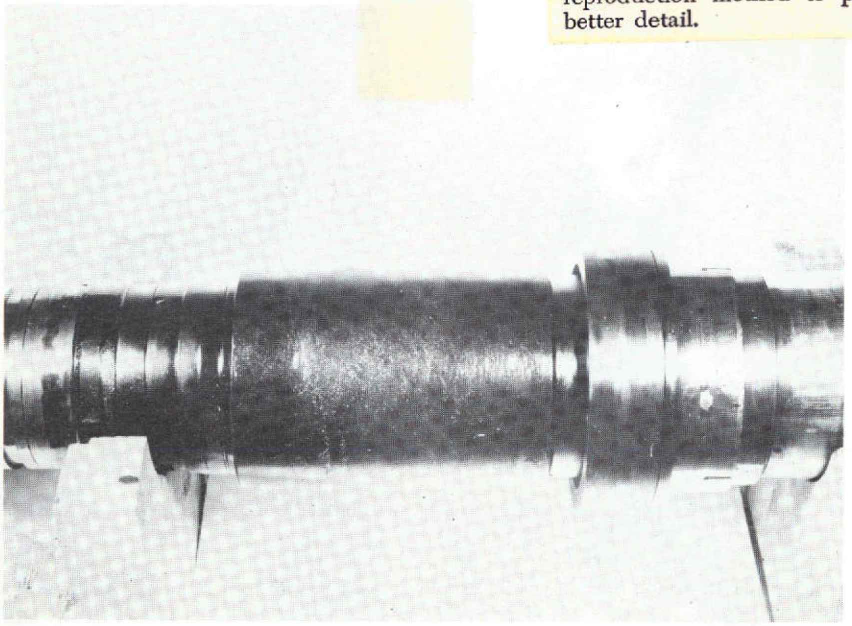
ENCLOSURE 40

TEST RIG PARTS AFTER SCREENING TEST USING CONOCO DN 600 FLUID AT SPEEDS TO 16,000 RPM



TEST BEARING HOUSING

This page is reproduced at the back of the report by a different reproduction method to provide better detail.

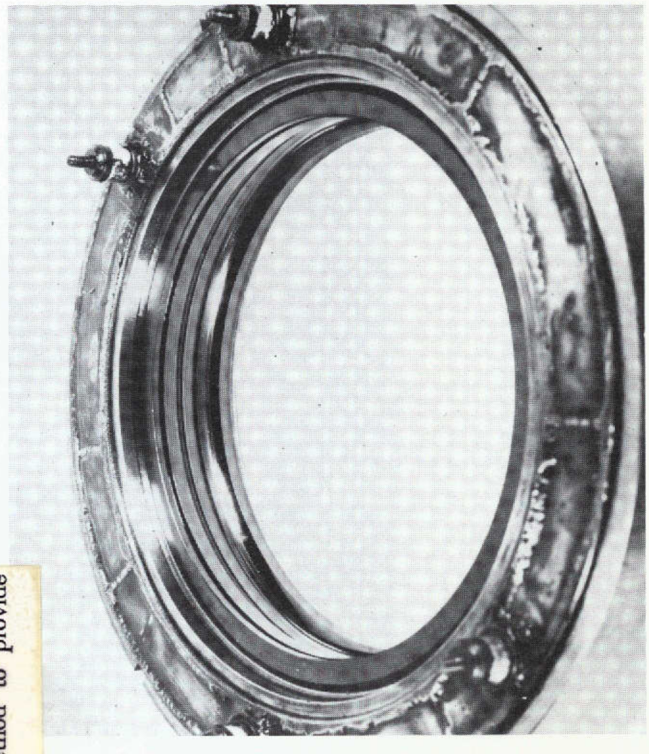


TEST SHAFT

ENCLOSURE 41

TEST BEARING COMPONENTS AFTER SCREENING TEST USING AEROSHELL TURBINE OIL 555 AT SPEEDS TO 21,400 RPM

INNER RING



OUTER RING

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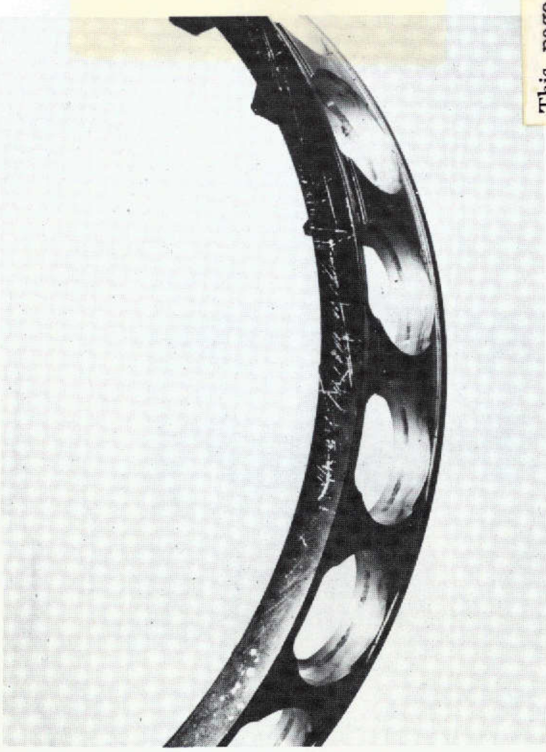
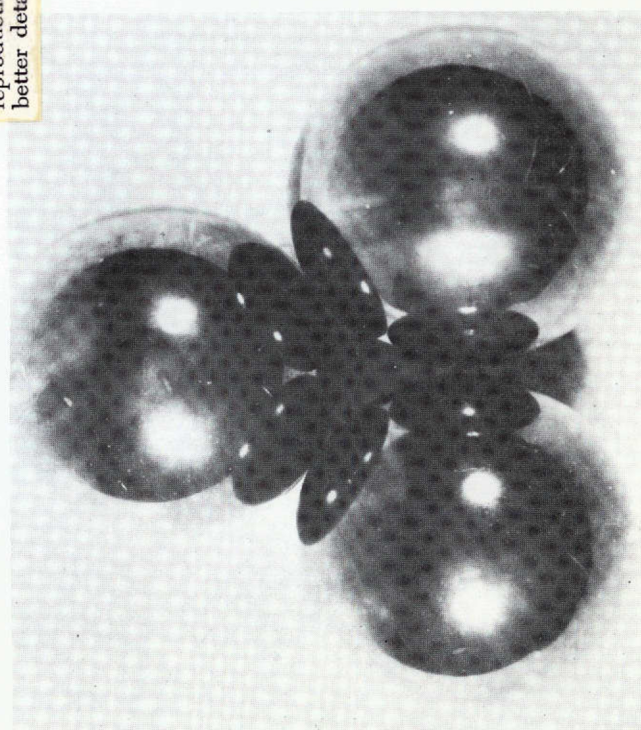


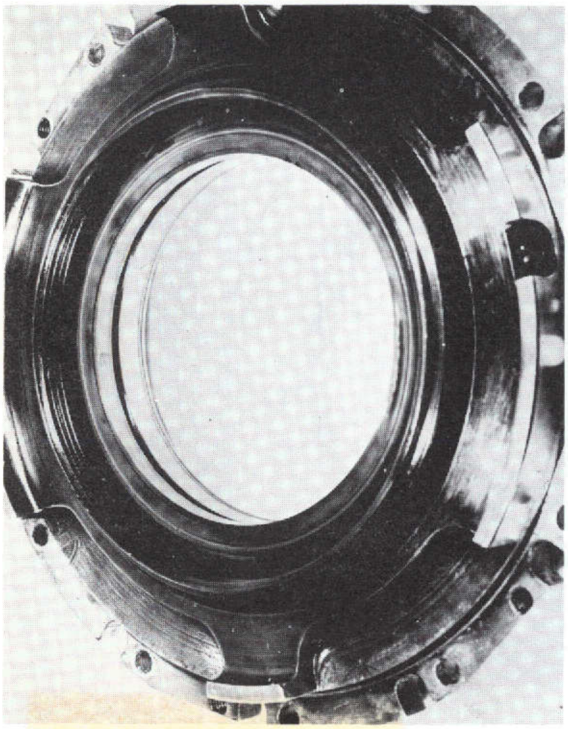
CAGE



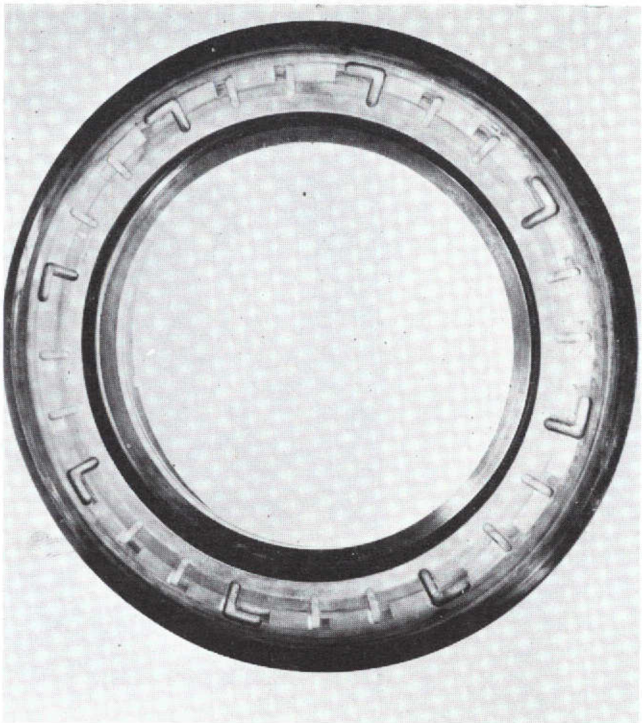
BALLS

ENCLOSURE 42

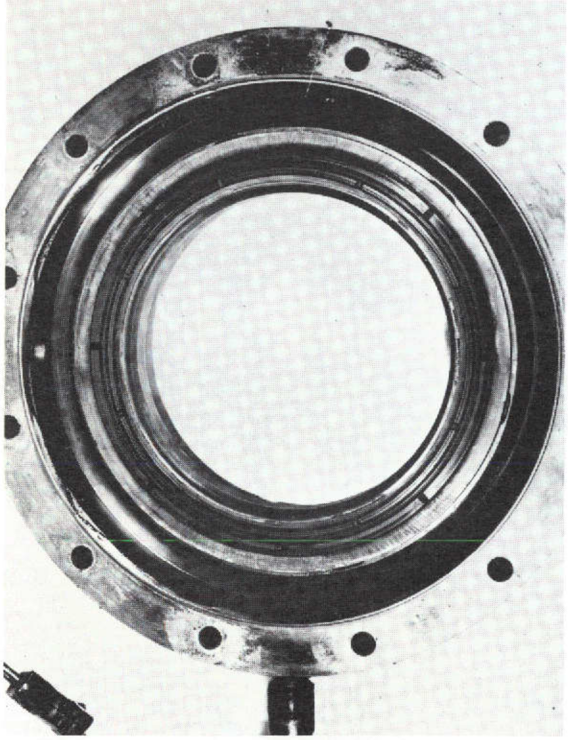
TEST SEALS AND MATING RINGS AFTER SCREENING TEST USING AEROSHELL TURBINE OIL 555 AT SPEEDS TO 21,400 RPM



KOPPERS OIL SEAL



OIL SEAL MATING RING



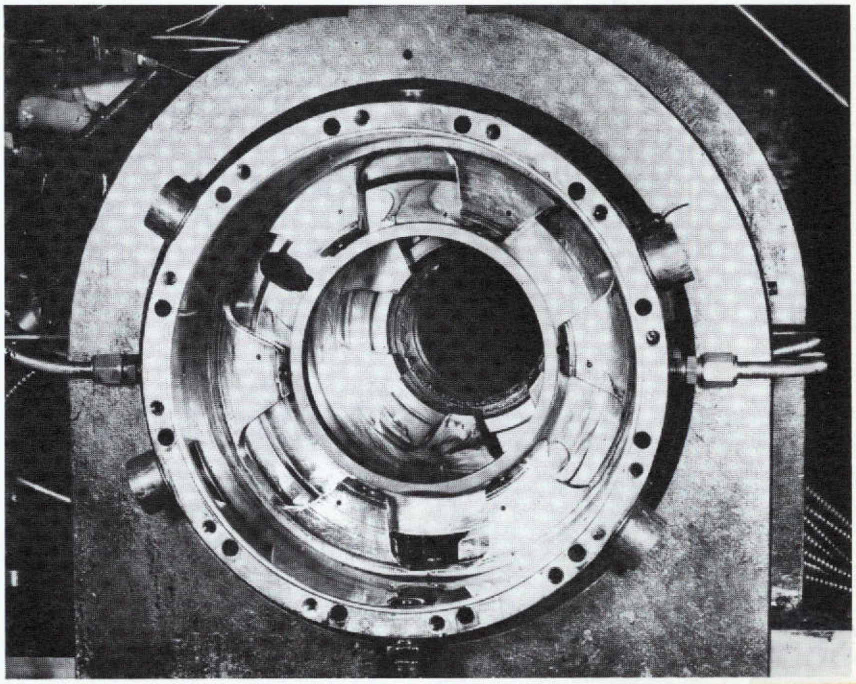
KOPPERS AIR SEAL

This page is reproduced at the back of the report by a different reproduction method to provide better detail.

AIR SEAL MATING RING

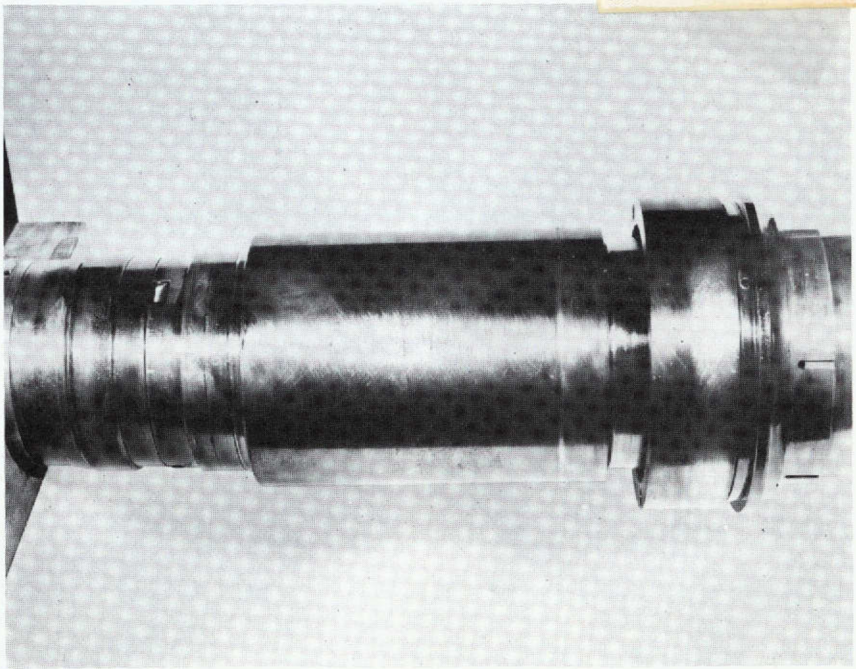
ENCLOSURE 43

TEST RIG PARTS AFTER SCREENING TEST USING AEROSHELL TURBINE OIL 555 AT SPEEDS TO 21,400 RPM



TEST BEARING HOUSING

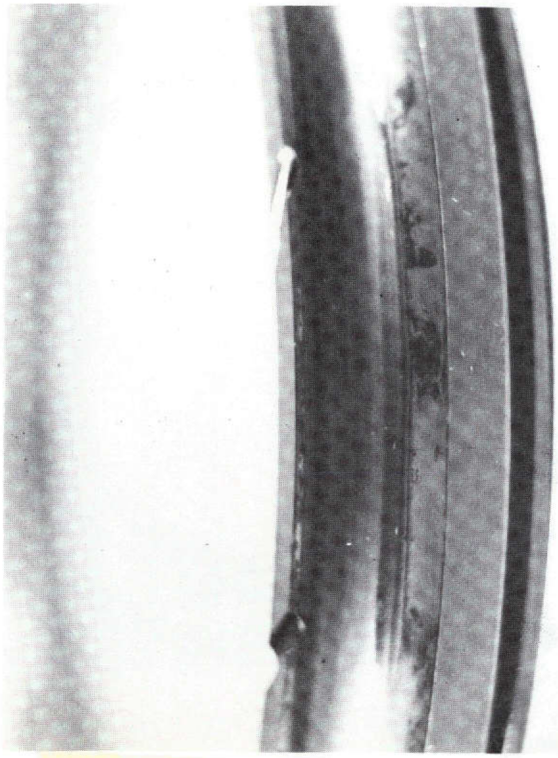
This page is reproduced at the back of the report by a different reproduction method to provide better detail.



TEST SHAFT

ENCLOSURE 44

TEST BEARING COMPONENTS AFTER EXTENDED DURATION TEST USING AEROSHELL TURBINE OIL 555 AT SPEEDS TO 20,000 RPM FOR 26.7 HOURS



INNER RING

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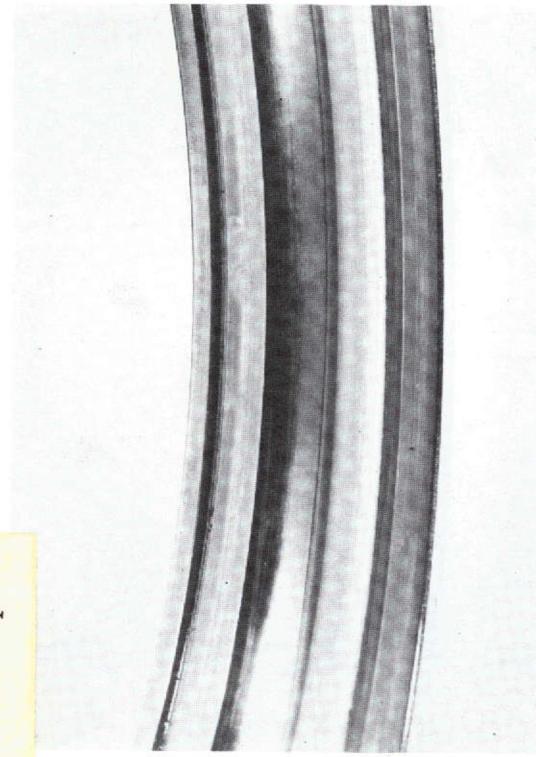
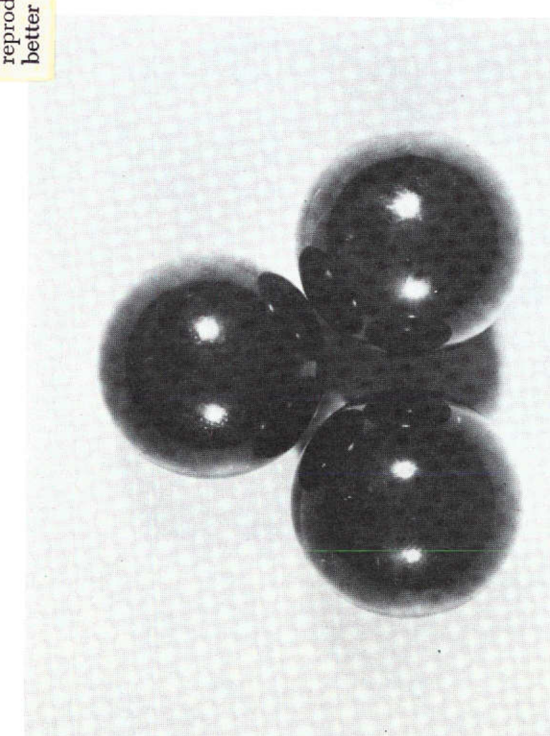


OUTER RING

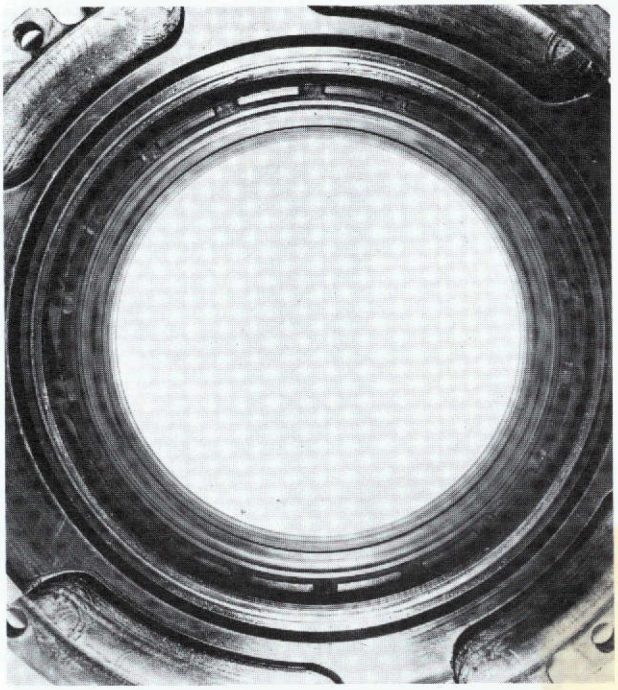
CAGE



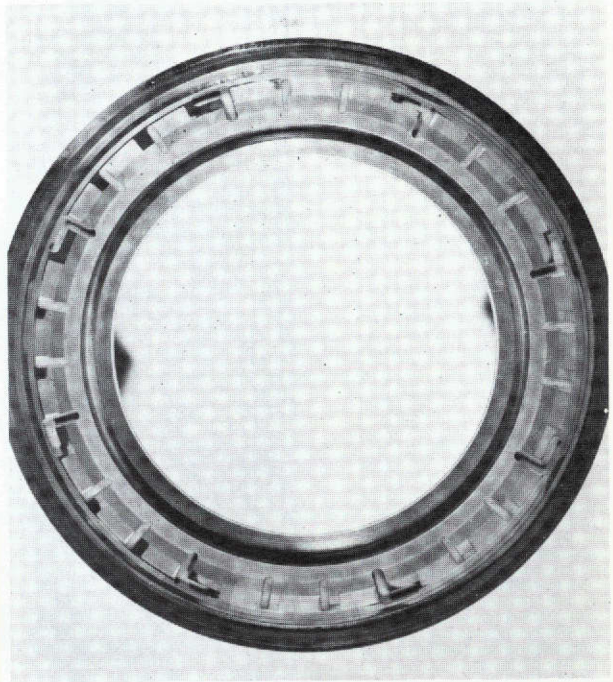
BALLS

ENCLOSURE 45

TEST SEALS AND MATING RINGS AFTER EXTENDED DURATION TEST USING AEROSHELL TURBINE OIL 555 AT SPEEDS TO 20,000 RPM FOR 26.7 HOURS



KOPPERS OIL SEAL



OIL SEAL MATING RING

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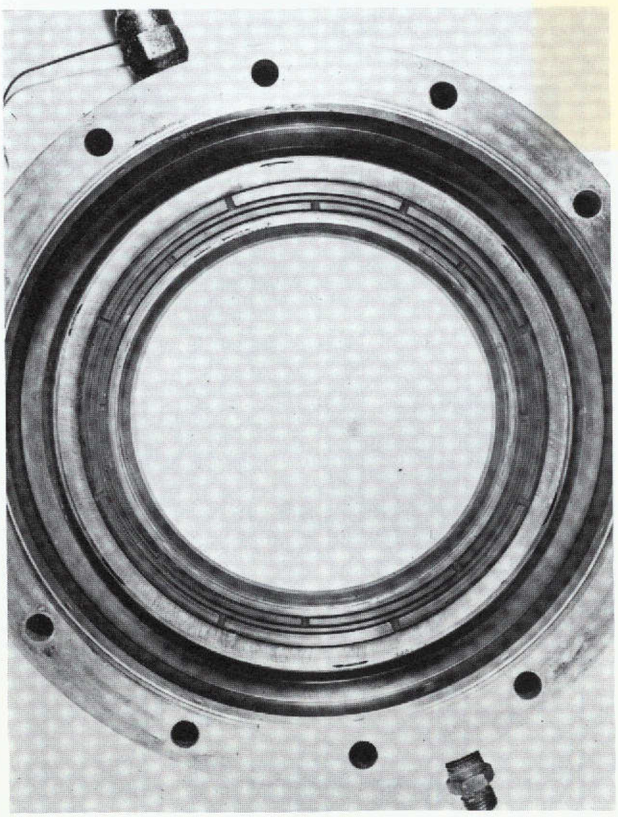
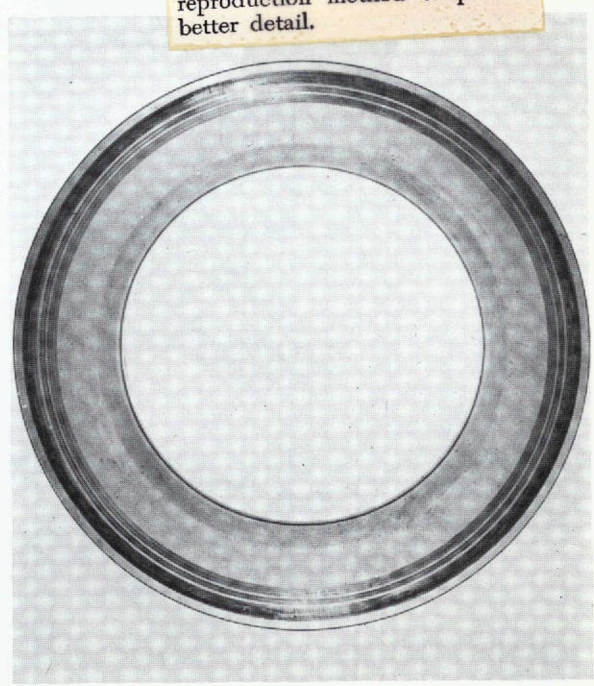


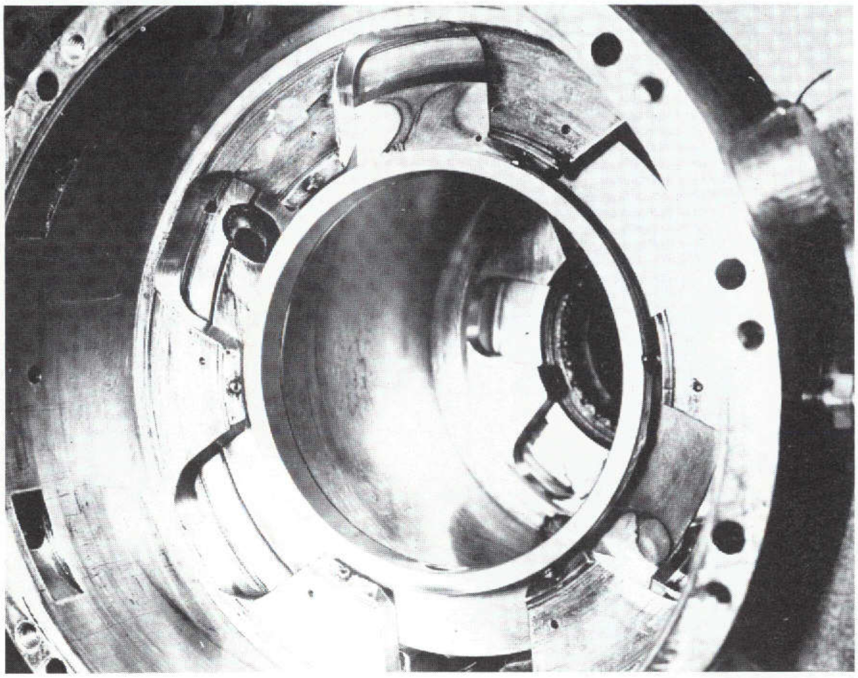
KOPPERS AIR SEAL



AIR SEAL MATING RING

ENCLOSURE 46

TEST RIG PARTS AFTER EXTENDED DURATION TEST USING AEROSHELL TURBINE OIL 555 AT SPEEDS TO 20,000 RPM FOR 26.7 HOURS



TEST BEARING HOUSING

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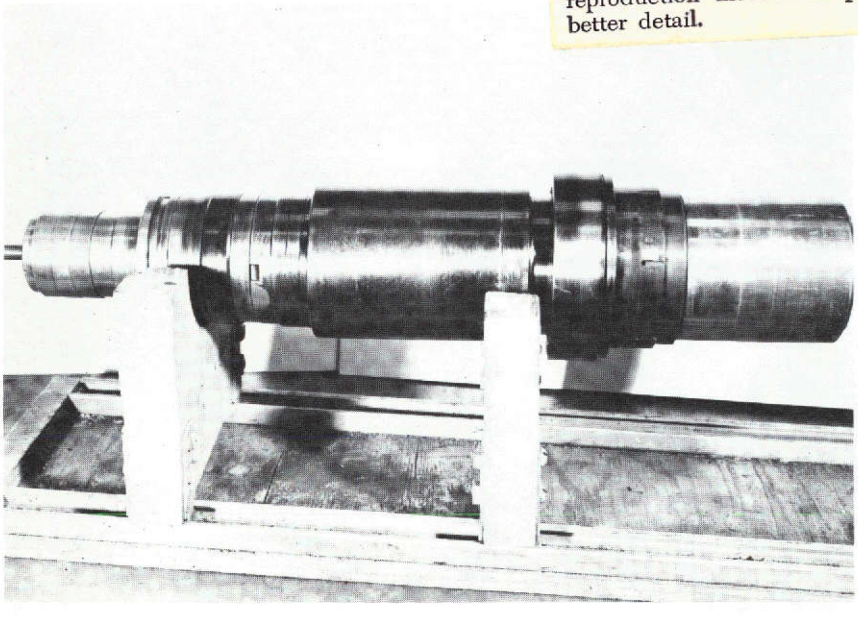
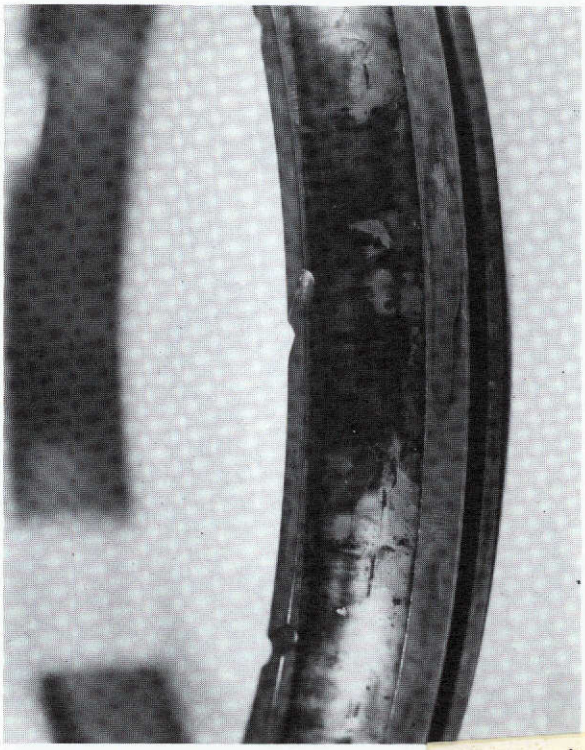


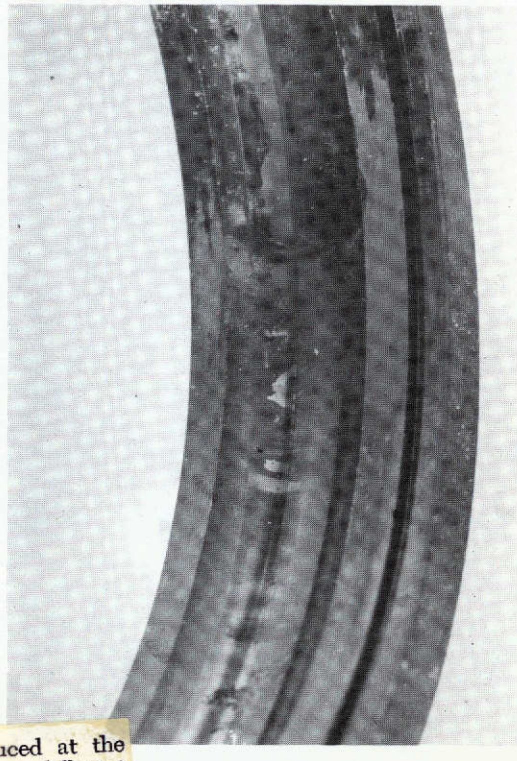
TEST SHAFT

ENCLOSURE 47

TEST BEARING COMPONENTS AFTER ABORTED EXTENDED DURATION TEST USING HUMBLE FN-3158 OIL BLENDED WITH 5% KENDALL 0839 RESIN AT SPEEDS TO 10,800 RPM FOR 2.1 HOURS



INNER RING



OUTER RING

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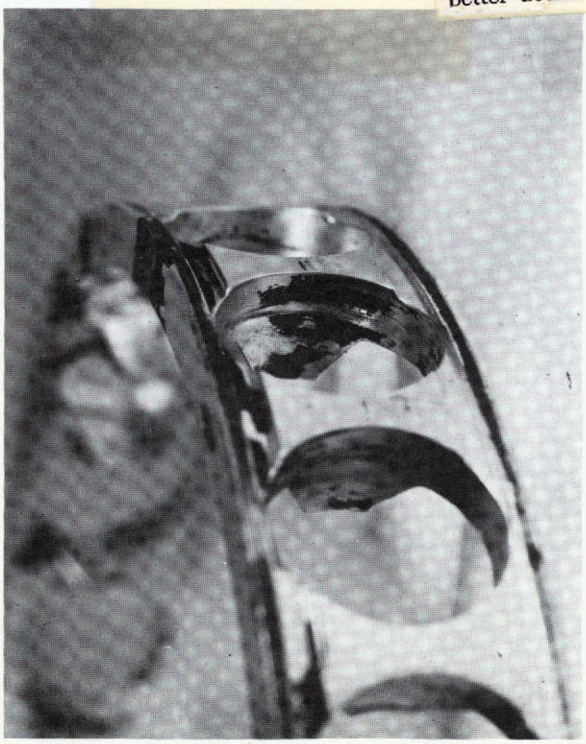
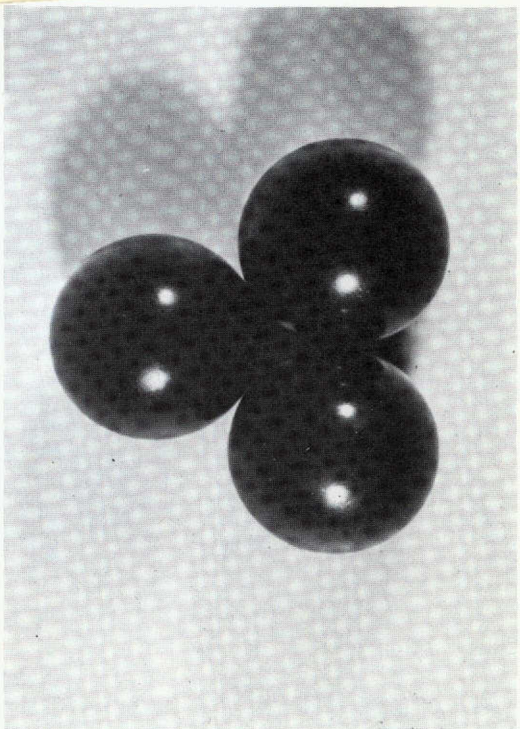


CAGE



BALLS

ENCLOSURE 48

TEST SEAL AND MATING RING PARTS AFTER ABORTED EXTENDED DURATION TEST USING HUMBLE FN-3158 OIL BLENDED WITH 5% KENDALL 0839 RESIN AT SPEEDS TO 10,800 RPM FOR 2.1 HOURS

NASA OIL SEAL MATING RING

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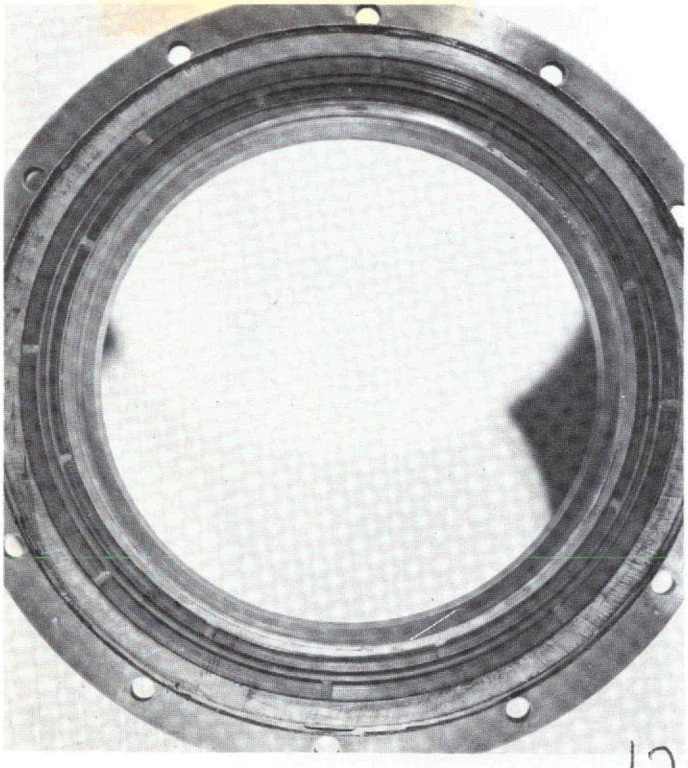
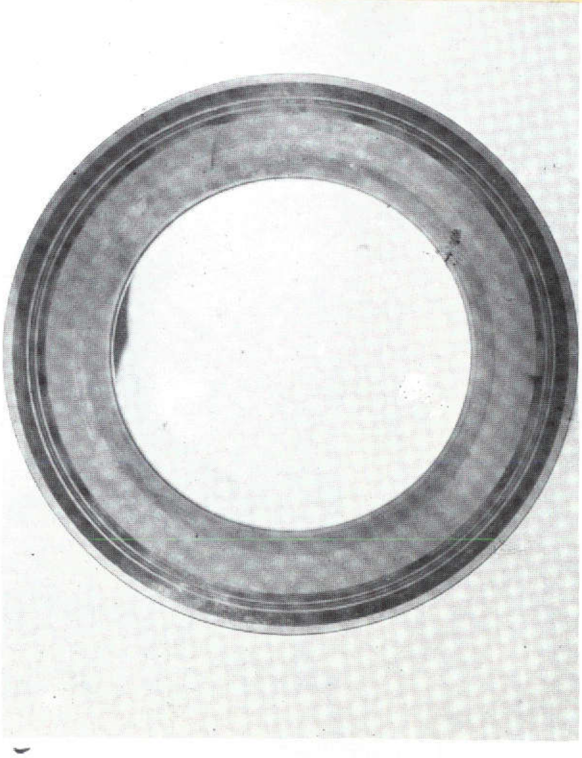


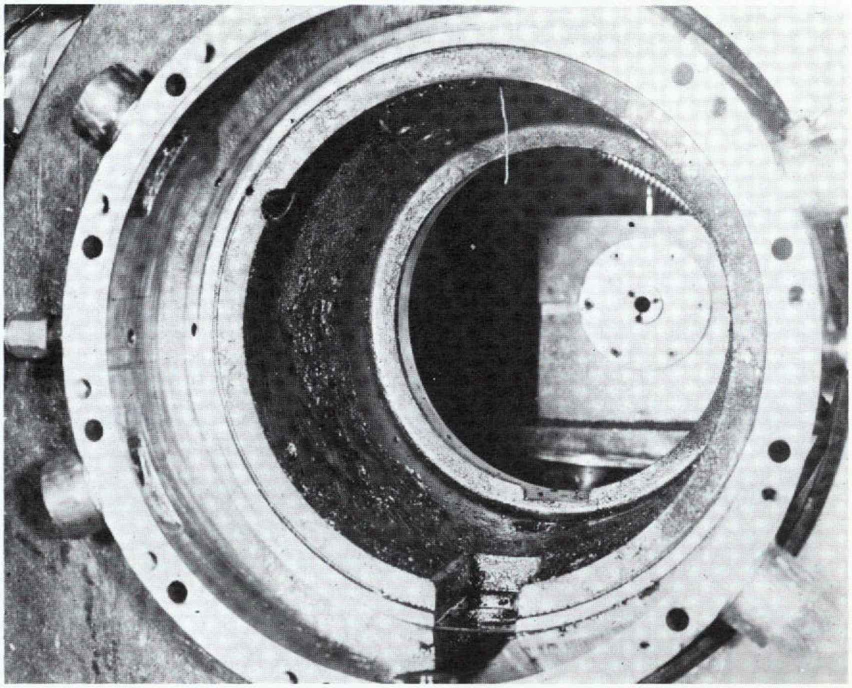
KOPPERS AIR SEAL



AIR SEAL MATING RING

ENCLOSURE 49

TEST RIG PARTS AFTER ABORTED EXTENDED DURATION TEST USING HUMBLE FN-3158 OIL BLENDED WITH 5% KENDALL 0839 RESIN AT SPEEDS TO 10,800 RPM FOR 2.1 HOURS



BEARING HOUSING

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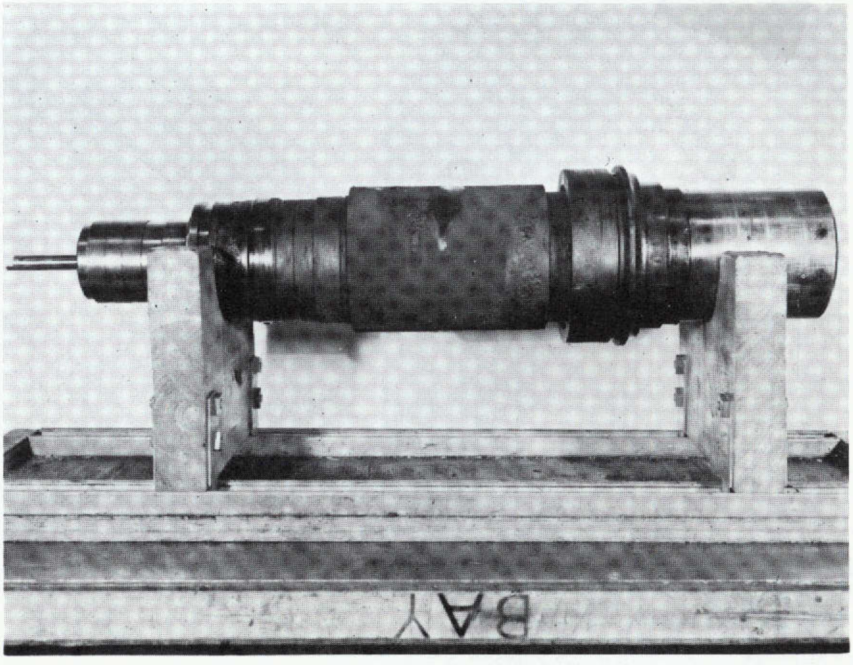
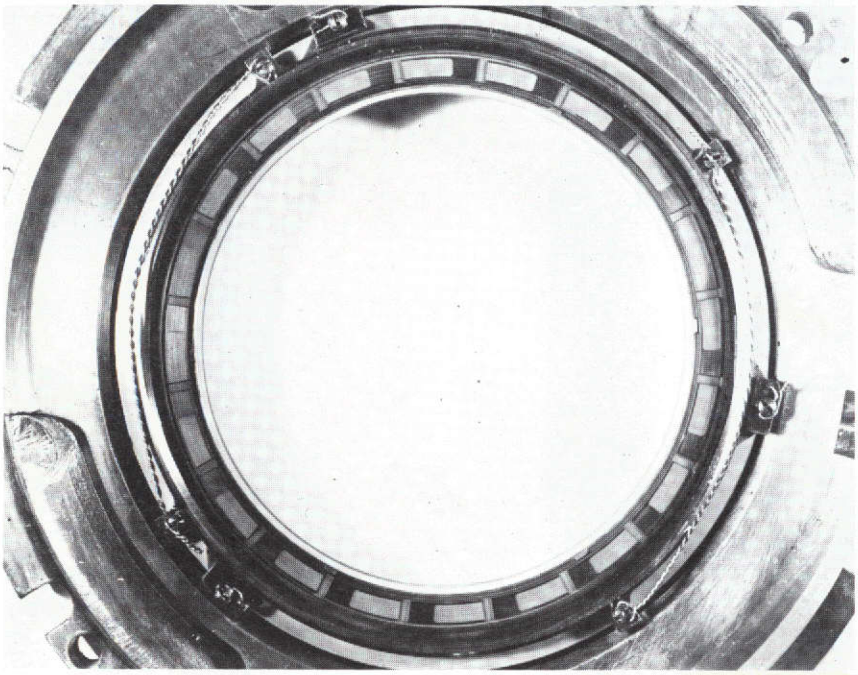


TEST SHAFT

ENCLOSURE 50

NASA OIL SEAL AND MATING RING DURING REPEAT EXTENDED DURATION TEST USING HUMBLE FN-3158 OIL BLENDED WITH 5% KENDALL 0839 RESIN, SHOWS CONDITION AT TIME OF ROLLER BEARING FAILURE AFTER 4.7 HOURS AT SPEEDS TO 19,000 RPM



NASA OIL SEAL

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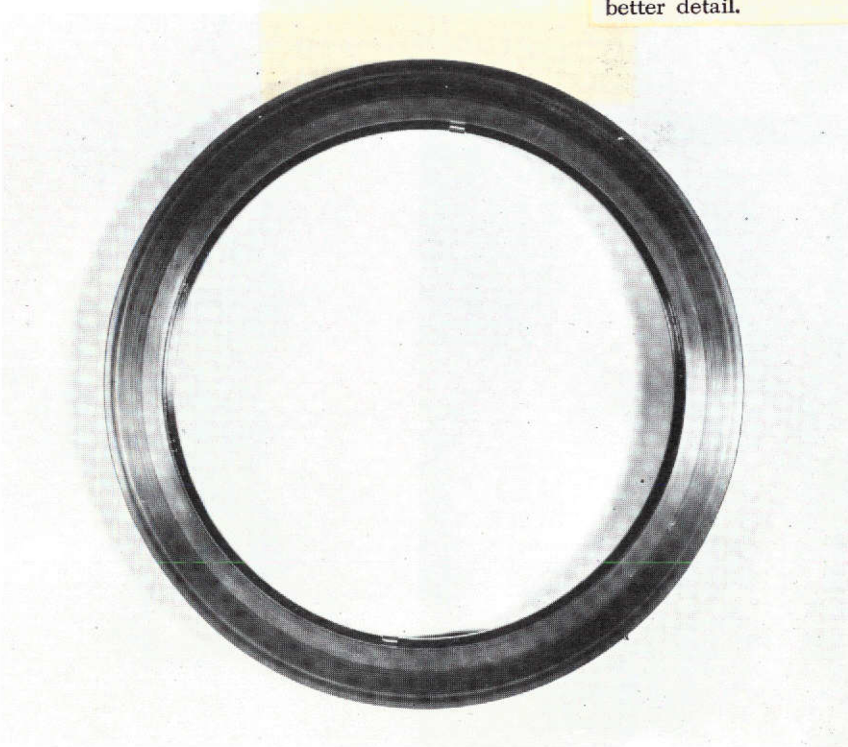
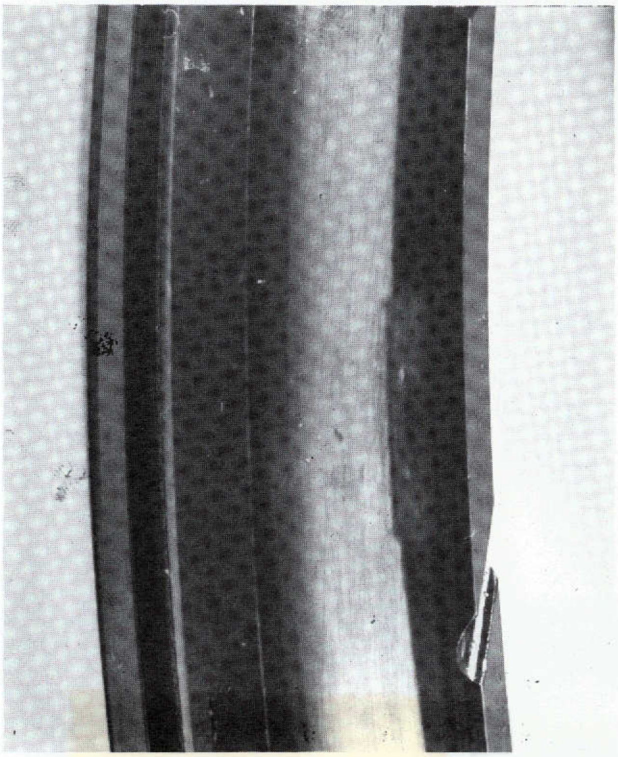


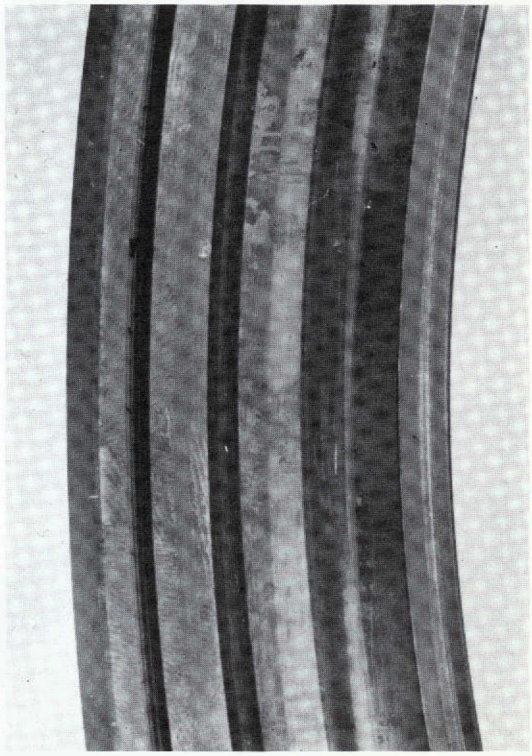
OIL SEAL MATING RING

ENCLOSURE 51

TEST BEARING COMPONENTS AT COMPLETION OF REPEAT EXTENDED DURATION TEST USING HUMBLE FN-3158 OIL BLENDED WITH 5% KENDALL 0839 RESIN AFTER 6.3 HOURS AT SPEEDS TO 20,100 RPM



INNER RING



OUTER RING

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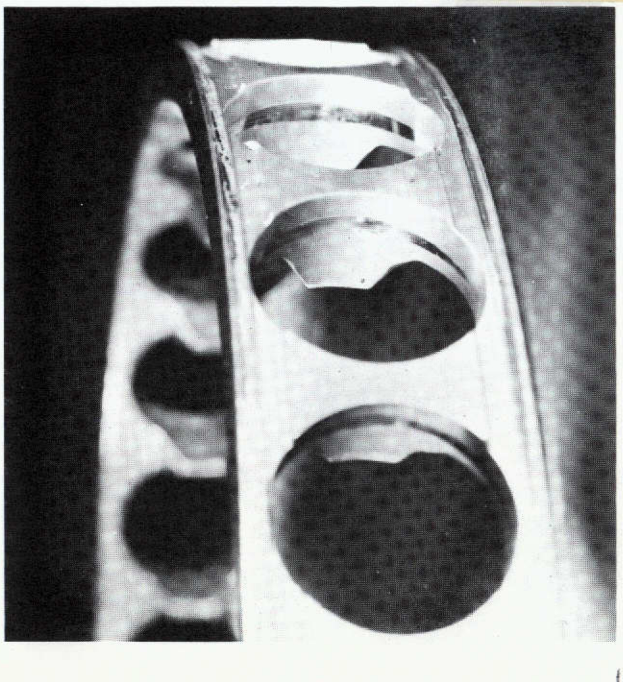
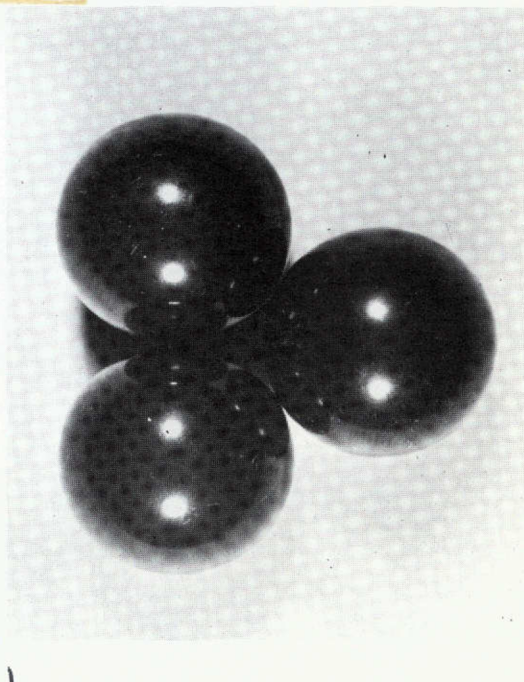


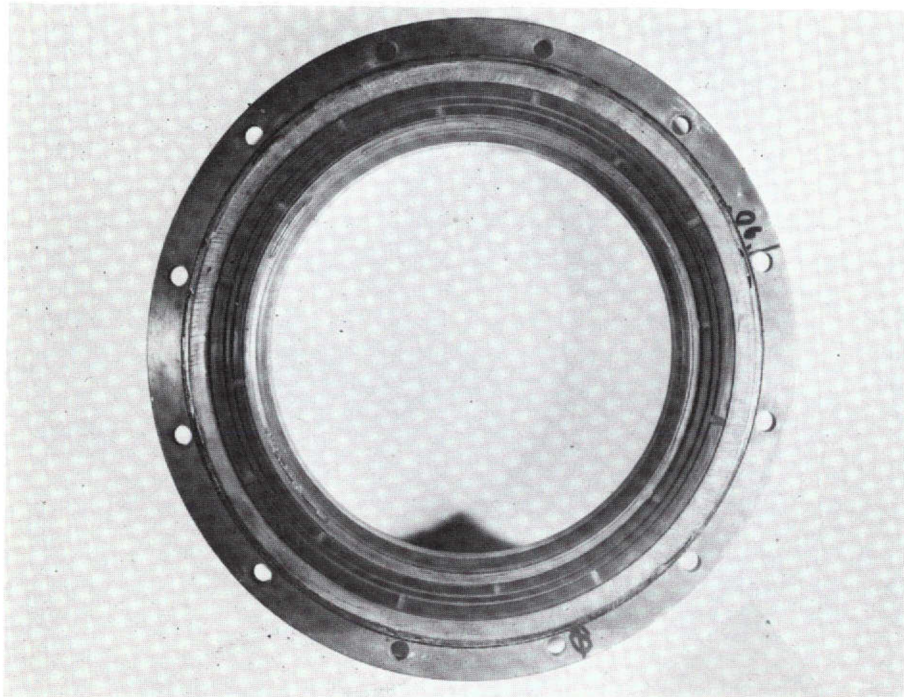
CAGE



BALLS

ENCLOSURE 52

AIR SEAL AND MATING RING AT COMPLETION OF REPEAT EXTENDED DURATION TEST USING HUMBLE FN-3158 OIL BLENDED WITH 5% KENDALL 0839 RESIN AFTER 6.3 HOURS AT SPEEDS TO 20,100 RPM



KOPPERS AIR SEAL

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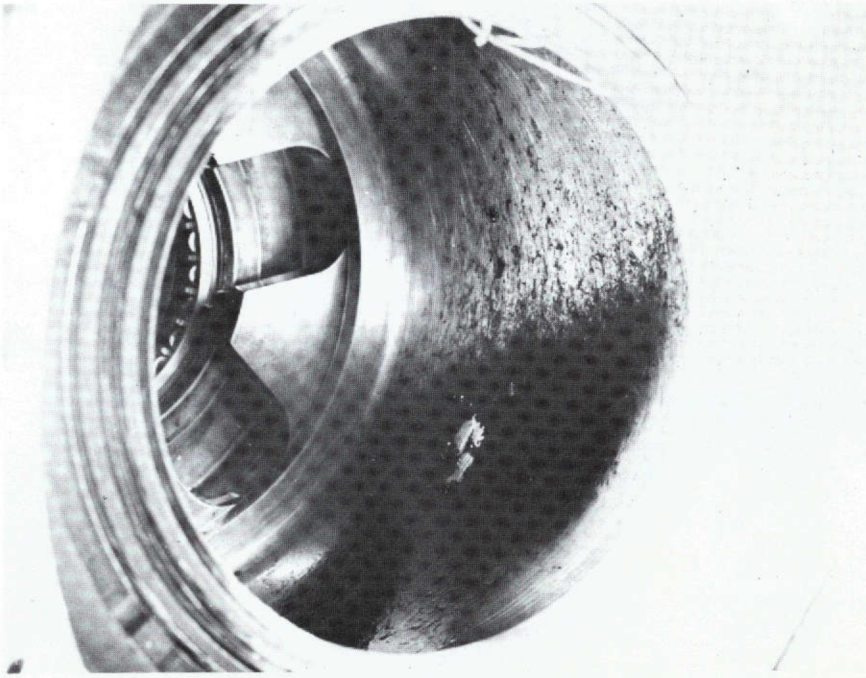
AIR SEAL MATING RING

ENCLOSURE 53

TEST RIG PARTS AT COMPLETION OF REPEAT EXTENDED DURATION TEST USING HUMBLE FN-3158 OIL BLENDED WITH 5% KENDALL 0839 RESIN AFTER 6.3 HOURS AT SPEEDS TO 20,100 RPM



BEARING HOUSING

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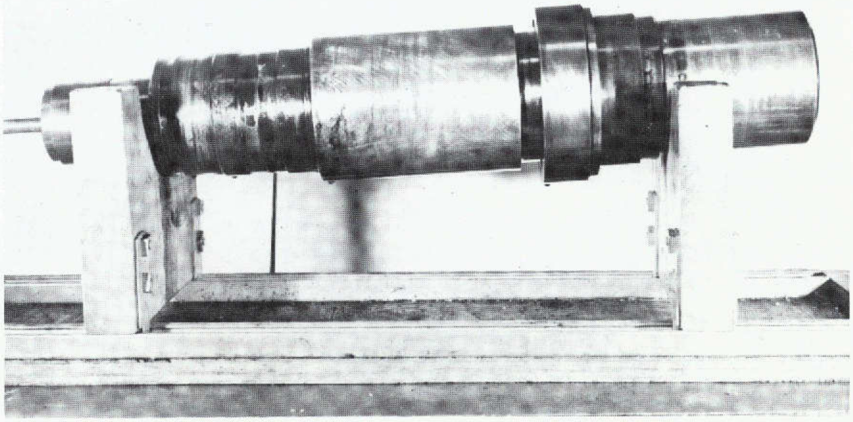
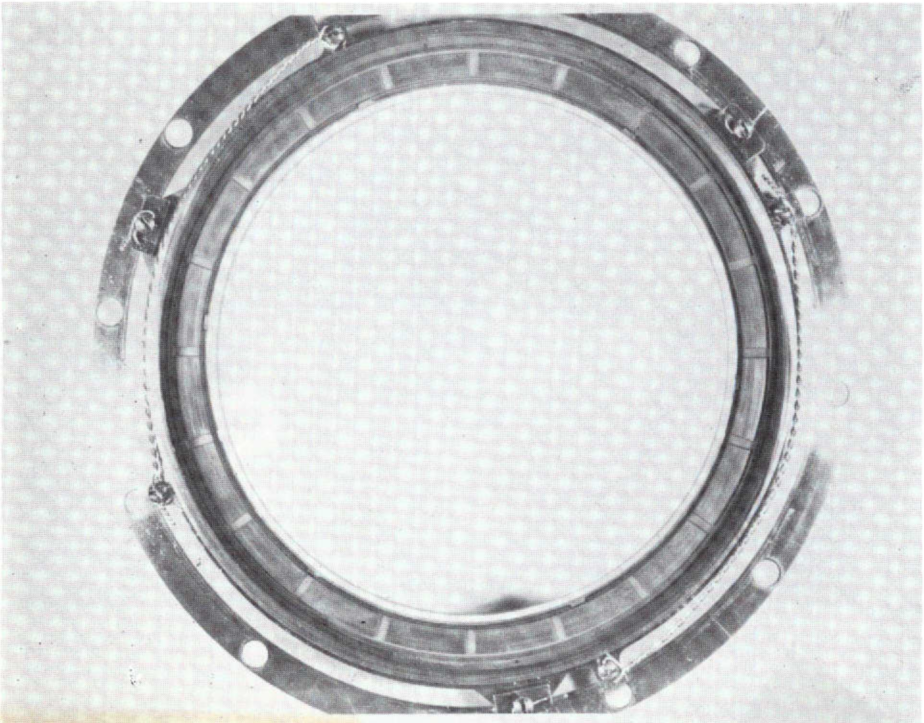


TEST SHAFT

ENCLOSURE 54

NASA OIL SEAL AND MATING RING AT COMPLETION OF REPEAT EXTENDED DURATION TEST USING HUMBLE FN-3158 OIL BLENDED WITH 5% KENDALL 0839 AFTER 6.3 HOURS AT SPEEDS TO 20,100 RPM



NASA OIL SEAL

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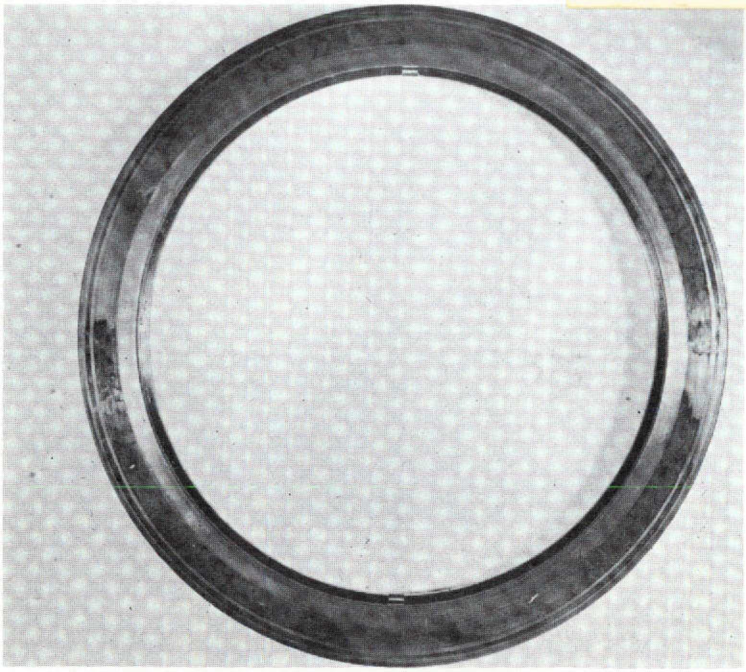
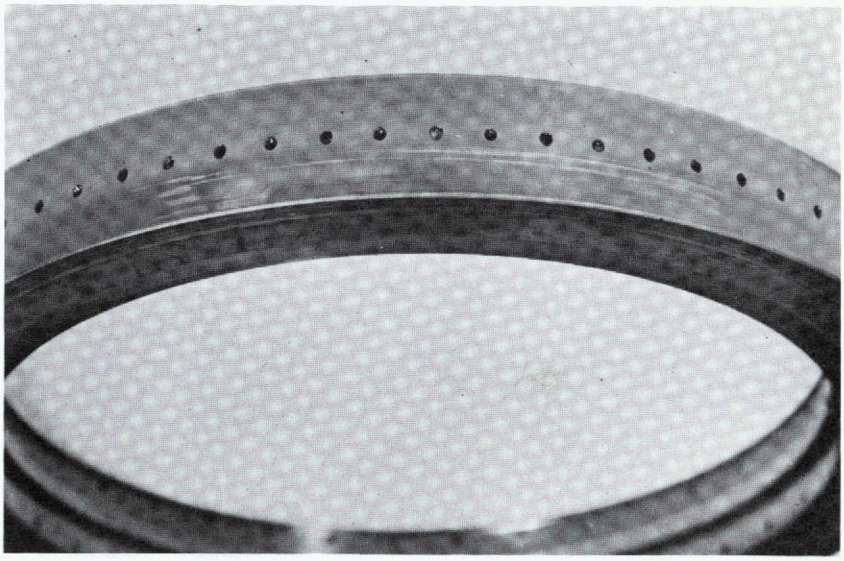


OIL SEAL MATING RING

ENCLOSURE 55

NASA OIL SEAL MATING RING AT COMPLETION OF REPEAT EXTENDED DURATION TEST USING HUMBLE FN-3158 OIL BLENDED WITH 5% KENDALL 0839 RESIN AFTER 6.3 HOURS AT SPEEDS TO 20,100 RPM



SCRATCHES ON O.D. DUE TO WINDBACK CONTACT

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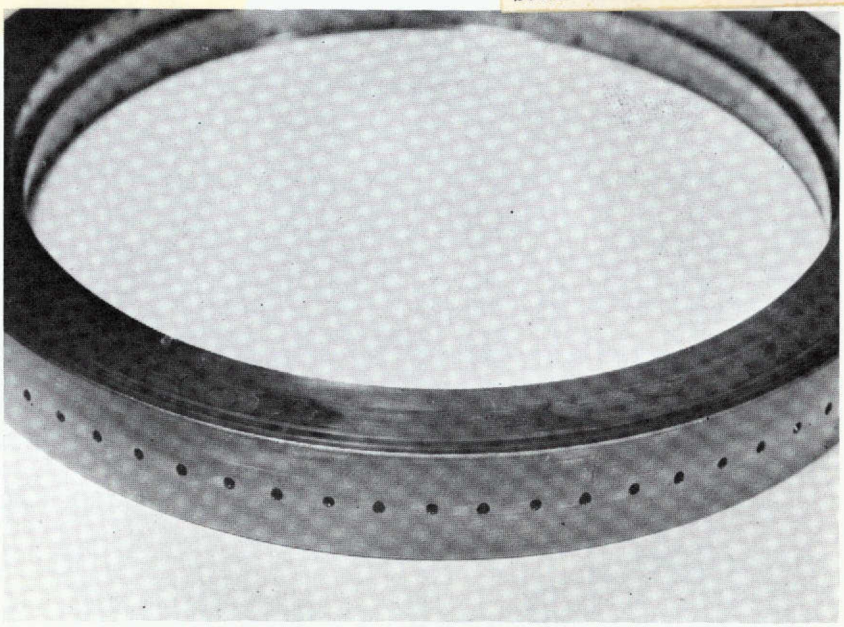
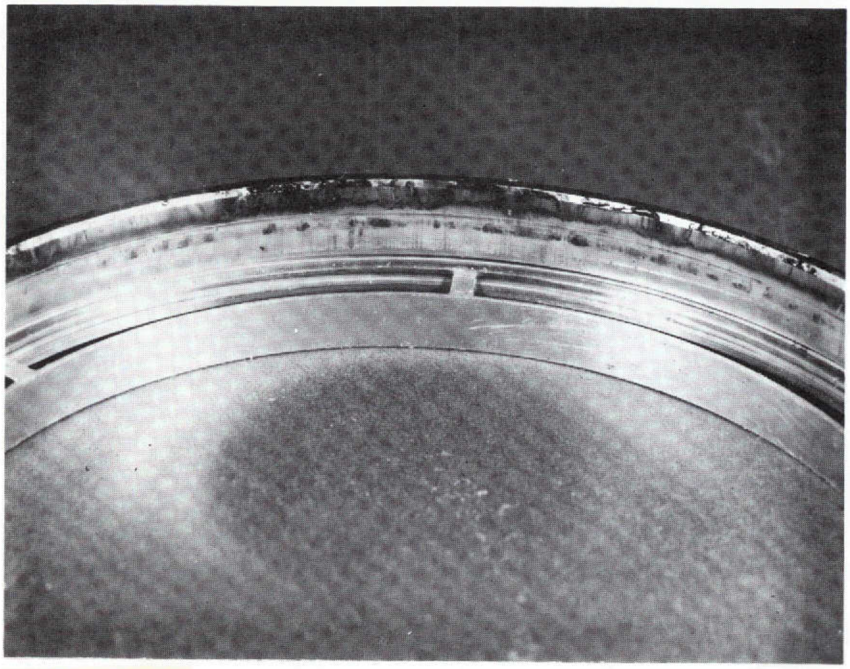


INTERMITTENT SMEARING ON FACE

ENCLOSURE 56

CF 848839 CARRIER OF NASA OIL SEAL AT COMPLETION OF REPEAT EXTENDED DURATION TEST USING HUMBLE FN-3158 OIL BLENDED WITH 5% KENDALL 0839 RESIN AFTER 6.3 HOURS AT SPEEDS TO 20,100 RPM



BORE OF CARRIER SHOWING COKING
IN PISTON RING SEALING AREA

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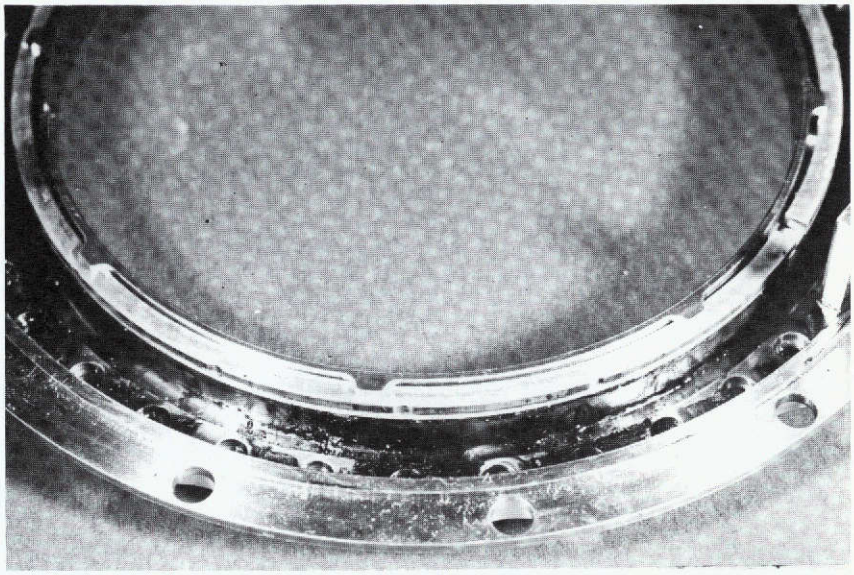
BACK OF CARRIER SHOWING LACK OF COKING
IN NOSEPIECE SEATING AREA

ENCLOSURE 57

CF 850687 SUPPORT OF NASA OIL SEAL AT COMPLETION OF REPEAT EXTENDED DURATION TEST USING HUMBLE FN-3158 OIL BLENDED WITH 5% KENDALL 0839 RESIN AFTER 6.3 HOURS AT SPEEDS TO 20,100 RPM



INTERIOR OF SUPPORT SHOWING COKING

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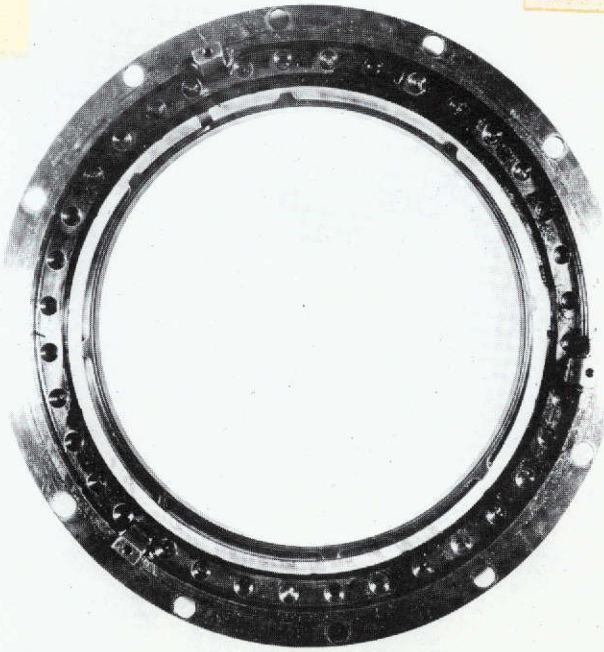
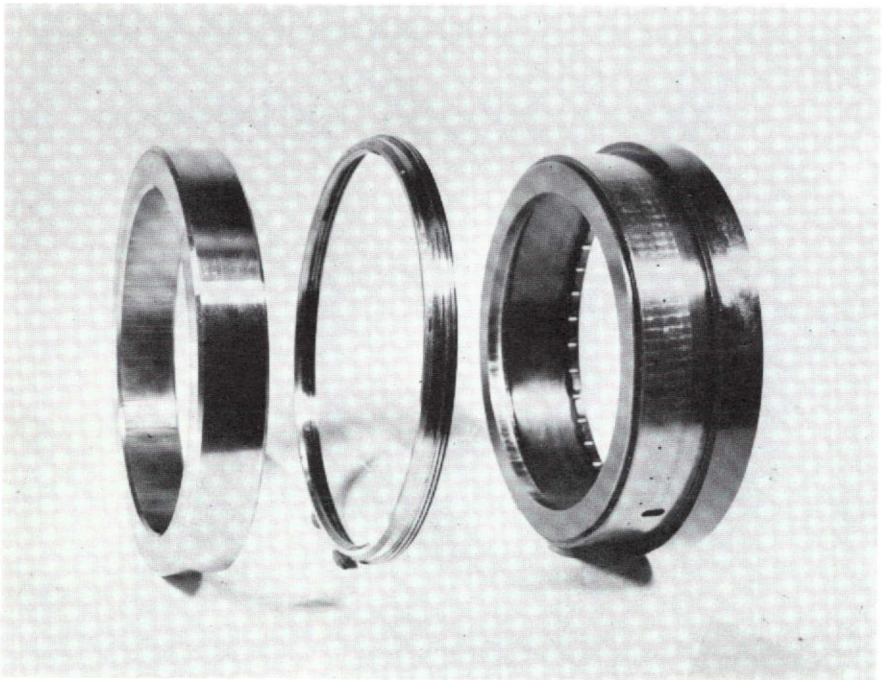


BACK OF SUPPORT SHOWING COKING

ENCLOSURE 58

NASA OIL SEAL MATING RING MOUNTING COMPONENTS AT COMPLETION OF REPEAT EXTENDED DURATION TEST USING HUMBLE FN-3158 OIL BLENDED WITH 5% KENDALL 0839 RESIN AFTER 6.3 HOURS AT SPEEDS TO 20,000 RPM



APPENDIX I

CONTRACT WORK STATEMENT

NAS3-14320

HIGH-TEMPERATURE LUBRICANT SCREENING AND SYSTEMS STUDIES

I

The Contractor shall perform the work described below:

Task I - 125-mm Bearing and Seal Assembly Test Rig and Test Elements

The test rig and test elements are as follows:

A. Test Rig

1. The Contractor shall provide the Contractor-owned 125-mm bearing and seal assembly test rig that is equipped with a recirculating lubrication system and was used previously under NASA contract NAS3-6267. The temperature distribution on the test bearing and seal assembly and test cavity shall be such that no part is at a temperature of more than 25°F below the outer ring bearing test temperature. To insure that the heat flow path will be from the inner to the outer ring of the test bearing, the inner ring temperature shall be maintained at least 10°F hotter than the outer ring temperature. Oxygen content shall be monitored in the test cavity.

2. Modifications to the test rig specified in paragraph 1 of this Task I shall be made as follows:

   a. Provide necessary hardware modifications to allow operation at increased speeds, up to and including 24,000 rpm (3×10^6 DN), if possible.

   b. Provide facilities (e.g., searchcoils) to monitor test bearing ball motion, using existing available electronic equipment. An existing pyrometer shall be used to determine cage temperatures, if feasible. The temperature capabilities and sensitivity limits of this equipment shall be determined during shakedown testing, and it shall be used within these capabilities.

   c. Provide necessary hardware to permit under race cooling for the bearing inner ring. The modified system shall be subjected to shakedown testing prior to the use in the subsequent tasks.

B. Bearings

The Contractor shall utilize fifteen (15) Government-Furnished Angular-Contact, split-inner-ring type, 125-mm bore diameter ball bearings. The bearing design shall be SKF Industries No. 459981G. The balls and races shall be of consumable-electrode vacuum melted M-50 steel. The cage shall be no. FS/W40 and shall be made of AISI 4340 steel and have electroplated silver coating. The bearings shall be designed for operation at shaft speeds up to 24,000 rpm (3×10^6 DN value) and at outer race temperatures of 700°F.

C. Seals

1. For initial check out and preliminary testing in this program one of the existing Government-owned double seal assemblies (mean face diameter of 6.33 inches) from contract NAS3-6267, to be selected by the Contractor with the approval of the NASA Project Manager, shall be used.

2. The test seal to be used in the step speed testing and fifty (50) hour evaluation tests of lubricants as specified in Task II shall be as follows: (If needed to maintain acceptable test schedules, existing Government-owned seals shall be used with the approval of the NASA Project Manager.)

   a. The test seal shall be constructed as a pressure-balanced double face seal assembly and be of the type depicted in Koppers Drawings 700495, 700513 and 101056B. There shall be an <u>air</u>-seal exposed to 1200°F air temperature and a pressure drop across the seal of nominally 5 psi; and an <u>oil</u> seal exposed to the bearing chamber oil temperature and a pressure drop across the seal of 105 psi. The seal springs or bellows shall be made of Inco 718 or equivalent high temperature material. Damping shall be used.

   b. All seals shall have sealcarbons of National Carbon CDJ-83 (or equivalent) and shall otherwise be made according to design and detail drawings approved by the NASA Project Manager under this contract. Two new oil seals shall be provided by the Contractor.

   c. The secondary seal shall be of a bellows type or piston ring type. Springing force exerted by the bellows shall be determined by the seal manufacturer.

3. A continuous leakage in excess of 5 scfm, or as directed by the NASA Project Manager upon recommendation by the Contractor, across any single seal under the conditions described above shall be considered a seal failure. Gas flow-meters in the nitrogen and high-pressure air lines and mass spectrometer monitoring of the interseal and bearing test chambers, using a helium tracer in the nitrogen blanketing gas, shall be used to provide maximum flexibility in tracing seal leakage paths.

4. Nitrogen gas shall be available for supply to the space between the two seals at a pressure of about 5 psi higher than the 1200° F air pressure.

5. In the event test seals become functionally inoperative during testing they shall be repaired up to two reworks on the oil seals and one air seal rework.

D. Test Lubricants

1. The following lubricant test fluids shall be provided by the Contractor for evaluation as described in Task II. These fluids shall be from a single controlled lot of each fluid.

   a. Humble FN-3158 formulated super-refined naphthenic mineral oil plus 5 percent by weight Kendall 0839 heavy paraffinic resin (baseline fluid).

   b. Monsanto MCS-2931, improved modified polyphenylether.

   c. A synthetic mineral oil derivative (e.g., Conoco DN-600 Fluid, Type 2, formulated synthetic hydrocarbon or Mobil XRM-214B, formulated synthetic paraffinic hydrocarbon) to be selected by the Contractor, subject to the approval of the NASA Project Manager.

   d. One additional fluid to be selected by the Contractor, subject to the approval of the NASA Project Manager.

Task II - Lubricant Evaluation, 125-mm Bearing and Seal Assembly Rig.

The Contractor shall perform tests, under the conditions and following the procedures as described below, using the test lubricants listed in Task I, paragraph D.1. in the 125-mm bearing and seal assembly rig to determine their relative lubricating abilities, extent of corrosion, system deposits, and modes of failure in an open atmosphere recirculating lubrication system.

A. Test Conditions

1. Bearing outer ring temperatures shall be maintained as close as possible to 500° F (±15° F). Test lubricant being supplied to the test bearings shall be maintained at a temperature of 400° F (±10° F) with the system open to the atmosphere.

2. Inner ring (shaft) rotational speed shall be at the highest feasible speed up to a maximum of 24,000 rpm as established in the Test Procedure, paragraphs B.1. to B.6 of this Task.

3. The bearing thrust load for all testing shall be 3280 pounds.

4. Total oxygen content of the test cavity during the tests shall be measured.

5. The moisture content of the incoming air to the test cavity shall be maintained at 50% (±10%) relative humidity at atmospheric conditions, unless total seal leakage exceeds 10 SCFM.

B. Test Procedures

1. The testing program shall consist of running each of the previously enumerated fluids listed in Task I, paragraph D.1. in the recirculating rig described in paragraphs A.1. and A.2. according to the following testing schedule:

Test Fluid

a. Humble FN-3158 plus 5% Kendall 0839 (baseline fluid)

b. Monsanto MCS-2931

c. A synthetic mineral oil derivative to be selected

d. One additional fluid to be selected.

2. Air at 50% (±10%) relative humidity shall replace the nitrogen previously supplied to the space between the two seals at a pressure of about 5 psi higher than the pressure of the 1200° F air, and the test cavity shall be vented overboard. Measurement of relative humidity shall be limited to operation when total seal leakage is 10 SCFM or less.

3. Using a new test bearing with each test fluid the bearing temperature shall be stabilized at a shaft speed of 14,000 rpm with the thrust load held at 3280 pounds. Time required for stabilization of temperature should be in the range of one to two hours. The speed shall then be slowly increased in increments of about 2000 rpm, test bearing temperature being allowed to stabilize after each speed increase, until bearing or seal failure occurs or until 24,000 rpm speed reached, or until uncorrectable rig limitations arise which are uncorrectable with reasonable effort. Power consumption, measured by power input or torque developed, shall be determined for each set of stabilized test conditions for each fluid. The Humble FN-3158 fluid plus 5% Kendall 0839 resin shall be tested first, and the power consumption required for stable operation of this fluid shall be determined as a function of oil flow at five flow levels (i.e., 2.0, 1.5, 1.0. 0.75 and minimum gpm) unless rig limitations prevent the lower flows. Subsequent testing with this fluid and the other test fluids shall be at the same minimum feasibile oil flow level to be chosen by the Contractor, subject to the approval of the NASA Project Manager. If feasible without interrupting the test sequence, and subject to the limitations given in Task I, paragraph A.2.b., cage temperature measurements using the pyrometer assembly or other suitable ball motion determination method shall be made.

4. Two fluids of the 4 tested as in the preceding paragraph shall be selected by the Contractor and approved by the NASA Project Manager. The test bearing, if serviceable, or a new bearing and serviceable seals, shall then be tested with each of these fluids at the most severe feasible speed condition, as approved by the NASA Project Manager, for a total running time of 50 hours each. These maximum speed tests shall be run in periods of from 5 to 10 hours duration between which the test rig shall be shut down and allowed to cool for a period sufficient to reach a bearing temperature not exceeding 200° F (probably within one to two hours) before starting another test cycle. Cage temperature measurements using the pyrometer assembly or other suitable ball motion determination method shall be made on one bearing with each of the two lubricants at each set of conditions selected for 50-hour testing, subject to the limitations given in Task I, paragraph A.2.b.

5. The experimental rig shall be operated under each set of conditions specified for a duration of 50 hours or until failure is indicated by (a) a sudden rise in the test bearing torque, temperature or vibration or (b) an increase in seal leakage in excess of 5 SCFM across any single seal at 100 psi Δp, or (c) excessive coking of the oil.

6. Prior to conducting these tests, the Contractor shall develop and submit a test plan to the NASA Project Manager which will be subject to NASA approval. Included in this plan shall be the designated order of running these tests.

C. Data Required

1. The Contractor shall submit to the NASA Project Manager photographs of the test cavity showing the test seals and test bearing documenting the extent and nature of the lubricant coking and the visible wear of the test components from each test run disassembly. Samples of system deposits shall be collected. Portions of those samples shall be analyzed and other portions shall be delivered, properly identified, to the NASA Project Manager, at his request. Sump oil samples shall be collected after each screening test run and after each 5 to 10 hour run during the 50 hour runs. Selected samples shall be analyzed as agreed with the NASA Project Manager.

2. Bearing ball motion shall be monitored at the highest feasible speed for each of the two lubricants selected for 50-hour testing within the limitations set forth in Task I, Paragraph A.2.b.

D. Calibration and Additional Data Requirements

1. Critical temperatures shall be monitored by thermocouple/ recorder systems which shall be calibrated at the start of the tests, and after each major repair, against systems whose calibration over the temperature range is traceable to the National Bureau of Standards. Pressure and flow sensors shall be calibrated by the Manufacturer or other suitable vendor at the beginning of the program and after all major repairs.

2. Equipment records shall be maintained. Dated entries shall be made for all calibration results, all inspection data, and all maintenance operations on the equipment.

3. Technical record logs shall be established by the Contractor and dated entries made to provide a means for documenting the history of each seal throughout its testing cycle.

4. The above logs shall be fully maintained and be available for review by the NASA Project Manager.

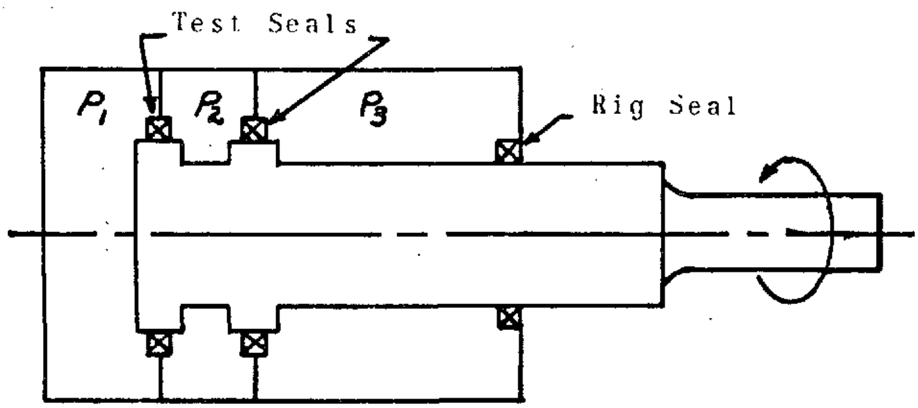
5. All data recorded under this contract shall be made available to the NASA Project Manager upon request.

APPENDIX II

CALCULATIONS OF THE LOAD APPLIED TO THE TEST BEARING BY PNEUMATIC PRESSURE

APPENDIX II

CALCULATIONS OF THE LOAD APPLIED TO THE TEST BEARING BY PNEUMATIC PRESSURE



Let P_1 be pressure in hot air chamber.
P_2 be pressure in inter-seal cavity.
P_3 be pressure in bearing chamber.

The pressure P_1 acts over the entire hot air end of the test rig as shown.

Assuming the hydraulic diameters of the two test seals are identical, then there is no net thrust due to the pressure P_2.

If d_T be the hydraulic diameter of a test seal and d_R be the hydraulic diameter of the rig seal then the thrust generated is given by:

$$T = P_1 \frac{\pi}{4} d_T^2 - P_3 \frac{\pi}{4}(d_T^2 - d_R^2)$$

$$= \frac{\pi}{4}\left\{d_T^2(P_1 - P_3) + d_R^2 P_3\right\}$$

APPENDIX II (Cont.)

Now d_T = 6.370"
and d_R = 4.500"

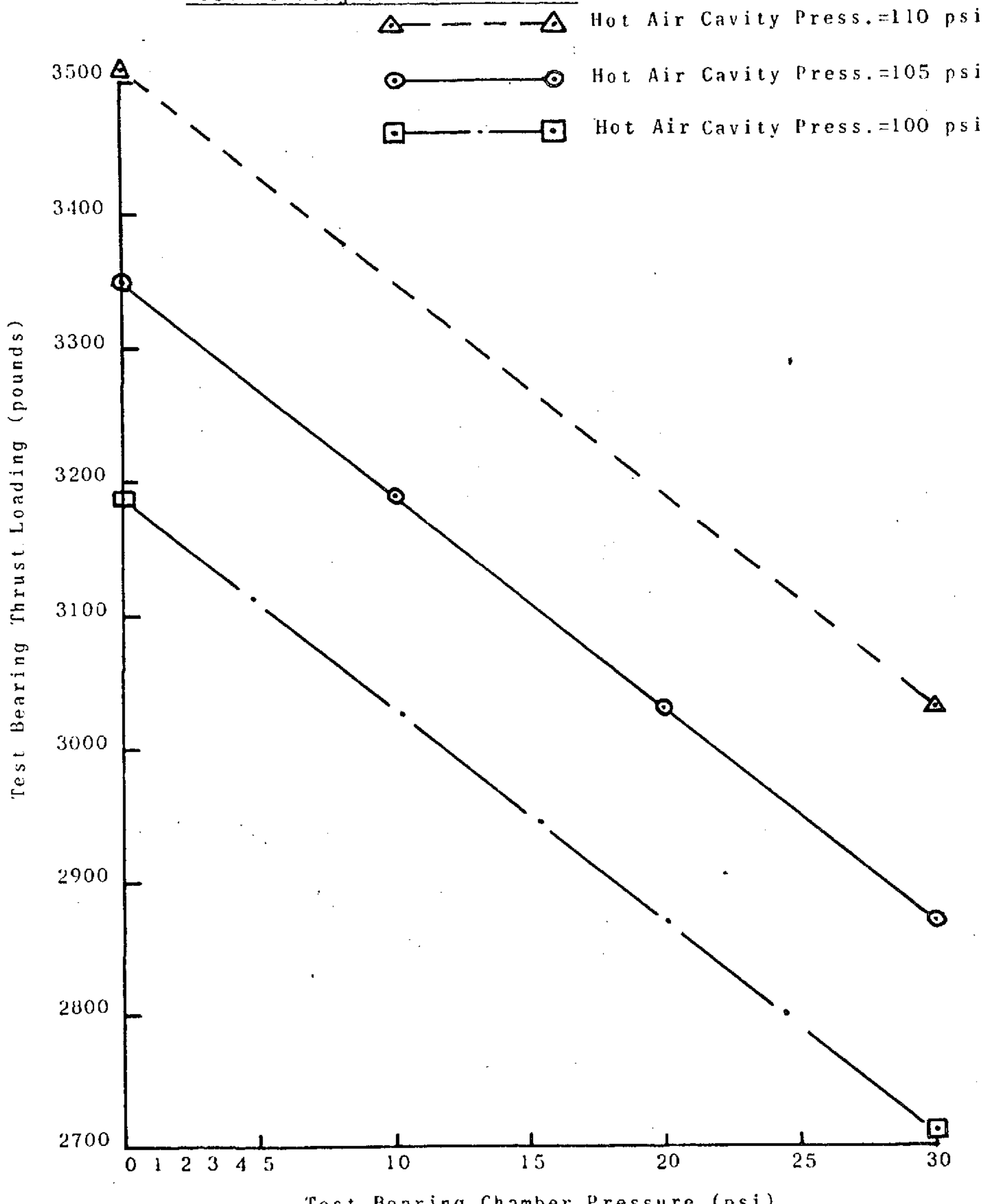
So that

$$T = 31.9\,(P_1 - P_3) + 16.0\,P_3$$

It has been found that P_1 is effectively constant during operation. Assuming P_1 is constant then and varying P_3 figure 1 is generated (for P_1 = 100, 105, and 110 psi).

APPENDIX II (Cont.)

Fig. 1 Test Bearing Thrust Loading
vs.
Test Bearing Chamber Pressure



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APPENDIX III

SKF REPORT NO. AL72Q021

EXPERIMENTAL OBSERVATION OF BALL KINEMATICS IN HIGH SPEED BEARINGS

REPORT ON TECHNOLOGICAL AREAS OF INTEREST

AUGUST, 1972

EXPERIMENTAL OBSERVATION OF BALL KINEMATICS IN HIGH SPEED BEARINGS

August 25, 1972

Contributor

T. E. Tallian

Released: T E Tallian

Report: AL72Q021

RESEARCH LABORATORY
SKF INDUSTRIES, INC.
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KING OF PRUSSIA, PA.

REPORT ON TECHNOLOGICAL AREAS OF INTEREST
AUGUST, 1972

EXPERIMENTAL OBSERVATION OF BALL KINEMATICS IN HIGH SPEED BEARINGS

In experimental studies of high speed bearings, the rotation of rings and cage is of course readily observed. It has become common to measure cage speed by proximity probe and utilize the results in the interpretation of bearing kinematics.

Recent kinematics analyses, taking into account elastohydrodynamic theory, have led to the abandonment of "raceway control" theories, which assumed that, in a thrust loaded ball bearing, ball spin takes place only with reference to one of the two races. Skidding and gyroscopic spin effects are also well demonstrated. All these factors suggest that any hypothesis about the position of the instantaneous axis of rotation of a ball is precarious. Without such a hypothesis, knowledge of the cage rotational velocity does not suffice to deduce the kinematics of the ball, and hence the distribution of sliding velocity in the ball to race contacts. For precise heat loss calculations in the bearing, as well as for the prediction of skid marking damage, knowledge of sliding velocity distributions is indispensable, and accordingly, there is increasing need to observe ball motions experimentally.

About a decade ago, Shevchenko and Bolan (1)* published experimental results of ball motion in an angular contact bearing, obtained by high speed kinematography of a ball on which black spots had been etched to identify its position. This information has long been the sole direct experimental source of ball motion data. The Shevchenko-Bolan method is questionable, however, because visible spots on a ball are likely to have unusual traction properties, even if they are not geometrical imperfections. They may, accordingly, modify the delicate balance of tractive forces on a ball. More importantly, the need for optical access to a ball severely limits the conditions under which the bearing can be operated, with respect both to bearing and cage configuration and to lubrication. Interpretation of results is lengthy and difficult, and records cannot practically be compiled to cover long periods of running.

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

More recently, Hirano and Tanoue (2) have applied a method repeatedly suggested before, i.e. the permanent dipole magnetization of a ball and the observation of its motion by winding a concentric sensing coil around the outer ring of the bearing. As the magnetized ball moves, the flux encompassed by the coil varies with time, and one obtains induced voltage/time traces, showing the periodicity of the ball rotation around its own axis, amplitude-modulated by lower frequency occurrences. For the latter, Hirano et al. propose interpretations based on a gradual relocation of the magnetic axis of the ball in the bearing. Hirano's method was developed for radially loaded bearings, and the kinematic theory used to interpret the output of his one coil is not clear. We now believe it to be a rather rough approximation.

In several programs of study for high speed thrust loaded angular contact ball bearings, we have attempted to improve on the Hirano method. We are using a single ball, permanently magnetized to form a dipole. In experiments to date, we have placed a single concentric coil of wire coaxially with the outer ring. Encl. 1 shows typical output voltage traces which appear to fall into three categories, viz:

Trace A, a sine wave reflecting ball autorotation with superimposed, relatively small random amplitude variations

Trace B, a similar sine wave but strongly amplitude-modulated with a lower-frequency sinusoid, and

Trace C, short duration, sharp disturbances of the sinusoidal fundamental frequency, extending over only a few cycles and having amplitudes up to twice the base amplitude, which appear at rare and irregular intervals, and only in certain tests.

Trace A is believed to reflect a ball rotating around an essentially unchanged axis.

Trace B shows a systematic change in ball axis with time. We have observed this type of behavior during a few minutes before a bearing underwent dramatic smearing failure, but also in other test conditions where there was no manifest failure but where the lubricant was unusual (polyphenylether). This harmonically modulated trace is thought to be that of a "precessing" ball, in which the rotational axis changes angular position periodically with respect to the ball magnetic axis. We do not know precisely how the precession is brought about, but it is noted that Hirano has also observed these precessions and has a kinematic theory to explain them. This theory does not involve "failure", i.e. an irreversible change in the operating mode of the bearing.

The excursions shown in trace C are tentatively attributed to microsmearing occurrences or similar events causing abrupt changes in ball axis. Assume that there is a microsmearing occurrence at one point of the elongated Hertzian contact in an angular contact ball bearing, and that this occurrence is not coincident with the point where the ball spin axis pierces the contact area. Then, one can postulate that the microsmearing occurrence will momentarily "lock" the ball and opposing race surfaces together at the point where the smearing occurs. Locking occurs with force sufficient to do permanent damage to the surfaces. It represents a discontinuous shift of the spin velocity vector from its previous position. When the smearing occurrence terminates (as soon as the asperity weld breaks upon lift-off of the contacting surfaces), the spin velocity vector will shift back to its previous position, assuming that the latter was stable. These two abrupt shifts in spin velocity vector, following in rapid sequence, can be visualized as imparting high momentary velocities to the ball motion, thereby creating a pulse of voltage in the sensing coil. (Of course, coil output voltage is proportional to the velocity of motion of the ball magnetic axis and not to its displacement.) We have observed C-type voltage disturbances only in tests where smearing was subsequently detected and only at points in time preceding the smearing occurrence by no more than a few minutes. Thus, we are conjecturing that the c-type trace excursions are indications of damage occurrences, which could be smearing, a rolling over a relatively large dirt particle, and perhaps others.

Traces such as those in Encl. 1 require, for the quantitative interpretation of ball motion, further development of both theory and experimental methods. We have embarked on such work as follows:

Our test rig in which these experiments are conducted is a high temperature rig in which the shaft supporting the bearing and the bearing housing are made of Inconel, a non-magnetic alloy. The bearing itself is made of martensitic steel and has a martensitic steel machined cage as shown in the product drawing of Encl. 2. One can explore the magnetic field that is generated around this bearing by a magnetized ball, by probing the surroundings of the bearing placed in air, since the shaft and housing will not channel the field appreciably. We have mounted a dipolar magnetized ball in the bearing, marking the location of its magnetic axis on the surface, and hand positioned this axis at various known angles to the bearing axis. A first series of tests was made with the magnetic dipole axis $\bar{M}$ in the radial plane of symmetry through the ball center and bearing axis, and positioned at various angles α with respect to the bearing axis. Further tests were made by placing the axis at different angles β away from this plane of symmetry (see Appendix I for definition of angles).

In each test, a Hall effect Gauss-meter probe was read at a series of positions on the bearing ring or cage surface, in the plane of symmetry going through the ball center and the bearing axis, to give a field map. Some measurements were also taken circumferentially away from this plane of symmetry. The resulting plots show these principal characteristics:

1. The magnetic flux densities at the points on the outer ring surface marked "N" and "J" on Encl. 2 and Encl. 6 are similar in amplitude and phase. The flux density at point "O" on the cage surface is similar in amplitude but out of phase with the outer ring surface as a function of ball axis angles.

2. Appreciable flux density exists over a circumferential distance of about one ball diameter in each direction from the plane of symmetry. Beyond this point, the field decreases rapidly.

3. Over small perpendicular distances away from the ring and cage surfaces, the flux density does not vary appreciably.

4. The flux density at all probe points is periodic as a function of the angles α and β describing the position of the magnetic ball axis. The nature of periodicities as a function of α and β respectively, are different.

We have constructed a first-order mathematical model describing the magnetic flux density measured at the surface of the outer ring at point N and at the surface of the cage at point O, as a function of the two angles α and β characterizing the position of the magnetic ball axis. The formulae obtained by curve fitting to experimental results are:

For Point N

$$B_N = B_{N,0}\left\{3.5\cos^{0.8}\left(\alpha-\frac{\pi}{4}\right)\cdot(\cos\beta-1)+(2.8-0.7\cos 2\alpha)(1.77-0.77\cos\beta)\right\} \quad [1]$$

For Point O

$$B_0 = B_{0,0}\left\{1.58\cos^{12}\left(\alpha-\frac{\pi}{4}\right)(1-\cos\beta)+(6+4.1\cos 2\alpha)(0.01\cos\beta+0.99)\right\} \quad [2]$$

Encls. 3 through 5 show plots of these equations and experimental points of measured magnetic flux density. The fit appears reasonable, although the substantial measurement scatter and the crudeness of the model cause considerable error.

Eqs. [1] and [2] provide us with relations connecting magnetic flux density outside the bearing (in the plane of symmetry of the magnetized ball) with ball magnetic axis position.

We have also performed a kinematic analysis describing the variation of the angles α and β in time, as a function of ball autorotation around an arbitrary but fixed momentary axis. Relationships given in Appendix I relate (a) the angular position (γ, δ) of the ball autorotation axis in the bearing; (b) the included angle ζ between dipole axis and autorotation axis (3);and (c) the rotational motion ωt of the dipole axis around the autorotation axis, to the directional angles α and β of the dipole axis, as a function of time.

Inserting the $\alpha(t)$ and $\beta(t)$ functions obtained into Eqs. [1] and [2], one obtains magnetic flux densities in the plane of symmetry of the magnetized ball (a) at the side face of the outer ring (B_N) and (b) at the side face of the cage (B_0) as a function of time.

We have devised sensing coils, the voltage output of which is related to these magnetic intensities, as follows:

Encl. 6 shows two concentric differential coils. Each of the differential coils consists of two thin cylindrical coils with an equal number of windings wound in opposite senses, and placed concentrically within each other, leaving a ring-shaped gap. When such a differential coil is placed in a magnetic field, which is approximately perpendicular to the plane of the coil, and the magnetic field changes with time, a voltage is induced by only that portion of the magnetic flux which penetrates the ring-shaped space between the two half-coils. Magnetic flux within the inner half coil or outside the outer half coil induces opposite voltages in the two half-coils which cancel. The magnetic flux in air near any ferromagnetic surface is essentially perpendicular to the surface, i.e. essentially perpendicular to the plane of the coils. The flux density is high near the plane of symmetry of the magnetized ball, and low elsewhere. The total flux penetrating the annulus of the differential coil is independent of the orbiting motion of the magnetized ball around the bearing. The region of high magnetic intensities orbits with the ball, but this does not change the total flux penetrating the coil, and therefore is not detected as induced voltage.

Appendix II provides approximate equations relating induced voltage in the differential coils to the flux densities B_0 and B_N given in Eqs. [1] and [2]. To the extent that the assumptions in that appendix are valid, one has

$$V_0(t) = K_0 M \frac{d}{dt}\left(\frac{B_0}{B_{0,0}}\right) \qquad [3]$$

$$V_N(t) = K_N M \frac{d}{dt}\left(\frac{B_N}{B_{N,0}}\right) \qquad [4]$$

where K_0, K_N depends only on bearing and coil geometry, and $B_0/B_{0,0}$, $B_N/B_{N,0}$ are taken from Eqs. [1] and [2] respectively.

The strength of the ball magnetization dipole M can vary slowly with time without influencing the V_0/V_N ratio.

The first approach to an interpretation of $V_0(t)$ and $V_N(t)$ plots in terms of ball motion will be to construct $\frac{d}{dt}(B_0/B_{0,0}(t))$ and $\frac{d}{dt}(B_N/B_{N,0}(t))$ plots, using the formulas in Appendix I and Eqs. [1] and [2] for postulated simple ball kinematic conditions. This yields two independent curves.

One can try to match the shape of these curves to experimental V_0, V_N curves.

Secondly, a ratio $dB_N/dt : dB_0/dt$ can be calculated and compared to the experimental V_N/V_0 ratio.

From Eqs. [3] and [4]

$$\frac{V_N}{V_0}(t) = \frac{K_N}{K_0}\left[\frac{dB_N}{dt} : \frac{dB_0}{dt}\right] \qquad [5]$$

i.e. the two ratios differ only by a constant. This should permit closer matching of the curves.

Finally, K_N/K_0 can be calibrated if a ball with known position of $\bar{\omega}_b$ and $\bar{M}$ is run in the bearing. This, of course, requires mechanical guidance of the ball, without change of its magnetic properties, e.g. by mounting it in the cage on a thin axle or pin.

Some possible ball kinematic conditions to be studied follow:

1. Ball rotation around a constant axis parallel to the bearing axis. ($\gamma = \delta = 0$, $\zeta \neq 0$).

2. Ball rotation around a constant axis, defined by the assumption of outer ring control in a bearing of slow speed. ($\gamma \neq 0$, $\delta = 0$, $\zeta \neq 0$).

3. An exploration of the effect of a change in ball rotational axis from outer ring control to inner ring control, and of the divergence of inner ring and outer ring contact angle arising from high speed operation. ($\gamma = \gamma(t)$; slow change, $\delta = 0$, $\zeta \neq 0$)

4. An exploration of the effect of the momentary "locking" of a point in the Hertzian contact as described above for a microsmearing occurrence. ($\gamma(t)$ "pulses").

5. An exploration of the effect of the superposition of gyroscopic rotation (around an axis perpendicular to the radial plane of symmetry of the ball). ($\mathfrak{z}=\mathfrak{z}(t)$, slow change or pulse)

If these exploratory studies are successful, we will use the method here described for the analysis of high speed bearing ball kinematics.

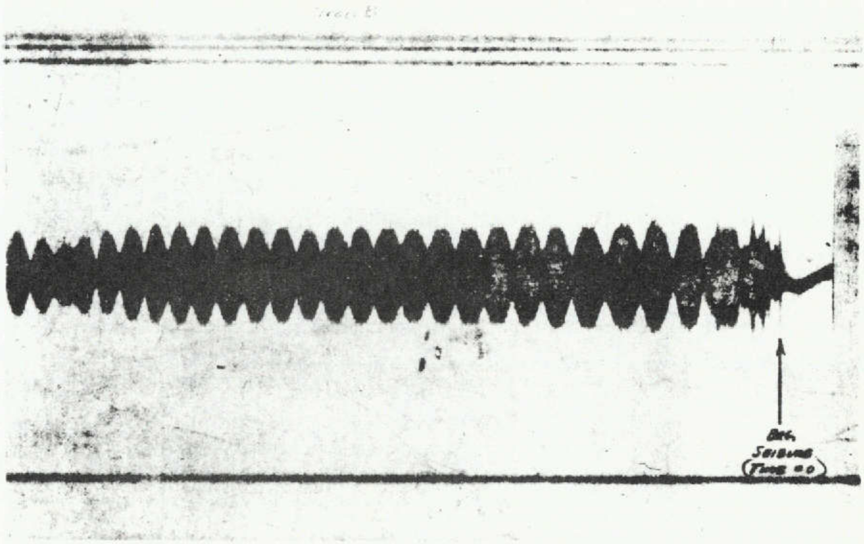
REFERENCES - APPENDIX III

1. Shevchenko, R. P., Bolan, P., "Visual Study of Ball Motion in High-Speed Thrust Bearing", SAE Paper 37, January 1957.

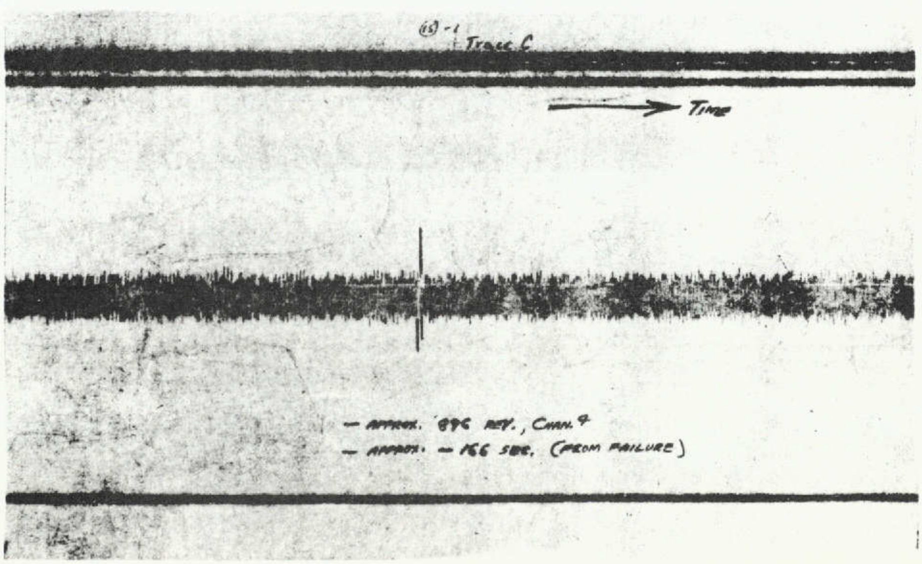
2. Hirano, F., Tanoue, H., "Motion of a Ball in a Ball Bearing", WEAR 4, 177-197, 1961.

ENCLOSURE III-1

Trace A



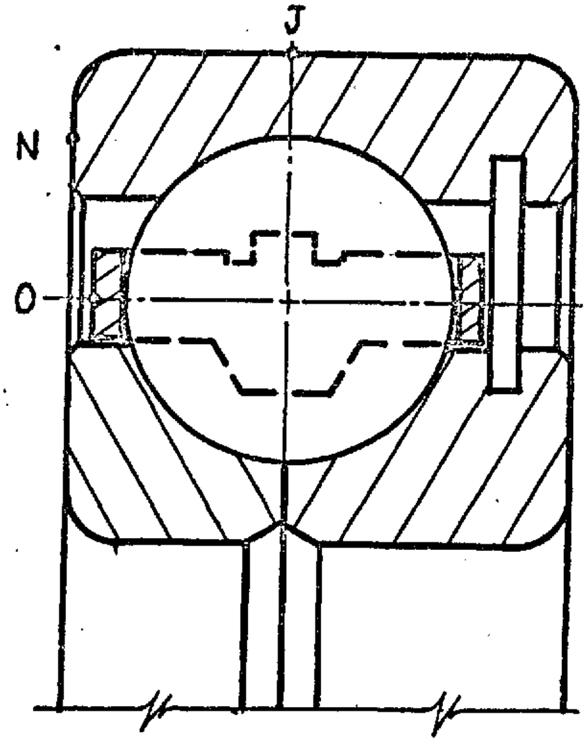
Trace B



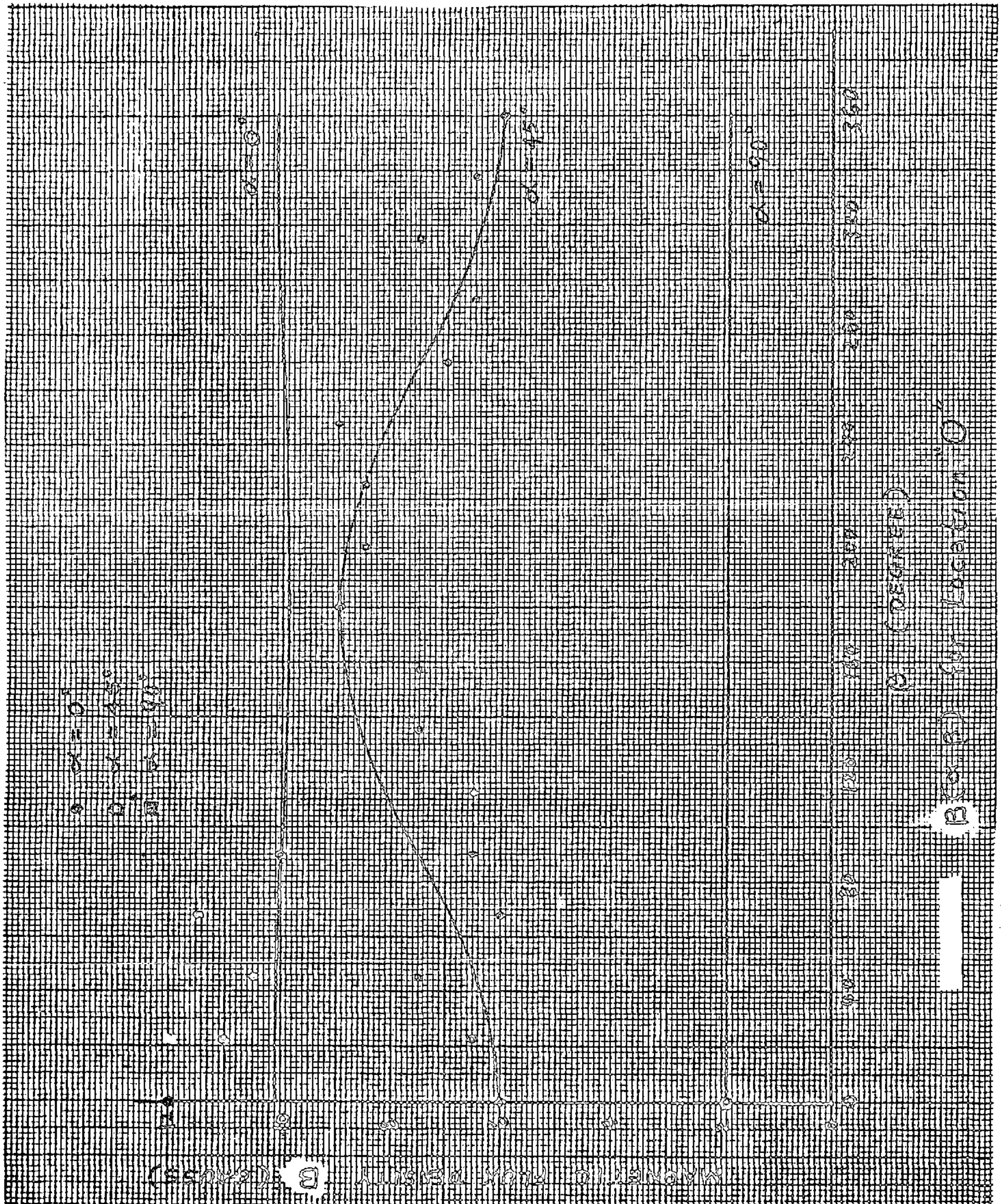
Trace C

ENCLOSURE III-2

CROSS SECTION OF EXPERIMENTAL
SPLIT INNER RING BALL BEARING

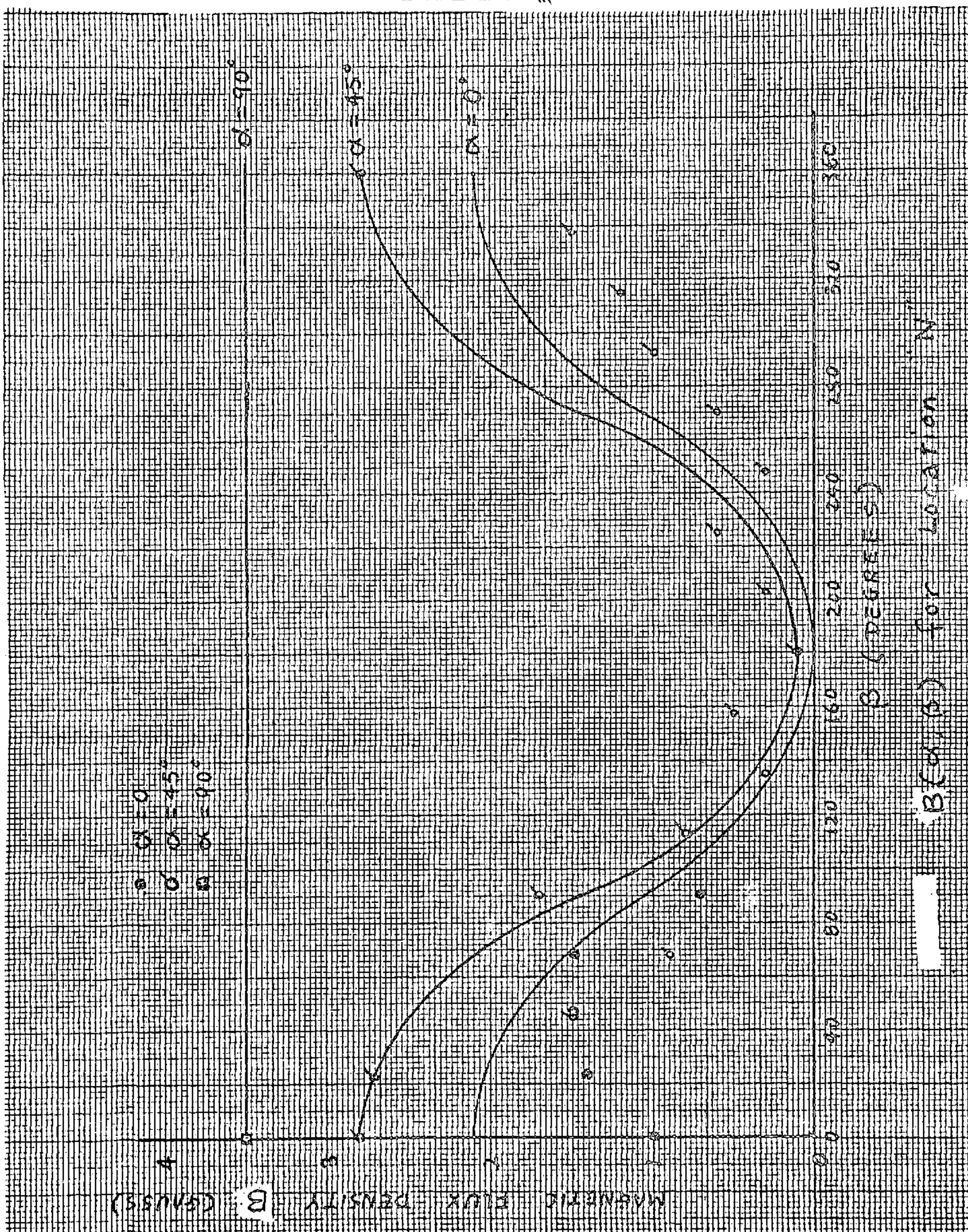


ENCLOSURE III-3

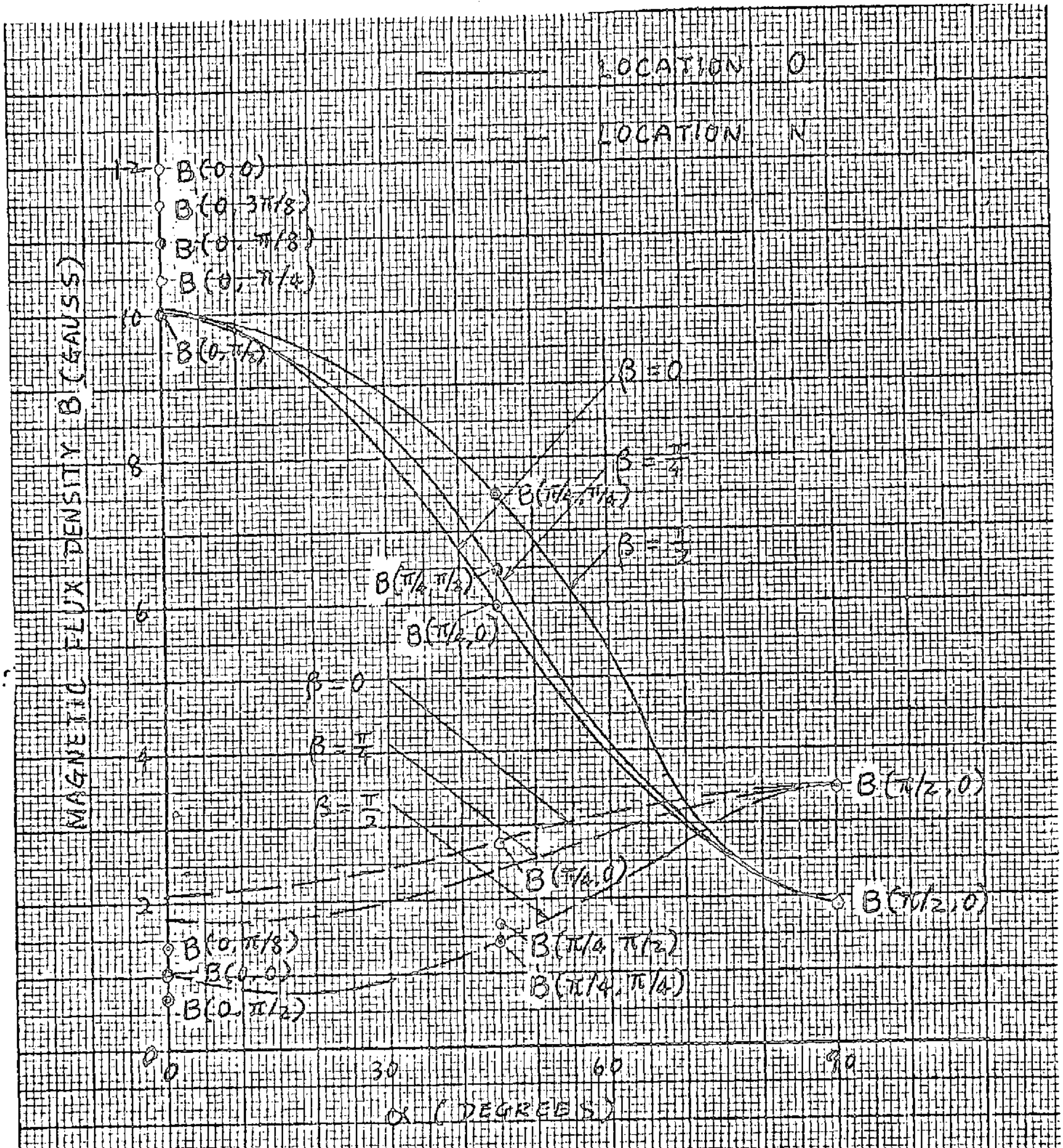


RESEARCH LABORATORY SKF INDUSTRIES, INC.

ENCLOSURE III-4



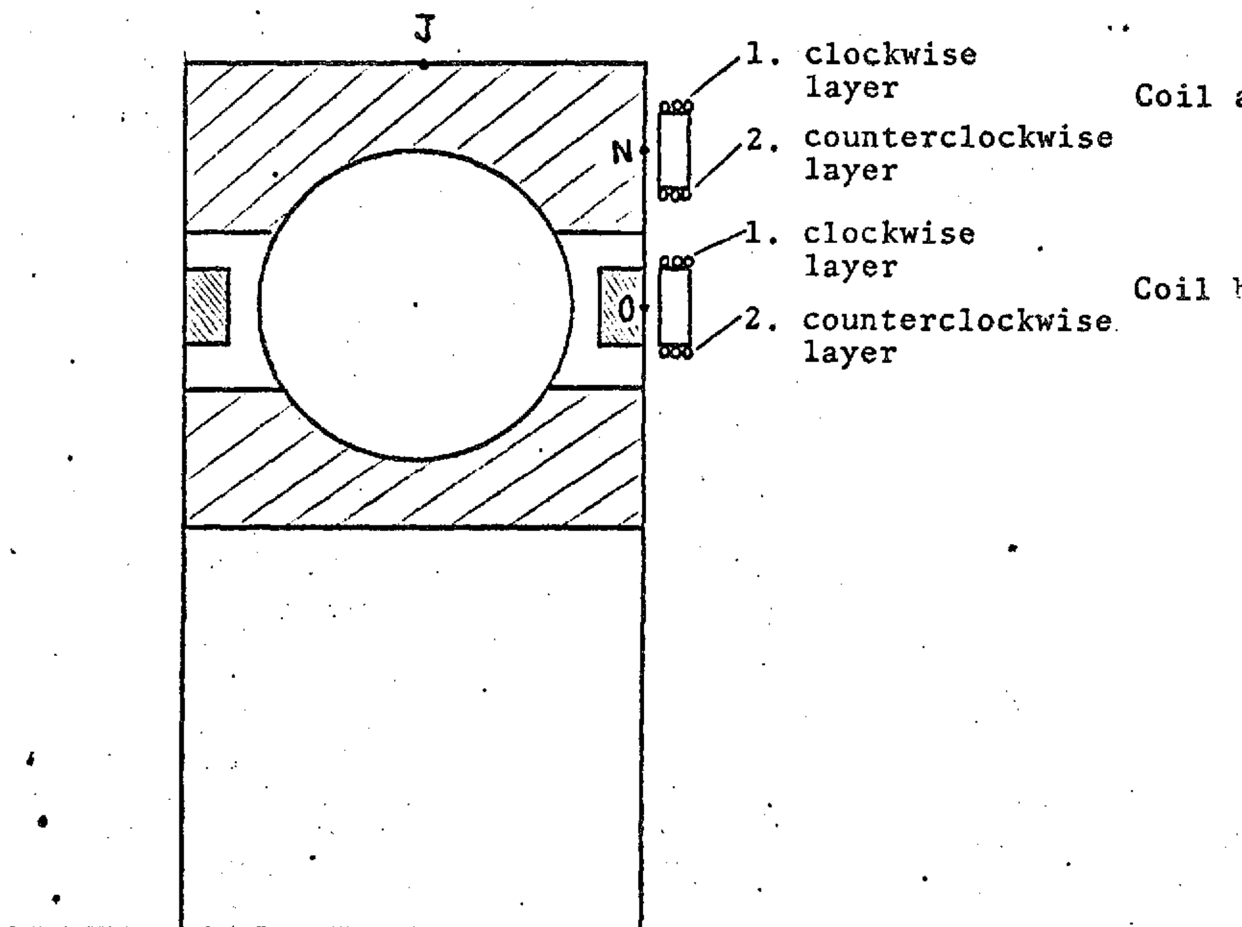
ENCLOSURE III-5



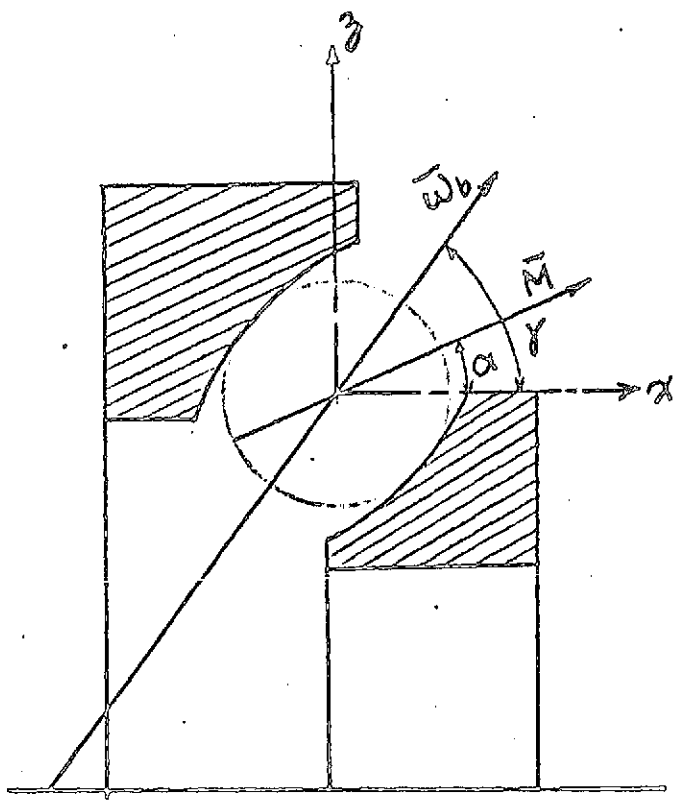
B(α,β) for Locations O and N

ENCLOSURE III-6

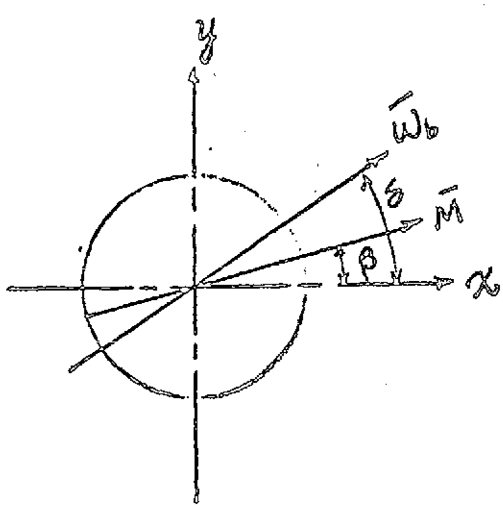
DIFFERENTIAL SENSING COIL ARRANGEMENT



RELATIONSHIP OF BALL MAGNETIC AXIS POSITION TO BALL MOTION



Bearing Axis



Absolute Value of $\bar{\omega}_b = \omega_b$

Angle$(\bar{M}, \bar{\omega}_b)$ = ζ

1) Directional angles of $\bar{\omega}_b$

$$\cos \Omega_1 = \cos \gamma \cos \xi \qquad [1-1]$$

$$\cos \Omega_2 = \cos \gamma \sin \xi \qquad [1-2]$$

$$\cos \Omega_3 = \sin \gamma \qquad [1-3]$$

2) Auxiliary angles

$$\cos \varphi_1 = -\cos \Omega_2 - \frac{\cos \varphi_3 \cos \Omega_3}{\cos \Omega_1} \qquad [1-4]$$

$$\varphi_2 = \Omega_1 \qquad [1-5]$$

$$\cos \varphi_3 = \frac{1}{\sin^2 \Omega_2}\left\{-\cos\Omega_1 \cos\Omega_2 \cos\Omega_3 + \cos\Omega_1\left[(1-\cos\Omega_2)(1+\cos^3\Omega_2)-\cos^2\Omega_1\right]^{1/2}\right\} \qquad [1-6]$$

$$\cos \lambda_1 = \cos \Omega_2 \cos \varphi_3 - \cos \varphi_2 \cos \Omega_3 \qquad [1-7]$$

$$\cos \lambda_2 = \cos \varphi_1 \cos \Omega_3 - \cos \Omega_1 \cos \varphi_3 \qquad [1-8]$$

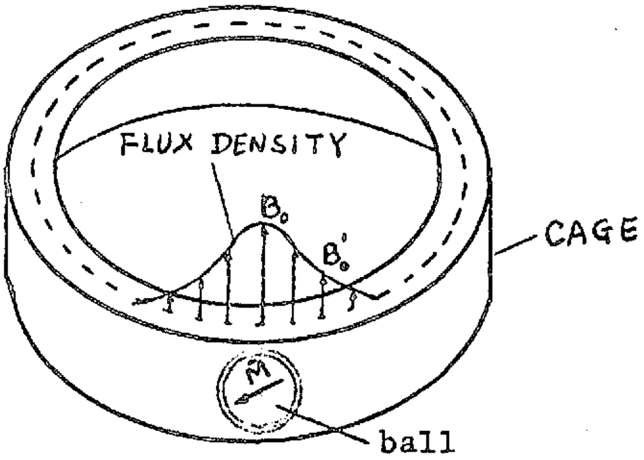
$$\cos \lambda_3 = \cos \Omega_1 \cos \varphi_2 - \cos \varphi_1 \cos \Omega_2 \qquad [1-9]$$

3) Equations for α and β

$$\sin\alpha = \cos \zeta \cos \Omega_3 - \sin \zeta \cos \varphi_3 \sin \omega_b t + \sin \zeta \times \qquad [1-10]$$
$$\times \cos\lambda_3 \cos \omega_b t$$

$$\tan\beta = (\cos\zeta \cos\Omega_2 - \sin\zeta \cos\varphi_2 \sin\omega_b t + \sin\zeta \cos\lambda_2 \times$$
$$\times \cos\omega_b t)/(\cos\zeta\cos\Omega_1 - \sin\zeta\cos\varphi_1 \sin\omega_b t +$$
$$+ \sin\zeta \cos\lambda_1 \cos\omega_b t) \qquad [1-11]$$

INDUCED VOLTAGE IN DIFFERENTIAL COIL



M strength of magnetic dipole $\bar{M}$ of ball

α, β attitude angles of $\bar{M}$

B_o' flux density at arbitrary position around annulus

B_o maximum flux density in the plane of symmetry of the magnetized ball

Assume a differential coil with annular surface S_o equal to the cage (or ring) side face is immediately adjacent to the side face. Assume, for any location on the side face, that $\frac{B_o'}{B_o}$ is independent of M, α or β. This is realistic if most of the flux is concentrated near the plane of symmetry of the ball and if the shape of the flux distribution does not vary with α or β.

We have

$$B_0 = f_0(M, \alpha, \beta) \tag{2-1}$$

Set

$$B_{0,0} = f_0(M, 0, 0) = \frac{M}{R_0} \tag{2-2}$$

with R_0 a magnetic resistance in the flux path of the cage, then

$$B_0 = B_{0,0} f_0(\alpha, \beta) = \frac{M}{R_0} f_0(\alpha, \beta) \tag{2-3}$$

Likewise at location N, with similar assumptions

$$B_N = B_{N,0} f_N(\alpha, \beta) = \frac{M}{R_N} f_N(\alpha, \beta) \tag{2-4}$$

$B_N/B_{N0} \equiv f_N(\alpha, \beta)$ is given in Eq. [1] and $B_0/B_{0,0} \equiv f_0(\alpha, \beta)$ in Eq. [2]

The induced voltage in the differential coils is

$$V_0 = \frac{d\phi_0}{dt} = \frac{d}{dt}\int_{S_0} B_0' \, dS_0 \tag{2-5}$$

$$V_N = \frac{d}{dt}\int_{S_N} B_N' \, dS_N \tag{2-6}$$

Since $\frac{B_0'}{B_0}$ and $\frac{B_N'}{B_N}$ do not depend on M, α, or β they are also time independent, so that, with [2-3]

$$V_0 \doteq \frac{dB_0}{dt}\int_{S_0} \frac{B_0'}{B_0} dS_0 = \frac{d f_0(\alpha, \beta)}{dt} \cdot \frac{M}{R_0} \int \frac{B_0'}{B_0} dS_0 = K_0 M \frac{d}{dt}\left(\frac{B_0}{B_{0,0}}\right) \tag{2-7}$$

and a similar equation holds for V_N.

APPENDIX IV

SKF REPORT NO. ALL72L083

ANALYSIS OF SEARCH COIL DATA

LETTER REPORT

TO: D. A. Jones

REPORT NO: ALL72L083
PROJECT CODE: LC36812
DATE: June 30, 1972
FROM: O. G. Gustafsson
COPIES TO: L. Sibley
J. McCool

TITLE: ANALYSIS OF SEARCH COIL DATA

REFERENCE: Contract NAS3-14320

1. Use of Search Coil.

In a paper by Hirano and Tanoue (1) the use of a search coil in a study of ball motions in a ball bearing with one magnetized ball is discussed both theoretically and experimentally. This paper applies to radially loaded bearings, while SKF Industries tests were performed on a thrust loaded bearing. Nevertheless, the approach in (1) was applied to the SKF Industries search coil data after is was found that many findings in the paper apply to axially as well as a radially loaded bearings.

One of the balls in the SKF Industries test bearing was permanently magnetized as a dipole (28 Gauss maximum residual flux density at the ball surface). An induction coil, wound concentric with the bearing axis, was placed laterally alongside of the magnetized ball. The alternating voltage emanating from this coil was tape recorded, and the tape analyzed using a high speed writing oscillograph (Honeywell Visicorder, Model 306A) at 0.2 and 25 inches/sec. paper speed.

According to (1) the amplitude of the search coil current depends on changes in the angle α between the magnetic axis of the magnetized ball and the axis of ball rotation. The induced voltage (or current) is at the maximum when the magnetic axis is perpendicular to the ball rotational axis; the current is at the minimum when the magnetic axis of the ball is parallel to the rotational axis of the ball. This was verified experimentally by SKF Industries, by static measurements of the magnetic flux density of a non-rotating bearing with one magnetized ball.

Enclosure 1 shows oscillograms taken from the Hirano and Tanoue paper (1). The frequiencies ω_s, ω_c and ω_b, and the times T_s, T_c and T_b are related to each other by:

$$\omega_s / \omega_c = \left| \omega_b / \omega_c - m \right| \qquad (1)$$

or

$$T_s = \left| \frac{2\pi}{\omega_b - \omega_c m} \right| = \left| \frac{1}{f_b - m f_c} \right| \qquad (2)$$

where
$\omega_s = 2\pi f_s = 2\pi / T_s$ = frequency of fluctuation with period Ts
$\omega_c = 2\pi f_c = 2\pi / T_c$ = cage rotational frequency
$\omega_b = 2\pi / f_b = 2\pi / T_b$ = ball rotational frequency with respect to the cage and m is the nearest integer to ω_b/ω_c. ω_s, ω_c and ω_b are expressed in radians/sec., f_s, f_c, and f_b in HZ. The times are expressed in seconds.

Equation 2 represents a beat phenomenon caused by the two frequencies f_b and mf_c

The amplitudes A_{max} and A_{min} in Enclosure 1, which define the beat period T_s, correspond approximately to the maximun and minimum of the angle α .

2. <u>Tests with Monsanto MCS-2931 Oil</u>.

The test bearing was a 459981 G-1 high speed split inner ring ball bearing operating under a 3280 lbs. thrust load. A sketch summarizing the tape recorded portions of the signal from the search coil is given in Enclosure 2. Visicorder traces of typical portions of the signals are given in Enclosures 3, 4 and 5. The paper speed used in these traces is 0.2 inches/sec. Enclosure 6 shows traces obtained with a paper speed of 25 inches/sec.

Enclosures 4 and 5 (at a paper speed of 0.2 inches/second) and 6 (at a paper speed of 25 inches/second) shows high amplitude excursions in the magnetic signal from the ball. Apparently, a very brief and violent change in magnetic axis location took place at that time when the excursions occur. We believe that these events are microsmearing occurrences which momentairly lock togetner the surfaces of the ball and ring at a point in the contact area where there was previous sliding. If this point is off-center in the contact ellipse, the locking forces a rapid change in ball axis direction. The change does not need to involve a large angular motion in order to show a high amplitude signal, since the induction coil is a velocity detector.

If our interpretation of these unusual kinematic signals is correct, they would indicate that micro-smearing occurrences have preceded the macro-smearing failure. The slow-speed modulations of the envelope of the signal can be read as an accumulation of repeated microsmearing events which force an "indexing" of the ball axis, by locking together the ball and ring surfaces in the same area of the contact every time the ball rolls over the smeared surface patch. The smearing occurrences are self-aggravating and at one point the temperature becomes uncontrolled and seizures take place. It is seen from Enclosure 2 that the first of the high amplitude excursions occurs at 45.5 minutes prior to failure. No excursions appear in the first three minutes of Trace 7 nor in the 5 minutes duration of Trace 5 of Enclosure 2. The amplitudes of some of the excursions are as large as twice the RMS value of the signal.

It is seen from Enclosures 2 to 5 that the signals in Trace 5, Trace 7 and the last minute of Trace 23 are clearly modulated. It is observed from Enclosure 5 that these modulations suddenly become much more pronounced only one minute before the smearing failure of the bearing. However, the modulations may also vary due to other reasons and are not necessarily indications of smearing failure. This may be the case with the modulation in Enclosure 3.

For further evaluation of the traces is Enclosures 3 to 6, the following frequencies are needed:

$f_c = \frac{f_r}{2}\left(1 - \frac{D}{d}\cos\alpha_c\right)$ = Frequency of the rotating ball set with respect to the outer ring (Cage frequency)

$f_b = \frac{f_r}{2}\frac{d}{D}\left(1 - \frac{D^2}{d^2}\cos^2\alpha_c\right)$ = Frequency of the polar rotation of the ball in a system attached to the cage.

f_r	=Rotational frequency of the inner ring.
D	=Ball diameter
d	=Bearing pitch diameter
α_c	=Contact angle

All frequencies are expressed in Hz (cycles/sec)

The beat period Ts in Enclosure 5 must satisfy Equation 2. In the beginning of the beat in Enclosure 5, Ts is approximately 1 second; at the end near the bearing failure Ts is approximately 2 seconds.

Computed values of fc and fb will not satisfy Equation 2 for Ts=1 or 2 seconds. In order to satisfy this equation it must be assumed that slip occurs at the ball to race contacts so that $f_b < f_b^*$. It is also assumed for simplicity that fc=fc*. The quantities with an asterisk (*) refer to computed values (without slip). The quantities without asterisk refer to values with slip, which satisfy Equation 2. Values of fb can be obtained from the traces of Enclosures 6. The spacing between two adjacent peaks in Enclosure 6 is $^1/fb$ seconds. The spacing $^1/fb$ was found to vary by as much as 15%, even on the same trace. Since the rotational speed fr varies considerably and since the traces do not contain a signal showing inner ring revolutions, it cannot be determined to what extent the frequency fb is influenced by slip and by speed variations of the inner ring.

In Table 1 below, fr and fc are assumed to have their computed theoretical values, while fb is computed from Equation 2 for known values of Ts, fc and m.

Table 1

ENCLOSURE	RPM	fr	fc	Ts	m	fb*	fb	S
5 (beginning)	18000	300	131.8	1	8	1141	1055	0.0754
5 (end)	18000	300	131.8	2	8	1141	1054.5	0.0758
4 (end)	18000	300	131.8	3	8	1141	1054.3	0.0760

S is the slip, computed from the Equation

$$S = 1 - \frac{fb}{fb^*}$$

It is seen that Ts in the range 1 to 3 seconds has only a small effect on S.

3. Tests with Conoco DN600 oil.

Only one trace at 0.2 inches/second paper speed (Enclosure 7) and one trace at 25 inches/second paper speed (Enclosure 8) are available.

The rotational speed of the bearing was 16000 RPM. It is seen that the trace in Enclosure 7 is more jagged than corresponding traces for the Monsanto oil, but no high amplitude excursions similar to those in the Monsanto test are seen. Enclosure 8 shows a few peaks, but they are not nearly as high as the peaks in Enclosure 6.

4. Tests with Aeroshell Turbo Oil 555

The last 11.25 minutes of the induced voltage were tape recorded. A typical trace is shown in Enclosure 9. The speed was 21400 RPM. No high amplitude excursions were observed.

REFERENCE
(1) F. Hirano and H. Tanoue: "Motion of a Ball in Ball Bearing". Wear, 4 (1961) 177-197, Printed in the Netherlands, Elsevier Publishing Company, Amsterdam.

Olof Gustafsson

O. Gustafsson

OG/rdb

ENCLOSURE IV-1

SEARCH COIL OSCILLOGRAMS

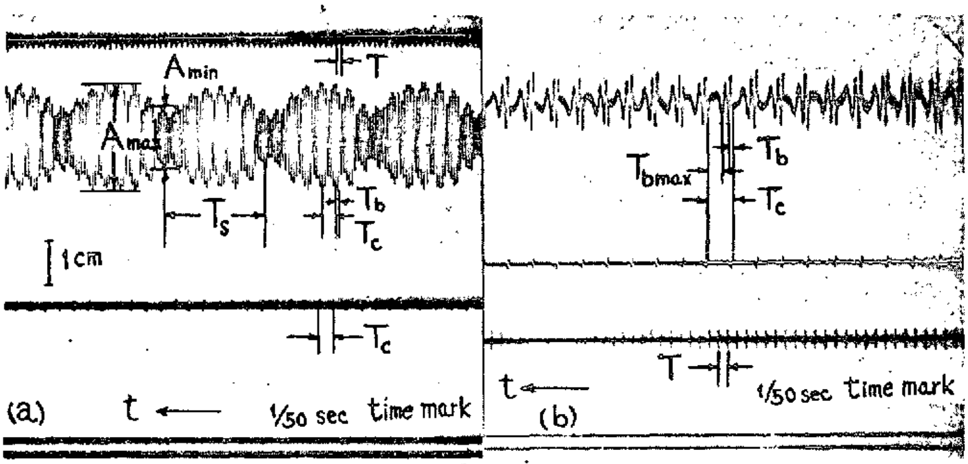
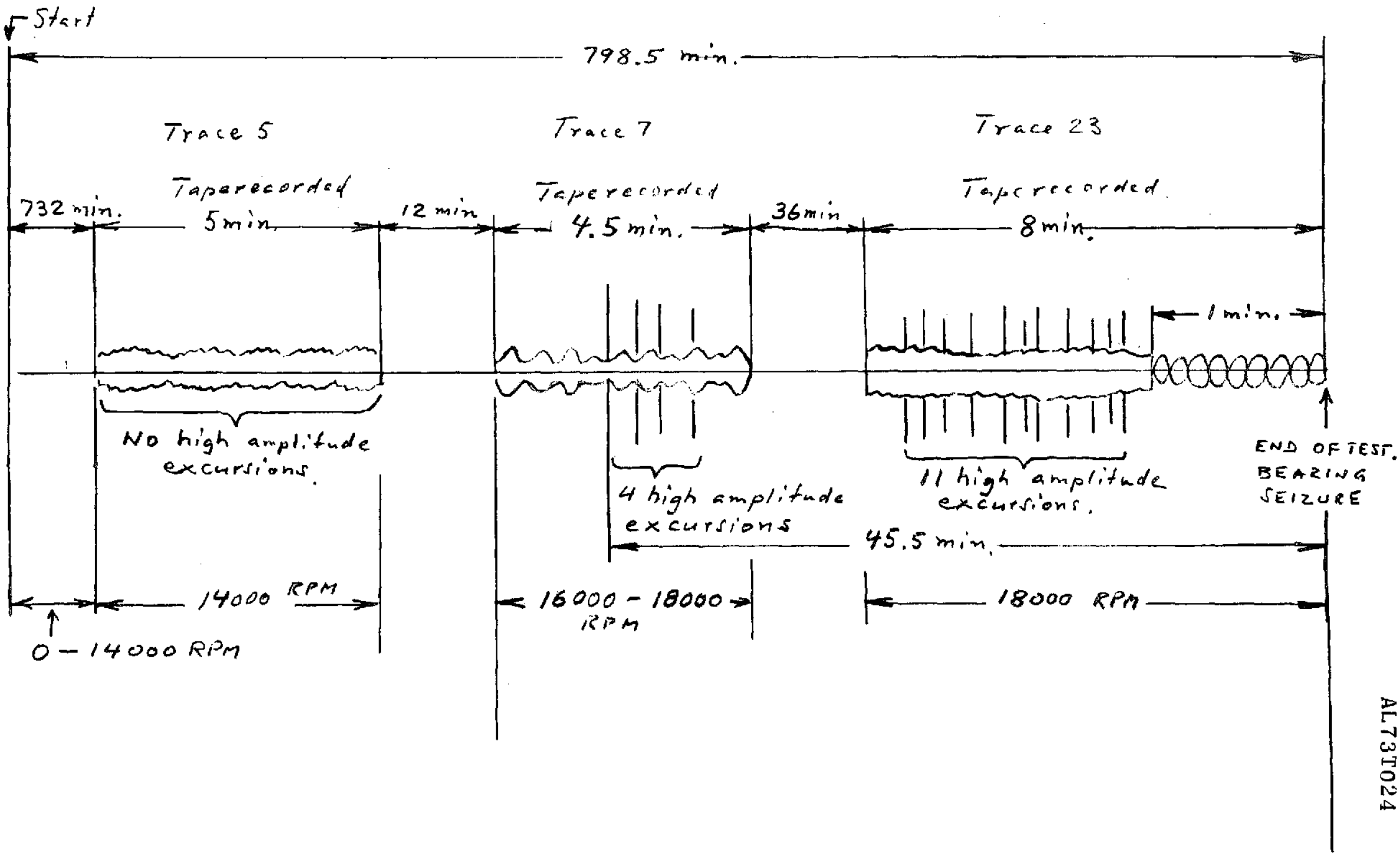


Fig. 6. Explanation of oscillogram. (a) No. 6307, $P = 100$ kg, $N = 1290$ rev/min. (b) No. 7307 $P = 100$ kg, $N = 750$ rev/min. T: Period of revolution of inner ring or shaft; T_b: Period of rolling ball, T_{bmax}: Maximum of T_b; T_c: Period of revolution of cage or ball centre; T_s: Period of fluctuation of amplitude.

ENCLOSURE IV-2

SUMMARY OF SEARCH COIL TESTS

MONSANTO MCS-2931 OIL



ENCLOSURE IV-3

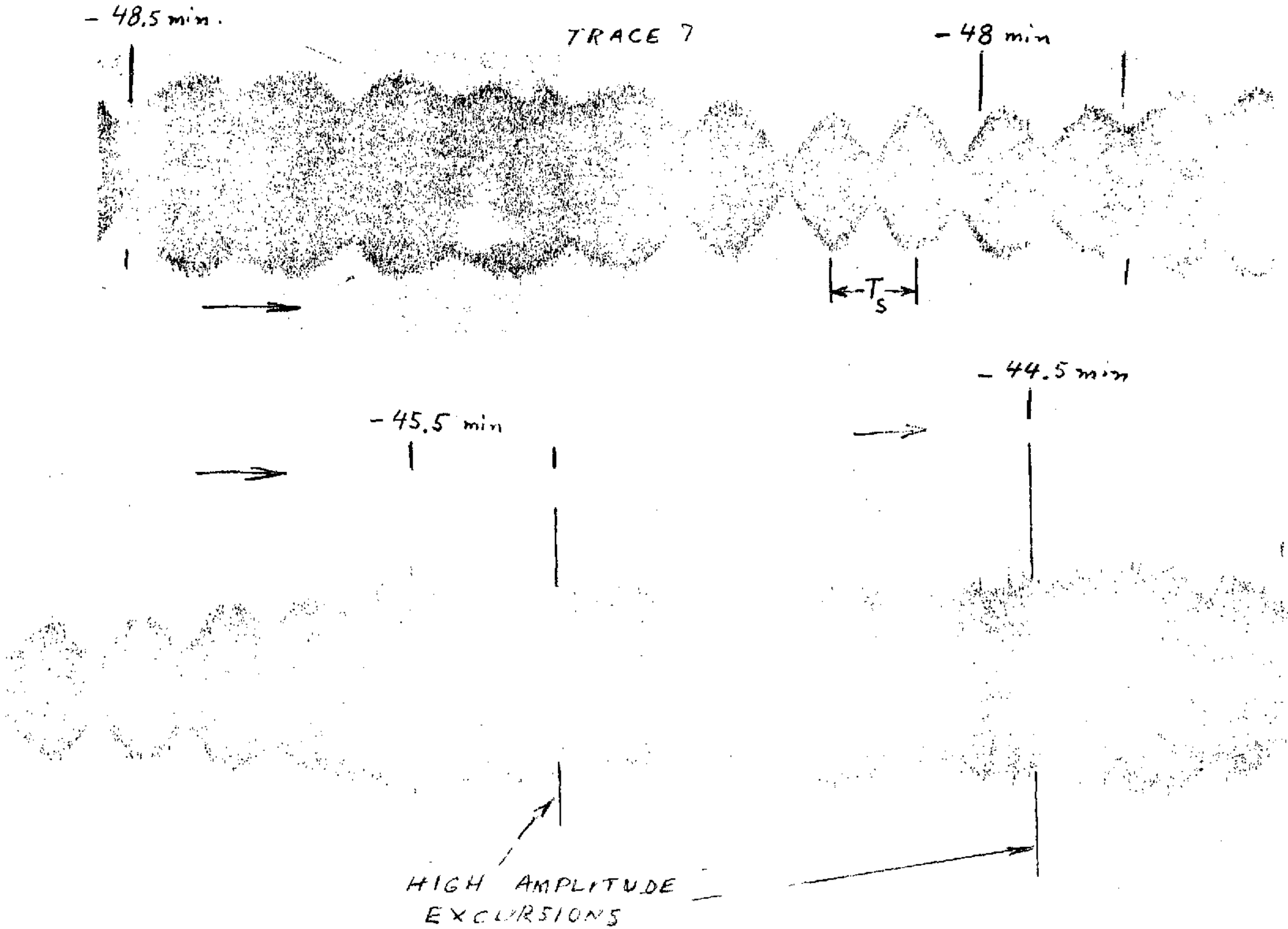
VISICORDER TRACES, 0.2 in/sec.

MONSANTO MCS-2931 OIL
16000 RPM
Trace 5

AL73T024

ENCLOSURE IV-4

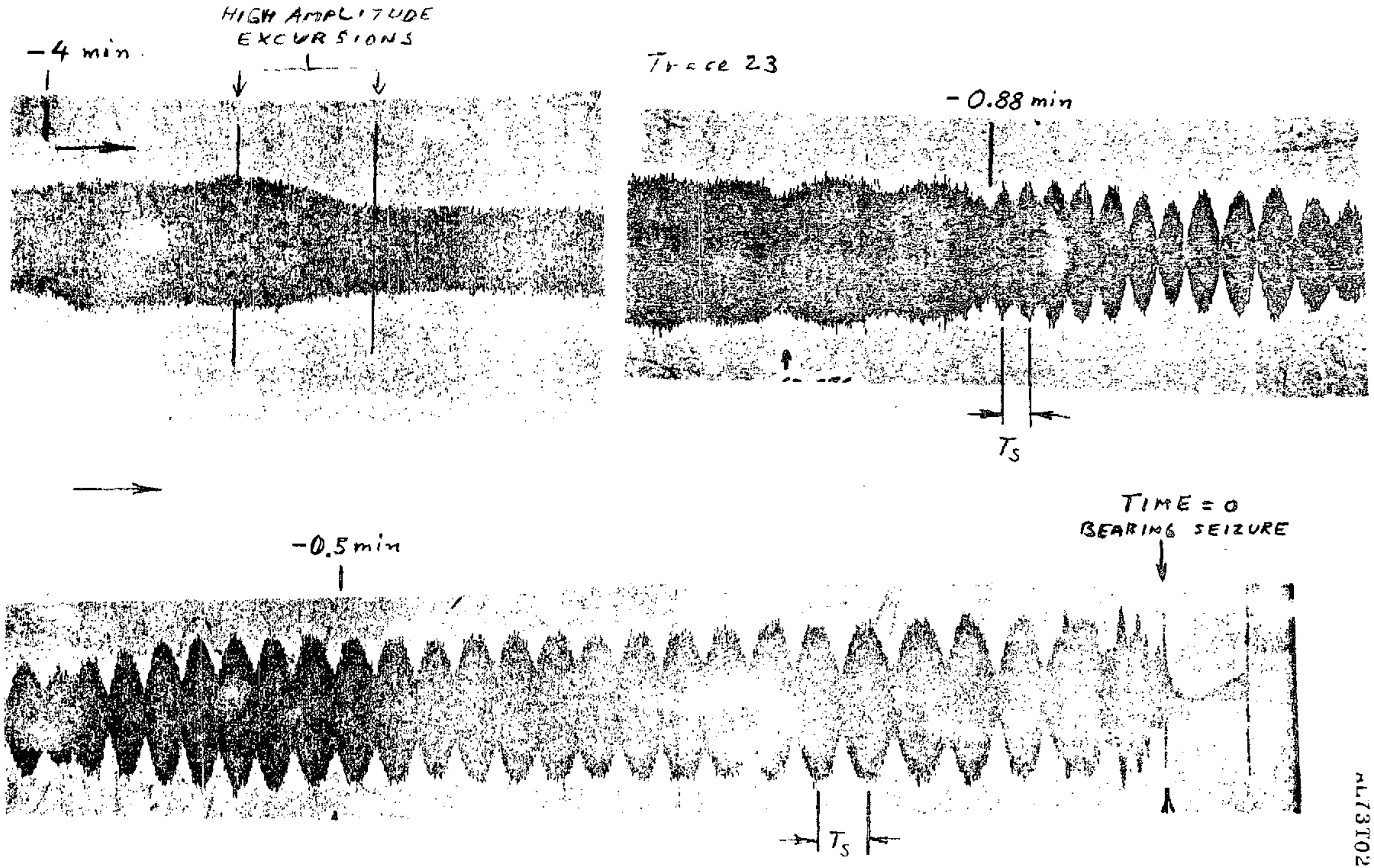
VISICORDER TRACES, 0.2 in/sec
MONSANTO MCS - 2931 OIL
16000 - 18000 RPM



AL73T024

ENCLOSURE IV-5

VISICORDER TRACES, 0.2 in/sec.
MONSANTO MCS-2931 OIL
18300 RPM



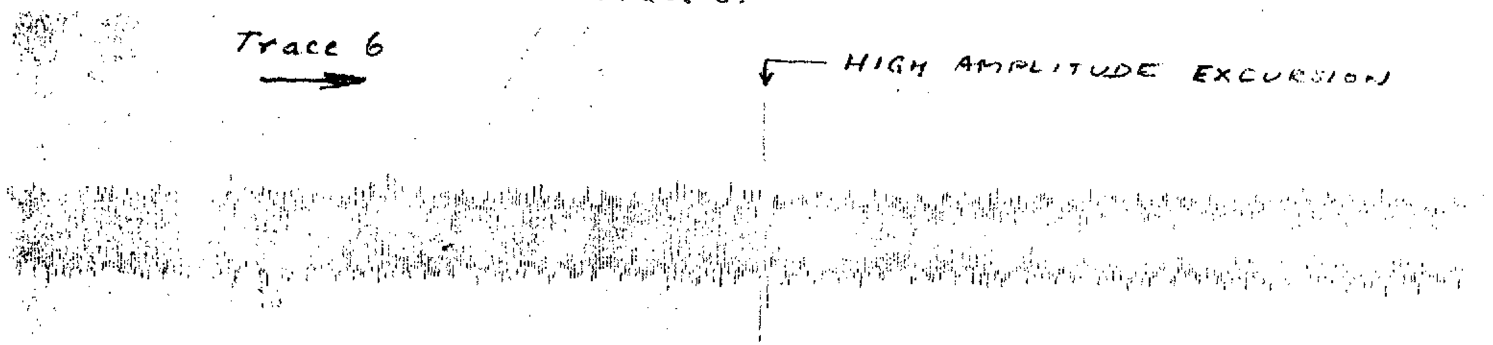
AL73T024

ENCLOSURE IV-6

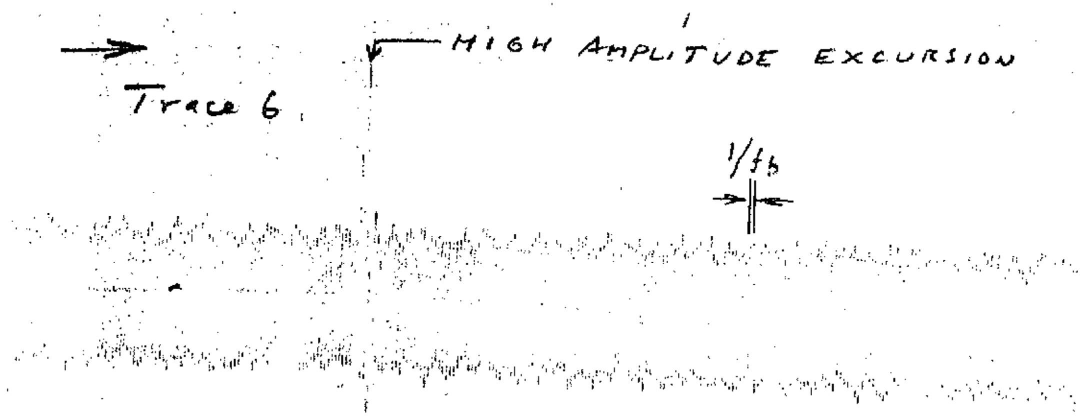
VISICORDER TRACES, 25 inches/sec.

MONSANTO MCS-2931 OIL

Trace 6.



APPROXIMATELY -2.75 MINUTES FROM FAILURE



APPROXIMATELY -3.85 minutes from failure

AL73T024

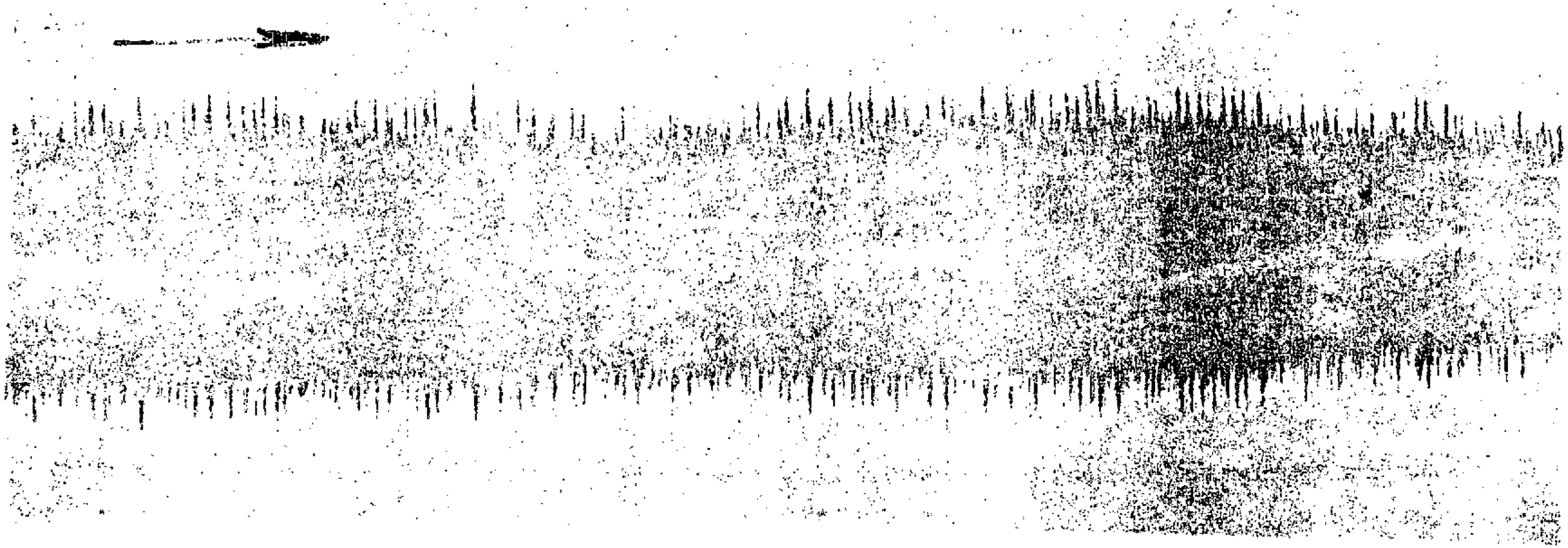
ENCLOSURE IV-7

VISICORDER TRACE, 0.2 inches/sec.

CONOCO DN 600 OIL

16000 RPM

TRACE 1.



AL73T024

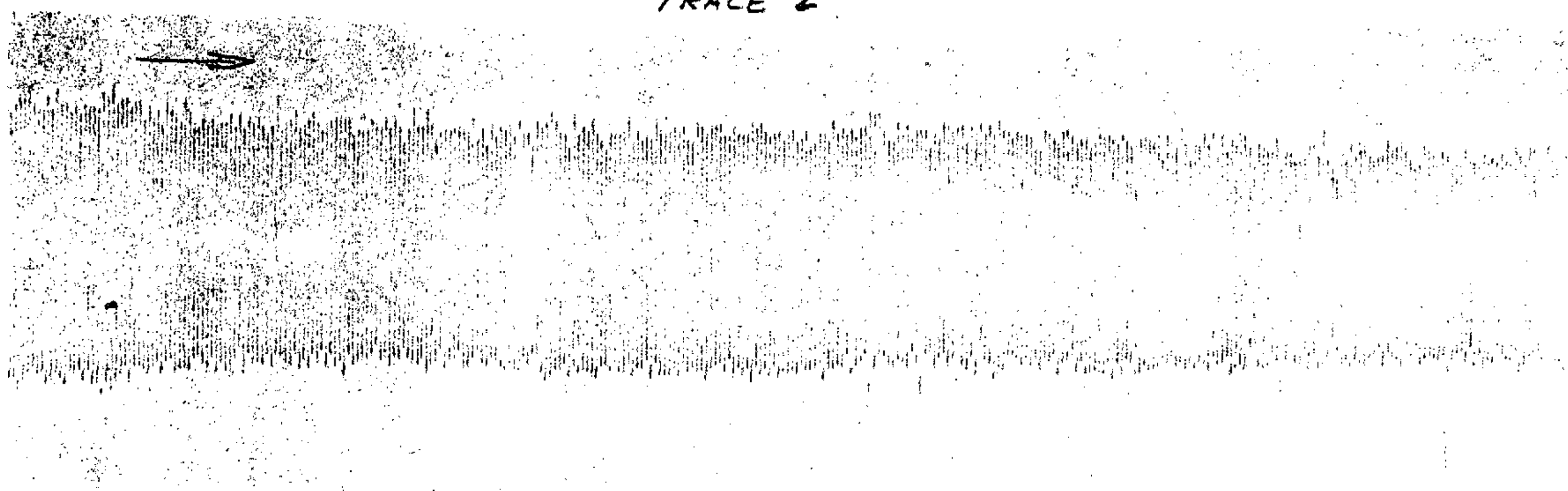
ENCLOSURE IV-8

VISICORDER TRACE, 25 inches/sec.

CONOCO DN 600 OIL

16000 RPM

TRACE 2



AL73T024

ENCLOSURE IV-9

VISICORDER TRACE, 0.2 inches/second

AEROSHELL TURBO OIL 555

21400 RPM

Trace 2

AL73T024

APPENDIX V

VIBRATION STUDY OF 125 MM BORE RIG

APPENDIX V

VIBRATION STUDY OF 125 MM BORE RIG

In order to establish the vibration mode and rpm level of the critical speeds of the test rig used for this program, a critical speed study was conducted by the NASA Lewis Research Center using a NASA developed shaft dynamics computer program.

Three separate shaft mounting situations were studied, one assuming "rigid" mounting (infinite bearing stiffness), a second assuming a "soft" mounting (low bearing stiffness) and a third assuming a "free" mounting (an unsupported shaft rotating in space).

The bearing stiffness values used in the second or "soft" mounting situation (1,192,341 lbs./in. and 3,384,000 lbs./in. for the test and rig bearings respectively) were calculated using the equations in Palmgren. The radial deflection was calculated assuming the maximum load occurs on the most heavily loaded ball. Radial load was taken as the weight of the shaft and the related components mounted on the shaft. Shaft imbalance and the loads developed as a result of shaft deflection were not taken into account. Shaft dynamics were studied for the speed range from 0 to 40,000 rpm.

The results of the computer study are shown plotted in Enclosures V-1 and V-2 which, respectively, show the rigid and soft mounting situations. The plots show shaft deflection vs. location along the shaft while operating in the critical speed vibration modes indicated. The "extra mass" notations along the bottom of these charts indicate the added mass of the components mounted on the shaft (i.e., main shaft end clamp, seal mating rings, spacers etc.)

As shown by Enclosure V-2, a "whip" mode critical speed was calculated to exist for the soft bearing mount situation at 24,897 rpm. This "whip" load critical speed could be lowered if the mounting stiffness is accounted for. Since three seals (two Koppers designs and the NASA design) have failed at about 20,000 rpm (operating during this contract and the previous contract), this critical speed problem might be the cause of the seal failures. The deflection of the shaft during this "whipping" motion might have caused the seal to rub which led to the seal failure. As might be expected no whip mode critical speed was found to exist through 40,000 rpm for the rigid bearing bearing mount situation indicating that the existence of this vibration mode is highly dependent on bearing stiffness.

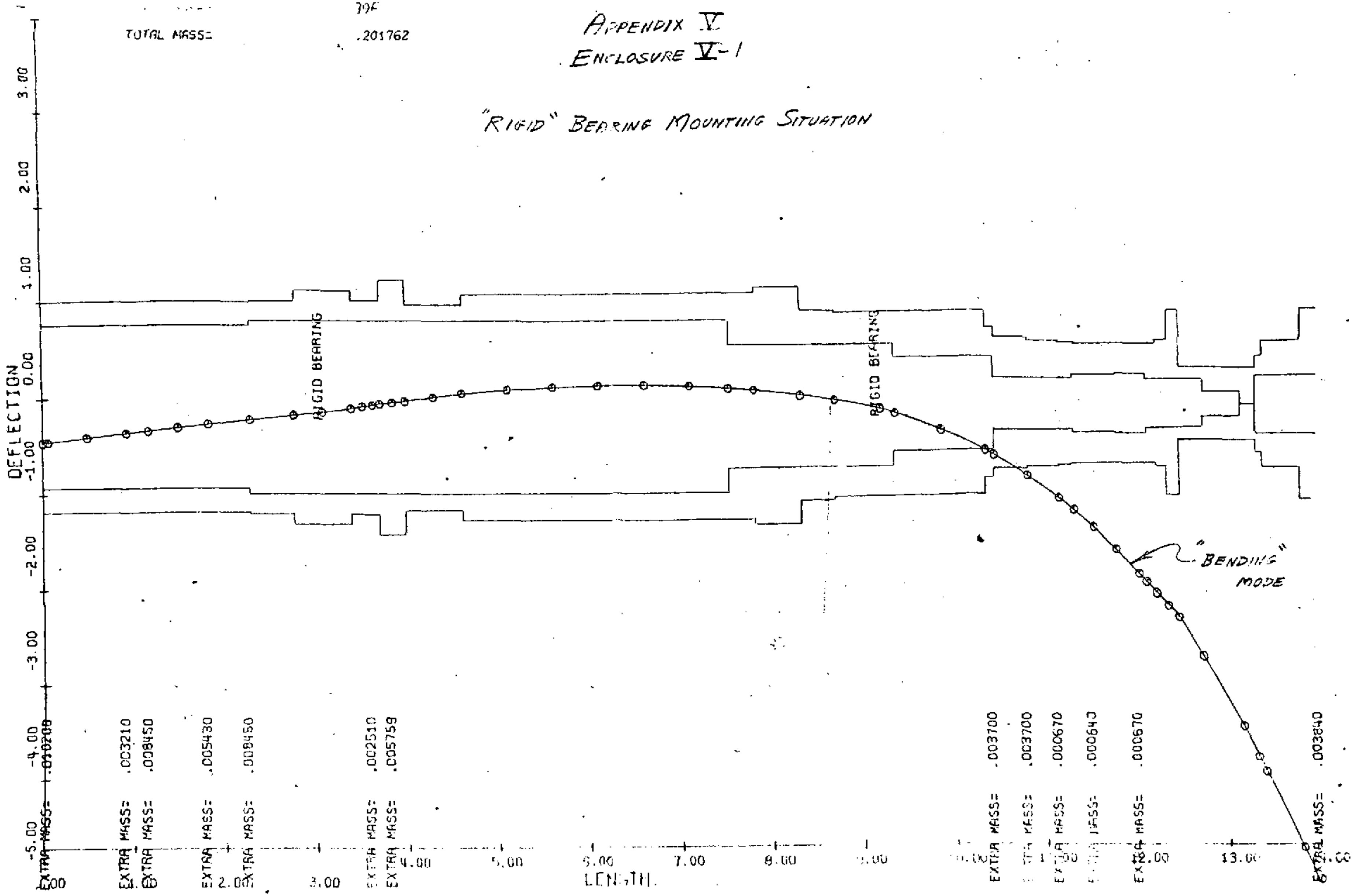
The rigid and soft mount situations show "bending" mode critical speeds to exist at 39,589 rpm. (Enclosure V-1) and and 35,970 rpm (Enclosure V-2) respectively. This indicates a much lesser influence of bearing stiffness for this mode.

The free mounting situation produced no critical speeds through 40,000 rpm.

CRITICAL SPEED 39589 RPM

APPENDIX V
ENCLOSURE V-1

"RIGID" BEARING MOUNTING SITUATION

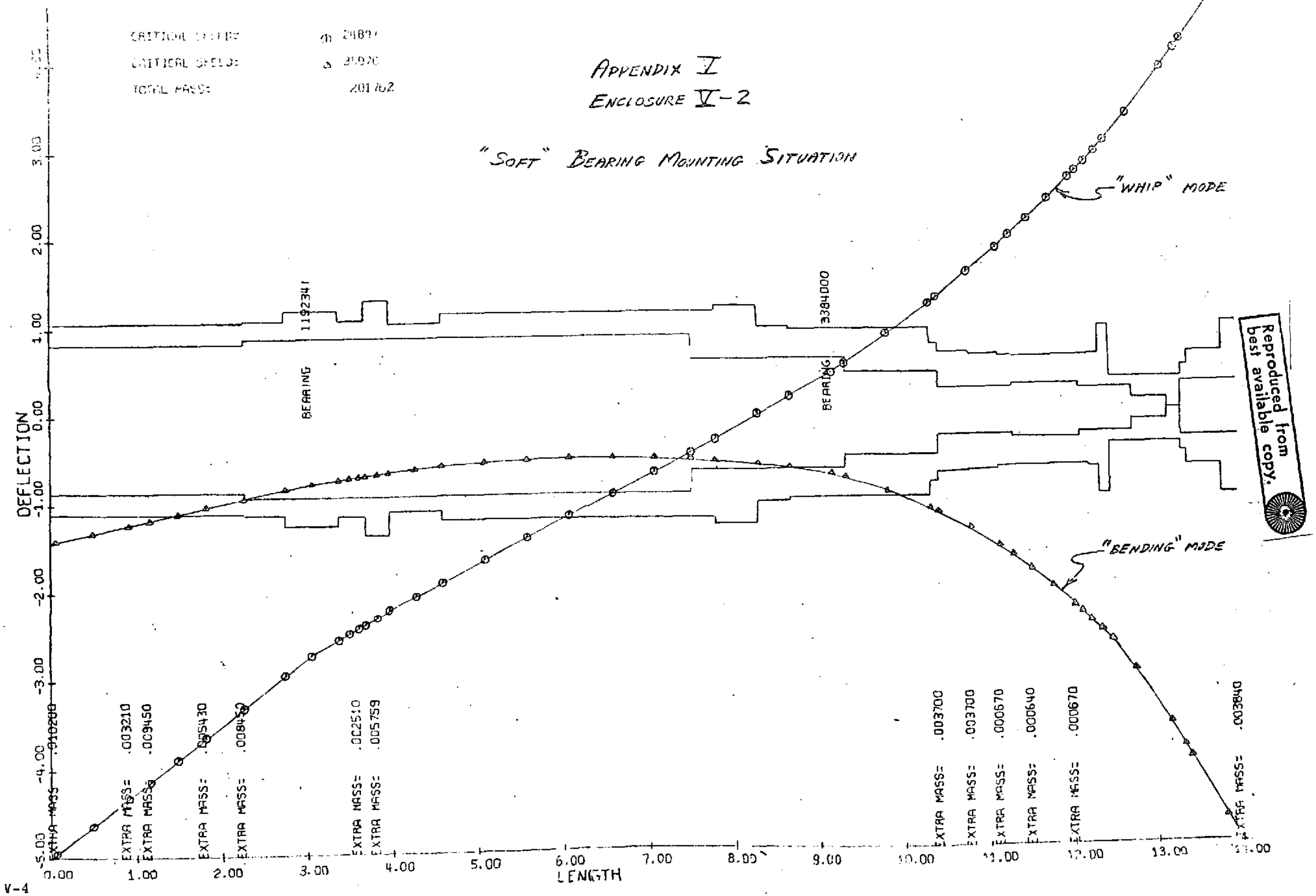


RESEARCH LABORATORY SKF INDUSTRIES, INC.

APPENDIX I

ENCLOSURE V-2

"SOFT" BEARING MOUNTING SITUATION



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2. Rhoads, W. L., and Sibley, L. B.; "Supersonic Transport Lubrication System Investigation", Final Summary Report on Phase I, NASA Contract NAS3-6267, NASA CR-54662, SKF AL67T060 (1967).

3. Rhoads, W. L., and Peacock, L. A.; "Advanced Turbine Engine Mainshaft Lubrication System Investigation", Final Summary Report on Phase I, Part 1 and Part 2, NASA Contract NAS3-6267, NASA CR-72854 and NASA CR-72873, SKF AL69T016, January, 1971.

4. Hirano, F and H. Tanoue; "Motion of a Ball in a Ball Bearing", Wear, 4, p. 177-197, (1961).

5. F. Hirano; "Motion of a Ball in Angular Contact Ball Bearing", ASLE Transactions 8, p. 425 - 434, (1965).

6. Povinelli, V. P. and A. H. McKibbin; "Development of Mainshaft Seals for Advanced Air Breathing Propulsion Systems, Phase II, Final Report", NASA CR 72737, PWA-3933 (1970).

7. Povinelli, V. P., and A. H. McKibbin; "Development of Mainshaft Seals for Advanced Air Breathing Propulsion Systems, Phase III, Final Report", NASA CR72987, PWA-4263 (1971).