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SUPERSONIC TRANSPORT LUBRICATION SYSTEM INVESTIGATION

by

C. G. Hingley and L. B. Sibley

prepared for

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

CONTRACT NAS3-6267

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

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SECOND SEMIANNUAL REPORT

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TABLE OF CONTENTS (Cont.)

LIST OF ENCLOSURES

LIST OF ENCLOSURES (Cont.)

LIST OF ENCLOSURES (Cont.)

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ABSTRACT

Ball bearings and bellows face seals for use on Mach 3 aircraft gas turbine engine mainshafts, and suitable test rigs to simulate engine operating conditions, have been manufactured and check-out testing conducted on all hardware and test systems. A development program is underway to determine the most suitable combination of presently available materials and lubricants to permit operation of these bearings and seals under typical engine load and speed conditions at the highest possible ambient temperature above 600°F with the seals exposed to 1200°F air and a pressure differential of 100 psi. Tests are conducted primarily using a nitrogen-blanketed bearing chamber and lubricant system.

Test lubricants are being screened for their potential performance under two different application techniques, namely, jet lubrication of the bearings with circulating oil, and once-through lubrication in the form of oil mist.

SUPERSONIC TRANSPORT LUBRICATION SYSTEM INVESTIGATION

by C. G. Hingley and L. B. Sibley

SKF Industries, Inc.

INTRODUCTION

This is the second semiannual report under NASA Contract NAS3-6267 and covers the work done from May 3 through October 31, 1965, on a 24-month research program.

The performance of aircraft gas turbine mainshaft ball bearings, seals, and lubricants under simulated supersonic transport engine conditions (Mach 3) is being studied using the most advanced materials, designs, and manufacturing techniques available. Both an oil circulating system and a once-through oil-mist system are under investigation, each with inert gas blanketing for high-temperature lubrication. Five candidate lubricants for each of the two systems have been selected for screening evaluation, and 1000-hour endurance tests will be conducted with one or two of the best oils in each system.

SUMMARY

Two test machines and related systems with recirculating oil and once-through lubrication respectively, for the evaluation of 125-mm bore ball bearings, two-stage bellows-type face seals (developed under subcontract by Koppers Co., Inc.) and candidate lubricants, have been manufactured and installed in test cells. Check-out testing without external heat added under speeds and loads up to essentially full design load and speed has been conducted successfully on both rigs.

All systems and components have been designed to operate at 14,000 rpm shaft speed, 3280 lbs. thrust load on the test bearing (AFBMA computed L_{10} life $\doteq$ 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. High temperature check-out testing has started in the recirculating oil rig using MIL-L-7808E lubricant (Esso Turbo Oil 4040), which was found in one hour tests to provide adequate lubrication of the bearings and seals in the recirculating rig at bearing temperatures of 500°F, using nitrogen blanketing.

CONCLUSIONS

1. According to the evidence from the check-out tests conducted, the test system for recirculating lubrication, together with its fluid and control systems, is operational.

2. A similar test system, but utilizing a once-through mist lubrication technique, is operational according to check-out tests.

3. Tests have shown that oil-mist lubrication is feasible for at least one hour's operation of an unheated gas turbine ball thrust bearing running at 14,000 rpm with 3,280 lbs. thrust load using a MIL-L-7808E fluid.

4. The experimental bellows face seals function adequately under unheated conditions when operated at 14,000 rpm under a differential pressure of 100 psi.

5. Rig check-out testing has uncovered some problems in extended operation of the various rig systems and bearings under the test conditions. However, the present designs of all systems and bearings are basically workable, and they will be used to start screening tests of the selected lubricants.

DETAILS OF PROGRESS

Background

In gas turbine engines designed for use in advanced generations of supersonic transport aircraft, the mainshaft thrust bearings are required to rotate at higher speed and to operate efficiently and reliably at higher temperatures than any current jet-engine bearings. Overall engine heat transfer and efficiency computations indicate that these thrust bearings must be capable of operating at 600°F and above, and as a consequence, the fluids used to lubricate them and the seals used to contain the lubricant in the bearing chamber, must be satisfactory at these temperatures. To minimize lubricant degradation at these high temperatures, nitrogen blanketing may be employed to reduce oxygen to a very low level in the bearing and lubricant system.

Two basic methods of lubrication present themselves. These are, firstly, the conventional recirculating oil-jet lubrication of the bearing, and secondly, mist lubrication using either nitrogen or air as a carrier for the oil, allowing a large portion of the heat generated by the bearing to be removed by gas cooling. In the second system, the oil is not recovered and thus the system is of the once-through type, dispensing with the need for oil coolers and scavenge pumps and also relaxing the thermal stability requirements necessary for recirculating oils.

The current state of development of bearings, seals and lubricants is such that operation, under the conditions specified for advanced supersonic transport engines, is definitely possible although extended operation of the proposed lubrication systems has not yet been reliably accomplished.

Research Objectives

It is the purpose of this program of research to investigate the limits of feasibility of using the best currently available bearings, seals, and lubricants in high-temperature lubrication systems under conditions simulating those expected in the main propulsion power units of an advanced Mach 3 supersonic transport aircraft. It is expected that this research will result in:

a) Data pertaining to maximum temperature capability of several of the most promising available lubricants in both circulating and "once-through" nitrogen-blanketed lubrication systems.

b) Operating experience with a lubricant having a Freon additive under supersonic transport engine conditions in both recirculating jet lubrication and oil-mist lubrication systems.

c) Comparison between an inerted and an open recirculating oil system under supersonic transport conditions using the most promising available lubricant.

d) Data on endurance of inerted recirculating and oil-mist bearing and seal systems under simulated supersonic transport engine conditions up to a maximum of 1000 hours.

Plan of Research

The research program is divided into six separate Tasks described in Exhibit A of the contract, which is given as Appendix I.

The phasing of the effort is shown in the PERT network given in Enclosure 1, which is current as of October 31, 1965. The project effort to date has been expended in Tasks I, II, IV, and VI concerned with construction of facilities and test specimens. During the six months covered by this report the major portion of the effort has been devoted to the installation and check-out testing of the two test rigs and the associated control equipment, the final selection of test fluids and the manufacture and quality control of bearings and seals.

Tasks II through V to be undertaken hereafter cover the screening of candidate lubricants in both the recirculating and once-through rigs, followed by two 1000-hour endurance tests with a selected lubricant in each rig, each 1000-hour test to be accumulated from short runs of about 10-hours duration each, with interposed cool-down cycles.

Base-line screening tests will also be run, using air instead of a nitrogen blanket to simulate the conditions of a conventional engine bearing lubrication system. The lubricant will be the one allowing the maximum bearing operating temperature in the inerted screening tests, provided that its properties permit operation in air. The operating conditions for these unblanketed tests will be chosen so as to avoid explosion hazards.

In addition to the base-line tests, one selected oil in each system will have a "Freon" additive added to it and will then be subjected to screening tests using air instead of nitrogen as an atmosphere.

The following lubricants, five in number for each lubrication technique, have been selected:

Recirculating Oils

a) Esso Turbo Oil 4040 (base-line fluid meeting MIL-L-7808E)
b) Sinclair Turbo S Oil, Type 1048, improved (ester-base)
c) Socony Mobil XRM 177F (hydrocarbon)
d) Monsanto MCS-293 (modified polyphenyl ether)
e) DuPont PR-143 (fluorocarbon)

Oil Mist Lubricants

a) Esso Turbo Oil 4040 (base-line fluid meeting MIL-L-7808E)
b) Union Carbide UCON 50-HB-5100 (polyalkylene glycol)
c) Sun Oil Sunthetic 18H (polyolefin)
d) Socony Mobil XRM 177F (hydrocarbon)
e) Hercules Powder Hercolube F (polyester)

Brief Description of Test Equipment and Conditions

The test apparatus is capable of operating an aircraft mainshaft ball bearing and face seal assembly with the candidate lubricants under the following initial conditions:

Oil-inlet temperature, 500°F* ± 10°F

Bearing outer-ring temperature, 600°F

Bearing inner-ring temperature, 610°F to 650°F

Bearing thrust load, 3280 lbs.

Air temperature at the outboard seal, 1200°F

Pressure drop across the test seal assembly, 100 psi

Shaft speed, 14,000 rpm

If satisfactory performance is obtained under the above conditions with any given lubricant, the bearing temperatures will be increased in 100°F increments to establish the maximum temperature under which the bearing-seal-lubricant combination will operate adequately.

These conditions are expected to occur in the gas turbine powerplants of advanced supersonic transport aircraft. The general plan of the test apparatus designed to provide these simulated engine conditions is shown schematically in Enclosure 2. The bearing-seal combination under test is mounted in a test rig which is supplied by several fluid systems to give the required simulation of environmental conditions.

Referring to Enclosure 2, the principal systems are:

a) A high-speed drive.

b) Lubrication systems for simulation of either a recirculating oil arrangement or a "once-through" oil-mist system.

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*This temperature is specified for the recirculating oil rig. For the oil-mist rig, the lubricant shall be supplied from a pressurized reservoir at a temperature of 200°F, minimum.

c) Nitrogen blanketing.

d) A nitrogen-tracer system for detecting the amount of leakage in each seal.

e) A hot-air supply to simulate the gases to be sealed out of the test bearing chamber.

f) Instrumentation for monitoring of test parameters (not shown in Enclosure 2).

More detailed discussions of the test apparatus and facilities provided are presented in the First Semiannual Report (1)*.

As shown in Enclosure 3, each rig, together with its associated equipment, is installed in a 20' by 15' test cell. An air compressor and its filtration system and reservoir, and also a bank of helium bottles together with the tracer mixing system are installed in a separate service cell. Each test cell is equipped with heavy-duty ventilation systems to minimize the risk of flammable vapors collecting and also to keep the cell air temperature as low as possible during rig operation.

The instrumentation and control panels for each rig are situated in the control corridor grouped about a reinforced glass window. Further control apparatus for the motor and air heater are situated against the other corridor wall, as is a mass spectrometer for analysis of helium tracer and oil degradation products.

Installation and Check-Out of Test Equipment

The equipment described above has been installed and checked out for satisfactory function with a few exceptions, which will now be discussed individually.

Test Rigs

Up-to-date assembly drawings of the test rigs are shown in Enclosures 4 and 5, which illustrate the two modes of operating used, to be described later. Photographs of the recirculating rig are shown in Enclosures 6 and 7.

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

Apart from the differences in the design of the lubricant nozzle rings, the two test rigs are identical, and the components interchangeable. During the development or debugging period modifications found necessary on the one were automatically carried out on the other as soon as the modification was proven effective. Only those modifications requiring significant rework of test rig components will be described.

Early attempts to run the test rigs, with provisional test bearings having ground and polished M-50 rings, 52100 balls and silver plated silicon-iron-bronze cages, revealed that adequate cooling of the load bearing inner rings could not be achieved. Increasing the lubricant supply rate tended to aggravate the situation and a number of bearing seizures, due to the thermal take-up of all internal clearance, were experienced. An under-race cooling system for the load bearing inner rings was designed and constructed. Due to the comparatively small amount of axial space, the design had to be compacted as much as possible, the final form being shown in Enclosure 4. While this expedient eliminated the thermal bearing seizures satisfactorily, it was found that the amount of lubricant that could be supplied to the load bearing unit was limited, due to the tendency for spillage to occur from the foreshortened collector cup. As a consequence the load bearing as a whole has been functioning at temperatures in the region of 300°F, which is approaching the limiting temperature of the unblanketed mineral oil supply. For this reason, if the load bearing is required in future testing (to be discussed later in this report), MIL-L-7808 oil having higher temperature capability than mineral oil will be used. However, it is now thought that no load bearing will be used at all, as described below.

The runner for the rig seal, located between a shoulder on the shaft and an annular lip on the load bearing mount, showed evidence of failure to rotate with the shaft, and consequently a certain amount of heating and surface damage occurred in the early running. This problem was eliminated by fitting of a suitable O-ring between the outer lip and the retainer ring on the load bearing mount. A special clamping sleeve and positive keying of the runner will be provided for high temperature running. During further runs, surface discoloration on the runner, together with a hard black "coke" from thermal degradation of the mineral oil then being used, indicated the existence of temperatures in the region of 600°F or more. Consultation with Koppers Company, Inc., the rig seal manufacturers, revealed that the high frictional heating at the seal-to-runner contact was probably due to the lack of a suitable pressure gradient across the seal dam. Satisfactory seal operating has been achieved in subsequent testing with a small positive pressure in the test bearing chamber.

During a number of the earlier rig runs, vibration of the rig seal was experienced, which gave rise under some conditions to a piercing squeal. A layer of damping material, held over the outside diameter of the seal bellows by a garter spring, eliminated the problem. Recently however, the heat generated by the rig seal and roller bearing has caused a deterioration in the condition of the damping material and so an alternative material will be provided. This will consist of an asbestos strap surrounding the outer surface of the bellows assembly, retained by the garter spring.

A mechanical damping arrangement, consisting of a close fitting sleeve over the seal bellows is being designed by Koppers Company, as a further back-up if oil degradation of the asbestos is found to be a problem.

Experiments with the screw pump-labyrinth seal designed to retain the air in the hot-air system at pressures up to 160 psia, showed that the differential expansions of the rotating and static components led to a loss of the very small allowable gap (.0005") and severe wiping of the silver surface of the static member, followed by seizure. The screw labyrinth component was replaced by one with annular grooves, mated against a leaded stator surface, used in cold running to test this labyrinth seal design. The differential expansion then resulted in grooves being formed in the lead at the points of contact, and satisfactory operating was achieved at air temperatures of up to 400°F, when the melting point of the lead was approached. Stator rings having a suitably increased thickness of silver layer are being procured for operating of this labyrinth seal at the design temperature.

The problem of the leakage of hot air through the labyrinth seal has now been solved by removing the internal shaft heaters and closing up the breathing holes in the heater mounting plate (Item 17 of Enclosure 4). This results in a build up of air pressure inside the hollow shaft and causes a net thrust on the shaft in a direction opposite to that imposed on the test bearing by the original loading system. Calculations, detailed in Appendix II, reveal that the 3280 lbs. thrust carried by the test bearing is exactly counterbalanced by the air pressure reaction thrust at a pressure that lies conveniently between the maximum and minimum pressures allowable in the hot-air chamber. This extra thrust would require approximately 6500 lbs. thrust load to be transmitted through the load bearing, if the original system were to be retained. This figure is clearly too great for the bearing under the speed conditions and so the obvious alternative was chosen; namely, to eliminate the load bearing and load cylinders of the original loading system, as shown in Enclosure 5. The selection of a particular hot air manifold pressure in the region of 106 psi, permits the test requirements of 100 psi differential pressure across the seal and a small positive pressure in the bearing chamber, to be satisfied, while the test bearing still carries the required thrust, but in a reverse direction to that originally applied by the load bearing system. The new system is calibrated, as the original system was, by the use of a strain gaged housing mount for the test bearing.

Approval has now been obtained from the NASA Project Manager to use this new pneumatic loading system. The entire original loading system, comprising the load bearing and housing, the three pneumatic cylinders and their load cells, have therefore been dispensed with. Removed also are the problems of load bearing lubrication and cooling and also the problem of the overheating rig seal runner, for there is immediately room available for ample air cooling. The change in the thrust direction alters the stresses in the test rig components and the bolts at the hot air end of the rig. A review of the stress calculations revealed that no parts would be overstressed, provided the heater mounting plate (part 17) was thickened for high-temperature operation, and the number of stainless steel mounting bolts increased to 54. Preliminary tests carried out, with no heat added, with the test rig modified as indicated on the previous page, (Enclosure 5) showed that the method was satisfactory, and that the existing control equipment could maintain sufficiently accurate pressures over a range of flow conditions to permit maintenance of a steady thrust load on the test bearing.

An additional pneumatic circuit has been conceived, shown in Enclosure 8, which either admits cold high-pressure air or permits the leakage of exhaust hot air from a tube, similar in length to that shown in the original design carrying the shaft heaters. By the use of this circuit, some degree of either heating or cooling of the test shaft may be achieved in the region of the test bearing inner rings, in a manner which is easier with the new loading design than by the use of the banks of electrical heaters inside the shaft, shown in Enclosure 4, from which electrical leads through a hot pressure wall would be needed.

Drive Systems

Each test rig is driven by a variable speed DC motor via a jackshaft assembly. A 75 HP motor powered by a silicon controlled rectifier arrangement, is fitted for the recirculating oil rig, and a conventional 50 HP motor supplied by a motor-generator system, drives the oil-mist rig.

Upon the installation and initial operation of the 75 HP motor, the control system hunted severely over wide ranges of speed, manifesting itself as violent fluctuations in the armature current consumption and in the torque output. Adjustments made to the controller by the manufacturers, Louis-Allis Company, Inc. have succeeded in minimizing the effect in the low and high speed ranges, but have not been able to eliminate it over the entire speed and power range of the motor. Provided care is used in accelerating through the critical region, no problems have so far arisen due to the phenomenon.

Trouble has been repeatedly experienced with the step-up flat-belt drive between the motor and jackshaft, due to belt slippage, the belt coming off the small pulley and the belt shredding. The difficulty has been minimized by increasing the crown slightly, and removing the guide flanges to give an increased effective face width to the small jackshaft pulley. New pulleys, as wide as can be placed between the jackshaft bearings are being obtained to eliminate the present tendency of the belt to leave the pulley at high speeds and torques.

Inter-seal Tracered Nitrogen

The apparatus designed to mix a given percentage of helium tracer with nitrogen (for detection of leakage through each test seal separately) over a range of flows has been installed and checked for basic operation. No gas mixing has been attempted due to the inoperability of the mass spectrometer unit (to be discussed later under Instrumentation). Rig testing thus far has been carried out with pure nitrogen, fed to the inter-seal cavity through the nitrogen section of the apparatus.

Test Oil Circulating and Conditioning System

When installed, this system was hydraulically static pressure tested to 145 psig. Persistent leaks were observed at all flange joints, which were eventually cured by the use of gaskets of a different construction, but still resistant to the corrosive attack of the fluorocarbon fluid, DuPont PR 143. After the static pressure had been maintained for some time, water weeping from the Monel positive-displacement oil-circulating pump revealed a fine crack in the body casting. The pump has been returned to the manufacturers and a stainless steel impeller-type pump installed as a temporary replacement. This pump is adequate for all the test fluids except PR 143. Despite the difficulties that have been experienced by the pump manufacturers in securing a crack-free replacement casting, the pump is now in shipment and will be available for the screening tests of PR 143 fluid. (Due to the long delivery on the DuPont material, this will be the last fluid to be screened.)

The oil-circulating system has functioned satisfactorily as designed; the four different operating phases described in (1) have been used and the control was straightforward. The liquid metal heating jacket has also functioned well and thermocouple indications show that the heating is very uniform over the hot surfaces exposed to the oil.

Oil-mist Lubrication and Gas Cooling System

Detailed considerations of the gas consumption requirements of the oil-mist test rig, and the costs and delivery times associated with the nitrogen reclamation equipment shown in Enclosure 34 of the First Semiannual Report (1), have resulted in a redesign of the oil-mist system, shown diagrammatically in Enclosure 9. Although this system employs a once-through gas flow concept, the design is such that the return or reclamation part of the circuit may be added at a later date, should the consumption of nitrogen prove prohibitive.

After the total gas flow is measured by the transmitting flowmeter (# 4 of Enclosure 9), it is divided three ways; to feed the large oil-mist generator for the test bearing, the blanket-cooling flow for the test bearing, and a combined mist and cooling flow through a small mist generator for the roller bearing. The gas flow to the oil-mist generator for the roller bearing is preset at a desired level and maintained by the setting valve, whereas the other two much larger flows may be varied to suit the requirements of the bearing under test. The actual flow rates in each of the channels may be obtained from the IBM computer record, described in the First Semiannual Report (1).

The mist generator for the test bearing has been supplied by Alemite Inc. and is composed of a commercial two choke atomizing head mounted in a heavily constructed, three gallon capacity, steel tank readily capable of withstanding a maximum operating pressure of 80 psig at the low oil temperatures of 200°F stipulated for the oil-mist tests. An oil gage, in the form of a sight glass, is fitted to the tank, and due to the extremely low rates of oil consumption, accurate readings can best be achieved by noting the amount of oil required to refill the tank to the initial mark. Four strap heaters supplied from a variable voltage AC source and attached to the outside cylindrical surface of the tank, provide the specified temperature of 200°F. Indications are that two of the mist test oils will require higher tank temperatures in the region of 300-350°F, due to their high viscosities at low temperatures. Simple electrical heating tape will be used for the roller mist generator reservoir.

Study of the characteristics of the oil-mist generator at room temperature, with Esso Turbo 4040 oil has shown that oil may be transferred to the test bearing at rates up to 0.0185 lb/min, and that a flow reduction ratio of 10:1 may be achieved. This reduction is accomplished by reducing the through-put of nitrogen and by restricting the oil flow to the choke by a needle valve. Care must be exercised not to attempt flow rate reduction below the minimum of the available range, due to a sharp cut-off, whereby no oil is delivered to the test bearing, even though a substantial quantity of gas is passed through the generator. This cut-off point seems to be highly dependent on the down-stream pressures and almost certainly on the density of the oil being misted, and therefore will be carefully defined with each new oil before starting the screening tests. The flow characteristics of the generator using MIL-L-7808E lubricant (Esso Turbo Oil 4040), with one and with both oil valves fully open are shown in Enclosure 10. The dashed portions of the curves indicate the cut-off.

The mist generator for the roller bearing is a small Norgren unit fitted into the pipe supplying cooling gas to the roller bearing. It is equipped with both by-pass and oil-flow valves to permit variation in the mist delivery to the roller bearing.

Tests with the oil-mist rig have shown that a readily visible mist issues from the nozzle rings when the reclassifier nozzles are not fitted. With the nozzles in place, reclassification is achieved, with oil droplets readily wetting out on a cold stationary bearing. Details of the actual operation of the test rig with this method of lubrication is described in a later section of this report on Experimental Results (pages 37-40).

Instrumentation

The instrumentation has been installed as planned and, after a few corrections and alterations, satisfactory operation achieved, except for the mass spectrometer, to be discussed later. The only design change made is in the method of sensing the oil level in the recirculating oil tank. The method now employed senses the differential pressure between the top and bottom of the oil tank and is shown diagrammatically in Enclosure 11. Nitrogen from a pressure reducing valve, is passed through a small flowmeter and into the bottom of the oil tank, and a panel mounted differential gage reads the pressure excess in this pipe over that in the head of the tank. The nitrogen flow forces the oil out of the lower pipe and then bubbles to the surface. The excess pressure due to surface tension and pipe losses may be zeroed out and then the gage reading, in inches of water, has only to be divided by the density of the oil in use to yield the actual head of oil. An alarm light draws the attention of the operator when the oil level is low, and the automatic rig shutdown function is energized if the level is allowed to approach the danger level.

Instrument Calibration

The pressure transmitters for the two test rigs have been calibrated, using the apparatus shown in Enclosure 12. In order to prevent oil from the certified dead-weight tester entering the test rig pressure transmitters, a back pressure was fed to both from a compressed air source. The air pressure was adjusted against the oil in the dead-weight tester until the balance piston floated. Readings of the pressure gage and the computer readout in millivolts were obtained at each calibration point. The output of each pressure transmitter was adjusted to produce an IBM readout of -10 mV at 0 psig and -50 mV at 160 psig or 75 psig, depending on the range of the particular transmitter being calibrated. The piston of the dead-weight tester was maintained in a floating position and rotated by hand during the entire calibration.

Calibration data and curves for each pressure transmitter are shown in Enclosure 13-18. It may be seen that the linearity is better than 1%. Accuracy of the pressure gage located on the test rig control panel is within 2% except for the gage indicating pressure in the test bearing cavity of the recirculating oil rig, which has a 6% error at 25 psig. No hysteresis was observed in any of the pressure transmitters calibrated. The task of calibrating the temperature sensing equipment is currently in progress, and again both the computer records and the visual readout will be calibrated. The flowmeters were certified accurate on delivery to 1% in most cases and 2% for non-critical flows, by the manufacturer, Brooks Instruments Company, Inc. Checks will be made to verify that the settings of the flowmeters were not disturbed during the installation and initial operation of the fluid systems. In the meantime, the test rig is being operated on the strength of the original calibration. For purposes of torque measurement, an aluminum spool, of the form shown in Enclosure 19, is inserted as the spacer element into the Koppers gear coupling. It has been equipped with a strain gage bridge sensitive to torsion. Static calibration of the first spool so equipped has been performed using weights and a level arm to apply torque, with the sensor mounted in its operational position. (The test shaft was locked and the drive flat belt removed.) The results shown in Enclosure 20, indicate a good linearity, repeatability and an accuracy within ½% at a full scale which was 110% of the maximum service torque. The output signal, at 33.8 mV maximum was somewhat less than the ideal for presentation to the IBM computer, but with the elimination of the strain gage load cells, by the use of a direct pneumatic loading system, the DC power supply voltage may be increased from 20 V to 24 V, yielding a maximum bridge output of better than 40 mV.

Whichever loading system is employed, an independent means of calibrating the thrust load on the test bearing is necessary. This has been provided by preparing an extra flexible housing, with I-section, which is used to carry the test bearing outer ring in the test rig. This component has been equipped with an asymmetrical strain gage bridge, sensitive to thrust loading, but almost completely insensitive to misalignment moments. Initial static calibration of the I-section housing (Enclosure 21) in a compression testing machine indicated an output of 0.4 mV; per 100 lbs. thrust, using a 20 V DC input. The output signal was linear and repeatable up to the maximum applied thrust of 3,500 lbs. The insensitivity to bending was demonstrated by superimposing a 50 lb. load on the end of a 20" lever arm, on a direct thrust of 3,500 lbs. The strain gage bridge output signal increased only by the expected 0.2 mV, associated with the increase the thrust component. Failure of one of the strain gage elements has prevented the use of this calibrated component and the part will be regaged and re-calibrated before use.

Owing to the elimination of the load bearing and pneumatic cylinders, at least in the immediate future, check calibration of the precision regulators and the tensile load cells is not immediately necessary.

Slip Ring and Connector Assembly

To achieve reliable monitoring of the test bearing inner ring temperature, and the temperatures of the two test seal runners, three pairs of thermocouples have been embedded in the test shaft. So as 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 rotating shaft are required.

Torque input to the shaft is monitored by a symmetrical strain gage bridge. Four contacts are required to supply power to and transmit the signal from this bridge circuit, so that a total of 16 separate slip rings are required. Due to the need for accurate temperature monitoring by these shaft mounted thermocouples, special low noise slip rings have been sought capable of continuous operation at 14,000 rpm.

The choice for this service has been slip rings of the mercury wetted type. At this time, a commercial 8-channel unit is on hand and operable, but this was found unreliable and difficult to maintain. Therefore, a unit was designed and its construction will be completed within 4-6 weeks. The general assembly of this 16-ring unit is shown in Enclosure 22.

The slip ring assembly is mounted at the free end of the jackshaft and is driven from it. A tachometer is, in turn, driven by the slip ring shaft. This layout necessitates the passage of the thermocouple leads through the torque spool, where they are joined by the strain gage bridge leads, and on through the hollow jackshaft. To enable the test shaft to be removed from the rig for either a bearing change or an inspection, the wires must be disconnected. This has been achieved by supporting the wires in a special ½" diameter probe extending from the slip ring assembly, through the jackshaft and torque coupling which has a twelve pin connector mounted in its forward end. The four strain gage contacts are made, at the appropriate points along the probe, by spring loaded contacts which mate with circumferential pick-up rings mounted two at each end of the torque spool. This arrangement is shown in Enclosure 23.

The mating component of the twelve-pin connector is carried on a short extension tube itself supported in a thermal insulating material. The purpose of this extension tube is to keep 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 24.

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.

Vibration Sensing

A crystal accelerometer, capable of operating at temperatures of up to 700°F, is available for monitoring the rig vibration. A suitable mounting point on the rig will be selected with the aid of a vibration sensor to yield the most sensitive location.

Mass Spectrometer

A mass spectrometer, operating on the time-of-flight principle and supplied by Nuclide Corporation of State College, Pa., has been set up to function under the control of the IBM data control system, described in the First Semiannual Report (1), so that sequential sampling from four chambers can be achieved automatically. The refilling of the cold traps has also been accomplished automatically so that the instrument may function for long periods without attention.

The signals from the instrument may be recorded in any combination of the following three ways:

1. Pen-chart record.
2. Visual read-out on an oscilloscope.
3. Selected peaks can be recorded by the IBM computer as a millivolt signal, which is a function of the concentration of the selected ion.

The details of the functioning of the electrical and mechanical linkages comprising the automatic read-out system for helium and oxygen concentration, were presented in the First Semiannual Report (1).

This instrumentation has been extensively tested partly in the presence of the Manufacturer's representative and it was found that the sample handling system (valves, programmer, pump) now functions satisfactorily.

However, the mass spectrometer itself, i.e. flight tubes, electron multiplier, and electronic readout gear are not satisfactory for these reasons:

a) The electron multiplier is unstable and loses sensitivity due to any ingress of hydrocarbons or air without provisions for reconditioning. Consequently, helium sensitivity for concentrations less than 500 ppm is not dependable.

b) The electronics show high noise levels and therefore no adequate signal/noise ratio can be produced for helium or oxygen concentrations of the magnitude expected.

c) The zero drift of the electronics is of an order comparable to the signals expected, preventing the computer from zero correcting the reading.

d) Electronic elements are overdriven by the N_2 peak, thereby causing a shifting zero-line for parts of the spectrum beyond N_2.

SKF Industries have now decided to reject the spectrometer portion of the Nuclide instrumentation.

The Bendix Corporation was invited and has quoted a substitute system which will be fitted to the Nuclide sample handling gear and has self-calibrating and maintenance provisions, by which Bendix quarantees 10% accuracy in reading 50 ppm helium or 700 ppm O_2, which is considered adequate. The new system is now being ordered and delivery within two weeks has been promised.

Arrangement of the Instrumentation and Control System

The instrumentation and controls as now existent are described as follows:

Three large vertical cabinets house the equipment associated with each test rig, shown in their test cell location in Enclosure 3. A further small unit for each contains the motor speed controls and, in the case of the recirculating oil test rig, the liquid metal oil-heater control gear. The layout is completed by a cabinet, common to the two rigs, containing the stablized DC voltage supply for the strain gage instruments, a digital tachometer, a 48-point temperature recorder, a 2 pen continuous temperature recorder and a 2 pen continuous millivolt recorder. A patch-panel is included to provide as versatile a recording system as possible.

Vacuum Degassing System for Lubricants

A 5-gallon capacity test-oil degassing apparatus is on hand to the design shown schematically in Enclosure 41 of the First Semiannual Report (1). Four batches of Esso Turbo 4040 oil have been degassed under nitrogen at pressures in the region of 0.5mm mercury and a nitrogen purge stream of 0.1-0.3 std. liter/hr. The oil temperature was maintained at 200°F and the degassing continued for at least 72 hours. A fifth batch of Esso Turbo 4040 oil was degassed under the same condition of temperature and pressure for 150 hours. A darkening of the oil occurred, similar in character to that observed when undegassed oil was heated in the Monel oil-recirculating system, apparently due to the heating of the oil with some oxygen present. In the future all test oils will be degassed for 72 hours without heating, or if more speedy processing is needed, only one hour at 200°F will be used, as specified by the NASA Project Manager.

Test Elements

Test Bearings

Eight test bearings, having a bore diameter of 125mm and a nominal mounted contact angle of 26°, have been supplied. These bearings, part of a batch of about thirty, conform to the design designated 459981A, the leading dimensions of which are shown in Enclosure 25. These bearings have balls and rings of CVM M-50 which retain suitable hardness up to approximately 600°F.

A further group of twelve test bearings, finished to 459980, that is with rings and balls of WB49 material, for operation at temperatures above 600°F, are being made available.

As a result of further studies of the kinematics and elastohydrodynamics of high-speed thrust bearings, a bearing design modification has been introduced, which increases the ratio of ball spin torque on the outer to that on the inner ring and reduces the frictional heat generation and thus the power consumption, when compared with the original design.

Due to the state of manufacture of the bearing rings, when the design modification was completed, 0.005" oversize balls were necessitated to provide sufficient excess material on the rings for the reworking. These new bearings have been designated by sequential serial letters for the two basic bearing groups, based on the material of which the rings are made, and the leading dimensions are shown in Enclosure 27. Twelve WB49 and twenty M-50 bearings of this type are on order. Extensions to the groups may be required if it is found during the experimental testing that the functioning of the bearings, made to the modified design, is limited by cage performance. In this case back-up cages, of the types provided for the first bearing designs, will be introduced. However, all initial screening tests are planned to be conducted with the 459980 and 459981A bearings, as agreed with the NASA Project Manager.

Bearing Cages

Eight cages, finished to the primary choice of material and design, have been supplied. They are made from M-1 steel, heat treated to 53-56 Rc to give the strength necessary for the anticipated severe conditions of SST operation. The outer-ring piloted design was chosen as the best arrangement to maintain adequate lubrication of the critical guide lands. Preliminary experimental evidence indicates that the frictional drag of the cage upon the balls and the outer race may create considerable heating of the cage. The consequent expansion would absorb the generous radial clearance provided and cause one or both lands to seize on the outer race, immediately producing bearing failure by preventing proper rolling motion of the ball complement.

This mode of failure has been observed in both recirculating oil and mist tests and so cannot simply be ascribed to inadequate lubrication or cooling. However, the inner rings have consistantly run much hotter than the outer rings in all tests so far, so the thermal conditions may not be representative of final test conditions. Back-up cages of the same material, silver-plated M-1, but of inner ring piloted design are being brought forward for comparative testing. The bearings then bear the designation 459981B, Enclosure 26. It will be noted that an even larger nominal clearance on the running diameters is allowed to accommodate differential thermal expansion of the inner ring.

Further M-1 forgings are being produced from which more cages may be made to either inner or outer piloted design, and having wider than normal guide lands, should land wear prove to be a problem once the expansion seizures have been overcome.

Test Bearing Dynamics and Elastohydrodynamics

In order to predict the performance characteristics of the test bearings shown in Enclosures 25, 26, and 27 under the test conditions imposed by this test program, kinematic analysis, elastohydrodynamic film thickness calculation, and heat generation analysis have been performed with the aid of a digital computer. The computer program, prepared and owned by SKF Industries, Inc., has been developed for the study of the parameters involved in the operation of high speed, highly loaded ball thrust bearings. The analysis takes into account the bearing geometry, the direct thrust loading, the speed, the gyroscopic and centrifugal forces on the balls, the elastic deformations at the contacts and (for heat generation and film-thickness computations) the lubricant properties at the estimated operating temperature. A sample of output data for the original and modified designs (Series I and II, respectively) is shown in Enclosure 28. The effects of varying the inner race and outer race conformities is clearly shown in the data plotted in Enclosure 29. The dependence of the bearing fatigue life and the spin-to-roll ratio is critical for bearing operation at high temperatures for the spin component represents frictional heat generation, whereas the rolling component is indicative of the rate of motion of the heated contact zone over the bearing surfaces and thus, heat removal from the contact. The design must, therefore, be a compromise between conformities leading to a low value of spin/roll ratio, with maximum spin-torque ratio (described in Appendix III of (1) and in (4,5)), and a minimum sacrifice in fatigue life. The life is predicted as a function of the contact stresses and stressed volume according to the Lundberg-Palmgren theory (2), which changes as the groove conformities are changed.

The results of the calculations yield the spin-torque ratio as a function of conformity as shown in Enclosure 30. This ratio is clearly improved in the new design, but at the expense of fatigue life. However, the life of the bearing is still adequate for the purposes of this project. Furthermore, the importance that may be attached to the small increases or decreases in the predicted fatigue life of a bearing is open to question, due to the complex effects of lubrication factors on fatigue, which are not included in the computation.

The computed minimum oil film thickness at the inner ring contact with the hydrocarbon lubricant (Socony 177F) at 600°F remained essentially constant, as expected, at approximately 13 microinches throughout the range of conformity variation. From previous experimental and theoretical work (3) the bearing can be expected to operate on an essentially complete oil film, which is conducive to long life. In order to take full advantage of the available lubricant film conditions, the grooves of these tool-steel bearings have been finished to a high degree of geometrical accuracy and a surface finish of better than 4 microinches, rms, in the cross-groove direction. All surfaces of the rings and the balls are black oxide coated, which serves as an aid to initial running-in of the contacting surfaces, and also as an aid to the post-run visual and photographic inspection.

Lubricants

The selection of test lubricants for this program, discussed in the First Semiannual Report (1), was made on the basis of published performance, experience under related conditions at SKF Industries, Inc. and properties data from the fluid suppliers. Temperature-viscosity data for the selected recirculating and mist lubricants are given in Enclosure 31 and 32, respectively. Since the last report, a few modifications and additions to the list of approved fluids were made and will be discussed below.

Selected Fluids to be Used as Recirculating Lubricants

1. MIL-L-7808E - Esso 4040 Turbo Oil - unchanged.
2. Ester-base lubricant with improved temperature properties - Sinclair Turbo 1048 - unchanged.
3. A highly refined mineral oil or hydrocarbon - Socony Mobil XRM 177F.

This product of the Socony Mobil Oil Company is composed of the synthetic paraffinic type oil designated Socony Mobil XRM 109F, previously selected in this category, to which a proprietary boundary lubrication additive has been added. This change was made in view of the results of endurance tests carried out in 7205 size bearings at SKF Industries, Inc., in which the XRM 177F lubricant showed itself to be superior to the base material, XRM 109F, when running at high speed (42,800 rpm) and thrust load (459 lbs.), at temperatures of 600°F (4). Having secured the approval of the NASA Project Manager, the entire batch of 109F on hand for this program was returned to the Socony Mobil Oil Company, and is being returned to SKF Industries with the additive package included.

4. A Polyphenyl Ether - Monsanto MCS-293

The substitution of Monsanto MCS-293 for the previous fluid, Monsanto OS-124 (modified) has been approved by the NASA Project Manager by reason of the better performance demonstrated by MCS-293 in the high speed bearing tests carried out under Contract NASw-492 (4). Properties data are given in Enclosure 33. The original fluid is being returned to the Monsanto Chemical Company on a direct exchange basis.

5. A Fluorocarbon Fluid - DuPont PR 143

Manufactured by E. I. duPont de Nemours and Company, Inc. this fluid has appeared favorable in preliminary bench-type screening tests both at NASA and SKF Industries, Inc. (4). Although the fluid is corrosive to ordinary steels at temperatures above 400°F, which dictates the use of special nickel alloys for all wetted parts, other than bearings and seals, it is extremely stable at temperatures of 600°F and above. The extrapolated viscosity at 600°F is 1.6 cs, which is superior to most other fluids available and probably adequate for the generation of a full elastohydrodynamic film in the bearing contacts. Arrangements have been made for the supply of 20 gallons of the fluid, taken from a single 80 gallon batch, for use in screening tests.

The remainder of the batch will be held for SKF Industries for potential use in the endurance runs of this program.

Fluids to be Used as "Once-Through" Lubricants

1. MIL-L-7808E - Esso 4040 Turbo Oil - unchanged.
2. A Polyalkylene Glycol - Ucon 50-HB-5100 - unchanged.
3. A Polyolefin - Sunthetic - 18H - unchanged.
4. A highly refined hydrocarbon - Socony Mobil XRM 177F. This fluid has been substituted for the XRM 109F, for the reasons explained in the discussion of the recirculating lubricants (page 26).
5. A polyester - Hercolube F.

This product of the Hercules Powder Company, Inc. has been approved by the NASA Project Manager as the fifth fluid to be tested in the oil-mist test rig. This material is an unformulated saturated straight fatty ester of pentarythritol, and was selected from five candidate polyester lubricants, on the basis of oil-mist tests, details of which are described in a later section of this report. Of two fluids tested, Hercolube A and Hercolube F, which did not deposit heavy varnish and/or gum, on the bearing surfaces, the grade F material was selected because of higher flash and flame points and the higher viscosity (1.0 cs at 600°F). The typical properties of this fluid is shown in Enclosure 34 and the temperature-viscosity is plotted in Enclosure 32.

Sufficient quantities of all the test fluids necessary for the performance of the screening tests are on hand or in transit, except the exchange of Socony 109F for 177F and Monsanto Skylube 600 for MCS-293, which are in process, and DuPont PR 143, which is expected by December 15, 1965, in time for the screening tests (see later section of this report on Test Scheduling). Written agreement of lot control has been received from all the lubricant manufacturers. This is an arrangement with the manufacturer whereby he either holds, for SKF's future purchase, a sufficient quantity of oil from a single lot to complete both screening tests and endurance tests on this program, or supplies the entire required quantity of fluid to SKF at once from a single batch on a use-or-return arrangement. The first alternative has been employed for the more expensive oils and the second for the lower cost, readily available oils, to minimize expenditure.

Freon Additives

Preliminary solubility and compatibility testing of candidate Freon materials in the above selected lubricants has been conducted at DuPont and the results are given in Appendix III. On the basis of these tests, and cost and delivery, DuPont Freon 113 is recommended for NASA approval for testing on this program in both the recirculating and the mist rigs, for those tests involving a Freon additive, and if approved, a sufficient quantity of this material will be procured in time for the screening tests.

Preliminary Oil Mist Studies

Preliminary studies of lubrication by oil mist have been made to determine the requirements of the mist generator and reclassifier units, when supplying oil mist into a chamber at a pressure considerably above ambient.

A small static test set-up, constructed for the purpose, is shown in Enclosure 35. A small Norgren Micro-Fog lubricator has been connected to a heavy, thick walled tubular oven in which static bearing rings could be placed so that the flow from the oil-mist reclassifying nozzle impinged directly upon them. The process of wetting-out of the reclassified oil droplets and the thermal degradation of the oil, if any, could be observed through a thick "Pyrex" glass window in the top of the oven. The system was piped and instrumented so that the pressures and flows of nitrogen and oil could be measured over a range of temperatures.

Initial setting-up tests established that a copious oil mist could readily be generated by the small lubricator at room temperature when delivering into pressures up to 60 psig using XRM 109F oil. A small fraction of this mist wet out on the stationary bearing rings, without the aid of the reclassifying nozzle, but very much more wetting-out was obtained when a suitable nozzle was fitted.

Having established that the apparatus was operating satisfactorily, the feasibility of generating an oil-mist from five candidate polyester lubricants, and the four fluids already selected for use in the oil-mist test rig, was studied. The oil reservoir of the generator was maintained at 200°F by means of a water bath to simulate the requirements of the full-scale test.

A satisfactory oil-mist was generated in each case, although the quantity varied from oil to oil. With the two most viscous oils, Sunthetic 18H and UCON 50-HB-5100, the reservoir temperature had to be raised from 200°F up to 300°F and 350°F, respectively, to lower the viscosity sufficiently for good oil misting.

During these tests it was noted that a pressure drop of 10 psi across the mist generator produced an adequate mist, and that this differential was independent of the delivery pressure up to the 60 psig tested. The studies have also shown that the nature of the mist is affected by the size of the reclassification nozzle.

A 1/8" diameter nozzle produced no discernible change in the 2-3 scfm of oil mist being discharged, viz., the oil was still suspended as a fine mist that would not wet out on bearing surfaces. However, a progressive reduction to a 1/16" diameter nozzle produced an increasing amount of visible droplets which readily wet out on any metal surface with which they came in contact.

Further tests have shown that all the oils tested could be misted and reclassified to wet out on a stationary bearing ring maintained at 600°F at a pressure of 45 psig. There was, however, a wide variation in the appearance of the bearing surfaces after exposure to these conditions for 30 minutes. The deposits ranged from a light tarnish to thick layers of varnish and/or gum. The results are summarized in Enclosure 36, and the typical appearance of the deposits formed on the bearing rings (using Sunthetic 18H) is shown in Enclosure 37.

On the strength of their ability to avoid heavy varnish formation in the above tests, the choice of a polyester lubricant was reduced from five candidate materials to two, Hercolube A and Hercolube F.

The effluent from the tubular oven of the apparatus was released to atmosphere through a small globe valve which tended to act as a further reclassifying nozzle for the mist that had not wetted out on the preceding surfaces. This effluent was extremely oily or smoky with all the oils used and posed a disposal problem. Passing the exhaust through a pipe stuffed with steel wool reduced the smoke density, but did not eliminate it, nor did passing it through a similar pipe maintained at red heat. The most satisfactory solution found was to pass the effluent jet through a high-temperature flame, whereupon the oil droplets combusted in the excess air induced by the flame, giving a clear exhaust.

The initial operation of the actual oil-mist test rig has shown the exhaust to be a faint mist which is barely visible, suggesting that a much greater degree of wetting-out is achieved in the rig, due to the high speeds of the moving surfaces and the tortuous path the gas is obliged to take.

Shakedown Testing

In order to prepare the two test rigs for the screening tests of Tasks II and IV of this program, a period of shakedown testing was undertaken. During this period the various facets of the equipment functioning were checked out, adjusted, or modified as necessary to ensure that it functioned in the proper manner. The tests were conducted according to the general plan given below. The plan applied to both test rigs, except where noted.

Rig Shakedown Program

1. Install test rig on its table and install low speed drive. Connect up temporary lubrication and instrumentation systems.

   - check out drive and jackshaft.
   - check out test rig for operation at low speed, 2000 rpm, 1000 lbs. thrust load, using mineral oil lubrication in air without external heating.

2. Initiate installation of full instrumentation.

3. Substitute high-speed drive.

   - gradually increase speed and load up to 14,000 rpm and 3000 lbs. thrust, no external heating.

4. Changeover to the experimental lubrication systems.

   a) - Recirculating Rig
   Check function and control of recirculating oil system.

   b) - Oil Mist Rig
   Check mist formation and reclassification, obtain oil delivery characteristics of oil mist generator.

- Complete unheated running by a sustained three hour test at 14,000 rpm and 3280 lbs. thrust (full load and full speed) using MIL-L-7808E baseline fluid as lubricant using nitrogen blanketing.

5. Complete installation of instrumentation and check operability of all final systems.

6. Repeat increasing speed and load tests, but with 100 psi pressure differential maintained across the test seal pair, with the inter-seal cavity held 5 psi above the air chamber pressure.

7. Check test oil heating, circulating and temperature control equipment.

8. Repeat increasing speed and load tests with a rig body temperature of 300°F with seal pressures applied but no air heating. Increase the rig temperature to 500°F and run at full load and speed.

9. Repeat runs with hot air at 300°F supplied to back of air seal.

10. Repeat runs with hot air at 900°F supplied to back of air seal. Strip the test rig and oil system down, examine, clean, and re-assemble using degassed baseline fluid MIL-L-7808E in preparation for the first official screening test.

A. Recirculating Rig Tests with Mineral Oil
Lubrication Supplied Through an Auxiliary System

Shakedown testing commenced as soon as the recirculating oil test rig was installed in its cell and temporary drive, lubrication system and instrumentation had been arranged, in the absence of the final equipment. In the early tests, low speed running from 1000 rpm up to a maximum of 2400 rpm indicated only minor problems with the load bearing unit and the rig seal. Both of these problems were brought about by the load bearing drain tube fouling the rig mount, when slight axial movement occurred under thrust loading. The bearing housing cocked and the clamping forces on the rig seal runner were removed, permitting that component to turn on the shaft resulting in galling of the register surfaces. The drain was altered to allow considerable clearance and smoother rig operation was immediately apparent.

The temporary low-speed drive was then replaced by a 50 HP motor intended as the final power unit for the oil-mist rig, so that trial runs at higher speeds could be made. Frictional heating of the test runners at increased speeds in the region of 4000 rpm to 6000 rpm, together with the heat from the test bearing, caused approximately a 200°F temperature rise in the test shaft. With no comparable heating of the hot-air manifold, the resulting differential thermal expansion absorbed the .001-.002" radial clearance in the hot-air labyrinth seal and caused a seizure of that assembly. The static silver-plated ring was removed from the labyrinth assembly to ensure adequate clearance under any thermal conditions until such time as air heating was available to raise the temperature of the labyrinth seal ring appreciably above that of the shaft, so that seizure could not occur.

Test loads and speeds were gradually increased up to 9000 rpm and 80% of full load (2600 lbs. thrust). Repeated sharp torque increases, as measured by the power input to the drive motor, together with an examination of the rig and bearing components, indicated that smearing had occurred in both the test and load bearings. A plausible cause of this phenomenon is that the inner rings of the bearing become much hotter than the outer rings, due principally to the spinning friction at the inner race/ball contacts and the lesser heat removal from the inner ring. The internal clearance of the bearing is taken up by the greater thermal expansion of the inner ring and this results in smearing of the balls and track, leading rapidly to complete seizure.

This situation is common in high-temperature testers operated at other than design temperatures. The solution is (a) to find temporary expedients for completing the cold test runs, e.g., increase bearing looseness, and (b) re-test at design temperature and then adjust internal clearances to the conditions prevailing.

Problems with the belt drive were also encountered, in that oil mist, emanating from the load-bearing housing reduced the friction of the belt, causing it to slip on the driven pulley, lose alignment and run off altogether. With the installation of ventilation equipment in the cells, and the construction of a suitable belt guard, the oil mist was subsequently prevented from fouling the belt. The belt ran satisfactorily at 10,000 rpm and 2,500 lbs. thrust for 3½ hours.

Initial attempts to run at higher speeds were thwarted by the load bearing overheating and seizing, despite the provision of greater oil drainage space beside the bearing and an increased oil throughput. Evidence was also found that the rig seal runner was running relatively hot at about 600°F. To check that this was not the cause of the excessive load bearing temperatures, tests were run with the seal runner replaced by a mild steel spacer ring not in contact with the seal. No difference in the condition of the load bearing was observed.

Using a cooling fan directed at the load bearing, the recirculating rig was operated at progressively higher speeds and loads with the maximum oil flow possible with the temporary oil system: approximately 4 gpm to the test bearing and 2 gpm to the load bearing. The desired conditions of 14,000 rpm and 3,000 lbs. thrust were maintained for just over ten minutes when a sudden increase of torque and a rise in the test bearing outer-ring temperature indicated incipient bearing seizure. As the speed was being reduced, the bearing seized. Later examination showed severe smearing of the balls and tracks, consistent with that which would be expected when all bearing clearance disappeared.

To determine if too much oil was being supplied to the test bearing causing excessive churning losses, the flow was gradually reduced from the maximum when the rig was operating at 6000 rpm, resulting in a rapid increase in the test bearing outer ring temperatures. The reason for this was found when the oil flow from the jets was observed with the test rig partially disassembled. At any setting much below the maximum flow, the jets failed to impinge upon the inner ring. All remaining tests were carried out at the maximum flow available from the temporary system.

Increased drainage provisions and the use of a separate scavenge pump for the load bearing permitted further runs in the speed range of 13,000 to 14,000 rpm, but at this time the problems with the flat belt had not yet been resolved and this caused drive problems.

During some of these tests plots of the power consumption of the DC motor were made, which are shown in Enclosure 38 (together with others made during operation of the oil-mist rig). The plots show that 45 HP is required to drive the recirculating rig at a rig speed of 14,000 rpm. This confirms the requirement of 40 to 45 HP estimated at the design stage, which led to selection of a 75 HP drive.

B. Oil-mist Rig Tests - Temporary Jet Lubrication System

Recirculating rig operation was suspended while the 75 HP drive and the special Monel oil system were installed. Meanwhile, study of the rig operating problems was continued on the oil-mist test rig to which the 50 HP motor was moved. For this rig, the estimated power consumption was less than for the recirculating rig, due to the elimination of oil churning friction losses, and thus the 50 HP motor was expected to be adequate. This was confirmed by the power consumption curves in Enclosure 38 plotted for the oil mist rig, both with and without the load bearing, taken later when the test bearing was mist lubricated.

The mist rig load bearing mount and housing were modified from the original design to that shown in Enclosure 4, to provide underrace cooling for the inner rings. Oil supplied through three jets is collected by a scoop on the shaft and pumped by centrifugal action to the bore of the loaded inner-ring half. The feed holes are so positioned that one third of the total oil passes between the two halves of the inner rings, into the bearing and one third flows over each end face to provide uniform cooling. Using this cooling system, the rig was run at progressively increasing speeds up to 13,000 rpm at 2600 lbs. thrust for over two hours. The outer-ring temperature of the load bearing (about 330°F) was higher than found with the original cooling arrangement, but the bearing gave no indications of distress. Strip down and subsequent examination revealed the bearing was in excellent condition. Thus the load bearing was found to operate satisfactorily with underrace cooling because the inner race receives better cooling (somewhat at the expense of the outer race.)

After reassembly, the oil-mist rig was brought up to full speed and full load, over a period of one hour, and run for over four hours at those conditions, until shutdown was caused by disintegration of the drive belt. Subsequent examination showed that all bearing and seal components were in good condition. The lubrication of all bearings in this successful test was still furnished by the temporary mineral oil jet lubrication system intended to be used later only for the lubrication of the jackshaft and load bearings.

Leakage from the load bearing housing has remained a problem, aggravated by bulk spillage when the rig comes to rest due to the loss of the centrifugal pumping action of the load bearing assembly. It is possible to eliminate the spillage by providing for a solenoid valve in the oil supply line, controlled by a centrifugal switch, built into one channel of the mercury slip ring unit which would cut off all but a small oil flow once the shaft speed had fallen below 500 rpm. However, the tests in the near future will be run with the entire load bearing arrangement removed (see page 10) in favor of pneumatic loading, thus eliminating the load bearing lubrication problem.

With the completion of the installation of the 75 HP motor and the Monel oil system for the recirculating rig, shakedown testing continued on the two rigs simultaneously, each with its own special test bearing lubrication system. For clarity, test series of the two rigs will be discussed separately.

C. <u>Oil-mist Rig Tests - Final Mist Lubricating System</u>

Having established that the oil-mist test rig was basically operable, conversion was made to the large rig oil-mist lubrication system using air and mineral oil. Static tests were first conducted to establish that oil from the supply tank of the mist generator was satisfactorily misted and transported in mist form to the test bearing region and that it wetted-out well on the bearing surfaces upon reclassification by the nozzles. These tests showed successful operation. The large rig mistor was then cleaned out and filled with Esso Turbo 4040 and the nitrogen supply system was connected up. The supply characteristics of this oil-mist generator in the test rig with this oil are shown in Enclosure 10. Testing then commenced with this oil-mist lubrication using, for the first time an actual, rather than a "set-up", test bearing of the 459981A design. (See Enclosure 25) This bearing has M-50 rings and balls, and silver plated M-1 outer land riding cage. After a short run of 15 minutes at 4000 rpm and 1000 lbs. thrust load, the rig was stopped and thetest bearing inspected. It appeared in good condition and had an oily appearance indicative that the oil mist was reaching the bearing and wetting out. The rig roller bearing at this time only received lubrication by way of the overspill of mist from the test bearing but it too, showed evidence of an adequate supply. The test was repeated at 6000 rpm, for thirty minutes with the same results upon inspection, but a load bearing failure occurred when attempting the next run at 8000 rpm, after 45 minutes of operation.

Examination of the failed bearing showed that all internal clearance had been taken up and suggested as a cause inadequate cooling of the load bearing due to continuing underrace cooling problems, as discussed previously in this report.

At this time, the final nitrogen blanketing and cooling system designed specifically for the task, became operable and so the tests were repeated using nitrogen for both oil-mist generation and for extra cooling flow over the bearing. A test bearing type 459981A was used, having M-50 rings and balls and a silver plated outer-land guided M-1 cage. Cooling air was also forced through the hollow test shaft to help prevent overheating of the inner ring. After two hours of satisfactory operation at 6000 rpm and 1000 lbs. thrust, the outer-ring temperatures of the test and roller bearings were relatively low and so both the supply of oil mist and the cooling nitrogen were gradually cut back from 100 scfm to 60 scfm total (of which the nitrogen to the mist generator was reduced from 50 scfm to 5 scfm) and as expected there was a slight increase in the bearing temperatures. As this nitrogen flow to the oil mist generator was being cut back to 5 scfm, the mist exhaust coming from the outdoor stack ceased to be visible, which was later found to be due to a cutoff in the oil supply within the mist generator caused by a lack of venturi lift at the low flow rate at the particular pressure prevailing within the mist generator. The bearings continued to function however, despite a further turndown of the total nitrogen flow to 45 scfm, until a sharp rise in friction of the test bearing terminated the test after a total of almost seven hours of operation, the last three of which had been apparently without effective oil-mist supply. Subsequent examination of the test bearing revealed the inner and outer races were not seriously damaged but the ball complement had become temperature discolored and the cage ball pockets and one land were severely smeared.

The evidence suggested that the lack of lubricant manifested itself initially in the sliding contacts occurring between the balls and the cage pockets and the cage lands rather than at the rolling and spinning ball-race contacts. The conditions of the active surfaces of this bearing is shown in Enclosure 39.

After establishing the cut-off characteristics of the oil-mist generator for the mineral oil used (See Enclosure 38) a further test was run using a new test bearing of the same design (459981A) under conditions of slowly increasing speed and load. The flow rate through the mist generator was held at 20 scfm throughout the test, as this figure was safely above the cut-off point and was predicted to yield

0.006 lbs/min of oil at the test bearing. A further 20 scfm of nitrogen was also supplied directly at the bearing through the cooling nozzle system. The test proceeded up to a speed of 9,600 rpm and a thrust of approximately 1,800 lbs. with the outer ring temperature of the load, roller and test bearings each being about 240°F despite their very different lubrication conditions. After twenty minutes at this condition the outer-ring temperatures of both the roller and the test bearings increased rapidly, rig seizure occurring as the speed was being reduced. Subsequent examination showed the test bearing had suffered smearing of the loaded half of the inner ring with the apparent transfer of silver from the cage pockets via the balls to the inner ring. Large brinell marks at the exact ball spacing all around the outer race suggested that cage seizure occurred, preventing the balls from rolling, and so the balls had produced the wear marks by rotating on a fixed spot. Examination of the severely worn cage pockets and the smeared condition of one guide land tended to confirm this.

The fact that the unloaded inner ring half showed no sign of contact, eliminated the loss of internal bearing clearance as a possible cause of failure. However, there is the possibility that the cage to outer ring land clearance was lost due to high cage temperature.

D. <u>Final Mist Rig Shakedown - MIL-L-7808E fluid - No heat</u>

After modification of the test rig to the pneumatic loading arrangement discussed earlier in this report (page 10), a series of tests culminating in one at full load and full speed has been made. In this test, the rig was run at 14,000 rpm and 3,280 lbs thrust, with MIL-L-7808E, using oil-mist lubrication and a 100 psi pressure differential across the test seal pair, but with no external heating. After an hour under the maximum conditions, distress in the roller bearing caused a shutdown. This roller bearing was of a standard aircraft type in current manufacture, not one of the special roller bearings designed for the project.

All unheated check-out testing of both rigs was conducted with these standard roller bearings, which differ from the special roller bearings manufactured for this project in that it (the standard bearing) has a round outer ring, whereas the special bearing has an out-of-round outer ring computed to prevent roller-race skidding at the high design speed and nominal roller-bearing loads in the test rigs (1). These standard bearings performed entirely satisfactorily for many hours running at 12-13,000 rpm, as well as for some short periods at 14,000 rpm, with the load bearing arrangement in the rigs shown in Enclosure 4. The smearing failure of this bearing described above occurred when the mist rig was run for the first time without the load bearing installed, as shown in Enclosure 5. It is conjectured that the weight of the load bearing housing and its fixtures produced sufficient radial load on the roller bearing in the early tests to prevent gross roller skidding (and the attendent smearing), whereas without this weight, excessive skidding occurs with the standard bearing design. The out-of-round outer ring design of the special bearing, therefore, should remedy this situation by providing sufficient traction at the small diameter of the outer ring to prevent roller skidding.

Early testing of the special roller bearing in the circulating rig (to be discussed in the next section of this report) showed unusually heavy contact on the roller and race tracks, with all of the black oxide coating completely removed from the roller paths and roller OD's after only short running periods. No smearing occurred on any contacting surfaces (even though the roller-ends, flanges, and cage pockets showed heavy, sometimes uneven contact, as discussed later), so the bearing seemed to be basically operable. However, when this special roller bearing was used in the mist rig, replacing the smeared standard bearing discussed above, two catastrophic smearing failures occurred after only short test runs. It is believed that these failures were due to thermal take-up of all internal clearance, leading to overloading of the rolling elements.

The radial looseness of the special rig roller bearing is now being examined and may be increased to eliminate the problem.

E. Recirculating Rig Tests - Final Oil System

The final oil circulating system for the recirculating rig was installed and a static oil heating test carried out. It showed the system to be operable.

The first running test employed a provisional test bearing having standard finished M-50 rings, stabilized 52100 balls and a silver plated silicon-iron-bronze cage. The test was made with Esso Turbo 4040 oil to MIL-L-7808E specification at an oil inlet temperature of 450°F which rose gradually to an intended 500°F as the rig speed was increased progressively up to 10,000 rpm at a thrust load of approximatly 2,400 lbs. The test was terminated after two hours running time by the load bearing temperature exceeding its preset limit of 300°F, and causing an automatic shutdown. Examination of the test bearing and the special out-of-round rig roller bearing (No. 459982) designed for this task, revealed no signs of distress in the test bearing. The roller bearing, however, indicated considerable severity of flange to roller contact.

Attempts to repeat the above test using a final design test bearing (No. 459981A) were impeded by electrical control problems. The test was eventually started with the oil-inlet temperature at 400°F which increased to 460°F due to mechanical power input to the test rig by the time the shaft speed was up to 8000 rpm. A seizure occurred as the speed was being raised to 10,000 rpm. A detailed examination showed severe local wear on the outer race groove of the test bearing and a uniformly smeared inner ring. The cage ball pockets were severely smeared and one cage land had smeared while the other appeared almost unworn.

In one test of over 8 hours total duration, full speed and load were achieved for a short period before the rig was stopped automatically by excessive load bearing temperature. The test bearing from this test was in good condition and is shown in Enclosure 41.

This was the last test of any significant duration on the recirculating rig. This rig has since essentially only been used for short low-speed testing of the test air labyrinth as described in the Section on Test Rigs earlier in this report.

Test Seals

During the initial stages of the shakedown testing when no differential gas pressures were maintained across the first generation test seals, no malfunction of any type was observed. The seal runners invariably showed highly polished annular rings on their lapped chromium surfaces. The carbon seal faces also developed a uniform polished appearance over the entire surface of the dam, and also over parts of the pads flanking each side of the dam.

Discoloration of the conical back of the runners indicated that a temperature rise had occurred during operation as expected.

After the application of nitrogen and air pressures to maintain the stipulated 100 psi differential pressure across the seal pair, total leak rates of the nitrogen from the cavity between the seals was measured. Generally this total leak rate was 6 scfm or less, which is an acceptable quantity for low temperature operation. Appreciably higher seal leakages have been found during the initial pressurized runs of the oil-mist rig, and were reduced by relapping both the seal runners and the carbons. The good condition of the oil side test seal after a run of eight hours duration, at speeds up to 14,000 rpm in unheated Esso Turbo 4040 oil, is shown in Enclosure 42.

The application of 400°F hot air to the back of the air seal had no detectable effect upon the leak rate.

Summary of Tests

A total rig operating time of approximately 160 hours has been accumulated to date during shakedown testing. This time has been divided almost equally between the two test rigs, the recirculating oil rig having slightly the greater time.

Reviewing the shakedown testing performed to date against the debugging program plan outlined earlier in this report, it can be seen that the test rigs have been operated "cold" at full load and speed for periods of something less than the desired three hours.

The fact that the last test reported on the oil-mist rig was terminated by a failure other than the test bearing is significant, however, and the low outer-ring temperature of 275°F at full load and speed is indicative that satisfactory test bearing operating is possible. The first seven parts of the debugging program have thus been completed with one test result indicative that Part 8 may be accomplished readily. Short runs, conducted during the labyrinth seal study on the recirculating oil rig indicate that the maintenance of a 400°F air temperature at the back of the air test seal is feasible and so Part 9 of the program has, in essence, been fulfilled.

There thus remains the final part 10, with the hot-air temperature increased in the region of at least 900° - 1000°F to be accomplished before the first screening test can be initiated.

The unreliable behavior of test and rig bearings, i.e. the random incidence of smearing failures is not unexpected. It is a rare event in high-temperature high-speed testing if a test system operates through its first runs without some bearing failures. The way to eliminate these is by patient modification of minor design parameters of the bearings, of lubrication and heating details and of methods of run-in. This method applied in the contract NASw-492 (4) has led to very high operating reliability of the bearings after considerable effort.

The scope and timing of the present project does not allow for any extended bearing or systems development to optimize performance prior to the screening tests.

It has been agreed with the NASA Project Manager that screening tests will be initiated immediately with bearings and systems in essentially design condition and early test experience carefully evaluated. If the tests appear to yield unsatisfactory information, making it mandatory to further match bearing and systems design to the lubrication and temperature conditions of the test, then NASA Project Management will be appraised and appropriate redirection of work sought.

Scheduling of Screening and Endurance Tests (Tasks II-IV)

Referring to the PERT network of the project tasks in Enclosure 1, it has been indicated that the check-out of the test bearings, seals, rig, and systems (Task I) is now completed although, as discussed in the previous sections of this report various deficiences in some components and systems have so far prevented extended periods of operation under full design conditions. However, all such deficiences have been, or are being remedied and, it is expected that screening tests (Tasks II and IV) at least in the recirculating rig, will begin by the end of November (week 57, which is at the latest allowable time indicated on Enclosure 1) under full specified conditions, with the following exceptions:

a) The mass spectrometer will not be operable, so that no real time analyses of oxygen and helium content of the test and seal chamber will be available before mid-December. In the meantime, gas samples will be collected during each screening test for gas analysis at a later time, which is being arranged. This will result in a lack of information regarding the individual leak rate through the air seal and the oil seal and only their sum will be known. It also supplies information on oxygen content in the bearing chamber only after the test so that excess oxygen if present cannot be corrected during testing.

b) The final slip-ring and connector systems will not be available before mid-December, so that the 8-ring assembly now on hand will be used for initial screening tests. Since this unit has not proved to be reliable in the testing so far, the results of each test in which it is used will be reviewed and a decision made regarding the acceptability of the data.

c) Some minor instruments have not yet been operated on a fully automatic basis, such as the vibration sensor, some housing thermocouples, and the nitrogen blanket flow controller. Final adjustment of these instruments will be attempted in the first tests, but if unsuccessful, the data will be taken manually and reviewed later for acceptability.

d) Modifications of some parts of the hot-air manifold for operation at full pressure and temperature with the new pneumatic test bearing loading scheme will be completed by week 57 for the first screening tests, and the success of these parts in attaining full specified test conditions will be reviewed at that time.

It is planned to present NASA Project Management with a detailed account of the screening tests run as described above and obtain a decision, on the basis of the results, as to the acceptability of these first tests under Tasks II and IV.

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3. Tallian, T. E., Brady, E. F., McCool, J. I., and Sibley, L. B., "Lubricant Films in Rolling Contact of Rough Surfaces", ASLE Trans. 4, (2), November, 1961.

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APPENDIX I

EXHIBIT "A"

SCOPE OF WORK

SST LUBRICATION SYSTEM INVESTIGATION

(NAS3-6267)

TASK I. Bearing and Seal Assembly Test Rig

The Contractor shall provide two bearing and seal assembly test rigs (one of which shall be designed and fabricated by the Contractor and shall be Contractor-owned equipment and the other shall be fabricated under this Contract and shall become the property of the Government) which shall simulate the bearing and seal assembly required in the propulsion system of a Mach 3 supersonic transport. Each rig shall be constructed of the same basic elements using dimensionally-stable corrosion-resistant structural materials, with appropriate modifications made for a recirculating lubrication system in one rig and a once-through system in the other. Provision shall be made for heating the test bearing and seal assembly, preheating the lubricant and using nitrogen gas blanketing in the test cavity. Quality of the nitrogen gas to be used for inerting shall be at least 99.99 percent by volume nitrogen, oxygen content of not more than 50 ppm by volume, hydrocarbon content (as methane) of not more than 5 ppm by volume, and a dew point of -90^{o} F. or better. 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^{o} 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^{o} F. hotter than the outer ring temperature. Oxygen content shall be monitored in the test cavity and the following additional requirements shall be met:

A. Bearings

1. The primary test bearing shall be a split-inner-ring ball bearing with a 26^{o} nominal contact angle manufactured from consumable-electrode vacuum melted M50 or WB49 steel, dependent upon the highest operating temperature required, suitable for use as an aircraft gas turbine engine bearing with a life of at least 3000 hours. The WB49 steel shall be used for test bearings required to operate at temperatures higher than 600^{o} F.

2. The bearing selected, SKF #459862, has a bore diameter of 125 mm.

3. The test shaft shall be run at 14,000 rpm to yield a dn value equal to 1.75 x 10^{6}.

4. The bearing shall be tested under a thrust load P=3280 lbs. (C/P = 7.49) equivalent to a L_{10} = 500 hours (420 mill. revs.) as calculated by AFBMA (Anti-Friction Bearing Manufacturers Association) methods. Bearing thrust load shall be applied by a system designed so that the load shall be independent of the pressure drop applied across the test seal assembly.

B. Seals

1. The test seal shall be constructed as a pressure-balanced double face seal assembly suitable for use as a mainshaft seal in an aircraft gas turbine engine with a design life of 3000 hours. Seal ring materials shall be selected on the basis of best wear life data available under SST conditions. The first trial run seal springs and bellows shall be made of AM350. As a backup material, Inco 718 shall be considered. Damping shall be used if excessive vibration of the bellows seal occurs.

2. The seal shall have a mean face diameter of 6.33".

3. The seal shall provide a mean interface speed of 388 fps.

4. A leakage in excess of 5 scfm across any single seal at 100 psiΔp with 1200°F. high-pressure air shall be considered a seal failure. Gas flowmeters in the nitrogen and high-pressure air lines and mass spectrometer monitoring of all test chambers, using a helium tracer in the nitrogen blanketing gas, shall be used to provide maximum flexibility in tracing seal leakage paths.

5. Nitrogen gas shall be supplied to the space between the two seals at a pressure of about 5 psi higher than the 1200° F. air pressure, and the test cavity shall be vented overboard, providing this system meets the nitrogen leakage and other requirements of an SST system. Otherwise, both seals shall be allowed to leak into a common cavity and shall be vented overboard. Total oxygen content of leakage gases into the test cavity shall not exceed 0.5%.

TASK II. Closed and Inerted Recirculating Lubrication System

The Contractor shall investigate the feasibility of a closed and inerted recirculating lubrication system under simulated Mach 3 SST engine flight conditions.

A. Fluid Selection

The following five fluid types shall be selected for investigation on the basis of lubricating ability, minimum residual deposits under test conditions, heat of vaporization, additive susceptibility, and present availability of lubricant. Consideration shall be given to those fluids with high thermal stability to operate satisfactorily in a low oxygen atmosphere, which would otherwise fail in an open circulating system at bulk oil temperatures in the range of 525° F. to 550° F. The pour point of candidate oils shall be at least as low as -30° F.

1. A baseline fluid meeting Mil-L-7808E specification, incorporated herein by reference and hereby made a part hereof, Humble Oil and Refining Company's 4040 Turbo-oil

2. A fluid as similar as possible to Mil-L-7808E, but with improved high-temperature lubricating properties such as Sinclair Turbo S oil, type 1048 (improved).

3. A highly refined mineral oil such as Socony Mobil XRM-109F.

4. A polyphenyl ether, Monsanto's OS-124 (modified according to a user's proprietary formulation) or a special blend designed to reduce the pour point without sacrificing high-temperature viscosity excessively.

5. A fifth fluid, such as Monsanto's MCS-525, as an improved version of MCS-258, or DuPont's MLO-64-9.

The NASA Project Manager will select one of the above base fluids and a Freon-type additive for running as described in Paragraph B.1. of this Task II. The final selection of all test lubricants shall be approved by the NASA Project Manager.

B. Screening Tests

The Contractor shall perform a series of screening tests using the five approved fluids in the bearing and seal assembly to establish the extent of corrosion, system deposits and mode of failure of the closed and inerted recirculating lubrication system under simulated Mach 3 SST engine flight conditions.

1. Test Procedure

The experimental rig shall be operated under each set of conditions (type of fluid and bearing temperature) for a duration of three 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 to an excessive leakage rate, or (c) excessive coking of the oil to the extent that oil flow to the bearing cannot be maintained. A run shall consist of using a lubricant from the selected list, previously degassed, in a nitrogen-inerted recirculating system at an oil-in temperature of 500° F. ± 10° F. This run shall be conducted with the bearing outer-ring temperature maintained at 600° F. If the full three-hour test is conducted without failure, the test elements and rig shall be cleaned and a new run conducted at an outer-ring temperature 100° F. higher than that obtained in the previous run. After a failure of a test part is obtained, another run shall be conducted with the outer-ring temperature maintained at 50° F. less than that at which the failure occurred. This procedure shall be followed for each lubricant to establish the maximum bearing operating temperature for each lubricant under inert blanketing conditions.

The effect of using a Freon-type additive, in the amount of 10% or less by volume, on system performance shall be explored in a separate test run in one degassed lubricant. Tests of the lubricant's spontaneous ignition temperature shall be conducted on the test lubricant prior to the screening test, according to ASTM specification D2155-63T, incorporated herein by reference and hereby made a part hereof.

The lubricant, which is expected to be in the range of 525° F. to 550° F. (or higher in some cases) as measured at the oil outlet from the rig test cavity, shall be cooled by a heat exchanger down to 500° ± 10° F. before being returned to the test cavity. The initial volume of each lubricant in the system shall be kept constant and as small as possible, preferably 4 to 6 gallons, and required system volume above 6 gallons shall be approved by the NASA Project Manager. The oil flow rates used shall be representative of advanced engine design practice as approved by the NASA Project Manager.

All lubricants shall be degassed for a 72 hour period immediately before running by means of a mechanical vacuum pump capable of maintaining a pressure of 10^{-3} torr. New bearings shall be used for each lubricant series and after each bearing failure. Rework of seals is permitted to insure maximum utility of useable seal components, provided such rework does not affect the test results. Reworked seals are subject to Task VI, Quality Assurance Provisions. Discrepancies in reworked seals are also subject to Task VI.

2. Data Required

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 in each test run. Samples of system deposits shall be collected. Portions of these samples shall be analyzed and other portions shall be delivered, properly identified, to the NASA Project Manager, at his request.

3. Base Line Test

The Contractor shall obtain base-line data with a circulating oil system open to the atmosphere as in conventional systems. (This test shall be run under conditions selected to avoid explosion hazards.) The lubricant found in the inerted screening tests to provide the maximum bearing operating temperature shall be used in this base-line test to determine the highest tolerable operating conditions in an open system by means of a series of screening tests similar to those employed in the closed system studies. Should the assembly run successfully at the maximum operable bearing temperature as determined for the fluid in the closed inerted system, the run shall be extended to failure or to a maximum of 100 hours at that maximum temperature.

If the polyphenyl ether lubricant does not exhibit the highest satisfactory operating temperature in the inert system screening, and if failure is obtained within the three-hour test period in the open system with the lubricant having the highest inert system pe perature capabilities, then the open system test shall be repeated with the polyphenyl ether.

TASK III. Endurance Tests of Closed and Inerted Recirculating Lubrication System

The Contractor shall perform endurance testing of the closed and inerted recirculating lubrication system. Based on the results of Task II, a lubricant shall be specified by the NASA Project Manager for extended endurance testing in the closed and inerted recirculating lubrication system. At least two tests shall be conducted for a total running time of 1000 hours each or until failure occurs as defined under Task II. These endurance tests shall be run in periods of from 5 to 10 hours duration running time between which the test rig shall be shut down and allowed to cool for a period sufficient to reach a temperature of all parts not exceeding 200° F. (probably a period of 1 to 2 hours) before starting another test cycle. These tests shall be conducted under operating conditions specified by the NASA Project Manager.

TASK IV. Once-Through Lubrication System Study

The Contractor shall investigate the feasibility of the "once-through" or "throw-away" lubrication system under simulated Mach 3 SST engine flight conditions. The "once-through" lubrication system is herein defined as a system where a small amount of lubricant is supplied in a liquid state as a gas-oil mist or in droplet form through an orifice close to the test bearing and then discarded.

A. Fluid Selection

The following five lubricant types shall be used in mist or drop-feed tests. For the mist application, the aspirating gas shall be nitrogen except for the lubricant with a Freon-type additive where air will be used. (This test shall be run under conditions selected to avoid explosion hazards.)

1. A fluid meeting Mil-L-7808E specification, incorporated herein by reference, Humble Oil and Refining Company's 4040 Turbo-oil.
2. A polyalkylene Glycol, such as Union Carbide's UCON 50-HB5100.
3. A polyolefin, such as Sun Oil's Sunthetic 18H.
4. A highly refined mineral oil, such as Socony Mobil XRM109F.
5. A polyester, such as Rohm and Haas Plexol 79.

The NASA Project Manager shall select one of the above type fluids and a Freon-type additive for running as described above. The final selection of all test lubricants shall be approved by the NASA Project Manager.

B. Screening Tests

The Contractor shall perform a series of tests to determine the characteristics of the candidate fluids in a "once-through" lubrication system. Data to be obtained include extent of corrosion, system deposits and mode of failure of the "once-through" lubrication system under simulated Mach 3 SST engine flight conditions.

1. Test Procedure

An oil mist system shall be designed and tested early in the program so that it shall be available for use in the screening tests. Once the mist system parameters (flow rates, and nozzle geometry) are established for each lubricant, the fluids shall be screened using a test procedure as outlined in Task II. If a fluid will not perform satisfactorily in a mist system, a drop-feed lubrication system shall be used. The test parameters shall be identical to those of Task II except the lubricant shall be supplied from a pressurized reservoir at a temperature of 200° F. $\pm$ 5° F.

NASA TN-D-1994, incorporated herein by reference and hereby made a part hereof, shall be used as a designing guide for the lubricant feed systems and extrapolation of these data shall be used as a guide to predict the minimum oil flow requirements of the specified conditions. The lubrication system shall supply a flow rate of the order not exceeding 0.1 lbs./min. with an absolute maximum of 0.25 lbs./min. oil flow.

2. Data Required

The data requirements shall be identical to those of Task II.

TASK V. Endurance Tests of "Once-Through" Lubrication System

The Contractor shall perform endurance testing of the "once-through" lubrication system. The best overall "once-through" lubricant shall be selected by the NASA Project Manager and two endurance runs containing at least 100 shutdown periods (thermal cycles) shall be conducted with this "once-through" system (the test to be run for a total of 1000 hours, or until failure occurs). Operating conditions shall be specified by the NASA Project Manager.

TASK VI. Quality Assurance

A. Raw Material and Purchased Article Control

Procurement documents for raw material and other purchased articles shall be prepared under Corporate Procedure SKF 1307, entitled "Processing of Orders Covered by a U.S. Government Contract", incorporated herein by reference and hereby made a part hereof.

B. Receiving Inspection

Inspection of raw material and purchased articles shall be accomplished under Corporate Procedure SKF 1308, incorporated herein by reference and hereby made a part hereof.

Test bearings used in the performance of this Contract shall be made by SKF Industries, Inc. 100% inspection, documentation, and certification of results for dimensions, serialization, hardness and process inspection shall be done by SKF according to requirements of product drawings and specifications and controlled through corporate quality control and inspection procedures which shall be in accordance with NASA Quality Publication NPC-200-3, incorporated herein by reference and hereby made a part hereof. The quality control and inspection procedures shall be available for on-site inspection by authorized NASA personnel.

Bearings shall be manufactured in accordance with Specification 471085 entitled "Identification Control of Critical Bearings", incorporated herein by reference and hereby made a part hereof.

Finished seals shall be purchased and the requirements of 100% dimensional inspection, serialization, documentation, and certification of inspections results shall be specified on Purchase Orders and shall be controlled by Procedures SKF 1307 and SKF 1308, incorporated herein by reference and hereby made a part hereof, and the supplier shall be bound by NASA Quality Publication NPC 200-3, incorporated herein by reference and hereby made a part hereof.

Lubricants shall be purchased under specifications prepared by SKF Industries with NASA approval, and certification of compliance with the specifications shown on the procurement document shall be required. Controls shall be effected through procedures SKF 1307 and SKF 1308, incorporated herein by reference and hereby made a part hereof, and the supplier shall be bound by NASA Quality Publication NPC 200-3, incorporated herein by reference and hereby made a part hereof.

Discrepant material shall be reviewed in accordance with Procedure SKF 1309 entitled "Identifying and Processing Non-conforming Supplies", incorporated herein by reference and hereby made a part hereof, with the exception of discrepant bearings, seals, and lubricants. The material review board for these items shall consist of the SKF Project Leader, an SKF Central Quality Control representative, and the NASA Project Manager.

C. Test Hardware Inspection

In support of the fabrication and assembly performed by the Contractor on test hardware the Contractor shall provide for 100% dimensional inspection as well as any other inspection necessary to insure compliance with engineering specifications supporting these processes.

D. Control of Data Acquisition Equipment

Data Acquisition Equipment. Equipment used in the acquisition of data shall be calibrated, evaluated, controlled, and maintained to ensure reliability and accuracy.

1. Calibration. Data acquisition equipment shall be calibrated at pre-established intervals or prior to and after usage. The equipment shall be calibrated against certified standards which are readily traceable to national standards.

2. Evaluation. Prior to use, data acquisition equipment shall be evaluated for accuracy and the results shall be documented. The evaluations required are dependent on the equipment and its use.

   a. Commercial equipment for which sufficient information is available relative to accuracy, stability, and repeatability, when used in a manner consistent with its design, need not be subjected to evaluation tests. However, a calibration shall be made and the results documented to indicate the amount of measurement error through its entire range.

   b. Special equipment designed to provide specific measurements shall be evaluated. In addition, this equipment shall be operated using actual test procedures and conditions to verify correctness of the procedure, ease of operation, and accuracy. Results of all evaluations shall be available for review by NASA-Lewis Research Center.

E. Cleanliness Control

The Contractor shall control the cleanliness of all components that will contain or otherwise contact the test fluids by enforcing a cleanliness specification covering:

1. The cleanliness level required.

2. The method for measuring and inspecting for cleanliness.

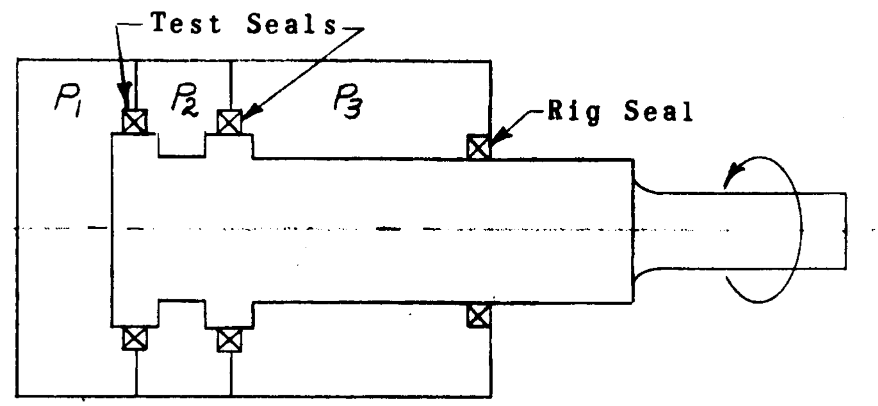
3. Component cleaning procedures.

4. Control of ambient test conditions.

5. Purge or other cleaning procedures to be used when changing from one test fluid type to another.

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.

Assuming the labyrinth seal has no significant pressure drop across it then 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:

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

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

Recalling that the stipulated operation pressure differential across the test seal pair is 100 psi

the $P_1 - P_3 = 100$ psi, and given that

$$d_T = 6.370'' \text{ and } d_R = 4.500$$

Consequently when $P_3 = 5$ psi (the minimum) $T = 3262$ lb.
$P_3 = 45$ psi (the maximum) $T = 3900$ lb.

Now the required bearing thrust load is 3280 lbs., thus it is possible to select a single cavity pressure P_3 which gives the exact loading required.

Thus $T = \frac{\pi}{4}\{6.370^2(100) + 4.500^2(P_3)\} = 3280$

and $P_3 = 6.19$ psig

This linear relationship of bearing load to bearing cavity pressure is shown in Enclosure 40 .

In practice there is some slight additional thrust from the spring forces of the three bellows seals in the test rig, which all act in a direction such as to increase the loading of the bearing. For this reason, and because the hydraulic diameters of the seals are not in reality identical, a direct thrust load calibration must be carried out and correction made to the pressure. This applies to either form of loading considered for these test rigs.

APPENDIX III

DU PONT REPORTS ON FREON ADDITIVE TESTING

"FREON" PRODUCTS LABORATORY

E. I. du Pont de Nemours & Company

TECHNICAL REPORT

Report Number: KSS-5103

Subject: Solubility of Fluorocarbons in Lubricants

Company: "Freon" Products Division

By: H. H. Seibert Date: January 18, 1965

Approved by: H. M. Parmelee

Object: To determine if several fluorocarbons are at least 50% soluble by weight at 70°F in a selected group of lubricants. An approximate solubility will be found when the solubility is is less than 50%.

Results:

Only one of the fluorocarbons ("Freon-114B2") was soluble to more than 50% in all of selected lubricants. Four of the fluorocarbons were soluble to more than 50% in seven of the lubricants. One of the fluorocarbons was soluble to more than 50% in three of the lubricants. Two of the fluorocarbons were soluble to more than 50% in only one lubricant. Data are tabulated in Table #I.

Comments:

The general procedure for mixing the fluorocarbon and lubricant was to slowly add approximately 1 ml increments of the fluorocarbon to 10 grams of the lubricant while being stirred in a 70°F bath. The flask was weighed at the first indication that the fluorocarbon was insoluble. Since only a small amount of perfluorodimethylcyclobutane was available, an ultrasonic bath had to be used to mix such small quantities. This dictated a temperature of approximately 85°F.

TABLE I

SOLUBILITY OF FLUOROCARBONS IN LUBRICANTS

Solvents / Solutes	Mobil XRM 109F	Sunoco 18H Bottoms (X424-125)	Monsanto OS-124	Esso 4040 Turbo Oil	Sinclair Turbo S1408 (Improved)	Union Carbide Ucon 50-HB-5100	Rohm & Haas Plexol 79	Du Pont PR 143
"Freon" E 3	6.2	5.8	5.4	12.8	7.2	10.5	5.6	50.8
"Freon-112"	50.5	50.5	54.3	50.7	51.3	51.7	50.5	11.8
"Freon-112a"	53.0	50.9	54.7	57.7	51.2	53.4	50.1	9.3
"Freon-113"	51.6	51.4	47.4	51.5	51.2	52.5	50.2	50.3
"Freon-114B2"	51.6	51.2	50.8	51.9	53.6	51.0	51.0	50.3
"Freon-214"	53.8	53.2	51.9	52.0	52.4	50.7	51.4	28.1
"Freon-216"	20.3	33.0	5.9	50.8	50.7	19.7	20.7	51.2
Perfluoro-dimethyl-cyclobutane	2.5	1.7	1.3	6.3	3.2	1.9	1.2	50.3

Note: All values are in percent by weight.
For those values which are more than 50, the actual solubility is more than the amount indicated.
For those values which are less than 50, the actual solubility is slightly less than the amount indicated.
All of the solubilities for perfluorodimethylcyclobutane were done at approximately 85°F.

"FREON" PRODUCTS LABORATORY

E. I. du Pont de Nemours & Company

TECHNICAL REPORT

Report Number: KSS-5103A

Subject: Solubility of Some Combinations of Stable Fluids and Refrigerants and their Stability with Metals

Company: SKF Industries, Inc.
King of Prussia
Pennsylvania

By: H. M. Parmelee Date: April 28, 1965

Object: To determine the solubility of "Freon-113" and "Freon-114B2" refrigerants, in four lubricants, at atmospheric pressure and 150°F and 250°F. The lubricants were: Mobil XRM-109F, Monsanto OS124, Union Carbide Ucon 50-HB-5100, and Du Pont PR-143.

A second object was to run sealed-tube stability tests with the two refrigerants and four lubricants in combination with each of six metals for 24 hours at 500°F. The metals were stainless steel 304, Inconel X750, Am 355 steel, SKF #2 Steel, Monel 505 and silver.

Results:

"Freon-114B2" was more soluble than "Freon-113" in all of the oils at corresponding temperatures. The order of decreasing solubility for "Freon-114B2" in the lubricants was as follows: 50-HB-5100 > XRM-109F > OS124 > PR143. The order for "Freon-113" was: XRM-109F > 50-HB-5100 > OS124 > PR143. The data are presented in Table I.

PR143 was the only lubricant which was reasonably stable with the two refrigerants and six metals at 500°F. The liquids in the PR143 tests were clear, though sometimes turned to an amber color, with tarnish on the metal and a thin, loose, dark film on the glass wall. All of the other lubricants either burst the tubes or turned to a dark tarry mass. The data are presented in Tables II to V.

Comments:

The solubilities were determined by placing about 100 ml of the lubricant in a 250 ml tube about 10" long. The tube was equipped with a sintered glass sparger and a gas outlet. The apparatus was tared, the oil weighed and the apparatus was immersed in an oil bath at controlled temperatures. The refrigerants were fed by gravity from an elevated reservoir, through a control valve, then to a flash coil located in the bath, and the resulting vapor bubbled through the lubricant to the air. At periodic intervals the apparatus was removed from the bath, cooled and weighed. When the weight was constant, the weight of refrigerant dissolved was determined by difference. (The small amount of gas remaining in the vapor was neglected.)

The sealed-tube tests were carried out in our usual manner except that the refrigerants were introduced as liquids and the contents were frozen in N_2 before degassing.

All of the metal test pieces except the silver were supplied by SKF and were segments of cylinders which were about 1" long. Each segment appeared to contain about 30° of arc of a cylinder 1/2" in diameter. The pieces appeared to have been considerably heated during cutting, as judged by the blue "burned" appearance. For this reason, the discolored parts were reground to give fresh clean metal surfaces. The silver was obtained from the Chambers Works PVI laboratory and was very pure silver, cut from material used by them in certain safety disks. It was polished with crocus cloth before use.

When the 500°F oven was opened to receive the cold tubes, the temperature fell to about 350°F. The 24 hour time was taken beginning when the temperature had returned to 500°F, about 4 hours after the tubes were placed in the oven.

TABLE I

The Solubility of "Freon-113" and "Freon-114B2" in Some Stable Lubricants

Oil	Refrigerant	Temperature (°F)	Weight % Refrigerant
OS124	"F-114B2"	150	31.0
	"	250	7.2
Ucon 50-HB-5100	"	150	54.9
"	"	250	10.8
XRM-109F	"	150	37.5
"	"	250	11.9
PR-143	"	150	26.1
"	"	250	5.1
OS124	"F-113"	150	23.0
"	"	250	5.2
Ucon 50-HB-5100	"	150	38.5
"	"	250	6.3
XRM-109F	"	150	40.1
"	"	250	8.3
PR-143	"	150	17.1
"	"	250	3.0

TABLE II

Sealed-Tube Tests on Mobil XRM-109F

Systems: 2 Ml XRM-109F + 2 Ml Refrigerant + Metal
Conditions: 500°F for 24 hours

Refrigerant	Metal	Appearance After 24 Hrs. Liquid	Metal
"F-113"	SS 304	Burst	--
"F-114B2"	"	"	--
"F-113"	Inconel X-750	Burst	--
"F-114B2"	"	Tarry	Black
"F-113"	Am 355	Burst	--
"F-114B2"	"	"	--
"F-113"	SKF #2	Burst	--
"F-114B2"	"	"	--
"F-113"	Monel 505	Tarry	Black
"F-114B2"	"	"	"
"F-113"	Silver	Tarry	Black
"F-114B2"	"	"	"

- - - -

TABLE III

Sealed-Tube Tests on Monsanto OS124

Systems: 2 Ml Monsanto OS124 + 2 Ml Refrigerant + Metal
Conditions: 500°F for 24 hours

Refrigerant	Metal	Appearance after 24 Hrs. Liquid	Metal
"F-113"	SS 304	Very dark with solids	Dark
"F-114B2"	"	Burst	
"F-113"	Inconel X-750	Very dark with solids	"
"F-114B2"	"	" " " "	"
"F-113"	Am 355	Very dark with solids	"
"F-114B2"	"	Black	"
"F-113"	SKF #2	Red Brown Tar Solids	Brown
"F-114B2"	"	Tarry	"
"F-113"	Monel 505	Tarry	Dark
"F-115B2"	"	"	"
"F-113"	Silver	Very dark with solids	"
"F-114B2"	"	" " " "	"

APPENDIX III (Cont.)

TABLE IV

Sealed-Tube Tests on Ucon 50-HB-5100

Systems: 2 Ml Ucon 50-HB-5100 + 2 Ml Refrigerant + Metal
Conditions: 500°F for 24 Hours

Refrigerant	Metal	Appearance After 24 Hrs. Liquid	Metal
"F-113"	SS 304	Burst	--
"F-114B2"	"	"	--
"F-113"	Inconel X-750	"	--
"F-114B2"	"	"	--
"F-113"	Am 355	"	--
"F-114B2"	"	"	--
"F-113"	SKF #2	"	--
"F-114B2"	"	"	--
"F-113"	Monel 505	"	--
"F-114B2"	"	"	--
"F-113"	Silver	"	--
"F-114B2"	"	"	--

- - - - - - - - - -

TABLE V

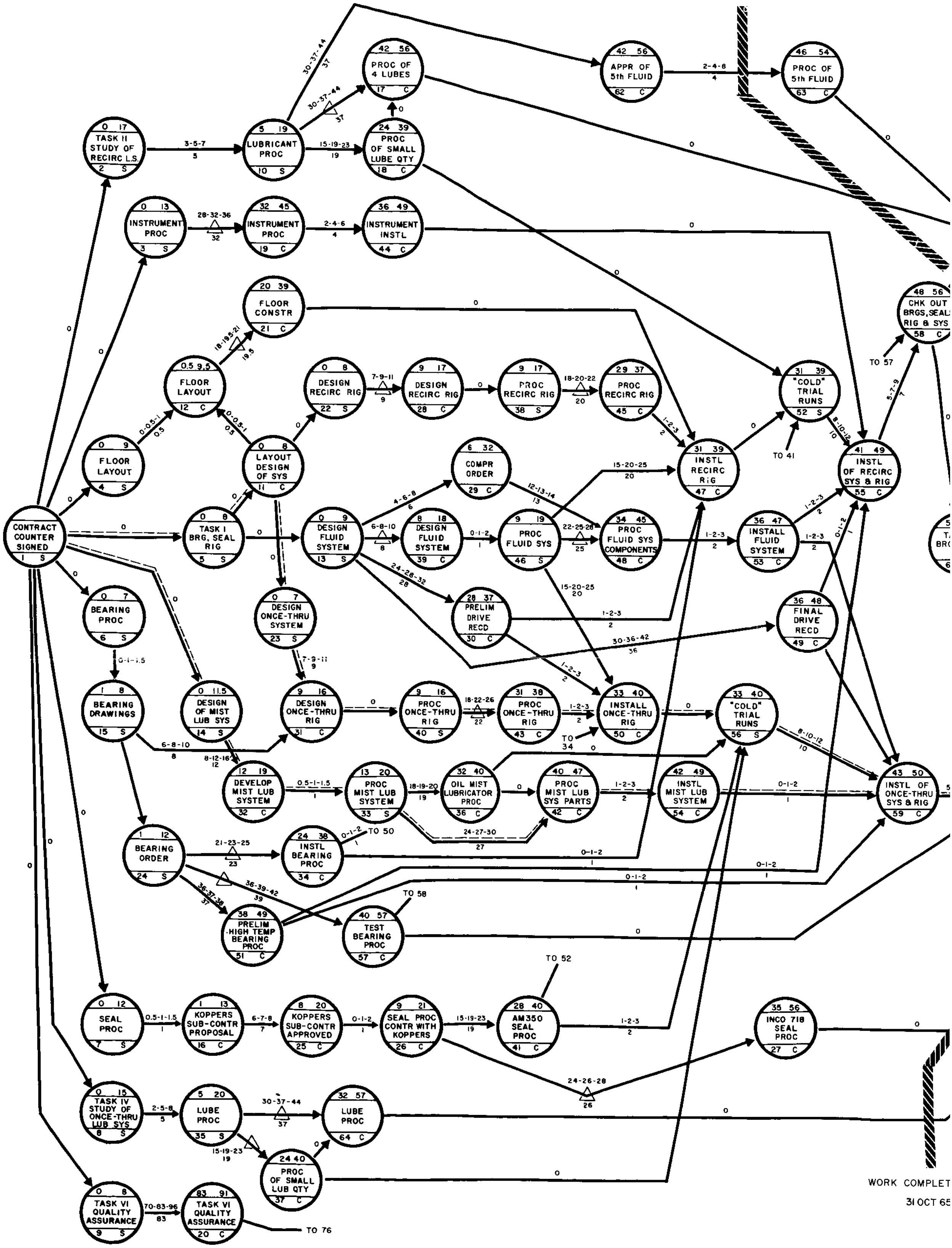
Sealed-Tube Tests on Du Pont PR-143

Systems: 2 ML Du Pont PR-143 + 2 Ml Refrigerant + Metal
Conditions: 500°F for 24 Hours

Refrigerant	Metal	Appearance After 24 Hrs. Liquid	Metal
"F-113"	SS 304	Clear - Sl. Film on Glass	Tarnished
"F-114B2"	"	Amber - " " " "	"
"F-113"	Inconel X-750	Clear Amber - Film	Bright
"F-114B2"	"	" " "	Tarnish
"F-113"	Am 355	Clear	Sl. Tarnish
"F-114B2"	"	" - Film on Glass	"
"F-113"	SKF #2	Clear	Brown Tarnish
"F-114B2"	"	" - Film on Glass	Gray Tarnish
"F-113"	Monel 505	Clear	Dark Gray Tarnish
"F-114B2"	"	" - Film on Glass	" " "
"F-113"	Silver	Clear - Film on Glass	Sl. Br. Tarnish
"F-114B2"	"	" " " "	Brown Tarnish

ENCLOSURE 1

PERT NETWORK OF SST LUBRICATION SYSTEM INVESTIGATION



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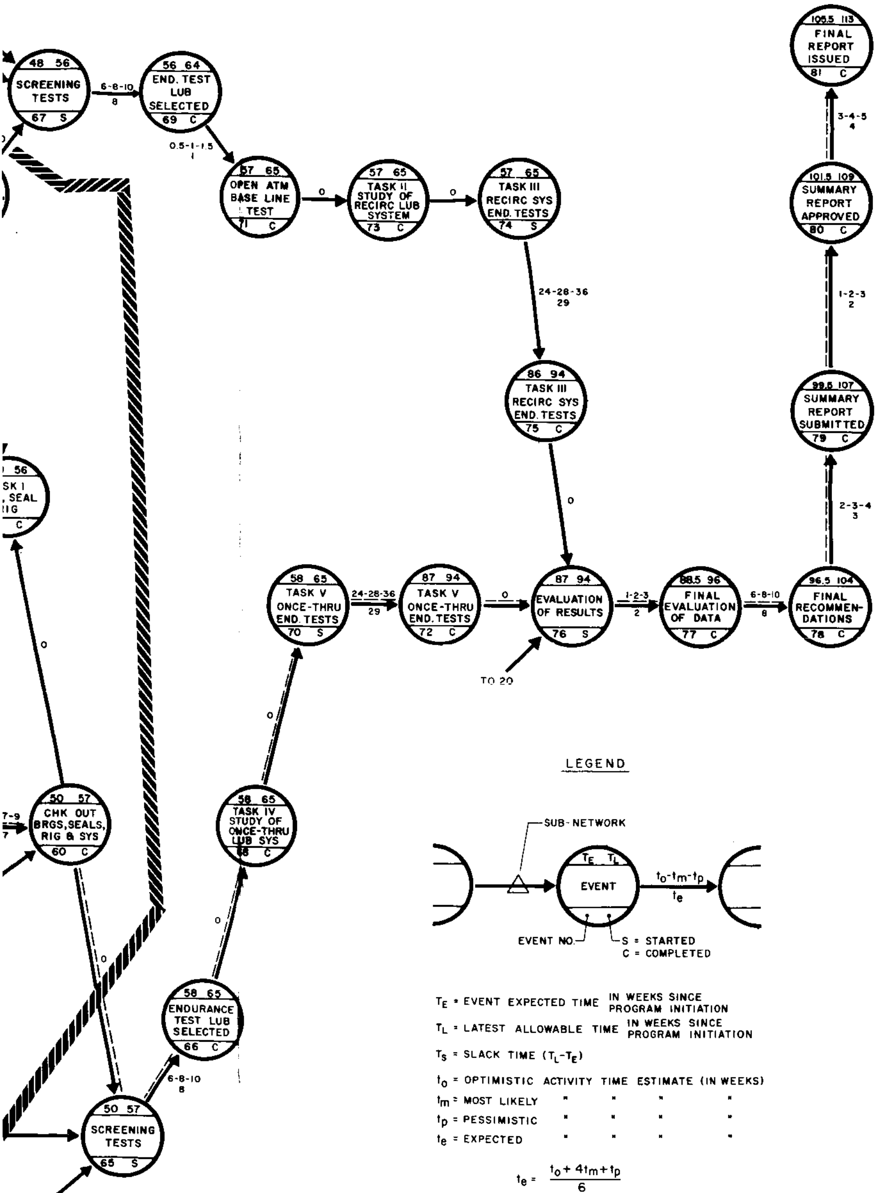
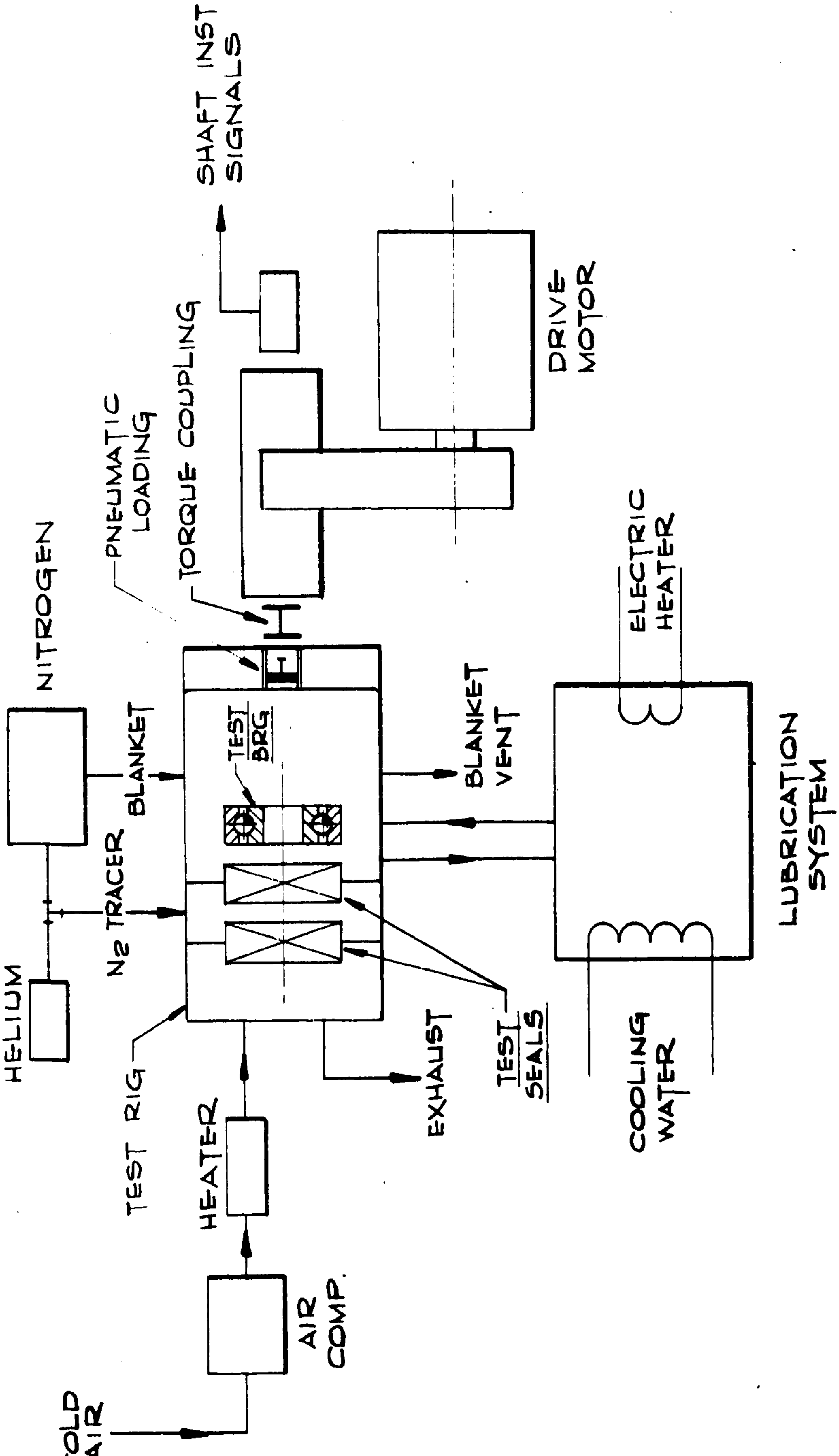

48 56 SCREENING TESTS 67 S
6-8-10 8
56 64 END. TEST LUB SELECTED 69 C
0.5-1-1.5 1
57 65 OPEN ATM BASE LINE TEST 71 C
57 65 TASK II STUDY OF RECIRC LUB SYSTEM 73 C
57 65 TASK III RECIRC SYS END. TESTS 74 S
24-28-36 29
86 94 TASK III RECIRC SYS END. TESTS 75 C
105.5 113 FINAL REPORT ISSUED 81 C
3-4-5 4
101.5 109 SUMMARY REPORT APPROVED 80 C
1-2-3 2
99.5 107 SUMMARY REPORT SUBMITTED 79 C
2-3-4 3
58 65 TASK V ONCE-THRU END. TESTS 70 S
24-28-36 29
87 94 TASK V ONCE-THRU END. TESTS 72 C
87 94 EVALUATION OF RESULTS 76 S
TO 20
1-2-3 2
88.5 96 FINAL EVALUATION OF DATA 77 C
6-8-10 8
96.5 104 FINAL RECOMMEN-DATIONS 78 C
50 57 CHK OUT BRGS, SEALS, RIG & SYS 60 C
58 65 TASK IV STUDY OF ONCE-THRU LUB SYS 68 C
58 65 ENDURANCE TEST LUB SELECTED 66 C
6-8-10 8
50 57 SCREENING TESTS 65 S
LEGEND
SUB-NETWORK
T_E T_L EVENT
t_o-t_m-t_p t_e
EVENT NO.
S = STARTED
C = COMPLETED
T_E = EVENT EXPECTED TIME IN WEEKS SINCE PROGRAM INITIATION
T_L = LATEST ALLOWABLE TIME IN WEEKS SINCE PROGRAM INITIATION
T_S = SLACK TIME (T_L-T_E)
t_o = OPTIMISTIC ACTIVITY TIME ESTIMATE (IN WEEKS)
t_m = MOST LIKELY " " " "
t_p = PESSIMISTIC " " " "
t_e = EXPECTED " " " "
$t_e = \frac{t_o + 4t_m + t_p}{6}$

ED TO

2#

ENCLOSURE 2

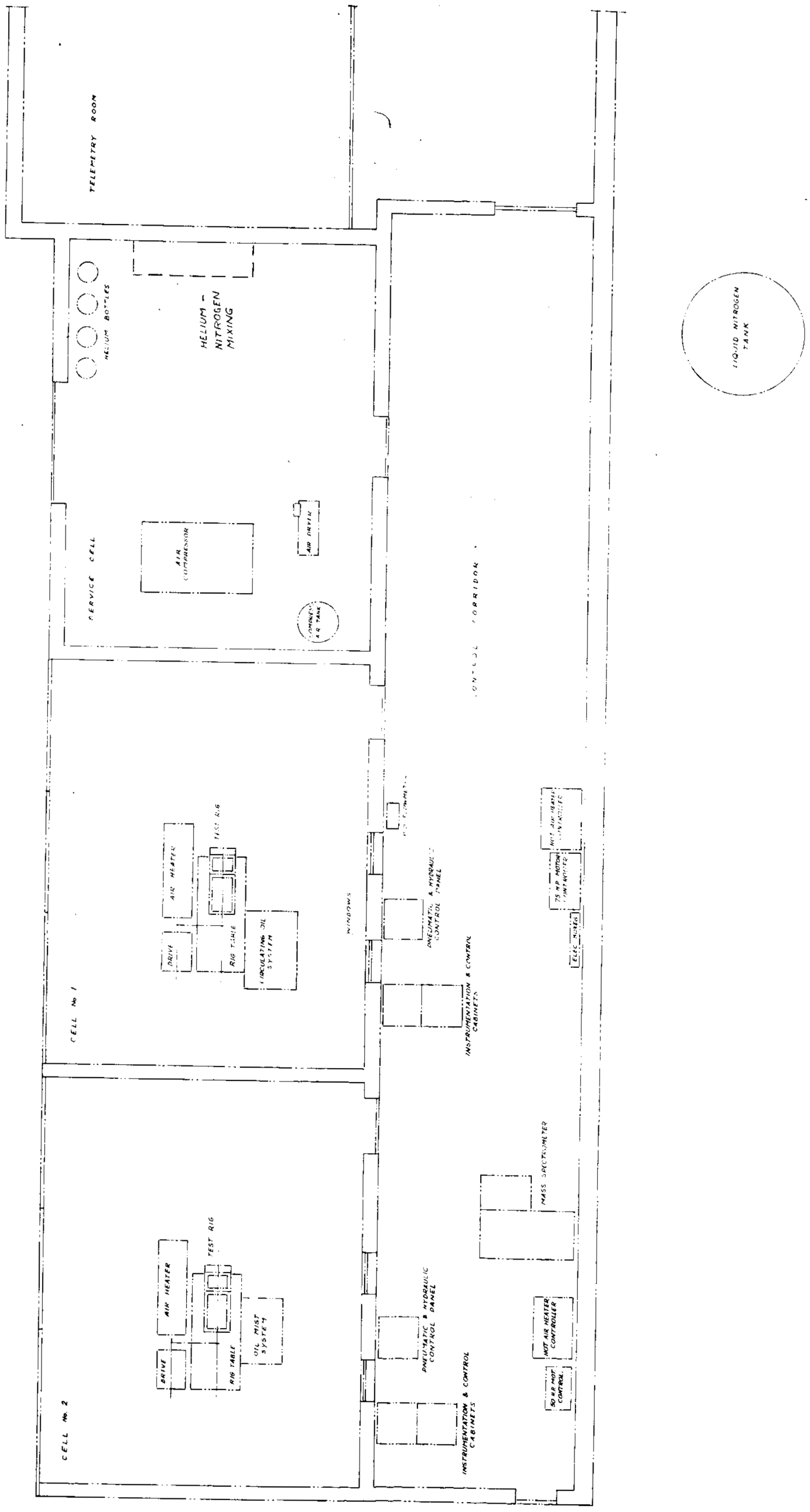
GENERAL TEST RIG LAYOUT SCHEMATIC



RESEARCH LABORATORY SKF INDUSTRIES, INC.

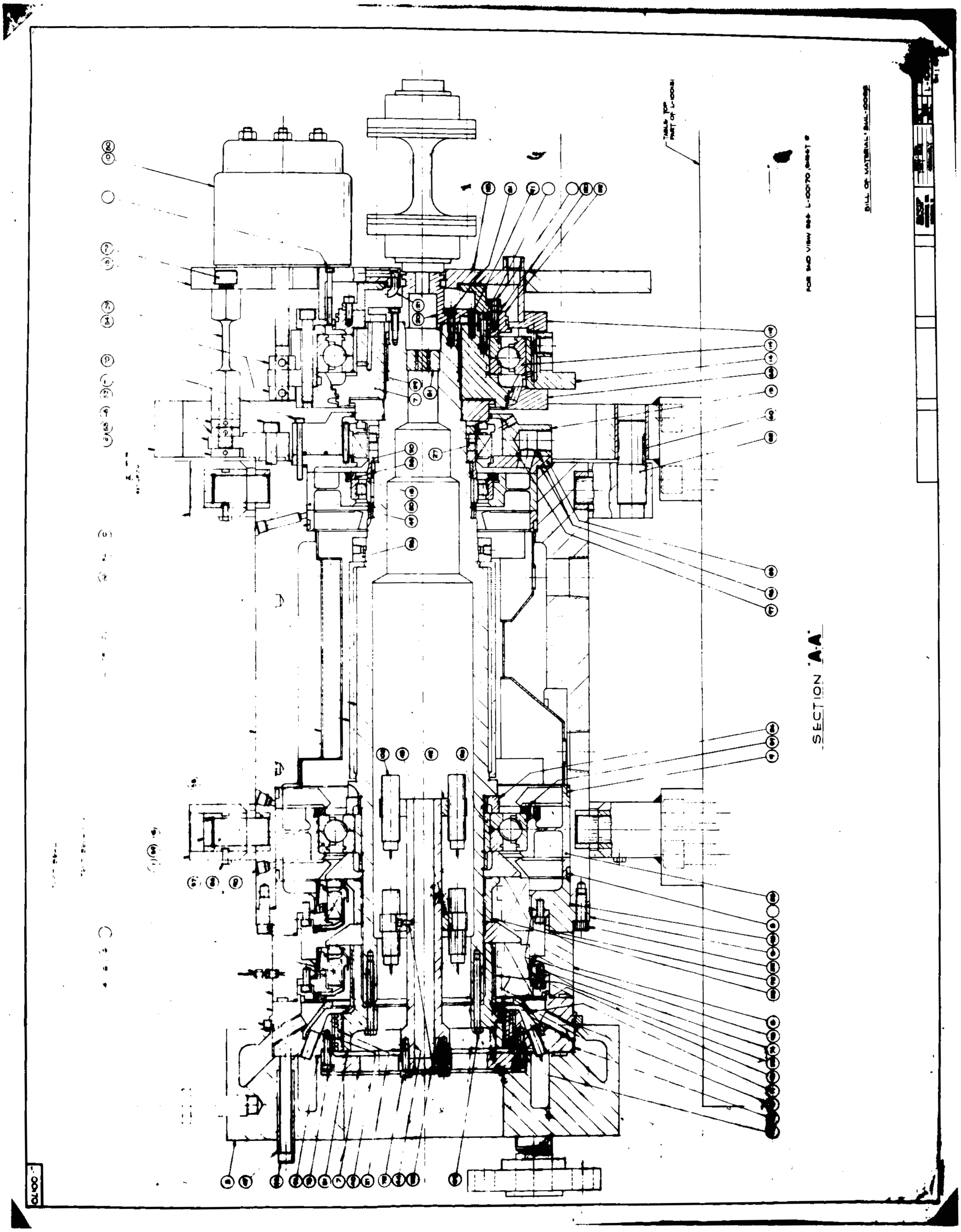
ENCLOSURE 3

TEST FACILITY - GENERAL LAYOUT



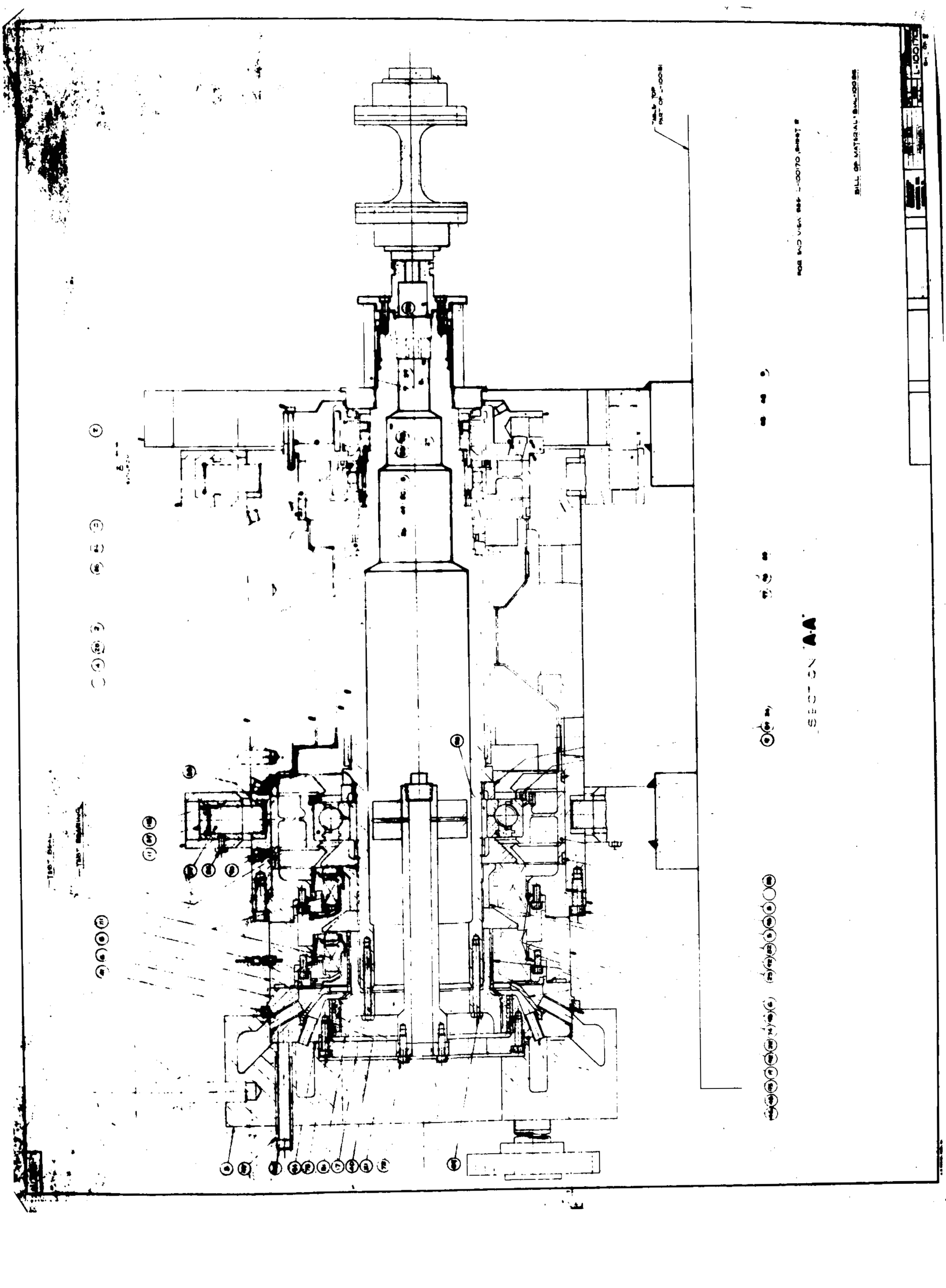
ENCLOSURE 4

TEST RIG ASSEMBLY - THRUST APPLIED THROUGH LOAD BEARING



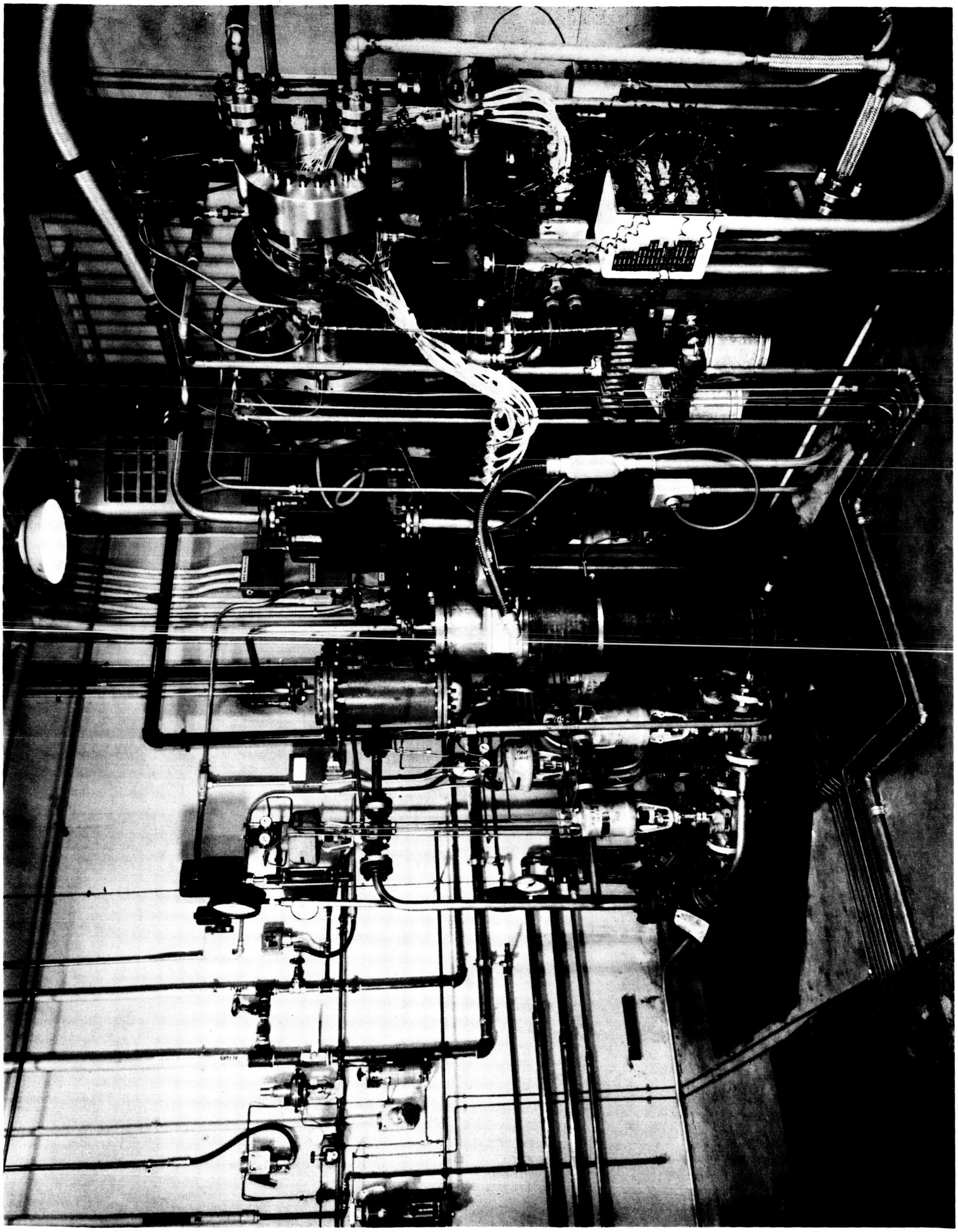
ENCLOSURE 5

TEST RIG ASSEMBLY - REVERSE DIRECTION PNEUMATIC LOAD



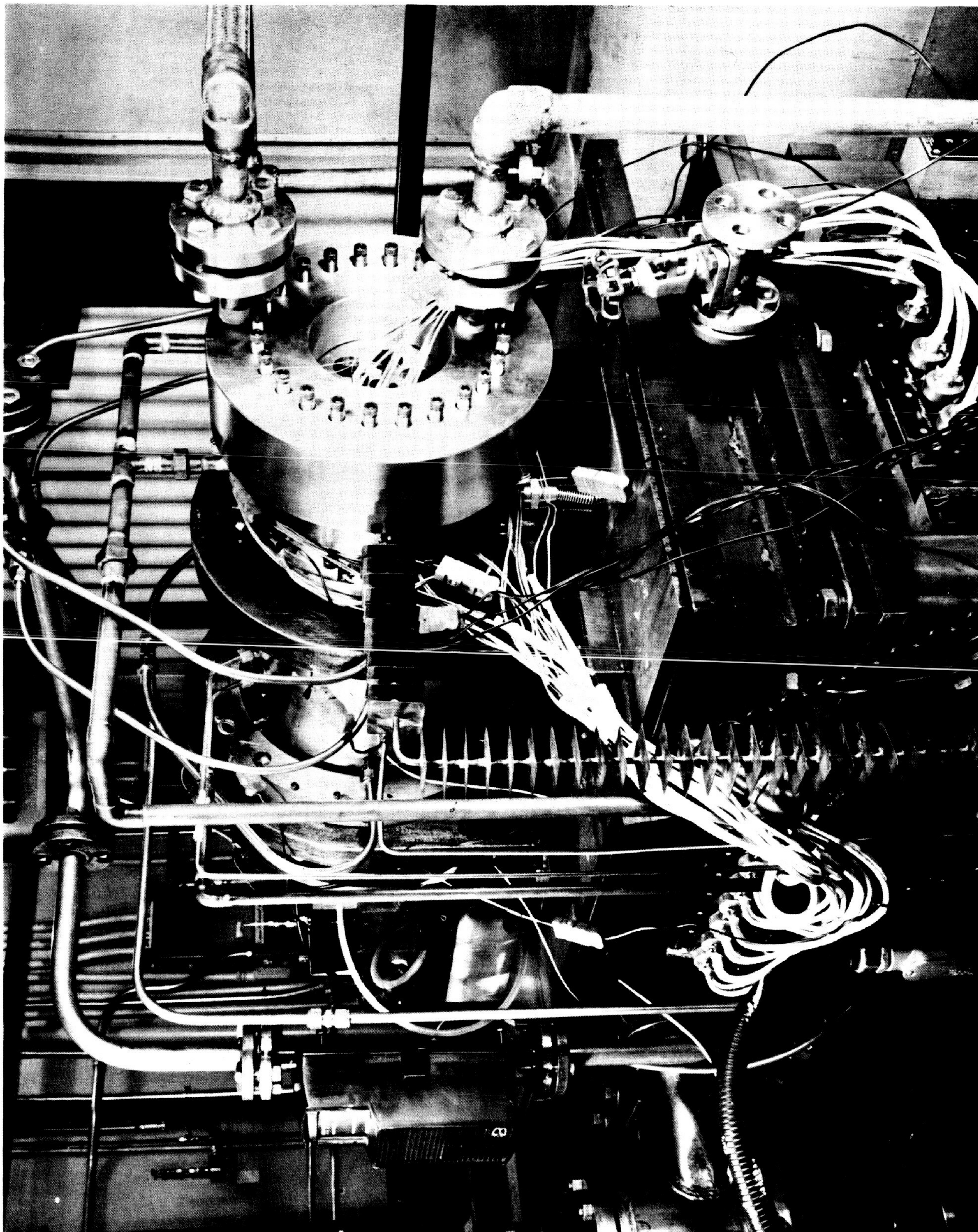
ENCLOSURE 6

GENERAL VIEW OF RECIRCULATING-OIL TEST CELL



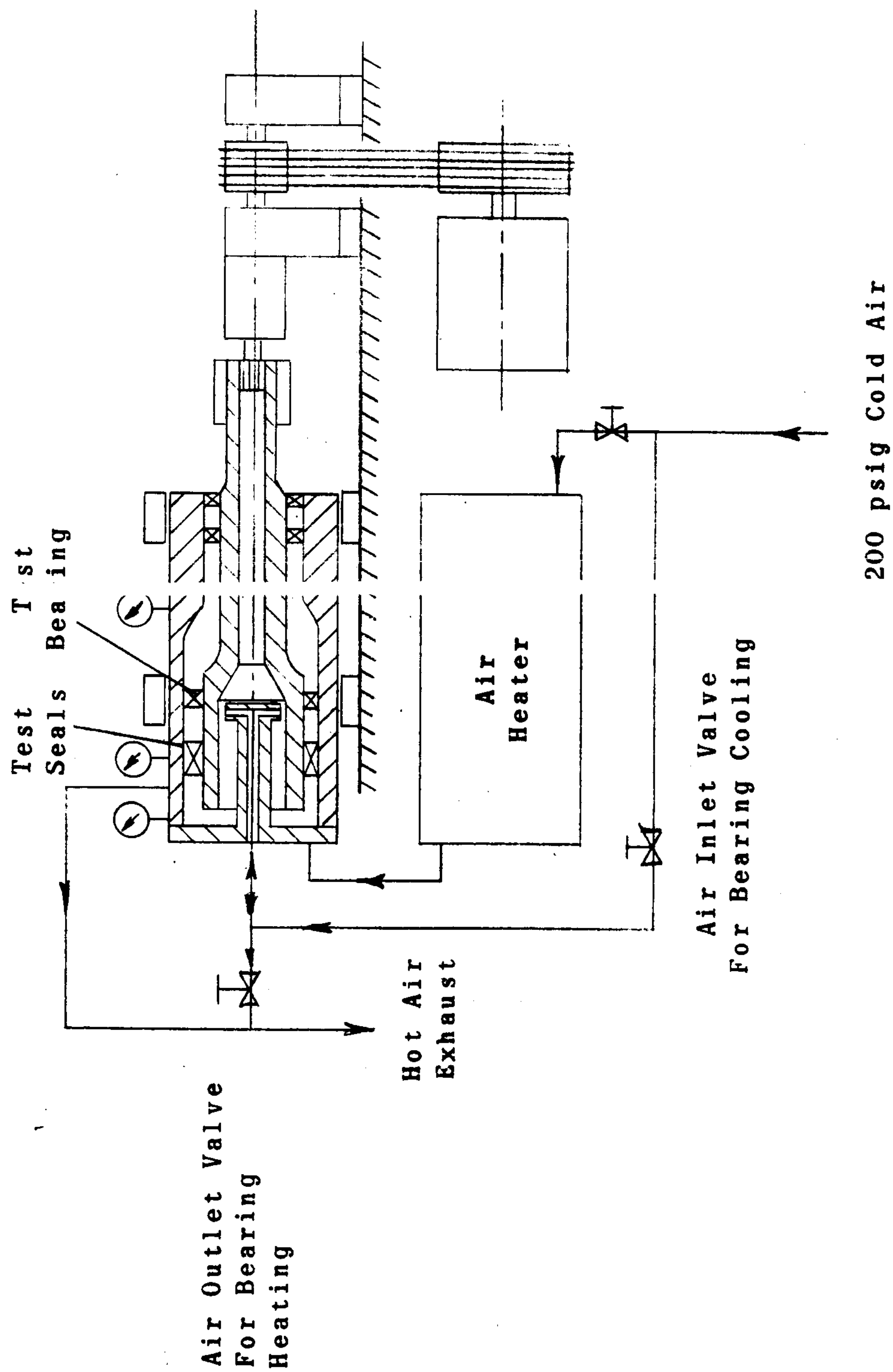
ENCLOSURE 7

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



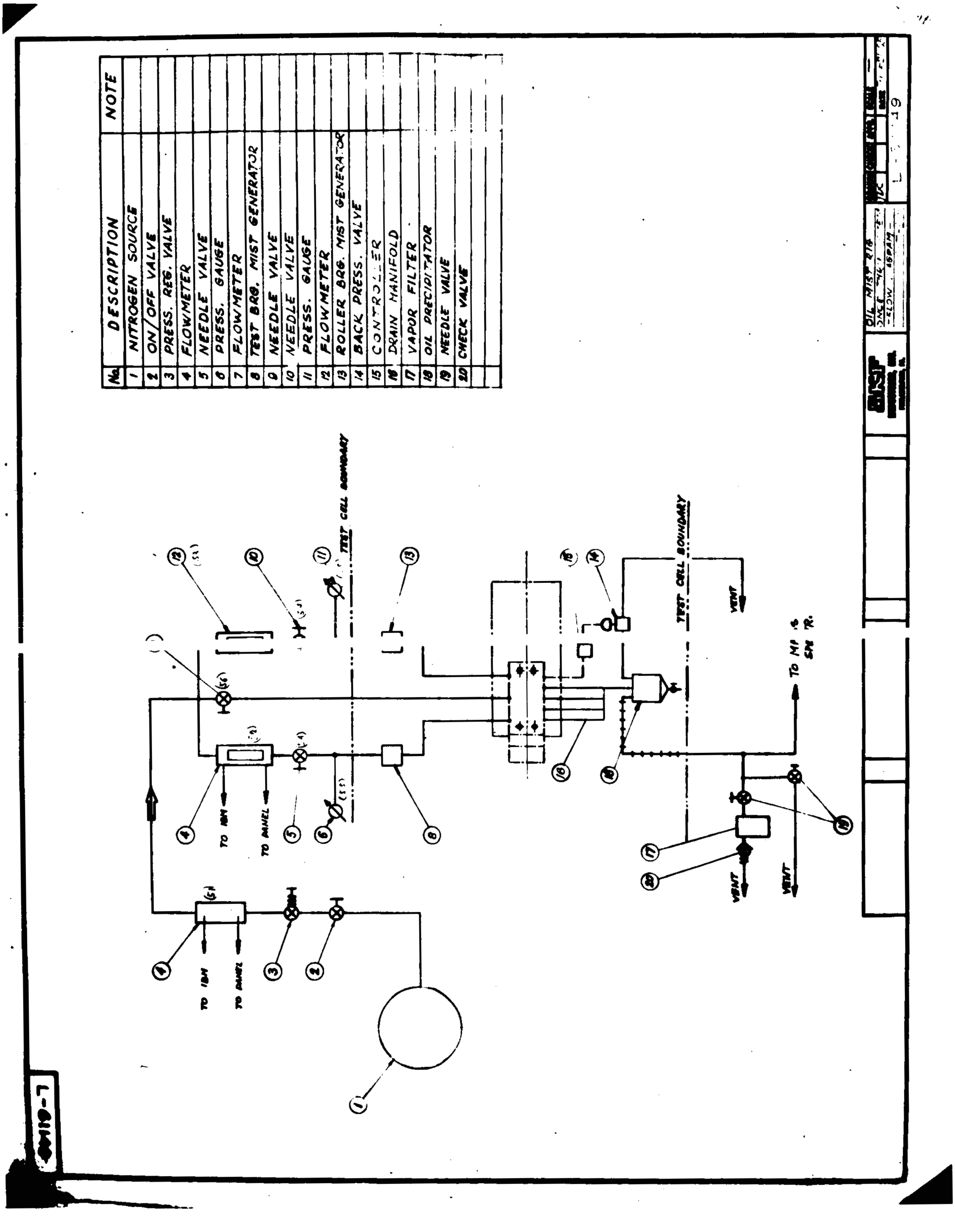
ENCLOSURE 8

PNEUMATIC REGULATION OF SHAFT TEMPERATURE



ENCLOSURE 9

OIL MIST TEST RIG - NITROGEN SUPPLY SYSTEM

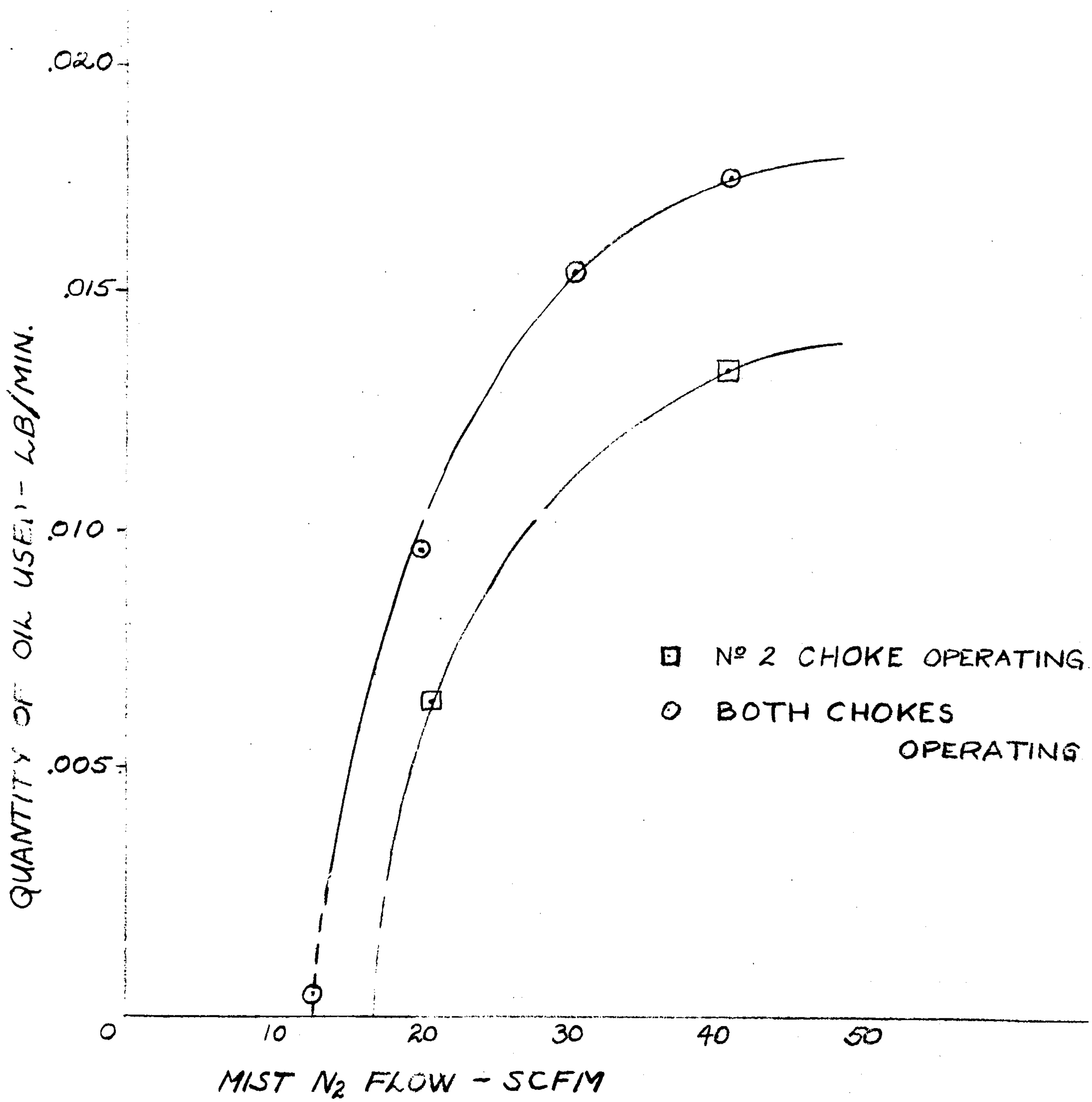


No.	DESCRIPTION	NOTE
1	NITROGEN SOURCE	
2	ON/OFF VALVE	
3	PRESS. REG. VALVE	
4	FLOWMETER	
5	NEEDLE VALVE	
6	PRESS. GAUGE	
7	FLOWMETER	
8	TEST BRG. MIST GENERATOR	
9	NEEDLE VALVE	
10	NEEDLE VALVE	
11	PRESS. GAUGE	
12	FLOWMETER	
13	ROLLER BRG. MIST GENERATOR	
14	BACK PRESS. VALVE	
15	CONTROLLER	
16	DRAIN MANIFOLD	
17	VAPOR FILTER	
18	OIL PRECIPITATOR	
19	NEEDLE VALVE	
20	CHECK VALVE	

ENCLOSURE 10

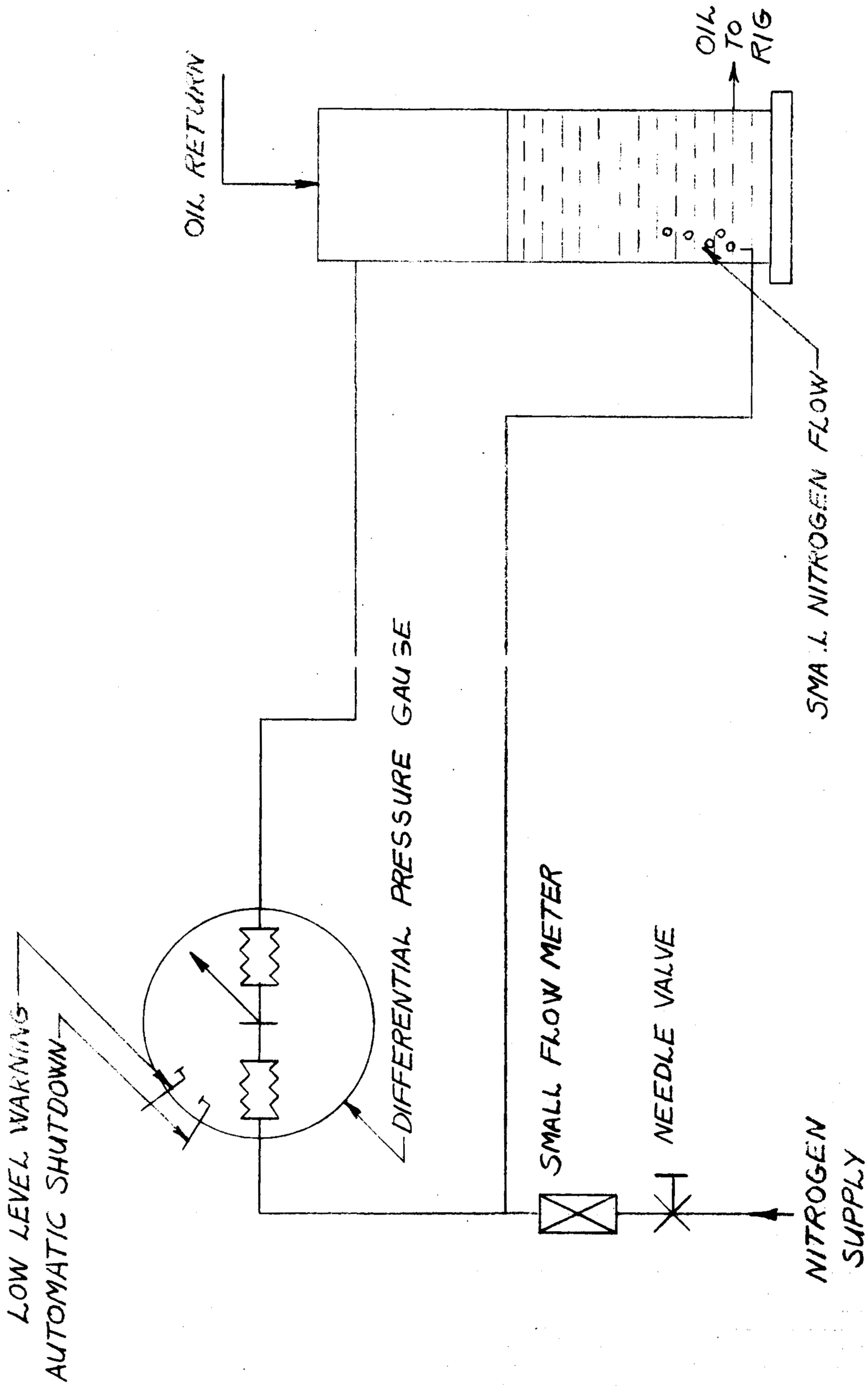
OIL MIST QUANTITY SUPPLIED BY MIST GENERATOR

Lubricant: Esso Turbo Oil 4040
Oil Reservoir Temperature: 70°F
Bearing Cavity Pressure: Atmosphere



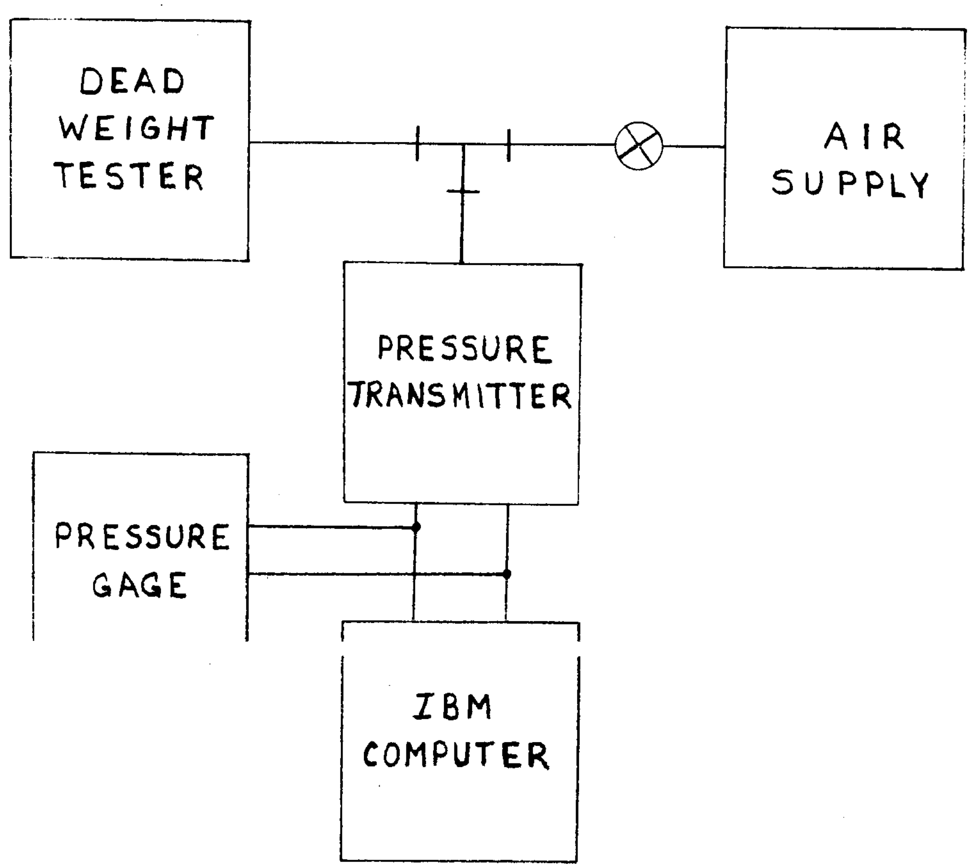
ENCLOSURE 11

REMOTE OIL LEVEL DETECTION



ENCLOSURE 12

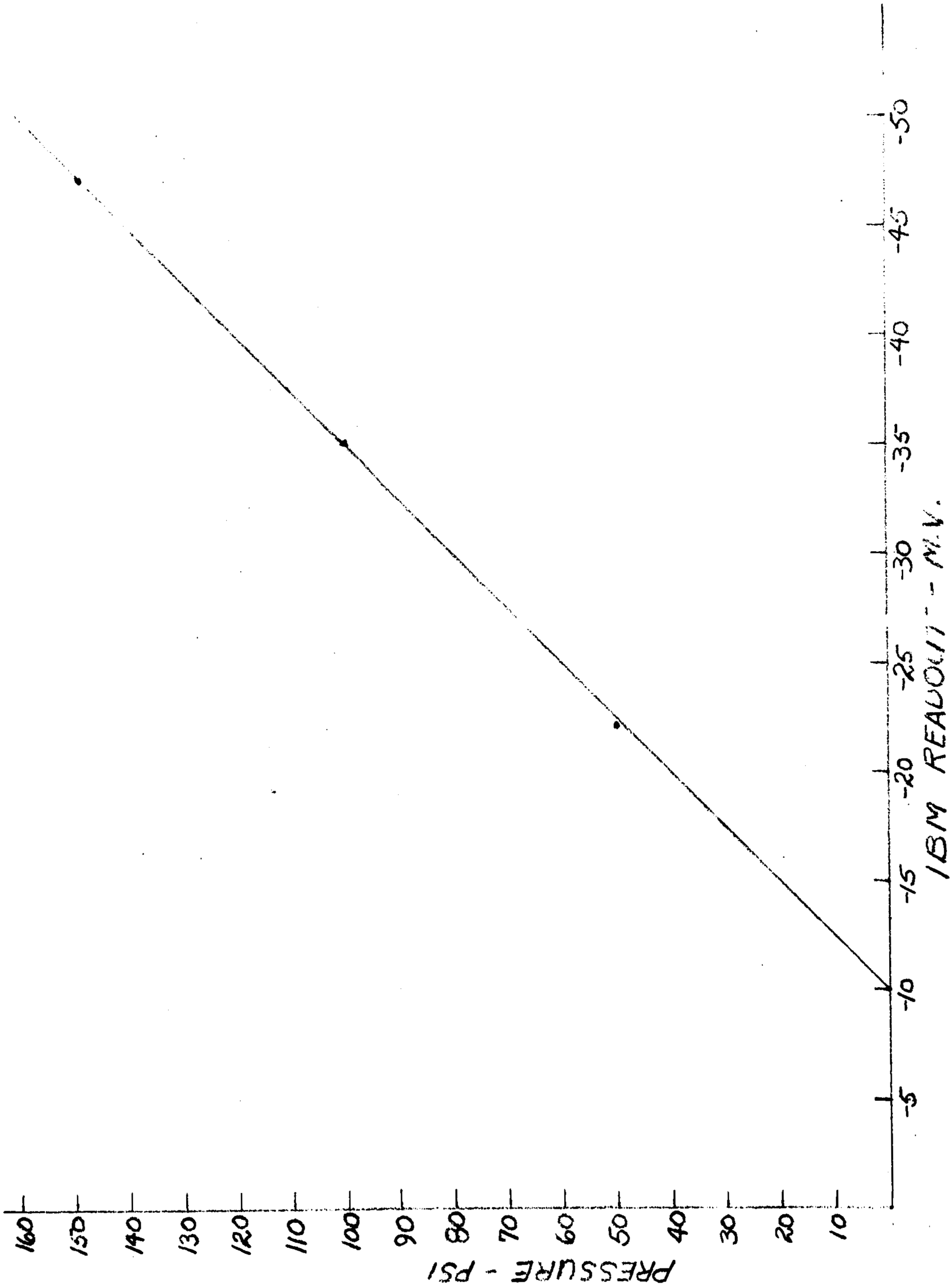
BLOCK DIAGRAM OF APPARATUS FOR SST PRESSURE CALIBRATION



ENCLOSURE 13

PRESSURE TRANSMITTER CALIBRATION - RECIRCULATING OIL RIG

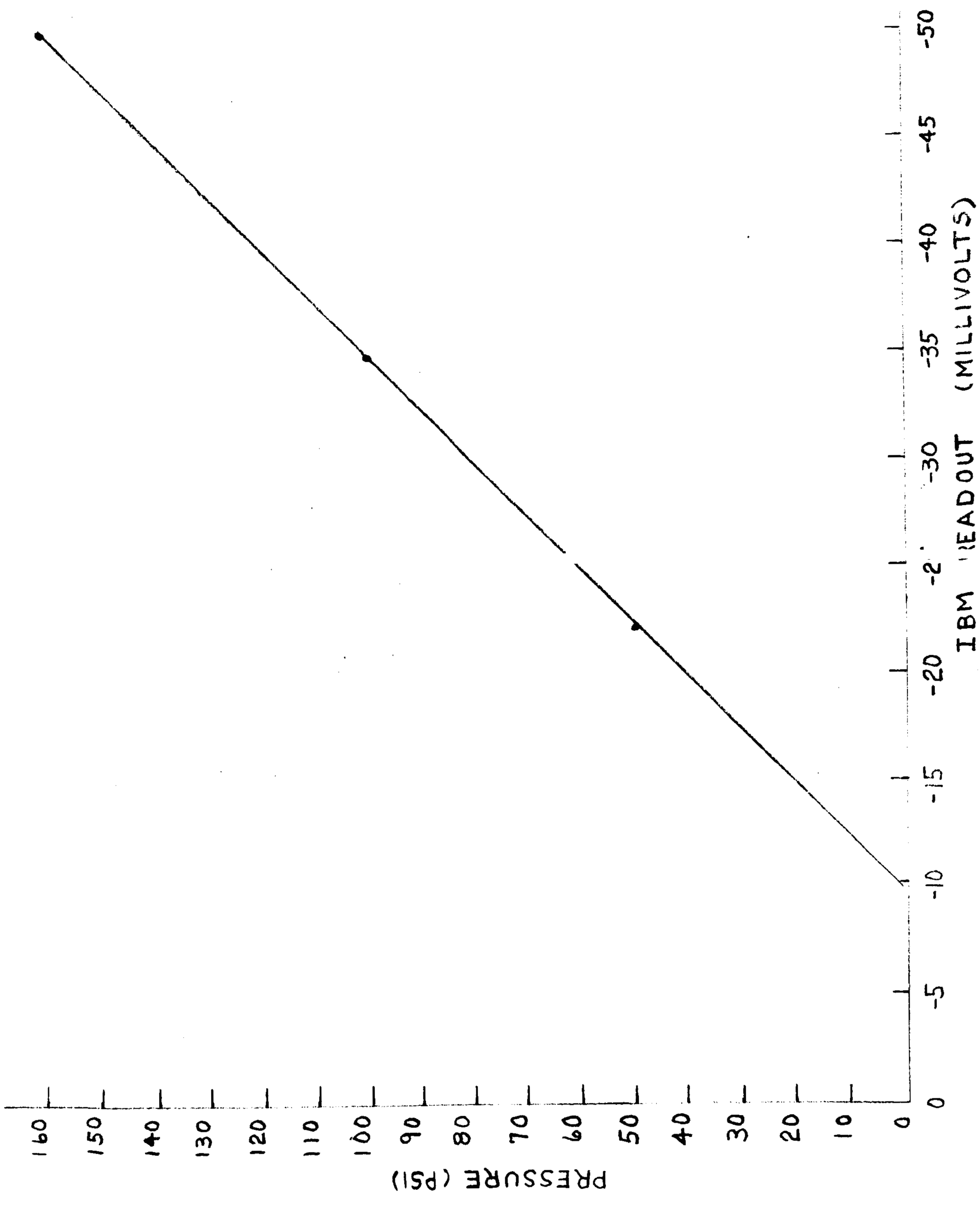
HOT AIR CHAMBER IBM POINT NO. 275



ENCLOSURE 14

PRESSURE TRANSMITTER CALIBRATION - RECIRCULATING OIL RIG

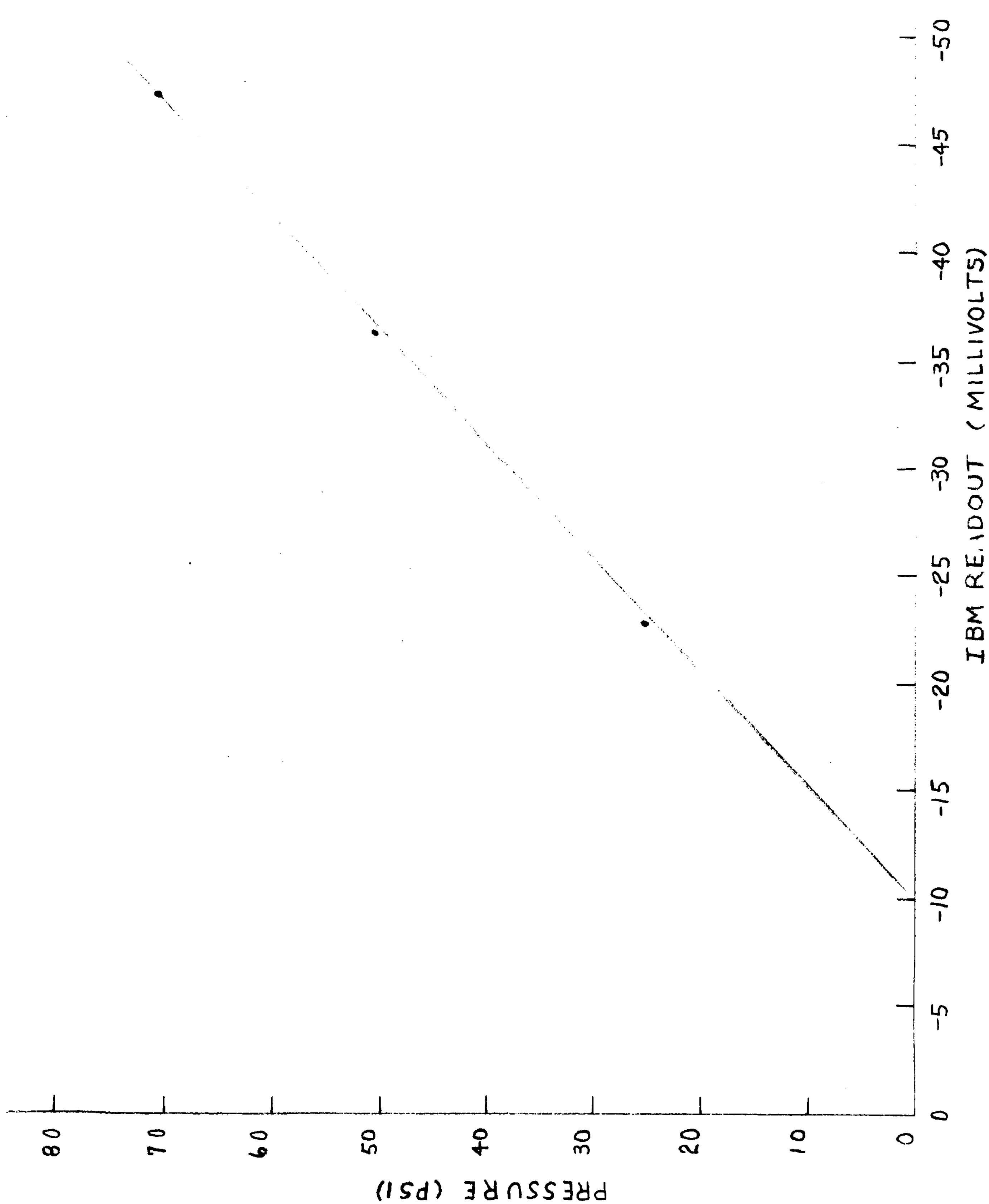
INTER SEAL CAVITY - IBM POINT NO. 273



ENCLOSURE 15

PRESSURE TRANSMITTER CALIBRATION - RECIRCULATING OIL RIG

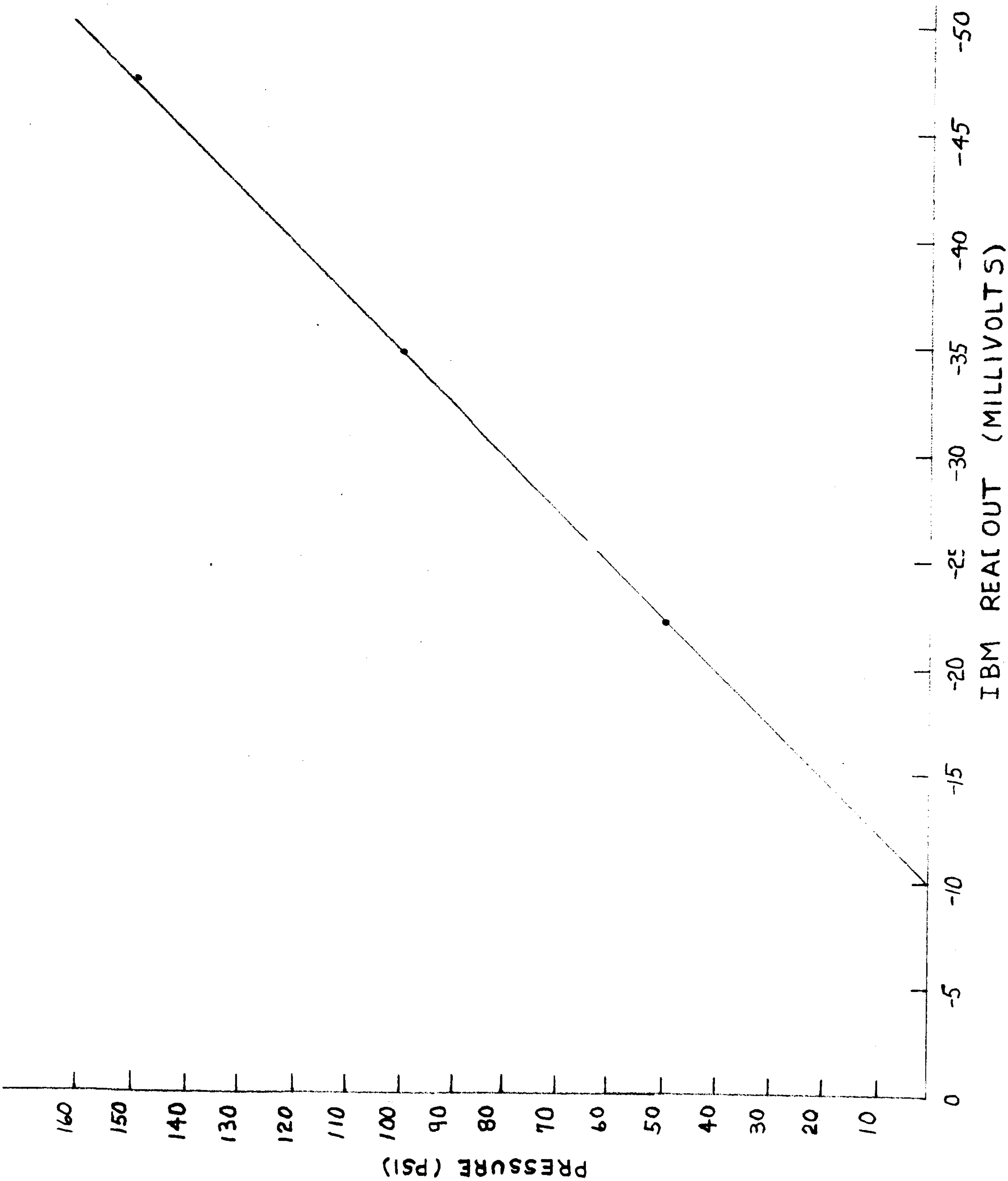
TEST BEARING CAVITY IBM POINT NO. 274



ENCLOSURE 16

PRESSURE TRANSMITTER CALIBRATION - OIL MIST RIG

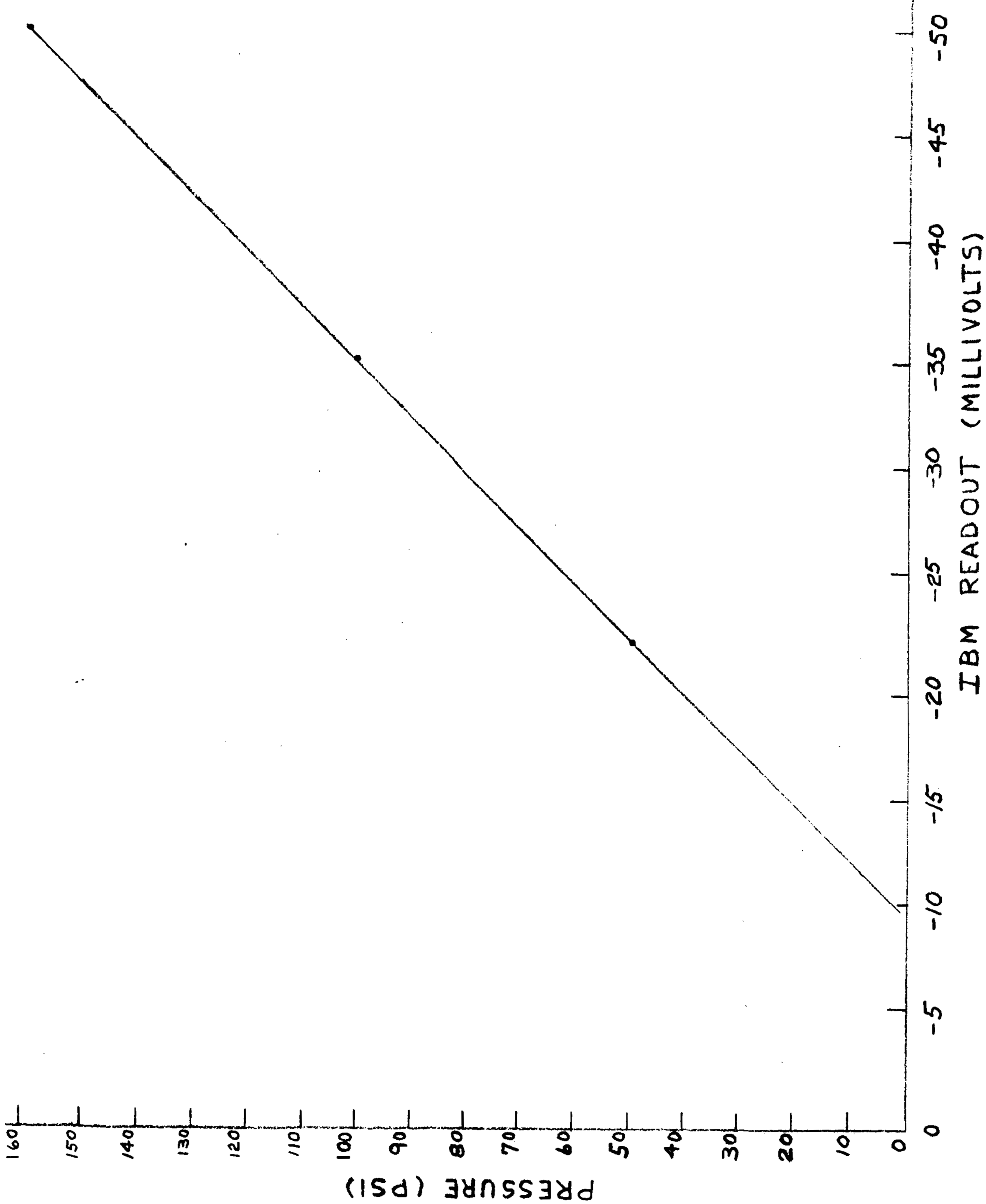
HOT AIR CHAMBER IBM POINT NO. 252



ENCLOSURE 17

PRESSURE TRANSMITTER CALIBRATION - OIL MIST RIG

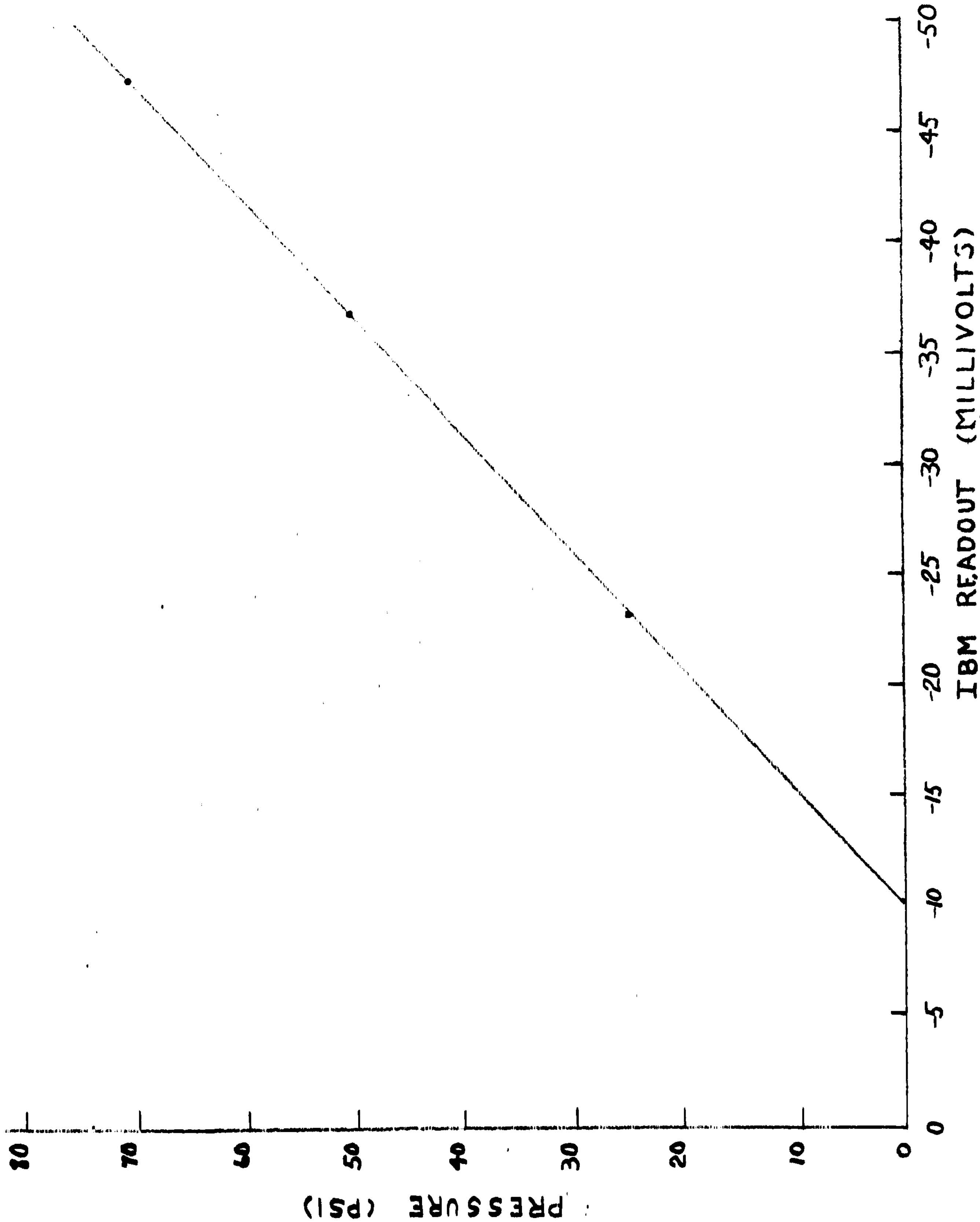
INTER SEAL CAVITY IBM POINT NO. 250



ENCLOSURE 18

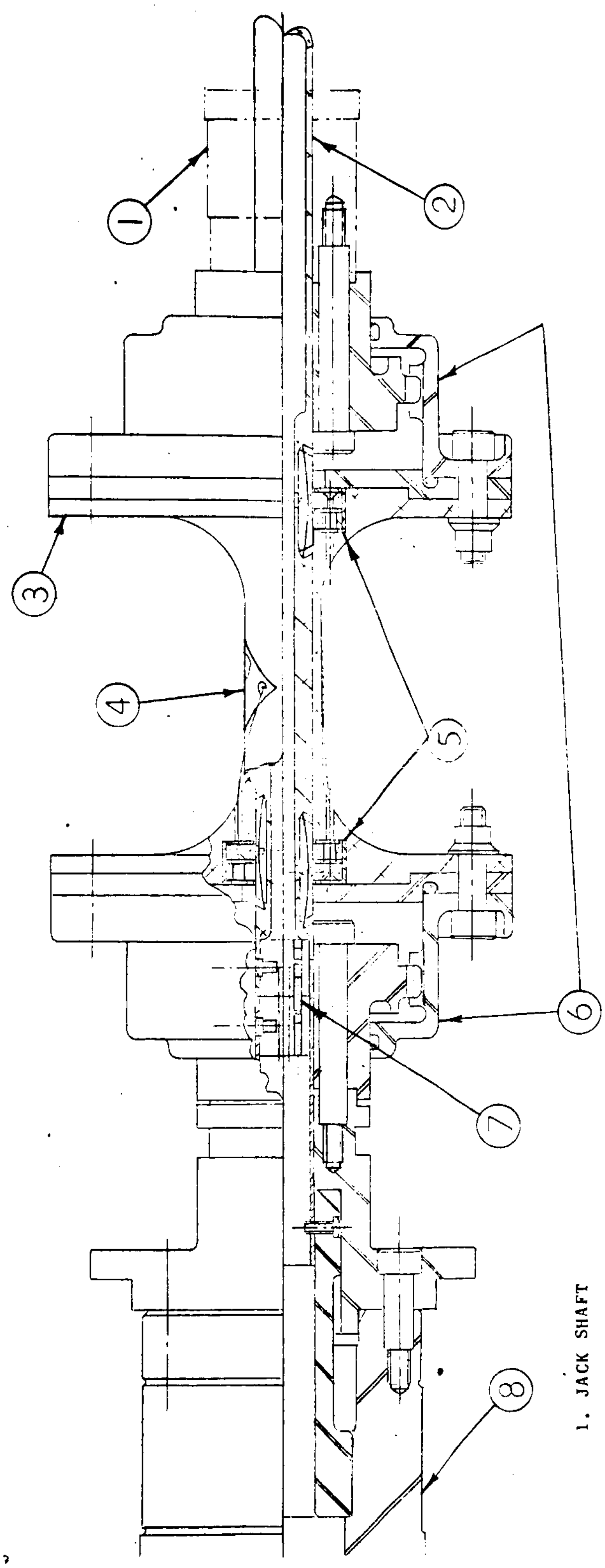
PRESSURE TRANSMITTER CALIBRATION - OIL MIST RIG

TEST BEARING CAVITY IBM POINT NO. 251



ENCLOSURE 19

TORQUE SENSOR ASSEMBLY



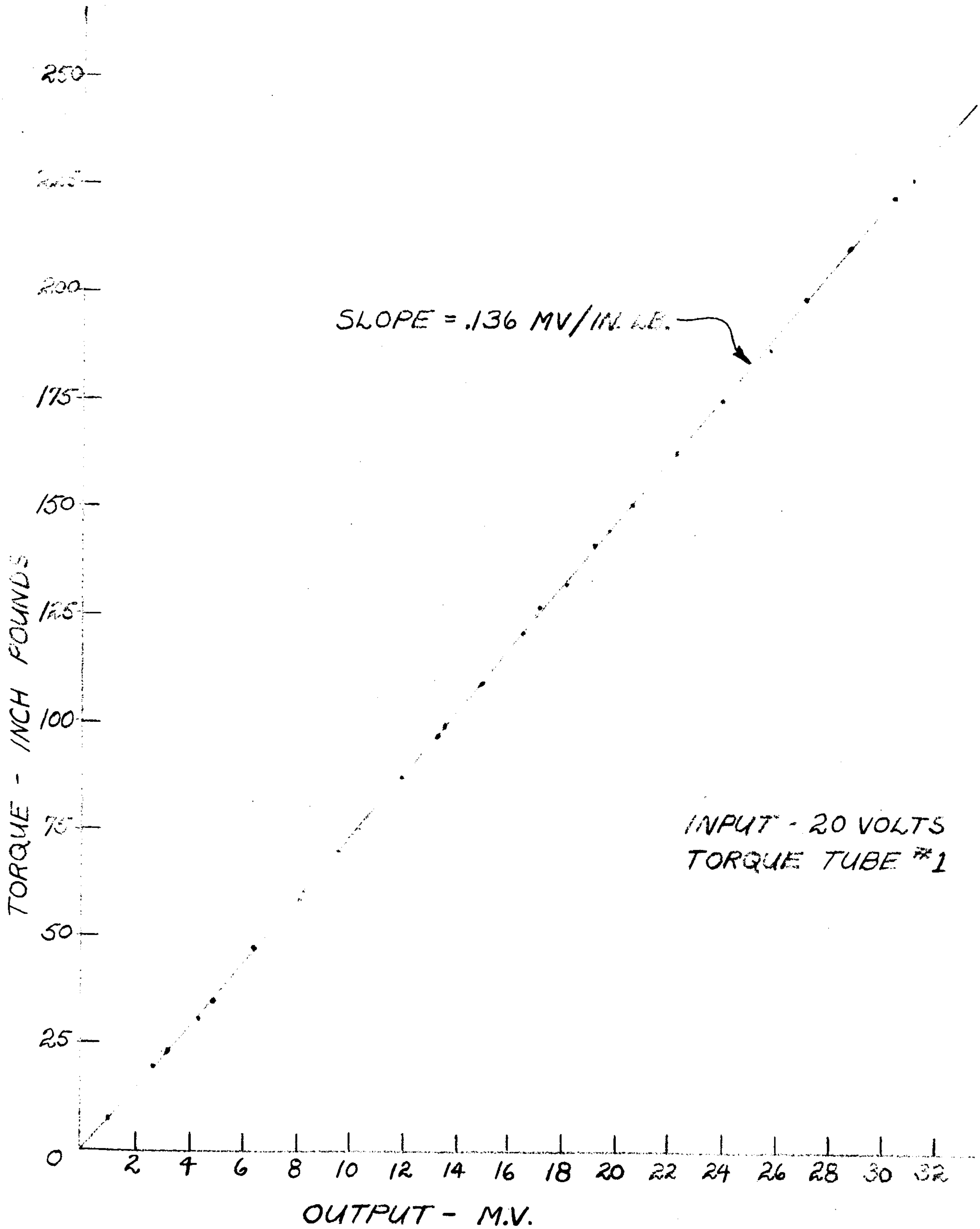
ASSEMBLY OF TORQUE SENSOR

1. JACK SHAFT
2. SLIP RING PROBE UNIT
3. ALUMINUM SPOOL TORQUE SENSOR
4. TWO-GAGE STRAIN ROSETTE
5. STRAIN GAGE CONTACTOR RINGS
6. KOPPERS GEAR COUPLINGS
7. SHAFT THERMOCOUPLE 12 WAY CONNECTOR
8. TEST RIG SHAFT

ENCLOSURE 20

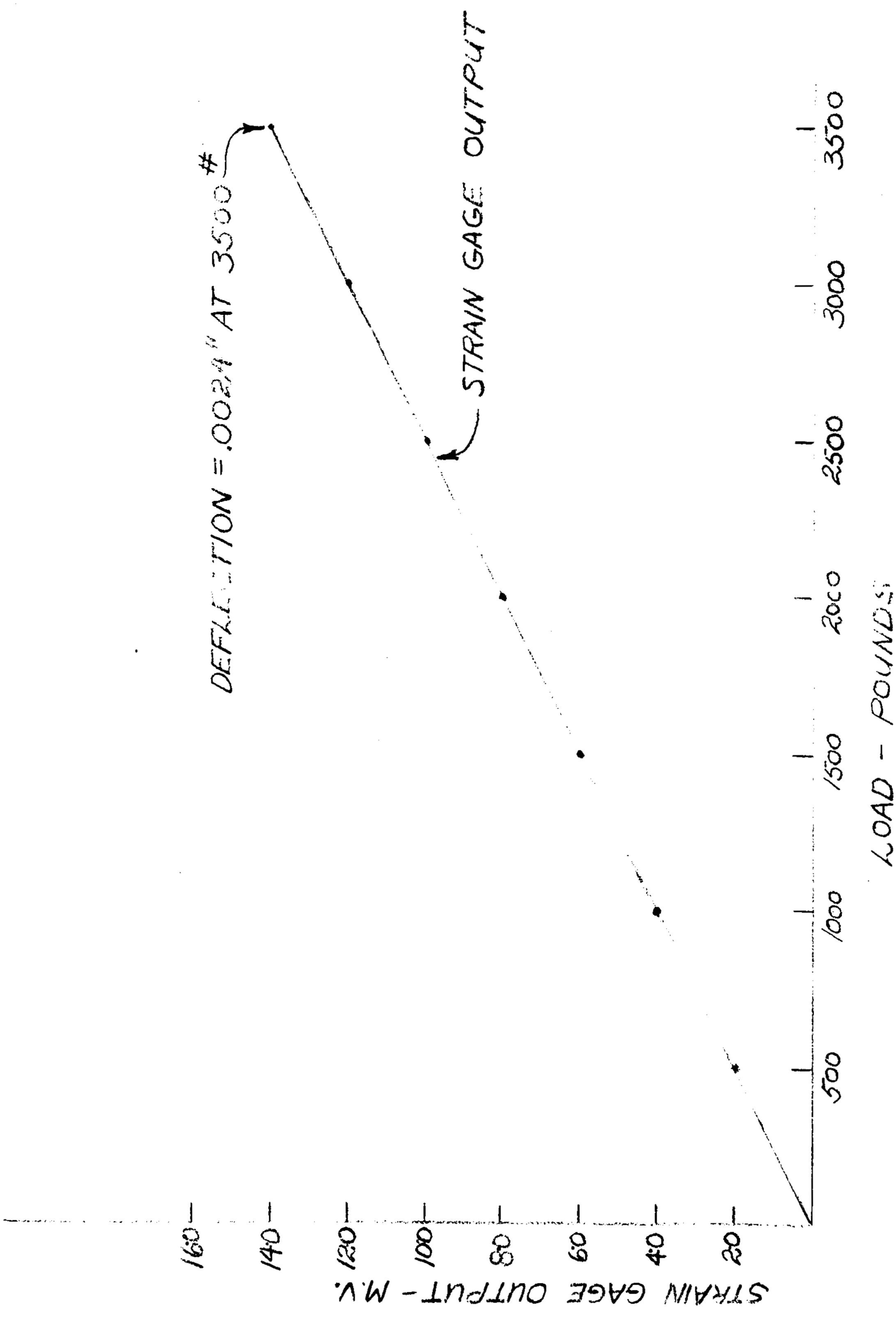
TORQUE SENSOR CALIBRATION

SPOOL NO. 1

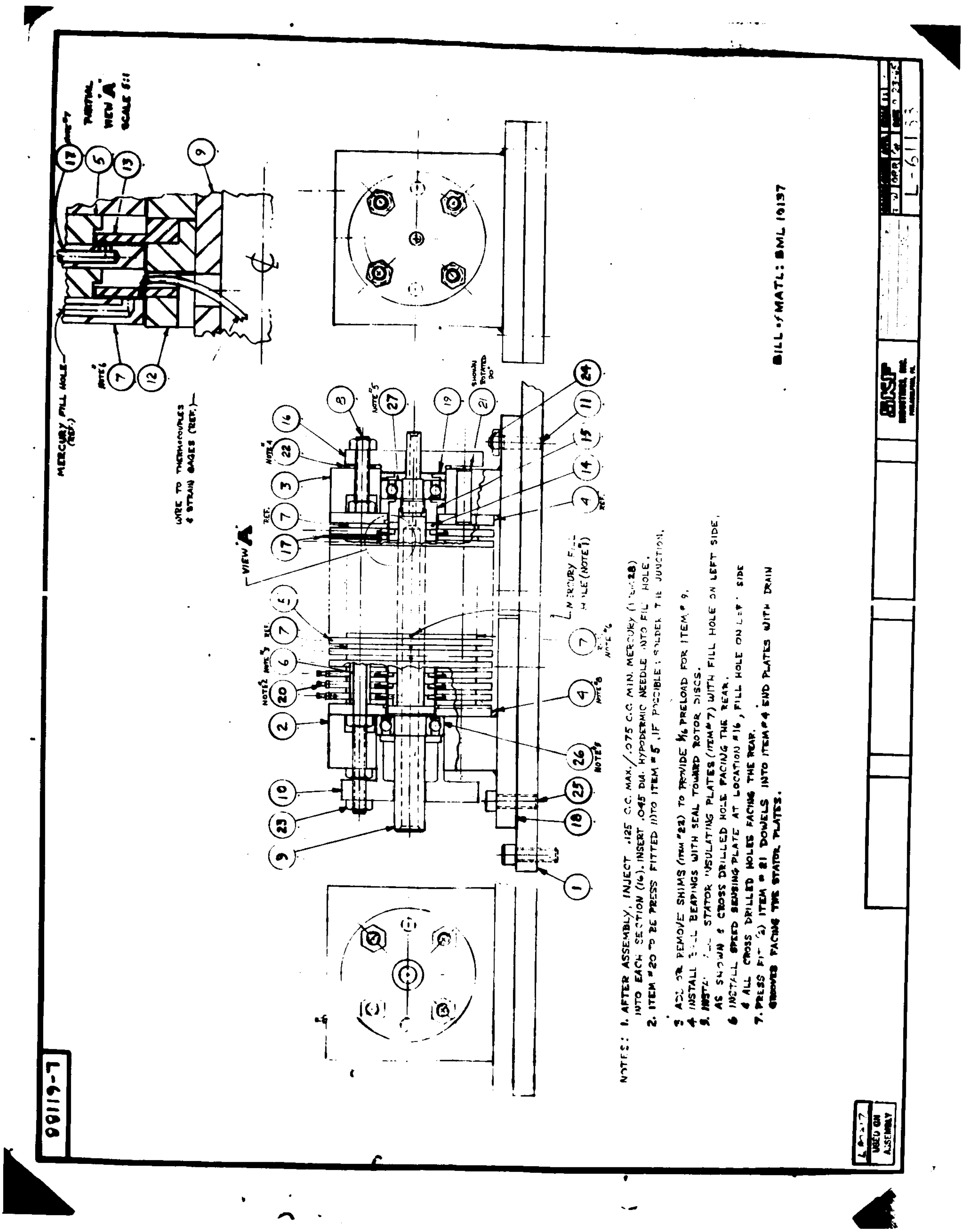


ENCLOSURE 21

TEST BEARING THRUST LOAD CALIBRATION

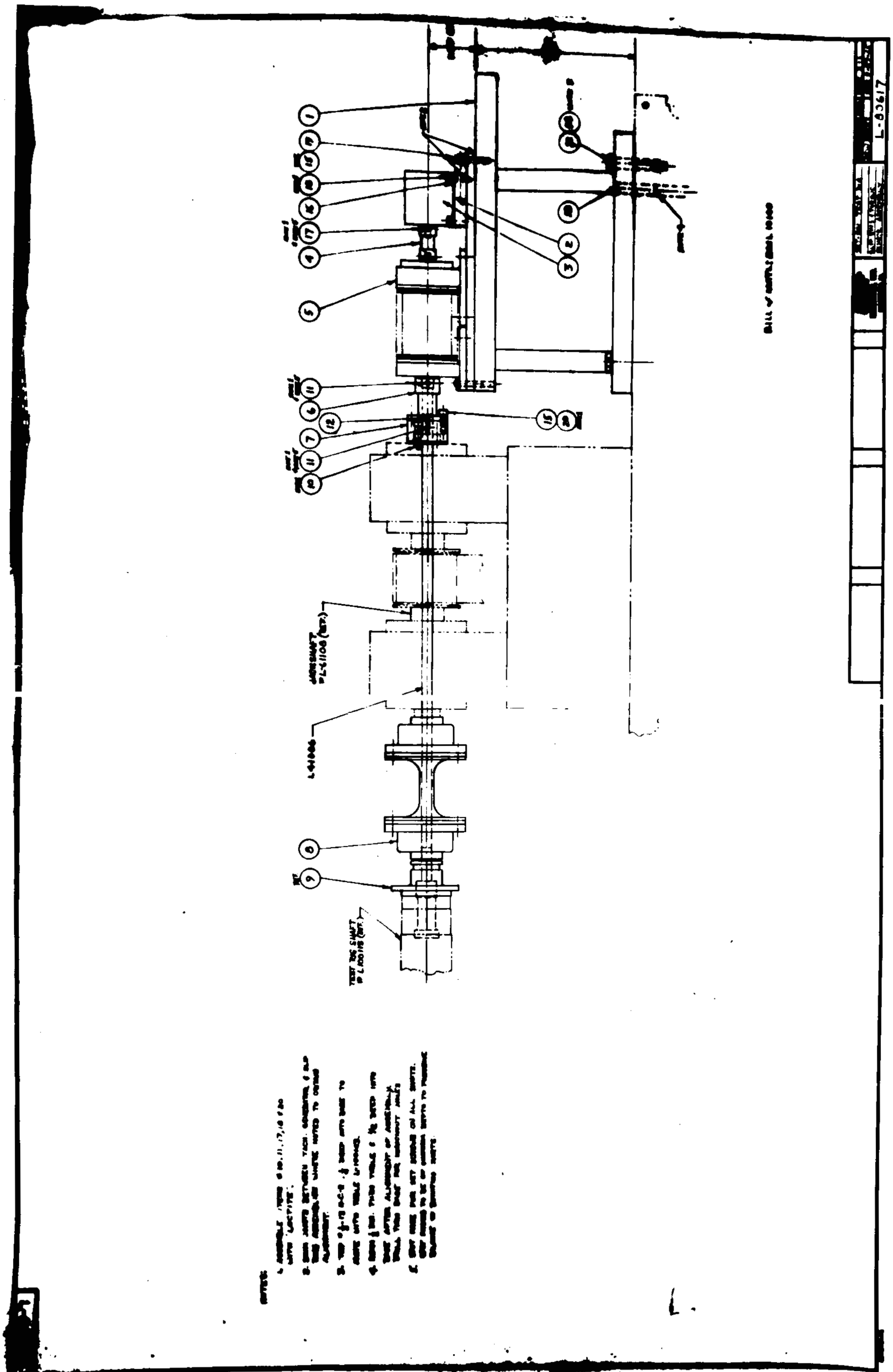


SIXTEEN CHANNEL MERCURY WETTED SLIP RING - ASSEMBLY



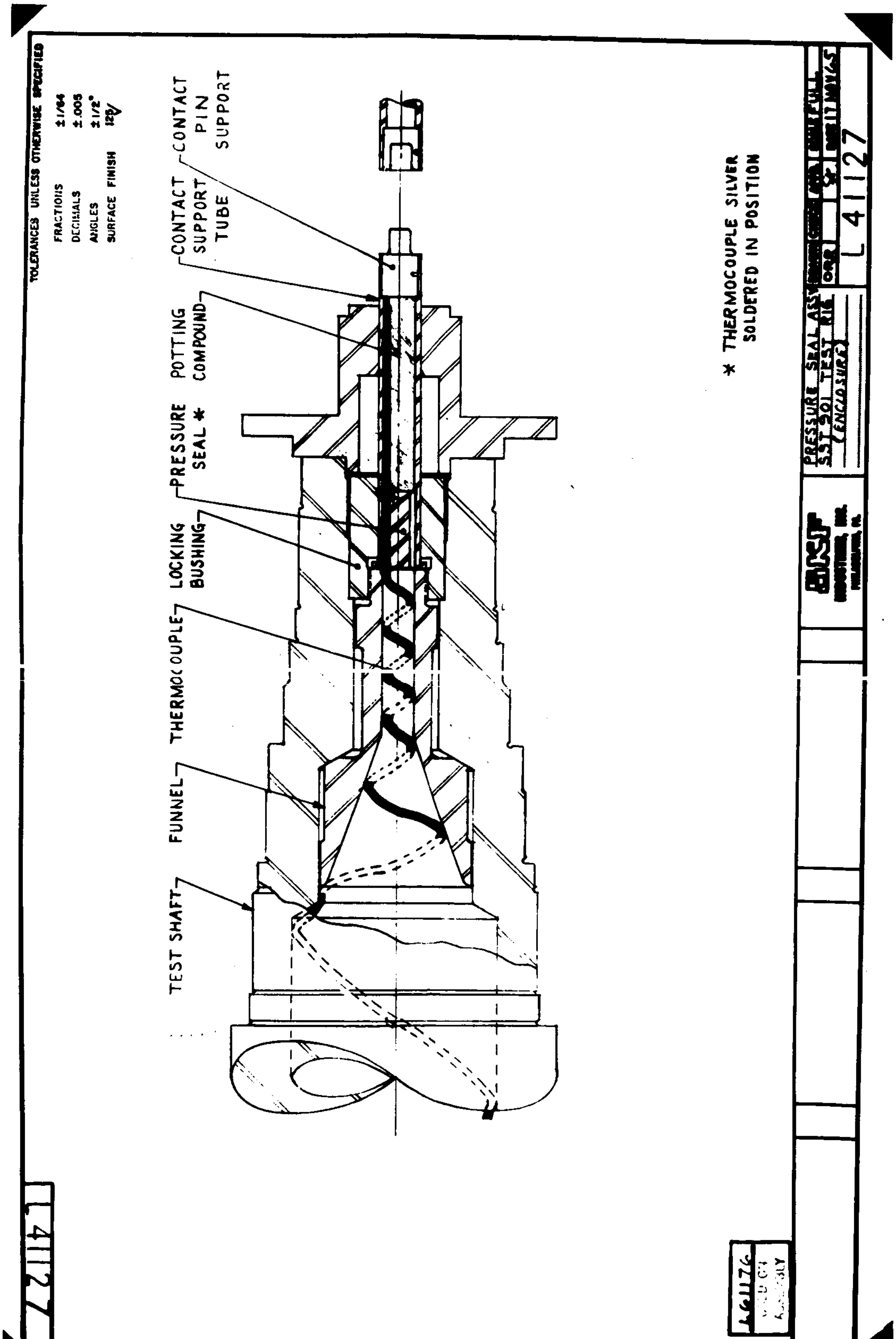
ENCLOSURE 23

ELECTRICAL CONNECTOR PROBE - ASSEMBLY



ENCLOSURE 24

TEST SHAFT ELECTRICAL EXTENSION TUBE



ENCLOSURE 25

TEST BEARING DATA - SERIES I DESIGN

76272

	BEARING DATA	SKF					
1	VENDORS PART NO.	455980	455980A	455980B	455980C	455981	455981A
2	VENDORS DRAWING NO.	76272	76272	76272	76272	76272	76272
3	BEARING TOLERANCE	ABEC-7	ABEC-7	ABEC-7	ABEC-7	ABEC-7	ABEC-7
4	NO. & SIZE OF BALLS	21-13/16 DIA.	21-13/16" DIA.	21-13/16" DIA.	21-13/16" DIA.	21-13/16" DIA.	21-13/16" DIA.
5	INTERNAL RADIAL LOOSENESS (UNMOUNTED)	.0099"-.0103"	.0099"-.0103"	.0099"-.0103"	.0099"-.0103"	.0076"-.0084"	.0076"-.0084"
6	MAX. END PLAY (UNMOUNTED)	.035"	.035"	.035"	.035"	.030"	.030"
7	DESIGN CONTACT ANGLE	31°30'-35°30'	31°30'-35°30'	31°30'-35°30'	31°30'-35°30'	28°30'-32°30'	28°30'-32°30'
8	OUTER GROOVE RADIUS AS PERCENT OF BALL DIA.	51.50-51.75	51.50-51.75	51.50-51.75	51.50-51.75	51.50-51.75	51.50-51.75
9	INNER GROOVE RADIUS AS PERCENT OF BALL DIA.	52.50-53.00	52.50-53.00	52.50-53.00	52.50-53.00	52.50-53.00	52.50-53.00
10	SHIM WIDTH	.012"	.012"	.012"	.012"	.012"	.012"
11	PITCH DIA	6.280"	6.280"	6.280"	6.280"	6.280"	6.280"
12	CAGE WIDTH (MAX.)	1.024"	1.020"	1.024"	1.020"	1.020"	1.024"
13	CAGE SPEC. (FOR OUT-OF-ROUNDNESS & SQUARENESS)	470939	470939	470939	470939	470939	470939
14	CAGE POCKET CLEARANCE	.008"-.022"	.013"-.022"	.009"-.022"	.013"-.022"	.013"-.022"	.009"-.022"
15	BALL GRADE (AFBMA)	15	15	15	15	15	15
16	INNER RING MATERIAL	WB49	WB49	WB49	WB49	M-50	M-50
17	OUTER RING MATERIAL	WB49	WB49	WB49	WB49	M-50	M-50
18	BALL MATERIAL	WB49	WB49	M-1	M-1	M-50	M-50
19	CAGE MATERIAL	M-1	M-1	M-1	M-1	M-1	M-1
20	CAGE TYPE	ONE PIECE MACHINED OUTER LAND RIDING	ONE PIECE MACHINED OUTER LAND RIDING	ONE PIECE MACHINED OUTER LAND RIDING	ONE PIECE MACHINED OUTER LAND RIDING	ONE PIECE MACHINED OUTER LAND RIDING	ONE PIECE MACHINED OUTER LAND RIDING
21	BEARING DESIGNED TO OPERATE AT (MAX.)	850° F	850° F	850° F.	850° F.	650° F.	650° F.
22	INNER RING HARDNESS	67 MIN.	67 MIN.	67 MIN.	67 MIN.	61 MIN.	61 MIN.
23	OUTER RING HARDNESS	67 MIN.	67 MIN.	67 MIN.	67 MIN.	61 MIN.	61 MIN.
24	BALL HARDNESS	67 MIN.	67 MIN.	62 MIN.	62 MIN.	61 MIN.	61 MIN.
25	CAGE HARDNESS	55 MIN.	55 MIN.	55 MIN.	55 MIN.	55 MIN.	55 MIN.
26	CAGE RUNNING CLEARANCE ON DIA.	.010"-.020"	.010"-.020"	.010"-.020"	.010"-.020"	.010"-.020"	.010"-.020"
27	CAGE O. DIA. "L"	6.681-6.673"	6.677-6.671"	6.681-6.673"	6.677-6.671"	6.677-6.671"	6.681-6.673"
28	CAGE I. DIA. "K"	6.111-6.121"	6.115-6.125"	6.111-6.121"	6.115-6.125"	6.115-6.125"	6.111-6.121"
29	CAGE PLATING TYPE	SILVER AMS 2410	———	SILVER AMS 2410	———	———	SILVER AMS 2410
30	CAGE PLATING THICKNESS	.001"-.002"	———	.001"-.002"	———	———	.001"-.002"

CAGE TO BE 100% FLUORESCENT PENETRANT INSPECTED (470908).
100% VISUAL INSPECTION (470653)
100% MAGNAFLUX INSPECTION (471682)
100% ETCH INSPECTION (471637).
MATERIAL AND IDENTIFICATION CONTROL AS PER SPEC. 471085.
INNER RING, OUTER RING AND BALLS TO BE BLACK OXIDE TREATED AS PER AMS 2485.
SURFACE FINISH REQUIREMENTS

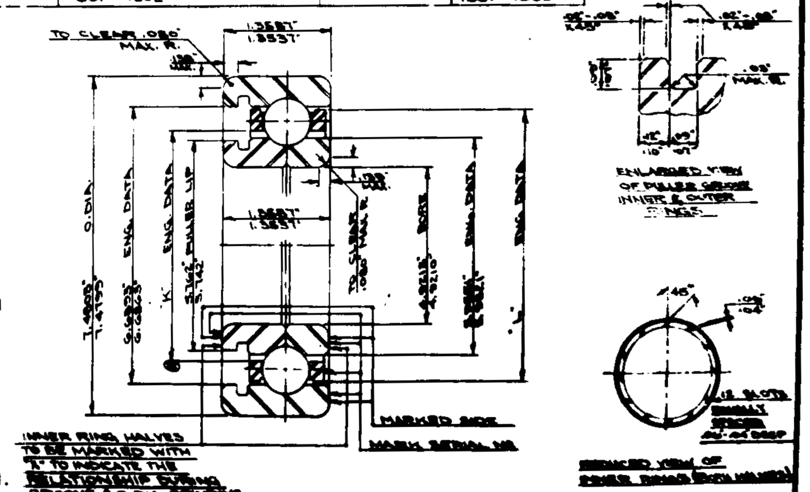
INNER RING BORE	16	RMS	MAX.
INNER RING GROOVE	4	"	"
OUTER RING O. DIA.	16	"	"
OUTER RING GROOVE	4	"	"
CAGE SUPPORT SURFACE	20	"	"
BALL	1.2	"	"
OUTER RING BORE	4	"	"
INNER RING O. DIA.	4	"	"

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

INNER & OUTER RING LAND TOLERANCES
A. 2 POINT OUT-OF-ROUND WITHIN .0004" F.I.R.
B. ECCENTRICITY TO O. DIA. WITHIN .0004" F.I.R.
C. SECTION HEIGHT OF LAND TO O. DIA. EQUAL WITHIN .0004" IN SAME AXIAL PLANE.
D. 90% CONTACT WHEN ROLLED ON A BLUED SUPPORT.

CAGE SUPPORT
A. 2 POINT OUT-OF-ROUND WITHIN .002" F.I.R.
B. 2 POINT DIAMETERS EQUAL WITHIN .002" IN SAME AXIAL PLANE.
C. SQUARENESS TO COMMON MACHINING REF. FACE WITHIN .002"
D. 90% CONTACT WHEN ROLLED ON BLUED FLAT PLATE.

O. DIA. AND POCKETS ARE SPRAYED WITH "S" MONEL (.005 MIN. THICKNESS).



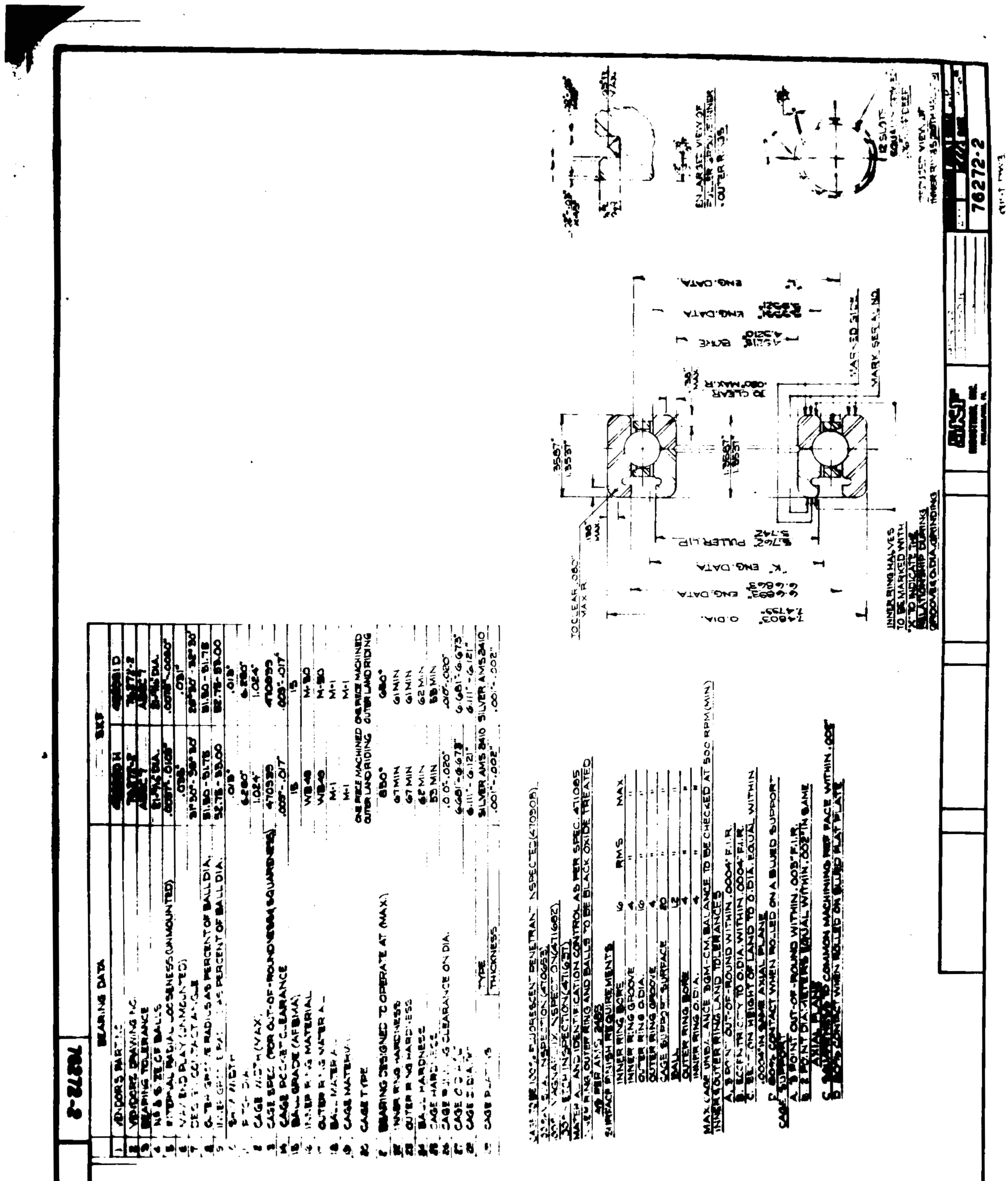
76272

CUST. DWG.

Encl 17

ENCLOSURE 26

TEST BEARING DATA - SERIES I DESIGN



ENCLOSURE 27

TEST BEARING DATA - SERIES II DESIGN

78272-1

	BEARING DATA	SKF					
1	VENDORS PART NO	4559ROD	4559ROE	[illegible]	4559RCG	4559RIB	[illegible]
2	VENDORS DRAWING NO	76272-1	76272-1	76272-1	76272-1	76272-1	76272-1
3	BEARING TOLERANCE	ABEC-7	ABEC-7	ABEC-7	ABEC-7	ABEC-7	ABEC-7
4	NO & SIZE OF BALLS	21-[illegible] DIA	21-[illegible] DIA	21-[illegible] DIA	21-[illegible] DIA	21-[illegible] DIA	21-[illegible] DIA
5	INTERNAL RADIAL LOOSENESS (UNMOUNTED)	.0099"-.0103"	.0099"-.0103"	.0099"-.0103"	.0099"-.0103"	.0076"-.0084"	.0076"-.0084"
6	MAX END PLAY (UNMOUNTED)	.035"	.035"	.035"	.035"	.030"	.030"
7	DESIGN CONTACT ANGLE	31°30'-35°30'	31°30'-35°30'	31°30'-35°30'	31°30'-35°30'	28°30'-32°30'	28°30'-32°30'
8	OUTER GROOVE RADIUS AS PERCENT OF BALL DIA	51.50-51.75	51.50-51.75	51.50-51.75	51.50-51.75	51.50-51.75	51.50-51.75
9	INNER GROOVE RADIUS AS PERCENT OF BALL DIA	52.50-53.00	52.50-53.00	52.50-53.00	52.50-53.00	52.50-53.00	52.50-53.00
10	SHIM WIDTH	.012"	.012"	.012"	.012"	.012"	.012"
11	PITCH DIA	6.280"	6.280"	[illegible]	6.280"	[illegible]	[illegible]
12	CAGE WIDTH (MAX)	1.024	1.020	1.021	1.020	1.020	1.024
13	CAGE SPEC (FOR OUT-OF-ROUNDNESS & SQUARENESS)	470939	[illegible]	[illegible]	470939	470939	[illegible]
14	CAGE POCKET CLEARANCE	[illegible] - .022"	.009" - .022"	.013" - .022	.013" - .022"	.009" - .022"	.013" - [illegible]
15	BALL GRADE (AFBMA)	15	15	15	15	15	15
16	INNER RING MATERIAL	WB49	WB49	WB49	WB49	M-50	M-50
17	OUTER RING MATERIAL	WB49	WB49	WB49	WB49	M-50	M-50
18	BALL MATERIAL	WB49	M-1	WB-49	M-1	M-50	M-50
19	CAGE MATERIAL	M-1	M-1	[illegible]	[illegible]	M-1	[illegible]
20	CAGE TYPE	ONE PIECE MACHINED INNER LAND RIDING	ONE PIECE MACHINED INNER LAND RIDING	ONE PIECE MACHINED OUTER LAND RIDING	ONE PIECE MACHINED OUTER LAND RIDING	ONE PIECE MACHINED INNER LAND RIDING	ONE PIECE MACHINED OUTER LAND RIDING
21	BEARING DESIGNED TO OPERATE AT (MAX)	[illegible] °F	[illegible]	650 °F	[illegible] °F	650° F	650 °F
22	INNER RING HARDNESS	67 MIN.	67 MIN	67 MIN	67 MIN.	61 MIN.	61 MIN.
23	OUTER RING HARDNESS	67 MIN	67 MIN	67 MIN	67 MIN.	61 MIN.	61 MIN.
24	BALL HARDNESS	67 MIN.	67 MIN.	62 MIN.	62 MIN	61 MIN.	61 MIN.
25	CAGE HARDNESS	53 MIN.	53 MIN.	53 MIN	53 MIN.	[illegible] MIN.	58 MIN.
26	CAGE RUNNING CLEARANCE ON DIA.	.016"-.026"	.016"-.026"	.020"-.030"	.020"-.030"	.016"-.026"	.020"-.030"
27	CAGE O. DIA "L"	6.438-6.448"	6.438-6.448"	6.663-6.671"	6.663-6.671"	6.438-6.448"	[illegible]
28	CAGE I. DIA "K"	5.971-5.979"	5.971-5.979"	6.115-6.125"	6.115-6.125"	5.971-5.979"	6.115-6.125"
29	CAGE PLATING TYPE	SILVER AMS 2410	SILVER AMS 2410	———	———	SILVER AMS 2410	———
	CAGE PLATING THICKNESS	.001"-.002"	.001"-.002"	———	———	.001"-.002"	———

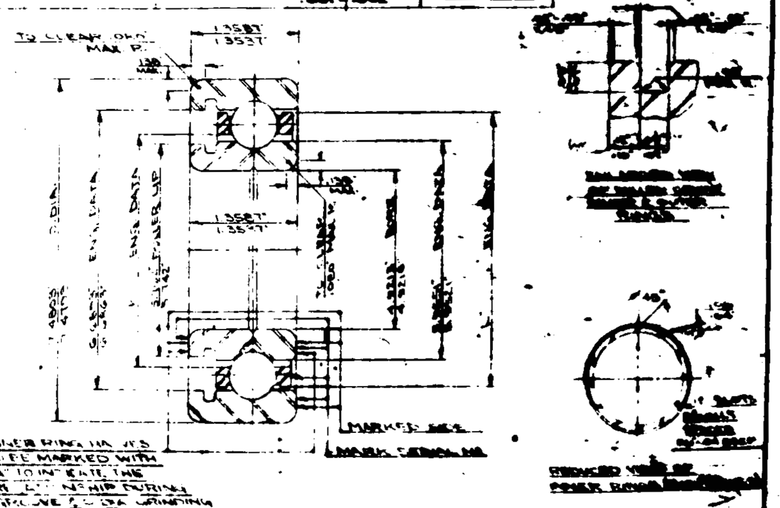
CAGE TO BE 100% FLUORESCENT PENETRANT INSPECTED (470[illegible])
100% VISUAL INSPECTION (4[illegible])
100% MAGNAFLUX INSPECTION (4716[illegible])
100% ETCH INSPECTION (47163[illegible])
MATERIAL AND IDENTIFICATION CONTROL AS PER SPEC 4110[illegible]
INNER RING, OUTER RING AND BALLS TO BE BLACK OXIDE 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 [illegible] SURFACE	20	"
BALL	1.2	"
OUTER RING BORE	[illegible]	"
INNER RING O. DIA.	[illegible]	"

MAX CAGE UNBALANCE 3 GM CM, BALANCE TO BE CHECKED AT [illegible] RPM (MIN)
[illegible] AND TOLERANCES
A. 3 POINT OUT-OF-ROUND WITHIN .004 F.I.R.
B. ECCENTRICITY TO C DIA WITHIN .004" F.I.R.
C. SECTION HEIGHT OF LAND TO O. DIA EQUAL WITHIN .0004" IN SAME AXIAL PLANE.
D. 80% CONTACT WHEN ROLLED ON A BLUED SUPPORT.

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



INNER RING [illegible] TO BE MARKED WITH "A" TO INDICATE THE RELATIONSHIP DURING GROOVE & [illegible] DIA GRINDING

78272-1

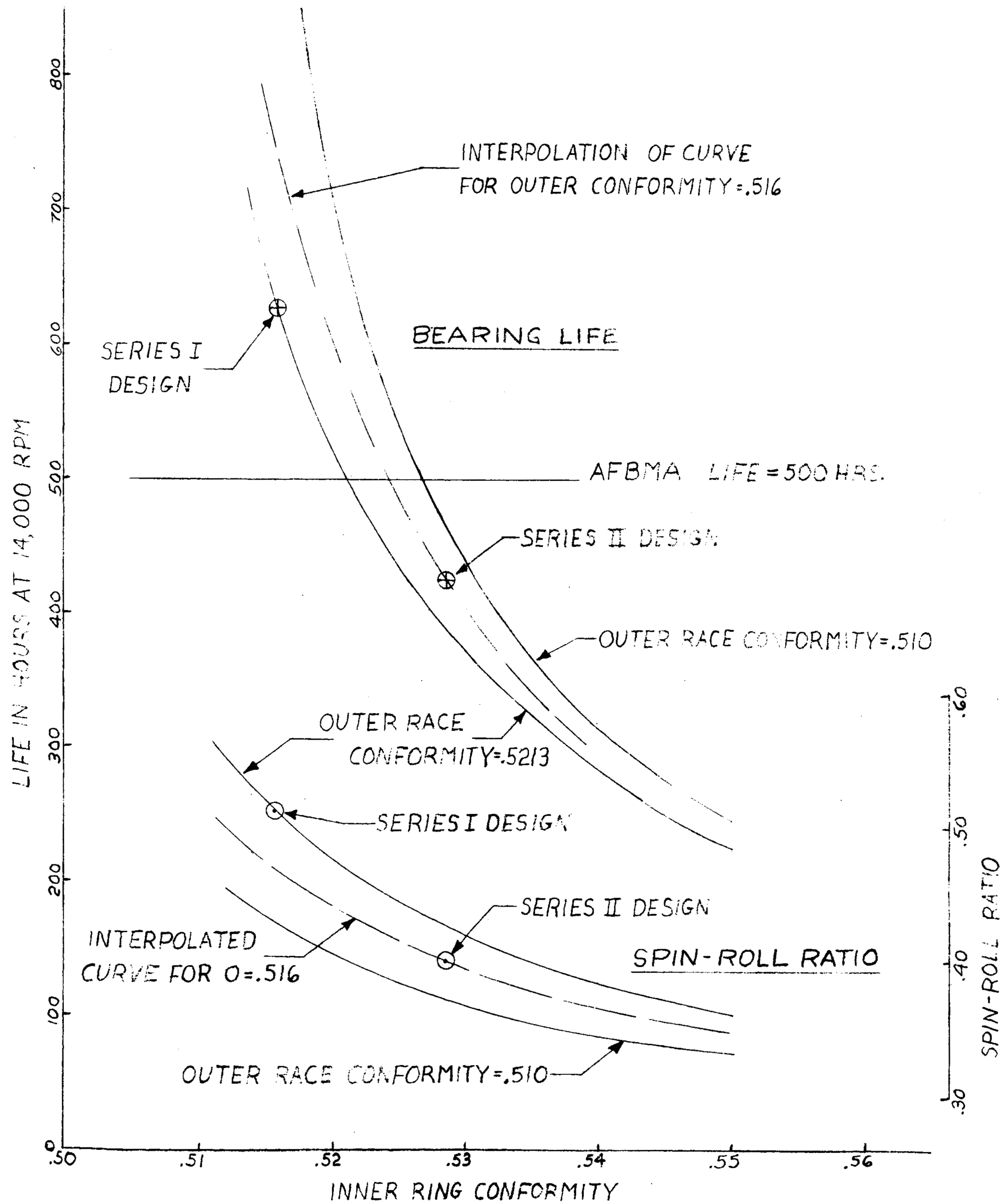
ENCLOSURE 28

HIGH SPEED THRUST BEARING - COMPUTER OUTPUT DATA

	26° Nominal Mounted Contact Angles	
	Original Design (Series I)	Modified Design (Series II)
Test Conditions		
Shaft speed, rpm	14,000	14,000
Thrust load, lbs.	3,280	3,280
Computed Operating Parameters:		
Inner ring contact angle, degrees	40.25	34.14
Outer ring contact angle, degrees	16.61	15.87
Axial bearing deflection, in. x 10^{-6}	447.	1053.
Ball centrifugal force, lbs.	305.	289.
Inner semi-major contact axis, in.	.0877	.0697
Outer semi-major contact axis, in.	.0930	.1041
Normal inner ring ball load, lbs.	221..	255..
Normal outer ring ball load, lbs.	448.	515.
Maximum inner contact stress, kpsi	136.	177.
Maximum outer contact stress, kpsi	197.	196.
Spin-to-roll ratio on inner	.514	.399
Cage speed, rpm	6585.	6412.
Ball rolling speed, rpm	50896.	49559.
Ball axis orientation angle, degrees	14.7	14.1
Ball spin torque on inner, in-lbs.	.513	.356
Outer ball spin torque projected on inner, in-lbs.	1.068	1.084
Spinning heat generated, Btu/hr.	3145.	2196.
Total heat generated, Btu/hr.	18931.	18834.
Minimum friction coefficient required to prevent gyro slip	.049	.041
Bearing life (Lundberg-Palmgren), hours	624.2	431.8
Life of inner ring contact, hours	1951.8	686.4
Life of outer ring contact, hours	848.9	955.4
EHD oil film thickness on inner for 2.5 cs viscosity lubricant, 10^{-6} in.	13.3	13.1
Ball gyroscopic moment, in.-lbs.	14.2	12.9

ENCLOSURE 29

EFFECT OF BEARING GROOVE CONFORMITY ON BEARING LIFE AND SPIN TO ROLL RATIO

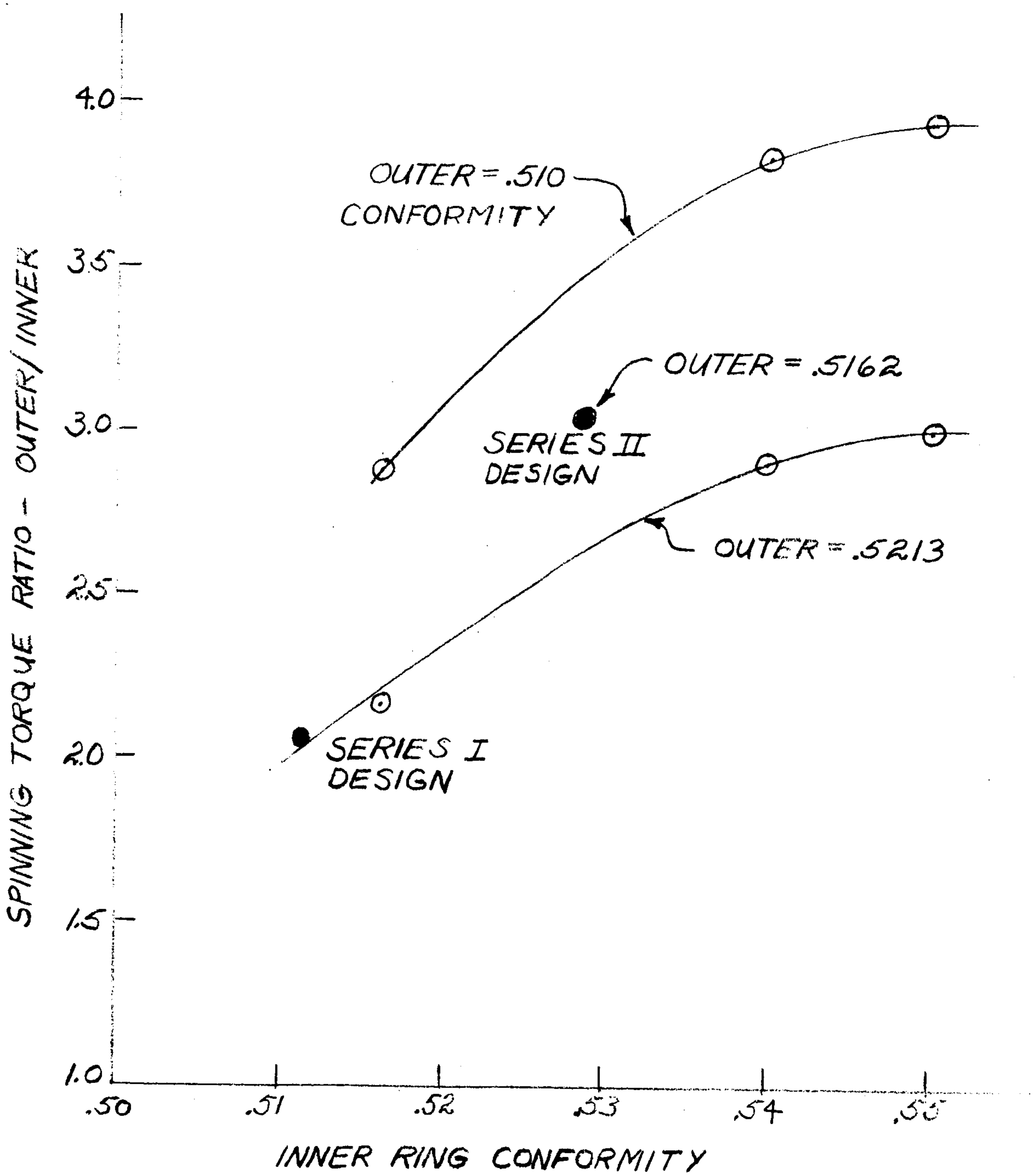


ENCLOSURE 30

THE DEGREE OF OUTER RACE CONTROL

SPINNING TORQUE RATIO AS A FUNCTION OF INNER AND OUTER RACE CONFORMITIES

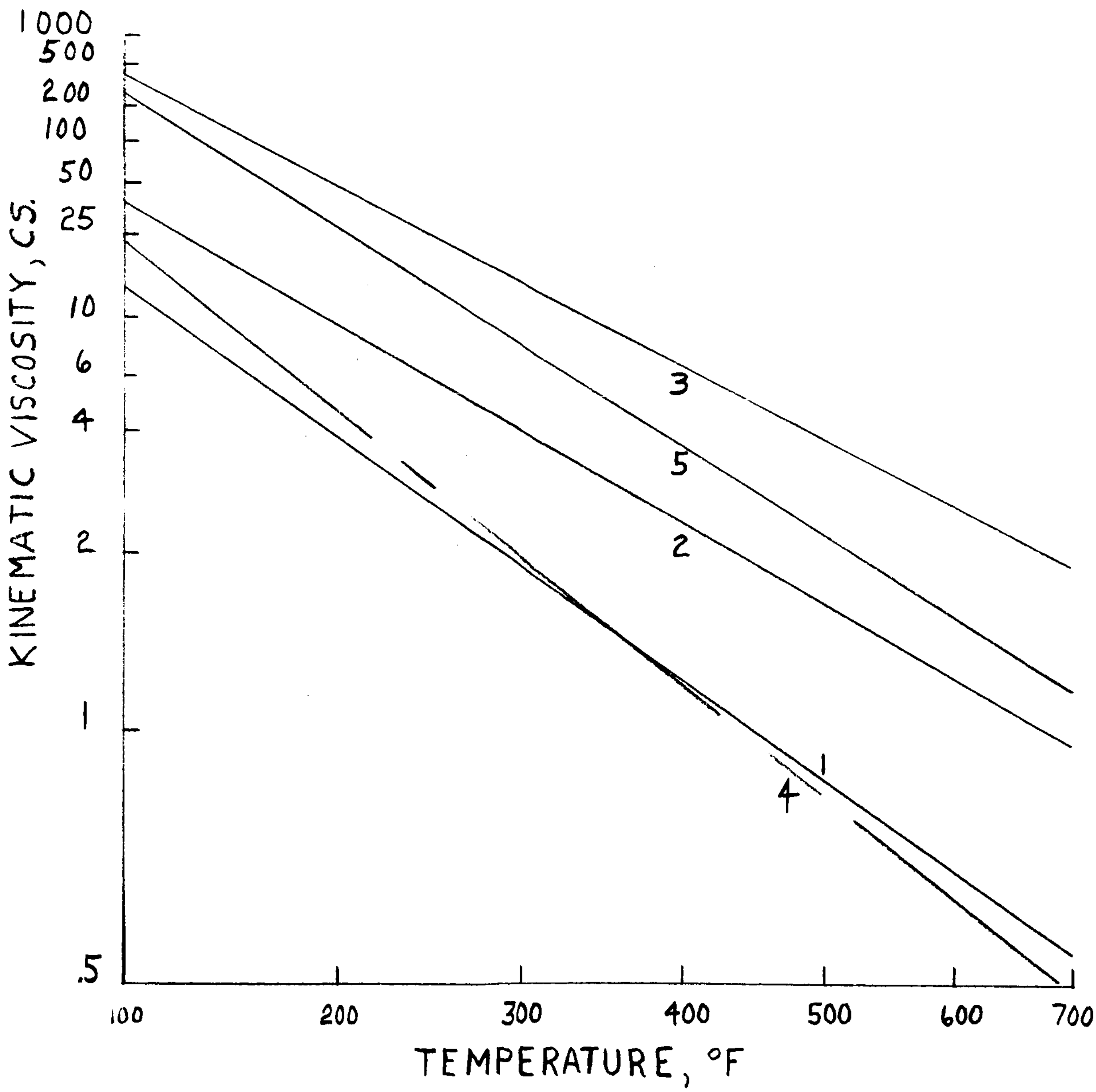
Conformity = $\frac{\text{groove radius}}{\text{ball diameter}}$



ENCLOSURE 31

VISCOSITY-TEMPERATURE RELATION FOR CIRCULATING OILS

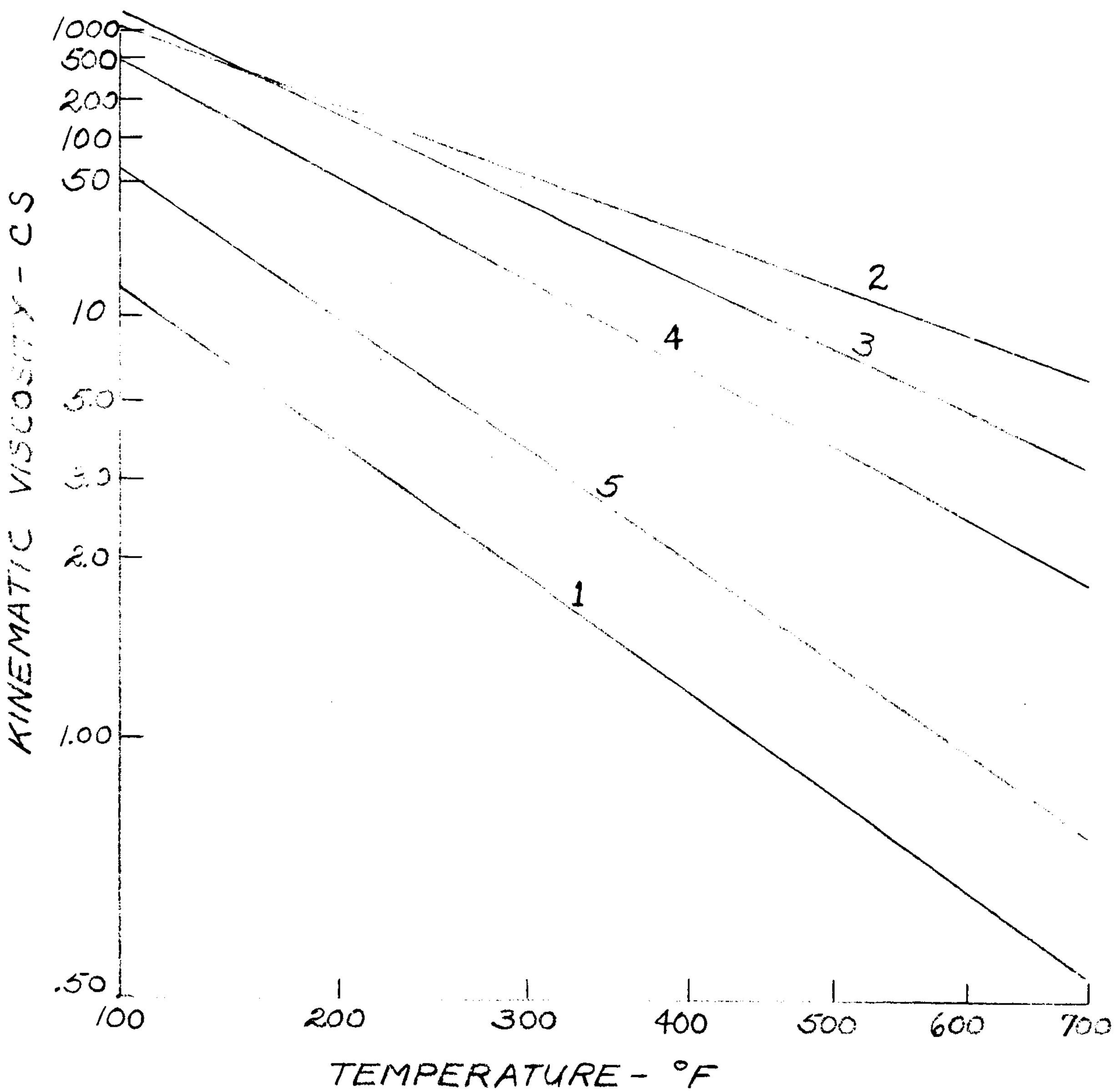
1. Esso 4040 MIL-L-7808E
2. Sinclair Turbo S 1048 Improved
3. Mobil XRM 177F
4. Monsanto MCS-293
5. DuPont PR 143



<u>ENCLOSURE 32</u>

<u>VISCOSITY-TEMPERATURE RELATION FOR MIST LUBRICATING OILS</u>

1. Esso Turbo Oil 4040 (MIL-L-7808E)
2. Union Carbide UCON 50-HB-5100
3. Sun Oil Sunthetic 18H
4. Socony Mobil XRM 177F
5. Hercules Hercolube F



ENCLOSURE 33

PROPERTIES OF MONSANTO MCS-293

Viscosity, Density and Compressibility

Temperature °F	Temperature °C	Kinematic Viscosity cs	Density gm/cc	Bulk Modulus* psi
0	-18	13,040	-	500,000
77	25	-	1.195	410,000
100	38	25.2	1.184	340,000
210	100	4.13	-	285,000
300	150	2.0	1.101	230,000
500	260	0.81	1.017	190,000
700	370	0.48	0.926	-

*Secant, Isothermal 0-7,500 psi range

Pour point	-20°F
Flash point	445°F
Fire point	540°F
Autogenous Ignition Temperature	940°F

Ryder Gear Scuff Test - at 167°F 3020 ppi (average)
at 400°F 1350 ppi (average)

Evaporation Loss 500°F at 760mm 50.7% (average)
500°F at 140mm 61.0% (average)

Oxidation - Corrosion Test (Modified FS 791, Method 5308.2; 48 hours at temperature SL/W air flow)

Temperature		500°F	600°F
Weight change of metals (mg/cm^2)			
iron		+0.04	+0.23
copper		-3.97	-8.37
silver		-0.71	-1.37
Change of Viscosity	at 6.5°F	+6.5%	+42.3%
	at 210°F	+4.4%	+20.8%
Neutralization Number			
Initial		0.01	0.01
Final		0.10	0.15

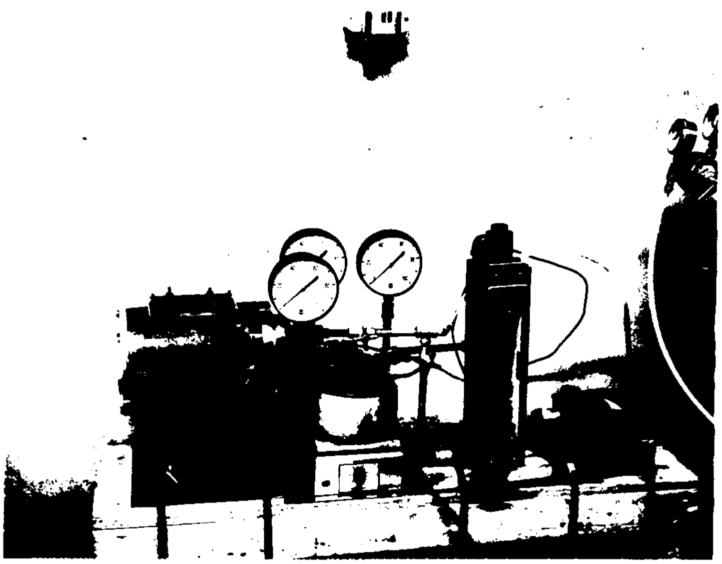
ENCLOSURE 34

GENERAL PROPERTIES OF HERCOLUBE F

Kinematic viscosity, cs		
	-40°F	38,000
	100°F	57
	210°F	8.7
Pour Point, °F		-60
Flash Point, °F		550
Flame Point, °F		620
Autogenous Ignition Point, °F		910
Evaporation loss, 6½ hrs. at 400°F,%		4
Specific gavity at 60/60°F		1.013
	lbs. per gallon at 60°F	8.50
Saponification Number		390
Acid Number, max.		0.1
Hydroxyl, max. %		0.2
Ash, max. ppm		50

ENCLOSURE 35

OIL-MIST STATIC TEST RIG



T-924010

This rig has been constructed to study the feasibility of oil-mist lubrication under high pressure and high temperature conditions.

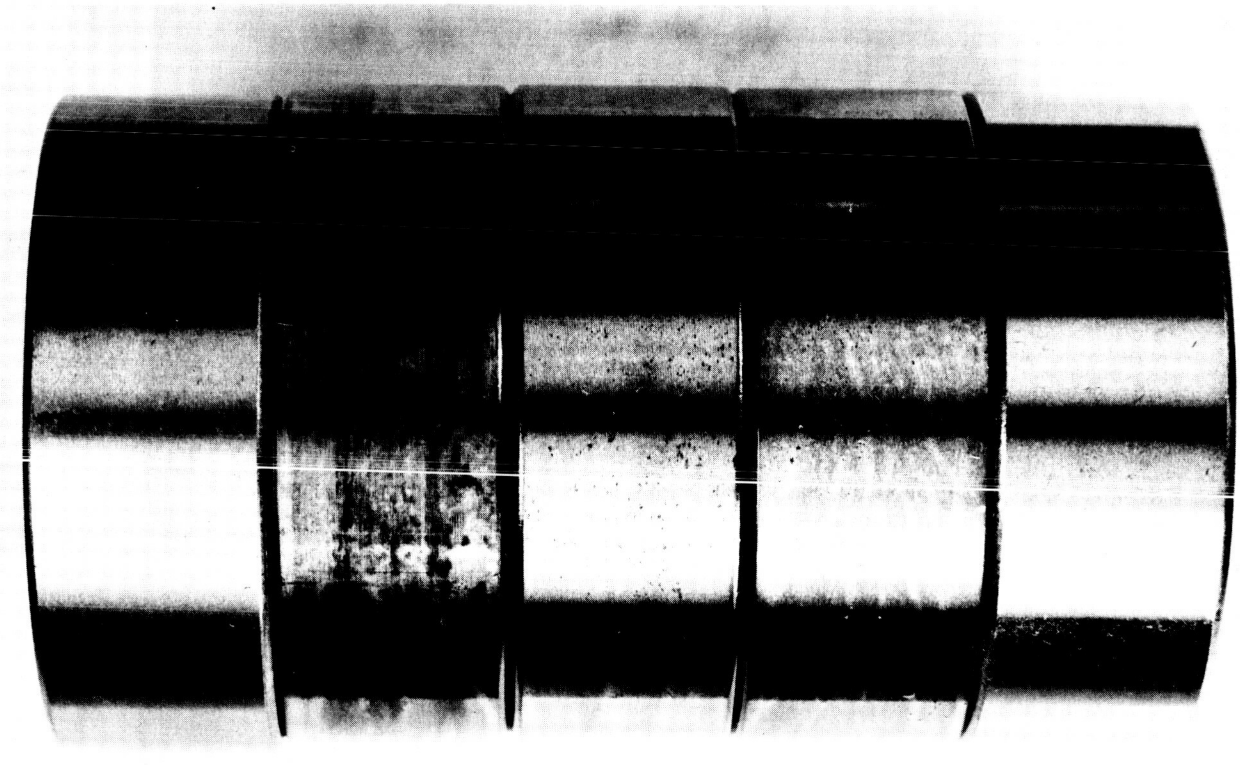
ENCLOSURE 36

SUMMARY OF CONDITIONS AND RESULTS OF OIL MIST DEPOSIT FORMATION TESTS

Oil	Oil Sump Temp °F	Generator Nitrogen Flow SCFM	Chamber Pressure psi	Pressure Drop Generator psi	Nozzle Dia. Inches	Chamber Temp. °F	Condition of Rings
Sunthetic 18-H	330	2.4	45	10	3/32	600	Oily with light tarnish
Hercolube A	200	3.1	49	9	3/32	600	Slight oiliness-light tarnish
Hercolube F	200	3.4	45	9	3/32	600	" " "
Esso Turbo WS 5435	200	3.4	45	9	3/32	600	Dark gum
Drew Syn. Lube. Base MW-3992-R	200	3.1	45	9	3/32	600	Dark gum
Drew Syn. Lube. Base MW-3992	200	3.4	45	8	3/32	600	Dark gum
Ucon 50-HB-5100	200	3.1	45	9	3/32	600	Oily-clear
XRM-177F	200	3.4	45	10	3/32	600	Oily with light tarnish

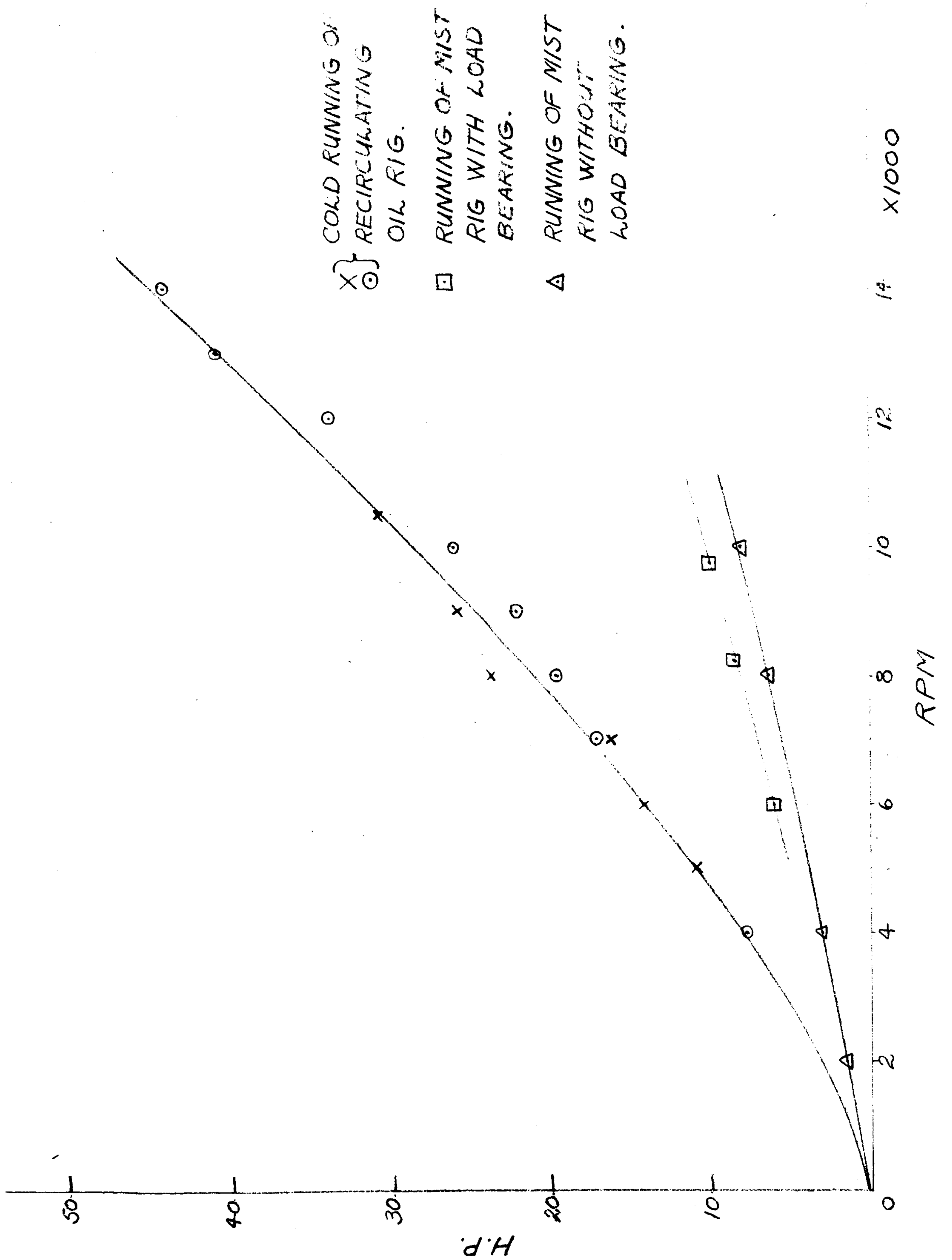
ENCLOSURE 37

DEPOSIT FORMATION ON 600°F STATIC BEARING RINGS
EXPOSED TO SUNTHETIC 18H IN OIL MIST FORM FOR 30 MINUTES



ENCLOSURE 38

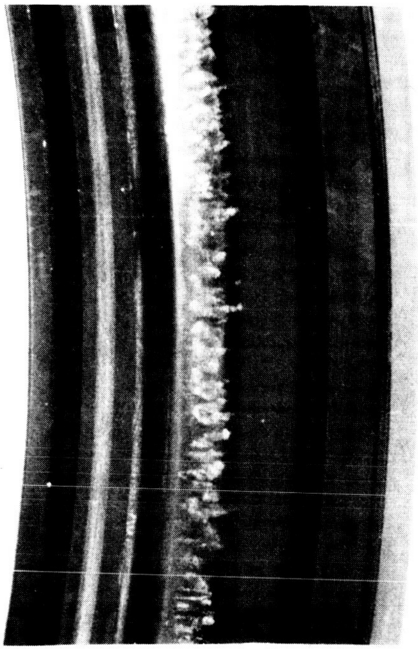
POWER CONSUMPTION CURVES FROM INPUT TO 50 HP DRIVE MOTOR



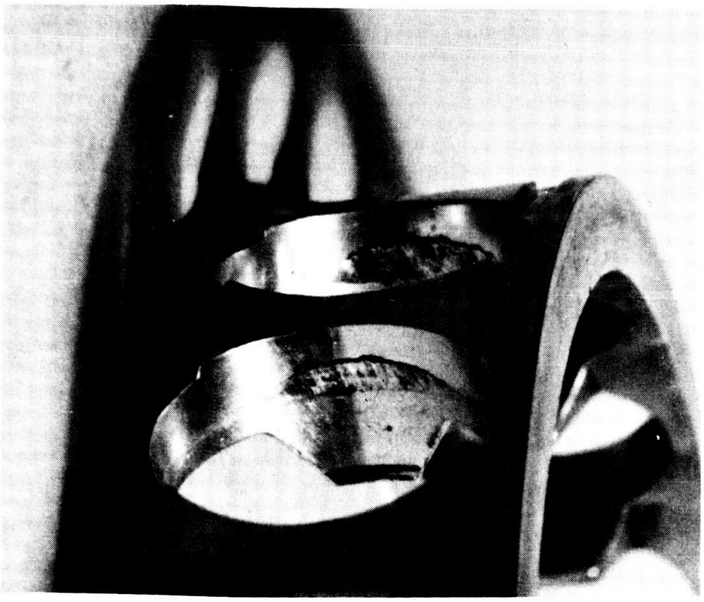
ENCLOSURE 39

LUBRICATION DISTRESS IN OIL MIST TEST BEARING COMPONENTS AFTER RUNNING 6.8 HOURS AT 6000 RPM, 1000 LBS THRUST (OIL MIST LUBRICANT SUPPLY CUT OFF AFTER 3.8 HOURS)

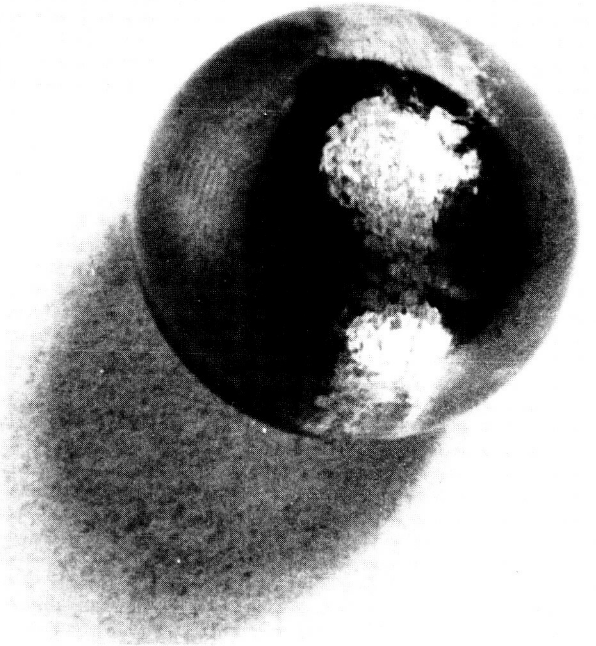
Loaded Unloaded
Inner Rings



Outer Ring



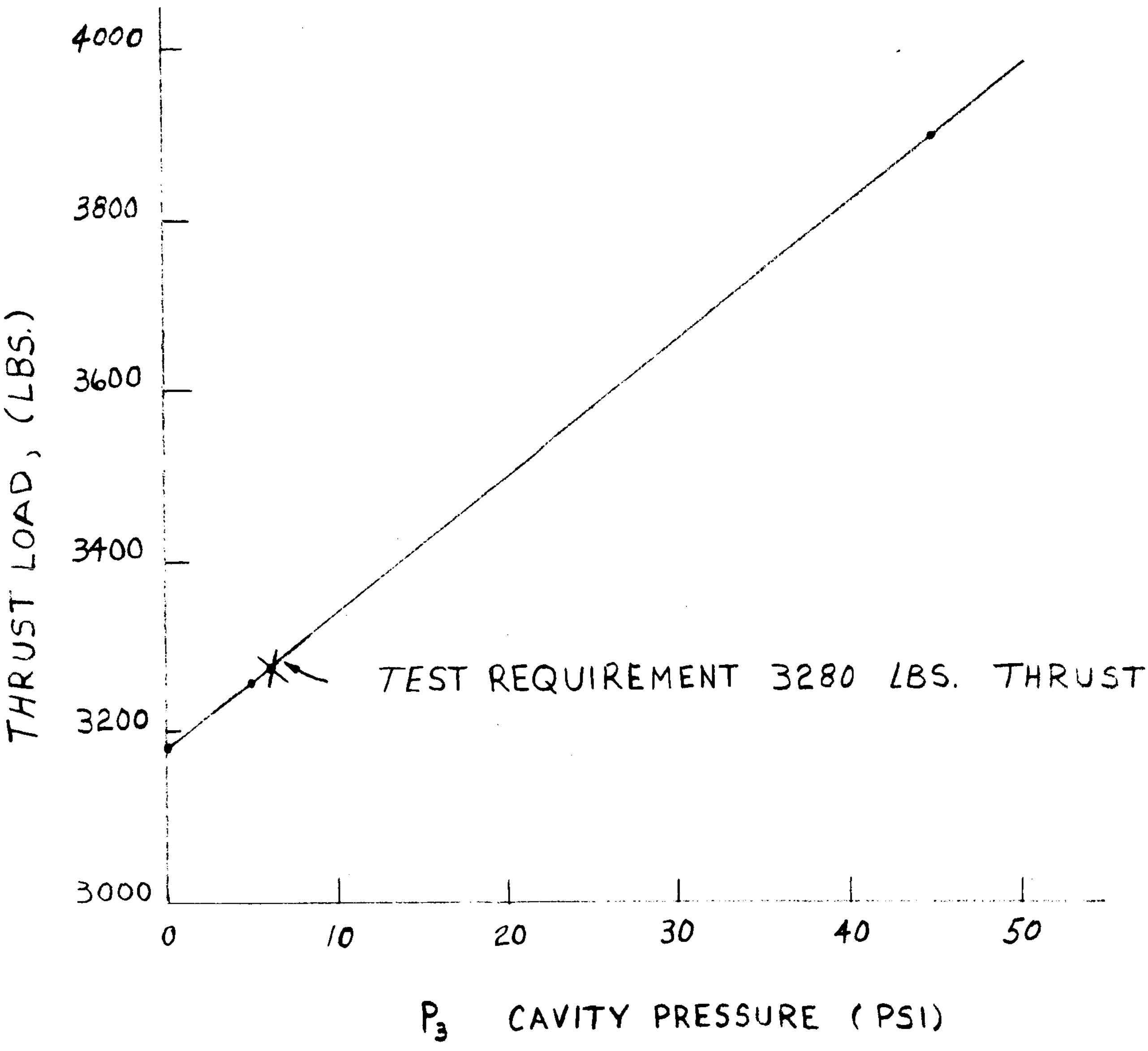
Cage Pocket



Typical Ball

ENCLOSURE 40

NET THRUST ON TEST BEARING AS A FUNCTION OF BEARING CAVITY PRESSURE



100 PSI PRESSURE DIFFERENTIAL MAINTAINED ACROSS TEST SEALS

ENCLOSURE 41

459981A TEST BEARING AFTER 8 HOUR RUN UP TO FULL SPEED AND LOAD IN RECIRCULATING RIG

LUBRICANT: Esso Turbo Oil 4040 Unheated
AIR TEMPERATURE: 400°F

ENCLOSURE 42

OIL SIDE TEST SEAL AFTER 8 HOUR RUN UP TO FULL SPEED AND LOAD IN RECIRCULATING RIG