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BEARING TESTS IN WATER

AEC RESEARCH AND DEVELOPMENT REPORT

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Pratt & Whitney Aircraft DIVISION OF UNITED AIRCRAFT CORPORATION

CANEL

MIDDLETOWN CONNECTICUT

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PRATT & WHITNEY AIRCRAFT
DIVISION OF UNITED AIRCRAFT CORPORATION
CANEL
MIDDLETOWN CONNECTICUT

SUBJECT Single Film Journal Bearing Tests in Water	TIM NO. 916
COPIES TO: AEC-NYOO CANEL Div., AEC-TID(2) A. D. Barton, C. C. Bigelow, H. C. Gray, R. W. Kelly, G. M. Wood, D. V. Manfredi, C. Merz, P. T. Heyl, W. Richards, B. Weinberg, H. Welna, D. Newey, D. Groll, L. Knudsen	DATE OF REPORT: July 1965 WRITTEN BY: B. Weinberg APPROVED BY: G. M. Wood GMW R. W. Kelly R. W. K.
CIRC: W. Doll, E. R. Dytko, R. I. Strough	

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Classification	Authorized Classifier	Date

ABSTRACT

This report describes a series of tests conducted with fluid-film bearings as part of a program for developing a shaft-bearing system for liquid metal pumps to be used in the SNAP-50/SPUR powerplant. Hydrodynamic, hydrostatic, and hybrid type bearings were tested using water to simulate the low kinematic viscosity condition of lithium.

The test unit, which was basically a shaft supported by a conventional single deep-groove ball bearing at one end and the test bearing at the other, was also operated to simulate the dynamic conditions of a simular test unit which was operated with lithium at 600F.

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SUMMARY

A series of tests were conducted with a single journal bearing test unit using water as the working fluid. Hydrostatic and step type hybrid bearings with a 2 3/8 inch diameter, length to diameter ratio of 1.0, and a radial clearance of 0.002 inch were tested from 0 to 10,000 rpm. Test results show the 4-pool hydrostatic bearing to have higher load-carrying capacity at 0 rpm than the step type hybrid bearing. However, at 10,000 rpm the step type hybrid bearing was found to have the higher load-carrying capacity. The optimum orifice size for the 4-pool hydrostatic was found to be a diameter of 0.060 inch at 10,000 rpm.

Test results of a 4-pool hydrostatic bearing with the same parameters as the above mentioned bearing, except for an increase of length-to-diameter ratio from 1.0 to 1.5, showed an increase of in load-carrying capacity of 35 percent over a range of 0 to 10,000 rpm. This bearing was tested with an 0.070 inch diameter orifice.

Test results of the step type hybrid bearings, with and without orifice-compensation, show the one with a step depth of 0.0006 inch to have a higher load-carrying capacity when orifice-compensation was applied. When orifice-compensation was omitted, the bearing with a step depth of 0.0012 inch was found to have the higher load-carrying capacity.

The test unit, operated to simulate the dynamic conditions of a similar unit design for liquid metal tests and using a hydrodynamic bearing, demonstrated excellent operation from start-up to 10,000 rpm.

I. INTRODUCTION

A program was initiated to develop liquid metal pumps for the SNAP-50/SPUR space powerplant [1, 2]* In order to provide a compact lightweight pump, the design concept selected was a hermetically-sealed unit which utilized the liquid metal being circulated by the pump as the lubricant for the shaft bearings. Because of the tendency for metals to self-weld in high temperature liquid metal environments, fluid-film bearings were selected for the prime development effort in preference to the rolling contact type. Further discussion related to bearing development problems and bearing selection are presented in TIM-722 (3).

A review of the literature indicated that shaft stability is the major problem with lightly loaded film bearings, as is the case in space applications. Although many sources have investigated this problem, an analytical model to predict shaft behavior has not as yet been established.

In support of the pump development effort, a bearing test program was started which included both single bearing tests and rotor dynamics tests (4, 5). This report covers tests conducted with the 2 3/8 inch diameter journal bearings.

The purpose of the single bearing tests was primarily to permit a detailed study of fluid film bearing characteristics and to permit correlation of test data with analytical predictions. These characteristics included bearing stiffness, load carrying capacity, attitude angle, lubricant flow for bearing operation and cooling, and optimization of orifice size and configuration. The main advantages of the single bearing test unit were the low down time for changing of the test item, the flexibility of design, and the ease of operating an identical unit in liquid metal in order to provide correlations between water and liquid metal bearing test results. The free movement of the shaft in these units was dependent on face seals which operated on a hydrodynamic film. The sealing system used in the single bearing test units was developed at CANEL as part of the Lithium Cooled Reactor Experiment Pump Development Program(6).

* Numbers in parenthesis designate references at the end of this report.

In addition to ease of handling, the advantages of using water rather than liquid metal as the test lubricant were that it permitted detailed instrumentation of the test item as well as visual observation of lubricant flow and problems related to bearing cavitation and ingassing of the test lubricant.

The most important piece of instrumentation was the inductance type distance detector which provided high frequency response and accurately sensed the shaft motion and position within the test bearing (7).

The single bearing program included tests of hydrodynamic, hydrostatic and hybrid type bearings. In addition to tests to investigate bearing characteristics, a test was conducted with a hydrodynamic bearing to establish the dynamic startup and operating conditions for a similar liquid metal bearing test unit. This test also provided shaft-bearing stability and whirl-data. The 10,031-hour liquid metal bearing test based on the these results is reported in TIM-915 (8).

This report includes a description of the test unit, equipment, test procedure, and test results.

II. DESCRIPTION OF APPARATUS

The bearing tests described in this report were performed in the PT-3 test cell of the CANEL Pump-Turbine Laboratory. Air and electric power were made available along with process and domestic water. A chemical demineralizer was used to render the domestic water non-conductive at two million ohms or higher for test use. A spray tower vacuum type deaerator was also used to further purify the test water. Process water was used for cooling purposes.

A. Test Unit

The test rig consisted of a modified TP-1 pump center section shown in Fig. 1. Fig. 2 shows the test rig configuration used during these tests. The test rig, was driven by an air turbine. Several nozzles of this turbine assembly were blocked off with epoxy resin to obtain approximately ten percent partial admission. This was done to facilitate smoother speed control since the turbine power requirements were appreciably reduced from those for the pump.

From the turbine section, the shaft extended through the ball bearing oil cavity and subsequently ended up at the test bearing. The ball bearing supported the shaft both radially and axially, and the test bearing provided only radial supported for the shaft. Sealing of the ball bearing oil cavity was provided by two bellows-mounted carbon face seals. A low pressure (1.0 psig or less) sweep gas system was used to purge out lower oil seal leakage to prevent mixing with the test water.

The test bearing was supported by the lower portion of the oil cavity housing using a combination hydrostatic shaft loader and bearing support as shown in Fig. 2. Bearing alignment was accomplished by the placement of shims above the upper flange of this bearing support near the hold down bolts. The test bearing - support combination was located within a housing designed to allow pass throughs for bearing instrumentation as well as supply bearing lubricant and pressurized fluid to the hydrostatic loader. A plastic bottom was bolted to this housing containing the test water to provide visual inspection.

The complete test rig assembly was bolted directly to an angle iron welded framework. This stand was, in turn, attached to the concrete cell floor via hard rubber pads. These were chosen to minimize the effects of stand vibration. The rig was attached directly to the remainder of the loop as shown by the flow diagram in Fig. 3.

A three horsepower Worthington centrifugal pump supplied pressurized water to both the hydrostatic loader and the test bearing. The pressurized water to the hydrostatic loader through a pressure-drop calibrated manifold within the test bearing housing. The test bearing supply flow was fed via removable, external orifices. Mounting the orifices external to the test bearing permitted quick change of their size. Both loader and test bearing supply pressures were controlled by air-to-open and air-to-close Research valves respectively. These valves were in turn controlled by manually operated air pressure regulators located at the test panel. Discharge flow from the hydrostatic loader and the test bearing was channeled back to the dump tank via a weir well. The weir well height was pre-cut to establish the water level within the test bearing housing.

Before the start of each test, water temperature within the loop was brought to operating conditions by increasing heat input to the system. This was accomplished by electrical heaters surrounding the deaerator tank. During testing, the test water was cooled to maintain operating conditions by a water to water heat exchanger.

Safe limits of bearing supply and lube oil pressures, and turbine speed were maintained using Honeywell Pressuretrols and an overspeed trip respectively. If pressures decreased below a preset value or if the turbine oversped, a swing check valve would automatically shut off the turbine air supply thereby terminating unsafe operation.

B. Instrumentation

The instrumentation in the PT-3 tests consisted of pressure gages, flowmeters, thermocouples, a speed pick-up, and electronically operated distance detectors. All pressure measurements were made with Heise gauges. A zero to 100 psig gage was used for the bearing supply pressure. Accuracies of these gages were within 0.1% of their full scale reading.

All flows were measured on Fisher and Porter float type flow meters. These ranged from 0.270 to 5.4 gpm capacity and have an accuracy of 2.0% of full scale reading. Temperatures of the test water, test bearing, and ball bearing were measured by Chromel-Alumel thermocouples. These read out on a Brown circular recorder giving an overall accuracy of 2.5% of the 600F full scale reading.

To obtain accurate measurement of the journal displacement within the test bearing, an electronically operated distance detector system was employed(7). This system worked on a variable inductance principle. The detector heads, each consisting of a pancake wound inductance coil, were placed every 90 degrees circumferentially around the upper and lower ends of the test bearing. As the shaft moved to and from a given detector head (the shaft being of conductive material), it changed the detector heads' coil inductance in proportion to the movement. This change in inductance was converted to a change in a D.C. voltage which was in turn displayed on an oscilloscope. Any two detectors that gave clean response and were located 90 degrees away from each other were used to obtain a trace in an X-Y plane. Also, a summing amplifier arrangement was utilized in order to reduce any temperature drift between readings. With this set-up two detectors, 180 degrees apart, were used in combination for each X-Y axis readout. The summing feature cancelled out temperature drift if the drift on two opposing detectors was the same. The summing amplifiers tended to add appreciable noise to the system. Also, frequent recalibration of the zero eccentricity position further reduced any temperature effects. A photograph of the distance detector components is shown in Fig. 17.

Accuracy of the detector system was approximately $\pm$ 0.0001 inch for a fullscale of $\pm$ 0.002 inch in absolute movement. However, relative changes in position of the shaft as small as 50 millionths could be observed. To hold accuracy of calibration, the resistivity of the test water was continuously monitored and maintained at two million ohms or higher. The detector units and summing amplifiers were manufactured by Bently-Nevada Corporation, Model D-152. The oscilloscope was a Techtronix with dual trace and storage features, Model 564.

The shaft speed was read on a Berkely Digital Counter. The signal fed to the counter came from a magnetic pickup in proximity to a six toothed gear on the test unit shaft. Also, test stand vibration was monitored with a Consolidated Electrodynamics Corporation vibration sensor unit.

C. Test Items

The test bearings included in these tests consisted of hydrodynamic, hydrostatic, and hybrid types. All of these bearings were internally chrome plated and machined to 2.3490-2.3492 inches in diameter. The test units shaft journal was 2.3450-2.3451 inches in diameter. This gave a radial clearance of 0.002 inches, from journal to bearing, for all bearings used.

1. The first bearing tested was a four feed port hydrodynamic bearing with an L/D ratio of 1.0. An example of this bearing is shown in Fig. 4. One purpose for running this bearing was to mock-up a similar bearing test configuration which was to be run with liquid metal lithium as a lubricant. Because of these requirements, the hydrostatic loader was not used with this bearing. Instead, loading was provided by inserting a given unbalance weight within the lower end of the shaft. The shaft unbalance was varied from 0.04 to 0.70 oz. inch. Also, for this mock-up test, the four 0.450 inch feed holes were connected to the supply loop, but were left open within the test bearing housing. This was to more closely duplicate the conditions under which the liquid metal hydrodynamic bearing would operate. The liquid metal bearing is described in TIM-915[8].

2. The second and third bearings tested were both hydrostatic bearings with L/D ratios of 1.0 and 1.5. Examples of these bearings are shown in Fig. 5. The design of these bearings was based on an analysis established by A. A. Raimondi and J. Boyd[9]. These bearings had 0.250 inch lands and four 0.450 inch feed holes feeding into four separate pools. Loading was accomplished by using a hydrostatic loader which also acted as a test bearing support. The 1.5 L/D bearing did not complete the test due to some chrome plating breaking away from the end lands with subsequent failure. The 0.050 inch, 0.070 inch, or 0.090 inch orifices were assembled external to the test bearings.

3. The fourth and fifth bearings tested were of the step hybrid type. A hybrid type bearing incorporates features from both hydrodynamic and hydrostatic type bearings. Both were of L/D ratio 1.0. The design of these bearings was based on an analysis established by R. R. Donaldson[10]. An example of these bearings is shown in Fig. 6. The first bearing had a single 0.0006 inch deep step with four 0.450 inch feed holes and end land lengths of 0.475 inch. The second bearing had a 0.0012 inch deep step. These bearings, being of hybrid type, were tested with and without the externally mounted orifices. Orifice sizes ranged from 0.015 inch to 0.090 inch in diameter as listed in Fig. 11. A third bearing of this type scheduled for a future test had a step depth of 0.002 inch. The machining accuracy of these bearings is illustrated in Fig. 16.

III. PROCEDURES

A. Assembly

For the assembly of the test bearing to the test unit, use was made of the test bearings distance detectors. These detectors displayed, on a dual beam oscilloscope, the position of the journal at the upper and lower planes of the test bearing. The bearing was considered aligned when the journal was centered within the test bearing at the bearing's upper and lower ends. Alignment was accomplished by placing shims between the sub-assembly of the test bearing and hydrostatic loader, and the lower part of the ball bearing oil cavity housing. By watching the scope while installing the shims, the assembler could actually see the alignment and determine immediately the correct shim thickness to use and its position. Upon completion of this procedure, alignment was verified by measuring the total shaft movement within the bearing with dial indicators.

B. Test

After a typical installation of the test unit in the PT-3 test stand, the dump tank was filled with demineralized water. The supply water pump, the deaerator and heater were then started. Th loop temperature was brought to approximately 120F to speed deaeration before testing began. Deaeration was necessary to prevent air entrainment problems with the distance detectors. Prior to test start-up the test water was cooled and maintained at 85F $\pm$ 5F. This temperature was chosen to match the kinematic viscosity of lithium in the 600F to 700F range.

At the start of the hydrodynamic bearing test, pressure feed lines were temporarily connected to the test bearing. This was done to prevent journal-to-bearing contact on start-up while checking out the turbine and also facilitated oscilloscope calibration. By pressurizing one of the four feed lines at a time it was possible to move the journal through its full travel within the test bearing which was the diametral clearance of 0.004 inch. The scope was then set up to show this displacement over a given grid interval for both the X and Y directions. Thus, a circle with a calibrated diameter of 0.004 inch would appear on the scope equal to the maximum movement of the shaft within the bearing.

After calibration, the feed lines were removed from the test bearing to mock-up the hydrodynamic liquid metal bearing test. Initial zero supply pressure startup with the hydrodynamic bearing was made with a very low 0.04 oz-in shaft unbalance. The unbalance was gradually increased to a maximum of 0.70 oz-in. The speed was varied from zero to 10,000 rpm. Data was taken of shaft displacement and speed, test unit and stand vibration, and lubricant temperature.

For the hydrostatic bearing tests, orifices were placed in suitable fittings in each of the four supply pressure feed lines. Oscilloscope calibration was carried out as mentioned previously. Start-up was no problem with these bearings since they were orifice compensated and had appreciable load carrying capability at zero shaft speed. A 50 psig supply pressure to the orifices was used since this was proposed for the SNAP-50 pumps. The bearing was tested at speeds from zero to 10,000 rpm. Load was applied to obtain eccentricity ratios of zero to 0.6 in increment of 0.1. Data was taken of speed, load, shaft position, supply and pool pressures, lubricant flow rate, and temperature.

Operating procedure for the hybrid step bearings was similar to that of the hydrostatic bearing. The only addition was that the hybrid bearings were tested both with and without orifices.

IV. RESULTS OF TESTS

The results of the single journal bearing tests are shown in Fig. 7 through 15 inclusive. Fig. 7 shows the displacement of the journal within the hydrodynamic bearing due to the shaft's speed and unbalance. This displacement is illustrated as synchronous whirl patterns in Fig. 8. The influence of stand resonance on bearing performance is also indicated in Fig. 7, as a peak in a slowly rising curve. At 10,000 rpm the displacement of the shaft was still within an eccentricity of less than 0.5. Also, many repeated start-ups were made, with zero supply pressure, without any detrimental affect on the bearing.

Figs. 9 and 10 show the load carrying capability and flow rates for the L/D 1.0 and 1.5 hydrostatic four pool bearings. Only limited data is shown for the L/D 1.5 bearing due to the premature bearing failure. The optimum orifice diameter was found to be between 0.050 and 0.070 inch. Flow in this range was between 1.6 and 2.7 gpm.

Figs. 11 through 14 show the load carrying capability and flow rates for the hybrid step bearings with 0.0006 and 0.0012 inch step heights. The no orifice data is shown with a 0.450 inch orifice as this was the bearings feed hole diameter. The curves show that the 0.0012 inch step bearing had a higher load carrying capability than the 0.0006 inch step bearing when no orifice compensation was applied. The 0.0012 inch bearing tended to be less responsive to orifice compensation than the 0.0006 inch step bearing. This resulted in the 0.0006 inch step bearing showing higher load carrying ability when orifices were installed. Since the hybrid bearings have a larger hydrodynamic component than the hydrostatic bearings, their stiffness was seen to change much faster with a change in shaft speed. Flow rates for the hybrid bearings were approximately one-third of the flow rate for the hydrostatic bearings when both types were compensated with optimum orifices.

Fig. 15 shows that the attitude angle of the hybrid step bearing was higher and increased at a faster rate than the attitude angle of the hydrostatic bearing. This is due to the higher hydrodynamic component of the step bearing.

An Indi-ron inspection report is included in Fig. 16 to illustrate the accuracy with which the test bearings were machined.

V. CONCLUSIONS

The following conclusions may be drawn from the study of these tests:

1. The hydrodynamic bearing test demonstrated successfully the ability to start, stop, and restart the test unit with no supply pressure.

2. The hydrodynamic bearing test also showed that a bearing of this type can operate with a substantial rotating load, but that test stand resonance may be a factor in successful operation.

3. The 4-pool hydrostatic bearing has a higher load-carrying capacity at 0 rpm than the step type hybrid bearing. The converse is true at 10,000 rpm.

4. When orifice compensation was applied to the hybrid step bearings, the one with a 0.0006 step depth had a higher load carrying capacity at speed than the one with a 0.0012 step depth.

VI. REFERENCES

1. G. M. Wood, "Status of SNAP-50/SPUR Pump Development Program", CNLM-6308, Pratt & Whitney Aircraft-CANEL, May, 1965.

2. P. T. Heyl, A. McKibben, J. Walton, H. Welna, "Summary of Design and Analysis of the SNAP-50/SPUR Reactor Coolant Pump", PWAC-467, Pratt & Whitney Aircraft-CANEL, August, 1965.

3. P. Odom, "Liquid Metal Bearings for Space Powerplant Applications", TIM-722, Pratt & Whitney Aircraft-CANEL, October, 1962.

4. D. V. Manfredi, "Summary of SNAP-50/SPUR Pump Bearing Development Program", CNLM-6342, Pratt & Whitney Aircraft-CANEL, July, 1965.

5. D. Groll, "Summary of Fluid Film Bearing Analysis", TIM-917, Pratt & Whitney Aircraft-CANEL, June, 1965.

6. D. V. Manfredi, "Summary of Shaft Face Seal Development Program", TIM-909, Pratt & Whitney Aircraft-CANEL, June, 1965.

7. J. L. Hoffman, "A Clearance Indicator for High Speed Bearing Journals", ISA Trans., October, 1962, Vol. 1, pp 329-339.

8. W. Richards, "Hydrodynamic Journal Bearing Tests in Lithium", TIM-915, Pratt and Whitney Aircraft-CANEL, July, 1965.

9. A. A. Raimondi and J. Boyd, "An Analysis of Orifice Compensated Hydrostatic Journal Bearings", Paper No. 54 LUB-17, ASME-ASLE Lubrication Conference, October, 1954.

10. R. R. Donaldson, Ph.D. Thesis, "Incompressible Journal Bearings with Combined Hydrostatic-Hydrodynamic Action", Department of Mechanical Engineering, Massachusetts Institute of Technology, June, 1965.

11. Groll, D., "Design Procedure for Externally Compensated Hybrid Bearings," TIM-826, Pratt & Whitney Aircraft-CANEL, May 1964.

12. Merz, C., "Straddle Bearing Rotor Dynamics Tests in Water," TIM-836, Pratt & Whitney Aircraft-CANEL, November, 1964.

13. Merz, C., "Straddle Bearing Rotor Dynamics Tests With Step Bearings in Water," TIM-921, Pratt & Whitney Aircraft-CANEL, July, 1965.

14. Newey, D. A., "Overhung Rotor Dynamics Tests Using Hydrostatic Water Bearings," TIM-874, Pratt & Whitney Aircraft-CANEL, June 1965.

15. Knudsen, L., "Single Hybrid Journal Bearing Tests in Water," TIM-925, Pratt & Whitney Aircraft-CANEL, July, 1965.

16. Knudsen, L., "Thrust Bearing Tests in Water," TIM-926, Pratt & Whitney Aircraft-CANEL, July 1965.

17. Richards, W., "Ball Bearing Test in Lithium," TIM-923, Pratt & Whitney Aircraft-CANEL, July 1965.

18. Wallace, M., "Summary Report of Potential Liquid Metal Bearing Materials for SNAP-50/SPUR Pumps," PWAC-468, Pratt & Whitney Aircraft-CANEL, August 1965.

FIG 1

SINGLE JOURNAL BEARING WATER TEST RIG

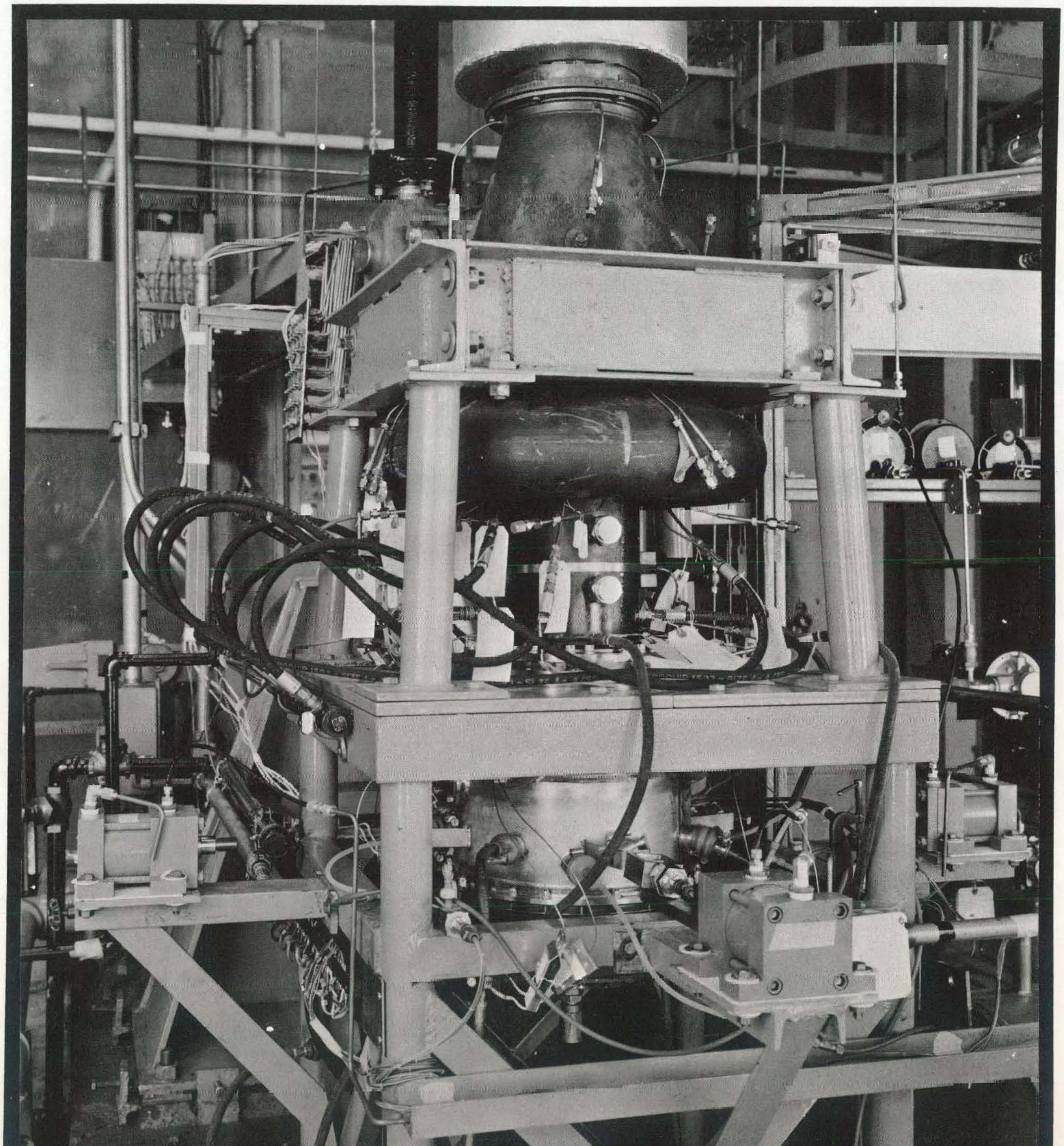


FIG 2

CROSS-SECTION OF SINGLE JOURNAL BEARING WATER TEST UNIT

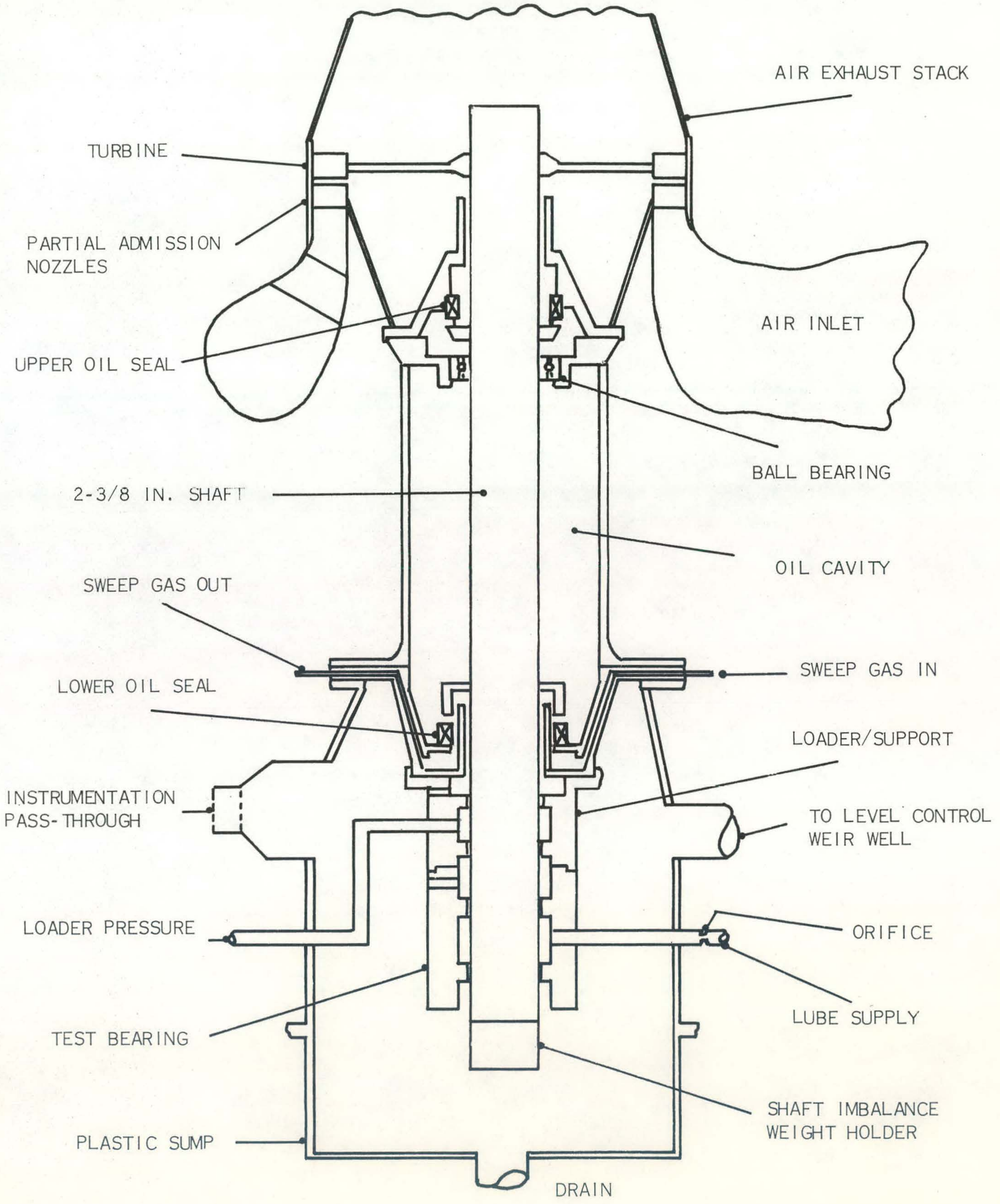


FIG 3

JOURNAL BEARING WATER TEST RIG FLOW DIAGRAM

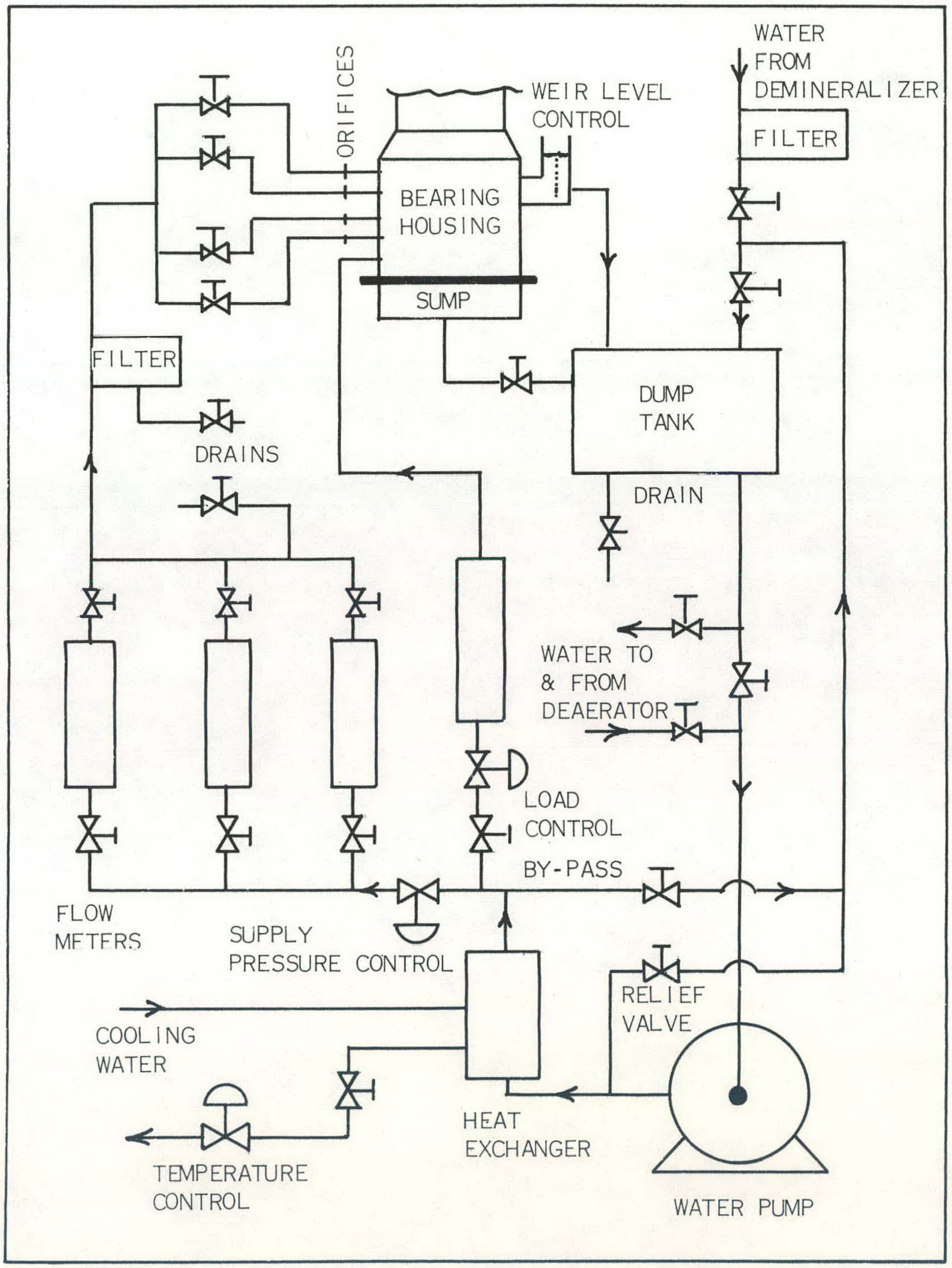
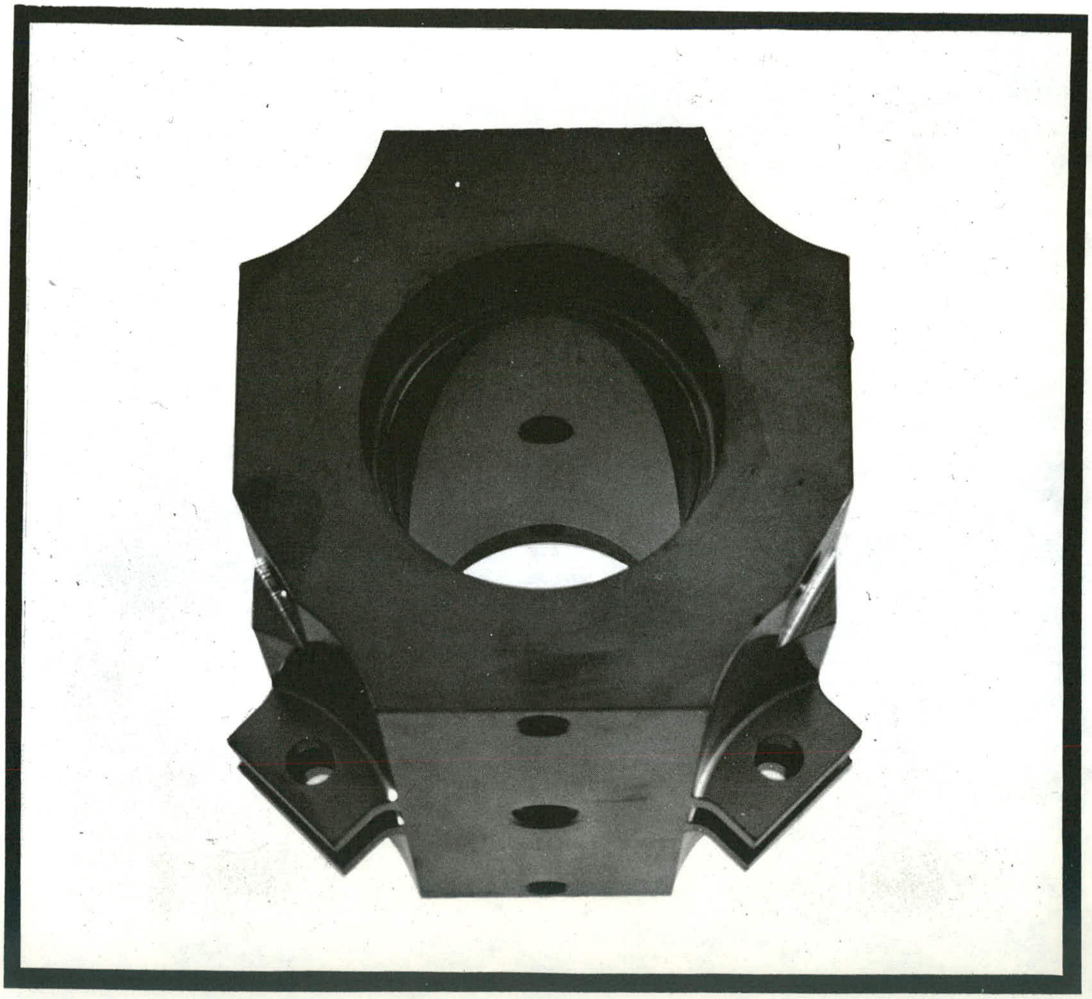


FIG 4

HYDRODYNAMIC BEARING CONFIGURATION

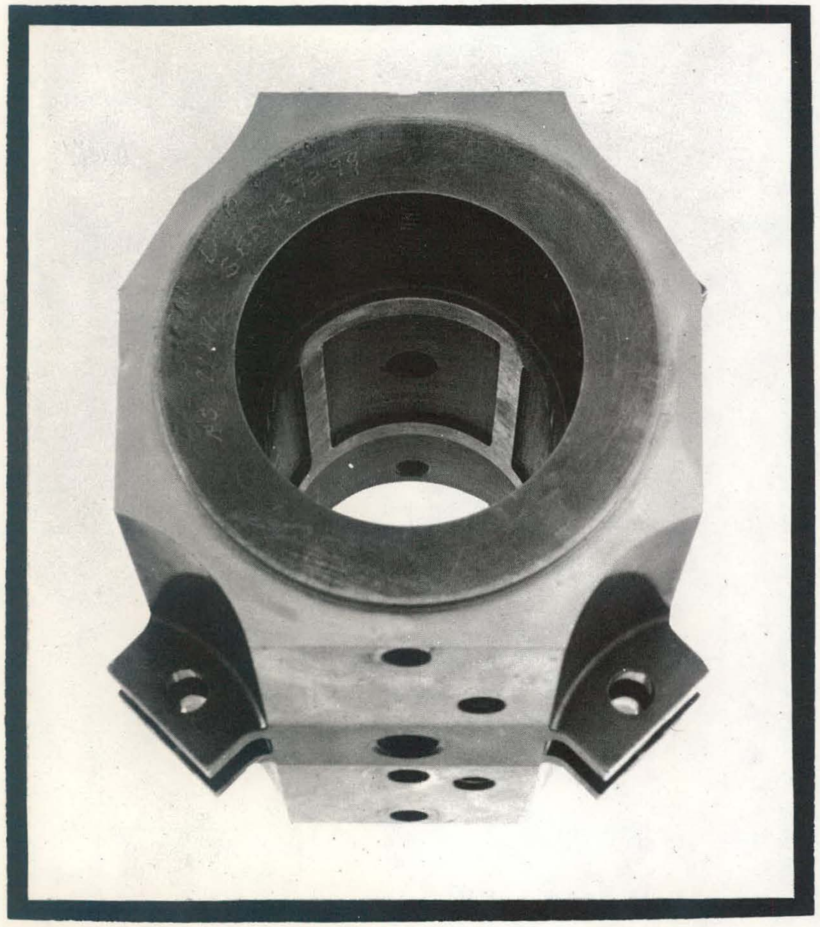


DIA. = 2.3490 IN.

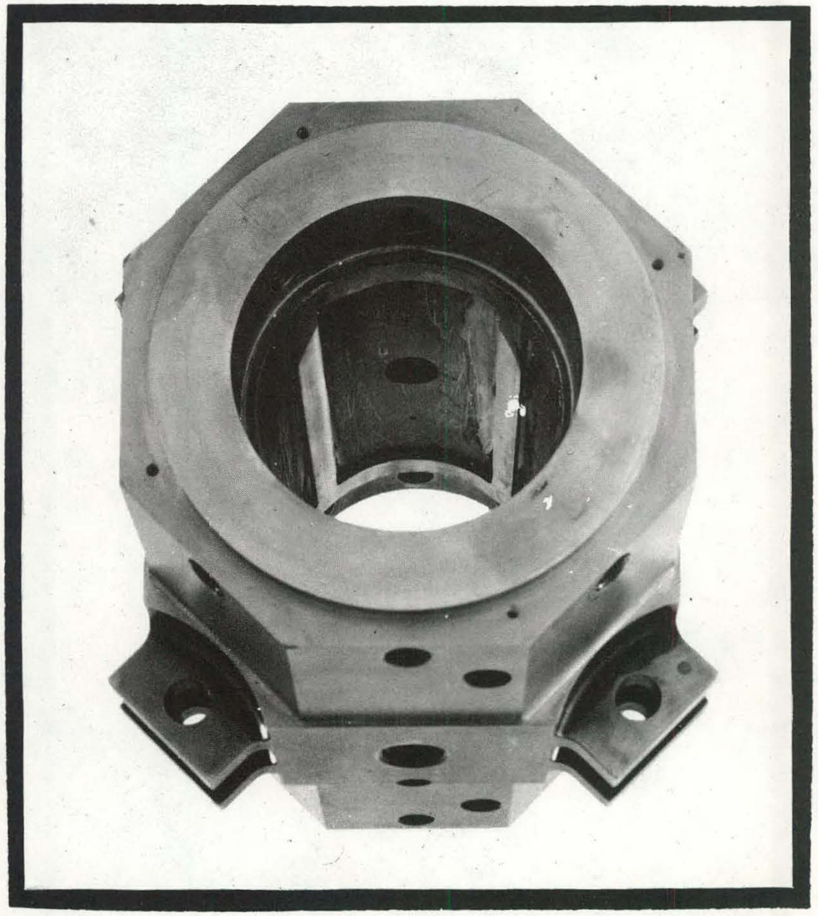
HYDROSTATIC BEARING CONFIGURATIONS

4-POOL, ORIFICE-COMPENSATED

DIA. = 2.3490"



L/D = 1



L/D = 1.5

FIG 5

FIG 6

STEP BEARING FOR WATER TEST

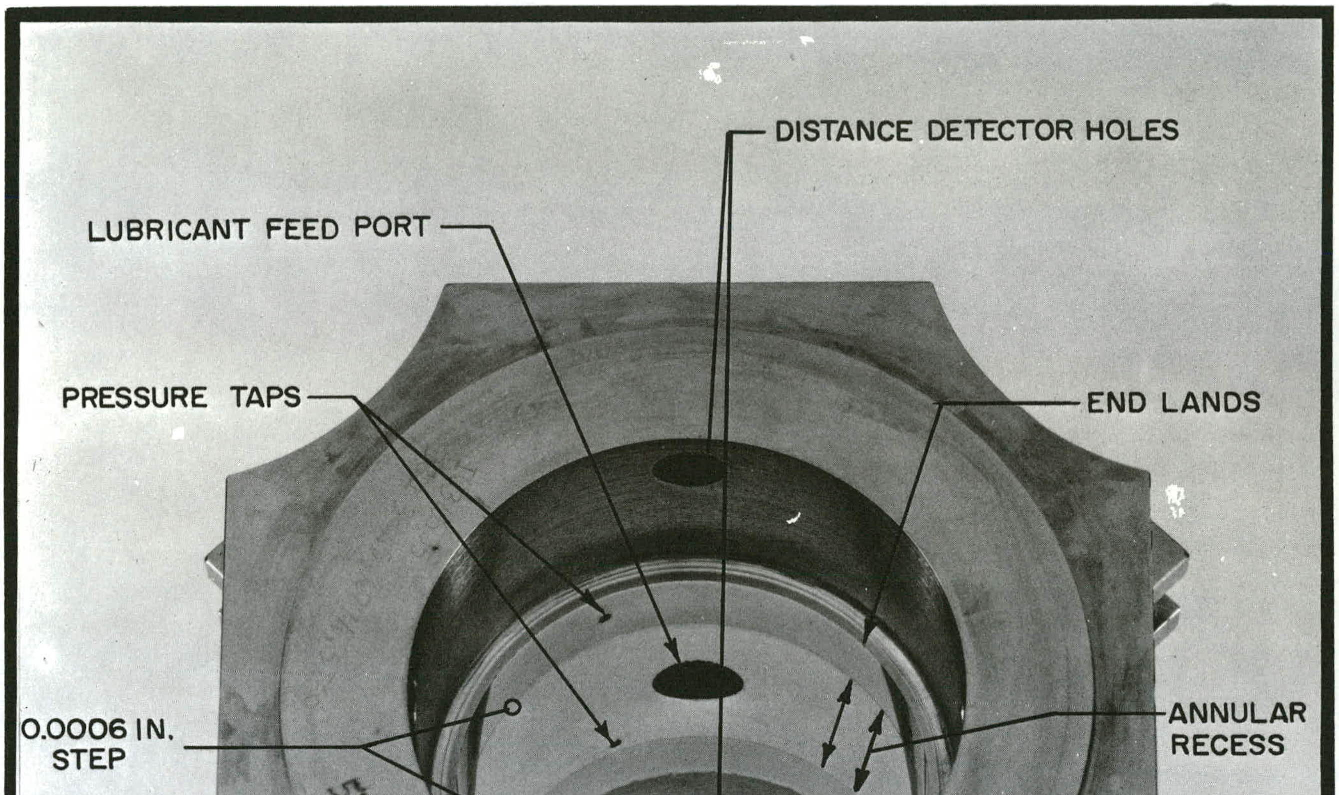


FIG 7

RADIAL SHAFT DISPLACEMENT AND TEST STAND VIBRATION VERSUS SHAFT SPEED FOR HYDRODYNAMIC BEARING

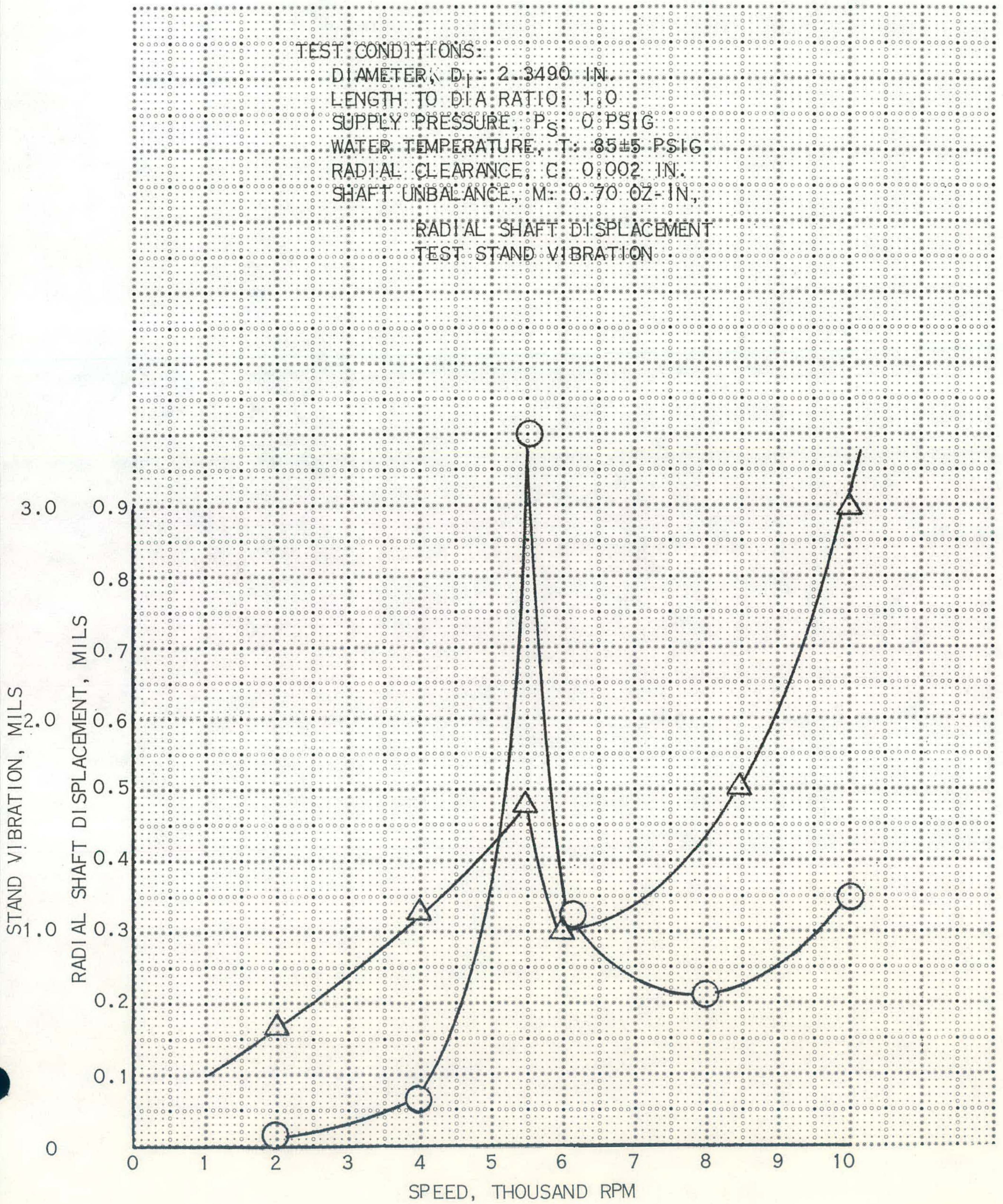


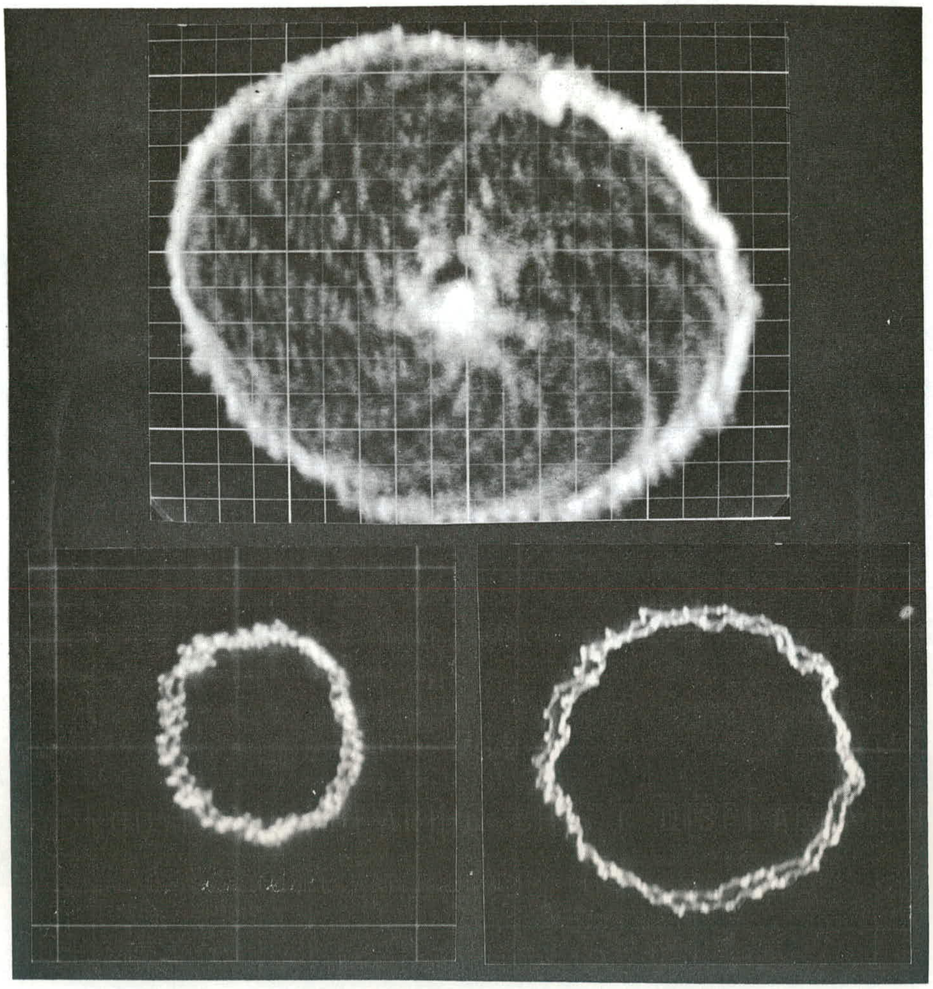
FIG 8

HYDRODYNAMIC BEARING SHAFT DISPLACEMENT

SINGLE WATER BEARING TEST

TEST CONDITIONS
DIAMETER, D_I: 2.3490 IN.
LENGTH TO DIA. RATIO: 1.0
RADIAL CLEARANCE, C: 0.002 IN.
SUPPLY PRESS, P_S: 0 PSI
WATER TEMP, F: 85±5F
UNBALANCE, M: 0.7 OZ-IN.

START-UP FROM ZERO
RPM



0.001 IN.

0.001 IN.

0.001 IN.

8500 RPM

10,000 RPM

FIG 9

LOAD VERSUS ORIFICE DIAMETER AT AN ECCENTRICITY RATIO OF 0.5

FOUR POOL HYDROSTATIC BEARINGS

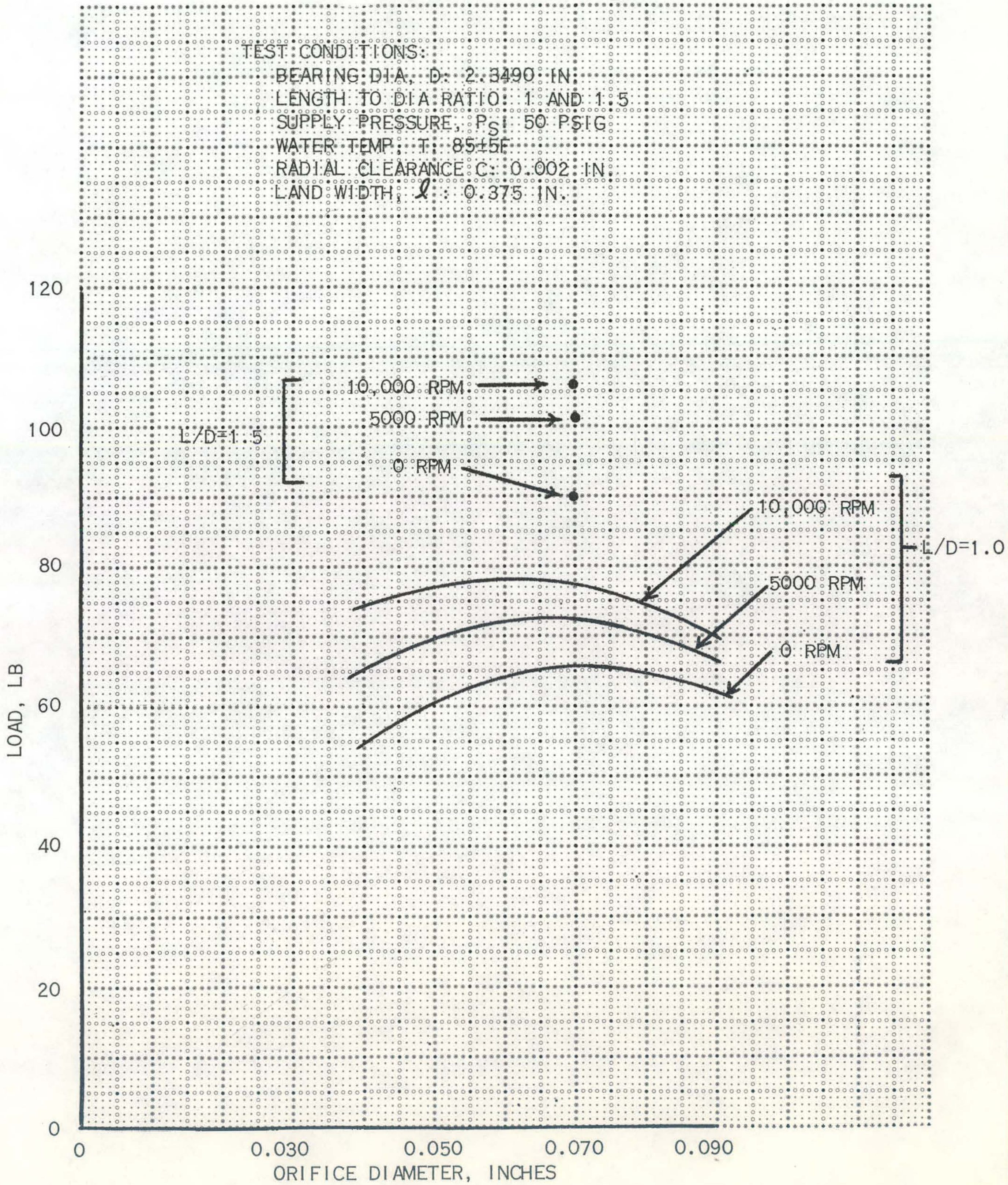


FIG 10

TOTAL FLOW RATE VERSUS ORIFICE DIAMETER

FOUR POOL HYDROSTATIC BEARINGS

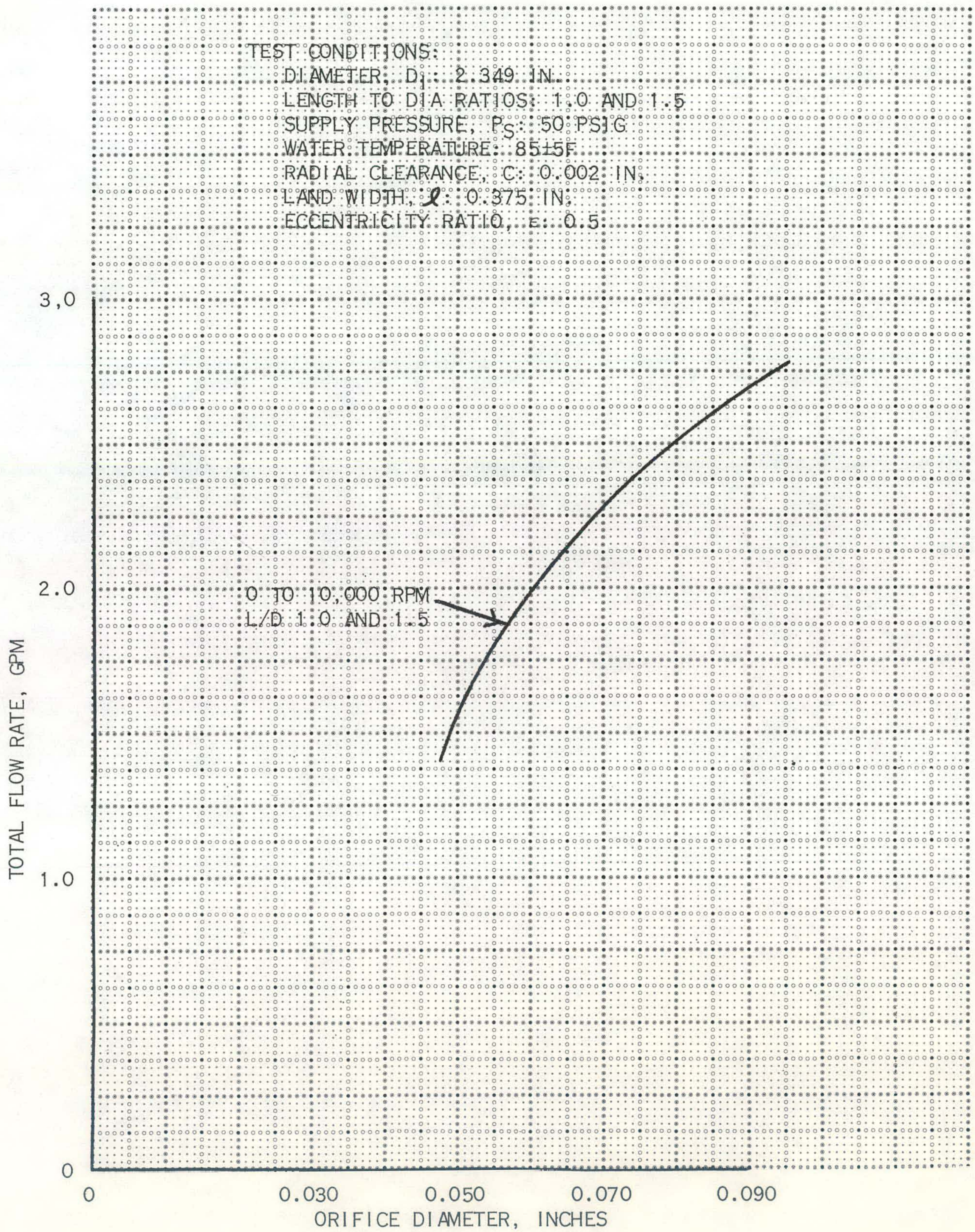


FIG 11

LOAD VERSUS ORIFICE DIAMETER FOR A STEP JOURNAL BEARING

SINGLE BEARING WATER TEST RIG

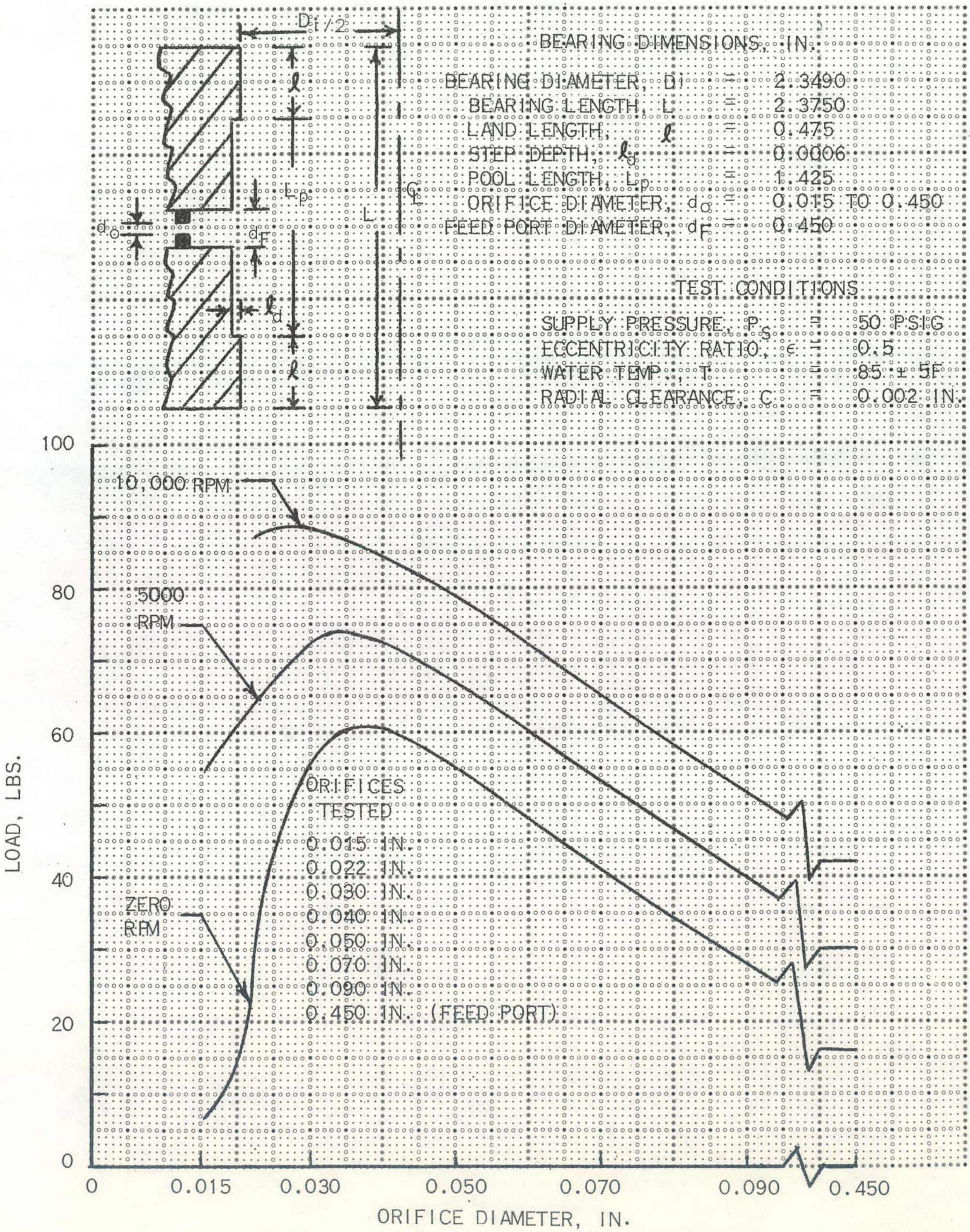


FIG 12

TOTAL FLOW RATE VERSUS ORIFICE DIAMETER FOR A STEP JOURNAL BEARING

SINGLE BEARING WATER TEST RIG

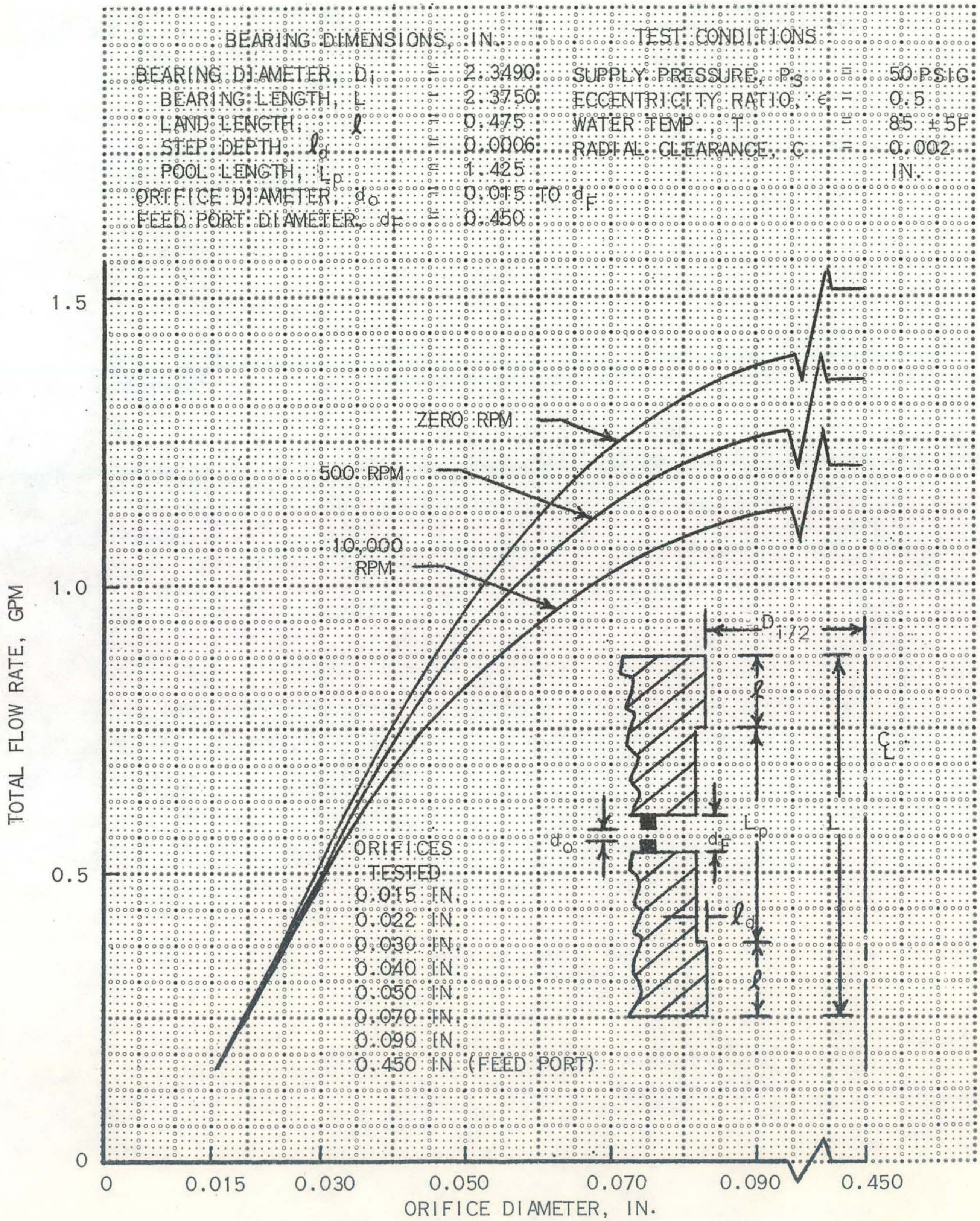


FIG 13

LOAD VERSUS ORIFICE DIAMETER FOR A STEP JOURNAL BEARING

SINGLE BEARING WATER TEST RIG

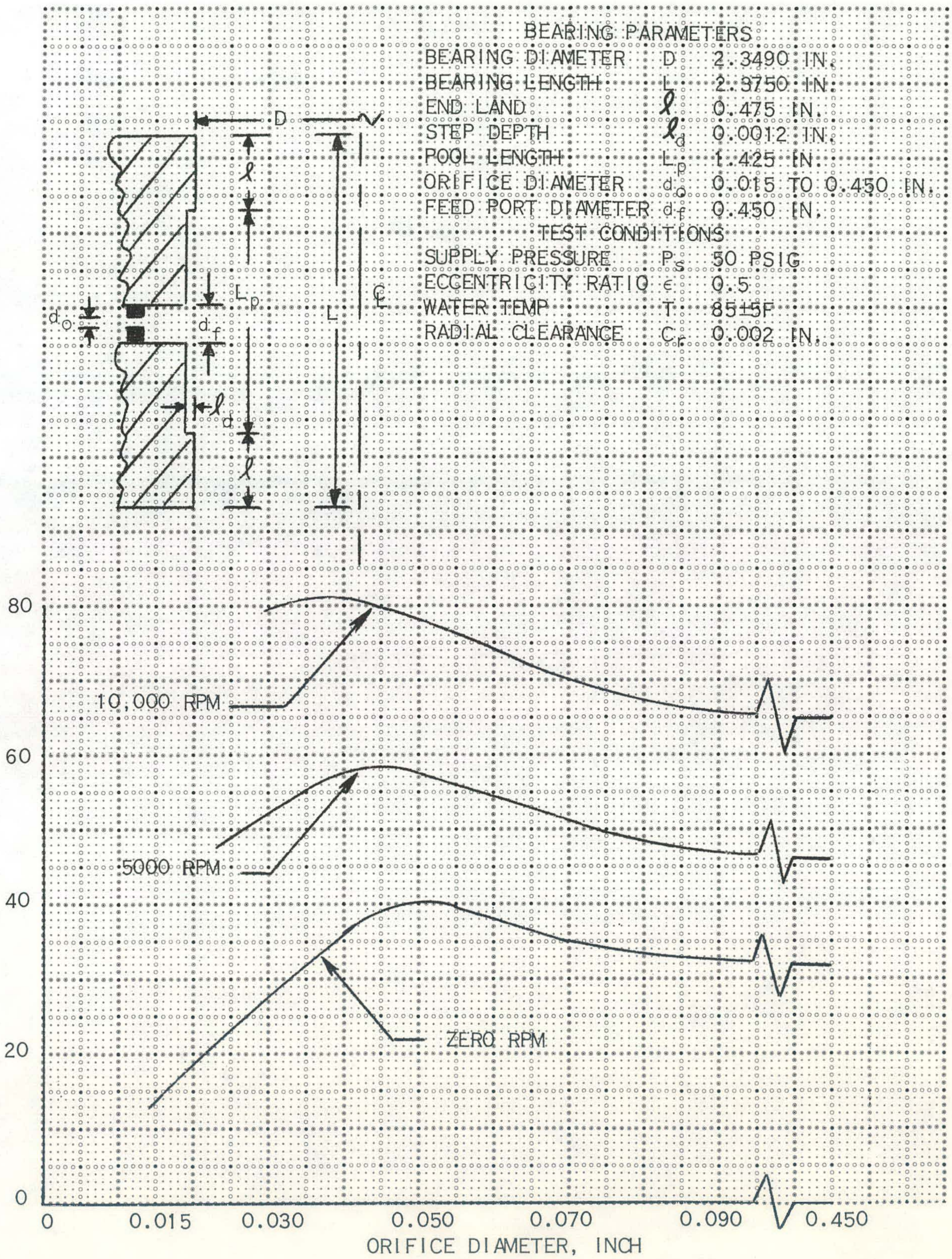


FIG 14

TOTAL FLOW RATE VERSUS ORIFICE DIAMETER FOR A STEP JOURNAL BEARING

SINGLE BEARING WATER TEST RIG

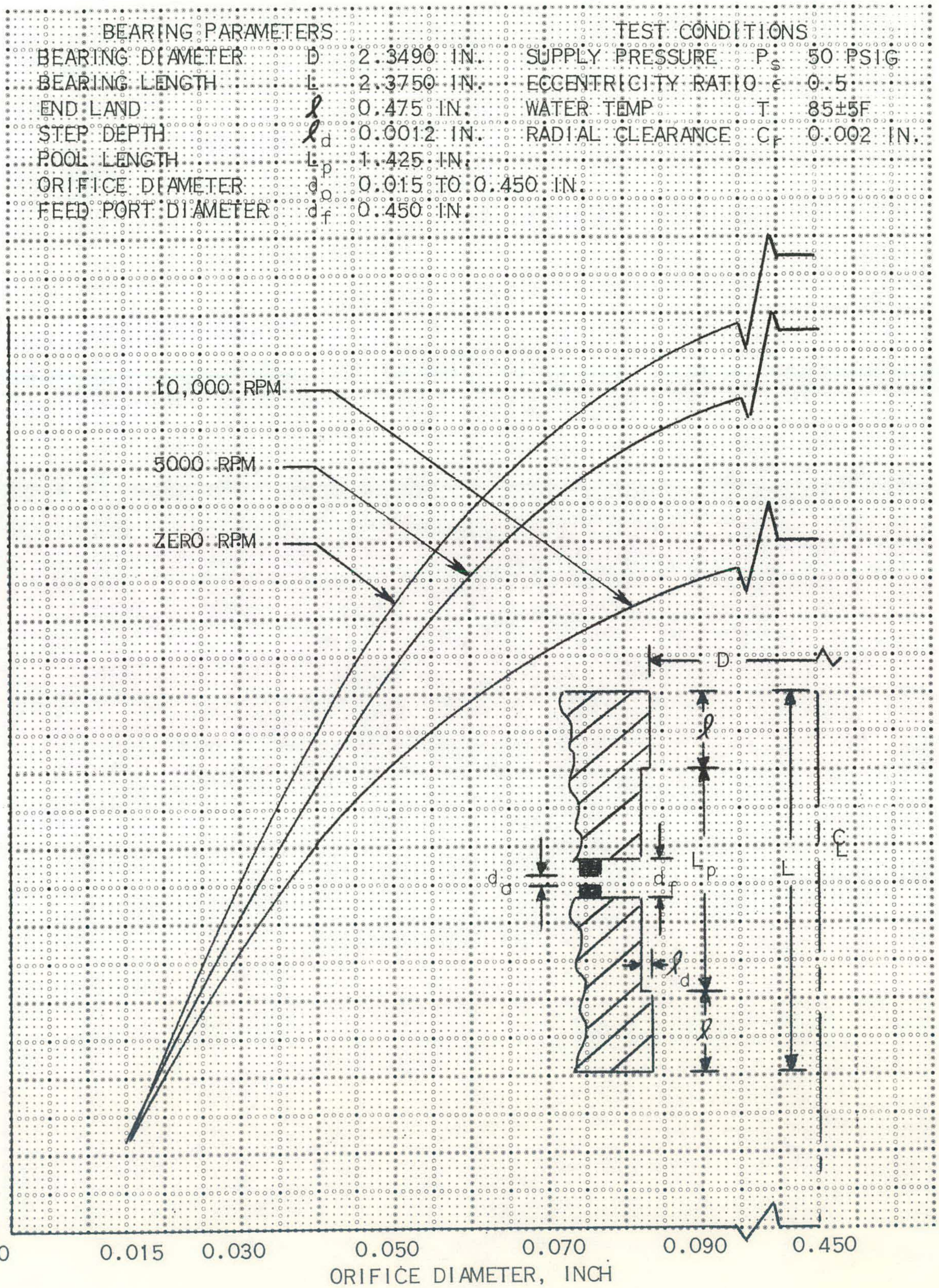


FIG 15

ATTITUDE ANGLE VERSUS SHAFT SPEED FOR WATER JOURNAL BEARINGS

STEP AND FOUR POOL HYDROSTATIC BEARINGS

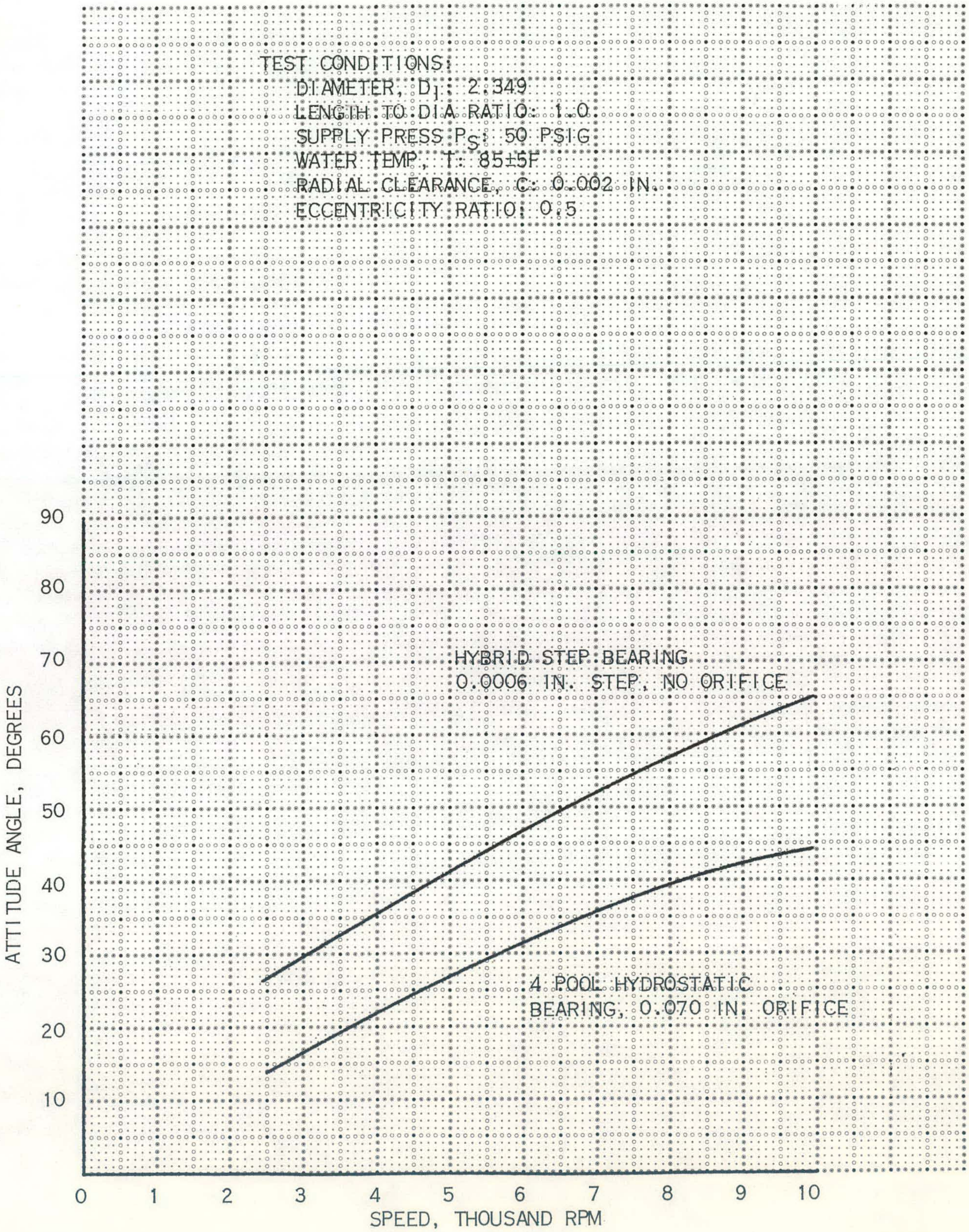
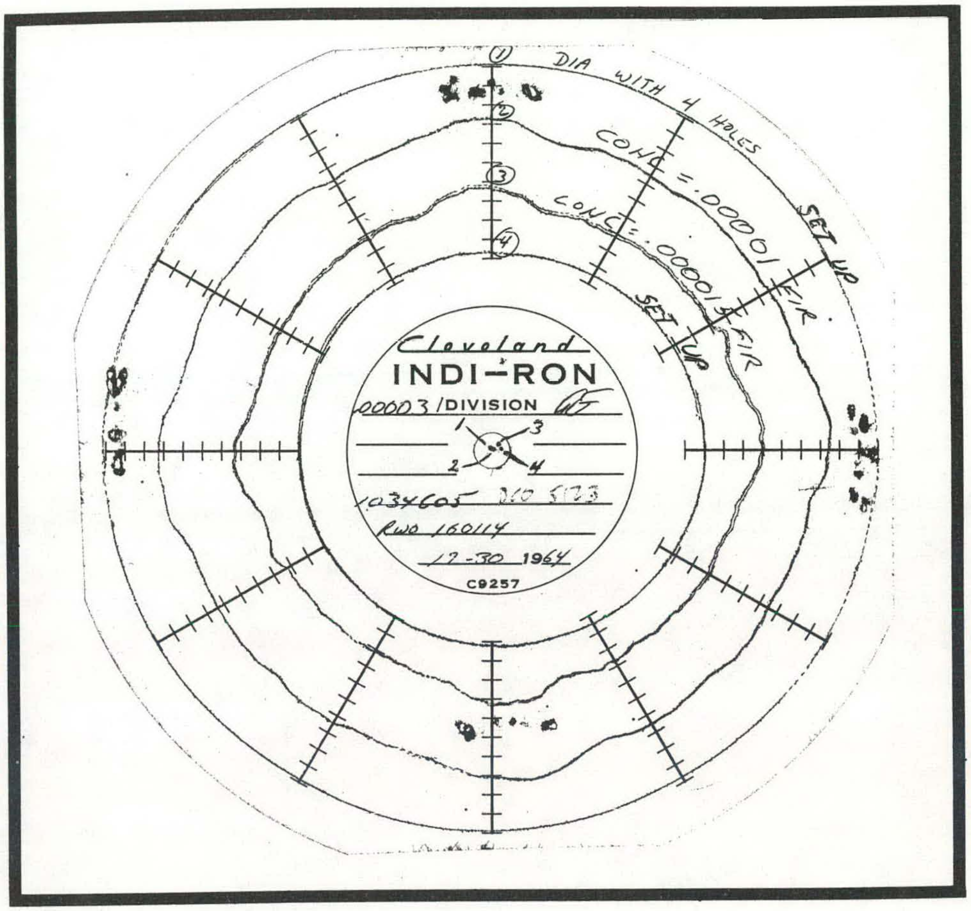
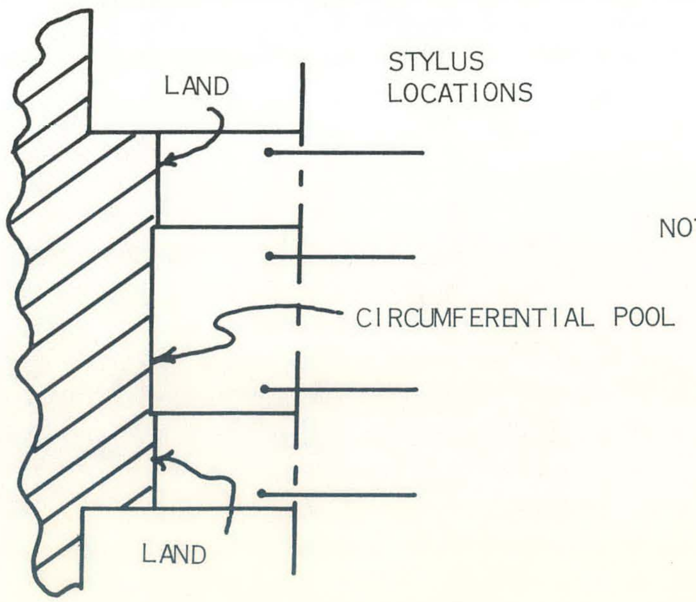


FIG 16

HYBRID STEP BEARING CONCENTRICITY MEASUREMENTS

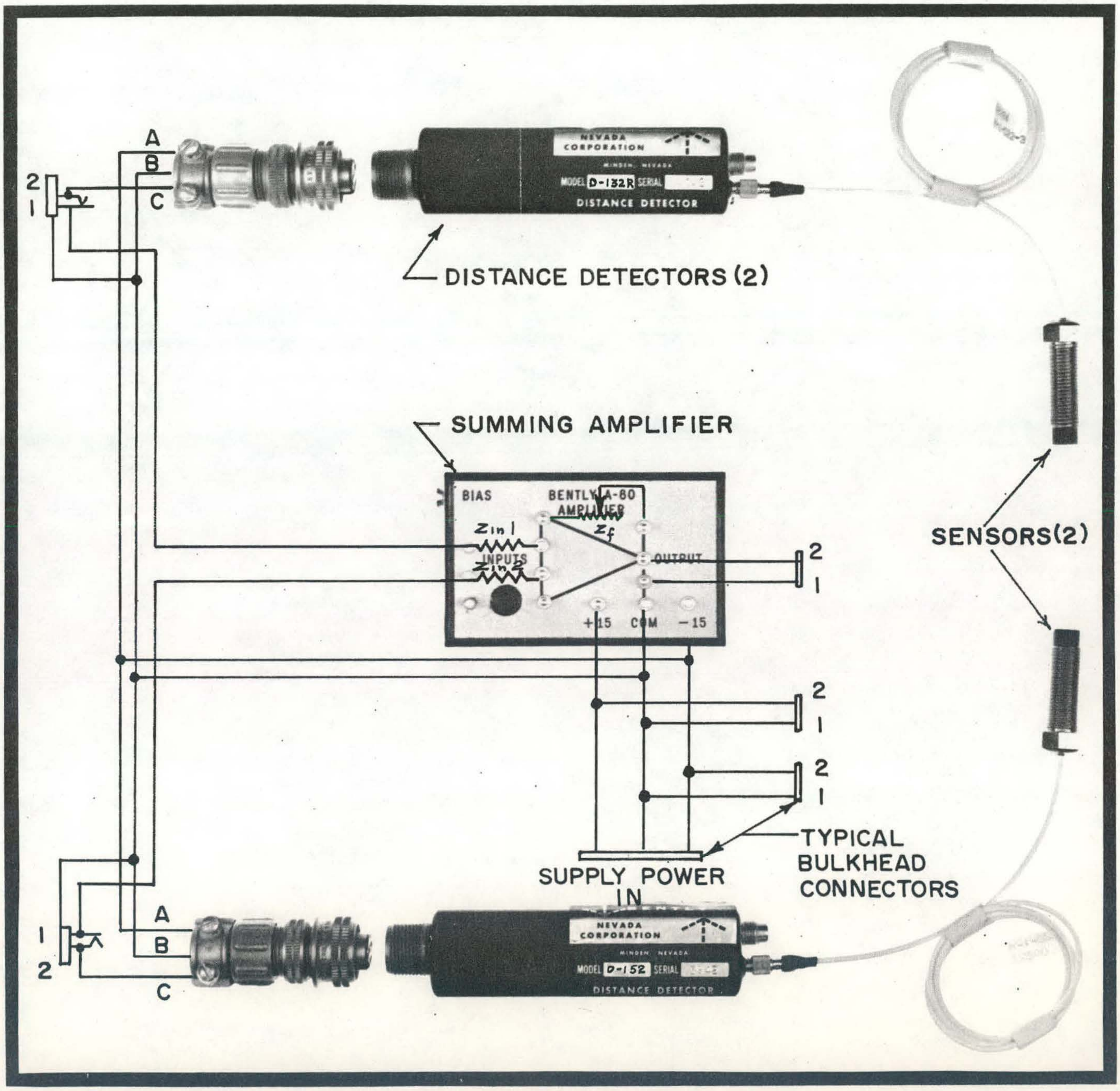




NOTE: THE CONCENTRICITY OF POOL TO LANDS WAS $15X10^{-6}$ IN. FIR.

FIG 17

BENTLY NEVADA DISTANCE DETECTORS AND SUMMING AMPLIFIER FOR WATER BEARING TEST UNITS



NOTE:

1. POINTS 1 AND 2 ARE VOLTAGE TAPS