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ORNL
CENTRAL FILES NUMBER
59 - 2 - 1

COPY NO. 40

DATE: February 1, 1959

SUBJECT: Hydrostatic Journal Bearing Water Tests Conducted in Modified PK-A Pump

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Abstract

A hydrostatic journal bearing mounted near the impeller of an overhung vertical shaft centrifugal pump was subjected to water testing as a part of the molten salt lubrication investigation at ORNL. Three tests were performed with bearings having radial clearances of 0.003 in., 0.0075 in., and 0.005 in. The first journal and bearing (0.003 in. radial clearance) were found to be heavily scored after testing. Only faint localized scratches were found on the second journal (0.0075 in. radial clearance) and these may have been caused by the many test starts and stops. Localized scratches, somewhat deeper than those on the second journal, were observed on the third journal, but no measurable wear had occurred from testing. An apparent inconsistency was noted in that at the same pump operating condition the bearing load as computed from pocket pressure data increased by a factor of 1.2 to 1.7 as bearing radial clearance was increased from 0.005 in. to 0.0075 in. Bearing load test data and their reduction are omitted from this report. Satisfactory operation with molten salt lubricated hydrodynamic journal bearings prompted a decision to de-emphasize hydrostatic bearing testing prior to resolving the bearing load inconsistency. The configuration of the submerged hydrostatic bearing used in these tests appears to be satisfactory for use as a lower journal bearing in this size and type of centrifugal pump, at least insofar as operation in water is concerned.

DISCLAIMER

Introduction

The purpose of the tests was to investigate the performance and reliability of the hydrostatic type journal bearing with water as the lubricant prior to testing it with molten-salt as the lubricant at elevated temperatures (1200°F and up). The hydrostatic bearing is one of three approaches to the development of a molten-salt lubricated journal bearing. The other two approaches are the hydrodynamic journal bearing and the rotating pocket hydrostatic journal bearing (with orifice compensation).

The tests were conducted by the Experimental Engineering Department of the Reactor Projects Division at Oak Ridge National Laboratory. They are a part of the molten-salt lubricated bearing development program which was evolved during the last half of FY-58, under auspicies of the Molten Salt Reactor Program.

Should molten-salt lubricated bearings, either hydrodynamic or hydrostatic, become a reality, they would be extremely useful elements in the design of a pump for a molten-salt reactor. They could be used to expand the usefulness and to increase the flexibility of design for the vertical overhung type of sump pump. In the event that high temperature, radiation-resistant electric motors become available, they would permit the realization of a completely submerged pump, requiring no auxiliary oil lubrication circuits with their attendant complexities.

A model PKA pump was modified to accommodate a hydrostatic journal bearing and to provide the lubricating water under pressure from the pump volute. Three water test bearings were operated. Each bearing was of identical configuration except that various bearing clearances and compensating orifice diameters were tested as outlined in "Selection of Test Bearings".

Selection of Test Bearings

The sizes of the test bearings were based on a report by Raimondi and Boyd[1]. This report includes theory and test results based on static tests with two cases of bearing loading: radially through the bearing land and

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[1]Raimondi, A. A., and J. Boyd, An Analysis of Orifice- and Capilliary-Compensated Bearings, ASLE-ASME Joint Conference on Lubrication, Baltimore, (October 18-19, 1954).

radially through the center of the bearing pocket. The analysis was thought to be general enough to use on a rotating bearing, since, in throry, the total flow through the bearing is little affected by rotation, and the load capacity is increased by rotation of the journal. Also, the analytical results indicated very little dependence of load capacity on the load angle relative to the pockets.

The first bearing chosen for test consisted of a bearing and journal having a radial clearance of 0.003 in. and 0.500 in. diameter orifices. The supply pressure to this bearing was significantly lower than was anticipated, probably due to pressure losses in the inlet passages. Computations indicated that a compensating orifice diameter of 0.375 in. and radial clearance from 0.004 to 0.010 in. would provide satisfactory bearing performance for pump speeds ranging from 600 to 3000 rpm and widely varying system flow resistances, i.e. bearing loads. Bearings and journals with radial clearances of 0.0075 in. and 0.005 in., and compensating orifice diameter of 0.375 in. were used in the second and third tests. Each of these bearings was tested at approximately 45 pump operating points with speeds varying from 600 to 3000 rpm, with loads varying from 3 to 160 lb_f., and with bearing supply pressures ranging from 3 to 65 psi.

Test Facility

The bearings were mounted in a modified PK-A pump, as shown in Figs. 1 and 2. One of the test bearings, which contains four pockets with orifice compensation, is shown in Fig. 3. The bearings and journals were constructed of stainless steel with an overlay of Stellite 6. The bearing was supplied with water from the pump volute. The loads on the bearings were provided by the pressure unbalance in the volute and by the dynamic unbalance in the rotating shaft impeller.

The pump drive motor was connected to the pump rotary assembly previous to installation in the test loop, and tests were made to ensure that no loads were applied to the hydrostatic bearing by misalignment of the motor shaft to the pump shaft. It was found that the flexible coupling used permitted radial displacement of the test journal in the bearing with very little resistance.

In the first bearing, sixteen pressure pickups were connected to the bearing system, four for each bearing pocket. For each pocket, a pressure line was connected to the feed manifold, the pocket manifold feeder, the pocket manifold, and the pocket itself (see Fig. 1). These were connected through a valving arrangement to four bourdon tube pressure gages. Due to space limitations, all four of the pocket manifold feeder pressure lines and one feed manifold line were removed for the second and third bearings tested. A U-tube manometer was connected to the system in such a manner that the pressure differential could be measured between opposite pockets, and across all supply orifices.

After tests on the first bearing were concluded, it was decided that an attempt to measure journal eccentricity during dynamic operation should be made. Air nozzles supplied at constant pressure, and having a range of sensitivity of output signal vs journal displacement large enough to encompass the full swing of the journal, were applied. These nozzles were individually calibrated, duplicating as nearly as possible the operating conditions that would be encountered in the pump. Three nozzles spaced 90° apart were used with the second and third bearing tests. Two adjacent nozzles would have been sufficient; the third nozzle was added as a check on the first nozzle. Due to severe space limitations the tubes connecting the nozzles to the supply lines were very small in diameter (an ID of 0.096 inches). The flow resistance in this tubing was so high that the measurement sensitivity was not as good as desired.

The test pump was installed in a modified test loop (PKA-2) having an additional throttling valve installed to increase the range of high resistance to pump flow. This was done to obtain greater bearing loads by operating the pump far from the design point.

All of the compensating orifices used in the tests were calibrated in place in the bearing housing, using a weighing tank and a stop watch. A typical calibration curve is shown in Fig. 4.

Test Methods and Test Operation

Three bearing designs were tested, each design being subjected to basically the same test program. Each bearing was tested at nine pump speeds ranging from 600 rpm to 3000 rpm in increments of 300 rpm along constant flow resistance lines (constant valve settings in flow loop). At least four constant resistance lines were identical for the three bearings. These resistance lines are shown in Fig. 5.

No long time tests were run during the test program. Steady state operation was obtained for each run prior to data taking. Each run took approximately 25 minutes, thus the total running time for each bearing was about 30 hours, including rerun time. All the data points were rerun for all the resistance lines, with the exception of lines N, O, and P. For these lines the data for pump speeds below 2100 rpm were omitted.

Test Results and Discussion

Upon completion of the tests on the first journal and bearing, a large amount of rubbing wear could be seen on the bearing and journal (Figs. 6 and 7). Data revealed wide variations in the instantaneous value of supply pressures to each of the four hydrostatic pockets. Prior to start of testing of the second bearing, the bearing supply passages were modified and the variation in the instantaneous value of pressure supply among the four pockets were reduced to 0.5 psi.

The bearing and journal used in the second tests (Figs. 8 and 9) experienced only slight wear from rubbing. Wear experienced with the third bearing and journal (Figs. 10 and 11) was only slightly more than that experienced with the second journal and bearing. The wear in these tests may be attributed to the operating condition during the many starts and stops.

The bearing loads were computed, using pocket pressure data, to be greater with the second bearing (radial clearance 0.0075 in.) than with the third bearing by a factor of from 1.2 to 1.7 at identical pump operating conditions (same flow resistance and speed). Each pocket pressure is, also, the downstream pressure of the complimentary compensating orifice. Each orifice had been calibrated to provide bearing flow measurements. Further testing would be required prior to reporting on bearing load and flow performance, therefore no bearing load and flow data are presented in this report.

Operating experience with the air gauge method of measuring the displacement of the rotating loaded journal indicates that some development would be required to bring this method to a state of usefulness. Therefore, no journal concentricity data are included in this report.

The test data and log books pertaining to this investigation are retained in the files of the Pump Development Group of Experimental Engineering, Reactor Projects Division.

At this time, further work with the hydrostatic bearing has been deferred in favor of testing a molten salt lubricated hydrodynamic type journal bearing[2], a molten salt lubricated thrust bearing, and an actual molten salt pump demonstration containing one salt-lubricated hydrodynamic journal bearing.

Conclusions and Recommendations

Satisfactory pump operation was obtained with the second and third bearing. The bearing loads computed from the experimental data appear inconsistent in that the loads with the 0.0075 in. radial clearance bearing exceeded those with the 0.005 in. radial clearance bearing by a factor of 1.2 - 1.7 at identical pump operating conditions of head, flow, and speed. Further study and possibly further experimental work are indicated to resolve this apparent inconsistency. A bearing configuration was developed for the test pump which should operate satisfactory in elevated temperature molten salt.

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[2] Smith, P. G., Salt-Lubricated Hydrodynamic Journal Bearing Tests Nos. 1 and 2, ORNL-CF 58-8-10 (August 7, 1958).

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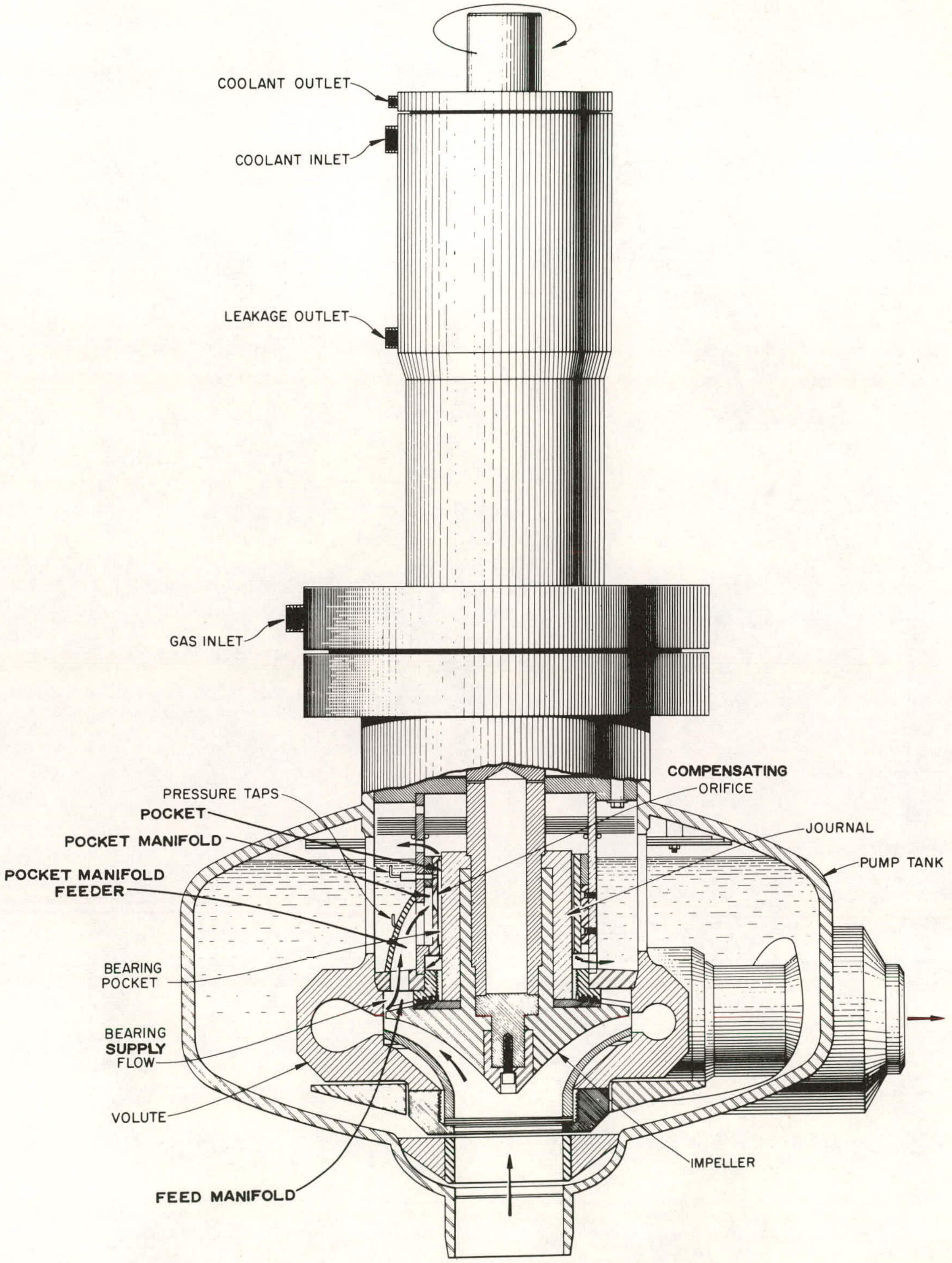

Fig. 1

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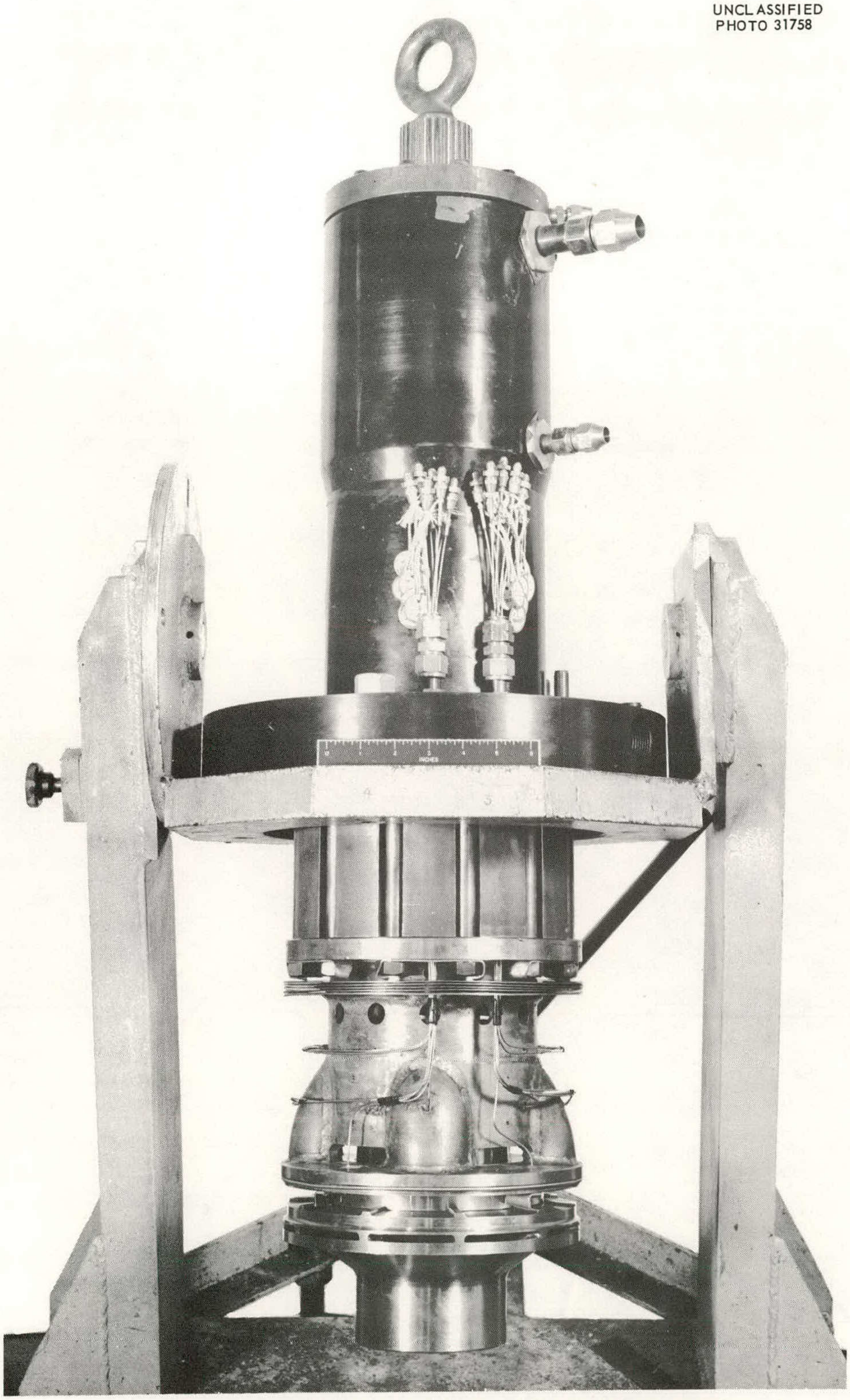

Fig. 2

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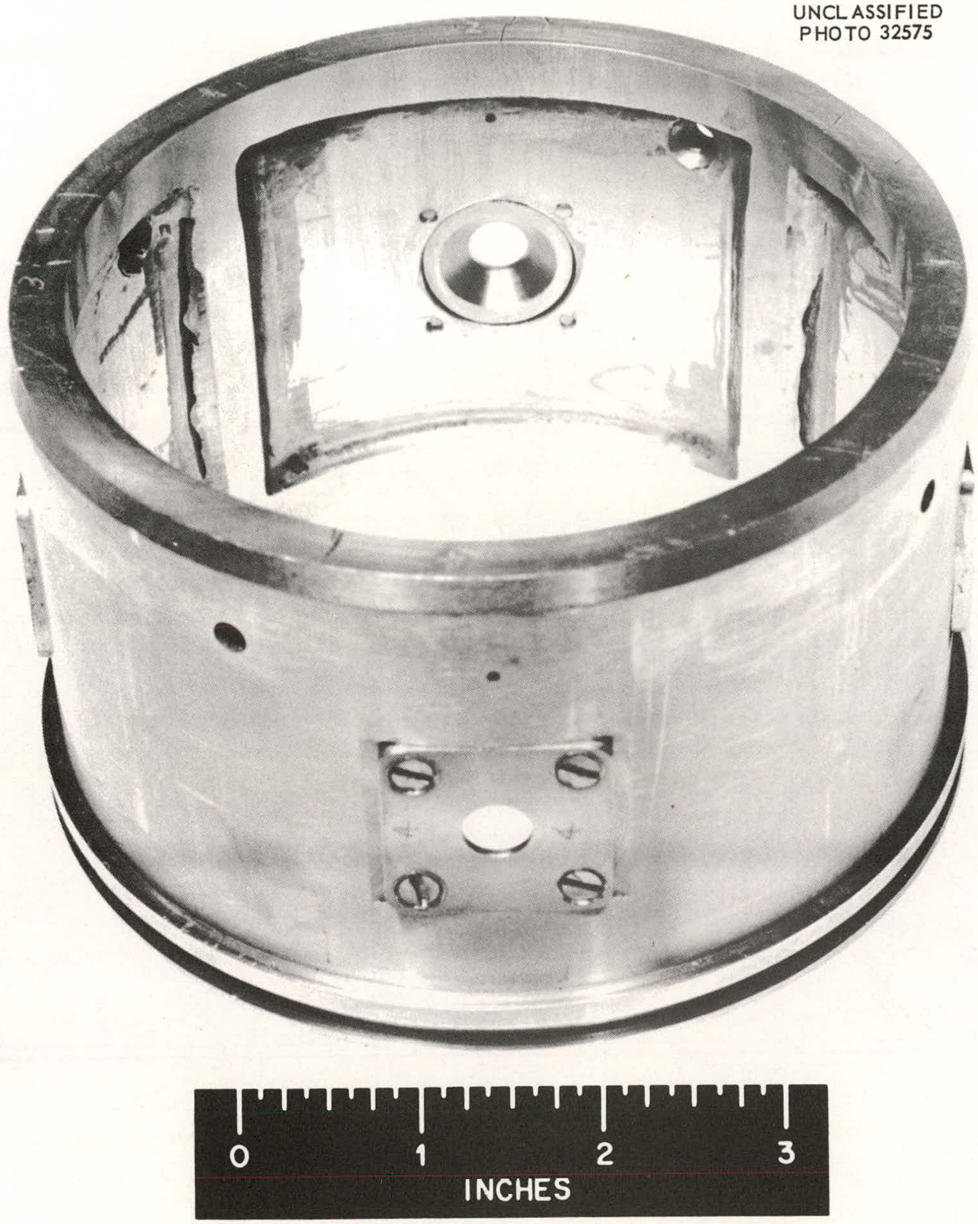

Fig. 3

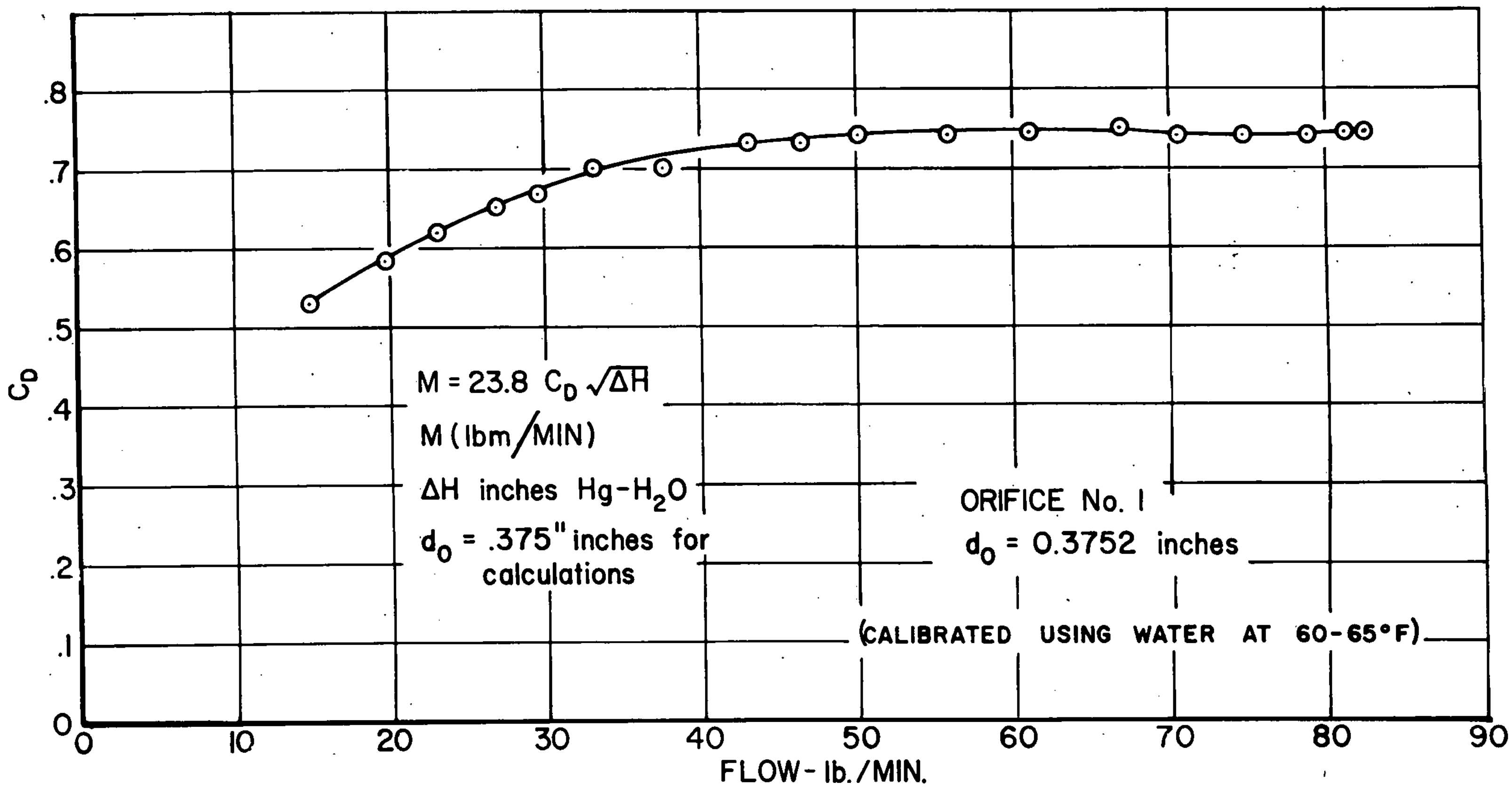
.8
.7
.6
.5
C_D
.4
.3
.2
.1
0
0
10
20
30
40
50
60
70
80
90
FLOW - lb./MIN.
$M = 23.8\ C_D \sqrt{\Delta H}$
M (lbm/MIN)
ΔH inches Hg-H_2O
d_0 = .375" inches for calculations
ORIFICE No. 1
d_0 = 0.3752 inches
(CALIBRATED USING WATER AT 60-65°F)

Fig. 4

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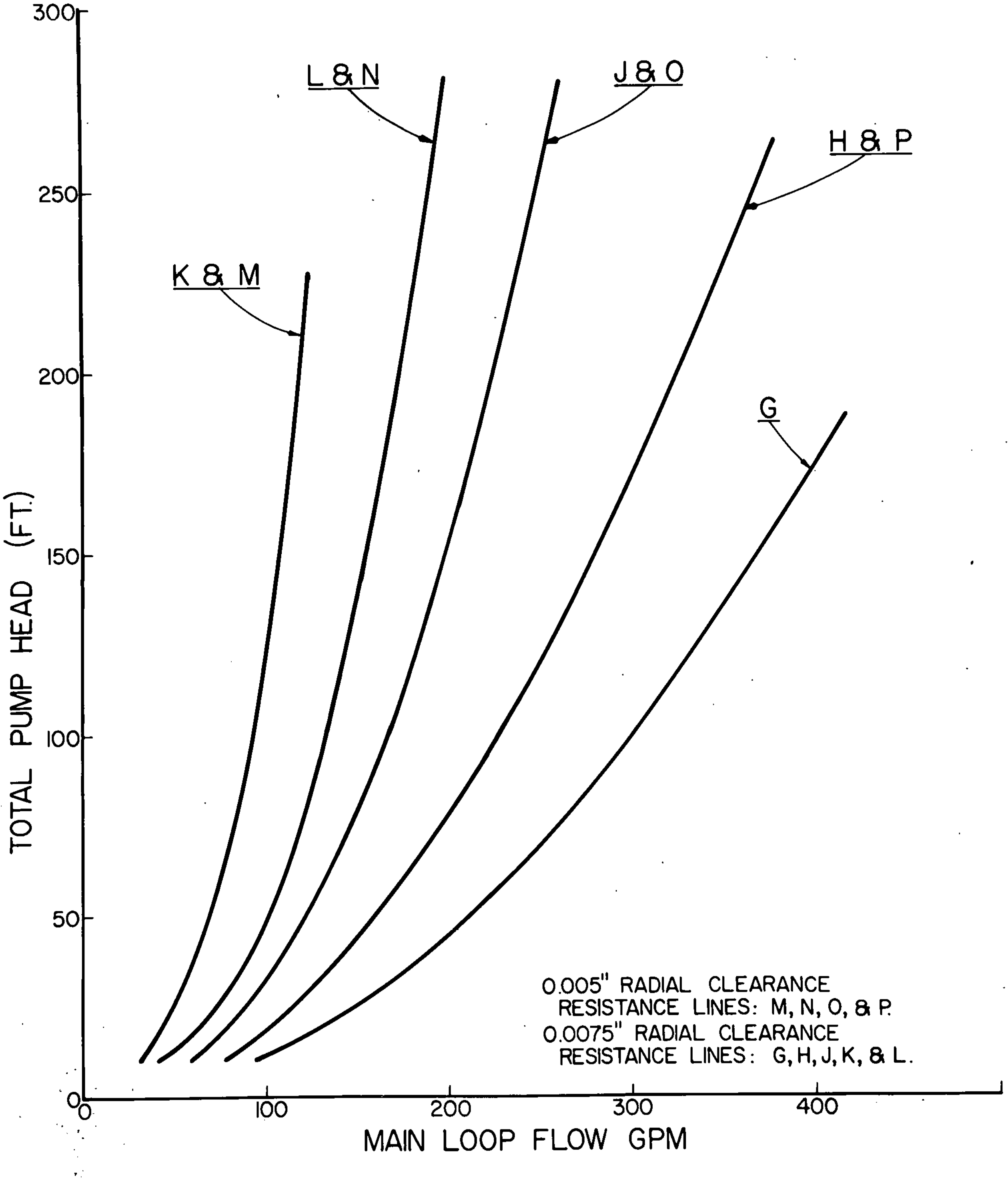

Fig. 5

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Fig. 6

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Fig. 7

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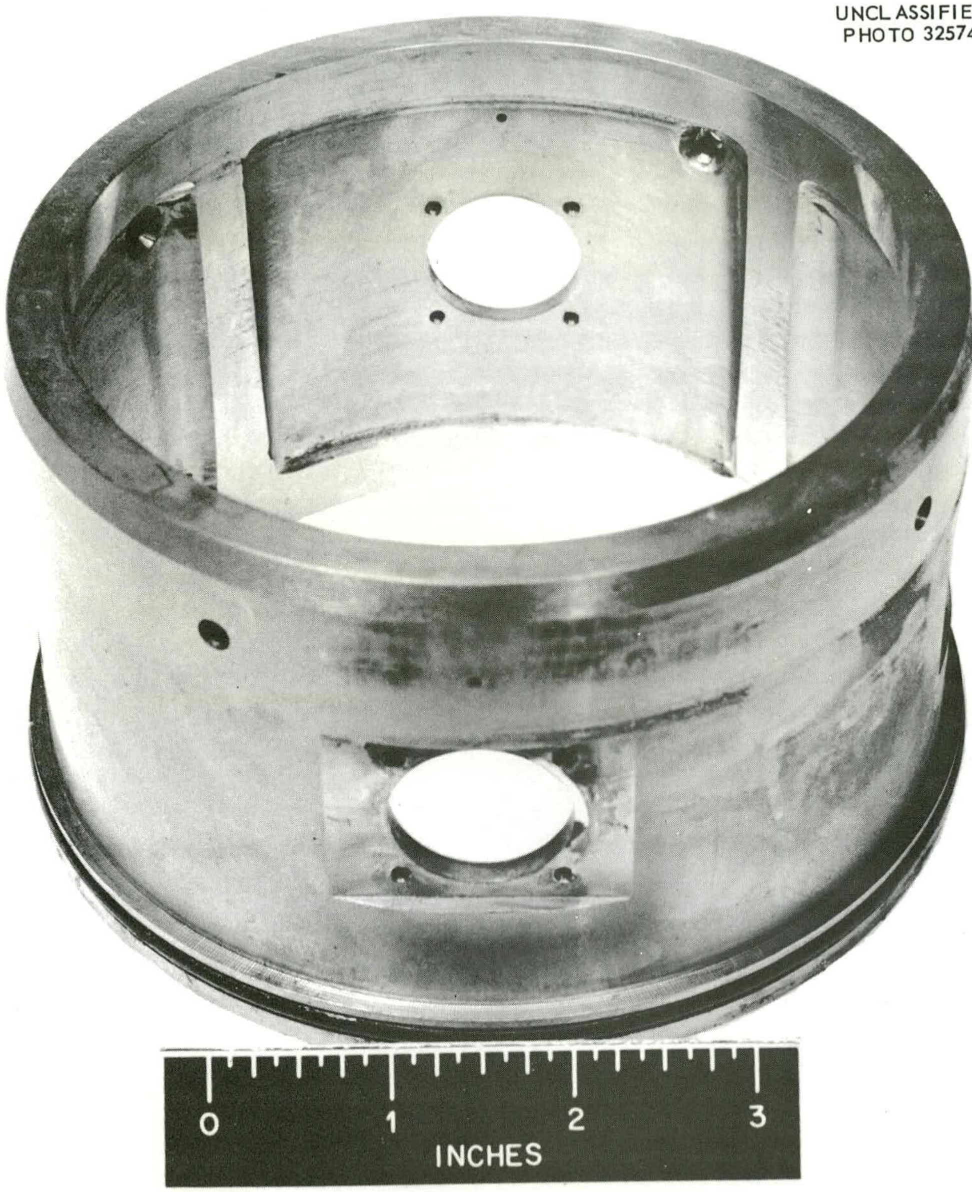

Fig. 8

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Fig. 9

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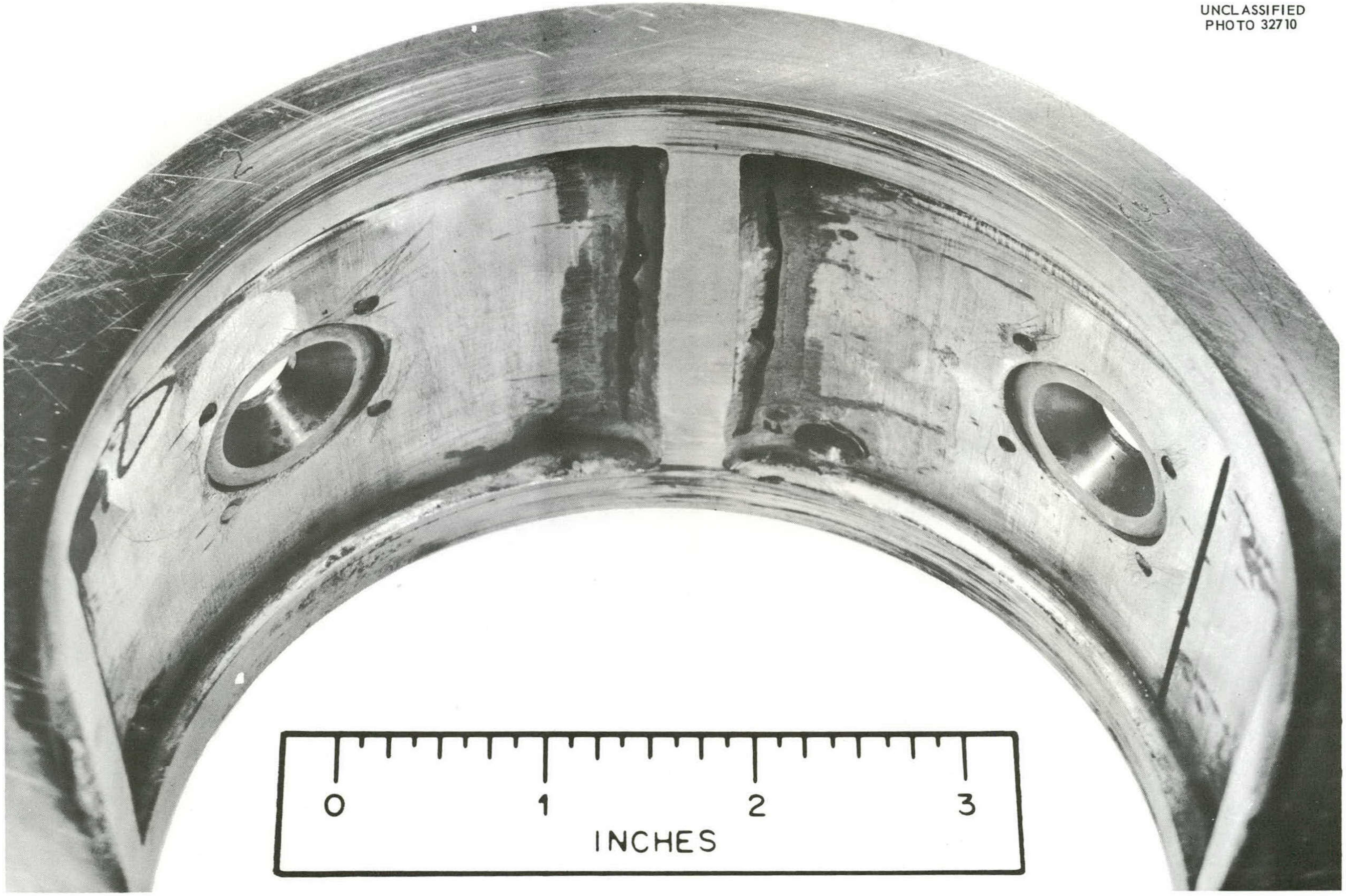

Fig. 10

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Fig. 11

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