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By *W. D. Waldron*

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DESIGN OF LIQUID METAL LUBRICATED BEARINGS
VOLUME 2: AN EXPERIMENTAL INVESTIGATION OF
NaK LUBRICATED TILTING PAD BEAR-
INGS

by

W. D. Waldron
E. B. Arwas

February 28, 1968

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NYO 3363-7
VOLUME 2

MECHANICAL TECHNOLOGY INCORPORATED

Final Report
on

DESIGN OF LIQUID METAL LUBRICATED BEARINGS
VOLUME 2: AN EXPERIMENTAL INVESTIGATION OF
NaK LUBRICATED TILTING PAD BEARINGS

W. D. Waldron
E. B. Arwas

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February, 1968

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investigation, was conducted in a test facility which is described in Section II of this volume. Very briefly, however, the principal characteristics of the test rig are:

Rotor weight - 57 lbs. symmetrically supported on two test bearings

Test bearing type - four-pad tilting pad bearings, with 0.55 pivot position

Bearing size - 2" diameter x 2" long

Machined pad-to-shaft clearance ratio: 0.004 $\frac{1}{16}$ "

Assembled bearing-to-shaft clearance ratios: 0.002 $\frac{1}{16}$ " and 0.004 $\frac{1}{16}$ "

Speed range - 3000 to 28,000 rpm, infinitely variable (design speed = 24,000 rpm)

Lubricant - NaK 78 at 110-120°F

Range of Reynolds Numbers - 1250 to 11,700 (10,000 at the 24,000 rpm design speed)

Steady state load application capability - 0 to 325 lbs./bearing (i.e. 0 to 81 psi on projected area).

Unbalance imposed on rotor - 0 and 0.412 oz.in.

The test rig, which is shown in Fig. 1, (and, in schematic form, in Fig. 38) has the rotor supported near its ends by the two NaK lubricated test bearings. A particular feature of the design was that the steady state load was applied through an externally pressurized, nitrogen-fed loader bearing at the center of the rotor. This load-transmitting bearing was connected to a pneumatic loading cylinder through a bank of 10, soft (40 lb/in), parallel tension springs which, in effect, dynamically decoupled it from the load cylinder and support structure. Thus, when a dynamic, once-per-shaft-revolution force was applied to the rotor by means of mechanical unbalance, the loader bearing vibrated essentially in phase and with the same amplitude as the rotor, so that the static-load it imposed on the rotor remained time-independent.

Dynamic (unbalance) loads were applied to the rotor by means of set screws introduced into two planes in the rotor located near the ends of the loader bearing. The test rig was installed in an existing, AEC owned dry box in which a nitrogen gas environment was maintained at about 5 psi above atmospheric. The rotor was driven from outside the dry box by means of an existing variable speed

(36,000 rpm max.) electric motor through a quill shaft with crowned spline connections. The drive coupling penetration of the dry box was sealed by a buffered gas labyrinth seal during operation, and by a positive contact seal. Axial positioning of the rotor was achieved by means of a double-acting, externally-pressurized, nitrogen-fed thrust bearing.

The program called for determination of the steady-state load capacity of the NaK lubricated bearings, as well as of their dynamic characteristics, i.e., the stiffness and damping properties of their liquid-metal lubricant films. Since these were to be compared with theoretical data, a high degree of accuracy and reliability in the measurement of film thickness and of dynamic shaft orbits within the bearing clearance was clearly necessary. Accordingly, considerable attention was paid to the proximity instrumentation used. Capacitance type probes using the Wayne Kerr amplification system were selected, the probes being located in the cover gas ambient outboard of the seals at each end of the two bearings. This proximity instrumentation system is very accurate and reliable, and it has the further important advantages of being highly linear thus greatly facilitating data reduction. It functioned very satisfactorily throughout the test program. In the later parts of the test program, when it was established that inductance type probes, capable of operation directly in the liquid metal environment, could be used, they were added to the bearing instrumentation to study the motions of the individual pads.

The sodium-potassium eutectic (NaK 78) was used as the test fluid because it permitted economical attainment of the task objective, which was to determine the accuracy with which the theoretically derived test data predicted the fluid-dynamic performance of liquid metal lubricated bearings. The program was limited to this and its scope did not include the equally important question of suitable materials for long term operation of liquid metal lubricated bearings in a high temperature environment.

Since NaK 78 is liquid down to very low temperatures (about 12°F), the tests were conducted at between 110 and 120°F. This had the dual advantage of minimizing test facility (and unit test) costs, as well as greatly simplifying the instrumentation and measurement problems inherent in liquid metal tests. Since NaK is comparable in viscosity and density to its constituent elements, the test could be

conducted in the range of Reynolds Numbers of interest in compact turbomachines. The non-wetting and low gas solubility of NaK are also characteristic of some other alkali metals. Thus, if these lubricant characteristics (which are neglected in lubrication analysis, save from the standpoint of the pressure at which the film will rupture) proved to significantly influence fluid-dynamic bearing performance, such effects would show up in the tests. No such effects were observed in the test program.

The test rig configuration used was representative of some compact, high rotational speed machines that were of interest when the program was initiated.

The material presented in this Volume of the report is essentially in two parts. The first part, consisting of Sections II through V, describes the selection and design of the test rig and NaK loop, as well as the check-out tests. These are all reviewed in detail and the problems encountered, as well as the corrective action taken, are described. In all cases, the operational problems became evident during the check-out tests and were corrected by re-design and modification of some rig and loop components. No limiting problems that would have jeopardized the test program were encountered.

The second part of this Volume of the report, comprising Sections V through VII, describes the actual test program, the data generated and its comparison with theory. In general, good agreement with the calculated design data was found. This was true, in particular, when the tests were conducted with the bearing pads assembled so as to produce a preload coefficient of 0.5. Here, the assembled bearing clearance ratio was 0.002 $^{\circ}$ / $^{\circ}$, and the preloading insured high test bearing stiffness under all conditions — even when the loader bearing was used to apply a load exactly equal and opposite to the rotor and loading mechanism assembly weights, so that the test bearings were effectively unloaded, simulating vertical rotor operation. Preloading of tilting pad bearings, when these are lubricated with low viscosity fluids, and particularly when they may operate with little or no external load, is strongly recommended.

When the tests were run with the bearings assembled without any preload, a marginal (but controlled and non-destructive) instability was observed and recorded, when the externally applied load on the rotor was at or near zero.

In these tests, the assembled clearance ratio was large (0.004 "/") and, in the absence of an applied bearing load, the fluid film stiffness was very low so that secondary effects dominated and the test data showed some scatter and apparent deviations from theoretical prediction. Even in this zero preload condition, however, the scatter was eliminated and the test data returned to agreement with theory when sufficient load was applied to the test bearings to induce an eccentricity ratio of 0.2 or greater. We concluded, based on the satisfactory correlation between the theoretical and test results, that the design data tables given in Volume 1 can be used for design of liquid metal lubricated bearings, within the limitations on clearance ratio, Reynolds Number and absence of pockets or sharp discontinuities in the film that were identified in Volume 1. These limitations were obtained in the study of the super-laminar flow regions of bearing operation, where fluid inertia effects become significant.

One final experimental task was conducted and is reported here in Section VIII. This task was the calibration of a high temperature inductance type probe, operating directly in liquid sodium at 500, 700 and 1000°F. This task was prompted by the well established need to develop proximity instrumentation capable of accurate and reliable operation in liquid sodium at elevated temperatures. Such instrumentation is needed in the development and prove-out testing of the process-fluid lubricated bearings needed in components of the sodium system of the Liquid Metal Fast Breeder Reactor. The task was added to the basic program after a high temperature version of the inductive probes, that had been used successfully in the rotor-bearings test rig, became available. The calibration tests confirmed the feasibility of inductance type proximity measurements in high temperature sodium, but they also showed that additional development work is needed to improve linearity, range, reliability and ruggedness of the system, including its leadwires.

II. TEST RIG DESIGN

1. Test Rig Design Criteria

The principal characteristics of the test rig were cited in the Introduction. As noted there, the test rig was designed to meet the objective of the test program, which called for accurate determination of the fluid-dynamic performance of liquid metal (NaK) lubricated bearings, for comparison with their theoretically predicted values. The use of NaK as the test fluid was also discussed in the Introduction, as was the selection of the test rig configuration.

At the time the program was initiated, there were questions regarding the performance of liquid metal lubricated bearings in compact, high rotational speed turbomachines. The test rig was made representative of such machines, partly in order that useful rotor-bearings dynamics design information could be gained, in addition to the fluid-dynamic bearing performance. Furthermore, with high rotational speeds, large Reynolds Numbers could be achieved with relatively small equipment and loop sizes and, hence, costs. Existing AEC owned equipment, remaining from previous liquid metal bearing investigations conducted by another contractor, could then also be used and did not have to be duplicated. This equipment included the dry box and vacuum system, as well as some instrumentation and controls.

The ranges of the operating parameters over which the tests were to be conducted were also established at the start of the program. These are listed below, together with the values actually achieved in the tests.

	<u>Initially Established</u>	<u>Achieved in the Test Program</u>
1. Clearance ratio	1×10^{-3} to 4×10^{-3} in/in	2×10^{-3} and 4×10^{-3} in/in
2. Reynolds Number	1000 to 12,000	1250 to 11,700
3. Surface Speed	10 to 200 ft/second	10 to 200 ft/second
4. Steady state, unit loading at de- sign speed	0 to 75 psi	0 to 81 psi

Two complete test sequences had to be conducted over the above ranges of parameters. The first sequence was to be with the rotor fully balanced and the second with a

large, known unbalance imposed on the rotor. An unbalance of 0.412 oz.in. was actually used in this second test sequence.

As noted above, a compact high rotational speed test rig was envisaged. The design speed was selected to be about 24,000 rpm. A design specification, however, was set that the test rig should be capable of operation at overspeed, up to 30,000 rpm, in order to provide a substantial margin of safety for the full design range. The actual test program was run at speeds up to about 28,000 rpm.

Since the dynamic characteristics of the fluid film (i.e. its stiffness and damping) were to be established, in addition to its steady state load capacity, a critical requirement was that the load mechanism used would not impose any extraneous motions or restraints on the rotor. Furthermore, the load applying mechanism was to be dynamically decoupled from the support so that the static load would be time-independent during the dynamic load tests.

Finally, a requirement was imposed on the rotor design, requiring the rotor to be essentially rigid over its design range, to minimize bending corrections that would have to be made on the test data, since the proximity measurement planes would be located outboard of the bearings, rather than in the mid-planes. (This was necessary to keep the capacitance probes in the cover gas atmosphere, rather than in a NaK ambient). In the final design, some bending of the test rotor occurred at the higher end of the speed range, above about 21,000 rpm, but the shaft curvature due to bending was small, so that the errors were less than 20 microinches, i.e., of the order of 1% of the smallest radial clearance (0.002") used in the tests.

2. Selection of Configuration

Three test rig concepts that were consistent with the above criteria were evaluated. They are summarized pictorally on Fig. 2. All the concepts shown incorporated the feature of applying load through a spring which is very much softer than the test bearing film so as to eliminate any coupling effects to the test bearing during dynamic load testing.

The concept shown in Fig. 2 (a) utilizes two rolling element bearings to support the shaft and fix the shaft axis in space. Steady state load is applied directly to the test bearing whereas dynamic load is applied by changing the radial position (runout) of the test journal relative to the shaft axis which is fixed by the rolling contact bearings. This concept, although theoretically sound, was discarded primarily because of the rolling contact bearings. To eliminate extraneous shaft motions during dynamic load application, the stiffness of the support bearings and support structure would have to be significantly larger than the test bearing. To accomplish this, very highly preloaded bearings would be required and the adverse combination of preload, applied load, and very high speed is well beyond rolling element bearing state of the art.

The second concept, shown in Fig. 2(b), relies on the fact that the spacing between the support bearing and the test bearing is very large, in comparison with the spacing between the test bearing and the loader bearing. Thus, the test bearing and the support bearing are essentially decoupled. With the steady state load applied through a loader bearing near the test bearing, and dynamic load applied by means of unbalance weights in approximately the same position, the test bearing receives, for all practical purposes, the entire applied load. This scheme, in effect, eliminates any extraneous effects of the support bearing upon the test bearing. Although this concept had been used with much success in earlier turbulent bearing test work at MTI, its application for the subject program was precluded for two main reasons:

- (a) because of the high test speed, up to 24,000 RPM or higher, the support and loader bearings, which would have been of the rolling element type would have required some type of jet lubrication to achieve the life requirements needed to conduct the tests with a high degree of reliability. Oil jet lubrication would pose sealing and dry box contamination problems.

- (b) the existing, AEC owned dry box was not large enough to achieve the necessary span between the support and test bearings. The added expense of constructing a new dry box would have been necessary and it was not felt to be justifiable.

The third concept shown in Fig. 2 (c), consisting of two rigidly mounted test bearings and a central, hydrostatic, nitrogen fed loader bearing, was selected for the test program. This concept not only completely eliminated the dry box contamination problem imposed by using an oil lubricated loader bearing, but also virtually eliminated any extraneous forces on the test bearings that would tend to compromise the data. This feature resulted from applying the load on the rotor through a gas film and a spring which was very much softer than either the nitrogen or liquid metal bearing films. A secondary advantage of this concept is that it has two, identical test bearings. Thus, two sets of data are obtained in each test for cross-check purposes. Also, in event of an instrumentation failure in one test bearing, a complete set of data could still be obtained from the second test bearing.

The final test rig configuration is shown on Fig. 1 which is a reproduction of a portion of the layout-assembly drawing. The test rig consists of two tilting pad test bearings located outboard of a central hydrostatic loader bearing. Steady state load is applied to the loader bearings by means of a nitrogen pneumatic cylinder through ten parallel tension springs. Dynamic load is applied by changing the rotating unbalance by means of set screws in the shaft face near the loader bearing. The test rig, located inside a nitrogen filled dry box, is driven from outside of the dry box by means of a high speed electric motor (rating: 30 hp at 36,000 rpm) through a quill shaft and labyrinth seal arrangement. There are virtually no thrust loads and axial positioning of the rotor is affected by means hydrostatic, nitrogen-fed thrust bearings located on the shaft faces at the loader bearing.

The mechanical design of these test rig components are described in detail in the next section.

3. Test Bearing Size

The selection of the test bearing size was based upon the following considerations:

1. Lubricant film Reynolds Numbers to at least 10,000 were desired with normal-practice clearance ratios.
2. Anticipated, compact turbomachinery speeds were to be simulated. A typical alternator speed of about 24,000 rpm was used as this design objective.
3. The bearing had to be large enough to permit accurate measurements of film thickness to be made.
4. The loads applied to the test bearing had to be consistent with the capabilities of a gas loader bearing.

Bearings ranging from 1 to 3 inches in diameter were evaluated in light of the above considerations. The length to diameter ratio was limited to unity because, at the time when the test rig was designed, extensive data was available only for this particular geometry*. The results of this evaluation indicated that a 2 inch diameter test bearing represented a satisfactory compromise. The computed performance of the 2 inch diameter bearing is summarized on Figs. 3 through 7 which show Reynolds Number, load capacity, power consumption stiffness and damping as functions of speed and/or eccentricity ratio. These curves are based upon the design data given in Volume 1 of this report. The data shown on Figs. 3 through 7 form the basis of the test rig design requirements.

4. Test Rotor

The significant requirements imposed upon the design of the rotor are summarized below:

1. The test bearing size was two inches x two inches as defined in the preceeding section.
2. The load was to be applied midway between the two test bearings resulting in equal bearing loading. This would permit the best cross check of the resulting test data.

*Subsequently, the design data was extended to L/D ratios of 0.25 and 0.5, as given in Volume 1 of this report.

3. The rotor was to be as rigid as practical to both minimize bending under applied load and to avoid crossing critical speeds in the region where the shaft no longer behaves as a rigid body.

The final rotor configuration shown in Fig. 1 consists of 2" diameter end sections with a built-up section in the center. The built-up section forms the hydrostatic loader bearing journal with the faces forming the thrust bearing surfaces. The test bearings are placed as close together as practical to give the shaft maximum rigidity under radial loading at the midplane.

The built-up center section starts immediately next to the capacitance type proximity transducers, providing additional shaft stiffness. Several undamped critical speed calculations were made to determine the optimum size of the built-up section for maximum shaft rigidity. The final configuration shown in Fig. 1 is the optimum considering all the design requirements. The following summarizes some of the trade-offs in changing the center section dimensions from those finally established.

1. Increasing the center section diameter, although increasing loader bearing load capacity, reduces the first flexural critical speed because of the increased mass.
2. Reducing the center section diameter has little effect on the first flexural critical speed but does reduce the loader bearing capacity.
3. Increasing the length of the built-up sections immediately inboard of the first bearings slightly increases the first critical speed but makes the test bearing spacing prohibitively long for installation into the dry box.
4. Reducing the mass of the center section by making it partially hollow increases the critical speed but results in large centrifugal growth of the loader bearing journal diameter.

It should be noted, that in making these critical speed calculations, the mass of the loader bearing must be added to that of the shaft since it is coupled to the shaft through a stiff spring and essentially decoupled from ground by a soft spring system.

The undamped critical speed curves for the final rotor configuration are shown in Fig. 8. Since it was anticipated that the test rig would be operated without the loader bearing in the initial check-out tests, the critical speed calculations without the loader bearing were also made and are shown in Fig. 9. Examination of these curves indicate that some shaft bending occurs at the high end of the speed range. This is evidenced by the fact that the bearing stiffness vs speed curves intersect the undamped critical speed curves in the region where the slope of the critical speed curves, plotted on logarithmic paper, are no longer linear*.

Since the operating range of the rotor-bearings system spans the system critical speeds, the damping introduced by the fluid films of the NaK lubricated bearings is particularly important as it is this damping that limits the vibration amplitudes when the rotor traverses the critical speeds. Accordingly, rotor-bearings system dynamic response calculations, rigorously including the calculated values of bearing damping, were made to determine the amplitudes of vibration during passage through the system critical speeds.

The calculations of rotor response to unbalanced load were made for the largest bearing clearance ratio (0.004in/in), since this provides the smallest value of bearing damping. The calculated amplitudes of vibration at the bearings and measurement planes as a function of speed are shown in Figs. 10 and 11. Unbalance was assumed to be located at the balancing planes which are located just outboard of the thrust bearings.

These plots showed that the rotor vibration amplitudes were acceptably low throughout the speed range up to 30,000 rpm so that the rotor configuration shown in Fig. 1 was entirely satisfactory. Figure 12 is a photograph of the finished shaft.

* Within the speed range in which the rotor behaves as a rigid body, the rotor-bearings system critical speeds are proportional to $\sqrt{K}$, where K is the bearing stiffness. Thus, within this range, the plots of N_{crit} vs K are linear with a slope of 1/2, when plotted on log-log paper. However, at higher speeds, when rotor flexibility effects become important, the system critical speeds become functions of both the bearing stiffness K and the rotor flexibility. The plots of N_{crit} vs K are then no longer linear when plotted on log-log paper.

5. Loader Bearing

The basic size of the hydrostatic, nitrogen fed, 4.5" dia. x 6" long journal bearing that served as the loader bearing, was established by the rotor design and other requirements discussed above. Within this envelope, the bearing was designed for maximum stiffness so as to obtain the greatest possible load capacity and to insure that the bearing would follow the shaft motions. It was also necessary to design the bearing for the minimum practical gas flow.

To minimize the gas flow, the design clearance ratio at 24,000 rpm was set at the relatively small value of 0.5×10^3 in/in. With this clearance ratio, 24 feeding orifices, each 0.030 inches diameter were used to obtain the maximum bearing stiffness. These holes are arranged in two planes halfway between the bearing center and ends (double plane admission). This arrangement is the best compromise between obtainable load capacity and minimum gas flow. Inherent compensation for this design was favored over orifice compensation because of the former's superior resistance to pneumatic hammer.

In calculating the design performance of the hydrostatic loader bearing it was necessary to consider the variations in bearing clearance due to manufacturing tolerances and journal centrifugal growth. Setting the total manufacturing tolerance at ± 0.0002 inches and adding the centrifugal growth from zero to 30,000 rpm, the anticipated range of clearance ratios to be considered was 0.348×10^{-3} to 0.701×10^{-3} in/in. The following table summarizes the bearing performance over this clearance range assuming N_2 at 120°F . The first tabulated number corresponds to the value at the smaller clearance.

In this table, which is given on the next page, a limiting value of 0.6 for the eccentricity ratio in the N_2 loader bearing was set. Actually somewhat larger values of eccentricity ratio up to about 0.8, are permissible with externally pressurized gas bearings, provided that a large number of feed holes are used to prevent lockup. In this case, however, since the loader bearing was merely the device for imposing static load on the NaK lubricated bearings, rather than being a central object of the test program, it was clearly desirable to operate in a conservative range, well short of its ultimate capacity.

TABLE 1

CALCULATED LOADER BEARING CHARACTERISTICS, WITH MAXIMUM AND MINIMUM VALUES
OF THE CLEARANCE RATIO

Supply Pressure (PSIG)	Flow (SCFM)	Load Capacity at $\epsilon = 0.6$ (LBS)	Radial Stiffness (10^6 lbs/in)	Angular Stiffness (10^6 in lbs/rad)	Damping (lbsec/in)
15	0.41 to 1.18	7 to 55	0.02 to 0.06	0.19 to 0.18	1380 to 284
45	0.57 to 2.94	73 to 236	0.16 to 0.25	0.63 to 0.74	1610 to 212
75	1.37 to 5.26	203 to 396	0.43 to 0.42	1.08 to 1.60	1650 to 160
105	2.22 to 7.63	374 to 534	0.80 to 0.56	1.38 to 2.42	1570 to 127
135	3.13 to 9.20	549 to 656	1.27 to 0.69	1.45 to 3.42	1380 to 104
165	4.12 to 10.90	804 to 731	1.71 to 0.77	1.56 to 4.75	1340 to 84
210	5.57 to 14.70	1010 to 815	2.36 to 0.86	1.81 to 6.41	1110 to 79

In order to reduce the supply pressure required in the loader bearing for any given load and, hence to minimize the flow of pressurized nitrogen, an additional design feature was used. This consisted of using two separate manifolds, the first supplying the feed holes in the lower half of the bearing and the second supplying feed holes in the upper half of the bearing. This permitted the two halves of the bearing to be supplied with different pressures such that the bearing could be operated as two separate 180 degree arcs. By operating the unloaded half at a lower supply pressure than the loaded half, the eccentricity at any given load, as well as the two supply pressures would then be less than for a single manifold bearing with equal supply pressures, on both halves. In practice, the supply pressure for the unloaded half need only be large enough to give the bearing the required stability and rotating load capacity for the particular operating condition.

Another benefit derived from splitting the supply manifold was the ability to control the onset of half frequency whirl if necessary. Since the bearing could develop significant hydrodynamic pressures at high speeds, the possibility of half frequency whirl at low loads existed. Should the phenomenon be experienced in operation, it could be suppressed by unbalancing the supply pressures to the bearing halves thus imposing an internal stabilizing force.

To aid in the control of the hydrostatic bearing supply pressures, proximity probes were placed in the bearing looking at the shaft such that, during actual operation, the relative position of the shaft and bearing as well as the vibration amplitudes could be monitored. With this information, the operator could easily determine the best magnitude and combination of bearing supply pressures for any operating condition. (The proximity instrumentation is discussed in more detail in a later section).

The loader bearing is attached to the loading mechanism through a ball end joint to eliminate moments applied to the bearing by the spring loader. The hydrostatic feed manifolds have been incorporated into the load adapter which transmits the load from the loader mechanism to the bearing. This arrangement permits introduction of the supply lines at the bearing centerplane thereby minimizing extraneous piping moments.

It was necessary to incorporate a heat exchanger into the loader bearing to remove the heat generated by the hydrostatic gas film which was calculated to be a maximum of about 16,000 BTU/hr at 30,000 rpm. In order to maintain reasonable control of the bearing clearance, it was determined that the maximum allowable gas film temperature was in the order of 250°F. At this temperature, less than 1% of the loader bearing heat can be expected to be taken out through the shaft and about 4% by the hydrostatic gas mass flow. The need for a significant amount of auxiliary cooling is therefore obvious.

NaK was selected as the heat transfer fluid because of its excellent thermal properties and its availability in the dry box. By using the discharge of the two test-bearing heat exchangers (17.6 gpm at 110°F), the entire loader bearing heat of 16,000 BTU/hr can be removed with only about a 10°F rise in the NaK.

The loader bearing is fabricated from 6061T6 aluminum because its low density, high thermal conductivity and high coefficient of thermal expansion. (The aluminum was electroless nickel plated to prevent caustic attack during clean up). The low density was required to reduce the bearing mass and mass moment of inertia so as to maintain a high resonant frequency of the bearing mass-hydrostatic film system. The high thermal conductivity was required to minimize the temperature gradient between the gas film and coolant fluid while a large expansion coefficient

was desired to minimize the clearance change resulting from the aforementioned gradient.

A thin (.125" thick) "Electrofilmed" bronze sleeve was originally fitted on the inside diameter of the aluminum housing to provide a better bearing-journal material combination and final machining surface. (This was subsequently changed to 0.375" thick aluminum sleeve because leakage of the high pressure gas on the back face of the thin bronze sleeve caused it to distort, as discussed later in the description of the check-out tests).

A thermal analysis of the loader bearing was completed assuming various values of contact resistance between the sleeve and housing such that an expected range in clearance variations could be determined. The results are tabulated below:

h_1	T_F	ΔC_r
250	283	-0.96×10^{-3}
500	235	-0.35×10^{-3}
1000	211	-0.05×10^{-3}
∞	187	$+0.25 \times 10^{-3}$

where: h_1 = interface contact conductance (BTU/ft²-hr-°F)

T_F = calculated film temperature (°F)

ΔC_r = change in radial clearance from assembly (inches)
(- sign indicates loss of clearance, relative to the value at assembly)

Average coolant temperature = 124°F NaK

Housing material = 6061T6 aluminum ($k=99$ BTU/ft-hr-°F; $\alpha = 13.1 \times 10^{-6}$ in/in°F)

Shaft material = 410 SS ($k=14$ BTU/ft-hr-°F; $\alpha = 5.6 \times 10^{-6}$ in/in°F)

Sleeve material = 544 Bronze ($k=50$ BTU/ft-hr-°F, $\alpha = 9.6 \times 10^{-6}$ in/in°F)

The above tabulation clearly indicates the influence of the aluminum to bronze contact resistance on the bearing clearance and the importance of maintaining a high contact conductance. A reasonable value of h_1 for an interference pressure of 500 psi; (resulting from about .0015 inches of radial interference) is about 500 BTU/hr.ft²°F. Using this value, the .00035 inch change in clearance out of

.00125 assembled clearance is tolerable although is considered a maximum desirable reduction. To insure that the clearance never is reduced below this value, it was desired to increase the contact conductance to the highest possible value. To do this, the aluminum and bronze mating surfaces were "tinned" with aluminum solder prior to assembly. The assembly of the two pieces was done at a temperature somewhat above the melting point of the solder (450°F) so that the solder would flow into all the surface irregularities thus materially decreasing the contact resistance. It was anticipated that this technique would increase the value of h_1 to well in excess of 1000 and so tend to make the bearing clearance grow slightly larger.

Load is applied to the hydrostatic loader bearing by a high pressure pneumatic cylinder through a nest of ten springs arranged in parallel. The piston of the cylinder is structurally attached to the main housing which supports the test bearings. The stiffness of the load spring assembly is 400 lb/in which is more than three orders of magnitude less than the hydrostatic bearing gas film stiffness. With this large difference in stiffness, the loading spring system will have no detectable influence on the loader bearing dynamics.

Figures 13 and 14 are photographs of the completed loader bearing. Figure 13 shows the assembled bearings with the manifold end pieces (Item 6 in the drawing of the bearing in Fig. 1), in place. The NaK coolant flow tubes can be seen exiting from these manifolds. The external load is applied through a spherical joint (not shown in the photograph), bolted to the top of the hexagonal shaped member at the center of the bearing. In Fig. 14, one of the coolant manifolds was removed to show the axial cooling passages. The radial holes on either side of the central hexagonal member are for the capacitance probes that were used to measure the bearing film thickness.

6. Test Bearing Housings

The bearing housings are bolted and keyed to the lower half of the cylindrical rig support structure. They contain the pivoted pads, adjustable pivots, a clearance type shaft seal, seal slingers and a multitube NaK-NaK heat exchanger. These are shown schematically on Fig. 15.

The use of a liquid metal as the test lubricant presented two problems which are not normally serious when using conventional lubricants. These were the shaft seal and the proximity instrumentation. Because of the high electrical conductivity of the NaK, no fully developed and reliable proximity instrumentation was available that could be placed directly in the NaK environment. For this reason, the probes had to be placed outside the housing, in the dry box N_2 environment. This put an additional burden on the seals in that even a light NaK spray from the seals could short out the probes. Secondly, it was necessary to gravity drain the seal drain cavity since no scavenge pump for liquid metal service was readily available. It was further desired not to incorporate a disk type scavenge pump within the seal drain cavity because of the additional mechanical complication, power consumption, and its wide performance variation with speed.

To make the very simple sealing arrangement shown on Fig. 15 practical, it was necessary to limit the bearing cavity pressure to a low enough value that the resulting seal leakage could be readily handled by gravity flow through the seal drain lines. On the other hand, the bearing cavity had to be maintained at a pressure high enough to insure that the cavity remained full (even at zero speed) and under no conditions could gas be pulled from the seal drain cavity to the bearing cavity.

To accomplish this, a restricted siphon arrangement was incorporated where a relatively small quantity of flow is introduced to the bottom of the bearing cavity and flows out the top where it is pumped through a restrictor to a drain sump within the dry box. The fixed restrictor negates the siphon effect caused by the height of the cavity above the sump and, for all practical purposes, sets the pressure level within the cavity for any given inlet flow. Since the leakage flow out of each seal is only 10% of the flow through the restrictor, large variations in the clearance seal pressure drop characteristics will have little influence on the total through flow.

The leakage value of 0.1 gpm shown on Fig. 15 was based upon an inlet flow of 1.2 gpm and a 0.26" dia. fixed orifice restrictor assuming a diametral seal clearance of .020 inches (fully eccentric) and zero speed. The drain cavity and lines were sized for 0.5 gpm gravity flow only. As the speed is increased

(or the clearance is reduced), the leakage flow will decrease to the point where it should become negligible at the test speed. As a result (assuming the supply value is unchanged) the cavity pressure will increase slightly (thus further insuring against gas suction) and the through flow will decrease by approximately 10%. The slingers shown on Fig. 15 are not intended to provide any pumping but were included as a block to line-of-sight flow to the outside through the outboard clearance.

Final adjustment of the flow balance, where necessary, was made prior to bearing testing by adjustments of the orifice size after flow checks using water as the calibration fluid.

The indicated lube flow of 1.2 gpm was established as a reasonable value for gravity draining both the bearing cavity and the dry box sump. It is, however, inadequate to remove the heat generated by the test bearing and seal. If sufficient flows to remove generated heat were passed through the bearing and drain sump, the line and valve sizes to permit gravity draining would have become prohibitively large.

Instead, a heat exchanger within the bearing cavity was incorporated to remove the 17,650 BTU/hr of generated heat (4.75 HP max bearing and 2.5 HP max seal losses). The heat exchanger is made of six parallel copper tubes wrapped around the inside of the bearing housing. The total NaK flow through the tubes is approximately 8.8 gpm. Phosphorous deoxidized copper, which is compatible with NaK up to about 800°F, was selected as the heat exchanger material because of its excellent thermal conductivity and its good forming and welding properties.

The aforementioned heat generation and the fluid temperatures shown on Fig. 15 are the maximum values calculated at 30,000 rpm. The average temperature within the bearing housing of 123°F should be uniform within a couple of degrees as a result of the relatively small through flow, the high thermal conductivity of NaK, as well as the fluid mixing afforded by the rotating shaft and by the end-leakage from the bearings.

The test bearing housings were fabricated from AISI stainless steel and were electroless nickel plated to inhibit corrosion (primarily from water which is

used to clean up the parts after test). AISI 410 was selected to match the thermal expansion characteristics of the shaft pads and pivots so as to minimize bearing clearance variations during the test. The seal plates were fabricated from 300 series stainless steel, again because of corrosion resistance. In this case the control of clearances was not as important as in the bearings so the use of electroless nickel could be avoided by selecting the higher expansion alloy.

A photograph of the bearing housing assembly is shown on Fig. 16(a). This picture, with the seal plates removed, shows the heat exchanger, the tilting pads, and the pivots. Each of the 80° arc length pads are located on the pivots by means of a collar around the pivot which is secured to the pad. The pivot position is 0.55 (i.e. the arc length from the leading edge of the pad to the pivot "point" is 55% of the total arc length of the pad). The design data given in Volume 1 was obtained for this value of pivot position.

7. Test Bearing Pivots

The successful and reliable operation of pad-type bearings, which employ a pivot configuration as a method of support, is, in part, dependent on the pivot characteristics. The pivot characteristics of primary importance to the NaK bearing test rig are as follows:

1. The pivot must impose a minimum of restraint on the bearing pad when operating conditions demand changes in pad attitude angle or bearing alignment.
2. The pivot must withstand extended periods of operation without significant wear or increase in pad restraint.
3. The pivot must be capable of operation without relying upon boundary lubrication other than metallic surface treatments.
4. The design configuration of the pivots must be integrated with the various mechanical requirements. These involve:
 - a. facilities for bearing clearance adjustment
 - b. keeping the mass of the pad to a minimum (to maintain a high pad resonant frequency)
 - c. Stiffness of the pivot must be high, compared with that of the fluid film, in order not to adversely affect the overall dynamic characteristics of the bearing.

Where the pivot does not satisfy the above criteria, the high damping, stability and alignment capabilities of tilting pad bearings are impaired. It may then become subject to the difficulties inherent in rigid bearings, such as fractional frequency whirl and inability to accommodate misalignment.

Relative motions between the mating elements of the pivot are characteristically small and under dynamic load are oscillatory with amplitudes in the range of 10^{-4} to 10^{-3} inches. Under dynamic loads on the shaft, as due to rotating mass unbalance, the pads will oscillate in the pitch direction in response to the translatory component of the shaft motion, and they will oscillate in the roll and yaw directions, in response to the conical component of the shaft motion. The pitch, roll and yaw axes are as defined in the sketch below.

In addition, there will be oscillatory radial deflection of the pivot due to the rotating load vector. The load on the pivot may, therefore, be considered as having a static component, which is the sum of bearing preload and the applied, steady-state load on the bearing, and a dynamic component which is that due to the rotating unbalance load. The angular, relative motions between the pivot elements are accommodated by the following types of motions:

1. rolling
2. sliding
3. microslip
4. elastic deformations

Ideally, for long term operation without damage, the pivot should be so designed that the relative motions are taken entirely in rolling as these have minimum friction. Gross sliding should be prevented, as it has high friction and can produce severe fretting type damage to the pivots. In practice, sliding cannot be altogether eliminated - e.g. yaw motions must to some extent be accommodated by relative sliding, but these have generally very low amplitudes, of the order of 10^{-4} radians, and can be tolerated without significant damage if hard compatible materials are used. Microslip is slip due to elastic deformation in the contact zone and can cause fretting around the fringes of the contact zone. Here again this can be minimized by materials choice and by designing the pivot to have low contact stress. Cyclic elastic deformations and, hence, stress variations are induced by the rotating load and these can cause fatigue problems if the contact stress is high.

To date, good pivot design has been largely empirical, though a systematic study of this aspect of bearing design currently in progress, for high temperature gas bearings, under NASA sponsorship (Ref. 1). For the NaK bearings in the present program, a conservative pivot design was used, which past experience in gas lubricated turbomachinery produced at MTI had proved to be entirely satisfactory for long term operation of the order of 15,000 hours or longer*. This design consisted of:

* Experience gained with pivot design for high temperature gas bearings, is in part applicable also to liquid metal bearings. In the former case the pivot is totally unlubricated. In the latter case, the poor boundary lubrication property of liquid metals offers little protection to the pivot.

Geometry: Sphere mating with a cylindrical socket, the socket radius being 2 to 5% greater than the radius of the ball to maximize rolling and minimize sliding.

Materials: M2 tool steel vs M2 Tool Steel

Hardness: RC 58 to 62

Maximum allowable contact stress: 120,000 psi

Surface finish: $< 8 \mu\text{in RMS}$

A 1" nominal radius of curvature of the pivot elements was used, to keep the maximum Hertzian contact stress below 120,000 psi as required. The calculated pivot stiffness is shown in Fig. 17. Comparison of this stiffness with that of the bearing film (Fig. 5) shows that it is 3 to 10 times higher than the film stiffness, depending on the test condition. It is important to note that the pivot stiffness should always be designed to be larger than the fluid film stiffness in order that radial deflections due to dynamic load are largely absorbed in the fluid film, which has high damping, rather than in the pivot, which has virtually no damping. Furthermore, in a test program such as the one reported here, it is important that pivot deflections be kept small, relative to deflections in the film since the housing-to-shaft proximity probes measure the sum of the two deflections. In order to determine the fluid film deflection, the component due to pivot deflection has to be substrated from the total reading, based on a static calibration curve of pivot deflection vs load. Clearly, when the pivot deflections are small, this will enhance the accuracy with which the fluid film deflections can be determined.

Figure 16(b) shows the bearing disassembled. The pads, pivots, seal rings and other components, including a capacitance probe are shown in this photograph.

8. Support Structure

In addition to providing a means of holding the test bearings relative to one another and connecting the loader mechanism to the bearing housings, the support structure must be so designed that no extraneous vibrations are excited within the structure that would influence the bearing operation and the resulting test data. To obtain the necessary rigidity within the limited height dictated by the existing dry box dimensions, a heavy wall, cylindrical cross section support

was selected. The bearings mount inside the lower half of the split cylinder. The upper half is secured by highly preloaded bolts which should allow bending moments to be transmitted across the split line. A partially assembled view of the test rig is shown in Fig. 18. This photograph shows the bearing housings mounted in the lower half of the split, cylindrical support structure.

The natural frequencies of the basic cylindrical structure are listed below:

Free-Free beam bending mode:	378,000 CPM
Hoop mode:	276,000 CPM
Ring bending mode:	48,000 CPM

The first two are about an order of magnitude above the maximum anticipated operating speed of 30,000 rpm and, hence, could be eliminated from concern. The calculated ring bending mode, however, was only about 1.6 times the maximum operating speed. Accordingly, provisions were made to weld on circumferential stiffeners in event that this structural resonance mode were detected in the tests. No evidence of this or other structural resonances were, however, encountered in the actual test program.

9. Drive Connection Seal

Because of both space limitations and potential oil contamination problems it was necessary to locate the drive motor outside the dry box. The motor was connected to the test shaft through a splined quill shaft to take the small misalignments between the motor and test rig, as illustrated in Fig. 1

The dry box shaft seal is a combination rotating and static seal arrangement incorporating a labyrinth for the operating condition and an "O" ring seal for the shutdown condition. The labyrinth seal can be center fed with nitrogen so that buffer gas leakage is both into the dry box and out towards the motor. Alternately, the labyrinth could be fed from the dry box ambient, which was maintained at about 5 psi above atmospheric, with leakage outward only. The labyrinth is rigidly mounted to the test bearing housing to insure good alignment with the rotating shaft. Secondary sealing between the labyrinth and dry box is accomplished by a static diaphragm assembly. To eliminate the need for buffer gas flow while the machine is not in operation, an "O" ring cylinder is provided which slides over the shaft and statically completes the seal between the shaft OD and dry box.

10. Drive-Motor Mount

Although the drive motor is located outside the dry box, it was necessary to physically attach it to the support structure so that the alignment of the motor and test shafts could be maintained. The space in the vicinity of the dry box wall penetration, which accommodates the structural coupling of the motor and rig structure, and the dry box seal, was limited. For this reason it was not possible for the structural coupling to possess the stiffness necessary for the motor support assembly to have a natural frequency above the operating speed. The coupling assembly was, therefore, designed to have a natural frequency below any anticipated operating speed so that the motor mount assembly could not respond to vibrations by the rotating shaft in the normal operating speed range. The calculated natural frequency of the motor assembly on the structural coupling was 4160 cpm which was well below the speed range of principal interest which is 6000 to 24,000 rpm.

The support structure of the test rig was vibration isolated from the dry box by spring type vibration isolators so that no dry box, dry box structure or piping resonances could be excited by the rig, and so that no extraneous excitations could be introduced into the rig. The natural frequencies of the entire test rig on its vibration isolators were:

translatory:	260 cpm
pitch	: 420 cpm
roll	: 700 cpm

These frequencies were very far below the minimum operating speed and far enough below the motor mount-coupling frequency so that the rig and motor were, essentially, decoupled.

11. Shaft, Test-Bearing and Pivot Materials

There were a number of factors that entered into the selection of the shaft and bearing materials. The more important of these factors are summarized below:

1. Since the parts are cleaned in water after exposure to NaK and all parts would be exposed to the atmosphere during periods of disassembly, materials resistant to normal water corrosion and caustic attack in addition to the obvious liquid metal corrosion were necessary.

2. The thermal expansion properties of the shaft, bearings and pivots, were required to be near identical to minimize the change in bearing clearance resulting from the difference in assembly and test temperatures.
3. The journal and bearing surface material combinations had to be compatible with NaK and exhibit anti-seize qualities such that no start-up damage would occur. The combination should also be able to tolerate an accidental high speed rub.
4. Pivot material combinations also had to exhibit good wear qualities when subjected to highly loaded relative motion in the NaK environment.

Of all the potential materials that were in the past subjected to various evaluations, only the hard refractory materials appear to possess the anti-gall qualities in liquid metal environment that are required for bearing applications. Unfortunately, these materials in the solid or sintered form are brittle, difficult to fabricate, and have much lower coefficients of thermal expansion (than the structural materials). For these reasons, it was desired to coat the bearing and journal surfaces with a refractory metal by means of flame or plasma spraying processes. The coating approach retains the wear and hardness qualities of the refractory metals and the desirable physical properties of the base structural material. (This approach is satisfactory for a test program, such as the one reported here. For long life turbomachinery applications, however, the feasibility of using ceramic coated bearings in high temperature, liquid metal environment requires much additional study).

As an aid in selecting the particular coating to be used, static test of several coatings in 200°F and 600°F were conducted to evaluate NaK corrosion resistance and resistance to caustic attack. The results of these tests, which were described in Ref. 2, indicated that the Linde LWS Tungsten Chromium Carbide coating offered the required resistance to chemical attack and was, therefore, selected for the rig test bearings.

Tungsten Carbide (run against itself) had been shown in previous tests at MTI (Ref. 3) to have excellent start-stop type wear resistance and acceptable, high-speed-rub wear resistance. (A superior material, from the standpoint of tolerance of high speed rubs - chrome oxide - did not exhibit the required corrosion resistance in the aforementioned tests, Ref. 2).

The base material for the pads which contains the cylindrical pivot member is M-50, tool steel which has been shown to have reasonable wear resistance in NaK and has been successfully used in other NaK bearing pivot designs. The pads were through-hardened to RC 54-56 as a compromise between the high hardness desired for the pivots (Rc 58-62) and the moderate hardness desired for the plasma coating process (Rc 38-42).

The AISI 418 stainless steel shaft was selected because of its good corrosion resistance and good thermal expansion match with the pad and pivot material (5.6×10^{-6} in/in^oF for the 418 shaft compared with 6.15×10^{-6} in/in^oF for the tool steel pads and pivots). The N₂ lubricated loader bearing and thrust bearing surfaces were electroplated to give a hard anti galling surface against the electroplated stationary members.

12. Instrumentation

In testing of bearing performance, the single most important piece of data is the position of the shaft relative to the bearing (including the amplitude of motion under dynamic loading). To measure this important test journal bearing parameter, capacitance type distance transducers were incorporated at four approximately equally spaced planes along the shaft. The planes (which are normal to the shaft axis) were so located that both shaft position and bending could be determined.

The measurement planes were located and fixed to each end of each test bearing housing (3.5 and 4.25 inches from the test bearing center plane) such that the transducers sense the distance from the housing to the shaft surface. Since the bearings are positively located in the housings by the pivots, the transducers also indicate the relative shaft to bearing position. Each measurement plane contains two transducers spaced at 90^o, with the output of the two connected to the horizontal and vertical amplifiers of a cathode ray oscilloscope. In this manner, the scope displays an accurate picture of shaft motion within the measurement plane. A single stationary dot on the scope indicates no relative motion of the shaft with respect to the housing, whereas a circle or other pattern indicates the amplitude and locus of dynamic motion. The frequency of motion is ascertained by switching one of the probes from the X-Y mode to a time basis display.

To monitor the loader bearing performance during test, two measurement planes of capacitance transducers were incorporated into the loader bearing such that the position of the floating bearing relative to the shaft was measured. A single transducer mounted on each bearing housing parallel to the shaft axis was incorporated to measure shaft axial position as an aid in controlling the thrust bearing.

The applied steady state load was measured by means of the nitrogen pressure imposed upon the calibrated pneumatic load cylinder which is attached through springs to the loader bearing. Rotating load was a known value based upon the unbalanced masses attached to the shaft at each of two balance planes.

Other instrumentation used to collect operational data included:

1. Speed - magnetic pickup on the drive motor coupling displayed on an electronic pulse counter.
2. Temperatures - immersion thermocouples in the bearing housing to measure bulk lubricant temperatures; surface thermocouples to measure:
 - a) lube inlet and outlet temperatures of each test bearing
 - b) coolant inlet and outlet temperatures of each test bearing
 - c) coolant inlet and outlet temperatures of the loader bearing
3. Flow - lube and coolant NaK flows to each bearing were measured by means of calibrated electromagnetic flowmeters with D.C. millivolt readout. Total N_2 flow to the loader and thrust bearings was measured on a calibrated rotometer.
4. Pressure - the supply pressures to the loader and thrust bearings were indicated on Bourdon tube pressure gages.

III. TEST LOOP DESIGN

NaK is supplied to the test rig bearings by means of the "NaK Circulation Assembly" which contains the circulating pump, NaK control valves, flowmeters, NaK cooler, filtering equipment, etc. The circulation assembly is connected to the test rig located inside the filled dry box by appropriate lines and bulkheads.

The information delineated on Fig. 19 forms the basis for the NaK circulation assembly specification. The flow rates, pressures, and temperatures indicated on Fig. 19 were derived from the test rig design analysis as discussed in the preceeding section of this report. The height limitations (which define the heads available for gravity draining) result from test cell space availability and the dimensions of the existing AEC owned dry box.

The NaK circulation assembly was designed and fabricated by MSA Research Corporation to MTI Specification. MSA Drawing AK 2418-1 (included as Fig. 20) defines all of the assembly components and shows the installation. A photograph of the assembly is shown in Fig. 21.

The schematic of the entire installation is shown on Fig. 22 which defines the NaK system, the test rig schematic, the inert gas system and all instrumentation. A photograph of the system, installed in the test cell is shown on Fig. 23. The control area and read-out instrumentation are shown on Figs. 24 and 25.

The inert N_2 which is used as the sump tank and dry box cover gas as well as the hydrostatic bearing lubricant is purified by a NaK bubbler type of purification system. The purpose of the purification system is to remove trace amounts of detrimental contaminants such as oxygen, water vapor, and oil. This is accomplished by passing the gas first through heated and then through ambient temperature NaK to react out the aforementioned contaminants prior to introducing the gas into the test system. The inert gas purifier was built by MSA Research Corporation to MTI specification. A schematic of the purifier is shown in Fig. 22.

IV. CHECK-OUT TESTS

Prior to initiations of full scale bearing testing in NaK, several component and subassembly checks and tests were planned. The purpose of these tests were to verify the basic design of critical components, establish the performance characteristics, and gain operating experience prior to committing the rig to the dry box and NaK when adjustments and modifications would be very difficult and time consuming.

These tests and their objectives are summarized below:

Loader and Thrust Bearing Static Tests

- a) verify design calculations
- b) determine pneumatic stability limits
- c) experimentally establish operating performance
- d) calibrate loader springs and pneumatic cylinder

Rig Tests using "Easy-to-Use" Low Viscosity Fluids

- a) determine effectiveness of drain system
- b) verify pressure drop estimates
- c) check-out shaft distance instrumentation
- d) shake down unanticipated problem areas

Facility Check-out Tests in NaK

- a) check-out operation of all components (e.g. NaK loop, EM pump and flowmeters, Nitrogen loop, test rig and instrumentation) on the NaK environment
- b) recheck adequacy of drains and of seals
- c) scan the full operating ranges to quickly find and correct any potential problem areas
- d) provide the operators with experience in the operation and control of the test facility.

This section describes the results of these tests and the problem areas that were solved during this phase of the program.

1. Thrust Bearing Static Tests

The thrust bearings, which are shown in Fig. 26, were calibrated over a load range from 4.44 lbs. to 40.1 lbs with supply pressures varying from 10 to 120 psig. Gas film thickness, which varied from 0 to 1200 microinches, were measured with an electronic indicator.

Thrust bearing film thickness as a function of supply pressure for several constant applied loads are shown on Fig. 27. The results are also plotted in dimensionless form on Figs. 28 and 29 as this type of plot is more convenient for comparison with theory. The theoretical curves on Figs. 28 and 29 are reproduced from Ref. 4. Since the measured film thicknesses were very small, some error in this measurement could be expected. The band of theoretical performance indicated on Fig. 29 shows the variation resulting from a $\pm 10\%$ variation in measured film thickness. Since 10% represents at most about 50 microinches, the correlation of the theoretical and experimental data is excellent.

As a result of these tests and the fact that no pneumatic hammer could be induced, it was concluded that the thrust bearing would operate in the test rig as intended.

2. Loader Bearing Tests

During the initial static calibration tests of the loader bearing it was found that the bearing bore contained several high spots in the order of up to 0.0015 inches in height. Since the bearing radial clearance was only 0.0013 inches, this condition was not acceptable. The exterior appearance of the bearing indicated that it had been mishandled with the result being distortion of the bearing bore.

To correct this situation, the bearing bore was reground and the journal surface was built up with a plasma spray coating of molybdenum which was then reground to attain the original bearing clearance.

Subsequent attempts to calibrate the bearing resulted in the reestablishment of the aforementioned high spots in the bore. Further examination revealed that the solder bond between the aluminum body and the 1/8" thick bronze bore liner was discontinuous. This situation permitted the high pressure nitrogen to enter the resulting voids and deform the bearing bore.

This loader bearing distortion problem was corrected by removing the bronze bore liner and machining 1/4" from the inside diameter of the aluminum bearing body. The thin bronze sleeve was replaced with an aluminum sleeve 3/8" thick. The increased thickness considerably stiffened the bore liner (by a factor of 27, since the stiffness is proportional to the cube of the thickness) thus minimizing the possibility of local distortions. Aluminum was used in the rework because it was compatible with the molybdenum coated journal surface, as well as being immediately available.

The results of the static tests of the reworked loader bearing are shown in Figs. 30 through 32. Figure 30 shows a typical dimensionless load vs eccentricity plot. It will be noted that the measured load capacity is somewhat higher than predicted. This is most likely caused by the higher pressure drop in the supply line feeding the unloaded side of the bearing (because of the higher flow to that side) which results in less internal preload and greater load capacity. The bearing flow, shown on Fig. 31 is about 20% higher than predicted but still well within the capability of the nitrogen system. The most obvious explanation for the higher flow probably lies in the value of the average bearing clearance. Since flow is a function of the cube of clearance, a variation in radial clearance of less than .0001 inches would account for 20% difference in flow. Figure 32 shows load vs eccentricity curves for several different supply pressures. It should be noted that, except at the lower supply pressures, the curves are quite straight indicating little or no tendency towards lock-up at eccentricity ratios up to 0.6.

A typical multi-exposure photograph of the oscillograph display of the loader bearing distance transducer outputs are shown on Fig. 33. The scribed circle represents the clearance circle and the two points lying on the circle show the shaft journal touching the bearing surface at the top and bottom of the bearing. (The two transducers are located 45° on either side of the vertical center line so that vertical motion appears at 45° angle on the oscilloscope). The center point was displayed by applying 100 psig to both the upper and lower supply manifolds with zero load applied to the bearing. The remaining points were taken by applying loads from 128 to 783 lbs. to the bearing with supply pressures of 75 to the unloaded side and 150 psi to the loaded side manifolds.

As in the case of the thrust bearings, the loader bearing tests showed it to be completely free of any pneumatic instability. As noted earlier, since the loader bearing was an accessory and not the central object of the test program, care had been exercised to design it conservatively, so that its region of operation would be far removed from any likelihood of instability or of lock-up.

3. Rig Check-out Tests Using Water and Alcohol

The first rotor-bearing dynamics tests were run in a water loop without the loader bearing, to check out the operation of the test rig and all its instrumentation, as well as to obtain bearing and rotor dynamics tests data over the full speed range for comparison with the theoretical data. The results were encouraging in that they showed that the principal components of the test rig, i.e. the bearings, seals and instrumentation, operated satisfactorily as noted below. The water tests, however, pointed out a number of practical problems, in time to correct them before starting the tests in liquid metal (NaK) ambient. These are also discussed below:

The test runs were made over the speed range of 3000 to 8000 rpm. These showed:

1. The operation of the bearings was satisfactory, with the predicted load capacity and without any tendency to instability.
2. The combination of clearance and centrifugal seals effectively prevented any leakage of lubricant from the bearing cavity. This is vital for the proper functioning of the proximity probes, as leakage of the liquid lubricant to the probe faces would either short out the probes or cause erratic readings. For this reason a substantial part of the design effort was devoted to the seals, to insure accurate measurement of the bearing and rotor-dynamic performance. Based on the tests conducted, the seals and proximity probes operated satisfactorily.

The three problem areas that were found in the check-out tests and the corrective action taken were:

1. The assembly procedure that had been developed prior to starting the dynamic tests permitted a misalignment between the axis of the drive motor and that of the test shaft that had exceeded the 0.020" tolerance

for the crowned face spline coupling. Basically, the reason for this was that the plane on the outboard bearing housing that served as the reference plane for the drive motor assembly, was itself misaligned relative to the rotor axis due to a stack up of tolerances. Accordingly, a new assembly procedure was developed which used the axis of the test rotor after its assembly, as the reference for the drive motor assembly. With the new assembly procedure, very good alignment was achieved and the spline coupling was always free.

2. Despite the use of a rust inhibitor (potassium dichromate) in the water loop, some corrosion did take place in the tool-steel bearing pivots near the contact area. With time this caused a change in the bearing clearance, requiring disassembly and re-lapping of the pivots. Accordingly, an alcohol loop was set up and was used in place of water for the balance of the tests, prior to the switch-over to NaK.
3. Galling was exhibited in the two slinger seal units, at their pilot diameters. In one case this was sufficiently severe that one of the slinger seal units cold-welded to the shaft at the pilot diameter and had to be milled out. The slinger seal units, which were previously made of 410 stainless steel, were replaced by units of carbon steel, with silver plating specified to prevent galling. Anti-galling coating compatible with NaK, was also used to prevent a recurrence of this problem during the NaK tests.

Two sets of tests using denatured ethyl-alcohol as the lubricant, were run over the speed range up to 24,000 rpm. In the first set of tests, the shaft was fully balanced and there were no measurable orbits at any speed.

In the second set of tests an unbalance of 0.16 oz.in. was imposed to operate the unit under dynamic load. This test was also conducted over the speed range to 100% of design speed. The calculated orbit for this unbalance was 0.5 mils at 24,000 rpm and this was also the measured orbit.

These two preliminary tests indicated smooth and stable operation over the complete speed range. They showed that the new motor alignment procedure was satisfactory, the spline coupling being completely free both before start-up and immediately

after shutdown. The bearing clearances were fully repeatable and there was no indication of corrosion. (The subsequent disassembly of the unit, after the liquid metal tests, confirmed this and also showed that there was no galling with the new seal rings).

Evaluation of the test data showed precise agreement between the steady-state theoretical and test data up to 15,000 rpm. Thereafter the measured film thickness was smaller than the calculated data and also smaller than at lower speeds. The largest difference between the calculated and measured film thickness was 0.4 mils. This was at 24,000 rpm where the calculated film thickness was 1.8 mils and the measured film thickness was 1.4 mils. There was no discrepancy between the calculated and measured orbit under dynamic loads

This difference between the calculated and measured film thicknesses, while small, is significant in that it illustrates, we believe, the effect of fluid inertia at the pad inlet at high speeds. To investigate this effect further during the subsequent tests in the NaK, loop, 1/16" radius chamfers were introduced at the leading and trailing edges of the pad to determine the effect on the entrance and exit losses.

Since the tests with the low viscosity fluid did not show any difficulties that would prevent liquid metal bearing tests, the conversion to liquid metal operation was then started.

4. Check-out Test Using NaK

The dynamic tests were conducted with a balanced shaft at speeds up to 12,000 rpm. The liquid metal bearing tests confirmed the smooth and stable operation of the unit that was anticipated on the basis of the low viscosity fluid tests. The test data obtained was generally in agreement with the theoretical data. The only significant difference noted between operation with NaK and with water and with ethyl alcohol was a larger starting torque with NaK.

The purpose of this first run with NaK lubrication was to rapidly scan the operation of the loop, cover gas system, purifier, test rig and instrumentation, as well as the interconnecting lines and controls and to identify problem areas. All components appeared to operate satisfactorily. Only some relatively minor problems were found. These were:

1. the dry box drain system was inadequate and a pool of NaK formed in the bottom of the dry box. There was some splatter of NaK in the dry box, aggravated by the high velocity nitrogen discharged from the centering thrust bearing. In consequence, after about 15 to 20 minutes of operation there was some NaK mist in the dry box which interfered with the operation of capacitance probes.
2. The collection of NaK in the dry box reduced the NaK level in the sump tank so that, also after about 20 minutes of continuous operation, the EM pump suction was not fully submerged and the lubricant and coolant flows become erratic.

To correct this situation the NaK drain plumbing was changed that the lubricant was guided directly out of the bearing cavities, through closed lines, into the drain line to the NaK pump.

The first set of tests after modifying the drains was conducted with a balanced rotor at up to design speed (24,000 rpm). Test data was collected over the speed range from 2600 rpm to 24,000 rpm and several runs over this speed range were made. An unbalance of 0.29 oz. in. was then introduced in the rotor and tests were repeated, this time at up to 26,400 rpm. In this latter test, the journal orbits over the speed range were recorded in both test bearings.

Typical reduced test data along with theoretical predictions are shown in Figs. 34 through 37. Figure 34 shows the locus of shaft motion for tests in both NaK and alcohol. It will be noted that the increase in pad leading edge radii accomplished after conclusion of the alcohol tests eliminated the deviation from the straight line shaft locus at small eccentricities (at speeds in excess of 15,000 rpm). Figure 35 which shows the measured and the theoretical steady state pivot film thickness as a function of speed still indicates a deviation from the theoretical of up to about .0003 inches at the higher speeds (small eccentricity ratios). The orbits resulting from the rotating load shown on Figs. 36 and 37, however, indicate reasonably good agreement between theory and test.

Part of the problem of lack of agreement at small eccentricity ratios was attributed to the high coefficient of friction of the bearing and pivot materials in the

NaK environment. The pivot configuration requires that the cylindrical member slide on the spherical member until the sphere contacts the cylinder at the bottom of the cylindrical surface. This is the theoretical pivot "point". If the coefficient of friction between the pivot members is too high, the pad (containing the cylindrical member) cannot slide all the way to the point where the spherical member contacts the cylindrical member at the bottom of the cylindrical surface. Under this condition, the actual pivot point is displaced from the theoretical point and the bearing clearance is smaller than theoretical (by the amount of the height of the segment defined by the displaced pivot chord length).

Several symptoms indicated a pivot friction problem as summarized below:

1. During initial assemblies of the rig with very clean parts, it was difficult to set the pads at the desired clearance and repeatability of the clearance measurement was poor. By applying a small amount of oil to the pivot surface, the problem was completely eliminated.
2. The shaft starting torque in NaK was excessive, occasionally to the point of lock-up. This was attributed to the fact, that without rotation, the upper pads fall away from the pivot surfaces and rest instead on the journal surface. When rotation is initiated, the upper pads rotate with the shaft until the cylindrical pivot member contacts the stationary spherical member at a point high on the cylindrical curvature. Under this condition, there is actually no bearing clearance and if the friction between the shaft and pads is also high, the pad can become wedged between the shaft and pivot causing the bearing assembly to lock up. (Reversing rotation direction freed the pads when this occurred). With high pivot friction coefficients, the angle between the tangent at the contact point and the line of action may not be large enough to cause the pad to slide back to the theoretical pivot point resulting in both reduced clearance and a displaced pivot position.
3. Analysis of the test data indicated that the shaft axis was not moving in a direction parallel to the axis of the bearing clearance that was established at assembly. Since the only forces acting upon the shaft were bearing and gravity forces, it was probable that the actual bearing axis was different from the axis obtained at assembly. This could

have been caused by changed pad radial positions resulting from the start-up pivot friction phenomenon described above.

The experience of a relatively high starting torque with NaK lubricated tilting pad bearings had previously also been encountered in the bearings of the NaK PMA pump for the SNAP 8 development. A hypothesis that explains this problem is the fact that NaK — a strong reducing agent — will rapidly dissolve any oxide or other contaminant films on the pad and journal surfaces, thus increasing the coefficient of friction at startups and, possibly, causing the pads to "adhere" to the journal surface when the rig is shut down. As shaft rotation is then restarted, and particularly, if this is done slowly, with a low applied torque at start-up, the pads will tend to rotate with the shaft causing the ball elements of the pivot to rise up on their spherical or cylindrical sockets thus reducing the effective bearing clearance as described above. This points up the need for a design and experimental study of suitable pivot configurations that, in addition to meeting the other pivot design criteria earlier discussed, will also prevent the pads from moving with the shaft at start up. This study was, however, outside the scope of this program whose principal aim was the study of the hydrodynamic behavior in the fluid film.

Accordingly, a modification of pivot design was made that would eliminate the problem in the subsequent, complete test program. This modification comprised changing the cylindrical socket element to a flat and installing a loose locating pin at the pivot point, which penetrated into both the spherical and flat elements of the pivot.

This modification proved to be entirely successful for the test program — in that no further starting difficulties were encountered at any time in the subsequent, complete test program. This modification, while acceptable for a test program of relatively short duration (of the order of 150 hours total test time), cannot, however, be recommended at this time for the long term operation of the order of 10,000 to as much as 50,000 hours required of practical turbomachinery.

A ball on flat pivot design is acceptable, provided that the Hertzian contact stresses are kept low (of the order of about 120,000 psi for hardened tool steels and somewhat higher for tungsten carbide elements) and the pivot stiffness is kept

substantially higher than the film stiffness. However, the use of a locating pin that interrupts the high stress contact zone is certainly not desirable in long life applications, it is likely to produce stress concentrations that will result, in time, in pivot damage.

After completing the pivot modifications described above, the test rig was re-assembled, with the loader bearing installed, in preparation for the full test program. Before this was undertaken, however, the check-out tests were completed by cycling the test rig, operating in NaK, over its full speed range from 4000 to 26,000 rpm and static load range from 0 to 325 lbs/bearing; both without any imposed unbalance and with an unbalance of 0.412 oz.in. During this check-out test sequence, and the test rig, instrumentation and test loop all operated satisfactorily and no further problems were experienced.

This completed the check-out of the test facility and the complete test program was initiated, as described in the next sections of this report.

V. SUMMARY REVIEW OF THE TEST APPARATUS

The detailed description of the test rig, NaK loop instrumentation and other components of the test facility was given in the previous sections. It is appropriate to briefly recall at this point the principal features of the test rig, before proceeding to a discussion of the test data and its correlation with theory.

The drawing of the test rig was given in Fig. 1. The rig is also shown schematically in Fig. 38. It consists of two tilting-pad test bearings located outboard of a central hydrostatic loader bearing. Steady-state load is applied to the loader bearing by means of a nitrogen pneumatic cylinder through ten parallel tension springs. Dynamic load is applied by changing the rotating unbalance by means of set screws in the shaft face near the loader bearings. The test rig, located inside a nitrogen filled dry box, is driven from the outside of the dry box by means of a high speed electric motor (rated at 30 hp and 36,000 rpm) through a quill shaft and labyrinth seal arrangement. Thrust loads are taken by means of hydrostatic nitrogen thrust bearings located on the shaft face at the loader bearing. A photograph of the assembled test rig is shown in Fig. 39.

The salient characteristics of the test rig are:

Rotor weight:	57 lbs (shaft) + 26 lbs. (loader bearing assembly)
Speed range:	3000 to 28,000 rpm, infinitely variable (design speed = 24,000 rpm).
Bearing size:	2" dia x 2" long
Bearing type:	four pad, tilting pad with 0.55 pivot position
Machined clearance ratio:	0.0004 "/"
Lubricant:	NaK at 110-120°F
Range of Reynolds Numbers:	1250 to 11,700 (10,000 at the 24,000 rpm design speed)
Steady state load application capability	0 to 325 lbs. per bearing, i.e., 0 to 81 psi on projected area

To measure the position of the shaft relative to the bearing (including amplitude of motion under dynamic loading), capacitance type distance transducers were incorporated at four approximately equally spaced planes along the shaft (see Fig. 38). These planes (which are normal to the shaft axis) are so located that both shaft position and bending can be determined.

The measurement planes are located and fixed to each end of each test bearing housing (3.5 and 4.25 inches from the test bearing center plane) such that the transducers sense the distance from the housing to the shaft surface. Since the bearings are positively located in the housings by the pivots, the transducers, in effect, measure the shaft position relative to the bearing pads, after correction is made for the deflection of the pivot elements. As earlier described, this correction is made from the calibration curve of load vs pivot deflection. Each measurement plane contains two transducers spaced at 90° with the output of the two connected to the horizontal and vertical amplifiers of a cathode ray oscilloscope. In this manner, the scope displays an accurate picture of shaft motion within the measurement plane. A single stationary dot on the scope indicates no relative motion of the shaft with respect to the housing, whereas a circle or other pattern indicates the amplitude and locus of dynamic motion. The frequency of motion is ascertained by switching one of the probes from the X-Y mode to a time base display.

To measure the motion of the unloaded pads during the sequence conducted with the bearing pads assembled with zero preload, inductive type transducers were placed directly in the bearing housings "looking at" one lower pad in each test housing*. These transducers sensed motion of the pad through a liquid metal gap. The output of these probes was read out on a cathode ray oscilloscope.

* Since the loader bearing applies a vertical "upward" force on the rotor, it is the lower pads, i.e., the pads located in the lower half of the bearing, that are "unloaded". While the vertically upward direction for the force transmitted by the loader bearing was governed by the need to locate the load mechanism at the top of the assembly, i.e., above any NaK level, as well as by the geometry of the existing dry box, it has the added advantage that the applied load can be made to exactly cancel out this rotor weight, thus permitting test data to be obtained at precisely zero load, as well as at higher loads. The zero load condition is, of course, representative of a vertical rotor operation.

The test rig had several subassemblies, which had been described in some detail earlier in Section II of this Volume of the report, and illustrated in Figs. 12 (the test rotor), 13 and 14 (the loader bearing) 15 and 16 (the bearing housing and bearing parts) and 18 (subassembly showing the lower half of the cylindrical support structure).

The NaK circulation assembly was also described earlier in Section III of this Volume of the report and illustrated in Figs. 19 (flow lines schematic), 20 (loop drawing), 21 (loop photograph) and, 21 (complete schematic). Figure 22 showed the installed loop, dry box and Wayne-Kerr amplifiers, in the test cell. Figures 24 and 25 respectively show the panel board and the visual readout equipment used.

VI. STEADY-STATE LOAD TESTS

The results of the steady-state load tests are shown on Figs. 40 through 46 with the bearings assembled for a 0.5 geometrical preload coefficient. Figures 47 through 50 show the similar data with the bearings assembled for zero preload. These curves denote the eccentricity in mils (.001 inches) of the shaft relative to the bearing geometric center.

Since the capacitance probes were mounted on the outside of the bearing housing (up to 4-1/4 inches away from the bearing center line) the probes measured shaft bending in addition to journal displacement in the bearing. To establish the bearing displacement, it was necessary to plot the measured shaft deflection curve at each load (using the probe output data). From these plots, the actual journal displacement could be determined at each bearing location. This procedure is illustrated in Fig. 51.

Because the pivots of tilting pad bearings have finite stiffness, the true bearing clearance will vary with the bearing load. Thus, at assembly, the pads are arranged to have a particular clearance C_{Bo} and, hence, a particular geometrical preload factor, $m_o = 1 - (C_{Bo}/C_p)$ where C_p is the difference between the radii of curvature of the pads and of the shaft. However, as the shaft is rotated at high speed, hydrodynamic pressures are generated under each pad. The resultant of these pressures, on each pad, acts as a radial force, whose line of action passes through the pivot. Since the pivot comprises two elements in Hertzian contact, with a finite radial stiffness, there will be some deflection of the pivot elements and, hence, an increase in the effective clearance in the bearing; i.e., if C_{Bo} is the clearance at assembly and Δ is the average deflection of the pivot members, then the true clearance will become $C_B = C_{Bo} + \Delta$, and the corresponding, true geometrical preload coefficient will be $m = 1 - (C_B/C_p) = m_o - (\Delta/C_p)$.

In the comparison of theoretical and test data given here, the true bearing clearance (C_B) and the true geometrical preload factor (m) were computed at each operating condition and used instead of their zero-speed, assembly values

(C_{Bo} and m_o), to insure that the theory to test data comparisons were accurately made. The corrected values of operating bearing clearance and preload factors were calculated using a small digital computer program that, for each given operating condition, computes:

- a) the load component carried by each pad and, hence, the deflection of each pivot. For a four pad bearing where the steady-state load line passes midway between the pivots of adjacent pads, the increase in radial clearance is then the average of the deflection of the four pivots.
- b) the vertical shift in the effective bearing center. This is due to the fact that the two pads located in the bearing half on which the applied load acts carry both the applied load and the bearing preload, whereas the two pads in the other half of the bearing carry only the bearing preload. Thus, the pivot deflection of the first pair of pads is larger than that of the second pair of pads. This results in a small, vertical shift of the effective center of the bearing which, for a four pad bearing where the load line passes midway between pivots of adjacent pads is:

$$\delta = \frac{W \cos^2 45}{2 K_{\text{pivot}}} = \frac{W}{4 K_{\text{pivot}}}$$

where, W is the total applied load on the bearing and K_{pivot} is the stiffness of each pivot. Since the film thickness measurements in the tests were made relative to the housing, they were also corrected by substituting the bearing center shift δ in order to establish the operating eccentricity, relative to the true bearing center. These corrections have been made in the data presented in Figs. 40 through 50.

The test results shown in Figs. 40 through 46, for the steady-state load tests with the bearing pads assembled so that the geometrical preload (at assembly) was 0.5, show generally good agreement with the design data obtained from the thin film, turbulent lubrication theory. In general, the test data shows slightly higher load capacity than theoretical value. This is due in part to two factors:

- a) the centrifugal growth of the shaft and, consequent slight reduction of the clearance amounting to 160 microinches at 26,000 rpm and,
- b) the actual pivot deflections were found in static tests to be a little smaller than those calculated from the Hertzian deflection equations (Ref. 5), the discrepancy being 140 microinches under an applied load of 300 lbs. This also makes the actual bearing clearance smaller than the calculated one, with an apparent increase in load capacity.

Generally, however, the test data agrees with the theory within about 10 percent, which is entirely satisfactory for bearing design purposes.

The results of the steady-state load tests of the zero preloaded bearing shown on Figs. 47 through 50 show fair agreement with theory and there is somewhat more scatter, than with the nominal 0.5 preload case. The zero preload bearing, being very much less stiff than the one assembled with a preload of 0.5, is much more susceptible to extraneous side loads, machining and assembly inaccuracies, and pivot friction. The effect of these inaccuracies is most clearly seen in the typical shaft center locus shown in Fig. 52. This figure shows that the bearing attitude angle is not zero and that it is, in fact, relatively large in the near centered position, i.e., under very light loads. It is precisely because of the relatively low stiffness characteristic of non-preloaded tilting pad bearings when: (a) operated at zero (or very small) applied loads and (b) lubricated with a low viscosity fluid, that preloading is strongly recommended as a good design practice. In contrast, the attitude angle for the 0.5 preload case is almost exactly zero as predicted by theory. This can be seen by the typical shaft locus shown in Fig. 53.

During testing of the bearings assembled with a preload of 0.5, no evidence of instability was ever experienced. Attempts at forcing the shaft into an instability by shocking the housing while operating the test rig under zero load conditions were totally unsuccessful.

In contrast, this inherent stability was not experienced with the bearings assembled with zero preload. In these tests, the bearing always exhibited some marginal instability at zero load and speeds greater than 11,000 rpm ($Re \approx 5000$).

Under these conditions, fractional frequency whirl was experienced although its amplitude was controlled and small compared to the clearance space. Furthermore, only small loads were required to stabilize the bearing. The magnitude of these loads were of the same order as those required to measurably increase the bearing stiffness from its centered position value. That is to say, bearing stiffness is nearly independent of load in the range of $e = 0$ to $e = 0.2$. Beyond 0.2, bearing stiffness begins to increase very rapidly with load. It was at about an eccentricity rate of 0.2 that the bearing became stabilized due to the applied load. The conclusion that can be drawn from this is that the non-preloaded bearing, although marginally unstable, requires only a small amount of load to make it stable. Thus, relatively small amounts of preload which are easily and automatically achieved in practice by assembling the bearing to have a small, positive geometrical preload factor. A preload factor of 0.3, should be adequate to insure stable operation for turbomachinery comparable in size, rotor mass, pad mass and speed range to the test rig used in the current tests.

The whirl frequency was found to coincide with the first or second rigid body critical speed of the shaft-bearing system as is shown in Fig. 54. For example, at an operating speed of 23,500 rpm (see dashed line in Fig. 54), the zero load bearing stiffness was calculated to be 1.74×10^4 lb/in. At this bearing stiffness, the second rigid body critical speed occurs at 7600 rpm, while the whirl frequency was measured to be approximately 7500 rpm.

The lower (i.e., the unloaded) pads were instrumented in the zero preload tests to determine their mode of vibration and the effect of shaft stability and load on their operation. Photographs of typical oscilloscope traces indicating pad motion are shown in Fig. 55. Figure 55(a) shows pad motion with a 40 lb. load directed towards the instrumented pad, while Figs. 55(b), (c) and (d) show the same pad with loads of 115, 168 and 227 lbs. respectively in the opposite direction. These figures show that while the unloaded pads vibrate with relatively large amplitudes — on the order of .001 to .0015 inches in this case — they vibrate at shaft frequency in all cases. That is, they follow the shaft and do not vibrate at their own natural frequency. Also, the amplitude of vibration, once started, is relatively insensitive to shaft position. This characteristic was seen at all other speed and load conditions.

Figure 56 shows typical shaft and pad motions at near zero load where the shaft was vibrating in a fractional frequency mode. Figure 56(a) shows pad motion while Fig. 56(b) and (c) show shaft motion. Figure 56(b) is a simultaneous display of the two probe outputs connected to the vertical and horizontal deflection axes of the scope which results in a true picture of the shaft motion in space. Figure 56(c) shows each of the two probe outputs as a function of time. Two things should be noted in Fig. 56. First, looking at Fig. 56(b) and (c), the mode in the "Y" direction is basically at shaft frequency while the mode in the "X" direction is at fractional frequency. This suggests that, contrary to theoretical predictions, the bearing does not have equal stiffness in all directions. This "preferential orientation" phenomenon is, however, most likely due to small assembly inaccuracies which result, in this particular case, in a higher stiffness in the "Y" direction. Referring back to Fig. 52 which shows the shaft locus with applied load, this reduced stiffness in the "X" direction can be readily seen in the large attitude angle near the center position. From a practical standpoint, this is only another illustration of the very low stiffness of tilting pad bearings when there is no externally applied load, and when the bearing does not have any preload.

It may also be noted in Fig. 56 that the dominant pad mode of vibration is primarily fractional frequency. The once-per-revolution frequency can also be seen superimposed on the basic fractional frequency wave. As in all cases, this once-per-revolution component is in the phase with the shaft motion, i.e., the pad is operating below its pad-fluid film resonance and it is trying to track shaft motions.

Referring to Fig. 57 which shows the whirl orbit and frequency at 23,470 rpm, it can be seen that, as the speed is increased, the whirl frequency, though not its amplitude, becomes the same in both directions.

VII. DYNAMIC LOAD TESTS

The purpose of the dynamic load tests was to determine, experimentally, the stiffness and damping characteristics of the bearings for comparison with their theoretical values which were calculated from the thin film turbulent lubrication theory and given in the design data tables in Volume I of this Report.

Knowledge of the fluid film stiffness and damping characteristics of journal bearings is required to practice to establish the critical speeds of the rotor-bearings systems of turbomachines, as well as to determine the response of such systems to dynamic loads, such as mass unbalance. The critical speeds of a rotor-bearing system are calculated using a digital computer program based on the well known Holzer-Prohl method (Ref. 6). The system response to dynamic loads is calculated from a forced vibration analysis which is basically an extension of the Holzer-Prohl method. The mathematical derivation of the rotor-bearings response analysis was earlier given in a paper of the Topical Report on the "Proceedings of the Technical Meeting on Super-Laminar Flow Bearings and Seals for Process Fluid Lubricated Turbomachinery" (Ref. 7). The numerical, rotor-bearings response calculations are obtained using a computer program based on this analysis.

Since the dynamic characteristics of the bearings, obtained from the thin film turbulent lubrication theory, are used for critical speed and rotor-response calculations, the preferred method for verifying their accuracy consists of using these calculated values, to compute the response orbits of a test rotor to known, imposed unbalances and then to measure these orbits. Comparison of the calculated and measured orbits then provides a measure of the accuracy of the dynamic bearing data and of the level of confidence with what it can be generally applied to practical turbomachinery rotors. In other words, the accuracy of the bearing dynamic characteristics is directly checked through an actual application, using a test rotor.

In the program reported here, known unbalance weights were applied to the test rotor which is supported in the NaK lubricated bearings, and the dynamic orbits,

over a range of speeds up to 26,000 rpm, as well as over a range of externally applied static load to 300 lbs. per bearing, were measured and compared with the calculated orbits. This same approach had been used in an earlier task of the program, with three different test rotor arrangements operated in turbulent flow bearings lubricated with an "easy-to-use" low viscosity fluid, and reported in Ref. 7.

The calculated and measured amplitudes of vibration, as functions of speed, are shown in Figs. 58 through 60. These are for the case where the load applied through the loader bearing was equal and opposite to the gravity load (i.e. to the weight of the rotor and the loader bearing assembly). Thus, the Nak lubricated test bearings were operating with no applied load, simulating vertical operation. The data given in Figs. 58 to 60, was obtained with the bearings assembled with 0.5 geometrical preload. The data is given for the inboard probe at the No. 1 bearing and for both the inboard and outboard probes at the No.2 bearing. Runout at the outboard probe of the No. 1 bearing, during this sequence of tests invalidated its readings.

This data was obtained with unbalance loads applied to the rotor in two planes near the ends of the loader bearing. Each of these two unbalances was 0.206 oz.in. and they were located at the same circumferential locations, i.e. the two unbalances were placed in phase with one another. The transmitted dynamic loads at each bearing due to these unbalances amounted to a rotating dynamic force of 240 lbs. at 26,000 rpm. This same magnitude and location of unbalance were used throughout in the dynamic load tests.

The measured amplitudes, given in Figs. 58 through 60, agree reasonably well with the calculated values, the maximum deviation being about 150 microinches. It should be noted that although the measured amplitudes are generally below the calculated amplitudes, they follow quite well the shape of the calculated curves. Two potential sources of error explain the magnitude and trend of the differences between the calculated and measured amplitudes.

First, extraneous external loads and moments imposed upon the loader bearing by the gas and coolant plumbing caused the test bearing to operate at an eccentricity ratio larger than zero during these tests. As a result, the

actual bearing stiffness was somewhat larger than calculated resulting in reduced synchronous whirl amplitudes. Since these extraneous external loads are not speed dependent, the actual eccentricity ratio will decrease with increasing speed and the actual stiffness will approach the stiffness used in arriving at the calculated amplitudes. Therefore, the error should be the largest at the lower speeds. This trend can be seen in the data presented in Figs. 58 through 60.

Secondly, two sources of extraneous unbalance exist in the rotor. These are at the splined coupling and the seal slingers, neither of which could be completely corrected for in the initial shaft balancing operation. Since the extraneous or residual unbalance should be small compared with the imposed unbalance, the effect should be small at the lower speeds where the shaft behaves as a rigid body. At the higher speeds, however, where some shaft bending occurs, these extraneous unbalances may slightly alter the shaft mode shape and the measured amplitudes.

The effect of coupling unbalance on the coupling end of the shaft at 18,000 and 26,000 rpm is also shown in Figs. 59 and 60. Here, 0.010 oz.in. of unbalance (the limit achievable within the specified limits on dimensional tolerances) was assumed at the coupling, both in and out of phase with the applied unbalance of 0.412 oz.in. It can be seen from Figs. 59 and 60 that the measured amplitudes fall within the calculated limits defined by the aforementioned coupling unbalance and that the shape of the calculated curve above 18,000 rpm is modified to appear more like that measured. The effect of No. 1 bearing end slinger unbalance of 0.006 oz.in. (the level of pretest balance repeatability) on the No.1 bearing inboard measurement plane amplitude at 26,000 rpm is shown in Fig. 58.

Considering the magnitude and direction of the effects of small extraneous load and unbalance on the calculated vibration amplitudes, the agreement between test and theory are satisfactory.

To verify the bearing dynamic coefficients at eccentricities other than zero, vibration amplitudes were also measured with finite static loads applied to the bearings. The results of these tests are presented in Figs. 61 through 65 for the zero preload case, and on Figs. 66 through 69 for the bearings assembled with a geometrical preload factor of 0.5. This data is presented as amplitude vs shaft

length for several discrete values of speed and load. These test points also agree reasonably well with the theoretical predictions.

This completed the test program in NaK. In general, the test results showed a satisfactory degree of agreement with the theory, particularly in the case of the bearings when assembled with some preload. When the bearings were assembled without any preload and, particularly, when little or not external load was applied by means of the loader bearing, the test data exhibited some scatter. As noted, however, this is due to the fact that, in this case, the NaK lubricated, test bearing pads were essentially unloaded and, furthermore, operated at a large clearance ratio (0.004 "/"). The fluid film stiffness was very small and secondary effects such as small pad assembly in accuracies and the minor side loads imposed by the NaK and nitrogen lines became significant. From a practical standpoint, preloading is recommended when designing tilting pad bearings lubricated with low viscosity fluids, as such preloading achieves high fluid film stiffness under all conditions, including the zero applied load case, thereby suppressing any tendency towards instability.

This was the calibration of high-temperature, inductance-type proximity transducers operating in a liquid sodium ambient at three temperatures up to 1000 F. These are described in the next section.

VIII. CALIBRATION TESTS OF HIGH TEMPERATURE, LIQUID METAL PROBES IN LIQUID SODIUM

As we noted earlier, at the time when the NaK lubricated bearing test program was planned, no proximity instrumentation was available that could be used to make the required static and dynamic bearing film thickness measurements directly in the liquid metal ambient. This required the probes to be mounted outboard of the bearing seals, in a region where the ambient was the cover gas. The well developed and highly reliable capacitance type probes using the Wayne Kerr amplification system could then be used and the shaft to housing motions could be calculated at the bearing centerline by interpolation between the readings of the inboard and outboard pairs of probes, taking account of shaft bending as previously described. Such an approach was satisfactory for this test program, where the test rig could be designed with effective seals that prevented any leakage of liquid metal into the probe locations, and where these probe locations could be selected close enough to the bearings to permit accurate interpolation.

Clearly, however, the absence of proximity transducers capable of operation in the liquid metal ambient of prototype, turbomachinery is a much more serious deficiency. Unlike a test rig, a pump or other practical machine, cannot as easily be designed to accommodate proximity instrumentation that can be used only in a cover gas ambient. Accordingly, some limited effort was devoted to locating proximity instrumentation that, unlike the capacitance type, would not be affected by the electrical conductivity of a liquid metal film, and thus, would offer potential for use directly in liquid metal ambients. Inductance type probes were known to have this property and, furthermore, to be little affected by the dielectric of the film, i.e., their calibration curve is essentially the same, whether the fluid is a gas or a liquid metal, or even if both were present in the gap. This was confirmed in tests conducted by MTI's Instruments Division, using inductive proximity transducers and electronic amplification and readout equipment manufactured by Associated Engineering Ltd. (England). Calibration tests conducted at low temperatures in air, water, mercury and NaK, showed no difference in the calibration curves.

Accordingly, a program modification was requested, and approved by the AEC, wherein inductive probes were added to the test rig and used to measure pad motions relative to the housing, to supplement the shaft-to-housing measurements made with the

capacitance probes. The measurement of the pad motions directly in the NaK ambient proved to be successful as described above, in the review of the static load tests.

Currently, it is known that a principal area of liquid metal lubrication in nuclear turbomachinery will be for the large sodium pumps and some other components of the sodium system of the Liquid Metal Fast Breeder Reactor. The sodium lubricated bearings that will have to be developed for use in this sodium system will be required to operate at temperatures ranging, in some cases, up to about 1200F. The bearing (and other component) developments will be greatly aided if proximity probes and other instrumentation (e.g. for fluid pressure measurements) were to be developed, capable of operating in liquid metal ambients at elevated temperatures. Since a high temperature version of the inductance probe, that had been successfully used in the pad motions studies was available, it was deemed advisable to attempt a calibration of this type of probe directly in liquid sodium at several temperature levels. Authorization for this was requested and received from the AEC and the calibration tests were conducted as described below. It should be emphasized, however, that these calibration tests were intended only to test feasibility and to identify further instrument development needed to arrive at a practical and reliable system for film thickness measurement. These preliminary calibration tests were conducted at three temperature levels, 500F, 700F and 1000F.

The transducer used in the tests consists of a ceramic encapsulated coil wound around iron wires shaped to form a double "U" section, the ends of which form the "working face". Supplying a high frequency a.c. signal to the coil produces an alternating magnetic field in the iron wire and across the working face. When a magnetic material intercepts this field, an inductance change occurs which varies the high frequency current in the coil. This current variation is a function of the proximity of the magnetic material and, thus, of the gap between the transducer working face and the target magnetic material. Since the liquid metal in the gap is non-magnetic, it is not "seen" by the probe.

The electronic gear consists of a regulated power supply, an oscillator to generate the probe exciting or carrier frequency and a detector-filter unit (DF-4). The DF-4 consists of a fixed inductance which is wired in bridge with the probe, a

filter circuit to demodulate the probe current, and controls to balance the bridge network and vary the D.C. output (gain) to the desired level.

The ceramic encapsulated probe (Model No. HT-6) was housed in a 300 series stainless steel body with a .001 inch thick 300 series stainless steel diaphragm covering its working face. This diaphragm, which was electron beam welded to the housing, produced the hermetic seal which separated the liquid metal from the probe and the atmosphere. Except for the fact that it represents a fixed gap of .001 inches, the non magnetic diaphragm does not influence the induced current in the probe.

An assembly drawing (160F70) of the calibration fixture with the probe installed as shown on Fig. 70. The fixture consists of a 300 series stainless steel tank (which contains the sodium) with the probes mounted in the bottom. The spring loaded magnetic 400 series stainless steel target is moved by means of a hand operated micrometer which is mounted on top of the fixture and is thermally isolated from the tank portion to keep it near room temperature. The target motion is measured by the micrometer. The tank portion of the fixture is heated by electrical resistance heaters strapped to the tank O.D. (not shown on Fig. 70) and the entire assembly is insulated with Vermiculite.

The fixture is provided with two probes, one of which is directly beneath the movable target and the other looking at a fixed gap. The purpose of the second or dummy probe is to provide temperature compensation. With the second probe wired into the bridge network with the measuring probe, changes in resistivity and permeability of the probe, target and diaphragm materials with temperature are cancelled. This temperature compensation feature was not tried on the subject calibration tests because of a leadwire failure on the dummy probe early in the test program.

A collar was incorporated on the measuring probe which provided a positive stop for the target short of the diaphragm. The resulting gap (0.0016 inches) was used as the "reference" zero to set-up both the micrometer and the electronic gear.

The results of the calibration tests in sodium at 500, 700 and 1000°F, are shown on

Figs. 71 through 73. For these tests, the DF-4 inductance bridge was balanced with the test set-up at room temperature and the probe gap set at .00165 inches (the target bottomed against the spacing collar). The carrier frequency for the tests was 50 KC. For calibrations at elevated temperatures, the DF-4 balance was not readjusted but the D.C. offset was adjusted at each temperature to produce zero output with the target bottomed against the collar.

At 1000⁰F two test series were conducted, one with a carrier frequency of 50KC and one with a carrier frequency of 125 KC. The 50 KC tests were conducted with the original room temperature balance adjustments while the 125 KC tests were conducted after rebalancing the DF-4 inductance bridge with the probe at 1000⁰F and the gap set at 0.00165 inches.

The results shown on Figs. 71 through 73 clearly indicate that the inductance system produces a good output at all temperatures to 1000⁰F with Na in the gap and that temperature has only a minor influence on sensitivity. Both linearity and repeatability leave something to be desired although this particular shortcoming is common with the type of electronic gear used in this test. It is also clear from Fig. 73 that the increased carrier frequency results in reduced sensitivity and shortens the linear range.

These tests demonstrated the feasibility of adapting the inductive measurement system to high temperature liquid metal machinery and pointed to improvements that will be necessary before such a system can be considered state-of-the art. These improvements are summarized below:

1. Improve the D.C. stability of the probe and electronic system to permit better repeatability (and accuracy).
2. Extend the useable range to approximately 0.040 inches.
3. Improve the linearity to at least 75% of full range.
4. Increase the temperature range to about 1200⁰F (current Ht-6 probe is limited to about 1100⁰F)
5. Develop a rugged high temperature encapsulation technique and leadwire system.

IX. CONCLUDING REMARKS

The technology program that was conducted under AEC Contract AT(30-1)-3363 comprised an investigation of the fluid-dynamic performance of liquid metal lubricated bearings operating in the turbulent flow regime. It was intended to develop theory and readily useable design data for bearings operating in this regime and to check out the validity of this data through rotor-bearings experiments first in easy-to-use fluids and, then, in a liquid alkali metal (NaK). The program generally followed the recommendations for such a fluid-dynamic study of liquid metal lubrication given in the report of I.D.A. Space Power Bearing Panel (which included MTI participation) and which was issued by the Institute for Defense Analysis in November 1962*.

The technology program was limited to the fluid-dynamic performance of liquid metal bearings and its scope did not include either of the other two important problem areas identified by the IDA Panel, notably: liquid metal bearing materials and liquid metal bearings instrumentation.

In Volume 1 of this report, extensive design data tables were given for tilting-pad bearings operating in the turbulent flow regime. These were obtained from the thin film turbulent lubrication theory that had been developed under the program and presented in an earlier Topical Report. This theoretical treatment took account of the developed turbulence in the fluid film, but it did not treat fluid inertia effects which are related to the conversion of static to velocity heads in the flow within the film, as well as at entrance to and exit from the bearing lands. Based on theoretical grounds, comparing the magnitudes of the fluid inertia and the turbulent stress terms of the turbulent-flow Navier Stokes equation, a map was given in Volume 1 which delineated the range of applicability of the theory and, hence, of the design data. Briefly, it was found that the theory and, hence, the design data obtained from it should be applicable when all the following conditions are observed.

* "A Program for Research and Development in Liquid Metal Lubricated Bearings," George C. Szego, Institute for Defense Analysis Report No. IDA/HQ-62-874, November 1962.

1. the bearing lands are smooth and do not contain pockets or other sharp discontinuities
2. assembled bearing clearance ratios do not exceed 0.002 $''/''$
3. the fluid film Reynolds Number (based on assembled clearance) is within the range from 1000 to 15,000. (When the fluid film Reynolds Number is less than 1000, laminar-flow theory should be used).

The experimental design data obtained with NaK lubricated bearings, and described in this second Volume of the report, showed generally good agreement with the calculated design data, confirming the validity of the theory within the operating region limits denoted above. In fact, at least as far as clearance ratio is concerned, the test data suggested that the above cited limit may be somewhat restrictive. Good agreement between theory and test was found at an assembled clearance ratio of 0.004 $''/''$, as well as at one of 0.002 $''/''$. The above cited restrictions are, nevertheless recommended as they are quite conservative.

The test program confirmed also the very satisfactory performance of tilting-pad type bearings in high speed applications. From a practical design standpoint, the following design features are recommended when tilting pad bearings are used with low viscosity fluids such as liquid metals.

1. Preloading is strongly recommended to achieve high stiffness under all conditions, including the zero load condition. In the absence of preloading and when the bearing load is very small, some marginal instability can be exhibited at high speeds, as the individual pads are then unloaded and lack radial stiffness.
2. Pad edges should be chamfered, to prevent entrance losses, which can reduce load capacity and stiffness, at high operating speed.

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(a) Fixed Shaft Axis- Floating Test Bearing

(b) Fixed Test Bearing - Pivoted Shaft

(c) Fixed Test Bearings - Floating Loader Bearing

Fig. 2 Test Rig Configurations

Fig. 3 Test Bearing Load Capacity

Fig. 4 Test Bearing Load Capacity

Fig. 5 Test Bearing Stiffness

Fig. 6 Test Bearing Damping

Fig. 7 Test Bearing Power Loss

Fig. 8 Shaft Critical Speed vs Support Stiffness - NaK Bearing Test Rig
 (with loader bearing)

Fig. 9 Shaft Critical Speed vs Support Stiffness - NaK Bearing Test Rig
(without Loader Bearing)

NaK Bearings Test Rig

Calculated Response at No. 1 Bearing

Unbalance = 0.206 oz.in. in each balance plane, in-phase.

Fig. 10 Rotor Response to Unbalance

NaK Bearings Test Rig

Calculated Response at No. 2 Bearing

Unbalance = 0.206 oz.in in each balance plane, in-phase.

Fig.11 Rotor Response to Unbalance

Fig. 12 Test Rig Rotor

Fig. 13 Assembled Loader Bearing

Fig. 14 Partially Disassembled Loader Bearing

Fig. 15 Schematic of Test Bearing

(a) Bearing Assembly

(b) Bearing and Seal Parts

Fig. 16 Bearing Housing Showing Heat Exchanger, Bearing Pads and Pivots, Seal Plates, and Proximity Probe

Fig. 17 NaK Bearing Test Rig - Pivot Stiffness and Stress

Fig. 18 Test Rig Partially Assembled

Fig. 19 NaK Circulation Assembly Interface Requirements

Fig. 20 NaK Circulation Assembly

Fig. 21 NaK Circulation Assembly

F 160FO4

LEGEND

- PS PRESSURE SWITCH
- S SWITCH
- L LIGHT
- T TEMPERATURE SENSOR
- P PRESSURE SENSOR
- VP VACUUM PUMP
- NaK
- NL GAS
- ELECTRICAL INSTRUMENT
- PNEUMATIC CONTROL LINE
- NEEDLE VALVE
- SELF-ACTUATED PRESSURE REGULATING VALVE
- SOLENOID VALVE NORMALLY CLOSED
- SOLENOID VALVE NORMALLY OPEN
- PRESSURE RELIEF VALVE

1	34	VACUUM VALVE	AEC 165619
1	33	VACUUM PUMP	AEC 165650
1	32	TUBING	COPPER
1	31	TUBING	ST STEEL
1	30	GAS PURIF. SYS.	MSAR MODEL 60-500
1	29	NAK CIRC. SYS.	MSAR 9K, YA 120395
1	28	RELIEF VALVE	1/4" NPT(F), 2-7 PSI RANGE, COMPOSITION DIAPHRAGM & SEAT - FISHER GOVERNOR CONTROLS TYPE 90H 30HP, 36, 600RPM, 600V 600Z, 3Ø, 31.5 AMPS
1	27	IND. MOTOR	ALLIS CHALMERS MODEL 1
1	26	EL. CONTROLLER	SUPPLIED WITH ITEM 1 MSAR NAK CIRCULATION SYSTEM
1	25	TEMP INDICATOR	24 POS, AEC HONEYWELL-BROWN TYPE U
1	24	POWER SUPPLY	30V DC, FILTERED, 115V 60HZ, AEC 165551
1	23	ELECT. COUNTER	ELECTRO MODEL N6R 115V 60HZ, AEC 165452, 1
2	22	ELECT. GLAND	HEWLETT-PACKARD MODEL 522B
2	21	PANEL METER	TYPE "A", LAVA SEALANT - CONAX CORP #EG-09-SS-L DC, 0-3 MILLIVOLT, ±2% ACCURACY
3	20	PANEL METER	ASSEMBLY PRODUCTS INC. MODEL 351. DC, 0-5 MILLIVOLT, 20A ±20%, ±2% ACCURACY -
1	19	PRESS. CONTROL	ASSEMBLY PRODUCTS INC. MODEL 351. 115V 1/4", OPEN HIGH/CLOSE LOW, 50-450 LBS RANGE - PENN CONTROLS, INC #4PIIA
1	18	SOL. VALVE	115V 60HZ, 2 WAY, N.O., 500 PSI OPERATING PRESSURE - ATKINSON VALVE, MODEL BN 4001
1	17	SOL. VALVE	115V 60HZ, 2 WAY, N.C., 500 PSI OPERATING PRESSURE - SKINNER ELECTRIC V-5 SERIES, #V52HDB2050Z
1	16	VACUUM GAGE	0-500 MICRONS, M1E0D TILT TYPE-STOKES MODEL 276-BB
1	15	T.C. WIRE	TYPE J (IRON-CONSTANTAN), 20AWG
2	14	BELLOWS VALVE	1/4" CV 11.2, BUTT WELD - FULTON SYLPHON I-3321-B
4	13	GLOBE VALVE	1/4" TUBE, BRASS
1	12	PRESS GAUGE	HOKE #D3212 G4-B (GYROLOK FITTINGS)
2	11	PRESS GAUGE	WALL MTD, BACK CONNECTED, 0-30" HG, 4 1/2" FACE, 1/4" NPT (M) - ASHCROFT QUALITY GAUGE MODEL 1010
8	10	PRESS GAUGE	WALL MTD, BACK CONNECTED 30-PSIG, 4 1/2" FACE, 1/4" NPT (M) - ASHCROFT QUALITY GAUGE MODEL 1010
1	9	PRESS GAUGE	WALL MTD, BACK CONNECTED, 0-60 PSIG, 4 1/2" FACE, 1/4" NPT (M) - ASHCROFT QUALITY GAUGE MODEL 1010
2	8	PRESS GAUGE	WALL MTD, BACK CONNECTED, 0-600 PSIG, 4 1/2" FACE, 1/4" NPT (M) - ASHCROFT QUALITY GAUGE MODEL 1010
2	7	PRESS GAUGE	WALL MTD, BACK CONNECTED, 0-300 PSIG, 4 1/2" FACE, 1/4" NPT (M) - ASHCROFT QUALITY GAUGE MODEL 1010
2	6	PRESS GAUGE	WALL MTD, BACK CONNECTION, 0-900 PSIG, 4 1/2" FACE, 1/4" NPT (M) - ASHCROFT QUALITY GAUGE MODEL 1010
8	5	PRESS. REG.	RELIEVING TYPE, 125 PSIG (MAX) INLET, 0-15 PSIG SUPPLY, MOORE PRODUCTS MODEL 40
1	4	PRESS. REG.	RELIEVING TYPE, N2 SYSTEM, 400 PSIG (MAX) INLET, 0-30 PSIG SUPPLY, 1/4" NPT (F) - NORGREN 11-004-016
2	3	PRESS. REG.	RELIEVING TYPE, N2 SYSTEM, 400 PSIG (MAX) INLET, 0-250 PSIG SUPPLY, 1/4" NPT (F) - NORGREN 11-002-017
2	2	PRESS. REG.	RELIEVING TYPE, N2 SYSTEM, 400 PSIG (MAX) INLET, 0-125 PSIG SUPPLY, 1/4" NPT (F) - NORGREN 11-002-013
2	1	PRESS. REG.	RELIEVING TYPE, 116 1/2 SCFM (MAX), 5000 PSIG (MAX) INLET, 3200 PSIG (MAX) SUPPLY, 1/4" NPT (F) - HOKE 920A10
ITEM NAME		DESCRIPTION	
F 160FO4		MECHANICAL MOUNTING INCORPORATED LATHAM NEW YORK	
F 160FO4		INSTRUMENTATION & CONTROL SCHEMATIC FOR LIQUID METAL BEARING TEST RIG	
F 160FO4		CODE REVISION NO. 1 SCALE: AS SHOWN DATE: 11/10/04	

Fig. 22 Test System Schematic

Fig. 23 Test System Installed in Cell

Fig. 24 Control Panel

Fig. 25 Control Area

Fig. 26 Test Rig Thrust Bearings

Fig. 27 Hydrostatic Thrust Bearing
Deflection vs Supply Pressure

Fig. 29 Dimensionless Flow vs Restrictor Coeff.
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Fig.47 Load vs Bearing Eccentricity

Fig. 48 Load vs Bearing Eccentricity

Fig.49 Load vs Bearing Eccentricity

Fig. 50 Load vs Bearing Eccentricity

Fig. 51 Diagrammatic Representation of Method for Obtaining Motions at Bearing Centerlines from Inboard and Outboard Probe Radians

$Re \approx 8000$
 $N = 18600 \text{ RPM}$

Fig. 52 Typical Shaft Locus for $m = 0$ Bearing

Re $\approx$ 8000
N = 18800 RPM

Fig. 53 Typical Shaft Locus for $m = .5$ Bearing

Fig. 54 Shaft Critical Speed vs Support Stiffness

(a)

Bearing Load = - 40 lbs.

(b)

Bearing Load = 115 lbs.

(c)

Bearing Load = 168 lbs.

(d)

Bearing Load = 227 lbs.

Sweep Time = 0.010 sec/cm
Speed = 18660 RPM
 $Re \approx 8000$
 $m = 0$

Fig. 55 Effect of Bearing Load on Pad Motions

Speed = 18660 RPM
 $Re \approx 8000$
 $m = 0$
 Bearing Load = 0

← 0.05 v/cm

(a)

Sweep Time = 0.010 sec/cm

2.2 mils/cm

(b)

← x

1.1 mils/cm

← y

(c)

Sweep Time = 0.005 sec/cm

Fig. 56 Shaft and Pad Motions During Fractional Frequency Instability

Speed = 23470 RPM
 $Re \approx 8000$
 $m = 0$
 Bearing Load = 0

(a)

Sweep Time = 0.010 sec/cm

(b)

2.2 mils/cm

(c)

Sweep Time = 0.005 sec/cm

1.1 mils/cm

Fig. 57 Shaft and Pad Motions During Fractional Frequency Instability

NaK Bearings Test Rig

Steady State Bearing Load = 0

Lubricant: NaK at 110-120F

Unbalance = 0.206 oz.in. in each of the two balance planes

Geometrical Preload Factor (Assembled) = 0.5

Amplitude Measurements made at the Inboard Probe of the No. 1 Bearing

Theoretical Data: —————

Experimental Data: +

Fig.58 Rotor Amplitude vs Speed

NaK Bearings Test Rig

Steady State Bearing Load = 0

Lubricant: NaK at 110-120 F

Unbalance = 0.206 oz.in. in each of the two balance planes

Geometrical Preload Factor (Assembled) = 0.5

Amplitude Measurements made at the Inboard Probe of the No. 2 Bearing

Theoretical Data: —————

Experimental Data: +

Fig.59 Rotor Amplitude vs Speed

NaK Bearings Test Rig

Steady State Bearing Load = 0

Lubricant: NaK at 110-120 F

Unbalance = 0.206 oz.in. in each of the two balance planes

Geometrical Preload Factor (Assembled) = 0.5

Amplitude Measurements made at the Outboard Probe of the No.2 Bearing

Theoretical Data: —

Experimental Data: +

Fig. 60 Rotor Amplitude vs Speed

Fig.61 Rotor Maximum Amplitude Mode Shape

Fig 62 Rotor Maximum Amplitude Mode Shape

Fig. 63 Rotor Maximum Amplitude Mode Shape

Fig. 64 Rotor Maximum Amplitude Mode Shape

Fig. 65 Rotor Maximum Amplitude Mode Shape

Fig.66 Rotor Maximum Amplitude Mode Shape

Fig. 67 Rotor Maximum Amplitude Mode Shape

Fig. 68 Rotor Maximum Amplitude Mode Shape

Fig. 69 Rotor Maximum Amplitude Mode Shape

Fig. 70 High Temperature Liquid Metal Proximity Probe Calibration Fixture

Fig. 71 Inductance Probe Calibration in Sodium

Fig. 72 Inductance Probe Calibration in Sodium

Fig. 73 Inductance Probe Calibration in Sodium

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NOMENCLATURE

a	Feeding radius, inches
C	Journal clearance, inches
C _B	Assembled, journal bearing radial clearance, inches
C _P	Machined radial clearance of bearing pad (radius of curvature of pad - radius of shaft), inches
D	Bearing diameter, inch
K	Stiffness, lb/in.
L	Bearing Length, inch
m	Geometrical preload coefficient $(\frac{C_P - C_B}{C_P})$
M	Mass, lb.sec ² /in.
N	Speed, RPS (also used in the dimensionless, gas thrust bearing charts in Figures 28 & 29 to denote number of feeding holes)
P _a	Ambient pressure, psia
P _s	Supply pressure, psia
Q	Flow, SCFM (also used in the dimensionless gas thrust bearing chart in Figure 29 to denote mass flow in: lb.sec/in)
R	Radius, inch
R ₁	Thrust bearing outer radius, inch
R ₂	Thrust bearing inner radius, inch
Re	Reynolds Number $(\pi \rho D N C_P / \mu)$
R	Gas Constant, in ² /sec ² -°R
T	Temperature, °R
v	Volts
W, W _B	Bearing load, lb.
$\bar{X}$	Pivot location (arc distance of pivot from leading edge ÷ pad arc length)

δ	Inherent compensation parameter ($\delta = a/2C$ where a = feeding hole radius and C = mean clearance)
Λ_s	Gas bearing feeding parameter
μ	Absolute viscosity, $\text{lb}\cdot\text{sec}/\text{in}^2$
ρ	Mass density, $\frac{\text{lb}\cdot\text{sec}^2}{\text{in}^4}$
ν	Kinematic viscosity, (μ/ρ) , in^2/sec .
ϵ_B, ϵ	Bearing eccentricity ratio

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