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PROGRESS REPORT

For the Period

August 1, 1968 - October 31, 1968

**LIQUID METAL BEARINGS TECHNOLOGY FOR LARGE,
 HIGH TEMPERATURE SODIUM ROTATING MACHINERY**

AEC CONTRACT AT (30-1)-3930

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**MECHANICAL TECHNOLOGY INCORPORATED
PROGRESS REPORT**

on

**LIQUID METAL BEARINGS TECHNOLOGY FOR LARGE,
HIGH TEMPERATURE SODIUM ROTATING MACHINERY**

For the Period

August 1, 1968 - October 31, 1968

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**Contract AT(30-1)-3930-3
U. S. Atomic Energy Commission
January, 1969**

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I. INTRODUCTION

The purpose of the program, currently in progress under Contract AT(30-1)-3930 is to develop the design data and instrumentation that will be needed for reliable design, optimization and experimental evaluation of the sodium lubricated bearings for the high capacity sodium pumps as well as other auxiliaries of the sodium circulation systems of large IMBR plants. To this end, the program has been planned to generate the following:

- a. extension of turbulent flow bearing theory to incorporate fluid inertia effects, followed by preparation and experimental verification of accurate bearing design data and procedures applicable to the large, highly turbulent bearings.
- b. accurate and reliable instrumentation with which proximity and film pressure measurements may be made directly in the high temperature sodium environment.

The scope of this program and its individual tasks are defined in the Project Plan and in the previously issued first progress report (Ref. 1).

During the period (August 1 through October 30) covered by this report, the principal effort was in the following areas:

1. Complete development of the turbulent flow lubrication analysis, including fluid inertia effects.
2. Continued development of the proximity probe for operation in high temperature liquid metal environment.
3. Continued development of the film pressure probe for operation in high temperature liquid metal environment.
4. Continued design and analysis of the liquid metal bearings test rig.

The progress in these areas is summarized in the next section and discussed in more detail in Section III of this report.

11. SUMMARY OF PROGRESS

The progress by task during the period covered by this report is discussed in detail in Section III. Briefly, however, this included:

1. Task II, the Fluid Inertia Investigation, which comprises the analysis and computation procedures necessary to prepare the design data for large sodium lubricated bearings that operate in the turbulent inertial flow regime was successfully completed. The work conducted under this task, and its results, are described in a separate Topical Report, Ref. 4.
2. Work under Task III, is starting with calculations to optimize shrouded step bearing configurations for operation with low viscosity lubricants (such as liquid sodium) and under high levels of turbulence.
3. Under Task IV-A, an in-house development of proximity (film-thickness) measuring transducers suitable for operation in high-temperature, liquid sodium ambient was started in September and is currently in progress. This development became necessary when the present probe manufacturer proved unable to deliver the required transducers. The availability of the probes is necessary for the forthcoming bearing tests in liquid sodium and they are also needed for future component tests and monitoring of bearing performance in prototype pumps. Accordingly, the internal development of the transducers is being pursued as a matter of urgent priority and is proceeding satisfactorily. Completed probes are expected to be available for evaluation in late January 1969.
4. Under Task IV-B, the development of the film pressure measurement transducers for operation in high temperature sodium was completed. A sample probe is being subjected to fatigue testing to evaluate the ability of the transducer, including its diaphragm and coated surface to withstand large, cyclic pressure changes, when the probe is installed in a rotating journal, to scan the circumferential pressure profile in the bearing.
5. Under Task V, the analysis and design of the test rig for conducting bearing tests with 12 inch diameter bearings in liquid sodium at up to 1200 F is proceeding. Analysis of the major components are substantially completed and preparation of manufacturing detail drawings was initiated. A design modification whereby the loader bearing was relocated to be in close proximity to the test bearing was also completed. This design modification was made in order to eliminate a shaft deflection problem and to achieve closer simulation of the loading conditions of the sodium lubricated bearings of vertical shaft sodium circulation pumps.

111. DISCUSSION OF PROGRAM PROGRESS

The progress during the current reporting period was summarized in the previous section. Following is a more detailed discussion of the work conducted in each task.

TASK I - Program Management and Reporting

Letter type monthly progress reports were issued and a formal report describing the progress of the program during its first six months (February 1 through July 31, 1968) was prepared and issued (Ref. 1). The quarterly review meetings initiated by NYOO were held.

TASK II - Fluid Inertia Investigation

The technical work under this task was completed at the end of October, 1961, in compliance with its Work Statement and which is defined as follows in the Project Plan:

"This is a fundamental task of the project where the theoretical basis for the design handbook on the large, liquid metal lubricated bearings will be prepared. Under this task, the turbulent lubrication theory will be extended to include fluid inertia effects. The pressure profiles generated in a 12" diameter bearing with different step geometries will be measured for comparison with the theory, specifically including the static to velocity head conversions at sharp discontinuities in bearing films. A non-corrosive, low viscosity fluid such as a light silicone oil will be used as the lubricant. The general equations for turbulent flow bearings, including fluid inertia effects will be generated and the numerical solution of these equations will be developed and programmed.

A Topical Report will be issued covering the results of this task".

This effort included:

1. An analysis was developed for hydrodynamic lubrication of bearings, that takes full account of fluid film turbulence and the effects of convective

fluid inertia forces. Turbulence was handled by means of the earlier developed analytical treatment of thin film turbulence (Refs. 2 & 3). Convective inertia was accounted for by means of the convective derivatives of the mean film velocities averaged over the bearing film. The work conducted included a direct, numerical solution of the simultaneous equations representing conservation of momentum and fluid continuity.

2. The analysis was applied to pad bearings of arbitrary geometry and incorporated in a computer program to calculate the film pressure developed in such bearings. The results of this program were compared with the pressure profiles measured in a 12 inch diameter, four-pad, shrouded step bearing lubricated with two silicone fluids.

The work conducted in this task and its results are fully described in a Topical Report (Ref. 4) and will not be duplicated here. Briefly, however, the conclusions of the work are as follows:

1. The theory for full, turbulent inertial conditions in pad-type bearings was verified by the experimental comparisons. The detailed correlation of the analytical and test data is given in Ref. 4, however, the mean discrepancies between the predicted and measured peak pressures were found to range from a low of 3 percent to a high of 8 percent. The integrations of the calculated and measured profiles showed that the differences between predicted and measured load capacities would be close to or slightly less than the above noted differences in peak pressures. Thus, pending the later comparisons of calculated and measured bearing performance in high temperature sodium ambient, it appears that the analysis and computer program developed in Task II may be used to generate accurate bearing performance data in turbulent flow regime and including fluid inertia forces.
2. Additional comparisons were made between the calculated bearing pad pressure data and that measured in earlier tests that had been conducted by another Investigator in an independent study (Ref. 5). These tests had been conducted in highly turbulent flow regime and where fluid inertia effects were very large; The analytical

results agreed quite closely with the measured ones accurately predicting the very severe pressure gradients that occurred at the sharp boundary between pocket and land, due to fluid inertia.

3. Both the effects of fluid inertia and the effects of film turbulence tended to increase the peak pressures attained in a bearing, As to which effect was most significant, this depended on the relative values of the Reynolds Number, N_{Re} , which is the parameter governing the level of turbulence and the reduced Reynolds Number, $N_{Re}(C/R)$, which is the parameter governing the relative importance of inertia effects. In tests conducted with $N_{Re} = 5000$ and $N_{Re}(C/R) = 16.7$ both effects were of roughly equal significance, and the two effects combined to increase the peak pressure rise to approximately four times the value predicted on the basis of laminar; non-inertial calculations.
4. A major characteristic of convective inertial effects is that they tend to produce a very sharp drop in pressure at the step at the end of a pocket, often giving rise to subambient pressures over the land following the pocket. This pressure drop is associated with the acceleration of flow into the relatively narrow clearance in the land region. These inertial pressure drops were predicted with very good fidelity by the analysis.
5. A relatively small fraction of the reference dynamic head is recovered as pressure at inlet to the pad. All comparisons under inertial conditions show this fraction (the recovery factor) to lie between 10 and 20 percent and the recommended value for the recovery factor was 15 percent. A brief investigation shows that potential errors in stiffness resulting from universal application of this recovery factor will be less than ± 15 percent.

TASK III - Preparation of Bearing Design Charts

This task comprises generation of design data for three self-acting bearing configurations selected for potential use in large sodium lubricated applications such as the sodium circulation pumps for LMFBR service, and preparation of the bearing design handbook. This design data will be generated using the analysis and computer program generated under Task II. The work on this task was started late in October, with calculations to optimize bearing pad geometry from the standpoint of maximum load capacity under highly turbulent flow conditions.

TASK IV - Development of Proximity and Film Pressure Instrumentation for High Temperature Liquid Sodium Environment

Part A: Inductance Type Proximity Transducer

Non-contact proximity sensors are used to measure the locus of the journal within the bearing clearance. Such measurements are required for experimental determination of the eccentricity ratio and attitude angle for steady-state loading. For accurate measurements of this type it is necessary for the sensors to exhibit very low DC drift, including the minimal drift due to temperature change. The sensors are used also to measure the motion of the rotor axis in response to dynamic loading, such as that due to unbalance. Accurate measurements of this type require moderate frequency response, low noise (at least in the frequency range of interest) and a linear relationship between output signal and displacement is extremely helpful for straightforward interpretation of the results.

The following target specifications were established for the proximity sensors at the onset of the program:

environment	-----	sodium
temperature	-----	50 to 1100°F
operating range	-----	0.004 to 0.040 in.
linear range *	-----	0.030 in.
sensitivity	-----	0.1 volt/mil
probe and leadwire temperature sensitivity *	-----	10^{-6} in/°F
readout equipment temperature sensitivity	-----	10×10^{-6} in/°F max.
DC drift *	-----	10×10^{-6} in/hr. max.
readout equipment line voltage sensitivity	-----	10^{-6} in/volt max.
noise	-----	30×10^{-6} in equiv.

* when wired in "push-pull"

Initially, the probe development program was centered on the further development, encapsulation and evaluation of an existing commercial, high-temperature inductance probe. Early experiments (Ref. 6) with this probe had shown promising results when operating encapsulated in a NaK environment at temperatures to 1000°F. The inductive probe has a major advantage over other non-contact proximity sensor concepts. This is the fact that its output can be insensitive to the presence of a conducting medium in the gap between the probe tip and the target (shaft) surface. This characteristic was demonstrated by the results given in Ref. 6. It is necessary, however, that the target surface be ferromagnetic for effective operation of the inductive probe.

The principle of operation of the inductive sensor is to detect the effect of target proximity on the reluctance of the probe magnetic circuit. A typical probe configuration is shown in Fig. 1. There is a core made of magnetic material which is typically of a Configuration known as "pot core". This consists of an outer tube and an inner, concentric post connected together by a solid web at one end. The open end of the core faces the target surface so that the magnetic flux path from post to outer tube is completed through the target and the intervening "air gap". A coil is wound around the center post of the core and the magnetic flux field is produced by passing an AC current through this coil. The self-inductance of the coils depends on the reluctance of the magnetic circuit and the latter is a function of the spacing between the face of the core and the target. If the permeabilities of the core and the target are constant, the relationship between coil inductance and gap is essentially parabolic with the change in inductance increasing more rapidly with gap as the gap closes.

Detection of the gap separation, and therefore of the target proximity is usually accomplished by placing the probe in an AC bridge circuit so that the changes in the probe coil inductance result in changes in the bridge output signal. Two probes are often used with one on either side of the shaft so that motion of the shaft produces opposing changes in the probe inductances. The probes are connected into the bridge so that opposing changes are additive while changes in the same direction, due to temperature effects for example, tend to be self-canceling. This arrangement, called "push-pull" also gives improved linearity of the gap-output characteristic.

Review of Results with Commercial (HT-6 and DHT-25) Probes

The promising outlook for the commercial (HT-6) probes, based on the test results in NaK, was diminished considerably by the results of elevated temperature testing in an air environment earlier in the present program (Ref. 1). Measurements were made of the pertinent electrical characteristics of several HT-6 probes during temperature cycles in an oven. The principal findings of these tests were:

- (1) There were irreversible changes in the coil resistance and in the inductance after temperature cycles going ^{above} about 800°F. After several cycles to temperatures of 900-1000°F, the probes failed because of open circuit of the coils.
- (2) The changes in coil inductance with temperature were large when compared to the change in inductance with gap. Changes with temperature (70-900 F) were of the order of 2 to 1 while the change in going from infinite to closed gap was typically 15 percent. This means that large temperature induced drifts must be expected even when using two probes in push-pull for compensation.

The HT-6 probe also has a rather small operating range. The range of gap variation for linear response with push-pull probes is about 10-15 mils. One sample of a larger probe (DHT-25) with a greater range (estimated to be 40 to 60 mils linear range for a push-pull pair) was received from the manufacturer. This probe showed much better temperature stability and more desirable electrical properties than the HT-6 probes. It was subjected to a number of temperature cycles to 600°F and showed a change in inductance with temperature of about 20 percent over that range. The upper temperature was limited in these tests because this was the only available sample and it was considered important to keep it operable and available for evaluation of the detection equipment. The DHT-25 probe showed a much lower coil DC resistance (by a factor of about 5) and higher Q (by a factor of nearly 2) than the HT-6 probe. These differences are considered to be desirable for good stability and performance of the detector.

Early in the current reporting period, the manufacturer repeatedly set back his delivery dates on additional HT-6 and DHT-25 probes for evaluation. Inquiries revealed that he was experiencing difficulties in obtaining satisfactory per-

* Q is defined as the ratio of the inductive reactance to the total AC resistance.

formance and life at elevated temperature in his own pre-shipment qualification tests. Because of these difficulties and considering the importance of a reliable, effective proximity sensing system to the conduct of liquid-metal bearing tests, it was decided that MTI would undertake the development of a high-temperature inductive probe as an independent effort parallel to the commercial probe evaluation. In September, the manufacturer of the HT-6 and DHT-25 probes informed us that they had not yet resolved their difficulties and they could not give any definite date for further deliveries. The internal probe development was also started at that time.

Internal Development of High-Temperature Proximity Probe

The failures of the commercial HT-6 probes were directly due to open circuits of the coils. These coils were made of ceramic-insulated, nickel or stainless steel clad copper wire. The ends of the wire were brazed or welded to tubular metal connectors passing through a ceramic seal at the back of the probe. The core of these probes was made up of a number of soft iron wires, each of which is bent into a U shape. The wires were used instead of a solid core because of the resulting reduction in internal core losses (eddy currents). However, the arrangement appeared to lack good mechanical stability, especially when subjected to thermal cycling. Also, sectioning of several probes after elevated temperature operation showed evidence of substantial core wire oxidation.

Based on the test results and post-operation examination of the HT-6 probes, the following guidelines and design improvement objectives were set for the internal probe development effort:

- (1) A larger probe size, comparable to the 1-inch diameter and 2.2 inch length of the DHT-25 probe, was an acceptable, and desirable, trade off for the better range and electrical characteristics which this made possible.
- (2) Significant improvements in the high-temperature life and stability of the coil wire and in the coil fabrication and connections were considered essential.
- (3) Greater oxidation resistance and higher Curie point of the core material were highly desired.

- (4) A core configuration with better mechanical stability was needed without too much of a penalty in internal core losses.
- (5) If possible, the probe should be fully metal encased with all joints, including the coil lead connectors sealed by welding.

Parallel and concurrent investigations were begun to (1) select a coil wire and develop the coil fabrication and connection techniques, (2) specify a core material and a core configuration, and (3) prepare a tentative design for the complete probe assembly including the coil, core, casing, leads and the methods of fabrication and assembly. Progress in each of these areas is described next.

Coil Development

Several alternative approaches were considered to achieve the objectives of improved coil life and stability at high temperatures. These included:

- (1) Continuing the use of ceramic insulated wire but using a superior wire material, preferably a single metal without cladding. The clad copper wires had good electrical properties (low DC resistance) but diffusion of the cladding metal into the copper at high temperatures and probable disruption of the protective cladding at the welded or brazed connections are major drawbacks.
- (2) Use of a high-temperature alloy wire, such as Nichrome, with a self-formed, adherent oxide coating for insulation.
- (3) Use of a metal sheathed, mineral-insulated wire of the type used for high temperature resistance heaters.

The sheathed, mineral-insulated wire concept was quickly rejected because it was realized that the metal sheath would result in large internal losses in the coil. Also the packing density (number of coil turns per unit volume) would be extremely poor with this approach.

The approach of using a high-temperature alloy with self-formed oxide protection was discarded when we found that most of the promising candidate wires are magnetic. A coil wound of one of these materials produces very

little external flux field. The remaining materials of this type have very high resistivity resulting in a very low coil Q , and therefore low probe sensitivity.

A survey was made of ceramic-insulated, high-temperature magnet wires including contacts with manufacturers and a review of published reports by other investigators working on high-temperature electrical components. We concluded that a ceramic insulated gold wire was the best choice for good electrical properties and demonstrated high-temperature performance. A quantity of 10-mil diameter ceramic-insulated (Secon D) gold wire was ordered and was received near the end of the report period.

A special coil-winding fixture has been designed, fabricated and assembled (Fig. 2). This fixture features close control of the wire tension during winding and permits visual inspection of the wire and its insulation through a stereo microscope during winding. Winding of the first gold wire coil (500 turns) was completed at the end of the report period. Considerable care was taken to insure an absolutely level wind. Nylon gloves are used when handling the wire to avoid possible contamination of ceramic insulation. As received, the ceramic is only partially cured to retain flexibility of the ceramic.

The coil is wound onto a coil form of machinable ceramic which is furnace cured after machining. The ceramic insulation of the wound coil wire was then fully cured by heating to 1500 F in the furnace. The ends of the wire are then anchored to the form by "Astrocera" cement (Type A-LP).

The DC resistance, inductance and Q of the coil (with air core) were measured before, during and after the 1500 F ceramic curing cycle. The results are given in Table 1.

Table 1: Electrical Properties of Ceramic-Insulated Gold Wire Coil Before, During and After Ceramic Curing Cycle (1,2)

<u>Temperature, F</u>	<u>DC Resistance, ohms</u>	<u>Inductance, mh</u>	<u>Q</u>
80	14.4	1.72	-
400	23.0	1.72	0.55
900	34.5	1.72	0.42
1300	46.7	1.70	0.30
1500	50.4	1.65	0.26
1250	43.2	1.74	0.32
1000	36.7	1.74	0.40
775	31.8	1.73	0.42
75	14.4	1.74	0.82'

- (1) Measurements made with Hewlett Packard Universal Bridge, Model 4260A.
- (2) 500 turn coil, 10-mil diameter wire, operated with air core, connected to the bridge by two .011" dia. alumel wires fused to the .010 gold wire termination.

The results are encouraging in that they show good stability and little or no irreversible change in properties in spite of the high peak temperature and the fact that this was a curing cycle and stability should improve for subsequent thermal cycles.

Another encouraging aspect of the curing Cycle measurements was the fact that the coil wire to lead wire connections stood up well. Prior to winding the coil, some trail connections were made between the gold wire and alumel lead wires. The melting point of alumel is well above that of gold and joints were made both by melting only the gold wire, and by melting both. Good results were obtained with both approaches but it was considerably easier to obtain a good connection by melting only the gold wire and this technique was used to make the connections to the wound coil before the ceramic curing cycle. These joints showed no signs of degradation or irregular behavior during or after the curing Cycle. The inductance and Q of the coil was measured with the open

end of the core in contact with a 416 stainless steel target surface (closed gap) and with no target (open gap). Different slot configurations were then evaluated by progressively machining slots into the cores. Results are given in Table 2. The desired properties are a high inductance and a high Q with a maximum change in inductance with gap. Based on the results of the measurements, it was decided to use a 50 mil wall thickness core since this gave the best combination of high inductance, high Q and high sensitivity. There appeared to be relatively little further improvement in magnetic properties when more than one slot was used and additional slots weaken the core.

Table 2 Coil Inductance and Q for Different Core Wall Thicknesses and Slot Configurations (1,2,3)

Core Wall Thickness inch	Open Gap Inductance, (L_o) mh	Q_o	Closed Gap Inductance, (L_e) mh	Q_e	Sensitivity $\frac{L_e}{L_o}$
<u>Without Slots</u>					
.030	6.0	4.2	8.0	3.4	1.33
.050	6.7	3.4	9.8	2.4	1.46
.080	7.2	2.6	10.0	1.5	1.39
<u>One Diametral Slot Through Walls Only</u>					
.030	5.8	5.4	8.0	4.9	1.38
.050	7.0	4.8	10.5	3.6	1.50
.080	7.7	3.8	12.1	2.5	1.57
<u>Two Diametral Slots Through Walls Only</u>					
.050	6.8	5.1	10.4	4.1	1.53
<u>One Diametral Slot Extending Through Walls and Web</u>					
.030	5.9	6.7	8.1	5.9	1.37
.080	8.1	4.7	13.4	2.9	1.65
<u>Two Diametral Slots Extending Through Walls and Web</u>					
.050	7.1	6.7	11.1	4.9	1.56

- (1) Excitation frequency 1 KHz
- (2) Cores and Targets - 416 stainless steel
- (3) Measurements made on a 500 turn, 10 mil diameter copper wire coil mounted on the cores.

A Hiperco 27 core of 50 mil wall thickness was made. A single radial slot was machined in steps beginning with the center post, continuing through the base web, and then extending on through the outer shell. Results obtained with this core are shown in Table 3.

Tabel 3: Coil Inductance and Q For a
Core of .050 Inch Wall Thickness
and Single Radial Slot (1.2)

	Open Gap	
	Inductance <u>mh</u>	<u>Q</u>
Without Slots	5.8	3.2
One Radial Slot in Center Post	6.1	4.0
Slot Continued Across Base Web	6.2	4.2
Slot Continued Up Outer Shell	6.4	6.2

- (1) Excitation Frequency - 1 KHz
- (2) Core Material - Hiperco 27

The single radial slot extending through both walls and the web gave very nearly as much improvement in electrical properties as even the two diametral slot configuration. For this reason, and because of superior structural stability, the single radial slot core configuration was chosen for further development and for the tentative probe assembly design.

The complete, high-temperature probe for use in liquid metal will be encapsulated in a non-magnetic casing with a thin non-magnetic diaphragm over the face of the core. The effects of placing the core (.050" wall) in an outer shell of 304 stainless steel was investigated. There was no measurable effect on the coil inductance or Q, even when the core was electrically in contact with the casing. The effects of the diaphragm were investigated by placing pieces of 304 stainless steel shim stock on the face of the 416 stainless steel core with two diametral slots through walls and web. Results are shown in Table 4. The effects on the open gap inductance and Q are negligible. There is some effect on the closed gap measurements which is attributed to the "standoff" caused by the presence of the shims between the core face and the target:

Table 4: Effects of 304 Stainless Steel Shim
on Face of 416 Stainless Core (1,2)

	Open Gap		Closed Gap	
	Inductance mH	Q	Inductance mH	Q
No Shim	7.02	6.6	10.8	5.0
4-mil shim over core sensing face	7.04	6.6	10.1	5.2
8-mil shim over core sensing face	7.04	6.6	9.5	5.4

(1) Excitation Frequency = 1 KHz.

(2) Core Wall Thickness = 50 mils, two diametral slots through walls
and base web.

The oxidation resistance of the Hipercor 27 cores will be evaluated by high-temperature furnace tests in the near future. If necessary, protective coatings, such as gold plating will be tried for improved oxidation resistance. The problem can be avoided by vacuum purging and backfilling the encapsulated probe with inert gas if necessary. However, it would be desirable to avoid the necessity for this, particularly since a requirement for encapsulation would significantly complicate elevated-temperature evaluations of the electrical properties and performance of the coil-core assembly prior to fabrication of a final probe assembly will be performed following thermal cycling test with the coil alone. These experiments will be aimed at:

- (1) Obtaining data on the effects of temperature and repeated thermal cycling on the open-gap inductance and Q of the coil-core assembly.
- (2) Establishing the degree of similarity of temperature effects for several coil-core assemblies to evaluate the effectiveness of a push-pull pair for temperature compensation.
- (3) Obtaining gap-output calibration data for a push-pull pair of assemblies over a range of temperatures.

Insofar as is possible, these experiments will be performed with coil-core assemblies alone, without the added complication of encapsulation.

Tentative Probe Assembly Design

The tentative design of the encapsulated probe assembly **is** shown in Fig. 4. The coil consists of 500 turns of 10-mil diameter, ceramic-insulated gold wire wound on a ceramic coil form. The coil fits over the center post of the Hiperc 27 core which has a single radial slot through both walls and the web. The center post of the core **is** hollow and the leads to the coil pass through the center hole. The lead wires are alumel. When the connections between the coil and lead wires have been made, the lead wires are pulled down into the center hole **so** the joints are inside the hole **and** protected. A ceramic plug fits into the back of the core. **This** plug positions the lead wires and provides strain relief since the leads are bent sharply just outside the ceramic body. The lead wires pass out of the back of the probe through a tube which **is** welded to the probe body. The leads are enclosed in the tube until some point outside the high-temperature, liquid-metal environment **is** reached where a lead-to-tube seal can be made.

The core-coil assembly **is** positioned within the probe body by four tapered pins which are inserted **radially** through the body and the outer wall of core near the sensing end. This arrangement is designed to maintain a fixed position of the core sensing face relative to the end face of the body, while allowing for the possibility of differential thermal expansion. The end of the probe **is** covered by a diaphragm cap of the same material as the probe body. The cap is welded to the body along a flange which is close to the face but behind the locating pins. **This** eliminates the need for a difficult sealing weld between the pins and the body. Tentatively, the diaphragm thickness **is** 10 mils.

During the next several months, we plan to procure a number of test pieces of caps and probe bodies and make a series of trial welds of the joints between these parts and of the tube-to-body joint. Metallographic sectioning and helium leak testing will be used to evaluate the welds. Several weld flange designs for the cap-to-body joint will be tried and evaluated by these means. This effort will be carried on in parallel with the evaluations of the coil and the coil-core assemblies.

Part B: Pressure Probe

The objective of this part of Task IV is to develop the instrumentation, suitable for the use in high temperature sodium environment, that is needed to measure the pressures in the bearing film. The target specifications that were set for this pressure measurement system are as follows:

Primary Sensor

Environment ----- Sodium
 Temperature ----- 50 to 1200 F
 Pressure Range ----- 0 to 500 psig
 No. of Pressure Cycles ----- 5×10^8

High Temperature Leadwire

Environment ----- Sodium
 Temperature ----- 50 to 1200 F
 Pressure Range ----- 0 to 500 psig
 Length ----- to 20 feet
 Minimum Bend Radius ----- 0.5 inch

Detector

Environment ----- Air or Inert Gas
 Temperature ----- 50 to 200 F

System

Operating Range ----- 0 to 500 psig
 Sensitivity ----- 10 MV/PSI
 Linearity ----- $\pm 5\%$ Full Scale
 Line Voltage Sensitivity ----- $\pm 10\%$ No Effect

Based on prior experience with pressure instrumentation for rigorous environments (specifically with development of pressure instrumentation for high temperature steam environment), a hermetically sealed capacitance probe that senses the deflections of an elastic diaphragm was selected for the high temperature sodium application.

The pressure sensors may be developed using this concept for placement either in the rotating shaft or in the stationary bearing, however, the physical arrangement of the completed transducer will be different in the two cases. For the purpose of the tests to be conducted on a large sodium-lubrication bearings under TASK VI of the program, the use of pressure transducers located in the rotating journal has the major advantage that the complete circumferential pressure profile around the bearing can be scanned in each revolution. For this purpose, dynamic pressure capability is required, to show the pressure variation around the bearing. The signal for the probe has to be transmitted to the amplifying and indicating equipment by means of slip rings.

During the previous reporting period, the pressure probe development work was concentrated on development of the manufacturing and hermetic sealing processes. During this development phase, the problem areas were identified, solutions were established and a detailed pressure transducer fabrication instruction was prepared.

The emphasis during this reporting period has been on cyclic testing of one of the probes that have been welded into a plate to simulate its application in a journal. We were particularly interested in identifying any fatigue problems that may be inherent with the transducer design because, in its final application, in the test bearing journal it will be subjected to at least one pressure cycle per revolution. Although the diaphragm stresses can be relatively accurately calculated, the end effects resulting from the swaging, flattening, and welding operations make it necessary to experimentally establish the diaphragm fatigue limit. Furthermore, the effect of diaphragm flexing could have a significant influence on both the coating adherence and the amount of support afforded by the dielectric.

To establish the magnitude of the fatigue problem, one of the transducers (No. 3-1) was subjected to cyclic pressure tests after it has been machined flush with the surface and the surface had been flame sprayed with chrome carbide (nickel chrome bonded) wear resistant coating. It was then pressure cycled between 0 and 700 psi and inspected periodically. (700 psi was established as producing at room temperature approximately the same diaphragm deflection

as 500 psi at 1200°F). At every inspection, a replica of the surface was made and the surface profile measured with a Talysurf. Also during these cyclic pressure tests, the transducer was connected to a capacitance meter and the output measured on a digital voltmeter. Any failure of the coating, or of the diaphragm would be detected from a change in the transducer output or a change of surface profile or both.

A total of more than 10,000 pressure cycles were accumulated during this period, interrupted for surface inspections after 100, 1100, 5000 and 10,000 cycles. The meter output was constant for slightly over 8,000 cycles after which there was some increase in sensitivity. Correlation of meter readings and surface profile measurements will be made. The tests are being continued to establish the fatigue limits of the transducer and to identify the weakest element.

Following the conclusion of this test, determination will be made on whether shaft mounted pressure probes (which have the advantage of providing cyclic scanning of the pressure profile, but which are thus exposed to cyclic stresses) would be used in the liquid metal bearing tests, or whether bearing mounted probes which are not exposed to large, cyclic pressure variations should be used instead.

TASK V - Test Rin Design and Fabrication

The objective of this task is to design and build a bearing test rig in which the theoretical and technology data gained on the other tasks can be verified in bearing test conducted in liquid sodium at elevated temperatures. The test rig must be capable of producing data which is sufficiently accurate to check the validity of the turbulent lubrication theory including fluid inertia forces and to verify the accuracy of the design charts. To accomplish this objective, the test rig design and instrumentation must be such as to permit accurate measurement of the thickness and pressure of the lubricant film at several positions around the clearance space under all operating conditions.

The following summarizes the test rig specification requirements that were established early in the program:

Bearing type -----	Flooded hydrodynamic
Bearing operation -----	Vertical and submerged
Bearing size -----	12" dia.x 12" lg. max.
Bearing C/R -----	.001 to .003
Lubricant -----	Sodium
Temperature -----	1200°F, max.
Speed -----	1200 RPM max.*
Unidirectional load -----	10,000 lb. max.
Rotating unbalance -----	75 in.lb. max.

These conditions were selected so that the operating conditions that the test bearings will be subjected to, will span or exceed those currently anticipated in the radial, sodium lubricated bearing of a 60,000 gpm capacity LMFBR sodium circulation system pump;

The basic test rig concept which was described in the previous progress report consists of a vertical shaft with the lower end submerged in sodium. The sodium is contained within a vessel which also contains the test bearing and appropriate instrumentation and heaters. The upper portion of the vessel is filled with an inert gas which separates the sodium from the atmosphere. A face type seal is provided between the shaft and the top of the vessel to minimize cover gas leakage to the atmosphere.

* Can be increased to 2400 rpm max. with modification of the drive, as noted on page 23.

During the previous reporting period and the early part of this period, several design analyses and design layouts were made which identified the significant mechanical design problem areas, established solutions, and led to finalization of overall arrangement and component detail designs.

This preliminary design phase was completed during this period and the component designs and sizes necessary to accomplish the objectives of the test program have been established. The final detail layouts of several of the components were initiated along with the manufacturing drawings of their parts.

It was noted in the previous report that the shaft bending and thermal stress problem had been made and showed the shaft bending deflection to be excessive. This is due to the fact that it is desired to conduct the bearing tests over a wide range of steady-state loads (0 to 10,000 lbs) on the 12 inch diameter sodium lubricated test bearing. With the loader bearing located midway between the test and support bearings, the maximum applied load at the load bearing is 20,000 lbs, resulting in shaft deflections of the order of 0.050" maximum. This makes it very difficult to insure accurate alignment between the test bearing and the rotor over the full load range. Since alignment is essential in order to obtain accurate bearing performance data, for comparison with the theoretical data of the design handbook, a design study was made to resolve this problem.

Based on the results of this design study, the loader bearing has been relocated so that it is now immediately above the test bearing. With this arrangement, the loader bearing load required to achieve a maximum applied load of 10,000 lbs on the test bearing is reduced to 12,000 lbs. Furthermore, since the loader and test bearings are now in close proximity, the shaft deflection is very small so that the flexure mount of the test bearing can readily insure alignment with the rotor. This relocation of the loader bearing also offered several test rig design simplifications and advantages including:

1. Low bending stress in the high temperature region of the rotor
2. Low thermal stresses in the rotor since a greater axial length is now available to accomplish the reduction in temperature from 1200 F to room temperature.

3. A 300 series stainless steel rotor can now be used because of the reduction in thermal and bending stresses. (An Inconel rotor had been considered in the earlier design due to the high thermal and bending stresses).
4. The removal of the loader bearing from above the vessel cover area makes considerably more space available for the shutdown seal, as well as for the instrumentation connections and instrumentation penetrations.
5. The location of the loading mechanism close to the test bearing simulates the load application in vertical sodium pumps where the radial load is generated in the impeller, which is close to the radial, sodium lubricated bearing.

The static load will now be applied to the rotor by means of a sodium lubricated pivoted pad type bearing that has higher load capacity and stability threshold than the test bearings. Bellows type seals have been incorporated to seal the vessel at the point of penetration of the load arm. Figure 5 shows the component arrangement of the final test rig concept.

The top portion of the test rig, which has been detailed, contains the upper support bearing and mounting structure. The test rotor is supported, or essentially hung, from an angular contact ball bearing. This bearing is mounted in a spherical member which permits both room temperature and high temperature alignment. Alignment of the shaft between the support and test bearings is accomplished by moving the socket member of the spherical support in two mutually perpendicular directions normal to the shaft axis. Alignment is measured by means of the proximity probes mounted on the test bearing.

The test rotor is driven by means of a variable speed drive (U.S. Motors Varidrive) through a right angle gear box which is mounted atop the spherical bearing member. This arrangement eliminates any misalignment of the coupling between the gear box and test rotor. The coupling is a splined type which is located in the same axial plane as the support bearing. This location minimized any coupling effects on the test rotor dynamics since it is at or very near the rotor nodal point under all anticipated operating conditions.

The present right-angle drive gear box has a 1:1 bevel gear arrangement. This drive is interchangeable with a 2:1 speed increaser which would raise the maximum speed of the rotor to 2400 RPM. The other elements of test rig system have been designed to operate at up to 2400 RPM. In this higher speed range (to 2400 RPM) the rotor bearing system tranverses the first, calculated system critical speed.

Mounted on tap of the gear box (which has a hollow, straight-through shaft) is the slip ring assembly through which the signals from the internal shaft thermocouples and pressure transducers are taken. The capacitance type dynamic pressure transducers are integral with the shaft within the area of the test bearing and continuously measure test bearing lubricant film pressure. The capacitance detectors, which demodulate and linearize the transducer signal, are located inside the shaft at about midplane where the temperature is well below their maximum ambient temperature limit of 250°F. The leads between the transducers and detectors are rigidly mounted in five plates axially placed along the shaft to minimize leadwire vibration and any resulting signal noise. These plates also act as shaft stiffening members to maintain the required shaft roundness under conditions of high applied load.

As indicated above, the load to the test bearing (up to 10,000 lbs) is applied through a sodium lubricated tilting pad bearing located within the sodium containment vessel. The load is structurally carried to the upper support and test bearings through a cylindrical member within the test vessel. This member also forms a portion of the vessel cover. By this arrangement, the primary sodium containment vessel is not subjected to any applied bearing loads or stresses. As indicated in the previous progress report, the vessel cover is heated and maintained at the same temperature as the sodium so that the vessel will not be subjected to any axial thermal gradients of stresses. A stress analysis of the pressure vessel to the requirements of Section VIII of the ASME unfired pressure vessel code was completed during the reporting period. The vessel has been designed for 1200°F and 25 psig and meets the code requirements for a design life in excess of 100,000 hours at these conditions. In addition the stress and deflection analysis of the internal cylindrical load carrying member was also completed.

The layout design of the loader mechanism was also completed during this period. The test rig will have two loaders so that test loads can be applied in two mutually perpendicular planes. Since the loader is a tension device, springs displaced 180° from each loader have been incorporated between the loader bearing and supporting structure. The purpose of these is to allow moving the shaft radially through the entire bearing space to permit calibrating the bearing proximity transducers in place. Figure 6 is a schematic of the loader mechanism.

Load is applied to the loader shaft which is supported on linear ball bearings by means of high pressure nitrogen (up to 200 psi) acting on a piston at the end of the loader shaft. Belloframs are used as piston seals to eliminate leakage while still minimizing friction and damping from the loader system as was discussed in the previous progress report. A large volume accumulator will be put into the nitrogen supply line to make the load system pneumatically soft and thus dynamically decouple the loader shaft and bearing from the supporting structure (this feature was also discussed in the previous progress report). The most difficult design problem that had to be solved with the loader mechanism was that of providing a fail safe seal scheme since the loader shaft penetrates the containment vessel below the sodium level. This was accomplished by incorporating double or redundant bellows seals for both the static seal (which must accommodate only static misalignment and tolerance start-up) and the loader shaft seal (which must take both shaft deflection and vibratory motions). Nitrogen at a pressure slightly above cover gas pressure is introduced into the space between each bellows pair. This nitrogen purge accomplishes the following:

1. Minimizes the pressure differential and, thus, the pressure stress across the primary seals.
2. Eliminates the primary bellows seal oxidation problem.
3. Minimizes the leakage of sodium in the event of primary bellows failure.
4. Eliminates the sodium oxidation problem in the event of primary bellows failure.
5. Provides a vehicle for both primary and secondary bellows leak detection (by indication of nitrogen flow).

Both bellows assemblies have been sized for a design life of 1000 hours at 1200 F and 5 psi with a safety factor of 2, and for 10^8 cycles at .018 inches, peak to peak, at 1200 F with a safety factor of 2.5.

Air cooled heat exchangers have been incorporated into the loader housing and shaft at the linear ball bearing nearest the sodium containment vessel. The purpose of these heat exchangers is to cool that portion of the loader so that conventional bearings and bellofram seals can be utilized. A final computer solved thermal elastic stress analysis of the loader mechanism has been initiated to confirm the design. If the results of this analysis indicate any problems in the critical thermal transition region, these can readily be corrected by modifications of the transition zone diameter and/or length.

The mechanical exciter which was discussed in the previous progress report- is attached to the end of the loader shaft. It consists of four rotating disks to which unbalance weights are applied. The four disks, each with an unbalance weight have been arranged in such a manner that the sum of the forces and moments transmitted to the support structure is always zero which eliminates the need to provide for any external loader mechanism support structure to take the vibratory forces. (This method for introducing synchronous or non-synchronous dynamic loads is in addition to the provisions for adding up to 75 in.lbs unbalance load).

Concurrently with the detail design of the test rig, the design of the test system and the facility to accommodate the rig and system continued. The piping schematic (sodium, nitrogen, pneumatic control, etc) was completed and appears as Fig. 7. Most of the equipment identified on the piping schematic has been released for procurement. In addition work was initiated on the electrical schematics which include heater circuits, motor circuits, instrumentation circuits and system and rig control circuits.

To minimize the amount of on-site sodium equipment (valves, pipes, pump. tanks, etc) fabrication, the facility and piping system has been laid out such that all significant equipment is located immediately adjacent to the Oxygen Control and Indicating (OCI) system or the sodium sump tank. In this way these items can be procured as integrated and checked out assemblies. Specification control drawings

for the OCI System and the sump tank which includes identification and location of all accessory equipment have been prepared and released for procurement bids.

V. SCHEDULE

The detailed Schedule and Pert Plan for the program were provided on the Project Plan issued on April 30, 1968. A summary schedule by task is given on the next page with the actual progress by task indicated by the shaded areas.

As shown on this schedule, the efforts on Tasks I, II and IV-B are on schedule.

The start of Task III was delayed approximately four weeks to mid-October, in order to permit completion of the computerized analysis required for this task and which was prepared under Task II. This analysis was completed, as described in Ref. 4 and is currently being used in Task III.

Tasks IV-A and V are approximately eight weeks behind schedule. The delay in Task IV-A is due to the inability of the probe manufacturer to deliver the required transducers. As noted earlier in this report, an internal transducer development was started in September, to insure their production. This development is being carried out on an urgent priority basis and is progressing rapidly. The delay in Task V was due to a greater need for detailed analysis of the test rig components than had been anticipated, and to the design modification involved in relocating the loader bearing to bring it in close proximity to the test bearing. This additional effort has been completed and detailing of the test rig parts is in progress are noted earlier.

At this time, no delay in the program completion date (March, 1970) is forecast.

SCHEDULE SUMMARY

	F.Y. 1968						F. Y. 1969						P. Y. 1970															
	D	J	F	M	A	M	J	J	A	S	O	N	D	J	F	M	A	M	J	J	A	S	O	N	D	J	F	M
TASK I - Program Management and Reporting																												
TASK II - Fluid Inertia Investigation																												
TASK III - Preparation of Bearing Design Charts																												
TASK IV-A Proximity Instrumentation for High-Temperature Sodium																												
TASK IV-B Pressure Instrumentation for High-Temperature Sodium																												
TASK V - Analysis, Design and Procurement of Test System																												
TASK VI - Test Program and Data Correlation

Fig. 1 Typical Inductive Probe Configuration

Fig. 2 Coil-Winding Fixture

Fig. 3 Some Single-Piece, Slotted Core Samples

4	7	#C21	PIC TAPER PIN .047 DIA., 1/4" / FT. TAPER
1	6	212 B 104	TRANSDUCER CAP
1	5	212 B 103	COIL FORM
1	4	212 B 100	MAGNETIC CORE
1	3	212 B 105	LEAD SPACER
1	2	212 B 101	TRANSDUCER BODY
X	1		ASSEMBLY
QTY.	ITEM	PART NO	DESCRIPTION

Fig. 4 Tentative Design of High-Temperature, Liquid-Metal Proximity Sensor Assembly

Fig. 6 Schematic of Loader Mechanism

Fig 7 Piping Schematic for 12" Diameter, Sodium Lubricated Bearings System

DESIGNED BY	MECHANICAL	DATE	11/10/66
CHECKED BY		DATE	11/10/66
APPROVED BY		DATE	11/10/66
FLUID SYSTEMS SCHEMATIC			
E 212-P-01			

NOMENCLATURE

C Radial clearance, inches

L Inductance, mh

N_{Re} Reynolds Number, ($= \frac{R\omega C}{\nu}$)

Q Ratio of inductive resistance to total AC resistance

R Bearing radius, inches

ν Kinematic viscosity, in²/second

ω Angular speed, radians/second

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