M
MECHANICAL
T
TECHNOLOGY
I
INCORPORATED

RECEIVED BY DTIC APR 28 1969

MASTER

DISCLAIMER

This report was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States Government nor any agency Thereof, nor any of their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States Government or any agency thereof. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency thereof.

DISCLAIMER

Portions of this document may be illegible in electronic image products. Images are produced from the best available original document.

LEGAL NOTICE

This report was prepared as an account of Government sponsored work. Neither the United States, nor the Commission, nor any person acting on behalf of the Commission:

- A. Makes any warranty or representation, expressed or implied, with respect to the accuracy, completeness, or usefulness of the information contained in this report, or that the use of any information, apparatus, method, or process disclosed in this report may not infringe privately owned rights; or
- B. Assumes any liabilities with respect to the use of, or for damages resulting from the use of any information, apparatus, method, or process disclosed in this report.

As used in the above, "person acting on behalf of the Commission" includes any employee or contractor of the Commission, or employee of such contractor, to the extent that such employee or contractor of the Commission, or employee of such contractor prepares, disseminates or provides access to, any information pursuant to his employment or contract with the Commission, or his employment or contract with the Commission, or his employment with such contractor.

Report No. NYO 3930-4
MTI 69TR18

MECHANICAL TECHNOLOGY INCORPORATED
968 Albany-Shaker Road
Latham, New York 12110

LEGAL NOTICE

This report was prepared as an account of Government sponsored work. Neither the United States, nor the Commission, nor any person acting on behalf of the Commission:

A. Makes any warranty or representation, expressed or implied, with respect to the accuracy, completeness, or usefulness of the information contained in this report, or that the use of any information, apparatus, method, or process disclosed in this report may not infringe privately owned rights; or

B. Assumes any liabilities with respect to the use of, or for damages resulting from the use of any information, apparatus, method, or process disclosed in this report.

As used in the above, "person acting on behalf of the Commission" includes any employee or contractor of the Commission, or employee of such contractor, to the extent that such employee or contractor of the Commission, or employee of such contractor prepares, disseminates, or provides access to, any information pursuant to his employment or contract with the Commission, or his employment with such contractor.

PROGRESS REPORT

For the Period

November 1, 1968 - January 31, 1969

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

AEC CONTRACT AT (30-1)-3930

RECEIVED
JAN 31 1969
24

NYO 3930-4
MTI 69TR18

MECHANICAL TECHNOLOGY INCORPORATED
PROGRESS REPORT

on

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

For the Period

November 1, 1968 - January 31, 1969

E. B. Arwas
L. Hoogenboom
C. Kissinger
R. Newell
F. K. Orcutt
W. D. Waldron

Contract AT(30-1)-3930-4
U. S. Atomic Energy Commission

March, 1969

TABLE OF CONTENTS

I.	INTRODUCTION	1
II.	SUMMARY OF PROGRESS	2
III.	DISCUSSION	4
	TASK I Program Management and Reporting	4
	TASK II Fluid Inertia Investigation	4
	TASK III Preparation of Bearing Design Charts	5
	TASK IV Development of Proximity and Film Pressure Instrumentation for High Temperature, Liquid Sodium Environment	7
	TASK V Test Rig Design and Fabrication	15
IV.	SCHEDULE	21
	REFERENCES	23

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 LMFBR 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 of the self-acting type.
- 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 covered by this Quarterly Progress Report, November 1968 through January 31, 1969, the principal effort was in the following areas:

1. Review and issue of the Topical Report, NYO-3930-2, covering completed work of Task II.
2. Bearing design optimization and preparation of computer programs for generating production data for the self-acting bearings design handbook being prepared under Task III.
3. Continued development of the proximity (film thickness) probe for operation in high temperature liquid metal ambient under Task IVA.
4. Continued development of the static (bearing mounted) and dynamic (shaft-mounted) film pressure probes for operation in high temperature, liquid metal ambient under Task IVB.
5. Continued analysis, design and procurement of the sodium lubricated test rig, under Task V.

II. 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. Following completion of the work under Task II of the program as noted in the last quarterly report, the Topical Report in this Task (NYO 3930-2), was submitted for approval in December 1968 and subsequently issued in January 1969.
2. Under Task III of the program, the optimization of the shrouded step bearing configuration was conducted, leading to the selection of the principal design parameters including step depth and sealing land geometries for the desired ranges of clearance ratio and Reynolds Numbers (up to 0.003 $\frac{1}{\text{in}}$ and up to 60,000 respectively). This optimization was made for maximum radial stiffness of each bearing pad for motions normal to the centerline of the pad.
3. Also under Task III, the computer program derived under Task II to calculate the performance of plain and stepped pads including turbulence and fluid inertia effects, was extended for production runs by addition of appropriate subroutines. With these subroutines, large amounts of production data can be obtained, cycling over specified ranges of eccentricity ratio, attitude angle and pad geometry. The assembly program that uses this pad data to provide performance data on complete bearings was also written.
4. Under Task IVA, the in-house development of proximity (film thickness) measuring transducers suitable for operation in liquid metal ambient at elevated temperatures, which was started in September 1968, was continued. Furnace tests of the completed transducers were made in January 1969 over the temperature range from room temperature to 1000 F, and showed stable and repeatable probe characteristics. Ageing tests of the core-coil combination were also made in cyclic temperature tests over a temperature range up to 1200 F. These showed inductance changes of less than 1 percent per cycle. Extended duration tests at elevated temperatures were also made showing no measurable inductance changes at 1000 F and approximately a 1 percent inductance change after 24 hours at 1200 F.
5. Under Task IVB, the cyclic pressure tests of the shaft mounted probe version were completed. These showed no damage to either the diaphragm or its coating. They did, however, indicate unacceptably high changes in sensitivity due to compacting of the dielectric material between the diaphragm and the motion sensing capacitance element. Accordingly, the design of the shaft mounted transducers was modified to eliminate this problem. Also, in the liquid metal bearing tests, the shaft mounted transducers will be backed up by bearing mounted, static pressure measuring transducers which are not subject to cyclic pressure variations. In addition to measuring the pressures

at two locations on the bearing, the bearing mounted transducers will also serve to provide a continuous calibration of the shaft mounted transducers that continuously scan the pressure probe around the bearing film.

6. Under Task V, the analysis and design of the test rig for conducting the tests of 12" diameter sodium lubricated bearings was continued. All analyses of the test rig elements were completed. Detailing and checking of the upper shaft support structure, the sodium containment vessel and the loader mechanism were completed and released for procurement. Detailing of the test bearing support assembly, the loader bearing assembly and the main load support assembly were also completed. Checking of these manufacturing detail drawings and release for procurement will be made during the current reporting period. Mechanical design of the rotor, cover gas seal and thermal transition piece was continued and these parts also will be completed and released for procurement during the current reporting period. The loop design and specifications were earlier completed and released for fabrication.

III. DISCUSSION OF PROGRAM PROGRESS

The progress during the period covered by this quarterly report 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 the quarterly report for the period August 1, 1968 through October 31, 1968 was prepared and issued in Ref.3 (Report No. NYO 3930-3). A review meeting with NYOO was held in December 1968.

Task II - Fluid Inertia Investigation

As noted in the last quarterly report, this task was successfully completed and reported in Topical Report NYO 3930-2 (Ref. 2) which was submitted for review in December 1968 and subsequently issued in January 1969. The first part of this effort had comprised the development of the lubrication analysis that accounts for turbulence and convective fluid inertia forces, as well as preparation of the computer program to obtain numerical solutions for plain or stepped pad journal bearings. Comparisons of the calculated pressures generated in the turbulent-fluid inertial flow regime were made with measured data obtained with a 12" diameter journal bearing having two different shrouded step pad configurations and lubricated with a low viscosity silicone fluid. A range of speeds under both concentric and eccentric operating conditions was run experimentally and the pressure profiles measured for the comparison with the analytical predictions. The comparisons showed consistent agreement over the full range of test conditions, with the discrepancies being well under 10 percent. Also included in the work under this task and reported in Topical Report NYO 3930-2, was an investigation to establish a means of prescribing the dynamic head of the fluid which is recoverable as static pressure at the pad inlet. Based on this investigation, a representative, empirical recovery factor was set for use in bearing design and the error limits associated with it were defined.

TASK III - Preparation of Bearing Design Charts

This task comprises the generation of design-data for three self-acting bearing configurations selected for potential use in large sodium lubricated applications, such as in sodium circulation pumps for LMFBR service. This design data is to be generated using the analysis developed in Task II.

During the first part of the current reporting period, the computer program for plain and stepped journal bearings derived under Task II was extended by addition of subroutines for production runs. This version of the program accepts non-dimensional input and computes normalized values of the load components, inlet and outlet flow and friction torque for individual bearing pads. The data is punched directly on cards when production runs are made. This output is then assembled in a separate assembly program for any arrangement of the pads into composite bearings.

It is planned that the shrouded step bearing production runs will be made for optimized pocket geometries. The basis of this optimization will be to achieve maximum stiffness of a single pad for motion normal to the center of the pad from the concentric position. An investigation to establish such geometries for a range of values of Reynolds number, clearance ratio and overall pad geometry has been undertaken and is about 80 percent completed. This investigation shows that, where inertia effects are not negligible, to achieve optimum values of stiffness, pocket width and length values should be only fractionally less than the pad width and length respectively.

From consideration of manufacture, effective life and also of dynamic stiffness characteristics, however, such a design would be unsatisfactory, and it is expected that the final pocket width and length decided upon will be between 60 and 70 percent of the pad width and length respectively.

The optimum pocket depth has been determined to be close to, or slightly less than twice the land clearance. This is shown in Figure 1, where, for a Reynolds number of 2×10^4 , stiffness and load are plotted against normalized pocket depth, δ/c . This curve for stiffness peaks close to $\delta/c = 2$. It is of interest

to compare the curve of load against δ/c , for which the peak is reached for δ/c close to 5.

The approach for producing useful design data for multi-pad journal bearings will be to use an economical assembly computer program separate from the film analysis program. This assembly program will use the single pad data generated by the film analysis program, and will produce eccentricity loci and plots of eccentricity, inlet flow, side leakage and friction torque against load. The advantages of the assembly program approach is that the same basic data can be used separately by this program for any conditions of load direction relative to the pads, and any preloading of the pads.

Naturally, an essential feature of such a program is an effective interpolation procedure. As a basis for testing contending interpolation procedures, a sample production run for a near optimum geometry was generated by the film analysis program and punched on cards. Some numerical experiments have been performed on this data. The conclusion from these experiments is that third order Lagrangian interpolation is very satisfactory for interpolation with respect to the two required location parameters of eccentricity ratio and pad inlet angle.

The adequacy of this interpolation procedure with respect to pad inlet angle is demonstrated in Figure 2. Here data points are given every thirty degrees, and interpolated values are given every ten degrees. A very smooth curve results, and equally good results are obtained for interpolation on eccentricity ratio.

During the next period, the detailed pad data for shrouded step bearings will be obtained and the preparation of design charts based on this data will be conducted.

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

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 measurement 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*	-----	10×10^{-6} in/ $^{\circ}$ F max.
readout equipment temperature sensitivity	-----	10×10^{-6} in/hr. max.
DC drift*	-----	10×10^{-6} in/volt max.
readout equipment line voltage sensitivity	-----	10^{-6} in/volt max.
noise	-----	30×10^{-6} in equiv.

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. 4) 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

* when wired in "push-pull"

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. 4. It is necessary, however, that the target surface be ferromagnetic for effective operation of the inductive probe.

As noted in the last quarterly progress report (Ref. 3), however, the commercial probe manufacturer encountered serious quality control problems and the test probes procured showed irreversible changes in coil resistance after temperature cycles up to 800 F and failed because of open circuits of the coils after several temperature cycles to 900 or 1000 F. Moreover, the changes in coil inductance with temperature were very large (characteristically of the order of 2 to 1 in the temperature range from 70 to 900F), as compared with the changes in inductance as a function of gap, characteristically, about 10 percent in going from infinite to closed gap.

Because of these difficulties, and considering the importance of a reliable and effective proximity sensing system to the conduct of liquid-metal bearing tests, an internal development of a high temperature inductive probe was undertaken as an independent effort parallel to the evaluation of commercial probes. This development was initiated in September, 1968.

The in-house development of the high temperature, inductive proximity probe, that was initiated during the previous quarter, was continued on an accelerated basis during this quarter. The final prototype design was firmed up early in November. Fabrication of coil, core and body parts for four complete transducers was initiated. Testing of the first two ceramic-insulated, gold-wire coils as wound on the specially built coil winding apparatus, was continued. They were subjected to repeated thermal cycles from room temperature to 1200 F while monitoring their D.C. resistance, inductance and A.C. resistance. The properties of the coils have been perfectly reproducible from cycle to cycle and there have been no observed anomalies or evidence of deterioration of either the gold-wire coil itself, the alumel extension leads or the gold to alumel connections. The inductance of the coil shows no measurable change from room temperature to 1000 F.

Core Fabrication

Two pot cores having .050" wall thickness and one radial slot, as described in the previous quarterly report, were fabricated from Hiperco 27. The Hiperco 27 was chosen as the core material because of its high curie point (approximately 1700 F) and its good magnetic properties. Its machining properties, although somewhat poor, proved to be tolerable.

Coil-Core Combination

One of the gold coils was mated with one of the Hiperco 27 cores. The assembly was temperature cycled, up to 1200 F, several times. Figure 3 shows the inductance of the coil-core combination as a function of temperature. It can be seen that the magnetic properties of the core contribute significantly to the overall results. For example, the inductance is no longer constant over the temperature range, as it was for the coil alone. Over the full temperature range the inductance of the coil-core combination changes by about 8 percent, this change occurring primarily in the temperature range above 900 F. It can also be seen from Fig. 3 that the inductance stability is improved by temperature cycling and seems to settle down to a minimum after approximately 6 cycles. There is at present very little specific information available on the effect of temperature on the permeability of the core material, or on the heat treatment that would optimize stability of the permeability at elevated temperatures. Accordingly, it was decided to adopt a somewhat longer heat treatment cycle than normally deemed necessary by the material vendor. Whereas the manufacturer simply specifies heating the Hiperco to 1500-1550 F in an inert atmosphere followed by oven cooling to room temperature, we elected to heat gradually to 1500 F, over a period of 2 hours, to hold at 1500 F for 2 hours, and to cool gradually, over 2 hours, to room temperature. In the interest of consistency, this heat treatment schedule was adopted for all the Hiperco parts. Some consideration was given to gold plating the Hiperco core to protect it from oxidation, but this was not done at this time since there appeared to be little evidence of accumulate oxidation. Protective plating for the core may, however, become necessary if it is decided to extend the continuous operating temperature of the transducer to 1200 F rather than the present 1000 F upper limit. It may also become necessary to investigate other core materials.

Concurrently with the oven testing of the first coil-core combination, two more coils were wound and tested. The inductance of the coils alone is approximately 1.7 mh and remains essentially constant, to within an experimental accuracy of approximately $\pm .01$ mh, throughout the range from room temperature to 1000 F. The resistance is approximately 10 ohms, at room temperature and increases linearly, to approximately 40 ohms at 1200 F. The resistance was repeatable to within an experimental accuracy of $\pm .1$ ohms. These tests were further evidence of the extremely stable and repeatable characteristics of the coils themselves.

Additional coil-core combinations were temperature cycled to 1200 F in order to accelerate the possible ageing effects that may occur in the room temperature to 1000 F range. These tests showed an inductance change of less than 1 percent per cycle. Tests of the effects of extended time periods at elevated temperature were also conducted. It was found that 15 hours at 1000 F gave no measurable change in inductance and 25 hours at 1200 F resulted in approximately a 1 percent increase in inductance.

A plot of the inductance versus gap for the coil-core combination at room temperature is shown in Fig. 4. It should be noted that the non-linearity for gaps smaller than approximately .020" is not significant since the minimum operating gap between the end of the core and the target will be at least this large. Also, we plan to use two probes in a push-pull configuration and this will greatly improve linearity of output.

After receiving the transducer parts, a trial assembly of the encapsulated probe, using a copper wire coil, was made. This resulted in several minor modifications to the design to insure proper venting of the area beneath the diaphragm and to allow more space within the ceramic coil form for positioning of the coil-to-lead wire joints. Several sample T.I.G. welds were made on various cap to body flange designs to determine the best configuration for the weld joint. The weld samples were sectioned for examination and were judged to be of excellent quality. A configuration having slightly tapered flanges were selected and incorporated into the final transducer design.

Evaluation tests were conducted on the modified Daytronic Model 300 D, 3 KHz carrier amplifier system being considered for use as the readout equipment. The output characteristics for a linear change in inductance are shown in Fig. 5. The results indicate that the modified Daytronic readout has a satisfactory linear response to linear changes in inductance. Curves B and C of Fig. 5 were taken with 20 and 40 ohms additional resistance inserted in series with the active and the compensating transducers. Curve B simulates the effect of the resistance change caused by approximately a 1000 F temperature rise in both transducers. This causes approximately a 6 percent decrease in sensitivity and the linearity is unaffected.

The first two prototype transducers were assembled and oven tested for several cycles before welding to the diaphragm cap to the body. An automatic recording system was set up to monitor the output from the modified Daytronic 300 D, 3 KHz carrier amplifier used to drive the transducers during extended-time temperature testing. Figure 6 shows the data obtained with these transducers on their third temperature cycle to 1000 F, without the diaphragm in place. The transducers were connected as a half-bridge, push-pull arrangement and the remaining half-bridge was made up of variable inductors and resistances. This arrangement is similar to that which will be used for the actual probe system. This results in cancellation of all like effects, such as temperature induced resistance changes in the coils and temperature induced permeability changes in the cores, and gives an output proportional to the difference in the tracking of the two transducers.

The upper curve in Fig. 6 shows that the two transducers track each other closely over the full temperature cycle. The tracking error is the deviation of this curve from a horizontal line drawn through the 100 percent scale point on the ordinate of the plot. The test data also showed that the displacement sensitivity of the transducers varies slightly with temperature. The vertical distance between the upper and lower curves of Fig. 6 is a measure of the displacement sensitivity of one transducer over the temperature range. Hiperco 27 was used as the target material and it should be pointed out that the displacement sensitivity includes the temperature induced permeability changes of the target material as well as the transducer itself. It should be noted that this data is to be considered as preliminary. Additional testing with the completed transducers and using an improved calibration fixture will be done in the next quarter.

The remaining two transducers were assembled and given a temperature cycle to 1100 F to cure the ceramic cement. Their characteristics were checked before and after this temperature cycle. All four transducers were then considered to be ready for welding of the diaphragm caps to the body.

A special fixture was built to hold the diaphragm cap in proper contact with the probe body during welding of the adjoining flanges. All four transducers were welded without difficulty. The transducers characteristics were found to be unchanged by the addition of the caps and the welding process.

Plans for the next quarter include additional and more complete temperature testing of the four completed transducers. To accomplish this we are designing a new, high temperature, horizontally mounted, calibration fixture that will be used for testing of the production probes as well as the prototype probes.

B. Sealed Diaphragm Capacitance Type 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 Pressure 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 the 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.

During the earlier work, the manufacturing process for both the bearing mounted and the shaft mounted probes were developed through manufacture of prototype probes and calculations were made over the temperature range up to 1100 F indicating satisfactory operation. Also, as reported in the last progress report, fatigue tests of the shaft mounted probe were made to determine the integrity and possible failure modes of this transducer which, since it is mounted in the shaft, is subjected to cyclic variations in film pressure.

During October, approximately 10,000 cycles were accumulated and these showed a small change in sensitivity. During November, the cyclic testing was continued until approximately 45,000 pressure cycles were accumulated. The sensitivity change continued until about 20,000 cycles had been completed whereafter the calibration was unchanged. The total sensitivity change, about 10 mV/PSI was unacceptably high.

Calibration tests made in the transducer after the fatigue tests showed almost identical sensitivity as that prior to the tests, the sensitivity change occurring in the range from 200 to 700 psi. There was no cracking or deformation of the diaphragm and the degradation of coating adherence, if any, had not

influenced the performance of the transducer. Moreover, microscopic and surface measurement evaluation of the flame sprayed coating in the exposed surface of the diaphragm revealed no coating distress such as cracking or pits.

The change in sensitivity between 200 and 700 psi was attributed to compacting of the ceramic powder dielectric between the diaphragm and the motion sensing capacitance element as a result of repeated pressure cycles. Accordingly, the design of the shaft mounted transducer was modified to eliminate the compacted dielectric between the diaphragm and the sensing element.

In the sodium lubricated bearing tests under Task VI of the program, two bearing mounted pressure transducers will be used. These are not subjected to cyclic pressures and they have been designed and tested satisfactorily for the liquid metal application. In addition, however, since full scan of the pressure profile around the bearing remains desirable, shaft mounted transducers of the modified design which eliminates the compacted dielectric material will be used. Measured errors due to sensitivity variations, if any, that are induced by repeated cyclic pressures will be minimized since the bearing mounted, static pressure transducers will promote a continuous calibration of the shaft-mounted dynamic transducers.

TASK V - Test Rig 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 tests 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 -----	Flood hydrodynamic
Bearing operation -----	Vertical and submered
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 has been described in detail in the previous quarterly report consists of a vertical shaft (approximately 9 feet long) with the upper end supported by a large rolling element bearing and the lower end supported by the test bearing. Both the test bearing and the hydrodynamic tilting pad type loader bearing are submerged in high temperature sodium which is contained within a 42 inch diameter vessel which forms the lower half of the test rig. The containment vessel has a free sodium surface approximately 3 inches

above the loader bearing (which is located above the test bearing) with the remaining space filled with nitrogen cover gas. Load is applied to the loader bearing by means of a loader mechanism which penetrates the vessel wall below the sodium level as was described in detail in the previous quarterly report.

During the previous reporting period the concept arrangement was frozen and all major components were located and sized. The major activities during this period have been directed toward detailing of the test rig components and performing those design analyses which are necessary assure the integrity of the components as finalized. As these final design analyses are completed and as machining and assembly operations become clear, it is necessary to make minor detail modifications to the test rig. The more significant ones that have been made are discussed later in this section.

Detail drawings of the following major subassemblies have been completed (including checking) and have been released by procurement:

- a) Upper shaft support structure which includes the rolling element bearing and housing, shaft alignment mechanism, drive coupling, gear box, and slip ring mount and drive assembly.
- b) Sodium containment vessel which forms the lower portion of the test rig.
- c) Loader mechanism including the link assemblies which attach to the loader bearing, the mechanical oscillators which permit imposing dynamic loads to the test shaft, and the primary and secondary closure seals.

The detailing and final analyses of these components caused only one minor revision in concept from that described in the previous report. That revision included the method of taking the thermal gradient and transmitting the load from the bearing support structure to the loader mechanism frame. The cylindrical member which separated the loader piston cylinder from the bearing support structure has been replaced by three columns in the area outside of the containment vessel as is shown in the assembly drawing, Fig. 7. This revision was made necessary because the cylindrical section between the vessel

and the loader rod support bearing has to take both the thermal gradient (1200 F to 150 F) and a high compressive load (12,900 lb) which resulted in excessive stresses. The columns, which replace the cylinder, are free to slide radially at the cold end which allows accommodation of the differential thermal growth and thus eliminates the thermal bending stresses in this highly loaded component.

Detail drawings were also completed (but not checked) for the following major subassemblies:

- a) Test bearing Support Assembly which includes the bearing housing (which accepts the test bearing insert), the alignment flexures, and the main support ring.
- b) The loader bearing assembly including housing, tilting pads, and pivots.
- c) The main load support assembly which includes the vessel cover and flange, and the internal load carrying cylinder.

Design analysis was completed on the test bearing self alignment flexure and the following summarizes the significant features:

Radial stiffness: $5 \times 10^6 - 7.5 \times 10^6$ lb/in (depending upon load direction)

Angular stiffness: $\approx 0 - 3.9 \times 10^5$ in-lb/rad

Maximum spoke stress: 18,000 PSI

The minimum radial stiffness of the support is more than an order of magnitude larger than the maximum anticipated test bearing stiffness that will be realized during dynamic testing which will result in small support effects on the dynamic characteristics of the bearing. The angular stiffness of the support, which is a function of the spoke assembly preload, has sufficient flexibility to assure that the deviation in film thickness from one end of the test bearing to the other will never exceed 10 percent of the minimum film thickness. The maximum spoke stress of 18,000 PSI under a 9000 lb. assembly preload and a 10,000 applied bearing load (the total of which results in an angular stiffness of 3.9×10^5 in-lb/rad) is well below the 1200 F, 2 percent yield strength of 148,000 PSI for the Inconel 718 spoke material.

The loader bearing has been designed to impose a 12,600 lb load on the test rotor (resulting in a 10,000 lb applied test bearing load) with an eccentricity ratio less than 0.85. The operating preload has been set at 0.4 such that, under conditions of elevated temperature (1200 F) and the 12,600 load, the unloaded pads will never lose all their preload, which eliminates the possibility of pad flutter. (Since the pivots, which are made of Stellite 6, have a lower thermal expansion coefficient than the 316 housing and pads the assembly preload is approximately 0.5). The pivots have been designed to experience a Hertz stress of less than 105,000 PSI which is sufficiently below the 1200 F compressive yield strength of 150,000 PSI for Stellite 6 to eliminate the possibility of brinelling. The maximum deformation of the loader bearing housing under the maximum design load is only .001 inches, as compared to the pivot clearance of .012 inches and should have negligible influence on bearing performance.

The main load support assembly has been modified in concept slightly since the last report as a result of the detail problem of instrument and heater penetrations that must pass through the vessel cover and be hermetically sealed. There are a total of 62 functional connections that pass through the vessel cover into the high temperature sodium environment. These include 16 bearing proximity probes, 2 bearing pressure probes, 8 - 2KW heaters, 1 liquid level detector, 32 thermocouples, and one cover gas line. Bringing all of these connections through the top was complicated not only by the hermetic seal requirement but also by the limited space on the cover because of the several bolts, cover heaters, and cooling passages (the latter is discussed later) that occupy cover space. To accomplish these hermetically sealed penetrations the load support assembly was made of two pieces bolted together just under the flange. (This assembly was formally conceived as a single weldment). A manifold has been incorporated on the underside of the cover inside the flange area. The purpose of this manifold is to accept the proximity probe leadwires which are sheathed in 1/4" diam. SS tubing. The probe tube sheaths are connected to the manifold by means of swagelok fittings and the lead wires are brought through the manifold to four individual headers where they terminate in standard hermetic electrical connectors. The manifold is purged with nitrogen at a pressure slightly higher than the cover gas pressure so that any leakage through the swagelok fittings will be into the vessel.

The remainder of the connections are through pipe bulkheads which pass through the manifold but do not communicate with it. The thermocouples are taken through 2 standpipes and terminate in standard Conax T/C fittings (16 each). All other connections are taken through individual bulkhead standpipes. Each is hermetically sealed by a seal weld below the manifold which is accomplished before attaching the lower cylindrical member. The heaters are inserted into pipe wells which terminate near the bottom of the vessel and are plug welded at the bottom to complete the hermetic seal. The hermetic seal of the other penetrations are completed by standard low temperature fittings or connectors at the top of the standpipes which are cool and made possible by their long length. A top view of the cover which shows these penetrations and standpipes is shown in Fig. 8.

Work continued on the mechanical design of the shaft with concentration on the details of the cover gas seal, the thermal transition zone, and the two piece joint. The major problems associated with the shaft arrangement described in the previous quarterly report were the shaft thermal stress at the high temperature shaft joint, the thermal distortions in the cover gas seal runner, the excessive nitrogen buffer flow through hydrostatic cover gas seal, and assembly difficulties. These problems were circumvented by moving the cover gas seal and the shaft joint out of the high temperature region (within the vessel) to an area higher on the shaft where the temperature is near room ambient.

This change permitted increasing the thermal transition length thus improving the thermal stress situation and moved the seal runner from the position of high thermal gradient thus eliminating most of the runner thermal distortion. Also, the move to the low temperature region permitted the use of a commercial face type seal instead of a special hydrostatic type seal with a significant reduction in seal gas flow requirements. Furthermore, the seal and removable runner are now accessible for maintenance without complete test rig disassembly.

Thermal designs of five integral test rig coolers were completed. Two of these; one located on the loader mechanism and the other located within the center section of the shaft are used during normal rig operation. The one on the loader is required to cool the area where the Bellofram seals and linear ball bearings are incorporated and the one on the shaft is used to cool the internal pressure

instrumentation detectors and seal runner.

The remaining coolers are located on the containment vessel OD, the containment vessel cover and within the shaft below the sodium level. The purpose of these coolers is to speed the test rig cool down after the sodium is dumped from the rig. It is planned to cool the rig from operating temperature to about 500 F by circulating Na through the OCI systems coolers and then dumping the Na at about 500 F. Cooling from 500 F to room temperature will be accomplished by using the forementioned air coolers. These coolers will cool most of the rig components to near room temperature in about 48 hours.

During the next reporting period, all detailing of the test rig components will be completed and procurement bids obtained. As noted in the previous report, the loop design and specifications have been completed and procurement bids have been obtained on these elements.

IV. SCHEDULE

A summary schedule chart for the various Program Tasks is given on the next page. As earlier noted, Task II has been completed and reported. The start of Task III was delayed to mid-October to permit completion of the computerized analysis required for this task and which was prepared under Task II. This task will be completed and reported in a Topical Report in May, 1969.

Task IVA, which comprises the development of the proximity (film-thickness) measurement transducer for operation in high temperature liquid sodium ambient, was enlarged to include the in-house development of the transducer as noted in the previous quarterly report (Ref. 3) and in Section III of this report. This task is now scheduled for completion and reporting in a Topical Report in June 1969. The development effort for its companion TASK IVA, which comprises the development of the film pressure measuring transducer for operation in high temperature sodium environment is about on schedule and will be reported in the same Topical Report as that for Task IVA.

As noted in the previous quarterly report, TASK V which comprises the analysis, design and procurement of the high temperature, liquid sodium bearings test rig was re-scheduled for completion in June 1969, instead of April 1969. This two month slip was necessitated by the need for more detailed analysis and design effort than had been anticipated and by the design modification which involved relocating the loader bearing to bring it close to the test bearing. This additional effort has been completed and the June 30 completion date for this task is being adhered to.

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

SCHEDULE SUMMARY

	F.Y. 1968						F.Y. 1969												F.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 - 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																														

== Program Plan Schedule

--- Forecasted Schedule Extension

REFERENCES

1. Progress Report for the Period February 1, 1968 - July 31, 1968. "Liquid Metal Bearings Technology for Large High Temperature Sodium Rotating Machinery". AEC Contract AT(30-1)-3930. Report No. NYO 3930-1.
2. First Topical Report, "Liquid Metal Bearings Technology for Large, High Temperature, Sodium Rotating Machinery". Prepared for U.S. Atomic Energy Commission. AEC Contract AT(30-1)-3930. Report No. NYO 3930-2.
3. 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. Report No. NYO 3930-3.
4. Final Report, "Design of Liquid Metal Lubricated Bearings". AEC Contract AT(30-1)-3363. Report No. NYO 3363-7 (Vol. 2).

Fig. 1 Normalized Stiffness and Load as a Function of
 Pocket Depth Ratio $N_{\text{Re}} = 2 \times 10^4 \cdot C/R = 2 \times 10^{-3}$

Fig. 2 Interpolation Test

$N_{Re} = 2 \times 10^4$	$L/D = 0.7$	$L_p/L = 0.7$	$\delta/C = 1.75$
$C/R = 2 \times 10^{-3}$	$\beta = 80^\circ$	$B_p/B = 0.6$	$\epsilon = 0.5$

Fig. 3 Inductance Change with Temperature, Coil-Core Assembly No. 1. Data taken with no target (Infinite Gap), showing results for repeated temperature cycles.

Fig. 4 Room Temperature Calibration of Proximity Probe No. 1

Fig. 5 Carrier Amplifier System Calibration

"Daytronic, 300D" Output Meter Reading, % of Full Scale

Figure 6 TRACKING OF #1 & #3 TRANSDUCERS BEFORE INSTALLATION & WELDING OF THE CAPS.

SEPARATE PARTS LIST ISSUED	
MECHANICAL	INCORPORATED
LOADER MECHANISM	ASSEMBLY
NO. BEARING TEST RIG	
DATE	J 212 J156
BY	
CHECKED	
APPROVED	

Fig. 7 Loader Mechanism

Fig. 8 Na Containment Vessel Cover Plate

DISTRIBUTION LIST FOR QUARTERLY PROGRESS REPORTS GENERATED UNDER
CONTRACT AT(30-1)-3930

U.S. Atomic Energy Commission
New York Operations Office
376 Hudson Street
New York, N.Y. 10014

Att: Leonard Wasser, Development Contracts Branch II
Joseph J. Dissler, Chief, Technical Branch

U.S. Atomic Energy Commission
Division of Reactor Development and Technology
Washington, D.C. 20545

Att: Milton Shaw, Director
Chief, Special Technology Branch (3 copies)
Chief, Liquid Metal Projects Branch
Chief, Components Branch
Chief, Instrumentation and Control Branch
Chief, Management Services Branch
Asst. Director for Plant Engineering
Asst. Director for Reactor Technology
Asst. Director for Project Management
Asst. Director for Reactor Engineering
W.R. Kornack, Liquid Metal Project Branch
R.F. Sweek, LMFBR Program Manager
J.J. Morabito, FFTF Program Manager
Chief, Fuel Handling Branch

U.S. Atomic Energy Commission
Division of Technical Information Extension
P.O. Box 62
Oak Ridge, Tennessee 37830
(2)copies with form AEC 426 (block I, No.3 will apply)

U.S. Atomic Energy Commission
RDT Site Representative
Canoga Park Area Office
P.O. Box 591
Canoga Park, California 91305

Argonne National Laboratory
9700 South Cass Avenue
Argonne, Illinois 60439
Att: Director, LMFBR Program Office

Brookhaven National Laboratory
Upton, N. Y. 11973
Att: Dr. O.E. Dwyer

Distribution List (Cont'd).

Pacific Northwest Laboratory
P. O. Box 999
Richland, Washington 99352
Att: FFTF Project Manager (2)

Liquid Metals Engrg. Center
P.O. Box 1449
Canoga Park, California 91304
Att: R.W. Dickinson, Director (2)

Byron Jackson Pump Division
Borg Warner Corporation
P.O. Box 2017 Terminal Annex
Los Angeles, California 90054

Westinghouse Electric Corporation
Advanced Reactors Division
P.O. Box 158
Madison, Pennsylvania 15663
Att: E.F. Beckett
Program Manager
LMFBR Programs

Westinghouse Electric Corporation
Atomic Equipment Division
Cheswick Avenue
Cheswick, Pennsylvania 15024
Att: C.E. Anthony

Chief of Naval Research
Department of the Navy
Washington, D.C. 20360
Att: Code 438

NASA, Lewis Research Center
21000 Brookpark Road
Cleveland, Ohio 44135
Att: Library MS 60-3

Science and Technology Division
Library of Congress
Washington, D.C. 20540

Engineering Societies Library
345 East 47th Street
New York, N.Y. 10017

Argonne National Laboratory
9700 South Cass Avenue
Argonne, Illinois 60439
Attn: Director, Reactor Engrg. Division (3)