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PROGRESS REPORT

For the Period

November 1, 1969 - April 31, 1970

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
November 1, 1969 - April 30, 1970

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Contract AT(30-1)-3930
U.S. Atomic Energy Commission
September, 1970

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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 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 sodium cooled, fast breeder reactors. To this end, the program was 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, sodium lubricated bearings.
- b. Accurate and reliable instrumentation with which proximity and film pressure measurements may be made directly in the high temperature sodium environment.

Specifically, the authorized program tasks are as follows:

- Task I - Program management.
- Task II - Development of the turbulent-inertial lubrication theory and experimental verification of the theory using a low viscosity, non-corrosive lubricant.
- Task III A - Preparation of a design data handbook for large, hydrodynamic bearings.
- Task III B - Preparation of a design data handbook for large, hydrostatic bearings.
- Task IV A - Development and evaluation of film thickness probes for liquid sodium bearing tests, including the amplification and read-out system.
- Task IV B - Development of film-pressure probes for liquid sodium bearing tests.
- Task IV C - Long duration (1000 hours) test of the rugged, extended

range (0.10 inches) proximity probes and the film pressure probes in liquid sodium at high temperatures up to 1200 F.

- Task V - Design and procurement of a 12 inch diameter, hydrodynamic sodium lubricated bearing test rig.
- Task VI A - Conduct evaluation tests of the 12 inch diameter, hydrodynamic bearings in liquid sodium at elevated temperatures up to 1200 F and compare the bearing performance characteristics with the design handbook data.
- Tasks VI B & C - Screening, selection, and evaluation of surface materials for high temperature sodium bearings.

During the period covered by this progress report, the principal effort was in the following areas:

1. Completion of the analysis and preparation of the computer program for large, externally-pressurized bearings (Task III B).
2. Conduct of the long term tests of the proximity and film pressure probes in sodium at 1150 F (Task IV C).
3. Installation of the large tapered metal bearings test rig (Task V).
4. Screening, corrosion tests of the selected bearing surface materials in 1150 F sodium (Task VI B).

In addition, under a separate task completed in April 1970, a design review of the bearings of the In-Vessel Fuel Handling Machine of the FFTF was conducted.

The progress by task during this period is described in the next section.

II. DISCUSSION OF PROGRAM PROGRESS

Following is a description by task of the work conducted during the current period.

Task I - Program Management and Reporting

Letter type progress reports were issued, as well as a formal report describing the work conducted during the period February 1, 1969 through October 31, 1969. A review meeting was held at AEC Headquarters on December 4, 1969, attended by representatives of NYO, RDT, ANL, and LMEC for the purpose of presenting the program results to-date and the future course of the program. At this meeting presentations were made - by the MTI personnel that are conducting the various tasks of the program - on:

1. The turbulent lubrication theory
2. The design handbook on large, hydrodynamic bearings
3. The planned effort for the handbook on large, externally pressurized bearings
4. The sodium bearing surface materials survey
5. The large sodium bearings test rig design
6. The film thickness and film pressure probes that have been developed for use in high temperature sodium bearings

A report on this technical review meeting that summarized the presentations was issued to all participants on December 19, 1969.

IVHM Design Review Task

Also during the current reporting period, MTI was requested by RDT to conduct a design review of the lift nut and bearings of the FFTF's In-Vessel Fuel Handling Machine (FFTF). This design review was handled as a task of the current program and was completed during the month of April 1970, with the final report issued on April 27, 1970.

Task II - Fluid Inertia Investigation

This was completed and described in Topical Report NYO 3930-2, issued in January 1969.

Task III A - Design Handbook - Self-Acting Bearings

This was completed and the design handbook issued as Topical Report NYO 3930-5, issued in November 1969. This handbook provided fluid-film design data for the following three, self-acting (hydrodynamic) journal bearing types:

- Four-axial groove
- Shrouded step
- Four-pad, tilting pad

The parameter ranges covered by the design data were:

- Length to diameter ratio: 0.5, 0.7, and 1.0
- Clearance ratio: non-inertial (i.e. small values of C/R),
0.002 and 0.004
- Reynolds number: laminar, 5000, 10,000, and 60,000
- Eccentricity ratio: 0 to 0.95

The data was presented in tabular form showing load capacity, attitude angle, friction and flow as functions of the above parameters. Plots of this data were also provided in many cases, to show performance trends and to facilitate design calculations. Numerical examples were worked out for each bearing type. This steady state data was generated for the turbulent flow conditions prevailing in large, sodium lubricated bearings, including the Bernoulli and convective inertia effects. The computations were based on the analysis developed earlier and described in Topical Report NYO 3930-2. Dynamic data, i.e. the fluid film and stiffness and damping coefficients were also presented for each of the three bearings over the above noted ranges of design parameters, but simulating the convective inertia effects. The effect of the time dependent inertia forces on fluid film stiffness and damping was investigated and described in an Appendix of the report for a simple bearing model that neglected side leakage flow.

Task III B - Design Handbook - Externally-Pressurized Bearings

The objective of this task is to prepare a design handbook for large, externally pressurized bearings operating under highly turbulent flow conditions. The handbook is to be similar to that earlier published under Task III A for self-acting bearings (Topical Report NYO 3930-5). Thus the two handbooks will form a set that provides the necessary information for designing either type of large, turbulent film bearings.

The externally-pressurized bearing has been used in earlier centrifugal, free-surface pumps. The FFTF pump will use an externally-pressurized bearing and this type has been selected also by the major pump contractors in their design studies for, at least the first, LMFBR pumps. This selection was based in part because of the lack of operational experience with large, sodium lubricated, self-acting and hybrid bearings. These latter, however, have some potential for future use due to their simplicity, low parasitic loss and high ultimate load capacities.

The effective operation of externally pressurized bearings rests upon their being fed through a flow restrictor, so that the total pressure drop between supply and discharge has two components:

1. The pressure drop across the restrictor system
2. The pressure drop in the bearing film

The division of total pressure drops between these two elements changes with displacement, so that the film pressure becomes a function of displacement. When the film thickness is large, the restrictor system accounts for the major part of the pressure drop so that the film pressure becomes small. When the film thickness is small the film becomes the dominant resistance to flow, so that the film pressure is high. This is the feature of the externally pressurized bearing that provides for positive film stiffness. Bearings in which a restricted supply must be considered, present some computational problems not encountered in self-acting bearings:

- First, the flow through the restrictor must be matched with the flow through the bearing film. The feeding source normally exhausts into a deep pocket of near uniform pressure, and the pocket pressure becomes a parameter which must be adjusted to satisfy overall flow continuity.
- A second problem is that the turbulence of the pressure induced flow introduces an additional nonlinearity into the equations to be solved.

The data presented in the self-acting bearing handbook (NYO 3930-5), as well as in the earlier Topical Report that described the analysis and verification tests (NYO 3930-2), had demonstrated that:

1. Turbulence radically affects the pressure distribution in the film.
2. Gross fluid inertia effects also greatly affect film pressures, wherever localized accelerations of the lubricant occur, such as at steps, pocket edges, and other discontinuities. These are termed Bernoulli effects.
3. Finally, the pressure distribution is influenced by the gradual accelerations and decelerations of the fluid through the film (convective inertia effects), but this influence is much less significant than that of Bernoulli type effects.

From these observations, it may be concluded that the convective inertia effects in the pocket and land areas of the externally pressurized bearings will be small, as they are in the case of the lands of the self-acting bearings, and may be neglected at this time. The inertia effects at the transition from pockets to land regions (Bernoulli effects) are, however, of first order importance and must be retained in the turbulent flow bearing analysis. With this simplification the analysis and numerical solution procedures for the large externally pressurized bearings were developed.

Work during the current reporting period was concentrated on completing the analysis, preparing the computer programs for numerical solutions, and

initiating design optimization calculations based on seeking maximum film stiffness.

The significant features combined in this computer program are the treatment of pressure flow induced turbulence; the treatment of Bernoulli-type inertia effects at points where the fluid accelerates into the film; and the calculation of both stiffness and damping coefficients. Novel numerical techniques were developed to achieve parallel iterative solution of the several different non-linearities involved in the governing equations. The resultant tool is capable of calculating performance over a wide range of operating conditions which fully encompasses the needs of the design handbook. The output from the program has been designed to form data sheets for direct reproduction in the design handbook.

The program was then used for design optimizations for maximum stiffness. A key objective of these computations was to establish quantitatively the influence of turbulence and inertia effects on bearing performance and to define optimum geometries for various levels of turbulence. Given this information, decisions can be made regarding the required volume of data, the best means of selecting optimum geometries and presenting information relating to performance under off-design conditions.

The following paragraphs give a brief description of the principles involved in the operation of a hydrostatic bearing, a qualitative description of the influence of turbulence and inertia effects on performance, and a discussion of the magnitude of these trends as demonstrated by output from a number of trial production runs covering a range of conditions.

The effective operation of a hydrostatic bearing results from the interrelationships between the resistance to flow of the lubricant supply orifice, and the resistance to flow of the bearing film. Under radially centered conditions the two resistances are generally of similar magnitude, so that the pressure in the pocket (assumed to be uniform) is about half the supply pressure. As the journal displaces from the centered position the film profile

around the bearing will be modified. In regions where the film height decreases the film resistance will increase relative to the orifice resistance, resulting in an increased pocket pressure. In regions where the film height increases, the film resistance will decrease resulting in a decreased pocket pressure. Thus a net pressure force opposing displacement results and, in this way, the bearing derives its stiffness and load capacity.

Under laminar conditions the resistance of the bearing film is independent of the bearing flow rate through the bearing film. However, use of a lubricant with a low kinematic viscosity such as liquid sodium will, in a large bearing, cause high levels of turbulence in the bearing film. Under turbulent conditions the film resistance increases as the flow increases.

Turbulence is likely to reduce stiffness in that a reduction in film height is accompanied by a reduction in turbulence level. The reduction in film height tends to increase resistance while the reduction in turbulence tends to decrease resistance. Thus stiffness may be expected to fall with increasing levels of turbulence.

Damping, on the other hand is expected to be significantly increased by turbulence. Damping represents the pressure force resistance of the bearing film to relative approach velocity between the journal and bearing surfaces. The film offers pressure force resistance because the relative velocity of the surfaces induces a change in flow. The change in flow is accompanied by a change in pocket pressure proportional to the film resistance to flow. The higher the film resistance to flow, the higher will be the pressure change. Hence turbulence, which increases flow resistance, increases damping in a hydrostatic bearing.

Inertia effects in the bearing film will be strongly predominated by the Bernoulli type pressure drop occurring as flow accelerates from the pocket into the film. For a given pressure differential between pocket and ambient, the fraction of this pressure differential accounted for by the Bernoulli drop will increase as the film clearance increases. The Bernoulli drop may

be considered as additional film resistance in series with the viscous resistance.

With respect to stiffness the Bernoulli effect may also be harmful in that its resistance decreases as the film height decreases. The Bernoulli effect will influence damping in a similar way to turbulence. Under laminar conditions, normalized performance of a hydrostatic pocketed bearing is a function of pad and pocket geometry, L/D ratio, eccentricity ratio, and some parameter relating orifice resistance to film resistance. Under conditions where turbulence and inertia effects are influential two additional parameters are necessary to define the performance. For the present study the additional governing parameters chosen are the groups $(P_s C^3 / (L \mu \nu))$ and (C/R) where: P_s is the supply pressure; C is the clearance; L is the bearing length; μ and ν are the absolute and kinematic viscosities respectively; and R is the bearing radius. The first group is analgous to a pressure flow Reynolds number, and is referred to as GRADPS. GRADPS governs the turbulence level, and (C/R) governs the level of inertia effects at a particular value of GRADPS. The parameter chosen to relate orifice resistance to film resistance is (d_o/C) where d_o is the orifice diameter. The parameter d_o/C is referred to as the orifice ratio.

Tables 1 through 4 are typical program output sheets and represent identical conditions apart from the level of turbulence (GRADPS). Comparison of the numbers for the four sets of data offer some confirmation of the trends anticipated above. Normalized damping (B_{XX}) and (B_{YY}) increases with increasing turbulence level, approximately as the square root of GRADPS. Inevitably the flow falls, again approximately as the square root of GRADPS. Centered stiffness falls mildly with turbulence, although at moderate and high eccentricity ratios the stiffness actually increases with turbulence. This reversal may well be a result of departure from optimum conditions. Optimization studies indicate that a significant penalty may be incurred for deviation from the optimum value of orifice ratio, and that the optimum value of d_o/C is appreciably influenced by the levels of both turbulence and inertia effects.

It may be seen that the load parameters at an eccentricity ratio of 0.9 does

be considered as additional film resistance in series with the viscous resistance.

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It may be seen that the load parameters at an eccentricity ratio of 0.9 does

Table 1

 DEEP POCKET HYDROSTATIC BEARING PERFORMANCE DATA

***** GEOMETRY AND OPERATING CONDITIONS *****

PAD EXTENT(DEG)	NO. OF PADS	NO. OF POCKETS	POCKET WIDTH RATIO	POCKET LENGTH RATIO	GRADPS	CLEARANCE RATIO	L/D RATIO	ORIFICE RATIO DQ/C
80.00	4	1	.800	.800	1,000E+03	.0020	1.000	60.000

***** STEADY STATE PERFORMANCE DATA *****

*****--LOAD OVER PAD CENTER--*****

ECCENTRICITY RATIO	LOAD PARAMETER	TOTAL FLOW PARAMETER
0.000	.0000	5.3865
.200	.1415	5.3298
.400	.2600	5.1773
.600	.3415	5.0076
.700	.3682	4.9553
.800	.3881	4.9384
.900	.4030	4.9574

*****--LOAD BETWEEN PADS--*****

ECCENTRICITY RATIO	LOAD PARAMETER	TOTAL FLOW PARAMETER
0.000	.0000	5.3861
.200	.1423	5.3260
.400	.2695	5.1718
.600	.3718	4.9902
.700	.4114	4.9267
.800	.4442	4.8926
.900	.4731	4.8828

***** DYNAMIC PERFORMANCE DATA *****

*****--LOAD OVER PAD CENTER--*****

ECCENTRICITY RATIO	* DYNAMIC COEFFICIENTS *			
	K(XX)	K(YY)	B(XX)	B(YY)
0.000	8.623E-01	8.611E-01	3.531E+00	3.531E+00
.200	7.811E-01	8.281E-01	3.355E+00	3.598E+00
.400	5.812E-01	8.051E-01	2.883E+00	3.645E+00
.600	3.612E-01	7.921E-01	2.317E+00	3.687E+00
.700	2.730E-01	7.743E-01	2.296E+00	3.760E+00
.800	2.105E-01	7.389E-01	3.137E+00	3.920E+00
.900	1.638E-01	7.155E-01	1.083E+01	4.431E+00

*****--LOAD BETWEEN PADS--*****

ECCENTRICITY RATIO	* DYNAMIC COEFFICIENTS *			
	K(XX)	K(YY)	B(XX)	B(YY)
0.000	8.596E-01	8.618E-01	3.531E+00	3.531E+00
.200	7.860E-01	8.243E-01	3.485E+00	3.467E+00
.400	6.857E-01	7.399E-01	3.271E+00	3.211E+00
.600	5.127E-01	6.112E-01	3.138E+00	2.886E+00
.700	4.313E-01	5.474E-01	3.360E+00	2.741E+00
.800	3.704E-01	4.876E-01	4.661E+00	2.700E+00
.900	3.506E-01	4.363E-01	1.507E+01	3.128E+00

Table 2

 DEEP POCKET HYDROSTATIC BEARING PERFORMANCE DATA

***** GEOMETRY AND OPERATING CONDITIONS *****

PAD EXTENT(DEG)	NO. OF PADS	NO. OF POCKETS	POCKET WIDTH RATIO	POCKET LENGTH RATIO	GRADPS	CLEARANCE RATIO	L/D RATIO	ORIFICE RATIO DD/C
80.00	4	1	.800	.800	1.000E+04	.0020	1.000	60.000

***** STEADY STATE PERFORMANCE DATA *****

*****--LOAD OVER PAD CENTER--*****

ECCENTRICITY RATIO	LOAD PARAMETER	TOTAL FLOW PARAMETER
0.000	.0000	2.0469
.200	.1319	2.0259
.400	.2550	1.9635
.600	.3550	1.8751
.700	.3909	1.8340
.800	.4176	1.7933
.900	.4346	1.7654

*****--LOAD BETWEEN PADS--*****

ECCENTRICITY RATIO	LOAD PARAMETER	TOTAL FLOW PARAMETER
0.000	.0000	2.0468
.200	.1323	2.0260
.400	.2603	1.9650
.600	.3758	1.8759
.700	.4255	1.8286
.800	.4674	1.7869
.900	.5006	1.7590

***** DYNAMIC PERFORMANCE DATA *****

*****--LOAD OVER PAD CENTER--*****

ECCENTRICITY RATIO	* DYNAMIC COEFFICIENTS *			
	K(XX)	K(YY)	B(XX)	B(YY)
0.000	8.171E-01	8.164E-01	1.086E+01	1.086E+01
.200	7.903E-01	8.128E-01	1.056E+01	1.089E+01
.400	6.900E-01	8.172E-01	9.550E+00	1.086E+01
.600	4.876E-01	7.887E-01	7.739E+00	1.104E+01
.700	3.701E-01	7.719E-01	6.801E+00	1.114E+01
.800	2.601E-01	7.711E-01	6.554E+00	1.122E+01
.900	1.724E-01	7.464E-01	1.326E+01	1.172E+01

*****--LOAD BETWEEN PADS--*****

* DYNAMIC COEFFICIENTS *			
K(XX)	K(YY)	B(XX)	B(YY)
8.151E-01	8.170E-01	1.086E+01	1.086E+01
7.976E-01	8.043E-01	1.075E+01	1.072E+01
7.354E-01	7.586E-01	1.038E+01	1.023E+01
6.373E-01	6.886E-01	9.842E+00	9.406E+00
5.368E-01	6.292E-01	9.825E+00	8.948E+00
4.432E-01	5.705E-01	1.066E+01	8.516E+00
3.670E-01	5.079E-01	2.052E+01	6.565E+00

Table 3

 DEEP POCKET HYDROSTATIC BEARING PERFORMANCE DATA

***** GEOMETRY AND OPERATING CONDITIONS *****

PAD EXTENT(DEG)	NO. OF PADS	NO. OF POCKETS	POCKET WIDTH RATIO	POCKET LENGTH RATIO	GRADPS	CLEARANCE RATIO	L/D RATIO	ORIFICE RATIO DO/C
80,00	4	1	,800	,800	1,000E+05	,0020	1,000	60,000

***** STEADY STATE PERFORMANCE DATA *****

*****LOAD OVER PAD CENTER*****

ECCENTRICITY RATIO	LOAD PARAMETER	TOTAL FLOW PARAMETER
0,000	,0000	,7026
,200	,1197	,6960
,400	,2378	,6756
,600	,3439	,6465
,700	,3865	,6293
,800	,4197	,6122
,900	,4425	,5977

*****LOAD BETWEEN PADS*****

ECCENTRICITY RATIO	LOAD PARAMETER	TOTAL FLOW PARAMETER
0,000	,0000	,7026
,200	,1196	,6960
,400	,2385	,6769
,600	,3534	,6470
,700	,4066	,6299
,800	,4550	,6128
,900	,4972	,5979

***** DYNAMIC PERFORMANCE DATA *****

*****LOAD OVER PAD CENTER*****

ECCENTRICITY RATIO	* DYNAMIC COEFFICIENTS *			
	K(XX)	K(YY)	B(XX)	B(YY)
0,000	7,692E+01	7,688E+01	3,155E+01	3,155E+01
,200	7,610E+01	7,664E+01	3,125E+01	3,161E+01
,400	7,104E+01	7,609E+01	2,968E+01	3,175E+01
,600	5,636E+01	7,499E+01	2,547E+01	3,200E+01
,700	4,593E+01	7,467E+01	2,250E+01	3,209E+01
,800	3,324E+01	7,398E+01	2,002E+01	3,243E+01
,900	2,171E+01	7,239E+01	2,397E+01	3,313E+01

*****LOAD BETWEEN PADS*****

* DYNAMIC COEFFICIENTS *			
K(XX)	K(YY)	B(XX)	B(YY)
7,676E+01	7,692E+01	3,155E+01	3,155E+01
7,615E+01	7,677E+01	3,151E+01	3,141E+01
7,324E+01	7,584E+01	3,121E+01	3,076E+01
6,709E+01	7,225E+01	3,044E+01	2,920E+01
6,078E+01	6,815E+01	3,019E+01	2,809E+01
5,335E+01	6,538E+01	3,116E+01	2,688E+01
4,465E+01	5,762E+01	4,005E+01	2,601E+01

Table 4

 DEEP POCKET HYDROSTATIC BEARING PERFORMANCE DATA

***** GEOMETRY AND OPERATING CONDITIONS *****

PAD EXTENT(DEG)	NO. OF PADS	NO. OF POCKETS	POCKET WIDTH RATIO	POCKET LENGTH RATIO	GRADPS	CLEARANCE RATIO	L/D RATIO	ORIFICE RATIO DQ/C
80.00	4	1	.800	.800	1.000E+07	.0020	1.000	60.000

***** STEADY STATE PERFORMANCE DATA *****

-----LOAD OVER PAD CENTER-----

ECCENTRICITY RATIO	LOAD PARAMETER	TOTAL FLOW PARAMETER
0.000	-.0000	.0754
.200	.0998	.0749
.400	.2050	.0730
.600	.3126	.0700
.700	.3620	.0690
.800	.4041	.0659
.900	.4362	.0639

-----LOAD BETWEEN PADS-----

ECCENTRICITY RATIO	LOAD PARAMETER	TOTAL FLOW PARAMETER
0.000	-.0000	.0754
.200	.0992	.0749
.400	.2021	.0731
.600	.3097	.0702
.700	.3642	.0684
.800	.4176	.0664
.900	.4687	.0644

***** DYNAMIC PERFORMANCE DATA *****

-----LOAD OVER PAD CENTER-----

ECCENTRICITY RATIO	* DYNAMIC COEFFICIENTS *			
	K(XX)	K(YY)	B(XX)	B(YY)
0.000	6.667E-01	6.666E-01	2.769E+02	2.769E+02
.200	6.823E-01	6.661E-01	2.786E+02	2.774E+02
.400	7.042E-01	6.653E-01	2.776E+02	2.786E+02
.600	6.507E-01	6.650E-01	2.569E+02	2.807E+02
.700	5.661E-01	6.643E-01	2.347E+02	2.825E+02
.800	4.500E-01	6.618E-01	2.080E+02	2.851E+02
.900	3.101E-01	6.562E-01	2.024E+02	2.898E+02

-----LOAD BETWEEN PADS-----

* DYNAMIC COEFFICIENTS *			
K(XX)	K(YY)	B(XX)	B(YY)
6.661E-01	6.667E-01	2.769E+02	2.769E+02
6.738E-01	6.762E-01	2.786E+02	2.778E+02
6.894E-01	6.989E-01	2.822E+02	2.787E+02
6.842E-01	7.121E-01	2.855E+02	2.753E+02
6.583E-01	7.027E-01	2.876E+02	2.704E+02
6.353E-01	6.833E-01	2.933E+02	2.629E+02
5.637E-01	6.413E-01	3.345E+02	2.553E+02

not vary strongly with turbulence. The reason is that the ultimate load capacity of a hydrostatic bearing is not a strong function of anything but the supply pressure and the pad and pocket geometries.

The influence of inertia effects on stiffness and damping is typified by Figures 1 and 2. The optimum stiffness is very slightly decreased by increasing clearance ratio. Peak damping is appreciably affected by clearance ratio, although not quite as strongly as by turbulence level (GRADPS). It is noticeable that the orifice ratio for optimum damping lies 15 to 20 percent below the ratio for optimum stiffness. However, since the penalty in damping for operating at optimum stiffness is not severe, optimization will be performed with respect to stiffness.

Initial optimization studies have shown that the optimum ratio of orifice diameter to clearance for stiffness under turbulent conditions lies between 40 and 100. However, the optimum is sufficiently dependent on operating conditions, and the penalties for off-optimum operation are sufficiently large that it will be necessary to present optimum orifice ratios which in general are a function of all operating parameters such as L/D, C/R, and GRADPS. An economical means of doing this is, at present, being developed.

Final optimization and the starting of production running will be performed during the next reporting period.

Tasks IV A, B, and C - Development and Long Term Tests of Proximity and Pressure Transducers for Operation in High Temperature Liquid Sodium

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, as well as for measurements of shaft orbits due to dynamic loads and to determine stability thresholds. 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

Fig. 1 Normalized Stiffness vs Orifice Ratio
 GRADPS = 5×10^6 ; $L/D = 1.0$ (Optimization)

Fig. 2 Normalized Damping vs Orifice Ratio
 GRADPS = 5×10^6 ; $L/D = 1.0$
 (Optimization)

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	: up 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.

Earlier in the current program, under subtask IV A, the proximity transducers and electronic read-out equipment were developed to meet the above specifications and tested first in nitrogen and then in sodium at temperatures in excess of 1100°F.

Similarly, under subtask IV B, film pressure instrumentation using a hermetically sealed capacitance probe that senses the electric deflection of a stainless steel diaphragm, was developed to meet the following criteria:

Primary Sensor

Environment: Sodium

Temperature: up to 1200°F

* When wired in "push-pull".

Primary Sensor (Cont.)

Pressure Range : 0 to 500 psig
 No. of Pressure Cycles : 5×10^8

High Temperature Leadwire

Environment : Sodium
 Temperature : up to 1200°F
 Pressure Range : 0 to 500 psig
 Length : to 20 feet
 Minimum Bend Radius : 0.5 in.

Detector

Environment : Air or Inert Gas
 Temperature : 50°F 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

During the present reporting period, extended duration tests were conducted on both the proximity and film pressure probes under subtask C. The objectives of this subtask were as follows:

1. To evaluate the performance of the proximity transducers over an extended range consistent with the clearances currently anticipated for large diameter (up to 24 in.) sodium pump bearings.
2. To subject the proximity and pressure transducers developed on Tasks IV A and IV B to long term (1,000 hrs.), tests in liquid sodium to give assurance to their long term operational capability and to identify the operational limitations of the instrumentation systems.

The range objective of the previously completed proximity transducer development, subtask IV A, was .040 in. with two diametrically opposed transducers electronically arranged in push-pull in the bridge circuit. After this work was initiated, it was determined that this range was somewhat restrictive when considering use of proximity transducers on large sodium pumps where bearing diameters approaching 24 inches are being studied. It is possible that bearing clearances of the order .002 to .003 in/in of diameter will be specified for the large pump bearings. This indicates that future proximity instrumentation systems would require near linear ranges approaching .080 in.

Because of these potential future requirements, the proximity transducers and its associated electronic system, developed under subtask IV A, was extensively tested to total target motions of .100 in. to determine its capability for extended range applications. These tests, discussed later in this section, demonstrated the applicability of the proximity instrumentation system for useable ranges in excess of .080 in.

Also during this report period, 684 hours of operation of both the proximity and pressure transducers was accumulated in liquid sodium at temperatures between 1050°F and 1170°F. Most of this time was at 1150°F $\pm$ 15°F. A severe cover gas leak in the test fixture caused a premature discontinuation of the endurance test. The instrumentation, however, suffered no distress at any time in the tests.

The test fixture used to perform both the extended range proximity transducer tests and the high temperature, long term testing of the proximity and pressure transducer tests was the same fixture as was used for the initial proximity transducer tests in sodium, conducted earlier and described in detail in the previous progress report (NYO 3930-6). The fixture was modified prior to this test series based on the experience of the earlier tests and to incorporate the pressure transducers. These modifications are briefly described below:

1. Two bulkheads were added to the fixture to which the test pressure transducers were attached such that they are fully submerged in

sodium during the high temperature test program.

2. The method of mounting the proximity transducers was changed to eliminate the spring retention method and to permit constant referencing of the target to the transducer face..
3. Vapor traps using stainless steel wool were incorporated into the target push-rod stand pipes to eliminate condensation of sodium on the cold surfaces of the test rig.

Photographs of the instrumentation test apparatus are shown on Figures 3 and 4 in which the principal test components are identified.

During the course of the high temperature endurance test, the test instrumentation was calibrated on a daily basis. The proximity transducers were arranged electronically in push-pull and the calibrations were made by moving the probe targets in opposite directions in .002 to .005 in. increments, depending upon the total test range, to simulate the motion of a shaft through a bearing clearance space. Typical proximity probe calibration curves are shown in Figs. 5 and 6 for a total test range of .100 in. ($\pm .050$ inches for each transducer) at 1150°F . These particular runs were made near the beginning and end of the endurance test (75:45 and 643:35 hours respectively). The output data as a function of transducer gap unbalance is also tabulated in Table 5 for these two test runs.

From an examination of Figures 5 and 6, and Table 5 it can be seen that the proximity instrumentation is extremely stable and repeatable at high temperature (1150°F) and that the output is near linear over the range that would be used for measuring bearing film thicknesses (of the order ± 0.040 inches).

Post test evaluation (following the high temperature test) of the proximity transducers revealed no change in electrical characteristics (DC resistance, inductance and Q) from the pre test values. The diaphragm which serves as a hermetic seal between the liquid metal in the gap and the probe coil-core assembly had been plastically deformed due to the 5 psig cover gas differential pressure across it during the test run (the inside of the transducer is

Fig. 3 High-Temperature Sodium Instrumentation Test Apparatus

Fig. 4 High-Temperature Sodium Instrumentation Test Apparatus

Fig. 5 Proximity Transducer Calibration in Sodium at 1152°F After 72 Hours (Data Point 11)

Fig. 6 Proximity Transducer Calibration in Sodium at 1150°F After 643 Hours (Data Point 46)

TABLE 5
 PROXIMITY TRANSDUCER DATA
 IN SODIUM AT 1150 and 1152 F

<u>Gap Unbalance (inches)</u>	<u>Output Voltage at</u>	
	<u>72 Hours (DP11)</u>	<u>643 Hours (DP46)</u>
+ .045	9.0	9.0
+ .040	7.33	7.29
+ .035	5.98	5.96
+ .030	4.85	4.84
+ .025	3.88	3.87
+ .020	3.03	3.01
+ .015	2.23	2.23
+ .010	1.49	1.50
+ .005	.75	.79
0	.05	.05
- .005	- .66	- .64
- .010	-1.37	-1.36
- .015	-2.13	-2.10
- .020	-2.92	-2.89
- .025	-3.80	-3.79
- .030	-4.77	-4.80
- .035	-5.92	-5.94
- .040	-7.33	-7.31
- .045	-9.01	-9.05

vented to atmosphere). This pressure load on the diaphragm had caused it to creep and plastically deform.

This particular situation will not be a problem in the bearing test rig because the inside of the transducer is being maintained at the same pressure as the outside by means of a pressurized leadwire junction box manifold arrangement. In other applications, where it may not be practical to maintain pressure equilibrium across the diaphragm, the problem can be readily solved by increasing the diaphragm thickness. The increased diaphragm thickness could either be compensated for by an equivalent decrease in standoff distance or an increase in gain setting to maintain the same sensitivity (0 to 10 volts full scale).

The capacitance type pressure transducers were calibrated in 10 psi increments on the same, daily interval basis as the proximity transducers. During the course of the high temperature test run the external bridge and signal amplification system lost sensitivity. The problem was traced to a faulty capacitance detector (Wayne Kerr Dm 100). After repair of the detector (approximately 100 hours before the conclusion of the test) the pressure measuring system performed in a stable and repeatable manner with good sensitivity as is evidenced by the calibration data shown on Table 6. The data shows that the output voltage is repeatable within 1%.

Post test evaluation of the transducer indicated that a zero shift had occurred when compared with pre test calibrations. This zero shift is a result of a measurable change in the pressure sensing capacitance gap. The exact cause of the change is at this time unknown but it could have been caused by either of the following factors:

1. Over pressurization - the diaphragms were designed for 120 psi operation at 1000°F. Many of the test calibrations were accomplished at 150 psi at 1150°F which resulted in a diaphragm maximum fiber stress of approximately 20,000 psi. Since the lower portion of the tensile yield strength scatter band of 347 stainless steel (the diaphragm material) is slightly under 20,000 psi, it is possible that some

TABLE 6
PRESSURE TRANSDUCER DATA
IN SODIUM AT 1130 F ($\pm$ 31 F)

Test Pressure (psi)	Output (Volts) at		
	<u>595 Hours</u>	<u>644 Hours</u>	<u>685 Hours</u>
0	1.053	1.056	1.053
10	1.107	1.111	1.107
20	1.159	1.163	1.161
30	1.211	1.215	1.214
40	1.261	1.264	1.264
50	1.309	1.311	1.312
60	1.355	1.358	1.361
70	1.399	1.403	1.407
80	1.442	1.445	1.452
90	1.486	1.488	1.495
100	1.526	1.530	1.537
110	1.569	1.570	1.579
120	1.612	1.606	1.618

plastic deformation occurred.

2. Thermal shock - after disassembly of the test equipment some residual sodium was left adhering to the transducer surfaces. The transducers were cleaned by reacting the residual sodium with water. The heat of reaction which was rapid and local could have caused high thermal stresses in the diaphragm resulting in permanent deformation of the order detected in the post test calibrations.

In summary, the tests demonstrated that:

1. The proximity instrumentation system has a useable range in excess of .080 inches ($\pm .040$) and that stable and repeatable high temperature performance can be expected. The only problem encountered with the proximity transducer was that of diaphragm creep which is readily solvable with either back pressurization through the leadwire sheath or by increased diaphragm thickness.
2. The pressure transducer performed in a stable and repeatable manner after the problem with the Wayne Kerr capacitance detector was isolated. Recurrence of this type of difficulty can be prevented by improved maintenance and periodic standardization procedures during operation.

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 technological 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	: Flooded hydrodynamic
Bearing Operation	: Vertical and submerged
Bearing Size	: 12" dia. x 12" lb. max.
Bearing C/R	: .001 to .003
Lubricant	: Sodium
Temperature	: 1200°F
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 effort on this task during the report period has been concentrated in the following areas:

1. Test rig manufacture and incorporation of minor modifications to ease and facilitate manufacture and assembly.
2. Design of test rig assembly tools and fixtures.
3. Installation of control room and facility components.

It was indicated in the previous progress report that the assembly procedures had been finalized and that the assembly tool requirements had been identified. During the course of this effort, several minor test rig component modifications to ease assembly and provide adaptation to standard tooling were identified. These modifications were incorporated into the detail drawings during this report period. In addition, the special assembly tools mentioned above were designed and released for manufacture.

The subcontract for manufacture of the rotor was let and delivery is anticipated in September. The delay in delivery from the April date projected in the

previous report is due mainly to the difficulty in procuring the basic austenitic stainless steel forgings of the required size as a result of the prolonged strike in the nickel industry. All of the test rig pressure vessel components which consist of all stationary structural and sodium containment components below the cover gas seal were received and have been installed in the test cell. Figure 7 shows the installation of the flange cover mounted bearing housings retainer cylinder into the test vessel.

Also during this report period the pneumatic plumbing and wiring of control room operating and control components was completed. In addition, wiring between control room and test cell junction boxes was completed. During the next period, control room - test cell pneumatic interconnections and test cell vacuum system installation will be initiated.

Task VI B - Screening of Bearing Surface Materials

The purpose of this initial task is to experimentally screen candidate bearing surface materials and coatings for liquid sodium lubricated bearings operating at temperatures up to 1200°F. Upon completion of the screening tests and establishment of favorable surface material types, methods of non-destructive testing of the adherence and integrity of the surface coatings are to be established.

The candidate materials including plasma sprayed, diffusion bonded and weld applied types were selected during the previous report period and identified in the previous report (NYO 3930-6).

During this period, samples for testing in high temperature sodium were made and pretest dimensional inspections and photomicrographs were prepared. Two samples of each candidate material was placed in the same test chamber and subjected to the same environment as the instrumentation described under Task IV. These samples have since been cleaned after their exposure to high temperature sodium and are currently undergoing post test evaluation.

The post test evaluation to be completed during the next period will include:

- Dimensional inspection
- Bond strength testing
- Sectioning and photomicrographs

III. PROGRAM STATUS

During the following period, the principal effort will be in the following areas:

1. Completion of the program on large, externally-pressurized bearings and completion of the optimization of these bearings for maximum fluid film stiffness. The production runs to generate the data for the design section of the handbook will also be started.
2. Continue the installation of the components of the large, self-acting, sodium bearings test rig.
3. Analysis of the data of the tests on corrosion of the bearing surface materials in high temperature sodium.

Also, a Topical Report has been prepared on the proximity and film pressure probe development and long term tests in liquid sodium, will be issued for technical and patent approval.

The bearing tests in liquid sodium are now scheduled to be started in November 1970 and completed in January 1971.

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