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NYO 3930-10

MTI 71TR11

PROGRESS REPORT

For the Period

August 1, 1970 - January 31, 1971

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

AEC CONTRACT AT(30-1)-3930

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PROGRESS REPORT

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

prepared for

U.S. Atomic Energy Commission

prepared under

Contract AT(30-1)-3930

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February, 1971

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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 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, self-acting bearings.
Task III B	-	Preparation of a design data handbook for large, externally pressurized 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. The design handbook on hydrostatic bearings was completed and submitted to the Commission for patent clearance and technical approval.
2. All remaining parts of the 12" diameter sodium bearings test rig were received and facility installation was completed. Decontamination of the sodium exposed parts of the test equipment was made.

Following is a description of the work by task.

II. DISCUSSION OF PROGRAM PROGRESS

Task I - Program Management and Reporting

Letter type monthly progress reports were issued, as well as a formal progress report (NYO 3930-8) for the period May 1, 1970 through July 31, 1970.

Task II - Fluid Inertia Investigation

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

Tasks III A and III B - Design Handbooks on (A) Self-Acting Bearings, and (B) Externally-Pressurized Bearings

Under Subtask A, the design handbook on self-acting bearings was completed and described in Topical Report NYO 3930-5 issued in November 1969.

Under Subtask B, a similar handbook was prepared for hydrostatic bearings. The analysis and the bearing optimization procedure had been completed earlier and reported in the previous progress report, NYO 3930-8.

During the period covered by this report, the effort on this task comprised:

1. The optimization calculations were completed and the data required for the design handbook was identified.
2. All the production runs needed to generate the design data tables for the handbook were conducted.
3. A review and analysis of the data generated was conducted to establish the effects of the geometrical parameters of the bearings and of the level of turbulence on bearing performance.
4. The design handbook was prepared and draft copies submitted to the Commission for patent clearance and technical review.

This design handbook provides tables of the steady state (load capacity and flow) as well as dynamic (fluid film stiffness and damping) characteristics

of externally pressurized bearings for a wide range of geometrical parameters and levels of turbulence. Numerical examples are provided to illustrate the utilization of the data.

Here, we will briefly review the results of the data analysis and optimization study that is treated in detail in the design handbook. As earlier noted (in Progress Report NYO 3930-8), the bearing performance was calculated as a function of the centered pressure ratio $\bar{p}_r$.

The advantage of this approach is that bearing characteristics are more consistently controlled by the pressure ratio than by a geometric parameter defining the hydrostatic inlet resistance, such as orifice diameter. Having defined the pressure ratio and obtained the bearing performance, including flow, the necessary orifice diameter to provide this performance may be calculated. In the following discussion, the results of the optimization study performed in terms of pressure ratio are summarized.

For the study, a wide range of operating conditions is examined as summarized in Table 1. Each of the influential parameters is varied, as shown, in turn, about the standard case value, while the other parameters maintain their standard case value. Thus nine conditions are examined in all. At each condition the pressure ratio is varied from 0.1 to 0.9 and corresponding performance parameters evaluated.

TABLE 1
RANGE OF CONDITIONS FOR OPTIMIZATION

Influential Parameter	Standard Case Value	Variations
L/D	1.	0.75, 0.5
GRADPS	10^5	10^3 , 10^7
C/R	0.002	0.001, 0.004
L_p/L	0.8	0.5, 0.2

The bearing performance parameters examined are stiffness, $\bar{K}$; flow, $\bar{Q}$; and damping, $\bar{B}$. Two further quantities are extracted, the ratio of stiffness to flow, and the ratio of orifice diameter to clearance which corresponds to the imposed pressure ratio and the calculated flow. Only a single bearing pad is examined for this study. It may be assumed that trends demonstrated by a single pad will correspond closely to the trends for a complete multi-pad journal bearing.

The variation of each performance parameter with pressure ratio has been plotted for the above conditions, and a number of representative plots are reproduced here (Figs. 1 through 4). The following trends have been observed from this optimization study:

1. Maximum values of stiffness, damping and stiffness to flow ratio occur at consistent values of centered pressure ratio, as defined in Table 2.

TABLE 2

Values of Centered Pressure Ratios for Optimum Performance

Centered Pressure Ratio, $\bar{p}_r$	Optimum Quantity
0.5	Stiffness
0.35	Damping
0.33	Stiffness to Flow Ratio

There are two exceptions to these patterns. Firstly, at a value of GRADPS equal to 10^3 , which is the lowest value considered in this study, the stiffness to flow ratio does not show a peak in the range of pressure ratios considered. A probable reason is that $GRADPS = 10^3$ is close to laminar conditions; and at the lower pressure ratios turbulence effects disappear. Due to this transition the rate of change of flow with pressure ratio will be greatest at low pressure ratios, and the peak in the stiffness to flow curve will be shifted to a lower pressure ratio. Under fully laminar conditions the curve may be expected to peak out at a

□	STIFFNESS	GRADPS = 10**3
○	STIFFNESS	GRADPS = 10**7
△	STIFFNESS	GRADPS = 10**5
+	ORIFICE RATIO	GRADPS = 10**3
X	ORIFICE RATIO	GRADPS = 10**7
◇	ORIFICE RATIO	GRADPS = 10**5

OPTIMIZATION. STIFFNESS AND ORIFICE RATIO VS
 CENTERED PRESSURE FOR A RANGE
 GRADPS VALUES
 C/R = .002 LP/L = 0.8
 L/D = 1.0

FIGURE 1

□	FLOW	C/R = .002
○	FLOW	C/R = .004
△	FLOW	C/R = .001

OPTIMIZATION. FLOW VS. CENTERED PRESSURE
 FOR A RANGE OF C/R VALUES
 L/D = 1. LP/L = .8
 GRADPS = 10**5

FIGURE 2

$\square$ DAMPING $L/D = .75$
 $\circ$ DAMPING $L/D = .5$
 $\triangle$ DAMPING $L/D = 1.$

OPTIMIZATION. DAMPING VS. CENTERED PRESSURE
 FOR A RANGE OF L/D VALUES
 $C/R = .002$ $LP/L = .8$
 $GRADPS = 10^{**}5$

FIGURE 3

- STIFFNESS/FLOW LP/L = .5
- STIFFNESS/FLOW LP/L = .2
- △ STIFFNESS/FLOW LP/L = .8

OPTIMIZATION. STIFFNESS/FLOW VS. CENTERED
 PRESSURE FOR A RANGE OF
 LP/L VALUES
 L/D = 1. C/R = .002
 GRADPS = 10**5

FIGURE 4

pressure ratio near 0.33 again. Secondly at the smallest pocket size, $L_p/L = 0.2$, the peak damping occurs at a pressure ratio of 0.8 which is markedly different from the value of 0.35 tabulated above. Even at $L_p/L = 0.5$, peak damping has shifted from a pressure ratio of 0.35 to 0.4. This effect results because, for the small pocket, the influence of the pocket on damping becomes very slight. Apart from these exceptions, the trends of Table 2 are consistent.

2. Flow, as would be expected, is a monotonically increasing function of pressure ratio. At low values of pressure ratio the rate of increase from zero is sharpest, since at sufficiently low pressure ratios the flow must become laminar.
3. The orifice diameter corresponding to the specified pressure ratio also increases monotonically with pressure ratio. The rate of increase is sharp at low pressure ratios, flattens off in the middle of the range, and rises again in the high pressure range. Clearly as the pressure ratio tends to one, the orifice diameter will tend to infinity.
4. The onset of turbulence tends to reduce normalized stiffness. However, once turbulence is developed, further increase in turbulence has a less marked effect. A greater loss in normalized stiffness occurs between the GRADPS values of 10^3 and 10^5 (23%) than between the values 10^5 and 10^7 (13%).
5. Inertia effects governed by clearance ratio also tend to reduce stiffness. Increasing clearance ratio from 0.001 to 0.004 decreases normalized stiffness by approximately 15%. This means that, in turbulent flow, the reduction in actual stiffness (K, lbs/in), resulting from increase in clearance (C, inches) will be somewhat greater than that predicted by the inverse relationship between stiffness and clearance that prevails under laminar flow.
6. L/D ratio has a negligible effect on stiffness or flow. However, damping is significantly affected by L/D ratio. Normalized damping is almost directly proportional to L/D ratio over the entire range of pressure

ratios considered. As damping is normalized with respect to bearing length amongst other parameters, it is clear that actual damping is proportional to the square of the bearing length.

7. Considering specifically the influence of pocket size, stiffness is strongly influenced - the bigger the pocket - the stiffer the bearing. Damping is also improved by a large pocket size. However, at the same time, flow increases strongly with pocket size, and a trade-off may be observed between flow on the one hand and stiffness and damping on the other hand. The ratio of stiffness to flow may be an additional parameter of interest in deciding pocket size, and the influence of pocket size on this parameter may be seen in figure 4. The apparent trend is a continuous increase in this ratio as pocket size falls, although there is a sharp leveling off of the effect below a 50% value of L_p/L .

In considering the above observations in the light of the objectives of the Handbook, the following decisions were made as to the performance data to be generated and presented.

1. The majority of the bearing data should be generated for a high stiffness bearing. This bearing should have a pocket axial length ratio of 0.8 and a centered pressure ratio of 0.5.
2. Secondary data should be run for a bearing with high stiffness to flow ratio. This bearing will have a pocket axial length ratio of 0.5 and a centered pressure ratio of 0.33.
3. The orifice ratio which will provide the desired pressure ratio and corresponding flow will be calculated and provided as an item of data. The value calculated will be based on a discharge coefficient of 0.6. This is a standard value often used in the absence of specific data (8). If a different value of C_d were to be favored by a designer the corresponding modification to orifice diameter is given by:

$$\frac{do'}{do} = \left(\frac{0.6}{Cd'} \right)^{1/2}$$

where do' is the modified orifice diameter, do is given by the performance data, and Cd' is the preferred coefficient of discharge.

4. In addition to the performance of the optimum bearing, the designer will require a basis for evaluating off-design performance. To provide this basis, the optimization data has been reordered and replotted. The ratio of performance under non-standard conditions to performance under standard conditions has been related to the corresponding ratio of non-standard orifice diameter to standard orifice diameter. Thus for the high stiffness bearing ($L_p/L = 0.8$)

$$\left\{ \begin{array}{l} \bar{K}/\bar{K}_{Pr} = 0.5 \\ \bar{B}/\bar{B}_{Pr} = 0.5 \\ \bar{Q}/\bar{Q}_{Pr} = 0.5 \end{array} \right\} \quad \text{have been plotted against} \quad (do)/(do)_{Pr} = 0.5$$

where $\bar{Pr}$ is the centered pressure ratio. For the bearing with high stiffness to flow ratio ($L_p/L = 0.5$)

$$\left\{ \begin{array}{l} \bar{K}/\bar{K}_{Pr} = 0.33 \\ \bar{B}/\bar{B}_{Pr} = 0.33 \\ \bar{Q}/\bar{Q}_{Pr} = 0.33 \end{array} \right\} \quad \text{have been plotted against} \quad (do)/(do)_{Pr} = 0.33$$

These curves are presented in an appendix of the Design Handbook. It is found that near universal curves of the performance ratios as function of $(do)/(do)_{Pr} = 0.5$ are obtained, as is seen typically in Figure 5, for flow. With the detailed performance data as a base, and with these curves for correction, the performance of bearings with a wide range of orifice sizes can be obtained.

The use of the performance data and off-design performance curves is discussed and demonstrated in detail in a section of the Design Handbook.

- STANDARD
- ⊖ L/D = .5 AND .75
- △ C/R = .004

- X GRADPS = 10**3
- ◇ GRADPS = 10**7

FIGURE 5

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

The development and testing of proximity (film thickness) and film pressure probes for long term operation in liquid sodium at temperatures up to 1100 F was completed. The Topical Report that describes the development and test effort was drafted.

Task V - Analysis, Design and Procurement of Sodium Test Bearing System

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" 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 rpm capacity LMFBR sodium circulation system pump.

The principal effort on this task during the report period has been in the following areas:

1. Assembly of the test facility, including the installation of all pumping and instrumentation was completed.
2. The rotor, which had been the last outstanding element of the test equipment, was delivered.
3. All the heaters and thermocouples were installed in the sump tank and test vessel and the insulation of these vessels, as well as of the sodium pumps was completed.
4. Cleaning and decontamination of the sodium exposed parts of the stator was completed and decontamination of the rotor parts was started.
5. The vacuum system including all valves was checked out.

Figure 6 shows the heaters and thermocouples installed in the sump tank, prior to insulation of the tank, and Figure 7 shows the insulated sump tank.

Completion of decontamination of the rotor parts and initiation of tests are scheduled for the next quarterly period.

Fig. 6 Sump Tank with Thermocouples Installed, Prior to Insulation

Fig. 7 Sump Tank After Insulation

NOMENCLATURE

$\bar{B}$	Normalized damping	$\frac{2\pi Bc}{\mu DL(R/c)^2}$	
c	Clearance, inches		
C_d	Discharge coefficient for orifice		
D	Bearing diameter, inches		
d_o	Orifice diameter, inches		
GRADPS	Supply turbulence parameter	$\frac{P_{sc}^3 \rho}{L\mu^2}$	(non-dimensional)
$\bar{K}$	Normalized stiffness	$\frac{cK}{P_s L D}$	
L	Bearing length, inches		
L_p	Pocket length, inches		
$\bar{P}_r$	Ratio of recess pressure to supply pressure under radially centered conditions of the journal		
P_s	Supply pressure, lb/in ²		
$\bar{Q}$	Normalized flow	$\frac{Q\mu}{P_{sc}^3}$	(non-dimensional)
Q	Flow, in ³ /sec.		
R	Radius of bearing, inches		
μ	Viscosity, lb.in ⁻² sec.		
ρ	Density, lb.in ⁻⁴ sec ²		

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