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VOLUME VI OF VI

TRW REPORT No. ER-6041

AEC CATEGORY C-92B, M-3679 (43rd EDITION)

MERCURY RANKINE PROGRAM (SNAP 2) TOPICAL REPORT No. 28

MERCURY RANKINE PROGRAM (SNAP 2)
DEVELOPMENT OF LIQUID-MERCURY-LUBRICATED BEARINGS
VOLUME VI
SPIRAL-GROOVE THRUST BEARING
EXPERIMENTAL RESULTS
(TITLE UNCLASSIFIED)

AEC RESEARCH AND DEVELOPMENT REPORT

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FOREWORD

This Development of Liquid-Mercury-Lubricated Bearings Topical Report defines the analytical and experimental results obtained with fluid film bearings lubricated by liquid mercury. This report includes the design approach, analytical techniques, procedures, test results, and experimental activity performed by TRW Inc. as a facet of the Mercury Rankine Power Conversion Program during the period of January 1961 through June 1963.

This report consists of the following six volumes:

Volume I	Analytical Design Approach and Status of Bearing Systems (With Appendixes)
Volume II	Plain Bearing Experimental Results
Volume III	Tilting-Pad Bearing Experimental Results
Volume IV	Three-Sector Bearing Experimental Results
Volume V	Three-Pad Bearing Experimental Results
Volume VI	Spiral-Groove Thrust Bearing Experimental Results

For convenience of the reader, citations of reference works are the same in each volume; e. g., Reference 6 cited in Volume I and Reference 6 cited in Volume II are the same publication.

Volume I contains the Nomenclature and the References on the study. All other Appendixes are also included in Volume I.

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ABSTRACT

This volume presents the results of the activity associated with the development of the double-acting spiral-groove thrust bearing.

On the basis of previous development work, which evaluated the hydrodynamic pocket type Rayleigh step bearing, the spiral-groove thrust bearing is selected, sized, and tested. Hydrostatic capacity is achieved by means of an annular orifice restriction to provide the required static load capacity. When combined with the hydrodynamic capacity generated as the speed increases and the other desirable performance characteristics such as flow-pressure requirements and power loss, the double-acting spiral-groove thrust bearing meets all CRU specifications as they exist at this time.

The experimental performance of this bearing is presented and compared with predicted performance. The primary dimensional parameter which is varied is total axial clearance. Its influence on load capacity is reviewed. For the selected size and considering supply pressure and flow requirements, the spiral-groove bearing with a total axial clearance of 0.001 inch has the largest minimum film thickness for the specified loads and is therefore selected for application to CRU's.

Some agreement is achieved with performance predicted by Whipple's theory. The influence of inertia and turbulence limits the agreement. Power loss in particular is influenced by high Reynolds numbers (Re at mean diameter greater than > 3000). No life limitation in terms of surface damage due to cavitation-erosion is anticipated.

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I. INTRODUCTION AND SUMMARY

The hybrid double-acting spiral-groove thrust bearing was developed to meet the CRU requirements after the hydrodynamic pocket-type Rayleigh step bearing was shown to have inadequate load capacity. It utilizes more of the available surface area since the spiral grooves guide the high inertia fluid in the clearance space and prevent the fluid from by-passing the land area of the bearing. Spiral grooves pumping outward were selected specifically to utilize the inertia of the fluid to get increased capacity. Analysis for compressible flow indicates that spirals pumping inward have superior load capacity. However, this analysis assumes that the inertia of the compressible fluid is small enough to be negligible. For incompressible flow, i. e., mercury, this assumption is not valid.

During the period covered by this report, the dual-face spiral-groove thrust bearing was used in all component and CRU tests. The bearing was designed for loads up to 30 pounds, flows of 6 to 8 lb/min at a temperature of $375^{\circ}\text{F} \pm 25^{\circ}\text{F}$. The major dimensions of the bearing are 0.625-inch OD and 0.3125-inch ID. The dual-face feature, with six spiral grooves per face, provides the bearing with load capacity in both directions. However, the slope of the spiral permits rotation in only one direction. Lubricant is introduced to the faces through an annular orifice formed by the outside diameter of the shaft and the inside diameter of the bearing.

To assure optimum load capacity, the annular orifice acts as a restrictor assuring build-up in pressure on the loaded side, while the pressure is reduced on the unloaded side.

A computer program was used to optimize bearing dimensions for maximum load capacity with the restriction of limited flow and fixed outside and inside diameters.

The spiral-groove thrust bearing was subjected to two distinct test programs aimed at establishing experimental performance. The initial test for overall performance evaluation was conducted to determine speed and axial load influence on flow-pressure characteristics. The second test was more comprehensive and established the influence of axial clearance on performance characteristics. The total axial clearance (TAC) was varied from 0.0007 to 0.0013 inch in three steps while all other dimensions were held constant. Additional instrumentation permitted measurement of load-deflection characteristics as well as determination of flow-pressure-speed relationship. A series of performance maps was generated from these tests. The performance maps relate load capacity, supply pressure drop, flow, and minimum film thickness for speeds from zero to 40,000 rpm over the tested total axial clearance range.

The spiral-groove thrust bearing selected for these tests met the CRU requirements. Optimum load capacity for an allowable minimum film thickness of 0.0002 inch was achieved with the bearing having a TAC of 0.001 inch. A maximum load of 68 pounds was supported satisfactorily at a speed of 40,000 rpm. A static load capacity of 19

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pounds was achieved at a supply pressure of 203 psi and a flow of 12.5 lb/min. The hybrid nature of the bearing was verified with supply pressure influencing load capacity at low speeds but having an insignificant effect at high speeds. Correlation between experimental load capacity and flow-pressure characteristics and theoretical predictions was satisfactory at zero and low speeds (below 10,000 rpm) and at high speeds (above 30,000 rpm). However, for intermediate speeds (10,000 to 30,000 rpm), the theoretical prediction was optimistic since the actual experimental capacity was lower. No evidences of cavitation-erosion damage were observed.

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II. SPIRAL-GROOVE THRUST BEARING

Two types of thrust bearings were evaluated for SNAP 2 application as of June, 1963. The hydrodynamic, pocket-type, Rayleigh, step bearing shown in Figure 1 was tested prior to the period covered by this report and its performance is presented in Reference 1. The hydrodynamic nature of the bearing and the centrifugal force causing the lubricant to bypass the pockets resulted in a poor load capacity of approximately fifteen pounds maximum. Consequently, the hybrid spiral groove thrust bearing shown in Figure 1 was developed. It utilizes more of the available surface area, the inertia forces of the lubricant film being converted to useful thrust capacity by the restricting dams around the outside perimeter. Spiral grooves pumping outward were selected specifically to utilize the inertia of the fluid to get increased capacity. The analysis indicates that spirals pumping inward have superior load capacity for compressible flow. This analysis neglects the small inertia of compressible fluids but takes into account the curvature of a circular bearing.

During the period covered by this report, the dual-face spiral-groove thrust bearing has served as the thrust bearing for all component tests and for all CRU models since CRU III. The thrust bearings are designed to withstand loads up to 30 lb, flows of 6 to 8 lb/min, and temperatures of $375^{\circ}\text{F} \pm 25^{\circ}\text{F}$.

The dual-face spiral-groove thrust bearing is shown in detail in Figures 2 and 3. The dual-face feature provides the bearing with load capacity in both directions, but the slope of the spiral grooves permits rotation in only one direction. Lubricant is fed through the annular orifice formed by the outside diameter of the shaft and the inside diameter of the bearing.

To obtain maximum thrust load capacity on the loaded side, the pressure on the unloaded side should be minimized. The annular orifice flow restriction between the bearing supply (P_S) and the spiral groove inlet pressure (P_O) at the minimum area accomplishes that objective. Consequently, as thrust load is applied and the film thickness of the loaded side decreases, the flow will decrease to that side tending to cause $P_O \rightarrow P_S$. The combination of higher inlet pressure (P_O) and minimum clearances increases the pressure developed on the loaded side and, therefore, increases the load capacity. Meanwhile the increased clearance on the unloaded side results in greater flow and a higher pressure drop across the annular orifice so that the developed pressure imposes a smaller contributing load on the loaded thrust bearing. The annular orifice, therefore, acts as a restriction and to some degree provides the type of orifice compensation found in journal bearings. The second restriction to flow, the clearance between the thrust faces, causes the two flow restrictions to be in series.

Optimizing the bearing for maximum load capacity and limited flow with outside and inside diameters fixed was accomplished with the help of a computer and the equations presented in Volume I and in Appendix J.

HOLE FED JOURNAL BEARING

AXIAL GROOVE JOURNAL BEARING

TANGENTIAL INLET JOURNAL BEARING

ANNULAR GROOVE BEARING

LOAD-PAD TYPE JOURNAL BEARING

THREE-SECTOR JOURNAL BEARING

THREE-PAD JOURNAL BEARING

TILTING-PAD BEARING

Figure 1. Typical Journal Bearing Geometries

(Actual Bearing has 6 Grooves)

All Dimensions in Inches

Figure 2. Spiral-Groove Thrust Face Layout

All Dimensions in Inches

Figure 3. Spiral-Groove Thrust Bearing Configuration

-

OD = 0.625 inch
 ID = 0.312 inch
 b = 0.110 inch
 c = 0.047 inch
 $\theta = 70^\circ$
 $h_2 = 0.0002$ inch
 Mercury at 430°F
 Speed = 40,000 rpm

*Thrust per inch of length along the mean diameter

Figure 4. Effect Of Groove Depth On Thrust Capacity
For A Spiral-Groove Thrust Bearing
(Predicted Performance)

*Per Inch Along the Mean Diameter

Figure 5. Effect Of Groove Depth On Mercury Flow For A
Spiral-Groove Thrust Bearing
(Predicted Performance)

OD = 0.625 inch
 ID = 0.312 inch
 $\theta = 70^\circ$
 $a_1/a_2 = 1.04$
 $TAC = h_1 - h_2 = 0.0010$ inch
 Speed = 40,000 rpm

*Per Inch Along Mean Diameter

Figure 6. Effect of Dam Width on Thrust Capacity and Mercury Flow
 For A Spiral-Groove Thrust Bearing
 (Predicted Performance)

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*Per Inch Along Mean Diameter

Figure 7. Effect of Groove Angle on Thrust Capacity and Mercury Flow For A Spiral-Groove Thrust Bearing

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These dimensions, when combined with fixed outside and inside diameters, dictated the following thrust bearing geometry:

Outside diameter (D_o)	0.625 inch
Inside diameter (D_i)	0.312 inch
Mean diameter (D_m)	0.469 inch
b	0.110 inch
c	0.047 inch
Groove depth ($h_1 - h_2$)	0.0005 to 0.001 inch
a_1	0.1035 inch
a_2	0.117 inch
θ	72 degrees
n	6 grooves
Annular orifice feed restriction	0.0048 inch nominal
Axial clearance (total)	0.001 inch nominal
Groove width	0.032 inch at mean diameter

The grooves are approximated by circular areas as shown in Figure 2.

A. Predicted Performance

As indicated in Volume I and Appendix J, predicted performance is based on the theory presented by R. T. P. Whipple (Ref. 20). No allowance was made for turbulence in the film for load capacity or flow. Calculation of the mean Reynolds number revealed

$$Re_{mean} = \frac{\rho h U_{mean}}{\mu} \quad . . . (1)$$

$$Re_{mean} < 3,000 \text{ at } 40,000 \text{ rpm}$$

Consequently, the anticipated increased load capacity as a result of this large Reynolds number was not considered, but it was assumed that calculated capacity was conservative.

The theoretical performance of the spiral groove thrust bearing defined above was evaluated with the aid of a Burroughs 205 digital computer. The objective of this analysis was to establish the effect of speed, supply pressure, temperature, and total axial clearance on the performance of the bearing. The two major performance parameters of interest were load capacity and flow-pressure characteristics; these were obtained and plotted versus total axial clearance (TAC) for given speeds and film thicknesses at constant supply pressures. From these curves, flow and load were cross plotted versus supply pressure, maintaining speed and clearance constant at

various film thicknesses. With these two sets of performance maps at each speed and temperature, it is possible to define bearing performance if the clearance is known. Design plots were made for speeds of 10,000; 20,000; 30,000; and 40,000 rpm; film thicknesses (h_{min}) of 0.0001; 0.0002; 0.003; and TAC/2 inch; temperatures of 100°F and 400°F; total axial clearances (TAC) of 0.0008; 0.001; 0.0012; and 0.0013 inch; and supply pressures of 0; 50; 100; 150; 200; and 250 psia.

Obviously a large number of curves were generated to illustrate bearing performance. Figures 8 through 11 are typical plots showing load capacity and flow versus TAC and supply pressure at 40,000 rpm. A wide variety of possible combinations of pressure and clearances are possible in the actual machine. Since flow is maintained constant in the CRU, it is possible to cross plot from the above curves the performance of the thrust bearing in terms of constant flow curves. This has been done for a 7 lb/min flow in Figures 12 and 13 to show load capacity and supply pressure requirements as functions of speed for various film thicknesses and TAC at both room temperature and 400°F.

Evaluation of the various performance curves indicates the following:

- 1) At a given speed, clearance and film thickness; the load capacity and flow vary linearly with supply pressure.
- 2) At a given speed and flow, the load capacity is almost independent of clearance.
- 3) At a given clearance and flow, the load capacity varies linearly with speed.
- 4) At a given film thickness, clearance, and flow; the supply pressure decreases with speed.

A typical performance map of the spiral groove thrust bearing derived from Figures such as 12 and 13 is shown in Figure 14 as a function of different TAC's. The following additional conclusions can be drawn from such a plot for a 40,000 rpm speed and 0.0002-inch minimum film thickness:

- 1) At a given flow an optimum TAC exists which produces a maximum thrust capacity. However, the maximum TAC varies slightly with flow. Figure 14 shows that between 6 and 14 lb/min, the optimum TAC varies from approximately 0.0008 to 0.00095 inch.
- 2) Increasing TAC above 0.0011 inch has little effect on thrust capacity for constant flow. Supply pressures will drop significantly.
- 3) Decreasing TAC below 0.0011 inch produces a modest increase in load capacity for constant flow, accompanied by a large increase in supply

Figure 8. Load and Flow vs Total Axial Clearance for Various Supply Pressures
 For A Spiral-Groove Thrust Bearing at $T_{in} = 100^\circ\text{F}$
 (Predicted)

Figure 9. Load and Flow vs Total Axial Clearance for Various Supply Pressures for a Spiral-Groove Thrust Bearing at $T_{in} = 400^\circ\text{F}$ (Predicted)

Figure 10. Load and Flow vs Supply Pressure for Various Film Thicknesses for a Spiral-Groove Thrust Bearing at $T_{in} = 100^\circ\text{F}$ (Predicted)

Figure 11. Load and Flow vs Supply Pressure for Various Film Thicknesses for a Spiral-Groove Thrust Bearing at $T_{in} = 400^{\circ}\text{F}$ (Predicted)

Figure 12. Load and Supply Pressure vs Speed for a Spiral-Groove Thrust Bearing for a Constant Flow of 7 lb/min and $T_{in} = 100^{\circ}\text{F}$ (Predicted)

Figure 13 Load and Supply Pressure vs Speed for a Spiral-Groove Thrust Bearing
for a Constant Flow of 7 lb/min and T_{in} = 400°F
(Predicted)

Figure 14. Load Capacity and Supply Pressure vs TAC at Various Flows for a Spiral-Groove Thrust Bearing

pressure. A maximum is reached beyond which further reduction in TAC is accompanied by a sharp decrease in load capacity.

Based on the predicted performance and the optimization study on groove depth, ratio of grooved area to ungrooved area and groove angle, a TAC of 0.0009 to 0.0011 inch appears most suitable for the CRU application.

B. Hydrostatic Effects

Lubricant is fed through the annular restriction (orifice) between the bearing inside diameter and the shaft outside diameter as shown in Figure 3. Theoretical flow through a concentric annular restriction is shown in Figure 15. On the loaded side of the thrust bearing the flow is small and, consequently, the inlet pressure to the grooves approaches the supply pressure. On the thrust bearing's unloaded side the flow is large, resulting in a larger pressure drop through the annular restriction. The result is a lower groove inlet pressure and, therefore, a higher net thrust capacity and a restricted flow through the bearing. The thrust bearing consequently acts somewhat like the orifice-compensated journal bearings with a hydrostatic load capacity in addition to the hydrodynamic load capacity. At zero speed when the hydrodynamic component is zero the spiral groove thrust bearing with its annular orifice has a hydrostatic load capacity which is a function of supply pressure.

The theoretical hydrostatic contribution to the net thrust load capacity is discussed in Appendix J. Predicted performance is shown in Figure 16 at constant pressure and compared to some experimental data that have been corrected for load required to center the shaft. Maintaining a constant annular orifice restriction, a performance map is made for the spiral groove thrust bearing which includes the hydrostatic component (see Volume I, Figure 75). For start and stop cycles, the hydrostatic component assures metal-to-metal separation, thereby eliminating startup wear and high friction torques.

C. First Experimental Program

The spiral groove thrust bearing has been used in all component and CRU tests during the period covered by this report. Specific thrust bearing component tests were conducted on several occasions as part of journal bearing evaluations.

Free Running Rig Test 21 (FRR 21) was one of the earliest tests in which thrust bearing performance was obtained. The absence of an axial capacitance probe prevented determination of film thickness, but the influence of speed and axial load on flow-pressure relationships was established.

The bearing tested had the following major dimensions:

Outside diameter	0.625 inch
Inside diameter	0.3125 inch

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$$\Delta P = 0.582 \times 10^{-6} \left[\frac{\lambda \frac{L}{2S} + 1.5}{\rho (D^2 - D_2^2)^2} \right] Q^2$$

S = Annular Orifice Radial Clearance

Diameter = 0.300-inch OD

Figure 15. Theoretical Flow Through a Concentric Annular Flow Restriction Versus Pressure Drop With Room Temperature Mercury

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Figure 16. Predicted Hydrostatic Load Capacity of Spiral-Groove Thrust Bearing

Total axial clearance 0.0012 inch

Annular orifice 0.0048 inch (diametral)

All other dimensions were essentially those shown in Figures 2 and 3.

The influence of shaft speed and axial load on flow-pressure characteristics is shown in Figures 17, 18, and 19. Figure 17 shows the effect of speed on pressure drop across the bearing for various constant total flows.

1. Correlation of Speed Influence with Predicted Performance

Without axial load the bearing is theoretically centered with a running clearance of 0.0006 inch on both sides. The performance parameters of this bearing for a clearance of 0.0006 inch have been computed (see Appendix J). At a shaft speed of 40,000 rpm, mercury at 70°F, $h_2 = 0.0006$, $E = 288$, $R = 10,000$, and $R^1 = 25,400$.

A high speed bearing such as this has friction losses with a resulting temperature rise of the mercury and reduced viscosity. Therefore, the power loss was estimated and found to be sufficient to raise the temperature about 15°F. The bearing parameters were then modified for viscosity of mercury at 85°F.

$$K = \frac{\mu_{85}}{\mu_{70}} = \frac{3.55}{3.65} = 0.975$$

With the new parameters, mercury is at 85°F, $h_2 = 0.0006$, $E = 286$, $R = 9,800$, and $R^1 = 24,900$.

Only E is a function of speed, and it is found by:

$$E_N = \frac{N}{40,000} E_{(40,000)} \quad \dots (2)$$

N, rpm	E	R	R ¹
0	0	9,800	24,900
10,000	71.5	9,800	24,900
20,000	143	9,800	24,900
30,000	215	9,800	24,900
40,000	286	9,800	24,900

Figure 17. Experimental Effect of Speed on Mercury Flow
For A Spiral-Groove Thrust Bearing

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Figure 18. Experimental Effect of Load on Mercury Flow for a Spiral-Groove Thrust Bearing With Thrust Applied Toward Alternator Bearing

A 10x10 grid of dots forming the letters 'L', 'E', 'S', 'S', 'E', 'S'.

Figure 19. Experimental Effect of Load on Mercury Flow for a Spiral-Groove Thrust Bearing with Thrust Load Applied Toward Turbine Bearing

Flow through the bearing clearance was calculated as before; it is shown in Table 1. Only two points were calculated for each speed because of linearity. This flow is plotted in Figure 20. The annular orifice curve for various supply pressures as tabulated in Table 2 is superimposed in this figure. The intersection of the curves are again the points of interest. These are plotted in Figure 21 and compared with the experimental data shown in Figure 17.

The correlation between theoretical and experimental lubricant flow curves is reasonably good. With increasing supply pressure, however, the difference becomes larger. This indicates additional losses in the bearing which seem to be functions of velocity head, $V^2/2g$, that had not been included in the analysis. Velocity head effects are thought to be due to sudden contraction losses as the fluid enters the running clearance from the annular orifice, as shown below:

The constant K_2 is a function of the amount of restriction.

TABLE 1

FLOW THROUGH CENTERED BEARING AT VARIOUS SPEEDS

N rpm	$(P_o - P_2)$ psi	J lb/sec-in.	Q lb/min
0	0	0	0
	200	0.089	7.8
10,000	0	0.0035	0.31
	200	0.0925	8.1
20,000	0	0.007	0.61
	200	0.096	8.4
30,000	0	0.0105	0.92
	200	0.100	8.75
40,000	0	0.0140	1.22
	200	0.103	9.0

TAC = 0.0012 inch
 $h_2 = 0.0006$ inch
 $C_D = 0.0048$ inch
 Mercury at 85°F
 $OD = 0.625$ inch
 $ID = 0.312$ inch

Figure 20. Mercury Flow Through Centered Spiral-Groove Thrust Bearing
 (Predicted)

TABLE 2

FLOW OF MERCURY THROUGH ANNULAR ORIFICE WITH VARIOUS SUPPLY
PRESSURES AND VARIOUS PRESSURE DROPS

P_s psig	$(P_o - P_2)$ psig	$P_s - (P_o - P_2)$ psig	Q lb/min
40	30	10	5.6
	20	20	7.6
60	40	20	7.6
	20	40	11.0
	10	50	12.0
80	60	20	7.6
	40	40	11.0
	20	60	13.3
100	80	20	7.6
	60	40	11.0
	40	60	13.3
	20	80	15.5
120	100	20	7.6
	80	40	11.0
	60	60	13.3
	40	80	15.5
	20	100	17.4
140	120	20	7.6
	100	40	11.0
	80	60	13.3
	60	80	15.5
	40	100	17.4
	20	120	19
160	140	20	7.6
	120	40	11.0
	100	60	13.3
	80	80	15.5
	60	100	17.4
	40	120	19
	20	140	21

Figure 21. Flow Characteristics of Spiral-Groove Thrust Bearing

TABLE 3

SPIRAL-GROOVE THRUST BEARING PERFORMANCE PARAMETERS

TAC = 0.0012 inch, C_D = 0.0048 inch, MERCURY AT 85°F, N = 40,000 RPM

h_2 , inch	E	R	R^1	Q lb/min				
				$P_o = 0$	50 psi	100 psi	150 psi	200 psi
0.0002	1830	56,700	670,000	0.46	0.58	0.70	0.815	0.93
0.0004	715	22,900	83,500	1.07	1.75	2.43	3.11	3.79
0.0006	287	9,770	24,800	1.24	3.20	5.15	7.1	9.02
0.0008	133	4,830	10,500	1.24	5.56	9.8	14.1	18.4
0.0010	126	2,670	7,380	1.89	8.75	15.6	22.3	29.2

2. Correlation of Influence of Axial Load with Predicted Performance

The experimental effect of axial load on flow-pressure characteristics is shown in Figures 18 and 19 for loads applied in both directions. Only minor deviations are observed when loading in either direction primarily because of small manufacturing variations. The spiral groove thrust bearing performance parameters were computed and are summarized in Table 3. Since experimental data on axial load (thrust capacity) versus minimum film thickness were not available, a graphical solution was developed which permitted determination of flow-pressure characteristics for various film thicknesses and supply pressures. These were then related to a given applied load. Table 3 (replotted as Figure 22). Table 4 (which represents the intersection points of Figure 22 and calculated P_1 values) and Table 5 (which represents calculated mercury flow and net thrust capacity) summarize the calculations. Figure 23 represents the performance map of the analyzed bearing showing net thrust and total flow as functions of running clearance for a range of constant supply pressures at 40,000 rpm. Figure 24 is the result of cross plotting to illustrate flow as a function of thrust load for constant supply pressure. Superimposed on Figure 24 are experimental results from Figure 18. Agreement is satisfactory over the pressure and flow range for which experimental data are available. No attempt should be made to obtain correlation at supply pressure of $P_s > 125$ psi by extrapolating the flow-pressure characteristics of Figure 18 as a straight line.

A cross plot of flow-pressure relationship at constant thrust levels from Figure 23 is shown in Figure 25 and compared to experimental results from Figure 18. Magnitude of flow agrees reasonably well with predicted theory although the experimental slope $\partial Q / \partial P$ is smaller than the theoretical value. This variation is assumed to be a function of the effects which were previously discussed. The design data must again be used with caution since agreement becomes poor beyond the range tested. The basic discrepancy appears to be a limitation of Whipple's solution which is good only for a low speed range or thrust bearing number.

3. Preliminary Conclusions

Several important conclusions were drawn from these early analytical studies and experimental results:

- 1) The dual face spiral groove thrust bearing, with lubricant flow to the outside (outward pumping action), will meet the load, flow, and pressure requirements specified.
- 2) Thrust load capacity was greater than that of the earlier pocket-land type bearing.
- 3) No evidence of cavitation-erosion was detected during early component or CRU tests.

Figure 22. Graphical Solution of Flow and Pressure Drop at 40,000 rpm for a Spiral-Groove Thrust Bearing

TABLE 4

INTERSECTION POINTS OF FIGURE 22 AND CALCULATED P_1 VALUES

P_s , psig	h_2 , inch	Q, lb/min	P_o , psi	P_1 , psi
50	0.0002	0.6	49.7	210
	0.0004	1.7	48.5	77
	0.0006	3.0	46	40
	0.0008	4.8	42	27
	0.0010	6.6	34	26
100	0.0002	0.7	99.6	251
	0.0004	2.2	97.7	106
	0.0006	4.8	92	64
	0.0008	7.7	78	45
	0.0010	10.0	64	42
150	0.0002	0.82	149.5	292
	0.0004	3.0	146.2	136
	0.0006	6.6	134	86
	0.0008	10.5	110	60
	0.0010	13.5	85	45
200	0.0002	1.0	199.3	334
	0.0004	3.7	195	166
	0.0006	8.2	176	108
	0.0008	13.0	140	74
	0.0010	16.5	106	57

TABLE 5

SPIRAL-GROOVE THRUST BEARING CALCULATED MERCURY FLOW AND THRUST CAPACITY

TAC = 0.0012 inch, $C_D = 0.0048$ inch, MERCURY AT 85°F, N = 40,000 RPM

h_{2a}	h_{2b}	P_s	Q_a	Q_b	Q_{a+b}	P_{oa}	P_{1a}	P_{ob}	P_{1b}	T_{net}
0.0002	0.0010	50	0.6	6.6	7.2	50	210	34	26	22.9
		100	0.7	10.0	10.7	100	251	64	42	27.3
		150	0.82	13.5	14.3	150	292	85	45	34.1
		200	1.0	16.5	17.5	199	334	106	57	39.7
0.0004	0.0008	50	1.7	4.8	6.5	48	77	42	27	6.3
		100	2.2	7.7	9.9	98	106	78	45	9.0
		150	3.0	10.5	13.5	146	136	110	60	11.7
		200	3.7	13.0	16.7	195	166	140	74	15.0

Figure 23. Theoretical Performance of Spiral-Groove Thrust Bearing at 40,000 rpm

OD = 0.625 inch
 ID = 0.312 inch
 $C_D = 0.0048$ inch
 TAC = 0.0012 inch

Figure 24. Load vs Flow for a Spiral-Groove Thrust Bearing at 40,000 rpm

Figure 25. Comparison of Experimental and Theoretical Mercury Flow as a Function of Supply Pressure for a Spiral-Groove Bearing at 40,000 rpm and Different Applied Loads

- 4) Supplying lubricant via an annular orifice at the inner diameter provides a useful hydrostatic load capacity in addition to the hydrodynamic component once rotation occurs.
- 5) The hydrostatic component can be predicted accurately by theory. For the bearing with an axial clearance of 0.001 inch, the hydrostatic capacity is approximately 17 lb at a supply pressure of 214 psig.
- 6) Lubricant flow-pressure characteristics can be predicted with reasonable accuracy. Between the supply pressures of 60 and 160 psig, the largest error was 16%.
- 7) An increase in speed causes an increase in lubricant flow (pressure decreases at constant flow). This has been demonstrated analytically and verified experimentally.
- 8) An increase in axial load causes an increase in lubricant flow (pressure decreases at constant flow). Minimum flow occurs at zero load, when the shaft is axially centered. This analytical prediction has been experimentally demonstrated.
- 9) Maximum applied loads of 35 lb at 40,000 rpm were successfully sustained although no film thickness measurements were made.
- 10) Spiral groove thrust bearing theory by Whipple can be used to predict performance with reasonable accuracy over the range of speeds, loads, and flows anticipated in this application.
- 11) Since theory is based on laminar flow in the clearance space, the reasonable agreement indicates little influence on load capacity and flow-pressure characteristics as a result of operation at $Re \approx 3000$.

D. Experimental Program to Establish Influence of Axial Clearance

The second major experimental program on the spiral groove thrust bearing on a component basis was conducted as part of a test series designated DBTR 4-1, -2, and -3. Specific dimensions of the single thrust bearing used for this series of three tests are shown in Figure 26. The dimensional variable was total axial clearance (TAC) which was changed as follows:

DBTR 4-1	0.0013 inch TAC	} Annular clearance =
DBTR 4-2	0.0011 inch TAC	
DBTR 4-3	0.0007 inch TAC	
		0.0039- to 0.004-inch diametral

All Dimensions in Inches

Figure 26. Dimensions of Spiral-Groove Test Thrust Bearing for DBTR 4-1, -2, and -3 Test Series

The test fixture used was the Dual Bearing Test Fixture modified to include a capacitance probe to measure axial displacement and minimum film thickness.

The experimental performance of the dual face spiral groove thrust bearing was established for a range of speeds (0 to 40,000 rpm), supply pressures (53 to 253 psi), and flows (3 to 16 lb/min), using ambient temperature mercury. The performance characteristics of interest included load versus minimum film thickness and flow-pressure relationship at speeds of 0; 10,000; 20,000; 30,000; 36,000; and 40,000 rpm and for supply pressures of 53; 103; 153; 203; and 253 psig. Experimental performance maps were compiled from the data; they are plotted in Figures 27 through 45. Any number of cross plots can be obtained from these maps since they cover the entire speed, flow, pressure, thrust load, film thickness, and total axial clearance range of interest. During an actual test only speed, flow, supply pressure, and built-in axial clearance are known to the operator. The operating film thickness, more important to him than actual thrust, can be obtained from the performance maps. Knowing total thrust, the film thickness can be established without difficulty.

Figures 27 through 45 represent the average performance measured by loading in both directions. Unfortunately manufacturing tolerances, of which the recess depth is most influential, prevent achievement of identical performances of both sides of the thrust bearing. In these three tests, the groove depth was 0.000575 inch (average for six grooves) on the side designated as the pump side and 0.00074 inch (average for six grooves) on the side designated as the turbine side. As a result, some performance variation resulted between the two sides, especially as the minimum film thickness became small. At larger film thicknesses, i. e., small applied loads, almost identical performance was measured for the two sides.

1. Correlation of Speed Influence with Predicted Performance

The design plots obtained with the aid of the Burroughs 205 digital computer had to be modified to take into consideration the smaller annular orifice (0.00395-inch diameter) of the test bearing. Direct comparison between earlier predicted and experimental performance and the data generated from the DBTR 4-1, -2, and -3 series cannot be made. Figures 46, 47 and 48 show experimental performance compared with predicted performance for 0.0013-, 0.00105-, and 0.0007-inch TAC, respectively, as functions of speed for different supply pressures at a minimum film thickness of 0.0002 inch. The figures clearly show the hydrostatic nature of the bearing at zero speed and the influence of supply pressure on load capacity. The hydrodynamic influence is fairly well pronounced at high speeds for the 0.0013- and 0.00105-inch clearance bearings because 0.0002 inch minimum film thickness represents a major displacement of the shaft, i. e., 70% and 60% respectively of the total. The 0.0002-inch film thickness of the 0.0007-inch clearance bearing represents only 40 percent of the possible shaft displacement. Therefore, the hydrodynamic influence is minor compared to the hydrostatic contribution. The linear relationship of thrust capacity with speed in Figure 48 verifies the hydrostatic tendency.

Figure 27. Performance Map of Spiral-Groove Thrust Bearing With a Total Axial Clearance of 0.0013 Inch at Zero Speed

DBTR 4-1 Test Series

Figure 28. Performance Map of Spiral-Groove Thrust Bearing with a Total Axial Clearance of 0.0013 Inch at 10,000 rpm

DBTR 4-1 Test Series

Figure 29. Performance Map of Spiral-Groove Thrust Bearing with a
 Total Axial Clearance of 0.0013 Inch at 20,000 rpm
 DBTR 4-1 Test Series

Figure 30. Performance Map of Spiral-Groove Thrust Bearing With a Total Axial Clearance of 0.0013 Inch at 26,000 rpm
DBTR 4-1 Test Series

Figure 31. Performance Map of Spiral-Groove Thrust Bearing with a Total Axial Clearance of 0.0013 Inch at 30,000 rpm

DBTR 4-1 Test Series

Figure 32. Performance Map of Spiral-Groove Thrust Bearing with a Total Axial Clearance of 0.0013 Inch at 36,000 rpm

DBTR 4-1 Test Series

Figure 33. Performance Map of Spiral-Groove Thrust Bearing With a Total Axial Clearance of 0.0013 Inch at 40,000 rpm

DBTR 4-1 Test Series

Figure 34. Performance Map of Spiral-Groove Thrust Bearing With a Total Axial Clearance of 0.00105 Inch at Zero Speed

DBTR 4-2 Test Series

Figure 35. Performance Map of Spiral-Groove Thrust Bearing With a Total Axial Clearance of 0.00105 Inch at 10,000 rpm

DBTR 4-2 Test Series

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Figure 36. Performance Map of Spiral-Groove Thrust Bearing With A Total Axial Clearance of 0.00105 Inch at 20,000 rpm

DBTR 4-2 Test Series

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Figure 37. Performance Map of Spiral-Groove Thrust Bearing With a Total Axial Clearance of 0.00105 Inch at 30,000 rpm

DBTR 4-2 Test Series

ID = 0.312 inch
 OD = 0.625 inch
 Annular Orifice = 0.00395 inch
 Ambient Mercury

Figure 38. Performance Map of Spiral-Groove Thrust Bearing With a Total Axial Clearance of 0.00105 Inch at 36,000 rpm

DBTR 4-2 Test Series

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ID = 0.312 inch
OD = 0.625 inch
Annular Orifice = 0.00395 inch
Ambient Mercury

Figure 39. Performance Map of Spiral-Groove Thrust Bearing With a Total Axial Clearance of 0.00105 Inch at 40,000 rpm

DBTR 4-2 Test Series

Figure 40. Performance Map of Spiral-Groove Thrust Bearing with a Total Axial Clearance of 0.0007 Inch at Zero Speed

DBTR 4-3 Test Series

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Figure 41. Performance Map of Spiral-Groove Thrust Bearing With a Total Axial Clearance of 0.0007 Inch at 10,000 rpm

DBTR 4-3 Test Series

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Figure 42. Performance Map of Spiral-Groove Thrust Bearing With a Total Axial Clearance of 0.0007 Inch at 20,000 rpm

DBTR 4-3 Test Series

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Figure 43. Performance Map of Spiral-Groove Thrust Bearing With a Total Axial Clearance of 0.0007 Inch at 30,000 rpm

DBTR 4-3 Test Series

Figure 44. Performance Map of Spiral-Groove Thrust Bearing With a Total Axial Clearance of 0.0007 Inch at 36,000 rpm

DBTR 4-3 Test Series

Figure 45. Performance Map of Spiral-Groove Thrust Bearing With a Total Axial Clearance of 0.0007 Inch at 40,000 rpm

DBTR 4-3 Test Series

Figure 46. Thrust Load Capacity vs Speed for a Spiral-Groove Thrust Bearing With a TAC of 0.0013 Inch

DBTR 4-1 Test Series

Figure 47. Thrust Load Capacity vs Speed for a Spiral-Groove Thrust Bearing With a TAC of 0.00105 Inch

DBTR 4-2 Test Series

Figure 48. Thrust Load Capacity vs Speed for a Spiral-Groove Thrust Bearing With a TAC of 0.0007 Inch

DBTR 4-3 Test Series

Agreement with predicted performance is good at zero; 10,000; and 40,000 rpm, but at 20,000 and 30,000 rpm the predictions are too optimistic. This is probably due to the smaller hydrodynamic component and the tendency to pump fluid out of the bearing, thereby creating a void or lower pressure at the exit of the annulus, resulting in lower hydrostatic performance. Figures 49, 50, and 51 present the influence of speed on total flow for the zero-load condition and at various constant supply pressures for $TAC = 0.0013$; 0.00105 ; and 0.0007 inch, respectively. Since total flow is minimum at zero load (when $h_{min} = TAC/2$, the flow will be greater for the loaded case. Previous performance maps can be used for these data. At constant pressure, flow becomes a fairly weak function of speed for the small TAC case. Predicted performance is superimposed on Figures 50 and 51; agreement is within 10 percent for the 0.00105 -inch clearance but is quite poor for the 0.0007 -inch clearance.

The influence of TAC on flow pressure characteristics for 36,000 and 40,000 rpm is shown in Figures 52 and 53 for the zero load case ($h_{min} = TAC/2$) and the $h_{min} = 0.0002$ inch case. These experimental data are again one method of presenting the performance of this type of bearing. Influence of load on performance can also be readily cross plotted from the performance maps. Data from Figure 53 are superimposed on Figure 14.

Since the actual thrust bearing must operate at a fixed flow rate and a design speed of 9 lb/min and 36,000 rpm, the experimental performance plot shown in Figure 54 may be most useful to the operator. These curves, shown for $TAC = 0.0007$; 0.00105 ; and 0.0013 inch, are quickly obtained from the performance maps, Figures 27 through 45. Performance at any other constant flow or pressure can also be obtained.

2. Film Stiffness

The film stiffness in the axial direction is generally discussed for a thrust bearing. The film stiffness obtained from the slope of the load-deflection curves (dW/dh) can lead to axial resonances just as the journal bearing may go through radial resonances if the thrust load, speed, and pressure result in a stiffness, K_{axial} which satisfies the equation $f_{cr} = \sqrt{K_a/M}$ where M is the mass of the rotor. The magnitude of stiffness is very sensitive to the thrust load as shown in the load-deflection curves. For zero load ($h_{min} = TAC/2$, the value of K_a is quite small since the slope W_T vs h_{min} is small. For large thrust loads where the value of $h_{min} \rightarrow 0$, the slope becomes very steep and $K_a \rightarrow \infty$. This behavior is identical to the hydrodynamic journal bearing. Since the spiral-groove thrust bearing is partially hydrostatic, the observed zero speed stiffness and the increase in stiffness with supply pressure is expected. To establish relative magnitudes of the axial film stiffness (K_a) Figures 55, 56, and 57 were obtained. The figures show the average axial stiffness between $h_{min} = 0.0002$ inch and $TAC/2$. These figures present the variation of the average axial film stiffness as a function of speed for the various supply pressures and for the three axial clearances. As with the hybrid journal bearing, the axial film stiffness for the spiral-groove thrust bearing increases as the clearance decreases for a fixed speed and supply pressure.

Figure 49. Flow vs Speed for a Spiral-Groove Thrust Bearing
 With a TAC of 0.0013 Inch

DBTR 4-1 Test Series

Figure 50. Flow vs Speed for a Spiral-Groove Thrust Bearing
With a TAC of 0.00105 Inch

DBTR 4-2 Test Series

Figure 51. Flow vs Speed for a Spiral-Groove Thrust Bearing with a TAC of 0.0007 Inch

DBTR 4-3 Test Series

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Figure 52. Pressure Drop Across Bearing vs Total Axial Clearance
For a Spiral-Groove Thrust Bearing at 36,000 rpm
(Experimental)

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Figure 53. Pressure Drop Across Bearing vs Total Axial Clearance
For A Spiral-Groove Thrust Bearing at 40,000 rpm
(Experimental)

Figure 54. Load and Supply Pressure vs Speed for a Spiral-Groove Thrust Bearing
For a Constant Flow of 9 lb/min and $T_{10} = 100^\circ\text{F}$

$C_D = 0.0030/0.0040$ inch
 Lubricant - Mercury
 (Inlet Temperature - 50°F)
 K_a is the Average Based on a
 Minimum Film from 0.00065
 to 0.0002 inch.
 Loading is on the Turbine Face
 of the Thrust Bearing.
 OD = 0.625 inch
 ID = 0.312 inch

Figure 55. Bearing Stiffness vs Speed for a Spiral-Groove Thrust Bearing at TAC = 0.0013 Inch

DBTR 4-1 Test Series

$C_D = 0.0039/0.0040$ inch
 Lubricant - Mercury
 (Inlet Temperature - 50°F)
 K_a is the Average Based on a
 Minimum Film from 0.000525
 to 0.00020 inch.
 Loading is on the Turbine Face
 of the Thrust Bearing.
 OD = 0.625 inch
 ID = 0.312 inch

Figure 56. Bearing Stiffness vs Speed for a Spiral-Groove
 Thrust Bearing at TAC = 0.00105 inch
 DBTR 4-2 Test Series

$C_D = 0.0039/0.0040$ inch
 Lubricant - Mercury
 (Inlet Temperature - 50°F)
 K_a is the Average Based on a
 Minimum Film from 0.000525
 to 0.00020 inch.
 Loading is on the Turbine Face
 on the Thrust Bearing.
 $OD = 0.625$ inch
 $ID = 0.312$ inch

Figure 57. Bearing Stiffness vs Speed for a Spiral-Groove
 Thrust Bearing at $TAC = 0.0007$ inch
 DBTR 4-3 Test Series

Since the axial film stiffness is near 5×10^4 lb/in. and since the rotor mass is 5 lb, it follows that the right combination of thrust load, pressure, and speed increase will result in an axial resonance below 36,000 rpm, where $f_{cr} = 18,800$ rpm.

3. Test and Condition of Hardware

During the DBTR 4-1 test, an overload at 40,000 rpm resulted in momentary contact on the pump (or alternator) side of the thrust bearing. At the time of the overload, the thrust load was 39 lb and the supply pressure was 103 psig. (Loads in excess of 39 lb were successfully supported at an equal pressure on the turbine side of the thrust bearing, indicating a higher load capacity in that direction.) The bearing appeared to recover completely, and the test was continued without interruption. Data were obtained with no problems at 40,000 rpm and 153 psig with loads of 42 lb. Some load variations were observed, however, and the flow decreased slightly, indicating that the resultant rub marks had created additional flow resistance.

Figure 58 shows the uncleaned, undamaged turbine side and Figure 59 the cleaned, damaged alternator side of the thrust bearing. The damage was very light but did reveal an approximate 0.0002-inch misalignment, confirming the inspection report which indicated that a 0.0002-inch total indicator reading initial misalignment existed. Consequently, all experimental data represent the actual performance of a spiral-groove thrust bearing with a 0.0002-inch TIR misalignment. The influence on flow-pressure relationship is small, but the misalignment can be expected to reduce the thrust load capacity.

No evidence of cavitation-erosion damage could be detected at the conclusion of this test series. Flow lines and some crud deposits were visible, but they did not appear to measurably influence load characteristics.

Prior to DBTR 4-2, the thrust bearing and runner were lapped to remove high spots. Since no measurable quantity of metal had been removed by the momentary rub, this cleaning up operation was considered sufficient even though all the rub marks were not eliminated. At the conclusion of DBTR 4-3 the condition of the turbine side of the thrust bearing was as shown in Figure 60, while the alternator side, Figure 61, still exhibited the rub marks suffered during DBTR 4-1. No additional rubbing occurred on this side during DBTR 4-2 or -3, even though loads as high as 58 lb were successfully supported at 40,000 rpm. This justified the assumption that the momentary contact and the light rub had little or no detrimental effect on the bearing's load capacity performance.

4. Conclusions on the Second Test Series

Based on the experimental performance measured during this test series and the analytical predictions, certain conclusions can be made about the dual-face, annular, orifice-compensated, spiral-groove thrust bearing.*

*These conclusions are made for an approximate misalignment of 0.0002-inch TIR.

Figure 58. View of Uncleaned and Undamaged Thrust Bearing Turbine Face
After DBTR 4-1

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Figure 59. View of Cleaned and Damaged Thrust Bearing Alternator Face After DBTR 4-1 Showing Score Marks

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Figure 60. View of Uncleaned Thrust Bearing Turbine Face
After DBTR 4-3

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Figure 61. View of Uncleaned Thrust Bearing Alternator Face
After DBTR 4-3 Showing Score Marks

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- 1) The spiral-groove thrust bearing with the dimensions shown in Figures 2 and 3 and with an assembled cold axial clearance of 0.001 inch is capable of meeting the specified SNAP 2 thrust conditions within the flow and pressure limitations of the over-all system.
- 2) The bearing supported a maximum thrust load of 68 lb at 40,000 rpm for the total axial clearance of 0.00105 inch.
- 3) The bearing has a hydrostatic (zero speed) load capacity because of the annular orifice restriction of 19 lb with ambient temperature mercury and a supply pressure of 203 psi for an axial clearance of 0.00105 inch. (This corresponds to approximately 12.5 lb/min flow.)
- 4) Increasing total axial clearance increases load capacity for the same supply pressure and film thickness (flow is increased). The load increase becomes asymptotic with increased clearance so that the increase is greater at a small total axial clearance than at a large one where the increase becomes almost negligible. For a film thickness of 0.0002 inch and a supply pressure of 153 psi at 40,000 rpm, the following loads can be supported with ambient temperature mercury:

<u>TAC</u>	<u>Thrust Capacity</u>	<u>Q</u>
0.0007 inch	17 lb	6.6 lb/min
0.00105 inch	25 lb	10 lb/min
0.0013 inch	28 lb	14.5 lb/min

- 5) The reverse holds true for the effect of total axial clearance on axial film stiffness. At 40,000 rpm with 153 psig supply pressure, the average axial film stiffnesses are:

<u>TAC</u>	<u>K_a</u>
0.0007 inch	92,000 lb/in.
0.00105 inch	65,000 lb/in.
0.0013 inch	62,000 lb/in.

The above behavior follows the anticipated trend of a hybrid bearing.

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- 6) The supply pressure has a large influence on thrust load capacity at low speeds, large total axial clearances, and relatively large film thicknesses. As speed increases and total axial clearance and minimum film thickness decrease, the influence of pressure on load capacity becomes small until it is negligible at $TAC = 0.0007$ and $N > 30,000$ rpm. In this respect the behavior of the spiral-groove thrust bearing is identical to that of the hybrid journal bearings.
- 7) The flow is minimum when thrust load is zero ($h_{min} = TAC/2$). This is not always true for different methods of orifice compensation.
- 8) Total axial clearance has a strong influence on flow through the bearing. For zero load, at 40,000 rpm and 153 psi, flow is given as:

<u>TAC</u>	<u>Q</u>
0.0007 inch	6.1 lb/min
0.00105 inch	8.2 lb/min
0.0013 inch	13.0 lb/min

- 9) Correlation between experimental load capacity and theoretical load prediction (based on laminar analysis) is good at zero and low speeds ($< 10,000$ rpm) and at high speeds ($> 30,000$ rpm). At intermediate speeds (10,000 to 30,000 rpm) the laminar analysis is optimistic since actual capacity is less than predicted. The influence of inertia and the pumping effect on reducing the hydrostatic component are assumed responsible for the variation. At high speeds (40,000 rpm), the bearing is predominantly hydrodynamic so that the influence of inertia and pumping effects is negligible.
- 10) The spiral-groove thrust bearing does not suffer cavitation-erosion type damage and is capable of continued successful operation after a momentary rub (breakdown in the fluid film lubrication) caused by an overload.

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III. CONCLUSIONS

The two test series on the double-acting spiral-groove thrust bearing demonstrated its ability to meet the specified loads applied during a normal CRU startup and shutdown as during design point operation. During the normal preheated startup the hydrostatic load capacity is adequate to support the anticipated transient loads. As the speed increases, the hydrodynamic capacity takes over to carry the imposed loads. Consequently, the spiral-groove thrust bearing utilized in the prescribed manner offers the advantage of hydrostatic performance during startup, when loads are higher than the hydrodynamic capacity, and hydrodynamic performance at design speed, which provides an increasing capacity as minimum film thickness become small.

Earlier CRU problems associated with the use of the spiral groove thrust bearing were confined to plugging of the annular orifice and the shallow grooves. The reason for this particular difficulty was traced to the choice of 18-4-1 tool steel (Bluechip). During shutdown a corrosion layer would build up making it difficult to restart. Selection of BG-41 or -42 (modified 440C stainless steel) overcame that problem.

No evidence of erosion and/or cavitation-erosion damage could be detected on the spiral-groove bearing during component or CRU tests providing assurance that life limitation was not a major concern.

Good alignment is essential to achieve maximum load capacity. Without it, the advantage of the hydrodynamic and hydrostatic performance cannot be realized. Good alignment, then represents a major requirement of this type of bearing. On the basis of the experimental component activity, alignment should be within 0.0002-inch TIR.

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