EVALUATION OF GAS LUBRICATED
HYDRODYNAMIC BEARINGS
IN A
GAS TURBINE ENVIRONMENT

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The Garrett Corporation
AEResearch Manufacturing Division

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This report covers the feasibility demonstration of gas lubricated hydrodynamic bearings in a gas turbine environment and covers the period from March 1971 to April 1972.

The specific type of compliant-foil gas bearings utilized in this demonstration program is a Garrett design covered by one or more of twelve U.S. patents, plus several more pending.

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Publication of this report does not constitute Air Force approval of the report's findings or conclusions. It is published only for the exchange and stimulation of ideas.

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This report contains a summary of the design analysis, development testing, and demonstration testing performed to demonstrate the feasibility of using gas lubricated hydrodynamic bearings in a gas turbine environment. The gas generator portion of the jet fuel starter used on the A7D aircraft was modified to incorporate compliant-foil gas bearings and subjected to varied phases of development testing and a 100 start plus 5 hr endurance demonstration test. In addition, engine starts were made under conditions which simulated normal starting gear loads. Results of the testing were very satisfactory. It is concluded that gas lubricated bearings can be utilized in a gas turbine application and further testing is encouraged.

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LIST OF SYMBOLS

\( g \) - acceleration due to gravity
\( I \) - polar moment of inertia, in.-lb-sec^2
\( K \) - bearing stiffness, lb/in.
\( N \) - rotational speed, rpm
\( N_c \) - shaft critical speed, rpm
\( P \) - pressure (units as noted)
\( T \) - gyroscopic moment
\( \omega \) - angular velocity, radian/sec
\( z \) - span between bearing extremes, in.
SECTION I
INTRODUCTION

The higher rotational speeds and increased temperatures required for upgrading the performance of the gas turbine engines has increased the demand for improved bearing designs and lubricating methods. Gas lubricated bearings offer a means of satisfying the demand by virtue of their independence from conventional lubrication requirements and their ability to cope with high speeds and temperatures. Gas lubricated bearings also have a significant cost and weight impact on gas generator design. Figures 1 and 2 show a comparison of conventional rolling contact systems versus foil gas bearing systems. Use of the latter results in both lighter and less expensive systems.

The program described herein was conducted to demonstrate the feasibility of a compliant-foil gas bearing concept, currently used successfully in air cycle cooling turbines, in the gas turbine engine environment.

To accomplish this, the gas generator portion of the jet fuel starter used on the A7D aircraft was modified to incorporate gas bearings in place of the existing rolling element bearings.

A demonstration test consisting of (1)-5 hours of endurance running, (2)-100 start-stop cycles and (3)-simulated geared starting was conducted. The successful completion of the test has demonstrated the feasibility of using gas bearings in gas turbine engines.
### ROLLING CONTACT VS FOIL GAS BEARING

#### A7D JFS GAS GENERATOR

<table>
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<tr>
<th>ITEM</th>
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<tr>
<td>ROTATING GROUP</td>
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<tr>
<td>HOUSINGS</td>
<td>1.0</td>
</tr>
<tr>
<td>SEALS</td>
<td>1.0</td>
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<tr>
<td>LUBE SYSTEM (OIL)</td>
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<tr>
<td>TOTAL UNIT</td>
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</tr>
</tbody>
</table>

*FOR CONTINUOUS DUTY MACHINES*

### WEIGHT COMPARISON

<table>
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<th>ITEM</th>
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</thead>
<tbody>
<tr>
<td>ROTATING GROUP</td>
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<tr>
<td>HOUSINGS</td>
<td>1.18</td>
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<tr>
<td>SEAL</td>
<td>0.25</td>
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<tr>
<td>LUBE SYSTEM (GAS)</td>
<td>0.02</td>
</tr>
<tr>
<td>TOTAL UNIT</td>
<td>0.68</td>
</tr>
</tbody>
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**FIGURE 1**
ROLLING CONTACT VS FOIL GAS BEARINGS
A7D JFS GAS GENERATOR

OPERATIONAL UNIT

MODIFIED UNIT

ROLLING ELEMENT BEARINGS

FOIL THRUST BEARINGS

FOIL JOURNAL BEARINGS

ITEM | COST FACTOR
--- | ---
BEARINGS | 1.0
SEALS | 1.0
SHAFTING | 1.0
HOUSING | 1.0
LUBE SYSTEM (OIL) | 1.0
BEARING FAILURE REPAIR | 1.0

COST COMPARISON

ITEM | COST FACTOR
--- | ---
BEARINGS | 0.25
SEALS | 0.20
SHAFTING | 0.20
HOUSING | 0.60
LUBE SYSTEM (GAS) | 0.05
BEARING FAILURE REPAIR | 0.30

FIGURE 2
SECTION II

SUMMARY

A program was conducted to demonstrate the feasibility of using compliant-foil gas bearings on small gas turbine engines. The gas generator from an A7D Jet Fuel Starter was modified to incorporate gas bearings, making it completely independent of an oil lubricating system.

Testing was divided into two phases: development and demonstration testing. Development testing included unfired cranking tests, bearing temperature rise tests, aerodynamic rotor thrust evaluations and cooling flow optimization. Demonstration testing included 135 start cycles and over 5 hrs of endurance running. In addition, simulated geared starting was demonstrated.

The successful completion of this program has demonstrated the feasibility and potential for utilizing the compliant-foil gas bearing concept in gas turbine engines.
SECTION III

TECHNICAL DISCUSSION

1. PROBLEM STATEMENT

The objective of this program was to demonstrate feasibility of a compliant foil hydrodynamic gas bearing system for use in the gas generator portion of auxiliary power units, missile/drone thrust engines, and jet fuel starters.

2. PROGRAM APPROACH

The approach to the feasibility demonstration consisted of applying existing technology to a small gas generator.

An existing small gas generator was selected and modified to replace the rolling element bearings with the compliant-foil gas bearing system. The gas bearing system was to be capable of withstanding a maneuver yaw rate of 3.5 rad/sec and a 5-g load in any direction while operating at maximum gas generator rotor speed.

The feasibility of utilizing compliant-foil gas bearings in a small gas generator was to be demonstrated by tests. These included: 5 hours continuous operation, 1 simulated geared start, and 100 start/stop cycles.

3. SELECTION OF DEMONSTRATOR SYSTEM

The vehicle selected for demonstration of the compliant-foil gas bearing system was the gas generator portion of the jet fuel starter used on the A7D Aircraft. Figure 3 is a photograph of the A7D Jet Fuel Starter.

The following parameters define the gas generator which was modified:

- Inlet airflow, lb/sec 1.7
- Pressure ratio 3.2:1
- Turbine inlet temperature, °F 1700
- Approximate weight, lb (including accessories) 40
- Length, in. 16
- Width, in. 10
- Height, in. 10
FIGURE 3. A7D JET FUEL STARTER

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Various design modifications were required to replace the rolling contact bearings in the existing jet fuel starter with compliant-foil gas bearings.

Since self-acting compliant-foil bearings require no externally supplied lubrication, the oil pressure and scavenge pumps in the accessory pack of the jet fuel starter were unnecessary (see Figure 4). The remaining functions of the accessory pack are PMG electrical power supply, fuel scheduling and electric starting. The PMG is superfluous in this application; fuel scheduling was supplied with a remote electronic fuel control, and starting was accomplished using the existing high pressure air impingement nozzles built into the first-stage turbine stator. Hence, the accessory pack was removed (Figure 4). To further simplify the demonstrator, the power turbine module was replaced with a thrust cone assembly. The thrust cone assembly was designed such that the pressures and temperatures present in the gas generator when the power turbine module was fitted could be duplicated, thus providing a basis of comparison with rolling contact bearing machines of the jet fuel starter type.

To achieve this goal, a computer model of the foil bearing demonstrator was assembled using "component maps" of proven accuracy from testing of existing jet fuel starter units. Since the aerodynamic components of the demonstrator are from a "stock" jet fuel starter, the only variable was thrust nozzle area. Figure 3 shows how a number of thermodynamic cycle parameters vary with changing nozzle area. The target effective nozzle area was 6.25 in.², yielding a turbine inlet temperature of 1660°F, and a projected static thrust level in the 90-95 lb range. The inner cone of the nozzle assembly is adjustable by means of 0.1 in. shim plates to vary the effective nozzle area.

Hardware changes to the gas generator module were kept to a minimum. A sheet metal housing was substituted for the aluminum one, only because it was easier to modify in the time span available.

The system of carbon-face seals, ball bearings, bevel gear, preload cage and spring and bearing spacer were replaced with a one-piece combination bearing journal shaft and thrust disk. Controlled-leakage labyrinth seals were fitted to both the turbine and compressor ends of the shaft to control bearing cooling airflow.

An eight-foil journal bearing was located at each end of the shaft and a twelve-foil thrust bearing rides against each thrust disk face. A new bearing support and compressor back-shroud were designed to locate and support the foil bearings (Figure 4).
FIGURE 5. SEA-LEVEL STANDARD-DAY PERFORMANCE CHARACTERISTICS,
FOIL BEARING DEMONSTRATOR
The instrumentation locations in the foil bearing demonstrator are shown on Figure 6. Thermodynamic cycle parameters to be studied included inlet air temperature and pressure, measured at the bellmouth; compressor discharge pressure and temperature, monitored at the diffuser exit; and turbine discharge temperature, measured in the exhaust cone. A comparison of the parameters with the computer model predictions was used to estimate the static thrust developed by the demonstrator.

Bearing performance parameters were sampled using proximity probes and thermocouples. Two thermocouples were installed on the foil back plate of each of the thrust bearings to monitor thrust bearing temperatures; and 4 thermocouples were installed in the bearing carriers of each journal bearing for monitoring radial bearing temperature.

Proximity probes, placed both axially and radially, were used to measure shaft excursions during transients, normal startup and shutdown, and during simulated geared starts.

The simulated geared start was accomplished by using an "unbalanced" impingement start nozzle system (see Figure 7). For starting normally, a "balanced" impingement start nozzle was used, as shown in Figure 8.

4. ANALYSIS

After selection of the gas generator, an analysis was initiated to determine the required bearing sizes. The selection of a bearing system was dependent on its capability to withstand all anticipated loads imposed by the rotating assembly.

The loads considered during the analysis included:

- Shaft criticals
- Rotor aerodynamic thrust
- Maneuver yaw rates up to 3.5 rad/sec
- Acceleration loads up to 5 g's in all directions

a. Analytical Approach

The approach to the analysis included the following steps:

(1) Journal Bearings

- Start with bearing sizes used on similar existing rotating assemblies (re: weight, bearing spans, speeds, etc.).
- Utilize bearing spring rates as determined empirically from existing designs.
FIGURE 7. UNBALANCED (SINGLE MANIFOLD) IMPINGEMENT NOZZLE STARTING SYSTEM
FIGURE 8. BALANCED (DUAL NOZZLE) IMPINGEMENT NOZZLE STARTING SYSTEM

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• Perform shaft critical speed versus bearing stiffness analyses.

• Determine optimum bearing size and locations based on results of critical speed analyses integrated with load factors for maneuver yaw rates, q-loads, and gear separating loads.

(2) Thrust Bearing

• Determine aerodynamic thrust on gas generator rotating assembly from test data previously obtained.

• Determine optimum size based on load factors for maneuver yaw rates, q-loads, and gear separating loads along with rotor aerodynamic thrust loads.

b. Analytical Results

(1) Critical Speeds and Resulting Bearing Loads

Rotor dynamics analyses were performed using computer models of the rotating assembly. The results of the final analyses were:

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<th>Mass and Inertia Properties</th>
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<tr>
<td>Rotor mass, in.-sec²/lb</td>
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<tr>
<td>Rotor weight, lb</td>
</tr>
<tr>
<td>Polar moment of inertia, in.-lb-sec²</td>
</tr>
<tr>
<td>Diametral moment of inertia, in.-lb-sec²</td>
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<table>
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<tr>
<th>Critical Speeds Versus Bearing Stiffness</th>
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<tr>
<td>Bearing Stiffness</td>
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</tr>
<tr>
<td>Kₜ, lb/in.</td>
</tr>
<tr>
<td>4000</td>
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<td>5000</td>
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An illustration of absolute bearing load versus shaft speed for a bearing stiffness of 5000 lb/in. is given in Figure 9.

(2) Maneuver Yaw

The bearing system must withstand gyroscopic loads resulting from yaw maneuvers of up to 3.5 rad/sec.

Total gyroscopic moment is equal to

\[ T = I_p \omega_r \omega_p \]

where

- \( I_p \) = polar moment of inertia, in.-lb-sec^2
- \( \omega_r \) = rotor angular velocity, rad/sec
- \( \omega_p \) = angular velocity of precession, rad/sec

Figure 10 illustrates the relationship between gyroscopic moment and gas generator shaft rotational speed for an angular velocity of precession \( (\omega_p) \) 3.5 rad/sec. From Page 14, it was shown that \( I_p = 0.013 \) in.-lb-sec^2. Based on test results for similar bearing systems, the thrust bearing absorbs up to 75-percent of the total gyroscopic moment with the journal bearing absorbing the remaining 25-percent. However, for design purposes, a 70-, 30-percent moment split was assumed.

(3) Maneuver Yaw Loads on Journal Bearings

The maximum gyroscopic moment occurs at 100-percent rotational speed and is equal to

\[ T = I_p \omega_r \omega_p \]

\[ T = (0.013) (72,506) (\text{3.5}) \]

\[ = 345.4 \text{ lb-in.} \]

Gyroscopic moment \( (T_y) \) on the journal bearings is

\[ T_y = 0.30 \times T \]

\[ = 103.62 \text{ lb-in.} \]
NOTES:
1. $K_1 = K_2 = 5000 \text{ LB/IN.}$
2. C.G. ECCENTRICITY = 0.0005 IN.

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FIGURE 3. BEARING LOAD VERSUS SHAFT SPEED, FOIL BEARING DEMONSTRATOR
FIGURE 10. BEARING LOAD VS GAS GENERATOR SHAFT SPEED FOR A 3.5 RAD/SEC MANEUVER
Load due gyroscopic moment on each journal bearing is

\[ T_g/T = 103.62/5.4 \]
\[ = 19.19 \text{ lb} \]

Where \( Z \) = span between bearing extremes.

(4) Maneuver Yaw Loads on Thrust Bearing

Gyroscopic moment \( (T_g) \) on the thrust bearing is

\[ T_g = 0.70 \times (T) \]
\[ = 0.70 \times (345.4) \]
\[ = 241.78 \text{ lb-in.} \]

Moment carried by each thrust bearing is 120.89 lb-in. Since the average radius of the journal bearing is 1.409 in., the load on each bearing, at full speed, is:

\[ \frac{120.89 \text{ in.-lb}}{1.409 \text{ in.}} = 85.5 \text{ lb} \]

(5) Load Due to Acceleration

Acceleration load = weight \( \times \) acceleration

Weight = 4.63 lb

Acceleration = 5 g's

Load = 4.63 \( \times \) 5

Load per journal = 23.15/2 = 11.57 lb (when applied normal to gas generator centerline)

Load on thrust bearing = 23.15 lb (when applied parallel to gas generator centerline)

(6) Load Due to Gear Separating Forces

In the jet fuel starter gas generator, a bevel gear set is used for starting and to drive gas generator accessories (oil pump, fuel pump, PMG, etc.). This gear set is typically found in nearly all gas generators currently in use.
To completely describe all loads which might be imposed on the bearings, the separating forces at this gear mesh were determined. The loads due to separating forces were considered at two operating conditions: (1) gas generator starting and (2) at full-speed operation. These loads are illustrated graphically in Figure 11.

(7) Aerodynamic Thrust Loads

Aerodynamic thrust loads were derived from test data correlated with a computer model of the gas generator spool. Figure 12 illustrates the aerodynamic thrust versus speed for the gas generator.

(8) Summed Bearing Loads

The total load for each bearing was derived by summing loads from each possible source (gear separating forces, maneuver yaw of 3.5 rad/sec, 5-g acceleration, aerodynamic thrust, and shaft dynamic loads). Figure 13 illustrates the total load versus speed for one thrust bearing and one journal bearing.

(9) Bearing Sizing

The bearing sizes were based on the maximum total load which the bearings might encounter, as shown in Figure 13. Shaft system criticals are of no concern in sizing the bearings because they occur early in the start sequence and very little time is spent at those speeds.

5. BEARING SELECTION

Based on the results of the analyses described in Section 4b, it was possible to select both thrust and journal bearing sizes from existing designs.

a. Thrust Bearing

The thrust bearing selected has a load area of 6.6 in.$^2$ and had previously demonstrated load capacities above that required for the gas generator application. Figure 14 shows details of the thrust bearing assembly.

b. Journal Bearing

The journal bearing selected has a load area of 2.4 in.$^2$ and has previously demonstrated load capacities in the range required for the gas generator application. Figure 15 shows details of the assembly.

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FIGURE 11. BEARING LOAD DUE TO GEAR SEPARATING FORCES VERSUS SPEED DURING START ACCELERATION
FIGURE 12. AERODYNAMIC THRUST VERSUS ROTOR SPEED FOR FOIL BEARING DEMONSTRATOR
FIGURE 13. MAXIMUM BEARING LOAD VERSUS GAS GENERATOR SHAFT SPEED
CONCEPTUAL DRAWING ONLY;
DIMENSIONS ARE APPROXIMATE
AND NOT NECESSARILY TO SCALE

FIGURE 14. COMPLIANT-FOIL AIR BEARING (THRUST) DETAILS

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FIGURE 15. COMPLIANT-FOIL AIR BEARING (JOURNAL) DETAILS
6. BEARING RIG TESTING

Both the journal and thrust bearings were subjected to verification testing before actual gas generator testing commenced.

a. Journal Bearing Rig

A schematic of the journal bearing test rig is shown on Figure 16. The test bearing was mounted on a variable speed shaft with a test bearing carrier around it. Slave bearings were mounted on each end of the shaft for support. Torque was measured by means of a strain gauge on the torque blade, fitting into the pegs on the side of the carrier. Side loads were applied to the test bearing by means of a piston-cylinder device. Air pressure in the lower cylinder cavity pushes the piston up against the bearing carrier, and high-pressure air is supplied between the piston and the carrier to produce an air film between the two surfaces. This allows frictionless loading of the test bearing without interfering with torque measurement. A photograph of the test rig is shown in Figure 17.

The rig tests were conducted to verify the predicted bearing performance and thus insure the bearing suitability for the gas generator application. The results of the journal bearing testing are shown in Figure 18.

b. Thrust Bearing Rig

Thrust bearing testing was also conducted for performance verification. Pictured in Figures 19 and 20 are the thrust bearing test rig and instrument panel. Loads were applied to the test bearing by means of a piston-cylinder device. A high-pressure air bearing was used to prevent surface contact between the piston and the thrust bearing. Torque was measured by adding weights to a spring balance device connected to the torque arm. A pulley-supported on air bearings was used to prevent any error in torque measurements due to friction.

The thrust bearing performance was verified to 60,000 rpm, confirming that the bearing had sufficient load capacity to be utilized in the gas generator. The results of the testing are presented in Figure 21.
FIGURE 16. COMPLIANT-FOIL AIR JOURNAL BEARING TEST RIG
FIGURE 18. JOURNAL BEARING RIG TEST RESULTS, FOIL BEARING DEMONSTRATOR
FIGURE 19. COMPLIANT-FOIL AIR THRUST BEARING TEST RIG
FIGURE 20. COMPLIANT-FOIL AIR THRUST BEARING TEST RIG CONTROL PANEL

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FIGURE 21. THRUST BEARING RIG TEST RESULTS FOIL BEARING DEMONSTRATOR
7. CONTROL SYSTEM

The control system for the foil bearing demonstrator was designed to provide the following functions:

- Initiate engine starting and lightoff
- Control rotor acceleration
- Prevent excessive turbine inlet temperature
- Prevent compressor surge
- Prevent overspeed
- Provide steady-state operation control with provisions for minor speed changes
- Shut down engine and drain fuel manifold

These functions were performed by a remotely driven fuel pump and an electronic control system.

Modification to the gas generator precluded use of the "stock" jet fuel starter control system. Therefore, a breadboard control system was utilized. This system was powered from test facility energy sources and utilized rotor speed and compressor discharge pressure for input signals. A schematic of the control system is given in Figure 22.

8. GAS GENERATOR TESTING

The testing of the modified gas generator was divided into two phases. The first phase, development testing, consisted of preliminary tests to verify the effectiveness of the modifications to the basic design and to make adjustments as required. The second phase consisted of a demonstration test.

a. Development Testing

A series of preliminary tests was conducted prior to initiating the demonstration test. These tests were designed to check out and calibrate various subsystems and components. Specifically, the development testing included:

- Instrumentation calibration
- Unfired cranking tests
- Bearing temperature rise testing (during extended cranking)
- Start testing
- Automatic breadboard control system checkout
- Aerodynamic thrust evaluation
- Bearing cooling flow optimization

(1) Instrumentation

The instrumentation utilized to monitor bearing and gas generator performance included (see Figure 6):
FIGURE 22. FOIL BEARING DEMONSTRATOR CONTROL SYSTEM SCHEMATIC
Calibrated inlet flow measuring section
- Compressor discharge total pressure probes
- Fuel flow meter
- Turbine discharge temperature thermocouples
- Bearing temperatures thermocouples
- Turbine disk static pressure taps
- Impeller disk static pressure taps
- Rotor axial clearance capacitance probes
- Rotor radial clearance capacitance probes
- Rotor speed monopoles

(2) Cranking Test
The first series of tests was directed toward establishing the impingement air start system performance. Impingement air pressure was varied from 200 psig through 900 psig. It was determined that the minimum pressure required to initiate rotation was 300 psig and the maximum cranking speed attainable with 900 psig was 30,600 rpm. The minimum required start assist speed for the gas generator is 16,000 to 17,000 rpm which is attainable with an impingement air pressure of 425 psig. Figure 23 shows the test data points plotted on the impingement start nozzle performance curve.

(3) Bearing Temperature Rise Testing
The second series of testing consisted of extended cranking, with no ignition, to determine bearing temperature variation with time. The purpose of this testing was to insure that bearings were aligned and provided with adequate cooling air. An illustration of the test results is given in Figure 24.

(4) Start Testing and Control System Checkout
Initial start testing was conducted to (1) monitor bearing temperature levels during hot operation, and (2) to checkout the automatic breadboard control system operation.

After approximately 50 seconds of run time, during the second start, a rapid bearing temperature rise, along with large shaft excursions, was experienced. The run was terminated.
**Figure 23.** Cranking Characteristics of Foil Bearing Demonstrator Using Impingement Air Nozzles

1. **Typical JFS100-13A Jet Fuel Starter Drag**
   - (Rolling Contact Bearings
     - Oil Pump, Fuel Pump, Etc.)

2. **Base Compressor Pumping Drag (No Help From Turbine)**

3. **Normal Lightoff Speed**

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FIGURE 24. FOIL BEARING DEMONSTRATOR BEARING TEMPERATURE VS TIME DURING COULD START
Subsequent inspection revealed that there was insufficient prestretch on the rotating group tie-bolt. This, along with thermal expansion of the tie bolt, caused a loosening of the clamping load on the rotating group. This allowed the rotating components to shift, resulting in an increased unbalance in the rotating group until failure occurred. As is characteristic with foil bearings, secondary damage was minimal.

The shaft bearing surfaces were refinshed and the bearings were replaced. The gas generator was reassembled for continued development testing.

(5) Aerodynamic Thrust Evaluation

Early in the development testing program, a thrust bearing failure occurred due to rotor thrust overloading. Figure 25 shows a condensed plot of the recording of an axial clearance probe through the duration of the run. Failure occurred at shutdown where a sharp rise in rotor thrust is normally experienced as the turbine is "unloaded". Using thrust bearing static spring rate curves, it was possible to calculate the forward thrust on the rotor. This thrust was approximately 160 lb, which would be sufficient for bearing overloading. Analytical thrust calculations confirmed the observed high rotor thrust values. The principle reason for the high load was that the bearing thrust runner had higher pressures on the turbine side than on the compressor side.

To alleviate the differential pressure condition, two changes were made to the system. Axial holes were drilled in the thrust runner (refer to Figure 26) to equalize the pressure on both faces. Also a seal was installed between the thrust runner and the impeller (Figure 26) to restrict the airflow and maintain constant pressure in the forward thrust bearing cavity.

Several static pressure probes were installed at appropriate locations throughout the unit to collect data for an accurate calculation of the aerodynamic rotor thrust for the new configuration, (see Figure 27). It was found that the modifications reduced the rotor thrust to about 25 lb, which is near the value used for the bearing design analysis.

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EVENT
1 - 2 IMPELLING AIR ON, NO ROTATION
2 - 3 IMPELLING AIR ON, ROTATION
3 - 4 LIGHT-OFF AND ACCELERATION
4 - 5 FULL GOVERNED SPEED RUN
5 SHUT DOWN

FIGURE 25. THRUST BEARING RUNNING CLEARANCE HISTORY FOR FOIL BEARING DEMONSTRATOR TEST
1/12/72
FIGURE 26. DESIGN MODIFICATIONS MADE TO FOIL BEARING DEMONSTRATOR
FIGURE 27. LOCATION OF STATIC PRESSURE PROBES ON FOIL BEARING DEMONSTRATOR
(6) Cooling Flow Optimization

As a result of rotor cranking and runs of short duration, it was evident that the bearing temperatures were rising to values higher than expected. In an effort to decrease the steady-state temperature of the bearings to 500°F, a cooling flow optimization study was undertaken. The unit was bench tested at zero speed utilizing external air to duplicate the pressures normally existing in the gas generator during operation. Seal clearances and cooling passage metering orifices were examined to determine their effect on cooling flow distribution. Orifice sizes were changed as a result of these tests and no further cooling problems were encountered.

b. Demonstration Testing

After the modifications were made as a result of the development testing, the demonstration testing was initiated. The total number of starts accumulated during the demonstration testing was 135. Data from all the starts were recorded, both manually and on Sanborn recorders. The list of parameters recorded is as follows:

- Speed
- Compressor discharge pressure
- Thrust axial bearing temperature
- Turbine journal bearing temperature
- Compressor journal bearing temperature
- Turbine discharge temperature
- Compressor discharge temperature
- Axial clearance probe

Figure 28 shows recordings of start numbers 6 and 135. These starts were typical and demonstrate the consistency of the start profiles.

The rotating group could be rotated freely in the bearings at the conclusion of all testing. Several starts were repeated because the unit experienced blow-out after reaching governed speed. The fuel control was modified after start number 35 and no subsequent blow-out problems occurred.

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SANBORN TRACES OF 6TH AND 135TH START CYCLES

Figure 28
Start number 103 was the initiation of the 5-hr endurance demonstration. Figure 29 shows the recording trace at the initiation of the 5-hr run and Figure 32 shows the recording trace at the conclusion.

Radial clearance probes installed 90-deg apart were utilized to monitor shaft excursion throughout the demonstration tests. The output of the probes was displayed as a miscellaneous pattern. The diameter of the circle indicated a shaft excursion of only 0.0005 in., which would result from an unbalance eccentricity of 0.00025 in.

Photographs of the demonstrator in the test configuration are shown in Figures 30 and 31.

The normal starting system for a small gas generator consists of a start motor coupled to the compressor/turbine spool via a bevel gear and tower shaft system. During a start, the bevel gears impart side and end loads to the rotating group. The necessary modification to the gas generator to incorporate foil gas bearings precluded the use of the geared system, therefore, an unbalanced high pressure air impingement nozzle arrangement was used.

The corrected thermodynamic performance of the unit was very close to that predicted. Figure 32 shows a number of performance parameters plotted as a function of effective thrust nozzle area for the inlet conditions in the test cell during the 5-hr endurance test. The gas generator speed was limited to 67,800 rpm because impeller blade interference analyses showed that the dual monopole struts in the instrumented bellmouth would cause resonant interference if the unit were run at 72,000 rpm. In an actual installation, however, a single monopole speed sensing system would be used, and the unit could be run at 72,000 rpm with no problem.

Comparing the observed through flow, compressor pressure ratio and exit temperature, and turbine discharge temperatures of 1.5 lb/sec, 3:1:1, 315°F, and 1360°F, respectively with the characteristics shown on the curve, an effective nozzle area of approximately 6.1 in.² is indicated. Entering Figure 5, (see Section 3, SELECTION OF DEMONSTRATOR SYSTEM) with this nozzle area, a sea-level standard-day thrust rating of 95 lb is indicated. The target range was 90-95 lb.

After the demonstration testing, the unit was disassembled for inspection. The inspection revealed the unit to be in excellent condition.
FIGURE 30. THREE-FOURTHS FRONT VIEW OF FOIL BEARING DEMONSTRATOR IN TEST CONFIGURATION

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FIGURE 31. THREE-FOURTHS REAR VIEW OF FOIL BEARING DEMONSTRATOR IN TEST CONFIGURATION
FIGURE 32. 5-HR ENDURANCE TEST, PERFORMANCE CHARACTERISTICS, FOIL BEARING DEMONSTRATOR
Teardown and inspection revealed no discrepancies with either the bearing system or the gas generator as a whole. There were no scratches, grooves, or deposits on any of the running surfaces.

Aside from normal burnish spots on the thrust bearings, no distressed areas were noted.

A minor amount of fretting was evident where the trailing edge of each journal-foil contacted its neighbor, but no coating chipping or peeling was evident, and bearing performance was not affected.
SECTION IV
CONCLUSIONS AND RECOMMENDATIONS

Compliant-foil gas bearings were successfully demonstrated in a gas turbine application. The bearings operated successfully in the high temperature environment of a gas turbine and withstood the aerodynamic thrust and critical speed loads that are present in gas turbines.

It is recommended that additional testing be conducted where the bearings can be subjected to more severe operating conditions typical of normal aircraft operation. This testing should include the following items:

- Establish the design load carrying capability of compliant-foil gas bearings; specifically, loadings due to aircraft yaw maneuvers and accelerations in all directions.
- Establish the degree to which the compliant-foil bearings are sensitive to normal aircraft induced vibrations.
- Establish the degree to which the compliant-foil bearings are sensitive to the ingestion of water and/or dust and dirt.
- Establish the effects of bearing overtemperature, as a result of insufficient cooling, on bearing performance.

The results of the gas foil bearing performance when subjected to the above conditions will prove its general feasibility for the small gas turbine application.

Since gas foil bearings have been restricted thus far to relatively small rotors, there may be problems in projecting the design criteria to the larger rotor systems. It is recommended that the compliant-foil gas bearings be evaluated for larger engines to broaden their application and potential to a larger percentage of engines in service today and in future designs.

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EVALUATION OF GAS LUBRICATED HYDRODYNAMIC BEARINGS IN A GAS TURBINE ENVIRONMENT

Final Report (March 1971 to April 1972)

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Heuer, Don F.

Approved for public release; distribution unlimited.

This report contains a summary of the design analysis, development testing, and demonstration testing performed to demonstrate the feasibility of using gas lubricated hydrodynamic bearings in a gas turbine environment. The gas generator portion of the jet fuel starter used on the A7D aircraft was modified to incorporate compliant-foil gas bearings and subjected to varied demonstration tests. In addition, engine stands were made under conditions which simulated normal starting gear loads. Results of the testing were very satisfactory. It is concluded that gas lubricated bearings can be utilized in a gas turbine application and further testing is encouraged.
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