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DYNAMIC AND ENVIRONMENTAL EVALUATION OF COMPLAINT FOIL GAS LUBRICATED BEARING

D. E. Heuer, et al

Airesearch Manufacturing Company of Arizona

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13. ABSTRACT
This report summarizes the work accomplished under USAF Contract F33615-72-C-1802 by the AiResearch Manufacturing Company of Arizona, during the period from April 1972 to May 1973. A series of demonstration tests, similar to those required for engine qualification, were conducted on a gas turbine engine utilizing gas lubricated compliant foil bearings.

The engine operated successfully throughout the water ingestion and dust ingestion tests. When subjected to high-yaw maneuver rates, which subjected the bearings to high gyroscopic moment loads, the ultimate load capacity of the bearings was reached. When subjected to high vibration, structural resonance was incurred which amplified the input vibration at the bearing carrier. The forward journal bearing did not survive the test but its vibrational loading was 5 times what was originally intended. Results of the overtemperature test indicated that the present bearing coating is acceptable up to approximately 550°F, but higher temperature coatings would be required if the bearings should be installed in new, advanced-design engines.

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Compliant Bearings						
Gas Lubricated Compliant Bearings						
Gas Lubricated Foil Bearings						
Hydrodynamic Gas Lubrication						
Turbine Engine Lubrication						

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D.F. HEUER
R.A. COLLINS

AIRESEARCH MANUFACTURING COMPANY OF ARIZONA

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FOREWORD

This report was prepared by The AiResearch Manufacturing Company of Arizona, A Division of The Garrett Corporation, Phoenix, Arizona. The activities discussed in this report were initiated under United States Air Force Contract F33615-72-C-1802. The contract was administered by the Aero Propulsion Laboratory, Aeronautical Systems Division, Air Force Systems Command, Wright-Patterson Air Force Base, Ohio with Mr. E.A. Lake acting as Project Engineer.

A series of demonstration tests, similar to those required for military engine qualification, were conducted on a gas turbine engine utilizing gas lubricated compliant foil bearings. The tests were conducted during the time period from April 1972 to May 1973.

This report is assigned a supplementary report number, 73-310169, by AiResearch Manufacturing Company.

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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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Chief, Fuels and Lubrication Division
Air Force Aero Propulsion Laboratory

ABSTRACT

Using the foil bearing demonstrator, which is a small, 95-pound thrust turbojet engine developed from an existing jet fuel starter under U.S. Air Force Contract F33615-71-C-1376 and described in Technical Report AFAPL-TR-72-41, a series of evaluation tests was conducted to demonstrate that a gas turbine engine, mounted on gas-lubricated, compliant foil bearings, has the potential for qualification in a military application.

The engine operated successfully throughout the water ingestion and dust ingestion tests.

When the engine was subjected to high-yaw maneuver rates, which subject the bearings to high gyroscopic moment loads, the ultimate load capacity of the bearings was reached, indicating additional development to increase the load-carrying capability of the bearings may be desirable.

When the engine was subjected to high vibrational loading, a structural resonance was incurred which amplified the input vibration at the bearing carrier. The forward journal bearing did not survive the test but its vibrational loading was 5 times what was originally intended. The ability of the foil bearings to withstand vibrational loading is dependent upon the engine structural hardware, which must isolate the external vibration from the bearing carriers. This phenomena becomes peculiar to the individual engine design.

Results of the overtemperature test indicate that the present bearing coating is acceptable up to approximately 550°F, but higher temperature coatings would be required if the bearings should be installed in new, advanced-design engines, which would have such high pressure ratios that the compressor discharge temperature would be hotter than the coating temperature limitation.

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SECTION I

INTRODUCTION

Gas-lubricated foil bearings show the potential for providing the advantages of simplicity, reduced maintenance, added reliability, reduced vulnerability, high-temperature operation, reduced weight, and cost savings to an advanced engine system.

1. ADVANTAGES

a. Simplicity

Simplicity of gas-lubricated bearings is effected by the absence of a complicated delivery and scavenge oil network, especially critical at the hot end journal bearing where oil crosses the engine exhaust path upon entering and leaving the bearing cavity. In this area, extensive development is sometimes necessary to prevent high oil temperature during soakback, which often reduces engine reliability and time-between-overhaul.

b. Reduced Maintenance and Added Reliability

The use of gas bearings will eliminate the possibility of oil contaminating the bleed-air supply (environmental control system) or the engine exhaust, as a result of oil seal leakage.

The endurance capability of the foil bearings has been proven by foil bearing-equipped production DC-10 cooling turbines which have accumulated more than 650,000 hours in scheduled airline service in more than 50 aircraft with a mean-time-between-unscheduled-removal (MTBUR) of 90,000 operating hours. The longest-mean-time unit has completed more than 8000 hours, with the number of starts estimated to be over 2000, conducted over a wide range of ambient temperature conditions.

c. Reduced Vulnerability

Since the foil bearings do not require external engine hardware, such as oil lines and an oil heat exchanger, the vulnerability of the engine is reduced.

d. High-Temperature Operation

At present, limitations on foil bearing temperature have been due to foil coating. Advancements in foil coating temperature tolerance will be achieved in time, as research on the problem continues. Because the air viscosity increases with temperature, a hotter-running bearing will provide higher load capacity, thus making high temperature operation desirable.

e. Reduced Weight

A weight saving can be effected by the reduction in oil capacity required by a system incorporating foil bearings, assuming that an engine gearbox is still required. The reduced oil capacity, in turn, would reduce capacity requirements for the oil heat exchanger and the oil pump. Elimination of oil delivery and scavenge networks results in a less complex, more uniform, and a lighter weight assembly. The weight saving, however, is compromised by the requirement for larger journal bearings and the addition of a thrust runner on the rotor for the thrust bearing.

f. Cost Saving

The inherent simplicity of the foil bearing lends itself to rapid and easy fabrication. After an initial investment in tooling, foils can be stamped or rolled out at low cost. The greatest cost saving is realized by elimination of the oil network.

2. BACKGROUND

a. Development Program

The gas-lubricated foil bearings utilized for this program are of a design patented by The Garrett Corporation. The bearings were initially developed for experimental use in cooling turbines of the air conditioning system onboard the Boeing 727 airliner.

Observing this successful application to the cooling turbines and recognizing the advantages foil bearings offer to advanced military engines, the U.S. Military sponsored a feasibility study of the application of gas-lubricated, compliant foil bearings to gas turbine engines. This initial program has been completed and is described in Technical Report AFAPL-TR-72-41, dated June 1972.

During this program, the foil bearing demonstrator was designed and fabricated from the gas generator of an AiResearch Model JFS100-13A Jet Fuel Starter, which is used on the Air Force A7D light-attack aircraft. Modifications to the original production equipment included replacing the rolling element bearings with compliant foil gas bearings and replacing the power turbine module with an exhaust thrust nozzle. The resultant engine is a 35-pound-static-thrust turbojet, mounted on compliant foil gas bearings, totally free of oil and external air* supply, and operating at a design speed of 72,000 rpm.

The demonstrator was operated continuously for five hours and underwent 135 start/stop cycles to full speed, as required by the demonstration contract.

b. <u>Evaluation Program</u>

This program was an extension to the development program. The evaluation program consisted of five tests, four of which are often required for engine qualification. These tests, listed below, were conducted to evaluate the potential of utilizing foil gas bearings on engines that must undergo this type of qualification testing.

1) Engine water ingestion

2) Engine dust ingestion

3) Engine maneuver (yaw and acceleration)

4) Engine vibration

5) Bearing overtemperature

This report provides a short description of the foil bearing demonstrator and the several configurations used during the test sequence. A description of each test is provided, along with the conclusions drawn from the test data. Recommendations are included on those areas of the bearings that could benefit from further development.

*An external impingement air supply is used to start the engine, but the bearings do not rely on this air for starting. After lightoff occurs, this external air is shut down.

SECTION II

PROGRAM DESCRIPTION

The program objective was to demonstrate the potential of a small gas turbine engine equipped with compliant foil gas bearings for qualification in military installations.

1. ENGINE DEVELOPMENT

The engine, called the foil bearing demonstrator, was designed and fabricated from the gas generator of an AiResearch Model JFS100-13A Jet Fuel Starter, the rolling element bearings being replaced by compliant foil gas bearings. The development of the engine and gas bearings was accomplished in a previous program, under U.S. Air Force Contract F33615-71-C-1376, and is described in Technical Report AFAPL-TR-72-41.

2. TEST PROGRAM DEVELOPMENT

A test program was established to determine the effect that environmental and loading conditions would have on the bearings during engine qualification. The program consisted of the following tests, each selected to evaluate potential areas of bearing vulnerability.

Test	Procedure
Water Ingestion	Water sprayed into inlet with engine operating at governed speed.
Dust Ingestion	Dust sprayed into inlet with engine operating at governed speed.
Maneuver Load	Engine operated on centrifuge, subjected to 3.5 rad/sec yaw rate simultaneous with 5-g acceleration.
Vibration Load	Engine operated on vibration table, subjected to a vibrational frequency sweep from 14 to 500 cps.
Bearing Overtemperature	Cooling air introduced at increasing temperature until bearing operation was impaired.

3. DESCRIPTION OF EQUIPMENT

The foil bearing demonstrator was constructed by modifying the gas generator of an AiResearch Model JFS100-13A Jet Fuel Starter. The modifications, depicted in Figure 1, consist essentially of replacing the conventional rolling element bearings with compliant foil gas bearings and replacing the original JFS power turbine module with a thrust nozzle.

The resultant "thrust" engine retains the aerodynamic components of the original gas generator, requires no oil to operate, and can generate up to 97 pounds of static thrust. A photograph of the engine is shown in Figure 2.

a. Engine

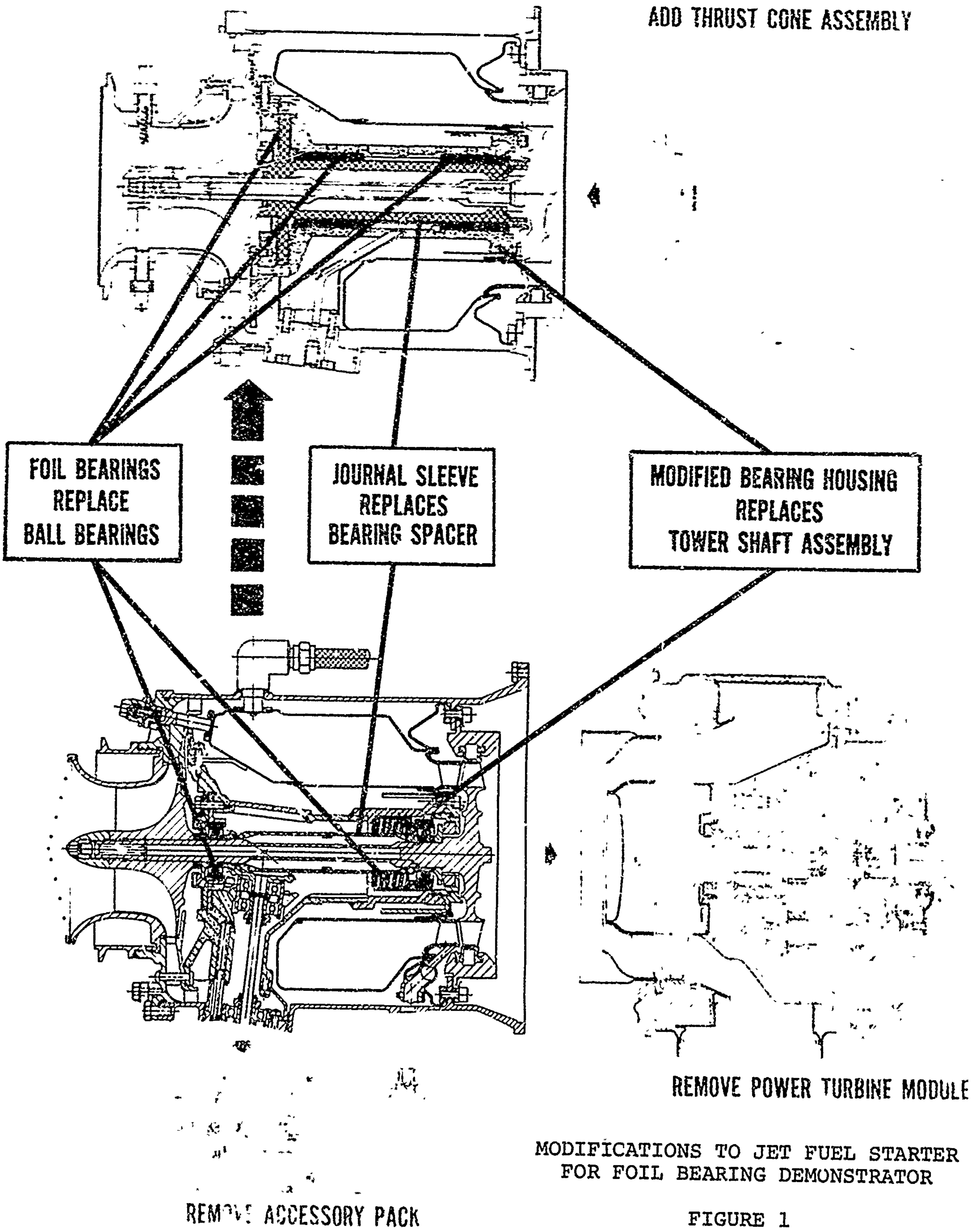
The engine layout is shown in Figure 3. The engine utilizes a single-stage centrifugal compressor, which generates a design pressure ratio of 3.2:1, and a through-flow of 1.72 pounds per second at the full-rated speed of 72,000 rpm.

The annular combustor, equipped with five fuel nozzles, provides the turbine inlet temperature of 1580°F. The axial turbine extracts a pressure ratio of 1.90 and the remaining pressure drop occurs through the exhaust nozzle (6.5-square-inch area) to provide the static thrust. Figure 4 presents the engine performance characteristics as a function of the exhaust thrust nozzle discharge area.

The geometric and mass properties of the engine are as follows.

Rotor Weight:	4.86 pounds
Polar Moment of Inertia:	0.0138 in-lb-sec^2
Design Speed:	72,000 rpm
Journal Bearing Diameter:	1.50 inches
Thrust Bearing Diameter:	3.54 inches
Engine Length:	17 inches
Engine Width:	10 inches
Engine Height:	10 inches

The design of the rotor/bearing system was dictated, in part, by the adaptability of the foil bearing system to existing parts of the jet fuel starter.



MODIFICATIONS TO JET FUEL STARTER
FOR FOIL BEARING DEMONSTRATOR

FIGURE 1

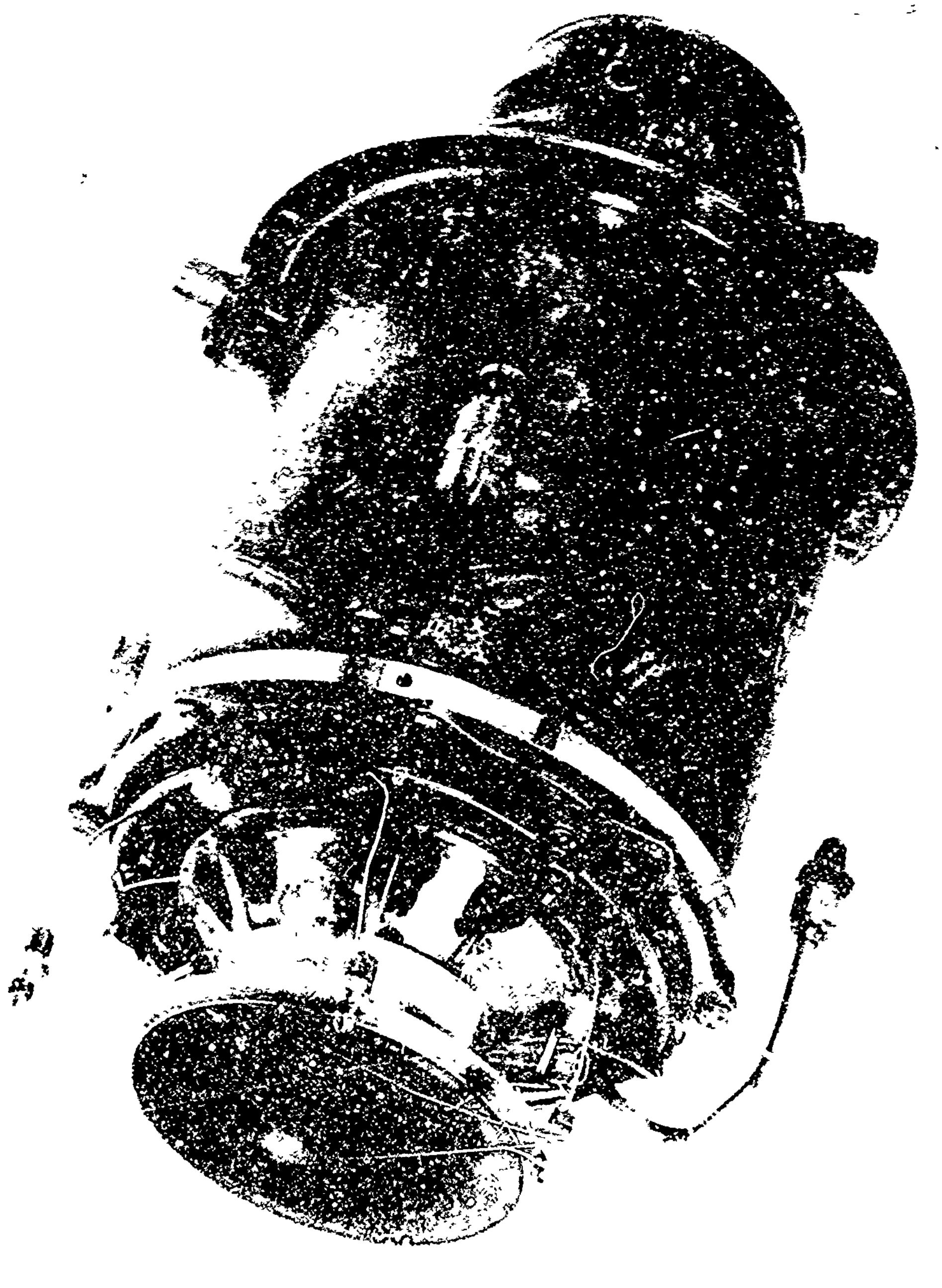


Figure 2. Three-fourths front view of foil bearing demonstrator in test configuration.

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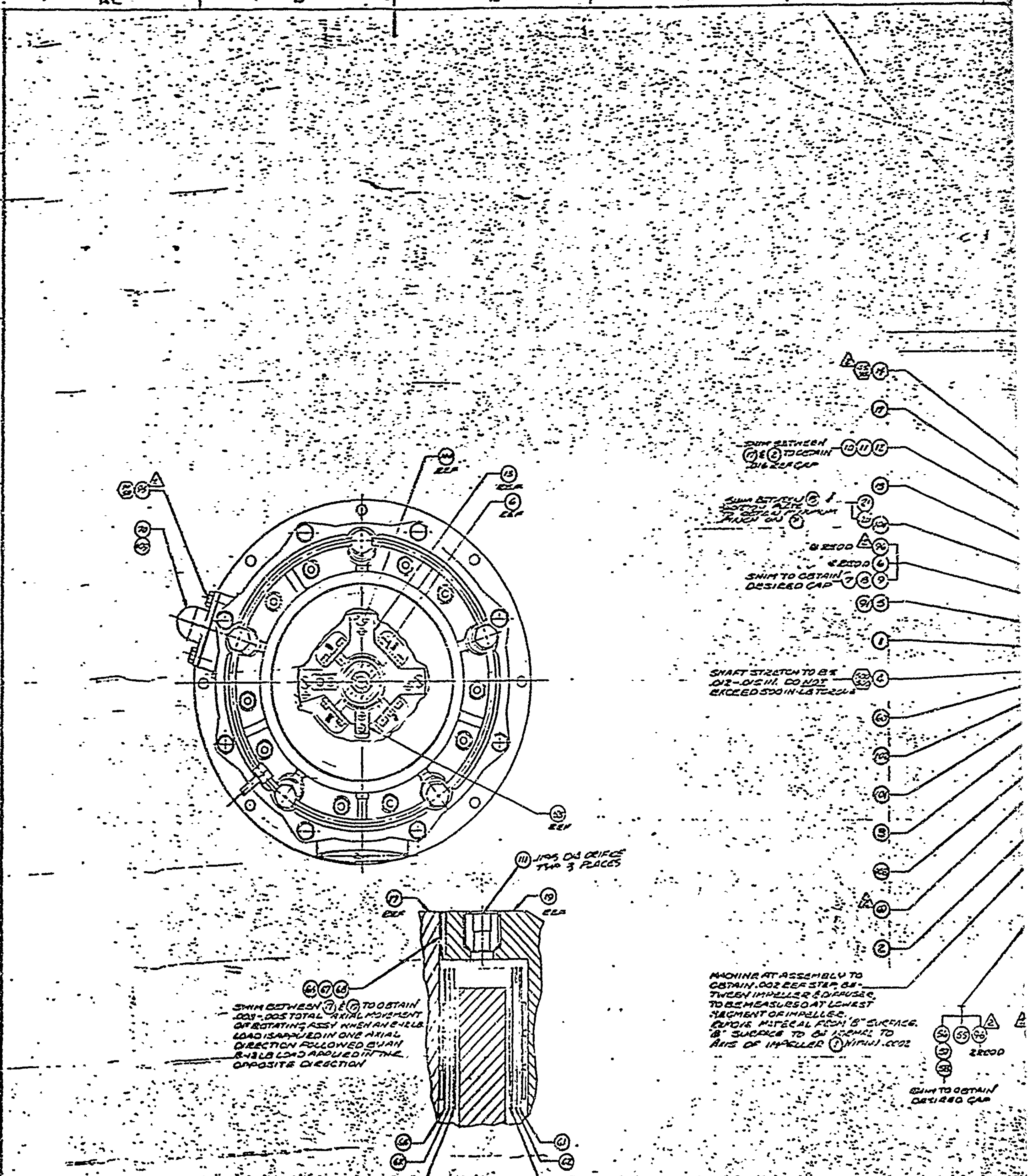
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EXCEED 500 IN-LB TORQUE
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LOAD IS APPLIED IN ONE AXIAL
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8-12 LB LOAD APPLIED IN THE
OPPOSITE DIRECTION
MACHINE AT ASSEMBLY TO
OBTAIN .002 REF STEP BE-
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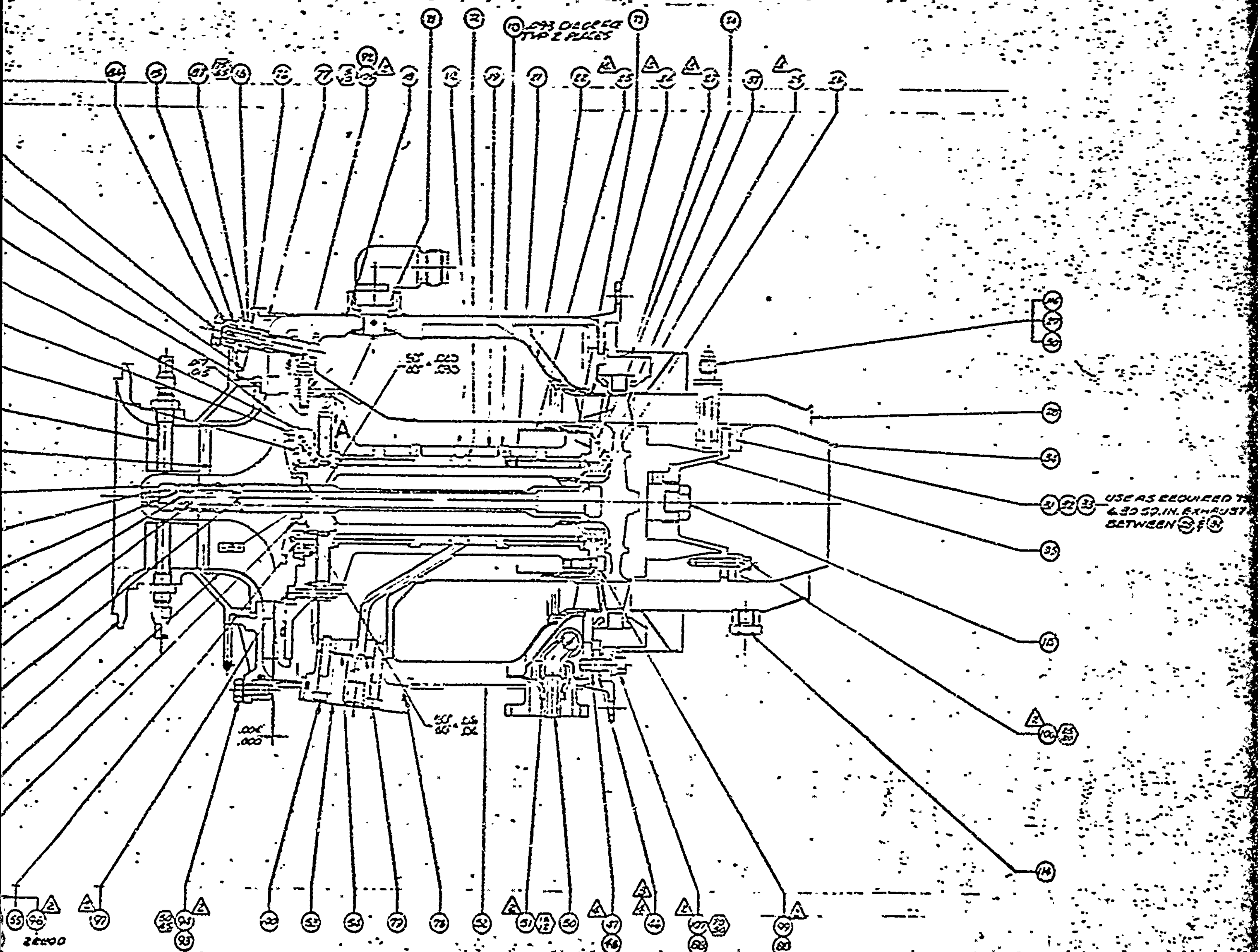

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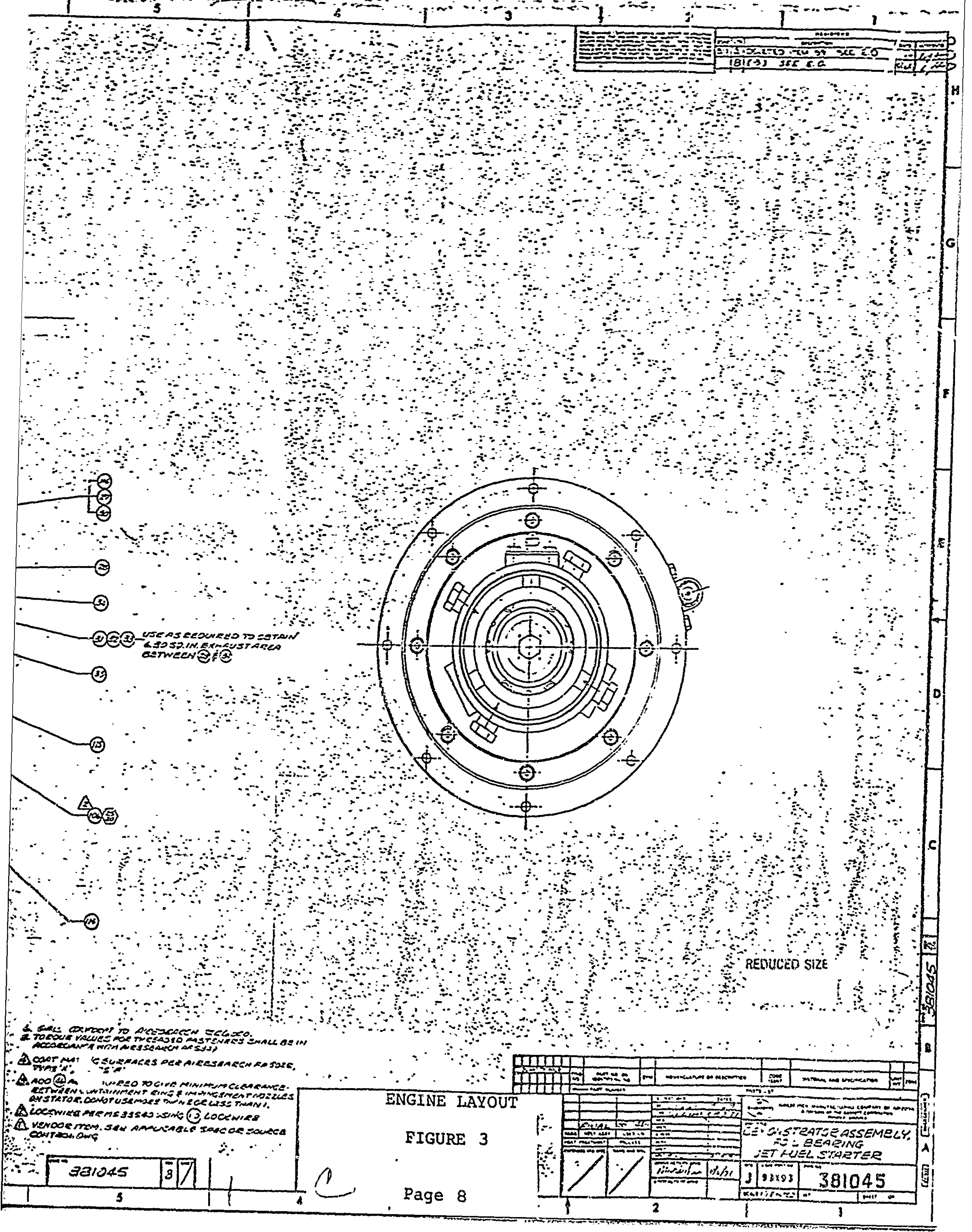

ENGINE LAYOUT

FIGURE 3

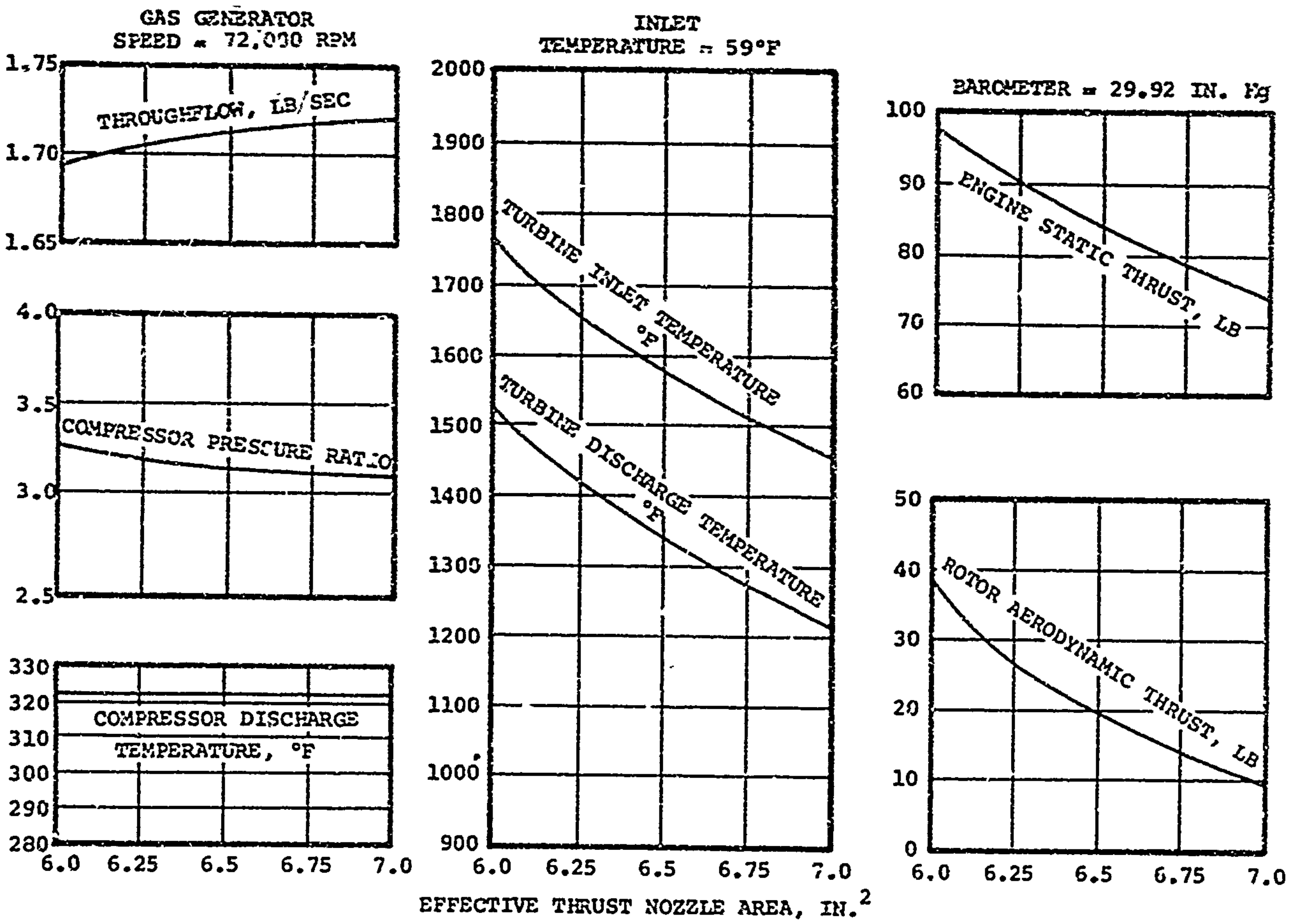

Figure 4. Sea-level standard-day performance characteristics, foil bearing demonstrator.

The bearing system is completely hydrodynamic and relies on compressor discharge air for pressurization and cooling. The cooling air flow path for the bearings is shown in Figure 5. The bearings utilize a foil coating that can withstand an ultimate temperature of 550°F but the bearings must be operated below 500°F to maintain an adequate margin for temperature rises resulting from hot spots, touchdown and soakback. The orifices, therefore, were sized to allow sufficient airflow through the bearings to maintain the bearing temperature at 500°F

The foil journal bearings have a relatively low springrate (5,000 pounds per inch) which drops the first two critical speeds (conical modes) below idle speed of the engine. Figure 6 presents calculated critical speed characteristics of the rotor/bearing system. The engine rapidly passes through the first two criticals during the start mode while accelerating to light-off speed from the air-impingement start system. Efforts to detect critical speed shaft excursion, using proximity probes, have been futile because of the rapid rotor acceleration. To overcome the static friction of the bearings during starting, the impingement air starting system must provide about 12 inch-pounds of torque. The bearing friction is greatly reduced after rollover and liftoff, resulting in rapid acceleration of the rotor. During an average start, the engine accelerates from approximately zero speed to 20,000 rpm in two seconds.

The third critical speed is much higher (134,000 rpm) than the normal engine operating speed (72,000 rpm) and presents no problem. The wide range of allowable operating speed between the second and third critical speeds is an inherent advantage of the lower bearing springrate and larger journal bearings.

The engine utilizes a remotely operated bench fuel control system that requires 115-vac and 24-vdc external electrical sources. There is no gearbox on the engine to drive accessories or provide a mechanical speed signal; therefore, these functions are performed by a remote, electrically driven fuel pump and an electronically operated speed sensor and governor system. The system control panel is shown in Figure 7.

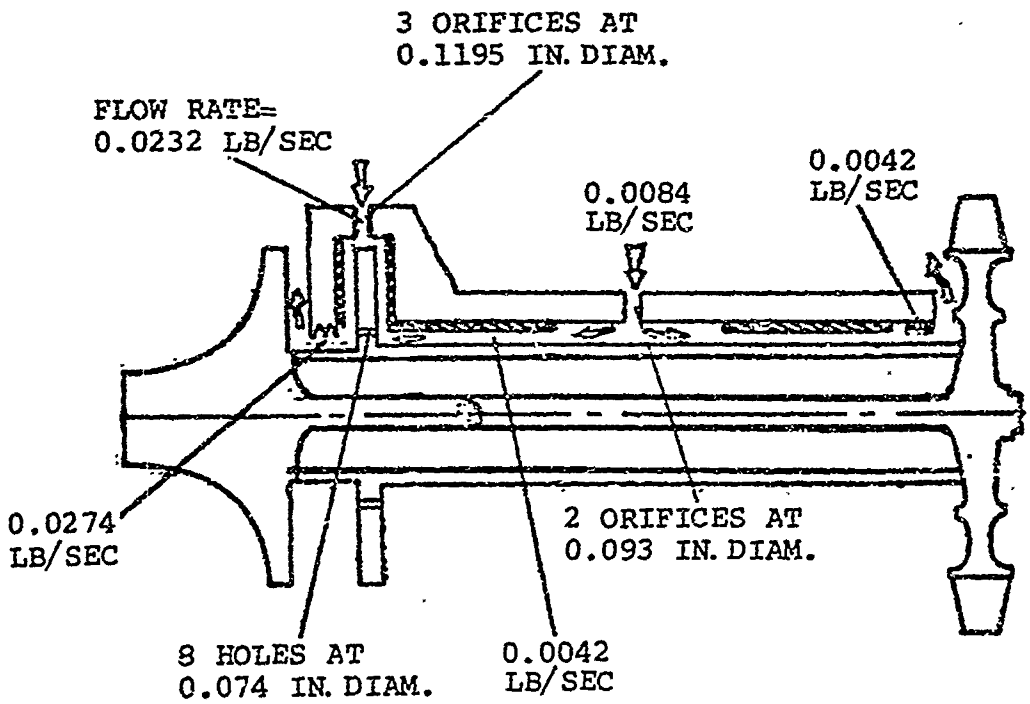

Figure 5. Cooling air flow paths and orifice sizes in foil bearing demonstrator.

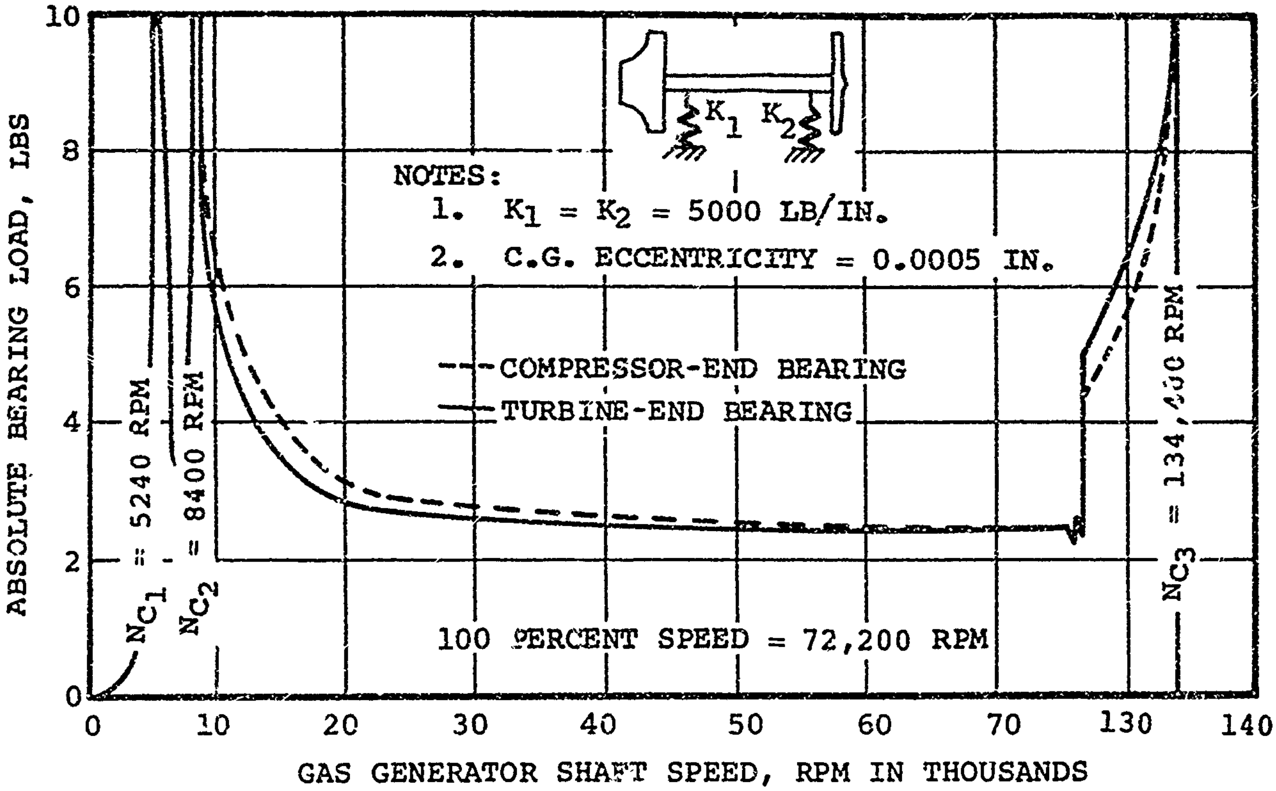

Figure 6. Bearing load versus shaft speed, foil bearing demonstrator.

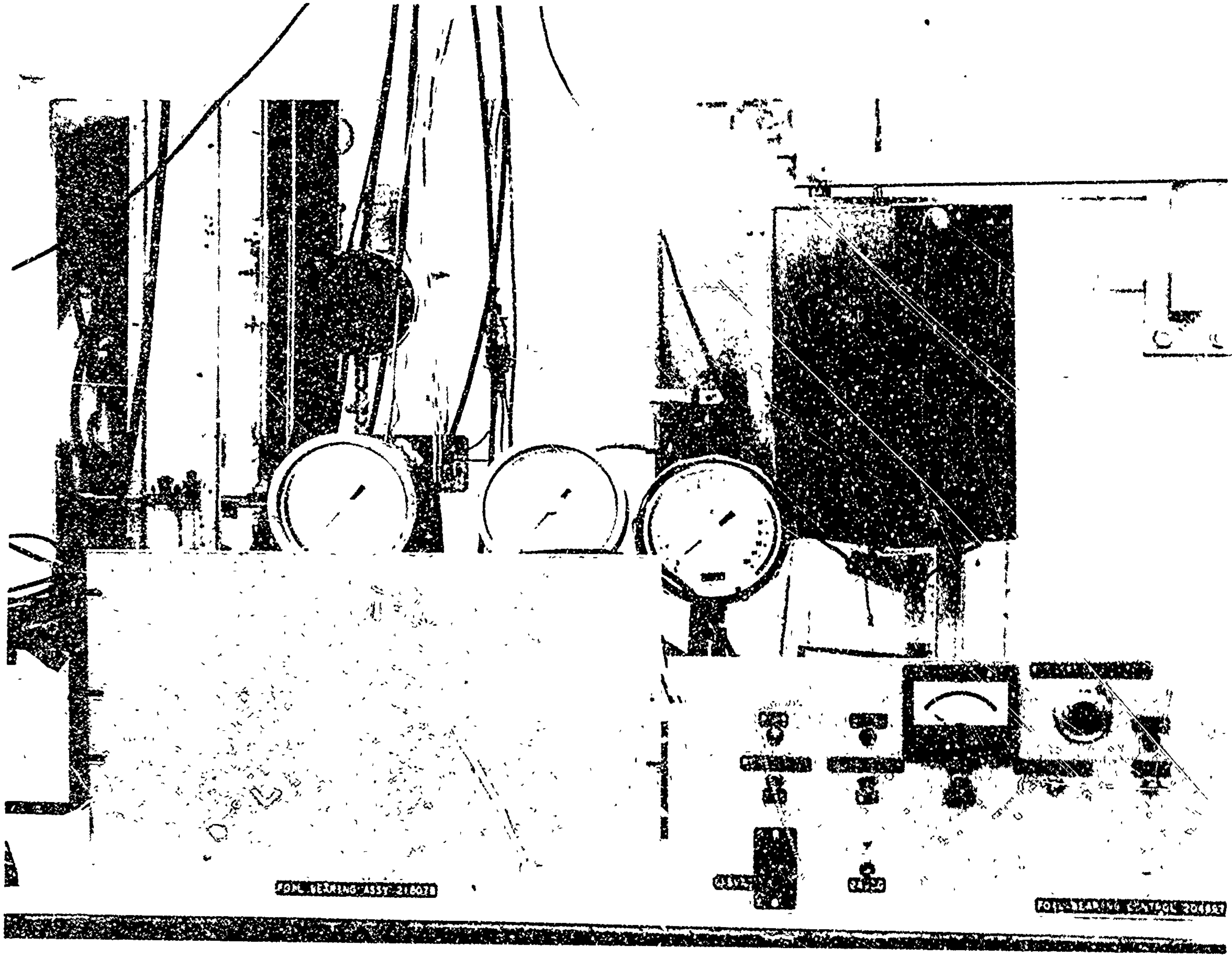

Figure 7. Foil bearing demonstrator control panel and electronic control assembly.

High pressure air supplied to the impingement nozzles is utilized to start the engine, which is automatically accomplished by the control system. The starting sequence is initiated when the engine start switch is actuated at the control panel. The high-pressure (1000 psia) impingement air is released to the nozzles, and the ignitor is energized. When the unit reaches 20-percent speed (14,400 rpm), fuel is supplied to the engine and lightoff accelerates the unit to governed speed, with the impingement air and the ignitor shut off at about 42,000 rpm. The fuel control governor maintains the desired speed via a fine-speed adjustment on the control panel.

Program tests scheduled for the engine included the costly procedure of operating and controlling the engine while on a centrifuge. To avoid the fuel plumbing, air starting, and remote engine control problems, the engine was modified to operate with external, pressurized air supplied to the engine plenum rather than the fuel engine mode. Although the motoring air pressure necessarily is somewhat lower than compressor discharge, the compressor discharge temperature was maintained. Figure 8 shows the engine design layout for the air-motored configuration. Figure 9 gives the plenum pressure of the modified engine as a function of speed. With no blades on the dummy compressor, the expansion of the plenum air across the turbine must overcome only the bearing power loss and aerodynamic drag.

The air motor configuration was utilized in the maneuver loading test, the vibrational loading test, and the bearing overtemperature test. This configuration was used for the overtemperature test to facilitate control of the bearing temperatures by increasing the plenum supply air temperature and, hence, increasing the bearing cooling air temperature.

b. Bearings

The foil bearing demonstrator is equipped with two 8-foil, 1.5-inch diameter by 1.6-inch long journal bearings and one double-sided, 12-foil thrust bearing with an outer diameter of 3.5 inches and a radius ratio of 0.6. Figure 10 shows the journal bearing and Figure 11 the thrust bearing used in the engine. The bearing system is completely hydrodynamic.

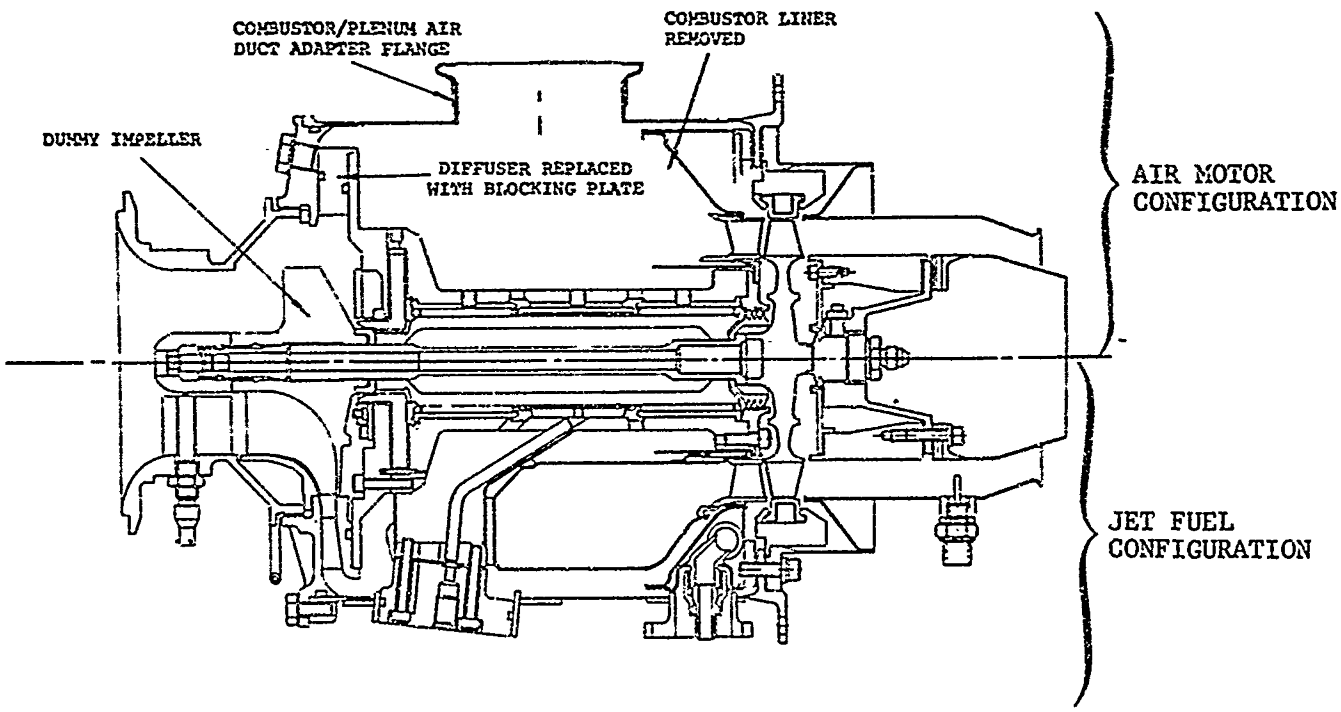

Figure 8. Changes to original foil bearing demonstrator for conversion to air motoring mode.

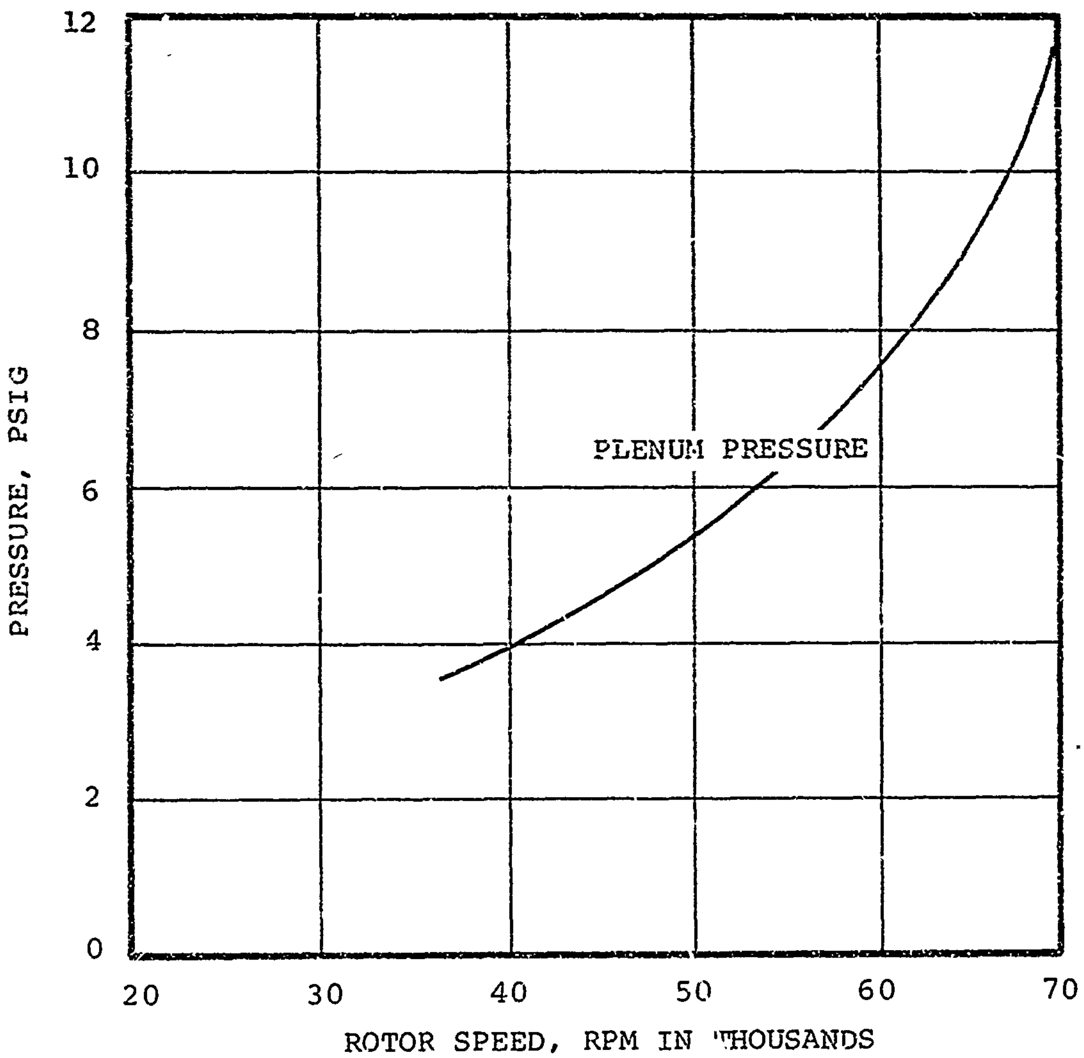

Figure 9. Plenum air pressure versus speed in the air motoring mode.

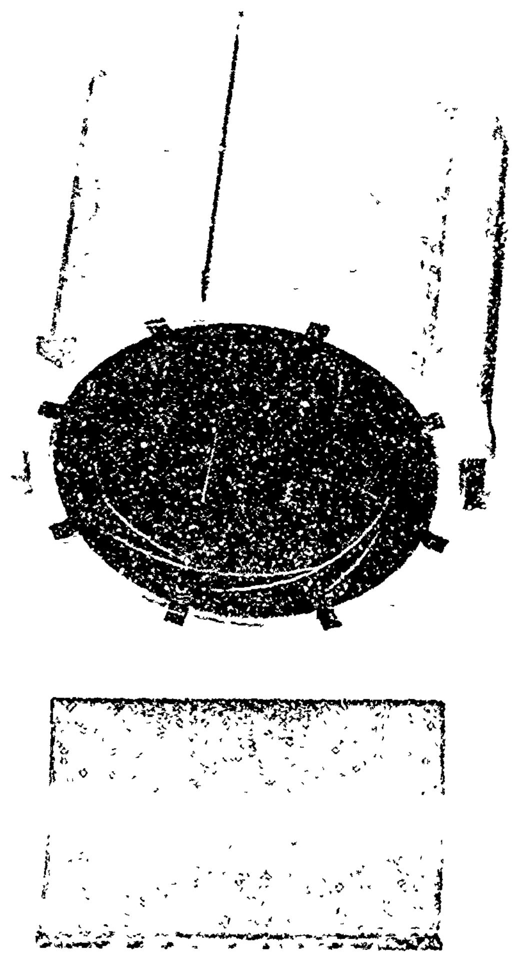

Figure 10. Journal bearing selected for foil bearing demonstration.

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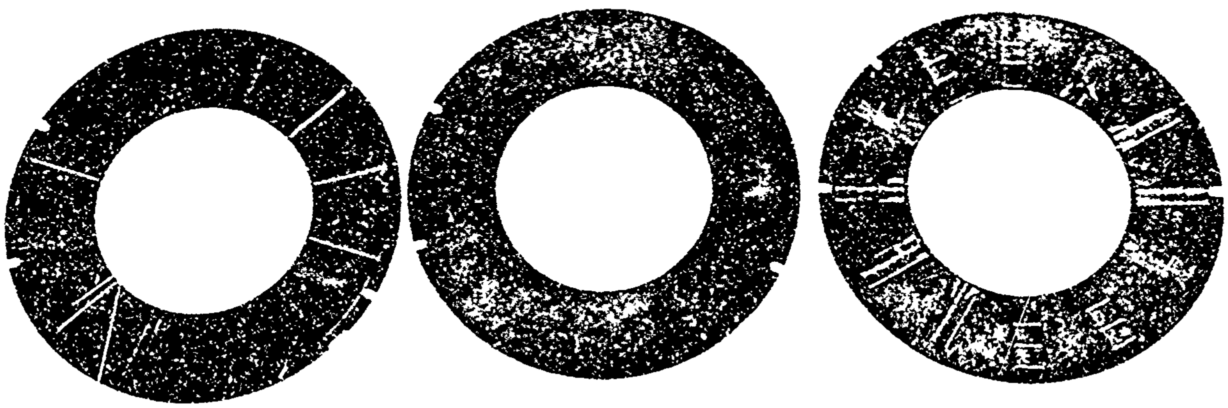

Figure 11. Thrust bearing set selected for foil bearing demonstrator.

The foil coating utilized for both the thrust and journal bearings is Teflon. Although it has a relatively low temperature limitation, the Teflon coating has exhibited low friction and good embedment and wear-in characteristics which other candidate coatings were unable to match.

The foil bearing system relies on compressor discharge air for pressurization and cooling. Higher air pressures and temperatures generally will provide higher load capacity. At sea level conditions, compressor discharge temperature will reach 280° to 320°F and will provide cooling to maintain the bearings at 480° to 525°F. Each journal bearing requires 0.004 pound per second cooling airflow and the thrust bearing requires 0.012 pound per second per side.

The breakaway torque for the rotor/bearing system is about 12 inch-pounds. A performance map for the journal bearings, established from test rig results in a previous program, is included in Figure 12. The bearing has a projected area of 2.4 square inches and, under good loading conditions, will generate about 10.5 psi or 25 pounds under ambient test conditions. The gyroscopic torque loading, combined with some degree of misalignment and thermal distortion unavoidable in an engine application, produced loads somewhat less than those obtained from the test rig, as indicated by the test results reported in Section 4.

The load capacity for the thrust bearing was also established from test rig results in a previous program and is shown in Figure 13. The bearing has a projected area of 6.35 square inches and, under good loading conditions, will generate 25.3 psi or 160 pounds under ambient test conditions.

The calculated aerodynamic thrust of the engine rotor is presented in Figure 14 as a function of speed. The maximum thrust is 30 pounds in the forward direction occurring at full speed.

c. Instrumentation

Instrumentation was installed on the engine to monitor bearing performance and observe general engine aerodynamic parameters to ensure proper operation of the engine. Figure 15 shows the location of the engine instrumentation.

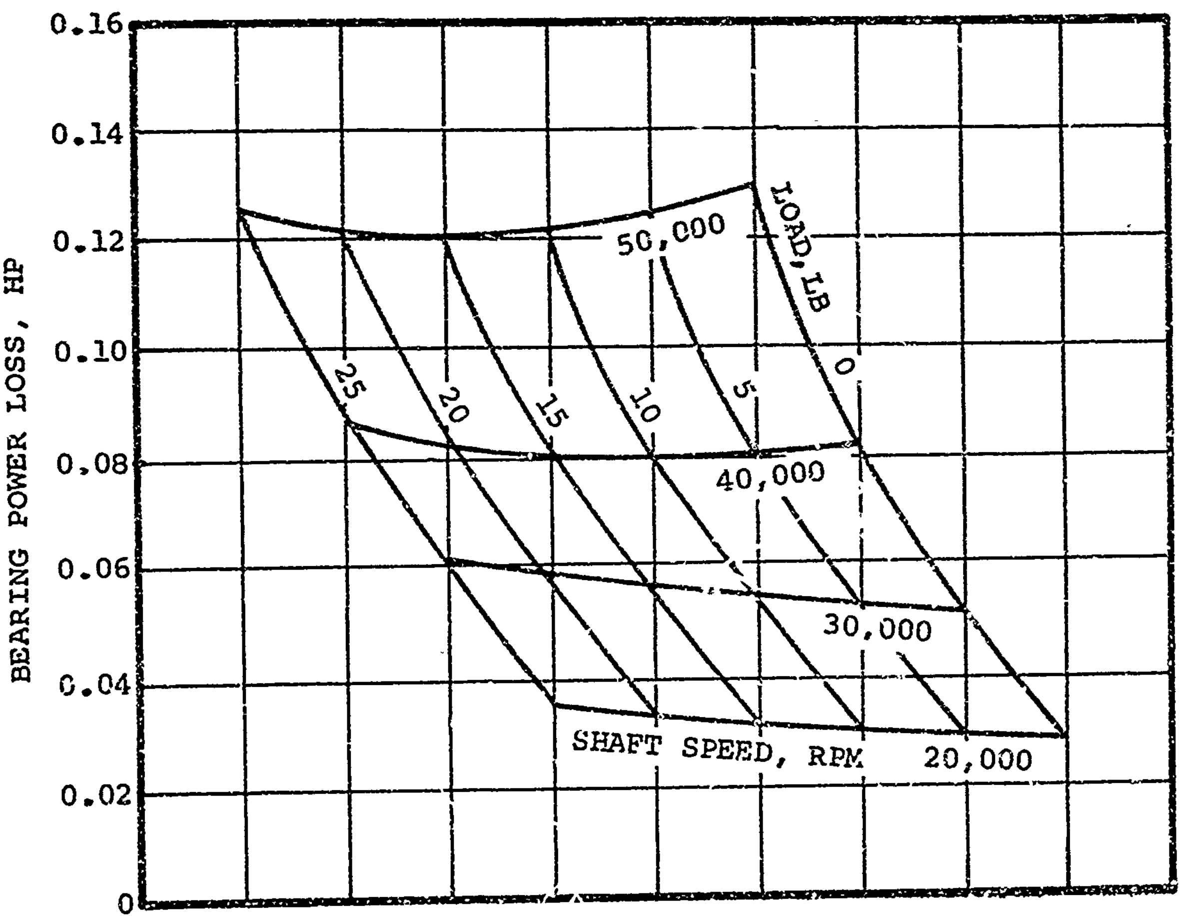

Figure 12. Performance map for journal bearings.

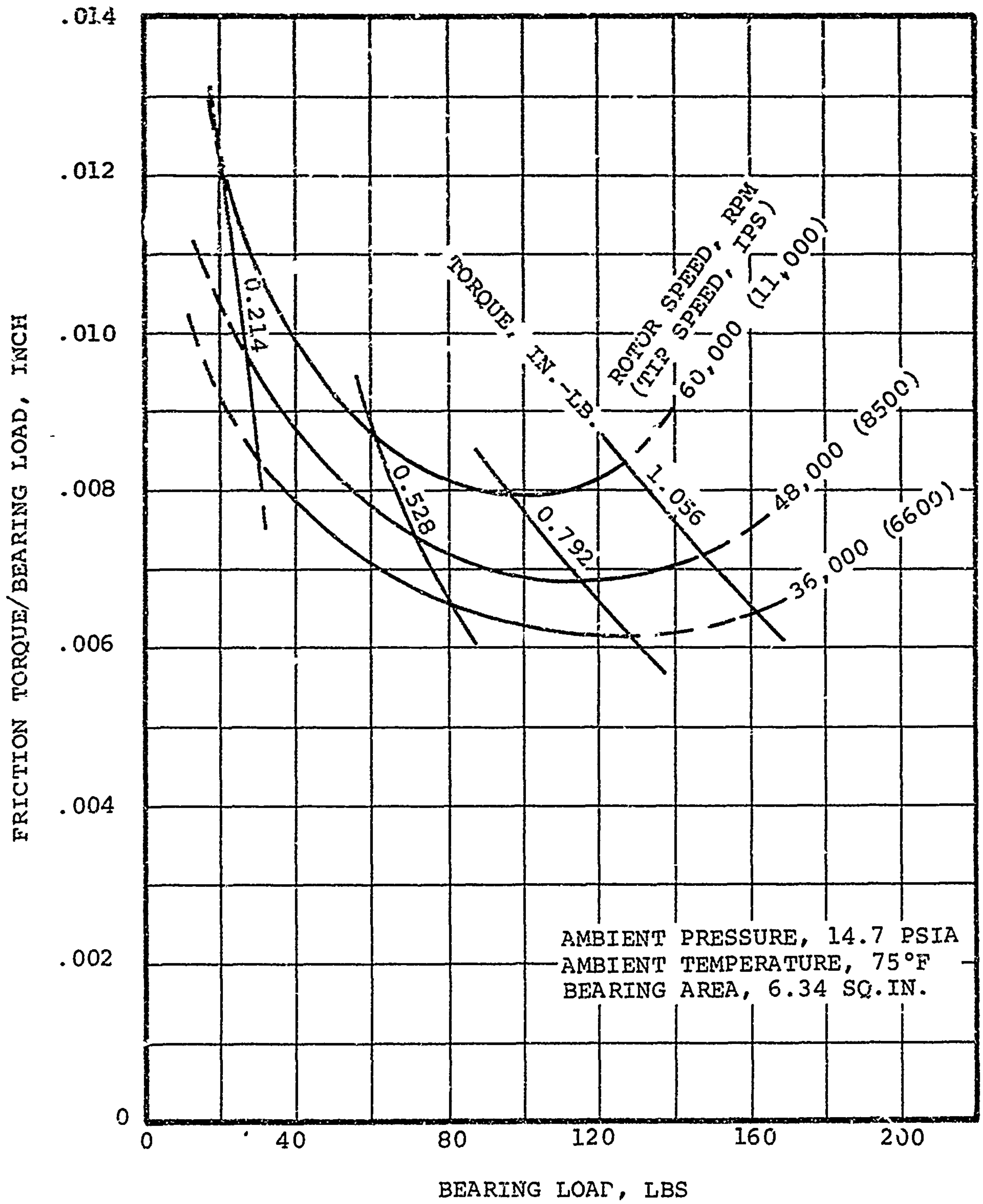

Figure 13. Load capacity of thrust bearing.

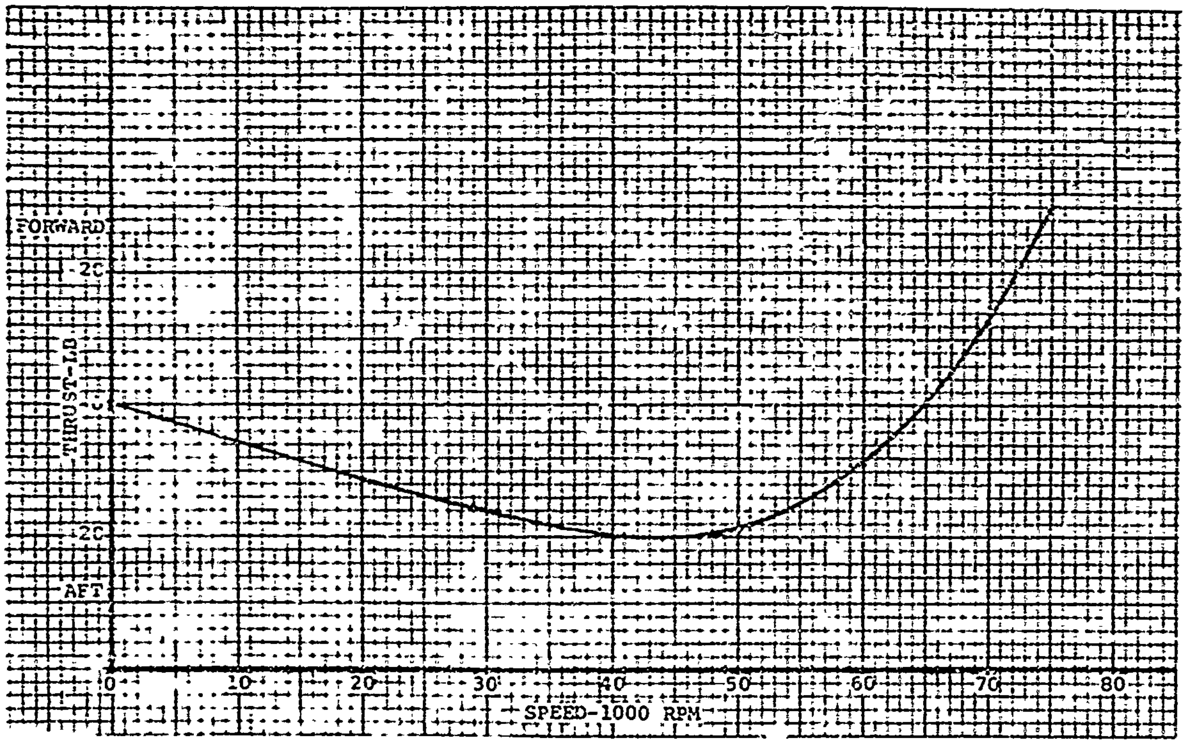

Figure 14. Aerodynamic thrust as a function of speed.

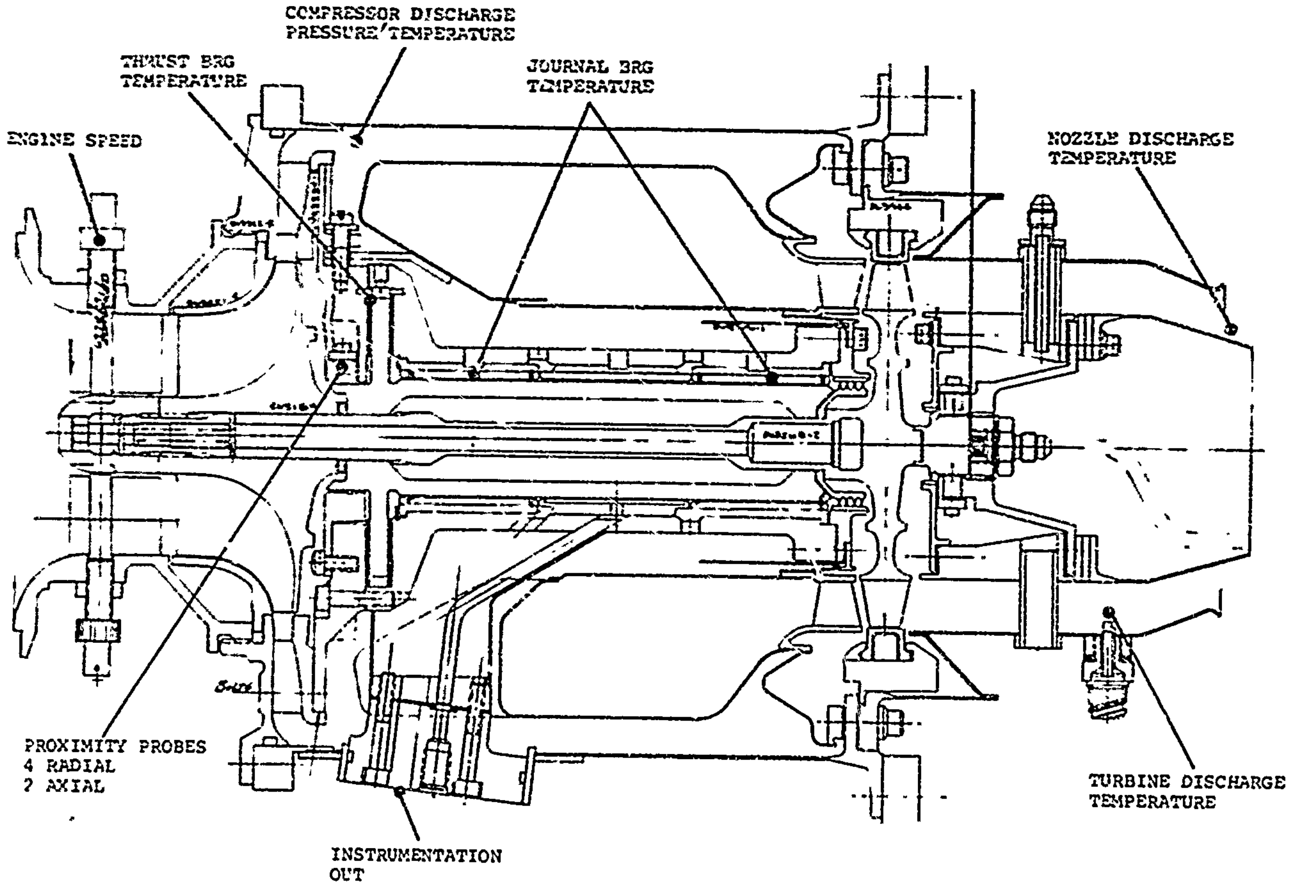

Figure 15. Engine instrumentation.

Wayne Kerr proximity probes were utilized to monitor shaft excursion. The probes were exposed to a temperature environment of about 400°F. Because of their insensitivity to temperature, Wayne Kerr probes were chosen over other types of proximity probes.

Instrumentation readout and recording were accomplished with instruments appropriate for each test. Sanborn and Offner recorders were used generally to record important engine aerodynamic data and bearing temperature. A tape recorder was utilized to record shaft excursion, the tape to be played back later through a Sanborn recorder or an oscilloscope.

4. TEST PROGRAM

The test program consisted of five tests, each necessary to determine the feasibility of utilizing foil bearings on engines requiring qualification testing.

a. Water Ingestion Test

Results of the water ingestion test conducted May 23, 1972, indicate that an engine utilizing foil bearings has strong potential to satisfy MIL-E-5007C specifications for water ingestion required to qualify a variety of military engines.

Figure 16 shows the engine installed in the test cell, with instrumentation connected and the water injection nozzle directed toward the inlet. The 0.078-inch-diameter nozzle was calibrated prior to the test, using a flowmeter. A pressure of 91 psia was necessary to attain the required 0.435-gallon-per-minute water flowrate.

After the unit accelerated to governed speed (67,500 rpm), the water injector was activated. Water ingestion continued for 35 seconds, followed by 1 1/2 minutes of operation to ensure that no damage was incurred. There were no visible signs of engine damage.

During the test, pertinent bearing and gas generator performance data were preserved by Sanborn recording and motion picture coverage was included.

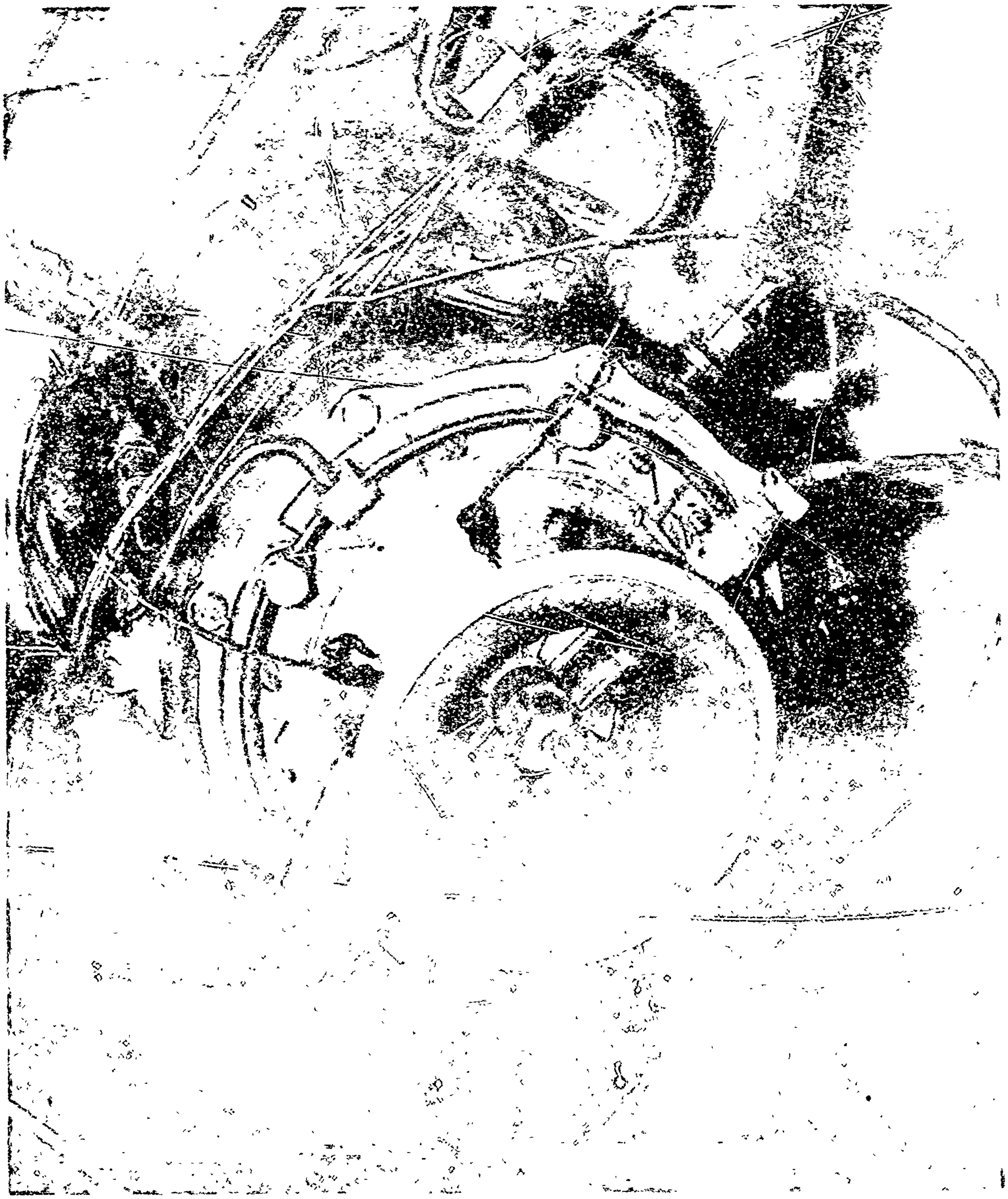

Figure 16. View of water nozzle directed toward inlet of engine prior to start of water ingestion test.

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Following the test, the unit was disassembled for bearing inspection and photography. Figure 17 shows a 10-power magnification of typical journal bearing foil segments. Other than a slight polish from shaft contact during starting and stopping there were no signs of damage to the bearings. Since the test was relatively short, the bearings did not reach high steady-state temperatures and the foil coating did not darken.

Figure 18 shows a 10-power magnification of a typical thrust bearing segment. Some scratches are evident, but these were attributed to dust inadvertently introduced during assembly. (Airborne dust entrained by the radially introduced bearing cooling air would have produced a helical scratch.) Local discoloration, common to all bearings as they "bed in", is evident, but as observed in previous running no performance deterioration resulted.

b. Dust Ingestion Test

Results of the dust ingestion test, conducted July 7, 1972, indicate that an engine utilizing foil bearings has strong potential to satisfy MIL-E-5007C specifications for sand and dust ingestion.

The sand and dust actually used in the test is classified from natural Arizona dust, prepared by the AC Spark Plug Division of General Motors. This grade of test dust has been used for dust ingestion on previous production gas turbine engines. Particle size distribution is as follows:

COURSE GRADE
PARTICLE SIZE DISTRIBUTION

0 to 5	Microns	12 ± 2%
5 to 10	"	12 ± 3%
10 - 20	"	14 ± 3%
20 - 40	"	23 ± 3%
40 - 80	"	30 ± 3%
80 - 200	"	9 ± 3%

A total quantity of 0.002 pound of test dust was introduced into the unit inlet over a period of 30 seconds. This was accomplished by utilizing a 14-compartment dust pan (Figure 19), mounted approximately two feet from the inlet bellmouth. The test dust was equally distributed among the compartments.

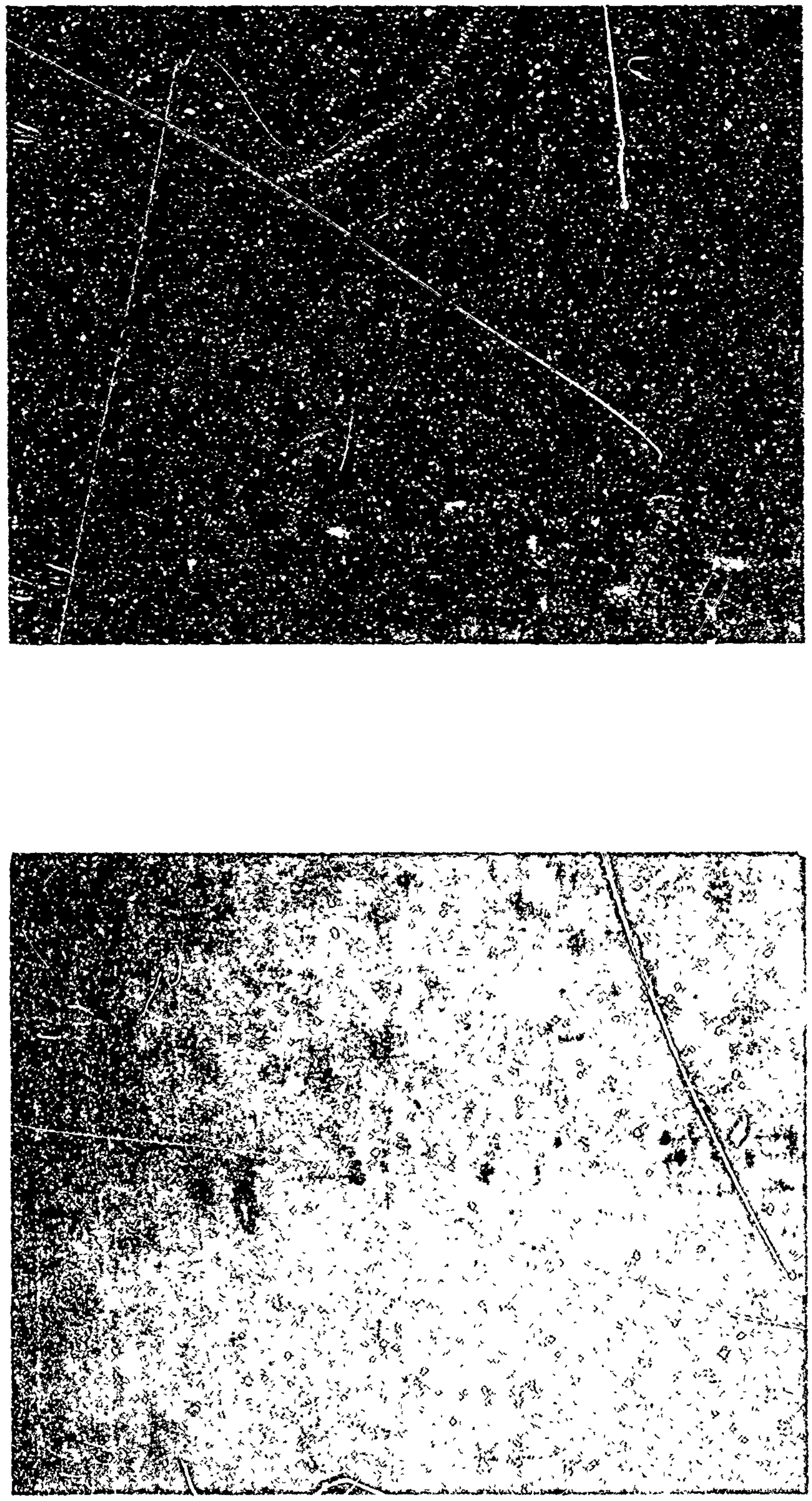

Figure 17. Ten-power magnification of journal bearing foil segments after water ingestion test.

MP-34326

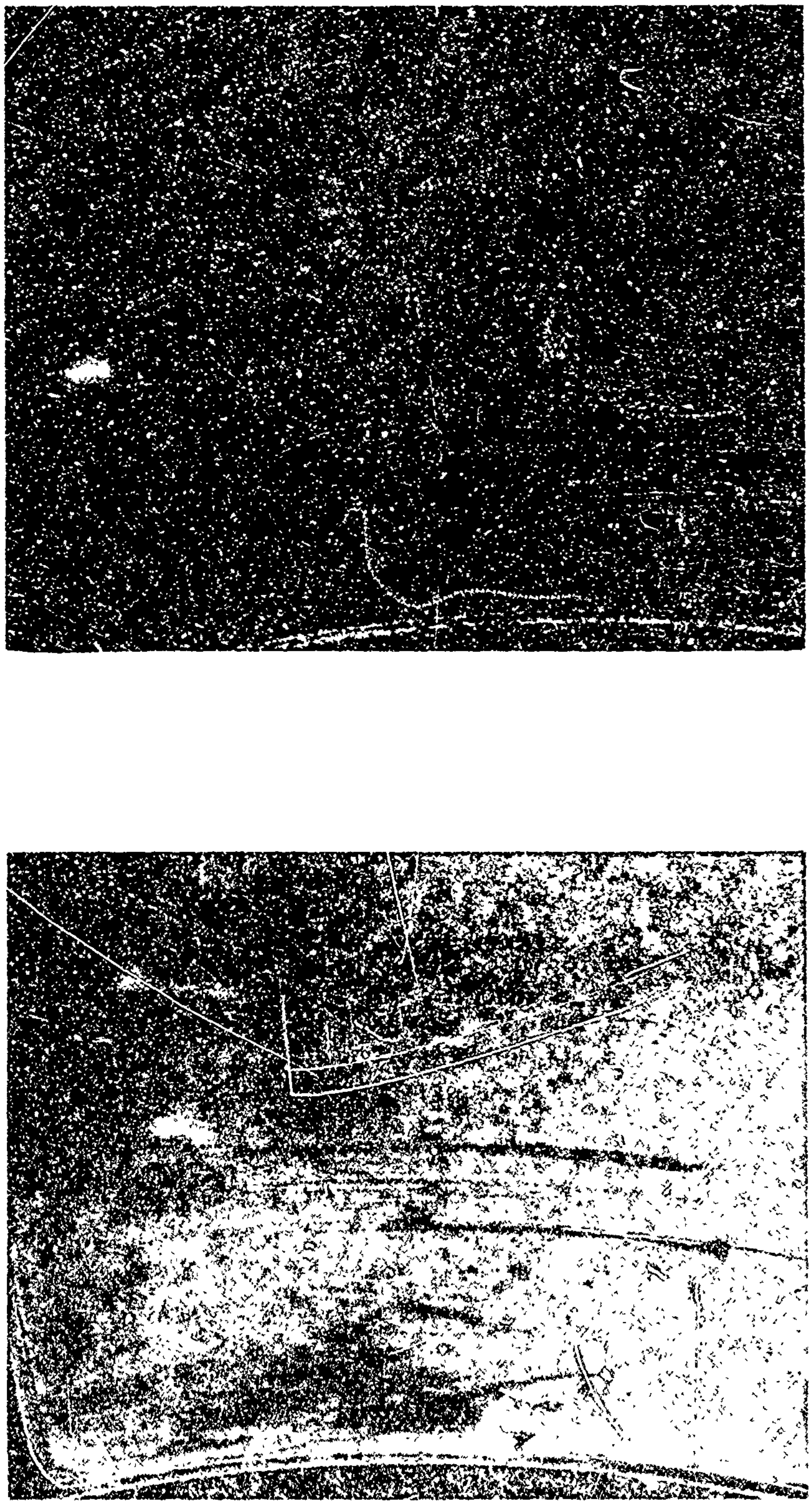

Figure 18. Ten-power magnification of forward thrust bearing foil segments after water ingestion test.

MP-34327

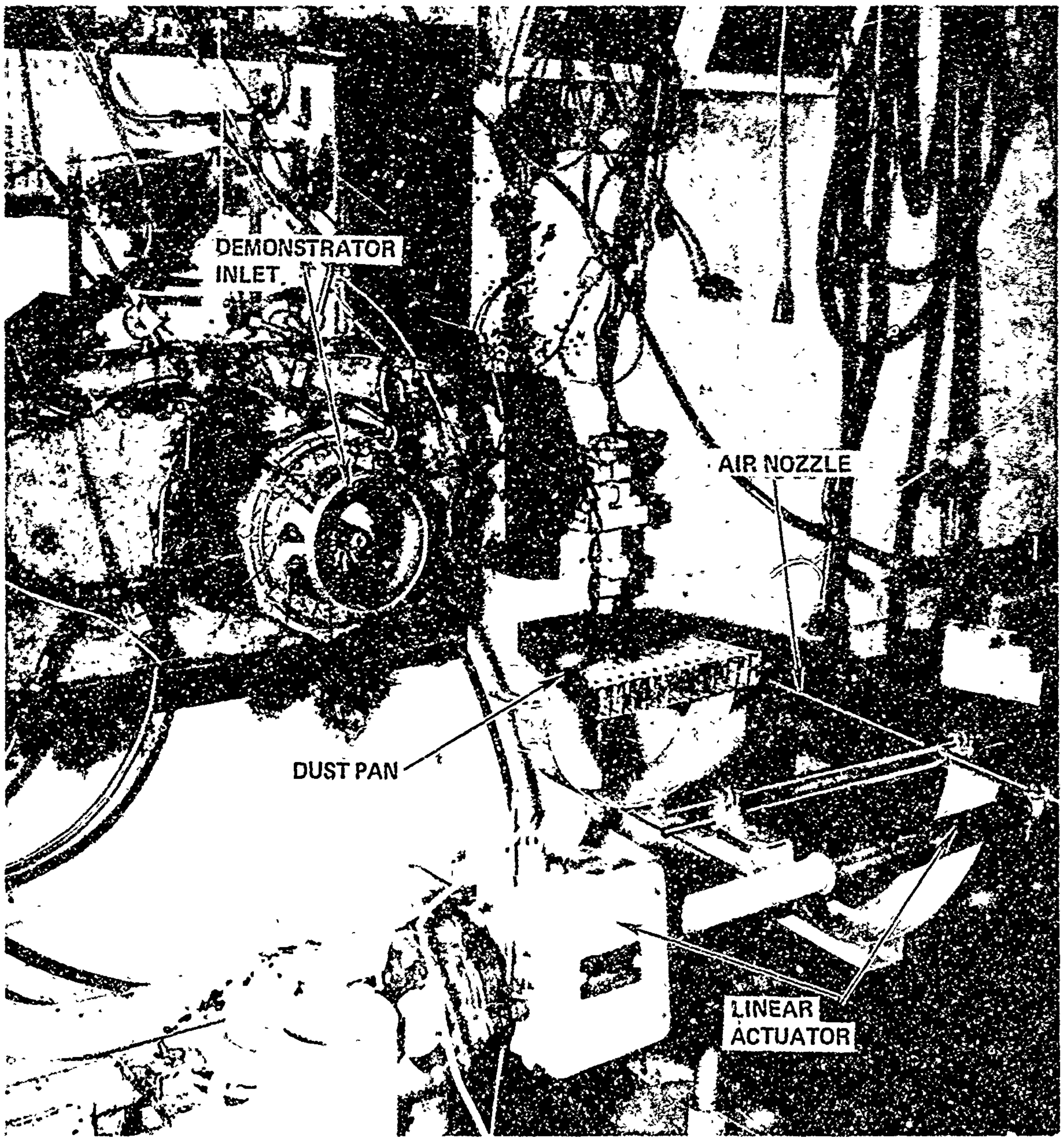

Figure 19. Foil bearing demonstrator and dust ingestion mechanism prior to dust ingestion test.

MP-34953

An additional quantity was added to allow for the dust film which constantly remained in the dust pan. The pan was set far enough from the unit inlet to prevent the test dust from being drawn into the inlet during engine starting, prior to initiation of the test.

A time-interval energized linear actuator was used to move an air nozzle past the dust pan compartments to blow dust from each of the compartments into the engine inlet during the 30-second period.

Before the actual test was initiated, a test run was made. The engine inlet was covered and dust was blown from the pan toward the inlet, establishing the direction of the spray and ensuring that all of the dust would be blown out within the required period. (All of the dust was blown from the pan in 29.2 seconds.) With the inlet cover removed, several engine starts were made for overall system checkout, after which the pan was refilled and the dust ingestion test was initiated.

With the engine operating at a governed speed of 67,500 rpm, the required 4.4×10^{-5} pounds dust per pound throughflow was introduced into the inlet for the 30-second period. The engine was operated for an additional 150 seconds to ascertain whether any immediate effects on engine operation or performance would result. No deterioration in either was detected.

Following the test, the unit was disassembled for inspection. Figure 20 shows two sample journal bearing foils: one from the turbine-end bearing and the other from the compressor-end bearing. The foils were in excellent condition. Normal minor discoloration at the foil tips, where they experience the greatest soakback temperatures (540°F in this test), and some glaze, which accompanies "bedding in", were evident. Shallow scratches, particularly the axial ones, could be seen on the foils. These may have been caused by trapped dust particles being rolled along the foils as the shaft was withdrawn during disassembly.

The forward and aft thrust bearing assemblies are shown in Figure 21. The circled portions of each, magnified ten times, are shown in Figure 22. As with the journal bearing foils, both thrust bearing assemblies were in excellent condition. Some of the dust particles were embedded in the foil coatings; the rest of the dust seemed to have had little effect on the bearings, other than leaving minor scratches. Some local charring was evident on all of the bearings, as was noticed after the water ingestion test.

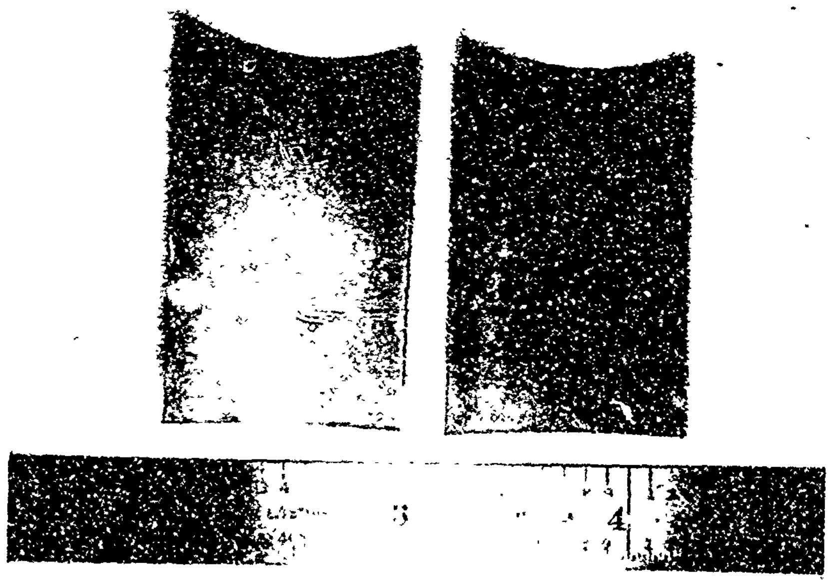

Figure 20. Journal bearing foil segments after dust ingestion test.

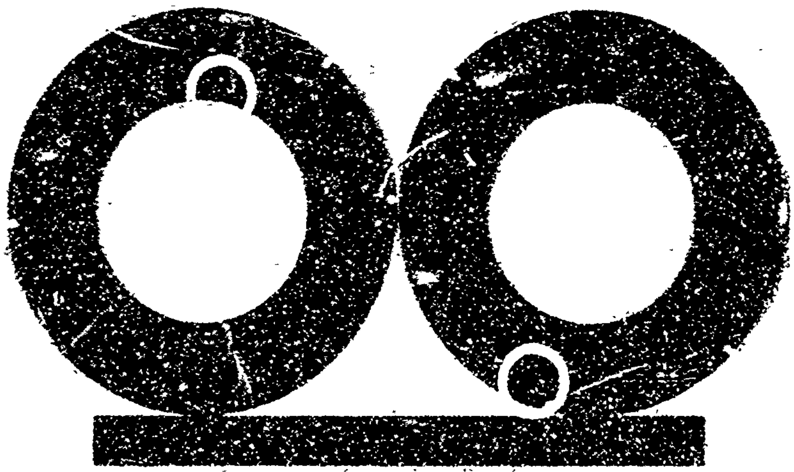

Figure 21. Forward and aft thrust bearing assemblies after dust ingestion test.

MP-34831

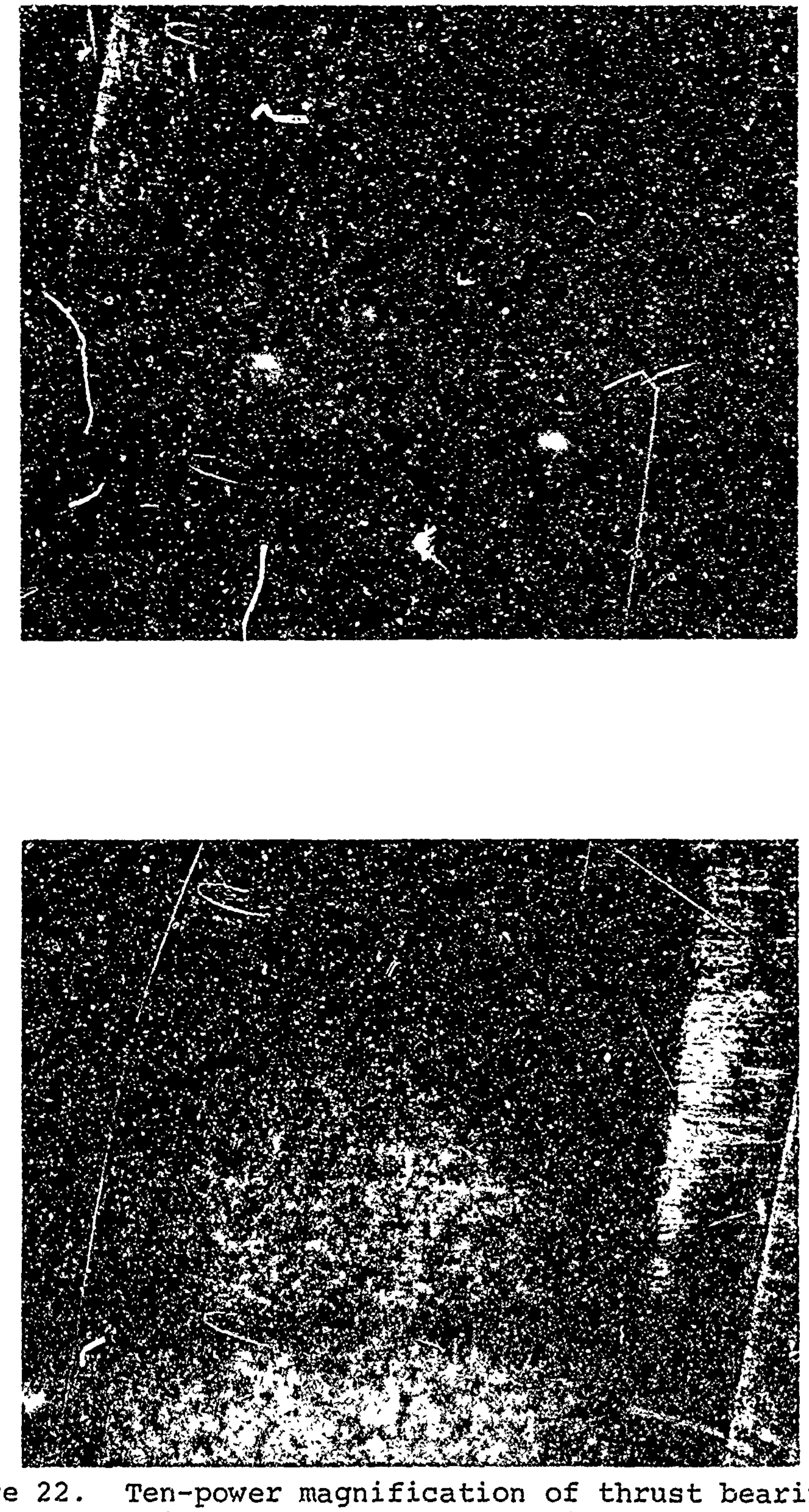

Figure 22. Ten-power magnification of thrust bearing foil segments after dust ingestion.

MP-34829

c. Maneuver Loading Test

Engine modifications as described in Paragraph 3 (a) were made prior to the initiation of the maneuver load test. Because this test was to be performed on a centrifuge test rig, the demonstrator was modified to operate with external, pressurized air (air-motoring mode) rather than the jet-fuel mode.

The purpose of this test was to operate the demonstrator under imposed loads from a 3.5-radians-per-second yaw rate simultaneous with 5-g acceleration. The modified demonstrator, mounted in the centrifuge adaptor, is shown in Figure 23.

The required 3.5-radians-per-second yaw maneuver rate combined with a 5-g centrifugal acceleration defines a unique centrifuge arm radius of 13.3 feet. Since the centrifuge planned for use was limited to 10 feet, a reduction in rotor speed from 67,500 to 52,000 rpm was necessary to provide a 3.5-radian equivalent gyroscopic load on the bearings.

The maneuver loading tests were conducted as follows.

1) Test No. 1

The original maneuver test was attempted August 23, 1972 at the AiResearch-Phoenix facility. The demonstrator was positioned 10 feet from the center of the centrifuge and was operated at an engine speed of 52,000 rpm. At approximately 11 rpm centrifuge speed, a noticeable drop in demonstrator speed was observed, followed by a partial recovery, another drop, a weak rise, and finally a decay to zero. The loss of hydrodynamic gas film occurred at about 11 rpm centrifuge speed, resulting in highly increased drag and subsequent loss in rotor speed. This delay in speed, however, reduced the "gyroscopic load" (a product of rotor speed and induced yaw rate) by a sufficient degree that the bearings could reestablish a gas film at the then existing rotor speed of 26,000 rpm. This, in turn, "freed" the shaft, allowing it to reaccelerate until the cycle was repeated, finally resulting in sufficiently high foil/shaft temperatures to cause total seizure.

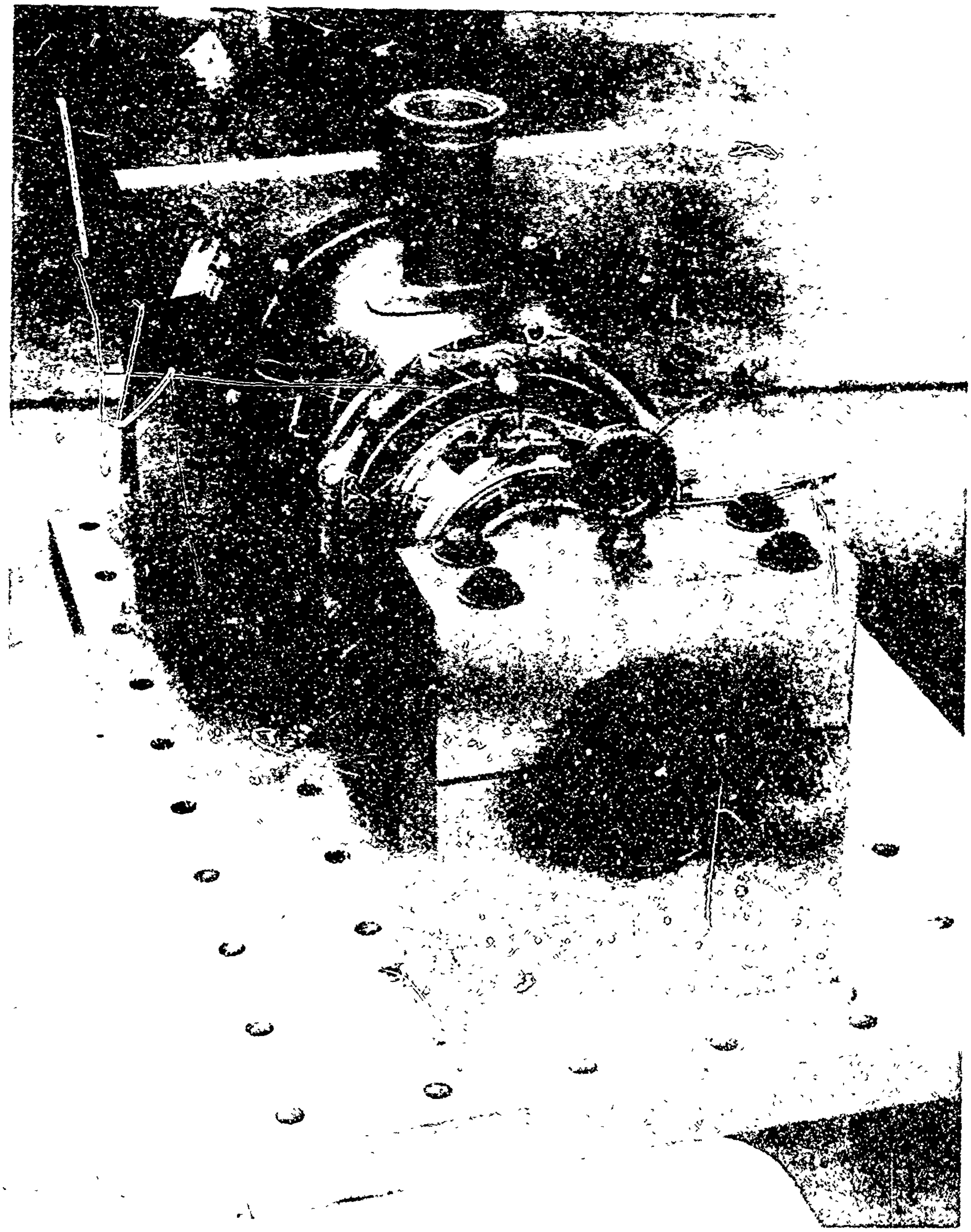

Figure 23. Front view of modified foil bearing demonstrator mounted on centrifuge adaptor prior to maneuver load test.

The decision to abort the run was made after the first cycle, but the low gain of the centrifuge drive/brake air-jet system, coupled with the high inertia of the arm, test piece, and counterweights, made it impossible to stop the simulated yaw maneuver in time to save the bearings.

Following the test, the unit was disassembled for inspection. The compressor-end foils are shown in Figure 24, the turbine-end foils in Figure 25, and the journal shaft/thrust runner assembly in Figure 26. Failure appeared to be the result of corner loading the bearings in excess of the load capacity that they could hydrodynamically sustain under the test shaft speed and bearing environment.

A number of bearing environment compromises, necessitated by the use of the 10-foot centrifuge, degraded the bearing performance to a higher degree than anticipated. These include the aforementioned reduction in shaft speed, bearing cavity pressure and inlet temperature from those seen when the demonstrator is operated in the jet-fuel mode.

It was decided, therefore, to repeat the test at the AiResearch facility at Mint Canyon, near Los Angeles, California. This is a new facility equipped with a 20-foot centrifuge that can easily provide the required motoring air supply up to 350°F. It also has an elaborate "brush" assembly that assures adequate signal transmission from the rotating assembly to ground, with minimum noise and high reliability.

The damaged parts of the demonstrator were replaced in preparation for the Mint Canyon maneuver test.

2) <u>Test No. 2</u>

After assembly, a bench test was conducted in Phoenix for checkout prior to shipment to Los Angeles. The bench test was conducted to accumulate information about the air motor configuration of the demonstrator, which was unobtainable while running on the centrifuge.

Static pressure taps were located at strategic positions throughout the engine (Figure 27) to monitor pressures at various speeds. Figure 28 contains plots of the pressures as a function of speed. The pressure data indicate the flow rate and the paths taken by the cooling airflow. Figure 5, showing the direction of flow of the cooling air and the approximate flow rate, was generated by the data from Figure 28.

Figure 24. Compressor-end journal bearing foil segments after maneuver load test (top of photograph oriented toward front of engine, as installed).

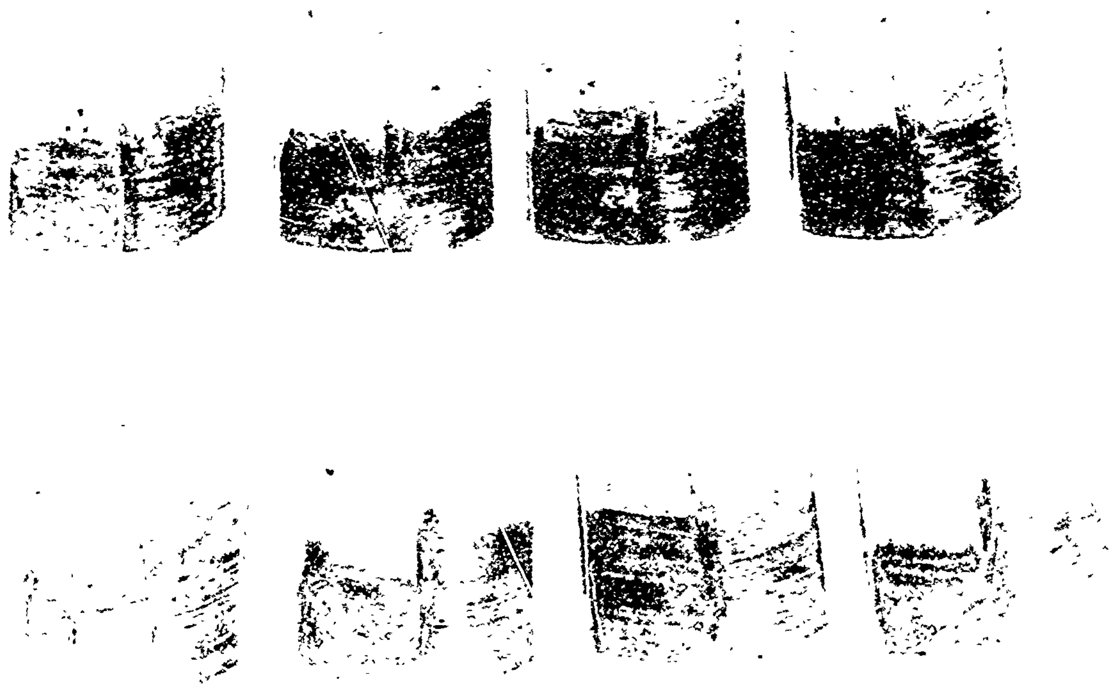

Figure 25. Turbine-end journal bearing foil segments after maneuver load test (bottom of photograph oriented toward rear of engine, as installed).

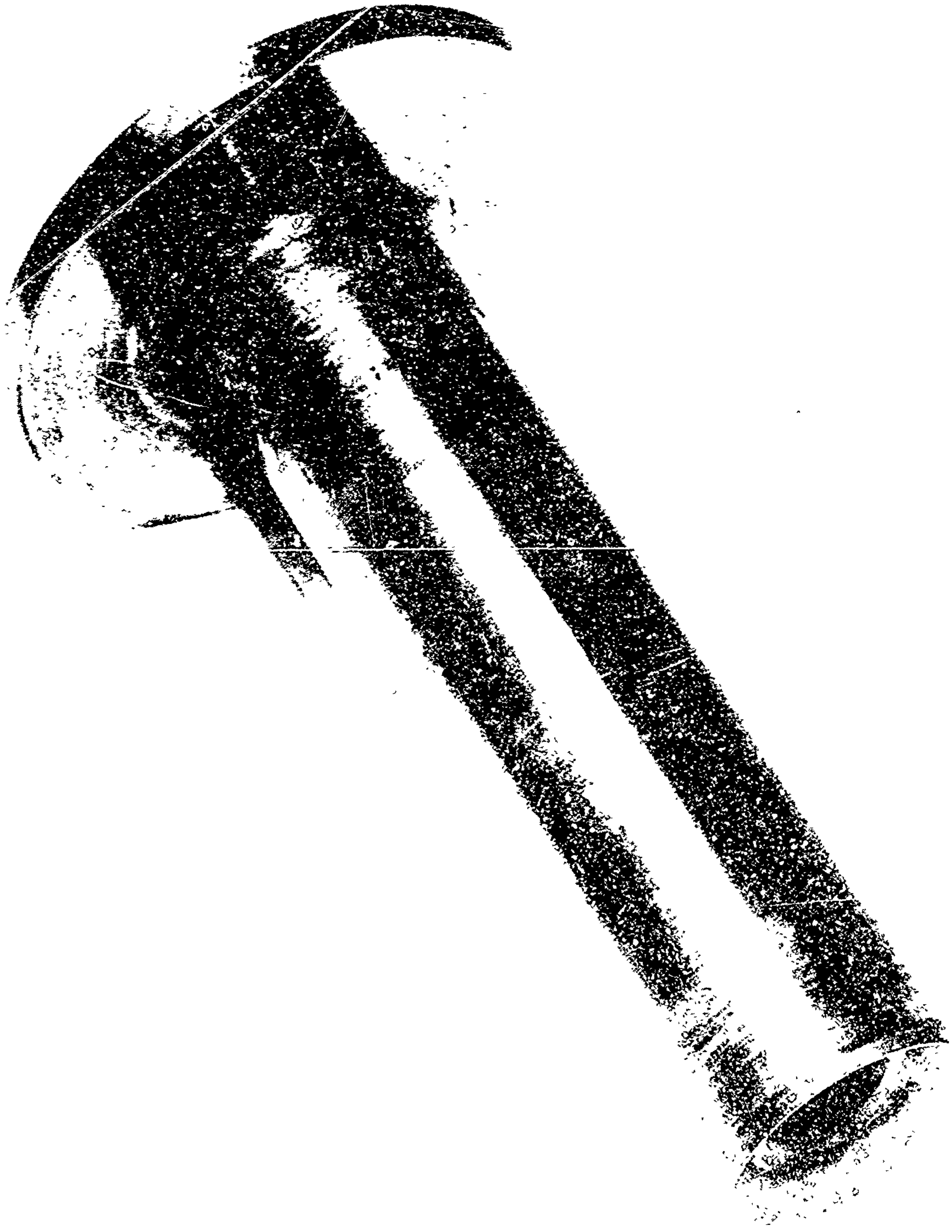

Figure 26. Journal shaft/thrust runner assembly after maneuver load test.

MP-35508

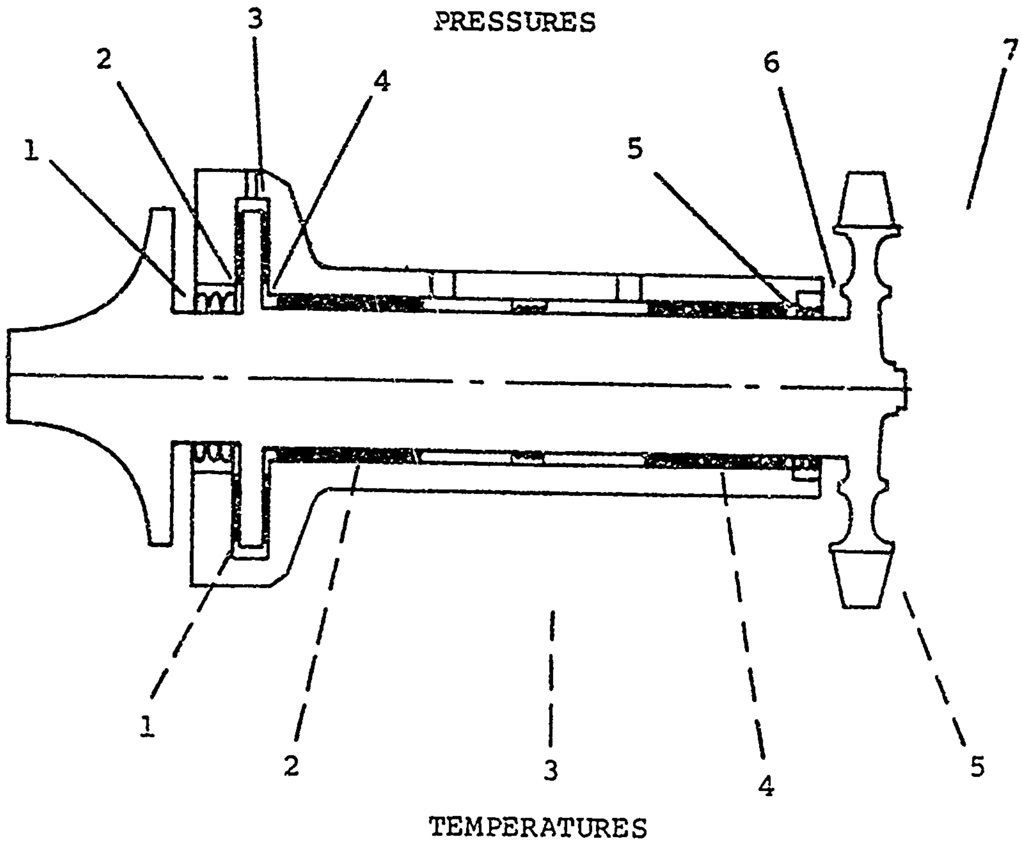

Figure 27. Instrumentation locations in foil bearing demonstrator.

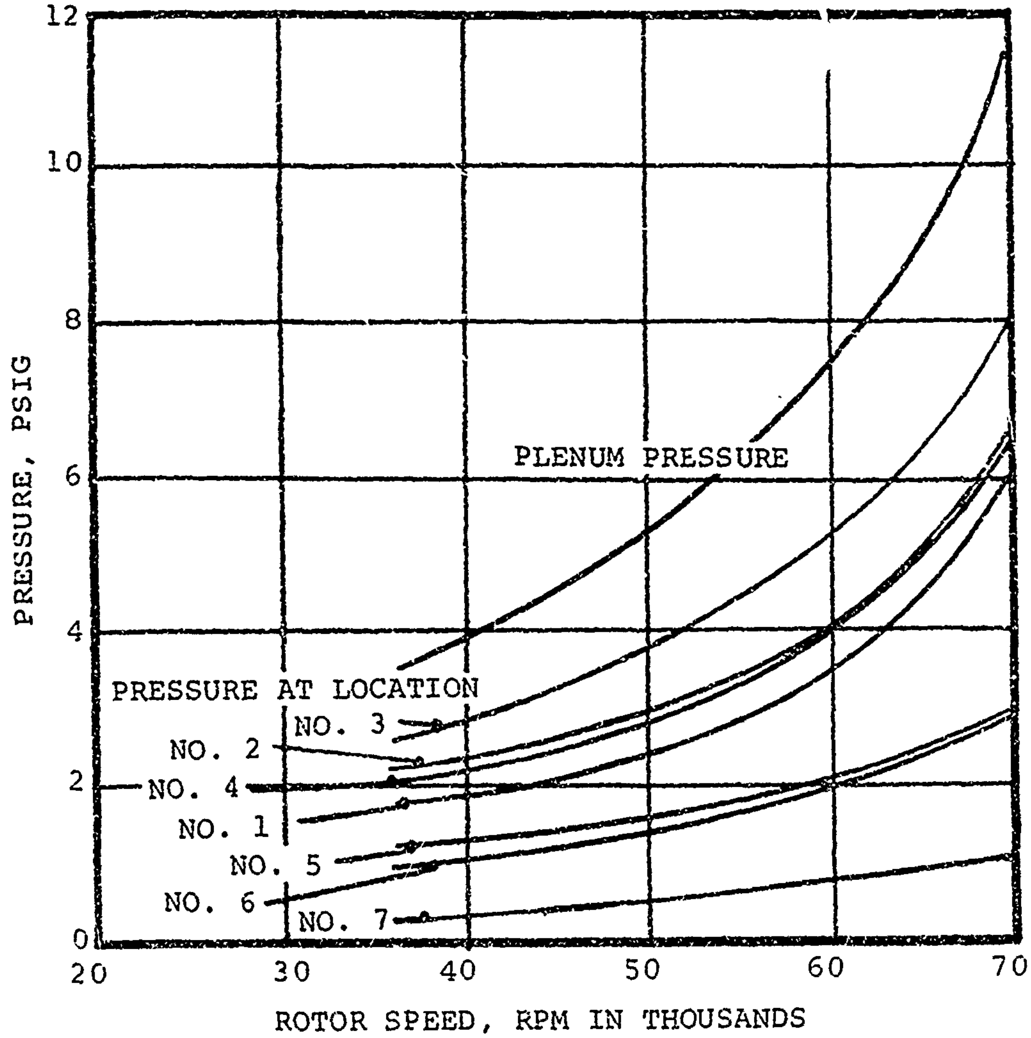

Figure 28. Cooling airflow pressure distribution in foil bearing demonstrator.

Proximity probes were utilized in the bench test to monitor shaft displacements and an oscilloscope monitored radial displacements by displaying a lissajous pattern.

When the demonstrator was started, it quickly accelerated to a steady-state speed of 48,000 rpm. A preload of 25 pounds was used for the thrust bearing, causing the rollover torque to be 24 inch-pounds. The relatively high plenum-air pressure required for breakaway resulted in the 48,000 rpm steady-state speed.

The demonstrator was operated at 71,200 rpm for about 10 minutes, until the bearing temperatures had completely stabilized. The shaft appeared to be stable as the shape of the lissajous pattern remained relatively constant throughout the speed range.

The thrust bearing temperature climbed to 460°F, the front journal bearing to 440°F, and the aft journal bearing to 385°F. It was felt that running the unit with the thrust bearing at 500°F or above constituted an unnecessary risk for the amount of potential additional load capacity. Thus, the test was concluded and the plenum air supply (290°F) was terminated. The unit rolled down to zero speed in excess of 11 seconds. During this time, the bearings received no cooling airflow* and the indicated thrust bearing temperature climbed to over 500°F. (Near the end of the rolldown and during the subsequent soakback period, the thermocouple was not suitably located to monitor the hottest part of the bearing, and it is estimated the bearing temperature was actually in excess of 600°F.) The high preload on the thrust bearing coupled with the 11-second loss of cooling flow allowed the bearing temperature to exceed the coating limitation, resulting in part of the coating adhering to the shaft. The damage was discovered when a rollover torque test was attempted and it required more than 30 inch-pound torque to rotate the shaft.

*This is not the case during a typical engine shutdown, however, where cooling air is continuously available, though in diminishing amounts, until the compressor stops rotating.

When the unit was disassembled and inspected, it was found that both sides of the thrust bearing were slightly damaged by overtemperature and some distress was noted in the backing plates, suggesting that failure occurred at the end of the rolldown. Both journal bearings were still in excellent condition and were left in the engine. The thrust bearing was replaced, and the preload was relieved somewhat to a value of 19 pounds, resulting in a rollover torque of 15 inch-pound. The unit was reassembled and shipped to Los Angeles.

3) Test No. 3

At the AiResearch Mint Canyon facility, the demonstrator was mounted on the centrifuge as shown in the schematic of Figure 29. The unit was oriented to discharge the exhaust in the direction of rotation of the centrifuge. This direction was chosen because the fluid coupling drive normally idles the centrifuge at 5 rpm, which was considered rather high for this test. Using demonstrator thrust to load the fluid drive, the idle was reduced to about 3 rpm; thus the test could be initiated at a lower centrifuge speed. Figures 30 and 31 show the test setup and Figure 32 shows the instrumentation and control consoles inside the remotely located blockhouse from which the test was monitored and controlled. An Offner 8-channel recorder was used to register the values of pertinent parameters during the test and a closed circuit television was set up for visual monitoring of the centrifuge.

The hot pressurized air supplied to the demonstrator was delivered by an AiResearch Model GTCP85-291 Auxiliary Power Unit (APU) with pressure and temperature conditioning accomplished by a remotely controlled pressure-regulating valve and a water-to-air heat exchanger. The air flowed through an overhead, insulated duct to the pivot axis of the centrifuge, through a rotating swivel joint to the centrifuge arm duct, and then to the unit.

When the installation was complete and a final calibration of the recorder was made, the demonstrator was accelerated to 30,000 rpm, using motoring air from the ground supply unit. The plenum air temperature gradually increased as the insulated air duct

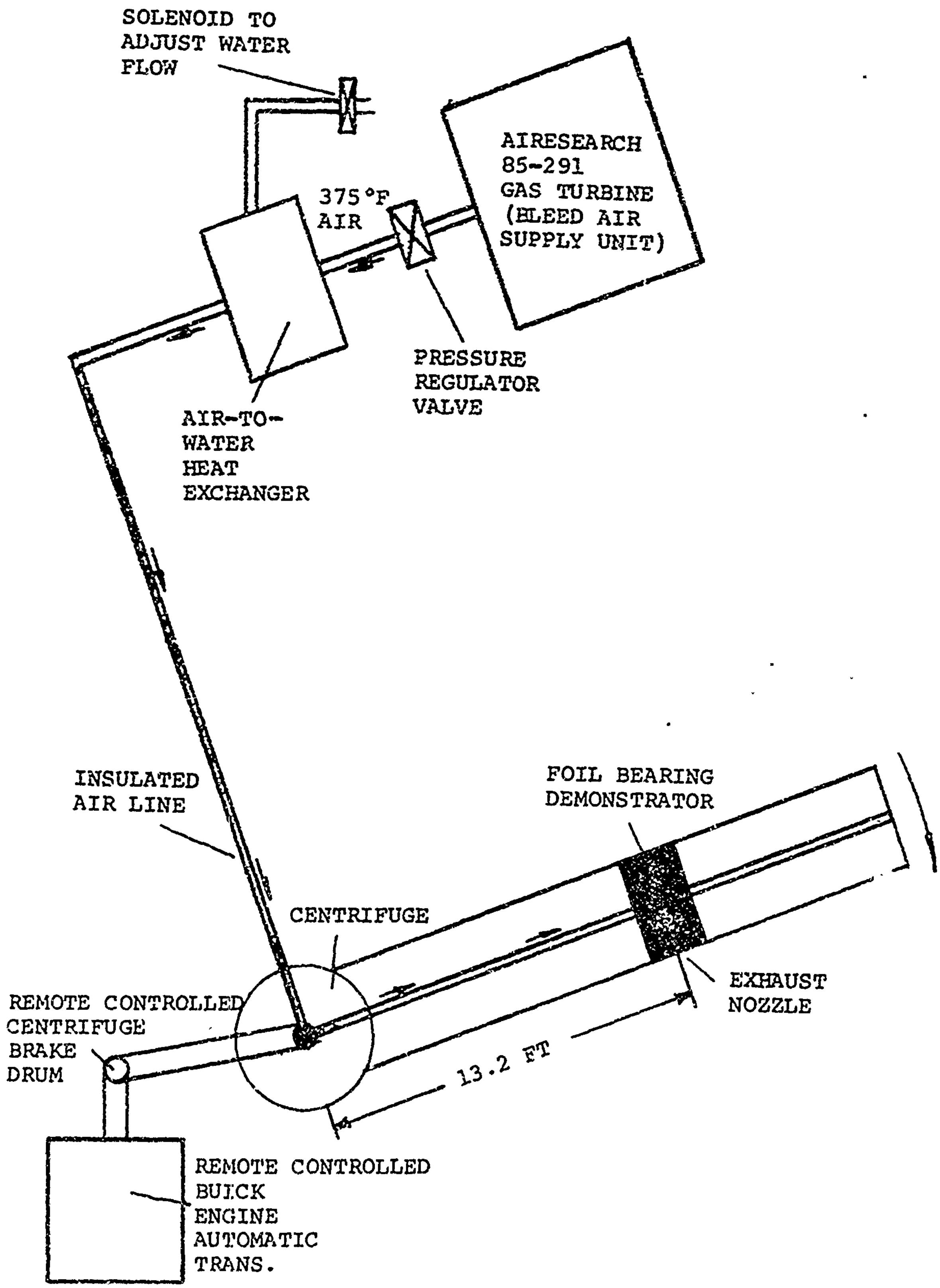

Figure 29. Schematic diagram of foil bearing demonstrator mounted in Mint Canyon centrifuge test facility.

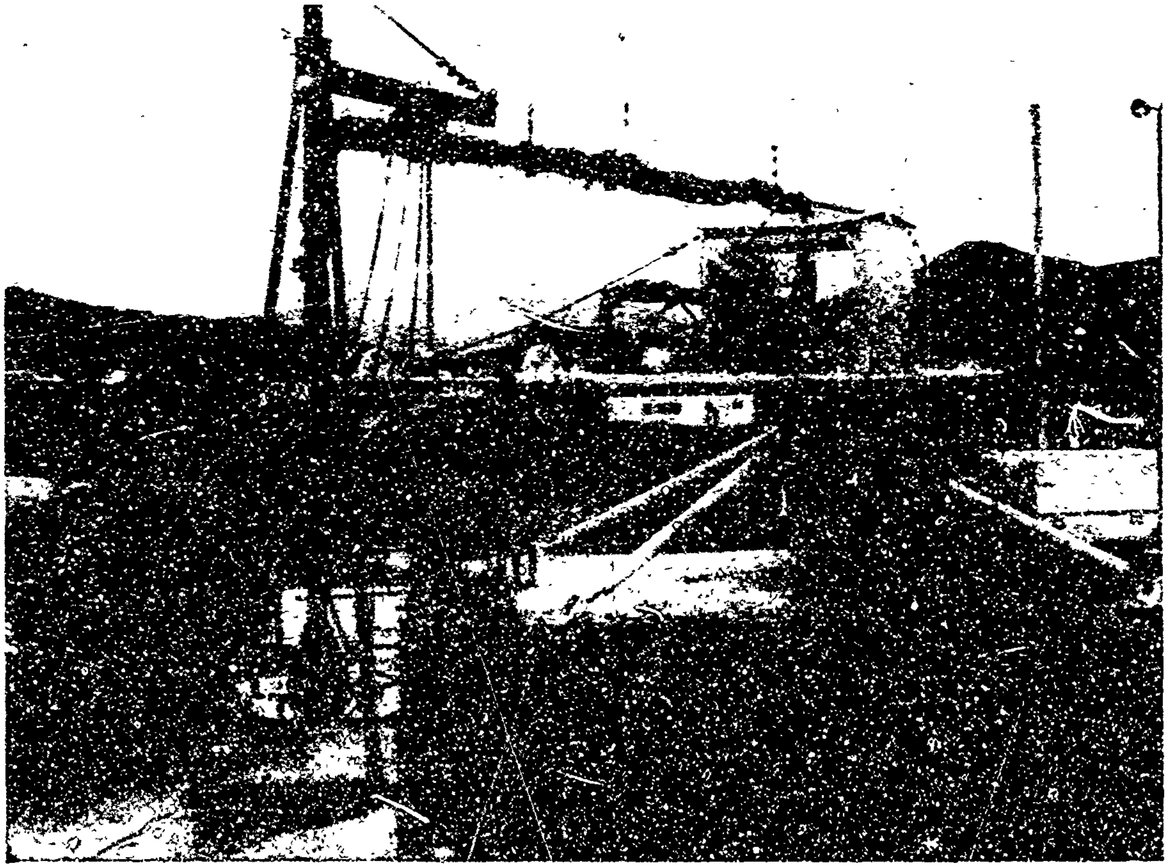

Figure 30. Overall view of centrifuge test facility.

Figure 31. Close-up view of foil bearing demonstrator mounted near end of centrifuge arm (note insulated air supply duct).

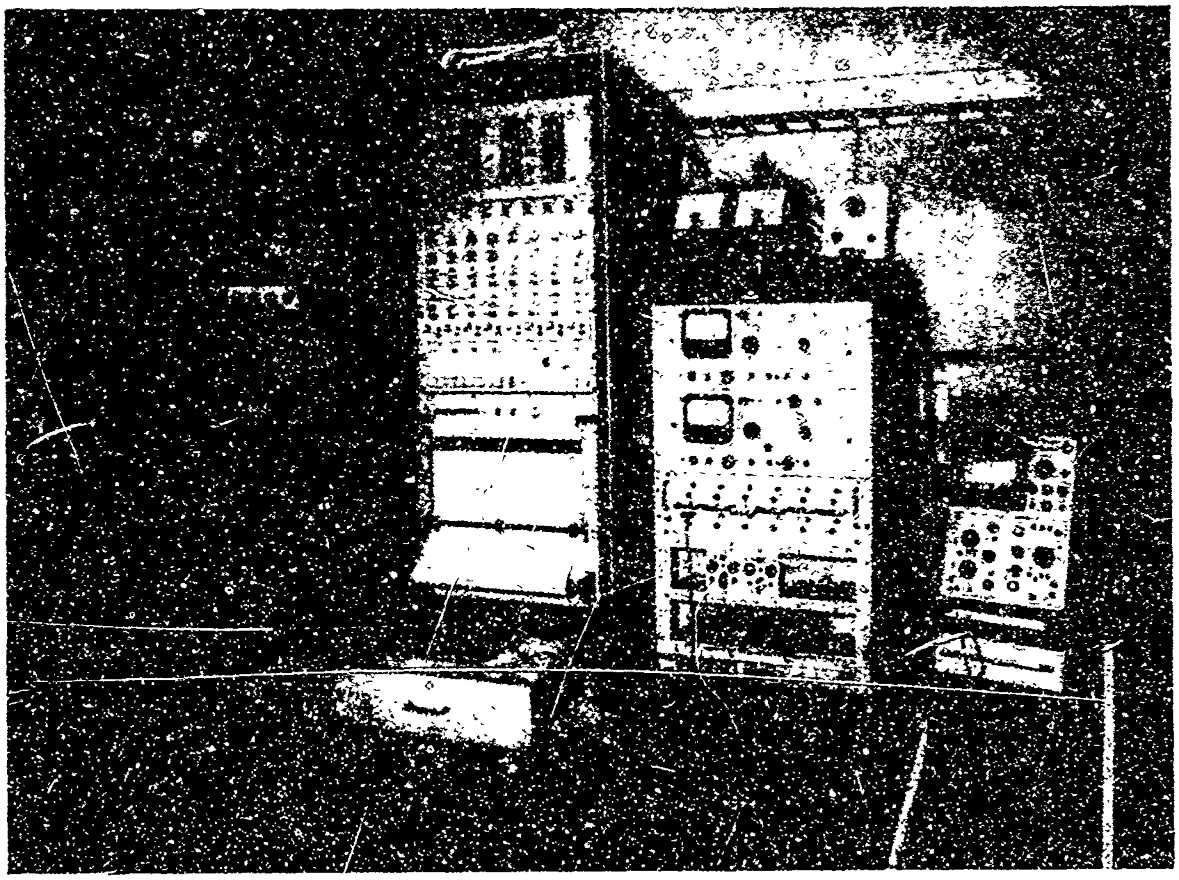

Figure 32. View of centrifuge test instrumentation showing centrifuge drive control, Offner 8-channel recorder, signal conditioner, and TV monitor.

warmed up, and the demonstrator speed stabilized at 42,000 rpm at a plenum air temperature of 260°F. Table I shows the values of pertinent parameters measured at various time intervals and taken from the Offner recording. The total run time of the demonstrator was 59 minutes, 42 seconds. The maximum speed of the demonstrator was 64,000 rpm. This speed was limited by the output pressure capability of the APU, because a large pressure drop through the air duct and the swivel joint limited the available pressure in the plenum to 8.6 psig.

The centrifuge was started, idled at 3 rpm, and then the speed was increased in increments of about 1 rpm. With the demonstrator oriented as shown in Figure 29, the compressor end of the rotating group was forced down and the turbine end up due to the induced gyroscopic movement. At a centrifuge speed of 13.6 rpm, the ultimate load capacity of the compressor-end journal bearing was exceeded. Shaft contact with the foils resulted in a sharp rise in temperature, culminating in seizure. The Offner recording of the last seconds of the test is presented in Figure 33.

The demonstrator was returned to Phoenix where it was disassembled and inspected. The failure of the compressor-end journal bearing was verified, but the turbine-end journal bearing was still in good condition. The coating on both sides of the thrust bearing was damaged from excessive heat, appearing somewhat like the thrust bearing that underwent the checkout bench test at Phoenix (Test 2). It is felt, however, that both these thrust bearings were damaged after the tests were completed, due to soakback of heat generated during the compressor journal bearing seizure, since neither the foil metal nor the backing plates displayed distress. Figure 34 shows both sides of the thrust foil bearing and Figure 35 shows the compressor side of the thrust runner. Figure 36 shows the turbine-side of the thrust runner, along with the turbine end of the journal shaft, which was still in excellent condition. The shaft was rigidly welded to the compressor-end foils and bearing carrier and could not be pressed out of the carrier. Although a certain amount of welding of the foils occurred during other bearing failures, the shaft could always be pressed out of the bearing carrier.

TABLE I

SUMMARY OF RECORDED PARAMETERS
DURING MANEUVER LOAD TEST
AT MINT CANYON FACILITY

Centrifuge Speed, rpm	Time, Sec.	Demonstrator Speed, rpm	Plenum Air Temp, °F	Thrust Bearing Temp, °F	Front Journal Bearing Temp, °F	Back Journal Bearing Temp, °F	Plenum Press, psig
	0	0	65	65	-	-	-
	8	30,000	75	65	65	70	2.1
	73	40,000	120	145	100	95	3.1
	508	50,000	240	375	290	260	4.8
	728	60,000	265	440	340	305	7.1
	1723	61,000	220	395	300	265	8.2
	1983	62,500	220	415	315	280	8.6
	2603	64,000	260	470	360	320	8.5
Start Centrifuge	2768	64,000	260	475	365	330	8.5
6 rpm	3108	64,000	255	475	360	320	8.5
12 rpm	3463	64,000	250	480	360	325	8.5
13.6 rpm	3582	64,000	250	480	365	330	8.4

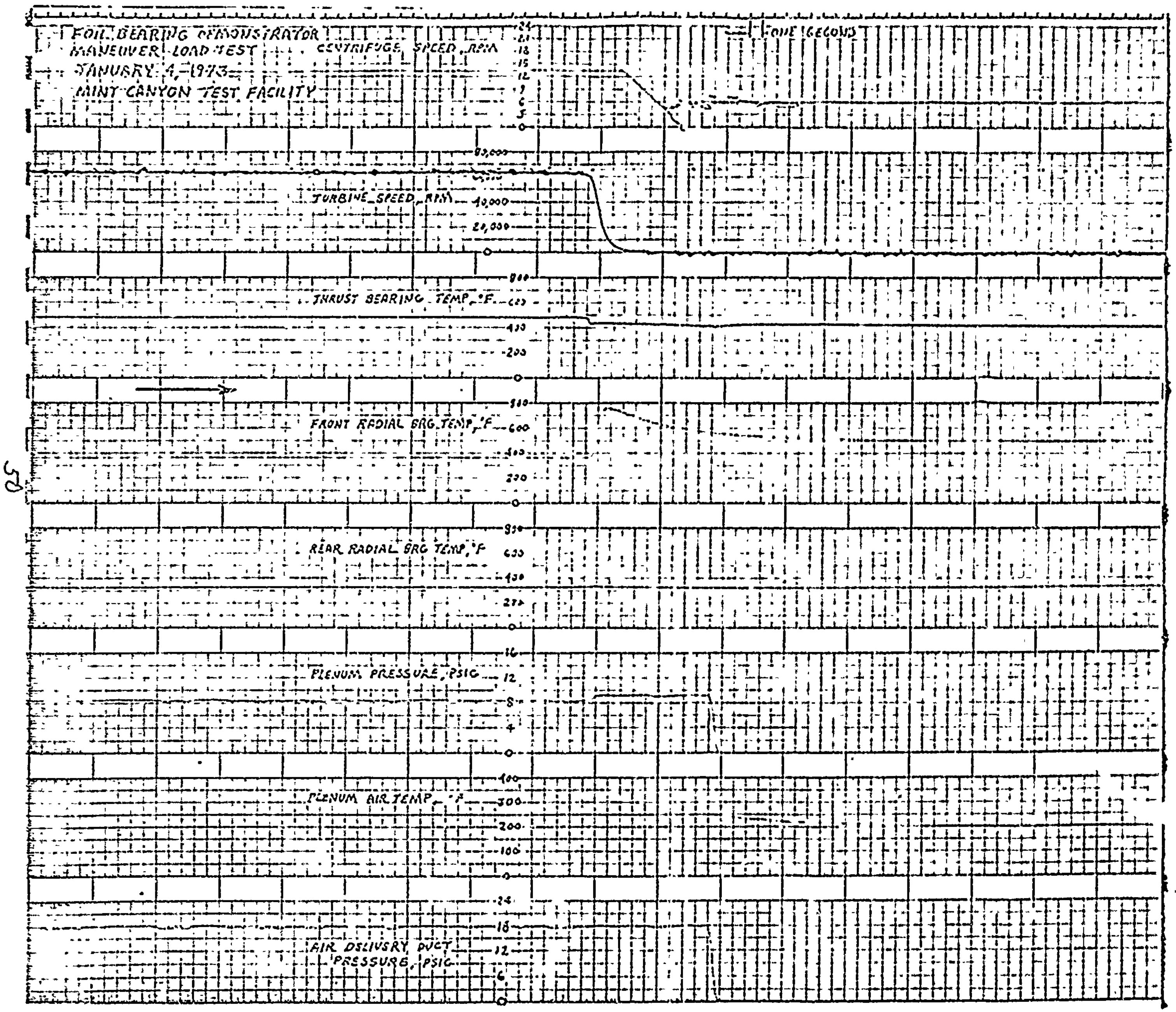
FOIL BEARING DEMONSTRATOR
MANEUVER LOAD TEST
JANUARY 4, 1973
MINT CANYON TEST FACILITY
CENTRIFUGE SPEED, RPM
ONE SECOND
TURBINE SPEED, RPM
THRUST BEARING TEMP, °F
FRONT RADIAL BRG TEMP, °F
REAR RADIAL BRG TEMP, °F
PLENUM PRESSURE, PSIG
PLENUM AIR TEMP, °F
AIR DELIVERY DUCT PRESSURE, PSIG

Figure 33

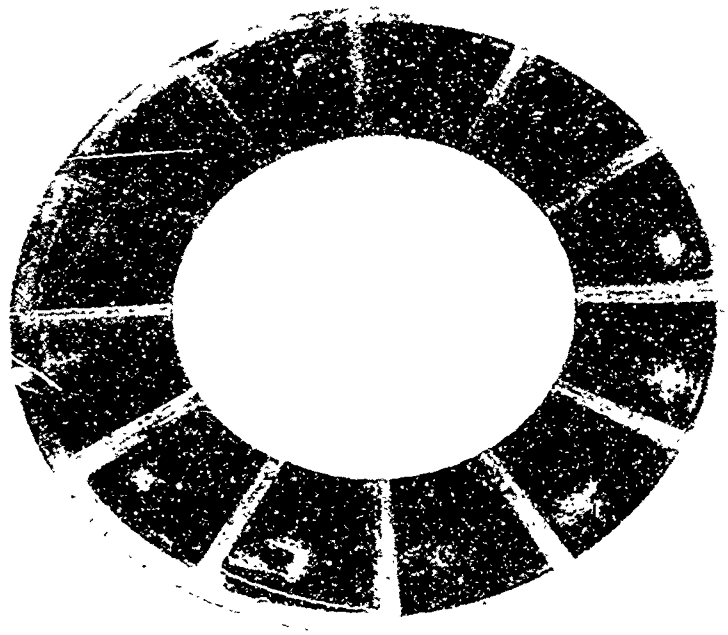

Figure 34a. Forward thrust bearing surface.

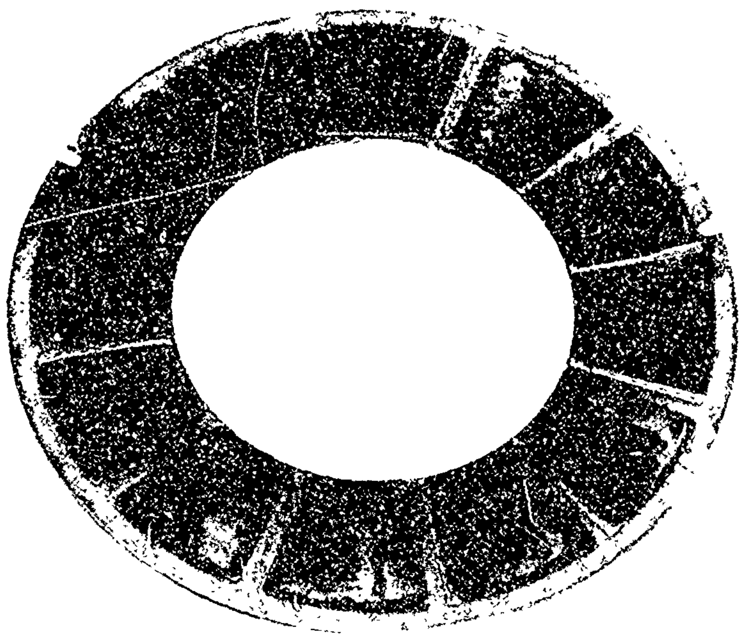

Figure 34b. Aft thrust bearing surface.

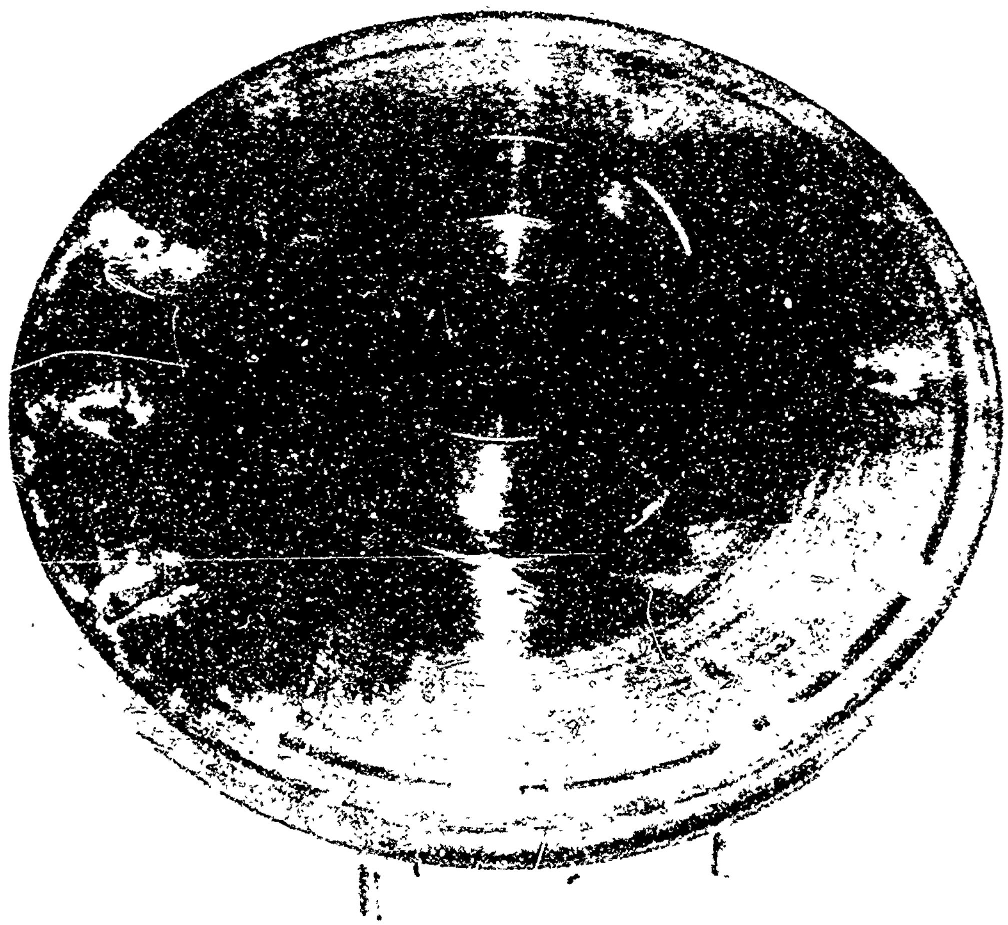

Figure 35. Forward thrust runner face.

MP-36885

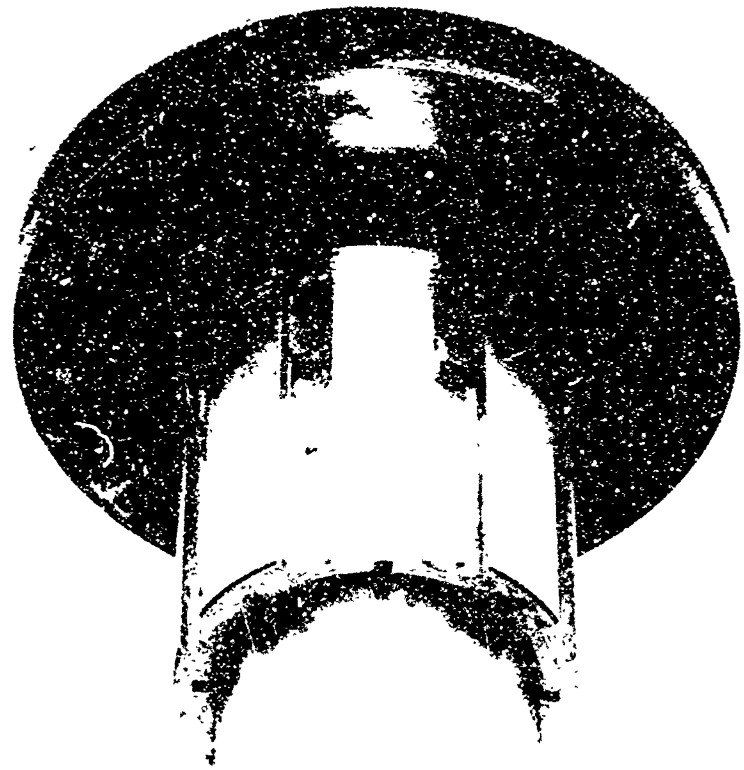

Figure 36. Aft thrust runner face and journal shaft.

In prior tests, however, the shaft outer surfaces had a hardness of Rockwell C-60 or better. For this test the outer surface hardness of t..e shaft was Rockwell C-40, and hence was less resistant to the tendency to "weld" than the other shafts.

Table II presents values of the pertinent parameters at the time of bearing seizure.

TABLE II

PERTINENT TEST INFORMATION
FROM PHOENIX AND MINT CANYON
MANEUVER LOAD TESTS

	Phoenix Test	Mint Canyon Test
Demonstrator speed, rpm	52,000	64,000
Terminal centrifuge speed, rpm	11	13.6
Centrifuge arm length, inches	120	158
Rotor Ip, in.-lb-sec^2	0.0138	0.01375
Rotor weight, lbs	4.88	4.86
Terminal-generated gyroscopic moment, in.-lb	86.61	131.7
Terminal-generated centrifugal load, g simultaneous with gyroscopic moment	0.41	0.83

In an effort to determine the load split between the journal and thrust bearings, a force diagram of the rotor was constructed as shown in Figure 37.

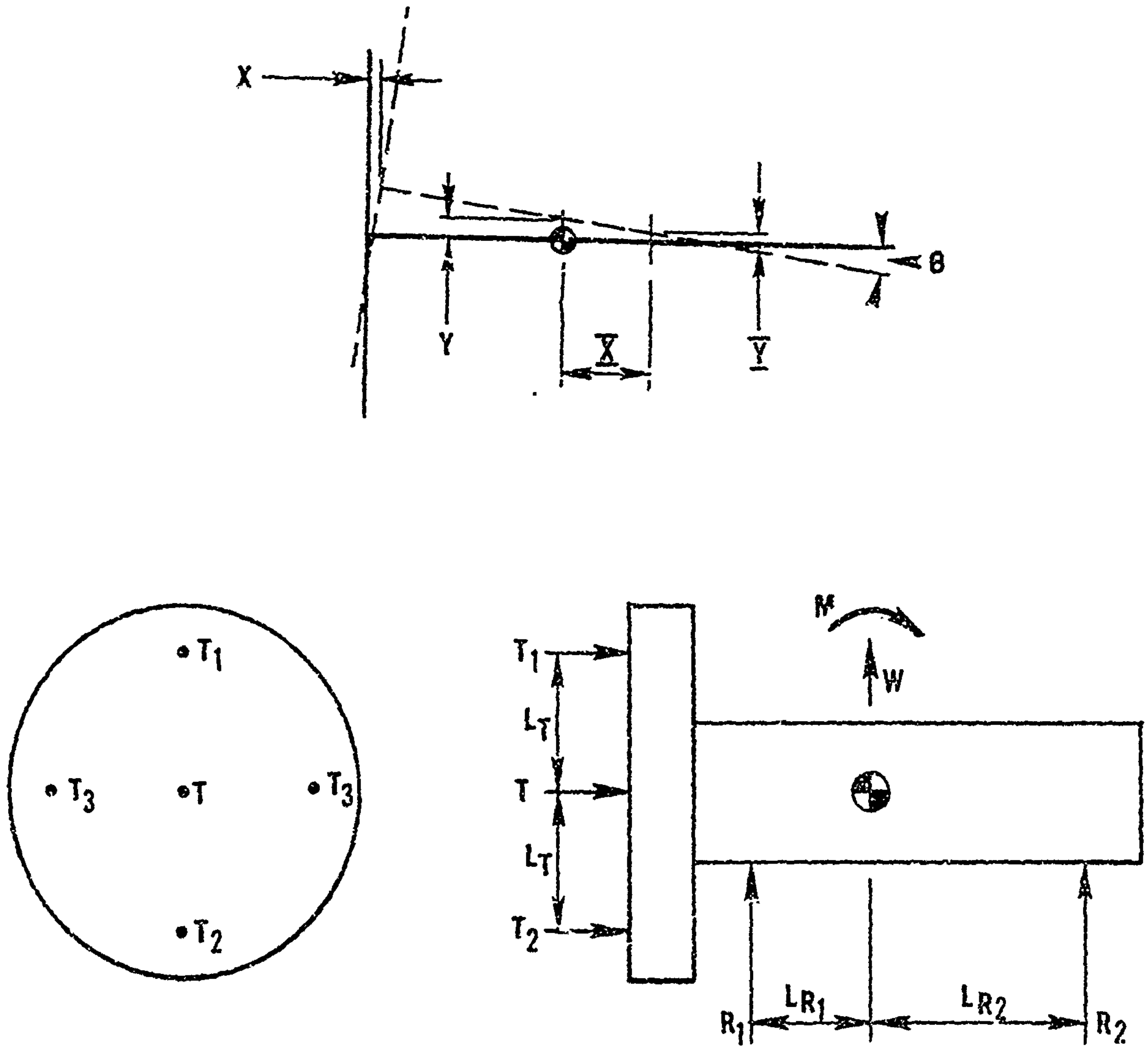

Figure 37. Analytic free-body of foil bearing demonstrator rotating group.

Before forces could be defined, however, load/deflection curves for both the thrust and journal bearings were required. These are presented in Figure 38 for the thrust bearing and Figure 39 for the journal bearing. The thrust bearing springrate was generated, assuming a 19-pound preload. There are 12 nonlinear springs on each side of the thrust bearing, but only half are effective for resisting moment loading. The amount of preload on the springs affects the load/deflection curve and is representative of a purely axial load, equal on both sides of the bearing.

From Table II, the imposed gyroscopic moment on the rotating group was 131.7 inch-pounds. Using a calculated aerodynamic thrust of 30 pounds acting forward on the rotating group, solutions of force and equilibrium equations yield the average loads on the thrust and journal bearings. The solutions are shown in Table III, which indicates the loads on each bearing resulting from the gyroscopic torque, g-loading, and aerodynamic thrust. The last row indicates the total unit loading on the bearings resulting from the combined loading. Figure 40 is an isometric view of the shaft with the direction and magnitude of each source of loading. Of the 131.7 inch-pounds of torque generated, the thrust bearings resisted 61.96 inch-pounds or about 47 percent and the journal bearings resisted 69.74 inch-pounds or about 53 percent of the load just before the load limit of the compressor journal bearing was exceeded.

The average unit load capacities are lower by about 15 percent than those predicted by test rig load capacity data, since the following engine phenomena were omitted in the accumulation of the data.

The demonstrator incorporated thrust and journal bearings, both of which carried relatively equal and heavy moment loads, meaning that both of their alignments and orthogonalities had to be held instead of just those of the journal bearing, as in the case of a journal bearing test rig.

Also, precautions are taken on a test rig to maintain a symmetrical load profile for the bearings and to strive for maximum magnitude of the entire bearing length. Misalignment, angularity, and unsymmetrical loading are avoided during bearing performance tests to obtain repeatable data. When the bearings resist an imposed moment on the shaft, the shaft assumes a skewed position relative to the bearings as shown (exaggerated) in Figure 41.

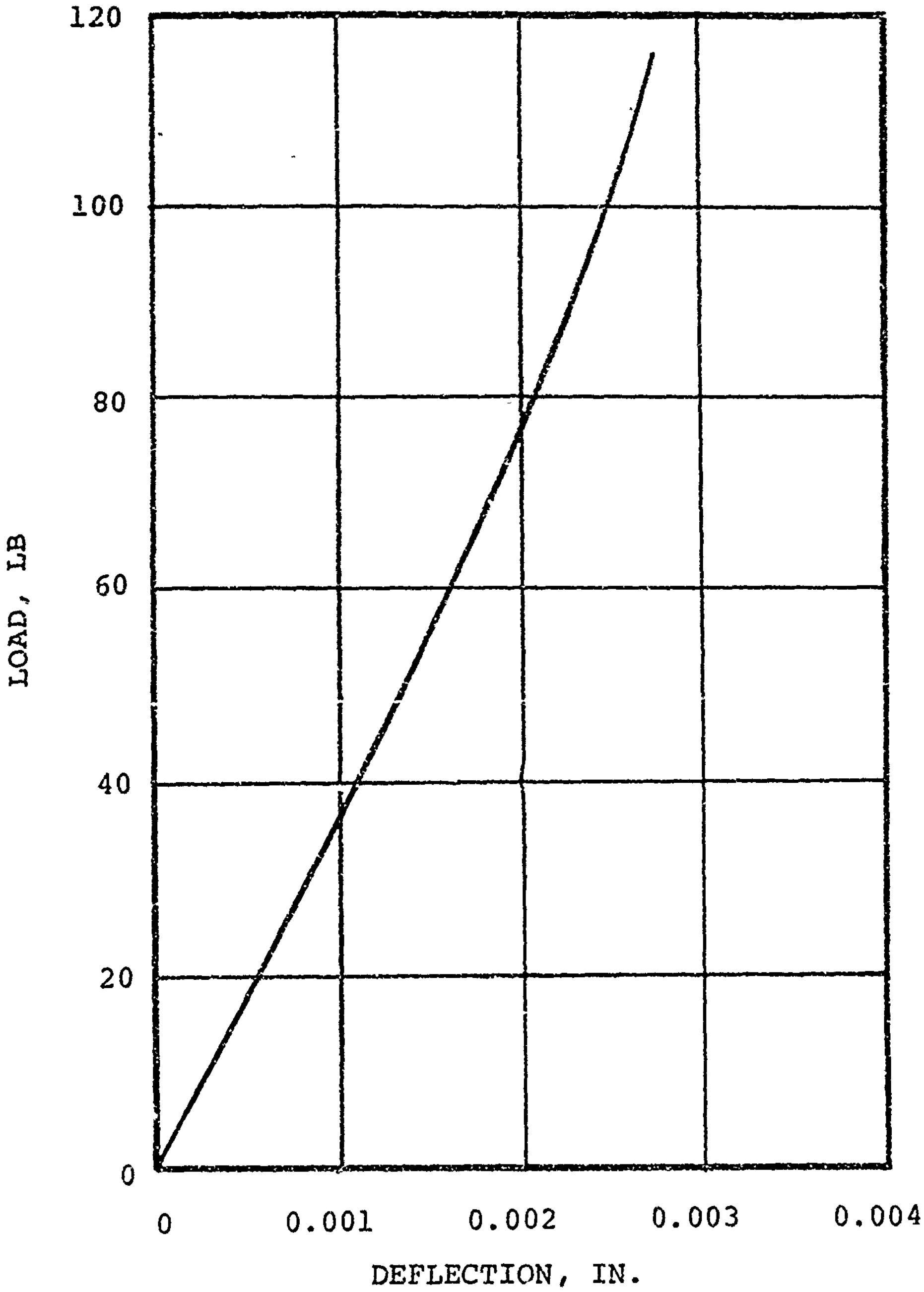

Figure 38. Axial thrust foil bearing load-deflection characteristics.

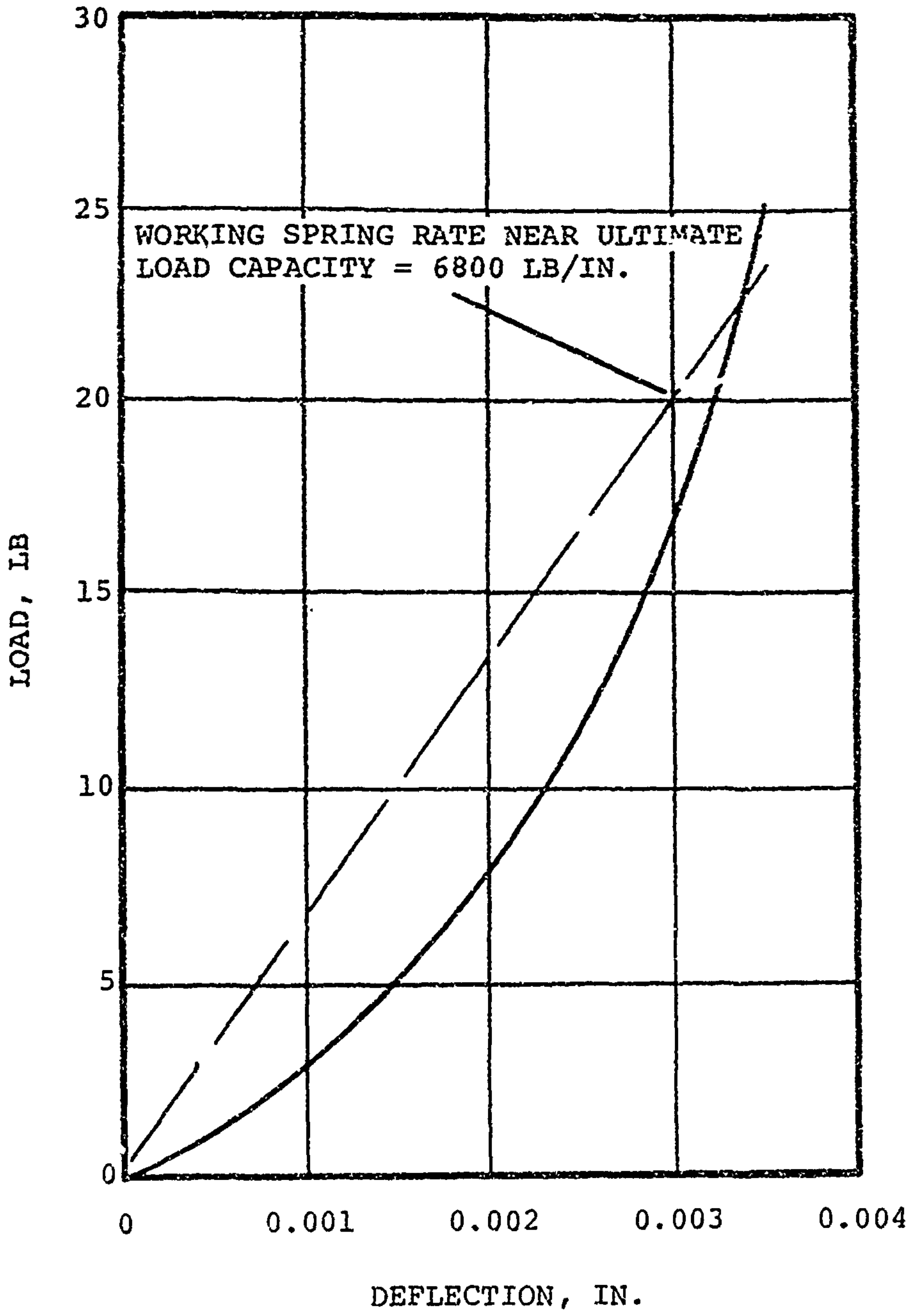

Figure 39. Radial thrust foil bearing load-deflection characteristics.

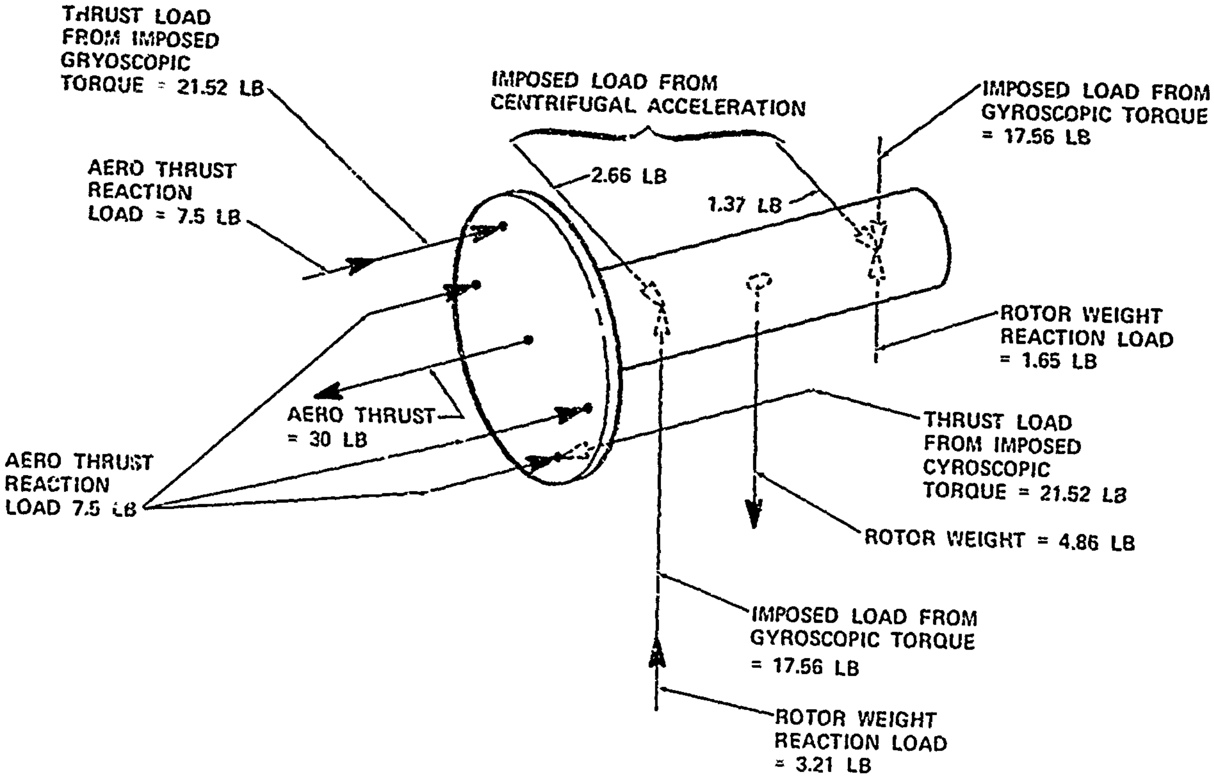

Figure 40. Isometric diagram of imposed loads upon compressor journal bearing seizure.

TABLE III

RESULTS OF BEARING FORCES
AND AVERAGE LOAD
CARRIED BY EACH BEARING
JUST BEFORE LOAD CAPACITY OF COMPRESSOR
JOURNAL BEARING WAS EXCEEDED DURING
MINT CANYON MANEUVER TEST
(LOADS SHOWN PICTORIALLY IN FIGURE 40)

Parameter	Compressor Journal Bearing	Turbine Journal Bearing	Bottom Turbine Thrust Bearing	Top Compressor Thrust Bearing	Middle Section Thrust Bearing
Load due to gyro torque (lb)	17.56	17.56	21.52	21.52	--
Average unit load from gyro torque (psi)	7.32	7.32	13.49	13.49	--
Load due to rotor weight (lb)	3.21	1.65	--	--	--
Average unit load from rotor weight (psi)	1.34	0.69	--	--	--
Load due to centrifugal "g" forces (lb)	2.66	1.37	--	--	--
Average unit load from centrifugal "g" forces (psi)	1.11	0.57	--	--	--
Load due to aero-dynamic thrust (lb)	--	--	7.50	7.50	15.00
Average unit load, due to aerodynamic thrust (psi)	--	--	4.70	4.70	4.70
Resultant load (lb)	20.94	15.97	14.02	29.02	15.00
Resultant average unit load (psi)	8.73	6.65	8.79	18.20	4.70

The loss of capacity in the bearing thus became a measure of its compliancy, or self-aligning capability. The skewed position of the shaft, when resisting a moment loading, decreases the effective area in the bearing that can generate the maximum load capacity. Hence the bearing demonstrates less overall load capacity than is observed in test rig data.

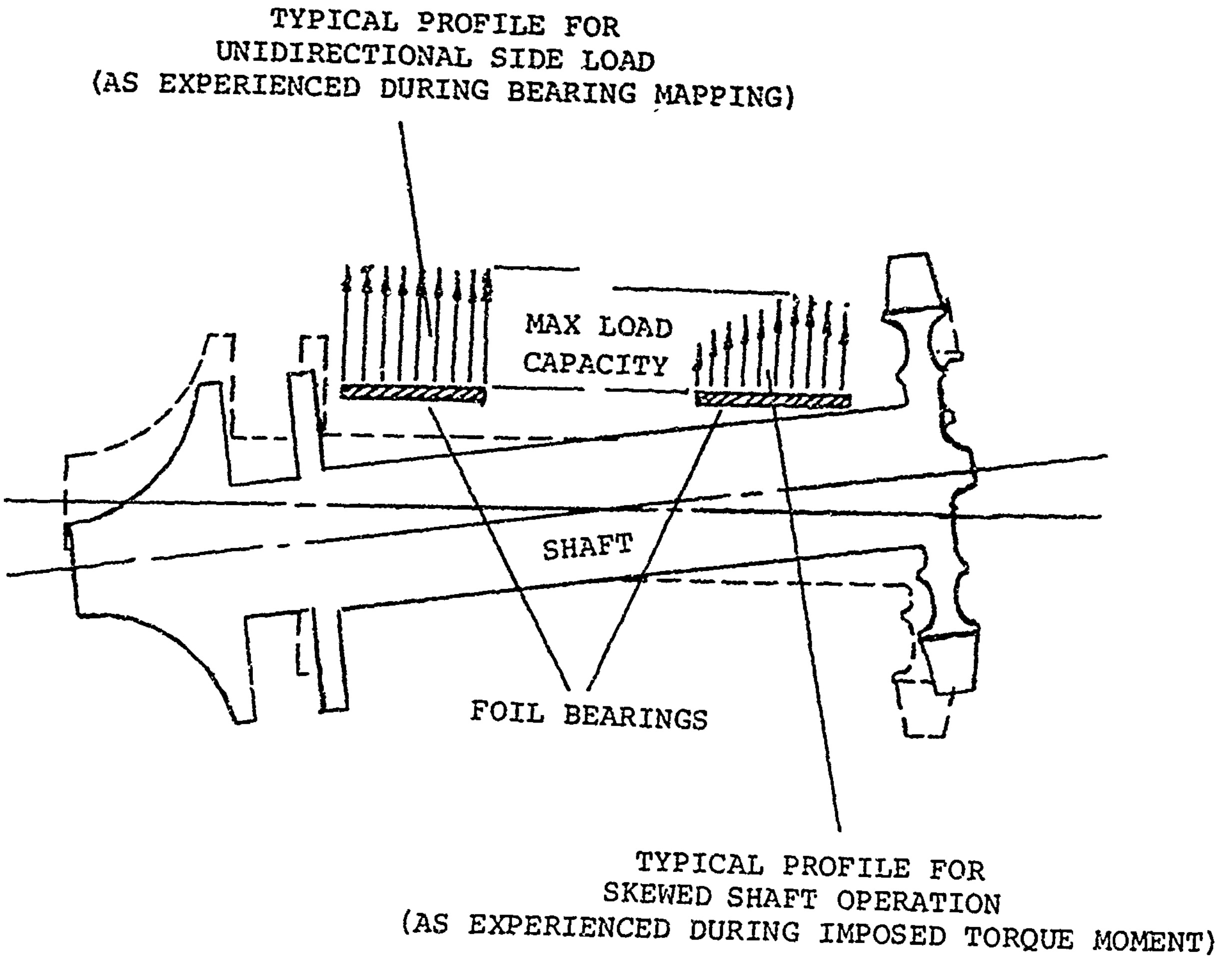

Figure 41. Coaxial and skewed shaft journal bearing load profile comparison.

d. Vibration Test

The vibration test for the foil bearing demonstrator was conducted March 13, 1973. The unit was built in the air-motoring configuration, as it was for the maneuver loading tests. Figure 42 shows the engine mounted in its adaptor on the vibration table.

The unit was operated at full speed (72,000 rpm) throughout the test. The vibration loading test program followed loading specifications of Curve C, Figure 514.1 of MIL-STD-810B, repeated here as Figure 43.

The large bearing carrier was equipped with one radial and one axial accelerometer, mounted inside the engine. Six proximity probes were installed just forward of the thrust bearing. Two probes (numbers 1 and 2) measured axial displacement. The remaining four probes (numbers 3 through 6) were radial probes mounted 90 degrees apart. Two probes (numbers 1 and 3) were inoperable at the time of the test. Hence, three radial probes and one axial probe were utilized to record the shaft excursion data.

The test was initiated by exciting the vibration table at the low frequencies, establishing the 0.10-inch double amplitude. After completion of the 1 g sweep, and while transferring to the 5 g line at about 53 cps, a structural resonance was encountered, multiplying the g-load from the shaker table into the bearing carriers. The high vibration forced the shaft into wide excursions, resulting in bearing rubbing and friction welding. Figure 44 presents the values of the pertinent parameters during the test as taken by a tape recorder. The tape was run through a Sanborn recorder and a real-time analyzer to aid in analyzing the test results.

Initial speed drop occurred at an excitation frequency of 53 cps and a table vibration of about 5 g. The total bearing carrier response indicated 28 g: 17 g at 1160 cps and 11 g at 53 cps. Note that 1160 cps corresponds to the engine speed and 53 cps the table-driving frequency. The corresponding amplitudes were 0.00012 inch at the high frequency and 0.038 inch at the low frequency. At the time of initial rotor speed reduction, shaft excursions relative to the carrier were decreasing, possibly indicating a stronger, more rigid bearing. This may have been caused by overheating and increased air viscosity or interference from the overheating foil coating.

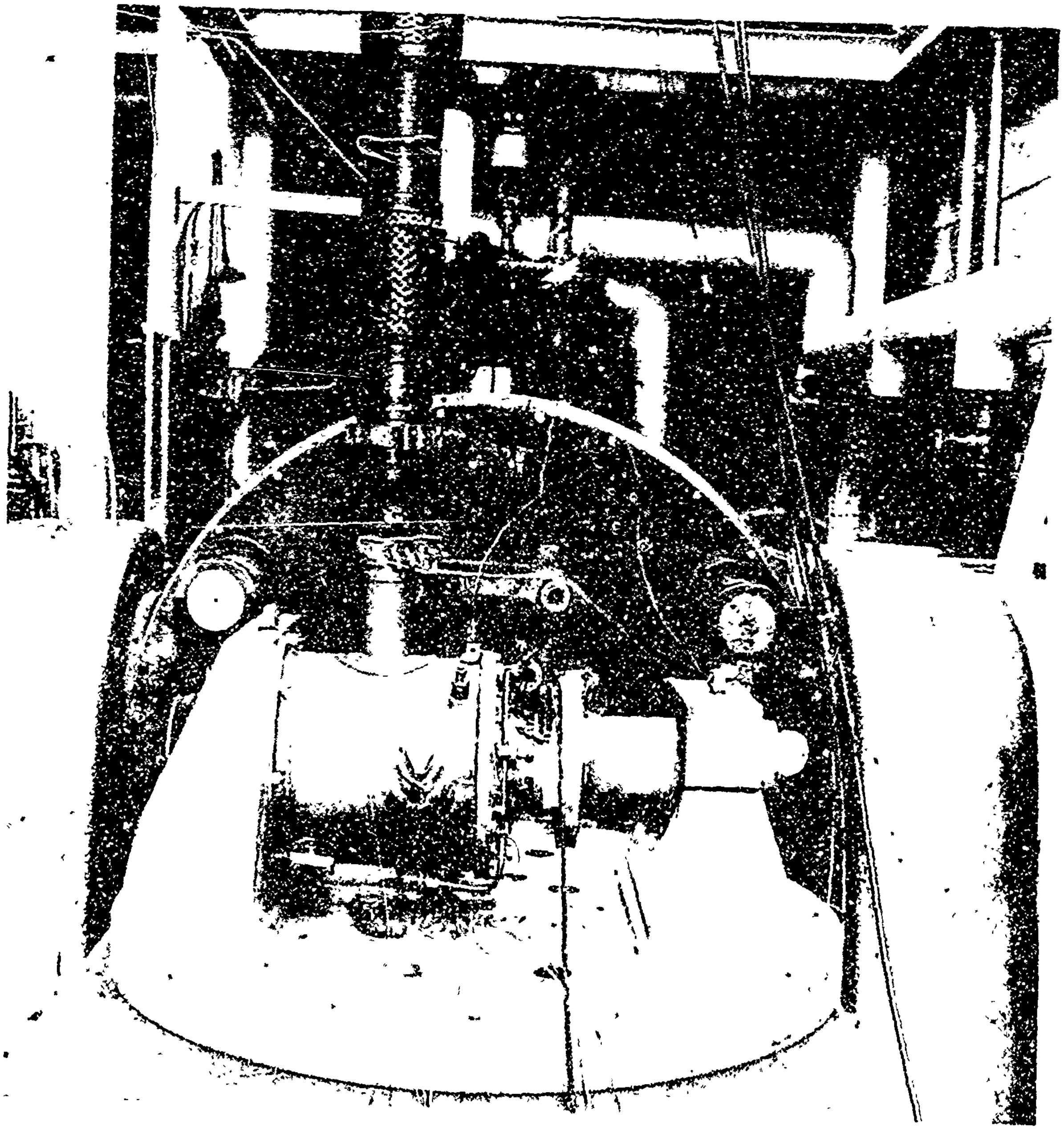

Figure 42. Side view of foil bearing demonstrator mounted on vibration table.

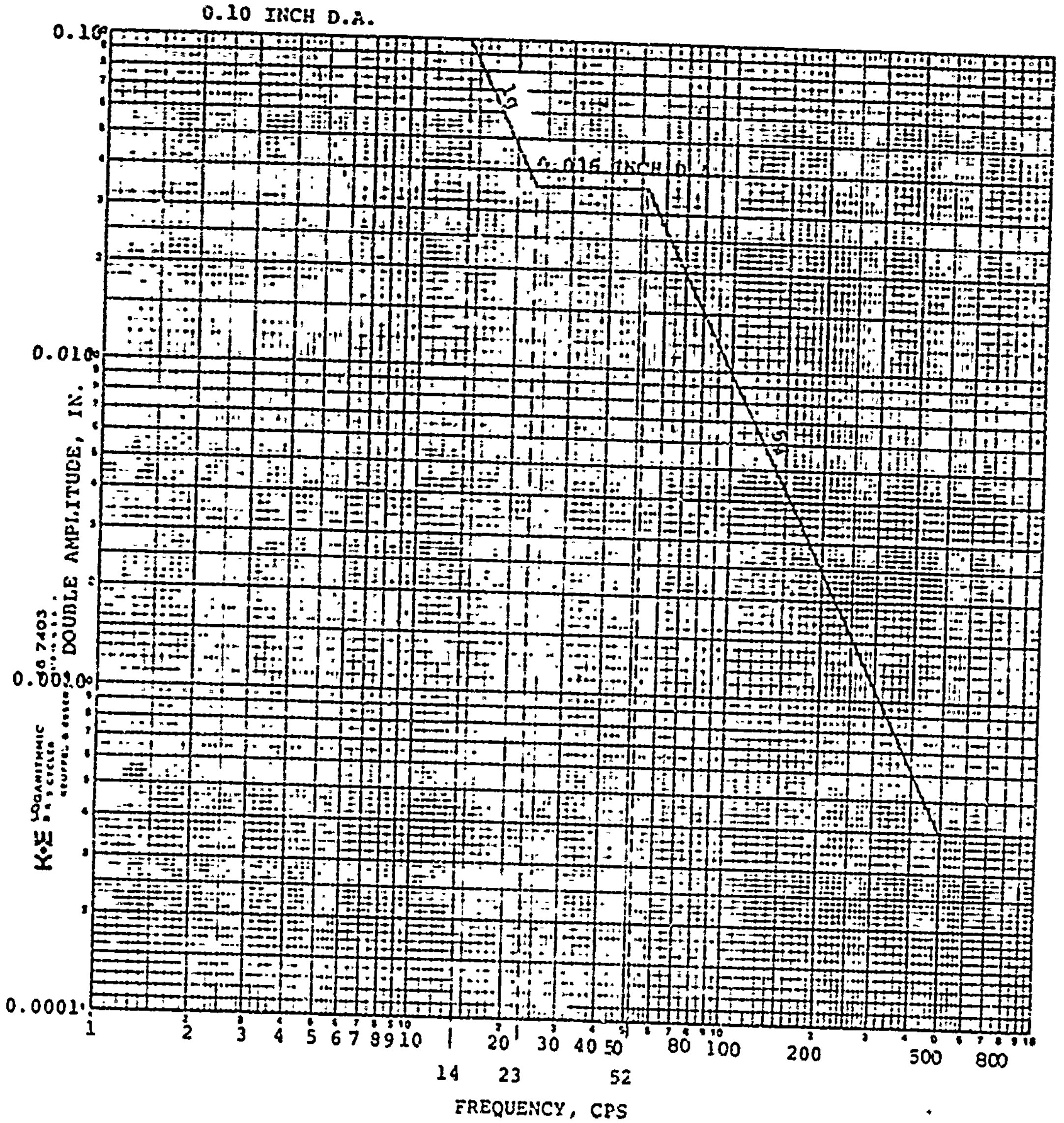

Figure 43. Vibration Curve "C" Figure 514.1 in MIL-STD-810B.

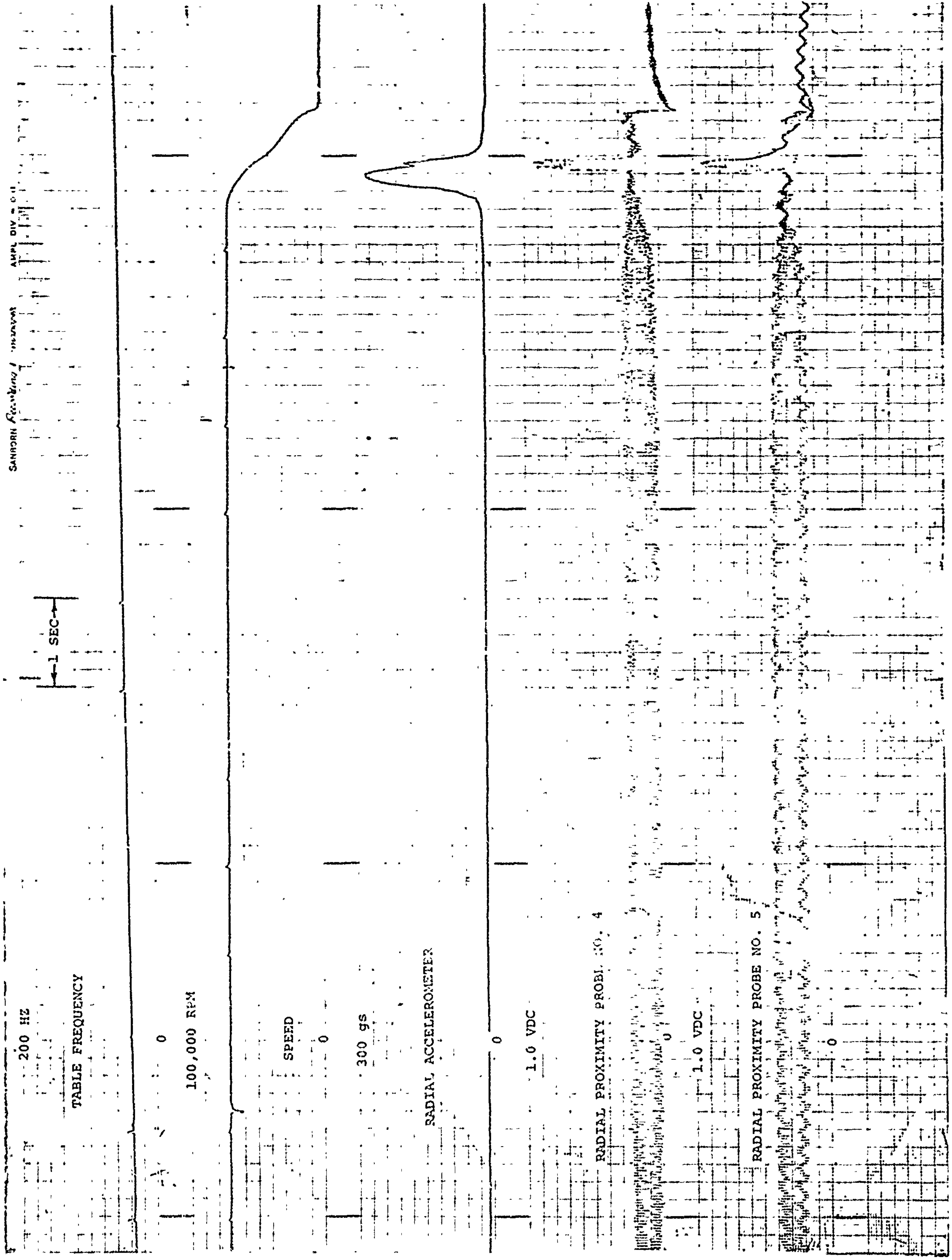

Figure 44. Oscillograph recording of vibration test.

After the test, the unit was disassembled and inspected. The compressor journal bearing had failed and the foils were inertia-welded into the shaft. The remaining bearings, however, were in relatively good condition. No damage was incurred by the turbine or other parts of the unit except for the foils, the immediate bearing carrier, and the shaft.

Figure 45 shows the shaft as it came out of the engine. Slight residual damage is visible at the turbine bearing end of the shaft, which probably occurred after the front bearing failed and the shaft experienced large displacements. Slight damage is also visible at the inner diameter of the turbine thrust bearing, which is adjacent to the failed journal bearing where high temperature soakback damaged the foil coating.

Figure 46 shows the remaining turbine journal bearing which is in relatively good condition although parts of the foil coating were scuffed. The bearing did not overheat and, from the color of the foil coating, did not experience temperatures above 500°F.

A renewed critical speed analysis was conducted with the bearing springrates increased to 10,000 lb/in. rather than 5000 lb/in. used in the previous analysis presented in Figure 47. The new springrate was considered to be more realistic in view of the high shaft displacements involved. The resulting critical speeds were 166 cps (9970 rpm) and 262 cps (15,700 rpm). Figure 48 shows the amplification factors in a simplified single degree of freedom system with the above natural frequencies. The structural response was detected from the data of the vibration test. The rotor resonances were predicted from the critical speed analysis. Using the data from Figure 2, Figure 3 was generated to show the amplification factor of the rotor with respect to the shaker table. The peak amplification factors are 9.0, 2.9, and 1.1. The high reduction of the two rotor resonances (critical speeds) results from isolation by the structure. This leads one to believe that the structural resonance of 53 cps at the bearing carrier is the worst vibrational loading condition for the bearings, had the unit survived the entire vibrational scan. However, there may be other structural resonances which more closely match the rotor critical speed frequencies, not providing the isolation shown in Figure 3. A problem such as this would require development to "tune" the structural frequencies away from critical speed frequencies.

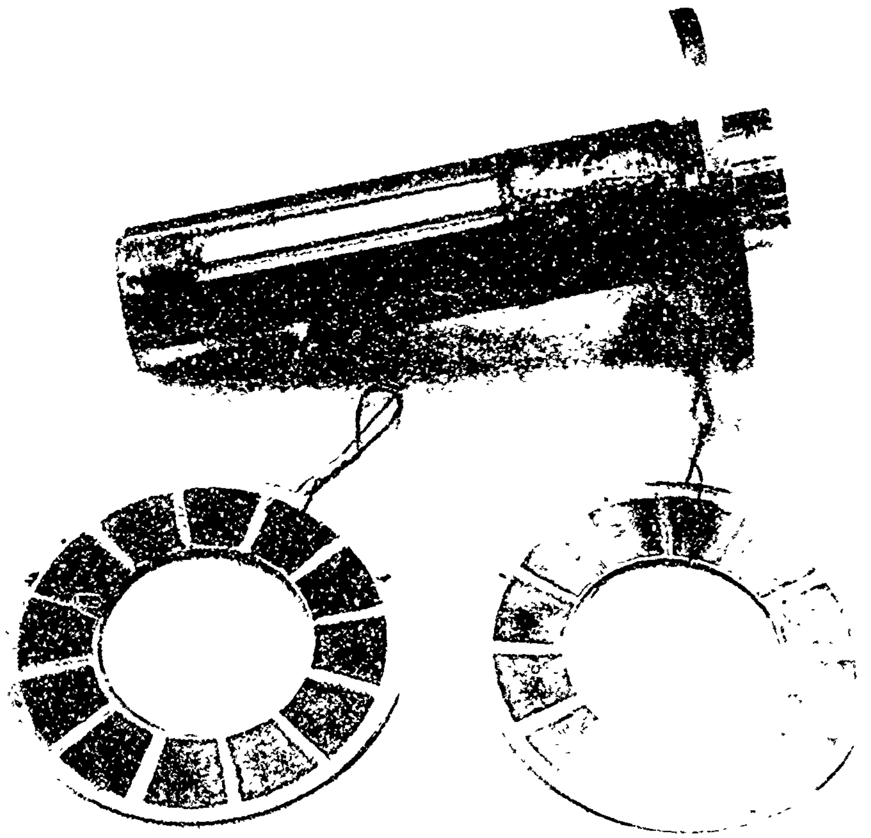

Figure 45. View of shaft and two thrust bearings after disassembly upon completion of vibration test.

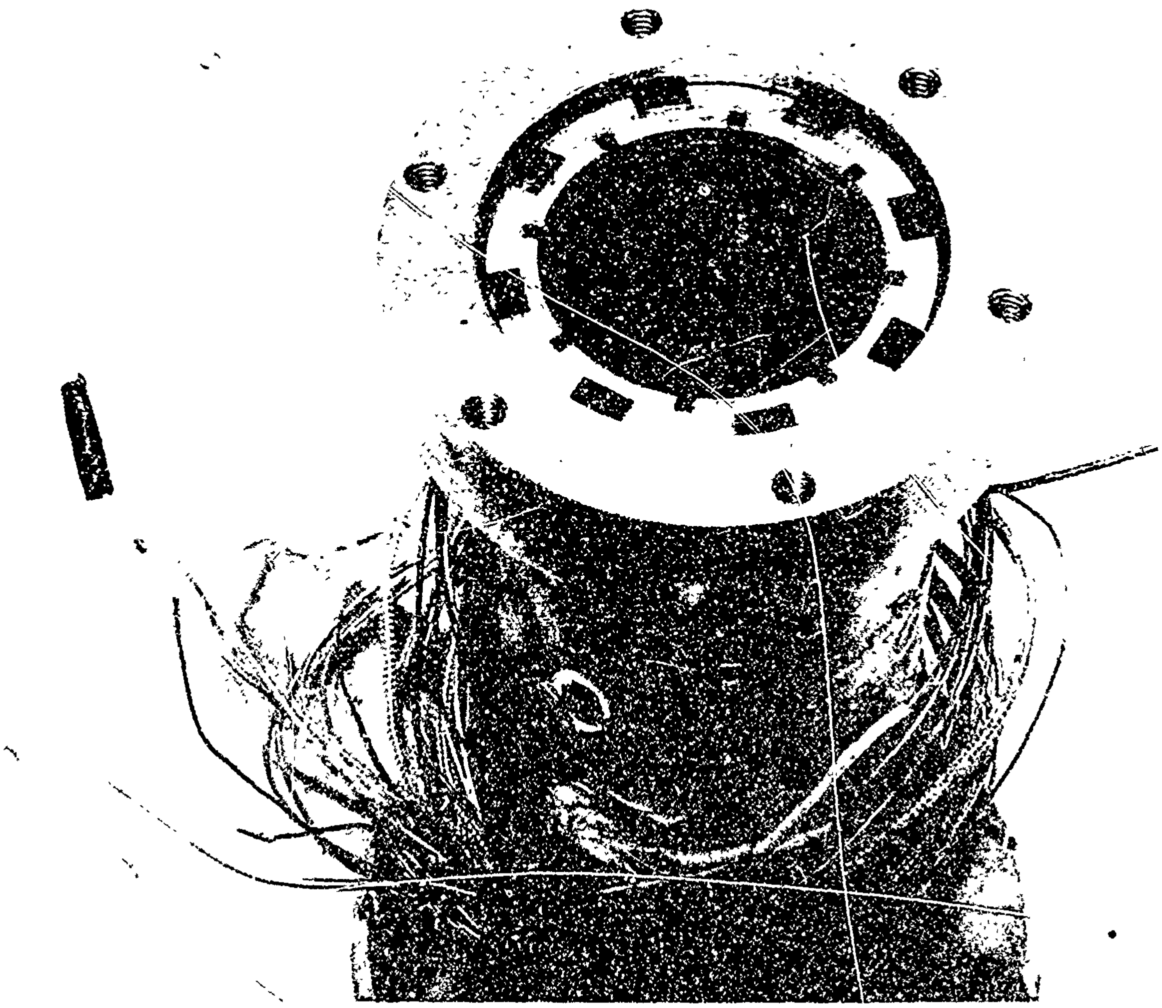

Figure 46. Close-up view of back (turbine) journal bearing.

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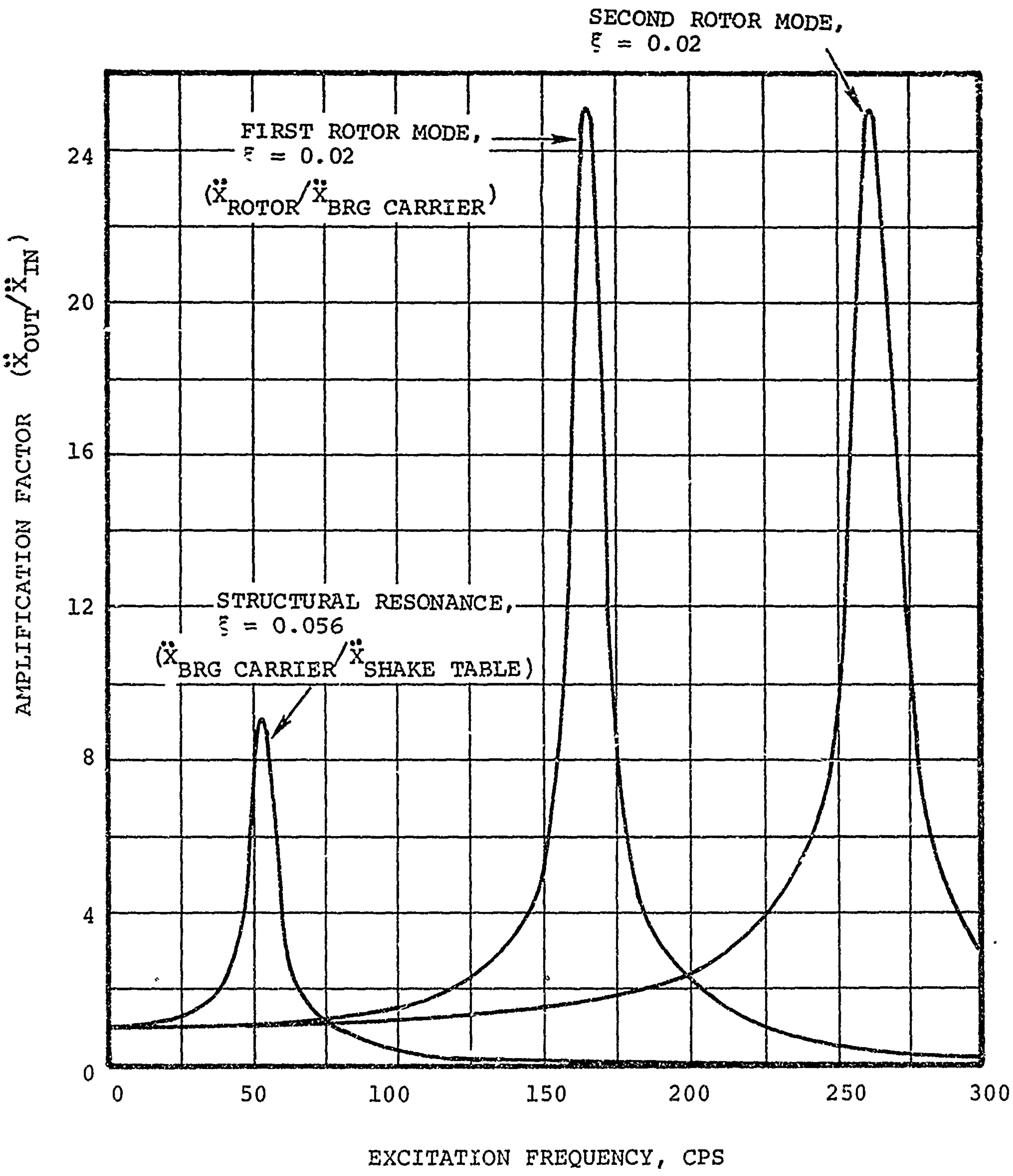

Figure 47. Single degree of freedom amplification factors.

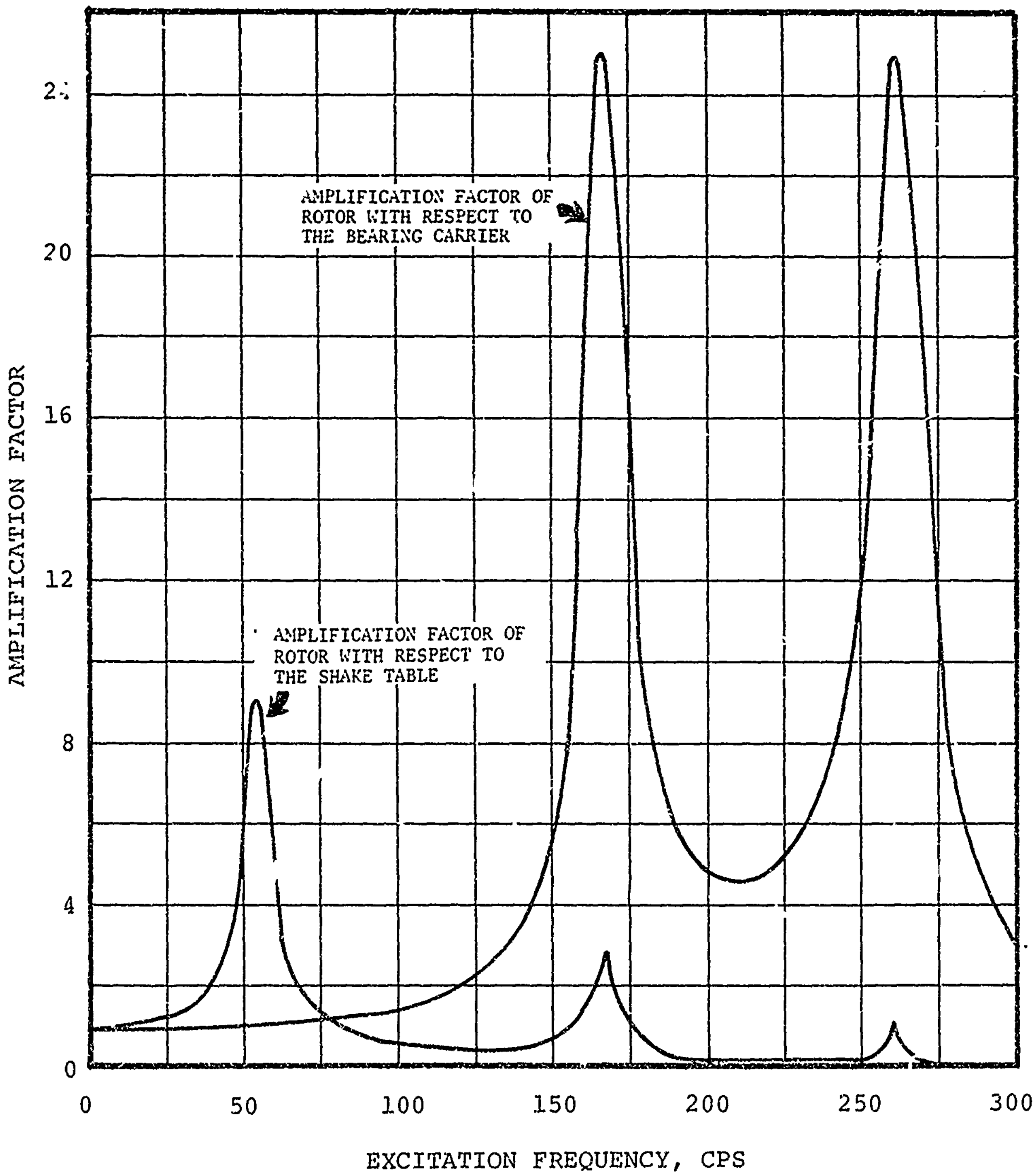

Figure 48. Estimated system amplification factors.

Structural frequencies in the engine, which can multiply the g-loading to the bearings occur on many engine structures and can become a development problem independent of the rotor/bearing natural frequencies. The original JFS100-13A utilizes a relatively thick-walled plenum made from a magnesium casting. The plenum used on the foil bearing demonstrator was made from Type 321 stainless steel, and had a much thinner wall than its cast production counterpart. The thinner wall consequently would exhibit sharper natural frequencies, with greater deflection and less damping.

It is believed that two phenomena caused the forward journal bearing to seize before either of the other bearings. The first is that the centroid of the rotating group is axially located closer to the forward journal bearing than to the aft journal bearing. Secondly, the vibrational load path on the engine structure starts where the engine plenum is bolted to the vibration fixture. The plenum amplifies the input vibration with minimum displacement of "pivoting" at the installation location, which is just above the turbine wheel and very close to the aft journal bearing.

Thus, the maximum vibrational loading occurred on the front, not on the aft, journal bearing.

e. Foil Bearing Overtemperature Test

The demonstrator, with new bearings installed, was assembled into the air motor configuration used for maneuver-loading and vibrational-loading tests. The thrust bearings were preloaded to 14 pounds, approximately the preload used on the maneuver and vibration tests. The measured breakaway torque was 16 inch-pounds, which indicated a rather high preload, similar to that of the maneuver test. A check was made on the journal bearing alignment by inserting a bushing tool with an outside diameter of 1.507 inches (0.007 inch larger than the engine shaft) through both journal bearings. Since it was necessary to rotate the tool to be able to insert it through the bearings, it was evident that the bearings were "bottomed out" and that the total sway space was 0.007 inch.

The proximity probes used on previous tests were removed from the engine at assembly to preclude the possibility of damage from either overtemperature or shaft rubbing at the time of bearing seizure, since, for this test, the bearings were to be operated at incrementally hotter temperatures until bearing seizure occurred.

Because the thrust bearing temperature could be measured much more accurately than the journal bearing temperature, it was decided to "overtemp" the thrust bearing.

Temperature measurements previously made on the journal bearings were actually a measurement of the bearing carrier, since the bead of the thermocouple was located flush with the inside diameter of the carrier at the center of the bearing, beneath the foil. It was felt that in this location the thermocouple was reading the temperature of cooling air flowing axially through the bearing, thus influenced by the plenum air and not providing as accurate a recording of the foil or coating temperature as could be obtained from the thrust bearing. Further attempts at measuring the foil temperature of the journal bearing were successful but it was felt that the method devised could impair the proper operation of the bearing and more development was required to perfect the method.

The thermocouple bead in the thrust bearing, however, is not necessarily at the hottest operational part of the bearing. It is recognized that bearing surfaces other than that at the thermocouple bead can be hotter, depending on bearing operation. For instance, in the Los Angeles maneuver test the thrust bearing temperature was measured in the same manner as in the overtemp test. With the verification test complete, all cooling (plenum) air was shut down and the thrust bearing operating temperature was recorded at 480°F. The thrust bearing had absorbed a relatively large amount of rotor kinetic energy in a short time (usually less than one second), especially when touchdown occurred and bearing friction was greatly increased. During rolldown, thrust bearing recorded temperature rose from 480°F to 505°F, a temperature considered safe for Teflon coating. Since there were no engine parts hotter than the bearings (the engine was in the air motor configuration), no soakback problems existed and the 505°F was the highest recorded temperature on the bearing.

However, after the bearings cooled, a rollover torque test was attempted on the rotating group, and the torque was found to be much greater than was measured prior to the verification test. Upon disassembly, it was found that parts of the thrust bearing foil coating had adhered to the thrust runner and it was evident that the coating had experienced temperatures beyond 600°F, which is considered the absolute temperature limitation of the foil coating. Hence, for the rolldown torque test, the thermocouple did not register the hottest parts of the bearing.

The small thermocouple, installed in the thrust bearing as shown in Figure 49 (as located in the overtemperature test), indicates temperature by air convection from the central part of the bearing. Temperature influence is obtained from the swirling air that is leaving the bearing and which has been heated from the constant viscous shearing inside the bearing. Due to high air speed, the convection (heat transfer) rate is expected to be high, and the temperatures indicated by the thermocouple should be approximately the same as those in the center of the bearing. During shutdown, however, and near the end of shaft rotation, air speed, and therefore convection, is low and the temperatures at the thermocouple are lower than the maximum temperatures generated in the bearing.

After assembly, the engine was installed in a test cell (see Figure 50) capable of providing temperature-controlled air at flows up to 110 pounds per minute. A 6-channel Sanborn recorder was used to monitor and record the pertinent bearing and engine parameters.

At a reference time "zero", cool, external pressurized air was delivered to the engine plenum, resulting in a breakaway speed of 34,500 rpm. Table IV gives the values of pertinent parameters as a function of the reference time taken from the Sanborn recorder traces.

After the engine and bearings had operated at the breakaway speed for several minutes, the plenum pressure was gradually increased until 50,000 rpm was achieved. At this speed, a hot air line was opened, increasing the temperature of the air to the plenum. The hot air temperature was 600°F and the desired air temperature to the plenum was effected by mixing in an amount of hot air to the plenum feed line, which normally provided cold air (60°F). The temperature regulation was accomplished by means of a remotely controlled solenoid valve.

After the delivery air temperature was increased to 260°F and it was verified that fine control of the air temperature was possible, the plenum pressure was increased until the unit reached the required speed of 70,000 rpm. The unit speed was maintained at 70,000 rpm for the remainder of the test by decreasing the plenum air pressure whenever the temperature was increased. The delivery air temperature was increased slowly until the thrust bearing indicated temperature reached 545°F. (A goal of 550°F was sought but a ±5°F tolerance was as close as could be reasonably controlled.) The contractual 10-minute overtemperature of the bearing at 550°F (actually 545°F) was

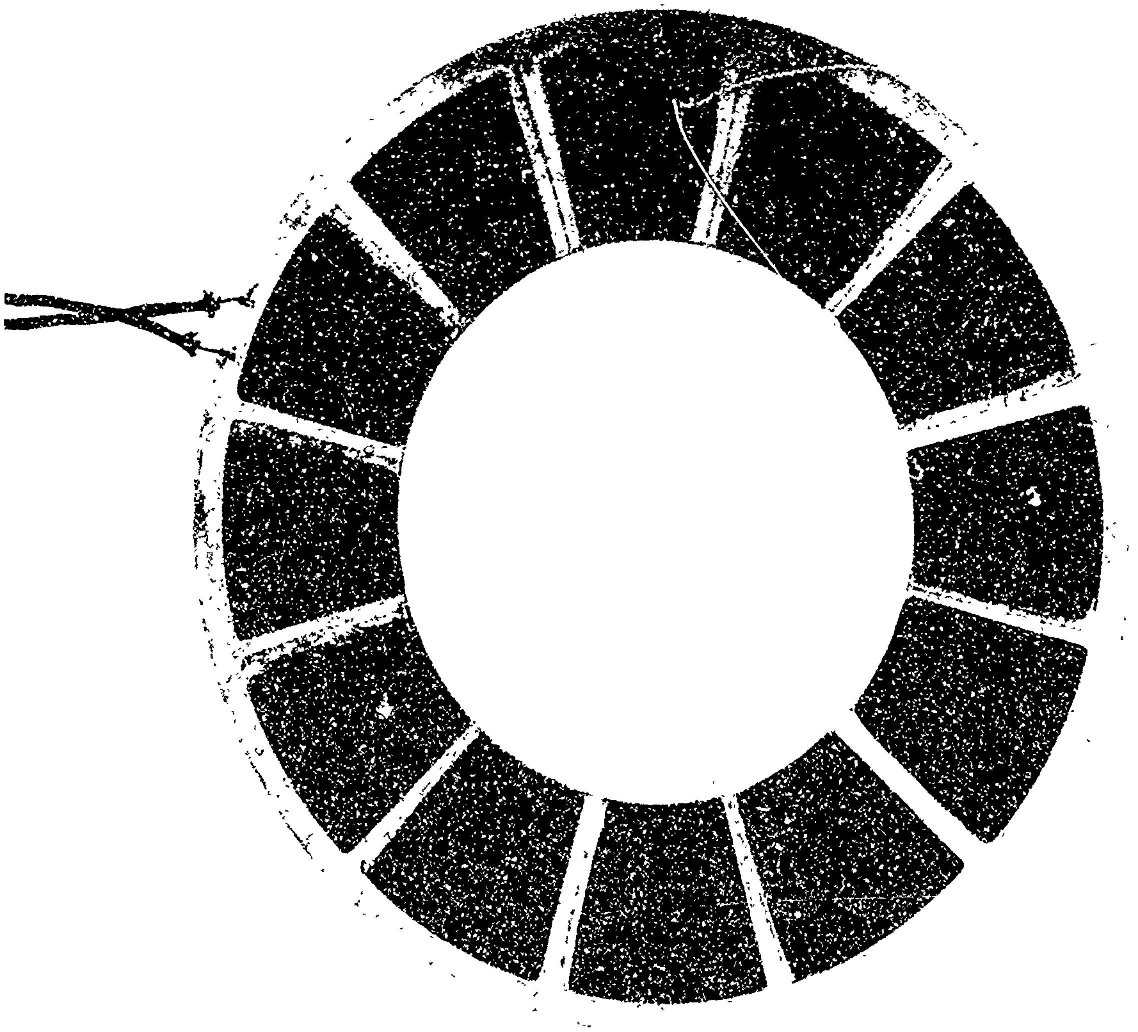

Figure 49. Thrust bearing thermocouple location.

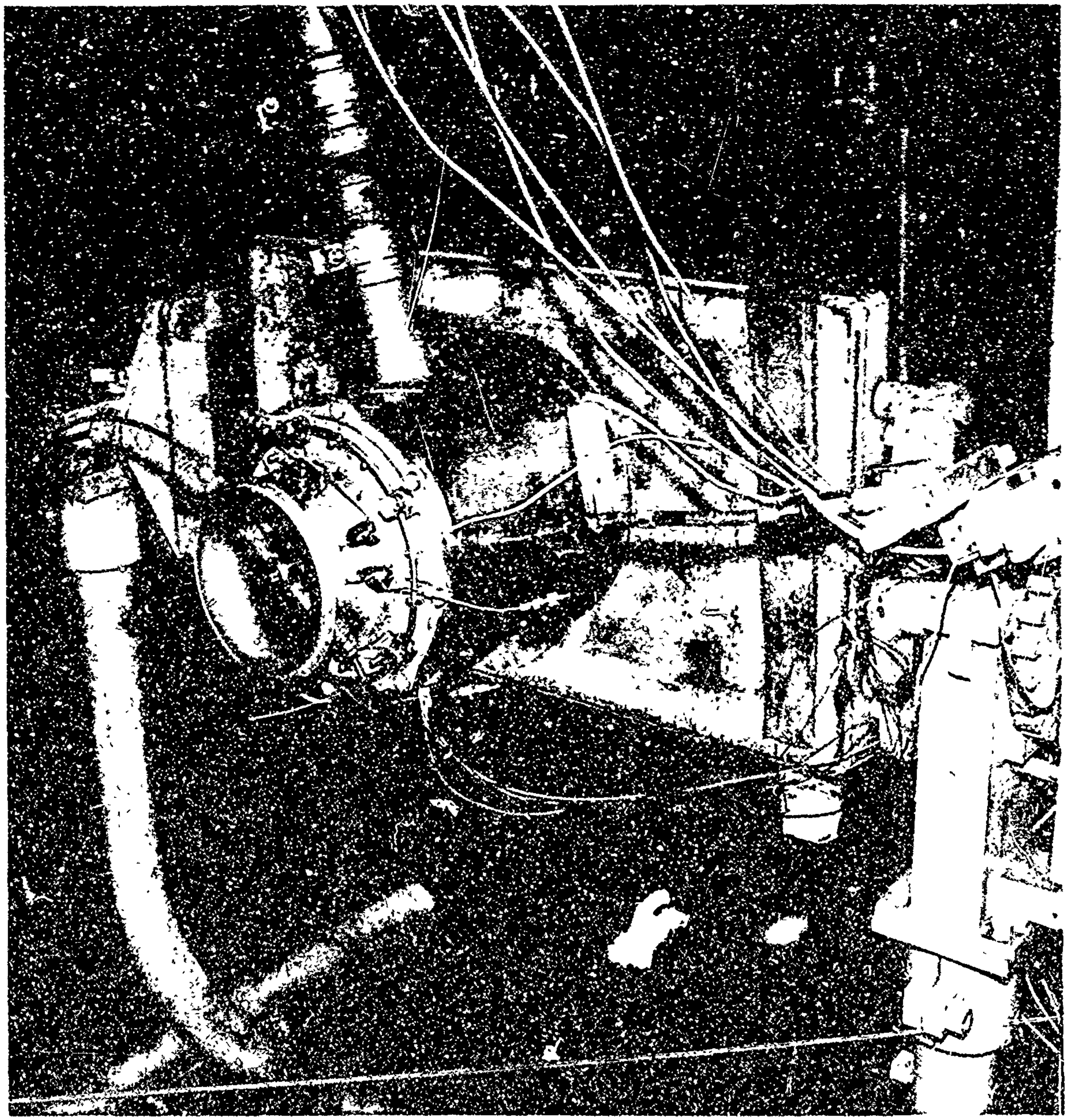

Figure 50. Front view of foil bearing demonstrator installed in the overtemperature test facility.

TABLE IV

DATA OBTAINED FROM
SANBORN RECORDING OF FOIL BEARING
OVERTEMPERATURE TEST

Time (sec)	Engine Speed (rpm)	Plenum Press (psig)	Plenum Temp (°F)	Axial Bearing Temp (°F)
0	0	0	86	90
20	34,500	2.5	88	110
180	34,000	2.5	80	150
260	48,000	5.5	78	160
380	50,000	6.4	70	170
500	52,000	6.4	110	210
620	54,000	6.4	150	260
740	56,500	6.4	210	340
860	62,000	8.2	210	380
980	65,000	9.2	215	405
1060	69,500	11.4	218	415 *
1180	70,000	11.5	220	430
1300	70,000	11.4	226	435
1420	70,000	10.7	250	460
1540	70,000	10.4	265	480
1660	70,000	10.1	280	505
1780	70,000	10.0	280	520
1900	70,000	10.4	296	530
2020	70,000	10.1	315	540
2140	70,000	10.1	320	540
2236	70,000	9.9	340	545 **
2300	70,000	9.9	340	545
2540	70,000	9.9	345	550
2660	70,000	9.9	340	550
2700	70,000	9.9	340	555
2740	70,000	9.9	340	560
2780	70,000	10.0	340	560
2800	70,000	10.0	340	565
2810	70,000	10.0	335	560
2820	69,500	10.0	330	560
2830	69,500	10.0	325	560 ***
2836	69,500	10.0	322	560

*Considered full speed
**Start overtemperature test
***Occurrence of bearing seizure

initiated at the reference time of 2236 seconds. (The unit had been operated for 2236 seconds prior to the initiation of the 550°F "overtemperature" test.)

As can be seen from Table IV, a plenum temperature of 340°F and a plenum pressure of 10 psig were required to maintain unit speed at 70,000 rpm and thrust bearing temperature at 550°F. During approximately the last half minute of operation, the bearing heat generation increased, which increased the temperature of the bearing. In an effort to reduce the temperature to 550°F, the delivery air temperature was reduced. This continual reduction resulted in a slight drop in unit speed and maintained the bearing temperature at 560°F. Other than the increase in bearing temperature, there was no warning prior to bearing seizure. The bearings operated in the 550°F high temperature environment for 600 seconds. Figure 51 shows the Sandborn trace at the end of bearing seizure.

After the test, the engine was disassembled and the bearings inspected. Both sides of the thrust bearing were discolored and the foil coating was dissipated, indicating overtemperature conditions. As in other bearing seizures, the shaft was welded to the foils, with damage sustained by both. No other engine components experienced rubs or damage and all parts except the foils and the spacer shaft were reusable without rework.

Figure 52 shows the forward thrust foil bearing after disassembly.

The mode of failure in the overtemperature condition began with the Teflon coating, which decomposes after reaching temperatures of 600°F. The gas given off by the Teflon will cause no harm and will be swept away from the bearing in the cooling airstream. There are cooler areas in the bearing, however, that will emit relatively large quantities of a decomposed Teflon powder, which acts as a bearing contaminant, like dirt and dust. This buildup of Teflon contaminant eventually disrupts the air film enough that the bearing cannot support the required load and it "touches" the shaft, causing simultaneous increase in temperature and bearing seizure.

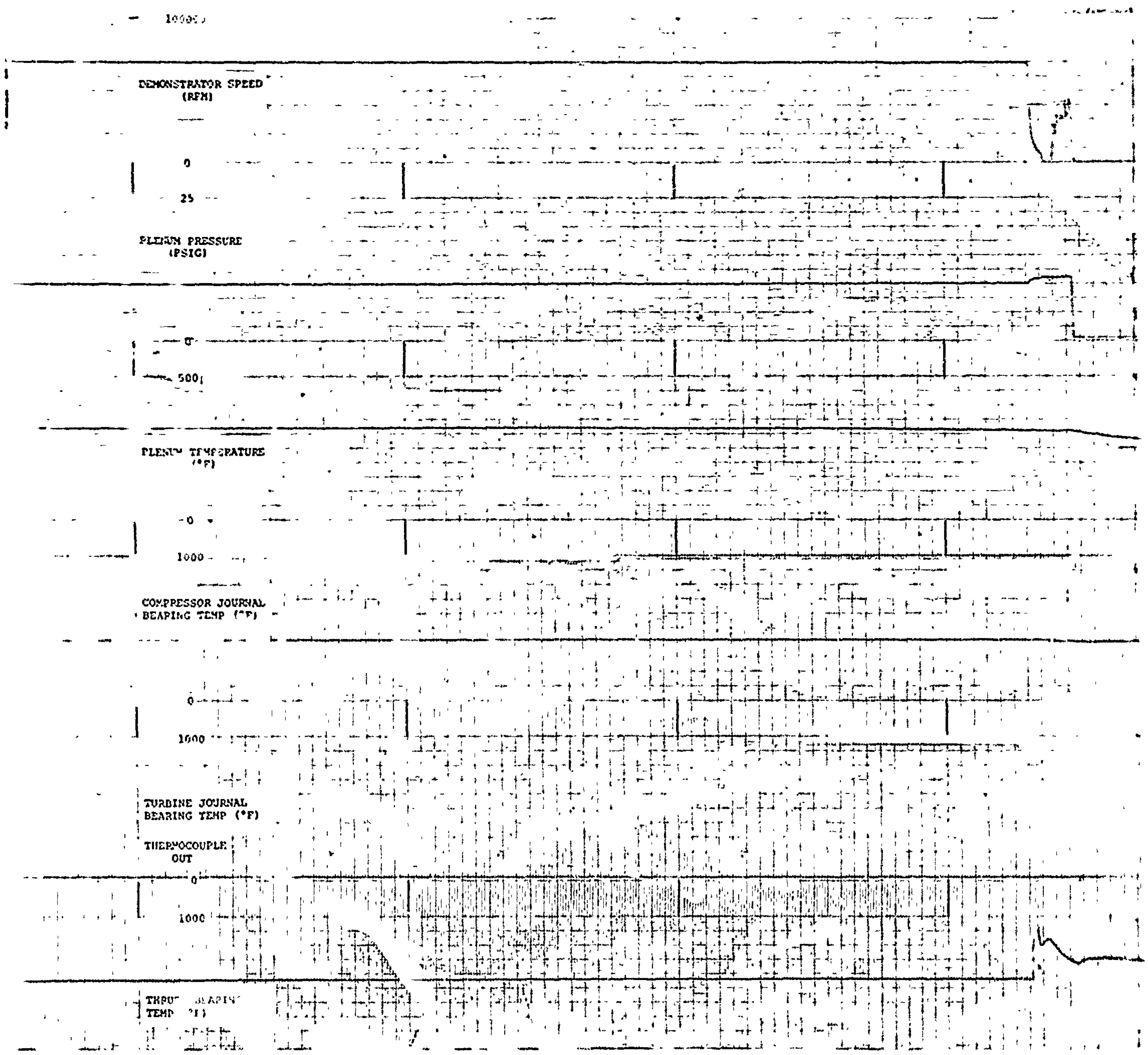

Figure 51. Sandborn trace showing final seconds of overtemperature test.

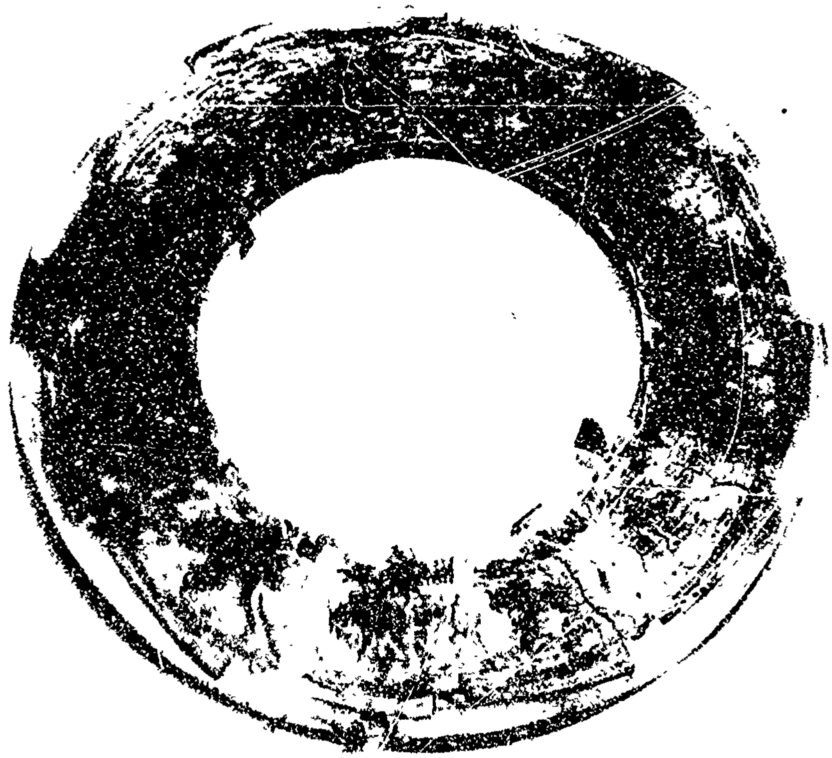

Figure 52. View of forward thrust foil bearing after disassembly.

The need for coatings that are tolerant to temperatures greater than those in this test is evident. Research in the development of pivoted-pad bearings has shown the possibility of acquiring coatings that will withstand up to 1200°F. The two most promising materials appeared to be chrome-oxide for operation up to 800°F and nickel-chrome/chrome oxide for operation up to 1200°F. Although not exhibiting the desirable embedment characteristic sought in the foil bearing development, these candidate materials are being considered for higher temperature operation capability.

SECTION III

CONCLUSIONS AND RECOMMENDATIONS

From the results of the program, it is concluded that the gas-lubricated foil bearing is a potential candidate for incorporation in advanced gas turbine engines. During the program, no problems were encountered in operating the demonstrator at the full speed of 72,000 rpm and the engine was impervious to the dust and water injected into the inlet.

Although no problem was encountered in operating the demonstrator engine at full speed, the maneuver test results indicated that an increase in foil bearing load capacity is desirable. While larger bearings and longer journal bearing spans could be utilized to accomodate greater yaw maneuver rates, the journal bearing diameters are often limited by flowpath restrictions and excessive surface speeds. To achieve yaw maneuver rates specified in MIL-E-5007A (3.5 radians per second), the load capacity of the foil gas-lubricated bearings must be increased.

Results of the vibration test reflect largely upon the engine frame rather than the bearings. Aircraft installations require that the engine frame prevent multiplication of vibrational amplitudes from the source to the bearing carrier by provisions for adequate damping. Hence, results of the vibration test do not effectively evaluate the bearings, since many simultaneous conditions contribute to the vibrational characteristics of the engine.

The overtemperature test reveals that an ultimate temperature of 550°F for the bearings is adequate for low-pressure-ratio engines and other machines that utilize cooler air. The present foil coating prevents higher temperature operation; hence the need for higher temperature coatings is evident.

In summary, the program has shown that gas-lubricated foil bearings have encouraging potential and offer attractive advantages for application in advanced gas turbine engines.

There are two areas where further development is required, namely:

- The load capacity of the bearings must be increased to support loads induced by MIL-E-5007C maneuver specifications.

- A higher temperature coating must be developed for bearing application in larger, higher pressure-ratio engines.

It is recommended that the foil bearing demonstrator be mounted on the wing of an aircraft and subjected to a wide variety of maneuvers, altitude operation, and vibrational environment that would be representative of an actual aircraft-mounted engine. Observing the bearings under these conditions will further justify their general feasibility for small gas turbine application.