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Zolton N. Nemeth

Lewis Research Center
Cleveland, Ohio

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EXPERIMENTAL EVALUATION OF FOIL-SUPPORTED RESILIENT-PAD GAS-LUBRICATED THRUST BEARING

by Zolton N. Nemeth

Lewis Research Center

SUMMARY

A previously analyzed new type of resilient-pad gas thrust bearing was tested experimentally to determine the feasibility of the design. The bearing consists of carbon graphite pads mounted asymmetrically on resilient foil beams. Two bearing configurations were tested. One configuration consisted of sector shaped pads mounted on sector shaped foils. The second configuration consisted of square pads mounted on rectangular foils. Each thrust bearing had an 8.9-centimeter (3.5-in.) outside diameter, a 5.1-centimeter (2.0-in.) inside diameter, and six pads mounted on 0.25-millimeter- (0.010-in.-) thick foils.

The bearings were run in air at thrust loads from 27 to 80 newtons (6 to 8 lb) (load pressure, 12.5 to 37.6 kN/m^2 (1.82 to 5.45 lb/in^2)) and at speeds to 9000 rpm.

Both configurations of the thrust bearings performed in a satisfactory manner over most of the load and all of the speed ranges tested. At the higher speeds tested, the sector pad bearing exhibited surface rubbing at low loads; increasing the load stopped the rubbing. No rubbing occurred under any test conditions for the square pad bearing. Bearing hydrodynamic torque was lowest for the sector shaped pad bearing.

INTRODUCTION

Mechanical systems are now being operated at increasingly severe conditions. This is especially true in gas turbine engines where higher temperatures are necessary. Even under normal conditions for automobile internal combustion engines the working life of both the mechanical system and the lubricant is being extended (ref. 1). Synthetic lubricants offer some help in that they can operate at higher temperatures than mineral oils. Future automotive gas turbine engines will operate above the lubricating ability and fire resistance of oils, and gas bearings seem to be the only alternative to oil lubricated bearings for this type of application. Until now oil lubricated bearings have been used in gas turbines because the thrust load is too high from aerodynamic thrust unbalance for gas bearings. There are other problems that gas bearings have to cope with - for example, dirt and thermal growth and distortion.

Dirt is a problem in the oil-lubricated film bearings for the passenger car engine even though these bearings are designed to tolerate a certain amount of dirt. Since high strength requirements of the bearing limit the thickness of the soft overlay to about 0.025 millimeter (0.001 in.), it is difficult for the bearing to absorb particles of greater size than this. If anything, modern oil-film bearings are more sensitive to dirt (ref. 2).

The rigid geometry gas bearings, such as the tilting pad bearing and the single disk spiral groove or Rayleigh step thrust bearing, are made of hard materials because of the precision required. Also, since the bearings are made of hard materials, they have poor embedability. The bearings tolerate dirt in the gas stream to a small degree because they can move slightly. They are, however, expensive to manufacture and assemble because of the multiplicity of the precision parts required.

Gas foil bearings seem more suitable to gas turbine application than the rigid geometry gas bearings because of their greater tolerance to dirt and thermal growth and distortion. The foil bearings are less expensive to manufacture because they are simpler and require a less severe manufacturing tolerance. Both the tensioned (refs. 3 and 4) and overlapping cantilevered (ref. 5) type bearings have been more successful as journal bearings than as thrust bearings. The foil principle is difficult to apply to the thrust bearing. In the foil journal bearing the film wedge forms naturally between the foil and the shaft. In the thrust bearing, however, the film wedge has to be formed by special foil support schemes which are usually inadequate in maintaining an undistorted film wedge. A bearing with rigid-geometry load-carrying surfaces (such as pads) mounted on a resilient support overcomes this difficulty.

In one investigation the rigid pads were mounted on a resilient layer of elastomer (ref. 6). This bearing performed well, but at high speed operation the friction heating in the air film caused the elastomer to increase in thickness. The bearing tended to loose its clearance. A heat exchanger had to be built in the bearing between the pad and the rubber mounting material.

A new type of resilient-pad gas thrust bearing was described and analyzed in reference 7. The bearing has all-metallic components and therefore can operate satisfactorily in a hot environment. The rigid flat pads are each mounted on a flexible metallic beam. The one-dimensional design analysis reported in reference 7 was straightforward and showed that the performance of the resilient-pad bearing was equal to that of a pivoted-pad thrust bearing. The present study is a continuation of this earlier investigation. The object of this investigation was to experimentally evaluate the new type of resilient-pad gas thrust bearing with metallic resilient-pad support and to compare two different configurations of pad geometry (but approximately equal areas) and resilient metallic support. Tests were performed with a 8.9-centimeter (3.5-in.) outside diameter bearing having six pads. Speed and thrust loads were varied to determine the feasibility of this design concept.

The resilient-pad gas test thrust bearing was evaluated in an air-bearing-supported and turbine-driven bearing test rig. The test conditions were rotor speeds to 9000 rpm, thrust loads to 80 newtons (18 lb) (or a load pressure of 37.6 kilonewtons per square meter (5.45 lb/in^2)), and air at ambient room temperature as the lubricant. The results were evaluated from the friction torque values produced in the test thrust bearing.

APPARATUS

Test Thrust Bearing

The resilient-pad gas thrust bearing consists of six pads mounted on a flexible metal foil which is in turn mounted on a supporting structure. The supporting structure has the same number of foil support areas as there are pads in the bearing. Each pad is located asymmetrically between foil supports.

Two configurations of the resilient-pad gas thrust bearing were tested. The first configuration is shown in figure 1. It resembles a conventional fluid thrust bearing since it is circular and has sector shaped pads. The foil support structure consists of metal radial spokes. Each radial spoke has a circular cross section at the foil plate contact.

The second configuration is shown in figure 2. The pads are square and are mounted on rectangular shaped foils. The foil support structure is triangular. In this configuration, the values of a and b, the lengths from the edge of the pad to the supports, are constant with radius.

The bearing material was chosen for operation in air at room temperature. The runner was made of hardened M-50 tool steel. The pads were made of medium hard carbon graphite (20 parts amorphous carbon and 80 parts graphite) pitch bonded (ref. 8). The foil was molybdenum metal. Epoxy cement was used to attach the carbon graphite parts to the foil. The bearing surfaces were lapped flat to within 2.5 micrometers (0.0001 in.) with the weight of the bearing assembly applied. The pads were held in position by this small initial thrust load of the bearing assembly.

Bearing Design

The design of the resilient-pad thrust bearing was reported in reference 7. A practical bearing configuration was suggested which has six rigid sector shaped pads (fig. 1) mounted on an annular flexible metal disk. The metal disk is supported on radial spokes to provide an asymmetric beam section for each pad. The desired pad angle inclination is thus obtained without using a pivot ball and socket.

The sector shaped pad on an annular disk results in a familiar circular type of thrust bearing. However, the sector shaped pad and the annular disk diverge from the assumptions of the analysis and results in a weak support of the pads toward the outside diameter of the bearing. This bearing configuration was the first one tested in the program.

A more desirable bearing configuration results when the pad is square and is mounted on a rectangular foil beam (fig. 2). The metal disk has a hexagonal shape. The stiffness of the rectangular beam for each elemental width is constant. This bearing configuration was the second to be tested.

The design procedure for the resilient-pad gas thrust bearing is given in reference 7, and the bearings tested in this program were made to the following dimensions:

Radius to outside pad, cm (in.)	4.45 (1.75)
Radius to inside of pad, cm (in.)	2.54 (1.00)
Length of pad in direction of motion (at mean radius), cm (in.)	1.91 (0.75)
Mean radius of pad, cm (in.)	3.49 (1.375)
Foil thickness, mm (in.)	0.254 (0.010)
Beam material	molybdenum
Leading edge foil length/trailing edge foil length, b/a	1.7

Test Apparatus

The test apparatus used in this investigation is a modified version of the apparatus used previously for the herringbone-grooved gas journal bearing investigation. The test apparatus before modification is fully described in reference 9. It consisted of a vertically oriented steel rotor mounted in two smooth bronze bushings. The rotor was 3.8 centimeters (1.5 in.) in diameter and 31.1 centimeters (12.25 in.) long. The bronze bushings were 3.8 centimeters (1.5 in.) long. An externally pressurized thrust bearing supported the rotor axially at its lower end.

The rotor was driven by an impulse turbine which consisted of six buckets cut into the upper end of the rotor and a nozzle ring surrounding the rotor.

Modification was necessary to accept a test thrust bearing, to add the capability of measuring the thrust bearing friction torque, and to provide thrust loading of the test thrust bearing.

The test apparatus in its present form is shown in figure 3. The herringbone grooved gas bearings were replaced by externally pressured gas journal bearings and a smooth surface shaft (fig. 4). At the comparatively low speeds used in this program, the self-acting herringbone grooved bearings had insufficient load capacity to prevent surface contact. The test thrust bearing was located between the two support journal bearings. An externally pressurized gas bearing load support with a self-alining gimbal and a bearing friction force transducer is used to hold the test thrust bearing stator. The thrust bearing runner was designed to clamp on the shaft. This was accomplished by a collect arrangement built into the runner. A pneumatic cylinder loaded the test thrust bearing in an upward direction through the externally pressurized rotor support thrust bearing. The support bearing was mounted on a spherical ball and socket pivot for alinement with the end of the rotor. The externally pressurized bushings were made of bronze and were a light push fit in the housing.

The lubricant for the test bearing was ambient air. The temperature of the room was kept at a constant 297 K (75° F).

Instrumentation

Orthogonally positioned capacitance probes were used to observe the motion of the shaft outboard of each journal bearing. An X-Y curve tracing cathode ray oscilloscope was used to display the signals generated by the capacitance probes. Thrust bearing friction was measured by an unbonded strain gage force transducer which was connected to the outer diameter of the bearing housing. Since the test thrust bearing and its housing floated freely on pressurized air, the test bearing friction force could be accurately measured provided the attached lines were arranged so they did not affect the torque measurement. Rotor speed was measured by a magnetic pickup facing the turbine buckets in the shaft and indicated on an electronic counter. Shaft speed was electronically controlled to an accuracy of ±1 percent. The bearing friction force and shaft speed were recorded on a two-pen strip chart recorder

PROCEDURE

Static load deflection test was performed on one bearing - configuration I. This bearing configuration has sector shaped pads mounted on sector shaped foils which are supported on radial spokes. Equal point loading was applied to each of the six pads at the center of the pressure location on the pads. The center of pressure chosen was 0.67. This is a point on the mid-radius of the pad whose distance from the leading edge of the pad to the length of the pad in the direction of motion equals 0.67.

To start a run, the test thrust bearing stator was lifted off the bearing runner by hand and held until the speed of the rotor reached approximately 2500 rpm. At this speed the bearing stator was let down onto the runner and the bearing was running self-acting on a gas film. The speed was rapidly increased to 5000 rpm and the load applied to a nominal value of 27 newtons (6 lb). Data were obtained at various loads and speeds thereafter. Ten minutes of running was accumulated at each condition to reach equilibrium conditions before data were taken. Bearing friction and speed were recorded continuously on a strip chart recorder.

Before and after each day's run the static friction was obtained with tare load of the stator assembly. The shaft was rotated slowly by hand for this measurement.

RESULTS AND DISCUSSION

The experimental results obtained are shown in tables I and II and in figures 5 to 9. Two bearing configurations were investigated. Both bearings had six carbon graphite pads mounted on a 0.25 millimeter (0.010 in.) thick molybdenum foil. One bearing configuration consisted of sector shaped pads mounted on sector shaped foils. The foil supports were radial spokes. The second bearing configuration consisted of square shaped pads mounted on rectangular foils. The foil supports were triangular. The bearings were operated in air. The bearings were run at thrust loads from 27 to 80 newtons (6 to 18 lb) (load pressure, 12.5 to 37.6 kN/m^2 (1.82 to 5.45 lb/in^2)) and speeds to 9000 rpm.

Sector Pad Thrust Bearing

The results of the load deflection experiment are shown in figure 5. Under a point load of 20 newtons (4.5 lb) at the pad center of pressure the various surface locations deflect as shown in the figure. The pad surface deflects downward (away from the load) in the radial direction toward the outside radius and in the circumferential direction toward the leading edge of the pad. There is then an undesirable amount of deflection radially outward for the pad giving a weak support in this direction.

Bearing torque is plotted against rotor speed in figure 6 for four thrust loads. Torque increases with an increase in speed and is greater for each higher load at the low speed end of the runs up to 6500 rpm. Bearing torque is very low (below 0.0065 N-m (0.058 in-lb)) in this speed range indicating that no physical contact is made between the stationary bearing pads and the rotating runner and that the bearing is operating in the hydrodynamic mode.

As the speed is increased beyond 7000 rpm a point is reached where bearing torque begins to increase more rapidly as can be seen in figure 6. This point occurs at higher speeds for increased thrust loads.

At 8000 rpm the bearing ran satisfactorily at the higher loads but rubbed at the low load of 27 newtons (6 lb). At this 27-newton (6-lb) load the instantaneous rise in bearing torque was to about 0.24 newton-meters (2.14 in-lb). Again at 9000 rpm the bearing ran hydrodynamically at loads greater than 44 newtons (10 lb). At 44 newtons (10 lb) the bearing rubbed and the bearing friction force jumped to about 0.31 newton-meter (2.74 in-lb). Increasing the thrust load produced a lower bearing torque at 9000 rpm. The bearing torque was 0.0100 newton-meter (0.089 in-lb) at a thrust load of 67 newtons (15 lb) and only 0.0076 newton-meter (0.067 in-lb) at a thrust load of 80 newtons (18 lb).

It is evident that larger thrust loads are required in order to obtain higher speeds without a rub between the bearing surfaces. Some difficulty was experienced early in the experimental program when this was not appreciated. The bearing was operated with light load initially at various speeds, and this mode of operation gave erratic torque results.

The pad, the foil support, and the gas film might have resonances at some speed within the speed range investigated. Resonances of the pad may occur in the radial, pitch, or roll directions. If the amplitude of motion is large enough, the pad could strike the runner and rub it.

Also, it is probable that at light load and high speed the pressure distribution in the film is such that it causes the pad to tilt so that one edge rubs. Pad tilt can occur in the roll direction because the foil stiffness is especially soft in that direction.

The bearing surfaces for the sector pad thrust bearing after test are shown in figure 7. Some rubbing and carbon graphite wear particles are seen on both the stator and runner surfaces but these are minimal and are to be expected with operation of the bearing until rubbing occurs. The runner surface is clean for the most part and the light source can be seen reflected from the surface. The wear particles appear in six equally spaced areas around the rotor. This is from the six pads coming to rest on the runner at zero speed after testing. Most wear occurred at the outer diameter of the bearing pads with a lesser amount at the inside diameter of the pads. The wear particles are thrown outward and eventually build up at the outer diameter. Contact of the pads with the rotor occurs at the outer diameter if the bearing surfaces are not completely parallel, as could be the case during an instability or resonance of the thrust bearing.

Square Pad Thrust Bearing

Bearing torque is plotted against rotor speed in figure 8 for the square pad thrust bearing for four thrust loads. In general, torque increases with increases in speed. An exception occurred for the highest load of 80 newtons (18 lb) at low speed. However, with a rerun of the 5000 rpm data the value of torque at 80 newtons (18 lb) dropped from 0.0198 newton-meter (0.175 in-lb) to 0.0128 newton-meter (0.113 in-lb) (see table I).

Bearing torque also increases with an increase in load. The rate of increase in bearing torque is low to a speed of about 7500 rpm; above this speed the torque increases more rapidly with speed. The bearing did not experience a rub over the load and speed range investigated. However, for a load of 27 newtons (6 lb) and a speed of 9000 rpm, the torque was higher than for greater thrust loads, indicating that the bearing may have rubbed if a further increase in speed had been attempted. It is evident that for the square resilient pad bearing to run successfully requires increased loading as speed is increased.

The bearing surfaces for the square pad thrust bearing after test are shown in figure 9. Light rubbing occurred between the mating surfaces. The photo light source can be seen reflected from the runner surface. The wear particles were deposited in six equally spaced areas on the runner surface by the pads at zero speed after testing. Most wear occurred at the outer diameter of the bearing pads. There was practically no wear at the inside diameter of the pads.

Bearing Static Torque

Before and after each run the static bearing torque was obtained by rotating the shaft slowly by hand (table II). The bearing load was the weight of the stator assembly, 12.5 newtons (2.8 lb). Also included in the table is the maximum bearing torque during coast down and rubbing of the surfaces. The static friction was lowest when the surfaces were clean at the beginning of the tests. After one coast down the static torque was at least double or more. By wiping the runner with a mineral solvent moistened swab the surface could be cleaned of the carbon graphite and restored to the original surface appearance. The static torque also dropped. A better bearing material or surface coating for low starting torques would be one that did not increase the static torque with running time.

The designs of the bearings tested were based on the very approximate one-dimensional analysis and thus compromised the designs. To achieve higher speeds than reached in this effort requires that the analysis be of an actual bearing configuration and include the dynamic response of the pad-support system.

SUMMARY OF RESULTS

Two configurations of the resilient pad gas lubricated thrust bearing were run in air at thrust loads from 27 to 80 newtons (6 to 18 lb) at speeds to 9000 rpm to ascertain the feasibility of this type of bearing experimentally. One configuration consisted of sector shaped pads mounted on sector shaped foils. The second configuration consisted of square pads mounted on rectangular foils. The pads were mounted asymmetrically between the foil supports with foil free length ratio of 1.7. The pad material was carbon graphite and the bearing runner was hardened tool steel. The following results were obtained:

1. Both configurations of the resilient pad gas lubricated thrust bearing performed in a satisfactory manner over most of the load and all of the speed ranges tested.

2. The square pad bearing operated to the maximum speed without surface rubbing at all loads tested.

3. Sufficient thrust load had to be applied to the sector shaped pad bearing to reach higher speeds without surface rubbing.

4. The hydrodynamic bearing torque for the square pad bearing with triangular foil supports was approximately two to three times that for the sector shaped pads with radial spoke foil support.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, June 1, 1977,
505-03.

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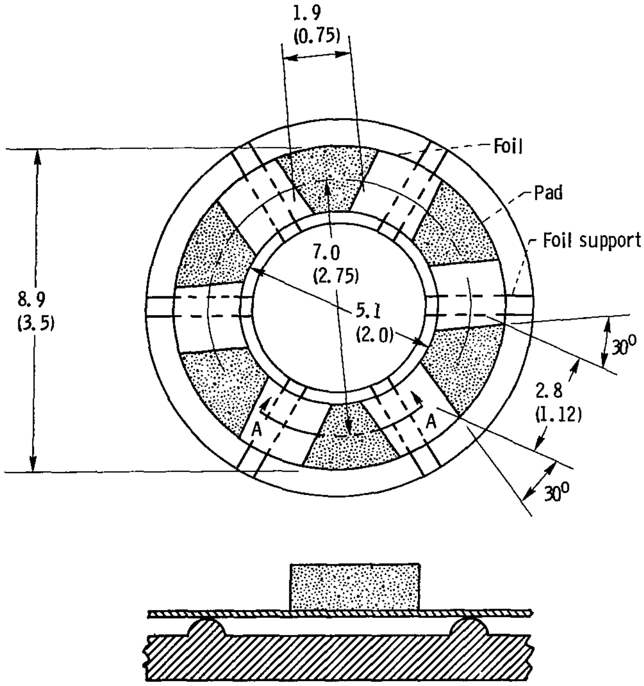
TABLE I. - SQUARE PAD BEARING TORQUE AT 5000 RPM

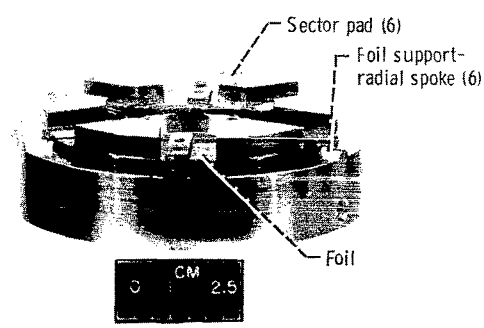
Pad load pressure		Thrust load		Bearing torque			
				Initial		Repeat	
kN/m^2	psi	N	lb	N-m	in-lb	N-m	in-lb
12.5	1.82	27	6	0.0068	0.060	0.0093	0.082
20.9	3.03	44	10	.0085	.015	.0106	.094
31.3	4.55	67	15	.0128	.113	.0117	.104
37.6	5.45	80	18	.0198	.175	.0128	.113

TABLE II. - BEARING TORQUE AT SURFACE CONTACT

Bearing	Run	Bearing torque					
		Static (before run)		At coast down (maximum)		Static (after run)	
		N-m	in-lb	N-m	in-lb	N-m	in-lb
Sector pad (radial spoke foil support)	1	0.07	0.6	0.12	1.1	0.26	2.3
	2	.41	3.6	.39	3.4	.32	2.8
	3	.26	2.3	.40	3.5	.43	3.8
Square pad (triangular foil support)	1	0.09	0.8	0.18	1.6	0.24	2.1
	2	.25	2.2	.20	1.8	.25	2.2

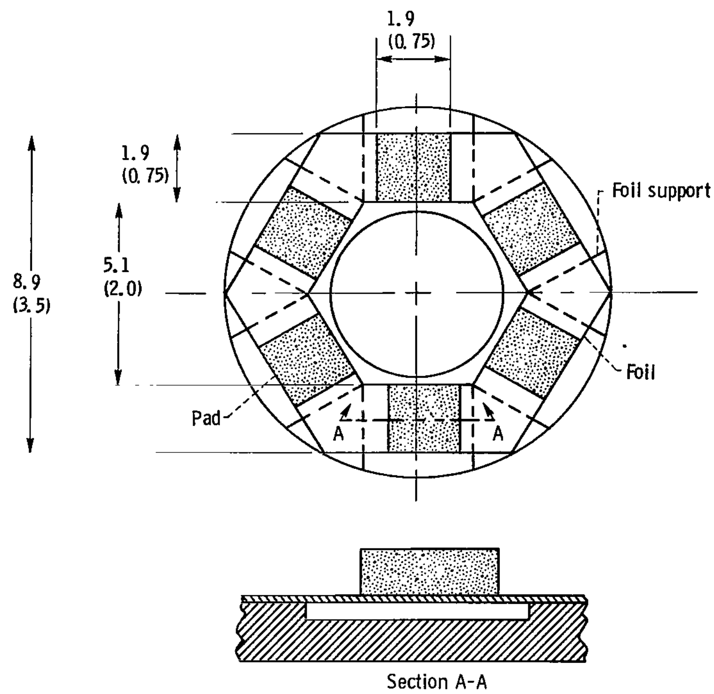


(a) Schematic.

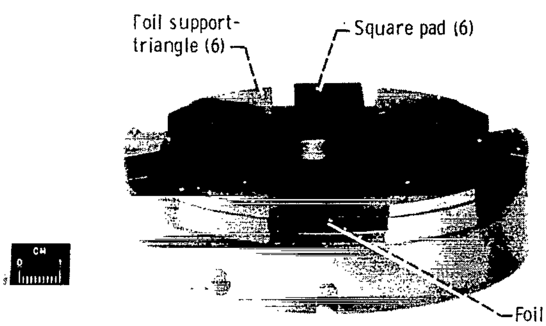


(b) Stator assembly.

Figure 1. - Resilient-pad gas bearing - configuration I.
All dimensions in centimeters (in.).

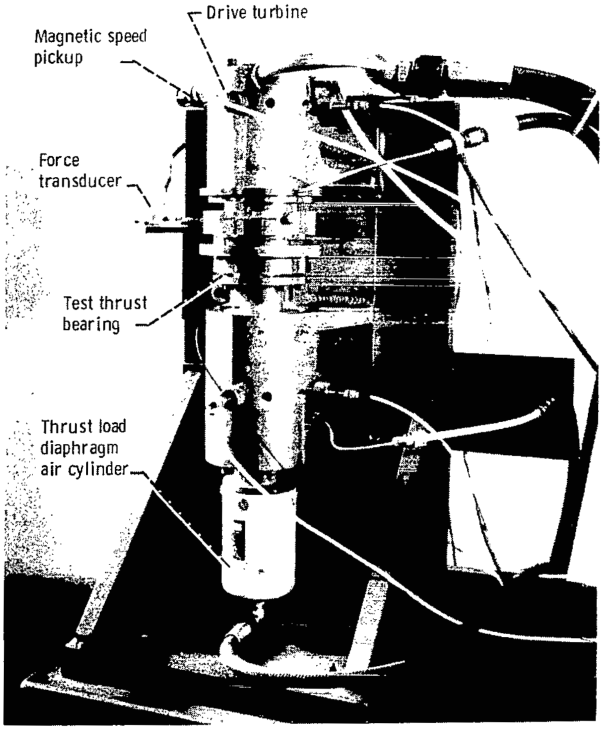


(a) Schematic.

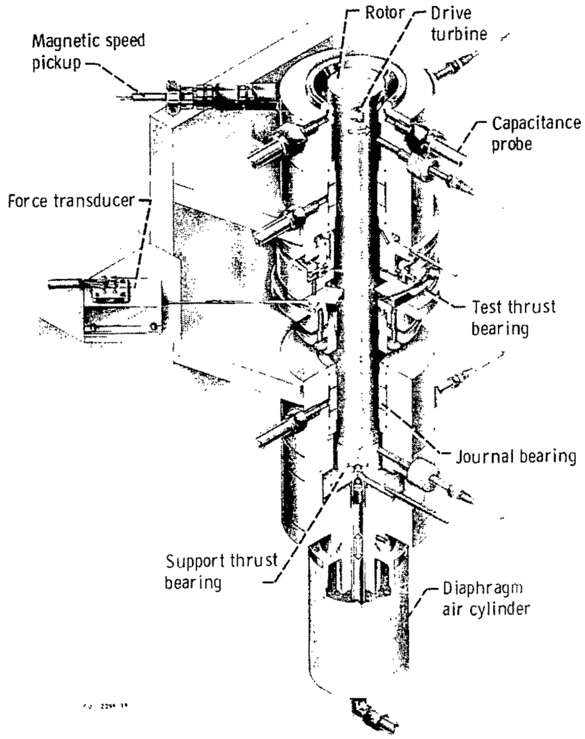


(b) Stator assembly.

Figure 2. - Resilient pad gas bearing - configuration II. All dimensions in centimeters (in.).



(a) Photograph of test apparatus.



(b) Schematic drawing of test apparatus.

Figure 3. - Gas thrust bearing test apparatus.

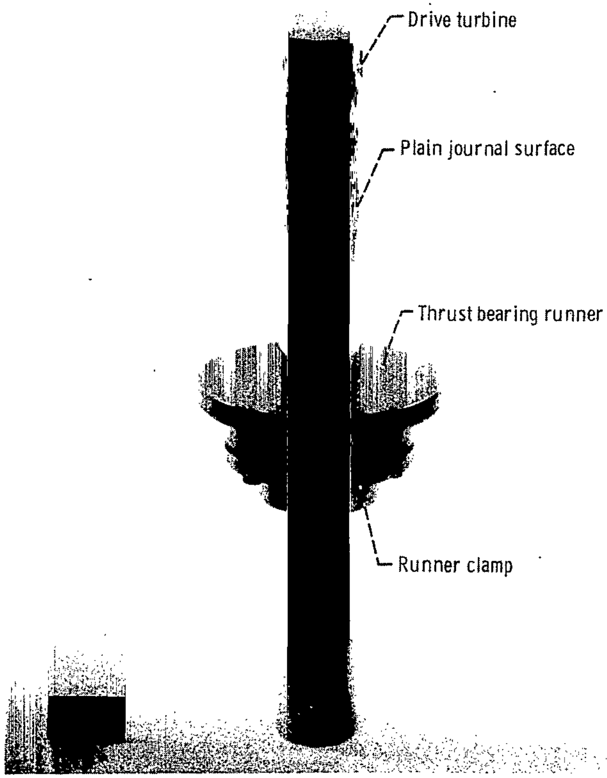


Figure 4. - Shaft assembly with clamp on thrust bearing runner. Externally pressurized journal bearings.

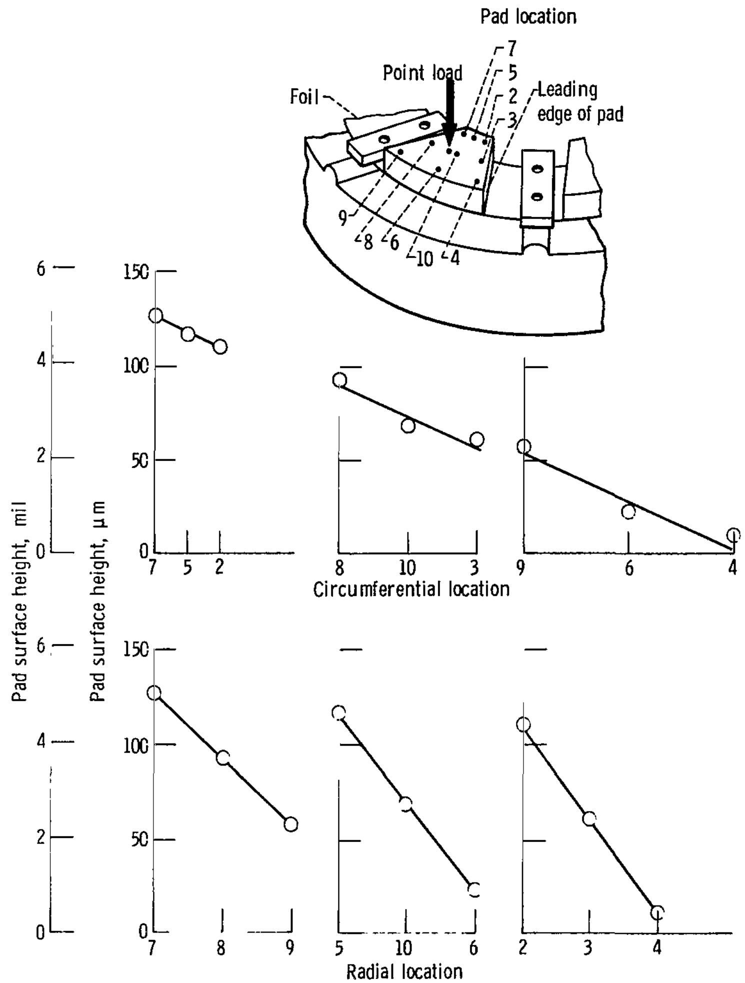


Figure 5. - Air thrust bearing pad surface deflection under point load of 20 newtons (4.5 lb) at pad center of pressure.

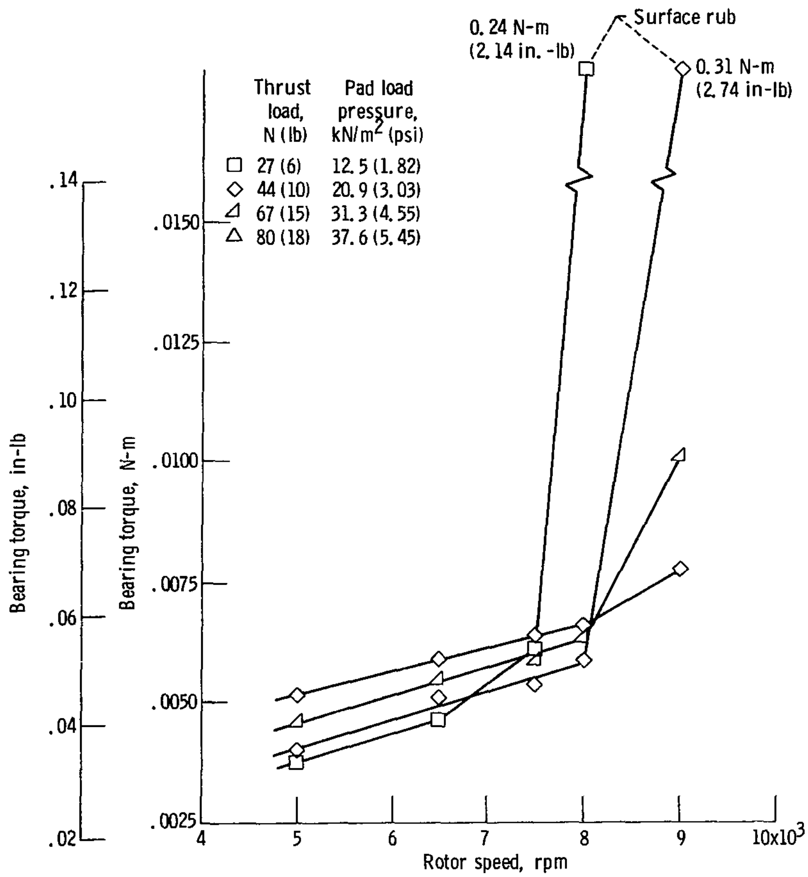
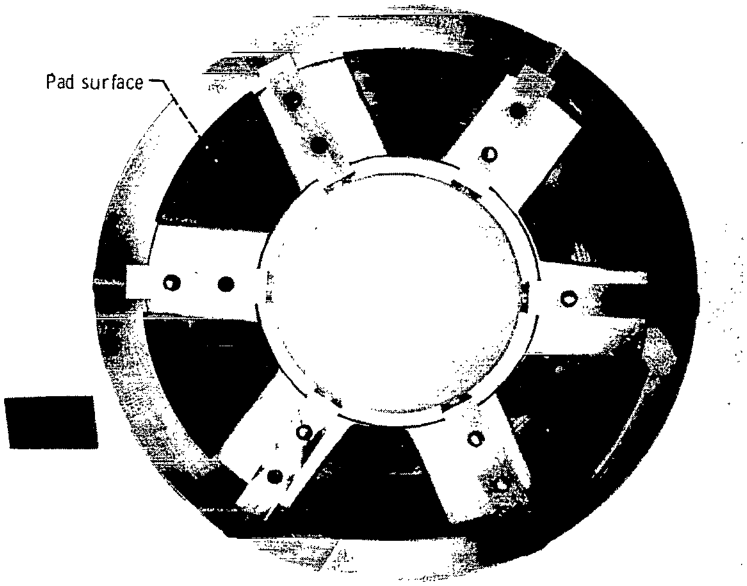
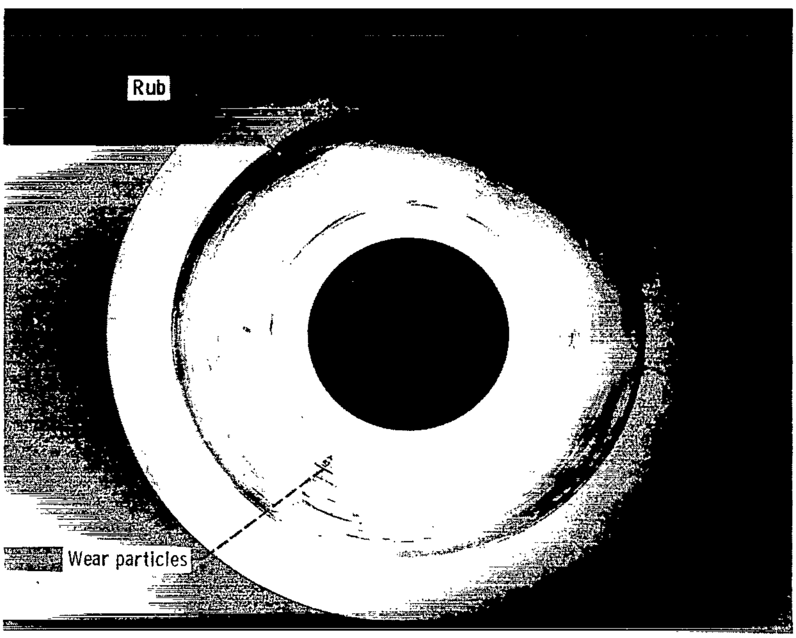


Figure 6. - Bearing torque as function of speed at various thrust loads. Resilient pad thrust bearing with sector pads.



(a) Stator.



(b) Runner.

Figure 7. - Sector pad resilient pad thrust bearing after test.

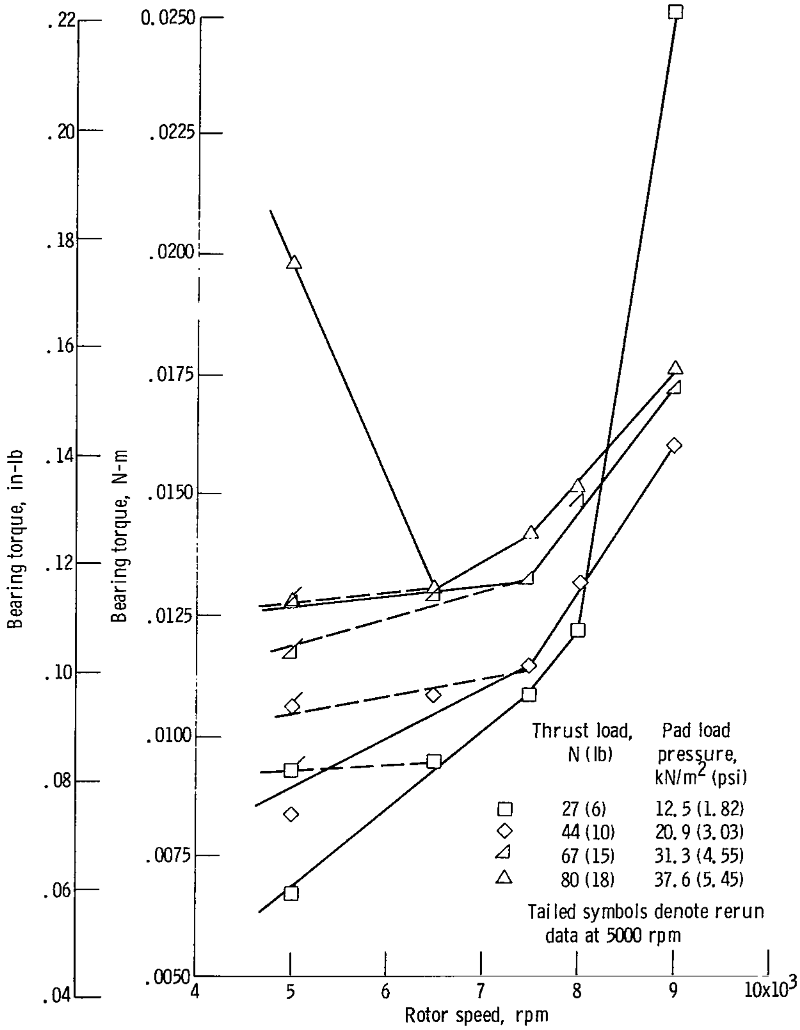
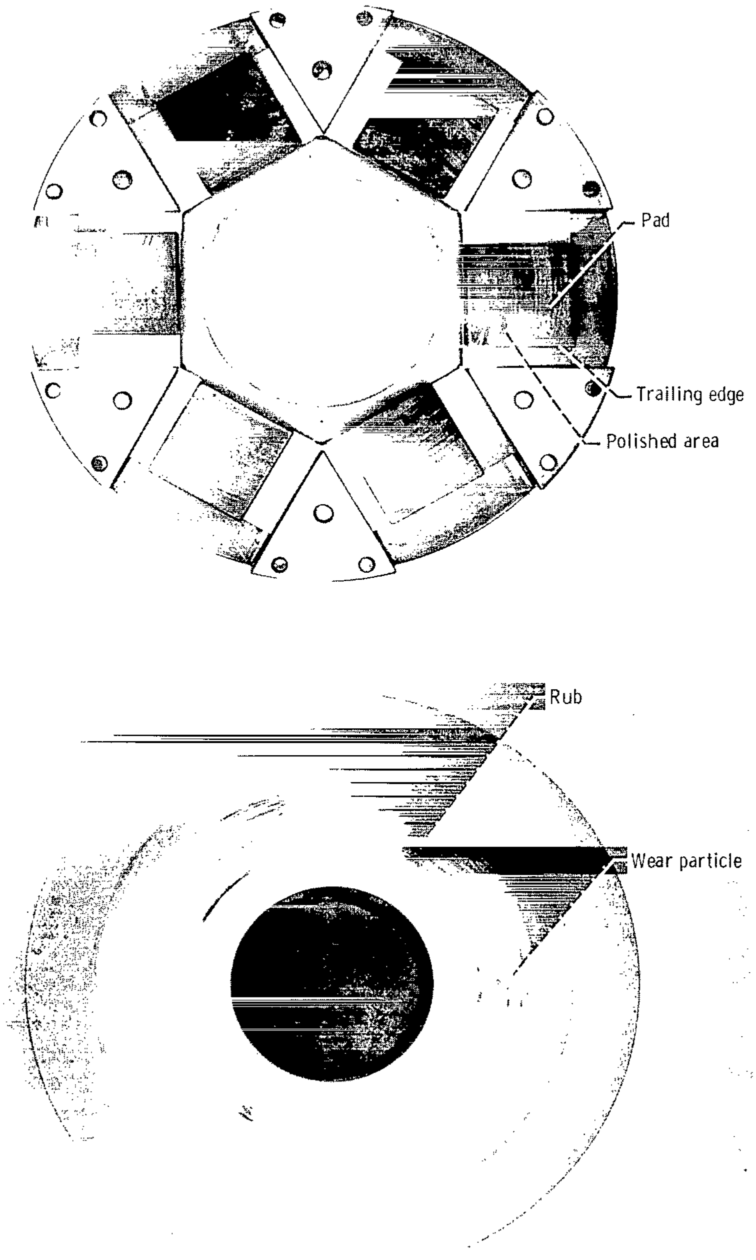


Figure 8. - Bearing torque as function of speed at various thrust loads. Resilient pad thrust bearing with square pads.



(b) Runner.

Figure 9. - Square pad resilient pad thrust bearing after test.

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15. Supplementary Notes

16. Abstract

A new type of resilient-pad gas thrust bearing was tested to determine the feasibility of the design. The bearing consists of carbon graphite pads mounted asymmetrically on foil beams. Two bearing configurations were tested at thrust loads from 27 to 80 newtons (6 to 18 lb) at speeds to 9000 rpm. The outside diameter of the bearing was 8.9 centimeters (3.5 in.).

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