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ANALYZATION, DESIGN, FABRICATION
AND TESTING OF A FOIL BEARING
ROTOR SUPPORT SYSTEM
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ABSTRACT

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The design of the high-speed, self-acting, gas-lubricated, foil-bearing, rotor-support test rig has been completed. This test rig has been designed for a maximum rotational speed of 570,000 rpm, and it is instrumented to continuously measure shaft speed and shaft motion in two planes, foil tension and wrap angle, and the torque exerted by the rotating shaft on the foil journal bearing. The fabrication and assembly of the test rig is nearly complete, lacking only the micrometer movements for the capacitance probes and the high-speed, turbine-rotor assembly.

A study of the stress and vibration characteristics of high-speed rotors showed that rotor design was often limited by the first free-free bending critical speed because conventional gas lubricated bearings cannot support a shaft during acceleration through this critical speed. A shaft was purposely modified to lower its free-free bending critical speed and it was accelerated to this speed (90,000 rpm) while supported on self-acting foil bearings. The shaft broke, but the bearing did not fail. Another shaft with high internal damping has been designed and will be similarly tested.

Author

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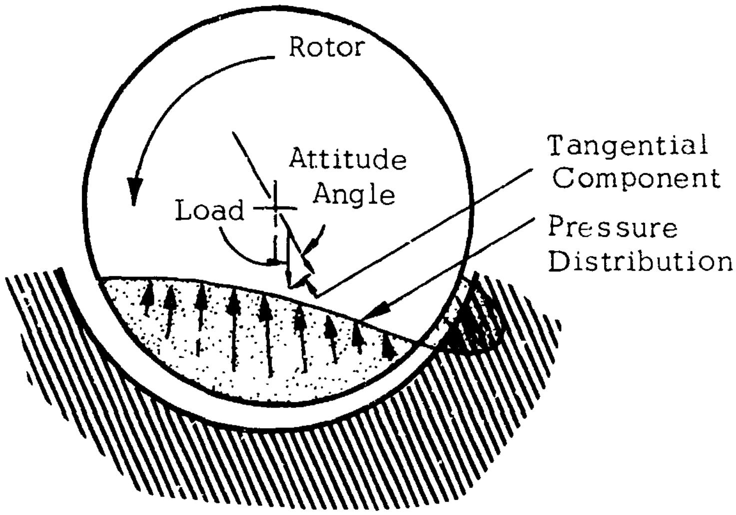
LIST OF FIGURES

I. INTRODUCTION

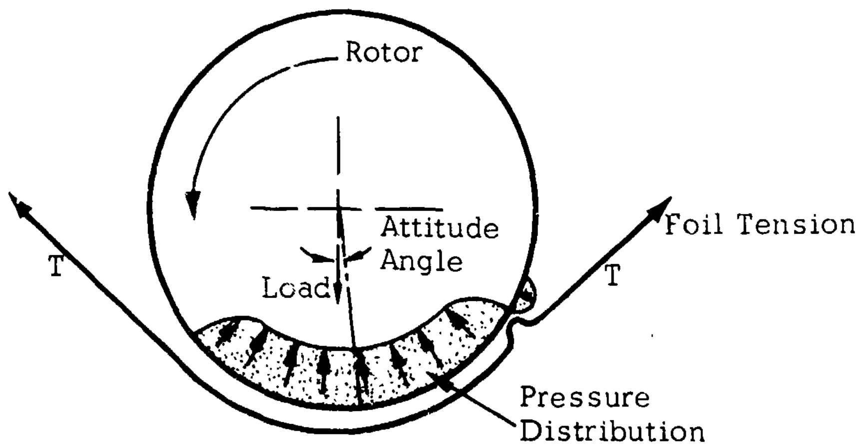
A foil bearing, as shown in Fig. 1, is a fluid bearing in which one member is flexible. Foil bearings differ from conventional bearings in which both surfaces are rigid, in that the flexibility of one member causes the pressure within the bearing to be almost constant. To illustrate this, the pressure distribution in a conventional, self-acting bearing and a self-acting foil bearing, each subtending an arc of 120°, is shown in Fig. 1. The pressure distribution in a bearing with rigid surfaces can be found in most standard texts on lubrication[1], and the pressure distribution of a self-acting foil bearing is described in reference 2. The force and displacement vectors of the same two bearings are also shown in Fig. 1. It can be seen that for the bearing with rigid surfaces, the misalignment between these two vectors is about 25°, while it is less than 5° for the foil bearing.

From Fig. 1, it can be seen that for the bearing with rigid surfaces, the resultant of the fluid film forces has a component which is tangential to the surface of the bearing. It is this tangential component which causes whirl of the rotor. All fluid film bearings, in which this tangential component is present, have a tendency to whirl. This tendency is aggravated when there is little damping, as is the case when the lubricant is a low viscosity fluid, such as gas or a liquid metal. The frequency of the whirl is approximately one-half the rotational frequency, due to the fact that pressure rise in a fluid film bearing is proportional to $U - 2V$, where U is the relative tangential velocity of the two bearing surfaces and V the relative normal velocity of the two bearing surfaces. For journal bearings, these two terms are of equal magnitude when the center of the journal orbits at one-half the spin frequency. Under these conditions, the bearing loses its load capacity, and failure often results.

However, in foil bearings the resultant fluid film force has essentially no tangential component, and there is therefore no tendency to whirl. It is this characteristic of the self-acting foil bearing that makes it particularly attractive for high speed rotor support applications.



Self-Acting Rigid Surface Bearing



Self-Acting Foil Bearing

Fig. 1 Load, Attitude Angle, and Pressure Distribution in Self-Acting, Rigid Surface Bearing and Foil Bearing

II. OBJECTIVES

The objectives of this study are to determine by analytical and experimental means the characteristics and limits of applicability of high-speed rotor supports using self-acting, gas-lubricated, foil journal bearings. This work can be segmented into three major categories:

A. The analysis of a self-acting gas lubricated foil bearing rotor support system consisting of three symmetrical steel foils operated in an inert gas atmosphere;

B. The design and fabrication of a self-acting, gas-lubricated, foil journal bearing rotor support system;

C. The test of this rotor support system with the primary emphasis on the determination of the maximum speed at which the rotor can be operated without encountering self-excited "whirl" instability.

III. PROGRAM DESCRIPTION

The self-acting, gas-lubricated foil journal bearing has been used to support rotors at high speeds, in excess of one-quarter million revolutions per minute[3]. The apparent absence of half-frequency whirl instability in this type of journal bearing has made it attractive for use in high-speed applications or under the conditions of light load, particularly such as would be found in a space environment. There are, however, some undesirable characteristics associated with foil journal bearings. These bearings obtain their freedom from self-excited, half-frequency whirl from the flexibility of the foil. It is this flexibility which creates most of the problems that have prevented the use of this type of bearing in practical applications.

There are four major problem areas associated with the support of high speed rotors on gas lubricated self-acting foil journal bearings:

1. The possibility of a limited rotational speed due to the occurrence of self-excited, half-frequency whirl instability

2. The high starting torque resulting from the "capstan effect"

3. The low frequency critical speeds associated with the mass of the rotor and the small positional stiffness of the foil journal bearings

4. Structural limitations which, thus far, have prevented the operation of rotors supported on gas lubricated journal bearings beyond their first free-free bending critical speed.

These four problem areas can be roughly separated into three categories:

1. High-speed investigations
2. Low-speed and starting investigations
3. Flexural critical speed investigations

The objectives of this program will be best satisfied where test equipment for each of the areas of investigation is designed specifically for testing in that area rather than using a single piece of test equipment for all studies.

1. High-Speed Test Rig

The design of the test rig and its instrumentation for high-speed operation of a rotor supported on self-acting, gas-lubricated foil bearings has been completed. The rig has been designed to support a shaft which can rotate at 500,000 rpm. During operation, continuous measurements can be made of the shaft motion and/or growth by capacitance probes located in two planes: one near each end of the shaft. Four capacitance probes are installed in each of these planes. The journal bearings used to support the shaft consist of three separate foils spaced at 120°, each with individually controlled foil tensions and wrap angles. The foils will be made of Havar*, 1/2 in. in width and of thicknesses from 0.0001 in. to 0.0005 in.

Figures 2 through 4 show the high-speed, self-acting foil bearing, rotor support test rig. The entire unit is enclosed within a 17 in. diameter cylinder of 6061-T6 aluminum alloy with walls 1 in. thick for blast protection.

The radial inflow turbine which drives the shaft is located on the upper end of the shaft. The shaft is supported from below by an externally pressurized, gas-lubricated thrust bearing. The foil journal bearings are located near the extreme ends of the shaft, and the planes of the

*Trade Mark, Hamilton Watch Company

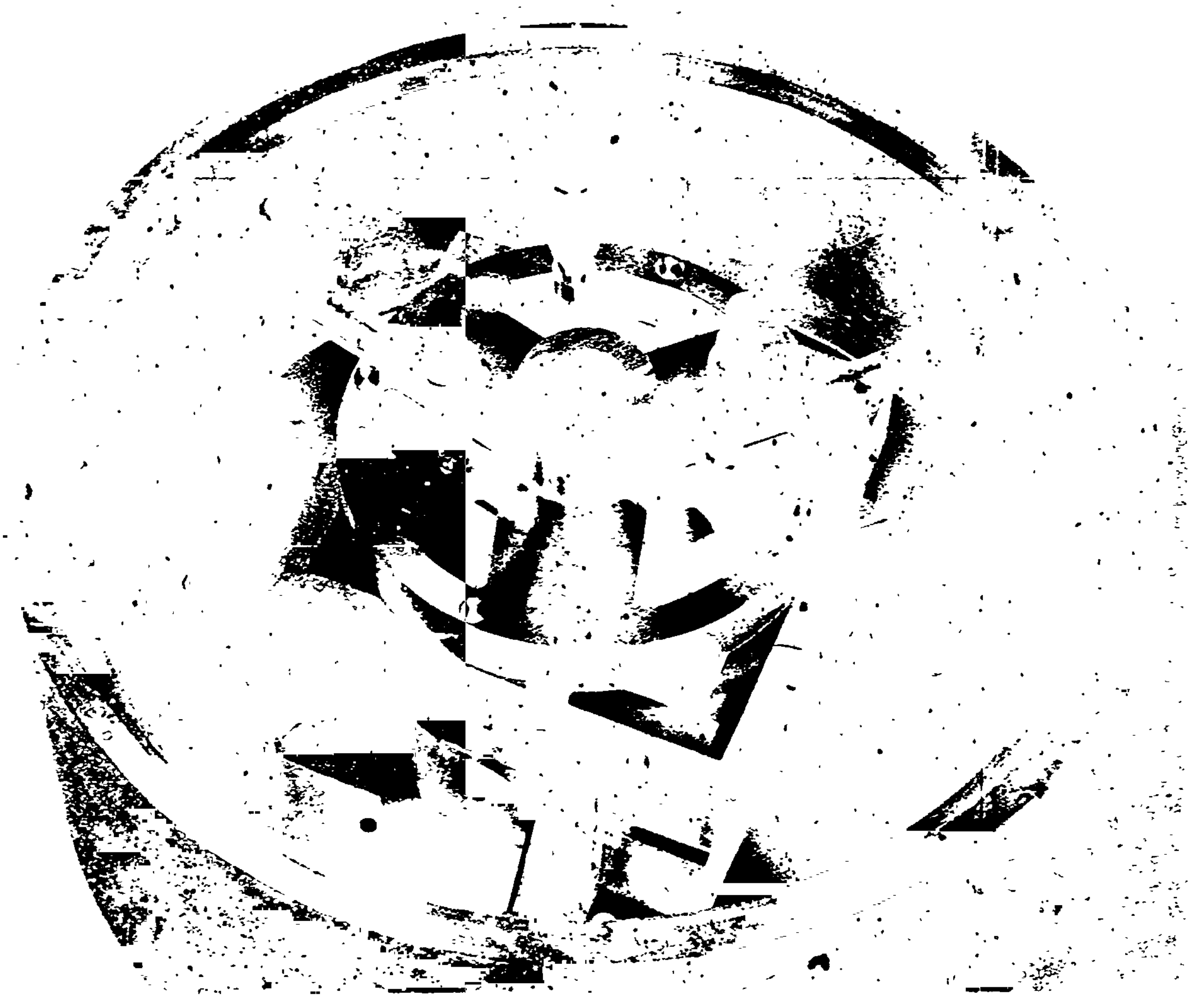


Fig. 2 The High-Speed, Self-Acting Foil Bearing, Rotor Support Test Rig

Fig. 3 High-Speed Test Rig, Disassembled View

Fig. 4 High-Speed Test Rig, Disassembled View Showing Means for Adjusting Foil Tension, Wrap Angle, and Radial Position

capacitance measuring probes are located adjacent to these bearings. Each of the foils can be individually adjusted to change wrap angle, foil tension, and radial position. All of these parameters can be measured, either by micrometer motion, strain gauge, or capacitance probe.

The rotor was designed to satisfy the requirements imposed by the high rotational speed and with sufficient length to allow for adequate instrumentation. There are two conditions which can limit the rotational speed of the shaft: the burst speed of the rotor and the first free-free flexural critical speed of the rotor. The effects of shaft length and diameter on these two criteria are shown in Fig. 5. It can be seen that very high rotational speeds are limited to short, small-diameter rotors. The design point shown in this illustration represents a compromise between high speed and sufficient length for instrumentation; i.e., the shaft has been designed to place its free-free and burst speeds close together and above 500,000 rpm. For the initial runs, the rotor will be made of brass (see Fig. 3). The brass rotor will later be used as the pattern for the electrical discharge machining of the high-speed rotor which will be made of heat-treated Ni-Mark 300*.

It can be seen from Fig. 5 that the occurrence of the free-free critical speed often places severe restrictions on the shape and size of the high-speed rotor. Conventional gas lubricated bearings cannot be used to support a rotor during acceleration through this critical speed. Since the foil-type bearing, because of its flexibility, may be able to provide this extra degree of freedom, an effort was started to investigate that possibility.

2. Low-Speed and Starting Test Rig

A separate test rig(s) will be required to investigate the characteristics of self-acting, gas-lubricated, foil journal bearings during start-up and acceleration through the low-speed critical speeds. Several methods of minimizing the high starting torques, caused by the capstan effect of the nonlubricated foil wrapped about the rotor, will be investigated. Possible methods are:

*Trade Mark, Carpenter Steel Co.

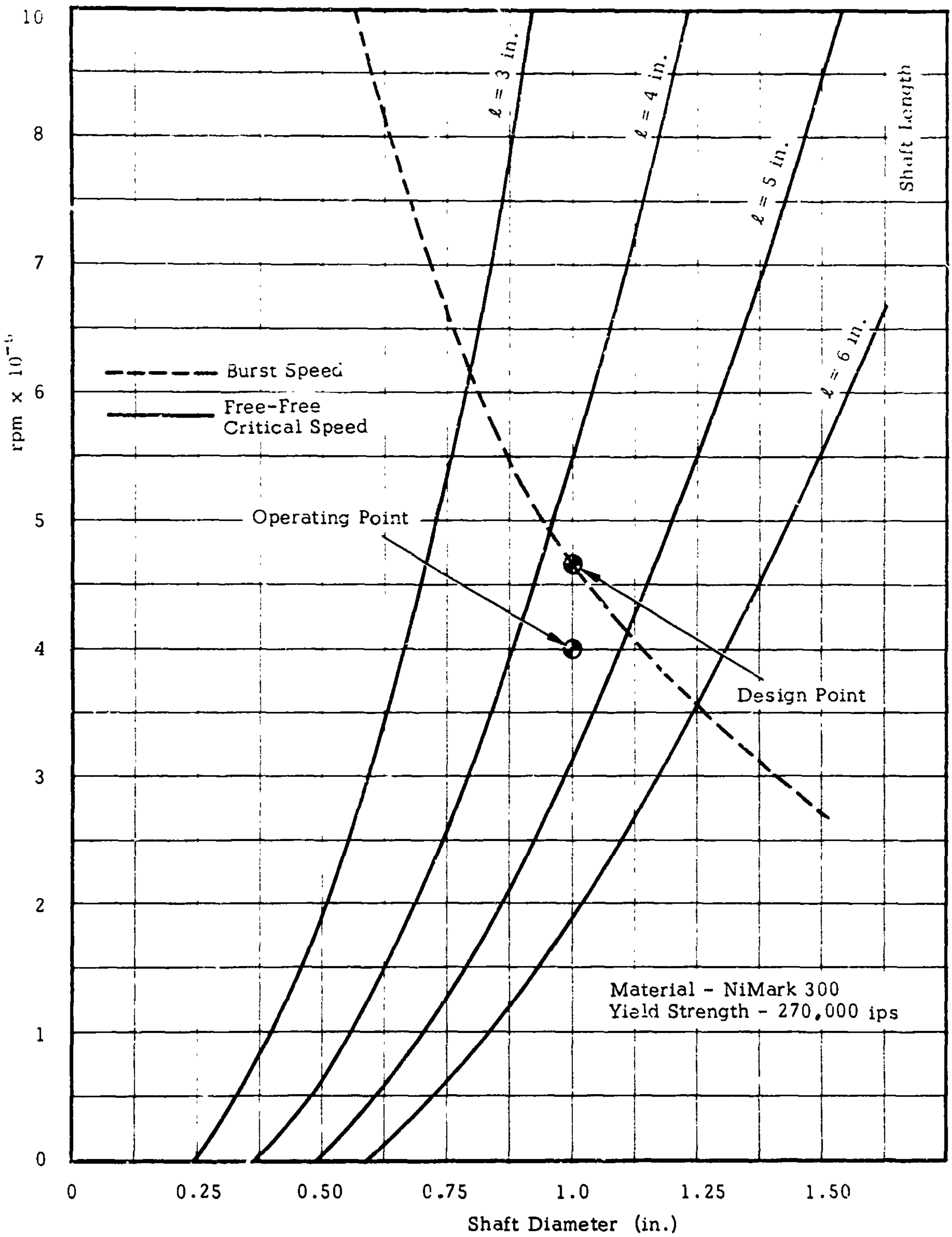


Fig. 5 Burst Speed and Flexural Vibration Characteristics of a High-Speed Rotor

(1) external pressurization of the foil bearing from within the shaft

(2) a second fluid, volatile lubricant

(3) pressurized gas jets applied externally to the shaft

(4) temporary relief of the tape tension and mechanical support of the rotor by auxiliary bearings

The rotor may be driven by an electric motor during some of these high-torque starting investigations.

3. Flexural Critical Speed Test Rig

The flexural critical speed test rig is a modified Ampex type that has been used to spin rotors supported on self-acting, air-lubricated foil bearings at speeds of approximately 250,000 rpm. This test rig is shown in Fig. 6. Here the rotor is supported axially by two externally pressurized thrust bearings, and radially by two sets of three foil journal bearings. The foils used may either be of metal or Mylar*, and the tension is adjustable as shown in Fig. 7. However, the tension cannot be measured. The shaft is rotated by means of air impingement on the blading cut into the center of the shaft. The speed is measured by means of a variable reluctance pickoff which senses two grooves cut into the end of the shaft. Figure 8 shows two of the rotors used with this test rig. The solid or cylindrical rotor is made of stainless steel, and it has been accelerated to the runaway speed of the turbine, i.e., approximately 250,000 rpm, with no sign of bearing instability. The necked-down rotor is also made of stainless steel, but it has not been tested. The necked-down rotor (Fig. 9) had the same length and diameter as the solid rotor (Fig. 8), except for the sections of smaller diameter. This shaft had a measured first free-free, flexural critical speed of 90,000 rpm. An attempt was made to accelerate through this critical speed. However, when this shaft was tested, it went into a violent vertical oscillation at approximately 90,000 rpm, breaking the shaft. Note that this was the natural

*Trade Mark, E. I. DuPont

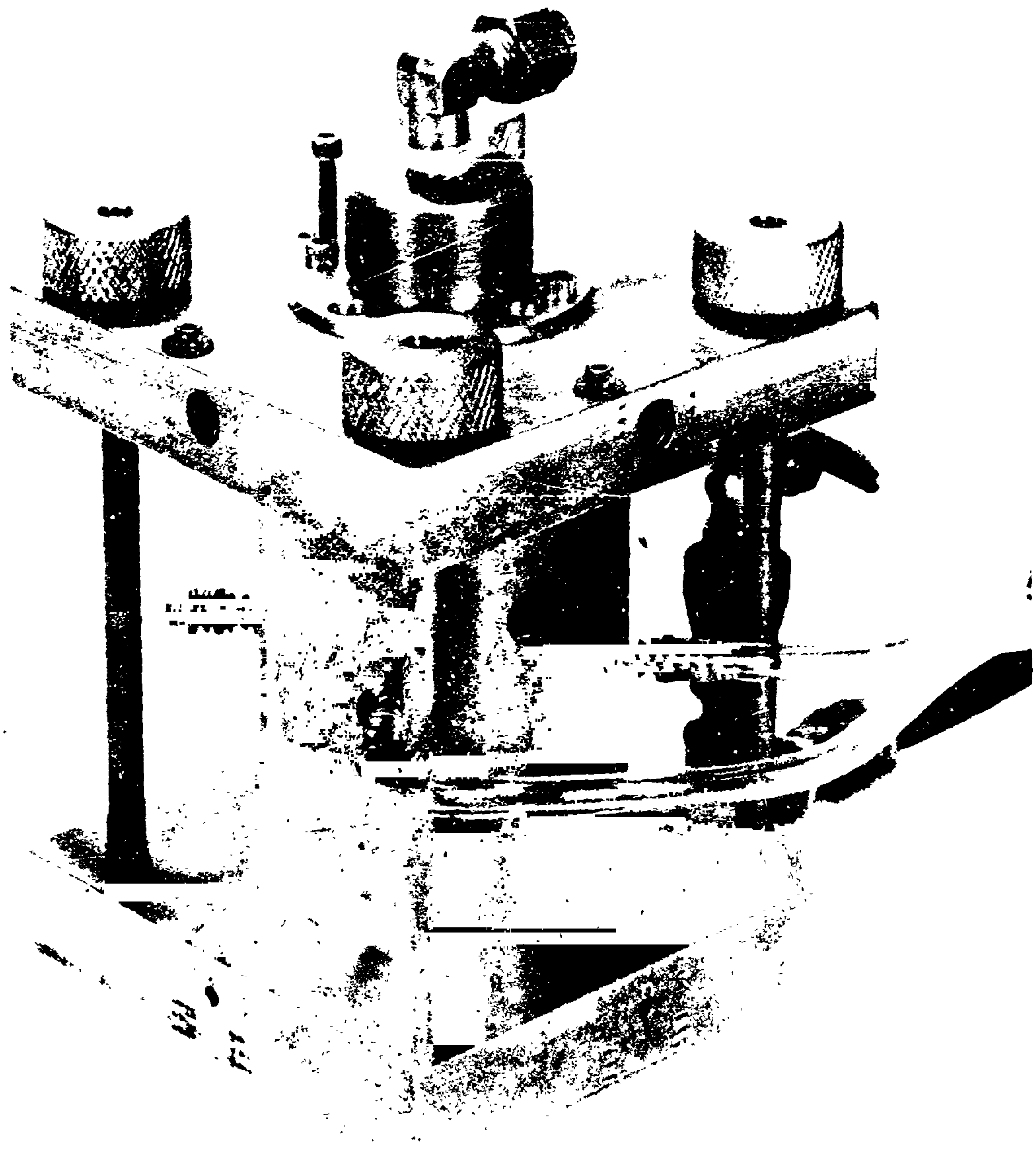


Fig. 6 Test Rig for Flexural Shafts Supported on Self-Acting, Gas Lubricated Foil Journal Bearings

Fig. 7 Disassembled View of Flexural Shaft Test Rig
Showing Externally Pressurized Thrust Bearing
and Foil Tension Device

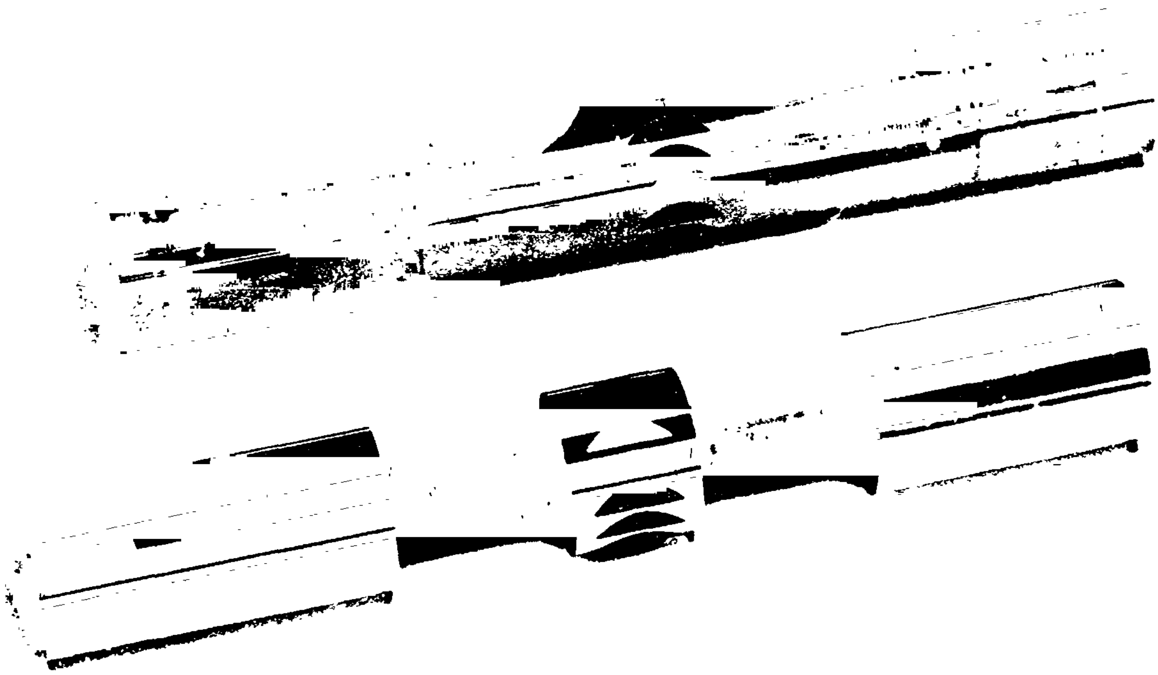


Fig. 8 Flexural Shafts with Free-Free Critical Speeds of 100,000 rpm and 336,000 rpm

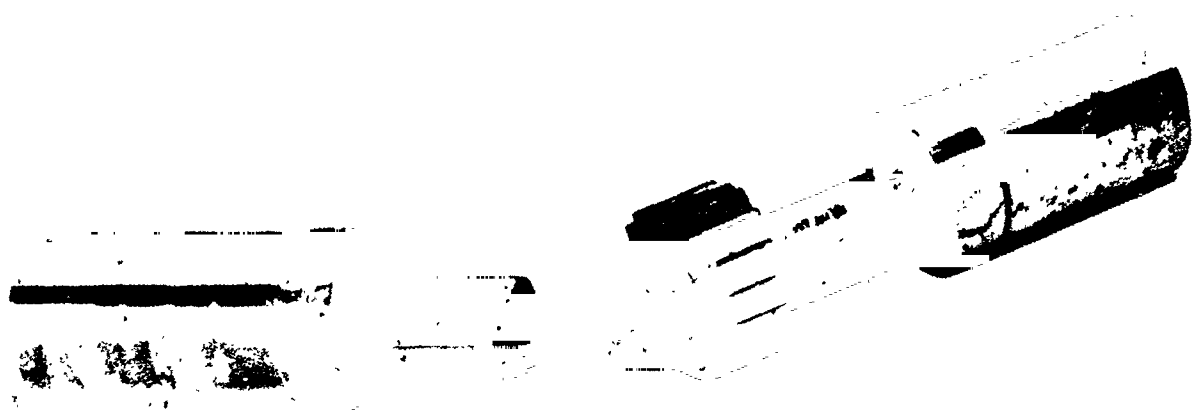


Fig. 9 Flexural Shaft after Operation at the 90,000 rpm Free-Free Critical Speed

frequency of the flexural vibration of the shaft, not a bearing critical or half-frequency whirl instability. The self-acting foil journal bearings did not fail, even though the shaft broke.

Two new shafts have been made to continue these flexural tests. One of the shafts is made of steel (Fig. 8) and the other of magnesium. The free-free critical speeds of these two shafts are 91,000 rpm for the magnesium shaft and 100,000 rpm for the steel shaft. It is hoped that the high internal damping of the magnesium will allow the shaft made from it to be accelerated through its flexural critical speed without shaft failure.

The design of these test rigs will be started during the second quarter of this effort.

IV. REFERENCES

1. W. A. Gross, Gas Film Lubrication, John Wiley and Sons, Inc., New York, 1962

2. A. Eshel and H. G. Elrod, Jr., "Theory of the Infinitely Wide, Perfectly Flexible, Self-Acting Foil Bearing," Paper No. 65-LUBS-3, 1965 ASME Spring Lubrication Symposium, New York, N. Y.

3. Experimental Test No. 4001, Ampex Research Laboratory, Mechanics Section, August 5, 1964