LIGHTWEIGHT MAGNETIC BEARING SYSTEM FOR AIRCRAFT GAS TURBINE ENGINES

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ABSTRACT

Gas turbine engines can benefit from replacement of oil lubricated ball bearings with magnetic bearings. Magnetic bearings can make possible (1) higher thermodynamic cycle operating temperatures, (2) longer life and reduced maintenance, (3) lower bearing power loss, (4) improved rotor dynamics control, and (5) diagnostics for monitoring bearing and turbine rotor operating conditions.

Two generations of lightweight, low power consumption radial magnetic bearing systems were developed to demonstrate the feasibility of using magnetic bearings in U.S. Air Force aircraft gas turbine engines. These designs employ a homopolar bias, actively controlled attraction type radial bearing made of high saturation vanadium-cobalt-iron alloy. Two radial bearings were combined with a permanent magnet bias actively controlled thrust bearing for complete shaft support and control.

One radial bearing design employed a hollow circular cylindrical permanent magnet to energize the radial air gaps. The second radial bearing design was for operation at temperatures over 800 °F and employed a coaxial electromagnetic bias coil instead of the permanent magnet.

The homopolar permanent magnet design was shown to be *only 26% to 28% of* the weight of commercial magnetic bearings[1] of silicon steel with the same load capacity. The power consumption of the permanent magnet bearing including control electronics is only *5%* of that required for a commercial magnetic bearing of the same load capacity. The all-electromagnetic bearing required only *16%* of the power of an equivalent commercial bearing.

Further, the five-axis electronic controller for this new bearing system weighs 30 pounds compared with 268 pounds for an equivalent commercial magnetic bearing controller[2]. These major improvements in reduced weight and power consumption have now made magnetic bearings a viable alternative for aircraft gas turbine engines.

ENGINE SIMULATOR DESIGN

The first step in developing bearing systems for aircraft engines is testing the system in a mass/stiffness simulator. Figure 1 illustrates a shaft mass simulator test rig developed for testing a single radial bearing. One end of the shaft is supported by a face-to-face (DF) duplex pair of ball bearings and the shaft is driven by an electric motor through a flexural coupling. The shaft of this rig has a mass of 90.8 Kg (200 pounds) and a first flexible critical speed of 18,100 rpm. This rig was used for testing and characterizing a single radial bearing at speeds up to 1000 rpm. The rig parts and the radial magnetic bearing parts are shown in Figure 2 prior to assembly.

Pratt & Whitney has developed a simulator for evaluating complete bearing shaft support systems at speeds up to 24,000 rpm as shown in Figure 3. This simulator consists of a steel shaft and bearing pillow blocks for supporting two radial bearings and one thrust bearing. Electric heaters are provided at one radial bearing location for elevating the temperature of the bearing to 800° F. The test shaft is driven by a 50 horsepower electric motor through a 4:1 gear box and quill shaft. Back-up bearings with 100 mm bore steel races and 12 mm diameter silicon nitride balls are provided to capture the shaft in the event of magnetic bearing system power loss

HOMOPOLAR BIAS RADIAL BEARINGS

Two different types of lightweight, low power consumption radial magnetic bearing systems were developed to demonstrate the feasibility of utilizing such bearings in aircraft gas turbine engines. These designs employ a homopolar bias actively controlled attraction type radial bearing with stator laminations of high saturation vanadium-cobalt-iron alloy. One design employed permanent magnet bias, while the second utilized a high temperature all-electromagnetic coaxial bias coil instead of a permanent magnet.

The principle of operation of the homopolar bias bearing [3,4] is shown in Figure 4. A hollow circular ring of axially polarized, rare earth alloy permanent magnet material is used to energize the working radial air gaps, and electromagnet coils are used only for stabilization and control. The permanent magnet produces an *axially* flowing magnet flux (Φ_{pm}) which flows through the cylindrical pole pieces, into the laminated field magnet assemblies and across the two radial air gaps. A multiplicity of electromagnetic coils in the fixed laminated field magnet assemblies generate magnet flux (Φ_{em}) that flows circumferentially around the laminated field magnet stators, but not through the permanent magnets. This approach permits very small electromagnet coils because they only provide control currents, not the primary magnetic field currents, and can operate at a low "duty cycle". The small coil windows in turn make possible a smaller bearing assembly.

Commercial all-electromagnetic bearings generally consist of a multiplicity of pairs of "horse-shoe" like magnets in a circular stator. The control coils are normally operated at a large constant bias current level to linearize the control law and to provide the levitation magnetic field. This requires significantly larger coils and stator field magnet frame to enclose the coils than is required for the new homopolar design coils which are operated at zero bias current.

The homopolar design only has four salient poles on each stator. Further, the homopolar bias magnetic field results in minimum variations in magnetic intensity in the shaft as it rotates. This feature greatly reduces the eddy current losses compared with conventional multiple "horse shoe" pole all-electromagnetic beairngs.

The permanent magnet bias radial bearing design is illustrated in Figure 5. The laminated stators and the outer stator housing were made of vanadium-cobalt-iron alloy. The maximum design magnetic flux density in the stators is 1.6 Tesla. The permanent magnet is an axially polarized hollow cylinder of 30 MgOe energy product samarium cobalt. The coils are Class K polyimide coated copper magnet wire vacuum impregnated with 220° C epoxy resin.

The high temperature radial bearing was designed to operate at 800° F which is above the operating limits of samarium cobalt material used in the permanent magnet bearing. In the high temperature design, the permanent magnet was replaced with a coaxial electromagnetic bias coil as shown in Figure 6. The laminated stators and the outer stator housing were made of vanadium-cobalt-iron alloy. The maximum design magnetic flux density in the stators is 1.1 Tesla. The coils were constructed of nickel copper alloy wire coated with ceramic insulation and potted in a ceramic compound using processes developed at AVCON for high temperature coils.

The design parameters of the two radial bearings are summarized in Table 1.

TABLE 1: Summary of design parameters of radial magnetic bearings.

DESIGN PARAMETER	HOMOPOLAR PERMANENT MAGNET BIAS RADIAL BEARING	HOMOPOLAR ELECTRO MAGNETIC BIAS RADIAL BEARING
Outer Diameter	22.329 cm (8.791 in)	22.329 cm (8.791 in)
Shaft Diameter	15.240 cm (6.000 in)	15.240 cm (6.000 in)
Length (L)	7.468 cm (2.940 in)	7.142 cm (2.812 in)
End Turn Stick-out	1.27 cm (0.5 in)	1.27 cm (0.5 in)
Coil Resistance (per axis)	0.58 Ohm	1.81 Ohm
Coil Inductance	8.0 milli H	3.5 milli H
Weight	9.58 Kg (21.1 lb.)	12.17 Kg (26.8 lb.)

The homopolar permanent bearing shaft position was measured using a pair of eddy current inductive position sensor. One sensor was used for each radial orthogonal axis.

The all electromagnetic radial bearing used a differential inductive ring sensor. This sensor was developed by AVCON and uses the same type of wire and insulation system used in the high temperature bearing.

THRUST BEARING

A permanent magnet bias thrust bearing was integrated into the high speed demonstration test rig (Figure 3). The thrust bearing is a novel design that places the permanent magnet on the stationary part of the assembly to avoid high stresses in the permanent magnet due to centrifugal loading.

The design parameters of the thrust bearing are summarized in Table 2.

TABLE 2: Summary of design parameters of permanent magnet bias thrust bearing.

DESIGN PARAMETER	PERMANENT MAGNET BIAS THRUST BEARING
Capacity	8896 N (2000 lb.)
Outer Diameter	18.8 cm (7.42 inch)
Inner Diameter	12.0 cm (4.75 inch)
Length	7.24 cm (2.85 inch)
Weight	6.81 Kg (15.0 lb
Power Consumption	
@ zero external load	10 watts
@ max. load	570 watts

SERVO CONTROL ELECTRONICS

The five-axis servo electronics assembly is shown in Figure 7. This unit uses five bi-polar PWM amplifiers that output up to 5000 volt-amps per channel and operate at a DC voltage of ± 160 volts. The permanent magnet bias (or homopolar electromagnetic bias) system only require five power amplifiers rather than the ten required by traditional commercial all-electromagnetic bearing systems. The total weight of the five-axis servo electronics is 13.6 Kg (30 lb). Five-axis servo electronics with equivalent volt-amp capacity for commercial all-electromagnetic bearings weigh about 123 Kg. (270 lb)

The open loop transfer characteristics of the radial bearing analog servo electronics controller is shown in Figure 8. The analog controller transfer function is now being programmed into a digital controller with a TMS 320 C31 digital signal processor (DSP). The same PWM power amplifier assembly shown in Figure 7 is used with the digital control.

TEST RESULTS

The two radial bearings were tested individually using the low speed test rig shown in Figure 1. Performance characteristics were determined while the shaft was statically levitated and while spinning at speeds up to 1000 rpm. The test results are summarized in Table 3.

TABLE 3: Summary of measured performance of radial magnetic bearings.

DESIGN PARAMETER	HOMOPOLAR PERMANENT MAGNET BIAS RADIAL BEARING	HOMOPOLAR ELECTRO-MAGNETIC BIAS RADIAL BEARING
Radial Load Capacity		
@ Ambient Temp.	2224N (500 pounds)	6227N (1400 pounds)
@ 800° F	not applicable	4448N (1000 pounds)
Radial Stiffness (Static)	1.75×10^3 N/mm (1×10^7 lb./in)	1.75×10^3 N/mm (1×10^7 lb./in)
Power Draw		
@ zero external load	2.0 watts	30 watts
@ maximum external load	85.1 watts	166 watts
Phase lead @ 300 Hz	31 deg.	30 deg.

CONCLUSIONS

This development effort has demonstrated the feasibility of utilizing magnetic bearings today in aircraft gas turbine engines. The mass of the turbine support system components are:

High Temperature Radial Bearing	12.17 Kg	(26.8 lb.)
Permanent Magnet Bias Radial Bearing	9.58 Kg	(21.1 lb.)
Thrust Bearing (8896^3N capacity - 2000 lb.)	6.81 Kg	(15 lb.)
Control Electronics	13.6 Kg	(30 lb.)
Total Magnetic Bearing System =	42.2 Kg	(92.9 lb.)

The mass of a complete shaft support system including control electronics was shown to be only 42.2 Kg (93 lb.) including two radial bearings, a bi-axis thrust bearing and a five-axis electronics controller. This weight can be further reduced by using flight type electronics packaging for the servo control electronics system and elimination of the 110 volt isolation transformer which weighs over 6.8 Kg (15 lb.). The maximum power consumption under fully loaded conditions was shown to be 166 watts for the all-electromagnetic radial bearing; 85 watts for the permanent magnet bias radial bearing; and the predicted power loss in the 8.9×10^3 N (2000 pound) load capacity thrust bearing at full load is 570 watts. The total input power required for the complete shaft support system is summarized in Table 4.

TABLE 4: Total power required for shaft support system.

COMPONENT POWER DRAW	**POWER DISSIPATED**	
	LEVITATED	**AT MAX. LOAD**
Permanent magnet bias radial bearing	2.0 watts	85 watts
Electromagnetic bias radial bearing	30 watts	166 watts
Permanent magnet bias thrust bearing	10 watts	570 watts
TOTALS	42 watts	821 watts

Design studies by Pratt & Whitney show that this new magnetic bearing system will reduce the weight of an aircraft gas turbine engine by 16% because of the elimination of the ball bearing oil lubrication and circulation system[5]. Consequently, the new homopolar magnetic bearing system with lightweight actuators, small lightweight servo control electronics is a viable option for aircraft engines now. Further work will be required to demonstrate high speed rotor dynamics control and verify low rotating losses in the shaft magnetic bearing target material. These tests will be conducted in late 1994 at Pratt & Whitney in East Hartford, Connecticut.

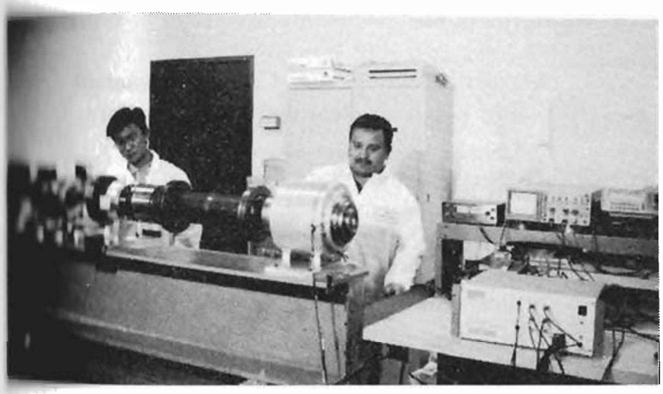

FIGURE 1. Low speed radial bearing test rig

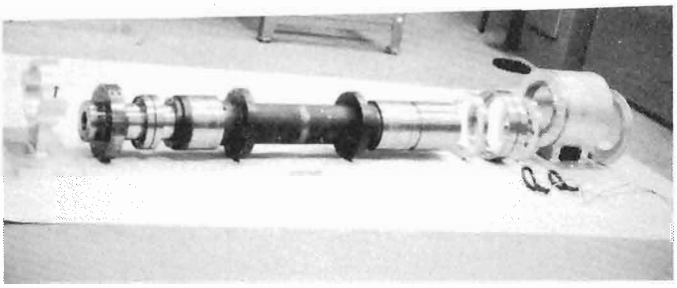

FIGURE 2. Permanent magnet bias radial bearing and turbine engine mass mock- up shaft system before assembly.

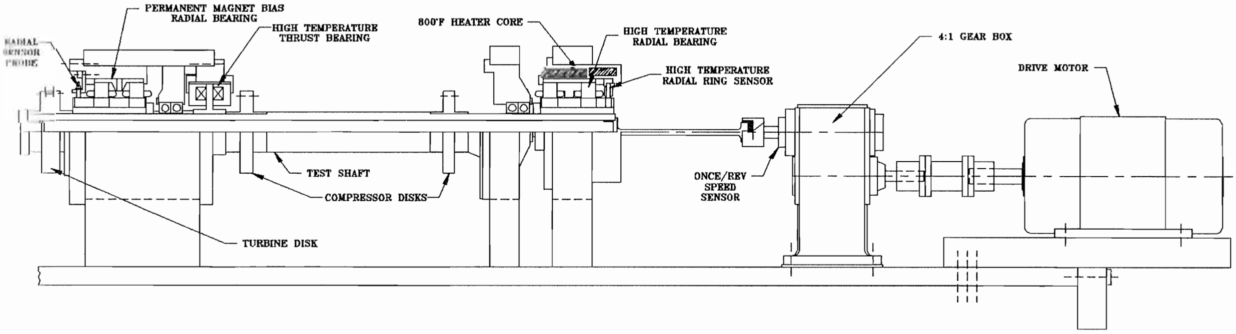

FIGURE 3. Cross section of high-speed magnetic bearing feasibility demonstration rig. (Courtesy Pratt & Whitney)

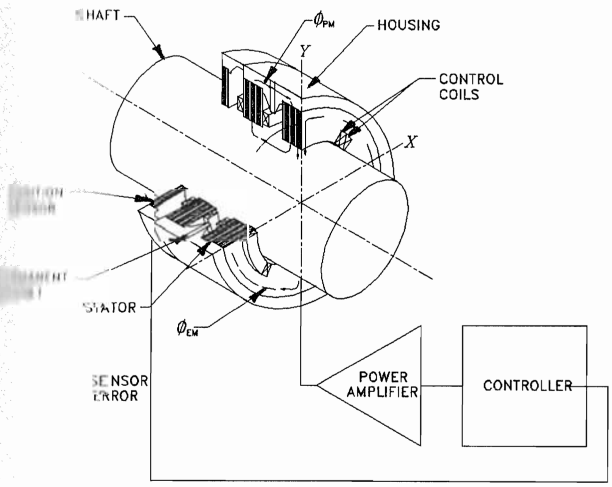

FIGURE 4. Principle of operation of homopolar permanent magnet bias radial magnetic bearing.

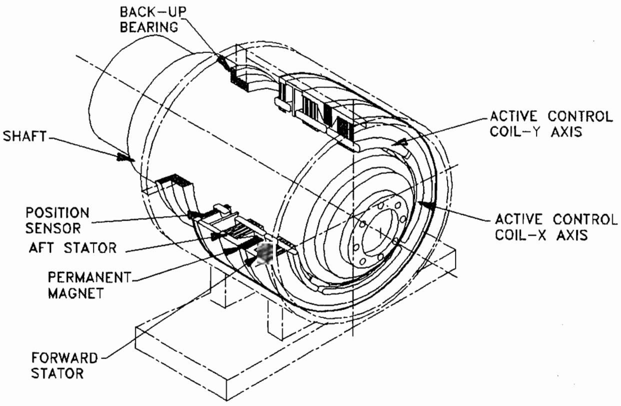

FIGURE 5. Design of homopolar permanent magnet bias radial magnetic bearing.

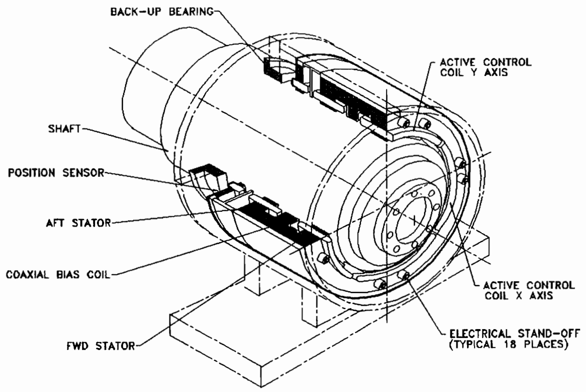

FIGURE 6. Design of coaxial bias coil electromagnetic bias radial magnetic bearing.

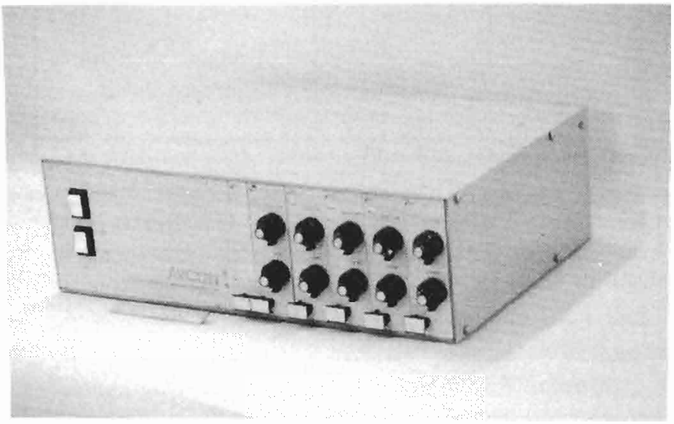

FIGURE 7. Five-axis servo electronics controller using analog compensation.

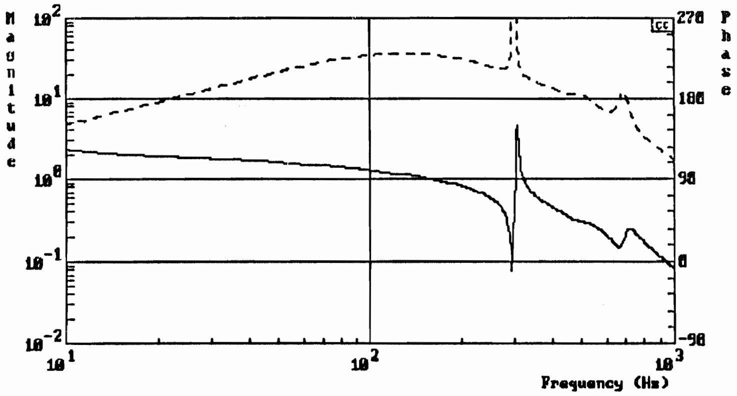

FIGURE 8. Open loop transfer function of radial bearing control.

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