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**HIGH TEMPERATURE, PERMANENT MAGNET BIASED, FAULT TOLERANT, HOMOPOLAR MAGNETIC
BEARING DEVELOPMENT**

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Abstract

This paper summarizes the development of a magnetic bearing designed to operate at 1,000F. A novel feature of this high temperature magnetic bearing is its homopolar construction which incorporates state of the art high temperature, 1,000F, permanent magnets. A second feature is its fault tolerance capability which provides the desired control forces with over one-half of the coils failed. The construction and design methodology of the bearing is outlined and test results are shown. The agreement between a 3D finite element, magnetic field based prediction for force is shown to be in good agreement with predictions at room and high temperature. A 5 axis test rig will be complete soon to provide a means to test the magnetic bearings at high temperature and speed.

NOMENCLATURE

HT – High Temperature
HTRMB – High Temperature Radial Magnetic Bearing
PM – Permanent Magnet
OD – Outer diameter
ID – Inner Diameter
RBTFR – Radial Bearing Force Test Rig
HTPM – High Temperature Permanent Magnet

INTRODUCTION

The purpose of this research is to design and develop a high-temperature (1000°F) Magnetic Bearing System using High Temperature Permanent Magnets (HTPM), developed by EEC. The system consists of two radial bearings, one thrust bearing, a motor and 2 sets of catcher bearings all operating at 1,000F.

This high temperature magnetic bearing system would be used in high performance, high speed and high temperature applications like space vehicles, jet engines and downhole drilling equipment. The bearing system has a target design load equal to 500 lb (2225 N). Another objective is to design and build a test rig fixture to measure the load capacity of the designed high temperature radial magnetic bearing (HTRMB). This is referred to as the Radial Bearing Force Test Rig (RBFTR).

The permanent magnet bias of the radial magnetic bearing reduces the amount of current required for magnetic bearing operation. This reduces the power loss due to the coil current resistance and also increases the system efficiency because the magnetic field of the HTPM is used to take up the major portion of the static load on the bearing. The bearing was designed to produce 500 lb (2225 N) of force at 1000°F. The bias flux of the Homopolar radial bearing is produced by the HTPM.

A procedure was developed using the RBTFR to measure actual load capacity of the designed bearing in the test rig. All the results obtained from the experiment were compiled and analyzed to determine the relation between bearing force, applied current and temperature.

Overview

Magnetic bearings are one of the biggest advancements in the area of bearings for rotary machinery. The major need for study in this area is because magnetic bearings have found their application in extreme environment. Some properties include the absence of contact between rotating members and active control of the dynamic parameters. Recent research has increased its application into the high temperature environment [1]. Magnetic bearings are a cutting edge technology that will allow engines to operate at speeds and temperatures well beyond the limits of current technologies. Magnetic bearings are well suited to operate at elevated temperature, higher rotational speeds, and extreme altitudes and are a promising solution to current limitations [2] [3].

Along with the higher speed capabilities from no contact between the rotor and stator, the elimination of the lubrication systems is also a big benefit [4]. The absence of lubrication and contaminating wear make the magnetic bearing system exceptional for use in clean rooms, sterile rooms, or vacuum rooms. The bearing losses are less than conventional bearings resulting in lower operating costs. Magnetic bearings are being implemented into turbo-machinery because of their lower maintenance costs and higher life span. It is also possible to adapt the stiffness and the damping of the bearing to absorb any vibration that might occur [5]. In addition to supporting loads, magnetic bearings directly measure bearing reaction forces. This enhances their application as engineering tools by providing a diagnostic and measurement tool for rotating machines. In order to obtain these loads the force as a function of current, air gap, operating frequency, and alignment must be known [6].

Objectives

The objectives of the research are as follows:

- Design and build a permanent magnet biased, radial magnetic bearing that can operate at 1000° F while supplying 500 lb (2225 N) load capacity.
- Testing of the radial bearing to determine the force to current relationship and correlate with the predicted relation.
- To determine the position and current stiffness of the radial bearing.
- Design and build a full magnetic bearing test rig broadly consisting of permanent magnet biased 2 radial bearings, 1 axial thrust bearing, 1 motor and 2 catcher bearings that can be used to test magnetic bearings and motors at 1000° F.

Literature Review

A permanent magnet biased magnetic bearing that has coplanar geometry is usually known as a homopolar

design (Meeks et al. [7]). such geometry was found in electromagnetic bearings. Homopolar bearing designs have permanent magnets in the back iron path to provide the bias flux. Electric coils can also be used in place of permanent magnets to provide the bias flux to the stators. Sortore et al. [8] have shown experimental data with significant reduction in power consumption utilizing permanent magnet biased homopolar bearings. Lee, Hsio, and Ko [9] provide analysis and advanced circuit models to predict the flux paths and other parameters for bearings using permanent magnets. A comparison between the predicted and measured force and stiffness characteristics of bearing were performed by Imlach, Blair, and Allaire[10]. A study of hysteresis effects was published by Fittro, Baun, Maslen, and Allaire [11].

Palazzolo and Kenny [12] of TAMU-VCEL described a single plane magnetic bearing with permanent magnet bias. Provenza, Palazzolo, Jansen et al. [3] described the open loop, experimental force and power measurements of a radial, redundant-axis, magnetic bearing at temperatures to 1000 °F and rotor speeds to 15,000 rpm. Mohiuddin [1], described and demonstrated the working of a high speed and high temperature electromagnetic axial bearing. Hossain [2], experimentally showed the results of a radial bearing that uses poles with permanent magnets which was designed for supplying 500 lb (2225 N) at 1000°F and permanent magnets provide the bias flux.

The focus of this paper is the measured response of the mechanical components. The fault tolerant capability of the magnetic bearing is covered in depth in [13]. Other papers on high temperature magnetic bearings include [14] and [15].

DESIGN OF HTRMB

Magnetic Design

Initially a HTRMB was designed and built, but due to certain factors like weight and the linearity of the load capacity in the design there was need to improve on the original design [2]. The following sections discuss the new Improved Design of the High Temperature, PM Biased Radial Magnetic Bearing (HTPMRMB) in comparison with the original design.

The dimensions for a lighter weight high temperature radial bearing with Hiperco components were calculated. This weighed 46.8 lb (213 N) including the wire coils, magnets, and stator, back-iron, and rotor laminates. The FEA model of the bearing confirmed the weight without the coils to be 42.4 lb (193 N). The FEA calculation of the load capacity with 15 Amp current was 656 lb (2978 N) at 1000 degrees F in the direction halfway between two coils separated by sixty degrees. The bias flux density in the air gap was 0.53 Tesla at 1,000F. The dimensions of this radial bearing are given in Table 1 .

A comparison between several key parameters of the original radial bearing and the new design of the radial bearing are shown in Tables 1 and 2. The new and original bearings have a similar outer diameter. The weight of the new design is predicted to be lower, although the actual weight of the new

design may be a little heavier due to the weight of the coil potting, etc. The main benefit of the new design, is that for a comparable weight it has a better linear load capacity which is over 600 lb (2670N) with 15 A. This is due to the new design possessing a higher bias flux in the air gap.

Table 1 HTPMRMB Dimensions

Rotor Laminate ID	3.81 cm
Rotor Laminate OD	8.13 cm
Stator Laminate ID	8.23 cm
Stator Slot Dia.	14.2 cm
Stator Laminate OD	17.51 cm
Stator Thickness	2.9 cm
Distance Between Stators	1.44 cm
Magnet ID	17.51 cm
Magnet Covered Angle	55 degrees
Magnet Thickness	2.9 cm
Magnet OD	21.17 cm
Backiron ID	21.17 cm
Backiron OD	23.74 cm
Number Turns/Coil	36
Inductance/Coil	.00206 H
Wire Diameter with insulation	0.16 cm
Inductance/Coil	.00206 H

Table 2 Comparison Between New and Original HTPMRMB

Parameter	New Design	Original Design
Bearing OD	23.75 cm	23.11 cm
Bearing Length	8.18 cm	10.08 cm
Bearing Weight	208 N	267 N
Air Gap Flux	.53 Tesla	.45 Tesla

Figure 1 shows the solid model of the new design for the *HTPMRMB*.

Figure 2 shows the flux density with both the permanent magnet flux and 15 amp coil current applied. The two top poles have bias flux with control flux added, while the two bottom poles have bias flux with control flux subtracted. The load capacity is determined from this current distribution. The units in figure 2 are Teslas.

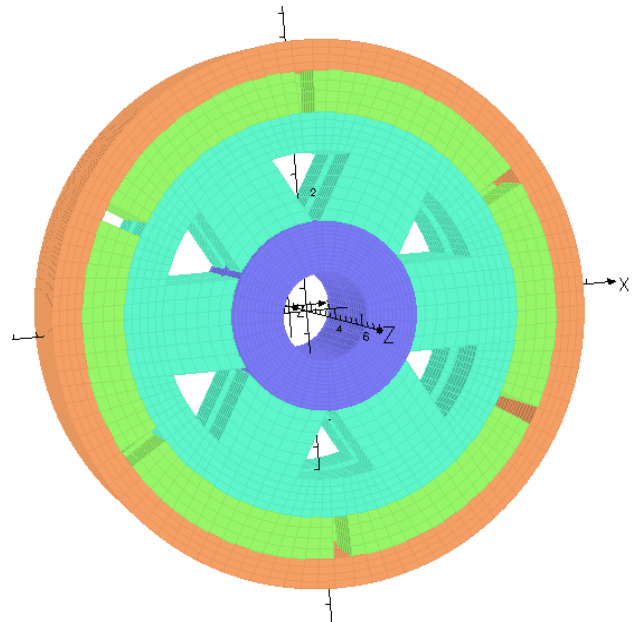


Figure 1 Solid Model of the Radial Bearing

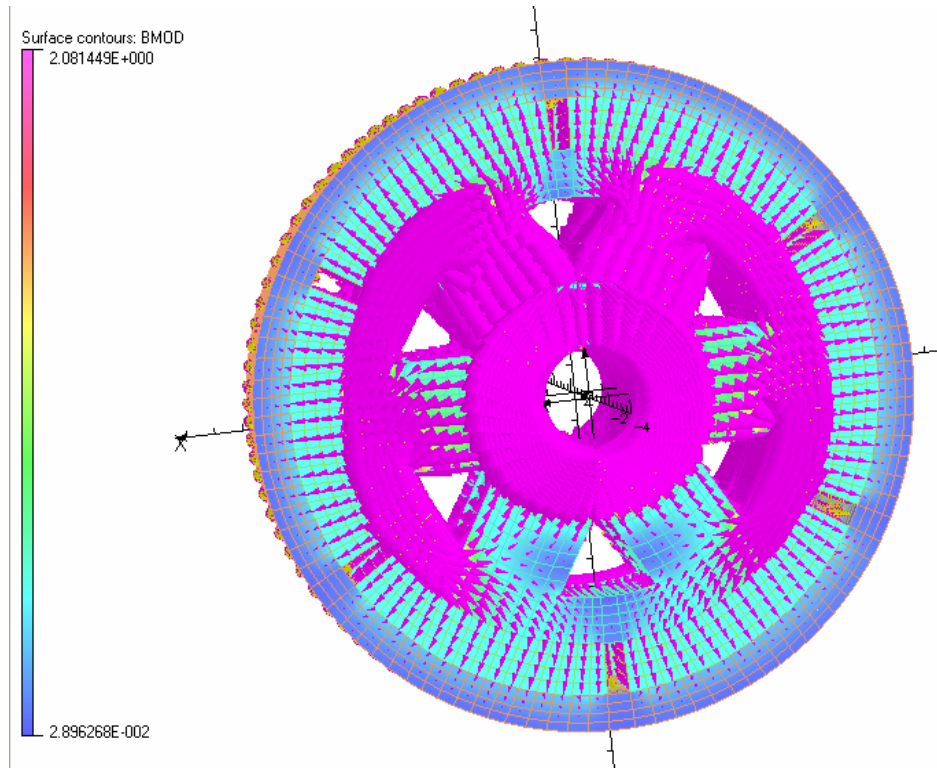


Figure 2 Results of FEA calculations with Bias and Control Flux

An FEA model was made for the high temperature radial bearing with a segmented backiron. Segmenting the backiron makes the bearing much easier to manufacture and assemble. The backiron segments were separated by .1 inch gaps. This is the same as a separation angle of 1.375 degrees. The force, according to the finite element analysis, was 656 lb (2919 N) with the conventional cylinder backiron and 602 lb (2679 N) with the segments. This force is with 15 A current applied to the coils on the top and -15 A current applied to the coils on bottom.

Figure 3 shows the segmented parts of the backiron.

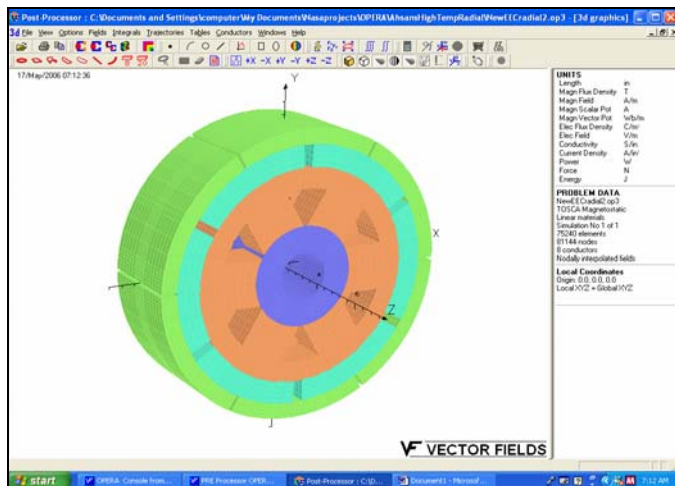


Figure 3 Segmented Back-Irons

Some model results are:

Radial bearing magnetic flux density information:

Bias flux density in air gap at 70 deg F = 0.98 T average

Bias flux density in air gap at 1000 deg F = 0.6 T average

Table 3 and Figure 4 show the control flux density in air gap when all twelve coils have equal current.

Table 3 Control Flux Density

Amp-Turns	Average Gap Control Flux (TESLA)	
	1000 F	70 F
0	0.00	0.00
135	0.20	0.23
270	0.31	0.40
405	0.37	0.45
540	0.42	0.50

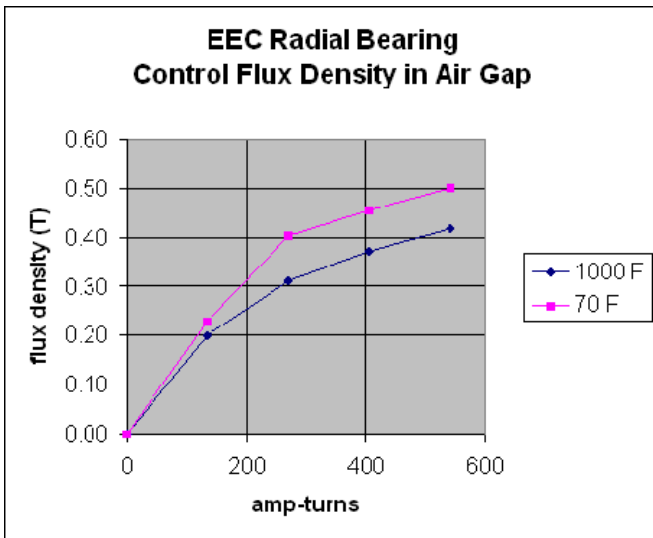


Figure 4 Flux density vs. Amp-turns

HTRMB Mechanical Design

After studying a number of designs with different considerations of size, weight, cost and most importantly the magnetic design feasibility as discussed above, the final design of the radial bearing stator was chosen. This section clearly explains the designs for all the parts that were used in the radial bearing fabrication.

The magnets that were finalized were made by EEC to serve as a high temperature source of magnetic flux for bias. Orientation of the magnetic field is such that the O.D. is a North Pole and the I.D. a South Pole. Six (6) magnets were made with such pole orientation for stator 1 and another 6 magnets with reversed polarity for stator 2. Each of these magnets consisted of 5 small segments that were cemented together with high temperature cement to obtain the arc shape.

The laminates are made from Hiperco50 and are stacked together. A dummy shaft is used to position the rotor laminate stack in the stator laminate stack. The dummy shaft is made of non-magnetic Titanium Ti-4AL-6V alloy. The backirons serve as a means to transmit bias flux along the axial direction from one stator to the other. They are also made from Hiperco 50 material.

Design of RBFTR

The apparatus shown in Figure 5 was fabricated and employed to provide accurate measurements of the radial forces produced by the radial bearings vs. temperature, current and relative position of the rotor in the bearing clearance between the stator and rotor laminate stacks. A previous build of this apparatus [2] exhibited excessive flexibility which caused the test results to possess a high uncertainty. This new design possesses the following improvements:

- Higher stiffness load cells
- Increased resolution on load cells due to internal amplification feature.

- Better quality dial gauges to indicate the radial gap and rotor/stator relative position at all temperature conditions.
- High stiffness bearing supports provided by solid ring plates that provide high load capacity and act as a thermal barrier. These plates may be easily modified to accommodate thermal expansions if required.

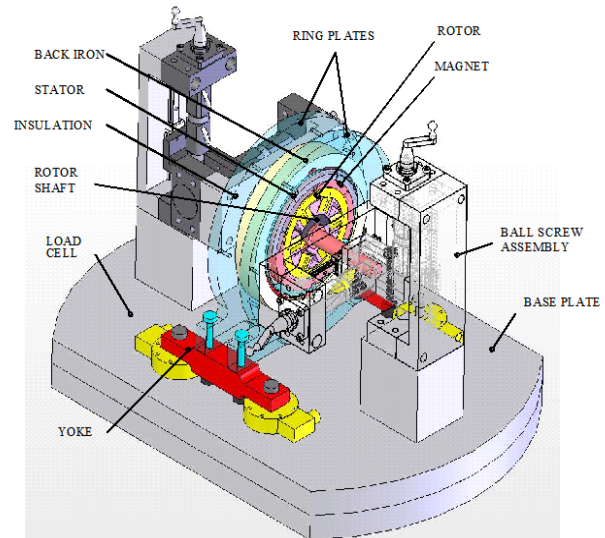


Figure 5 Solid Model of Radial Bearing Force test rig

A cooling base plate is employed to provide a heat sink to maintain acceptable temperatures for the load cells. This table will also be employed in the actual spin rig for the complete bearings / motor / catcher bearing system. The base consists of two base plates made of steel that are sandwiched and sealed together for cooling water to be passed in between them. The ring plates provide support for the bearings and serve as a load transmission path from bearings to the load cells. The plates are manufactured from a special Titanium alloy material (Ti4Al6V) which provides a thermal barrier to keep the bearings hot and the load cells cool. The yoke shown in Figure 5 is used to transmit the bearing load equally to the two load cells. The ball screw assembly shown was designed to provide both vertical and horizontal displacement to rotor shaft at both of its ends, with minimal support deflection. This provides micrometer stage action with a very rigid mechanism. The ball screw system was replaced with monolithic shaft support blocks to improve the systems stiffness and consequently reduce the deflections.

Figure 6 shows the entire exploded view of the stator sub assembly and Figure 7 shows the rotor sub-assembly that is inserted through the stator.

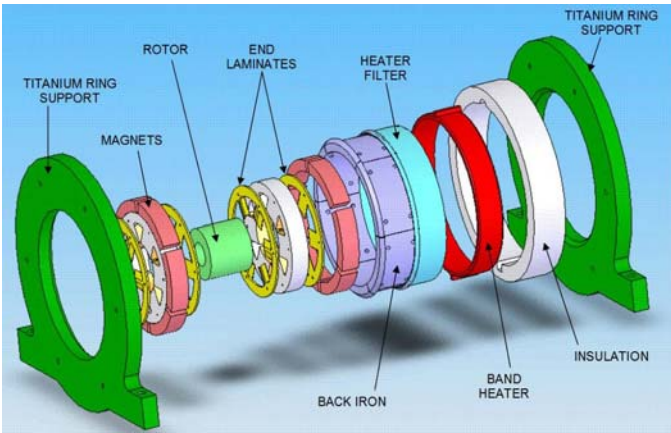


Figure 6 Stator Sub-Assembly

The rotor laminates and the shaft is inserted into the bearing. This was done using plastic shims and a trolley mechanism to align the shaft in the X and Y directions as the shaft gets drawn into the stator. Figure 9 shows the dial indicators used to measure the shaft position relative to the stator.

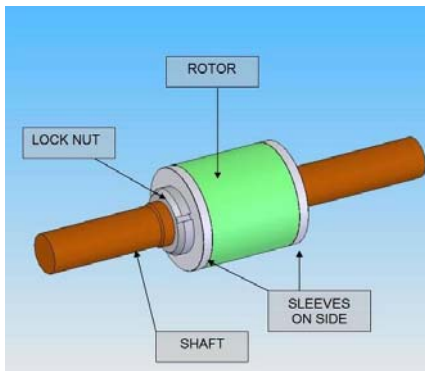


Figure 7 Rotor Sub-Assembly

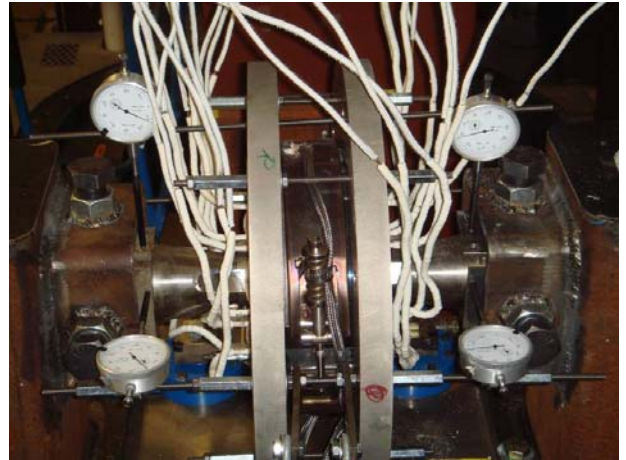


Figure 9 Alignment of the Force Test Fixture

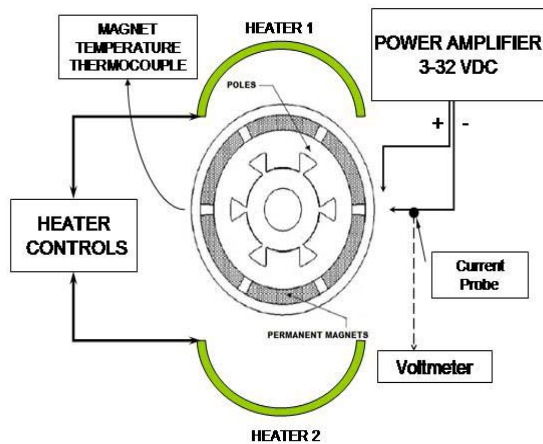


Figure 8 Schematic of the test setup



Figure 10 Monolithic shaft support

Figure 8 shows the schematic diagram of the complete test setup which consists of heater controls and power supply to the stator coils.

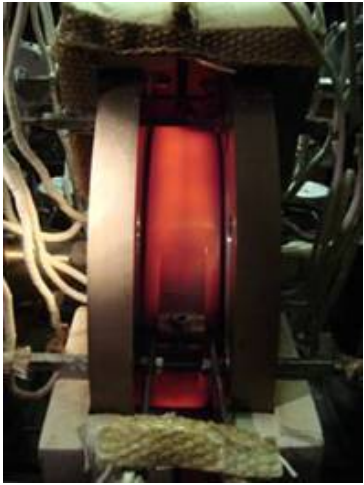


Figure 11 Band Heater at 1,000F



Figure 12 View of Inner Face of Stator with High Pot Instrument

Insulation plates are used to provide a thermal barrier to efficiently heat the system. Figure 10 shows another view of the test rig from the side. The band heaters used to heat the stator become red hot at 1,000 F as shown in Figure 11.

High Potting Test Results

The “High Potential” test is utilized to check for breakdown voltage and effective resistance. Stator 1 is shown in front of the high potting instrument in Figure 12.

Tables (4) and (5) below show measurements of current with 500V applied between the magnetic bearing stator lamination stack and the individual coils. This “High Potting” procedure is utilized to check for breakdown voltage and effective resistance. The results appear excellent even at

1,000F since the actual maximum operating voltage will not exceed 180V.

No signs of breakdown occurred during all of the High Pot testing. The trend surely shows a very significant increase in current between the room temperature and 1,000F states. This verifies that high potting must be performed at high temperature in order to check the integrity of the insulation at high temperatures. The tables correspond to the 2 radial bearings with 2 stators per bearing

Table 4: HTRMB-1, Stator-1 High-pot results

	Temp degrees F	Pole No.	V_{DC} volts	I_{DC} amps $\times 10^{-6}$	R_{eff} Mega- Ohms	V_{AC} volts	I_{AC} amps $\times 10^{-6}$
HIGH TEMPERATURE	1000	1	500	25	20.0	500	400
	1000	2	500	140	3.6	500	520
	1000	3	500	120	4.2	500	480
	1000	4	500	180	2.8	500	520
	1000	5	500	200	2.5	500	580
	1000	6	500	200	200	2.5	500

ROOM TEMPERATURE	76	1	500	0	∞	500	28
	76	2	500	0	∞	500	30
	76	3	500	0	∞	500	30
	76	4	500	0	∞	500	30
	76	5	500	0	∞	500	30
	76	6	500	0	∞	500	30

Table 5 : HTRMB-1, Stator-2 High-pot results

	Temp degrees F	Pole No.	V_{DC} volts	I_{DC} amps $\times 10^{-6}$	R_{eff} Mega-Ohms	V_{AC} volts	I_{AC} amps $\times 10^{-6}$
HIGH TEMPERATURE	1000	1	500	50	10	500	420
	1000	2	500	100	5	500	400
	1000	3	500	150	3.3	500	420
	1000	4	500	180	2.8	500	450
	1000	5	500	180	2.8	500	420
	1000	6	500	150	3.3	500	460
ROOM TEMPERATURE	76	1	500	0	∞	500	32
	76	2	500	0	∞	500	35
	76	3	500	0	∞	500	34
	76	4	500	0	∞	500	34
	76	5	500	0	∞	500	28
	76	6	500	0	∞	500	30

Radial Bearing Force Test Results

The radial bearing force was measured to be 629 lbs (2799 N) at a magnet temperature of 920 F and a current of 13.3 amps. This value is considered to be conservative (low) for the following reason:

- 2 of the 4 load cells failed due to excessive loading. These were replaced with solid steel “dummy” load cells of dimension similar to the actual load cells. The moment M_D results from the bolted connection of the dummy load cell and the stator support stand. This moment reduces the load on the actual load cell A. The moment M_D is neglected and moments are summed about D to obtain a conservative (low) estimate of the magnetic force F_M which has twice the measured force R_A .

Experimental Procedure

The experiment on the test rig was carried out at room temperature and at higher temperatures. The steps below explain how the data was taken for the test rig.

1. First the shaft is brought to its center by adjusting the bolts on the shaft support tower and checking if the shims are free.
2. Then the through bolts for the stator and the stator support rings are loosened so as to allow for thermal growth during the heating.
3. The heaters are then powered up and the desired temperature is set using the controllers.
4. Once the desired temperature is reached we tighten all the bolts at this steady state.
5. Again the shaft is centered to free the shims.
6. Then the current is supplied through the magnetic coils and at every data point the shims are made free and the force data is measured.

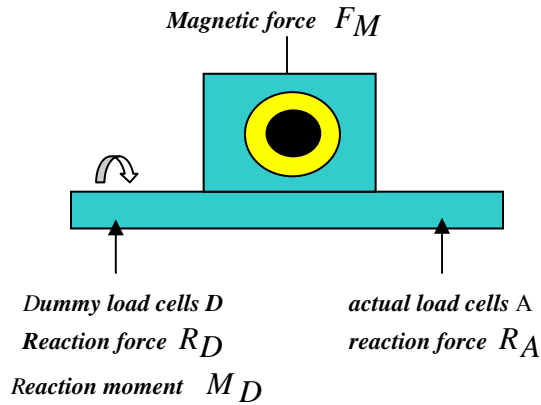


Figure 13 Schematic of 2 load cell setup

Test Results and Correlation

A significant effort was made to obtain force vs. current and temperature for the radial bearing. This resulted in a measurement methodology that yields quite repeatable and reasonable results. Error! Reference source not found.14 shows measured force vs. position with zero current for the radial bearing. This data was taken at room temperature.

- Max. position related force will be approximately 500 lbs at 15 mils.

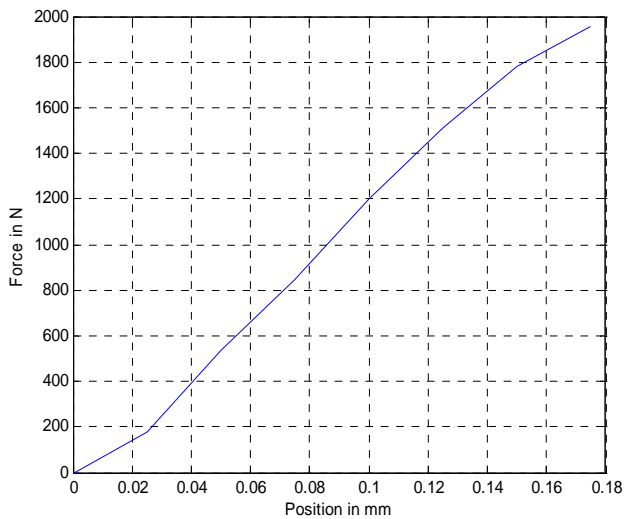


Figure 14 Measured Force vs. Position with Zero Current

The position stiffness is seen to be approximately 66 lbs per mil. The current stiffness is approximately 50 lbs. / amp (227 N/amp) for this room temperature test. For comparison, the predicted current stiffness equals to 40 lbs/amp (182 N/amp) at 1,000 F.

A high temperature test was performed and the results are given below:

- Temperature of Permanent Magnets = 920 F
- Temperature of Shaft = 660 F
- Temperature of Ceramic Layer covering the poles = 690 F
- Force at 13.3 amps = **629 lbs. (2800 N)**

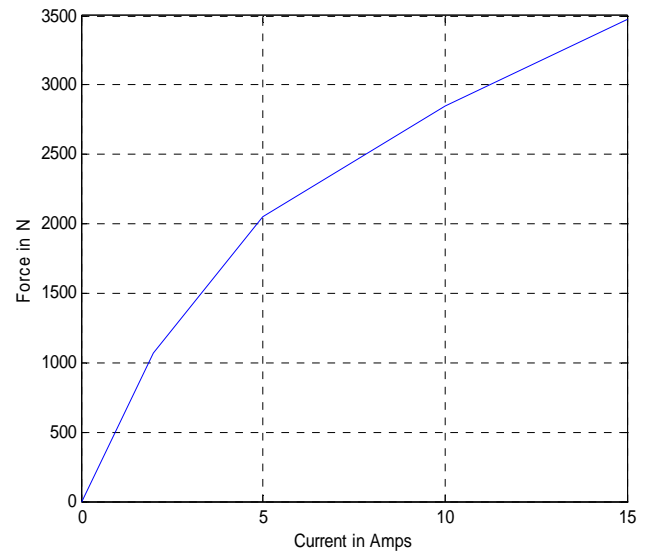


Figure 15 Measured Force vs. Current with Fixed Position

Forces are measured with various currents for the radial bearing at room temperature. All twelve coils on two stators are energized with equal current, but the opposite polarity is given to the groups of top three coils and bottom three ones. Sixteen mil thick shims which are put in the gaps between the top / bottom poles and the shaft are kept free while forces are read from the four load cells. For the remaining four gaps between the poles and the shaft, 13 mil shims are checked to be free. The results are shown in Figure 15.

Figure16 shows a comparison among the measured data, FEA, and the calculation from magnetic circuit modeling.

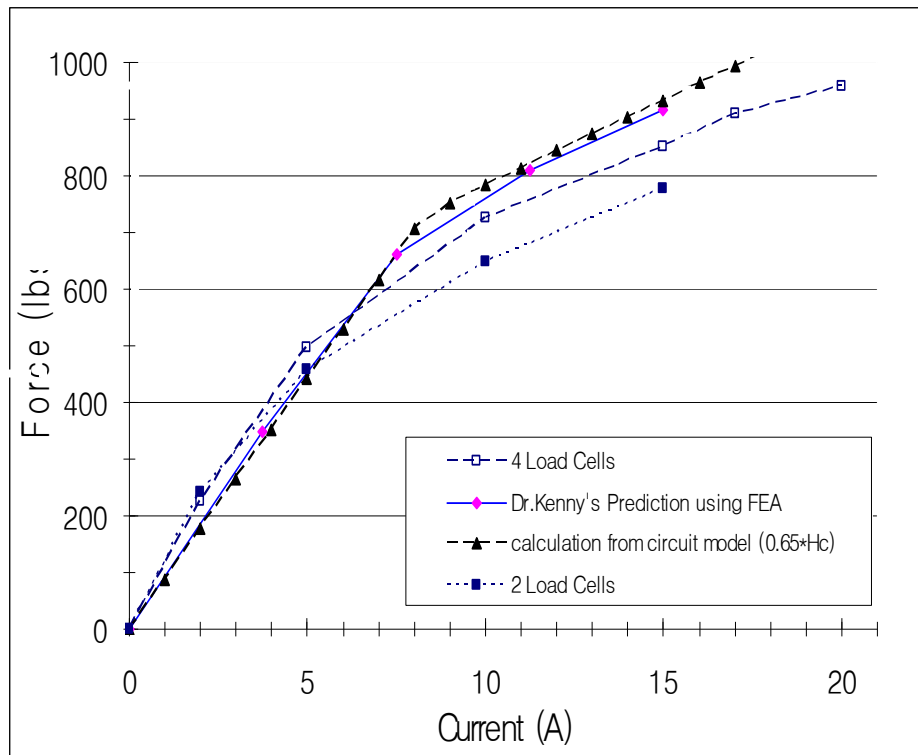


Figure 16: Force vs. Current at room temperature (4.45* lbs equals N)

For 1,000 F, the theoretical, 3D Finite Element, load capacity is 602 lbs (2733 N), position stiffness is 66 lbs/mil (12,000 N/mm) and current stiffness is 40. Lbs/amp (182 N/amp).

FUTURE WORK

The test rig will first be mounted and operated on a floor mounted stand. The motor system will operate at low speed and low temperature test conditions and on an assembly stand. The stand will mount securely to the floor and will allow the slow speed at moderate temperature operation. For high speed operation at high temperatures the assembly will use the stand to transport the system to the spin pit. The subsystems (radial bearings with their shaft components, thrust bearings with its shaft thrust disk and catcher bearings, and motor with its rotor components) will be stacked using the stand as a secure base fixture. Stand components are added and removed as necessary by disconnecting them at their interfaces along the shaft. Access to RTD's, heaters, and

displacement sensors are easily accommodated using the window ports in the outer housing.

Once lowered into the spin pit the support structure occupies most of the available space. This will provide an ability to add soft supports and have a range of support stiffness that can be used during balancing for vibration control.

The high temperature test rig solid model is shown in Figure 17. The components and access can be seen with the insulation and heater assemblies removed. The motor was rotated to better work with the room available in the spin pit.

Sensors are to be mounted on the outside of the support housing. This will allow much easier access to make needed adjustments. The sensors will be of the eddy current type and are made by KAMAN for ultra high temperature applications. Figure 18 shows a calibration curve at high temperature..

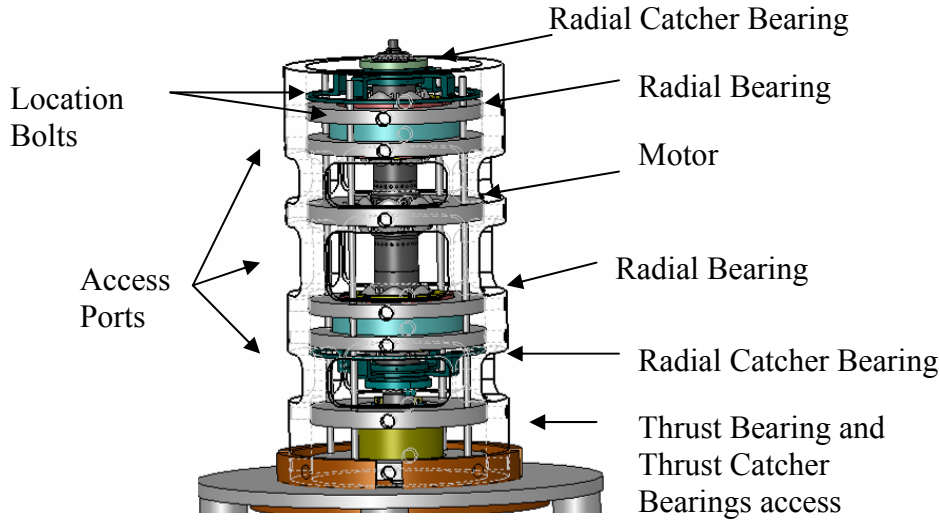


Figure 17 High Temperature Test Rig Components

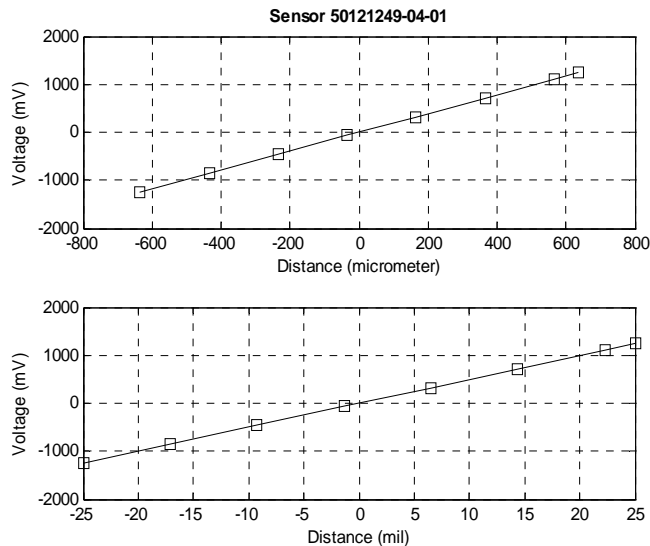


Figure 18 High Temperature Displacement Sensor Calibration

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