# 5-AXES LEVITATION OF A ROTOR BY ACTIVE MAGNETIC BEARINGS EMPLOYING DIRECT OUTPUT FEEDBACK CONTROL

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**Abstract**

*In recent years the application of active magnetic bearings (AMB), as supporting element as well as vibration controller for rotors, has attracted considerable attention from researchers. In this work an attempt has been made for the complete levitation of a rotor weighing 4 kg, supported on two radial active magnetic bearings and two axial passive magnetic bearings (PMB) resulting in 5-axes levitation. The active bearings working in the mode of attraction make the system inherently unstable. Herein, a direct output feedback control scheme based on proportional and derivative of position signal has been employed to achieve stability. The active magnetic bearings consist of electromagnetic actuators, proximity probes for position measurement, feedback controller and current feedback type power amplifier. The passive bearings consisting of axially magnetized circular permanent magnets work in the mode of repulsion. The design of the electromagnetic actuator has been carried out considering load carrying capacity of the AMB, core loss and the resistive loss in the coil, so that the temperature rise in the core as well as in the coil is within allowable limit. Design of the feedback controller is based on direct output feedback control scheme, which makes the controller design easier and implementation of the controller hardware becomes simpler. The feedback controller parameters have been selected appropriately to generate required stiffness and damping in the AMBs for stable levitation of the rotor. The rotor has been driven up to a speed of 2000 rpm.*

## Nomenclature

| | |
|---|---|
| $A_g$ | = cross sectional area of the flux path at the air gap, m$^2$ |
| $Bg$ | = flux density in the air gap, Tesla (Wb/m$^2$) |
| $[C]$ | = damping matrix, N-s/m |
| $\{F(t)\}$ | = control forces, N |
| $f_L$ | = leakage factor |
| $[G1]$ | = proportional gain matrix |
| $[G2]$ | = derivative gain matrix |
| $h_g$ | = air gap, m |
| $I$ | = coil current, ampere |
| $[K]$ | = stiffness matrix, N/m |
| $[M]$ | = mass matrix, kg mass |
| $N$ | = number of turns in the coil |
| $W$ | = load on actuator, N |
| $\{x\}$ | = system coordinate vector |
| $\mu_0$ | = permeability of vacuum, Henry/m |
| $H_\infty$ | = control scheme based on infinity norm |

## Introduction

Though the use of magnetic bearings has been discussed as early as 1842 [1], these bearings have found wide applications with the advent of reliable electronic components in recent years. The magnetic bearings consist of mechanical components as well as electronic components such as sensors, power amplifiers, control circuitry. Thus, it is a mechatronic product and the development of active magnetic bearings requires knowledge in multiple disciplines.

In magnetic bearings there is no physical contact between the stator and the rotor. Owing to the absence of contact these bearings have several advantages over conventional (slider and rolling element) bearings. The principle advantages are numerous, such as, low power loss, no lubrication, no leakage problem, suitability for application in high pressure or vacua, high rotor speed and longer bearing life. Furthermore, the stiffness and damp-

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ing characteristics of the magnetic bearings are easily adjustable according to the requirement. Magnetic bearings can operate in a wide range of temperature from -250°C to 450°C [2].

Application of magnetic bearings in various fields may be found in the literature employing different control strategy [3, 4, 5]. It may be observed from the literature that works in this area have been carried out abroad. However, a few research groups within the country are involved in the development of magnetic bearings. Kamala et. al. [6] have developed magnetic bearing (active in vertical direction) which supports a test rotor weighing 25 kg. In another attempt [7] two radial active magnetic bearings have been developed to support a test rotor weighing 0.5 kg.

In the present work, an attempt has been made to develop a magnetic bearing system for 5-axes levitation, i.e. complete levitation, of a test rotor. The support system consists of two radial active magnetic bearings employing direct output feedback control scheme and passive magnetic bearings in axial direction. Direct output feedback control scheme [8] appears to have advantage over other control scheme in respect to simplicity in designing the controller and ease of implementation. The present work involves the design and fabrication of electromagnetic actuator, feedback controller, power amplifier, mechanical components of the test rotor rig. Finally, all the sub-components are assembled together to achieve complete levitation.

## Control Strategy

The basic requirement for 5-axes levitation of a rotor is to control the rigid body movements in 5 degrees of freedom of the rotor leaving the rotational degrees of freedom. There are several control techniques such as pole placement, optimal, $H_\infty$, Modal, Output Feedback, which could be employed for stable levitation in a magnetic bearing system. In the present work direct output feedback control scheme has been employed, wherein control signals are generated directly from sensor outputs without the use of any observer or state estimator for reconstruction of state variables required for control action in a large system. The main assumptions in this control scheme are the numbers of actuators are same as that of the sensors and the location of the sensor-actuator pair is same, i.e., the sensor and actuator pair is collocated.

The principle of direct output feedback can be explained considering the equations of motion for the system either in state space or in configuration space. Let us assume that equation (1) represents the system in configuration space as follows:

$$[M]\{\ddot{x}\} + [C]\{\dot{x}\} + [K]\{x\} = \{F(t)\} \tag{1}$$

where, $[M]$, $[C]$, and $[K]$ are the system mass, damping and stiffness matrices respectively. The system matrices can be obtained from finite element method or any other suitable method. The vector $\{F(t)\}$ includes the control forces. The external excitation or unbalance forces, if considered in the model, can be included in $\{F(t)\}$. Let us consider only the feedback control forces in $\{F(t)\}$. The control force vector may be constructed according to the following control law.

$$\{F(t)\} = -[G1]\{x\} - [G2]\{\dot{x}\} \tag{2}$$

where, $[G1]$ and $[G2]$ are the feedback gain matrices for the displacement and velocity signals respectively. The feedback gain matrices have to be diagonal since the control action depends only on the response at the same location in direct output feedback control scheme. Moreover, the element in the gain matrices corresponding to non-controlled states would be zero. The closed loop equations of motion can be obtained by substituting equation (2) in to equation (1) as,

$$[M]\{\ddot{x}\} + \left([C] + [G2]\right)\{\dot{x}\} + \left([K] + [G1]\right)\{x\} = 0 \tag{3}$$

External forces are considered to be zero in the above equation. Now the task involved in designing the controller is to find out suitable feedback gain matrices to make the closed loop system stable. The selection of the gain matrices depends on the nature of the system. In the present case the rotor and the magnetic bearing system is inherently unstable without feedback control, as mentioned earlier. Therefore, both displacement and velocity feedback are necessary to make the system stable. Detailed discussion on controller design has been presented in [7].

## Rig Description

A suitable rotor rig has been designed and fabricated to demonstrate 5-axes levitation by magnetic bearings. The photograph of the rotor rig is shown in Fig. 1 The rotor rig consists of several mechanical components such as mild steel shaft, active radial magnetic bearing stands, passive axial magnetic bearing stands, air turbine wheels, auxiliary bearings etc. It may be seen from Fig.1 that two passive thrust bearings are mounted on the rotor shaft at the two ends. Passive thrust bearings consist of two axially magnetized permanent ring magnets with like poles facing each other. Thus, a repulsive force is generated which is capable of restricting the shaft movement in the axial direction. The desired strength of the passive thrust bearing can be obtained by selecting suitable ring magnets. The rotor shaft diameter and length were selected in view of achieving the rotor vibration in relatively a rigid mode shape considering the order of the bearing stiffness obtainable for the AMBs. The first three mode shapes are shown in Fig.2. The modal frequencies are 35.5 Hz, 95.8 Hz, and 167.3 Hz corresponding to 1st, 2nd and 3rd mode, respectively.

There are two active magnetic bearings supporting the rotor in the radial direction. The stator part of the radial bearing is an electromagnetic actuator designed in a way to provide supporting force in both the orthogonal radial directions (horizontal and vertical). Design of the electromagnetic actuator is described in detail in the following Section - Electromagnetic Actuator. A rotor disk is provided at the radial bearing location in order to have more surface area for magnetic actuator, which in turn develops more load carrying capacity of the bearing. Non-contact displacement measuring probes are fixed close to the bearing. Turbine wheels are mounted on the rotor and air jets on the wheels act as driving force to rotate the shaft.

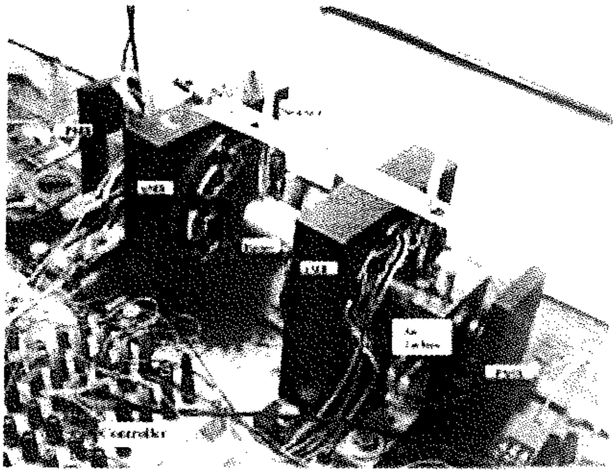

*Fig. 1 Photograph of the rotor-magnetic bearing test rig*

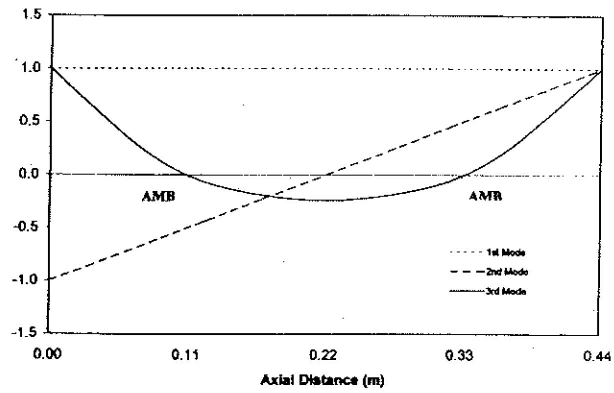


*Fig. 2 First three mode shapes of the rotor*

## Electromagnetic Actuator

AMB design process involves selecting many parameters satisfying the performance and constraints of the actuator. The proper selection of the free parameters leads to a superlative design. The different parameters selected in actuator design may be mentioned as poles/coil configuration, symmetry of poles, magnetic material used, complexity of coil control, coil space factor, amplifier current capacity. The main performance of the actuator is the load carrying capacity other than the actuator weight, power consumption, cost, reliability, safety (high voltage). Lastly, the design constraints are saturation flux density of the material, current capacity of the amplifier, force slew rate of the bearing, space availability.

In the present actuator design importance is given to the losses such as $I^2R$ losses in the coil and hysteresis losses in the core. Temperature rise has been considered as an important parameter for checking the design besides load carrying capacity of the bearing. While designing the actuator coil, factors such as optimum diameter, full utilization of the space available, coil shapes, coil interconnection are considered. The detailed design procedure of the electromagnetic core is given in [9]. However, the design procedure in brief is presented as follows.

In the present work the electromagnetic actuator has been chosen to have eight poles. Each AMB restricts the motion of the rotor in two mutually orthogonal directions at the bearing positions. Two U-shaped electromagnets are employed in each direction totaling to four electromagnets i.e. eight poles are required. The advantage of eight pole radial bearing is that two pole pairs can be oriented face to face in the direction of each of the two orthogonal radial coordinates. Modelling of the mechanical system, control design and measurement of rotor motion are usually based on these coordinates, which yields simplified bearing control. In the present configuration eight equispaced (symmetric) poles with NSSNNSSN pole placing is used. Actuator core is made up of laminated sheet of magnetic material, which is insulated. Laminations are used to reduce the eddy current formation, which reduces the eddy current loss in the actuator. The temperature rise due to eddy current loss in the core material may affect the performance of the actuator besides supplementing the heating of the coil winding.

The load carrying capacity of the bearing is a primary factor in deciding the geometry of the electromagnetic actuator. Thus, evaluation of the load carrying capacity of the electromagnetic actuator for a given geometry of the actuator core is one of the major steps in designing the actuator. The electromagnetic actuator must satisfy the required load carrying capacity. In order to calculate the load carrying capacity of the actuator, first the flux density required to carry the given load '$W$' for a specified air gap is calculated. The calculated flux density should be well below the saturation limit of the core material. It can be calculated as,

$$B_g = f_L \sqrt{\frac{W\mu_o}{A_g}} \tag{4}$$

Leakage factor $f_L$ is taken in to account for the flux leakage and fringing effect of the magnetic field at the intersection of two mediums through which flux travels. It's value varies from 1.1 to 1.3. Flux density in the air-gap calculated ($B_g$) using equation (4) is used to calculate the flux density in the pole cross section and is checked for the flux saturation.

The first step of coil design is to calculate the required amount of ampere-turns ($NI$) which is known as 'magneto motive force' (mmf) to carry the given load. Ampere-turns can be calculated using the following expression,

$$NI = \frac{2B_g h_g}{\mu_o} \tag{5}$$

The number of coil turns and diameter of the conductor wire can be decided based on the available space for coil and satisfying the required amount of ampere-turns. The conductor diameter decides the maximum current capacity of the coil.

### Sensor

In the present experimental rotor rig non-contact type displacement measuring probes (Vibrometer make) were used. These probes are also called proximity or eddy current type probe. It is required to measure the rotor displacement in two orthogonal radial directions at the active radial bearing location for the feedback control action. Therefore, two proximity probes are required for each active radial bearing. There are total four numbers of eddy probes, one each in two orthogonal radial directions in the two radial bearings in the rig. Displacement of the rotor in the two orthogonal transverse directions at the bearing location are continuously sensed by these probes and corresponding output is generated in terms of volts. Sensitivity of the probe is 4mV/micron and operating frequency range is from DC to 2kHz. The outputs of the eddy probes are given as input to the controller.

### Feedback Controller

In the present scheme the radial magnetic bearings are active and work on the principle of electromagnetic attraction. Therefore, the radial magnetic bearings are inherently unstable as mentioned earlier. Appropriate control strategy has to be employed to make the bearings stable. It has been mentioned earlier that a direct output feedback control scheme based on proportional and derivative of the position signal measured at the bearing location has been used for the radial active magnetic bearings. The basic idea in direct output feedback control scheme is that the control signal fed to the actuator is constructed from the measured signal at the same location. Therefore, the respective sensors and the actuators are to be collocated. The design of the output feedback controller for a rotor-bearing system has been carried out following the method discussed in Section - Control Strategy.

The control scheme has been implemented using analog control circuitry. Schematic diagram of the controller circuit is shown in Fig.3. It may be seen from the schematic that the sensor output is fed to the offset adjustment circuitry. Output voltage of this is proportional to the rotor displacement about the bearing center. In the next step the displacement signal is differenciated to obtain voltage signal corresponding to the velocity of the rotor. Displacement and velocity signals are multiplied with proper feedback gain to generate control signal. Feedback gains, as obtained from the controller design based on the rotor dynamic requirements of the test rig, are realized through operational amplifiers. Finally, the control signals are given as input to the power amplifiers, which in turn drives the electromagnetic actuators to produce required control action on the shaft. The schematic of the control loop with sensor, controller, power amplifier, magnetic actuator for a single active radial magnetic bearing is shown in Fig.4.

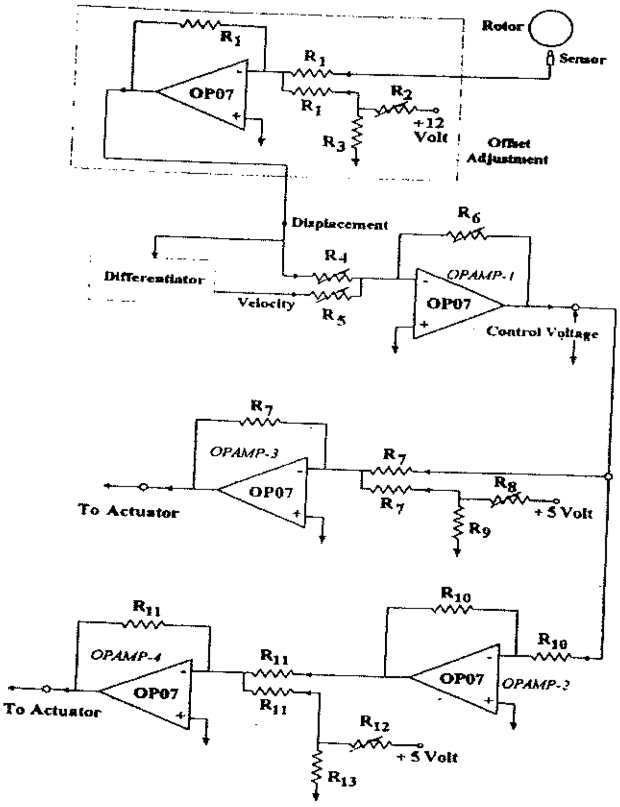


*Fig. 3 Schematic diagram of feedback controller*

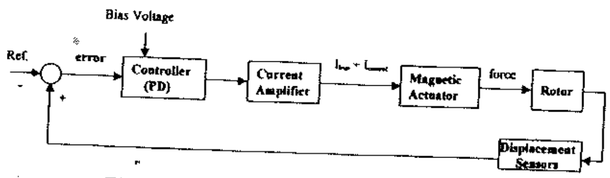


*Fig. 4 Schematic of the control loop for a single radial active magnetic bearing*

## Results

The stable 5-axes levitation of the test rotor has been observed in the experimental rig. The direct stiffness of the active radial bearings in vertical direction is measured and found to be in the range of 60-80 kN/m. This is for one set of feedback gain parameters. Different stiffness and damping coefficients of the bearing can be achieved by changing the feedback gain parameters. The damping ratio, which represents the system damping can be calculated from the transient response measured at AMB location.

Transient responses of the Rotor-AMB rig during switching on the AMBs have been recorded. The transient responses in terms of control voltage in horizontal and

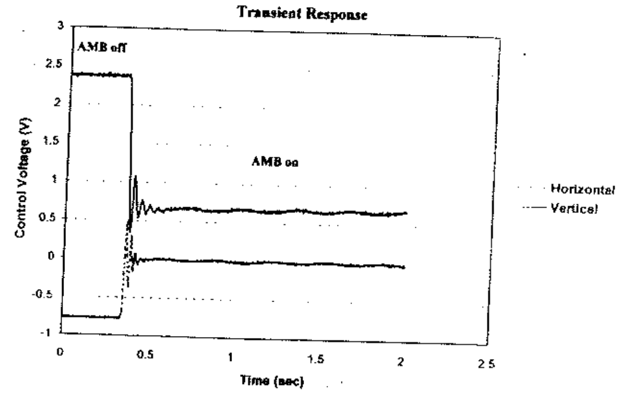


*Fig. 5 Transient response of the rotor-AMB rig after switching on the AMB*

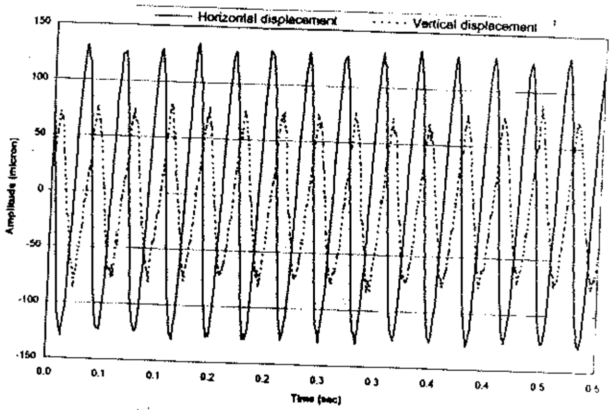


*Fig. 6 Rotor unbalance response at AMB location at 1800 rpm*

vertical directions have been shown in Fig.5. It may be observed from the figure that transient vibration settle down to steady state level within a short span of time. Therefore it may be concluded that sufficient damping has been provided to the system by the feedback controller. The damping ratios in horizontal and vertical direction are 0.083 and 0.122 respectively as calculated from the transient response using logarithmic decrement method. In levitated state in horizontal direction, the rotor is positioned almost at the center of the AMB as it is evident from the control voltage being almost zero. In vertical direction the non-zero control voltage is due to the rotor self weight.

The rotor has been driven up to a speed of 2000 rpm. The rotor response in horizontal and vertical directions at AMB location due to residual unbalance is shown in Fig.6. The response was recorded at the rotor speed of 1300, 1500, 1800 rpm. It may be observed from response plots that horizontal direction amplitude is more than that of

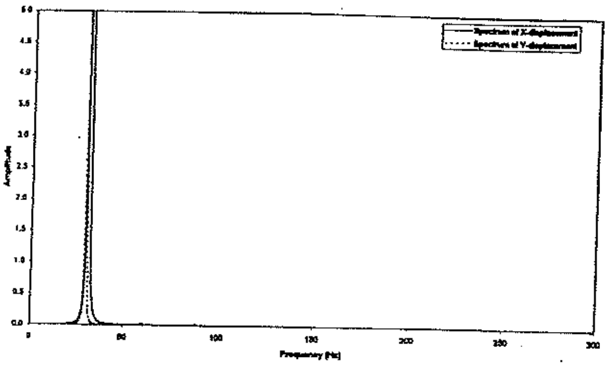

*Fig. 7 Frequency spectrum of rotor displacements at AMB location*

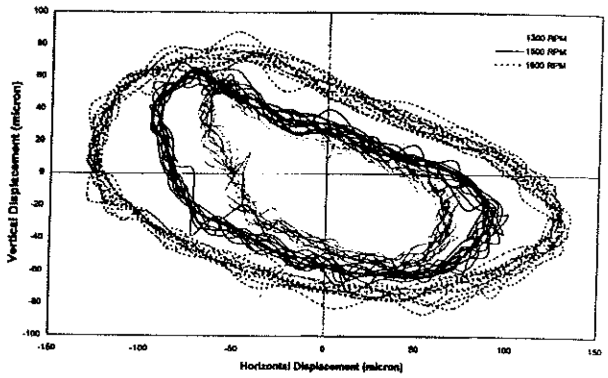


*Fig. 8 Polar plot of rotor displacement at AMB location*

vertical direction. This can be attributed to the unequal stiffness in the two orthogonal directions. The unequal stiffness in orthogonal radial directions may be due to the asymmetric actuator parameters in the orthogonal radial directions. The response is mostly dominated by lx component of vibration, i.e. the rotational frequency component. This is evident from frequency spectrum as shown in Fig.7. The locus of the shaft centre is shown in Fig.8 for three different speeds. It is clearly observed that as the rotor speed goes up the vibration amplitude is also increasing due to larger unbalance forces at higher speed. The elliptic shape of the orbit is due to unequal stiffness value offered by the active bearing in the orthogonal directions. The amplitude of vibration could however be reduced by suitably monitoring the bearing coefficients on line. This happens to be the major advantage of using active magnetic bearings.

## Conclusion

An attempt has been made to develop a magnetic bearing support system for the 5-axes levitation of a test rotor employing direct output feedback control scheme. The test rotor weighing 4 kg has been supported on two active radial magnetic bearings and two passive magnetic thrust bearings, so that the test rotor completely hangs in air in levitated state.

The direct output feedback control scheme based on proportional and derivative of the error signal has been employed for the active radial bearings. The analog controller hardware has been designed and fabricated. The electromagnetic actuator is of hetero-polar type and has eight poles. This has been fabricated from non-grain oriented chromium steel sheets. A current feedback power amplifier has been designed and fabricated to drive the electromagnets with high current.

It has been observed that the active radial magnetic bearings and passive magnetic thrust bearings have successfully supported the test rotor in levitated state without any physical contact. It may be concluded that the proposed control scheme, simple in design and implementation, is sufficient to achieve complete levitation.

## Acknowledgements

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