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VIBRATION ANALYSIS AND CONTROL OF ROTATING COMPOSITE SHAFT USING ACTIVE MAGNETIC BEARINGS

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ABSTRACT

In this article the fibre reinforced polymer (FRP) composites shaft analysed as a Timoshenko beam. The prime objective of this research to reduce the vibration in composites shaft system. To utilize the active vibration control technique. Three different isotropic rigid disks are mounted on it and they also supported by two active magnetic bearing at its ends. The analysis work involves finite element, vibrational and rotor dynamic analysis of the system. Rotary inertia effect, gyroscopic effect kinetic energy and strain energy of the shaft are derived and studied. The governing equation is obtained by applying Hamilton's principle using finite element method in which four degrees of freedom at each node is considered. Active control scheme is applied through magnetic bearings by using a controller containing low pass filter, notch filter, sensor and amplifier which controls the current and correspondingly control the stability of the whole rotor-shaft system. Campbell diagram, stability limit speed diagram and logarithmic decrement diagram are studied to establish the system stability. Effect of different types of stacking sequences are also studied and compared.

KEYWORDS Vibration of Composite Shaft & Vibration Analysis of the Rotating Shaft.

I. INTRODUCTION

Vibration needs to be reduced in most of the rotor-shaft system so that an efficient functioning of the rotating machines is attained. Almost all rotating parts should be vibration free as it causes a lot of problems leading to instability of the system. Therefore there is a necessity to reduce the vibration level in rotating bodies for proper functioning of the system and different researchers are aiming for this. In the present days, composite materials are widely used for the manufacturing of rotor .It is because composites have light weight, high strength, high damping capacity. Weight of the composite materials is less because long stiff fibres are embedded in very soft matrix. Composites are made by at least two materials at macroscopic level. This type of unique reinforcement gives a lot of advantage for different applications. Fiber reinforced polymer (FRP) composite is a polymer matrix in which the reinforcement is fiber. The reinforcement of fiber can be done either by continuous fibre or by discontinuous fiber. The active magnetic bearings present an important progress and a new concept in bearings technology, such thathave been motivating the development of this work. The active magnetic bearings are also efficient as actuators instrategies of active control of vibrations of rotating systems and they present several advantages respect to the conventional bearings for a variety of practical applications. One important advantage is that the system doesn't uselubricating oil, reducing significantly the maintenance operations of the machines. Due to this advantage, the magneticbearings can be applied in sealed pumps, turbo molecular pumps, turbo expansions and centrifuges pumps, where the lubricating oil cannot be used by the fact of reaching high temperatures, or, where the environment requests the minimal of maintenance.

Active materials like piezoelectric material, magneto-strictive material, electromagnetic actuator and micro fibre carbon are also used for the vibration control of rotating parts. Piezoelectric material property to develop charge when mechanically stressed is utilized to bring control of vibration in moving parts. It is used as actuator as well as sensor in the system. Magnetostictive materials are like ferromagnetic material. Materials like cobalt, nickel and iron are magnetostictive materials and therefore change in the shape and size occurs when they are magnetized. Electromagnetic actuator is used very often as it gains the magnetic property when its coils are supplied with current and the displaced position of the rotor can be adjusted according to the current supply.

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The active magnetic bearings present an important progress and a new concept in bearings technology, such that have been motivating the development of this work. The active magnetic bearings are also efficient as actuators in strategies of active control of vibrations of rotating systems and they present several advantages respect to the conventional bearings for a variety of practical applications. One important advantage is that the system doesn't uselubricating oil, reducing significantly the maintenance operations of the machines. Due to this advantage, the magnetic bearings can be applied in sealed pumps, turbo molecular pumps, turbo expansions and centrifuges pumps, where the lubricating oil cannot be used by the fact of reaching high temperatures, or, where the environment requests the minimalof maintenance. Another advantage is that there is no contact between the stationary part of the bearing and rotor ,offering low power loss and large useful life, being especially adapted in applications where the machines operate with higher rotation speeds. Other important benefit of the technology of active magnetic bearings is its capability of operating as a system of active control of vibrations, once the shaft position inside of the bearing can be corrected thousands of times per second .Schweitzer and Lange (1976) recognized the potential of the active vibration control of rotors using active magnetic bearings technology. Since then, significant works in active vibration control area have been published. An overview about magnetic bearings applications was presented by Kasarda (2000). Commercial applicationsexamples of pumps and turbo machines are described. In research fields, researchers have been analyzing applications in bearing less motors, biomedical applications (artificial hearts), tool machines, aircraft jet engines, rotating systems for energy storage and miniature systems.

This work uses the transfer functions of each electronic device that constitute the control circuit of these bearings to determine the global transfer function of the circuit. With the purpose of obtaining the dynamic characteristics of the magnetic bearings, a mathematical model was developed for a rotor magnetic bearing system, which allowed obtaining the expressions to calculate the equivalent stiffness and damping of the bearings though its characteristics and the global transfer function of the control circuit. As the global transfer function of the electronic circuit depends on the frequency, then the equivalent stiffness and damping of the magnetic bearing also will depend. A theoretical model for a rotating system supported by magnetic bearings is presented. The model has been developed by the impedance matrix method. A computational routine has been implemented employing the software and the dynamics of the rotating system can be analyzed in terms of natural frequencies and mode shapes by taking different values for the parameters of the control circuit.

The free oscillatory motion of any rotor bearing system is defined by its amount of stiffness and damping. In using mechanical bearings, the stiffness and damping characteristics of the system are generally fixed by the bearing project. Differently, in using magnetic bearings the stiffness and damping characteristics can be adjusted choosing a set of control parameters for a given control algorithm. Thus, the magnetic bearings have the capability to change the dynamic of the rotors to operate in better dynamic stabilities. Reduction in rotor vibrations can be obtained by application of an ‘‘open-loop’’ or feed forward control strategy superimposed on the ‘‘closed-loop’’ control strategy necessary to keep the rotor suspended. In this work the filtered XLMS algorithm will be presented. It is a time domain based adaptive feed forward approach widely used in the active control of sound and vibration. The X-LMS is a least mean squares approach where a reference signal, typically +denoted by “x”, is filtered before the LMS operation is performed.

II. RESEARCH AIMS

1. To prepare a mathematical model of active bearing for composite shaft.
2. To reduce the vibration of a fiber reinforced polymer (FRP) rotor shaft system by using PID control technique in active magnetic bearing.
3. To decrease stability limit speed has by applying control force through proportional derivative and integral control.

Mathematical Modelling of FRP Composite Shaft

The shaft is modelled as a Timoshenko beam considering both rotary inertia and gyroscopic effect. The shaft is rotating at a constant angular speed along the longitudinal axis and has uniform circular cross section along the length. The displacement field is described by assuming the coordinate axis coincides with the shaft axis. The strain energy equation of the composite shaft can be derived by assuming the displacement fields as follows,

$$\begin{aligned} u_x(x,y,z,t) &= u(x,t) - y\beta_y(x,t) \\ u_y(x,y,z,t) &= v(x,t) - z\emptyset(x,t) \qquad (1) \\ u_z(x,y,z,t) &= w(x,t) + y\emptyset(x,t) \end{aligned}$$

The stress-strain relationship considering cylindrical coordinate system is

$$\begin{bmatrix}\varepsilon_{xx}\\ y_{x\theta}\\ y_{xr}\end{bmatrix}=\begin{bmatrix}0 & 0 & r\sin\theta\frac{\partial}{\partial x} & -r\cos\theta\frac{\partial}{\partial x}\\ -\sin\theta\frac{\partial}{\partial x} & \cos\theta\frac{\partial}{\partial x} & \cos\theta & \sin\theta\\ \cos\theta\frac{\partial}{\partial x} & \sin\theta\frac{\partial}{\partial x} & \sin\theta & -\cos\theta\end{bmatrix}\begin{bmatrix}v_0\\ w_0\\ \beta_x\\ \beta_y\end{bmatrix}\left(\frac{\pi}{2}-\theta\right) \qquad (2)$$

The stress-strain relations for the r^{th} layer can be expressed in the cylindrical coordinate system as

$$\begin{bmatrix}\sigma_{xx}\\ \tau_{x\theta}\\ \tau_{xr}\end{bmatrix}=\begin{bmatrix}\bar{C}_{11r} & k_s\bar{C}_{16r} & 0\\ k_x\bar{C}_{16r} & k_s\bar{C}_{66r} & 0\\ 0 & 0 & k_s\bar{C}_{55r}\end{bmatrix}\begin{bmatrix}\varepsilon_{xx}\\ y_{x\theta}\\ y_{xr}\end{bmatrix} \qquad (3)$$

Strain Energy Expression for Composite Shaft
The strain energy equation of FRP composite shaft can be obtained as

$$U_s=\frac{1}{2}\int_v [\sigma][\varepsilon]dV=\frac{1}{2}\int_V (\sigma_{xx}\varepsilon_{xx}+2\tau_{xr}\varepsilon_{xr}+2\tau_{x0}\varepsilon_{x\theta})dV$$

After simplification and taking first variation of above strain energy expression can be written as,

$$\delta U_s=\left[\begin{array}{l} B_{11}\left[\begin{array}{l}\int_0^L\left(\frac{\partial\beta_x}{\partial x}\frac{\partial\delta\beta_x}{\partial x}\right)dx\\ +\int_0^L\left(\frac{\partial\beta_y}{\partial x}\frac{\partial\delta\beta_y}{\partial x}\right)dx\end{array}\right]+\frac{1}{2}k_sA_{16}\left[\begin{array}{l}\int_0^L\left(\beta_y\frac{\partial\delta\beta_x}{\partial x}+\frac{\partial\beta_x}{\partial x}\delta\beta_y\right)dx-\int_0^L\left(\beta_x\frac{\partial\delta\beta_y}{\partial x}+\frac{\partial\beta_y}{\partial x}\delta\beta_x\right)dx\\ -\int_0^L\left(\frac{\partial v_0}{\partial x}\frac{\partial\delta\beta_x}{\partial x}+\frac{\partial\beta_x}{\partial x}\frac{\partial\delta v_0}{\partial x}\right)dx-\int_0^L\left(\frac{\partial w_0}{\partial x}\frac{\partial\delta\beta_y}{\partial x}+\frac{\partial\beta_y}{\partial x}\frac{\partial\delta w_0}{\partial x}\right)dx\end{array}\right]\\ +k_s(A_{55}+A_{66})\left[\begin{array}{l}\int_0^L\beta_x\delta\beta_x dx+\int_0^L\beta_y\delta\beta_y dx+\int_0^L\left(\frac{\partial v_0}{\partial x}\frac{\partial\delta v_0}{\partial x}\right)dx\\ +\int_0^L\left(\frac{\partial w_0}{\partial x}\frac{\partial\delta w_0}{\partial x}\right)dx-\int_0^L\left(\beta_y\frac{\partial\delta v_0}{\partial x}+\frac{\partial v_0}{\partial x}\delta\beta_y\right)dx+\int_0^L\left(\beta_x\frac{\partial\delta w_0}{\partial x}+\frac{\partial w_0}{\partial x}\delta\beta_x\right)dx\end{array}\right]\end{array}\right]$$

Kinetic Energy Expression for Composite Shaft
Kinetic energy expression and the first variation of the kinetic energy of FRP composite shaft can be obtained as follows,

$$\int_0^L\left[I_m(\dot{v}_0^2+\dot{w}_0{}^2)+l_d(\dot{\beta}_x^2+\dot{\beta}_y^2)+\Omega^2 l_p-2\Omega l_p\beta_x\dot{\beta}_y^2+\Omega^2 l_d(\beta_x^2+\beta_y^2)\right]dx$$

$$T_x=\int_0^L\left[l_m\left(\dot{v}_0\frac{\partial\delta v_o}{\partial t}+\dot{w}_0\frac{\partial\delta w_0}{\partial t}\right)+l_d\left(\beta_x\frac{\partial\delta\beta_x}{\partial t}\right)-\Omega l_p(\dot{\beta}_y\delta\beta_x+\beta_x\frac{\partial\delta\beta_y}{\partial t})\right]dx$$

Where, the gyroscopic effect is denoted by $2\Omega l_p\beta_x\dot{\beta}_y$ and rotary inertia effect is represented by $l_d(\dot{\beta}_x^2+\dot{\beta}_y^2)$. As the term $\Omega^2 l_d(\beta_x^2+\beta_y^2)$ is very small compared to $\Omega^2 l_p$, it has been neglected in the further analysis. The terms l_m, l_d and l_p are elaborated in the Appendix.

Kinetic Energy Expression for Disks
In a similar way the kinetic energy expression and the first variation of kinetic energy of the disk are obtained as follows:

$$T_d=\frac{1}{2}\int_0^L\sum_{i=1}^{N_D}\left[I_{mi}^D(\dot{v}_0^2+\dot{w}_0{}^2)+I_{di}^D(\dot{\beta}_x^2+\dot{\beta}_y^2)-2\Omega I_{pi}^D\,E\beta_x\dot{\beta}_y^2+\Omega^2 I_{pi}^D\right]\triangle(x-x_{Di})dx$$

$$\delta T_d=\int_0^L\sum_{i=0}^{N_D}\left[I_{mi}^D\left(\dot{v}_0\frac{\partial\delta v_0}{\partial t}+\dot{w}_0\frac{\partial\delta w_0}{\partial t}\right)+I_{dt}^D\left(\dot{\beta}_x\frac{\partial\delta\beta_x}{\partial t}+\dot{\beta}_y\frac{\partial\delta\beta_y}{\partial t}\right)-\Omega l_{pi}^D(\dot{\beta}_y\delta\beta_x\frac{\partial\delta\beta_y}{\partial t})\right]\triangle(x-x_{Di})dx$$

Where, disks position denotes by i (=1, 2, 3….) and the symbol $\triangle(x-x_{Di})$ denotes the one dimensional spatial Dirac delta function.

Work Done Expression Due to External Loads and Bearings
Here R_y and R_z are assumed as the external force intensities (force per unit length) subjected to the shaft and M_x, M_y are the externally applied torque intensities (moment per unit length) distributed along the shaft. Now virtual work done by the external loads can be represented as follows:

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$\delta W_E = \int_0^L (R_y \delta v_0 + R_z\, \delta w_0 + M_x \delta \beta_x) dx$

In the present analysis, bearings are modeled by springs and viscous dampers. Virtual work done by springs and dampers can be obtained as,

$$\delta W_B = \int_0^L \sum_{i=1}^{N_B} \begin{pmatrix} -K_{yy}^{Bi} v_0 \delta v_0 - K_{zy}^{Bi} v_0 \delta w_0 - K_{yz}^{Bi} w_0 \delta v_0 - K_{zz}^{Bi} w_0 \delta w_0 \\ -C_{yy}^{Bi} \dot{v}_0 \delta v_0 - C_{zy}^{Bi} \dot{v}_0 \delta w_0 - C_{yz}^{Bi} \dot{w}_0 \delta v_0 - C_{zz}^{Bi} \dot{w}_0 \delta w_0 \end{pmatrix} \triangle (x - x_{Bi}) dx$$

The governing equations of the spinning shaft system can be obtained using equations. (9), (11), (13), (14), (15) and applying Hamilton's principle which is,

$\int_{t1}^{t2} [\delta(T_x + T_d) - \delta U_x + \delta W_E + \delta W_E] dt = 0$ Type equation here.

After simplifying and arranging the above equation, the equations of motions can be obtained as,

$$\delta v_0 :\ l_m \frac{\partial^2 v_0}{\partial t^2} + k_x (A_{55} + A_{66}) \left(\frac{\partial \beta_x}{\partial x} - \frac{\partial^2 V_0}{\partial X^2}\right) + \frac{1}{2} k_s A_{16} \frac{\partial^2 \beta_x}{\partial x^2} + \sum_{I=1}^{N_D} I_{mi}^D \frac{\partial^2 v_0}{\partial t^2} \Delta(x - x_{Di}) + P_{v_0}^b = R_y$$

$$\delta w_0 :\ I_m \frac{\partial^2 w_0}{\partial t^2} - k_s (A_{55} + A_{66}) \left(\frac{\partial^2 w_0}{\partial t^2} + \frac{\partial \beta_x}{\partial x}\right) + \frac{1}{2} k_s A_{16} \frac{\partial^2 \beta_x}{\partial x^2} + \sum_{i=m}^{N_D} I_{mi}^D \frac{\partial^2 w_0}{\partial t^2} \triangle (x - x_{Di}) + P_{w_0}^b = R_z$$

$$\delta \beta_y :\ I_d \frac{\partial^2 \beta_y}{\partial^2} - I_P \Omega \frac{\partial \beta_x}{\partial t} + \frac{\partial \beta_x}{\partial t} \frac{1}{2} k_s A_{16} \frac{\partial^2 W_0}{\partial x^2} - B_{11} \frac{\partial^2 \beta_y}{\partial x^2} - k_s (A_{55} + A_{66}) \left(\beta_x - \frac{\partial v_0}{\partial x}\right) + \sum_{i=1}^{N_D} \left(I_{dt}^D \frac{\partial^2 \beta_y}{\partial t^2} - \Omega I_{pi}^D \frac{\partial \beta_x}{\partial t}\right) \triangle (x - x_{Di}) = M_y$$

$$\delta \beta_x :\ I_d \frac{\partial^2 \beta_x}{\partial t^2} - I_p\, \Omega \frac{\partial \beta_x}{\partial t} + \frac{1}{2} k_s A_{16} \frac{\partial^2 v_0}{\partial x^2} - B_{11} \frac{\partial^2 \beta_x}{\partial x^2} - B_{11} \frac{\partial^2 \beta_x}{\partial x^2} - k_s (A_{55} + A_{66}) \left(\frac{\partial w_0}{\partial x} + \beta_x\right) + \sum_{i=1}^{N_D} \left(I_{di}^D \frac{\partial^2 \beta_x}{\partial t^2} - \Omega I_{pi}^D \frac{\partial^2 \beta_x}{\partial t^2} - \Omega I_{pi}^D \frac{\partial \beta_x}{\partial t}\right) \Delta(x - x_{Di}) = M_x$$

FE Modelling of FRP Composite Shaft

Here, finite element analysis is aiming to find out the field variable (displacement) at nodal points by approximate analysis. In the present FE model, the three-nodded one dimensional line elements are consider, each node having four degree of freedom (DOF). The Lagrange interpolation functions are used to approximate the displacement fields of shaft. The element's nodal DOF at each node is v_{0}, w_0, β_x, and β_y. now displacement field variable can be represented as,

$v_0 = \sum_{k=l}^{n} v_0^2\ (t)\psi_k\ (\eta), w_0 = \sum_{k=1}^{n} w_0^k (t)\psi_k\ (\eta), \beta_x = \sum_{k=1}^{n} \beta_x^k\ (t)\psi_k(\eta), \beta_x = \sum_{k=1}^{n} \beta_y^k\ (t)\psi_k(\eta)$

Now one dimensional Lagrange polynomial is defined as,

$$L_k(\eta) = \prod_{\substack{m=k \\ m \neq k}}^{n} \frac{\eta - \eta_m}{\eta_k - \eta_m}$$

For three nodded element, the shape functions or interpolating functions can be expressed as,

$$\psi_1 = \frac{-\eta(1-\eta)}{2}, \qquad \psi_2 = 1 - \eta^2, \qquad \psi_3 = \frac{\eta(1+\eta)}{2}$$

Now putting the displacement field variables and shape function expressions into governing equations, the equation of motion for a element can be written as,

$[M]\{\ddot{q}\} + ([C] + \Omega[G])\{\dot{q}\} + [K]\{q\} = \{F\}$

The rotor dynamics equation of motion including both internal viscous and hysteretic can be extended as

$$[M]\{\ddot{q}\} + ([C] + \Omega[G] + \eta_V [K])\{\dot{q}\} + \left[\left(\frac{1+\eta_H}{\sqrt{1+\eta_H^2}}\right)[K] + \left(\eta_V \Omega + \frac{\eta_H}{\sqrt{1+\eta_H^2}}\right)[K_{Cir}]\right]\{q\} = \{F\}$$

The equation of motion of the FRP composite shaft can be obtained after assembly of all the elemental matrices. Where, $[M], [C], [G], [K], \{q\}$ all elemental matrices are given in Appendix

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III. MATHEMATICAL MODELLING OF ACTIVE MAGNETIC BEARING

A magnetic bearing is used to carry a load by magnetic levitation technique. The main advantage of the bearing is that it runs without any surface contact with the stator, hence the operation is friction less. Magnetic bearings can run at higher speeds without any problem of mechanical wear.

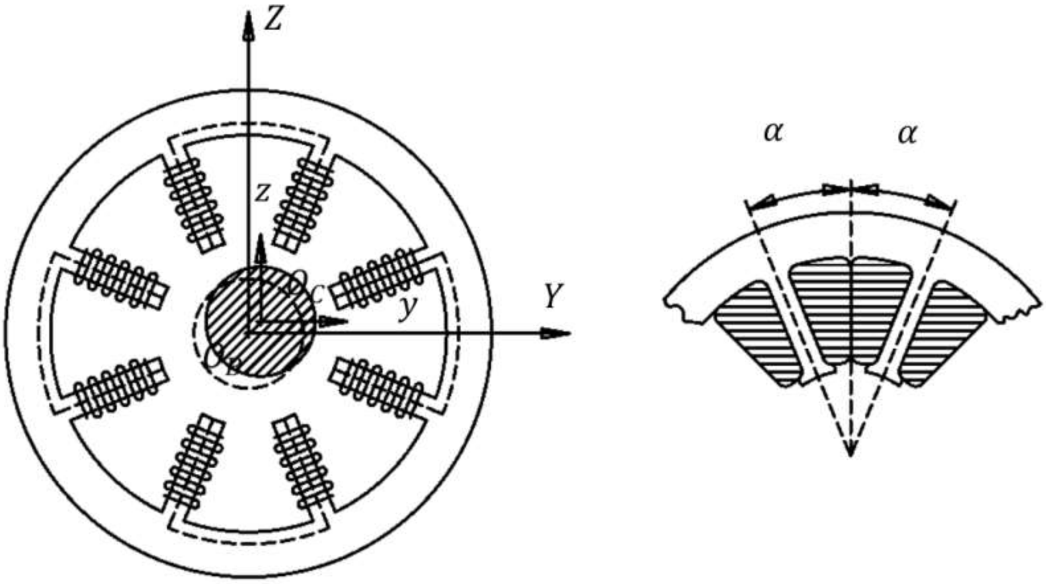

Fig.1. Geometrical representation of the stator, electromagnets and the included angle for ABM

The principle of working of the magnetic bearing is to provide suspension through electromagnetic current. For this purpose a complete electromagnet assembly is used. Two power amplifiers, one controller and two gap sensors are used in the assembly. The power amplifier sends control current to the electromagnet while the set of gap sensors and controller provides the control feedback according to position of the moving rotor within the gap Equal bias currents are sent from the power amplifier into the electromagnet from two opposite directions of the rotor. Function of controller is to add the bias current with positive and negative values of control currents as the rotor changes its centre position while running as shown in the Fig

The Electromagnetic Force

The detail description of the magnetic force created inside the electromagnet is given in detail below.

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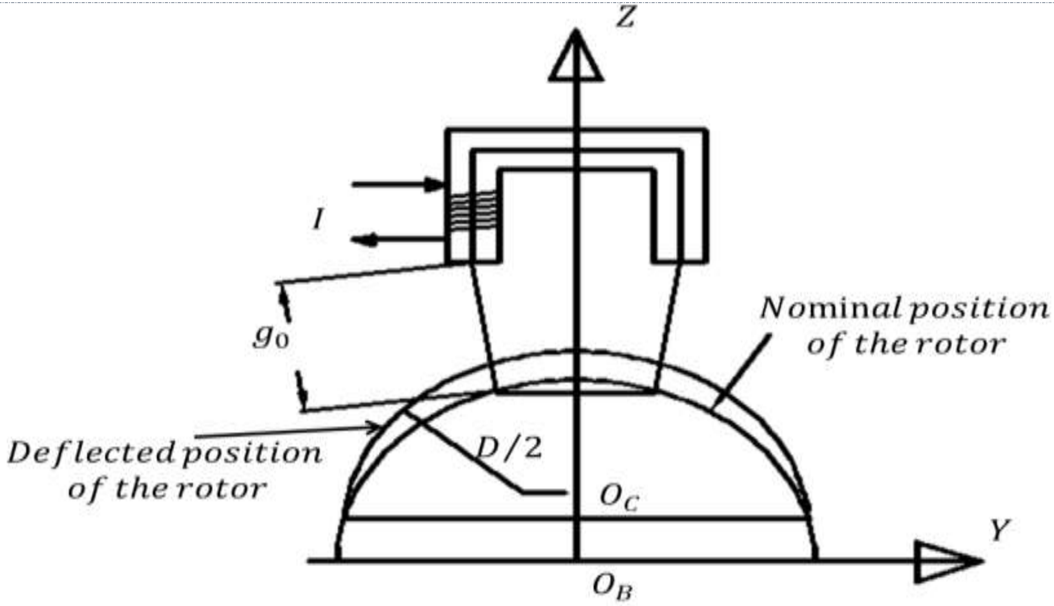

Fig.2. Diagram showing magnetic circuit formed between shaft and electromagnet

The above Fig. 2 is clearly showing how a magnetic circuit comes into existence in between the rotor and electromagnet when current is passed through the electromagnet. OB and OC are the centre of the shaft at its nominal (shaft is stationary) and the deflected positions. Some assumptions are taken into consideration like (1) the gap between rotor and stator is very small as compared to the radius of the rotor (2) Fringing effect as well as flux leakage is negligible at the pole face.(3) the length of lines of magnetic flux are equal (4) Flux density and the intensity of the magnetic field of the material follow a linear relationship (5) Magnetic permeability is constant within the operating range (6) lastly, magnetic hysterics of the material is also assume to be negligible.

$$F_{mag} = -\frac{\varepsilon\mu_0 A_g N^2 {i_0}^2}{4g_0^2}$$

Where μ_0, A_g, N, i_0 and g_0 are the absolute permeability of free air, face area of pole, and number of turns of coil, current and initial gap between the rotor and stator respectively. The force increases as the gap decreases due to negative sign in the equation. As the rotor starts rotating with its centre $O_{B,}$ it has uniform air gap g_0 and bias current of magnitude i_0 flows steadily through the coils of the electromagnet initially but as the time passes the eccentricity developed and the centre of the rotor changes. Unbalance magnetic forces start developing inside the rotor-shaft system due to the change in the position of the rotor, when it comes closer to one pair of pole; it goes farther from the opposite one. Because magnetic force is inversely proportional to the square of the gap exist between the rotor and shaft, the control force is necessary to keep the constant gap. The control force inside the system develops according to the gap between rotating and stationary parts. Deflection of the rotor either in Y or Z direction can be controlled by supplying appropriate control current. If the shaft comes closer to one pole of electromagnet, the current is reduced in order to decrease the magnetic force value. Simultaneously the shaft goes farther from the opposite pole of the same electromagnet; the magnetic force is increased by increasing the control current. The forces in either Y or Z directions are completely uncoupled.

IV. CONCLUSION

It may be concluded that the active magnetic bearing used in the rotor-shaft system where rotor is made up of fibre reinforced polymer (FRP) composite, brings the system into a stable position. The active technique is achieved by bringing a controller into the picture which senses the displacement and controls the corresponding current in order to achieve stability. The different parameters of PID filter are tuned manually to get the optimum results. The Campbell diagrams prove that the critical speed of the controlled system has been increased to a higher value and the stability limit speed (SLS) diagrams show that the system can run at much higher speed when operated with active control method. The active magnetic bearing also provides a contact-free operation which reduces rotor vibration. The control action is free from the problem of maintenance, wear and tear and power loss due to friction..

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