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Vibration Reduction in Rotating Systems Using Squeeze Film Dampers and Magnetic Bearings

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Abstract: This study investigates the vibration reduction performance of bearings integrated with squeeze film dampers (SFDs) and active magnetic bearings (AMBs) in a simplified rotor-bearing system. The integration of SFDs and AMBs is proposed to leverage their complementary characteristics and enhance vibration suppression. A theoretical model is developed to analyze the dynamic behaviour of the system, and the effects of various parameters on vibration reduction are examined. The results demonstrate that the combined use of SFDs and AMBs can significantly reduce vibration amplitudes and improve system stability. The findings of this study provide valuable insights into the design and optimization of rotor-bearing systems with enhanced vibration reduction capabilities.
***Keywords:** Squeeze film dampers, Active magnetic bearings, Vibration amplitudes, Rotor-Bearing system.*

I. INTRODUCTION

Active Magnetic Bearings (AMBs) are a type of bearing that uses magnetic forces to levitate a rotor, eliminating the need for physical contact. This technology offers several advantages over traditional bearings, including high speed, tunability, and low power consumption. The working principle of AMBs is based on a closed-loop control system. Sensors detect changes in the rotor's position and send signals to the controller. The controller then sends a signal to the power amplifier, which supplies current to the actuator coils. These coils create a magnetic field that attracts the rotor back to its equilibrium position, maintaining stability and control.

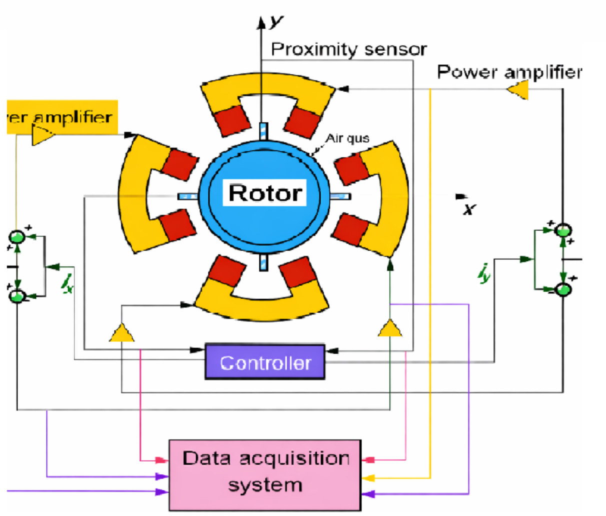

Fig 1: Working principle of AMB–rotor system

The way AMBs work is based on a closed-loop control system. Sensors pick up on changes in the rotor's position and send signals to the controller. The controller then sends a signal to the power amplifier, which gives the actuator coils power. These coils create a magnetic field that attracts the rotor back to its equilibrium position, maintaining stability and control. The control system is critical to the stable operation of AMBs. Feedback control laws are used to maintain rotor position and stability. PID (Proportional-Integral-Derivative) control is a common method used in AMBs.

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HARDWAREFOR AMB:

The simplest AMB stator can be made using two electric coils while a minimum of three coils are necessary in order to generate forces in two perpendicular directions. The most common configuration used is AMB with four coils. There are two configurations in Radial AMBs based on their construction called homopolar and heteropolar. In homopolar AMB shown in Figure 2 stator poles have the same magnetic polarities while in heteropolar AMB shown in Figure 3 poles the varying polarities.

In AMB applications, each bearing axis has a pair of power amplifies. They supply the actuator coils with the current required to generate the forces that act on the rotor. Thus, they function between controller and the coils of electromagnetic actuator.

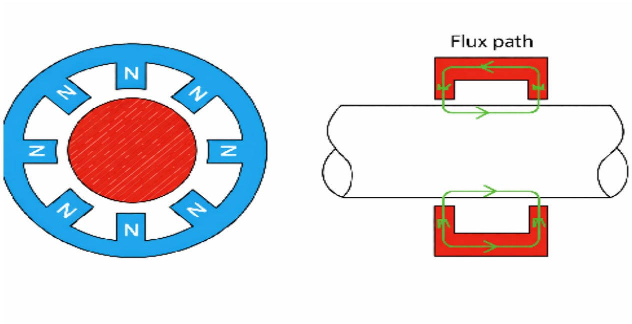

Fig 2: Homopolar radial magnetic bearing (HOMB)

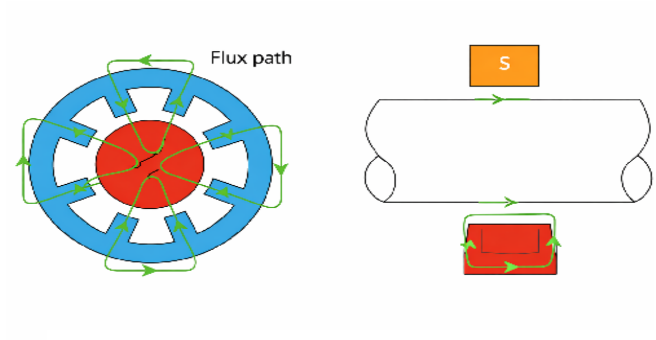

Fig 3: Heteropolar radial magnetic bearing (HEMB)

Eddy current proximity probes, capacitive sensors, inductive pick-ups, magnetic pick-ups, optical sensors, photo-electric sensors, laser type sensors and Hall sensors are the most used sensors in AMB applications. Eddy current sensors have a displacement range of 2 mm and possess good frequency response and are most used. Touch down bearings also called auxiliary, or backup or retainer bearings are mounted at one half of AMB air gap to protect the AMB in the event of power failure or load saturation. During normal operation they are not in contact with the rotor.

II. LITERATURE REVIEW

This literature review presents an overview of the research on Active Magnetic Bearings (AMBs) and their applications in rotating machinery. AMBs have been extensively studied for their vibration reduction capabilities and fault detection and diagnosis in various applications. Active Magnetic Bearings (AMBs) are a type of bearing that uses magnetic forces to levitate a rotor, eliminating the need for physical contact. AMBs have been widely researched for their vibration reduction capabilities and fault detection and diagnosis in various applications. Several studies have investigated the use of AMBs for vibration control in rotating machinery. Allaire et al. [1] used proportional and derivative control to reduce vibration in a flexible rotor. Burrows et al. [2], [3] controlled vibration in a flexible rotor using AMBs as exciters. Nonami and Ito [4] used µ-synthesis control for a flexible rotor-AMB system. Various control methods have been used for vibration control in AMBs, including PID control [5], [6], Linear Quadratic (LQ) control [5], and Model Predictive Control (MPC) [7]. AMBs have been used in various applications, including high-speed machinery, centrifugal pumps, compressors, and helicopters [8]. AMBs have also been used for fault detection and diagnosis in rotating machinery, including crack detection [9], [10] and bearing fault detection [11]. This literature review highlights the potential of AMBs in vibration control and fault detection in rotating machinery. Further research is needed to address the gaps in literature and explore new applications of AMBs.

III. OBJECTIVES OF THE WORK

This study presents a comprehensive dynamic analysis and performance comparison of two rotor-bearing stabilization systems: Squeeze Film Dampers (SFDs) and Active Magnetic Bearings (AMBs). Detailed mathematical models are formulated for each configuration, designated as Rotor-1 (SFD-supported) and Rotor-2 (AMB-supported), to establish a theoretical basis for their dynamic behaviour. The developed models are then implemented in Simulink, enabling us to generate and analyze time-domain responses that provide a deeper understanding of the system's dynamic behavior.

IV. DEVELOPING MATHEMATICAL MODELS FOR DYNAMIC SYSTEMS

The study examines two distinct rotor systems. The first system, shown in Figure 4, consists of two discs mounted on rolling element bearings at each end of a simple rotor. At both bearing locations, squeeze film dampers provide viscous damping. Figure 5 shows a second system with the same rotor configuration but with Active Magnetic Bearings (AMBs) on both ends.

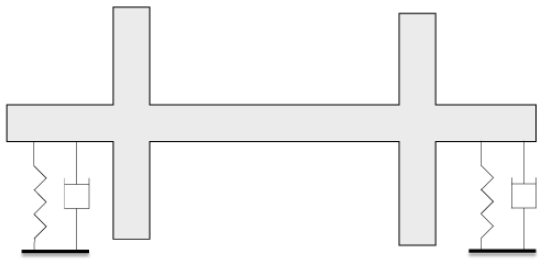

Fig 4: Rotor-Bearing-SFD system

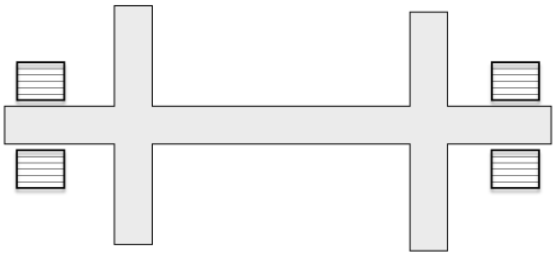

Fig 5: Rotor-AMB system

A. Shaft and disc model:

The matrices of shaft, disc, bearing, squeeze film damper, and active magnetic bearing are presented in this section. In order to obtain the global equations of motion (EOM) for the Rotor Bearing-SFD system and Rotor-AMB system, the matrices for the subsystems are compiled together.

$u^e = u_x + j\,u\_z = \{ x_1\ \varphi_{x1}\ x_2\ \varphi_{x2} \}^T + j\,\{ y_1\ \varphi\_z_1\ y_2\ \varphi\_z_2 \}^T$ ----------------(i)

(a) Rotational mass matrix for shaft

$$\mathbf{M}_R = \frac{\rho I_e}{30 l_e (1+\Phi)^2} \begin{bmatrix} n_1 & n_2 & -n_1 & n_2 \\ n_2 & n_3 & -n_2 & n_4 \\ -n_1 & -n_2 & n_1 & -n_2 \\ n_2 & n_4 & -n_2 & n_3 \end{bmatrix} \quad \text{———} \quad (2)$$

(b). Translational mass matrixfor shaft

$$\mathbf{M}_T = \frac{\rho A_i l_e}{420 (1+\Phi)^2} \begin{bmatrix} m_1 & m_2 & m_3 & m_4 \\ m_2 & m_5 & -m_4 & m_6 \\ m_3 & -m_4 & m_1 & -m_2 \\ m_4 & m_6 & -m_2 & m_5 \end{bmatrix} \quad \text{———} \quad (3)$$

(c) Gyroscopic matrix for shaft

$$\mathbf{G}_s = \frac{\rho I_e}{15 l_e (1+\Phi)^2} \begin{bmatrix} n_1 & n_2 & -n_1 & n_2 \\ n_2 & n_3 & -n_2 & n_4 \\ -n_1 & -n_2 & n_1 & -n_2 \\ n_2 & n_4 & -n_2 & n_3 \end{bmatrix} \quad \text{———} \quad (4)$$

(d). Stiffness matrix for shaft

$$\mathbf{K}_s = \frac{E I_e}{l_e^3 (1+\Phi)} \begin{bmatrix} k_1 & k_2 & -k_1 & k_2 \\ k_2 & k_3 & -k_2 & k_4 \\ -k_1 & -k_2 & k_1 & -k_2 \\ k_2 & k_4 & -k_2 & k_3 \end{bmatrix} \quad \text{———} \quad (5)$$

(e). Gyroscopic matrix for shaft(f) Mass matrix for disc

$$\mathbf{G}_s = \frac{\rho I_e}{15 l_e (1+\Phi)^2} \begin{bmatrix} n_1 & n_2 & -n_1 & n_2 \\ n_2 & n_3 & -n_2 & n_4 \\ -n_1 & -n_2 & n_1 & -n_2 \\ n_2 & n_4 & -n_2 & n_3 \end{bmatrix} \text{———} (6)$$

$$\mathbf{M}_d = \begin{bmatrix} m_{d_1} & 0 \\ 0 & I_{d_1} \end{bmatrix} \quad \text{———} \quad (7)$$

Where md and Id are the mass and diametrical mass moment of inertia of disc.

(g) Gyroscopic matrix for disc

$$\mathbf{G}_d = \begin{bmatrix} 0 & 0 \\ 0 & -I_{P_1} \end{bmatrix} \quad \text{———} \quad (8)$$

where Ip is the polar mass moment of inertia of disc.

B. BEARINGS

(a) Force due to bearing stiffness

$$\mathbf{f}_{b_{k_i}} = \begin{Bmatrix} 0.5\left(k_{rad}(u_{b_i} + \bar{u}_{b_i}) + k_{rad}(u_{b_i} - \bar{u}_{b_i})\right) \\ 0 \end{Bmatrix} \quad \text{——— (9)}$$

(b) Force due to bearing damping

$$\mathbf{f}_{b_{c_i}} = \begin{Bmatrix} 0.5\left(c_{rad}(\dot{u}_{b_i} + \dot{\bar{u}}_{b_i}) + c_{rad}(\dot{u}_{b_i} - \dot{\bar{u}}_{b_i})\right) \\ 0 \end{Bmatrix} \quad \text{——— (10)}$$

C. SQUEEZE FILM DAMPER

SFD shown in Figure 6 is commonly used in rotor systems in conjunction with a rolling element bearing for the purpose of attenuating rotor vibration. The empirical relations provided here are for the short bearing approximation (Barrett & Gunter, 1975). When the oil film completely fills (2π film, no cavitation) the annulus then the stiffness and damping of SFD are given by

When the oil film partially fills the annulus (π film, cavitation) then the stiffness and damping of SFD are given by

$$K = \frac{2\mu R L^3 \varepsilon \omega}{C^3(1-\varepsilon^2)^2} \quad \text{——— (13)}$$

$$K = 0 \quad \text{——— (11)}$$

$$C = \frac{\mu R L^3 \pi}{2C^3(1-\varepsilon^2)^{3/2}} \quad \text{——— (14)}$$

$$C = \frac{\mu R L^3 \pi}{C^3(1-\varepsilon^2)^{3/2}} \quad \text{——— (12)}$$

where μ is the viscosity of oil, C is the SFD clearance, ε is the eccentricity ratio (e/c; e is the journal eccentricity), ω is the angular speed of rotor, R is SFD radius and L is SFD land length. The above expressions are valid for the cases of circular synchronous radial motion about the origin with no precession.

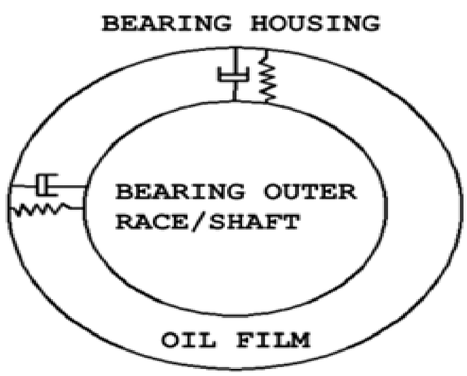

Figure 6: Arrangement of SFD

D. ACTIVE MAGNETIC BEARING

The lateral force exerted by the Active Magnetic Bearing (AMB) on the rotor at its nodal location is given by

$$\mathbf{f}_{AMB} = \begin{Bmatrix} k_s u_{AMB} - k_i i_c \\ 0 \end{Bmatrix} \quad \text{——— (15)}$$

where $$i_c(t) = K_p u_{AMB} + K_i \int u_{AMB}\, dt + K_D \dot{u}_{AMB}$$

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E. UNBALANCE FORCE

The unbalance force vector due to the mass eccentricities at the nodal locations of discs is

$$\mathbf{f}_{unb} = m_d e \omega^2 e^{j\omega t} e^{j\beta} \quad \text{——— (16)}$$

where e is the disc unbalance eccentricity located at the phase angle β .

V. RESULTS AND DISCUSSIONS

The response of the rotor-bearing system without external viscous damping through SFD is studied. The rotor is discretized into 25 elements, each 20 mm in length. The nodal locations of bearings are (1, 21) and those of discs are (6, 21). Table 1 shows the geometric properties of the shaft, discs and bearings.

Discs		Bearings	
Parameter	Value		
md1 (kg)	1.2	kbxx=kbyy (N/m)	6×10^5
md2 (kg)	1.5	cbxx (Ns/m)	400
Ip1 (kg·m²)	0.0014	cbyy (Ns/m)	600
Id1 (kg·m²)	0.0020		
Ip2 (kg·m²)	0.0028		
Id2 (kg·m²)	0.0039		
Shaft		Unbalance	
Diameter, ds (m)	0.022	m1e1 (kg·m)	5×10^{-5}
length, ls (m)	0.6	m2e2 (kg·m)	4.5×10^{-5}
		β1 (rad)	π / 4
		β2 (rad)	π / 3

Table1Propertiesofshaft,discs,and bearings

A. Disc Mass Moment of Inertia

For a solid disc:

$Ip = \frac{1}{2} mr2$; $Id = \frac{1}{4} mr2$

disc radius r = 0.045 m

B. Bearing Stiffness Increase

kb = 6×105 N/m

C. Unbalance Force

Fu= meω2

$\omega = \frac{2\pi N}{60} = 314$ rad/s

D. Shaft Stiffness Increase

Bending stiffness:

$$EI = \frac{\pi E d^4}{64}$$

$$= \left(\frac{0.022}{0.02}\right)^4$$

$$= 1.46$$

Shaft stiffness increases by 46%, improving realism for industrial rotor. The critical speeds of the rotor system are obtained from Figure 11 which is the hilbert envelope of vibration in x direction that is obtained by ramping up the rotor from 0 rpm to 12000 rpm at a rate of 40π rad/s for 10 seconds. It can be seen from that the resonant speeds of rotor are present at 3090 rpm and 8234 rpm. The orbits of the rotor at critical speeds are shown in Figure 7.

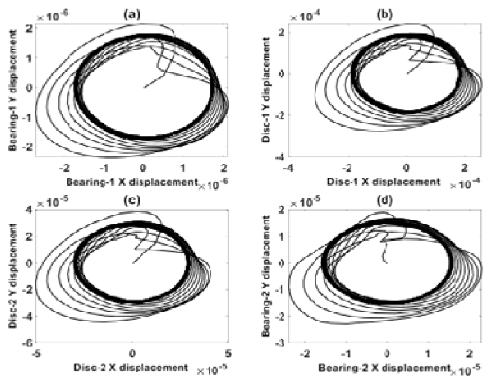

Fig 7: Orbits at various locations (a) Bearing-1 (b) Disc-1 (c) Disc-2 (d) Bearing-2 for c = 7430 N-s/m

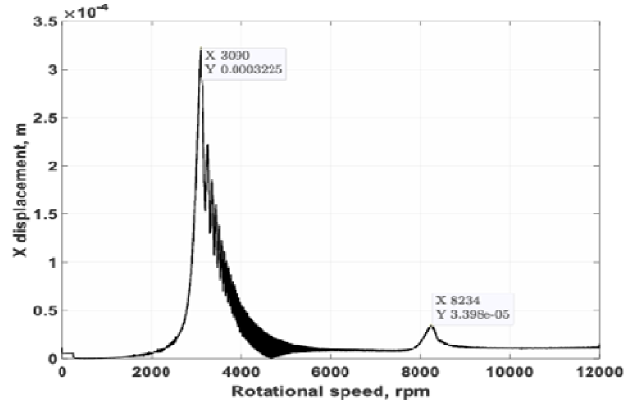

Fig 8: Hilbert envelope of rotor vibration in x direction for nominal unbalance at node-21

1) RESPONSE OF ROTOR SYSTEM WITH SQUEEZE FILM DAMPER (ROTOR-1)

Parametric studies are conducted by varying the radial clearance of the squeeze film damper to 100 µm, 175 µm, and 225 µm, and the land width to 22 mm and 30 mm. A full 2π oil film (assuming no cavitation) is considered around the annular region of the SFD. The damping coefficients corresponding to each configuration are evaluated using Eq. (4.11). The updated SFD parameters used in the analysis are summarized in Table 2.

Parameter	Value
Oil viscosity, µ (N·s/m²)	6×10^{-3}
SFD radius, R (mm)	80
Land width, L (mm)	22, 30
Radial clearance, c (µm)	100, 175, 225

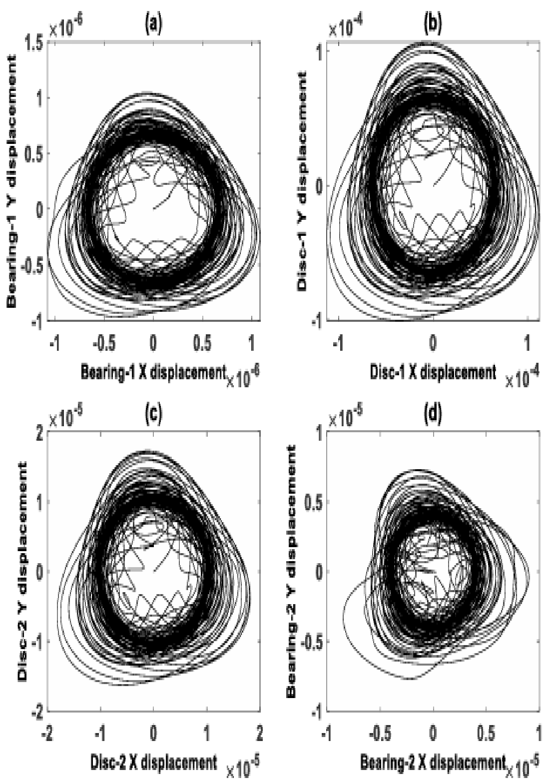

Fig 10: Orbits at various locations (a) Bearing-1 (b) Disc-1(c) Disc-2 (d) Bearing-2 for nominal unbalance

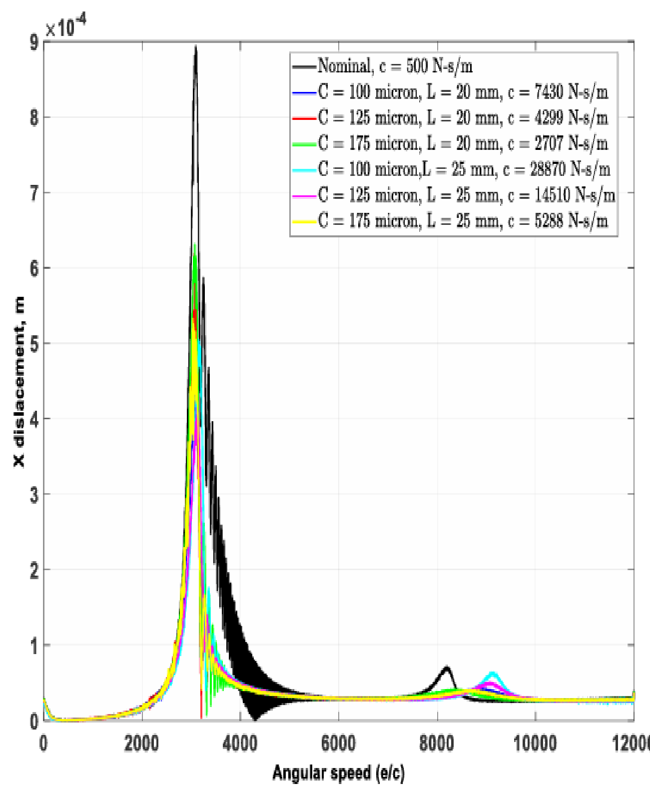

Fig 11: Response of various SFD configurations at Disc-2 location for 10 cm.gm unbalance

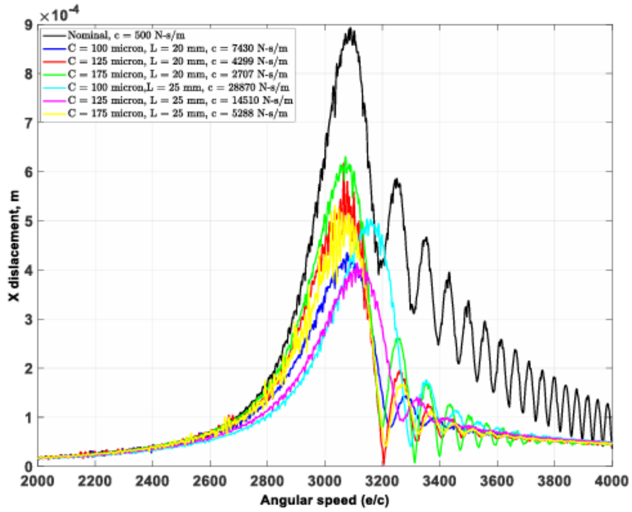

Fig 12: Response of various SFD configurations at 1st critical speed for 10 cm.gm unbalance

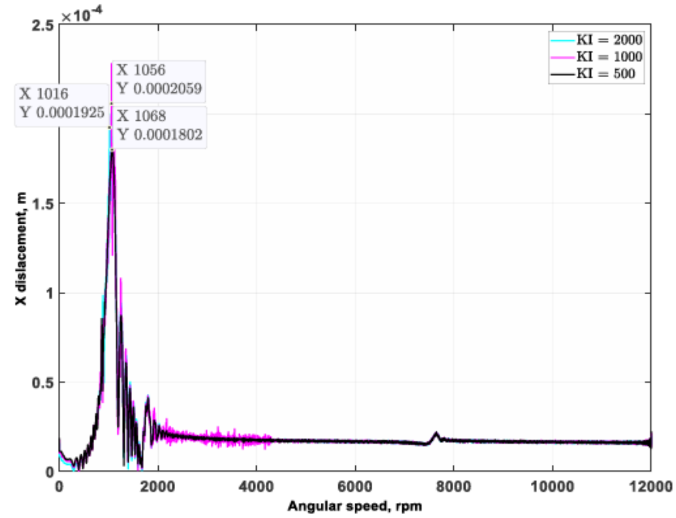

Fig 13: Sensitivity of Rotor-AMB system to K i(x-displacement at Disc-1)

The response curves exhibit linear behavior up to an eccentricity ratio (e/c) of 0.5, beyond which strong nonlinear characteristics are observed. In practical applications, squeeze film dampers (SFDs) generally operate within an eccentricity ratio range of 0.4–0.5. Figures 7(a)–7(f) show that, for a constant land width, an increase in radial clearance leads to a higher damping coefficient. In a similar way, a fixed radial clearance results in increased damping.

Figures 11 and 12 present the orbital responses at various rotor locations, including bearing-1, disc-1, disc-2, and bearing-2.

A noticeable reduction in orbit size is observed with increasing damping, indicating improved vibration suppression.The linear behavior observed with an eccentricity ratio up to 0.5 is explained by the balanced interaction between damping forces and fluid film pressures at lower eccentricities. This balance allows for predictable and proportional responses.

Thereafter, nonlinear characteristics result from increased fluid turbulence and asymmetrical pressure distributions caused by higher eccentricity ratios. SFDs typically operate within 0.4–0.5 ranges to optimize damping efficiency while maintaining stable performance. The increase in damping coefficient with radial clearance or land width arises because larger clearances or widths enhance the fluid's ability to dissipate energy, resulting in improved vibration suppression, as shown in the reduced orbit sizes. As the rotor accelerates above the second critical speed, figure 16 illustrates the response at disc-2 for six configurations.

As shown in Figure 13, higher damping shifts the critical speeds to higher values. Initially, vibration amplitudes decrease with increased damping; however, beyond an optimum value, further damping causes the amplitude to increase. For example, the x-direction displacement increases from approximately 400 µm at a damping coefficient of 14,510 N·s/m to about 500 µm at 28,870 N·s/m. This indicates that excessive damping stiffens the support system, resulting in increased vibration levels.

VI. CONCLUSIONS

This study dives into how squeeze film dampers (SFDs) and active magnetic bearings (AMBs) impact a rotor system featuring two discs—think of it as a high-speed spinning shaft setup common in turbines or engines.Modeling the Rotor We broke down the rotor using Timoshenko beam elements for accuracy, accounting for shear and rotary inertia. Unbalance forces from uneven mass distribution drive the external excitation. Global equations pull together the mass, stiffness, and gyroscopic matrices from both the shaft and discs.Baseline Critical Speeds The undamped version shows critical speeds at roughly 3090 rpm and 8234 rpm. At the first one, vibrations hit a peak of 322 microns—enough to worry about in real machinery.Squeeze Film Dampers in Action Placing SFDs at the bearings introduces viscous damping. Parametric tests varied clearances and lengths, revealing linear damping up to 0.5 eccentricity ratio; past that, nonlinear effects are significant. While these dampers initially reduce vibrations, excessively increasing the damping coefficient can induce "lock-up," causing the damper to behave as a rigid link, which in turn increases vibration amplitudes. Active Magnetic Bearings EdgeAMBs at the bearings offer tunable control via PID gains. Proportional gain shifts resonant speeds effectively; integral gives a slight shift plus amplitude reduction; derivative mirrors that. The AMBs adapt dynamics without hardware swaps, positioning them as a flexible SFD replacement for smarter rotor control.

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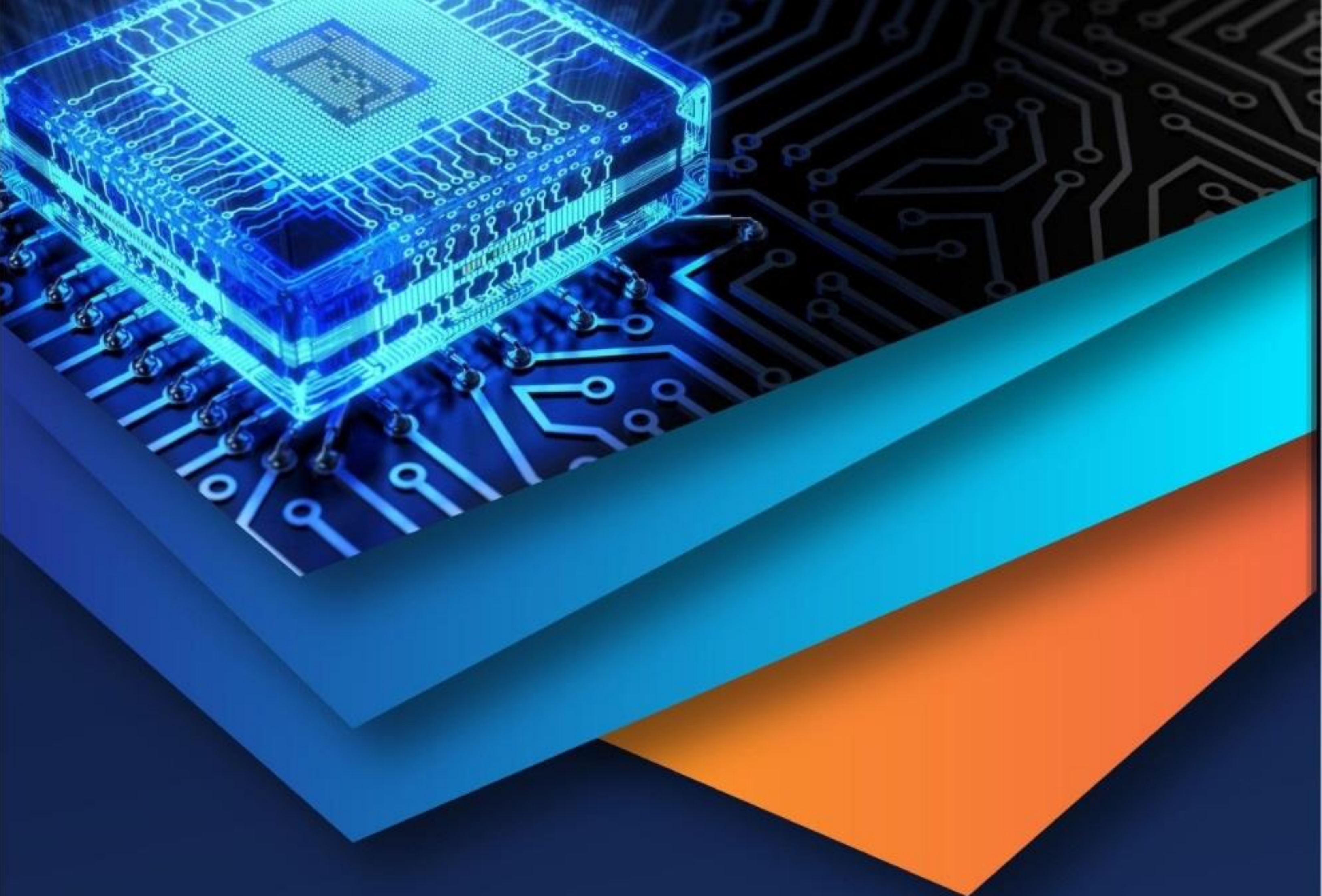

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