iJRASET
International Journal For Research in
Applied Science and Engineering Technology

INTERNATIONAL JOURNAL FOR RESEARCH
IN APPLIED SCIENCE & ENGINEERING TECHNOLOGY

Volume: 13 Issue: VIII Month of publication: August 2025

DOI: https://doi.org/10.22214/ijraset.2025.73752

International Journal for Research in Applied Science & Engineering Technology (IJRASET)
ISSN: 2321-9653; IC Value: 45.98; SJ Impact Factor: 7.538
Volume 13 Issue VIII Aug 2025- Available at www.ijraset.com

Comparative Analysis of Fixed and Tilting Pad Bearings in Rotordynamic Applications

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Abstract: Bearings constitute one of the most critical components in turbo and rotating machinery today. Bearings are fundamental elements in rotating machinery such as turbines, compressors, motors, and pumps, as they support the rotor, control vibration, and influence system stability.

Their stiffness and damping properties critically affect rotordynamic behaviour, including critical speeds and overall stability. Among the various types, fluid film bearings are widely used due to their ability to separate surfaces with a thin lubricant film, providing near-infinite life when properly maintained.

Operating regimes range from dry friction and mixed lubrication to full hydrodynamic lubrication, with the latter playing the most significant role in rotordynamic performance. Fluid film bearings exist in fixed-geometry (sleeve) and variable-geometry (tilting pad) configurations.

Fixed-geometry bearings are simpler and provide higher damping but are more susceptible to instability, while tilting pad bearings offer superior stability by eliminating cross-coupling effects, albeit at higher cost and complexity. The orientation (load-on-pivot vs. load-between-pivots), preload, and design parameters of tilting pad bearings strongly influence stiffness, damping, and thermal behaviour.

This paper highlights the rotordynamic implications of bearing design, configuration, and operating regimes, emphasizing their role in vibration control, stability enhancement, and overall machine reliability.

Keywords- rotordynamic, system stability, damping, critical speeds, hydrodynamic lubrication.

I. INTRODUCTION

Bearings are critical components in rotating machinery (turbines, compressors, motors, pumps, etc.) because they support the rotor, control vibration, and influence stability. Their stiffness and damping properties directly affect the dynamic behaviour of the rotor. Bearings are crucial in rotordynamics as they support rotating shafts and influence the dynamic behaviour of rotor systems, including critical speeds and stability.

The type of bearing (e.g., journal, thrust, magnetic) and its characteristics (stiffness, damping) significantly impact the rotor's response to imbalances and other disturbances. Improper bearing selection or wear can lead to excessive vibrations, reduced performance, and potential system failure.

A. Fluid Film Bearings

In a fluid film bearing, the stationary and rotating surfaces are separated by a thin film of lubricant, such as oil, air, water or process fluid. In a hydrodynamic fluid film bearing, the film pressure that separates the surfaces is created by the relative motion (rotation) of the surfaces as the lubricant is pulled into a converging geometry between the surfaces.

No contact of the surfaces takes place except during start-up and shutdown. Fluid film bearings have been around for centuries, but the understanding of how they actually worked did not come about until 1882–1883.

Fluid film bearings operating in the full hydrodynamic regime support the load on a very thin film and thus there is no contact between the shaft and the bearing.

It is for this primary reason that fluid film bearings offer infinite life provided the lubricant is kept clean and the machine is operating in a safe dynamic range away from stability and critical speed problems.

The primary requirement for hydrodynamic lubrication is that sufficient lubricant (typically mineral or synthetic oil, but it could be any fluid with the proper characteristics for the application) exists between the shaft journal and the bearing at all times. The formation of an oil wedge to lift the shaft journal is similar to hydro planning (controlled hydro planning in the case of bearings) and is dependent on the speed (relative speed between
the shaft and the bearing), load (weight of the rotor or any additional side loads from process fluid, or gear loads, or side loads due to
These parameters are combined and presented by the ZN/P curve shown in Fig. 1 The symbol Z represents viscosity, N is the speed in rpm, and P is the unit loading in lb/in2. This curve describes the three regimes of operation a bearing passes through while the machine accelerates to operating speed or decelerates from the operating speed to stand still conditions.

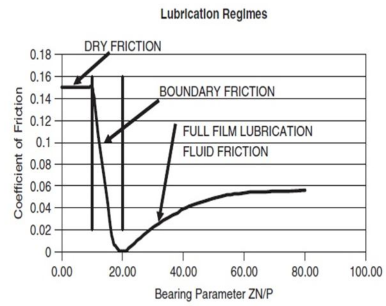

Fig-1: ZN/P Curve and their lubrication regime

These regimes are dry friction, where contact between the asperities of the shaft and the bearing exist; mixed lubrication regime, or boundary friction, and full hydrodynamic lubrication, fluid friction, where a thin film exists between the shaft and the bearing, which supports the static and dynamic loads in the rotating shaft. These three regimes are also shown graphically with an exaggerated clearance in Fig. 5-2 as the shaft accelerates from standstill to full operating speed. This regime has the most influence on the rotordynamic characteristics of the machine as it speeds up from stand still to possibly traverse one or more critical speeds on its way to reaching the design operation speed. Fluid film bearings develop pressure in the converging wedge, which supports the radial load on a thin oil film, Most fluid film bearings are of the hydrodynamic type.

where the film pressure is produced by the shaft rotation dragging the oil into the converging wedge formed by the shaft and the bearing surface. In hydrodynamic bearings the oil pressure is just enough to keep the bearing supplied with oil and to remove the frictional heat generated by the viscous shear in the thin film. A less common type of fluid film bearing is the hydrostatic type, where the film pressure is provided through external pressurization and feeds a recess in the bearing that supports the load. Such a bearing requires pressurization that is orders of magnitude higher than that needed for a hydrodynamic bearing and a significant power is required to drive the high-pressure oil pump.

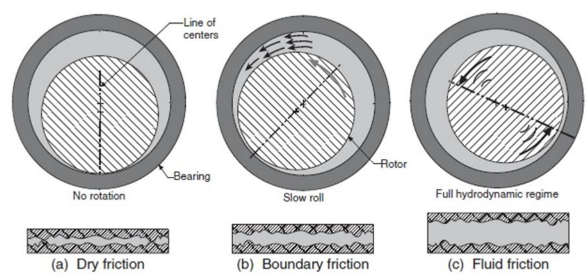

Fig-2: Lubrication regimes in fluid film journal bearings

Hydrostatic bearings are also used to provide a very stiff support and to allow rigid rotors to operate subcritical. This approach of providing very stiff bearings is successful only in limited applications. From the standpoint of rotordynamics, the main advantage of fluid film bearings is their inherent damping characteristics. Fluid film bearings exist in a variety of configurations depending on the application, space availability, and the rotordynamic requirements.

B. Fixed-Geometry Sleeve Bearings

The trend over the last few decades has favoured the variable geometry tilting pad type bearings over the fixed-geometry sleeve type bearings, particularly in high-speed, supercritical machinery due to their inherent stability characteristics. However, these variable-geometry tilt pad bearings are more complex, contain more parts, and generally have lower damping than fixed-geometry bearings. A fixed-geometry or fixed-profile sleeve bearing supports the weight of the rotating shaft by developing a hydrodynamic pressure in the converging wedge formed by the shaft and the bearing surface, shown in Figure-3. The asymmetric pressure profile in the oil film is characteristic of fixed geometry journal bearings. This gives rise to an attitude angle formed by the line of centres (line connecting the centre of the shaft and the centre of the bearing) and the load vector.

This characteristic, present in all fixed geometry journal bearings, is indicative of the presence of cross-coupling in the bearing. The load, which in this case is due primarily to the weight of the rotor, is acting directly downward on the journal bearing. Integrating the pressure profile results in a net force that balances the weight of the rotor. This force generated in the film is accompanied by a shaft displacement with a component that is along the load direction (direct) and a displacement that is orthogonal to the load direction and is shifted in the direction of rotation (cross). This cross-coupling characteristic is present in all rotating machinery where a fluid or gas is rotating with the shaft in a small annulus.

This phenomenon is unique to rotating machinery and is not present in other nonrotating structures or mechanisms. It is this cross-coupled stiffness force that promotes self-excited, subsynchronous vibrations and instability in rotating machinery.

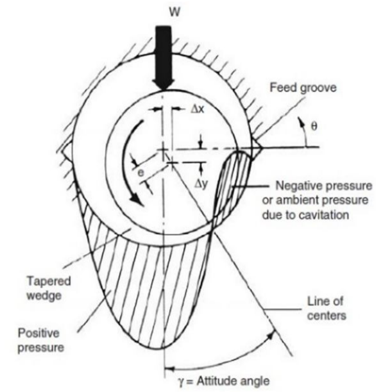

Fig-3: Pressure profile and attitude angle in sleeve bearing

The frequency of the subsynchronous vibrations in the oil whirl region is dependent on the L/D ratio of the bearing. The higher that ratio is, the closer the subsynchronous oil whirl is to half running speed. This gives rise to the thought that the subsynchronous frequency is directly related to the average circumferential flow of the lubricant between the shaft and the bearing. This circumferential flow is closer to half running speed as the length of the bearing increases and the effect of the side leakage is relatively smaller.

the cross-coupling in the journal bearing tends to always move or push the shaft to the opposite side of the converging wedge.

Therefore, this cross-coupling effect is always biased by the direction of rotation as shown in Fig-4.

The term rotational bias is often used to describe this phenomenon, but what is also relevant is the fact that it is forward driving. Therefore, when diagnosing subsynchronous vibrations it is important to note if the whirl of the subsynchronous vibrations is forward or backward to help distinguish oil whirl and whip from rub and other anomalies that produce backward whirl. Positive damping dissipates energy and, in this manner, it reduces vibrations and whirl amplitude, while the negative damping adds energy to the dynamic system and therefore increases vibrations and amplitude of motion. What may determine the final stability of the rotor–impeller–bearing–seal dynamic system is often dependent on the net effect of these two opposing forces—direct damping and the destabilizing cross-coupled stiffness (negative damping) present in the bearings, seals,

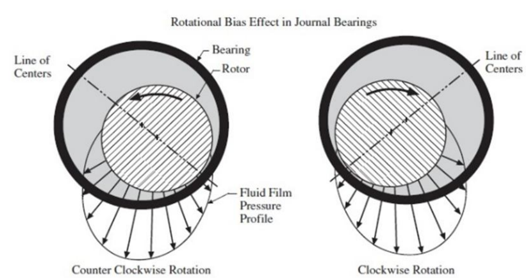

Fig-4: Rotational bias caused by cross-coupled

Although load and eccentricity also play a role in the stability of these bearings, the circular geometry and fluid rotations are directly related to the bearing design and configuration. These two parameters can be changed and/or modified to help an unstable or marginally stable bearing become more stable. Therefore, fixed-geometry bearings can achieve a higher stability threshold by adding axial grooves as shown in Fig-5 to reduce fluid rotation. The more grooves that are added, the more the net fluid rotation in the bearing is reduced, and the higher the stability threshold.

There is a limit to the addition of grooves in the bearing where it can become counterproductive. This is because as we add more grooves, we are effectively reducing the load area, increasing the bearing temperature, and eventually reducing the direct damping. There is also the possibility of the load vector being directly in line or very close to the groove, which will affect the flow of oil into the bearing. Thus, it is important to note that improving one aspect of the bearing performance may have deleterious effects on other operating parameters. Therefore, a complete bearing design must examine all aspects of the bearing performance and

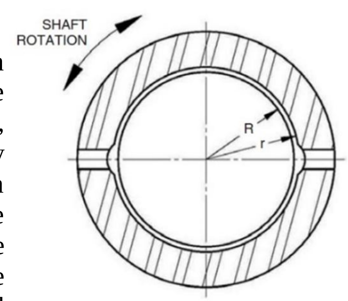

Fig-5: Sleeve bearing with two axial grooves

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C. Variable-Geometry Tilting Pad Bearings

Variable-geometry tilt pad bearings are characterized by the inherent stability that arises from the low or negligible cross-coupling present in these bearings. The pads tilt or rotate about their pivot in response to the radial load applied by the shaft journal, and will always produce a reaction force that is in line with the shaft centres, as shown in Fig-6.

There is no attitude angle (attitude angle is zero) and therefore no corresponding cross-coupled stiffness. The attitude angle is almost zero provided the pad inertia and the friction in the pivot are low or negligible.
Conventional tilt-pad bearings use a point or a line contact for the pivot and achieve the tilt motion through this mechanism, The contact stresses can be very high on these bearings particularly when used in integrally geared compressors or when the radial load or weight is high. Depending on the proximity of the critical speed to the operating speed, this may result in a loss of the separation margin and a further degradation in the bearing pivot leading to further increase in synchronous vibrations. The pivot wear can increase the bearing set clearance, and in many instances could significantly reduce the preload and cause the rotor–bearing dynamic system to become unstable.

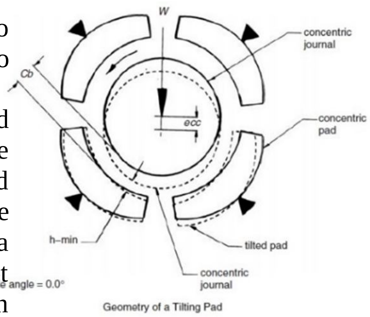


Fig-6: Tilting Pad Bearing

The conventional variable-geometry tilt pad bearings offer greater stability than the fixed-geometry sleeve bearings, but this benefit comes at a cost and brings some additional drawbacks that we need to be aware of. By virtue of having multiple parts, the tilt pad bearings are more costly to make and require longer lead time.

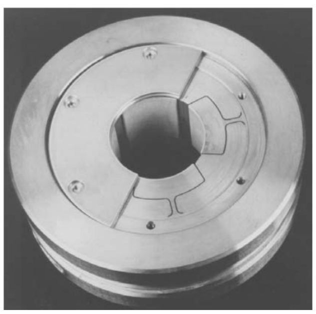
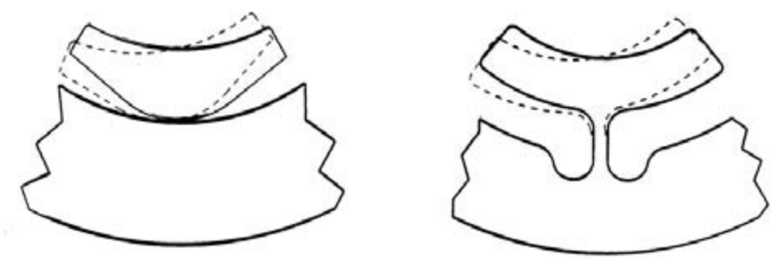


Fig-7: Flexure Pivot tilt pad bearing



(Rocker Back Tilt Pad) (Flexure Pivot Tilt Pad)

Fig-8: Rocker back and Flexure pivot tilt pad bearings

Flexure Pivot tilt pad bearing shown in Fig-7. This bearing achieves the tilt necessary for bearing stability through flexure of the web supporting the pad as shown in the schematic of Fig-8.

D. Load Between Pivots Versus Load on Pivot

Although tilt pad bearings, like fixed-geometry sleeve bearings, come in many variations and configurations, they have many parts and therefore more variations are available. These variations allow greater flexibility for the designer and rotordynamicist to tailor the bearing for the application. These variables include the orientation of the bearing for a load on pivot (LOP) versus a load between pivots (LBP), the number of pads used, the pad pivot offset angle, axial length of the pad, clearance, lubricant viscosity, and preload. They are all factors that can modified to achieve the desired thermal and rotordynamic characteristics.
The orientation of a tilt pad bearing with respect to the radial load plays an important role as this directly relates to the stiffness and damping characteristics of the bearing. The schematic shown in Fig-9 illustrates the orientation with respect to the load vector for a LOP and a LBP tilt pad bearing.

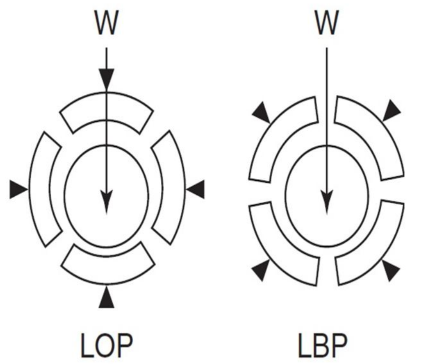


Fig-9: Schematic of LOP versus LBP in a 4-pad

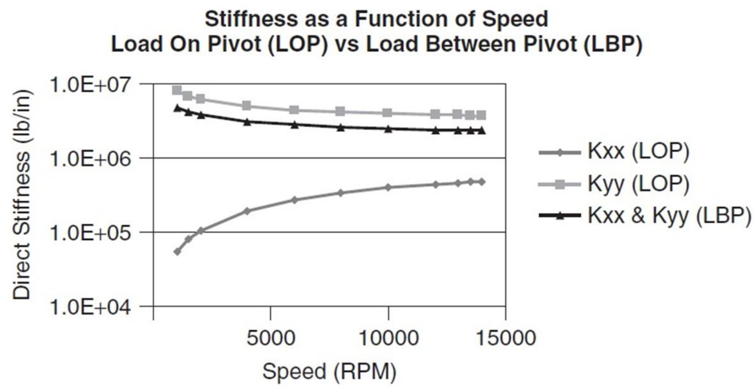


Fig-10: Direct stiffness coefficients for a 4-pad bearing in LOP versus LBP

The vertical and horizontal stiffness shows a large stiffness asymmetry for the LOP orientation, as shown in the plot in Fig. 5-24, whereas the LBP shows very symmetric characteristics and the vertical and horizontal stiffness curves are virtually identical and cannot be discerned in the plot. Furthermore, the LBP configuration tends to allow both bottom pads to share the load and thus this configuration offers higher load capacity than the LOP configuration.

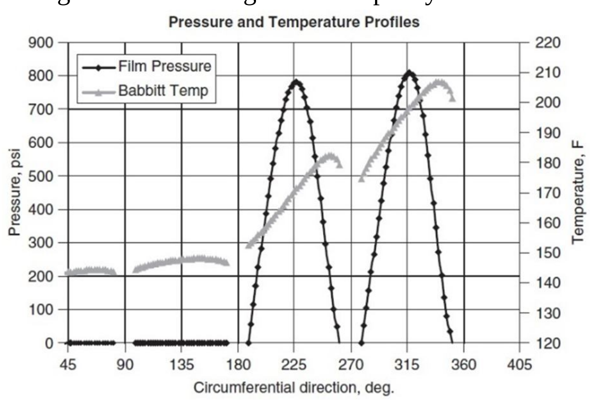


Fig-11: Pressure and temperature profile on a 4-pad LBP bearing configuration

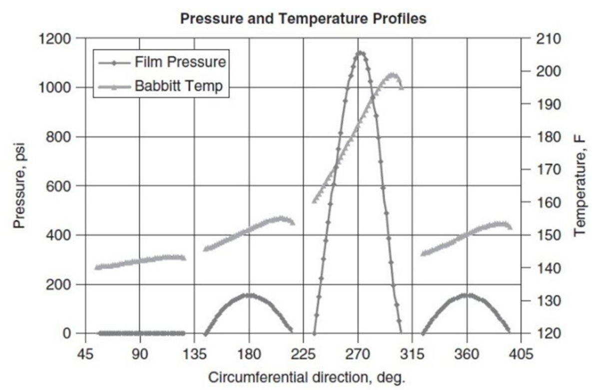


Fig-12: Pressure and temperature profile on a 4-pad LOP bearing configuration

This is apparent in the plot shown in Figs-11 and 12 for the LBP and LOP, respectively. In the LBP configuration the peak pressure in either of the two loaded pads is around 800 psi. In the LOP case the peak pressure reaches 1140 psi, which exceeds some of the conservative limits often employed for long term operation.

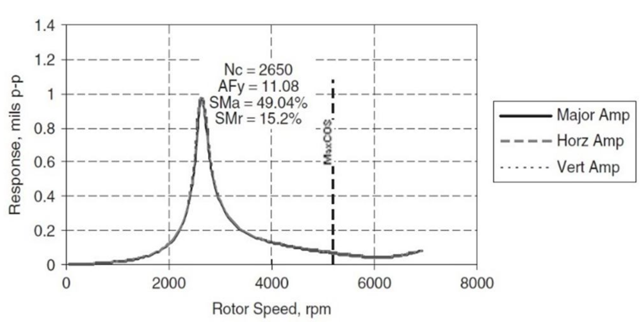


Fig-13: Unbalance response with LBP bearings

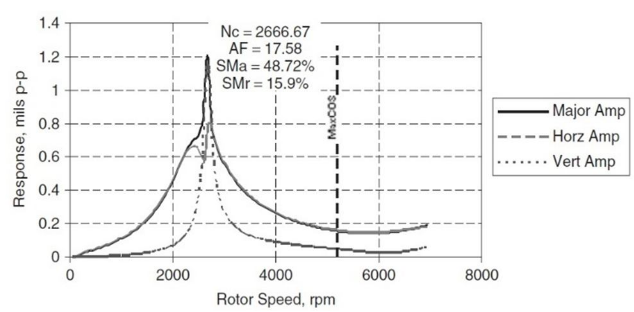


Fig-14: Unbalance response with LOP bearings

However, when we examine the response to imbalance, it is clear there is a distinct advantage for the LBP configuration. The response plot in Fig-13 clearly shows a more damped and symmetric response exhibited by the identical amplitudes predicted for the vertical and horizontal response. The amplification factor is also lower for the LBP with 11.08 as opposed to the 17.58 for the LOP configuration shown in Fig-14. The LOP response also shows a split critical which is introduced by the asymmetric stiffness characteristics for this bearing configuration.

E. Influence of Preload on the Dynamic Coefficients in Tilt Pad Bearings

The geometric preload in tilt pad bearings is a critical and effective parameter to alter the magnitude of the force coefficients. Preload can be explained in the graphic shown in Fig-15.

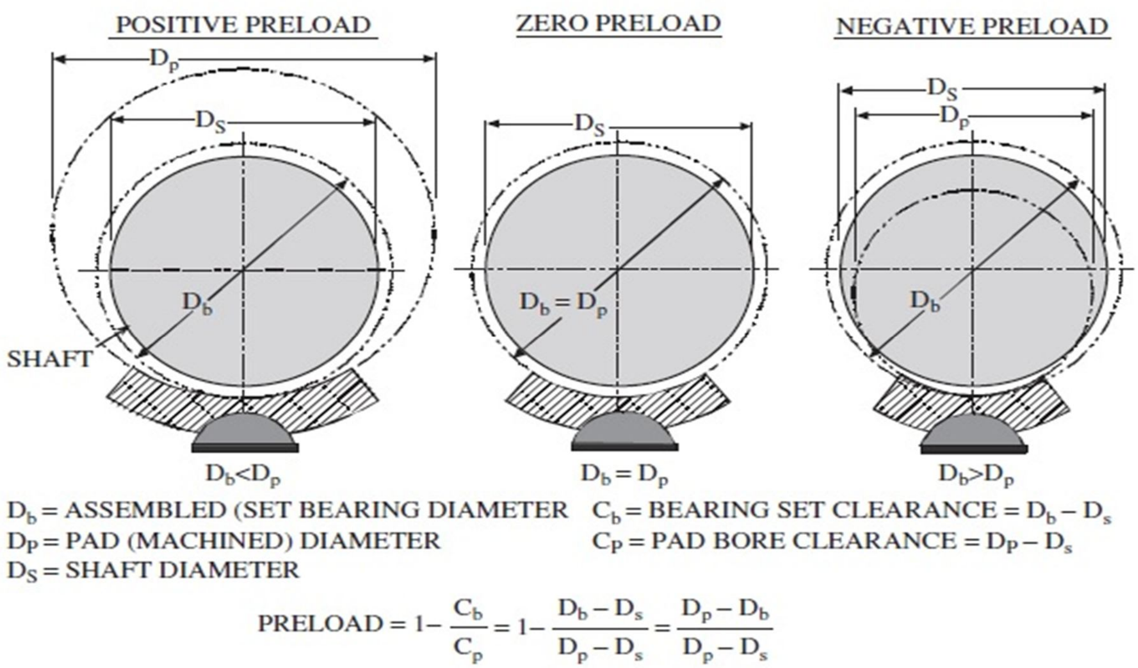


Fig-15: Schematic representation of positive, zero, and negative preload in tilt pad bearings

A positive preload is essential also from the standpoint of providing a larger inlet for the oil at the leading edge of the bearing. In contrast, zero preload due to manufacturing tolerance can quickly lead to a negative preload. This will reduce the effectiveness of the oil entering the leading edge of the bearing and will cause pad flutter or pad instability

II. CONCLUSIONS

The reliability issues at Hindalco's Mahan plant clearly show that soft foot—a condition where one or more motor feet fail to Bearings are fundamental to the performance, reliability, and stability of rotating machinery. Their stiffness and damping properties strongly influence rotor dynamics, critical speeds, and vibration behaviour. Fluid film bearings, particularly hydrodynamic types, provide long life due to non-contact operation and inherent damping, though stability concerns such as oil whirl must be addressed. Fixed-geometry sleeve bearings offer simplicity and higher damping but can be prone to instabilities unless modified with features like grooves. On the other hand, variable-geometry tilting pad bearings deliver superior stability by minimizing cross-coupling effects, with load orientation (LOP vs. LBP) and preload significantly affecting their dynamic response. While more complex and costlier, tilting pad designs allow greater flexibility to tailor performance for specific applications.

Overall, proper bearing selection and configuration—balancing stability, damping, load capacity, and cost—are critical to ensuring safe, efficient, and reliable operation of high-speed rotating machinery.

III. ACKNOWLEDGE

I would like to express my sincere gratitude to Mr. Pankaj Mishra (Vibration Expert BHEL), whose expert guidance, encouragement, and insightful feedback were invaluable at every stage of this work.

Special appreciation goes to my colleagues and peers for their constant support, lively discussions, and collaborative spirit, which greatly enriched this study.

Finally, I am profoundly thankful to my family and friends for their patience, understanding, and unwavering encouragement throughout this research.

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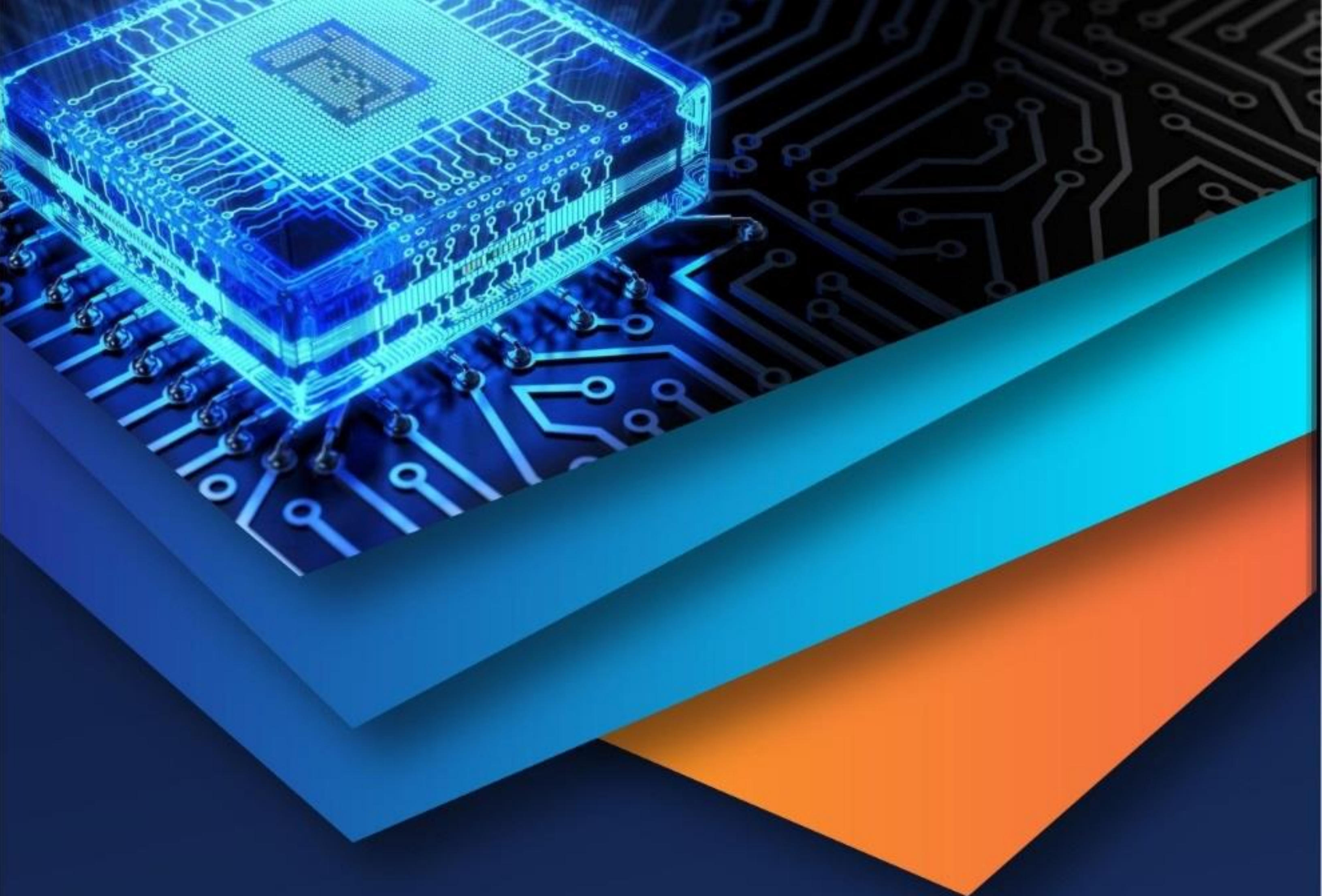

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