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Modal Analysis of Rotor System Using Line Body Elements in ANSYS Workbench

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Abstract: Rotordynamics deals with the vibration and dynamic behavior of rotating structures and plays a crucial role in the design of high-speed rotating machinery such as turbines, compressors, pumps, and electric motors. Accurate prediction of rotor dynamic characteristics during the design stage is essential to avoid excessive vibration, resonance, and catastrophic failures. The Finite Element Method (FEM) has become one of the most reliable techniques for analyzing rotor-bearing systems. Traditionally, rotor dynamic analyses have been performed using specialized commercial software; however, the continuous advancement of general-purpose finite element software such as ANSYS Workbench has provided an efficient and user-friendly platform for such analyses. This study investigates the applicability of line body (beam) elements in ANSYS Workbench for the dynamic analysis of a rotor system. Unlike three-dimensional solid elements, line body elements significantly reduce computational effort while maintaining satisfactory accuracy for slender rotor shafts. A finite element model of the rotor system is developed using line body elements, and its dynamic characteristics are evaluated through modal, harmonic, and transient analyses. The numerical results are validated with experimental measurements to assess the accuracy and reliability of the proposed modeling approach. The study demonstrates that line body elements in ANSYS Workbench provide an efficient and computationally economical alternative for rotor dynamic analysis without compromising the accuracy of the predicted dynamic response.

Keywords: *Rotordynamics, Line Body Elements, Finite Element Method, Modal Analysis, Harmonic Response, Transient Analysis, Rotor System.*

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I. INTRODUCTION

Rotating machinery, including turbines, compressors, pumps, generators, electric motors, and machine tools, plays a crucial role in power generation, manufacturing, aerospace, and automotive industries. The rotor is the primary rotating component of these machines, and its dynamic behavior directly affects system performance, operational reliability, and service life. As rotational speeds increase, rotor systems become more susceptible to excessive vibration, resonance, shaft instability, and fatigue failure. Therefore, accurate prediction of natural frequencies, critical speeds, and vibration characteristics during the design stage is essential to ensure the safe and efficient operation of rotating machinery.

The Finite Element Method (FEM) has become one of the most reliable numerical techniques for rotor dynamic analysis due to its capability to model complex rotor-bearing systems with high accuracy. Traditionally, rotor dynamic analyses were performed using specialized commercial software; however, the continuous development of general-purpose finite element software such as ANSYS Workbench has provided an efficient and user-friendly platform for analyzing rotor systems. ANSYS Workbench offers advanced capabilities for modal, harmonic, transient, and Campbell diagram analyses, enabling comprehensive evaluation of rotor dynamic characteristics. Among the available modeling techniques, line body (beam) elements provide a computationally efficient alternative to three-dimensional solid elements by significantly reducing computational time while maintaining acceptable accuracy for slender rotor shafts.

The development of rotor dynamics has progressed from analytical formulations to advanced finite element and experimental approaches. Early investigations by Genin and Murphy (1971) and Mingori (1973) established the theoretical foundations of rotor stability by examining the influence of material damping and shaft whirl. Muszynska (1995) further advanced the understanding of nonlinear rotor behavior by investigating forward and backward whirl in anisotropically supported rotors. With the advancement of numerical methods, Andrés (2006) reviewed the rotor dynamic behavior of turbomachinery and emphasized the importance of accurately modeling bearing and seal dynamics, while Vatta and Vigliani (2007) demonstrated the influence of internal material damping on rotor stability. Nelson (2007) provided a practical interpretation of rotor dynamic principles, and Das and Dutt (2008) proposed a reduced-order finite element model using the modified System Equivalent Reduction Expansion Process (SEREP) to improve computational efficiency. Samuelsson (2009) and Nassis (2010) further demonstrated the application of finite element techniques and ANSYS software for modeling and analyzing industrial rotor systems.

Subsequent research has focused on improving the accuracy of finite element models through model updating and experimental validation. Chouksey et al. (2012) investigated the combined effects of shaft material damping and fluid-film bearing forces on the modal characteristics and stability of flexible rotor systems. Later, Chouksey et al. (2013, 2014) proposed an Inverse Eigen Sensitivity Method (IESM) for finite element model updating of ball-bearing- and journal-bearing-supported rotor systems using experimentally measured modal parameters, significantly improving the prediction of rotor dynamic responses. Fegade and Patil (2013) carried out harmonic response analysis and design optimization of rotor systems using ANSYS, while Wagner and Hübner (2013) investigated the dynamic behavior of rotors integrated with complex supporting structures. Saxena et al. (2016) analyzed the modal characteristics of geared rotor systems and demonstrated the influence of gyroscopic effects and bearing stiffness on natural frequencies and critical speeds.

Recent studies have incorporated advanced numerical techniques to improve the modeling accuracy of rotor-bearing systems. Kang et al. (2021) developed a finite element model updating approach based on Particle Swarm Optimization (PSO) for identifying the dynamic coefficients of journal bearings. Choe et al. (2022) proposed a computational framework for nonlinear vibration analysis of rotor systems equipped with squeeze film dampers using Component Mode Synthesis (CMS) and the Incremental Harmonic Balance (IHB) method. Afane et al. (2023) evaluated the influence of different shaft materials on the modal and harmonic responses of a Jeffcott rotor using ANSYS Workbench. More recently, Li et al. (2024) presented an improved concentrated-mass formulation for modal analysis of centrifugal pump rotor systems, while Zhang et al. (2024) developed a nonlinear finite element model of a double-row angular contact ball bearing–rotor–disk system to investigate the effects of bearing configuration and rotational speed on system dynamics.

The reviewed literature indicates that significant progress has been achieved in analytical modeling, finite element analysis, nonlinear vibration analysis, and finite element model updating of rotor-bearing systems. However, most previous studies have investigated modal analysis, harmonic response analysis, or experimental validation independently, and only a limited number have integrated these analyses within a single computational framework. Moreover, studies on ball-bearing-supported rotor systems modeled using line body (beam) elements in ANSYS Workbench remain limited, despite their computational efficiency and practical engineering advantages. Consequently, there is a need for an integrated and experimentally validated numerical approach capable of accurately predicting the dynamic behavior of rotor systems using line body elements.

Hence, the present study develops a finite element model of a ball-bearing-supported rotor system using line body elements in ANSYS Workbench. The proposed model is employed to perform modal, harmonic, and transient analyses, and the numerical predictions are validated through experimental measurements. The study aims to demonstrate that line body elements provide an accurate, computationally efficient, and reliable alternative to solid element modeling for rotor dynamic analysis, thereby offering a practical methodology for the design and analysis of rotating machinery.

II. NUMERICAL MODEL ANALYSIS

➢ *Geometry of Rotor Bearing System*

The rotor system considered in the present study consists of a uniform steel shaft carrying a centrally mounted rigid disc and supported by two identical rolling-element bearings located at both ends, as shown in Figure 1. The numerical model is developed based on the experimental rotor test rig reported by Chouksey et al. (2013) and Kang et al. (2021). The shaft has a total length of 908 mm and a diameter of 20.62 mm, while the disc has a diameter of 144.12 mm and a thickness of 40.14 mm. The disc is positioned asymmetrically at distances of 407 mm and 501 mm from the left and right bearings, respectively.

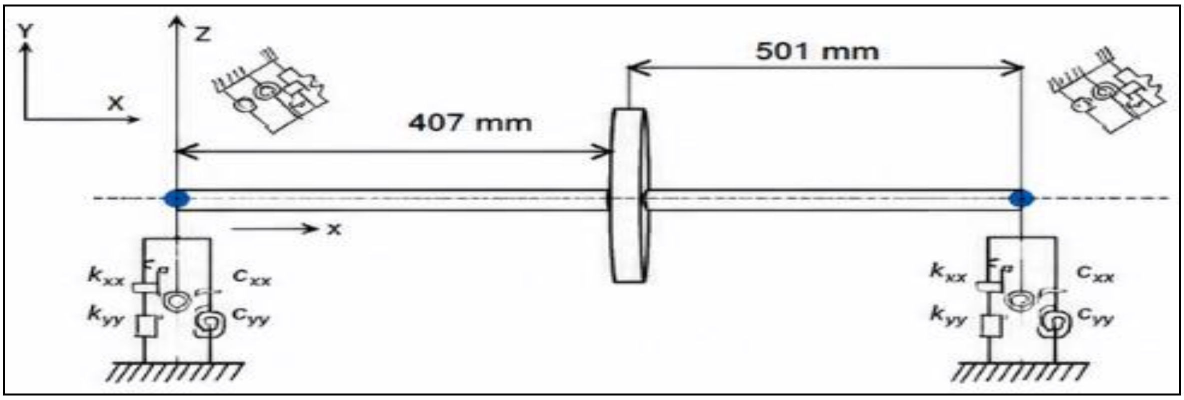

Fig 1 Geometry of Rotor Bearing System

The shaft and disc are assumed to be made of structural steel with homogeneous and isotropic material properties. The material properties used in the analysis are listed in Table 1.

Table 1 Material Properties of Rotor System

Property	**Value**
Young's Modulus, E	200 GPA
Density, ρ	7800 kg/m^3
Poisson's Ratio, ν	0.30
Material Damping	9.5×10^{-6} s

The rotor-disc is considered to be rigid and has been represented by its inertia properties, i.e., mass of disk is 5.29 kg, moment of inertia about diameter is 0.007517 $kg.m^2$, and polar moment of inertia is 0.01312 kg m^2s, which are found by generating a solid model of the disc.

- *Mathematical Formulations and Boundary Conditions*

The rotor-bearing system is modeled using the finite element method based on Timoshenko beam theory, which accounts for rotary inertia, shear deformation, gyroscopic effects, and distributed shaft damping. The governing equation of motion of the rotating system is expressed as:

$$[M]\, q(t)+[C+G]\, q(t)+[K+H]\, q(t) = F(t) \qquad (1)$$

Where q is the generalized coordinate matrix, M is the mass matrix, C is the symmetric damping matrix, G is the gyroscopic skew-symmetric damping matrix, K is the symmetric stiffness matrix and H is the skew-symmetric circulatory matrix.

For free undamped vibration,

$$[M]\, q(t) +[K]\, q(t) = 0 \qquad (2)$$

For a single-degree-of-freedom rotor model, the undamped natural frequency, damping ratio and damped natural frequency are given by:

$$\omega_n = \sqrt{(k/m)}, \qquad \zeta=c/2m\omega_n, \qquad \omega_d = \omega_n \sqrt{(1- \zeta^2)} \qquad (3)$$

Where k, c, and m denote stiffness, damping coefficient, and rotor mass, respectively.

The rotor shaft is supported by two identical rolling-element bearings located at the two ends. The bearings are modeled as linear elastic supports incorporating both translational and rotational stiffness and damping coefficients, as given in Table 2.

Table 2 Bearing Stiffness and Damping Coefficients

Parameter	**Value**
Translational bearing coefficients	
$K_{yy} = K_{zz}$	1×10^8 N/m
$C_{yy} = C_{zz}$	200 Ns/m
Rotational (tilting) bearing coefficients	
$K_{\theta y}$	1.5687×10^3 Nm/rad
$K_{\theta z}$	1.2686×10^3 Nm/rad
$C_{\theta y}$	0.89 Nms/rad
$C_{\theta z}$	0.89 Nms/rad

- *Line Body Modeling of Rotor System*

The finite element model is developed in ANSYS Workbench using Beam188 line body elements to represent the rotor shaft. Beam188 is a two-node Timoshenko beam element capable of accurately predicting bending deformation, rotary inertia, and shear deformation while significantly reducing computational cost compared with three-dimensional solid elements. The rigid disc is modeled as a Point Mass (Mass21) located at the actual disc position on the shaft. This approach accurately reproduces the mass and inertia effects without increasing the number of finite elements. The shaft is discretized into 27 Beam188 elements connected through 28 nodes, as shown in Figure 2. A circular beam cross-section corresponding to the shaft diameter is assigned to all beam elements. The finite element discretization enables accurate prediction of the rotor dynamic characteristics while maintaining computational efficiency.

Although the rotor model is developed in ANSYS Workbench, the rotational stiffness and damping associated with rolling-element bearings cannot be directly assigned through the graphical user interface. Therefore, ANSYS Parametric Design Language (APDL) commands are incorporated within the Workbench environment. The translational stiffness and damping coefficients are assigned through body-ground spring connections. Rotational stiffness and damping are introduced using APDL Commands inserted under the bearing connections, allowing stiffness about the local Y and Z rotational degrees of freedom to be defined. The APDL implementation provides realistic simulation of bearing flexibility while preserving the convenience of the Workbench environment. Figure 3 illustrates the implementation of bearing parameters through APDL commands.

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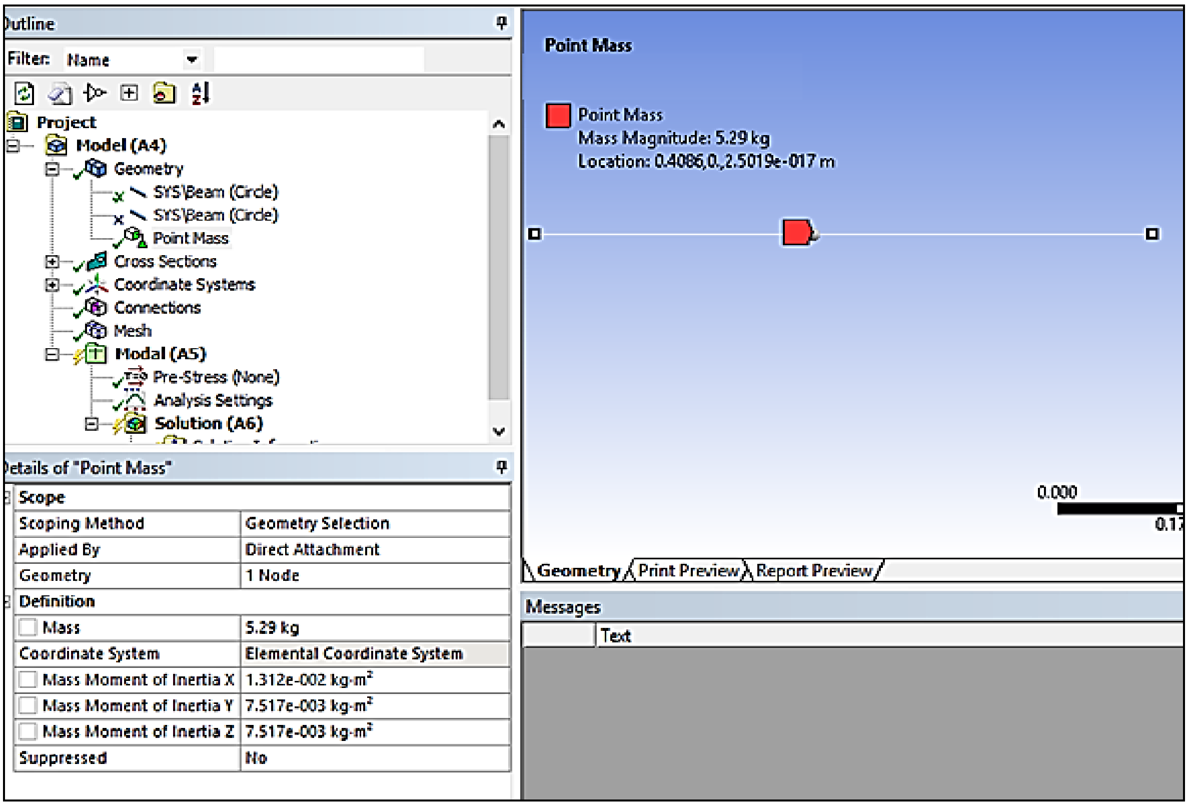

Fig 2 Real Constants for Mass21 Element

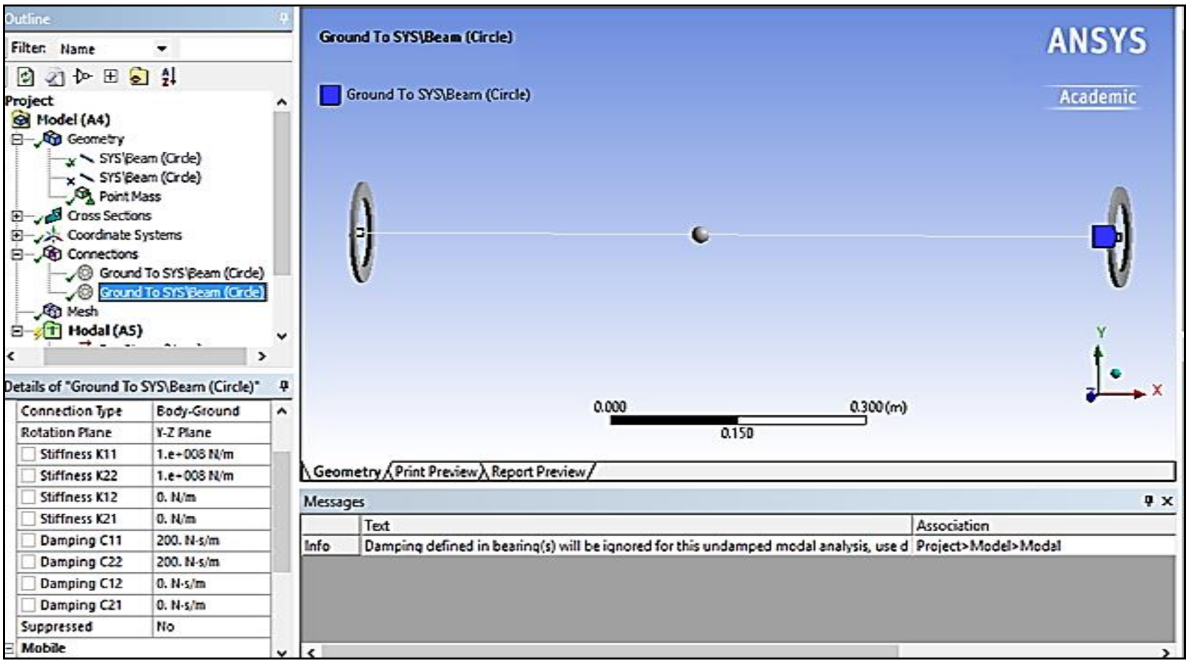

Fig 3 Bearing Parameters

➢ *Numerical Methodology*

The numerical analysis was performed in ANSYS Workbench by developing a line body finite element model of the rotor-bearing system. The shaft was modeled using Beam188 elements with a circular cross-section, while the rigid disc was represented by a Mass21 point mass. The rolling-element bearings were modeled using body-ground spring connections, where translational stiffness and damping were assigned through the Workbench interface and rotational (tilting) stiffness and damping were incorporated using APDL commands. After defining the material properties, inertia properties, bearing parameters, and boundary conditions, modal, harmonic, and transient analyses were performed to determine the natural frequencies, mode shapes, and dynamic response of the rotor system. Finally, the numerical results were validated against experimental modal data to assess the accuracy of the proposed line body finite element model.

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III. RESULTS AND DISCUSSION

The modal characteristics of the ball-bearing-supported rotor system were investigated using the finite element model developed in ANSYS Workbench with line body (Beam188) elements.

➢ *Model Analysis and Validation*

The modal analysis was performed to determine the natural frequencies and corresponding mode shapes of the non-spinning rotor. The predicted natural frequencies were subsequently compared with the experimentally measured values reported by Chouksey et al. (2013) to evaluate the accuracy of the proposed numerical model.

Table 3 presents the comparison between the experimentally measured natural frequencies and those predicted by the present finite element model. The numerical results exhibit excellent agreement with the experimental measurements. The maximum percentage error is only 2.89%, while most of the predicted frequencies differ by less than 1%, demonstrating that the line body modeling approach accurately captures the dynamic characteristics of the rotor-bearing system.

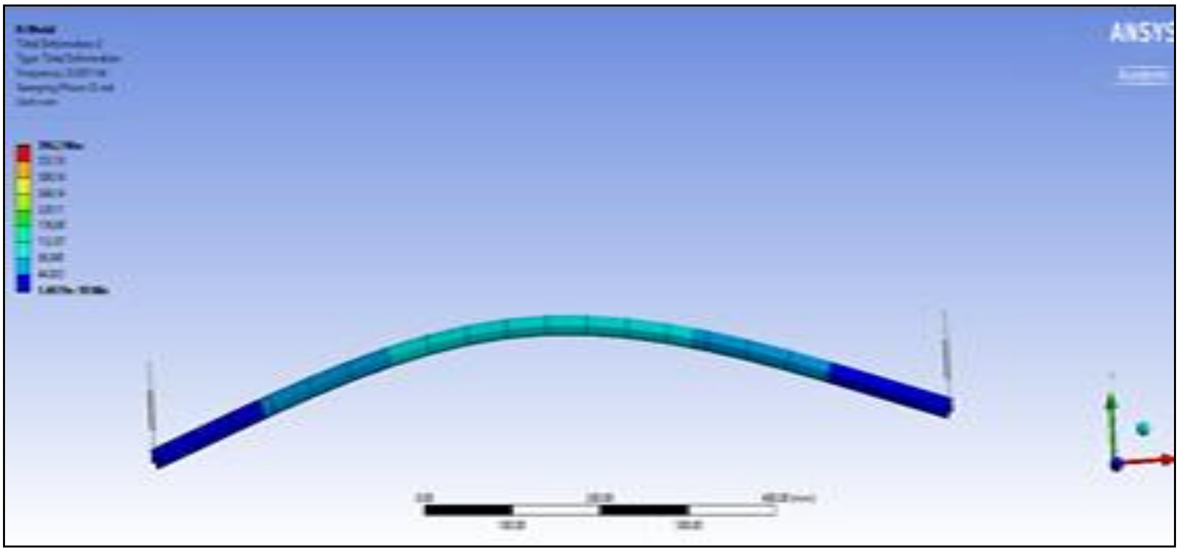
Table 3 Comparison of Experimental and Numerical Natural Frequencies

Number of Frequencies	Natural frequency (Hz)		Error (%)
	Experimental	Numerical	
1	23.75	23.611	0.59
2	24.19	24.083	0.44
3	166.32	171.13	2.89
4	168.17	172.1	2.34
5	327.51	327.33	0.05
6	331.32	328.49	0.85
7	498.23	498.02	0.04
8	502.8	499.04	0.75

The close agreement confirms that the shaft modeled using Beam188 elements together with concentrated disc mass and equivalent bearing stiffness provides an efficient representation of the actual rotor system. The results further indicate that line body elements significantly reduce computational effort while maintaining high prediction accuracy.

➢ *Mode Shape Analysis*

The first eight vibration modes obtained from the modal analysis are shown in Figures 4(a)-(d) and 5 (a)-(d). Since the rotor is supported by identical bearings, the natural frequencies appear as closely spaced pairs corresponding to vibrations in the horizontal and vertical planes. The first two modes correspond to the fundamental bending vibration of the rotor in two mutually perpendicular planes, with maximum deformation occurring near the central disc location. These two modes possess nearly identical natural frequencies due to the geometric symmetry of the rotor-bearing system. The third and fourth modes represent the second bending modes, exhibiting one intermediate node along the shaft length. Higher bending stiffness results in a substantial increase in natural frequency compared to the first bending modes. The fifth and sixth modes correspond to higher-order bending vibrations characterized by multiple inflection points along the shaft. The seventh and eighth modes exhibit more complex deformation patterns with increased curvature and additional nodes. These higher modes become important for high-speed rotor operation because they may be excited when the operating speed approaches their corresponding critical speeds.



(a)

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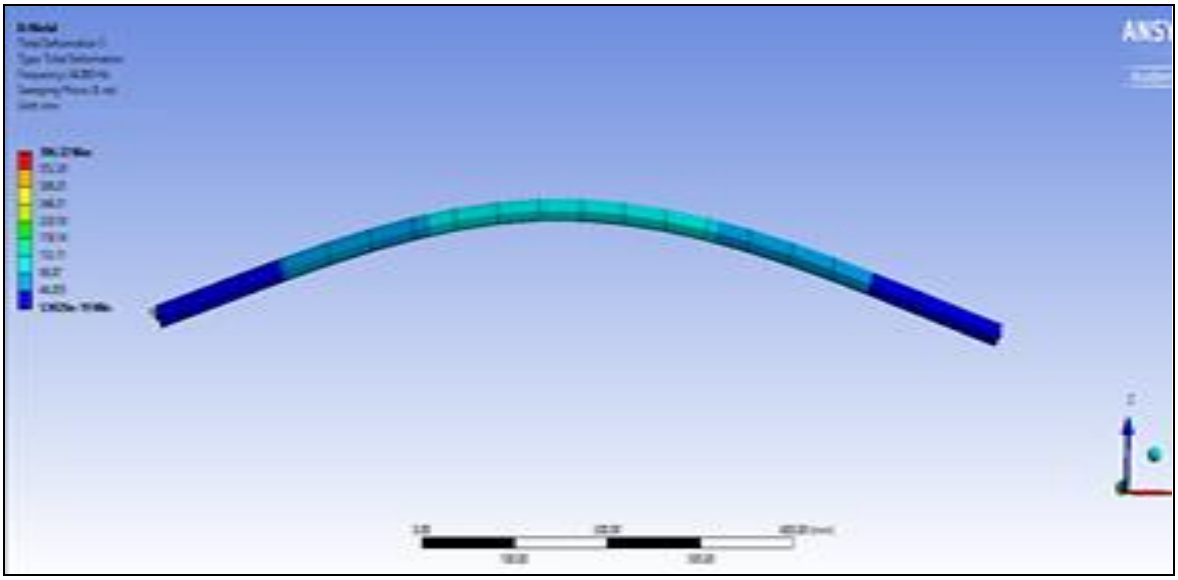
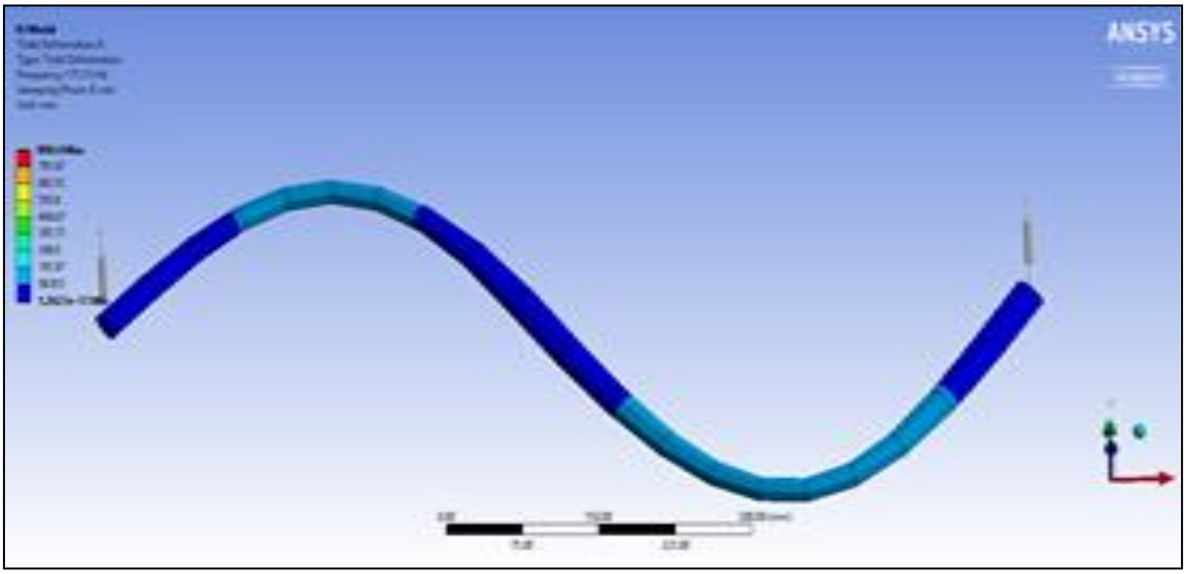
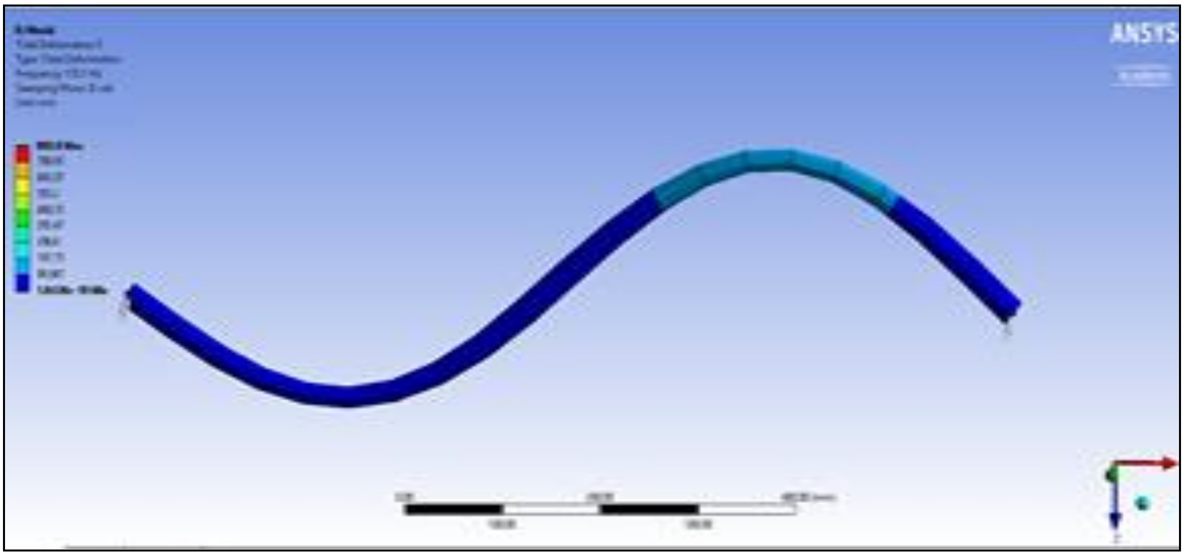


(b)



(c)



(d)

Fig 4 First four Mode Shapes of the Non-Spinning Rotor System

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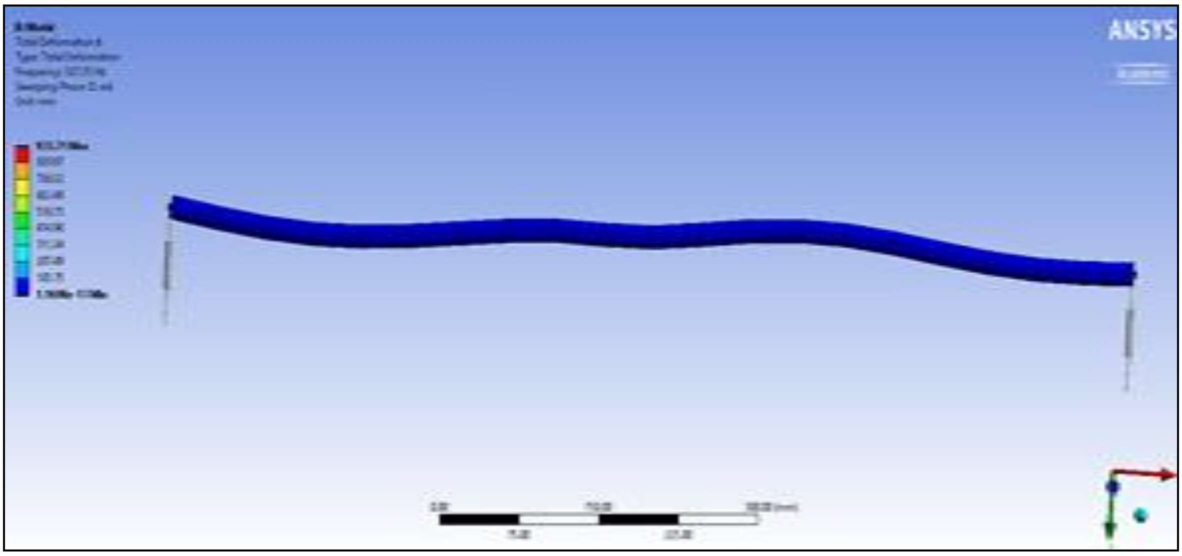
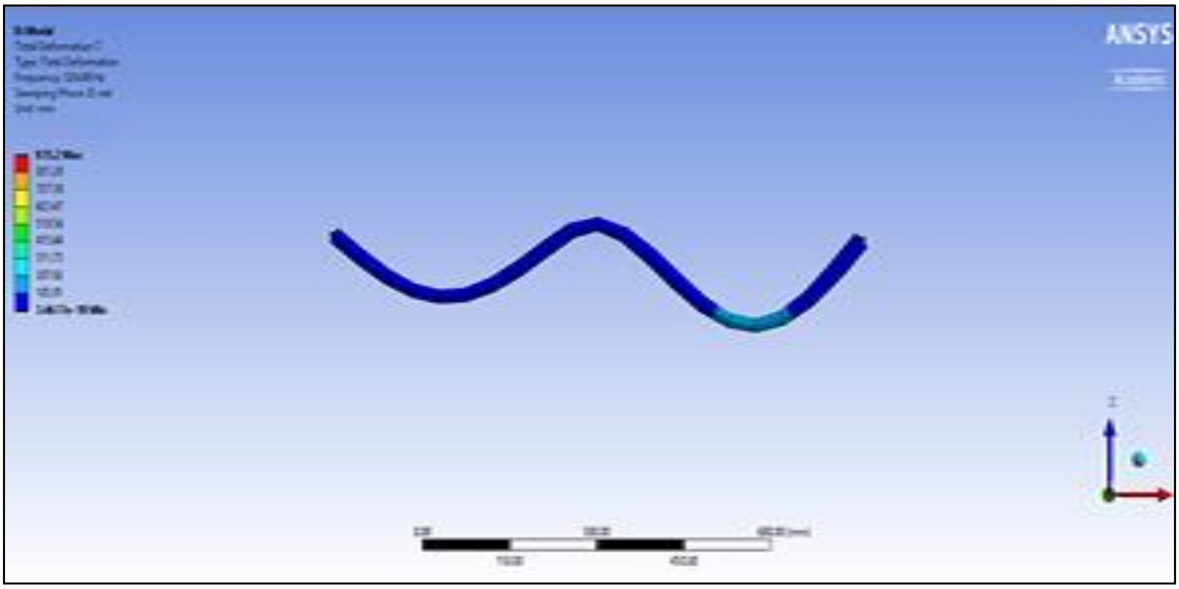
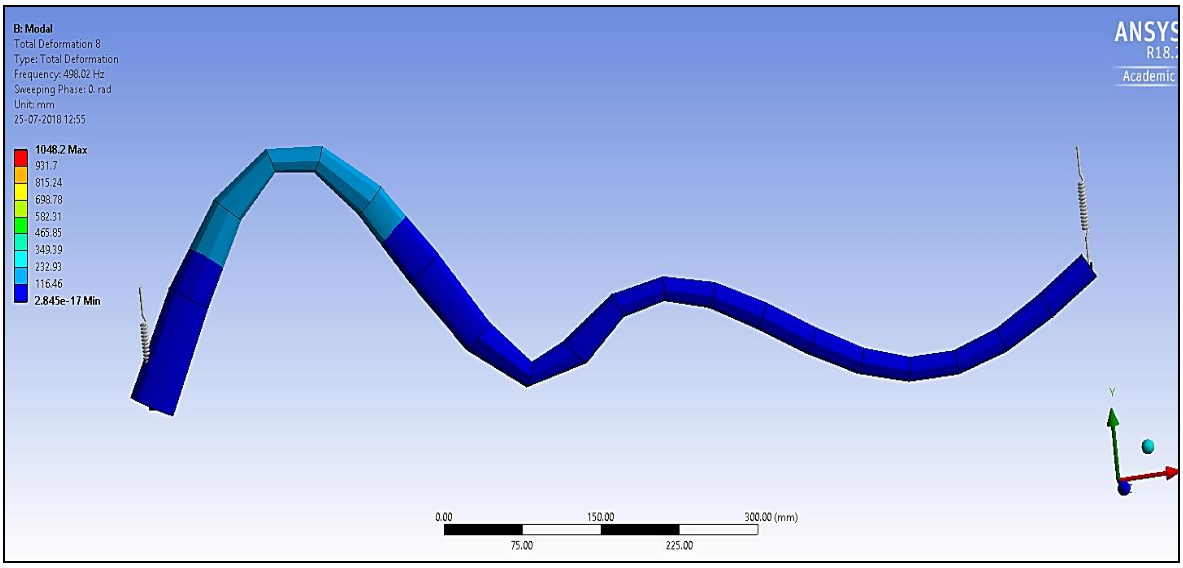


(a)



(b)



(c)

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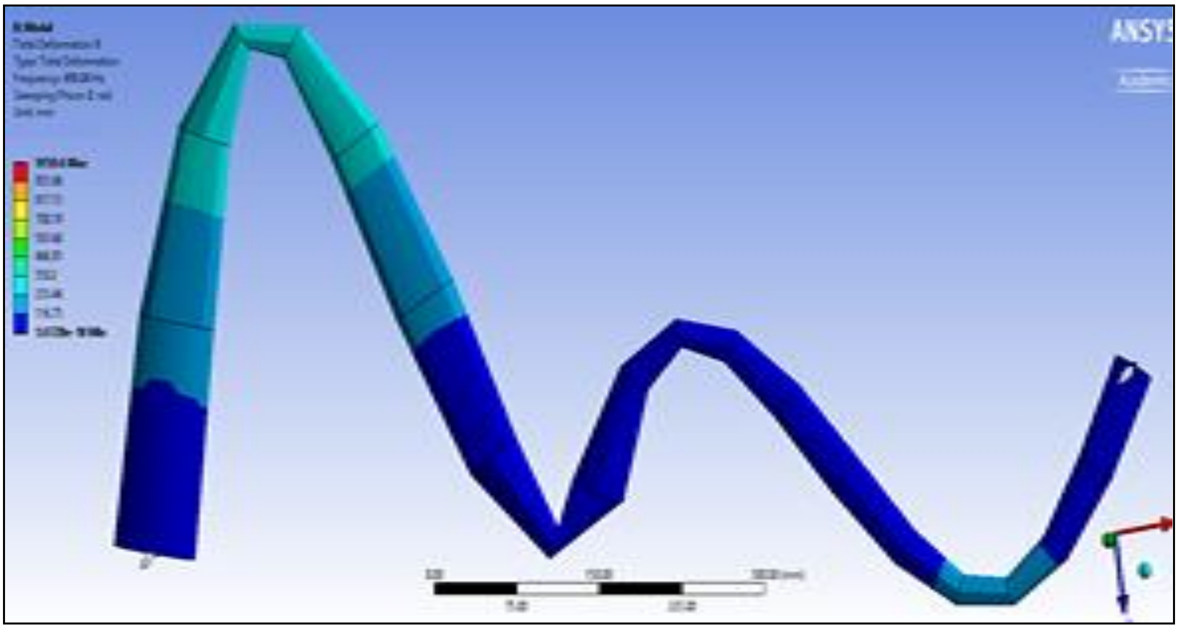

(d)
Fig 5 Higher Mode Shapes of the Non-Spinning Rotor System

Overall, the obtained mode shapes are physically consistent with classical beam vibration theory and agree well with experimentally observed vibration patterns reported in the literature.

- *Campbell Diagram Analysis*

The variation of natural frequencies with rotor rotational speed is illustrated through the Campbell diagrams shown in Figures 6–9. The Campbell diagram is an essential tool in rotor dynamic analysis because it identifies the critical speeds at which resonance may occur.

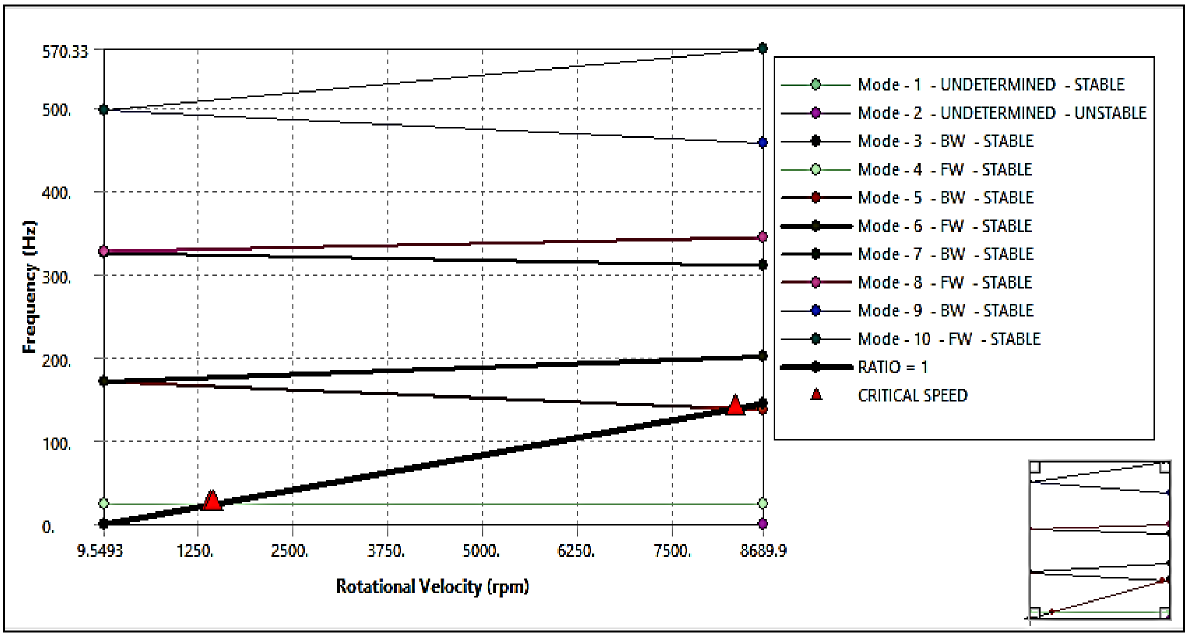

Fig 6 Campbell Diagram Showing Variation of Natural Frequencies with Rotational Speed

As the rotational speed increases, the natural frequencies split into forward whirl (FW) and backward whirl (BW) branches due to the gyroscopic effect introduced by the rotating disc. The forward whirl frequencies increase with increasing shaft speed, whereas the backward whirl frequencies decrease gradually. This behavior agrees well with the theoretical characteristics of flexible rotor systems.

The intersection of the synchronous excitation line (1× operating speed) with the forward whirl branch identifies the rotor critical speed. From the Campbell diagram, the first critical speed is approximately 1416 rpm, indicating the speed at which resonance is expected. Therefore, practical operating speeds should avoid this region or pass through it rapidly during machine startup and shutdown.

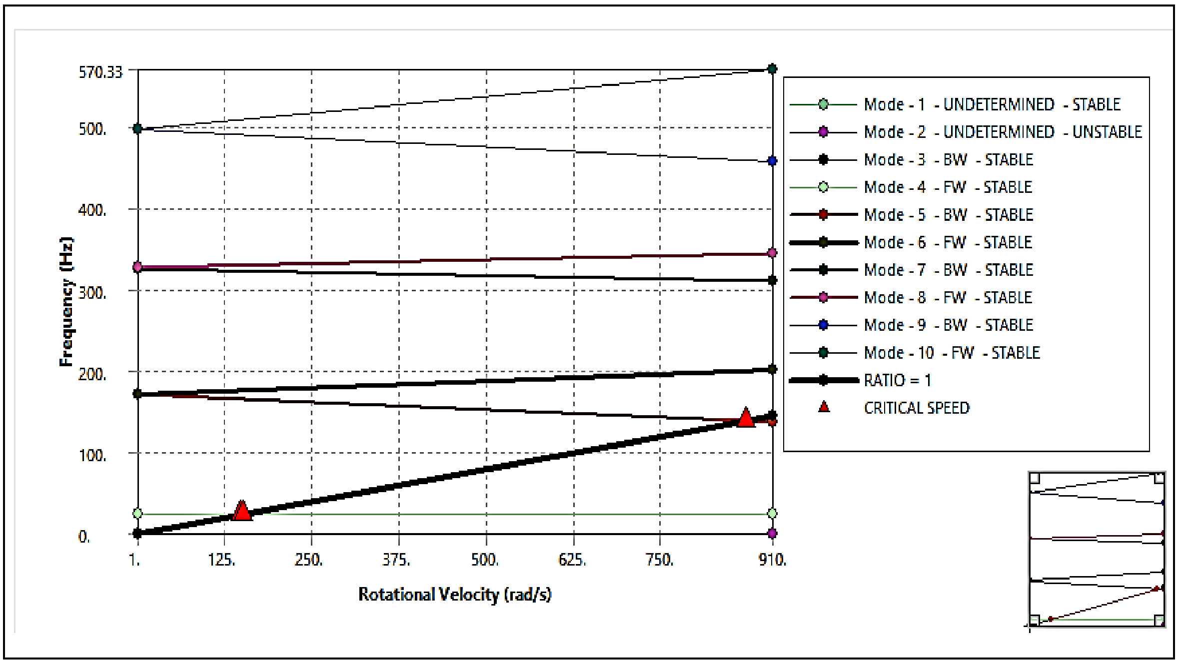

Fig 7 Campbell Diagram in Frequency Domain

The frequency-domain Campbell diagram further confirms the progressive separation of forward and backward whirl frequencies as rotational speed increases. The gyroscopic effect becomes increasingly significant at higher operating speeds, resulting in a noticeable divergence of the paired natural frequencies.

The logarithmic decrement obtained from the complex eigenvalue analysis is presented in Figure 8. The damping values remain nearly constant over the investigated speed range, indicating stable dynamic behavior of the rotor-bearing system.

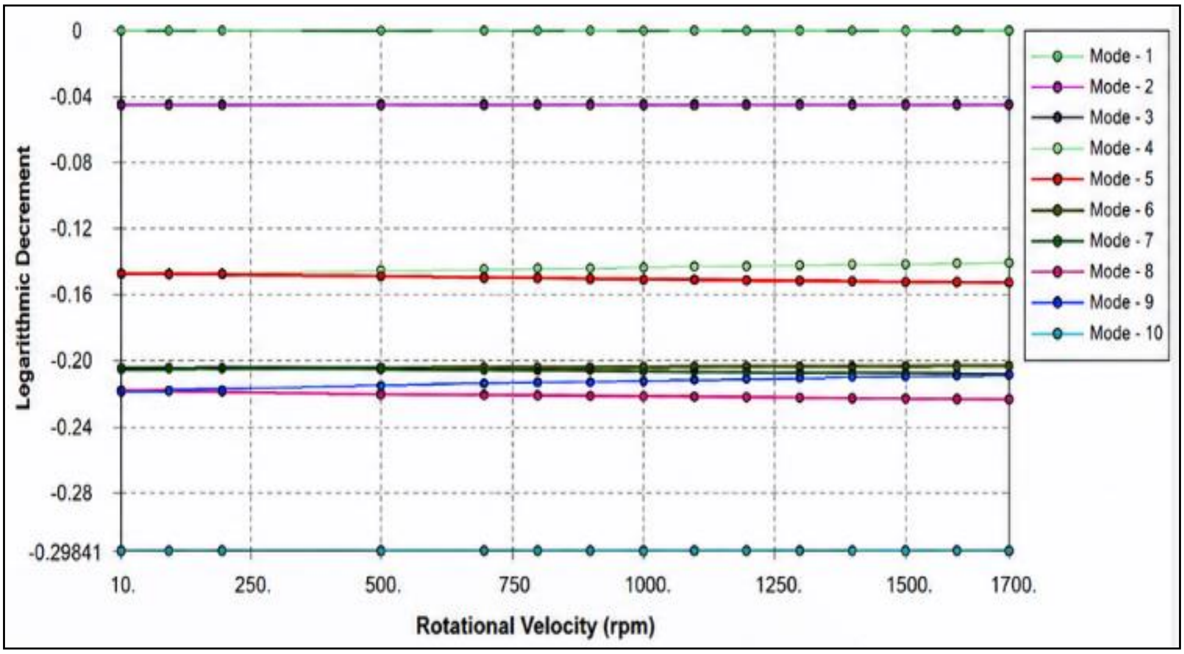

Fig 8 Campbell Logarithmic Diagram

The negative logarithmic decrement values indicate that all forward and backward whirl modes remain dynamically stable within the operating speed range considered in this study. No unstable whirl modes or self-excited vibration characteristics are observed, demonstrating that the selected bearing stiffness and damping parameters provide sufficient dynamic stability.

The Campbell diagram expressed in terms of rotational speed is shown in Figure 9. The intersection between the synchronous excitation line and the first forward whirl mode occurs at approximately 1416 rpm, representing the first critical speed of the rotor system.

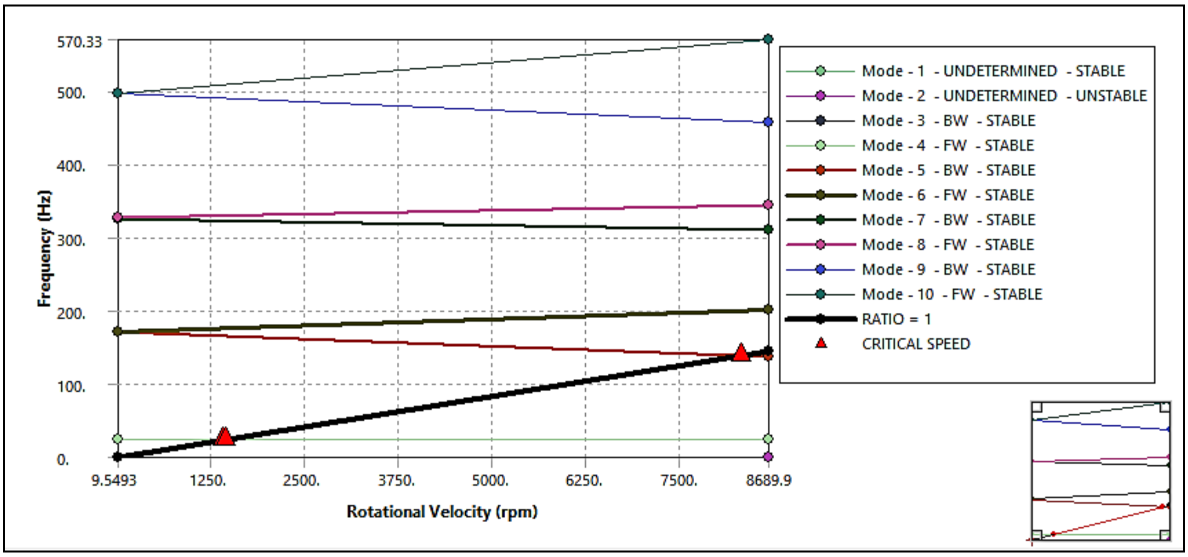

Fig 9 Campbell Logarithmic Diagram

The predicted critical speed agrees well with the expected dynamic behavior of the experimental rotor-bearing system reported in the literature. This agreement further validates the effectiveness of the proposed line body finite element model in predicting rotor dynamic characteristics while significantly reducing modeling complexity and computational cost compared with conventional three-dimensional solid element models.

The results demonstrate that the proposed line body finite element model developed in ANSYS Workbench accurately predicts the modal characteristics of the rotor-bearing system. The maximum error in natural frequency prediction is less than 3%, confirming excellent agreement with experimental measurements. The obtained mode shapes, frequency pairing, and gyroscopic splitting observed in the Campbell diagrams are consistent with classical rotor dynamic theory. Furthermore, the predicted first critical speed of approximately 1416 rpm closely matches the experimentally observed behavior. These findings establish that the Beam188 line body modeling approach provides a computationally efficient and reliable alternative to detailed three-dimensional solid modeling for rotor dynamic analysis, making it well suited for preliminary design, optimization, and dynamic performance evaluation of rotating machinery.

IV. SUMMARY AND CONCLUSIONS

A finite element model of a ball-bearing-supported rotor system was developed in ANSYS Workbench using line body (Beam188) elements to investigate its modal characteristics. The shaft was modeled using beam elements, the disc was represented as a concentrated point mass, and the bearing supports were modeled using equivalent translational and rotational stiffness and damping coefficients. The numerical model was employed to determine the natural frequencies and mode shapes of the rotor system, and the results were validated against experimental data reported in the literature. Campbell diagrams were generated to study the variation of natural frequencies with rotational speed and to identify the critical speed of the rotor system. The numerical predictions showed excellent agreement with the experimental results, demonstrating that the proposed line body modeling approach provides an accurate and computationally efficient alternative to conventional three-dimensional solid element modeling for rotor dynamic analysis.

The major conclusions of the present study are summarized as follows:

- A finite element model of a ball-bearing-supported rotor system was successfully developed in ANSYS Workbench using Beam188 line body elements.
- The predicted natural frequencies agreed well with the experimental results, with a maximum deviation of less than 3%.
- The computed mode shapes accurately represented the bending behavior of the rotor system and validated the proposed modeling approach.
- Campbell diagram analysis successfully identified the forward and backward whirl modes and predicted the first critical speed at approximately 1416 rpm.
- The logarithmic decrement results confirmed the dynamic stability of the rotor system over the investigated speed range.
- The line body finite element model effectively incorporated gyroscopic effects, shaft material damping, and bearing stiffness and damping characteristics.
- Compared with three-dimensional solid element models, the proposed approach significantly reduced computational effort while maintaining high prediction accuracy.
- The developed methodology provides a simple, accurate, and computationally efficient tool for the modal analysis and preliminary design of rotor-bearing systems and can be extended to more complex rotating machinery.

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