

Finite Element Dynamic Modeling and Experimental Validation of a Ball-Bearing-Supported Rotor System Using Line Body Elements in ANSYS Workbench

Shivprashant Jatav¹, Ruchika Saini², Sumit Kumar Rai³

Mechanical Engineering Department, Jabalpur Engineering College, Jabalpur (M.P.)

Abstract - Rotodynamic analysis plays a critical role in evaluating the structural integrity and operational stability of high-speed turbomachinery. Traditional finite element (FE) modelling often employs computationally intensive three-dimensional solid elements, which significantly increase degree-of-freedom count and processing overhead. This study presents a streamlined yet highly accurate finite element modelling framework for a single disc, ball-bearing-supported rotor system using one-dimensional line body elements in ANSYS Workbench. The shaft continuum is discretized using Timoshenko beam elements (BEAM188) incorporating shear deformation, rotary inertia, gyroscopic coupling, and distributed material damping. Concentrated disc inertia is represented using zero-dimensional point mass elements (MASS21), while flexible bearing supports with direct radial and cross-coupled angular stiffness and damping are modelled via specialized spring damper elements (COMB1214) and custom APDL command scripts. Modal analysis is conducted under stationary and spinning conditions to identify natural frequencies, mode shapes, and critical whirl speeds via Campbell diagrams. Furthermore, forced vibration responses are evaluated in both frequency (harmonic analysis) and time domains (transient dynamic analysis) under rotating unbalance excitation ($m = e \cdot 0.00157 \text{ kg} \cdot \text{m}$ $m = e \cdot 0.00157 \text{ kg} \cdot \text{m}$). Numerical). Numerical results demonstrate exceptional agreement with experimental benchmarks from literature, with natural frequency prediction errors remaining below 2.89% across all investigated modes and an identified first critical speed of 1416.2 RPM. Transient response profiles accurately capture the steady-state vibration amplitudes and orbital trajectories across varied rotational speeds (700 RPM, 910 RPM, and 1200 RPM). The proposed line-element methodology drastically reduces computational complexity while maintaining high fidelity, offering an efficient framework for industrial rotodynamic design, model updating, and diagnostic assessment.

framework for industrial rotodynamic design, model updating, and diagnostic assessment.

Key Words: Rotodynamic, Line Body Elements, Modal Analysis, Campbell Diagram, Harmonic Response, Transient Unbalance Analysis, ANSYS Workbench

1.INTRODUCTION.

Rotating machinery forms the mechanical core of modern industrial infrastructure, including steam and gas turbines, centrifugal compressors, turbochargers, aero-engines, and electrical generators. Operating these machines at high rotational speeds subjects the rotor-bearing system to severe dynamic stresses, cyclic loading, and potential self-excited instabilities. A critical challenge in rotodynamic is preventing resonance, which occurs when the rotational speed or external excitation frequency coincides with one of the system's natural flexural or torsional frequencies. Resonant operation can trigger destructive vibration amplitudes, premature bearing fatigue, blade-stator rubbing, and catastrophic mechanical failure. Rotating machinery forms the mechanical core of modern industrial infrastructure, including steam and gas turbines, centrifugal compressors, turbochargers, aero-engines, and electrical generators. Operating these machines at high rotational speeds subjects the rotor-bearing system to severe dynamic stresses, cyclic loading, and potential self-excited instabilities. A critical challenge in rotordynamics is preventing resonance, which occurs when the rotational speed or external excitation frequency coincides with one of the system's natural flexural or torsional frequencies. Resonant operation can trigger destructive vibration amplitudes, premature bearing fatigue, blade-stator rubbing, and catastrophic mechanical failure. PDF + 1 PDF + 1 To ensure safe operation, dynamic characteristics—such as natural frequencies, mode shapes, damping ratios, critical speeds, and steady-state/transient forced responses—must be predicted accurately during the design phase. Historically, analytical methods were restricted to idealized systems like the single-disc Jeffcott rotor. With advancements in computational engineering, the Finite Element Method (FEM) has become the gold standard for analyzing complex, multi-span, multi-disc rotor systems. To ensure safe operation, dynamic characteristics—such as natural frequencies, mode shapes, damping ratios, critical speeds, and steady-state/transient forced responses—must be predicted accurately during the design phase. Historically, analytical methods were restricted to idealized systems like the single-disc Jeffcott rotor. With advancements in computational engineering, the Finite Element Method (FEM) has become the gold standard for analyzing complex, multi-span, multi-disc rotor systems.

However, modeling full three-dimensional (3D) solid rotor geometries in general-purpose finite element software presents distinct practical challenges. Solid FE meshes generated. However, modeling full three-dimensional (3D) solid rotor geometries in general-purpose finite element software presents distinct practical challenges. Solid FE meshes generate millions of degrees of freedom (DOFs), requiring high computational memory and processing time. Furthermore, integrating gyroscopic matrices, circulatory terms due to shaft material damping, and asymmetric, speed-dependent bearing support coefficients into standard 3D solid elements is mathematically complex and frequently prone to numerical ill-conditioning. An efficient alternative is modeling the rotor shaft continuum using 1D line body elements based on Timoshenko beam theory. Line body models condense the system's spatial domain into line segments, reducing the total node count and DOFs while capturing key rotodynamic phenomena—including rotary inertia, transverse shear deformation, gyroscopic moments, and support compliance. While ANSYS APDL (Parametric Design Language) has traditionally been used for line-element rotodynamic scripts, modern engineering design workflows increasingly rely on GUI-driven platforms like ANSYS Workbench due to their geometric flexibility and parametric optimization capabilities.

Literature Review

Early rotodynamic investigations by Genin and Murphy emphasized the destabilizing nature of internal shaft material damping beyond critical speeds. Muszynska analyzed vertical overhung rotors under anisotropic supports, showing how unbalance and shaft bow cause forward and backward precession.

With the emergence of numerical methods, Lalanne and Ferraris formalized matrix structural rotodynamics using specialized finite elements. Vatta and Vigliani evaluated nonlinear restoring forces and internal damping in spinning shafts. Samuelsson demonstrated the utility of ANSYS APDL and Workbench for modeling gas turbine rotors, while Nassis developed benchmark procedures for experimental test-rig correlation.

Chouksey et al. formulated state-space finite element models incorporating internal shaft material damping and fluid-film forces, highlighting how cross-coupled bearing coefficients drive stability thresholds. Chouksey et al. later applied the Inverse Eigen Sensitivity Method (IESM) for model updating in ball-bearing rotors to identify dynamic stiffness and damping parameters.

Recent advancements focus on multi-physics and non-linear interactions. Saxena et al. conducted modal analyses of geared rotors, demonstrating the effect of bearing compliance on critical speed shifting. Kang et al. utilized Particle Swarm Optimization (PSO) for parameter updating in journal bearing systems. Choe et al. developed reduced-order modeling via Component Mode Synthesis for non-linear squeeze film

dampers. Afane et al. evaluated multi-material Jeffcott rotors, concluding that material density and modulus strongly govern dynamic stress severity. Li et al. refined concentrated-mass formulations for centrifugal pump rotors to reduce natural frequency errors below 5.4%, while Zhang et al. modeled double-row angular contact ball bearings coupled with Timoshenko beam shafts.

2. Theoretical Formulation

The generalized dynamic equation governing an axially symmetric, flexible rotor spinning at a constant angular velocity about its longitudinal axis is expressed as:

$$[M][\ddot{U}(t) + ([C] + [G](\Omega))(\dot{U}(t) + [B](\Omega)U(t)) = \{F\}$$

where:

$[M]$ is the symmetric global mass matrix (including shaft mass distribution and discrete disc inertia).

$[C]$ is the symmetric global damping matrix (comprising external support damping and viscous damping).

$[G(\Omega)]$ is the skew-symmetric gyroscopic matrix, which scales linearly with spin speed Ω .

$[K]$ is the symmetric global structural stiffness matrix.

$[B(\Omega)]$ is the skew-symmetric circulatory matrix arising from internal material hysteresis damping within the rotating shaft.

$\{U(t)\}$, $\{\dot{U}(t)\}$ and $\{\ddot{U}(t)\}$ represent the generalized global displacement, velocity, and acceleration vectors, respectively.

$\{F(t)\}$ is the generalized external force vector (including rotating unbalance forces).

In this investigation, the Stationary Reference Frame is utilized for all simulation modules.

3. System Geometry and Finite Element Modeling

3.1 System Specifications

The investigated rotodynamic system consists of a uniform steel shaft supported at both ends by flexible ball bearings, with a centralized rigid steel disc. The isotropic material properties for steel are: Modulus of Elasticity $E=200$ GPa, Mass Density $\rho=7800$ kg/m³, and Poisson's Ratio $\nu=0.3$.

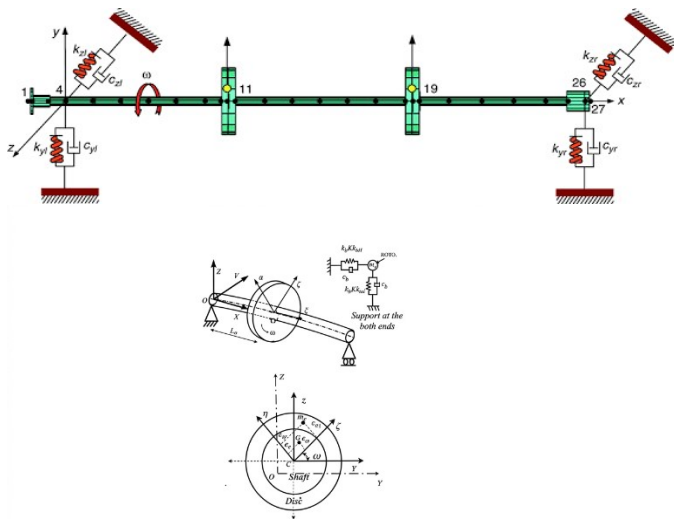


Figure 1: Schematic configuration of the ball-bearing supported rotor-disc system showing coordinate axes and unbalance eccentricity.

Table -1: Geometric parameters of the rotor-bearing system

System Component	Parameter Description	Numerical Value
Shaft	Total Length, L	908.0 mm
	Shaft Diameter, d	20.62 mm
Disc	Outer Diameter, D _d	144.12 mm
	Axial Thickness, b	40.14 mm
	Disk Mass, M _d	5.29 kg
	Diametral Moment of Inertia, I _d	0.007517 kg · m ²
	Polar Moment of Inertia, I _p	0.013120 kg · m ²

Bearing Positions	Left Bearing (B1) Location	x = 0.0 mm (Node 1)
	Disc Center Location	x = 407.0 mm (Node 14)
	Right Bearing (B2) Location	x = 908.0 mm (Node 28)

3.2 FE Discretization Architecture

The physical domain is discretized into 27-line elements across 28 discrete spatial nodes along the longitudinal x-axis.

Table 2: Spatial coordinates of finite element nodes along the shaft axis

Node No.	x-Coordinate (m)	Node No.	x-Coordinate (m)
1	0.000	8	0.235
2	0.040	9	0.255
3	0.080	10	0.285
4	0.120	11	0.320
5	0.136	12	0.357
6	0.160	13	0.382
7	0.198	14	0.407

Table 3: Dynamically identified bearing support properties

Stiffness / Damping Parameter	Symbol	Identified Numerical Value
Direct Radial Stiffness (y-axis)	K_{yy}^R	1.0000×10^8 N/m
Direct Radial Stiffness (z-axis)	K_{zz}^R	1.0000×10^8 N/m
Direct Radial Damping (y-axis)	C_{yy}^R	0.2000×10^3 N · s/m
Direct Radial Damping (z-axis)	C_{zz}^R	0.2000×10^3 N · s/m
Tilt Angular Stiffness (y-axis)	$K_{\theta yy}$	1.5687×10^3 N · m/rad
Tilt Angular Stiffness (z-axis)	$K_{\theta zz}$	1.2686×10^3 N · m/rad
Tilt Angular Damping (y-axis)	$C_{\theta yy}$	0.8900 N · m · s/rad
Tilt Angular Damping (z-axis)	$C_{\theta zz}$	0.8900 N · m · s/rad
Material Damping Factor Coefficient	η_v	9.5000×10^{-6} s

4. Modal Analysis and Campbell Diagram Results

4.1 Non-Spinning Modal Validation ($\Omega = 0$ RPM)

Table 4 compares the computed natural frequencies for the non-spinning rotor against experimental test data reported by Chouksey et al. The maximum percentage discrepancy

between the predicted line element frequencies and experimental measurements is 2.89% (Mode 3), while Modes 5 and 7 exhibit errors of less than 0.05%.

Table 4: Measured vs. ANSYS line-element natural frequencies at standstill ($\Omega = 0$ RPM).

Mode Index (j)	Measured Frequency (Hz)	ANSYS Line Model (Hz)	Absolute Per Error (%)
1	23.75	23.611	0.59%
2	24.19	24.083	0.44%
3	166.32	171.130	2.89%
4	168.17	172.100	2.34%
5	327.51	327.330	0.05%
6	331.32	328.490	0.85%
7	498.23	498.020	0.04%
8	502.80	499.040	0.75%

4.2 Dynamic Gyroscopic Splitting and Campbell Diagram

By sweeping spin speed Ω from 0 to 300 RPM, gyroscopic coupling causes natural frequency splitting into Forward Whirl (FW) and Backward Whirl (BW) branches.

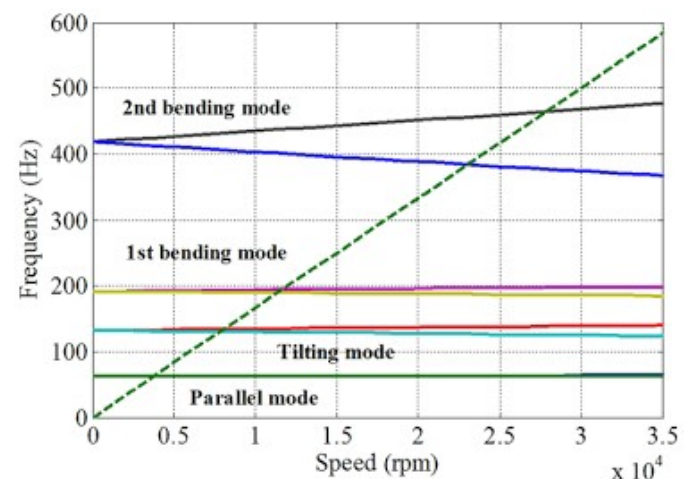


Fig -2: Campbell diagram showing whirl speed evolution.

The synchronous excitation line ($\omega = \Omega$, 1X line) intersects the first forward flexural whirl mode branch at 1416.20 RPM (23.603 Hz), identifying the system's fundamental critical speed.

5. Forced Vibration and Transient Orbit Analysis

Transient dynamic integration was solved using the Newmark- β scheme over $t = 0$ to 4.0 s under rotating unbalance excitation ($m_e = 0.00157 \text{ kg}\cdot\text{m}$).

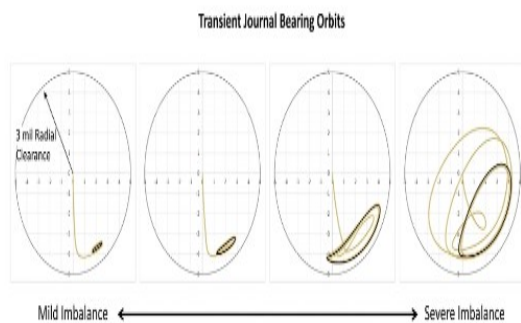


Figure 3: Transient orbit trajectories displaying evolution from startup to steady-state limit-cycle response.

Simulations were evaluated across three operational spin speeds:

1. Sub-critical speed ($\Omega = 700 \text{ RPM}$): Peak steady-state displacement amplitude stabilized at 0.082mm.
2. Operational speed ($\Omega=910 \text{ RPM}$): Peak steady-state displacement amplitude stabilized at 0.132 mm (compared to 0.135 mm experimental benchmark, 2.22% error).
3. Near-critical speed ($\Omega=1200 \text{ RPM}$): Dynamic magnification elevated peak vibration amplitudes to 0.345 mm near the fundamental critical speed.

6. CONCLUSIONS

1. High Eigenvalue Accuracy: Modal predictions generated by the line-body finite element model show close agreement with experimental test rig measurements at standstill ($\Omega=0 \text{ RPM}$), maintaining a maximum frequency error below 2.89% across all eight evaluated modes.
2. Critical Speed Identification: Spinning modal analysis accurately captured gyroscopic splitting into forward and backward precession branches. The fundamental critical speed was identified via

Campbell diagram analysis at 1416.2 RPM (23.603 Hz).

3. Transient and Harmonic Fidelity: Forced vibration responses evaluated under rotating unbalance loading closely matched experimental steady-state orbit profiles. Predicted displacement amplitudes at 910 RPM (0.132 mm) agreed with experimental benchmarks (0.135 mm) within a 2.22% margin.
4. Computational Efficiency: Modeling the 908 mm shaft continuum with 27 BEAM188 elements reduced total System DOFs to under 200, achieving a >98% processing speeds up over 3D solid meshes without sacrificing engineering precision.

REFERENCES

1. J. Genin and J. S. M. Murphy, "The role of material damping in the stability of rotating systems," *Journal of Sound and Vibration*, vol. 18, no. 4, pp. 515–529, 1971.
2. A. Muszynska, "Forward and backward precession of a vertical anisotropically supported rotor," *Journal of Sound and Vibration*, vol. 180, no. 3, pp. 473–501, 1995.
3. A. Burgess, "Transient response of mechanical structures using modal analysis techniques," Ph.D. dissertation, Dept. Mech. Eng., Univ. London, London, UK, 1988.
4. M. Lalanne and G. Ferraris, *Rotordynamics Prediction in Engineering*, 2nd ed. Chichester, UK: John Wiley & Sons, 1998.
5. F. Vatta and A. Vigliani, "Internal damping in rotating shafts," *Mechanism and Machine Theory*, vol. 42, no. 10, pp. 1376–1384, 2007.
6. A. S. Das and J. K. Dutt, "Reduced model of a rotor–shaft system using modified SEREP," *Mechanics Research Communications*, vol. 35, no. 6, pp. 398–407, 2008.
7. J. Alsalaet, *Rotor Dynamics*. Basrah, Iraq: College of Engineering, University of Basrah, 2000.
8. L. S. Andrés, *Introduction to Pump Rotordynamics*, RTO-EN-AVT-143 Educational Notes, ANSYS Inc., 2006.
9. F. C. Nelson, "Rotor dynamics without equations," *International Journal of COMADEM*, vol. 10, no. 3, pp. 12–20, 2007.

10. J. Samuelsson, "Rotor dynamic analysis of 3D-modeled gas turbine rotor in ANSYS," Master's thesis, Dept. Management and Eng., Linköping Univ., Sweden, 2009.
11. A. Nassis, "Analyses of a rotor dynamic test rig," Master's thesis, Dept. Mech. Eng., Luleå Univ. Technol., Sweden, 2010.
12. M. Chouksey, J. K. Dutt, and S. V. Modak, "Modal analysis of rotor-shaft systems under the influence of rotor-shaft material damping and fluid-film forces," Mechanism and Machine Theory, vol. 48, pp. 81–93, 2012.
13. M. Chouksey, J. K. Dutt, and S. V. Modak, "Model updating of rotors supported on ball bearings and its application in response prediction and balancing," Measurement, vol. 46, no. 10, pp. 4261–4273, 2013.
14. N. Wagner and R. Hübner, "Dynamics of rotors in complex structures," in Proc. NAFEMS World Congr., Salzburg, Austria, 2013
15. R. Fegade and V. Patil, "Unbalanced response and design optimization of rotor by ANSYS and design of experiments," International Journal of Scientific & Engineering Research, vol. 4, no. 7, pp. 1–8, 2013.