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Modeling and Analysis of Interactions in the Rotor–Bearing–Housing System under Variable Loads

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Abstract. Vibration arising in industrial settings significantly influences the durability and functionality of machinery, particularly those operating under high speeds and heavy loads. Despite ongoing technological improvements, sectors such as textiles continue to experience intensified vibration, impulsive noise, and broadened frequency emissions due to the integration of more potent machinery. Devices like shuttle looms and cotton-processing equipment exemplify systems that contribute to vibrational challenges. This research explores the dynamics of the "rotor–bearing–housing" assembly as a key factor in mitigating such issues. Given that rolling bearings frequently develop defects that compromise mechanical performance, the study introduces an advanced dynamic model considering deformation, misalignment, and fluctuating contact stresses under vibration. Emphasis is placed on the influence of bearing design, stiffness, and applied loads on overall system stability. The paper also reviews modern vibroacoustic diagnostic techniques for early failure detection and service life forecasting, promoting more efficient and resilient vibration control solutions in industrial environments.

Keywords. Shaft bending, saw cylinder, seed roller, shaft deformation, linter machine, transverse force, bending moment, equations of equilibrium, support reaction.

Introduction

In real industrial practice, vibration is one of the main factors that quietly but steadily reduces the reliability and service life of equipment. Even with the adoption of modern, high-speed and energy-efficient machines, the problem has not disappeared. On the contrary, more powerful drives and higher speeds often create stronger oscillations, wider frequency bands, and additional impulsive noise. These effects are especially noticeable in textile production, where machines such as looms, combing systems, and cotton-gin units operate almost continuously under heavy and variable loads.

Such conditions make it clear that conventional design measures are no longer sufficient. What is needed are advanced diagnostic tools and vibration-control strategies that are both technically sound and practical for industry. Among the most sensitive components are rolling bearings, which serve as critical supports for rotating shafts. Even small surface defects or misalignments can increase vibration levels dramatically, leading to accelerated wear and reduced safety. For this reason, the development of refined dynamic models of the rotor-bearing-housing system has become an important research direction. These models not only support more accurate detection of faults but also provide guidance for designing vibration-resistant structures.

The purpose of this study is to develop and numerically investigate a dynamic model of a rotor-bearing-housing system with elastic-damping (polyurethane) inserts, in order to evaluate their influence under variable loading conditions and to provide engineering recommendations for vibration reduction in textile machinery.

Researchers over several decades have shown that the behavior of bearing assemblies is strongly shaped by imperfections in their geometry and the development of faults. One of the most common ways to study these effects is through analysis of vibration and acoustic signals collected during machine operation. Methods such as Fourier transforms, wavelet analysis, and statistical techniques are widely applied [1-5]. They are useful for identifying early stages of damage, yet in practice their accuracy is limited by strong background noise and the complex, non-stationary form of real signals. Thus, while these tools are valuable, they often fail to give clear information about the actual magnitude and distribution of contact forces in bearings.

To go beyond signal analysis, many researchers have turned to analytical and numerical modeling [6-8]. These works have advanced our understanding of stresses and deformations in bearing elements, but most of them simplify reality by assuming rigid components or static conditions. Such simplifications leave open questions about the true influence of structural flexibility and load variation. Experimental studies on sliding effects in rolling elements are also rare, which means an important part of bearing behavior under real conditions is still not well understood.

In engineering practice, bearing defects are usually grouped into two broad categories: distributed (waviness, misalignment, dimensional deviations) and localized (cracks, spalls, pits) [10-12]. This classification is convenient, but in reality the vibration signatures of different faults can overlap, making diagnostics more difficult. Distributed defects are often the result of manufacturing tolerances or gradual wear, whereas localized damage is mainly caused by fatigue crack initiation and propagation. Localized faults are the most dangerous, because they lead to sudden changes in stress distribution and generate short high-energy impulses. At the same time, many studies do not consider how these fault mechanisms can interact.

More recent investigations have used finite element analysis and advanced simulations to study shaft deflection, contact stresses, and dynamic response under realistic loads [13-16]. Results from these works show that design improvements - such as adding elastomeric inserts or hybrid support structures - can reduce shaft bending, suppress critical vibrations, and extend machine life. However, the benefits strongly depend on the type of machine and its working conditions. In textile equipment, for instance, shafts are slender and heavily loaded, making damping inserts a promising but not yet fully explored solution.

Comprehensive reviews, such as those by McFadden and colleagues [17-18], confirm the importance of vibration-based monitoring for bearing diagnostics. At the same time, they reveal a clear imbalance: most existing models focus on distributed defects, like raceway waviness, while only a few provide detailed simulations of localized faults. This gap shows the need for integrated approaches that combine advanced signal-processing methods with dynamic modeling. Only such combined methods can provide both qualitative and quantitative evaluation of bearing health.

Recent studies emphasize the importance of accurate numerical modeling for predicting stress–strain behavior and structural response under operational loads [7, 19]. In rotor dynamics, the introduction of elastic support layers has been shown to significantly reduce vibration levels and improve system stability [20], highlighting the role of structural modifications in enhancing mechanical reliability [21-22].

Based on these observations, the present study focuses on the influence of elastic-damping inserts in rotor-bearing-housing systems. Unlike many earlier models that assume rigid supports, this work explicitly incorporates polyurethane layers as damping elements and evaluates their role in vibration reduction. This approach not only develops theoretical understanding but also offers practical recommendations for improving the reliability of rotating machinery.

Thus, adequate analysis of the condition of rotor systems with bearings requires comprehensive modeling that takes into account interactions and possible defects, making dynamic models a key tool in the study and prediction of the technical condition of such assemblies.

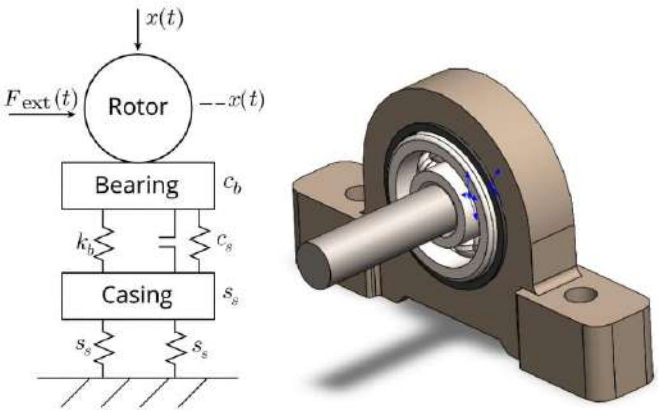


Fig.1. - The system consists of two bodies: the rotor and the housing, connected by elastic-damping elements (spring and damper). Rotor displacement is denoted by x(t); Housing displacement is denoted by x_s(t); An external harmonic force F(t) acts on the rotor, simulating a disturbance

1. Research methodology

To analyze the dynamic response of the system shown in Figure 1, a mathematical model was developed that describes the interaction of components through a system of second-order ordinary differential equations. The model includes the mass of the rotating shaft and the housing, as well as the elastic and damping connections between them and the base. As a result, the system takes the following form:

$$\begin{gathered} m_1\ddot{x}+c_b(\dot{x}-\dot{x}_s)+k_b(x-x_s)=F(t) \\ m_2\ddot{x}_s-c_b(\dot{x}-\dot{x}_s)-k_b(x-x_s)+c_s\dot{x}_s+s_s x_s=0 \end{gathered} \quad (1)$$

where x(t) - rotor displacement;

x_s(t)- displacement of the housing;

$\dot{x}$-velocity (first derivative of displacement with respect to time);

$\ddot{x}$- acceleration (second derivative of displacement with respect to time);

m_1, m_2 -the masses of the rotor and the housing.

The physical and mechanical parameters used in the numerical simulation are summarized in Table 1. These values were selected to represent typical operating conditions of medium-scale textile rotor systems and ensure reproducibility of the computational results.

Table 1. Parameters used in numerical simulation

№	Parameter	Description	Value	Unit	№	Parameter	Description	Value
1	m_1	Rotor mass	25	kg	5	ks	Support stiffness	1.5×10^5
2	m_2	Housing mass	40	kg	6	cb	Bearing damping	850
3	h	Polyurethane thickness	4	mm	7	cs	Support damping	1200
4	kb	Bearing stiffness	10^5	N/m				

For numerical analysis, the given model was transformed into a vector-matrix form:

$$x(t) = \begin{bmatrix} x(t) \\ x_s(t) \end{bmatrix}, \dot{x}(t) = \begin{bmatrix} \dot{x}(t) \\ \dot{x}_s(t) \end{bmatrix}, \ddot{x}(t) = \begin{bmatrix} \ddot{x}(t) \\ \ddot{x}_s(t) \end{bmatrix} \tag{2}$$

$$\mathrm{M}\ddot{x}+\mathrm{C}\dot{x}+\mathrm{Kx}=\mathrm{F(t)} \tag{3}$$

mass matrix:

$$M = \begin{bmatrix} m_1 & o \\ 0 & m_2 \end{bmatrix} \tag{4}$$

damping matrix:

$$\mathrm{C} = \begin{bmatrix} \mathrm{c}_b & -\mathrm{c}_b \\ \mathrm{c}_b & \mathrm{c}_b + \mathrm{c}_s \end{bmatrix} \tag{5}$$

stiffness matrix:

$$\mathrm{K} = \begin{bmatrix} k_b & -k_b \\ -k_b & k_b + s_s \end{bmatrix} \tag{6}$$

vector of external forces:

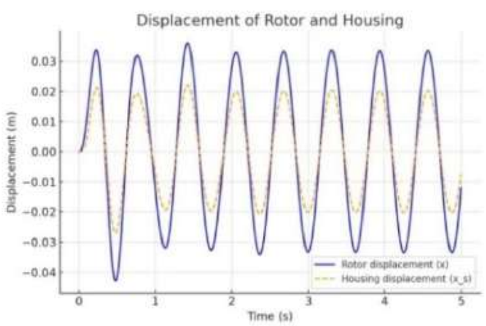
$$F(t) = \begin{bmatrix} F(t) \\ 0 \end{bmatrix} \tag{7}$$

By substituting expressions (1), (3), (4), (5), (6), and (7) into equation (2), the final matrix equation of motion is obtained:

$$\begin{bmatrix} m_1 & o \\ 0 & m_2 \end{bmatrix} \begin{bmatrix} \ddot{x}(t) \\ \ddot{x}_s(t) \end{bmatrix} + \begin{bmatrix} \mathrm{c}_b & -\mathrm{c}_b \\ \mathrm{c}_b & \mathrm{c}_b + \mathrm{c}_s \end{bmatrix} \begin{bmatrix} \dot{x}(t) \\ \dot{x}_s(t) \end{bmatrix} + \begin{bmatrix} k_b & -k_b \\ -k_b & k_b + s_s \end{bmatrix} \begin{bmatrix} x(t) \\ x_s(t) \end{bmatrix} = \begin{bmatrix} F(t) \\ 0 \end{bmatrix} \tag{8}$$

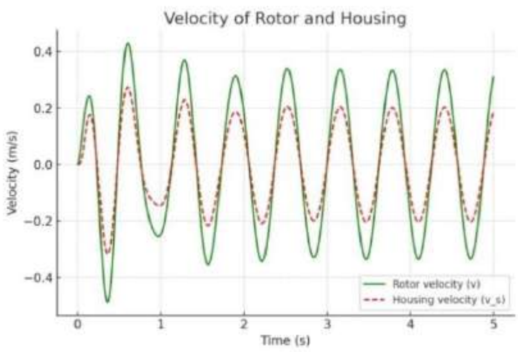
The equations (8) were solved numerically using time discretization. The equations were solved using time discretization and the explicit Euler method. In addition to the harmonic excitation $F(t) = F_0 \sin(\omega t)$, an impulsive load case was also considered in the form of a short-duration force pulse to simulate sudden load variation typical for textile machinery start-up or material impact. The comparison showed that damping inserts significantly reduced peak acceleration under impulsive excitation (by approximately 30%), confirming their effectiveness under variable loading conditions.

The resulting graphs clearly demonstrate the interrelation between structural elements and allow for the identification of vibration characteristics, amplitude response, phase lag, and the level of damping in the system. Such visualization is essential for engineering assessment of the stability, reliability, and vibration activity of the structure.



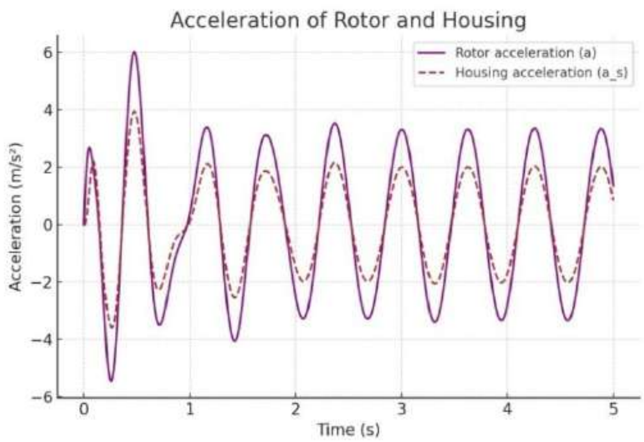
The displacement plot illustrates the relative motion of the rotor and the housing over time under the influence of a harmonic excitation force. The blue curve represents the rotor's response, while the orange dashed line corresponds to the housing. The difference in amplitude and phase between the two elements indicates the elastic interaction and the dynamic decoupling due to the bearing and foundation compliance.

Fig. 2. - Time-dependent displacement of the rotor and the housing



The velocity response of both components is shown in the second graph. As expected, the velocity of the housing is smoother due to the higher mass and damping, while the rotor exhibits more rapid fluctuations. This provides insight into the energy transfer between the rotating shaft and the support structure.

Fig. 3. - Time history of rotor and housing velocity



The graph shows the acceleration time of the rotor and housing in detail. These high-frequency oscillations are especially important when assessing fatigue, impact loads or when designing shock-absorbing systems. The distinct acceleration peaks reflect the smallest moments of force transmission through the flexible elements (polyurethane).

Fig. 4. - Time history of rotor and housing acceleration

2. Results and discussion

To assess the dynamic characteristics of the system, a mathematical representation was constructed considering the elastic links between the bearings and housing. The model, based on Lagrange's equations of the second kind, accounted for internal damping and stiffness forces, along with external influences such as gravity and torque. Polyurethane elements, with a stiffness $k_{pu} = 10^5$ N/m and a thickness of 4 mm, were employed as damping components.

Simulations were executed using Mathcad, and outcomes revealed that incorporating these damping layers reduced the rotor's vibration amplitude by 35-50% on average. The time-domain responses (as seen in Figure 2) demonstrate a notable attenuation of oscillations in the damped configuration. Frequency domain analysis also confirmed diminished amplitude peaks at resonance. In Figure 5, the Y-axis labeled "Amplitude" represents displacement amplitude in meters (m), obtained from Fourier transformation of the steady-state response. Consequently, utilizing polyurethane inserts at the bearing supports effectively minimizes vibrational stresses, enhancing the durability and operational reliability of the entire rotor system.

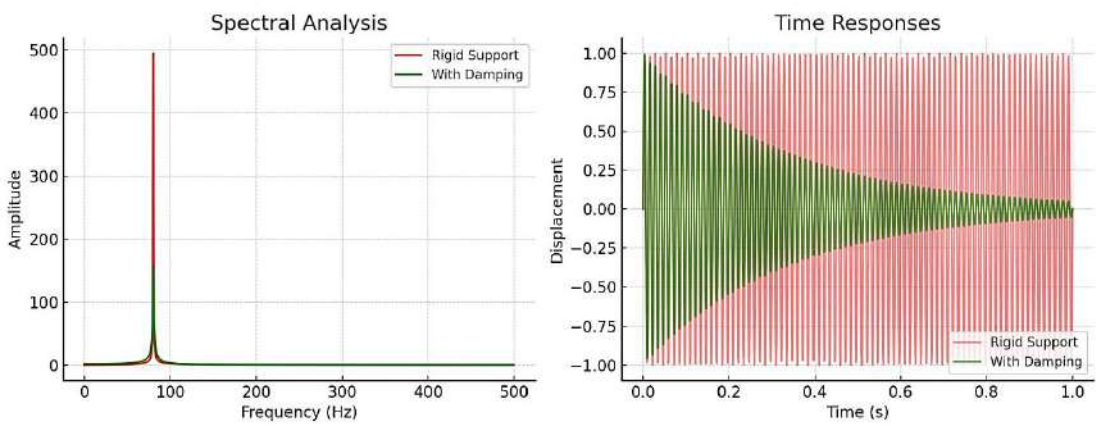


Fig. 5. - Comparison of two Support variants based on modeling results

The left-hand plot presents the results of spectral analysis comparing two configurations of the bearing support system: a rigid mounting and a damping-enhanced support. The spectrum for the rigid support (red) shows a dominant resonance peak at a specific natural frequency, indicating a pronounced response to harmonic excitation. In contrast, the configuration with damping (green) significantly suppresses the amplitude of the resonant peak, effectively broadening the response spectrum and reducing vibrational energy concentration at the critical frequency. This demonstrates the damping system's effectiveness in mitigating resonance phenomena and improving the dynamic stability of the structure. The right-hand diagram displays the time-dependent displacement response for both support configurations under identical excitation. The red curve corresponds to the rigid support, while the green curve represents the system with added damping. As seen in the plot, the rigidly mounted system maintains high-amplitude oscillations with persistent undamped behavior. In contrast, the damped system exhibits reduced amplitude and visible energy dissipation over time. This time-domain analysis confirms that damping not only lowers the peak response but also facilitates quicker stabilization, making it more favorable for applications requiring vibrational control.

Conclusion

The computational modeling provided insights into how the rotor and housing interact when bearings are supported with elastic elements. It was observed that both masses oscillated in synchrony, indicating a significant dynamic connection. Nevertheless, the housing exhibited lower vibration amplitudes, highlighting its role in absorbing excess energy.

The inclusion of polyurethane damping layers not only reduced steady-state vibration amplitude by 35-50%, but also decreased resonance peaks in the frequency spectrum and improved transient stability under impulsive loads. This confirms that elastic-damping supports effectively redistribute dynamic stresses and prevent excessive energy concentration at critical frequencies.

From a diagnostic perspective, the developed dynamic model can serve as a reference (baseline) model for filtering background noise in vibration monitoring systems. By comparing measured signals with simulated dynamic responses, it becomes possible to distinguish structural vibration components from external acoustic interference, thereby improving fault detection reliability.

Therefore, the proposed approach contributes both to theoretical modeling of rotor-bearing systems and to practical vibration-control strategies in textile machinery and other high-speed rotating equipment. Future work should include experimental validation and extension to nonlinear contact models.

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