Vibration analysis to predict the Transmissibility for Rotating Machines

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ABSTRACT

Springs being an important member of any mechanism serve the purpose of absorbing shocks and vibrations of the applied load due to the inbuilt characteristics called stiffness which they possess. The stiffness of any spring may be defined as the force required per unit deflection. It is one of the primary characteristics of a spring. The project work shows the application of a spring by using a spring-mass system. The current experimental analysis focuses on determining the transmissibility ratio for a given natural frequency for different harmonic motions obtained by varying speeds. The inertia plate is mounted on 4 springs whose stiffness is experimentally determined. Mass (electric motor) is kept on this inertia plate. The main idea is to obtain the plots showing the transmissibility ratio for various excitation frequencies. As an application, the setup comprising of spring mounts can be used as an additional feature while installing any machine tool on the base to absorb the mechanical vibrations induced by the machine tool and keep them to a minimum.

Keywords: Vibration, transmissibility, natural frequency, rotating machines

1. INTRODUCTION

Any system when displaced from its equilibrium position by the application of external force is said to undergo vibrations. Mechanical vibrations are defined as any unbalanced forces on the rotating or reciprocating machine which are the periodic oscillations of elastic bodies with reference to an equilibrium point. These oscillations may be linear or Non-linear. The Primary causes of vibrations include unbalanced forces in rotating and reciprocating parts, dynamic effects of rolling and sliding contacts of the members of the machine etc.,

Vibration is the motion of a particle or a body or system of connected bodies displaced from a position of equilibrium. Most vibrations are undesirable in machines and structures because they produce increased stresses, energy losses, cause added wear, increase bearing loads, induce fatigue, create passenger discomfort in vehicles, and absorb energy from the system.

2. Literature Review

To evaluate the predictive maintenance module's efficacy, validation experiments were conducted on drilling and milling operations. According to the ISO 10816 vibration severity chart, the present module's results can be used to monitor and give machine conditions at four different levels: good, satisfactory, unsatisfactory, and unacceptable for rotating equipment status [2].

When defects are identified early on, vibration analysis can assist extend the life of the machinery. A revolving tabletop model is also subjected to vibration analysis to demonstrate the possibility that certain flaws may persist despite being invisible to the unaided eye [3]

Frequency and phase analysis are used to identify the precise spectrum patterns that are produced by each machinery issue. I have provided a detailed explanation of frequency analysis, phase analysis, unbalancing, and the balancing technique here. Furthermore covered are acceptability standards, data gathering tools, and vibration measurement tools. In the several case studies I conducted across various industries, we addressed resonance, misalignment, and balancing through alignment across coupling and dynamic vibration absorbers [4].
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In order to diagnose a rotating machine effectively, effective vibration signal extraction techniques are essential. Over the past few years, numerous methods for extracting vibration signals have been proposed. Maintaining a machine's optimal level of efficiency and performance is made easier with condition monitoring. While a rotating machine's condition monitoring is an effective maintenance duty, it is frequently difficult and labor-intensive [5].

Vibration analyzers and the Ansys software V16.0 are used in our current study to perform vibration analysis on the cone crusher and shaft bearing assembly. Because of the eccentric sleeve, the cone crusher is a sophisticated, high-power hydraulic crusher that produces intense vibrations. The foundational reasons for the vibrations in the crusher shaft have been found through investigation, and the base frame has been modified to mitigate the vibrations by about 20%. By using spectrum analysis on the basis frame, these findings were verified [6].

This apparatus filters wheat. There are multiple mechanical issues with this machine, including unbalance, bearing heating, shaft bending, and bearing flaws that developed while it was in use. It is a delicate depiction of spectrals. The faults appear at the same components of high amplitudes that correspond to the harmonies of the frequency equal to the rotor's rotation frequency. They are overlaid [7].

In order to diagnose a rotating machine, effective vibration signal extraction techniques are essential. Over the past few years, numerous methods for extracting vibration signals have been proposed. This study reviews a few vibration feature extraction techniques used on various rotating machine types [8].

The effectiveness of the dynamic analysis approach suggested in this work is demonstrated by the results of an experiment and computer simulation utilizing the same rotating machine. Using symptom characteristics and vibration signal spectra collected in these states, the technique for differentiating between rotating machine structure problems (shaft misalignment state, unbalance state, and looseness state) is finally presented [9].

Because of the study conducted in these areas, predictive maintenance was created to help maintenance engineers schedule when repairs should be performed. By doing this, the chance of a forced failure breaking the machine is reduced. For vibration analysis, it is possible to measure the acceleration, displacement, and speed of vibrations produced by rotating machine parts. Consequently, vibration changes may be monitored and equipment repairs can be properly scheduled. Our project has involved looking into possible fixes for problems caused by the vibration of rotating equipment used in the process industry, including motors, forced draft fans, centrifugal pumps, and rotary compressors [10].

In this research, a three-layered cloud computing platform is given, and unlabeled data is received to produce an online judgment that can be interpreted. The vibration signal's features, including its range, mean, root mean square, standard deviation, and crest values, are extracted. The model's efficacy is assessed through the application of traditional statistical criteria, such as the vibration signal's Mean Absolute Error (MAPE) and Root Mean Square Error (RMSE). The suggested method is shown to be 25% and 90% more effective [11].

Selecting the characteristics impacted by errors is often done by artificial intelligence (AI) techniques. A thorough review of recent studies on fault diagnosis for various rotary machine elements, including the type of failure, feature extraction method, and classification technique performance, is presented in this paper along with a discussion of three different types of artificial intelligence methods: Artificial Neural Networks, K-Nearest Neighbor, and Support Vector Machine [12]

The actual readings were examined using a vibratory indicator, which allowed for the identification of the weak places in the machine (the rolling bearings) that were causing the malfunction. As a result, an online control system was used to monitor the degradation and optimize maintenance. Once the predetermined matching vibration level threshold has been reached, the analysis of these vibrations allows for the potential of locating and detecting the problematic components [13].

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3. NUMERICAL ANALYSIS OF VIBRATIONS

For performing numerical analysis of mechanical vibrations the following points are taken into consideration. (1) Mathematical Modelling of a Physical System, (2) Formulation of Governing Equations (3) Solution of the governing equations mathematically (4) Interpretation of the results.

3.1 Mathematical Modelling of a Physical System

The features and aspects of any mechanical system can be studied by mathematically modeling it. Also, mathematical modeling can find out the physical elements that need to be considered for the mathematical analysis.

Formulation of Governing equation: Considering the vertical motion the equation of motion for a single degree of freedom system is given as

$$m\frac{d^2x}{dt^2} + kx = F(t)$$

3.2 NATURAL FREQUENCY OF VIBRATIONS

The frequency at which the system starts to oscillate without any driving or damping force is known as the natural frequency of vibrations.

Free vibrations of an elastic body occur at a frequency called natural frequency.

The natural frequency (ω_n) for a spring mass system is given by: $\omega_n = \sqrt{\frac{k}{m}}$

where, k is the spring stiffness in N/m, m is the mass attached to the spring in kg

3.4 Magnification Factor

The ratio of maximum displacement of the forced vibration (x_{max}) to the deflection due to static force is known as Magnification factor

Mathematically$x_{max} = \frac{x_0}{\sqrt{\frac{c^2.\omega^2}{s^2}+\left(1-\frac{\omega^2}{(\omega_n)^2}\right)^2}}$

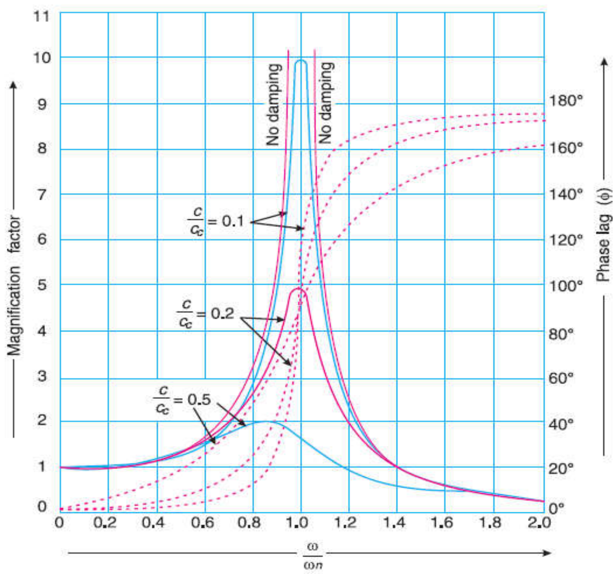

Fig.1 Relationship between magnification factor and phase angle for different values of $\frac{\omega}{\omega_n}$

3.5 Vibration Isolation and Transmissibility

When an unbalanced machine is placed upon the foundation then the force is transmitted to the foundation due to any unbalanced dynamic forces set up due to the reciprocating machine parts. But if the springs are installed in between them then the force due to unbalanced forces is absorbed by the springs and then the remaining minimum force is transmitted to the foundation, In this way the vibrations are minimized by isolating them.

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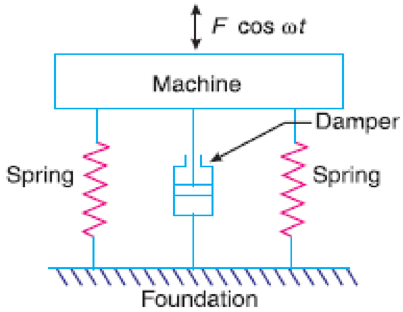


Fig.2 Vibration isolation

4. Experimental Setup

4.1 Setup Introduction: As shown in figure 3, an inertia block of mass m kg is mounted on 4 springs. These springs are attached onto the base plate. A supporting plate is used to serve as base support. A rotating machine of mass m kg (motor in our case) is placed upon the inertia block.

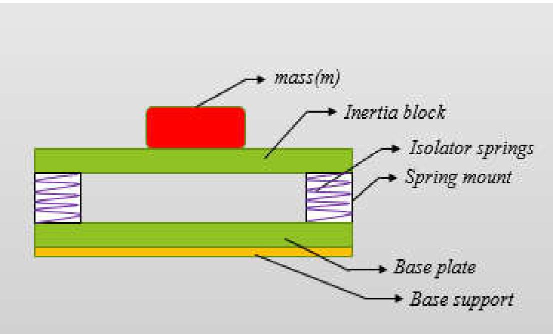


Fig.3 Experimental setup

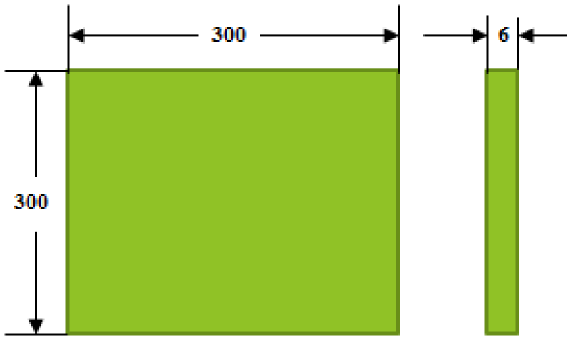


Fig.4 Dimensions of inertia block and base plate

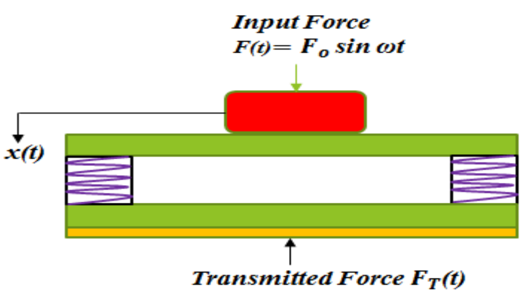


Fig.5 Forces acting on the system

Fig.6 Inertia block

Inertia plate: The inertia plate which is made of cast iron whose thickness is 0.6mm and is of square section 300mm x 300mm.The weight of the inertia plate is 2kg.

Base Plate: The base plate has the same dimensions as that of the inertia pate and is supported on 4 supports as shown in figure 7 and is made up of cast iron

Springs: Wire Springs are coiled spring products in a variety of cross-section forms.When comparing coil springs to flat/stamped springs, they offer increased travel, but with lower loads.The springs used for the experiment havea wire diameter of 3 mm and a coil diameter of 50mm

Spring mount: The mount serves the purpose of providing support to the springs. Each spring has two mounts one lower and one upper. The lower mount is fixed on the base plate whereas the inertia plate is mounted on the upper spring mount.

Fig.7 Base plate

Fig.8 Springs

Fig.9 Spring mount

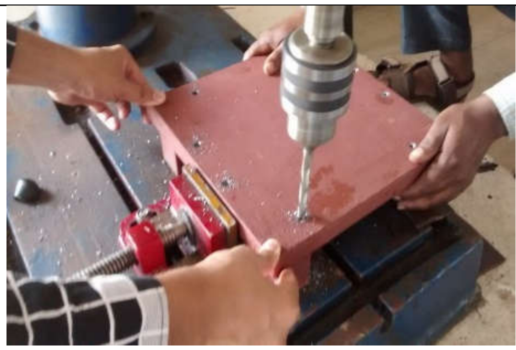


Fig.10 Hole drilling on the base plate

Hole drilling on the base plate: 8 holes of mm dia are drilled onto the base plate for the purpose of installing the spring mounts on it. This can be seen from figure 10.

Electric motor: The specifications of the motor are as follows: Speed N = 1480 rpm

4.2 Methodology

Maximum displacement or Amplitude is given by $x_{max} = \frac{F}{\sqrt{(k-m\omega^2)^2+c\omega^2}}$

Where F is the maximum force applied $F = F_0 \sin \omega t$

ω = Angular velocity of the machine

In the absence of damping the above equation reduces to $x_{max} = \frac{F}{\sqrt{(k-m\omega^2)^2}}$

The force transmitted to the base (F_t) due to the excitation force acting on the machine is given as $F_t = kx$

The ratio of Force transmitted to the base to the Maximum force is given by transmissibility (T)

Mathematically $T = \frac{F_t}{F}$

The transmissibility ratio gives the response of the machine to the displacement of the base occurring due to the excitation force.

5. Results and Discussions

The springs used for the analysis are computed for its stiffness experimentally for different loads and corresponding deflections are noted down. The spring testing machine is shown in the figure 11. Load on the spring is applied by rotating the lever and the corresponding deflections are measured on the scale for different loads.

Fig.11 Set up for the experimental determination of spring stiffness

Table shows the different values of loads and corresponding deflections. The stiffness is found out for each spring and the average stiffness if found out.

Table 5.1 Spring stiffness measurement

Sl.No	*Load (N)*	*Deflection (m)*	*Stiffness (N/m)*
1.	4.90	0.001	4900
2.	9.81	0.002	4905

3.	14.71	0.003	4903
4.	19.62	0.005	3924
5.	24.52	0.007	3502
6.	29.43	0.008	3678
7.	34.33	0.01	3933
8.	39.24	0.011	3567
9.	44.14	0.012	3678
10.	49.05	0.014	3503

The spring stiffness is given by $K = \frac{F}{\delta}$

Where F is the load applied on the spring in N, δ is the static deflection of the spring in m

The average stiffness of the springs $K = \frac{K_1+K_2+K_3+K_4+K_5+K_6+K_7+K_8+K_9+K_{10}}{10}$ = 3999.3 N/m

Angular velocity $\omega = \frac{2\pi N}{60} = \frac{2\pi X\ 1480}{60}$ = 154.956 rad/s

The transmissibility ratio is given by $\varepsilon = \frac{1}{1-\left(\frac{\omega}{\omega n}\right)^2}$

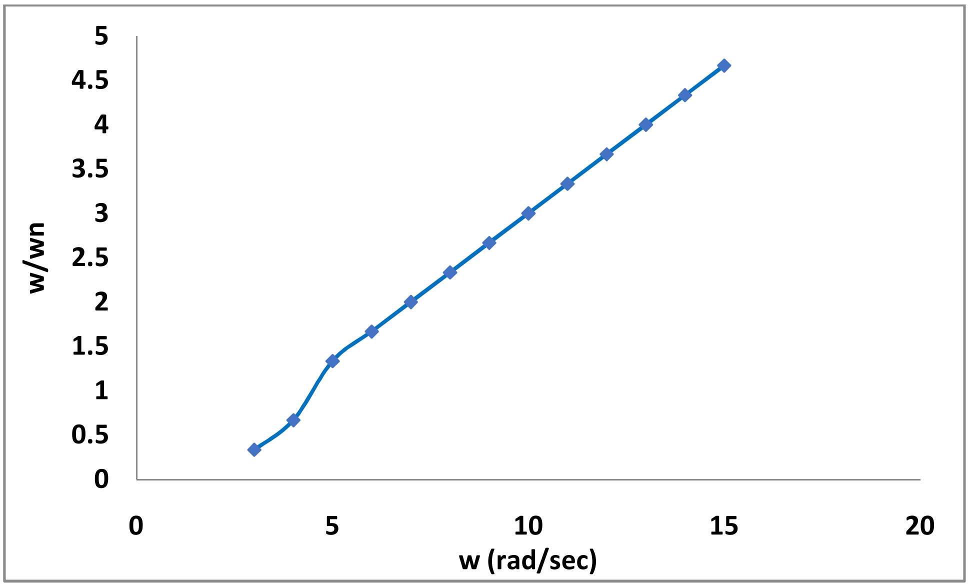
Calculating the transmissibility ratio for different excitation frequencies

1) Angular velocity ω = 100 rad/s, $\varepsilon = \frac{1}{1-\left(\frac{\omega}{\omega n}\right)^2}$ = 1.125

Table 5.2 Tabular column for transmissibility ratio

Sl.No	***Speed (RPM)***	***ω (rad/s)***	***ω_n(rad/s)***	$\frac{\boldsymbol{\omega}}{\boldsymbol{\omega_n}}$	***Transmissibility ratio (ε)***
1.	100	10.46667	31.38549	0.333487	1.125130071
2.	200	20.93333	31.38549	0.666975	1.801332763
3.	400	41.86667	31.38549	1.33395	-1.283002108
4.	500	52.33333	31.38549	1.667437	-0.561688321
5.	600	62.8	31.38549	2.000925	-0.332922797

6.	700	73.26667	31.38549	2.334412	-0.224745378
7.	800	83.73333	31.38549	2.6679	-0.16346045
8.	900	94.2	31.38549	3.001387	-0.124870079
9.	1000	104.6667	31.38549	3.334874	-0.098800686
10.	1100	115.1333	31.38549	3.668362	-0.080276933
11.	1200	125.6	31.38549	4.001849	-0.066600965
12.	1300	136.0667	31.38549	4.335337	-0.056195105
13.	1400	146.5333	31.38549	4.668824	-0.048081734



The plots of $\left(\frac{\omega}{\omega_n}\right)$ vs transmissibility ratio is shown in the figure 12.

The relation of the ratio of $\frac{\omega}{\omega_n}$ with the transmissibility ratio is shown in the figure 12. As can be seen from the figure that the ratio $\frac{\omega}{\omega_n}$ increases initially when the transmissibility ratio is in the range between 1 to 2 and reduces for the negative values of the transmissibility ratio.

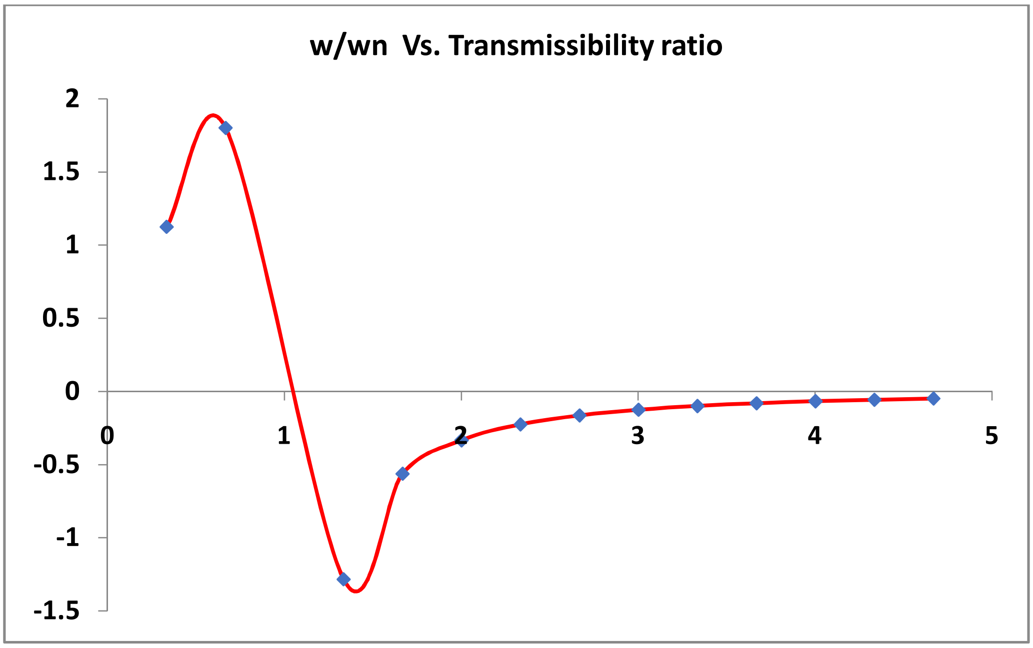


Fig.12 Graph for $\left(\frac{\omega}{\omega_n}\right)$ vs transmissibility ratio is shown in the fig.

6. CONCLUSIONS

The motor aligned on the frame supported by four helical springs has been operated at different speeds. The natural frequency of the system has been determined to avoid resonance. The transmissibility ratio has been determined for different angular velocities of the system. With increase in ratio $\frac{\omega}{\omega_n}$, initially the transmissibility ratio increased to 1.8 and later it was reduced to negative value -1.5 and reached near to the x axis. The transmissibility ratio reached a stable condition after the ratio $\frac{\omega}{\omega_n}$ increased more than 2.

The Experimental analysis can be extended with varying masses.

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