"ANALYTICAL & EXPERIMENTAL INVESTIGATION OF EFFECT OF MATERIAL & GEOMETRY OF SHAFT"

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ABSTRACT

In Fault detection of bearing, detection of misalignment and condition monitoring of bearing or gearing system, the system has to rotate at different speeds. If specific rpm matches with critical speed which is nearer to first bending natural frequency of shaft will generate excessive vibrations due to resonance. If the objective of whole analysis is to identify the fault with the help of vibration spectrum, excessive vibrations may dominate the spectrum which may be useful for fault detection. Hence it is necessary to avoid critical speeds, or detect the change in spectrum due to critical speed. In this paper the work will be focused on estimation of bending natural frequencies which is nearer to critical speeds of shaft and mode shapes by using FEM software and experimental verification of the same.

1. Introduction

Bending vibrations and critical speeds of rotating shafts is perhaps the most common problem that is discussed by a vibration engineer, as it is a vexing day-to-day problem in design and maintenance of the machinery. Some of the rotors weigh as much as 100 tons as in the case of big steam turbines and obviously they deserve utmost attention in this regard. The rotors have always some amount of residual unbalance however well they are balanced, and will get into resonance when they rotate at speeds equal to bending natural frequency. These speeds are called as critical speeds by Rankine [7] as far as possible they should be avoided. Even while taking the rotor through a critical speed to an operational speed, special precaution should be taken. In Fault detection of bearing detection of misalignment and condition monitoring of bearing or gearing system is necessary because the system has to rotate at different speeds. If specific r.p.m.matches with critical speed which is nearer to first bending natural frequency of shaft will generate excessive vibrations due to resonance. If the objective of whole analysis is to identify the fault with the help of vibration spectrum, excessive vibrations may dominate the spectrum which may be useful for fault detection. Hence it is necessary to avoid critical speeds, or detect the change in spectrum due to critical speed. In this dissertation the work will be focused on estimation of bending natural frequencies which is nearer to critical speeds of shaft and mode shapes by using FEM software and experimental verification of the same.

The calculations of bending natural frequency of simple shaft in rigid bearings is somewhat an easy matter, the problem in practice becomes complex because of following.

1. Gyroscopic effects of disks,
2. Dissimilar moments of area of the shaft,
3. Stiffness and damping properties of oil film bearings, and
4. Coupling between two rotors.

To avoid failures of shafting, the general practice in the design of rotors is determine the bending critical speeds, check the out-of –balance response and adopting a suitable balancing procedure. Different parameters can be varied such as geometrical and material for the calculation of natural frequency and the critical speed of shaft to study the effect of same. Here the Efforts are made in the same direction to calculate natural frequencies with variation in geometrical and material parameters to avoid resonance. This is carried out by using the finite element analysis and experimental with FFT analyzer.

2. Objectives

1. Study of vibration spectrums of shaft without variation in geometrical and dimensional parameters (Without variation in Diameter and Support Length)
2. Study of vibration spectrums of shaft with variation in geometrical and dimensional parameters (With variation in Diameter and Support Length)
3. Study of vibration spectrums of the shaft with change in material parameters.
4. Effect of variation in geometrical and dimensional parameters on vibration spectrum.

3. Literature Survey

A large number of methodologies for calculation of critical speed of shaft have been reported in the previously published literature. In this, previously published research work related to critical speed calculation of shaft has been reviewed.

Dunkerley [4] has proposed a very convenient method to determine the fundamental critical speed a shaft carrying number a components. The method is quite simple and consists a reducing the actual system into a number a simple subsystems, calculating the critical speeds a each by a direct formula, and combining these critical speeds according to the following equation.

$$\frac{1}{\omega^2} = \frac{1}{{\omega_1}^2} + \frac{1}{{\omega_2}^2} + \ldots\ldots + \frac{1}{{\omega_n}^2}$$

To obtain actual critical speed the system. For a normal proof to the above equation and the approximation involved, reference may be to Rao and Gupta [18]. The value obtained is always a lower bound approximation.

Rayleigh [20] has proposed another simple method which is based on the fact that the maximum kinetic energy must be equal to maximum potential energy for a conservative system under free vibration conditions. For a shaft carrying several components, we can use static deflection or any other suitable function to represent the fundamental mode of the shaft.

The fundamental frequency may be obtained from,

$$\omega^2 = \frac{g\ My}{\sum My^2}$$

Where M_1, M_2 are masses a different components and y_1, y_2 are deflections of the shaft at the locations a these components since the frequency is a minimum it is always an upper bound value. The main advantage a Dunkerley's and Rayleigh's methods are that they use simple strength a material formula for beams (Either analytical or graphical methods can be chosen) and that Dunkerley's method Rayleigh's method gives an upper bound value.

Therefore the exact range where the critical speed use is well established by these two methods taken together and thus they are very popular. There are several other methods that can be used to determine critical speeds, such as Stodola's method, Ritz method etc.

In a manner similar to Holzer method [8] for torsional vibrations, Myklestad [13] and Prohol [15] developed highly successful methods a computation for the bending critical speeds a shafts and these methods are used in transfer matrix form, for bending critical speed and out a balance response calculations. A polynomial frequency equation for Myklestad equations is given by Rao and Reddy [19].

G.N.D.S. Sudhakar and A.S.Sekhar in their paper "Identification of unbalance in a rotor bearing system" have given model based methods for fault detection by using equivalent loads minimization method. They have identified fault in a rotor bearing system by minimizing difference between equivalent loads estimated in the system due to the fault and theoretical fault model loads. Two different approaches: Equivalent loads minimization and vibration minimization methods are applied for identification of unbalanced fault in a rotor system, fault identified by measuring transverse vibrations at only one location [5].

Hsaing-Chieh Yu, Yin-Hwang Lin, Chin Liang chu, in their paper "Robust modal vibration suppression of a flexible rotor" studied active robust model vibration control of a rotor system supported by magnetic bearings. Finite element method is applied to formulate the rotor method. The Themoshenko Beam theory, Effects of shearing deformation is considered in their work. This study allies the independent modal space control (IMSC) approach. This approach is effective for vibration suppression when the system is subjected to impulsive or step loading, speed variation and sudden loss of disc mass [6].

R. Tiwari and V. Chakravarti in their paper "Simultaneous estimation of residual unbalance and Bearing dynamic parameters from the experimental data in a rotor bearing system" given two separate methods. The first method uses the impulsive response measurements of the journal from bearing housing in horizontal and vertical directions. Time domain signals of impulse forces and displacement responses are transformed to the frequency domain and are used for estimation of the residual imbalance and bearing dynamic parameters.

Experimental measurements responses have been fed to identify the residual unbalance and bearing dynamic parameters by both the methods. The simulated responses are in fairly good agreement with experimental responses in terms of mimicking predominant responses [16].

J.S.Rao in his book “Rotor Dynamics”, has given details of bending critical speeds of simple shafts. The phenomenon of bending vibrations and critical speeds of rotating shaft is perhaps the most common problem discussed by a vibration engineer as it is regular problem in design and maintenance of the machinery. The rotors have always some amount of residual unbalance however well they are balanced, and will get into resonance when they rotate at speeds equal; to bending natural frequency the speeds are called critical speeds and as far as possible they should be avoided. Even while taking the rotor through a critical speed to an operational speed, special precaution should be taken. In this dissertation the theory and methodology suggested by J. S. Rao [10] will be utilized.

Kimball and Hull [11] explain the physical action which takes place while the shaft passes through the critical speed. They show that below the critical speed the centre of mass rotates in a circle about the geometric centre, whereas above it the shaft rotates about the centre of mass. Thus the axis of rotation is changed at the critical speed from the geometric centre of the mass centre. When the shaft rotates at the critical the restoring force of the shaft is neutralized and the action is dynamically unstable, hence large amplitudes of vibration may occur.

4. FEM Analysis of Shaft For Rotor Bearing System

Finite Element Method has becomes a powerful tool for the numerical solution of a wide range of engineering problems due to large memory digital computers. The basic idea of FEM is to find the solution by spending more computational efforts. Since it is difficult to find the exact analysis of complicated shape is approximated as composed of several finite elements and inter convenient approximate solution is assumed and the conditions of overall equilibrium of the complicated shapes are derived. With the advance in computer technology and CAD system, the complex problem can be modeled with relative ease. Several alternate configurations can be tried on computer after prototype is built. All of this suggests that there is need to keep pace with these developments by understanding the basic theory, modeling techniques and computational aspects of FEM. In these days, the developments in mainframe computers and availability of powerful microcomputers have brought this method within reach of students and engineers working in industries.

5.1 ANSYS Software

The ANSYS computer program is ANSYS large-scale multipurpose finite element program that may be used for solving several classes of engineering analysis. The analysis capabilities of ANSYS include the ability to solve static and dynamic structural analysis, bucking given value problems, static or time varying magnetic analysis and various types of field and coupled field applications.

This program contains many special features, which allow nonlinearities or secondary effect to be include in the solutions, such as plasticity, large strain, hyper elasticity, creep, swelling, large deflection, contact, stress stiffening, temperature dependency, material anisotropy and radiation.

5.2 Modal Analysis

Modal analysis determines the vibration characteristics (natural frequencies and mode shapes) of a structure or machine components while it is being designed. . The natural frequencies and mode shapes are important parameter in the design of a structure for dynamic loading conditions. Same set of command is used for modal analysis that used in any other type of Finite Element Analysis. Likewise, choose similar option from the graphical user interface (GUI) to build and solve models.

For FEM analysis of the intermediate shaft as per geometry of the shaft which is carrying two three stepped pulleys and the shaft is resting on two bearings at the ends so number of nodes are twelve and so the number of elements are eleven.

The intermediate shaft is made up of solid steel material with inner diameter and outer diameter equals to diameter of the shaft. As the shaft is cylindrical shape the element type is considered as pipe with inner diameter considered as zero and outer diameter considered as diameter of the intermediate shaft.

As the intermediate shaft is resting on the bearings at the ends the boundary conditions are considered according to first and last element. The total number of nodes in the analysis for intermediate shaft is twelve and number of elements is eleven at the ends between first and second and between last and second last nodes the elements. Thus the boundary conditions are considered fixed at the left end as well as fixed at the right end also.

Steps Involved In Modal Analysis:

The procedure for a modal analysis consists of main steps.

1. Build the model.
2. Apply loads and obtained the solution.
3. Expands the modes.
4. Review the results.

1. Build the Model
This step includes the job name and analysis title and then uses PRER7 to define the element types, element real constant, material properties, and the model geometry.
2. Apply loads and obtained the solution.
In this step, define analysis types and options, apply load, satisfy load step options, and being the finite element solution for the natural frequency.
3. Expands the modes.
Specify the number of mode that, you have to expand. If frequency range is selected, only modes within that range mode results are appeared.
4. Review the results.
Results from modal analysis are written to the structural results file. Results consist of natural frequency; expand mode shapes relative stress and force distribution. Those results wish to see, database must contain the same modal for which the solution was calculated.

5. Experimental Analysis

5.1 Result Tables

Table 1 Effect of Geometric Parameters on Natural Frequencies for Intermediate Shaft

Shaft Geometry	Natural Frequencies in Hz		
	ω1	ω2	ω3
No change in geometry	433.15	1368.3	2775.3
10% reduction in shaft diameter	362.90	1167.1	2043.0
15% reduction in shaft diameter	337.48	1100.0	1936.7
10% reduction in shaft length	716.58	2222.1	3899.0
15% reduction in shaft length	755.47	2345.4	4041.3

The above Table shows Geometric Parameters influence on Natural Frequencies for Intermediate Shaft in which we can take various types of Steel Shaft Geometry by changing its geometry as follows: No Change in Geometry, 10% Reduction in Shaft Diameter, 15% Reduction in Shaft Diameter and also 10% Reduction in Shaft Length, 15% Reduction in Shaft Length and their respective Natural Frequencies. Effects of geometrical parameter of Shaft like change in diameter and length on natural frequency are found significantly at higher modes. As we go on reducing shaft diameter and shaft length there are not so much significant variations at higher mode.

Table 2 Effect of Material Parameters on Natural Frequencies for Intermediate Shaft

Shaft Geometry	Natural Frequencies in Hz		
	ω1	ω2	ω3
Steel	433.15	1368.3	2775.3
Brass	320.89	1027	1783.9
Aluminum Alloy	305.54	940.85	1799.0

The above mentioned Tabular 2 data shows Material Parameters influence on Natural frequencies for Intermediate Shaft in which we can take Shaft Geometry of Steel, Brass and Aluminum alloy having their different Natural Frequencies. It is an intermediate shaft made of Brass material and Aluminum alloy. Table number 2 shows the three Natural Frequencies in Hz. of intermediate shaft with no change in geometry having significant value of Brass and Aluminum alloy, Than Steel. From Above Table 2 we select steel shaft for experimental work as compare to steel. We have selected steel shaft for experimental work. From the assistance of the above tabular data.

The vital reason for this selection criteria is that the steel shaft natural frequencies are higher in all three modes of shapes as compare to Brass and Aluminum.

Table 3 Comparison between Experimental and Software Results

Natural Frequency Hz	Experimental	FEM	% Difference
ω1	424	433	0.979
ω2	1334	1368	0.975
ω3	2317	2375	0.975

The above Table 3 shows comparison between Experimental and (FEM)Software Results .We can come to the conclusion from the above tabular data that almost all the natural frequencies are constant and significant difference is found only at higher modes of vibrations.The Difference has been found Significant and it concludes that selection of steel shaft is better .From the above Table 3 it is possible to avoid critical speeds and hence the resonance can also be avoided.

6.3 Graphical Representation of Results

6.3.1 Effect of Shaft Diameter on Natural Frequency

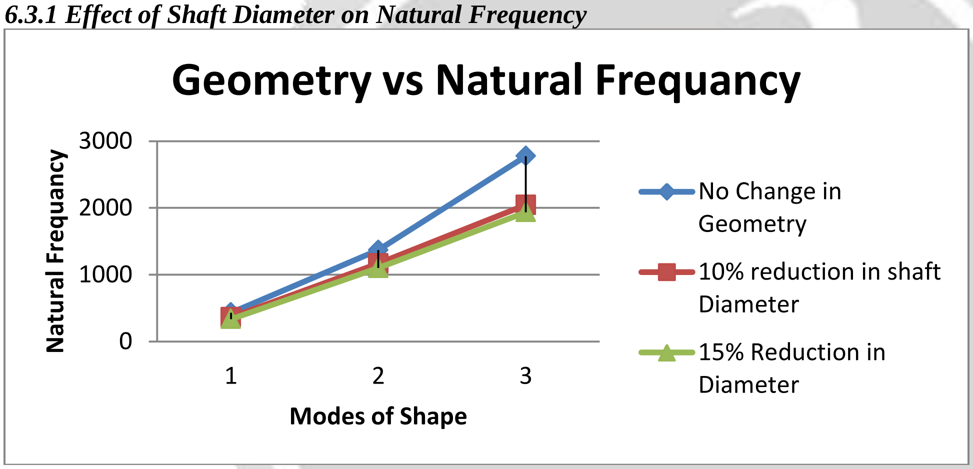

Fig Effect of Shaft Diameter on Natural Frequency

From the above figure it is observed that as the change in shaft diameter with natural frequency by No change in Geometry, 10% Reduction in shaft diameter and 15% Reduction in shaft diameter the effect of geometrical parameter like change in diameter on natural frequency is found significantly at higher modes.

6.3.2 Effect of Shaft Length on Natural Frequency

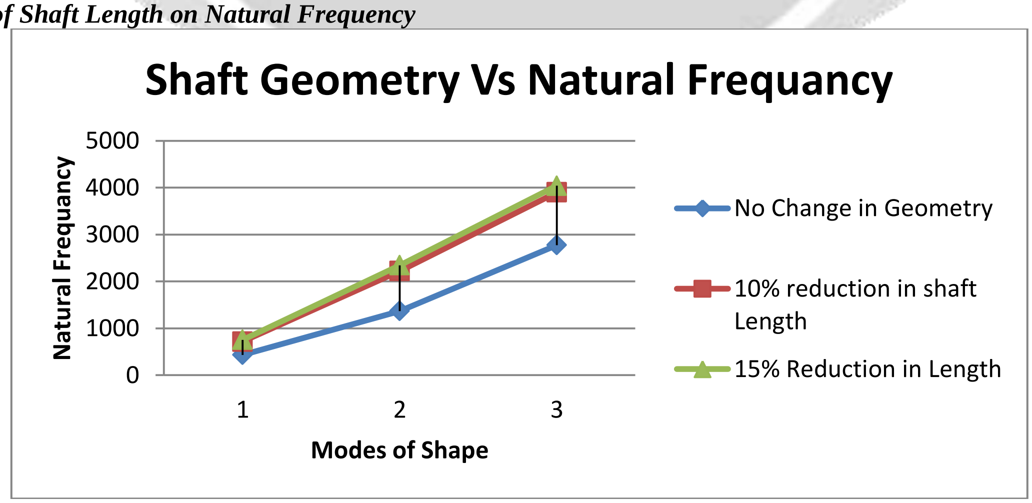

Fig 2 Effect of Shaft Length on Natural Frequency

From the above figure of change in shaft length with natural frequency, effect of geometrical parameter like change in length on natural frequency is found significantly at higher modes **.**Intermediate shaft having 10% and 15% reduction in shaft length with pulleys. Fig shows the three mode shapes of intermediate shaft with change in geometry that is reduction in length. From above figure it is observed that 15 % Reduction in shaft length shows higher natural frequency than other two natural frequency of shaft.

6.3.3 Effect of Material properties on Natural Frequencies

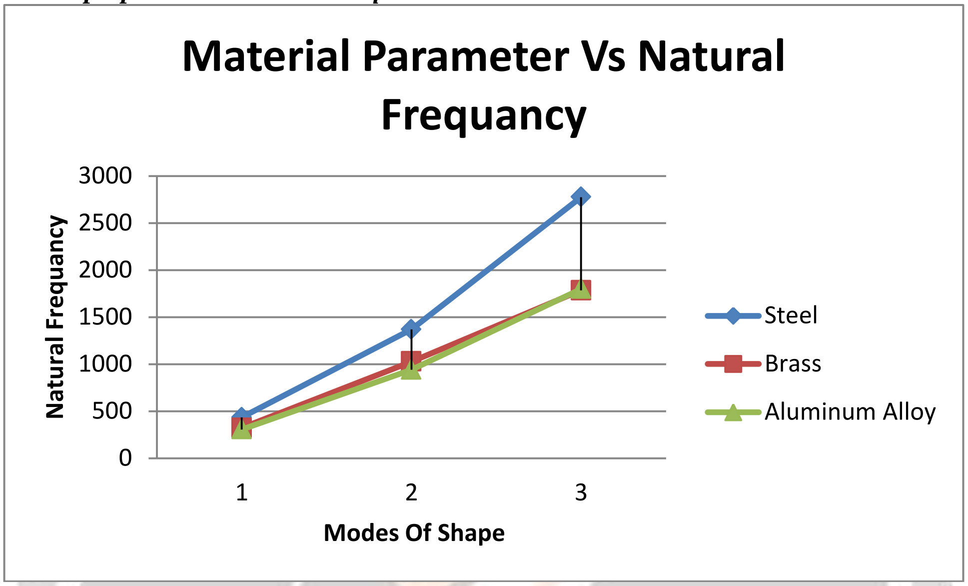

Fig 3 Effect of Material Properties on Natural Frequencies

From the above figure it has been observed that effect of material parameters on natural frequencies for shaft having significant change in natural frequency is observed between aluminum alloy and brass but the steel having higher natural frequency among aluminum, brass and steel.

6.3.4 Comparison Between Experimental and Software Results

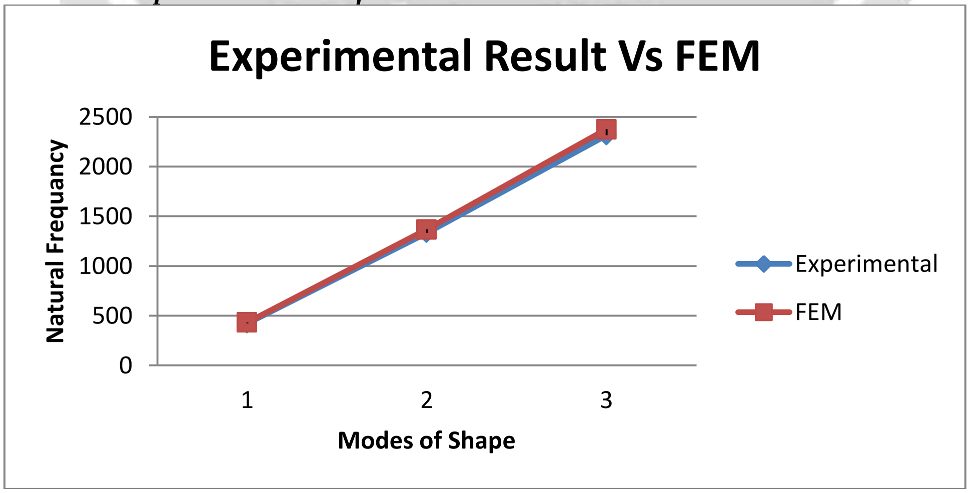

Fig 4 Comparison Between Experimental and FEA Results

From the above figure It has been observed that the results obtained from experimental is close agreement with the result obtain from FEM which is found at higher modes of vibrations. From the comparison of FEM and Experimental results, in case of intermediate shaft average deviation between the results is 0.976. From the analysis it is possible to avoid critical speeds and hence to avoid resonance.

6. Conclusion

The art of machine condition monitoring is knowing what to look for, and successful diagnosis is having the ability to measure it and correlate the result with known failure mechanisms. This report has attempted to address these issues in detail for geometrical and material parameters concerned with shaf.

In this paper a range of signal processing techniques has been discussed covering time domain and frequency domain analysis as well as FEM analysis.

1) From the analysis effect of change in diameter of the intermediate shaft on natural frequency is found significant at higher modes.
2) From the analysis effect of change in length of the intermediate shaft is found significant at 15% of the original length.
3) From the analysis of change in materials significant difference in natural frequency is observed between steel and brass material.
4) From the comparison of FEM and Experimental results, in case of intermediate shaft average deviation between the results is 0.976.
5) From the analysis it is possible to avoid critical speeds and hence to avoid resonance.

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