Optimisation of shafts through vibration analysis

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ABSTRACT

The detection of misalignment and condition monitoring of a bearing or a gearing system is necessary because the system has to rotate at different speeds. If a specific r.p.m. of the shaft matches with critical speed, which is nearer to the first bending natural frequency of the shaft, the shaft will generate excessive vibrations due to resonance. 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 the spectrum due to critical speed. In this paper, the work will be focused on the estimation of bending natural frequencies which are nearer to critical speeds of shaft and mode shapes by using FEM software and experimental verification of the same.

Keywords — Finite Element Analysis (FEA), FFT Analyzer.

I. INTRODUCTION

In rotary-dynamic systems, or systems which involve rotation of various components such as bearings, it is of prime importance to monitor the operating conditions of the components in order to avoid the untimely failure thereof. Care should be taken that the rotation speed of the rotating component does not match with the critical speed that may be close to the natural frequency thereof, so as to avoid excessive and undesired vibrations. The objective of the present work is to identify the faults in the vibration spectrum of a component. If excessive vibrations are dominating the vibration spectrum, this will be an indication of a fault in the rotating component. The present work focuses on the estimation of the natural frequencies which are closer to the critical speeds of a shaft. These are estimated first using by performing simulations on a FEM software and the experimental verification of the same are done with the use of an FFT analyzer.

II. LITERATURE REVIEW

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.

Hsaiing-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.

R. Tivari 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.

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 will be utilized.

III. OBJECTIVES

A. Study of vibration spectrums of shaft without variation in geometrical and dimensional parameters (Without variation in Diameter and Support Length)

B. Study of vibration spectrums of shaft with variation in geometrical and dimensional parameters (With variation in Diameter and Support Length)

C. Study of vibration spectrums of the shaft with change in material parameters.

D. Effect of variation in geometrical and dimensional parameters on vibration spectrum.

IV. EXPERIMENTAL ANALYSIS

Analysis was carried out with an FFT analyzer, an impact hammer, and the response is received from the use of an accelerometer in frequency domain and time domain.

1. Bearing Support
2. Mass 1
3. Mass 2
4. FFT analyzer
5. Impact Hammer
6. Accelerometer

V. ACTUAL EXPERIMENTAL SETUP WITH THE ACCELEROMETER USED

VI. PROCEDURE OF EXPERIMENTATION

A. Connect the accelerometer and the impact hammer to appropriate channels of the FFT through cables.
B. Prepare the set-up for modal analysis and in-pulse software.
C. Mount accelerometer on the shaft with the help of a magnetic base.
D. Excite the shaft by giving an in-pulse by the impact hammer.
E. Record the response received from the accelerometer in frequency domain and in time domain.
F. Identify the natural frequencies and corresponding phase in the FFT software and record it.
G. Repeat the procedure for different positions of accelerometer to record all the vibration modes of shaft.

VII. RESULTS AND DISCUSSION

A. Intermediate Shaft with no Change in Geometry
Analysis of Results for Intermediate Shaft with No Change: The above figure is a color coded plot of a Finite Element Analysis. It is an intermediate shaft with pulleys. Figures show the three mode shapes of the intermediate shaft with no change in geometry. The mode shapes were obtained similar to the first three ideal mode shapes of a simply support beam. Also, the maximum displacement is observed for the second and the third mode around the pulley location.

B. Intermediate Shaft with 10% and 15% reduction in shaft diameter

Fig. 4 First Mode (10% reduction in shaft diameter)

Fig. 5 Second Mode (10% reduction in shaft diameter)

Fig. 6 Third Mode (10% reduction in shaft diameter)

Fig. 7 First Mode (15% reduction in shaft diameter)

Fig. 8 Second Mode (15% reduction in shaft diameter)
Analysis of Results for Intermediate Shaft with Change in Diameter: Fig. 4 – Fig. 9 show an intermediate shaft having 10% and 15% reduction in shaft diameter with pulleys. The three mode shapes of the intermediate shaft with a change in geometry, that is, reduction in diameter. The mode shapes were obtained similar to the first three ideal mode shapes of a simply support beam. Also, the maximum displacement is observed for the second and third mode around the pulley location.

A. Intermediate Shaft with 10% and 15% reduction in shaft length

- Fig. 10 First Mode (10% reduction in shaft length)
- Fig. 11 Second Mode (10% reduction in shaft length)
- Fig. 12 Third Mode (10% reduction in shaft length)

- Fig. 13 First Mode (15% reduction in shaft length)
- Fig. 14 Second Mode (15% reduction in shaft length)
- Fig. 15 Third Mode (15% reduction in shaft length)
Analysis of Results for Intermediate Shaft with Change in Length: Fig. 10 – Fig 15 show an intermediate shaft having 10% and 15% reduction in shaft length with pulleys. Figures show the three mode shapes of the intermediate shaft with a change in geometry, that is, reduction in length. The mode shapes were obtained similar to the first three ideal mode shapes of a simply support beam. Also, the maximum displacement is observed for the second and the third mode around pulley location.

D. Intermediate Shaft with Steel Material

Fig. 16 First Mode (Steel shaft)

Fig. 17 Second Mode (Steel shaft)

Fig. 18 Third Mode (Steel shaft)

Analysis of Results for Intermediate Shaft of Steel: Fig 16 – Fig 18 show an intermediate shaft made of steel with pulleys. Figs show the three mode shapes of the intermediate shaft with no change in geometry. The mode shapes were obtained similar to the first three ideal mode shapes of a simply support beam. The frequency values obtained for first mode is 433 Hz which is close to value of the experimental natural frequency 424 Hz. Similar results were obtained for second and third mode shapes with values of natural frequencies 1368 Hz and 2375 Hz respectively.

E. Intermediate Shaft with Brass and Aluminum Alloy Material

Fig. 19 First Mode (Brass shaft)

Fig. 20 Second Mode (Brass shaft)

Fig. 21 Third Mode (Brass shaft)

Fig. 22 First Mode (Aluminium alloy shaft)

Fig. 23 Second Mode (Aluminium alloy shaft)
**Analysis of Results for Shaft with Brass and Aluminum Alloy Material**: The intermediate shaft is made of a Brass material and Aluminum Alloy with pulleys. Fig. 19 – Fig. 24 show the three mode shapes of the intermediate shaft with no change in geometry. The mode shapes were obtained similar to the first three ideal mode shapes of a simply support beam. For the Brass material, the first three mode shapes are obtained at 320 Hz, 1027 Hz, and 1784 Hz respectively. Similarly for the Aluminum Alloy, first three mode shapes are at 305 Hz, 941 Hz and 1799 Hz respectively.

**VIII. RESULTS TABLES**

**A. Effect of geometrical parameters on natural frequencies of intermediate shaft**

<table>
<thead>
<tr>
<th>Shaft Geometry</th>
<th>Natural Frequencies in Hz</th>
<th>(\omega_1)</th>
<th>(\omega_2)</th>
<th>(\omega_3)</th>
</tr>
</thead>
<tbody>
<tr>
<td>No change in geometry</td>
<td></td>
<td>433.15</td>
<td>1368.3</td>
<td>2775.3</td>
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<tr>
<td>10% reduction in diameter</td>
<td></td>
<td>362.90</td>
<td>1167.1</td>
<td>2043.0</td>
</tr>
<tr>
<td>15% reduction in diameter</td>
<td></td>
<td>337.48</td>
<td>1100.0</td>
<td>1936.7</td>
</tr>
<tr>
<td>10% reduction in length</td>
<td></td>
<td>716.58</td>
<td>2222.1</td>
<td>3899.0</td>
</tr>
<tr>
<td>15% reduction in shaft length</td>
<td></td>
<td>755.47</td>
<td>2345.4</td>
<td>4041.3</td>
</tr>
</tbody>
</table>

**B. Effect of Material Parameters on Natural frequencies for Intermediate Shaft**

<table>
<thead>
<tr>
<th>Shaft Material</th>
<th>Natural Frequencies in Hz</th>
<th>(\omega_1)</th>
<th>(\omega_2)</th>
<th>(\omega_3)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Steel</td>
<td></td>
<td>433.15</td>
<td>1368.3</td>
<td>2775.3</td>
</tr>
<tr>
<td>Brass</td>
<td></td>
<td>320.89</td>
<td>1027</td>
<td>1783.9</td>
</tr>
<tr>
<td>Aluminum Alloy</td>
<td></td>
<td>305.54</td>
<td>940.85</td>
<td>1799.0</td>
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</table>

**C. Comparison between Experimental and Software Results**

<table>
<thead>
<tr>
<th>Natural Frequency Hz</th>
<th>Experimental</th>
<th>FEM</th>
<th>% Difference</th>
</tr>
</thead>
<tbody>
<tr>
<td>(\omega_1, \omega_2, \omega_3) (steel)</td>
<td>424</td>
<td>433</td>
<td>2.25</td>
</tr>
<tr>
<td>No change in geometry</td>
<td>1334</td>
<td>1368</td>
<td>2.76</td>
</tr>
<tr>
<td>10% reduction in diameter</td>
<td>2317</td>
<td>2375</td>
<td>2.66</td>
</tr>
<tr>
<td>15% reduction in diameter</td>
<td>354</td>
<td>362</td>
<td>2.53</td>
</tr>
<tr>
<td>10% reduction in length</td>
<td>1138</td>
<td>1167</td>
<td>2.9</td>
</tr>
<tr>
<td>15% reduction in length</td>
<td>1988</td>
<td>2043</td>
<td>2.84</td>
</tr>
<tr>
<td>10% reduction in shaft length</td>
<td>328</td>
<td>337</td>
<td>3.1</td>
</tr>
<tr>
<td>15% reduction in shaft length</td>
<td>1078</td>
<td>1100</td>
<td>1.96</td>
</tr>
<tr>
<td>10% reduction in shaft length</td>
<td>1870</td>
<td>1936</td>
<td>2.58</td>
</tr>
<tr>
<td>15% reduction in shaft length</td>
<td>698</td>
<td>716</td>
<td>2.66</td>
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<td>10% reduction in length</td>
<td>2170</td>
<td>2222</td>
<td>2.53</td>
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<tr>
<td>15% reduction in shaft length</td>
<td>3803</td>
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<td>2.9</td>
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<td>10% reduction in shaft length</td>
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<td>755</td>
<td>2.84</td>
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<tr>
<td>15% reduction in shaft length</td>
<td>2276</td>
<td>2345</td>
<td>3.1</td>
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<tr>
<td>10% reduction in shaft length</td>
<td>3937</td>
<td>4041</td>
<td>2.66</td>
</tr>
</tbody>
</table>

**IX. GRAPHICAL REPRESENTATION OF RESULTS**
A. Effect of Change in Shaft Diameter with Natural Frequency

Effect of change in shaft diameter with natural frequency: From the graph of change in shaft diameter with natural frequency, the effect of geometrical parameter like change in diameter on natural frequency is found significantly at higher modes.

B. Effect of Change in Shaft Length with Natural Frequency

Effect of change in shaft length with natural frequency: From the graph of change in shaft length with natural frequency, the effect of geometrical parameter like change in length on natural frequency is found significantly at higher modes.

C. Effect of Material Parameters on Natural frequencies for Intermediate Shaft

Effect of material parameters on natural frequencies for intermediate shaft: From the graph of effect of material parameters on natural frequencies for intermediate shaft, significant change in natural frequency is observed between aluminum alloy and brass.

D. Comparison Between Experimental and Software Results

Comparison between Experimental and FEA results: From the graph of comparison between FEM and Experimental results almost the natural frequencies are constant and significant difference is found at higher modes of vibrations.

X. CONCLUSION

i. From the analysis, the effect of the change in the diameter of the intermediate shaft on natural frequency is found to be significant at higher modes.
ii. From the analysis, the effect of the change in the length of the intermediate shaft is found to be significant at 15% of the original length.
iii. From the analysis of change in materials, significant difference in the natural frequency is observed between steel and brass material.
iv. From the comparison of FEM and Experimental results, in case of intermediate shaft, the average deviation between the results is 0.976.
v. From the analysis it is possible to avoid critical speeds, and hence to avoid resonance.
REFERENCES


