ISSN 2395-1621

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.**

ARTICLE INFO

Article History

Received :18th November 2015

Received in revised form :

19th November 2015

Accepted : 21st November , 2015

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.

I. SCOPE OF WORK

The calculations of bending natural frequency of simple shaft in rigid bearings are somewhat an easy matter. The problem in practice becomes complex because of the gyroscopic effects of disks, dissimilar moments of area of the shaft, stiffness and damping properties of the oil film of bearings, and coupling between two rotors. To avoid the failures of shafting, the general practice in the design of rotors is to determine the bending critical speeds, checking the out-of-balance response, and adopting a suitable balancing procedure. Different parameters can be varied such as geometrical parameters and material parameters for the calculation of the natural frequency, and the critical speed of the shaft, to study the effect of same. Here, efforts are made in the same direction to calculate the natural frequencies with variation in geometrical and material parameters to avoid resonance. This is carried out by using the finite element analysis and experimental verification with 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 .

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.

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)

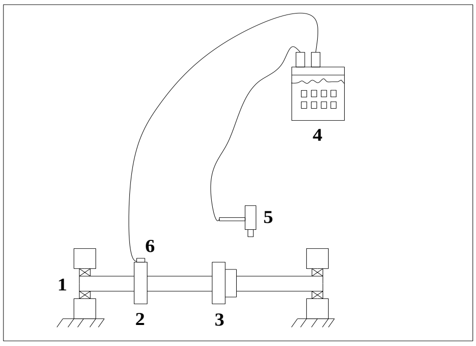
C. Study of vibration spectrums of the shaft with change in material parameters.

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. Acceleromete



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

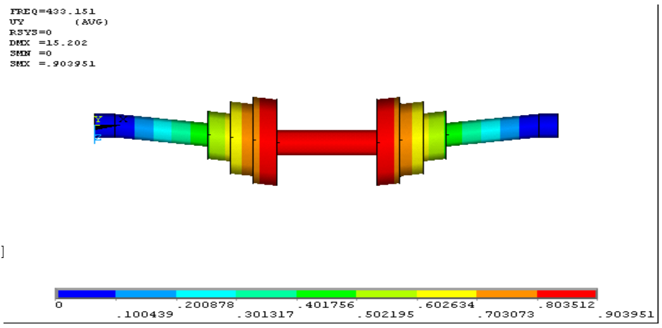


Fig. 1 First Mode

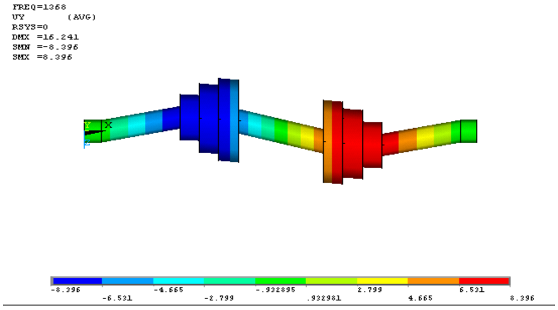


Fig.2. Second Mode

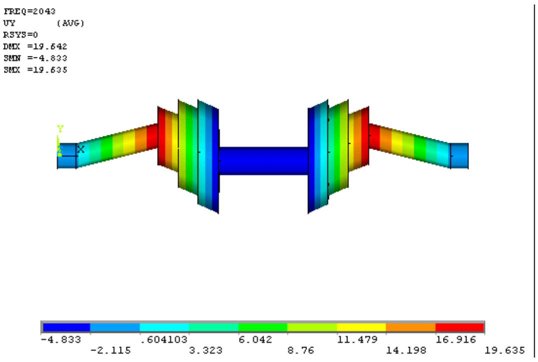


Fig.2. Third Mode

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*

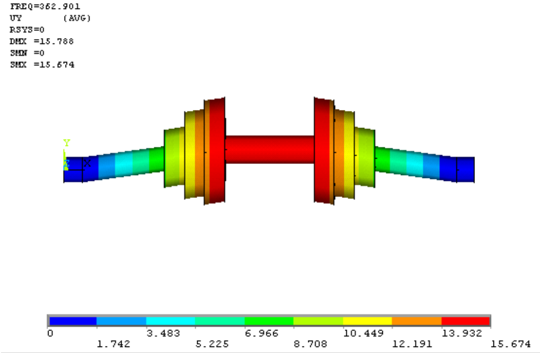
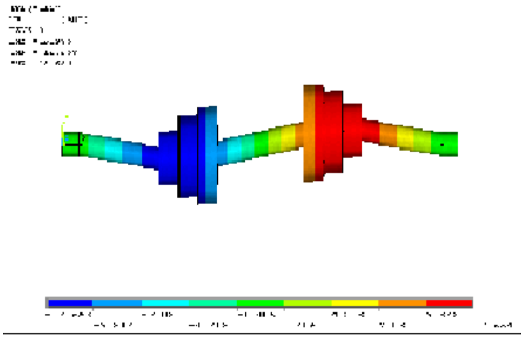
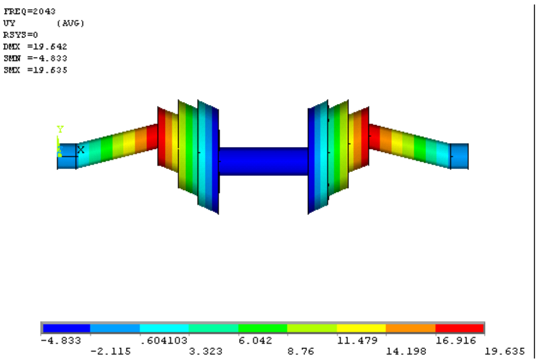


Fig. 4 First Mode (10% reduction in shaft diameter)





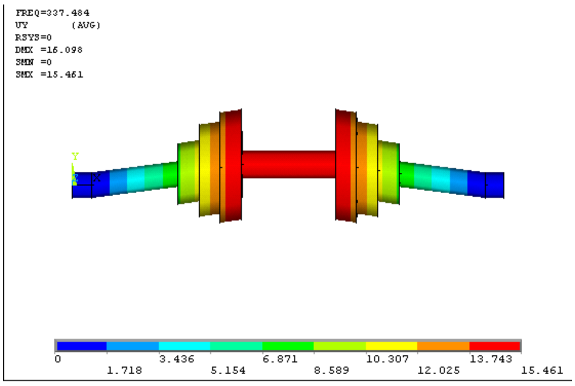
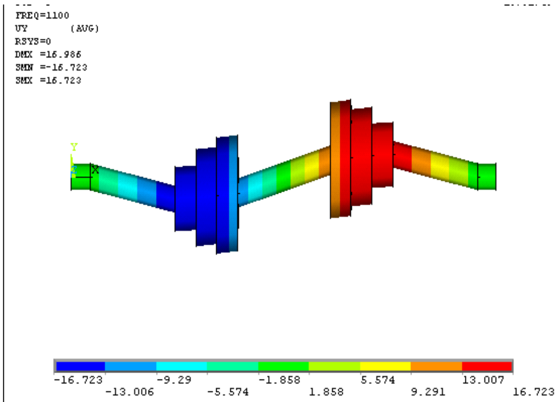


Fig. 5 Second Mode (10% reduction in shaft diameter)

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)



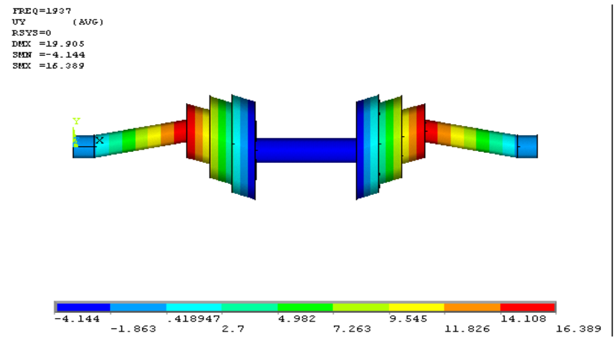


Fig. 9 Third Mode (15% reduction in shaft diameter)

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 third mode around the pulley location.

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

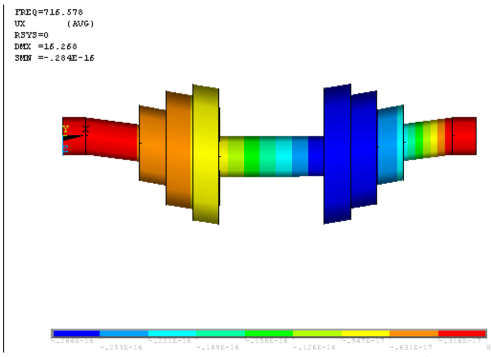


Fig. 10 First Mode (10% reduction in shaft length)

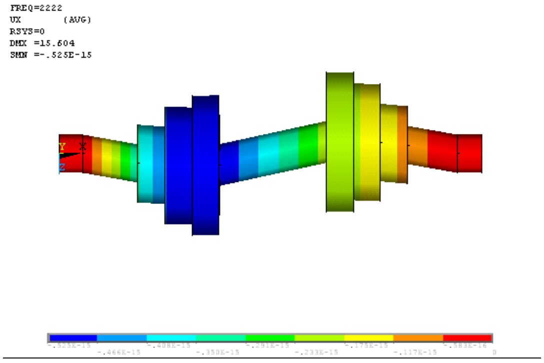


Fig. 11 Second Mode (10% reduction in shaft length)

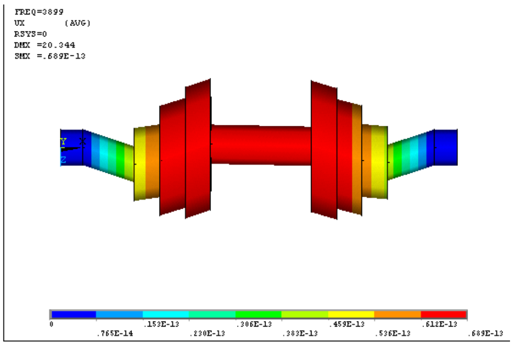


Fig. 12 Third Mode (10% reduction in shaft length)

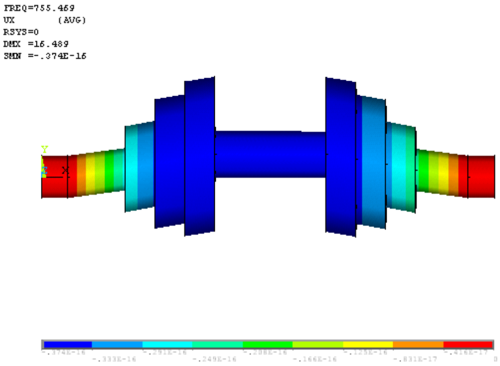


Fig. 13 First Mode (15% reduction in shaft length)

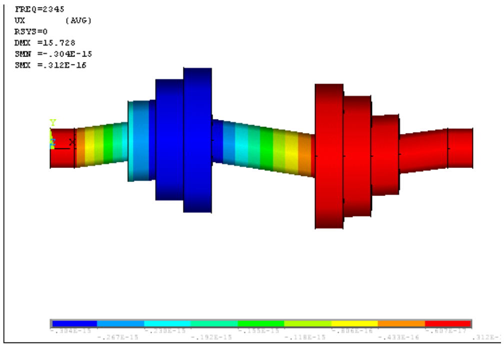


Fig. 14 Second Mode (15% reduction in shaft length

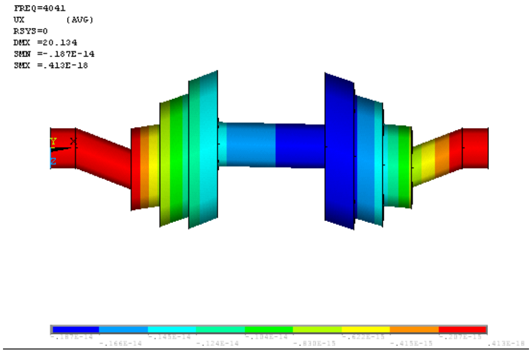


Fig. 15 Third Mode (15% reduction in shaft length

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

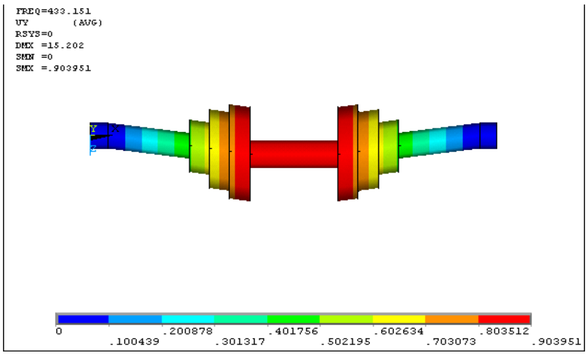


Fig. 16 First Mode (Steel shaft)

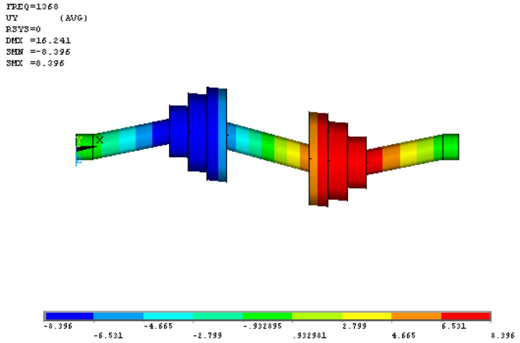


Fig. 17 Second Mode (Steel shaft)

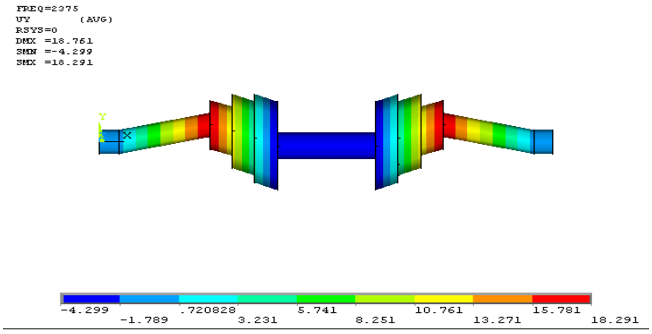


Fig. 18 Third Mode (Steel shaft)

Analysis of Results for Intermediate Shaft of Steel: Fig 16 – Fig 18 show an intermediate shaft made of steel with pulleys. Fig shows 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

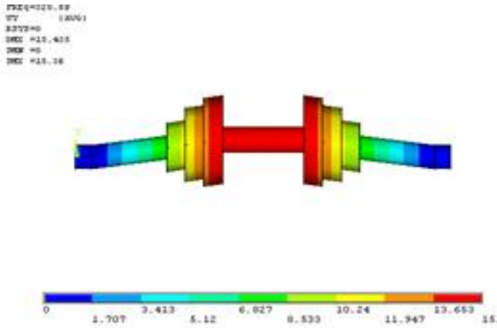


Fig. 19 First Mode (Brass shaft)

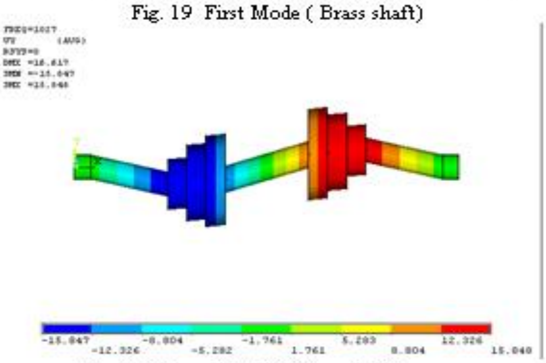


Fig. 20 Second Mode (Brass shaft)

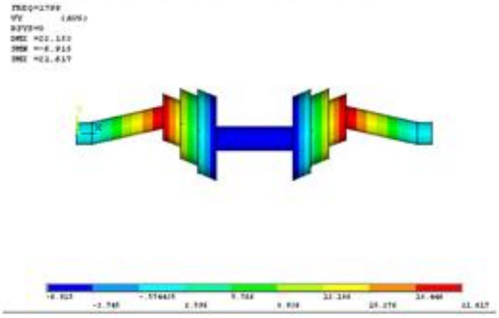


Fig. 21 Third Mode (Brass shaft)

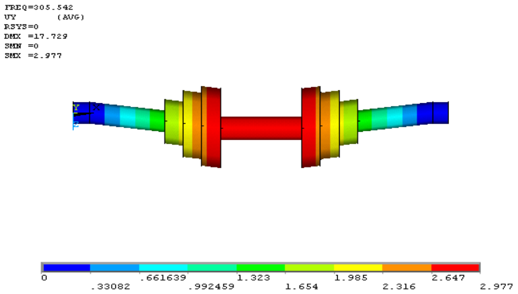


Fig. 22 First Mode (Aluminium alloy shaft)

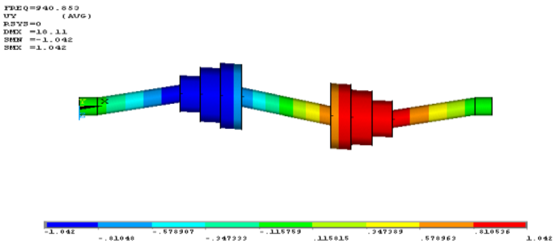


Fig. 23 Second Mode (Aluminium alloy shaft

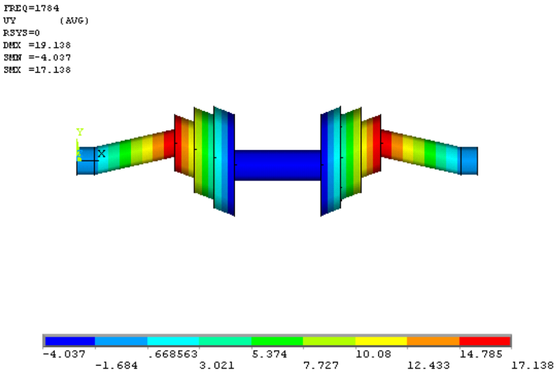


Fig. 24 Third Mode (Aluminium alloy shaft)

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

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

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

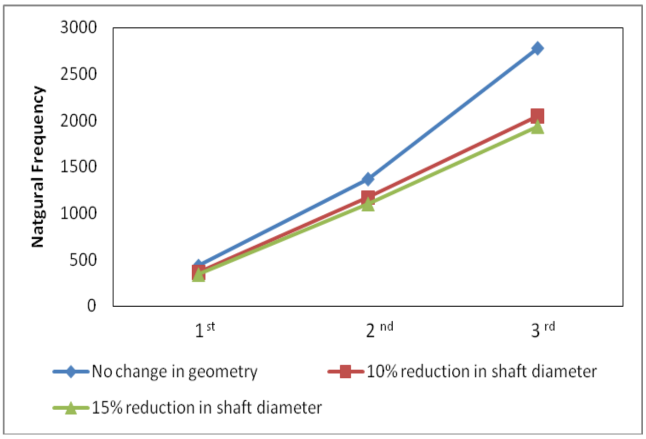
Shaft Geometry	Natural Frequencies in Hz		
	ω1	ω2	ω3
Steel	433.15	1368.3	2775.3
Brass	320.89	1027	1783.9
Aluminum Alloy	305.54	940.85	1799.0

C.Comparison between Experimental and Software Results

Natural Frequency Hz	Experimental	FEM	% Difference
ω1,ω2,ω3(steel) No change in geometry	424	433	2.25
	1334	1368	2.76
	2317	2375	2.66
10% reduction in diameter	354	362	2.53
	1138	1167	2.9
	1988	2043	2.84
15% reduction in diameter	328	337	3.1
	1078	1100	1.96
	1870	1936	2.58
10% reduction in length	698	716	2.66
	2170	2222	2.53
	3803	3899	2.9
15% reduction in shaft length	732	755	2.84
	2276	2345	3.1
	3937	4041	2.66
Brass	312	320	2.53
	997	1027	2.9
	1740	1783	2.84
Aluminum Alloy	294	305	3.1
	918	940	2.66
	1751	1799	2.53

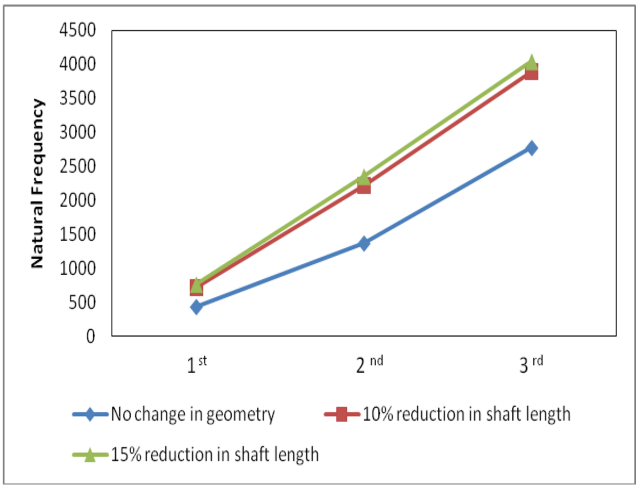
IX. GRAPHICAL REPRESENTATION OF RESULTS

A. *Effect of Change in Shaft Diameter with Natural Frequency*



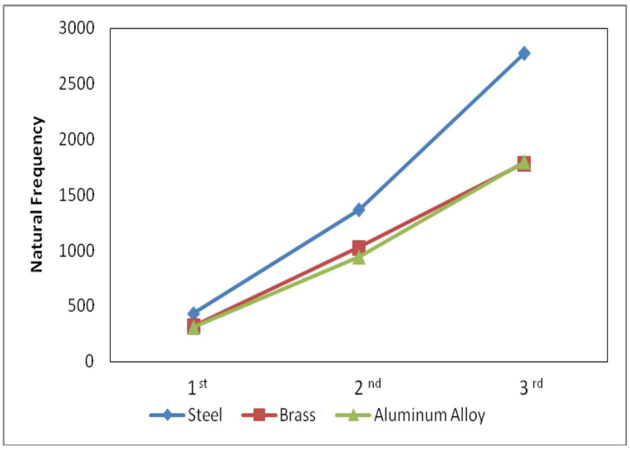
Effect of change in shaft diameter with natural frequency: From the graph of change in shaft diameter with natural frequency, the effect of geometrical parameter like change in diameter on natural frequency is found significantly at higher modes.

B. *Effect of Change in Shaft Length with Natural Frequency*



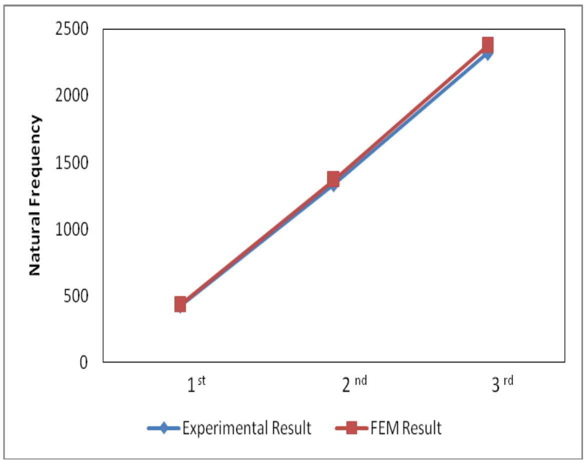
Effect of change in shaft length with natural frequency;From the graph of change in shaft length with natural frequency, the effect of geometrical parameter like change in length on natural frequency is found significantly at higher modes.

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

[1] Dunkerley, S. whirling of shafts, Phil. Trans. Roy. Soc., Series A, Vol. 185, 1894, P. 279.

[2] Prohol, M. A. A general method for calculating critical speeds of flexible rotors, Trans. ASME, 1945, P. A. – 142.

[3] Rao, J. S. and Reddy, K. B. V. Flexural Vibrations of simply supported beams, Proc. 11th Cong. Ind. Soc. Theo. and Appld. Mechs., 1966, P. 128.

[4] G.N.D.S. Sudhakar, &A.S.Sekhar, "Identification of unbalance in rotor bearing system", Journal of Sound and Vibration (2010)

[5] Hsiang-chieh Yu,Yin-Hwang Lin,Chih-Liang Chu,"Robust modal vibration suppression of a flexible rotor"

[6] Kimball and Hull, "Vibration Phonomena of Loaded Unbalanced Shaft," A. S. M. E. Trans, 1925, P. 673

[7] Jeffcot, H.H. The lateral vibration of loaded shafts in the neighborhood of a whirling speed, Phil. Mag., series 6,vol.37,2008 p,p.304.

[8] Rao J. S. and Gupta, K., A n introductory course on theory and practice of mechanical vibrations, Wiley publication, 2006.

[9] Reyleigh, J. W. S. Theory of sound, Dover Publication,2008.

[10] Jacobsen, L.S.and Aure, R.S.Engineering vibrations, McGraw–Hill Book Co, 2010.

[11] Myklestad, N. O. A new method of calculating natural modes of uncoupled bending vibrations of airplane wings and other type of beams, J. Aero. Soci.,2009, P. 153.