International Journal of Scientific Research & Engineering Trends
Volume 4, Issue 1, Jan-Feb-2018, ISSN (Online): 2395-566X

Experimental Investigation of Vibration Localization in Flexible Rotating Shafts

Pradip G. Gavade[1]., Vivek V. Patil[2]
[1]Government Polytechnic Dhule, Maharashtra, India
[2]Government Polytechnic Awasari, Maharashtra, India

Abstract- This study presents an experimental investigation of vibration localization in flexible rotating shafts under simply supported boundary conditions. Three shaft configurations were considered: circular stainless steel 304, non-circular stainless steel 304, and circular brass. The experimental program used an accelerometer, impact hammer, FFT analyzer, DC motor, coupling, and bearing supports to determine the natural frequencies and steady-state vibration responses. The non-circular shaft introduced a small difference between the two lateral moments of inertia; therefore, it represents a mistuned configuration. The experimental natural frequencies of 85.23 Hz, 89.84 Hz, and 75.78 Hz were obtained for circular stainless steel, non-circular stainless steel, and circular brass were 85.23, 89.84, and 75.78 Hz, respectively. The normalized forced responses increased with rotational speed, demonstrating the sensitivity of the vibration response to rotational speed and structural asymmetry. At 1000, 2000, and 3000 rpm, normalized responses of 0.03384, 0.18, and 0.5259 were obtained for circular stainless steel; 0.03565, 0.159, and 0.448 for non-circular stainless steel; and 0.0508, 0.2398, and 0.7706 for circular brass.

Keywords- Rotating shaft; Vibration localization; Experimental modal analysis; FFT analyzer; Accelerometer

I. INTRODUCTION

Rotating shafts are fundamental structural members in turbines, generators, motors, pumps, and internal-combustion-engine systems. Increasing the rotational speed and reducing the structural stiffness make the dynamic behavior of flexible shafts increasingly important. Excessive vibration can produce high stresses, accelerated wear, and reduced machine reliability. Vibration localization is particularly important when a shaft has structural asymmetry, because a small difference in the lateral moments of inertia can cause the vibration energy to become preferentially concentrated in one principal direction [1–7].

Previous studies have established that structural irregularity can produce localized vibration modes and that the interaction between disorder and internal coupling is important [1–3]. Studies of rotating shafts further showed that gyroscopic coupling varies with rotational speed and can produce directional localization in flexible shafts with dissimilar lateral moments of inertia [4–7]. The present work focuses on the experimental side of this problem. The objective was to measure the natural frequencies and steady-state responses of three shaft configurations and examine how the response changes with rotational speed.

The novelty of this study is the focused presentation of the experimental methodology and measured response data from the supplied shaft study, organized as a research-paper format. The work provides an experimentally grounded basis for subsequent analytical and finite-element comparison.

II. METHODOLOGY

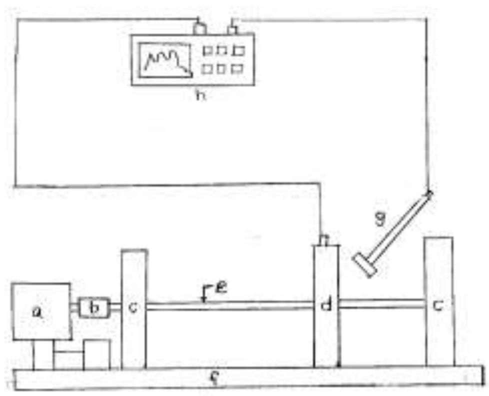

Figure 1. Schematic Diagram of the Experimental Setup

a- D.C. Motor, b- Rubber coupling , c- Bearing support, d- Transducer stand , e- Rotating member , f- Foundation g- Hammer, h- F-F.T analyzer

The experimental arrangement consisted of a DC motor, coupling, bearing supports, rotating shaft, accelerometer, impact hammer, and FFT analyzer. The accelerometer was mounted near the shaft center to capture the vibration response. The impact hammer provided broadband excitation during modal testing, whereas the FFT analyzer converted the measured time-domain signal into a frequency-domain spectrum. The experimental procedure also included rotation tests at 1000, 2000, and 3000 rpm. The supplied study reports that good test quality is associated with coherence values close to unity

Measurement procedure

First, the accelerometer was mounted on the shaft and connected to the second FFT channel. The impact hammer was connected to the first channel. The shaft was excited, and the frequency-response spectrum was recorded. The natural frequency was identified from a prominent spectral peak. The same procedure was repeated for different shaft materials and cross sections. The shafts were then rotated at selected speeds, and the steady-state vibration spectra were recorded. The response was normalized to enable consistent comparison of the results for different configurations.

III. MODELING AND ANALYSIS

Natural-frequency basis of Rayleigh model metho

The theoretical investigation of the mode localization of the phenomenon is analyzed. Compared to Galerkin’s method and the perturbation method, the Rayleigh model method is very simple. Therefore, a natural frequency analysis was performed using the Rayleigh model method. The natural frequency of simply supported beam is given by

$$\omega_n = \sqrt{\frac{EI}{m}}\,\frac{\pi^2}{l^2}$$

Where, E= Young’s modulus (N/m2)
I=Mass moment of inertia (m4)
l length of beam.(m)
m=mass per unit length

Now, if M_eq is the equivalent mass at the centre of the beam then the natural frequency of this equivalent system is given by

$$\omega_{EB} = \sqrt{\frac{48EI}{M_{Eq}l^3}} \qquad \ldots 2$$

_(EB)is fundamental natural frequency of a non-rotating circular Euler-Bernoulli beam with simply supported boundary conditions ,while calculating natural frequencies for the different materials different dimensions and structural properties has to be considered which are given below [Table 1].

Table 1: Dimensions and Structural properties of shaft

Sr.No.	Parameter	Model 1	Model 2	Model 3
1	Cross section	Circular shaft	Non circular shaft	Circular shaft
2	Material	Stainless steel(304)	Stainless steel(304)	Brass
3	Density , ρ (kg/m^3)	7920	7920	8400
4	Young’s modulus , E(N/m^2)	190 x 10^9	190 x 10^9	110 x 10^9
5	Total length L(m)	0.3302	0.3302	0.3302
6	Length between support (m)	0.30	0.30	0.30
7	Diameter (m)	8.9 x 10^{-3}	8.9x 10^{-3} and 8.8x 10^{-3}	8.9x 10^{-3}
8	Area of cross section , ((m^2)	7.202 x 10^{-4}	6.991 x 10^{-4}	7.202 x 10^{-4}
9	Total mass , M (kg)	0.147	0.145	0.155
10	Moment of inertia , I(m^4)	3.079 x 10^{-10}	3.079x $10^{-10}$and 3.011x 10^{-10}	3.079 x 10^{-10}
11	Radius of gyration , r_k (m)	2.225x 10^{-3}	2.225 x $10^{-3}$and 2.20 x 10^{-3}	2.225x 10^{-3}
12	Aspect ratio , $A_r = r_k/L$	7.416x 10^{-3}	7.416x $10^{-3}$and 7.33x 10^{-3}	7.416x 10^{-3}
13	Ratio of I , D= I_2/I_1	1	0 .9555	1
14	Disorder δ=1-D	0	0.0445	0

International Journal of Scientific Research & Engineering Trends
Volume 4, Issue 1, Jan-Feb-2018, ISSN (Online): 2395-566X

Rotational-speed normalization

The measured rotational speed N is converted to rotational frequency Ω in cycles per second by $\Omega = N/60$. The normalized speed is then defined as $\Omega = \Omega/\omega\_EB$. This non-dimensional representation allows responses from shafts with different material properties and natural frequencies to be compared on a common basis.

Experimental response metric

The steady-state response due to an unbalanced mass is represented by a normalized displacement response. The experimental study evaluates the response at the shaft midpoint and compares the response with normalized rotational speed. A rising normalized response with speed is interpreted as increasing dynamic excitation and increasing localization tendency, especially for the mistuned non-circular shaft.

IV. RESULTS AND DISCUSSION

Experimental natural frequencies

. For experimental analysis FFT analyzer is used with impact hammer as an excitation source which is connected to channel 1. Accelerometer which is output device is connected to channel 2. Natural frequencies of the system can be obtained by proper positioning of accelerometer test would be conducted as follows:

Accelerometer mounted on shaft with help of casing and connected to channel 2.
Impact hammer is connected to channel 1.
The F-F-T analyzer and impact hammer are used to excite and to acquire the steady state response due to an unbalanced mass for different stationary member, viz stainless steel (304), brass etc. The natural frequencies of the shaft at point where amplitude verses frequencies plot picks.

Results obtained from experimental analysis with help of FFT analyzer are shown in following Steady state response graph as fig. 2

Circular Stainless steel (304)

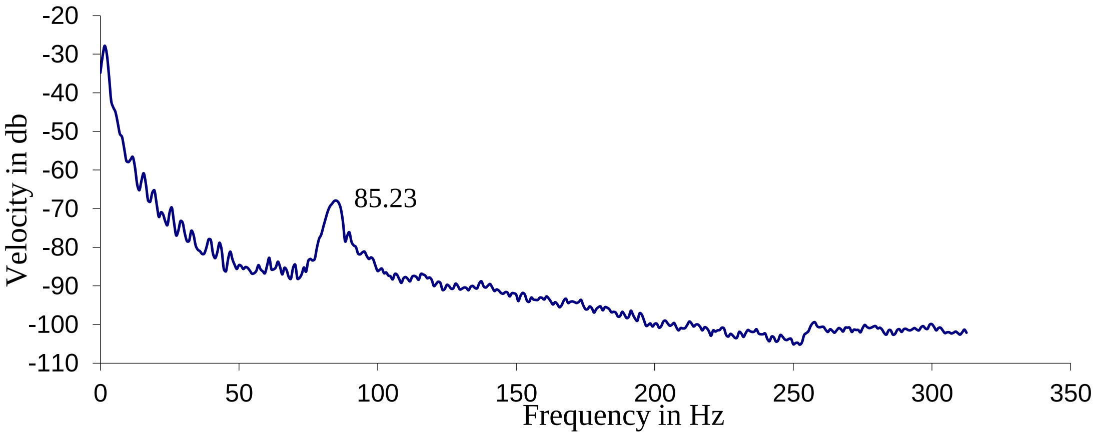

Fig 2 (a)

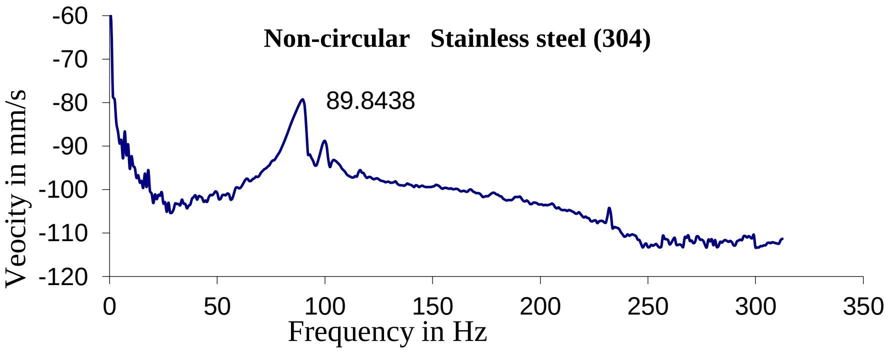

Fig 2 (b)

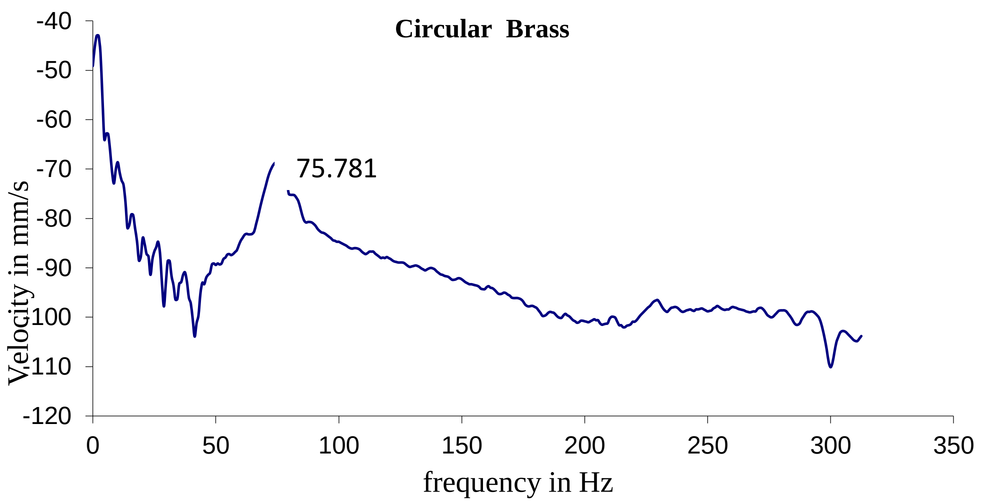

Fig 2 (c)

Fig 2 Steady state response graph

Table 2: Experimental Steady state response results

Sr. No.	Material	Cross section	Natural frequency (Hz)
1	Stainless steel (SS304)	Circular	85.23
2	Stainless steel (SS304)	Non circular	89.84
3	Brass	Circular	75.78

The circular SS 304 shaft exhibits a natural frequency of 85.23 Hz, while the non-circular SS 304 shaft gives 89.84 Hz. The difference is associated with the changed cross-sectional properties. The brass shaft has the lowest measured natural frequency, 75.78 Hz, consistent with its lower Young's modulus compared with SS 304. The results therefore demonstrate the strong influence of material stiffness and cross-sectional properties on shaft dynamics.

Again, the different shafts are rotated by using suitable motor to different speeds such as 1000 rpm, 2000 rpm, 3000 rpm respectively and the data is recorded by using F-F-T analyzer.

Results obtained from experimental analysis with help of FFT analyzer are shown in following graph as fig.2

Effect of rotational speed on normalized response

Experimental response and Normalized speed calculation given below

1) Circular stainless steel 304

$$u_{1p} = \frac{\Omega^2}{\left(\pi^2\sqrt{\frac{EI^2}{eAl^4}}\right)^2 + ((n\pi A_{r2})^2 - 1)\Omega^2}$$

Speed N= 1000 RPM, Real Speed $\Omega = \frac{1000}{60} = 16.667$Hz

$\omega_n = 85.516\ Hz$, Normalized speed =Ω/ωEB=0.1957

$u_{1p} = \frac{16.667^2}{(85.516)^2+((1\times\pi\times7.416\times10^{-3})^2-1)(16.667)^2} =$ 0.03984

2) Non Circular stainless steel 304

$$u_{1p} = \frac{\Omega^2}{\left(\pi^2\sqrt{\frac{EI^2}{eAl^4}}\right)^2 + ((n\pi A_{r2})^2 - 1)\Omega^2}$$

Speed N= 1000 RPM, Real Speed $\Omega = \frac{1000}{60}$= 16.667Hz

$\omega_n = 89.84\ Hz$, Normalized speed = Ω/ωEB=0.1855

$= \frac{16.667^2}{(89.84)^2+((1\times\pi\times7.416\times10^{-3})^2-1)(16.667)^2} =$ 0.03565

3) Circular brass

$$u_{1p} = \frac{\Omega^2}{\left(\pi^2\sqrt{\frac{EI^2}{eAl^4}}\right)^2 + ((n\pi A_{r2})^2 - 1)\Omega^2}$$

Speed N= 1000 RPM, Real Speed $\Omega = \frac{1000}{60} = 16.667$Hz

$\omega_n = 750781\, Hz,$, Normalized speed = $\Omega/\omega EB$=0.2199

$$u_{1p} = \frac{16.667^2}{(750781)^2+((1\times\pi\times7.416\times10^{-3})^2-1)(16.667)^2}$$

= 0.0508

Similar calculations have been done for different values of speeds and presented in table 3

Table 2: Experimental normalized response and Normalized speed

Shaft	Speed (rpm)	Ω (Hz)	Normalized speed	Experimental normalized response
Circular SS 304	1000	16.67	0.1957	0.03384
Circular SS 304	2000	33.33	0.3910	0.1800
Circular SS 304	3000	50.00	0.5871	0.5259
Non-circular SS 304	1000	16.67	0.1855	0.03565
Non-circular SS 304	2000	33.33	0.3709	0.1590
Non-circular SS 304	3000	50.00	0.5565	0.4480
Circular brass	1000	16.67	0.2199	0.0508
Circular brass	2000	33.33	0.4398	0.2398
Circular brass	3000	50.00	0.6597	0.7706

Above results are shown experimental graphs as fig.3

Three configurations showed an increase in the normalized response with increasing rotational speed. For the circular SS 304, the response increased from 0.03384 at 1000 rpm to 0.5259 at 3000 rpm, representing an approximately 15.5-fold increase. For non-non circular304, the response increased from 0.03565 to 0.448, approximately 12.6-fold. The circular brass shaft exhibited the largest response increase, from 0.0508 to 0.7706, approximately 15.2-fold. These trends are consistent with the general behavior reported in the supplied study: increasing speed increases the forced response and can intensify localization in flexible rotating shafts.

The non-circular shaft is particularly important because its unequal lateral moments of inertia create structural disorder. Theoretical work in the literature shows that disorder combined with gyroscopic coupling can produce directional vibration localization [2, 6, and 7]. The experimental data therefore support the physical interpretation that rotational speed is an important control parameter for the vibration response. Near critical-speed conditions, a rapid rise in response is expected; consequently, operation close to such conditions should be avoided unless the rotor system is specifically designed for them.

Circular stainless steel (304)

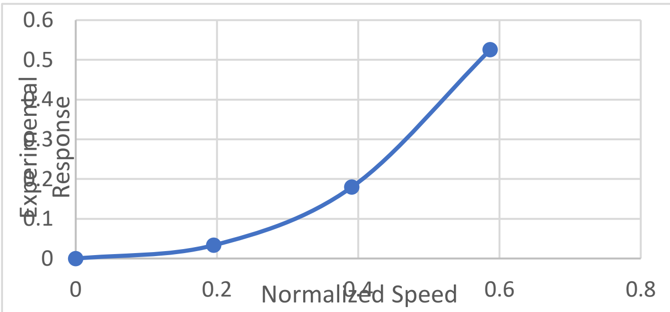

Fig 3 (a)

2) Non circular stainless steel (304)

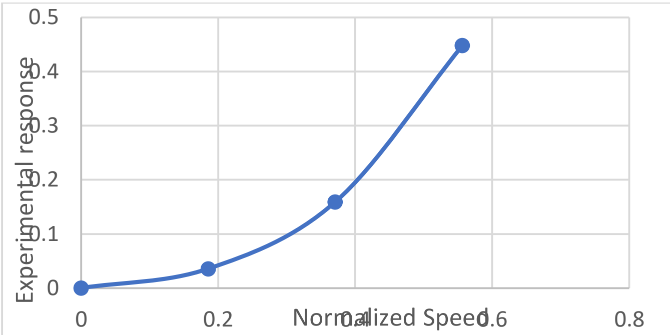

Fig 3 (b)

3) Circular Brass

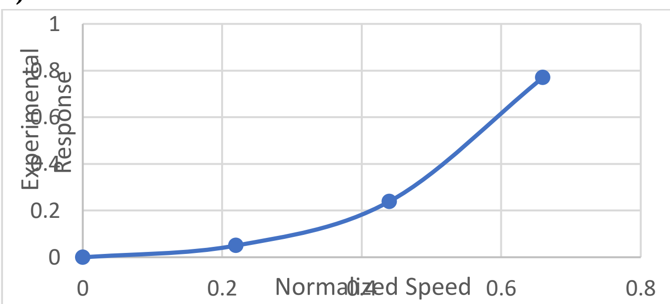

Fig 3 (c)

Fig 3 Experimental response Vs Normalized speed

V. CONCLUSION

An experimental investigation of vibration localization in flexible rotating shafts was presented using FFT-based modal testing. Three configurations—circular SS 304, non-non circular304, and circular brass—were evaluated. The measured natural frequencies were 85.23 Hz, 89.84 Hz and 75.78 Hz, respectively. The results demonstrate that material stiffness and cross-sectional configuration strongly influence the natural frequency. The normalized steady-state response increased systematically with rotational speed for all three shaft configurations. The largest response at 3000 rpm was measured for the circular brass shaft, while the non-non circular304 shaft provided a representative mistuned configuration for studying directional localization. The experimental methodology provides a practical validation route for analytical and finite-element models and can be extended to a wider range of rotor speeds, shaft geometries, and boundary conditions.

REFERENCES

1. C. H. Hodges, “Confinement of vibration by structural irregularities,” Journal of Sound and Vibration, 82(3), 1982.
2. C. Pierre and E. H. Dowell, “Localization of vibrations by structural irregularity,” Journal of Sound and Vibration, 114(3), 549–564, 1987. Doi: 10.1016/S0022-460X (87)80023-8.
3. J. Woodhouse and C. H. Hodges, “Vibration isolation from irregularity in a nearly periodic structure: theory and measurement,” Journal of the Acoustical Society of America, 74(3), 1983.
4. C. W. Lee, R. Katz, A. G. Ulsoy and R. A. Scott, “Modal analysis of a distributed parameter rotating shaft,” Journal of Sound and Vibration, 122(1), 119–130, 1988. Doi: 10.1016/S0022-460X (88)80010-5.
5. S. H. Choi, C. Pierre and A. G. Ulsoy, “Consistent modeling of rotating Timoshenko shafts subject to axial loads,” Journal of Vibration and Acoustics, 114(2), 249–259, 1992.
6. A. G. Ulsoy, S. H. Choi and C. Pierre, “Vibration localization in rotating shafts, Part I: Theory,” Journal of Vibration and Acoustics, 120(1), 138–148, 1998. doi:10.1115/1.2893796.
7. A. G. Ulsoy, S. H. Choi and C. Pierre, “Vibration localization in rotating shafts, Part II: Experiment,” Journal of Vibration and Acoustics, 120(1), 149–155, 1998. doi:10.1115/1.2893798.
8. G. K. Grover, Mechanical Vibrations, Nem Chand & Bros., Roorkee, 7th ed., 2003.
9. V. P. Singh, Mechanical Vibrations, Dhanpat Rai & Co., Delhi, 2008.
10. V. B. Bhandari, Design of Machine Elements, Tata McGraw-Hill, Delhi.