DEVELOPMENT OF EXPERIMENTAL SETUP AND FEA INVESTIGATION OF TORSIONAL VIBRATIONS OF TWO MASS ROTOR SYSTEM

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ABSTRACT- Precise prediction of vibration characteristics is complicated phenomenon. This necessity leads to the development of experimental set-up for determining the response of damped, undamped, forced, and free vibrations on the system. The primary objective of this work is to build up experimental set-up to calculate the changes in natural frequency and mode shapes for different shaft diameters, for various material changes and for varying damping. In addition, consequence of system parameters combination can be evaluated to recognize the performance of the system. FEA results are listed to realize the effects on two mass rotor systems. FFT is been coupled with experimental set-up to compute frequency response of the system. As a result the developed setup is proficient to find out experimental vibration results.

Keywords- Two mass rotor, ANSYS, Torsional vibrations, Modal Analysis, FEA, Mode shapes, Natural frequency.

INTRODUCTION-

Study of torsional vibration is very significant aspect as many of applications where rotary movement of shaft been concerned. In order to recognize the dynamic performance of shaft under the influence of torsion the need of experimental arrangement is required. In general torsional vibrations referred as the vibrations with reference to the axis of shaft. It can be depicted as the oscillatory angular motion of shaft. In turbo-machineries, the driving force drives number of component, as a result rotational components gets rotated so they are frequently subjected to such periodic torsional vibrations. It is very important to investigate the behavior of shaft under such periodic oscillatory motions. This paper is concerned with the experimental and numerical investigation of torsional vibrations of two rotor shaft system. For this, a work of setup development for vertical arrangement of two rotor system is undertaken in order to facilitate further studies. Purpose of this paper is to discover the frequency response and mode shapes of vibrations and to study the influence of the system's parameters on frequency response.

From the theory of torsion of shaft-

$$\mathrm{Kt} = \frac{GJ}{l} \qquad \ldots (1)$$

Where, Kt = Torsional stiffness, G = Modulus of rigidity, l = Length of shaft, J = polar second moment of area of the shaft cross-section.

$$\mathrm{J} = \frac{\pi}{32} \times \mathrm{d}^4 \qquad \ldots (2)$$

For Torsional natural frequency-

$$\omega n = \sqrt{\frac{Kt}{Ip}} \qquad \dots (3)$$

Ip = Polar mass moment of inertia,

For the determination of fundamental natural frequency for mode shapes below equation is used.

$$\text{Fn} = \left(n - \frac{1}{2}\right) \times \frac{1}{2l} \times \sqrt{\frac{G}{\rho}} \qquad \dots (4)$$

Where Fn- natural frequency, l= length of shaft, ρ = density.

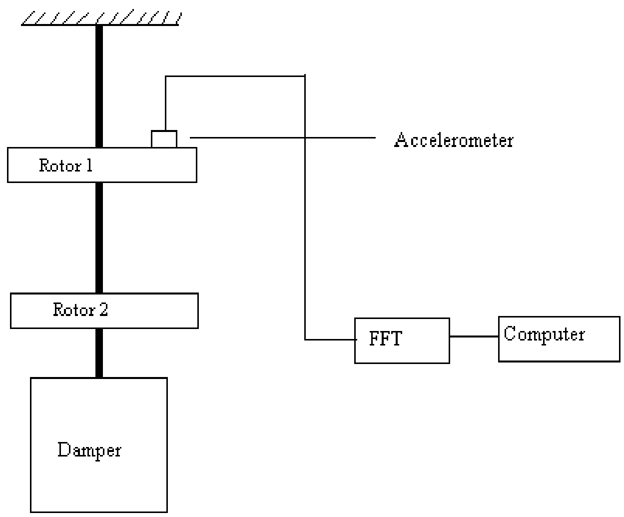

Fig 1. Schematic of experimental setup

Torsional vibrations occur because of inertia forces, impulsive loads coming on normal machine cycle and torques related to gear meshing but some forcing component functions are same as the undamped system in which excitation is free [1, 11]. Torsional behavior of system is analyzed by the frequency response and the mode shapes. The mode shapes of second, third, higher mode natural frequencies are taken into consideration very rarely and these higher-mode natural frequencies are very high and the frequency of actual excitation will not be within resonant range [2, 10, 12]. Machinery with various shafts, difficulty increases as natural frequency of the system comes close to the operating frequency range. Therefore tracking of torsional natural frequencies can be used as a health understanding feature in rotary machines [3, 9]. In such cases, analysis of torsional vibration becomes necessary to understand the behavior of complex system.

EXPERIMENTAL SETUP-

Fig. 2 shows the CAD model of experimental set up established. The setup consist of torsional vibration module in which major parts are two discs of different mass moment of inertias, three different chucks, damping pot. Chuck used have opening of 1.5mm to the 13 mm. Using single disc chuck arrangement we can change system to single degree of freedom system. For damping pot effortless removal and engagement, ring of 100 mm diameter is joined on base plate using adhesive joint method.

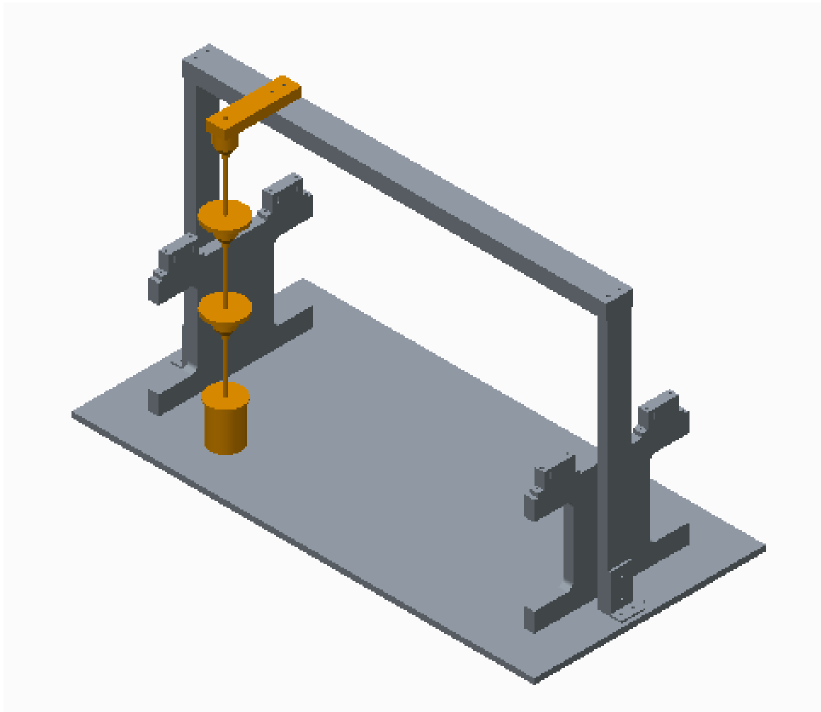

Fig 2. Model of experimental setup.

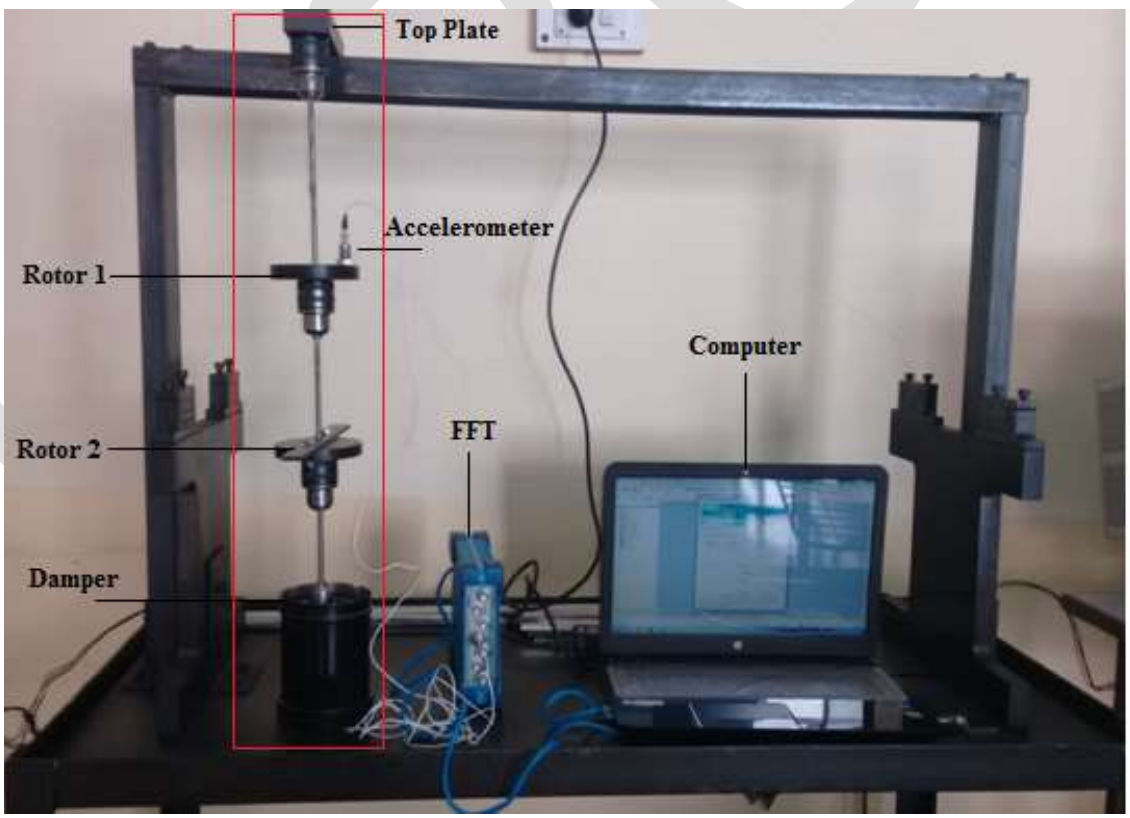

Fig 3. Manufactured experimental setup

For data acquisition as shown in figure 2 FFT is used. FFT of OROS made OR34 used which is having 4 channels. Accelerometer (DYTRAN) is connected to the FFT through which vibration data is captured. Accelerometer used is having sensitivity of 103.7mV/g. To acquire the data NV gate software used in connection with FFT.

FEATURES OF EXISTING SETUP-

Current presented setup consists of a variety of features. To make a model interchangeable every part of assembly is independent and has adjustment characteristic. With concern of building setup as a brief, wider provision of shaft diameter is been provided. Shaft diameter is key factor while taking into account the torsional behavior. It is achievable to perform experimentation with diameter ranges from 1 to the 10mm range to recognize the effects of torsional vibration. Interchangeability factor considered for damping pot also. A fixture of ring is been provided which holds the damping pot. Whenever the damping oil change requirement arises we can perform it with just replacing the pot, or replacing the oil in the pot. Rotor of different mass moment of inertia is been attached and it has adjusting characteristics, so we can adjust the rotor over the length of shaft wherever we want to fix. Rotors provided with holes on its peripheral in order to placement of extra mass to increase the mass moment of inertia. Provision for forced vibration arrangement by mounting motor can be possible. Along with the mass attachment the provision for making unbalance the extra arrangement specified. It is possible to measure the vibration level with balanced and unbalanced rotor system. Change in unbalance can be done with simply placing unbalance mass attachment on any of the disc. Overall length of shaft is been taken 700mm but can be increased if required. Each part of torsional assembly is distinct with the removal of damper pot and motor, system undergoes free vibration. Using above developed setup check for damped, undamped, free and forced vibration is possible in addition effect of parameter combination can be evaluated. Shaft crack detection can be possible on same experimental set-up to demonstrate the feasibility of torsional vibration as diagnostic method for shaft cracking detection and monitoring in rotating equipment in nuclear power plants [4].

ACTUAL SET UP WITH PARTS-

Fig 4. Torsional set up with parts

1 Top plate is made with mild steel material and chuck is integrated to the top plate using bush. Top plate is assembled on column where 700 mm height of shaft achieved. Mild steel material is used as it has low cost and sufficient hardness.

2 Rotors of two different diameters are manufactured with Mild steel material. Inertia effects kept different by varying the thickness of the plate. As shown in fig 5 both the rotors has adjusting characteristic and mentioned features by which we can fix the position of disc over the length of shaft.

Fig 5. Two rotors with chuck.

3 Damping pot of stainless steel is used to avoid the chemical effects of oil on damping pot. To increase the surface area of damping another disc is coupled with chuck and used to freely oscillate in the system to damp vibrations in the damping pot.

4 Shaft crack detection is possible through combining the parameters of the experiment and there relative frequency response gives the investigative response of crack behavior. [5, 6, 7, 8]

EXPERIMENTS TO BE PERFORMED-

1 One degree of freedom free vibration and forced vibration.

2 Two degree of freedom free vibration and forced vibration.

3 One and two degree of freedom free and forced with and without damping.

MERITS OF THE EXPERIMENTAL SET UP-

1 Cost of setup available in market is very high as compared to this set up. Up to 40% reduction in cost achieved.

2 Parameters range is very high compared with available setup in the market. Shaft diameter from 1mm to 10 mm can be varied.

FEA ANALYSIS-

To calculate the frequencies corresponding to the mode shapes, FEA of two rotor system is accomplished. Modal analysis is implemented to calculate the response of the system. ANSYS 14.5 version used for simulation and results were plotted for the developed system.

Material used for modal analysis of shaft has the following material properties.

Table 1. Material property of mild steel

Mild Steel	
Young's modulus (GPa)	210
Poisson's ratio	0.3
Density ($kg/m^{3)}$	7850

Result for first ten fundamental frequencies been extracted.

Table 2. FEA Result for mild steel

Modes (n)	Frequency by FEA (Hz)
1	3.579
2	3.786
3	23.197
4	25.241
5	26.718
6	63.366
7	70.695
8	74.681
9	154.220
10	163.945

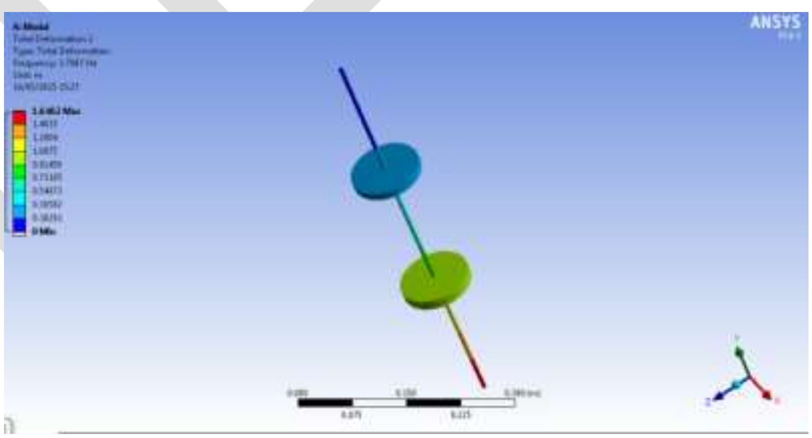

Fig 6. Frequency for mode shape 1

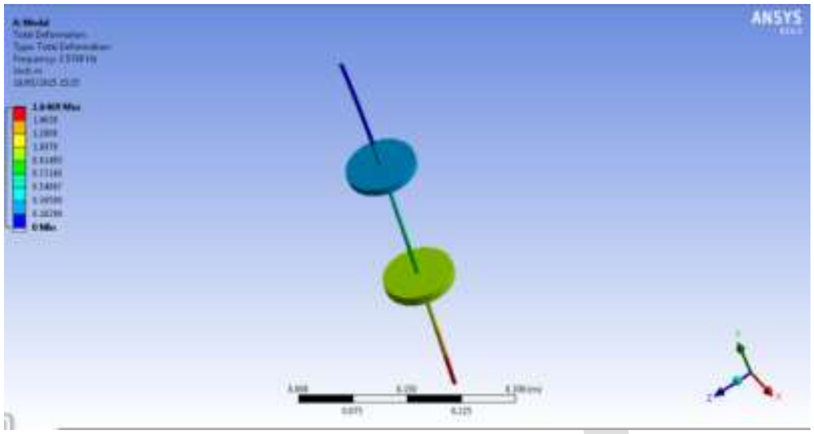

Fig 7. Frequency for mode shape 2

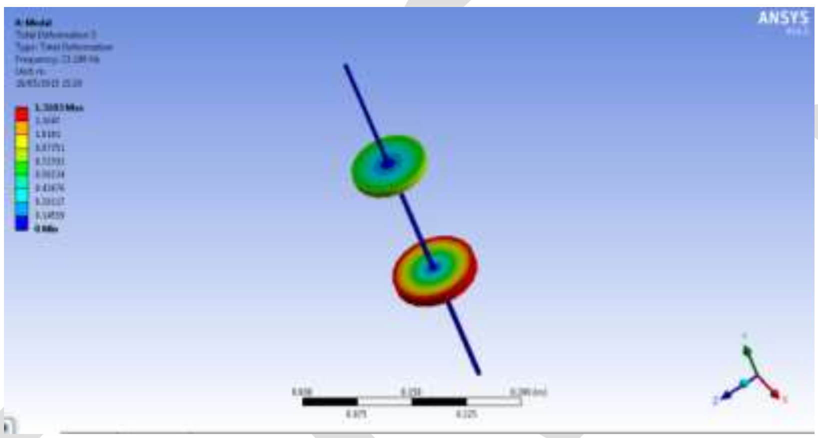

Fig 8. Frequency for mode shape 3

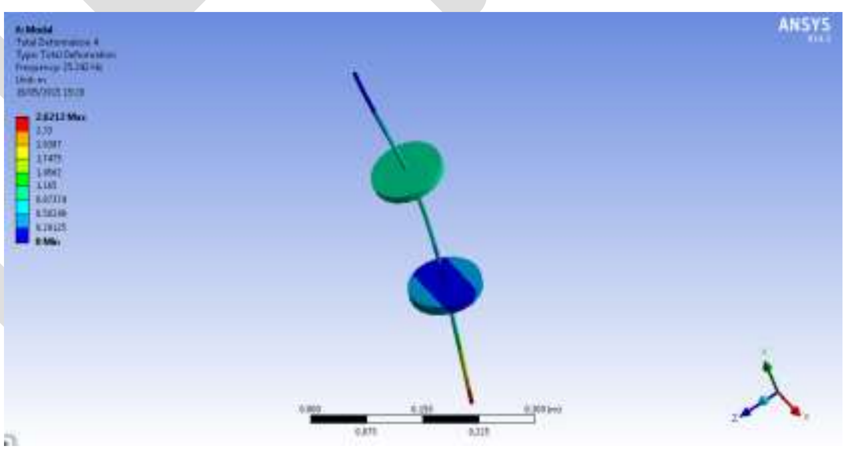

Fig 9. Frequency for mode shape 4

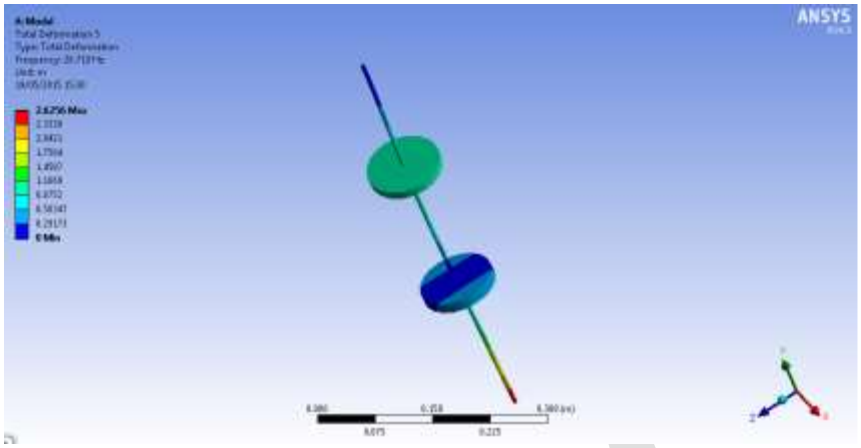

Fig- 10. Frequency for mode shape 5

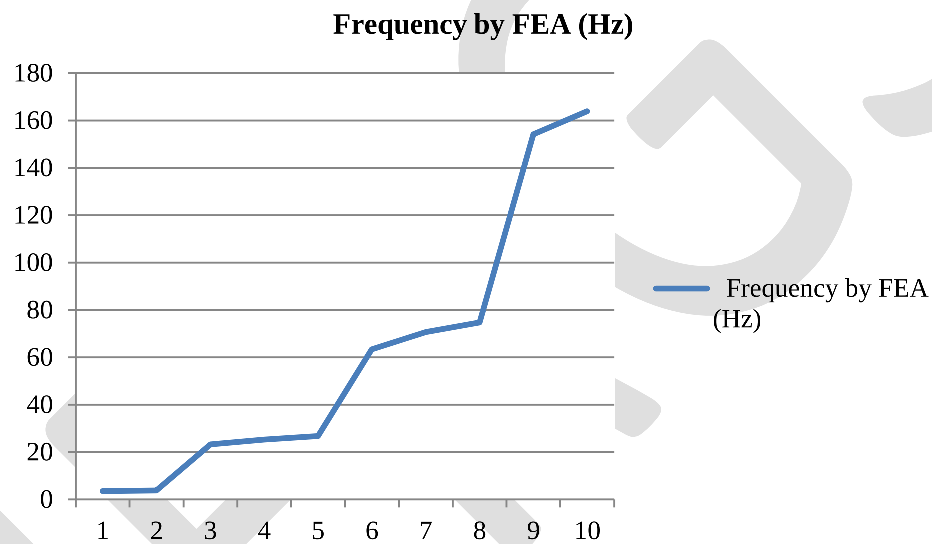

Fig 11. Graph for frequency vs. modes.

CONCLUSION-

Experimental setup for torsional vibration measurement has been successfully developed. The developed set up is simple in construction, accurate and least expensive. The results obtained are having good correlation with the results of FEA. Vertical arrangement of torsional vibration module is developed which defines the advantages of module and various experiments possible through above developed setup. In future shaft crack monitoring can be possible with same module.

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