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Experimental Investigation of Transverse Vibration Characteristics of Beams

Mahesha. T. M[1], Ranjith. K[2]
[1]M.Tech Scholar (MMD), [2]Assistant Professor, P.E.S College of Engineering, Mandya-571401

Abstract: This Article presents on Experimental Investigation of Transverse Vibration Characteristics of Beams. The fixture designed for 3 boundary conditions such as cantilever, simply supported and fixed-fixed end conditions and also to conduct set of experimental cases to determine the nature of forced vibration under the effect of varying length, effect of centrifugal force acting on the beam, effect of the location of excitation on beam. The experimental excitation frequency is found on the FFT graph and it is compared with theoretical natural frequency calculated from the Euler's Bernoulli's Beam Theory. The main objective is to analyze the transverse vibration characteristics of a flexible rectangular stainless steel beam. This determining the natural frequency of any system helps us to find how the system will behave when disturbed and left, and to find what kind of excitation frequency to be avoided in the system. The experiment was conducted in the PES college of engineering, Mandya. The results obtained are tabulated.
Keywords: Beams, Natural frequencies, Forced vibration,

I. INTRODUCTION

When external forces act on a vibrating system during its motion, it is termed Forced Vibration. Under this condition, the system will tend to vibrate at its own natural frequency superimposed upon the frequency of the exciting force. After a short time, the system will vibrate at the frequency of the exciting force only, irrespective of the initial conditions or natural frequency of the system. The latter case is termed steady-state vibration. In fact, most of the vibrational phenomena present in life are considered under forced vibration. When the excitation frequency is very close to the natural frequency of the system, vibration amplitude will be very large and damping will be essential to maintain the amplitude at a certain level. The later case is called "resonance" and it is very dangerous upon mechanical and structural elements. An example of structural failure under dynamic loading was the failure of well-known Tacoma Narrows Bridge U.S state of Washington during wind-induced vibration. Thus, care must be taken when designing a mechanical system by choosing the proper natural frequency that is suitably spaced from the exciting frequency.

In particular, models based on beam-like elements, with different boundary conditions, can be used to simulate the response of structures in engineering applications.

For example, we can model the vibrational response of spacecraft antennae, robot arms, building parts, bridge constructions and parts of musical instruments. Vehicle induced vibration of the bridge and various structures which may be simulated as beams and the effect of various parameters, such as suspension design, damping vehicle mass and velocity, matching among bridge and vehicle natural frequency, floor roughness etc., these dynamic behaviour of such structure where extensively investigated by a excessive number of researcher.

The entire research will certainly remain a major topic for future scientific researches, due to containing developments in design technology and application of new materials with better quality enable the construction of lighter and more slender structures, susceptible to dynamic and especially moving loads. One of the examples of Forced vibration is due to rotating eccentric masses, which is a centrifugal type constant periodic harmonic excitation. It is a common source of vibration excitation that one sees in everyday life like electric motors, pumps, fans, cloth dryers, vibrators, and compressors. It is caused when the centre of mass is out of alignment with the centre of rotation, resulting in the exciting centrifugal force acting radially outwards, with a maximum value of sinusoidal excitation in any direction. The main objective of the Experiment is to fabricate the fixture with the capability to perform almost all forced vibration experiments, to analyse the forced vibrations characteristics of the flexible rectangular stainless steel beam, to the theoretical calculation of natural frequency using Euler's Bernoulli's Beam theory, to Compare Theoretical obtained natural frequency with Experimental frequency obtained from the FFT graph, to Study the effect of centrifugal force, effect of the location of excitation, effect of boundary condition on vibration response of a beam. The present study deals with the experimental investigation on transverse vibration characteristics of the beam under different end conditions such as cantilever, simply supported and fixed-fixed condition.

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II. EXPERIMENTAL DETAILS

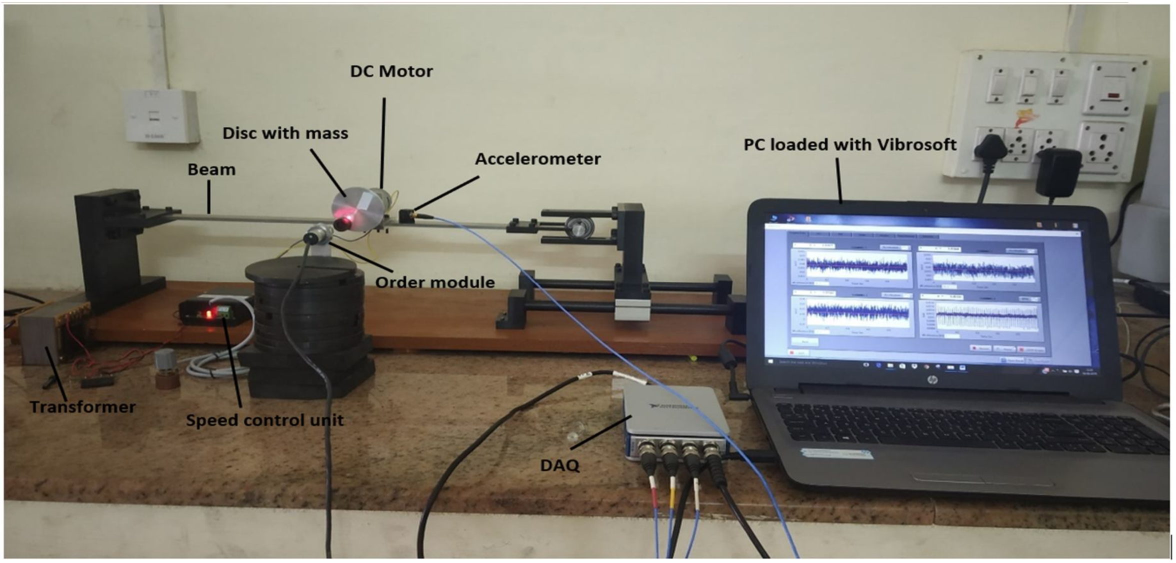

Fig.1: Experimental setup

Fig.1 shows the experimental setup used in the present investigation. The beam is fixed to the fabricated fixture and DC motor is rigidly bolted on 0.5*length or 0.6*length of the beam. The motor can be operated to a maximum speed of 3000rpm and DC motor runs with different speed of 400, 500 and 600RPM. speed is controlled with the help of the speed control unit. Power supply to the speed control unit is given by a transformer to reduce the voltage of alternating current. Motor shaft end attached with an aluminum disc of diameter 65mm, thickness of 5mm and the unbalance mass of 35 grams at an eccentricity of 32.5mm. order tracking module is placed in front of the rotating disc to analyze the speed of the rotating DC motor. Tri-Accelerometer is used to measure the vibration characteristics mounted on a beam near the location of DC motor with adhesive has paraffin wax and accelerometer is connected to a data acquisition system for acquiring vibration data. The vibration data acquired for different cases is as shown in the table 3. The signal acquired in the time domain is processed and converted into frequency domain signals using a Fast Fourier Transform(FFT) technique. Displacement of vibration in frequency domine is used for the analysis of the response of the beam.

TABLE 1
Specifications of beam

Material of beam	Stainless steel 304
Dimensions of beam	475*30*5
Mass	0.552 kg
Density	7750 kg /m^3
Youngs modulus	193 GPa

TABLE 2
Specifications of DC motor

Product name	Lunar CR 775
Weight of motor assembly	0.55 kg
Power	240 watts
Length including shaft	95 mm

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A. *Experimental Cases*

The experiment was conducted for transverse vibration of flexible rectangular stainless steel beam for different length of the beam, speed and location of the motor were investigated experimentally by considering 36 cases and each boundary condition of 12 cases as discussed below

TABLE.3
Experimental cases for studying the effect of centrifugal force, length and location of excitation on vibration characteristics

Case no.	End condition	Material	Dimensions of beam (L,W,T)mm	Location of motor	Speed of motor (RPM)
C-1	Cantilever	Stainless steel 304	475,30.5	0.5*length	400
C-2	Cantilever	Stainless steel 304	475,30.5	0.5*length	500
C-3	Cantilever	Stainless steel 304	475,30.5	0.5*length	600
C-4	Cantilever	Stainless steel 304	475,30.5	0.6*length	400
C-5	Cantilever	Stainless steel 304	475,30,5	0.6*length	500
C-6	Cantilever	Stainless steel 304	475,30,5	0.6*length	600
C-7	Cantilever	Stainless steel 304	450,30,5	0.5*length	400
C-8	Cantilever	Stainless steel 304	450,30,5	0.5*length	500
C-9	Cantilever	Stainless steel 304	450,30.5	0.5*length	600
C-10	Cantilever	Stainless steel 304	450,30.5	0.6*length	400
C-11	Cantilever	Stainless steel 304	450,30.5	0.6*length	500
C-12	Cantilever	Stainless steel 304	450,30.5	0.6*length	600
SS-1	Simply supported	Stainless steel 304	475,30.5	0.5*length	400
SS-2	Simply supported	Stainless steel 304	475,30.5	0.5*length	500
SS-3	Simply supported	Stainless steel 304	475,30.5	0.5*length	600
SS-4	Simply supported	Stainless steel 304	475,30.5	0.6*length	400
SS-5	Simply supported	Stainless steel 304	475,30,5	0.6*length	500
SS-6	Simply supported	Stainless steel 304	475,30,5	0.6*length	600
SS-7	Simply supported	Stainless steel 304	450,30,5	0.5*length	400
SS-8	Simply supported	Stainless steel 304	450,30,5	0.5*length	500
SS-9	Simply supported	Stainless steel 304	450,30.5	0.5*length	600
SS-10	Simply supported	Stainless steel 304	450,30.5	0.6*length	400
SS-11	Simply supported	Stainless steel 304	450,30.5	0.6*length	500
SS-12	Simply supported	Stainless steel 304	450,30.5	0.6*length	600
FF-1	Fixed-Fixed	Stainless steel 304	475,30.5	0.5*length	400
FF-2	Fixed-Fixed	Stainless steel 304	475,30.5	0.5*length	500
FF-3	Fixed-Fixed	Stainless steel 304	475,30.5	0.5*length	600
FF-4	Fixed-Fixed	Stainless steel 304	475,30.5	0.6*length	400
FF-5	Fixed-Fixed	Stainless steel 304	475,30,5	0.6*length	500
FF-6	Fixed-Fixed	Stainless steel 304	475,30,5	0.6*length	600
FF-7	Fixed-Fixed	Stainless steel 304	450,30,5	0.5*length	400
FF-8	Fixed-Fixed	Stainless steel 304	450,30,5	0.5*length	500
FF-9	Fixed-Fixed	Stainless steel 304	450,30.5	0.5*length	600
FF-10	Fixed-Fixed	Stainless steel 304	450,30.5	0.6*length	400
FF-11	Fixed-Fixed	Stainless steel 304	450,30.5	0.6*length	500
FF-12	Fixed-Fixed	Stainless steel 304	450,30.5	0.6*length	600

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B. Theoretical Calculation of Natural Frequency

Calculating the natural frequency for different mode, by using Euler's Bernoulli's Beam Theory

$$f_n = \frac{1}{2\pi}(\beta l)^2\sqrt{\frac{EI}{\rho A L^4}} \quad \ldots\ldots (1)$$

Where, I= moment of inertia of beam in m^4, ρ= density in kg/m^3 , L= length in m, E= young's modulus in N/m^3, A= area in m^3, (βl)=constant for end condition.

$$f_n = \frac{1}{2\pi}(\beta l)^2\sqrt{\frac{EI}{mL^3}} \quad \ldots\ldots (2)$$

Where m is the total mass,

From the textbook of engineering vibration Daniel J. Inman 3rd edition, rotating unbalance pg. no. 143. The following equation is taken

$$\text{Total mass, m} = \frac{m_b}{3} + m_m \quad \ldots\ldots (3)$$

Where m_b = mass of beam, m_m = mass of motor

$$m = \frac{0.552}{3} + 0.55$$

$$= 0.73 \text{ kg}$$

$$\text{Moment of inertia of beam, } I = bd^3/12 \quad \ldots\ldots (4)$$

$$= 0.03*0.005^3/12$$

$$= 3.125\times 10^{-10}\ m^4$$

βl = Constant Relative to Vibration Bound Condition, values are shown in table.5

TABLE 5

Values of $(\beta l)^2$ for the various end condition

Beam configuration	$(\beta l)^2$ First mode	$(\beta l)^2$ Second mode	$(\beta l)^2$ Third mode
cantilever	3.52	22.0	61.7
Simply supported	9.87	39.5	88.9
Fixed-fixed	22.4	61.7	121

TABLE 6

Natural frequency obtained for beam dimension of 475*30*5mm

Beam configuration	Natural frequency (Hz)		
	1^{st} Mode	2^{nd} Mode	3^{rd} Mode
Cantilever condition	15.55	97.21	272.65
Simply supported condition	43.61	174.54	392.84
Fixed-fixed	98.98	272.65	534.69

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III. RESULTS AND DISCUSSIONS

A. Effect Of Boundary Condition On Beam

Experimental results for different cases are compared with end conditions and plotted as shown in the fig.2. Figure shows that variation amplitude subjected to different end conditions for the beam is same as the excitation force. Therefore maximum amplitude of vibration occurs in cantilever condition.

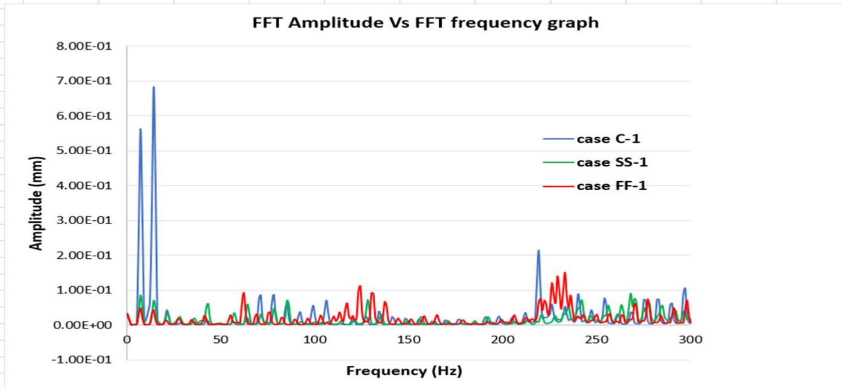

Fig.2: FFT Amplitude Vs FFT frequency graph for case C-1, SS-1 and FF-1

B. Effect of Varying Length of the Beam

The stainless steel beams with the same cross-section but the different length of 475mm and 450mm. we're an exciting same centrifugal force and results are shown in fig.3 and fig.4.

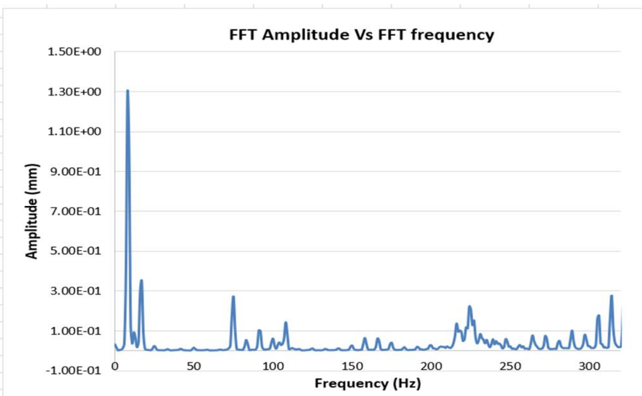

Fig.3: beam length of 475mm for case C-2

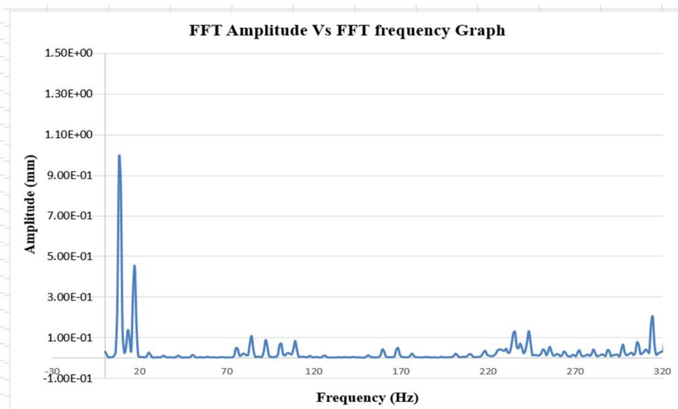

Fig.4: beam length of 450mm for case C-8

From the above figure length increases with amplitude is also increases, similarly for different cases are shown in fig.5 and fig.6

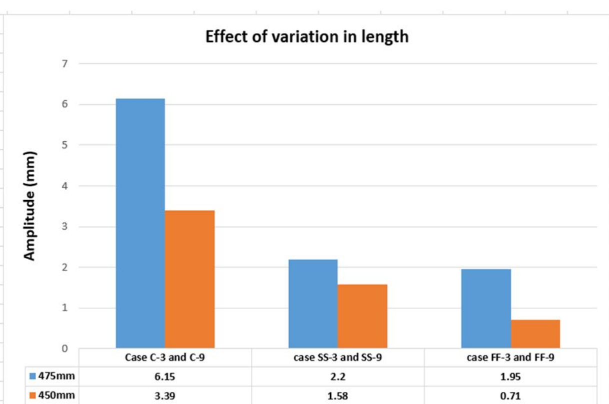

Fig.5: Effect of variation in length Fc acting at midpoint

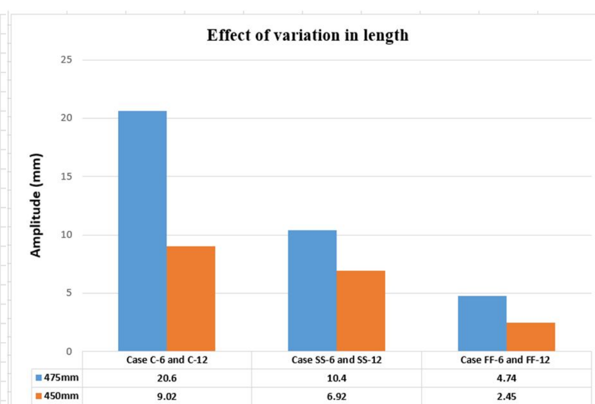

Fig.6: Effect of variation in length Fc acting at 0.6*length

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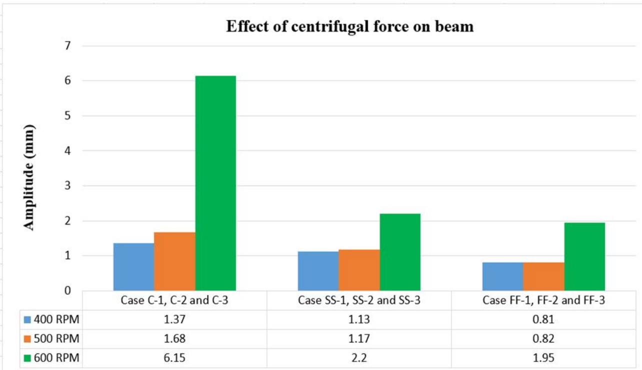
C. Effect Of Centrifugal Force Acting On The Beam

The highest amplitude of vibration occurs in 600RPM speed when compared to 500 and 400RPM speed, where least Amplitude of vibration occurs in 400RPM. Therefore amplitude of vibration increases with the increase of speed, as shown in fig.7 and fig.8.



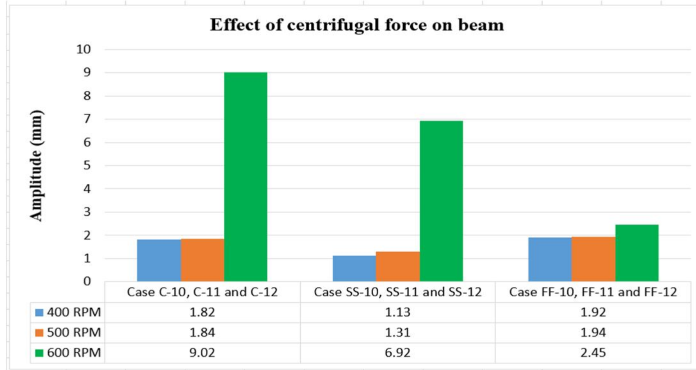


Fig.7: Effect of centrifugal force acting at midpoint

Fig.8: Effect of centrifugal force acting at 0.6*length of beam

D. Effect of location of excitation on beam

Location of centrifugal force is changed by changing the location of DC motor along the length of beam from the midpoint (0.5*length) to 0.6*length of the beam. Fig.9 shows that, even a little change in location of DC motor as a considerable change in amplitude of vibration.

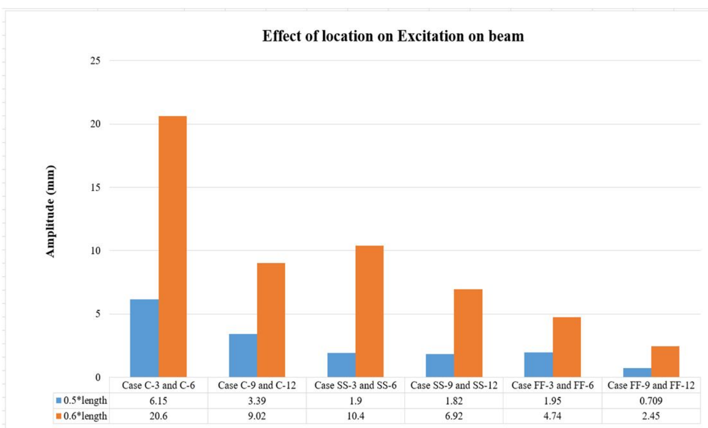
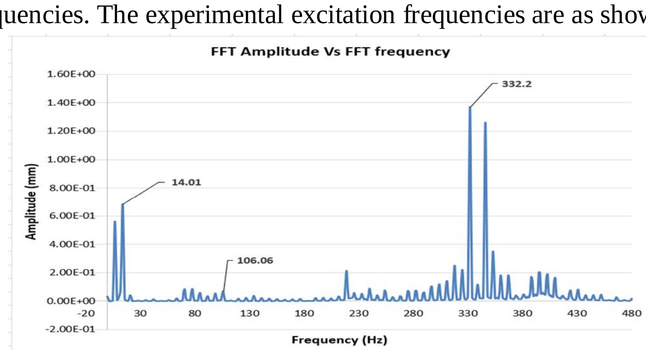


Fig.9: Effect of location of excitation for different cases

E. Experimental Natural Frequency

The results obtained for different cases as for the table. In FFT, amplitude Vs frequency graphs are plotted to obtained the peak line, this peak line is considered as experimental excitation frequency. These frequencies are compared with Theoretical natural frequencies. The experimental excitation frequencies are as shown in the fig.10 and fig.11



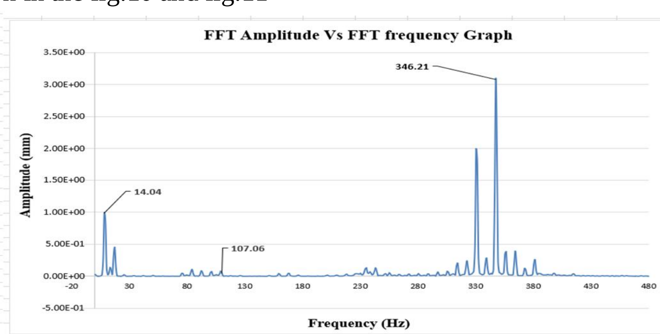


Fig.10: FFT graph to obtain peak line for case C-1

Fig.11: FFT graph to obtain peak line for C-7

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TABLE 12
The experimental frequency for Cantilever condition

Modes	Theoretical frequency (Hz)	Experimental frequency (Hz)					
		Case C-1	Case C-2	Case C-3	Case C-4	Case C-5	Case C-6
1st mode	15.55	14.01	17.01	14.04	14	17.01	14.04
2nd mode	97.21	106.06	108.06	113.07	106.06	105.06	106.26
3rd mode	272.65	332.2	331.2	329.2	318.19	339.21	331.2

TABLE 13
Experimental frequency for Simply supported condition

Modes	Theoretical frequency (Hz)	Experimental frequency (Hz)					
		Case SS-1	Case SS-2	Case SS-3	Case SS-4	Case SS-5	Case SS-6
1st mode	43.61	50.03	51.03	52.23	50.24	51.03	53.04
2nd mode	174.54	221.13	218.13	206.12	200.02	204.12	203.12
3rd mode	392.84	410.25	411.25	411.25	411.25	458.28	430.26

TABLE 15
Experimental frequency for Fixed-fixed condition

Modes	Theoretical frequency (Hz)	Experimental frequency (Hz)					
		Case FF-1	Case FF-2	Case FF-3	Case FF-4	Case FF-5	Case FF-6
1st mode	98.89	123.07	120.07	112.07	119.07	121.07	124.07
2nd mode	272.65	304.19	314.19	339.21	332.2	330.20	329.2
3rd mode	534.69	609.38	601.37	606.37	615.38	609.38	606.37

F. Percentage of Error

In the current study the theoretical work was carried by using Euler's Bernoulli's beam theory. It was seen that theoretical values are in agree with the experimental values and percentage of error are calculated by using the following formula

$$\text{Percentage of Error}(\%) = \left|\frac{\text{Theoretical Frequency} - \text{Experimental Frequency}}{\text{Theoretical Frequency}}\right| \times 100$$

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TABLE 15
The percentage of error for Cantilever condition

Modes	Percentage of Error (%)					
	Case C-1	Case C-2	Case C-3	Case C-4	Case C-5	Case C-6
Mode1	9.9	9.38	9.71	9.96	9.38	9.71
Mode2	9.1	11.16	16.31	9.1	8.07	9.3
Mode3	21.84	21.47	20.74	16.7	24.41	21.47

TABLE 16
The percentage of error for Simply supported condition

Modes	Percentage of Error (%)					
	Case SS-1	Case SS-2	Case SS-3	Case SS-4	Case SS-5	Case SS-6
Mode1	14.72	17.01	19.76	15.20	17.01	21.62
Mode2	26.69	24.97	18.09	14.59	16.94	16.37
Mode3	4.43	4.68	4.68	4.68	16.56	9.52

TABLE 17
The percentage of error for Fixed-fixed condition

Modes	Percentage of Error (%)					
	Case FF-1	Case FF-2	Case FF-3	Case FF-4	Case FF-5	Case FF-6
Mode1	24.45	21.41	13.32	20.40	22.42	25.46
Mode2	11.56	15.23	24.41	21.84	21.10	20.74
Mode3	13.96	12.47	13.4	15.09	13.96	13.4

IV. CONCLUSION

Developed test fixture with the capability to perform transverse vibration experiment for unbalanced rotating mass at different frequencies. A stainless steel beam was considered for different boundary conditions such as cantilever, simply supported and fixed-fixed condition by varying the length of the beam . Theoretical natural frequency were compared with the frequency obtained from the experiment. There is an error between theoretical and experimental value because of neglecting the damping force and loss of energy due to friction. Also studied the effect of various characteristics on the response of the beam, are listed below,

A. Among the boundary condition, cantilever condition gives the high value of the amplitude of vibration when compared to simply supported and fixed-fixed condition.
B. Due to the effect of centrifugal force , amplitude of vibration increases with increase in the speed of motor.
C. From the location of dc motor, it is seen that when the motor is placed at 0.6*length of the beam gives a high value of the amplitude of vibration occurs when compared to the midpoint of the beam.

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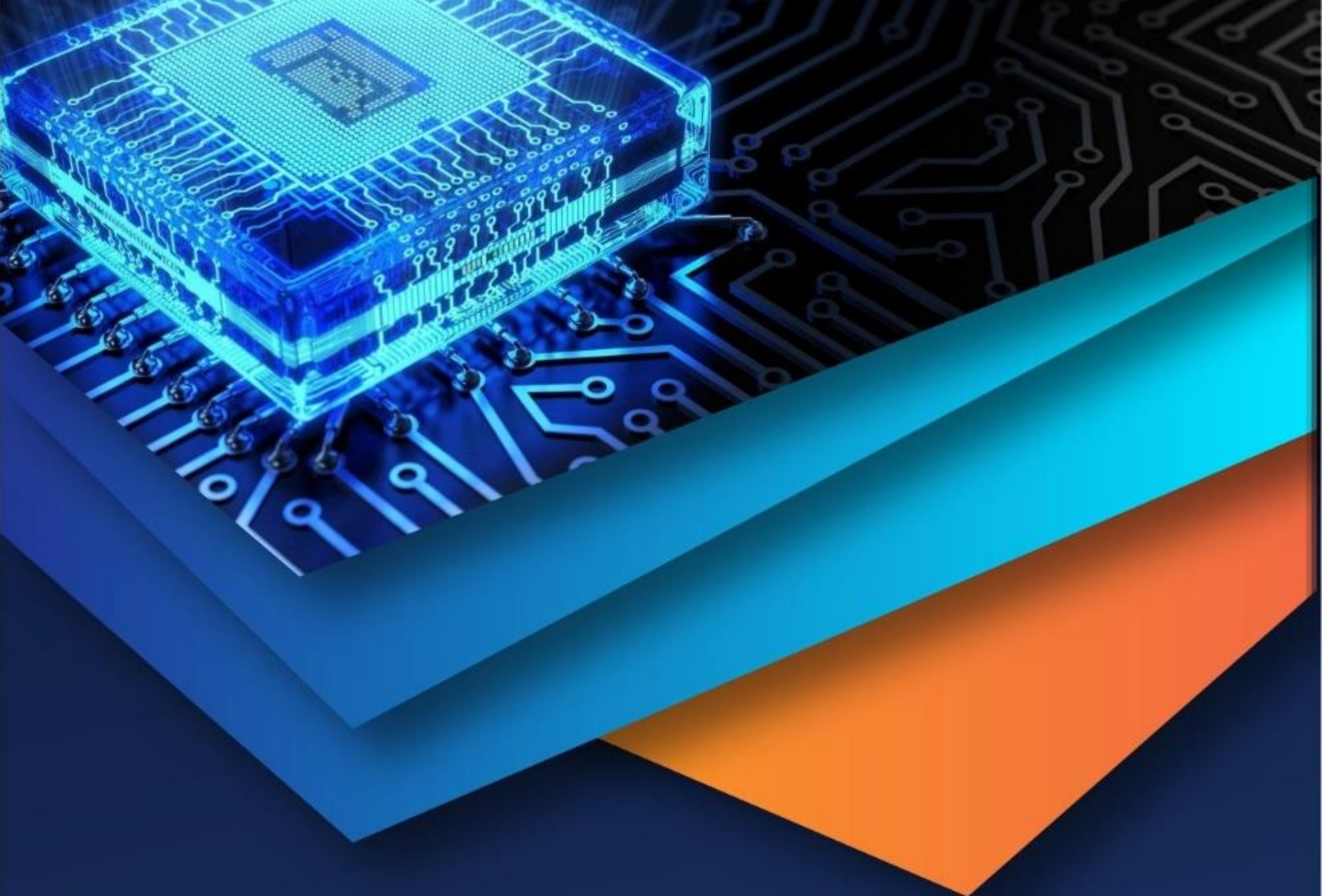

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