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Design & Testing Of Semi-Automatic Vibration Absorber

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Abstract: When machines are operating they create vibrations. In some cases these vibrations have a negative influence on the performance of the machine. For instance, in a milling machine the whole construction vibrates when the tool rotates and material is cut. This can have a negative impact on the precision of the manufactured products, the lifetime of the machine and the environment. A solution to this problem can be found in the application of dynamic vibration absorber (DVA). A DVA is a construction that can be mounted on the vibrating machine and which is connected to a controller. Whenever the vibration characteristics of the machine change, the controller adjusts the DVA so that the vibrations get damped as good as possible. The aim is to design active DVA

1. MODAL ANALYSIS OF BEAM.

In order to predict frequencies and mode shapes of beam modal analysis is carried out using ansys software. The beam is modeled using SOLID45 element.

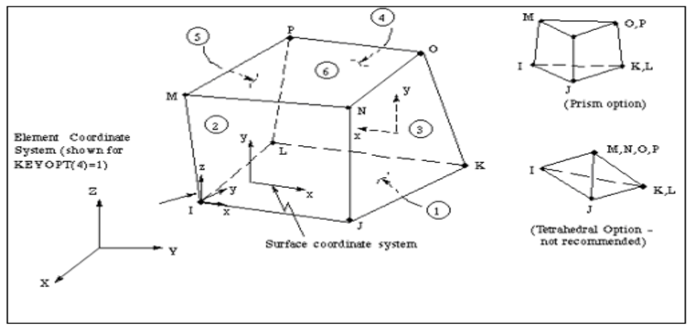


Fig 1.1: solid45 –structural solid

SOLID45 is used for three dimensional modeling of solid structure the element is defined by eight nodes: translations in the node x, y and z direction. The nodes at the one end of beam are fixed i.e. all degree of freedom of the nodes are zero.

Dimension and properties of the beam

Length of beam = 1000mm,

Thickness of beam = 5mm,

Width of the beam = 50mm,

Density of material = 7800 kg/m^3,

Modulus of elasticity = 200Gpa.

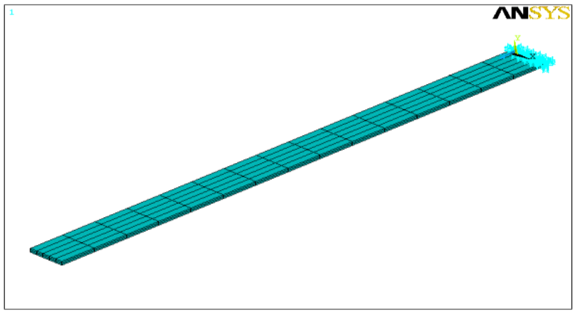


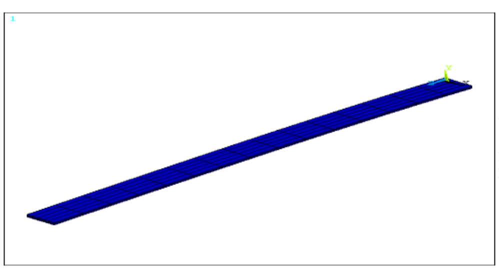
Fig 1.2: Beam meshed with Solid 45 element

1.1 Result and Discussion of Beam by Using Ansys Software

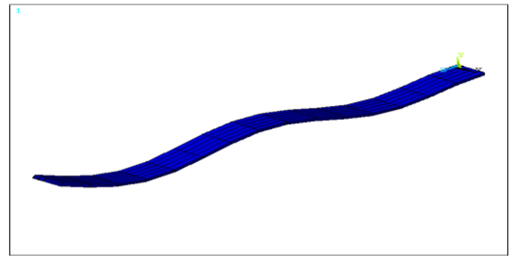
The first four frequencies of beam are listed in Table 2; the respective mode shapes are as shown in Fig 1.1.1

Table2: First five frequencies of beam

Serial no.	Frequencies (Hz)
1	8.28
2	52.42
3	150.01
4	304



a) First mode shape at 8.28 Hz



b) Fourth mode shape at 304 Hz

Fig 1.1.1: Mode shapes of beam for first four natural frequencies.

2. MANUFACTURING OF ABSORBER

The absorber consists of following different parts manufactured from mild steel material.

1) Square Housing

2) End masses

3) Sliding rod

4) Threaded Rod End Mass

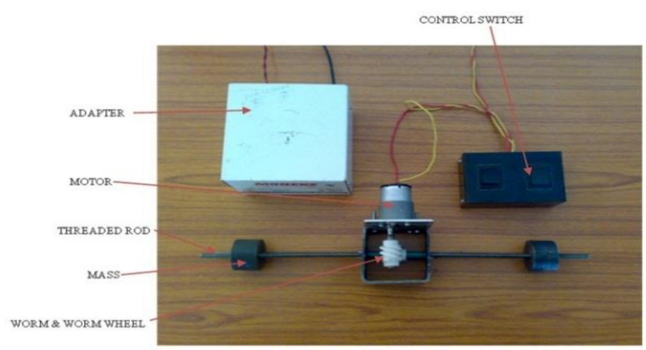


Fig 2.1 Dual Mass Vibration Absorber

The detailed dimensions of the parts of the absorber are as follows

1) SQUARE HOUSING

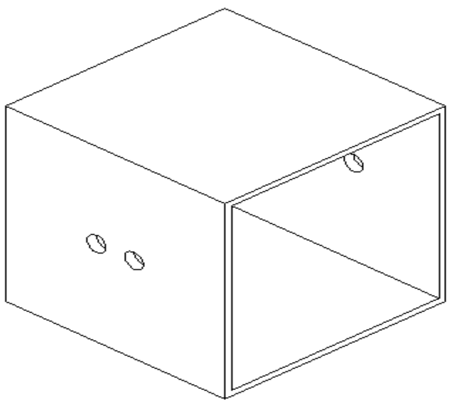


Fig 2.2 Square housing

square housing	
Length	70mm
Width	42mm
Height	70mm
Diameter of hole	6mm
Center distance between two holes	12mm

2) END MASSES

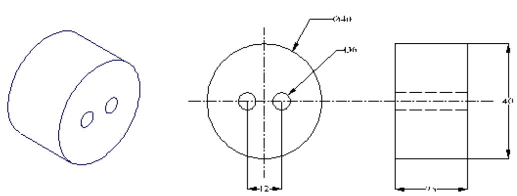


Fig 2.3 End Masses

Diameter of mass	40 mm
Hole diameter	6 mm
Length of mass	25 mm
Center distance between two holes	12 mm

3) SLIDING ROD

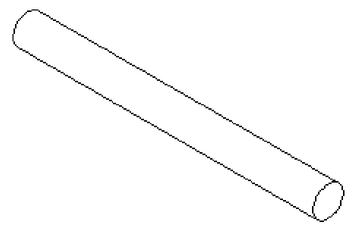


Fig 2.4 Sliding Rod

1. Diameter of rod = 6 mm
2. Length of rod = 160 mm

4) THREADED ROD

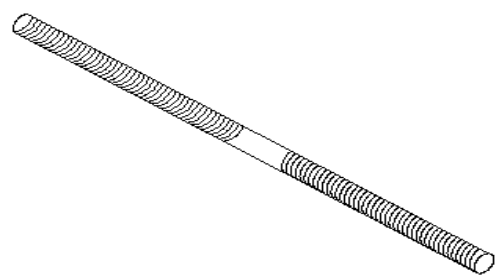


Fig 2.5 Threaded rod

1. Diameter of rod = 6 mm
2. Length of rod = 345 mm
3. One end of rod is left hand threaded & other end of rod is right hand threaded

The dimensions of beam used for testing are.

1. Length of Beam 1000mm
2. with of beam 65mm
3. Thickness of beam 10mm

Fig 2.6 Beam used for testing

2.1 Torque required for driving threaded rod

Known,

Pitch (p) = 5 mm,

Mean diameter (dm)= 14mm,

Number of teeth on worm (zw) = 4

Pressure angle (Φ)

$\tan \Phi = \mu = 0.3$

$\boxed{\Phi = 16.7^{\circ},}$

Lead angle (α)

$\tan \alpha = L / (\pi \times dm)$

$L = p \times zw$

$\tan \alpha = (p \times zw) / (\pi \times dm)$

$= (5 \times 4) / (\pi \times 14)$

$= 0.4547$

$\boxed{\alpha = 24.44^{\circ}}$

Tangential force required (F)

$F = w \times \tan(\alpha + \Phi)$

Where,

$w = 2 \times m \times g$

$= 2 \times 0.235 \times 9.81$

$\boxed{w = 4.6107 \text{ N}}$

$F = w \times \tan(\alpha + \Phi)$

$F = 4.6107 \times \tan(16.7^{\circ} + 24.44^{\circ})$

$\boxed{F = 4.03 \text{ N}}$

Torque required

$T = F \times (dm/2)$

$= 4.03 \times (14/2)$

$\boxed{T = 28.204 \text{ N-mm}}$

2.2 Specifications of motor

Type of motor - Johnson's d.c motor

Fig 2.2.1 D.C. motor high torque mini 12v D.C gear motor 200rpm

3. EXPERIMENTAL SETUP

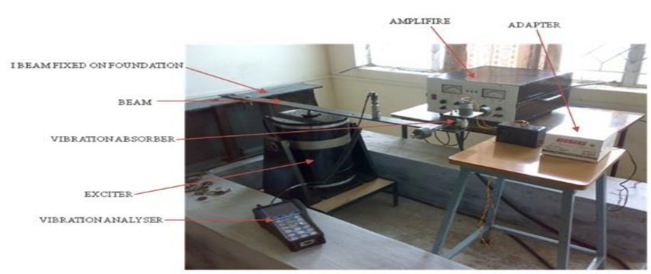


Fig 3.1 Experimental setup

The testing of absorber is carried on cantilever beam. The cantilever beam requires one end of beam fixed to the rid support and other end free. To create the fixed support condition for beam I -section is used. This I-section is fixed to the foundation with the help of nut bolts. Initially the I section is found to be less stiff to support the beam for testing. In order to increase the stiffness of I section the stiffeners are welded to the section as shown in Fig 4.2. To check the strength and stiffness of the support the beam required to use for testing is fixed to the I section with the help of nut bolt as shown in Fig 4.2. Then the beam is given harmonic excitation from 1

Hz to 60 Hz by vibration exciter. The vibration of the I section are observed at different locations, which are found to negligible. Thus the I section acts as a rigid support to the beam.

Fig 3.2 Semi-automatic absorber attached to the beam

3.1 Specification of the instruments

The beam is given harmonic excitations by vibration exciter. The excitation frequency and amplitude can be varied by power amplifier. The displacement of beam at various positions along the length of the beam can be measured with the help of accelerometer and vibration meter. The specifications of various instruments used for testing are as follows

1) VIBRATION EXCITER

Vibration Exciter	SI - 230
Pack Sine Force	200 Newton
Maximum Displacement	12mm (Peak to Peak)
Maximum Allowable Pay Load	3 Kg. Nominal
Power Requirement (Available from Power Oscillator)	250 VA Nominal
Weight	75 Kg. Nominal
Maximum power output	250 VA
Frequency response	1Hz. to 10kHz Within ± 1dB
Harmonic distortion	Less than 1%
Input impedance	10k ohms nominal
Gain at 1 kHz.	20 dB ± 1dB
Current indication	Load current is indicated on a panel meter

2) VIBRATION ANALYSER

- Dwell channel fast Fourier transform (F.F.T) spectrum analyser
- Model – Adash 4300 – VA3

3) ACCELEROMETER

- Accelerometer sensor with magnetic base 100 mv/gm
- AC 102

3.2 EXPERIMENTAL PROCEDURE

1. The beam is fixed with the help of nut –bolts to I beam support mounted on foundation, which acts as fixed support.
2. To find natural frequencies of beam experimentally, excitation is given to beam at

suitable location i.e. at 0.40 L and the frequency of excitation is varied from 1Hz to 60 Hz. As per theory of vibration of cantilever beam; displacement of beam is measured at various positions as follows for different frequencies.

3. As per theory when the beam is vibrating at first natural frequency the displacement at the end of beam is maximum. For second natural frequency as the node is at 0.86 L, so the frequency at which vibration of beam are minimum at 0.86 L is the second natural frequency of beam.
4. The beam is exited at 37 Hz at 0.4 L, Which is in between second and third natural frequency of beam.
5. The displacement of beam is recorded at various points along the beam.
6. The absorber is attached at the end of beam i.e. at 1 L and again excitation of 37 Hz is given to beam
7. For tuning the absorber to the excitation frequency i.e. 37 Hz the masses are move in and out as shown in Fig 4.4 and displacement of the beam at the absorber attachment location is recorded continuously. The mass position at which the vibration of beam is found to be minimum is recorded. Now for this mass position the frequency of absorber is equal to excitation frequency i.e. 37 Hz.
8. The displacement of beam with absorber is recorded at different positions along the length of beam.
9. Now the excitation frequency is given 42 Hz to beam
10. For tuning the absorber to the excitation frequency i.e. 42 Hz the masses are move in and out as shown in Fig 4.4 and displacement of the beam at the absorber attachment location is recorded continuously. The mass position at which the vibration of beam is found to be minimum is recorded. Now for this mass position the frequency of absorber is equal to excitation frequency i.e. 42 Hz.

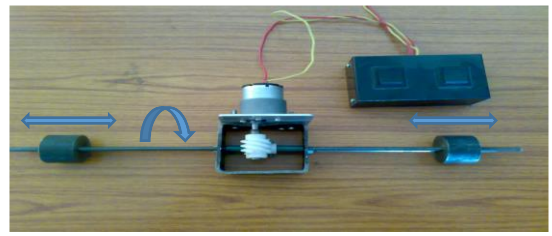


Fig3.2.1 Tuning of Absorber

3.3 RESULTS & DISCUSSION

Table No. 4 Displacement of beam with & without absorber attached at different positions along the beam at excitation frequency = 37 Hz & Excitation at 40 mm

Beam Length	Displacement of Beam (Micron)	
	Without Absorber	With Absorber attached at End
0	1	0
100	70	10
200	148	30
300	219	32
400	267	35
500	296	102
600	312	175
700	194	230
800	73	240
900	157	154
1000	264	15

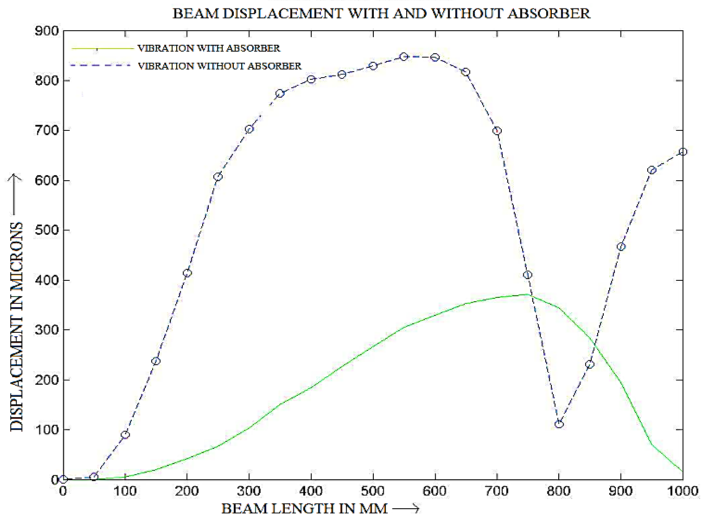


Fig. 3.3.1 Graph of displacement of beam with &without absorber at 37 Hz

It is observed from table that the when beam is excited at 37 Hz the beam vibrates with high amplitude without absorber. The vibrations of beam are less near support only. When the absorber is attached at the end of beam i.e.1L, the vibration near the attachment point becomes negligible from 657 micron to 15 micron.

3.4 COMPARISION OF RESULTS

1) Natural frequency of beam Hz

	Analytical	Ansys	Exprimental
First		8.28	8
Second		52.42	48
Third		150.01	--
Fourth		304	--

2) Natural frequency of Absorber Hz

	Analytical	Ansys
First	34.17	40.88

CONCLUSIONS

The dual mass vibration absorber can be effectively used to reduce the vibration of the beam. This absorber is tuned to excitation frequency by moving the dual masses in or out easily. The experimental result shows that around the absorber attachment location the vibration of beam become negligible. This absorber also reduces the vibration of beam other than attachment point.

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