DESIGN AND DEVELOPMENT OF LABORATORY MODELS OF DYNAMIC VIBRATION ABSORBER

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ABSTRACT:

Vibration study has become an integral part of system analysis these days. Most of the system failures occur due to excessive, unwanted vibrations. A Dynamic Vibration Absorber (DVA) is designed fabricated and tested for primary systems having single degree of freedom. The aim of the present study is to design DVA for two primary systems (laboratory models) i.e. Cantilever Beam and Vertical motion model (spring mass system) subjected to forced vibrations. Numerous papers published in international journals were studied and it was found that the research over DVA is not new and in fact there are a variety of practical situations in which DVAs are used. Innovative designs of DVAs which people have come up with were studied from the literature. While trying out various excitation mechanisms it was found that using cam was the best option. With the help of this project the students of our college can understand the concept of which DVAs work in practical situations and which may even inspire them to take up vibration control as their field of study.

INTRODUCTION:

The main reason of system failure due to vibration is Resonance. Resonance is the tendency of a mechanical system to respond at greater amplitude when the frequency of its oscillations matches the systems natural frequency of vibration (its resonance frequency) than it does at other frequencies. [8]It may cause violent swaying motions and even catastrophic failure in improperly constructed structures including bridges, buildings and airplanes - a phenomenon known as resonance disaster

In vibration analysis, a Dynamic Vibration Absorber or vibration neutralizer is a tuned spring-mass system which reduces or eliminates the vibration of the harmonically excited system. [8] A dynamic absorber can be affixed to rotating machine and tuned to oscillate in such a way that exactly counteracts the force from the rotating imbalance. This reduces the possibility that a resonance condition will occur, which can cause rapid catastrophic failure. Properly implemented, a DVA will neutralize the undesirable vibration, which would otherwise reduce service life or cause mechanical damage.

An absorber consisting of a spring mass system is shown in figure-1[5]

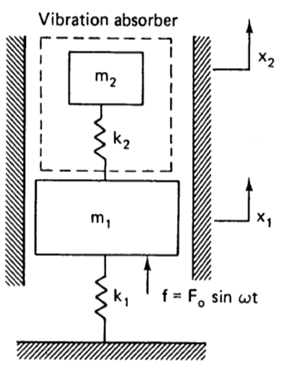

Figure 1- Dynamic Vibration Absorber

m_1, k_1 - Mass and spring constituting the primary system
m_2, k_2 - Mass and spring constituting the absorber
f - Force applied on the primary system with amplitude F_0 and frequency ω
x_1 - Displacement amplitude of primary system
x_2 - Displacement amplitude of secondary system

The above figure consists of a primary system whose mathematical model is equivalent to a mass and spring system, on which an external periodic force is applied.[5] In order to reduce the vibrations in the system due to this force, a DVA consisting of a spring and a mass is incorporated in the primary system at the top. The DVA will absorb the vibrations caused by the force to a large extent. As a result of this, the amplitude of vibration of the primary system will be reduced and the absorber system will vibrate with greater amplitude.

It was seen from the literature survey that there are many applications in which a dynamic vibration absorber can be used to suppress the vibration of a system and hence to increase its efficiency. Also to make the students understand properly about the working principle of dynamic vibration absorber a model of the same can be developed. Thus the problem can be defined as 'Design and Development of laboratory models of a Dynamic Vibration Absorber for a single degree of freedom system'.

The considerations and limitations which were proposed during the analysis are: the primary systems to be experimented on are a Cantilever Beam and a spring-mass system both having vertical motion, the primary system is a single degree of freedom system and the absorber to be designed is without damper.

EXPERIMENTAL SETUP OF A CANTILEVER BEAM:

A prototype model of a cantilever beam is designed, fabricated and tested.[1]

Specifications for the cantilever beam:

- Material : Aluminium
- Modulus of elasticity (E): 0.69 × 105 N/mm2
- Density (ρ) : 2700 × 10-9 kg/mm3
- Length (l) : 540 mm
- Width (b) : 38 mm
- Depth (d) : 4 mm

Natural frequency of cantilever beam with end mass: Resonance will occur when the frequency of the external applied force matches the natural frequency of the cantilever beam. The frequency of the force will depend on the speed of the motor. Hence, the specification of the motor was found out by calculating the natural frequency of beam.

Equivalent mass for cantilever beam (m_{eq}) = M+ (0.23×m)

Where,

M = Mass of end mass

m = mass of cantilever beam

Calculation of m_{eq}:

Mass of strip + nuts and bolts = 0.0465 kg

Mass of dead mass of the absorber beam = 0.0025 kg

Therefore,

M = 0.049 kg

Mass of 540 mm of beam is 0.2138kg

Therefore, m_{eq} = 0.09817 kg

Calculation of Moment of Inertia:[7]

$I = \frac{b\times d^3}{12} = 202.67\ mm^4$

Calculation of stiffness of the beam:[8]

A cantilever beam can be considered as a spring mass system with mass as the equivalent mass and stiffness of the spring as

$k_1 = \frac{3\times E\times I}{L^3} = 266.092$ N/m

Hence, the natural frequency would be

$\omega_1 = \sqrt{\frac{k_1}{m_{eq}}} = 52.062$ rad/s

For resonance,[7]

$\omega_1 = \frac{2\times\pi\times N}{60}$

$N = \frac{\omega_1\times 60}{2\times\pi} = 497.414$ rpm

- Fabrication and mounting of cantilever beam:
- A cantilever beam is bought of the specifications given in the above section. A rigid foundation in the vibration laboratory is chosen for this prototype. It is mounted on the metal structure with the help of clamps. The beam is placed between the two wooden plates on the metal frame.

SELECTION OF AN EXCITER FOR CANTILEVER BEAM:

An exciter system is designed, fabricated and mounted to give excitation to the cantilever beam in such a way that the natural frequency of vibration of a cantilever beam is equal to the excitation frequency. Different types of exciters are selected and analysed. The various exciters tested are as follows:

1. Scotch Yoke Mechanism
2. A Rotary Striker
3. Cam

Design, Fabrication And Mounting Of Exciter: A radial cam which would give simple harmonic motion to the beam is designed for giving excitation. [4]The cam is mounted on the motor using a coupler. The beam is considered as a flat faced follower.

Following are the terms related to radial cams which are used in our design-[9]

1. Base Circle: It is the smallest circle that can be drawn to the cam profile.

2. Lift or Stroke: It is the maximum travel of the follower from its lowest position to topmost position.

The values selected for these two are:

Base Circle Radius = 25mm

Lift = 10mm

The following displacement curve of displacement of the cam v/s angle through which the cam rotates is drawn.

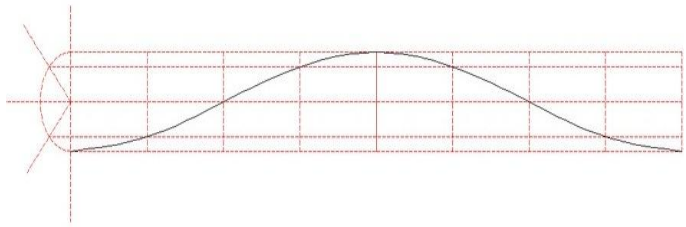

Figure 5- Displacement Diagram

Shown below is the required cam profile. The fabrication of the radial cam is done.

Figure 6- Radial Cam Profile

Mounting of motor: In order to mount the motor a wooden structure is made with holes drilled at the bottom to fix it to the end of the horizontal part of the channel using nuts and bolts. The top surface of the box is made plain on which the motor is mounted

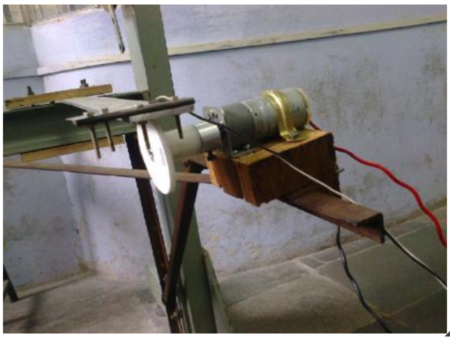

Figure 7- Wooden mounting for the motor

SELECTIONOF A DYNAMIC VIBRATION ABSORBER FOR THE CANTILEVER BEAM:

A DVA is designed and fabricated which can then be fixed on the primary system (cantilever beam) to neutralize or absorb the vibrations of the primary system.[6] The DVA to be designed is also a cantilever beam which will vibrate in the same plane as that of the primary system.

- The details of Dynamic Vibration Absorber is as discussed in the following sections.
- Specifications for the absorber system:
- Material: Stainless Steel (HS 304)
- Modulus of elasticity (E): 180 × 103 N/mm2
- Density (ρ): 8.03 × 10-6 kg/mm3
- Width (b): 22 mm
- Depth (d): 0.5 mm

Length of the absorber beam: In order to make the amplitude of primary system equal to zero, the natural frequency of the absorber beam (ω_2) should be equal to the frequency of the applied force (ω)

As we take the resonance condition,

$\omega = \omega_1 = \omega_2$

As the absorber beam is also a cantilever beam made of SS

$\omega_2 = \sqrt{\frac{k_2}{m_2}}$

Let L_2 be the length of the absorber beam.

$m_2 = \rho \times A \times L_2 = 0.088 \times L_2$

Calculation of Moment of Inertia:

$I_2 = \frac{b \times d^3}{12} = 2.29 \times 10^{-13}\ m^4$

$k_2 = \frac{0.1236}{L_2{}^3}$

Equating ω_1 and ω_2, we get

$L_2 = 0.2178$ m

Fabrication and mounting of Dynamic Vibration Absorber:

A stainless steel strip of above calculated length is cut by using abrasive cutter. It is bolted at the free end of the cantilever beam. The plane of vibration of both the beams is same.

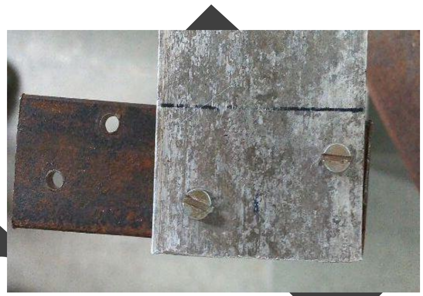

Figure 8 – Provision for attaching the absorber beam

Figure 9 – Primary system with the absorber beam

Theoretical analysis of the first setup

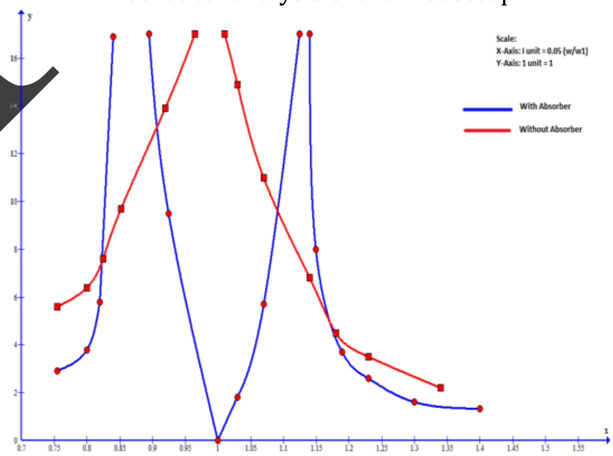

Figure 10- Response of the First Setup

According above plot, at resonance condition ($\omega = \omega_1 = \omega_2 = 52.062$ rad/s) the amplitude of vibration of the cantilever beam is most without absorber but is equal to zero with the absorber

EXPERIMENTAL SETUP OF THE VERTICAL MOTION MODEL:

This prototype consists of a foundation made of MS on which a primary mass is mounted using four springs. It consists of the following components:

1. Foundation
2. Rubber pads
3. Primary mass

4. Four springs for the primary system
5. Secondary mass
6. Spring for the absorber
7. Ten adapters
8. Motor for giving excitation

Design of foundation: A foundation, which is used as the base for this setup is designed such that it would act as a rigid support. Also, rubber pads are bolted at the bottom of the foundation which would act as isolators to absorb any vibrations transferred from the foundation to the table.

Specifications for the foundation:

- Material: Mild Steel(MS)
- Length : 200 mm
- Width : 200 mm
- Height : 10 mm

Specifications for the primary mass:

- Material : Mild Steel(MS)
- Length : 120 mm
- Width : 120 mm
- Thickness : 5 mm

Specifications of Spring for primary system

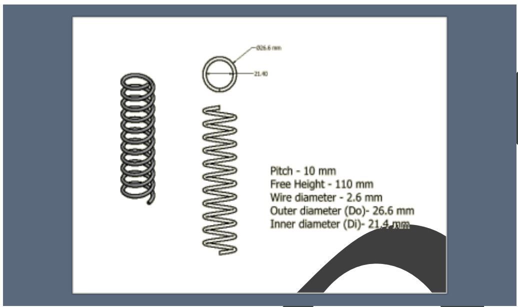

Figure 11 – Spring for primary system

Calculation of stiffness (theoretical):

We know that the stiffness of the spring is given by:[8]

$$k = \frac{G \times d^4}{8 \times D^3 \times N}$$

$= 4.80325$ N/mm

$= 4803.25$ N/m

Design of Adapters for primary system: Adapters are cylindrical metal pieces with grooves similar to the coils of the spring on them. The grooves are made such that only two coils of the spring will be inserted into the grooves. Hence the effective number of coils will be 11-4 = 7

The diameter of the adapter cylinder is taken as 26.6 mm

Calculation of natural frequency of the primary mass:

Mass (M_1) = Main mass + Mass of the five adapters + Mass of the five nuts + Mass of 8 coils of the spring

Main mass = 0.728 kg

Mass of one adapter = 0.094 kg

Mass of one nut = 0.004 kg

Mass of spring = 0.032

$M_1 = 0.728 + (5\times0.094) + (5\times0.004) + 0.029 + 0.027$

$= 1.274$

As the springs are in parallel, the total stiffness of the primary system will be

$k_1 = 4\times k$

$= 19213$ N/m

ω_1 = Natural Frequency of the primary mass

$$\omega_1 = \sqrt{\frac{k_1}{M_1}}$$

$$= \sqrt{\frac{19213}{1.274}}$$

$= 122.8$ rad/s

SELECTION OF A DYNAMIC VIBRATION ABSORBER FOR THE PRIMARY SYSTEM:

In order to reduce the amplitude of forced vibrations, a dynamic vibration absorber a spring mass system is designed and fabricated.

The value of the absorber mass is taken as 0.5 kg.

For reduction of amplitude of primary mass to zero (at the condition of resonance), the following condition should be satisfied:

$\omega = \omega_1 = \omega_2$

Where,

ω_2= Natural Frequency of the absorber system

Equating ω_1 and ω_2 we get the stiffness of the spring for absorber system as 7886.42 N/m

Specification of spring for absorber system

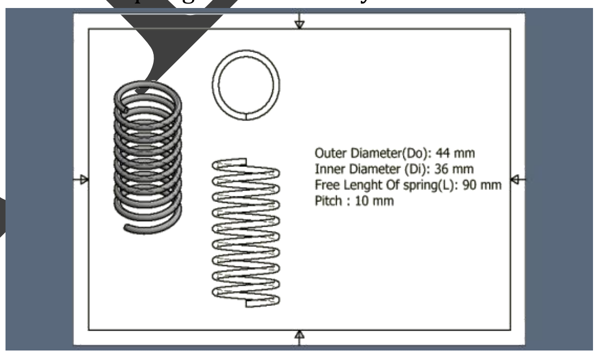

Figure 12 – Spring for absorber system

Figure 13- Second Model Theoretical analysis of the Second setup

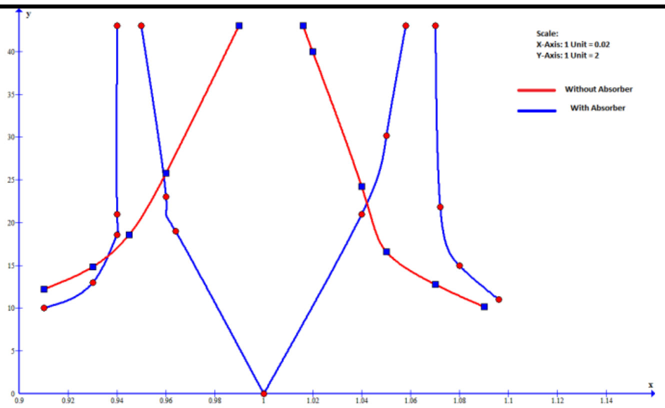

Figure 13- Response of the Second System

According to the above plot the amplitude of vibration of the vertical motion model is maximum without the absorber while it's zero with the absorber at resonance condition. ($\omega = \omega_1 = \omega_2 = 122.8$ rad/s).

CONCLUSION:

In this study, we accomplished the fabrication of the two laboratory models. Theoretical analysis of both the models has been completed. Based on it, graph of response of the two models has been plotted. It is seen that without the absorber, there is only one frequency at which we get resonance. But with the absorber, at the same resonant frequency, amplitude of the primary system is zero and there are two peaks.

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