Design and Development of Active Dynamic Vibration Absorber

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One of the important methods by which vibrations of a machine are reduced is by the use of a vibration absorber. This paper presents the design and development of an semi-active dynamic vibration absorber. The uniqueness of the design lies in coupling of tuned resonant device and actuating mechanism. This paper discusses the concept, design and experimental results of a vibration absorber mechanism. The solenoid valves with plunger and spring arrangements have been used to keep absorber mass fixed with the beam. The sequential control, ie, the energisation/de-energisation of solenoids coupled with the drive steps for the stepper motor (both clockwise and anti-clock directions) has been executed using a PC based software. The necessary drive circuits for stepper motor and solenoid valves have also been developed. The absorber is best for those machines, which run at variable speeds, but remain at a particular speed for a considerable amount of time. Interestingly this arrangement converts the system into a system with three- degree-of freedom. From the analysis of the system, it is noted that one of the natural frequencies of this mechanism is that of the absorber but no resonant conditions are created at this frequency.

Keywords : Vibration absorber; Active spring; Variable stiffness

INTRODUCTION

The control techniques used to control noise and vibrations can broadly be classified into active, passive or semi-active and hybrid control. It is known that there are fundamental limitations in reduction of the vibrations that can be achieved with passive control measures. Hence, a suitable active control system is required to achieve the greater reductions.

An active control system requires an actuator that can deliver a dynamic force. A semi-active control system requires a passive device where the passive properties are changed, so that, the device delivers a dynamic force with appropriate magnitude and phase. Important examples of the applications of dynamic vibration absorber have been cited earlier[1]. Brennen[2] described examples of variable tuned dynamic absorber but here, the method for varying the stiffness has not been indicated. Nolgac and Howsen[3] described a delayed resonator which is a nonlinear device. Cross[4] discussed the recent developments in the use of smart materials for smart structures. In this paper, the concept of tuning the absorber by changing the stiffness of the beam by varying its moment of inertia has been presented.The paper describes the design and development of a resonant device to generate the required secondary force. The reason for using such a device is that the construction and control of this type of device is much easier and can be controlled by using a PC.

One of the important methods by which the vibrations can be reduced is vibration absorption, which is also called 'energy sink' technique. In this technique, the energy absorbing mechanism which is a spring mass system is attached to the structure. For fixed excitation frequency, the absorber is tuned with the machine excitation frequency for the maximum attenuation. However, for changing excitation frequencies, self-tuning vibration absorbers are necessary. A dynamic vibration absorber[5] is an auxiliary mass-spring system that tends to neutralize the excitation of a structure to which it is attached. The basic principle of operation is that the vibration of the auxiliary system is out of phase to that of main system excitation thereby applying a counteracting force. This is called tuned absorber since it is tuned for an excitation of particular frequency.

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So far these systems are primarily used for those systems which are excited by a force that is a constant multiple of the rotational speed. Here it is possible to modify the vibration pattern and reduce its amplitude significantly by use of auxiliary mass tuned to the excitation frequency.

The main disadvantage of the undamped dynamic vibration absorber is that it adds another natural frequency to the system and is effective only at one frequency to which the auxiliary system is tuned. Hence, this type of absorber is suitable for constant operating frequency and where variation of this frequency is negligible, *eg*, diesel alternator where the operating parameters are desired to be kept constant.

Tuning Resonant Devices

The limiting factor in active control of the vibrations is gene-rally the actuator which requires appreciable power and complicated controls. To overcome this problem, a novel way is to use simple beam and attachment of mass configuration. The actuator used in the present experiment is based on the parallel beam arrangement. The purpose of the actuator is to change the distance between the beams and it is not meant for supplying any force to primary vibrating system. In this configuration, two beams of equal and fixed length carry a mass at their end. Since the mass is rigidly attached at the end, two beams form a single cantilever beam. Distance h separates the two beams of width b and thickness d. The stiffness is changed by changing the second moment of area, I, which is given by

$$I = 2bd\left[\frac{d^2}{12} + \frac{(h+d)^2}{4}\right] \qquad (1)$$

When the beams touch each other, *ie*, h = 0, it is the case of minimum second moment of area for the beam. The upper limit is decided by the maximum value of the h. By substituting this into equivalent stiffness $K_{eq} = 3\,EI/l^3$, we get the ratio of the upper to lower frequencies, which is a measure of the usable frequency range, and is expressed as

$$\frac{\omega_u}{\omega_l} = \left\{\frac{1}{4} + \frac{3}{4}\left(1 + \frac{h}{d}\right)^2\right\}^{0.5} \tag{2}$$

If the distance through which the beams are moved is small compared to the thickness of an individual beam, then the above equation simplifies[2] to:

$$\frac{\omega_u}{\omega_l} \approx 1 + \frac{3}{4}\left(\frac{h}{d}\right) \tag{3}$$

It has been found that for a given distance between the beams, moving beams that remain parallel is the best option[2].

In many practical situations, it is necessary to adjust the opera-ting frequency of the tunable vibration absorber by only a few percent-age, as the excitation frequency is restricted to a relatively small frequency range. Further, if the vibration absorber is working in a changing environment but the excitation frequency is fixed, then the tuned frequency can change depending upon the ambient conditions. A tunable device can be adopted in this case which adjusts itself, in order to tune the excitation frequency, always.

DESCRIPTION OF THE ABSORBER

The concept of the vibration absorber using a variable stiffness beam has two important requirements.

(a) The two absorber beams should deform as a single beam in bending as a cantilever. Thus, the two beams should act as a top and bottom fibre of a cantilever and no movement should take place with respect to each other of these two fibres.

(b) The opening/closing movement between the rods should be independent of the attached mass. Also, there should not be any relative motion between the beam and the mass, when the machine is running.

The weight of the absorber should be as low as possible. The resonant device used is a parallel beam combination with one common lumped mass at each end. A mechanical system has been used to change the distance between the parallel beams and hence the moment of area about the neutral axis and thus the natural frequencies. Figure 1 shows the mechanism used in the proposed device. The rotary drive from the stepper motor is converted into opening and closing linear motion of the absorber rods. This has been achieved by a gear drive arrangement. A gear mounted on the stepper motor drives another gear mounted at the centre of the shaft. This shaft has right hand and left hand threads on the top and bottom portions. Accor-dingly two internally threaded (right and left hand) blocks have been mounted on the shaft's top and bottom portions. Further, the sub-assembly of the vibration ab-sorber has been mounted on these two blocks. For one complete rotation of the gears, the total linear distance covered is twice the pitch of the thread.

In the above arrangement, the masses are having a central slot and the rod ends are made free to move in the slot. Solenoids with plunger and spring arrangement have been used at the end of each

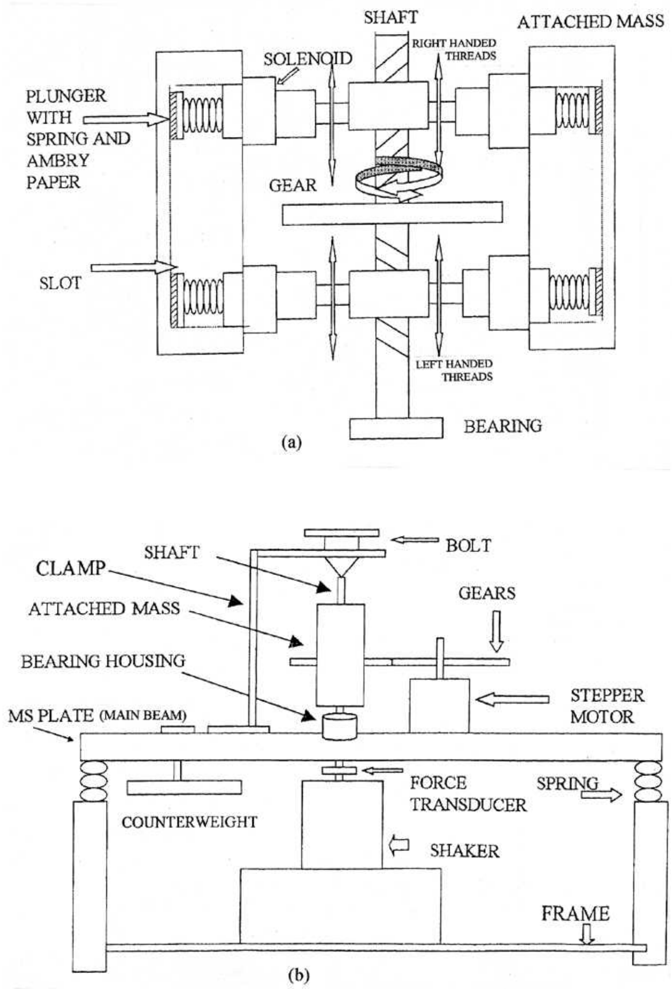

Figure 1 (a) Vibration absorber and (b) elevation of experimental set up

rod. When the solenoid coil is energised, the plunger is pulled back and it gets unattached with the inside slot surface of the attached mass. On de-energisation of the coil, the plunger due to spring tension retracts back and gets attached on the inside surface. At the end face of the plunger, a coarse ambry paper has been glued and the inside surface of the slot has been knurled. This ensured better gripping of plunger and attached mass and thereby the mass was kept fixed at the end.

The change in excitation frequency results in the transmission of appropriate signals for both solenoids and stepper motor through the control programme. The change in the geometry is executed, such that, the natural frequency of the absorber is equal to that of the excitation frequency. The results are obtained by analysing the effects on the resonant peaks before and after the movement of the absorber rods.

The natural frequency of the primary system is 130Hz. The mass of the primary system is 7.6 kg and that of absorber is 0.76 kg. The natural frequencies of the absorber alone for the maxi-mum and minimum distance between two rods of the absorber h are as follows

For h { = 0.028 m (minimum distance) }, f = 101.54 Hz.

h { = 0.064 m (maximum distance) }, f = 222.2 Hz.