ARPN Journal of Engineering and Applied Sciences

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EXPERIMENTAL AND FINITE ELEMENTS ANALYSIS OF A TUNED MASS ABSORBER FOR VIBRATION ISOLATION

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ABSTRACT

A structure under the influence of a harmonic excitation may start vibrating violently when the applied frequency is slowly approaching any one of its natural frequencies. Resonance results in large amplitude levels and vibrations. The possible alternatives to avoid resonance are either to shift the external harmonic frequency or to change the natural frequency of the structure itself. If both the alternatives are not feasible then tuned mass absorbers can be used. The emphasis of the current work is on modeling of a tuned mass absorber to suppress the vibrations of a single degree of freedom system operating at its fundamental natural frequency. System analysis was performed using ANSYS, a commercial finite elements package. ANSYS harmonic analysis was effective in predicting safe operating frequency range for the modified structure which was also verified experimentally. Hence it can be concluded that Finite Element Analysis can be effectively used to model dynamic systems and its response, hence possible solutions could also be sought. The adaptability of the current tuned mass absorber system when subjected to varying external frequencies is a new feature of the design.

Keywords: tuned mass absorber, vibration isolation, natural frequency, harmonic frequency.

1. INTRODUCTION

Vibration often becomes a problem due to unpleasant motions, noise, dynamic stresses that could lead to fatigue and failure of the structure or machine, energy losses, decreased reliability, and degraded performance. The natural frequency of the system is established by its mass and stiffness distribution. Usually there are two cases of vibration. They are free vibration and forced vibration. Free vibration (Clarance, 2007) occurs when the system oscillates due to an initial transient disturbance and henceforth there is no external agency influencing the oscillation. But forced excitation (Riviere, 2003) unlike free vibration is influenced totally by external forces. When the external excitation is repetitive or cyclic in nature, the system is forced to vibrate at this applied frequency. Rotating members in machineries are the major contributors of harmonic frequencies. But at some instance if the frequency of excitation coincides with one of the natural frequencies of the system, a condition called resonance occurs which results in very large amplitudes or oscillations. Failure of major structures like bridges, buildings and other super structures is more often related to vibration aspects and can be correlated to resonance phenomenon. Consequently one of the major reasons for vibration analysis is to predict when resonance may occur and to determine what steps to take to prevent it from occurring. Adding damping (Bert, 1973) can significantly reduce the magnitude of the vibration even at resonance. Magnitude can be reduced if the natural frequency can be shifted away from the forcing frequency by changing the stiffness or mass of the system. If the system cannot be changed, perhaps the forcing frequency can be shifted. If both the cases are not feasible, then tuned mass absorbers can be used to suppress the magnitude of vibrations. Tuned mass absorbers are now being effectively used in structures to absorb wind induced vibrations (Matsui *et al*., 2008; Liu *et al*., 2008; Chen *et al*., 2008) and transient vibrations induced due to earthquakes, cyclones etc (Wong, Cheung, 2008). One real time application of tuned mass damper is in TAIPEI 101, one of world's tallest buildings in Taiwan. The building has 101 floors with a total height of 509 metres equipped with 800 ton tuned mass damper at the 87th floor. It stabilizes the building against violent wind forces and also against possible earthquakes. Tuned Dampers are also being used in machineries running at fixed rotational speeds to curb large vibrations (Moradi *et al*., 2008). Stock bridge dampers in power transmission lines are commonly used to absorb transient vibrations set in the high tension wires due to high velocity winds.

In the current work a tuned mass damper system was designed to suppress the vibrations of a single degree of freedom system by making use of a commercial Finite Elements Method Package ANSYS 10. The results obtained by Finite Elements Package were verified experimentally and found to be accurate in predicting the modal frequencies.

2. THEORY OF TUNED MASS ABSORBER

Tuned Vibration Absorber (Frahm, 1909) is a system which is designed to reduce the amplitude of a Primary System operating at resonance or near to it by attaching an absorber. The principle is that, if a system (Primary system) has undesirable vibrations at an operating frequency which approaches its natural frequency, then to avoid vibrations a secondary system (Absorber System) having its natural frequency equal to the operating frequency, is coupled (Ormondroyd, 1928) to the Primary System to reduce its amplitude to zero. It is a passive vibration damping system [Kashani]. Basically a tuned damper can simply be attached to the system. It can be used to dampen the vibrations of any of the resonant

VOL. 6, NO. 1, JANUARY 2011 ISSN 1819-6608
ARPN Journal of Engineering and Applied Sciences

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modes by choosing appropriate value of stiffness and mass. Concept of a tuned mass absorber system attached to a single degree of freedom system is shown in Figure-1.

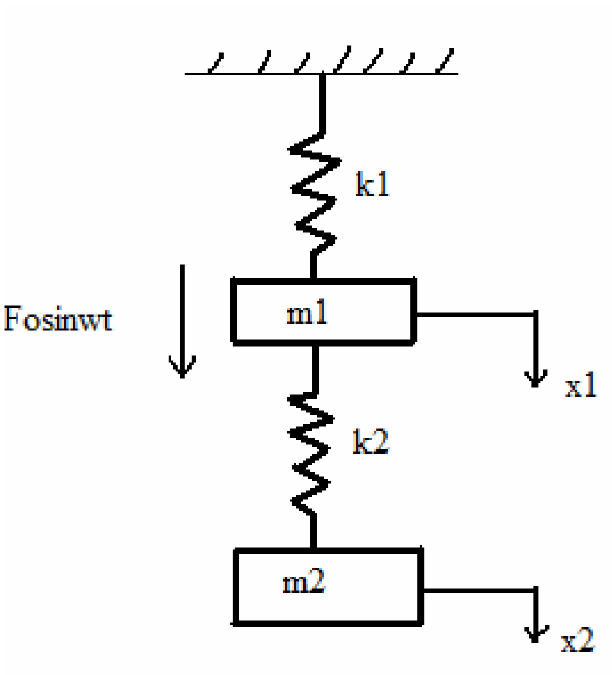

Figure-1. Two degree of freedom system.

k_1 and m_1 represents the primary system,
k_2 and m_2 represent the absorber system.
$F_0 \sin\omega t$ is the harmonic force acting on the primary system

When the primary system is attached with a tuned mass absorber or an absorber system whose natural frequency is equal to that of primary system, then even at the resonating frequency a condition of minimum amplitude is achieved at the primary system. The primary system amplitude is theoretically zero despite of being excited by a harmonic excitation (Thomson, 1990). The equation of amplitude of primary system is governed by

$$x_1 = \frac{F_0\left[1-\left(\frac{\omega}{\omega_2}\right)^2\right]}{k_1\left[1+\frac{k_2}{k_1}-\left(\frac{\omega}{\omega_1}\right)^2\right]\left[1-\left(\frac{\omega}{\omega_2}\right)^2\right]-\frac{k_2}{k_1}} \quad (1)$$

To make x_1=0, $\omega = \omega_2$

Hence for a tuned mass damper,

$$\sqrt{\frac{k_1}{m_1}} = \sqrt{\frac{k_2}{m_2}} = \omega \quad (2)$$

3. EXPERIMENTAL SETUP

An outline of the experimental setup is shown in Figure-2.

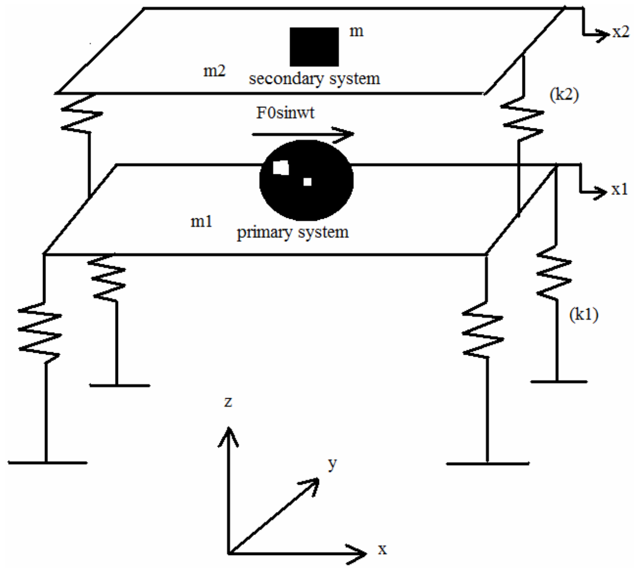

Figure-2. Outline of the experimental setup.

In Figure-2, k_1 and m_1 represent the stiffness and mass of primary system and k_2 and m_2 are stiffness and mass of the secondary system. Additional mass m is the extra mass necessary to tune the absorber system.

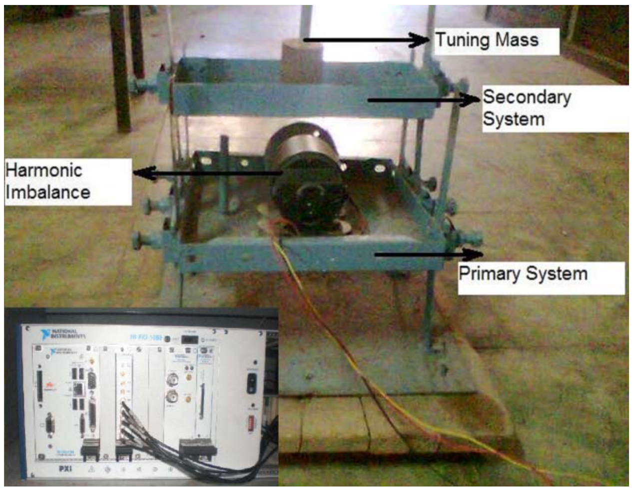

Figure-3. Experimental setup.

Figure-3 shows the experimental setup of a primary single degree of freedom system with a tuned mass absorber system (secondary system) attached to it to suppress the vibration amplitude at the operating frequency which is achieved by small imbalance attached to the DC motor. The system is constrained to move in x-direction due to lower stiffness in comparison with stiffness in y-direction. The rotary mass produces 2 components of centrifugal force one in x-direction and other in z direction. But due to lower stiffness along z-direction the x-direction component assures structure motion along x-direction.

Accelerometers were mounted on both the systems to study the vibration behaviour during various stages of the experiment. Two Accelerometers A and D

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1221 and A and D 3101with sensitivities 10 mV/g and 9.8 mV/g were used in the experiment. Data Acquisition was done through National Instruments PXI 4472 Sound and vibration module with 24 bit resolution and 102.4 kS/s acquisition rate capability. Data acquisition software used was National Instruments Lab VIEW 8.6. Figure-4 shows the Lab VIEW block diagram for vibration measurement of the tuned mass damper system.

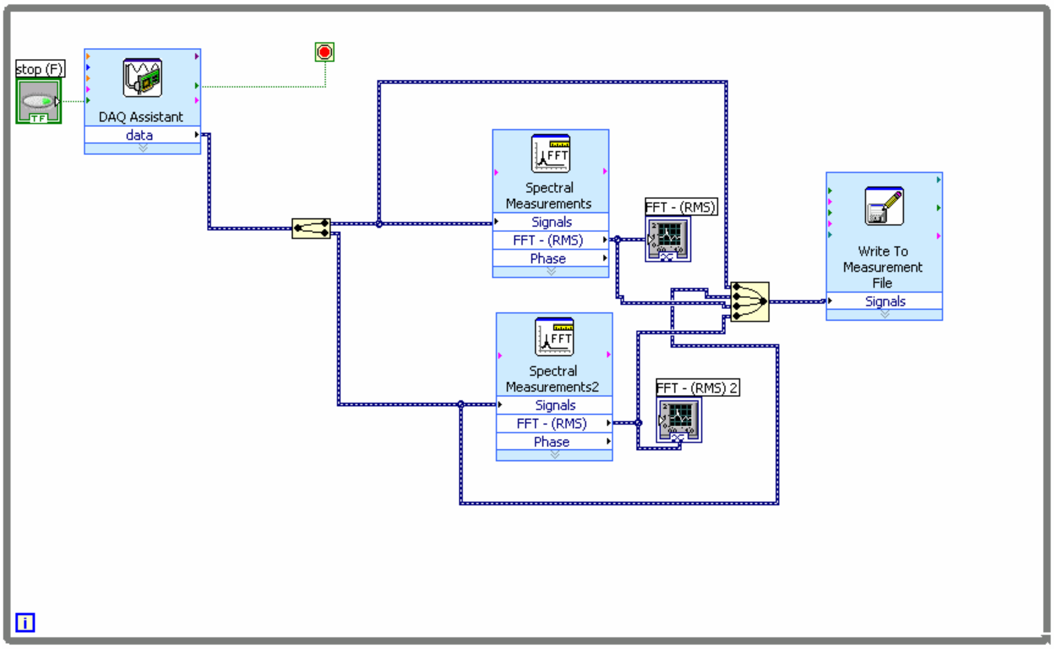

Figure-4. Lab VIEW block diagram for vibration measurement component specifications.

All beams are assumed to have the same stiffness and the beam specifications used in the experiment are as follows:

Breadth (B): 28 mm
Depth (D): 1.21 mm
For the primary system, Length (L1): 14.5 cm
Mass of the motor: 3.8 kg
Young's Modulus: 210 GPa
Density: 8000 kg/m^3

3.1 Features of the current experimental model

The current structural model has many advantages. The stiffness of the primary system can be changed by changing the span length of the four supporting beams. There is a provision to add mass to the absorber system which enables the absorber system to adapt to situations where primary system has to be operated at varying frequencies. Hence the absorber system has to be flexible to adapt itself to varying operating conditions also. In reality the experimental model can be viewed as a simple structure being operated by a harmonic disturbance which can be viewed as a prime mover which is often used in many industrial scenarios. In most of the cases the problem arises when the natural frequency of the structure is very close to rotational speed of the prime mover giving rise to resonance condition. Hence designing structures with a keen knowledge on its dynamic response characteristics is also necessary. The FEM analysis predicts the higher modes of vibrations of the structure also thereby indicating the safe mode of operation. This setup has been designed to be used as a lab experiment for under graduate students to understand the potential of absorber systems.

4. ANSYS MODELING

Model analysis was performed on the Structure using Block Lanczos as extraction method for solving the Eigen Values. The element type used was SOLID 92. The mass element used was STRUCTURAL MASS 3D. Results are interpreted separately for the primary and secondary systems which are shown in Figure-5 and Figure-6, respectively.

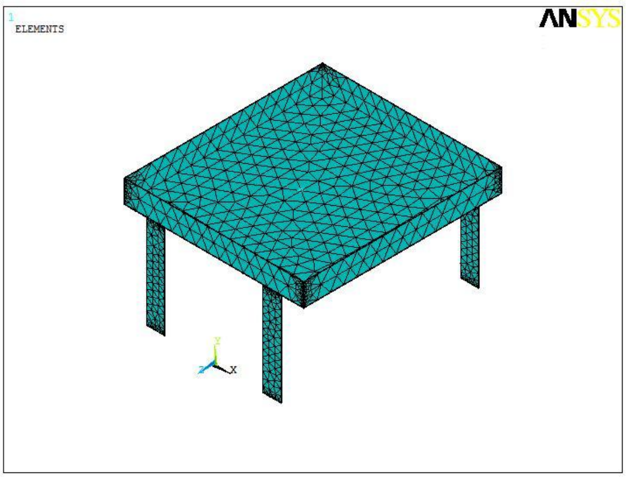

Figure-5. Primary system after meshing.

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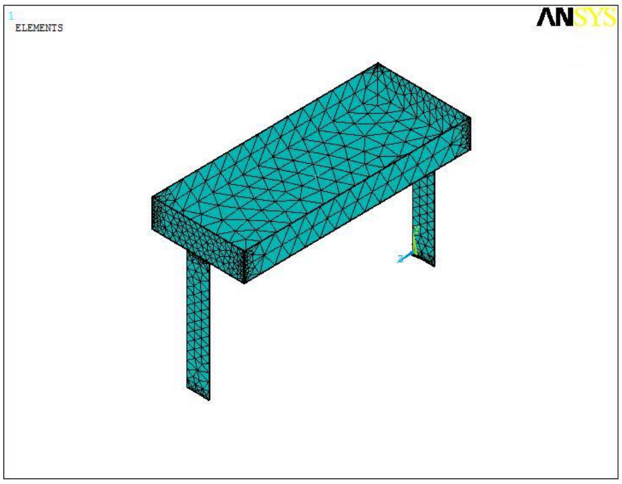

Figure-6. Absorber system after meshing.

Then secondary system or the absorber system was so designed that its natural frequency is equal to the natural frequency of the primary system. After several iterations optimal solution was obtained. Finally the total system was modeled along with the absorber system and new mass necessary for tuned condition was also computed. The model of the total system is shown in Figure-7.

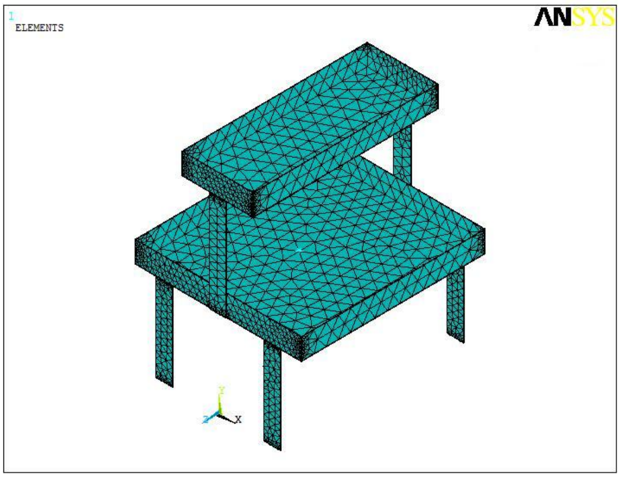

Figure-7. Total system after meshing.

It is important to note that modal analysis only gives the natural frequencies of the structures. Hence keeping an insight on the operating frequencies, the structure has to be designed in such a way that operating frequency is far away from the structural natural frequencies. Hence modal analysis can be a powerful tool to analyze vibration behaviour of structures.

5. RESULTS AND DISCUSSIONS

Table-1 compares different modes of vibration for primary and secondary system and for the total system.

Table-1. Different modes of vibrations of the total system.

System	Primary system	Absorber system	Total system
1st mode	8.3 Hz	8.3 Hz	7.3 Hz
2nd mode	18.5 Hz	59. 52 Hz	19.5 Hz

Table-1 indicates that fundamental frequency of the primary system has come down after attaching the absorber system. The first mode of vibration of the primary system obtained by modal analysis is shown in Figure-8.

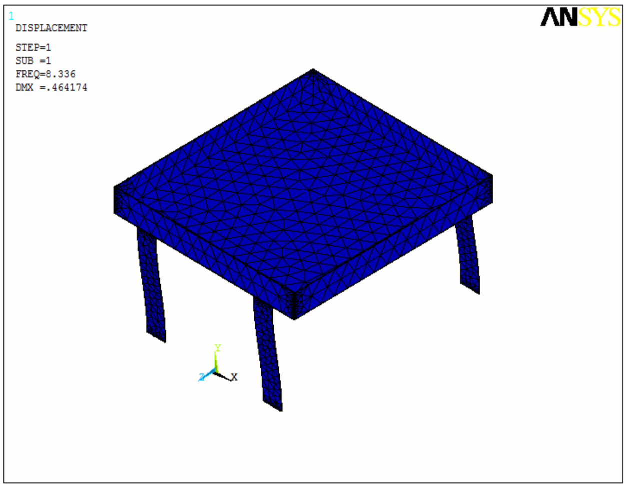

Figure-8. First mode of vibration of primary system.

The first mode of vibration of the secondary system obtained by modal analysis is shown in Figure-9.

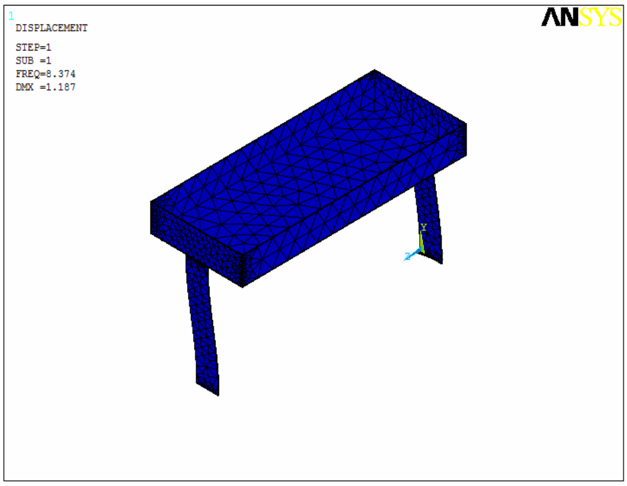

Figure-9. First mode of vibration of absorber system.

First mode of vibration is the lowest natural frequency of the structure which is also known as the fundamental natural frequency. ANSYS parametric design language (APDL) was made use of to make iterations easier. After several iterations it was found that natural

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frequency of the primary system is equal to natural frequency of the secondary system. This was achieved by placing a mass of 0.6Kg at the absorber system as described in the experimental setup. Now the absorber system was attached to the primary system and the total natural frequency was obtained which is shown in Figure-10.

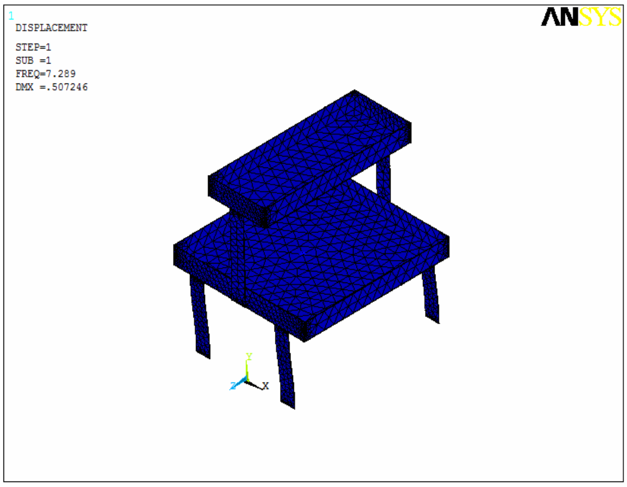

Figure-10. First mode of vibration of total system.

At the fundamental mode of vibration both the systems move in the same direction which is evident from Figure-10.

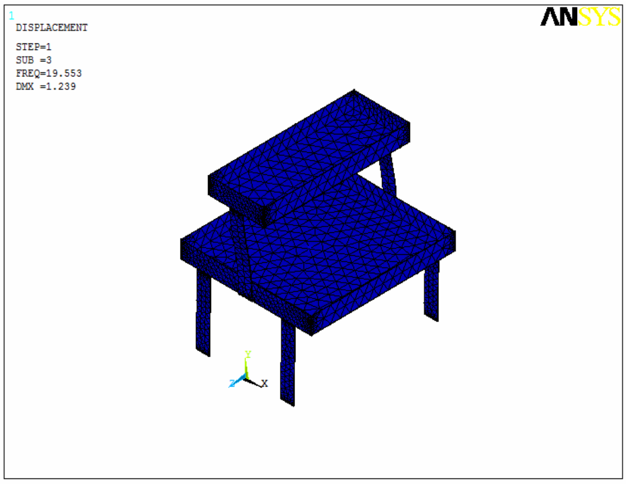

Figure-11. Second mode of vibration of total system.

At the second mode both the systems vibrate out of phase which is evident from Figure-11. To study the system response at various operating frequencies ANSYS harmonic analysis is used. Harmonic force with frequency varying between two limits is chosen and the system response to these frequencies can be studied. Harmonic analysis clearly indicates the safe operating limit which is evident by amplitude levels of resonating frequencies. By introduction of the tuned mass absorber the primary system's first mode natural frequency is shifted by a small unit. The harmonic response of primary system and secondary system clearly depicts the change in systems natural frequency after the introduction of tuned mass absorber. Figure-12 shows harmonic response of primary system to a unit force.

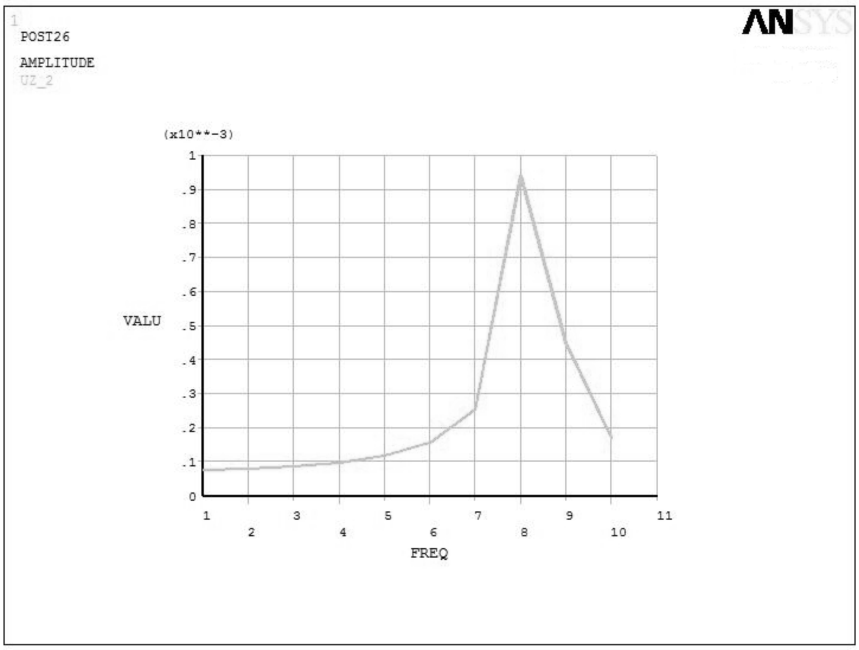

Figure-12. Response of the primary system to harmonic varying frequency.

Now after the introduction of the tuned mass absorber the harmonic response of the total system is shown in Figure-13.

ISSN 1819-6608
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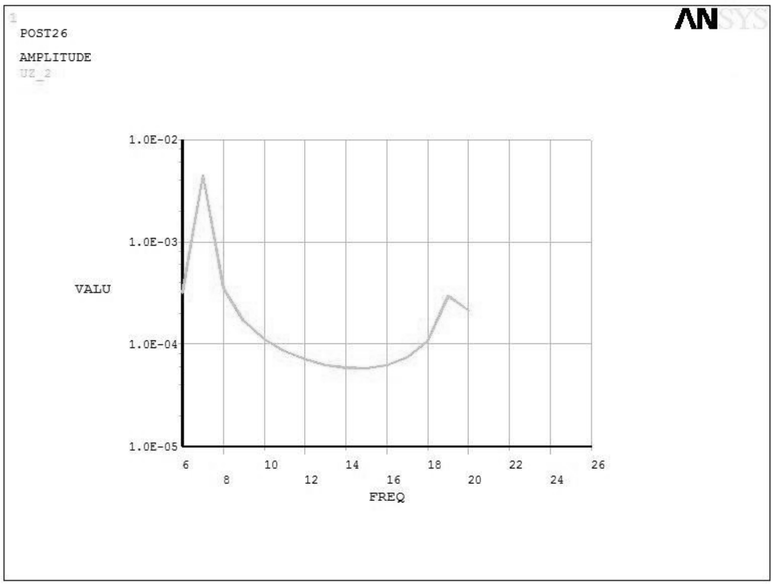

Figure-13. Response of the total system to harmonic varying frequency.

The safe operating frequency can be depicted looking at the response of the total system in Figure-13.

The heights of the two peaks can be adjusted by changing the stiffness of the spring in the damper. Changing the damping also changes the height of the peaks. The split between the two peaks can be changed by altering the mass of the damper (m_2).

Experiments were also conducted on the system with the absorber system attached to the primary system. The experimental setup is shown in Figure-3. Accelerometer readings were taken before and after tuning of the absorber systems. Tuning implies that the natural frequency of the absorber system is equal to the natural frequency of the primary system. This was achieved by placing mass on the absorber system. The accelerometer readings before and after tuning are shown in Figure-14 and Figure-15 respectively.

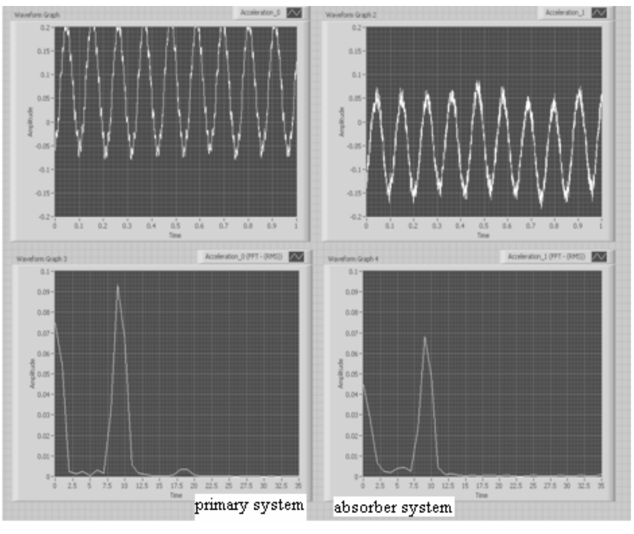

Figure-14. Vibration signal before tuning.

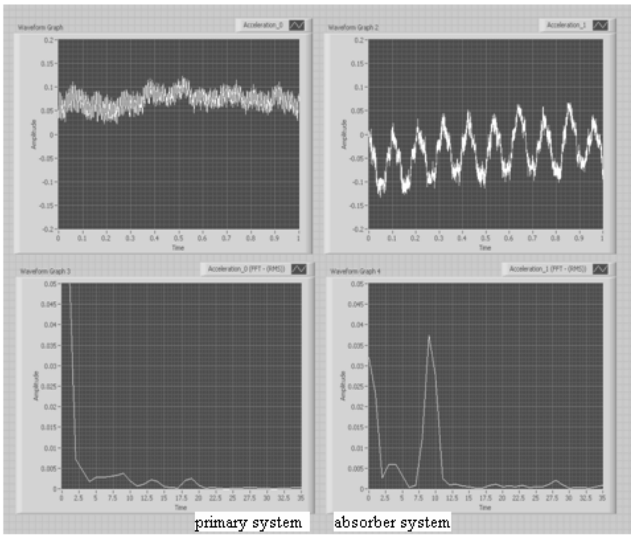

Figure-15. Vibration signal after tuning.

The two windows on the left side show the time domain and frequency domain signal coming from the primary system. The two windows on the right side show time domain and frequency domain signal coming from absorber system. It is clear that without tuning there is significant amount of vibration in the primary system. It can be noted from Figure-15 that after tuning the vibration level is negligible which is evident from the frequency domain window. Table-2 compares natural frequencies obtained by experiment and ANSYS.

Table-2. Comparison of natural frequency by ANSYS and experiment.

Analysis	ANSYS	Experiment
Natural frequency of primary system	8.3 Hz	8.5 Hz
Natural frequency of total system	7.3 Hz	7.6 Hz

6. CONCLUSIONS

A single Degree of Freedom system is chosen for analysis. The system is initially operated at its natural frequency. A tuned mass absorber system is so chosen as to suppress the vibration of the primary system at the operating frequency. It is observed that after introducing the tuned mass system the vibration amplitude has been suppressed significantly. The system was also modeled in ANSYS 10.0 a commercial FEM package and found to be reliable in predicting the natural frequencies. The operating range of frequencies could also be predicted by the analysis. Hence tuned mass absorbers could be effectively used to suppress vibrations at resonance frequencies. Finite elements analysis can help in optimum design of structures from vibration point of view. The flexibility of the structure to adapt to varying external frequencies is another key aspect of the design.

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