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Design and Analysis of Tune Mass Damper System

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***Abstract*— Vibrations are one of the major environmental factors that act on buildings and mechanical structures, potentially reducing their lifetime. Current trends in construction industry demand taller and lighter structures with more flexibility, yet they have a quite low damping value. This increases failure possibilities and also create problems from serviceability point of view. Many building structure and bridges fall because of vibration, if frequency of excitation coincide with one of the natural frequency of system. Now-a-days several techniques are available to minimize vibrations of a structure. Out of these, concept of using tuned mass damper is a recent trending innovation. This study is conducted to research the effectiveness of using tuned mass damper for controlling vibration of a structure. The report proposes a comparative analysis of passive vibration control system with un-damped system, on single degree of freedom structural frames subjected to external excitation. A combined unit of an accelerometer and an Arduino Uno R2 aids in measurement of vibrations and processes it in the form of acceleration vs. time graph, the preliminary results of which help us reach a conclusion. A working model was practically designed and manufactured to carry out the experiment for studying the TMD.**

Keywords*— Vibration control, Tuned mass damper (TMD), Single Degree of Freedom, Arduino Uno R2, Accelerometer

I. INTRODUCTION

Vibration control has its roots primarily in aerospace technologies to overcome problems such as tracking and pointing. Speaking specifically of the flexible space structures, vibration control technology has quickly moved into civil engineering and infrastructure-related industry to deal with issues like the protection of buildings and bridges from extreme loads of earthquakes and winds. Mostly these structures have low natural damping. So increase in the damping capacity of a structural system through various mechanical means has become a vast area of research in the technologically advanced era of tall and super tall buildings.

The control of structural vibrations produced by earthquake or winds can be done by various means such as modifying rigidities, masses, damping, or shape, and by providing passive or active counter forces. All vibrating structures dissipate energy due to internal stressing, rubbing, cracking, plastic deformations, and so on. The larger the energy dissipation capacity, the smaller the amplitudes of vibration. Some structures have very low damping of the order of 1% of critical damping and consequently experience large amplitudes of vibration even for moderately strong earthquakes. Active (external power source), Passive (no external power source), hybrid (combined use of active and passive control) and semi active (external energy requirements are less than typical active control) are the methods which can be used to control vibrations. Implementation of such solutions either individually or in a combination may prove to be effective in vibration control. Let's now get into the details of how a passive vibration control system like damping (especially tuned mass dampers) help modulate vibrations.

Tuned mass dampers (TMD) have been widely used for vibration control in mechanical engineering systems. In recent years, TMD theory has been adopted to reduce vibrations in tall buildings and other civil engineering structures. Dynamic absorbers and tuned mass dampers are the advanced versions of tuned absorbers and tuned dampers for structural vibration control applications. The inertial, resilient, and dissipative elements in such devices are mass, spring and dashpot (or material damping) for linear applications; and their rotary counterparts in rotational applications. Depending on the application, these devices are sized from a few ounces (grams) to many tons. Other configurations such as pendulum absorbers/dampers, and sloshing liquid absorbers/dampers have also been realized for vibration mitigation applications. TMD is attached to a structure in order to reduce the dynamic response of the structure. The frequency of the damper is tuned to a particular structural frequency so that when that frequency is excited, the damper will resonate out of phase with the structural motion. The mass is usually attached to the building via a spring-dashpot system and energy is dissipated by the dashpot as relative motion develops between the mass and the structure.

Objective of this paper is-

- To design spring-mass vibration systems.
- To measure the vibrations of that system.
- To reduce the vibrations by applying the concept of Tuned mass damper
- To verify result by measurement.

II. METHODOLOGY

Vibration measurement is an important area to determine the parameters of a vibrating body experimentally. During the analysis, number of assumptions are made; but practically all these assumptions are not satisfied. Thus, it is necessary to measure small and rapid vibrations, mainly in high speed machines to obtain the required data for analysis. To achieve this purpose, combination of an accelerometer and an Arduino is used. A single degree of freedom structure is made, to which the accelerometer is connected. The accelerometer measures the amount of vibrations with respect to time. Arduino further collects and processes this data to give us desired output graphs for study. The experiment is carried on the structure twice- once without any damping, while the other time with a Tuned Mass Damper (TMD). Comparative analysis of results provide a clear view of efficiency of TMD in vibration control of structures.

III. PRACTICAL IMPLEMENTATIONS OF TMD IN STRUCTURES

Till date TMD has been installed in large number of structures all around the globe. The first structure in which TMD was installed is the Centre point Tower in Sydney, Australia. There are two buildings in the United States equipped with TMDs; one is the Citicorp Centre in New York City and the other is the John Hancock Tower in Boston. The Citicorp Centre building is 279 m high and has a fundamental period of around 6.5 s with an inherent damping ratio of 1% along each axis. The Citicorp TMD, located on the sixty-third floor in the crown of the structure, has a mass of 366 Mg, about 2% of the effective modal mass of the first mode, and was 250 times larger than any existing tuned mass damper at the time of installation. Designed to be bi-axially resonant on the building structure with a variable operating period of adjustable linear damping from 8 to 14%, and a peak relative displacement of the damper is expected to reduce the building sway amplitude by about 50%.

IV. HARDWARE AND SOFTWARE

A. Arduino

To measure the vibrations of tuned mass damper we have used Arduino Uno. An official Arduino Uno R2 with descriptions of the I/O locations. The Arduino board exposes most of the microcontroller's I/O pins for use by other circuits. The current Uno provide 14 digital I/O pins, six of which can produce pulse-width modulated signals, and six analog inputs, which can also be used as six digital I/O pins.

Arduino Software-

A program for Arduino may be written in any programming language with compilers that produce binary machine code for the target processor. Atmel provides a development environment for their microcontrollers, AVR Studio and the newer Atmel Studio.

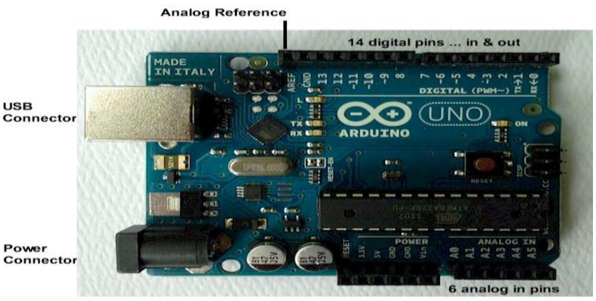

Fig.1 Arduino Uno

B. Accelerometer

ADXL335 is used along with Arduino. It is a small, thin, low power, complete 3-axis accelerometer with signal conditioned voltage outputs. The product measures acceleration with a minimum full-scale range of ±3 g.

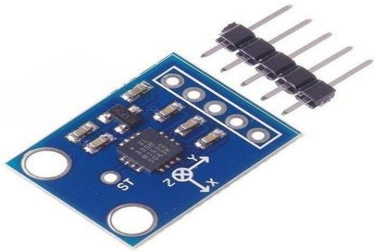

Fig.2 Accelerometer

V. MATHEMATICAL MODELING

It is a necessary to eliminate undesirable vibrations of externally excited body, by coupling some vibrating system to it. This attached system is called as vibration absorber. In that case, excited frequency should equal to natural frequency of body. Vibration absorber is used to control structure resonance.

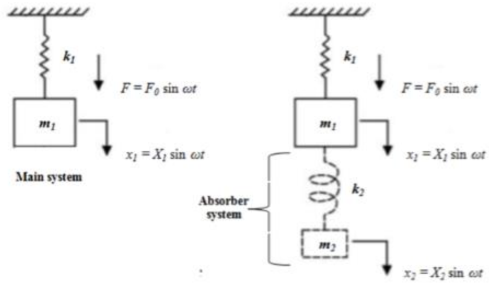

Fig.3 TMD arrangement

For example, if the excitation frequency ω is equal to the natural frequency $\omega = \omega_n = \sqrt{\left(\frac{k1}{m1}\right)}$ of the system amplitude of the vibrations would be very large because of resonance. Spring mass system ($m_2 - k_2$) is coupled to the main system as shown in figure. This spring mass system act as vibration absorber and reduces the amplitude of m_1 to zero, if its natural frequency is equal to excitation frequency i.e. $\omega = \sqrt{\left(\frac{k2}{m2}\right)}$ thus $\frac{k1}{m1} = \frac{k2}{m2}$.

When this condition is full filled, the absorber is called tuned absorber. Now, system becomes double degree of freedom.

$$m_1\ddot{x}_1 + k_1x_1 + k_2(x_1 - x_2) = F\sin\omega t \quad (1)$$
$$m_2\ddot{x}_2 + k_2(x_2 - x_1) = 0 \quad (2)$$

Solution will be,

$$x_1 = A_1\sin\omega t \quad (3)$$
$$x_2 = A_2\sin\omega t \quad (4)$$

Since,

$$\ddot{x}_1 = -\omega A_1 \sin\omega t \text{ and } \ddot{x}_2 = -\omega A_2 \sin\omega t$$

Putting values of $\ddot{x}_1$ and $\ddot{x}_2$ in eqn. (1) & (2)

$$(k_1 + k_2 - m_1\omega^2)A_1 - k_2A_2 = F \quad (5)$$
$$-k_2A_1 + (k_2 - m_2\omega^2)\, A_2 = 0 \quad (6)$$

Solving above eqn. We get,

$$A_1 = \frac{(k2 - m2\omega 2)F}{\beta} \qquad A_2 = \frac{k2F}{\beta}$$

Where, $\beta = [m_1 m_2 \omega^4 - \{m_1 k_2 + m_2(k_1+k_2)\}\omega^2 + k_1 k_2]$

To get the amplitude of mass m1 as zero, let us consider Eqn. (5),

$$A_1 = \frac{(k2 - m2\omega 2)F}{\beta} = 0$$

$$\omega = \sqrt{\left(\frac{k2}{m2}\right)} = \omega_n$$

From this it is found that the amplitude of absorber mass (m_2) is always much greater than that of the main mass (m_1). Thus, the design should be able to accommodate the large amplitudes of the absorber mass. Also as the amplitudes of m_2 are expected to be large, the absorber spring (k_2) needs to be designed from a fatigue point of view.

Arduino Program-

```
const int xpin = A0;
const int ypin = A1;
const int zpin = A2;
void setup()
{
 // initialize the serial communication:
 Serial.begin(9600);
 }

void loop()
{
 int x = analogRead(xpin);
 delay(10);
 int y = analogRead(ypin);
 delay(10);
 int z = analogRead(zpin);

float zero_G = 345.0;
float scale = 102.3;

Serial.print(((float)x - zero_G)/scale);
 Serial.print("\t");

 Serial.print(((float)y - zero_G)/scale);
 Serial.print("\t");

 Serial.print((((((float)z - zero_G)/scale)+0.29)*9.81)-9.81);
 Serial.print("\n");
 }
```

VI. DESIGN

To understand the concept of TMD experimental setup is designed with following components-

- Primary system – Any actual system can be considered as equivalent spring mass system. A primary system consists of spring mass arrangement whose amplitude of vibration is to be reduced. Four springs are selected for primary system.

Following table shows the spring stiffness calculation with constant weight of 1.2753 N for attachment to spring.

#	Total length (Meter)	Length with load (Meter)	Deflection (Meter)	Stiffness (N\M)
1	0.035	0.07	0.035	36.43714
2	0.039	0.08	0.041	31.10488
3	0.031	0.068	0.037	34.46757
4	0.035	0.071	0.036	35.425

Table No. 1 Spring Stiffness Calculation

Equivalent Stiffness of the springs (Ke) is given by

$K_e = k_1 + k_2 + k_3 + k_4$ …………… (Parallel Attachment of spring)

$K_e = 137.43459$ N/M

- Tuned mass damper (secondary system) - It is another spring mass system attached to primary system which act as a damper.
  Spring selected for secondary system having stiffness of 26.7 N/M therefore by relation we have,

$$\frac{Ke}{m1} = \frac{Kd}{m2} \quad \frac{137.43}{0.350} = \frac{26.7}{m2}$$

$m_2 = 0.068$ Kg

- Motor with eccentric mass – A motor with eccentric mass is used for external excitation force. Due to eccentricity of mass attached to motor it vibrates with large amplitude and act as source of vibration for primary system.
- Speed regulator - It used to regulate speed of motor in order to vary the excitation force.
- Frame- All system components are supported by a rigid frame.

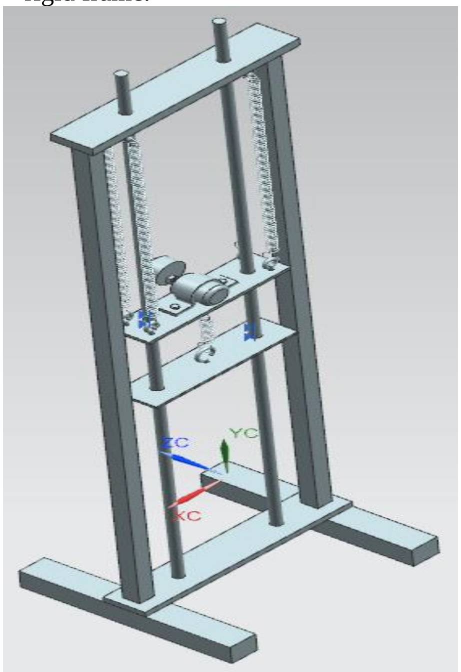

Fig.4 3-D Design model

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VII. OBSERVATION & RESULT

Results are obtained by using open source software provided by the manufacturer of Arduino. In the software we can get either serial plotter or serial monitor display. In this case we use serial plotter display to get simple graphical structure.
Here the results were concluded on the basis of comparison of two graphs as follows:

GRAPH FOR PRIMARY SYSTEM

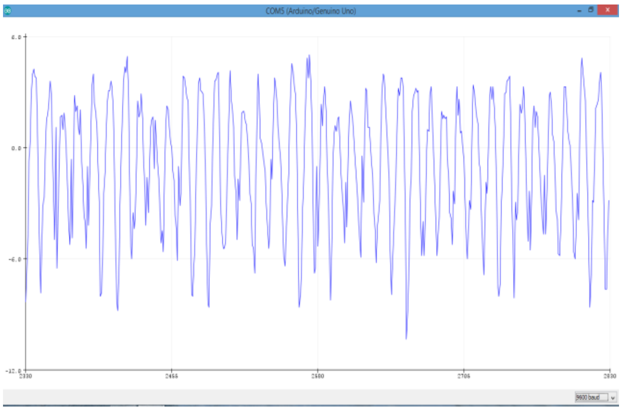

Fig.5 Graph for Primary System

The maximum peak for this primary system without attaching the damper plate is shown in this graph.

GRAPH FOR SECONDARY SYSTEM (WITH DAMPER PLATE)

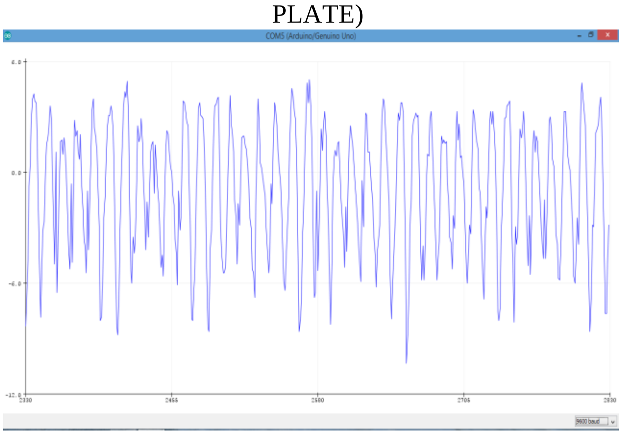

Fig.6 Graph for Secondary System

The maximum peak for this secondary system with attaching the damper plate is shown in this graph

- Maximum peak without damper: '-12'
- Maximum peak with damper: '-8'

VIII. CONCLUSION

The vibrations of a single degree of freedom system can be damped by converting it into two degree of freedom system and by comparison of the graphs between damped and un-damped condition, it has been concluded that when damper is attached to the primary system, the amplitude of vibration get reduced.

IX. FUTURE SCOPE

- To apply the concept to tuned mass damper on other mechanical system.
- To study the effect of different mass ratio on performance of tuned mass damper.
- To prepare more stable data acquisition system for vibration measurement.

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