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An Overview of Design, Behavior and Applications of Tuned Mass Vibration Absorber

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Abstract:** The tuned mass vibration absorber (TMVA) has been used for vibration control purposes in many sectors of mechanical/civil/automotive/aerospace engineering. A TMVA in its most generic form is a secondary system whose parameters can be tuned to suppress the vibration of a primary system. The secondary system is commonly a spring-mass-damper system and the TMVA suppresses the vibration at its point of attachment to the primary system. It is observed that by minor modification integrates very accurate results. It can be obtained for the variance of the response to excitation. It is found that the design can be based on the frequency tuning, leading to equal damping ratio and an accurate explicit approximation is found for the optimal damping parameter of the absorber and the resulting damping ratio for the response. In this paper the approach is discussed to the design of a TMVA, and a numerical procedure is elaborated to determine the optimum amount of damping. This paper presents an overview of literature related to the design, applications and behavior of Tuned Mass Vibration Absorber.

1. INTRODUCTION

Throughout the life, engineering machines undergo multiple sources of vibrations.The control of these vibrations in mechanical systems is still a thriving research field as it enables improvement in life as well as noise reduction and thereby comfort enhancement.

Vibration is the study of the repetitive motion of objects. Vibration is caused by anything that oscillates and it is transmitted through continuous media. Therefore, vibrations caused by a system effect other systems, unless the systems are separated by vacuum, as in space. It is observed in nature as well as in man-made devices and structures. When talking about vibration, one usually refers to the displacement, velocity and acceleration of a point on the vibrating structure. These quantities are functions of time, and are referred to as signals. Throughout the text, vibration signals will be referred to simply as vibration. Vibration, or vibration signals, can be divided into two main categories with respect to the characteristics of signals, namely, deterministic and nondeterministic. Deterministic vibration which can be categorized further, which can be periodic or non-periodic. Furthermore, on one hand periodic vibration is classified as sinusoidal or complex periodic, on the other hand non-periodic vibration is classified as almost periodic or transient (Bendat and Piersol). Sinusoidal vibration has the form of a sine signal, while complex periodic vibration may have the form of a combination of sine signals of different frequency and amplitude, or it may be any other signal that is periodic. Vibration caused by the rotation of unbalanced machinery such as turbines, wheels and shafts is an example to sinusoidal vibration. The vibration of an engine, however, is complex periodic since the components of the engine move periodically with different periods. Vibration that is present in the nature, e.g., vibration caused by earthquakes, often shows non-deterministic characteristics, hence the term random vibration. For repeated observations under basically identical conditions, random vibrations do not depict the same time histories. Therefore, statistical and probabilistic tools, e.g., mean and correlation, are used to study these vibrations. Transient vibration is another case of non-periodic vibration. In this case, an initial displacement, velocity or acceleration, or a sudden change in one of these quantities caused by an external element, causes the structure to experience vibrations that decay with time. Transient vibration is also referred to as free vibration, since the primary structure is not forced to move continuously, but instead, it vibrates freely due to initial conditions that are imposed on the structure. Vibration is used in different fields for numerous applications. Vibrating mixers and sieves are used in production systems for use in automated production, e.g,, automatic feeders. Vibrating sieves and tools are used in construction, e.g., pneumatic drills. In medicine, vibrating machines are used by doctors to cure diseases, e, g,, sound waves are used to break kidney stones, as in the lithotripter, which is used for Extracorporeal Shock Wave Lithotripsy (ESWL) treatments.

Although there are some applications that benefit form vibrations, in general, vibration is not desired in mechanical systems and engineering structures. The cyclic characteristic of vibration introduces dynamic stresses in members, which cause fatigue and can eventually lead to failure. Vibrations caused by earthquakes may cause structures to fail unless they are designed properly. Vibration of the passenger compartments of vehicles cause discomfort to the passengers.

The concept of resonance is very important and has a high priority in the study of forced vibrations. The term forced vibration is used for cases where the structure is excited continuously by an external source, e.g., a sinusoidal force. The forcing frequency is the frequency of this forcing function, e.g., the frequency of the sine wave.

For a vibratory system, resonance occurs when the forcing frequency is equal to the natural frequency, also referred to as the resonant frequency, of the structure. Under resonance, displacement of a nodal point reaches its highest value. Since displacements are maximum at resonance, resonance imposes the highest strains and stresses on the structural members, and therefore causes the most damage. The idea in using a vibration absorber in a system is mainly to reduce the destructive vibrations that the primary system experiences when it is excited at or around its natural frequency, i.e., when it experiences resonance, A vibration absorber can be described as an auxiliary system connected to the primary system of concern, which attenuates the vibrations of the primary system by absorbing and/or dissipating its kinetic energy. Vibration absorbers have found extensive use in engineering systems for about a hundred years. The first hydraulic vibration absorber was actually used in 1909 to reduce the rolling motion of ships, as the technology advances; applications require more efficient, smaller and lighter vibration absorbers.

2. TYPES OF VIBRATION

2.1 Free vibration

It occurs when a mechanical system is set off with an initial input and then allowed to vibrate freely. Examples of this type of vibration are pulling a child back on a swing and then letting go or hitting a tuning fork and letting it ring. The mechanical system will then vibrate at one or more of its "natural frequency" and damp down to zero.

2.2 Forced vibration

It is when a time-varying disturbance (load, displacement or velocity) is applied to a mechanical system. The disturbance can be a periodic, steady-state input, a transient input, or a random input. The periodic input can be a harmonic or a non-harmonic disturbance. Examples of these types of vibration include a shaking washing machine due to an imbalance, transportation vibration (caused by truck engine, springs, road, etc.), or the vibration of a building during an earthquake. For linear systems, the frequency of the steady-state vibration response resulting from the application of a periodic, harmonic input is equal to the frequency of the applied force or motion, with the response magnitude being dependent on the actual mechanical system.

3. VIBRATION REDUCTION METHODS

Damping is the conversion of mechanical energy (vibrations) into heat. Five basic methods exist for vibration control of industrial equipment, as Force Reduction of excitation inputs due to, for example, unbalance or misalignment, will decrease the corresponding vibration response of the system. Mass Addition will reduce the effect (system response) of a constant excitation force. Tuning (changing) the natural frequency of a system or component will reduce or eliminate amplification due to resonance. Isolation rearranges the excitation forces to achieve some reduction or cancellation. Damping is the conversion of mechanical energy (vibrations) into heat.

Vibration reduction methods are classified into three distinct categories:

1. Active control has been developed throughout the last two decades, despite the simplicity of the underlying principle. Schematically, the objective lies in reducing an undesirable vibration by generating an out-of-phase motion so that destructive interferences are generated. Active control generally gives the best vibration reduction performance, but it is not widely used due to its cost, the necessity to have an external energy supply, its lack of robustness and reliability in the industry.

2. Semi-active control of machines using electro- and magneto-rheological fluids was recently proposed. The particularity of these fluids lies in their varying viscosity with respect to the electric or magnetic field in which they are plunged. Since no energy is transferred to the controlled system, these techniques are robust and reliable while offering a vibration reduction level similar to active techniques. However, the modeling of the fluid behaviors as well as the development of the controller represent major challenges that still complicate the use of the systems for real-life structures.

3. Passive vibration method simply a structural modification by adding either a mass or a dynamical vibration absorber (DVA). It represents a very interesting alternative to the aforementioned methods as its performance is acceptable without requiring external energy supply.

3.1 Tuned Mass Vibration Damper

The tuned mass vibration damper (TMVD) is probably the most popular device for passive vibration of mechanical machines. It is commonly used for civil (e.g., Millenium bridge, Taipei 101 and Burj-el-Arab buildings) and electromechanical engineering machines (e.g, cars and hightension lines). Its broad range of applications is mainly due to its linear character and thereby the solid theoretical and mathematical foundations on which it relies. Despite the well-established theory for simple primary systems, the design of such an absorber is still a challenging problem when it is coupled to more complex machines. The present section aims to review the existing tuning procedures when a TMVD is attached to single-degree-of-freedom (SDOF) and multi-degree-of-freedom (MDOF) systems presenting linear and nonlinear characteristics.

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.

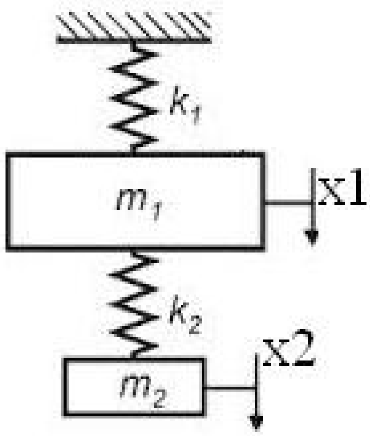

Figure 1. Tuned Mass Absorber

For example, if the excitation frequency ω is equal to the natural frequency $\omega=\omega_n=\sqrt{\frac{k1}{m1}}$ 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. $\omega=\sqrt{\frac{k2}{m2}}$

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.

$m_1 \ddot{x}_1 + k_1 x_1 + k_2(x_1-x_2)= F \sin\omega t$ (1)
$m_2\ddot{x}_2 + k_2(x_2-x_1)= 0$ (2)
Solution will be,
$x_1 = A_1\sin\omega t$ (3)
$x_2 = A_2\sin\omega t$ (4)
SINCE,
$\ddot{x}_1= -\omega^2 A_1\sin\omega t$ and $\ddot{x}_2= -\omega^2 A_2\sin\omega t$
Putting values of $\ddot{x}_1$ and $\ddot{x}_2$ in eqn. (1) & (2)

$(k_1+k_2-m_1\omega^2)A_1-k_2a_2=F$
$-k_2a_1+(k_2-m_2\omega^2)A_2=0$
Solving above eqn. We get,
$A_1 =\frac{(K2-m2\omega2)F}{\beta}$ (5)
$A_2=\frac{K2F}{\beta}$ (6)
Where, $\beta= [m_1m_2\omega^4-\{m_1k_2+m_2(k_1+k_2)\}\omega^2+k_1k_2]$

To get the amplitude of mass m_1 as zero, let us consider Eqn.(5),
$A_1 =\frac{(K2-m2\omega2)F}{\beta} =0$
$\omega = \sqrt{\frac{k2}{m2}} = \omega_2$

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 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.

3.2 Tuned Liquid Mass Dampers (TLMDs)

Tuned liquid mass damper basically consists of liquid tanks and liquid depth. Due to liquid, the damper response of TLMDs is nonlinear in nature and also frequency dependent device. The effectiveness of TLMD is increased by using multiple tuned mass dampers (MTLMDs) in which number of liquid tanks are increased to minimize the dynamic response of the machines. These MTLMDs can be used for high rise buildings to reduce the earthquake vibrations. Advantages of TLMDs are low initial and maintenance cost, easy to install as compared to TMDs. Fujino et al., [9] have developed rectangular model of the tuned liquid damper (TLMD) to reduce the dynamic response of structures. Experiments were carried out to make out the characteristics of TLMD and the relationship between the TLD and structure using the shaker test with a harmonic external loading. Chakraborty [10] have examined the uncertainty of the bounded system parameters to study the optimum design of liquid column vibration absorber for seismic vibration control of structures. The LCVA is modeled as a SDOF system. Results show that LCVA tends to reduce the level of uncertainty. It was also observed that neglecting the effect of system parameter uncertainty may overestimate the damper performance

3.3 Applications of TMVD

Tuned mass vibration dampers (TMVDs)[11] are comprising a spring, mass attached to the structure and are used for vibration control of structures when subjected to earthquake excitations. It is a frequency reliant device. The passive tuned mass vibration damper (PTMVD) was developed and implemented by Lin et al., [12] for seismic reduction of irregular buildings. Here, numbers of real earthquakes were considered for analysis (numerical and statistical) of five storied building. Results demonstrate that PTMVD effectively damps the response on building during earthquake. Two pedestrian bridges, equipped with tuned mass dampers almost two decades ago, have been evaluated for their long-term performance. The response due to pedestrian action, forced vibrations as well as deck response is compared to the response at installation. Since both inherent bridge damping and absorber effects reduce the overall response, it is difficult to separate the two effects. The concept of effective damping is used to show the correlation between the two effects. This concept permits separating the effects of bridge damping, absorber mass, and detuning in an approximate manner. The assessment shows that both bridges and tuned mass dampers are still working satisfactorily. Detuning has been observed in one of the two bridges, where it was existed already at installation. Parameters of the tuned mass dampers depend significantly on seasonal temperatures to give some practical limits to exact optimal tuning.

The combined effect of detuning at installation and detuning due to temperature effects can be expressed as loss in effective damping. Under summer and winter conditions, the loss in effective damping for the girder bridge was more than that for cable-stayed bridge. The temperature effects thus reduced the effective damping by in both cases. [2]Zuo et al., [13] have developed multi degree of freedom tuned mass vibration damper. To obtain the exact solution experiments were conducted sequentially to optimize the 2 degrees of freedom system. TMVD can be tuned to damp vibrations of the two flexural modes of a free-free beam. Pinkaew et al., [14] have reported that structure with tuned mass damper was not so effective for seismic damage reduction. Peter, [15] has discussed the theoretical and experimental studies on tuned mass vibration damper for the seismic retrofitting of existing structures. A simple and practical hybrid vibration absorber (HVA) was proposed for global vibration control of flexible structures under random stationary excitations.

A linear translational feedback signal used to synthesize active moment via a moment actuator. The passive tuned mass damper (TMD) can suppress vibration in the primary structure at the neighborhood of the coupling point but enhances vibration at other locations and other frequencies. The presented HVA provides more broadband attenuation on vibrations at multiple points when compared with the passive TMD in experimental verification on a beam structure. The proposed HVA is a simple and economic so it can retrofit the conventional TMD. For the design and firm execution of active controllers with optimum accuracy used to suppress multi-mode vibrations are not available, eigen-functions and eigen-state feedback of primary structures in general applications. so that's why, there is a need to generate unusual methods for design firm active controllers for global and broadband active vibration control in flexible structures. For example, instead of a coupling point HVA is capable to damp multi-mode vibrations in flexible structures. [3] To control the first three modes of the beam by using the independent modal space control method on the surface of a flexible beam with piezoelectric pieces for active vibration control. The dynamic equation of the beam is deduced by Hamilton's principles, and numerical simulation of the active vibration control of the first three modes of the beam are very significant, when compared with before and after control of flexible beam vibration suppression. Low energy consumption, light weight, and high energy efficiency, smart structures are the advantages of piezoelectric having very effective and good at suppressing the vibration of a flexible structure. It have spacious application in aerospace industry.[4]

Almazan et al., [16] have observed that new bi-directional and homogenous Tuned mass vibration dampers are very effective in controlling the seismic response of structures. Nonlinear vibration absorber was designed to satisfy the primary resonance vibrations of a SDOF weakly nonlinear oscillator having cubic non- linearity. The linear natural frequency of the nonlinear absorber was tuned to be approximately one-third the linearised natural frequency of the primary nonlinear oscillator. The low frequency mode for the absorber is favorably considered based on the fact that the nonlinear absorber can be easily realized in practice by using a light-weight mass attachment with small values of linear and nonlinear stiffness of coupling.

For a given primary nonlinear oscillator and absorber mass, implementation of three-to- one internal resonances requires the smallest value of the absorber linear stiffness among three options for utilizing internal resonances to design nonlinear absorber. The method of multiple scales is used to obtain the averaged equations that determine the amplitudes and phases of the first-order approximate solutions to the vibrations of the primary nonlinear oscillator and nonlinear absorber. It is found that the absorber response may admit either forced vibration having the forcing frequency or a combination of forced vibration and free-oscillation term having one third the forcing frequency. The nonlinear absorber can effectively suppress the amplitude of primary resonance response and eliminate saddle-node bifurcations occurring in the frequency-response curves of the primary nonlinear oscillator. Numerical results are given to show the effectiveness of the nonlinear absorber for suppressing nonlinear vibrations of the primary nonlinear oscillator under primary resonance conditions (5)

Marano et al., [7] have proposed a linear tuned mass vibration damper for seismic control of structures by using constrained reliability based on optimization technique. The optimum design of TMD was based on the natural frequency and damping ratios. In this, by experimentally analyses carried out under different earthquakes records for SDOF structures with various periods, the selection of mass ratio takes place. According to these information; the mass ratio, period of the structure and the excitation must be taken into consideration during optimization. So that, the performance of the TMD is based on the parameters like mass ratio, external excitation and period of the structure. In addition to these optimization techniques, for MDOF structure a meta- heuristic algorithm called Harmony Search was proposed and compared with the other methods. For Harmony Search (HS) approach, they compared proposed method and simple expressions showed that the optimum parameters are more economical and feasible. The HS was conducted under six different earthquakes and the results were tested under benchmark earthquakes in order to show that the TMD parameters are true optimum. In that increase of the damping ratio was directly effective on the performance. In this case, the damping ratio can be thought as an unnecessary parameter to be optimized. But, the optimum mass and frequency ratios must be found corresponding to the damping ratio. Due to, security reasons, Especially, the mass ratio must be kept in minimum levels for high rise buildings.[6]Further, Marano et al., [8] have reported the optimum parameter of tuned mass damper for minimization of displacement of the structure.

4. CONCLUSION

In this paper, we take the overview of vibration theory, various vibration reduction methods like Passive, Semi-Active, Active and applications in details. For effective working of TMVA, natural frequency of primary, secondary and forced frequency must be equal. In Passive TMVA the limited range of vibrating frequency is effective and Active TMVD is used for random vibration in SDOF and in few special cases of MDOF.

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