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Shape Memory Composite as Actively Tuned Vibration Absorber

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Abstract—*Shape memory materials (SMMs) are known for their excellent vibration absorption properties. The current work is focused on study of vibration absorption characteristics of Shape Memory Composites (SMCs), as tuned vibration absorber has been taken up. Thin cantilever plate of SAC works as the absorber system in the configuration.*

1. INTRODUCTION

A well-established vibration control device is the tuned vibration absorber (TVA). Even though such a device may be have different shapes it acts like a spring-mass system. For simplicity, a beam-like TVA can be used. One of the drawbacks of such a device, however, is that it can detune during operation because of changes in forcing frequency. To maintain its tuned condition a variable stiffness element is required so that the natural frequency of the absorber can be adjusted in real-time.A beam-like TVA has been realized using shape memory alloy (SMA) by Rustighi et al [1]. Such a material changes its mechanical properties with temperature. By varying the temperature of the beam the absorber can be tuned in order to maintain the vibration of the host structure to be very small. Many researchers studied vibration control using SMA. Baz et al. demonstrated SMA actuation capability to control the vibration frequencies of composite beams by activating optimal sets of embedded SMA wires [2]. The presented work focuses on the application of Smart adaptive composite (SAC) or shape memory composite (SMC) in active vibration control.

2. OVERVIEW OF TUNED VIBRATION ABSORBER

None of the structures such as buildings, bridges, towers, or, machines or machine partsthat may be composedof many sub-assemblies can be made completely rigid [3]. They move in response to natural disturbances. In particular, every member has a set of special frequencies called natural frequencies, or resonance frequencies, at which it will respond particularly severely. When subjected to periodic forces at one of these natural frequencies, themember may respond with vibrations of amplitude large enough to affect the normal functioning of the member itself or its assembly or perhaps even dangerous to itself or to the assembly. If engineers expect a member to be subject to a periodic force at or near one of its natural frequencies, they may incorporate into its design a special device called a vibration absorber which is just a secondary spring mass system attached to the main system as shown in Fig. .(1). This is a device that suppresses vibration of the structure at one of its natural frequencies by transferring the energy that would cause such a vibration into vibration of a secondary mass.

However complicated a system may bewe are concerned only with only oneof its natural frequencies, and hence it can be modeled as a spring masssystem whose resonant frequency is the troublesome natural frequency of the system. Moreover, though there can be any degree of sophistication in design, a tuned vibration absorber is in essence a secondary spring mass system attached to the primary system. Therefore the system (mass m_1 and spring constant k_1) coupled with tuned vibration absorber (mass m_2 and spring constant k_2) can be modeled using the coupled spring-mass system in Fig. 1.The system of differential equations describing the motion of such a spring mass system is given by,

$$m_1\ddot{x}_1 = -k_1x_1 - k_2\left(x_1 - x_2\right) + F_1\sin\omega t$$
$$m_2\ddot{x}_2 = -k_2\left(x_2 - x_1\right) \quad (1.1)$$

A particular solution of this system can be obtained by the method of undetermined coefficients for a single second order equation, in which x_1, and x_2 are each of the form $c_1\sin\omega t + c_2\cos\omega t$. But since there are no first order derivatives, and the second derivative of a sine function is still a sine function, we can simplify the solution in the form:

$$x_1 = A\sin\omega t\,,\; x_2 = B\sin\omega t \quad (1.2)$$

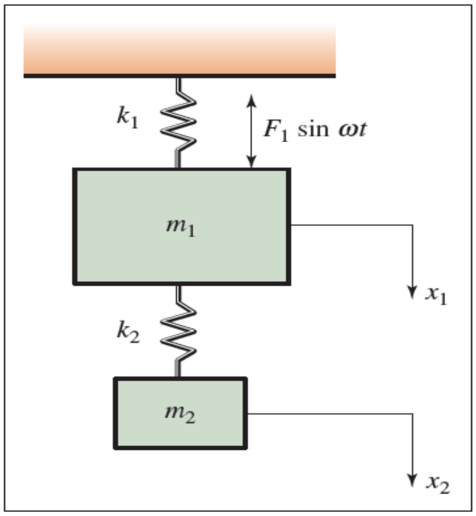

Fig. 1: Tuned Vibration Absorber (without damping)

Where A and B are constants to be determined.Using the forms indicated in (1.2) into (1.1), the forms of A and B that satisfy the system of equations are obtained.

$$\left(k_1 + k_2 - m_1\omega^2\right)A - k_2 B = F_1$$

$$-k_2 A + \left(k_2 - m_2\omega^2\right)B = 0$$

And

$$A = \left(\frac{F_1}{k_1}\right)\left[\frac{1-\left(\omega/\omega_2\right)^2}{\left(1+\mu\left(\frac{\omega_2}{\omega_1}\right)^2 - \left(\frac{\omega}{\omega_1}\right)^2\right)\left(1-\left(\frac{\omega}{\omega_2}\right)^2\right) - \mu\left(\frac{\omega_2}{\omega_1}\right)^2}\right] \quad (6.3)$$

Where $\omega_1 = \sqrt{k_1/m_1}$ and $\omega_2 = \sqrt{k_2/m_2}$ are the natural frequencies of the systemalone and of the absorber alone respectively; $\mu = m_2/m_1$ is the ratio of the mass of the absorber to the mass of the structure; and F_1/k_1 is the static displacement of$m_1$under a constant force F_1.

Note that the masswill not vibrate at all when natural frequency ω_2 of the absorber equals the forcing frequency ω. So to eliminate vibrations of the system at its natural frequency, we tune the absorberto the natural frequency of the member by setting $\omega_2 = \omega_1$. It is convenient to measure the amplitude of the vibration of the structure with the quantity in LHS, which is dimensionless and therefore independent of particular units of measurement:

$$\frac{A}{F_1/k_1} = \frac{1-\left(\omega/\omega_1\right)^2}{\left(1+\mu-\left(\frac{\omega}{\omega_1}\right)^2\right)\left(1-\left(\frac{\omega}{\omega_1}\right)^2\right)-\mu} \quad (1.4)$$

To suppress the resonance of the system-absorber assembly at its two natural frequencies, and to dissipate the energy of the system into vibrations of the absorber mass m_2, we can incorporate damping into the absorber. This is represented by a dashpot providing viscous damping with a damping constant of β Fig. (2). The resulting damped vibration absorber is sometimes called a tuned mass damper. To write its equations of motion, we add the appropriate damping terms to (1) and obtain:

$$m_1\ddot{x}_1 = -k_1x_1 - k_2\left(x_1 - x_2\right) + \beta\left(\dot{x}_2 - \dot{x}_1\right) + F_1 \sin \omega t$$
$$m_2\ddot{x}_2 = -k_2\left(x_2 - x_1\right) - \beta\left(\dot{x}_2 - \dot{x}_1\right) \quad (1.5)$$

A sinusoidal steady state solution can be derived for this system of the form

$$x_1 = A\sin\left(\omega t + \phi_1\right)$$
$$x_2 = B\sin\left(\omega t + \phi_2\right) \quad (6.6)$$

Where $A$and$B$ are constants to be determined. We forgo this calculation here (because it is rather long and messy), but the result is that the amplitude A with which mass $m_1$vibrates satisfies:

$$\left|\frac{A}{F_1/k_1}\right| = \sqrt{\frac{\left(2\frac{\beta}{\beta_c}s\right)^2 + \left(s^2 - r^2\right)}{\left(2\frac{\beta}{\beta_c}s\right)^2\left(s^2 - 1 + \mu s^2\right)^2 + \left(\mu r^2 s^2 - \left(s^2-1\right)\left(s^2 - r^2\right)\right)}} \quad (6.7)$$

Where $r = \omega_2/\omega_1$, $\beta_c = 2m_2\omega_1$ and $s = \omega/\omega_1$ The classical formulae given in many vibration engineering textbooks and handbooks for optimal tuning of the tuned mass damper are:

$$r = \frac{1}{1+\mu} \text{ and } \frac{\beta}{\beta_c} = \sqrt{\frac{3\mu}{8\left(1+\mu\right)^3}}$$

Finally we note that just as one can eliminate the dashpot from the tuned mass damper toproduce produce an undamped tuned vibration absorber based on the spring alone, as is analyzed, one can also eliminate the spring to a vibration absorber based on viscous damping alone called a viscous vibration absorber. Viscous vibration absorbers are less effective than tuned mass dampers, but are simpler to construct.

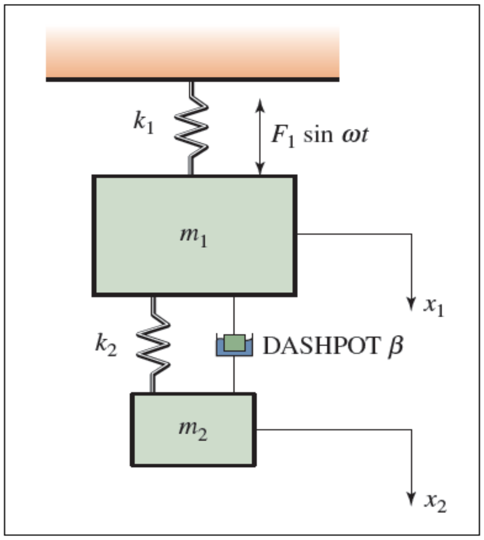

Fig. 1.2: Conventional Vibration Absorber (with damping) [34]

3. ACTIVELY TUNED VIBRATION ABSORBER:

Quite often in real time systems subjected to vibrations, we come across problems wherein the system's resonant frequency changes may be due to a change in inertia or elastic properties of the system. In such situations tuned vibration absorbers would cease to function. Now the problem requires that the inertia or the elastic properties of the tuned vibration absorber should change suitably in accordance with the changes in the main system parameters so that the vibration absorption is achieved in all changing conditions.

It has been evident from the work that the elastic properties of the SMC change when activated by thermal and/or mechanical parameters. Hence proper control of these parameters gives us good control over the elastic properties. Therefore there arises a possibility of using an SMC in place of secondary spring mass system which would lead to a more advanced vibration absorbers termed as actively tuned vibration absorbers.

The SMC secondary system has been modeled as a cantilever plate and coupled to the primary spring mass system as shown in Fig. (3). The SMC composed of NITINOL SMAfibers in Epoxy SMPwith SMA volume fraction of 0.5 has been taken for the study. The stress strain curve for SMC during loading and unloading, forms a parallelogram exhibiting definite residual strain which is recovered at the end. And the elastic modulus necessarily remains constant on the sides of these parallelogram. Hence the SMC can offer four elastic moduli which can be controlled by suitably activating the SMC, and which is the basis for actively tuned vibration absorption. The stress strain curve for SMC for loading and unloading cycle at temperatures over 55^0 C forms a closed loop having four slopes as shown in Fig. (5) with 100% strain recovery. The four elastic moduli of SMC for thermo-mechanical loading and unloading cycle at 55^0C are 33581.41 N/m^2, 26844.36 N/m^2, 13231.41 N/m^2, and18684.62 N/m^2.

4. FINITE ELEMENT MODEL BUILDING AND ANALYSIS

The absorber system for the tuned vibration absorber was modeled as a cantilever plate of SAC with the dimensions shown in Fig. 6.4.

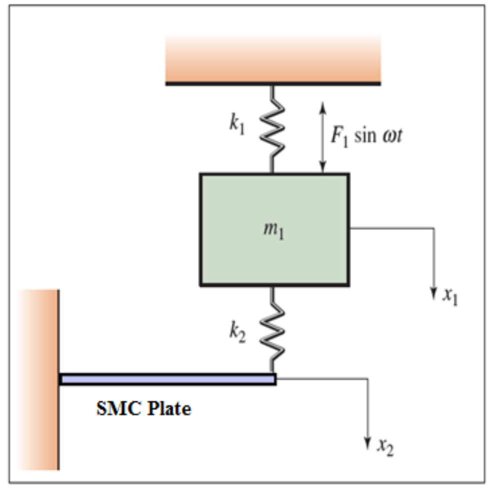

Fig. (3): SMC Plate as absorber

FEA model of the SAC plate was built using Hypermesh 9.0 as a two dimensional plate with Quad4 elements. The spring mass system was modeled using concentrated mass elementCONM2 and spring element CELAS2. The spring mass system was coupled with the SMC plate by means of another spring and solved using Nastran. The FEA model so generated has been given in Fig. (4).

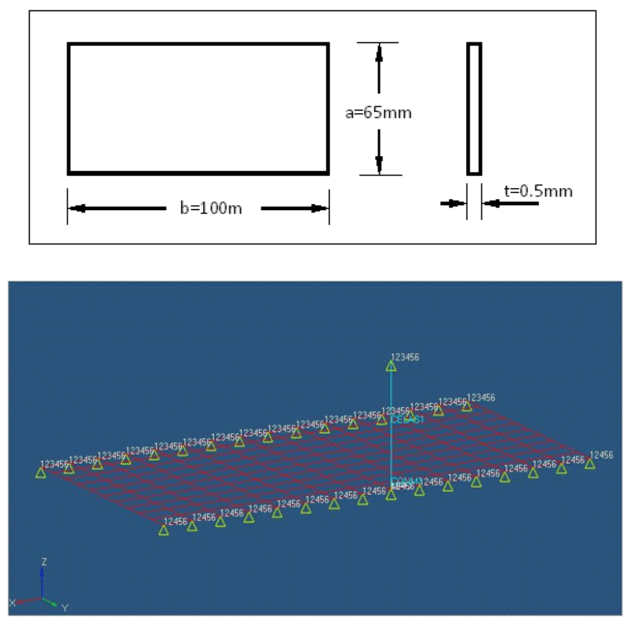

Fig. (4): Finite element model of ATVA

Frequency response analysis was carried on the spring mass system coupled with cantilever plate or the Actively tuned vibration absorber (ATVA) for fourelastic moduli obtained over the four regions of loading and unloading cycle of SMC.The results obtained were taken analyzed in post-processing and frequency response curves were generated using Hypergraph 9.0.the same are given in Fig. s (6), (7), (8) and (9).

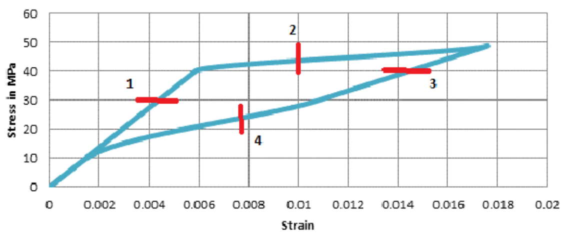

Fig. 5: Stress strain curve for SMC at 55^{0}C

5. RESULTS AND DISCUSSION

For studying the vibration absorption characteristics of SAC, the FEA model of ATVA was run on FEA solver NASTRAN. From the results of the FEA analysis, frequency response curves are plotted. The activation of SAC when subjected to external stimulus (like thermal, electrical etc), changes the elastic properties of the SAC and because of this the vibration absorptivity of SAC will also change accordingly. Which means the response of the overall system should change with change in elastic properties of SAC facilitated due to activation of SAC.

The stress strain curve for SAC Fig. .(5) reveals that the complete strain recovery occurs only above 40^{o}C, below which there is a residual strain in the SAC. Hence to demonstrate the change of System response with the mechanical loading, four sets of elastic properties were taken from loading and unloading cycle at 55^0 C. The frequency response curves so generated are presented in the Fig. s (6), (7), (8)and (9). For the 1st set of properties the maximum response occurs between 650 to 700 Hz, for the 2nd set of properties the maximum response occurs between 550 to 600 Hz, for the 3rd set of properties the maximum response occurs between 500 to 550 Hz and for the 4th set of properties the maximum response occurs between 550 to 600 Hz. From these curves it is evident that the resonant frequency changes with change in the properties of SAC. The change in resonant frequency of the system is the benchmark indication of the fact that the activation of the SAC is changing the response of the system. By suitably controlling the activation of the SAC, vibration absorption can be achieved over a range of resonant frequencies a vibrating system would likely undergo.

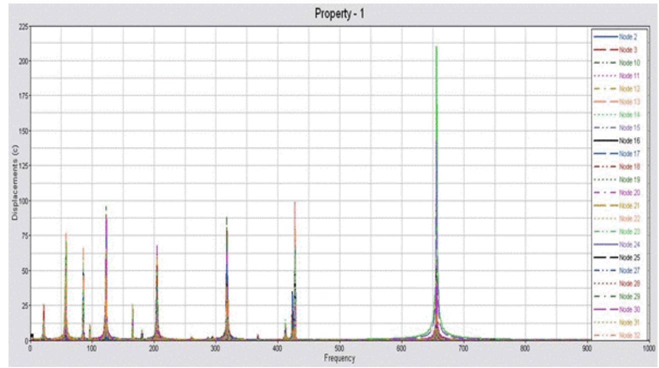

Fig. (6): Frequency response curve for property set - 1

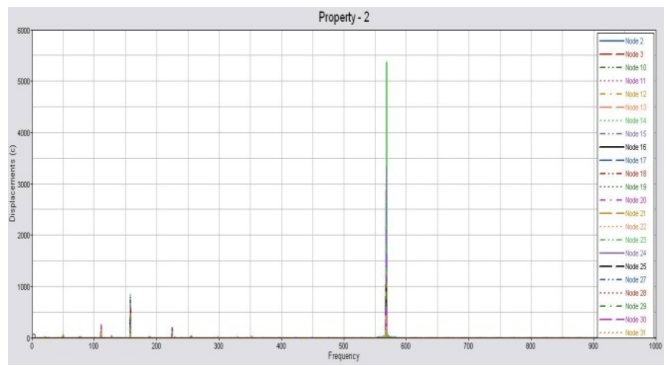

Fig. (7): Frequency response curve for property set - 2

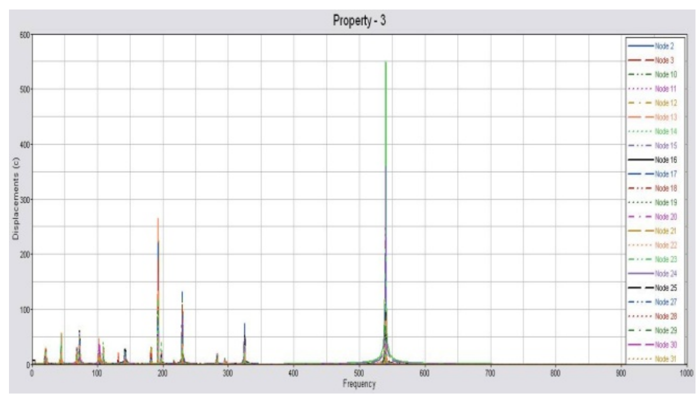

Fig. (8): Frequency response curve for property set – 3

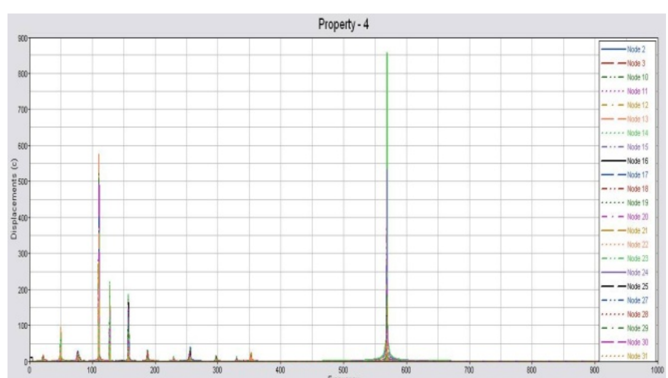

Fig. (9): Frequency response curve for property set – 4

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