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A semi-active smart tuned mass damper for drive shaft

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Key Word: Tuned mass damper, Drive shaft, Natural frequency, Shape memory alloy.

ABSTRACT

Tuned mass damper is widely used in many applications of industry. The main advantage of tuned mass damper is that it can increase the damping ratio of system and reduce the vibration amplitude. Meanwhile, the natural frequency of system will be divided by two peaks, and the peak speeds are closely related to the mass and the stiffness of auxiliary mass system added. In addition, the damping ratio will also affect the peak frequency of the dynamic response. In the present research, the nonlinear mechanical characteristics of rubber is investigated and put into use, since it is usually manufactured as the spring element of tuned mass damper. By the sense of the nonlinear stiffness as well as the damping ratio which can be changed by preload applied on, the shape memory alloy is proposed to control the auxiliary mass system by self-optimizing. Supported by the experiment data of rubber, the 1 DOF theoretical model and finite element model based on computer simulation are implemented to perform the feasibility of the proposed semi-active tuned mass damper working on the drive shaft.

1. INTRODUCTION

Tuned mass damper (TMD) is an attractive device for both the mechanical engineering and civil engineering. It is an auxiliary mass-spring-damping device attached to the primary system. The performance of the auxiliary part can reduce the dynamic response as well as the natural frequency of the primary system. Frahm [1] proposed the TMD system in 1909, and found that when the natural frequency of auxiliary system is very similar to the dominant mode of the primary system, the vibration response of the primary system will decrease a large amount. It is the basic concept for designing TMD.

To get the suitable mass, stiffness and damping, the optimal design method is usually used. In 1940, Den Hartog [2] found the analytical method to obtain the optimal tuning ratio and damping ratio of a single degree of freedom (SDOF) TMD connected to an undamped SDOF system. From then on, many researches are conducted to optimal design TMDs for various systems, such as two-degree-of-freedom system [3], certain mode of multi-degree-of-freedom systems [4, 5]. However the effectiveness of a Passive TMD is limited in a narrow frequency band. To change some parameters of the TMD structures,

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such as the natural frequency or damping, the variable TMD is developed and is called semi-active tuned mass damper (SATMD). As the realization of smart materials has been more and more, the SATMD is well developed. A classical type of SATMD is a damping variation control system like a magnetorheological (MR) or electrorheological (ER) fluid damper [6, 7]. The other type of SATMD is dry-friction damper, and the usage of piezoelectric actuators has made this kind of SATMD easily adjustable and quick responding [8].

In industry of automotive, TMD are widely used on the transmission shaft to avoid resonance and decrease the vibration. The TMD used on the shaft are usually a block of mass packaged by the rubber. It is manufactured by injecting the ribbon-shaped rubber into the model. Then the development of this kind of TMD is to improve the structure and compatibility [9, 10]. In the present work, an SATMD is designed to shift the resonance frequency of the propeller shaft actively. The nonlinear property of the rubber is used to generate the variable stiffness and damping ratio for the auxiliary system. The shape memory alloy (SMA) spring actuators are used to control the rubber. In this paper, theoretical analysis for the SATMD will be explained, and the structure of the SATMD will be introduced. The finite element analysis will be operated to verify the feasibility.

2. SMART TUNED MASS DAMPER

The new type of SATMD proposed in this paper mainly uses the specific characteristic of shape memory effect. The shape memory effect was discovered in 1951 in Au-Cd system [11]. Then it was developed to TiNi alloy and was widely used in the industry and civil engineering as actuators or dampers. The shape memory alloy (SMA) has many advanced mechanical properties, as shape memory effect, pseudo-plasticity and Pseudo-elasticity [12]. The shape memory effect is the SMA can recover its original shape when the temperature is increased above the transformation temperature. Based on this effect, it is well know that if a pre-strained shape memory alloy (SMA) is constrained when the Martensite is transformed to Austenite, a large recovery stress is generated [13]. This property makes the SMA used as the actuators, fasteners, connectors, seals and clamps [14].

The available structure designed for the SATMD is shown in Fig.1. For the proposal of change and recover the stiffness of rubber, the SMA actuator will be used to apply on and off the preload. As shown in Fig.1, the SATMD is composed by three components. One is a tube shape rubber, and the other two are half- cylindrical mass. The two mass are fixed by the bolts. While between the bolt and the mass, there are two SMA springs which are totally compressed at the beginning. Then between the two mass, there is one steel spring, which is acting as the recover spring. As the spring constant of the steel spring and the two phases SMA spring has the following relation: $K_a > K_s > K_m$, the actuator can well perform the force on and off.

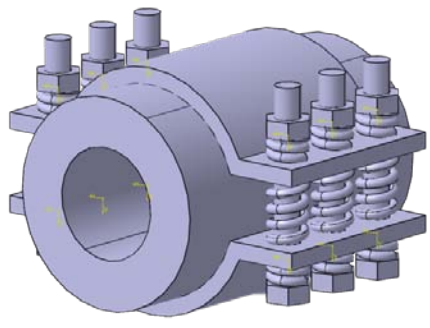

Fig.1 The sketch of the proposed SATMD.

3. THEORETICAL ANALYSIS

The proposed SATMD attached on the propeller shaft can be seemed as a SDOF system as shown in Fig.2.

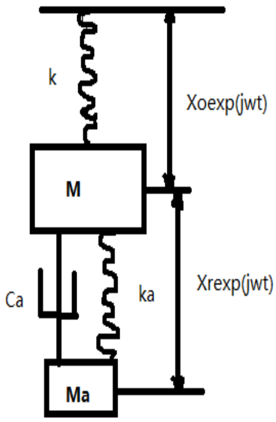

Fig.2 The sketch for a SDOF system

In the analysis, the shaft is seemed as a no damping system, while the TMD is seemed as the auxiliary system Ma. Assumed the displacement of the two mass as $x_0 e^{j\omega t}$ and $x_r e^{j\omega t}$, and harmonic exiting force is applied on the propeller shaft, the dynamic function for the whole system is shown as following:

$$\begin{bmatrix} M & 0 \\ 0 & m_a \end{bmatrix} \begin{Bmatrix} x_0 \omega^2 e^{j\omega t} \\ x_r \omega^2 e^{j\omega t} \end{Bmatrix} + \begin{bmatrix} c_a & -c_a \\ -c_a & c_a \end{bmatrix} \begin{Bmatrix} x_0 \omega e^{j\omega t} \\ x_r \omega e^{j\omega t} \end{Bmatrix} + \begin{bmatrix} k+k_a & -k \\ -k & k \end{bmatrix} \begin{Bmatrix} x_0 e^{j\omega t} \\ x_r e^{j\omega t} \end{Bmatrix} = \begin{Bmatrix} F e^{j\omega t} \\ 0 \end{Bmatrix}$$

Then the spring and damping forces acting on m_a can be shown the following motion equation:

$$(-ka \cdot x_r - c_a \cdot j\omega \cdot x_r) \cdot e^{j\omega t} = -m_a (x_0 + x_r) \omega^2 e^{j\omega t}$$

The exiting force can also be expressed as following:

$$F = \frac{(k_a + jc_a \omega) m_a \omega^2}{-m_a \omega^2 + jc_a \omega + k_a} x_0$$

Connected by the exiting force, the auxiliary mass system can be exerted by an equivalent mass m_{eq} rigidly attached to the moving foundation:

$$m_{eq} = \frac{1 + 2\zeta \frac{\omega}{\omega_a} j}{(1 - \left(\frac{\omega}{\omega_a}\right)^2) + 2\zeta \frac{\omega}{\omega_a} j} m_a$$

Where $\omega_a = \frac{k_a}{m_a}$ is the natural frequency of the auxiliary system. A damping parameter $\zeta = \frac{c_a}{c_{ca}}$ is the non-dimensional parameter with $c_{ca} = 2\sqrt{k_a m_a}$ is critical damping of the auxiliary system. Then if there is no auxiliary system, the motion equation of the forced vibration SDOF can be expressed as:

$$-m\omega^2 x_0 e^{j\omega t} + kx_0 e^{j\omega t} = Fe^{j\omega t}$$

The effect of the auxiliary mass system is to increase the mass m of the primary system by the equivalent mass of the auxiliary system [15], so the vibration amplitude of the whole system, compared with the static deflection is shown as following:

$$\frac{x_0}{\delta_{st}} = \frac{\left(1 - \left(\frac{\omega}{\omega_a}\right)^2\right) + 2\zeta \frac{\omega}{\omega_a} j}{\left(1 - \left(\frac{\omega}{\omega_a}\right)^2\right) + 2\zeta \frac{\omega}{\omega_a} j - \left(\frac{\omega}{\omega_0}\right)^2 \left[\left(1 - \left(\frac{\omega}{\omega_a}\right)^2\right) + 2\zeta \frac{\omega}{\omega_a} j + \mu\left(1 + 2\zeta \frac{\omega}{\omega_a} j\right)\right]}$$

where the mass ratio $\mu = m_a / m$. The natural frequency of the auxiliary system ω_a can be changed with the stiffness of the rubber changed. From the research of Yan [13], the recover stress of the SMA can be in the range of 300 Mpa to 400 Mpa according to the extent of pre-strained. In the study of the nonlinear stiffness of the rubber element, Rivin et. Al show the load-deflection characteristic of a rubber, and the according to the strain is change from 0.1 to 0.2, the stiffness can change 15% [16].

For the proposed rotation propeller shaft, the natural frequency can be obtained from the beam theory. The theoretical natural frequency is 1381.6

rad/s from the simplified shaft model. It can be obtained that as the mass of the auxiliary system increase, the differences between peak speeds of the resonance frequency of the composite system will increase. However, as the natural frequency of the auxiliary system increase, the low resonance frequency of the composite will be closer to the original one. Considering the damping ratio of the system, when the rubber is assumed to have the damping ratio of 0.025, the dynamic vibration response on the shaft can be obtained as shown in Fig.3. The SATMD which is similar to the self-optimization support system is proposed. When the rotation is about to suffer its resonance, the SMA will be active, and the resonance frequency of the composite will be changed. Thereby the strong vibration at resonance frequency will be avoided.

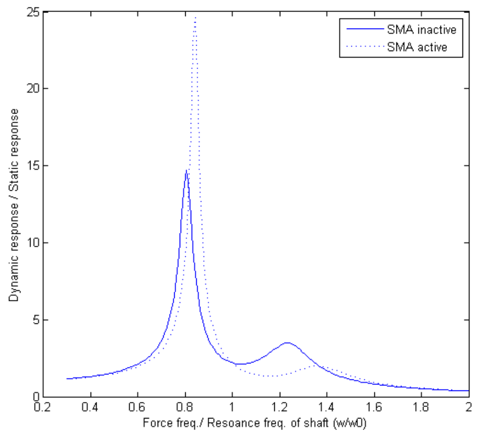

Fig.3 Dynamic response of the composite system.

4. NUMERICAL ANANLYSIS

The numerical analysis is conducted by the finite analysis software ABAQUS. The simplified model of the composite system is mesh in the preprocessing software HYPERMESH, and the shaft is divided into 47676 elements with the type of C3D8I. Meanwhile the rubber and the mass are divided into 8976 elements of C3D8I and 12256 elements of C3D8I respectively. The two mass are combined with 6 truss 1-dimension elements. The boundary condition is set as the two ends of the shaft are fixed.

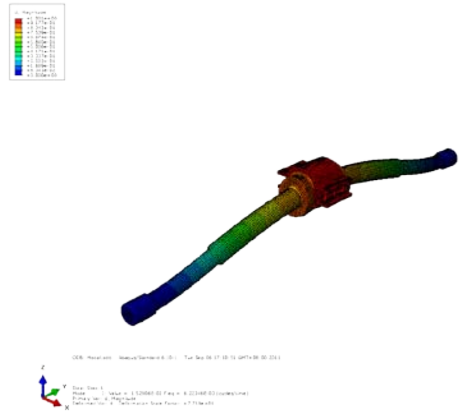

Fig.4 The bending mode for the composite system.

The eignvalue of the system will be analyzed first for detecting the natural frequency of the composite system and the vibration mode. As shown in Fig.4, the bending mode for the composite system have the natural frequency of 187.6 Hz. Compared with the natural frequency of the shaft which is 224.03 Hz, the first resonance frequency of the composite system is decreased.

Next the simulation of the dynamic vibration response is conducted. The exiting force is applied beside the SATMD on the shaft, and has the amplitude of 1000N. Then the response is checked at the center part of the shaft. In the present work, the SMA recover force is instead by the preload inside the truss element and the nonlinear stiffness data of rubber from experiments in research [16] has been input into the material properties. The damping ratio of the rubber is chosen to be 0.025 from the experience. The results of the dynamic vibration analysis are shown in Fig.5. It is obvious that the SATMD has damped the amplitude of the vibration, and change the resonance frequency. When the SMA is active, which means the preload inside the truss is working. It is clear that the resonance frequency of the system is changed, from186 Hz to 195 Hz. With the same amplitude of exiting force, when the resonance frequency has changed, the vibration response at the original point will be decreased. It is well coincide with the theoretical analysis. Thereby, when the rotation speed is close to one of these resonance frequencies, with the effect of the SATMD, the vibration can be well damped.

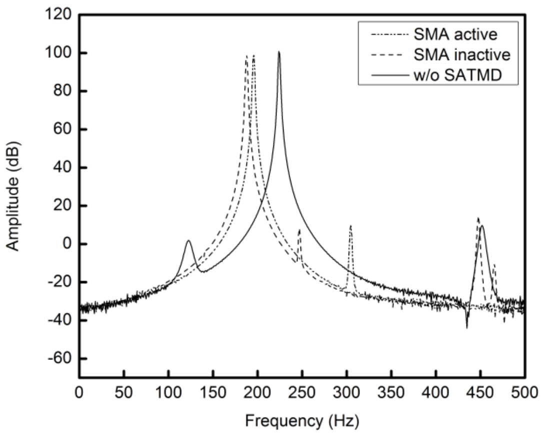

Fig.5 The dynamic response of the shaft.

5. CONCLUSION

In the present work, the concept of a new SATMD is introduced. From the analysis of a SDOF system, according to the stiffness of the auxiliary system will be changed, the resonance frequency of the whole composite system will be changed. Then with the same damping effect, the dynamic response of the system will be decreased when the resonance frequency has changed from one to another. The prototype of the SATMD is introduced, with the assembled effect of the SMA and steel springs. At last the finite element analysis is conducted and the results are quite following the theoretical prediction. Meanwhile, the numerical results will be referred for the real experiment. In the future, the exact SMA spring and steel spring structure can be modeled into the finite element analysis, and the preload can be changed to the pre-heat to get a more accurate simulation. The experiments of this new device will also be conducted in the lab and on the real vehicle.

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