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Active-adaptive Vibration Absorbers and Its Vibration Attenuation Performance

Chao Peng, Xinglong Gong

CAS Key Laboratory of Mechanical Behavior and Design of Materials,
Department of Modern Mechanics, University of Science and Technology of China, Hefei 230027, China

pchao@mail.ustc.edu.cn, gongxl@ustc.edu.cn

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Abstract. To improve the working frequency band and the damping effect of vibration absorber, an active-adaptive vibration absorber (AAVA) was presented. The AAVA can be considered as the integration of adaptive tuned vibration absorber (ATVA) and active vibration absorbers (AVA). The principle and the dynamic character of the proposed AAVA were theoretically analyzed. Based on the analysis, a prototype was designed and manufactured. Its dynamic properties and vibration attenuation performances were experimentally investigated. The experimental results demonstrated that the damping ratio of the prototype was significantly reduced by the active force. Consequently, its vibration attenuation capability was significantly improved compared with the ATVA.

Introduction

The tuned vibration absorbers (TVAs) are widely used in industries to suppress undesired vibrations of some machines excited by harmonic forces. However, the conventional TVA is a single-frequency vibration control device and it only works in a narrow frequency range. If the exciting frequency is in a wide range, the vibration attenuation effect of the TVA often decreases or even collapses because of the mistuning. This problem is the major limitation of the TVAs in many practical applications. One of the solutions is to develop the devices which are able to attenuate the vibration over a wider frequency range. Currently, the competent devices include the AVA [1, 2] and ATVA [3-5].

The AVA can be considered as a TVA with an active element attached. By controlling the active force, the AVA exhibited better vibration attenuation performances than TVA. The active resonator absorbers [1] and the delayed resonators [2] are two major types of AVAs, both of which were demonstrated to be equally effective in suppressing undesired oscillations. The AVA can achieve good vibration attenuation effect, but it still needs large active forces. The ATVA improves the vibration attenuation performance by adjusting its resonant frequency on-line to track the excitation frequency. Without an active force, it consumes less power compared with the AVA. Moreover, the ATVA is a fail-safe device as it can work as a TVA in case of loss of power. The heart of the ATVA is a stiffness element whose stiffness can be adjusted in real time. A variety of methods have been proposed to vary element's stiffness, such as variable stiffness element through mechanical mechanisms [3], variable magnetic spring controlled by current [4] or using controllable new materials [5]. For the ATVA, the damping of the ATVA has greatly influenced the vibration attenuation performance, as the smaller the damping is, the higher the vibration attenuation performance can be achieved. However, most of the materials used in ATVA have a stable damping and it is very difficult to decrease it. To solve this problem, much attention has been concentrated on the combination of ATVA and AVA. Kidner and Brennan [6] designed an active vibration neutralizer to control stiffness and damping in real time. It used the piezoelectric elements as the active element to control the damping. The experimental results showed that significant improvements in the vibration attenuation of a host structure were achieved compared with a passive one. An adaptive active resonator absorber based on the relative velocity feedback was proposed in theory [7]. The theoretical analysis showed that with smaller control effort the device can obtain the same vibration attenuation effect with the ARA.

The aim of this research is to improve working frequency band and vibration attenuation performance of a vibration absorber. Towards this end, a mechanical AAVA was designed. This paper is split into four sections. Following the introduction section, the principle and the vibration attenuation performance of the AAVA are analyzed in Section 2. Section 3 describes the prototype of the AAVA and its dynamic property testing. Section 4 presents the experiments of the vibration attenuation effect of AAVA based on a clamped-clamped beam. The conclusions are summarized in the final section.

Analysis on AAVA

The Principle of the AAVA

To illustrate the working principle, a single degree of freedom primary system with the AAVA is described in Fig. 1. The active element is placed between the absorber mass and the primary system. The m_p, c_p, k_p are the mass, damping and stiffness of the primary system, respectively; The m_a, c_a, k_a are the mass, damping and stiffness of the AAVA, respectively; The x_p, x_a are the displacements of the primary system and the absorber mass, respectively; The f_{act} is the active force; The f is the harmonic excitation force applied on the primary system.
According to the references [7], different values of the parameters in Fig.1 represent different types of vibration absorbers, as shown in Table 1.

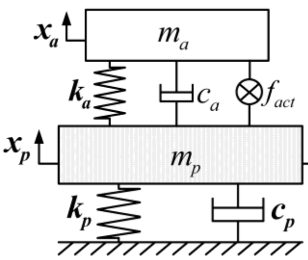

Fig.1 Schematic view of the primary system with a tuned vibration absorber

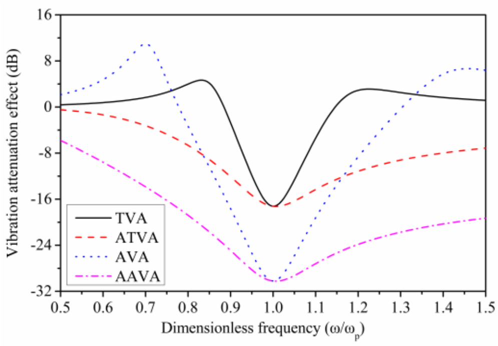

Fig.2 Vibration attenuation effect of different types of vibration absorbers

Tab.1 Several typical vibration absorbers

TYPE	k_a/m_a	f_{act}
TVA	constant	0
ATVA	ω^2	0
AVA	constant	$\eta[(k_a - m_a\omega^2)x_a + c_a\dot{x}_a]$
AAVA	ω^2	$\lambda c_a \dot{x}_a$

The ω is the frequency of excitation force; The η and λ are control efficient of AVA and AAVA, respectively. According to Newton's law, the equations of motion can be expressed as follows:

$$\begin{cases} m_a\ddot{x}_a + c_a(\dot{x}_a - \dot{x}_p) + k(x_a - x_p) = f_{act} \\ m_p\ddot{x}_p + c_p\dot{x}_p + c_a(\dot{x}_p - \dot{x}_a) + k_p x_p + k(x_p - x_a) = f - f_{act} \end{cases} \tag{1}$$

From Eq. (1), the amplitude magnification coefficient of the primary system with different type vibration absorbers attached can be obtained as follows:

$$X_p = \frac{(Z_m + Z_k)F - Z_m F_{act}}{Z_m Z_p + Z_m Z_k + Z_p Z_k} \tag{2}$$

Where,

$$\begin{cases} Z_m = -m_a\omega^2 \\ Z_k = 2jm_a\xi_a\omega_a\omega + m_a\omega_a^2 \\ Z_p = -m_p\omega^2 + 2jm_p\xi_p\omega_p\omega + m_p\omega_p^2 \end{cases} \tag{3}$$

Where $\omega_p = \sqrt{k_p/m_p}$, $\xi_p = c_p/(2m_p\omega_p)$, $\omega_a = \sqrt{k_a/m_a}$, $\xi_a = c_a/(2m_a\omega_a)$; The ω_p and ξ_p are the natural frequency and damping ratio of the primary system, respectively. The ω_a and ξ_a are the natural frequency and damping ratio of vibration absorber, respectively. For the AAVA and ATVA, ω_a tracks the frequency of excitation force. For the TVA and AVA, ω_a is fixed at the natural frequency of the primary system.

Comparison of vibration reduction performance

The vibration attenuation effect of the vibration absorbers is defined as the ratio of the amplitude magnification coefficient of the primary system with and without vibration absorber, the dB expression of the effect is:

$$\gamma = 20\lg\frac{X_{p\_with}}{X_{p\_without}} \tag{4}$$

Where, the X_{p_with} and $X_{p\_without}$ are the amplitude magnification coefficient of the primary system with and without vibration, respectively. According to the following parameters, $m_a/m_p = 0.1$， $\eta = \lambda = 0.8, \xi_a = 0.04, \xi_p = 0.08$, the effect of vibration absorption capacity can be calculated.

The comparison of four types of vibration absorbers with the same mass ratio and damping ratio is shown in Fig.2. For the TVA, the best vibration attenuation effect occurs at its natural frequency. When the excitation frequency is away from this frequency, the effect decreases quickly. The AVA has better effect than TVA close-by the natural frequency, but the effect also becomes bad. For ATVA, its vibration attenuation effect is equal to that of the TVA at its natural frequency, and it has better effect than TVA in the whole adjustable frequency band. Nevertheless, the superiority of the ATVA compared with the TVA is not evident because of the large damping. At the whole frequency range, the curve of AAVA's attenuation effect is below that of the other types of vibration absorbers, which indicates that the AAVA has the best vibration attenuation effect.

The design and dynamic property of AAVA

The prototype of AAVA

According to previous analysis, the AAVA has the best vibration attenuation effect compared with other types of vibration absorbers. It means that the AAVA is more suitable for engineering application. In this paper, a mechanical AAVA is designed. Fig. 3(a) shows a scheme of the prototype and Fig. 3(b) shows a photograph of the prototype.

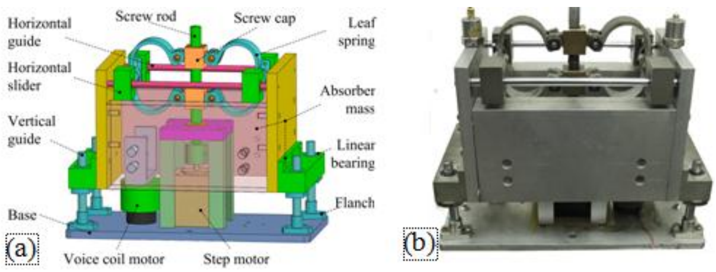

Fig. 3 The prototype of AAVA. (a) Scheme; (b) Photograph.

The absorber mass of the prototype is a closed configuration which is composed of four individual masses. There are two horizontal guides mounted on the absorber mass. Four horizontal sliders can move in horizontal direction along the two horizontal guides. This configuration can make the best use of the space and reduce the size of the AAVA. The leaf spring is chosen as the spring pole because of the large bear load and lateral rigidity. The stiffness element of the AAVA is composed of four leaf springs. The span between leaf springs can be changed, which causes the natural frequency of the AAVA changed. If the span keeps at a fixed value, the AAVA can be regarded as ATV. Four vertical guides, mounted on the base, are used to make the absorber mass to move only in vibration direction. The mass of the absorber mass is about 4 kilograms, and the other mass is about 1 kilogram.

The active force is provided by a small voice coil motor. The motor consists of a magnet and motor coil. The magnet is fixed to the absorber mass, and the motor coil is fixed to the base. When a current flows through the motor coil, an ampere force will be generated between the magnet and the motor coil. The amplitude and direction of the force can be controlled by the current. If the voice coil motor stop working, the AAVA can be regarded as a translational ATVA.

Dynamic characteristics of the prototype

Using the approaches have been proposed in references [8, 9], the dynamic characteristics of the prototype can be obtained. Fig. 4 shows the frequency-shift property curves and damping property of the prototype.

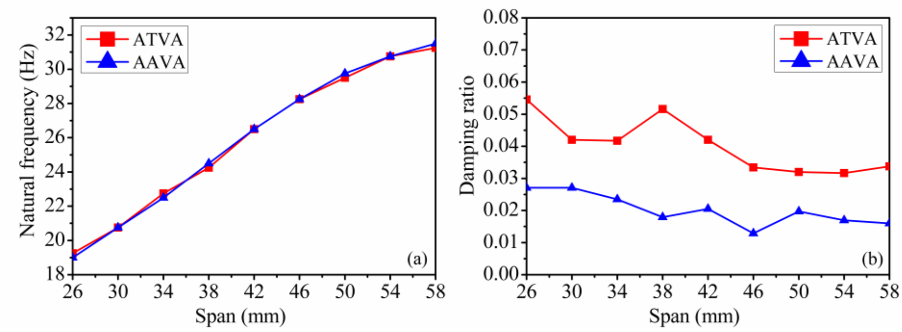

Fig.4 The experimental results of dynamic property.
(a) The frequency-shift property (b) The damping property

As described in this figure, for either the ATVA or AAVA, the natural frequency of prototype is from 19.25Hz to 31.25Hz when the span changes from 26mm to 58mm. The damping ratios are obtained by using the half power bandwidth method. It is shown that the average damping ratio of ATVA are 0.04, and this value is 0.02 for AAVA. As the AAVA introduces the active force, it can reduce the damping.

Experimental evaluation of the vibration attenuation performance of the AAVA

To evaluate the vibration attenuation effect of the prototype of AAVA, a clamped-clamped beam with a pullback weight attached was used as the primary system. Fig.5 shows the schematic of the experimental set-up for evaluation of the vibration attenuation performance of the AAVA. The first mode of the beam is bending vibration and the natural frequency of mode is about 26Hz. The AAVA was mainly used to control this mode vibration, so it was placed at the center of beam where the vibration is the largest.

As shown in Fig.5, an impedance head connects with the beam and the exciter, and it measures the force signal and the acceleration signal which is used as the input of the controller. The ratio of the force signal and the acceleration signal is the acceleration admittance of the beam. With an absorber attached, the dynamic property of the beam changes and hence the admittance of the beam changes. Therefore, the vibration attenuation effect of an absorber can be represented by comparing the admittance of the beam with and without absorber.

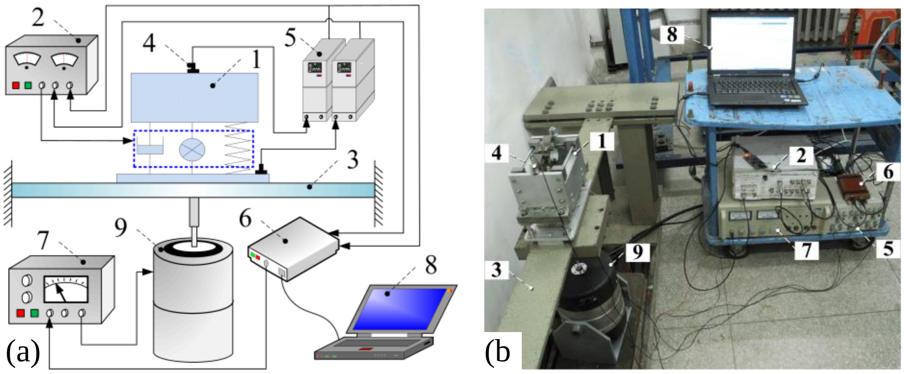

Fig.5 Experimental set-up. (a)Scheme; (b) Photograph.
1. Prototype; 2. Control box; 3.Clamped-clamped beam; 4. Accelerometer;
5.Charger amplifier; 6. Dynamic signal analyzer; 7. Power amplifier; 8. Computer; 9.Exciter;

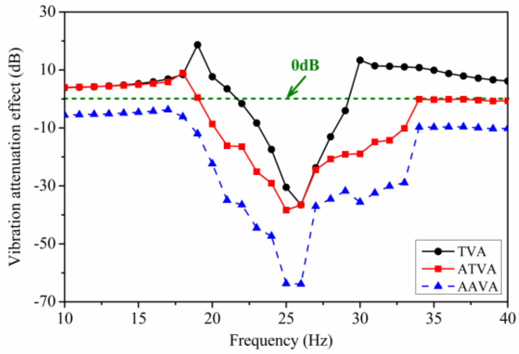

Fig.6 Experimental results of the vibration attenuation effect

Fig.6 shows the experimental results of the vibration attenuation effect, If the prototype is not controlled and its resonant frequency is fixed at 26Hz (the resonant frequency of the beam), it can be regarded as a TVA. While the prototype is only controlled to trace the excitation force frequency, it can be looked as an ATVA. As shown in Fig.6, for TVA, the best vibration attenuation effect occurs at its natural frequency. The effect goes down when the excitation frequency is far from this frequency. At some frequencies, the values of the effects are even larger than 0, indicating that the vibration of the system is increased by the TVA. For ATVA, whose natural frequency is tuned to trace the excitation frequency, its vibration attenuation effect is better than that of the TVA within the entire adjustable frequency band except at 26 Hz. Nevertheless, the superiority of the ATVA compared with the TVA is not evident because of the large damping. The AAVA has the best vibration attenuation effect. It can result in a reduction of vibration of the beam by up to average 15 dB, which is better than that achieved with the ATVA.

Conclusions

In this work, we developed a mechanical AAVA, whose natural frequency can be tuned from 19.25 Hz to 31.25 Hz by adjusting the span between the ends of two spring poles from 26cm to 58cm. The damping of the prototype is rather small, and the average damping ratio was reduced to 0.02 from 0.04 by the active force. Experimental studies were conducted to evaluate the vibration attenuation performance of the AAVA on a clamped-clamped beam. The results show that the AAVA has better vibration attenuation performance than ATVA and TVA.

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