A control strategy for adaptive absorber based on variable mass

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Abstract. The tuned vibration absorber (TVA) has been an effective tool for vibration control. However, the application of TVA can cause resonance of the primary system and increase its vibration when the absorber is mistuned. In this paper, a novel control strategy based on adaptive tuned vibration absorber (ATVA) of variable mass is proposed to reduce the resonance of the primary system. Unlike most ATVAs suggested by other researchers which adjust the absorber natural frequency by changing the stiffness, the variable mass ATVA varies its natural frequency by changing absorber mass to match the excitation frequency. Some simulations and experiments were conducted to test the performance of the control strategy. The results show that the proposed control plan can widen the frequency bandwidth of the absorber, as well as suppress the resonance of the primary system significantly. This implies that the work is useful for practical applications of ATVA.

1. Introduction

As an effective vibration control device, the tuned vibration absorber (TVA) has been used extensively in the vibration control fields since it was invented by Frahm in 1911 [1]. When the TVA is tuned to the excitation frequency of the primary system, it can suppress the host vibration greatly. However, the effective frequency bandwidth of the conventional TVA is very narrow. Once the excitation frequency drifts slightly, the TVA will lose effectiveness and probably increase the primary system vibration. In practical applications, the vibration source is complicated and its frequency often changes with time. This greatly limits the wide applications of the TVA. In order to deal with this problem, many new tuned vibration absorbers have been proposed in recent years, such as active tuned vibration absorber, and adaptive tuned vibration absorber.

The adaptive tuned vibration absorber (ATVA) can adjust its dynamic parameters (and then its natural frequency) in real-time to match the excitation frequency and significantly improve the performance of absorber in practical engineering applications. Many ATVA configurations have been studied these years. Franchek et al. designed an ATVA with variable stiffness by changing the number of active coils of a spring [2]. Walsh et al. used a pair of leaf spring as the variable stiffness element and adjusted the stiffness by changing the gap between the two beams [3]. Smart materials such as shape memory alloy (SMA), magnetorheological elastomer (MRE), have been applied on absorber design as well. Williams et al. have made a series of studies on SMA-based ATVA in which the
stiffness can be adjusted by changing the temperature of SMA [4]. Deng and Gong suggested an ATVA that used a MRE as the stiffness element. The stiffness of the absorber can be adjusted by controlling the magnet field applied on MRE [5]. To widen the effective frequency bandwidth of the absorber, the authors of this paper suggested the variable mass (VM) ATVA where a variable mass element is introduced in previous paper [6]. Different from other approaches, the natural frequency of the VM ATVA is adjusted by changing the absorber mass rather than the stiffness. Some simulation and experiments have been carried out to verify that the VM ATVA can suppress the vibration in a wider frequency range.

Besides the narrow effective bandwidth, it should be noticed that a new resonance peak is introduced into the frequency response of the primary system when the ATVA is attached to the primary system. Evidently, this will cause the serious vibration of the primary system when the excitation frequency is equal to this resonant frequency. However, to the best of authors’ knowledge, few studies have been reported to reduce this resonance. A novel control strategy based on the VM ATVA is proposed in this paper to prevent this big vibration of the primary system. Following the introduction, the traditional control strategy of the TVA is introduced briefly and then the novel control strategy is proposed in section 2. Section 3 presents the vibration attenuation effectiveness of the control strategy via some experiments. And then, the conclusions are drawn in the last part of this paper.

2. Novel control strategy of the VM ATVA

2.1. Traditional control strategy

Usually, a TVA consists of a mass, a spring and a damper. In the VM ATVA, a variable mass element, i.e., a liquid tank is adopted to change the absorber mass by adding or releasing some liquid from the tank, as shown in Figure 1, where \( m_1, k_1, c_1 \) are the mass, stiffness and damping of the primary system, respectively; \( k_2, c_2 \) are stiffness, damping of the ATVA, respectively; the absorber mass includes two parts: the constant mass \( m_2 \) and the variable mass \( m_v \), whose value ranges from 0 to \( m_{v_{\text{max}}} \). When the variable mass changes, the natural frequency of the ATVA can be computed following the equation:

\[
\omega_c = \sqrt{\frac{k_2}{m_2 + m_v}} \tag{1}
\]

It is obvious that the maximal and minimal natural frequencies of the ATVA, \( \omega_{v_{\text{max}}} \) and \( \omega_{v_{\text{min}}} \), occur while the liquid tank is empty and filled. Hence, the effective frequency band of the adaptive absorber ranges from \( \omega_{v_{\text{min}}} \) to \( \omega_{v_{\text{max}}} \) by changing the variable mass. When the excitation frequency \( \omega_e \) satisfies the equation of \( \omega_{v_{\text{min}}} \leq \omega_e \leq \omega_{v_{\text{max}}} \), the variable mass \( m_v \) can be computed by the next equation to keep the VM ATVA tuned:

![Figure 1. Model of VM ATVA attached to a primary system](image-url)
To obtain a wideband absorber, many researchers suggested a control strategy, which can be formulated as followings:

$$m_e = \frac{k_v}{\omega^2} - m_2$$

To obtain a wideband absorber, many researchers suggested a control strategy, which can be formulated as followings:

$$m_v = \begin{cases} m_{v_{\text{max}}} & 0 \leq \omega < \omega_m \\ m_e & \omega_m \leq \omega < \omega_n \\ 0 & \omega \geq \omega_n \end{cases}$$

where $\omega_m$ and $\omega_n$ are the minimal and maximal natural frequencies of the ATVA, i.e. the frequencies of the bottom point of the frequency responses of the primary system when the absorber obtains its minimal and maximal mass, as shown in Figure 2. Point P and Q are the peaks of these two frequency responses.

The dot dash curve (red) in Figure 2 illustrates the frequency response of the primary system while the control strategy above is applied. One can see that the primary system vibration is suppressed by 12.2 and 9.1 dB at points M and N with the frequencies of 7.1 and 10 rad/s, respectively. Moreover, when the variable mass is within the range of $0$ to $m_{v_{\text{max}}}$, a frequency bandwidth from $\omega_m$ to $\omega_n$ is obtained.

**Figure 2.** Frequency responses of the primary system with a VM ATVA attached. 
--- (green): liquid tank filled; — (blue): liquid tank empty; ·· (red): old control plan used.

2.2. *New control strategy to reduce the resonance*

As discussed in the previous section, the VM ATVA can widen the effective bandwidth of the absorber by using the old control strategy. However, there is still a resonance peak at point P in the frequency response of the primary system, as shown in Figure 2. That means the primary system vibration will increase while the excitation frequency is equal to $\omega_p$.

To suppress this big vibration, a new control strategy based on the VM ATVA is proposed, which can be expressed as follows:
where $\omega_s$ is the frequency of point S that is the intersection point of the frequency responses of the primary system while the variable mass obtains its maximal and minimal value.

**Figure 3.** Frequency responses of the primary system with a VM ATVA. --- (green): liquid tank filled; — (blue): liquid tank empty; -· (red): proposed control plan used

**Figure 4.** Vibration acceleration signal of the primary system. — (green): vibration curve at point P; -·· (blue): vibration curve at point S.
Different from the previous control strategy, the liquid tank keeps empty while the excitation frequency is less than $\omega_s$ when the new control plan is used. Figure 3 shows the frequency response of the primary system. It can be seen that the vibration of the primary system is attenuated by 12.2 and 9.1 dB at 7.1 and 10 Hz while the liquid tank is filled and empty, hence the bandwidth of the ATVA is from 7.1 to 10 Hz. When the excitation frequency lies within this range, the vibration of the primary system can be suppressed significantly. It is same as that when the old control plan is applied. However, the resonance at point P with amplification ratio of 16.3 dB is eliminated to approximate 2.1 dB, meanwhile a small lump about 2.3 dB at point S appears. This implies that the primary system vibration is suppressed about 14.0 dB when the new control plan is adopted. Figure 4 presents the vibration acceleration signal of the primary system while it is working at point P and S. It can be seen that the vibration of the host at point S is remarkably reduced and the amplitude is suppressed by 70.4% comparing with that at point P. Therefore, the proposed control plan can significantly reduces the resonance of the primary system caused by using the absorber, as well as widen the effective bandwidth of the ATVA.

3. Experiment tests

To verify the performance of the proposed control strategy, an experimental system was designed and some experiments, in which a glass bottle was used as variable mass element and water worked as the medium to change the absorber mass, were conducted. The frequency responses of the primary system when the glass bottle is filled and empty are shown in Figure 5. The frequency responses while the glass bottle is one-third and two-thirds water filled, and the excitation frequency varies from 10 Hz to 13.5 Hz are also conducted to obtain more information. From Figure 5 one can see that the vibration of the primary system is reduced by 27.0 and 29.0 dB at frequencies of 12.3 and 13.6 Hz while the glass bottle is filled and empty. Hence, by changing the water mass, the variable mass ATVA can effectively suppress the primary system vibration from 12.3 to 13.6 Hz.

![Figure 5](image_url)

**Figure 5.** Frequency responses of the primary system with a VM ATVA. ••• (blue): empty bottle; - - - (green): water-filled bottle; --×-- (red): with no VM ATVA mounted; 1: one-third water-filled bottle; and 2: two-thirds water-filled bottle.

In order to show the performance of the new control strategy more directly, a comparative figure of the old and new control strategies is illustrated in Figure 6. The previous resonant peak with the value of 18.1 dB at 10.8 Hz is suppressed to approximate 9.9 dB (at 11.0 Hz). This demonstrates that a resonance reduction of 8.2 dB is achieved by using the suggested control plan.
4. Conclusions
Although the ATVA can widen the absorber bandwidth by adjusting its natural frequency to match the excitation frequency, using the ATVA can cause the resonance of the primary system as well when it is mistuned. A new control strategy based on variable mass ATVA is proposed in this paper to suppress the resonance of the primary system. Some simulations and experiments were conducted to test the performance of the control plan. The results show that the suggested control plan can reduce the resonance of the primary system remarkably, meanwhile extend the effective frequency bandwidth of the absorber. Therefore, the performance of the ATVA can be promoted by using the proposed control strategy.

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Figure 6. Frequency responses of the primary system. — (blue): proposed control plan used; … (red): old control plan used.