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A novel kind of active resonator absorber and the simulation on its control effort

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Abstract

In this research, the advantages and limitations of an adaptive-passive vibration absorber (APVA) are analyzed in detail. Based on the analysis, a novel kind of active vibration absorber is proposed. It can be considered as the integration of APVA and active resonator absorber (ARA), so their advantages are inherited. Its control effort is theoretically analyzed. The results show that it needs much smaller control force than ARA.

1. Introduction

Dynamic vibration absorbers (DVA) have been successfully used to attenuate the vibration of many structures. The DVA usually consists of a mass attached to the structure to be controlled through a spring-damper system. It can be tuned to suppress vibration at single frequency harmonic excitation. The principal drawback of tonally tuned absorbers is that they require very low absorber damping to achieve good performance, which causes the effective bandwidth to be quite small. When the excitation frequency is unsteady or varying, traditional DVAs will become ineffective and potentially increase the base vibration.

In recent years, semi-active and active–passive vibration absorbers have been proposed to suppress harmonic excitations with time-varying frequency [1–13]. A semi-active vibration absorber achieves vibration control by changing its dynamic parameters, such as the stiffness or damping. Some advantages of semi-active control are that it requires less power, costs less, and has reduced complexity versus active systems, while being nearly as effective. Semi-active vibration absorbers can be separated into several types: variable stiffness through mechanical mechanisms [1,7,10,14,15] or using controllable new materials [5,9,12,20], variable inductor connected in series with the piezoelectric patch for piezoelectric absorbers [8,16–19]. A lot of adaptive vibration absorbers with variable stiffness have been proposed and verified experimentally and shown that these devices can effectively suppress the vibration of the primary structure. However, when the exciting frequency deviates from the resonant frequency of the primary structure, the effect of vibration absorber reduces obviously [7,8]. Furthermore, semi-active vibration absorber behaves as a passive one when its

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stiffness or damping is fixed, so it is not suitable for multi-frequency or broad-band excitation. In addition, these semi-active implementations have had difficulty in achieving fast and accurate tuning [16]. Active–passive vibration absorber is the integration of a passive vibration absorber (PVA) with a force actuator. Active control allows for the direct control of the absorber's transmitted force as well as modifying the dynamic properties of the device. This is achieved using the force actuator that essentially modifies the effective stiffness and damping of the absorber. As a result of the direct control of the absorber's transmitted force, the absorber can suppress the vibration of multi-frequency or broad-band excitation and tune rapidly with high accuracy. The active absorbers with different actuators and control laws have been investigated and implemented [2–4,6,11,13]. It has advantages such as large bandwidth, high vibration reduction level and fine robustness. But it needs large power requirement.

As active–passive absorber, the delayed resonator and active resonator absorber (ARA) have been investigated by many researchers [3,6,11]. Both of them consist of a proportional position or acceleration feedback and the only difference between them lies in that the feedback of the delayed resonator has a time delay. The reason they are called as resonator is that resonance is created in these absorbers by using the feedback control to place the dominant characteristic roots of absorber subsection on imaginary axis. In fact, these devices are similar to auto-tuning un-damping vibration absorbers, whereas the performance is realized only by the active force. Although they are effective for wide band frequency excitation, they still need large control effort. In this paper, a novel implementation of tunable vibration absorber is proposed. It is called an adaptive active resonator absorber (AARA). The AARA consists of two parts. The first part is an adaptive-passive vibration absorber (APVA) with variable stiffness, which can be adaptively tuned to the correct frequency. The second part is an actuator which provides control force to cancel the damping force applied on the absorber, hence leading to resonance. Once the AARA becomes resonant, it create the perfect vibration absorption at given frequency. The concept of the AARA is similar to the adaptive active–passive piezoelectric absorber [16–19], but the proposed absorber is spring-mass system which has quite different properties such as its configuration, system equation and applied field. The adaptive active–passive piezoelectric absorber is limited to vibration control of structure with elastic deformation, whereas the proposed absorber has no limitation and can also be applied to vibration reduction of rigid body or structure with no elastic deformation. In Section 2, the advantages and limitations of an APVA are analyzed in detail. In Section 3, the principle of the AARA is introduced. The control effort of AARA and ARA are theoretically analyzed in Section 4.

2. Adaptive-passive vibration absorber

Different APVAs are distinguished mainly by variable stiffness element and control law. However, all APVAs can be described as the same model shown in Fig. 1. In this paper, the performance of a general APVA

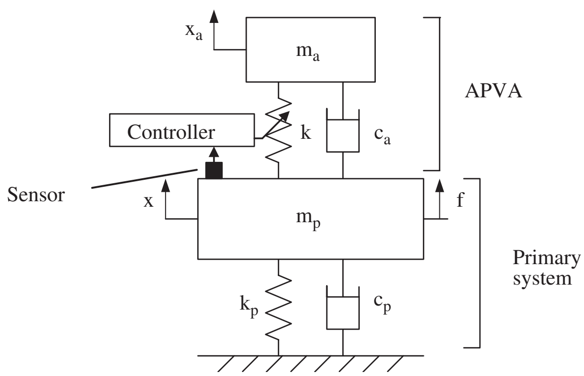


Fig. 1. A single-d.o.f. primary system with an APVA.

is discussed and the details of the variable stiffness are not involved. A single-degree-of-freedom primary system with a APVA is depicted in Fig. 1, where x, m_p, c_p k_p, are the displacement, mass, damping and stiffness of the primary system, respectively; x_a, m_a, c_a, are the displacement, mass and damping of the APVA, respectively; k is the variable stiffness of the APVA and its initial value or value without control are called k_a; f is the excitation force applied on the primary system. The principle of APVA is that according to the vibration signal of the primary system measured by the sensor, the controller analyzes the excitation frequency and tunes the APVA to this frequency. The performances and limitations of the APVA are investigated as follows.

When the APVA is removed, the driving point mobility of the primary system can be written as

$$H_1 = \frac{x}{f} = \frac{1}{-m_p\omega^2 + \mathrm{j}c_p\omega + k_p}, \tag{1}$$

where ω is the frequency of the harmonic excitation force f.

The driving point mobility of the primary system with an APVA is

$$H_2 = \frac{x}{f} = 1\Big/\left(\frac{1}{-1/(m_a\omega^2) + 1/(\mathrm{j}c_a\omega + k)} - m_p\omega^2 + \mathrm{j}c_p\omega + k_p\right), \tag{2}$$

where k is a variable and can be tuned in real time so that the natural frequency of AVPA coincides with the exciting frequency.

Nagarajaiah and Varadarajan developed a frequency tuning algorithm based on short time Fourier transform (STFT) to retune the natural frequency of a semi-active variable stiffness tuned mass damper in real time [14]. The basic idea of STFT is to break up the signal into small time segments and Fourier-analyze each time segment to ascertain the frequencies that exist in it. Similar to work in Ref. [14], STFT can be used to identify the dominant frequency of response measured by the sensor. For the response under harmonic excitation, the dominant frequency is equal to the exciting frequency ω. It is assumed that the excitation frequency is changing relatively slowly compared to the system dynamics (quasi-steady state). So the natural frequency of the APVA can be tuned to equal the exciting frequency ω exactly. Namely, the adaptive control law of APVA is

$$k = m_a\omega^2. \tag{2a}$$

The vibration reduction coefficient β is defined as the ratio of the driving point mobility of the primary system without and with the APVA, i.e.

$$\beta = \frac{H_1}{H_2} = \frac{\dfrac{-m_a\omega^2(\mathrm{j}c_a\omega + k)}{-m_a\omega^2 + \mathrm{j}c_a\omega + k} - m_p\omega^2 + \mathrm{j}c_p\omega + k_p}{-m_p\omega^2 + \mathrm{j}c_p\omega + k_p} = \frac{\dfrac{-m_a\omega^2(2\mathrm{j}\xi_a\omega_a\omega + k/m_a)}{-\omega^2 + 2\mathrm{j}\xi_a\omega_a\omega + k/m_a} + m_p(-\omega^2 + 2\mathrm{j}\xi_p\omega_p\omega + \omega_p^2)}{m_p(-\omega^2 + 2\mathrm{j}\xi_p\omega_p\omega + \omega_p^2)}, \tag{3}$$

where $\omega_a^2 = k_a/m_a$, $\omega_p^2 = k_p/m_p$, $c_a/m_a = 2\omega_a\xi_a$, $c_p/m_p = 2\omega_p\xi_p$. ω_a, ω_p are, respectively, the natural frequency of the APVA without control and the primary system and ξ_a, ξ_p are the damping ratio. Let $\mu = m_a/m_p$, Eq. (3) can be further expressed as

$$\beta = \frac{\dfrac{-\mu\left(2\mathrm{j}\xi_a\dfrac{\omega_a}{\omega} + \dfrac{k/m_a}{\omega^2}\right)}{\dfrac{k/m_a}{\omega^2} - 1 + 2\mathrm{j}\xi_a\dfrac{\omega_a}{\omega}} + \left(-1 + 2\mathrm{j}\xi_p\dfrac{\omega_p}{\omega} + \left(\dfrac{\omega_p}{\omega}\right)^2\right)}{-1 + 2\mathrm{j}\xi_p\dfrac{\omega_p}{\omega} + \left(\dfrac{\omega_p}{\omega}\right)^2}. \tag{4}$$

For APVA, the adaptive control law in Eq. (2a) is used to vary its variable stiffness. In addition, ω_a is generally designed to be equal to ω_p. Therefore Eq. (4) can be simplified as

$$\beta = \frac{-\mu\left(1 + \dfrac{\Omega}{2j\xi_a}\right) + \left(-1 + 2j\xi_p\dfrac{1}{\Omega} + \left(\dfrac{1}{\Omega}\right)^2\right)}{-1 + 2j\xi_p\dfrac{1}{\Omega} + \left(\dfrac{1}{\Omega}\right)^2}, \tag{5}$$

where $\Omega = \omega/\omega_p$ is the dimensionless frequency. In order to reflect the vibration reduction contribution of the AVPA, dimensionless impedances are defined as

$$Z_a = -\mu\left(1 + \frac{\Omega}{2j\xi_a}\right), \quad Z_p = -1 + 2j\xi_p\frac{1}{\Omega} + \left(\frac{1}{\Omega}\right)^2, \tag{6}$$

where Z_a, Z_p are, respectively, introduced by the AVPA and primary system. Substituting Eq. (6) into Eq. (5) gives

$$\beta = \frac{Z_a + Z_p}{Z_p} = 1 + \frac{Z_a}{Z_p}. \tag{7}$$

From Eq. (6), given ξ_a and μ, Z_a is a monofonic function of the dimensionless frequency Ω. For a given primary system, Z_p is a determinative function of the dimensionless frequency Ω. When Ω is near unity, Z_p has the minimum value and increases rapidly with respect to Ω deviating from the unity. From Eq. (7), the vibration reduction coefficient β is in inverse proportion to Z_p if the effect of phase is neglected. Therefore, when the disturbance frequency is near the resonance frequency of the primary system, the AVPA has the best vibration reduction effect, but it decreases rapidly with the disturbance frequency deviating from the resonance frequency. That is one of the limitations of AVPA. However, for a given Z_p, the larger modulus Z_a has, the larger β is. That is, the dimensionless impedance Z_a is a good metric representing the performance of an AVPA. From Eq. (6), increasing μ can enhance the APVA's performance but the larger mass ratio will make it heavier and unpractical. Moreover, from Eq. (6), the smaller damping ratio ξ_a is, the larger modulus Z_a has. It seems as if decreasing the damping ratio is a good choice but will make the APVA become less robustness and not fail-safe. That is due to that for the APVA with smaller damping ratio, it will seriously deteriorate the vibration of primary structure when the part of variable stiffness is disabled and the excitation frequency deviates the natural frequency of APVA. In next section an alternative method will settle these problems.

2.1. Comparison of APVA and PVA

A PVA means k in Eq. (4) is fixed and equal to $k_a = m_a\omega_a^2$. The concept of dimensionless impedance can also represent the performance of PVA. From Eq. (4), the dimensionless impedance of PVA can be written as

$$Z_{PVA} = \frac{-\mu\left(2j\xi_a\dfrac{\omega_a}{\omega} + \left(\dfrac{\omega_a}{\omega}\right)^2\right)}{\left(\dfrac{\omega_a}{\omega}\right)^2 - 1 + 2j\xi_a\dfrac{\omega_a}{\omega}} = \frac{-\mu\left(2j\xi_a\dfrac{1}{\Omega} + \left(\dfrac{1}{\Omega}\right)^2\right)}{\left(\dfrac{1}{\Omega}\right)^2 - 1 + 2j\xi_a\dfrac{1}{\Omega}}. \tag{8}$$

The comparison of the modulus of Z_a and Z_{PVA} is shown in Fig. 2. The three curves almost hold constants at the low dimensionless frequency range and increase rapidly at the high dimensionless frequency range. There are sharp troughs near $\Omega = 1$. Their values of the low dimensionless frequency range are smaller than that of the high dimensionless frequency range. Moreover, the curve with small damping ratio is above that with large damping ratio. The results show that the modulus of Z_a is much larger than that of Z_{PVA} except the point $\omega = \omega_a$. That is, the APVA has much better performance than PVA. Moreover, the smaller the damping ratio is, the more obvious the superiority of APVA is. In addition, the performance of the APVA for the low frequency excitation is less than that for the high frequency excitation.

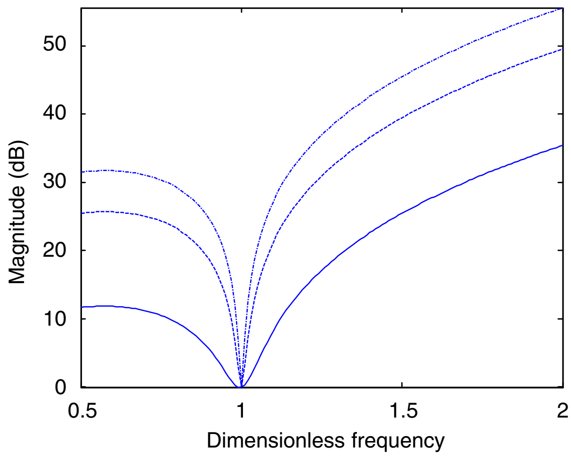


Fig. 2. Influence of the damping ratio ξ_a on $|Z_a/Z_{\mathrm{PVA}}|$. Solid line: $\xi_a = 0.05$; dashed line: $\xi_a = 0.01$; and dash-dot line: $\xi_a = 0.005$.

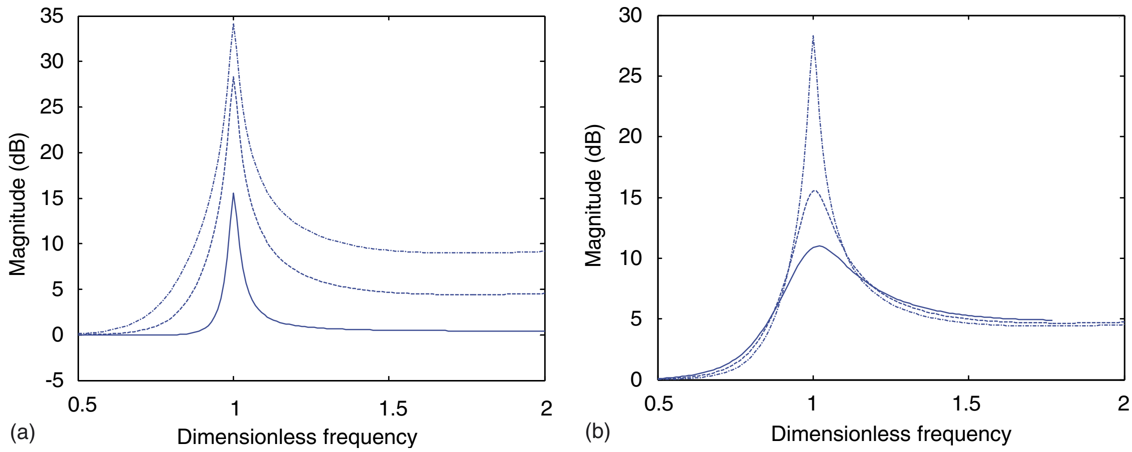


Fig. 3. Vibration reduction effect comparison: (a) the case APVA with different damping ratio, $\mu = 0.01$, $\xi_p = 0.01$, solid line: $\xi_a = 0.005$; dashed line: $\xi_a = 0.01$; dash-dot line: $\xi_a = 0.005$; and (b) the case primary system with different damping ratio, $\mu = 0.01$, $\xi_a = 0.01$, solid line: $\xi_p = 0.1$; dashed line: $\xi_p = 0.05$; dash-dot line: $\xi_p = 0.01$.

2.2. *Effect of the damping ratios on the performance of APVA*

Let $\gamma = 20\log_{10}|\beta|$ as the index of vibration reduction effect. The relationship between γ and dimensionless frequency Ω is demonstrated in Fig. 3. The three curves in Fig. 3(a) all have sharp peaks whose values increase rapidly as the damping ratios ξ_a decrease. Moreover, the curve with small damping ratio is above that with large damping ratio. The three curves in Fig. 3(b) also have peaks whose values increase rapidly as the damping ratios ξ_p decrease. The curve with small damping ratio is still above that with large damping ratio near $\Omega = 1$, but the order is adverse at the low and high frequency range. In addition, for all the curves in Fig. 3, their values of the low dimensionless frequency range are smaller than that of the high dimensionless frequency range. The results show that given the mass ratio and the damping ratio of the primary system, the smaller the damping ratio of APVA is, the better the vibration reduction effect is. Moreover, the frequency range of vibration reduction is wider for the smaller ξ_a. It is the same with the above analysis that when the dimensionless frequency Ω is near unity, the AVPA has the best vibration reduction effect, but it decreases rapidly with the disturbance frequency deviating from unity, which is coincided with the experimental results in literatures [7,8]. For the primary system with different damping ratio ξ_p and given the mass ratio and the damping ratio of APVA, γ is larger for the case with smaller ξ_p near $\Omega = 1$, but γ is smaller when Ω is away from unity.

3. The principle of the AARA

From Section 2, it can be obtained that APVA has a wide absorber band but its vibration reduction effect is limited by its damping. In order to avoid the limitation, a new absorber called AARA is proposed. It consists of two parts. The first part is an APVA with variable stiffness, which can be adaptively tuned to the correct frequency. The second part is an actuator which provides control force to cancel the damping force applied on the absorber, hence leading to resonance. In fact, AARA can be considered as the integration of APVA and ARA, so their advantages such as low cost, high performance and fail-safe are inherited.

A single-degree-of-freedom primary system with an AARA is shown as Fig. 4. It is similar to that shown in Fig. 1. The main difference between them lies in that an actuator is introduced in Fig. 4. The actuator provides the active control force f_{act}.

The equations of motion for the system in Fig. 4 can be written as

$$m_a\ddot{x}_a + c_a(\dot{x}_a - \dot{x}) + k(x_a - x) = f_{\text{act}}, \tag{9}$$

$$m_p\ddot{x} + c_p\dot{x} + c_a(\dot{x} - \dot{x}_a) + k_p x + k(x - x_a) = f - f_{\text{act}}, \tag{10}$$

where k is the function of the exciting frequency ω and equal to $m_a\omega^2$ in order to make AARA track the exciting frequency adaptively. The adaptive control law of variable stiffness part of AARA is the same as that of APVA described in Section 2. In order to cancel the damping force and obtain a resonator absorber, the active control force f_{act} should be equal to the damping force applied on the absorber, i.e.,

$$f_{\text{act}} = c_a(\dot{x}_a - \dot{x}) \tag{11}$$

which can be realized easily by velocity feedback. When the frequency of the external force f changes, it can be tracked by adjusting the variable stiffness k adaptively. For resonator absorber, the vibration of the primary system can be reduced to zero in theory, so AARA can get high performance in wideband frequency.

4. Control effort of AARA

For active absorbers, all kinds of them can obtain high performance as long as their control forces are enough. However, the absorber with too large control effort is impractical. So the control effort of active absorber is an important performance index. In this section, the control effort of AARA is discussed and compared to ARA.

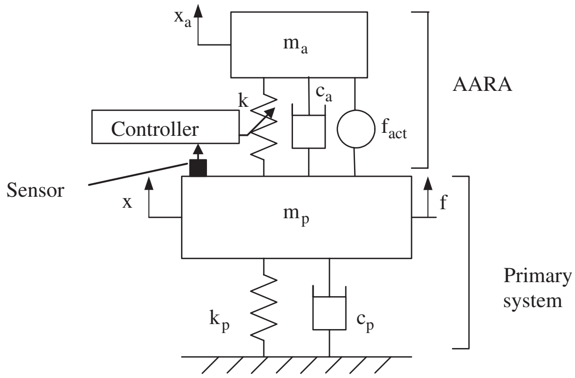


Fig. 4. A single-d.o.f. primary system with an AARA.

Considering the system shown in Fig. 4, the vibration of the primary system can be reduced to zero in theory. Therefore it can be assumed that the displacement x of the primary system is zero in the steady state. Then Eqs. (9)–(11) can be written as

$$m_a\ddot{x}_a + c_a\dot{x}_a + kx_a = f_{\text{act}}, \tag{12}$$

$$-c_a\dot{x}_a - kx_a = f - f_{\text{act}}, \tag{13}$$

$$f_{\text{act}} = c_a\dot{x}_a. \tag{14}$$

Eq. (12) plus Eq. (13) gives a simple equation,

$$m_a\ddot{x}_a = f. \tag{15}$$

Taking into account the harmonic exciting force with frequency ω, let $f = A_f e^{j\omega t}$, $x_a = A_{x_a} e^{jwt}$, $f_{\text{act}} = A_{f_{\text{act}}} e^{j\omega t}$, where A_f, A_{x_a} and $A_{f_{\text{act}}}$ are the complex amplitudes of exciting force f, displacement x_a and control force f_{act}, respectively. Substituting them into Eqs. (14) and (15), the relationship of the three amplitudes can be easily obtained as follows:

$$A_{x_a} = -\frac{1}{m_a\omega^2}A_f, \tag{16}$$

$$A_{f_{\text{act}}} = -\frac{jc_a}{m_a\omega}A_f = -\frac{2\xi_a j}{\Omega}A_f. \tag{17}$$

If the natural frequency of AARA cannot be adjusted, then AARA becomes an ARA and its stiffness is a constant labeled as k_a. In order to track the exciting frequency, ARA must have displacement or acceleration feedback besides velocity feedback. For the two cases, their control force can be, respectively, expressed as

$$f_{\text{act}}^d = c_a\dot{x}_a + \Delta kx_a, \quad f_{\text{act}}^a = c_a\dot{x}_a + \Delta m\ddot{x}_a, \tag{18}$$

where Δk, Δm are the gains of displacement and acceleration feedback. Substituting Eq. (18) into Eq. (12) and making ARA track the exciting frequency, Δk, Δm are found to be

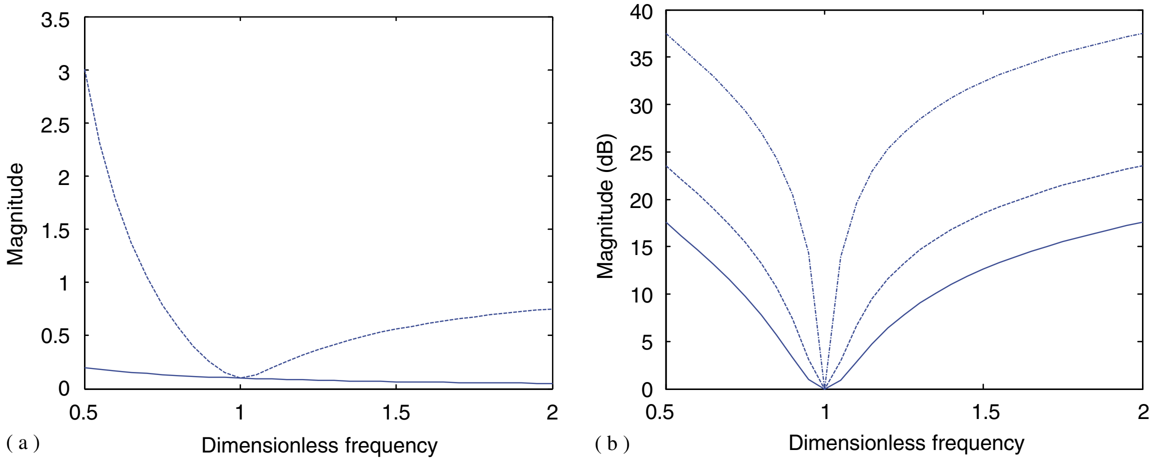
$$\Delta k = k_a - m_a\omega^2, \quad \Delta m = m_a - \frac{k_a}{\omega^2}. \tag{19}$$

From Eqs. (15), (18) and (19), the corresponding amplitudes of the two control forces, $A_{f_{\text{act}}^d}$, $A_{f_{\text{act}}^a}$, are identical and can be written as

$$A_{f_{\text{act}}^d} = A_{f_{\text{act}}^a} = \left(-\frac{jc_a}{m_a\Omega} + \left(1 - \left(\frac{\omega_a}{\omega}\right)^2\right)\right)A_f = \left(-\frac{2\xi_a j}{\Omega} + \left(1 - \left(\frac{1}{\Omega}\right)^2\right)\right)A_f. \tag{20}$$

The comparison of vibration control efforts is presented in Fig. 5. It is shown in Fig. 5(a) that the control force needed by ARA is much larger than that by AARA except the point $\Omega = 1$. The control force needed by ARA has a minimum value at $\Omega = 1$ and increases rapidly when the exciting frequency deviates from its natural frequency. In addition, the control force needed by ARA at the low frequency range is larger than that at the high frequency range. Nevertheless, for AARA, the control force decreases slowly as the exciting frequency increases, so the change is small. The results show that the control effort of AARA is much smaller than that of ARA and AARA has wider effective frequency range than ARA.

The amplitude ratios of the control force of AARA and ARA with different damping ratio are presented in Fig. 5(b). The three curves all have sharp troughs near $\Omega = 1$ and the curve with small damping ratio is above that with large damping ratio. The results demonstrate that the smaller the damping ratio is, the more obvious the advantage of AARA is. However, the adjustment of the variable stiffness for AARA also need control effort but this part is very small relative to the part of the control force and can be neglected in general. Therefore the control effort of AARA is much lower than that of ARA.



(a) (b)

Fig. 5. Vibration control effort comparison: (a) the amplitude ratios of the control force and exciting force: solid line: AARA; dashed line: ARA; and (b) the amplitude ratios of the control force of AARA and ARA: solid line: $\xi_a = 0.1$; dashed line: $\xi_a = 0.05$; dash-dot line: $\xi_a = 0.01$.

5. Conclusions

This investigation performs a thorough analysis on the advantages and limitations of adaptive-passive vibration absorber (APVA). In order to avoid the limitations of APVA, a novel kind of adaptive active resonator absorber (AARA) has been proposed. It can be considered as the integration of APVA and active resonator absorber (ARA), so their advantages such as low cost, high performance and fail-safe are inherited. The theoretical analysis on the control effort of AARA and ARA shows that AARA need much smaller control force than ARA.

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