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Some recent developments in adaptive tuned vibration absorbers/neutralisers

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Abstract. The vibration absorber has been used for vibration control purposes in many sectors of engineering from aerospace, to automotive to civil, for the past 100 years or so. A limitation of the device, however, is that it acts like a notch filter, only being effective over a narrow band of frequencies. Recent developments have overcome this limitation by making it possible to tune the device over a range of frequencies. This has been achieved by incorporating a variable stiffness element that can be adjusted in real-time. In this paper, some ways in which stiffness change can be achieved in practice are reviewed and some examples of prototype adaptive tuned vibration absorbers (ATVAs) are described. A simple control scheme to automatically tune an ATVA is also presented.

Keywords: Vibration, absorber, neutraliser, control

1. Introduction

The tuned vibration absorber (TVA) has been used for many years since its inception by Ormondroyd and Den Hartog [1]. The device acts like a notch filter; it is required to have a very narrow bandwidth when it is tuned to suppress vibration at a troublesome forcing frequency and a larger bandwidth when it is used to suppress vibration at a troublesome resonance frequency. The concern in this paper is the former. The main disadvantage in using a TVA to control vibration is that it can actually increase vibration of the host structure if the forcing frequency drifts so that the natural frequency of the TVA no longer coincides with the forcing frequency. This can also occur if the natural frequency changes because of environmental factors such as temperature change. To overcome this Brennan [2] has designed a wideband absorber, but the disadvantage of this is that it requires the addition of mass for it to be effective. To avoid this, adaptive tuned vibration absorbers (ATVAs) have been developed. An overview of some of these devices is given by Brennan [3], Sun and Jolly [4], and von Flotow et al. [5].

At the heart of an ATVA is a stiffness element, whose stiffness can be changed in real-time. Inevitably the stiffness element will have some structural damping and this will be a limiting factor in the performance of such a device [6]. The aim is to design a stiffness element that has low structural damping, whose stiffness can be adjusted quickly and easily with minimum power. In this paper some ways in which stiffness can be changed in practice are described and some examples of prototype ATVAs are presented. A simple control scheme to automatically tune an ATVA is also described.

2. Review of variable stiffness mechanisms

One early variable stiffness element used in a vibration absorber was described by Longbottom et al. [7]. A mass was sandwiched between a pair of pneumatic rubber bellows, and the stiffness was adjusted by changing the air pressure inside the bellows. Further work on this device by Long et al. [8] resulted in a simple means of automatically adjusting the stiffness. Franchek et al. [9] proposed a very simple way of changing stiffness by simply adjusting

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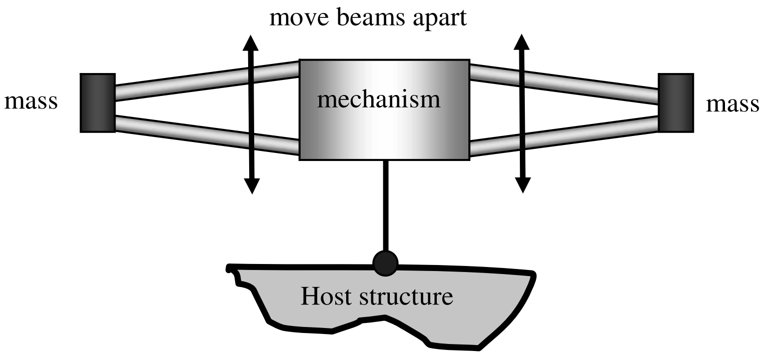


Fig. 1. A beam-like vibration absorber where the stiffness is adjusted by moving the beams apart.

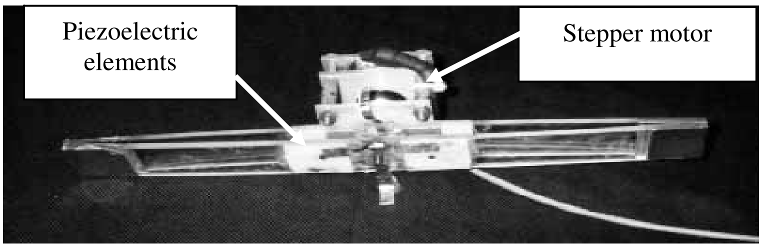


Fig. 2. Experimental beam-like ATVA.

the number of active coils in a spring. Walsh and Lamancusa [10] used a compound leaf spring for a stiffness element and adjusted the gap between the two beams. This had the advantage of having low damping and hence could potentially be more effective than the design described by Longbottom et al. [7]. This design was modified by Brennan [3], and developed by Kidner and Brennan [11–13] with a system to control both stiffness and damping. This device involved a change in shape to control the stiffness, but the actuation mechanism was relatively slow to operate. A development on this theme was described by Bonello et al. [14], who controlled the curvature of beams using piezoelectric actuators to the stiffness elements of an ATVA. This device had a very quick response time, and is discussed later in this paper. Inspired by the mechanical linkage used in the flight mechanism of small insects described by Brennan et al. [15], Bonello et al. [14] designed an ATVA that had a wide frequency range of operation and required virtually no force to adjust the stiffness of the device. The disadvantage of the device, however, was that it required an actuator with a large stroke. More details of ATVAs that exploit shape-change to change natural frequency are given in Section 3 of this paper.

A large-scale device where a mass is supported in the middle of a beam, and the length of the beam (which provides the stiffness element) is adjusted is described by Hong and Ryu [16]. A similar but small-scale device has recently been used by Carneal et al. [17] to control sound radiation from a plate. Their device was particularly effective as the actuator (a stepper motor) was also used as the absorber mass. Novel devices using electromagnetic forces to counteract a passive stiffness have been described by Abu-Akeel [18] and Waterman [19]. More recently Hagood and von Flotow [20], and Hollkamp [21] have designed a vibration absorber that uses a piezoelectric element which is bonded to a structure and connected to an inductor and a resistor to complete a resonant electrical circuit, which can be tuned to the required frequency.

In recent work, Williams et al. [22] have used shape memory alloy as a variable stiffness element. The Young's modulus (and hence the stiffness) of the element is changed by adjusting the temperature of the alloy by passing an electrical current through it. A device similar to this has been developed by Rustighi et al. [23] and is described in Section 4 of this paper. Hirunyapruk [24], has demonstrated that an ATVA can be made using a fluid-filled beam. A magneto-rheological fluid was used in its pre-yield state, and the shear stiffness of the fluid was adjusted by changing the magnetic field applied to the fluid. This is described briefly in Section 5 of this paper.

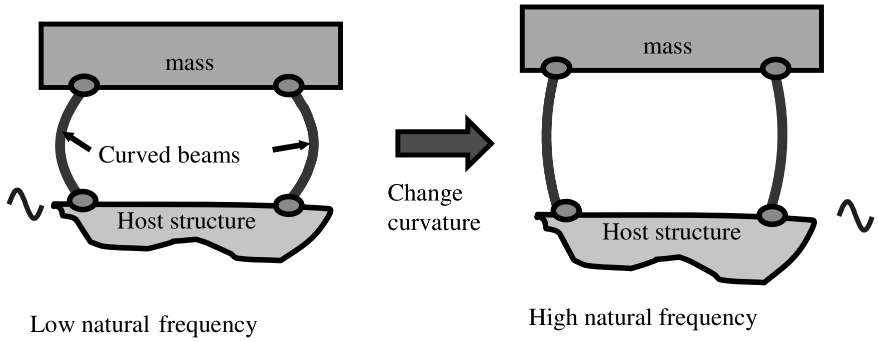


Fig. 3. ATVA with stiffness element formed from parallel curved beams. As the beams straighten, the natural frequency increases.

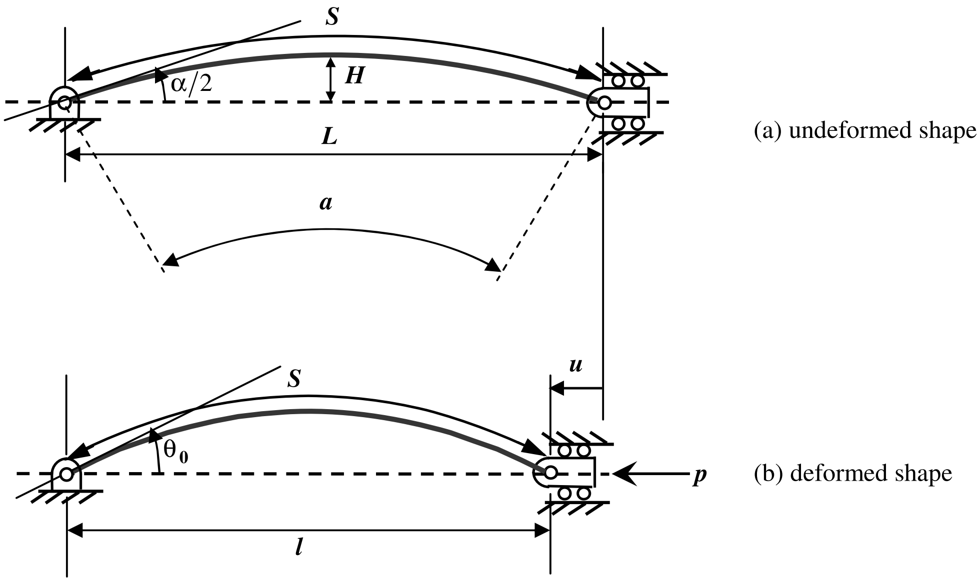


Fig. 4. Curved beam loaded along its span.

A control algorithm to automatically tuned the ATVA using the phase of the acceleration measured either side of the stiffness element is outlined in Section 6. This algorithm was based on the work of Long et al. [8], and was substantially modified by Rustighi et al. [25] to take into account the nonlinearity of the control system; it also added damping to the controller to suppress control activity without penalising performance. This control algorithm was also applied to the ATVA developed by Bonello et al. [26], to test its agility with an ATVA that can change its stiffness rapidly.

3. ATVAs using a variable stiffness element realised by shape control

A simple way of designing an ATVA is to manufacture it from two beams that can be separated so that the stiffness is a function of the distance between the beams. Such a system is shown in Fig. 1. For this device the lowest natural frequency, ω_l, occurs when the distance between the beams, h, where they attach to the mechanism, is zero. The ratio of the natural frequency of the device, ω_n, to the lowest natural frequency is given by [3]

$$\frac{\omega_n}{\omega_l} = 1 + \frac{3h}{8d} \tag{1}$$

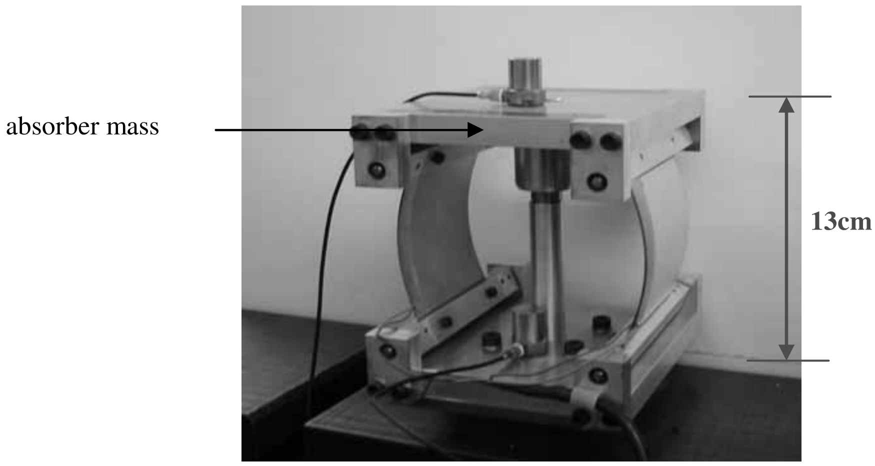


Fig. 5. Prototype ATVA using curved beams as a variable stiffness element.

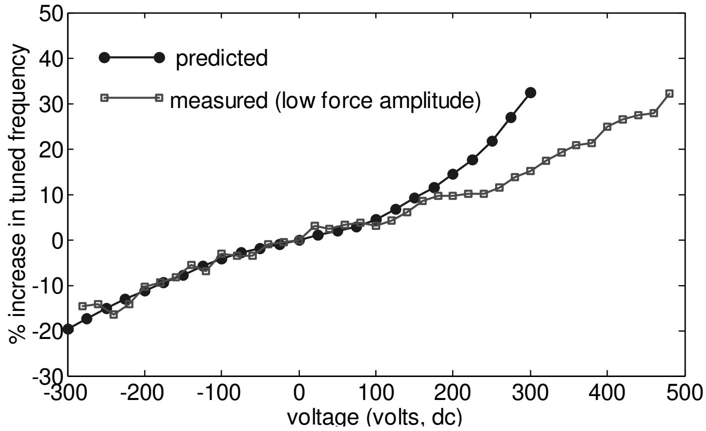


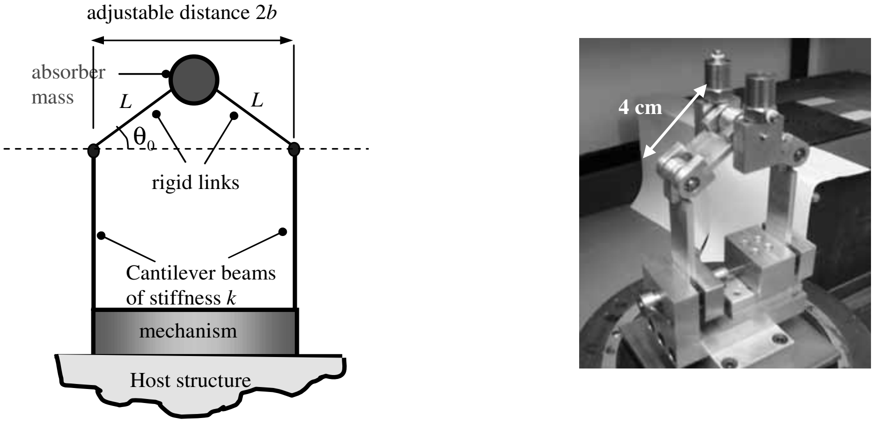
Fig. 6. Tuning characteristic for the prototype ATVA shown in Fig. 5.

where d is the thickness of an individual beam and $h < d$. It can be seen that this design is attractive in that only a relatively small movement is required to achieve a reasonable change in natural frequency. Kidner and Brennan [12, 13], modified this design using two perspex beams that were connected by tape at the ends and were allowed to bend in the centre as shown in Fig. 2. The corresponding change in natural frequency is given by

$$\frac{\omega_n}{\omega_l} = 1 + \frac{1}{2\sqrt{2}}\frac{h}{d} \tag{2}$$

A stepper motor was used to drive the beams apart, resulting in a change in natural frequency of 35% (100–135 Hz). The piezoelectric elements fitted to the beams shown in Fig. 2 were incorporated into a separate control system. They excited the beams dynamically at the operational frequency of the ATVA to counteract the effects of passive damping in the device in a way described by Kidner and Brennan [11]. This was done to improve the effectiveness of the ATVA. Although the ATVA in Fig. 2 was effective, the stiffness change was sluggish due to the relatively slow response of the stepper motor.

Another design of ATVA that exploits shape-change is shown in Fig. 3. The stiffness element consists of two identical curved beams and the stiffness is varied by adjusting the curvature of each beam. This is relatively easy to do because the ends are pivoted and are free to move in the vertical direction. To achieve this, a layer of piezoelectric ceramic is bonded to each beam and the dc voltage level applied to these actuators is adjusted. Such a simple arrangement has the advantage of not having redundant mass introduced by an actuator.



(a) diagram of the ATVA (b) photograph of the prototype

Fig. 7. Alternative design of an ATVA using shape change to realise a variable stiffness element.

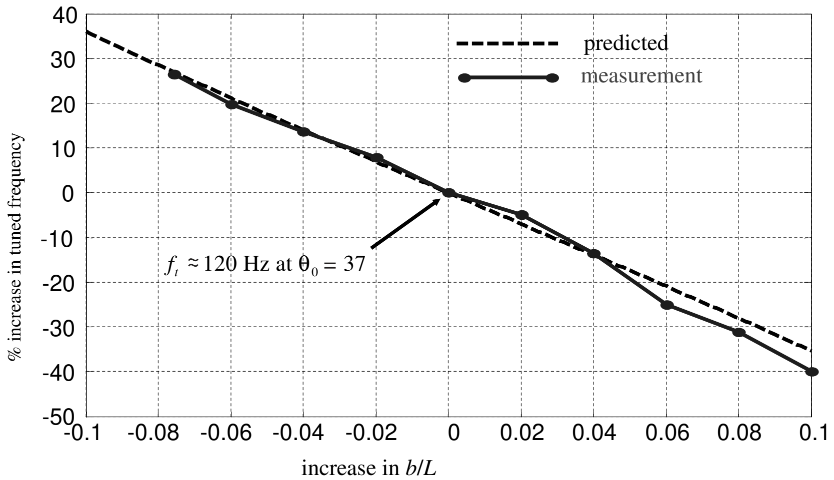
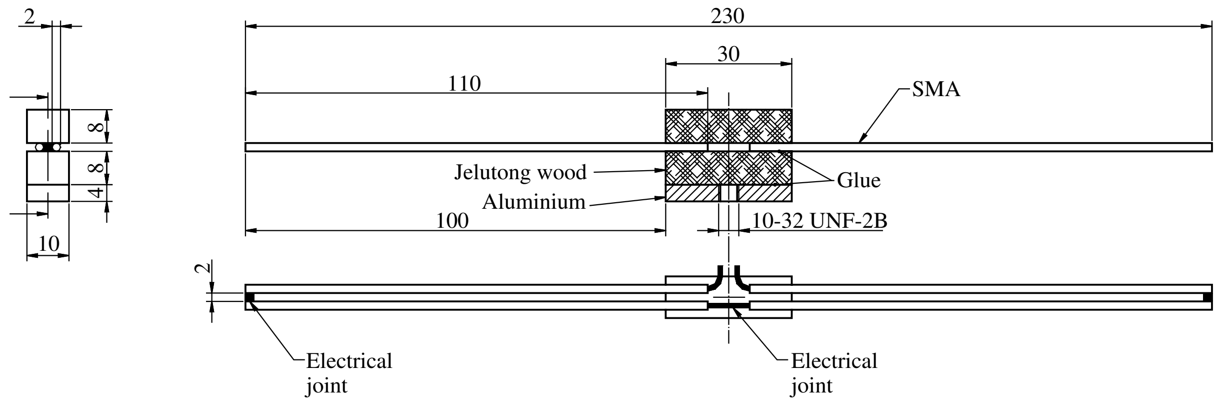


Fig. 8. Tuning characteristic of the prototype ATVA shown in Fig. 7.

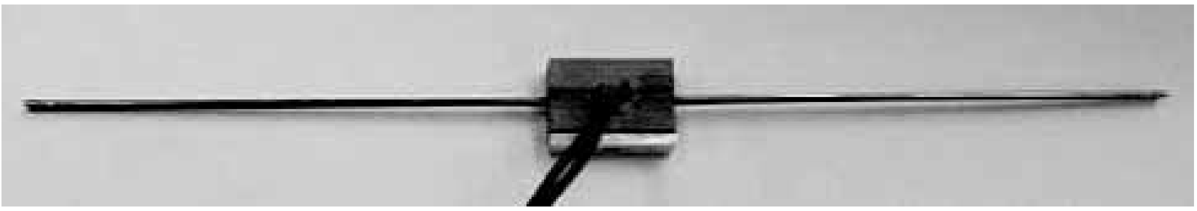
Consider one beam from the ATVA shown on its side for convenience in Fig. 4(a). It is pivoted at each end and has an initial curvature in the form of a circular arc of length S subtending an angle a at the centre. The beam is subjected to the compressive load p along its span and deforms as shown in Fig. 4(b). Using Castigliano's theorem, assuming the beam is slender and considering bending deformation only, neglecting any deformation along the curved longitudinal axis of the beam (S constant), the non-dimensional static stiffness is approximately given by [26]

$$\tilde{k} = \frac{p/u}{p_E/S} \approx \frac{2}{\pi^2 (H/S)^2} \tag{3}$$

where $p_E = \pi^2 EI/S^2$ is the Euler buckling load of a straight beam of the same length S, E is the Young's modulus and I second moment of area of cross-section of the beam. The natural frequency of the ATVA in Fig. 3 is proportional to $\sqrt{\tilde{k}}$, and so it is approximately inversely proportional to the crown height H of the beams. The curvature of the beams in the ATVA pictured in Fig. 3 can be adjusted by using a piezoelectric actuator fitted to each beam. In a prototype device pictured in Fig. 5, which had a lower natural frequency of 36 Hz, the natural frequency could be adjusted by about 56% to 56 Hz with a dc voltage between -280 V to $+480$ V applied to the piezoelectric actuators.



(a) Drawing of the ATVA



(b) Photograph of the ATVA

Fig. 9. Prototype shape memory alloy ATVA.

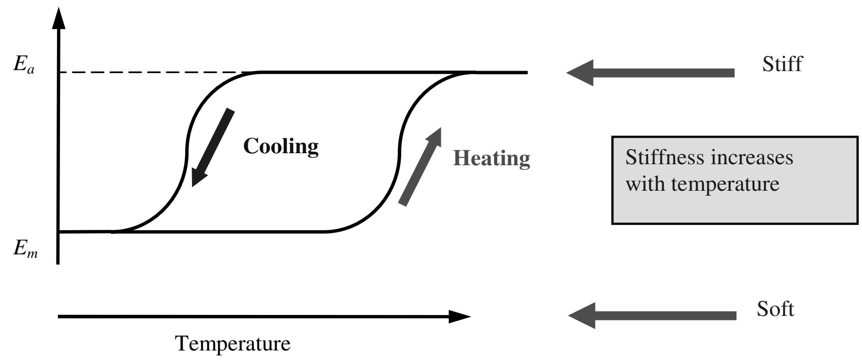


Fig. 10. Young's modulus of NiTinol SMA wire as a function of temperature. E_a and E_m denote the Young's modulus of the wire in its austenitic and martensitic states respectively. For the wire used in the prototype depicted in Fig. 9, the Young's modulus changes from 40 to 59 MPa.

The prototype ATVA depicted in Fig. 5 was placed on a shaker and excited with random vibration and its natural frequency measured as the dc voltage applied to the device was adjusted. The predicted and measured relationship between the natural (tuned) frequency and the applied dc voltage is shown in Fig. 6. In the predictions the inertia of the curved beams was neglected. This affects the upper frequency limit of the device and is the cause of the deviation between the predicted and the measured results.

The ATVA based on curved beams has the disadvantage that the actuators have to act against the stiffness required for the device to have a reasonably high natural frequency. This means that they have to have the capability to generate large forces, and in the case when piezoelectric actuators are used, high voltages are required. To overcome this problem it is possible to decouple the stiffness required for a high natural frequency and the stiffness that the actuators have to act against to change the natural frequency. An example of such a device is shown in Fig. 7, where

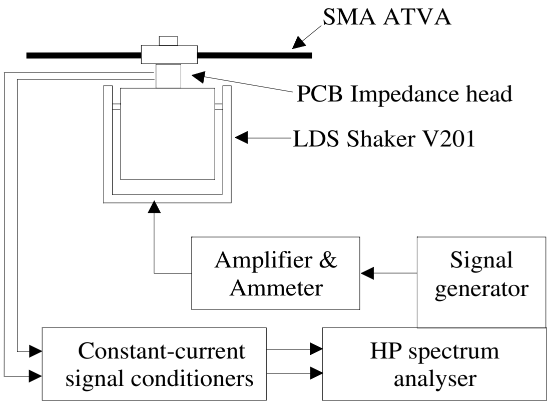


Fig. 11. Experimental set-up to measure the dynamic characteristics of an SMA ATVA.

Fig. 7(a) shows a diagram of the ATVA and Fig. 7(b) is a photograph of a prototype. When the mass moves in the vertical direction, the cantilever beams bend laterally, and the effective stiffness for small vertical motion of the mass is given by [15]

$$\frac{k_{\text{effective}}}{k} \approx 2\left\{\frac{1}{(b/L)^2} - 1\right\} \tag{4}$$

Consequently, by adjusting the distance b using a suitable mechanism the effective stiffness and hence natural frequency can be adjusted. Care has to be taken in the design of the device so that "snap through" does not occur. The actuating mechanism in Fig. 7(a) is a redundant mass that degrades the vibration attenuation. A better configuration would be to invert the device so that the actuator is now part of the effective mass of the device. The tuning characteristic of the ATVA in Fig. 7 is shown in Fig. 8, giving the percentage change in the natural (tuned) frequency from a nominal configuration. The operational frequency range for the device is limited to frequencies for which inertia effects of the components forming the linkage are negligible.

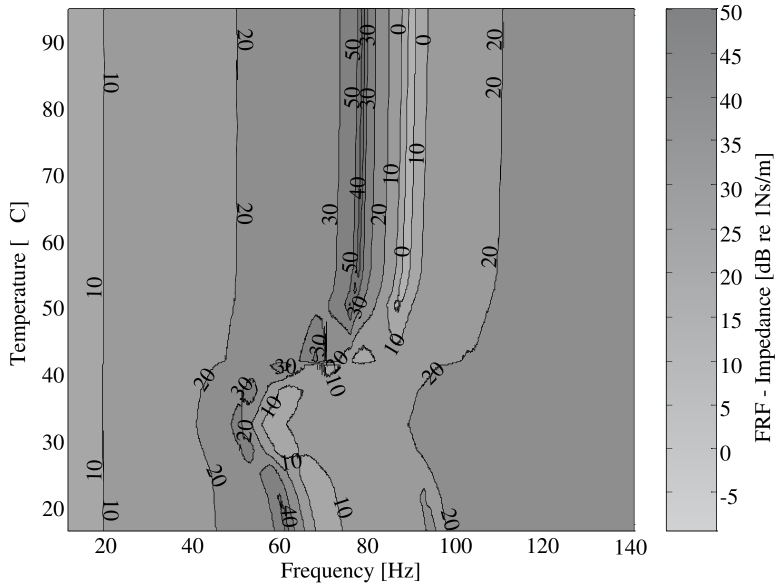
4. ATVA using a shape memory alloy stiffness element

Rather than change stiffness by shape change, another approach is to change material property. One way of doing this is to use shape memory alloy in a beam-like configuration as discussed by Williams et al. [22] and Rustighi et al. [23]. The device described by Rustighi et al. is shown in Fig. 9.

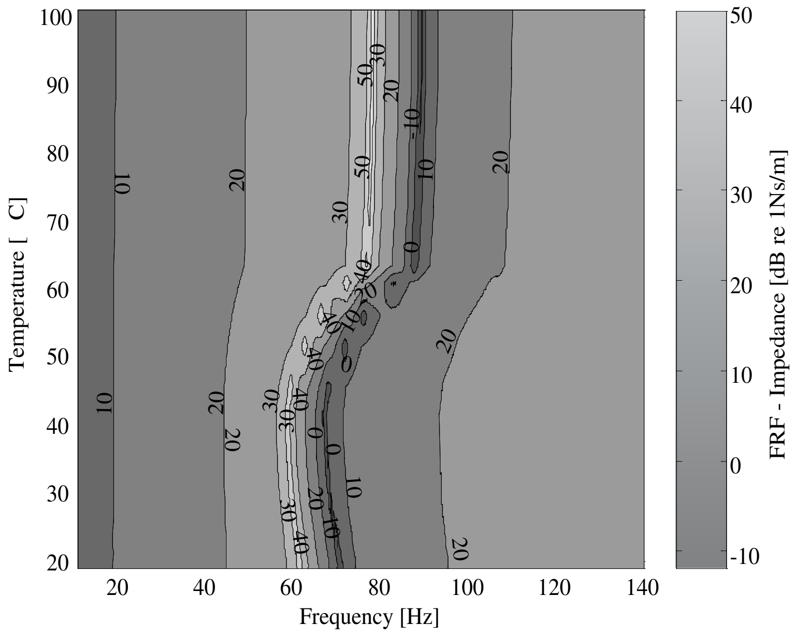
The device consists of two SMA wires and is connected to a host structure at its centre. Shape memory alloy has a low stiffness when it is in its martensitic state and a high stiffness when it is in its austenitic state. To transit from one state to the other requires a change in temperature, which is achieved by passing a current through the wires. The stiffness of the wire as a function of temperature is shown graphically in Fig. 10. It can be seen that there is hysteresis in the SMA wire (about 10°C for the NiTinol wire used in the prototype depicted in Fig. 9) and any control system employed to control the natural frequency of an SMA ATVA has to take account of this behaviour.

The SMA ATVA was tested by placing it onto a shaker, as shown in Fig. 11, and measuring the mechanical impedance (applied force/resulting velocity) using a PCB impedance head type D288 D01. The SMA ATVA was excited over a range of frequencies as the temperature of the SMA was increased incrementally from 20 °C to 100°C so the SMA wires changed their stiffness. A peak in the mechanical impedance denotes the natural frequency of the device. A similar experiment was conducted during the transition from hot to cold. The results of these experiments are shown in Fig. 12.

It can be seen from Fig. 12 that the lowest natural frequency was about 72 Hz and that the highest natural frequency was about 88 Hz, giving a change of about 22% in natural frequency. It can also be seen from Fig. 12 that the transition from one material state to the other occurred at different temperatures on heating and cooling. The temperature difference due to hysteresis was about 10 °C. Transient testing of the device showed that the SMA ATVA took a relatively long time to respond. It took about 2 minutes for the device to adjust its natural frequency from 72 Hz to 88 Hz using a current of about 9A.



(a) cooling

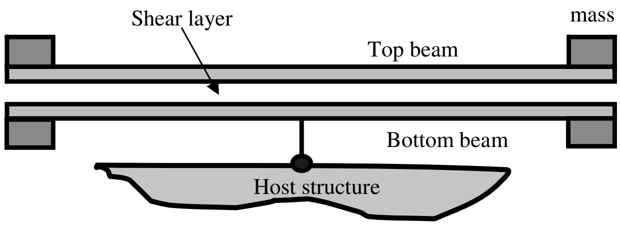


(b) heating

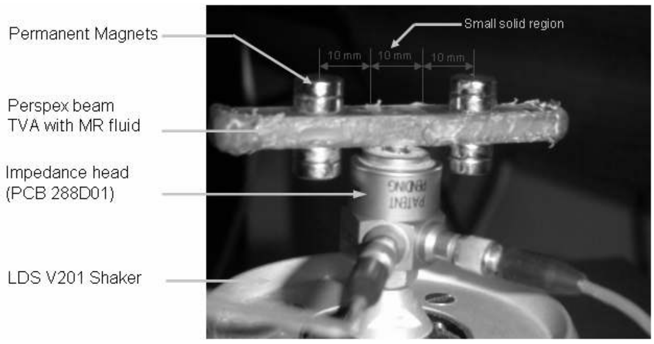
Fig. 12. Mechanical impedance of the SMA ATVA during cooling and heating.

5. ATVA using tunable fluid-filled beam

Hirunyapruk [24] has recently proposed a using a tunable fluid-filled beam as an ATVA. Such a device is shown in Fig. 13. The fluid was magneto-rheological fluid and was used in its pre-yield state. It was controlled in the proof-of-concept device by using permanent magnets to change the magnetic field applied to the fluid. The lowest frequency is governed by the smallest achievable stiffness; this is when the shear stiffness of the core is zero and



(a) schematic



(b) experimental setup to verify the proof of concept

Fig. 13. A tunable fluid-filled beam ATVA.

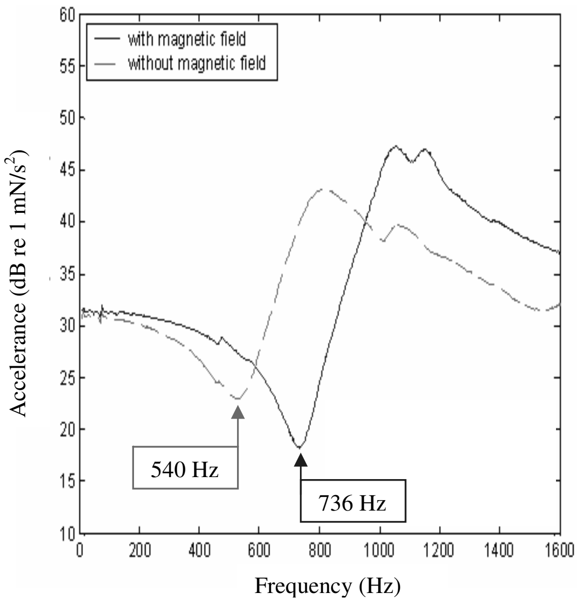
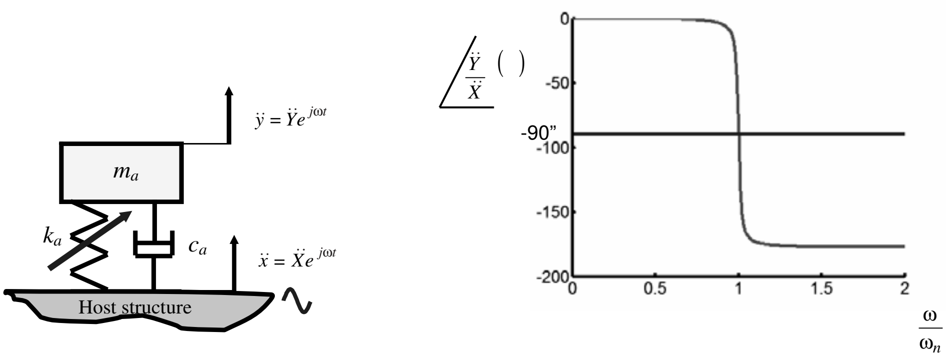


Fig. 14. Measured accelerance of the tunable fluid-filled beam ATVA.

the two beams vibrate independently. The highest natural frequency occurs when the shear stiffness is infinite. In a practical device these conditions cannot be achieved and so the frequency range of the device is somewhat less than the ideal.

To demonstrate that control of the shear stiffness of a sandwich beam is an effective way of implementing an



(a) schematic of an ATVA (b) phase relationship

Fig. 15. Schematic of an ATVA on a structure and the phase relationship between the acceleration of the host structure and the ATVA mass.

ATVA an experiment was setup as shown in Fig. 13(b). The accelerance (resulting acceleration/applied force) of the ATVA was measured using PCB impedance head type D288 D01 over a range of frequencies. A trough in the measured accelerance denotes the natural frequency of the ATVA (as opposed to the peak in the impedance in the previous section). To determine the natural frequency of the device with no magnetic field applied the permanent magnets were replaced by brass pieces of the same mass. The experimental results are shown in Fig. 14. It can be seen that a change in natural frequency of about 37% was achieved.

6. A controller for an ATVA

For the purpose of control, an ATVA can be considered to be a single degree of freedom system such as that shown in Fig. 15(a). When it is tuned, the natural frequency of the device ω_n is adjusted so that it is the same as the forcing frequency ω. When this condition occurs the phase between the acceleration of the absorber mass and the host structure is very close to 90° as shown in Fig. 15(b). Thus, a control system has to measure the phase between acceleration of the host structure and the ATVA mass and then adjust the stiffness until the phase between these two accelerations is 90°. This strategy was first proposed by Long [8] and subsequently modified by Rustighi et al. [25] who applied it to the SMA ATVA in Fig. 9. Bonello et al. [26] also applied it to the much more agile ATVA shown in Fig. 5 and the controller performance with this device is discussed later.

The phase angle can be determined from the measured accelerations by

$$\phi = \cos^{-1}\left(\frac{\frac{2}{T}\int_0^T \ddot{x}\ddot{y}dt}{|\ddot{X}||\ddot{Y}|}\right) \tag{5}$$

A simple proportional controller is not effective in this application because of the non-linear characteristic of the phase. It is desirable for the controller to make large adjustments to the stiffness when the natural frequency of the ATVA is very different from the forcing frequency, and to make small adjustments when the difference between these two quantities is small. Thus a suitable discrete time control law has been found to be

$$V_{n+1} = V_n - [P(\phi_n + \phi_n^3 + \phi_n^5) + Dd_n] \tag{6}$$

where V_n is the control signal at the n^{th} time step, d_n is the derivative of the phase angle at the n^{th} time step and P and D are constants which need to be determined empirically for each type of ATVA (This can be done either by simulation or experiment). The step size is also dependent upon the application and the type of ATVA. Some experimental results are presented in Fig. 16 for the ATVA shown in Fig. 5. The ATVA was placed on a large shaker and subjected to the frequency sweep shown in Fig. 16(a). This was a rapid sweep of 7 Hz/s from 38 Hz to 52 Hz. Figure 16(b) shows the acceleration of the shaker mass (host structure) with the control system switched off. The low vibration level at about 6 seconds is when the forcing frequency is coincident with the natural frequency of the

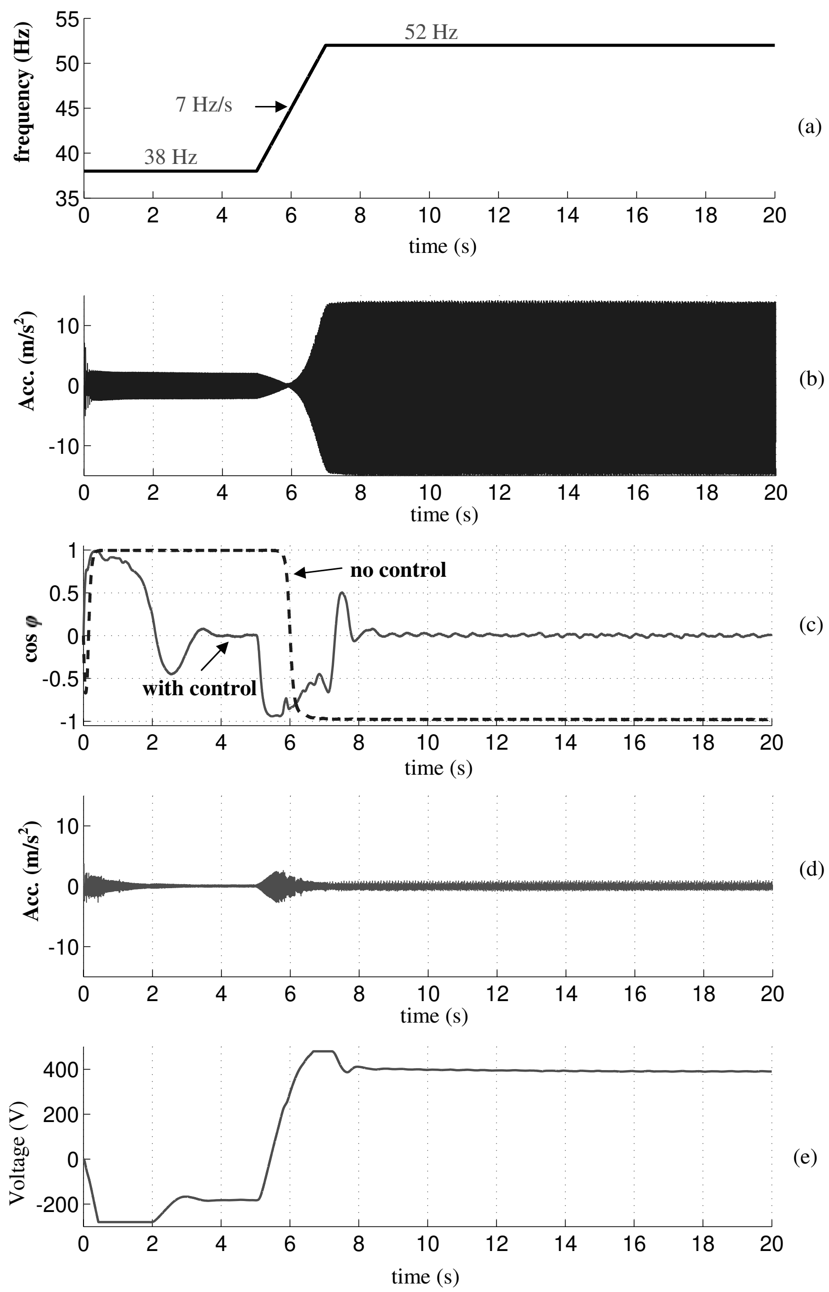


Fig. 16. Experimental results for variable frequency harmonic excitation. (a) frequency variation; (b) vibration $\ddot{x}$ without control, piezo-actuators at 0 V; (c) cosine of ϕ; (d) controlled vibration $\ddot{x}$; (e) variation of controlled voltage applied to piezo-actuators; for control results $P = 4 \times 10^{-2}$, $D = 8 \times 10^{-3}$.

ATVA. The cosine of the phase angle is depicted in Fig. 16(c). It can be seen that when the controller is switched off, this is zero when the vibration of the host structure is at its minimum at about 6 seconds. Figure 16(d) shows the vibration of the host structure when the controller is switched on and subject to the frequency sweep depicted in Fig. 16(a). It can be seen that the ATVA is effective at maintaining a low level of vibration of the host structure during and after the frequency sweep. The control activity and effort to achieve this can be seen in Figs 16(c) and (e) respectively.

7. Conclusions

This paper has described some recent developments in adaptive tuned vibration absorbers (ATVAs). Most devices of this type are realised by using a variable stiffness element, and several ways of achieving this in practice have been presented. These include shape change, material property change using shape memory alloy and a tunable beam filled with magneto-rheological fluid. A way of automatically tuning the devices has also been briefly presented. There is not a single "best" way of making an ATVA. It depends upon the required frequency range, the agility (speed of reaction) and cost. The various devices described in this paper should be viewed as prototype or proof-of-concept rather than refined devices, but they have demonstrated that "smart materials" can be incorporated into ATVAs as variable stiffness elements.

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