A HYBRID CONTROL SYSTEM FOR DISTRIBUTED ACTIVE VIBRATION AND SHOCK ABSORBERS

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Abstract

The control methods used for shock or free vibration are usually different from those for forced vibration, because shock vibration can be regarded as a type of transient vibration that is different from steady-state forced vibration. In reality, however, both steady-state and transient excitations may occur in transport vehicles, thus there is a need to control both types of vibration. To show the integration of different vibration control strategies, a hybrid control system including a distributed resonant absorber and a distributed shock absorber is proposed. The hybrid system is governed by a control arbitrator that switches a set of sensors and actuators between the two active vibration absorbers according to various external excitation conditions. The effectiveness of the integrated system is shown through simulations and experiments.

Nomenclature

$\mathbf{w}(x,t)$	displacement vector of the equilibrium position of the structure
x	spatial displacement variable
t	time variable
$\mathbf{m}(x)$	mass density matrix
ζ	damping coefficient matrix of the structure
$\phi_k(x)$	the k^{th} mode shape function
λ_k	the k^{th} eigenvalue
ω_k	the k^{th} mode frequency
$u_k(t)$	the k^{th} mode amplitude of the structure
H	the Hilbert space
f_j	amplitude of the j^{th} external harmonic excitation
Ω_j	angular frequency of the j^{th} external harmonic excitation
ζ_j	damping ratio of the distributed Vibration Clamping Absorber (VCA)
ξ_j	design parameter of the Distributed Active Shock Absorber (DASA)
v_j	control output of the VCA or DASA
K_1, K_2	feedback gains of the VCA
K	feedback gain of the DASA

Introduction

Trends in manufacturing technology towards the use of lightweight materials to achieve a high strength-to-weight ratio have resulted in modern transport vehicles with low structural damping, low stiffness, and low natural frequencies. Because of these characteristics, under some types of external excitation, transport vehicles may experience severe structural vibration that can cause damage effects to the vehicle's payload. An example is the malfunction of optical or communication devices attached to aircraft wings.

As externally induced vibrations can be categorised as either free or forced, the corresponding vibration control methods can also be classified into two groups, i.e., shock (or free) vibration control and forced vibration control [1]. Free vibration can be regarded as a type of transient vibration that is caused by a shock pulse. A shock pulse is defined as an event that transmits kinetic energy into a system in a relatively short interval compared with the system's greatest natural period [2]. A natural decay of oscillatory motion usually follows a shock. Forced vibration occurs when a structure is subjected to a continuous external excitation; for example, a harmonic excitation in rotating machines. Forced vibration often develops a steady-state oscillation of the same frequency as that of the excitation. Although pure harmonic excitation is less likely to occur than periodic or other types of excitation, most of vibration problems in structures are related to resonance phenomena.

Common sources of forced excitation are unbalanced rotating machines or reciprocating machines. If the system's natural frequencies are being excited by these forces, high-amplitude resonance response may occur. Common types of shock pulse are external impacts and sudden changes of the system's state during normal operation. These may cause severe transient free vibration with long decay times that can lead to fatigue, instability, or degradation of the performance of the structure and the payload.

It is known that, in reality, both steady-state and transient excitations may occur in transport vehicles; for example, an external excitation followed by a free excitation as a result of a shutdown of the external excitation. Therefore, it is necessary for systems to have absorbers to suppress both forced and free vibrations during operation.

The control of low frequency noise and vibration in lightweight transport vehicle structures has traditionally been difficult and expensive because of the long acoustic wavelengths involved. Active noise and vibration control technology [3] provides promising solutions for suppressing both forced and free vibrations.

As the status of external excitation for shock vibration is different from that for resonant vibration, the design methodology for shock absorbers is basically different from that for resonant vibration absorbers. The dynamic motion of shock vibration depends totally on the structural natural frequencies, damping ratios and initial conditions. However, the dynamic motion of resonant vibration depends on the characteristics of both the structure and the excitation.

Current technologies usually design resonant and shock absorbers (or isolators) either separately or in a combination form, i.e., either using one absorber for forced resonant vibration and one for free vibration, or using one absorber for both types of vibration.

To improve the performance of a shock isolator, Chandra, et al. [4] considered some combinations of a shock isolator and a dynamic vibration absorber (DVA). They showed that both the steady state and transient shock responses can be reduced by using a shock isolator along with a DVA (parallel combination) or a two-stage isolator (serial combination).

To overcome the possible low frequency amplification of DVAs, Babitsky and Veprik [5] designed a bumper vibration isolator including an undamped vibration isolator and two damped bumpers. They achieved a significant attenuation of vibration transmission under shock and random excitation.

Research on vibration and shock absorbers such as mentioned above has been mainly focused on lumped-parameter resilient elements. In this paper, the traditional approach to the design of vibration and shock absorbers is extended from lumped-parameter systems to distributed continuous systems using active vibration control technology. Distributed vibration and shock absorbers can be built as integrated elements of a structure by using the so-called smart materials, such as piezoelectric materials, magnetostrictive materials, and shape memory alloys. Especially, piezoelectric materials such as Lead Zirconate Titanate (PZT) or Polyvinylidene Fluoride (PVDF) can be produced as thin films that can be bonded to the surface of structures using strong adhesive materials [6].

The principle of distributed vibration and shock absorbers is to use the smart materials, such as PZT, to transfer energy between mechanical structures and electrical sources. The vibration energy can then be dissipated or absorbed via electrical impedance. From the analogous principle of physical systems, a mechanical spring-mass-damper system is equivalent to an electrical Resistor-Inductor-Capacitor (RLC) circuit, because the differential equations describing the dynamic performances of the two physical systems have the same form. Piezoelectric materials are inherent capacitive elements, therefore, they can be implemented with RLC resonant circuits to form distributed absorbers. If a PZT is connected with a simple resistor, the circuit acts similarly to viscoelastic damping device. If the circuit consists of an inductor and resistor in series, combined with the capacitance of the PZT, the whole device creates a damped electrical resonance. The resonance can be tuned so that the piezoelectric device acts as a damped distributed DVA.

Alternatively, distributed active DVAs can be designed to imitate the presence of an inductance or impedance by using integrated electronic circuits, digital signal processing system, and smart sensors and actuators that can be embedded into the structure. In this way, the properties of distributed active DVAs can be changed and tuned on-line by embedded controllers. Due to this distinct feature, distributed active DVAs have been seen as having the potential to replace the existing physical circuits, and provide effective structural damping without using direct RLC electrical circuit implementations [7,8].

To show the integration of different vibration control strategies, a hybrid control system, including a resonant absorber and a shock absorber, is described in this paper. The hybrid system has the advantages of the resonant absorber and shock absorber but avoids their disadvantages.

The remainder of this paper is organized as follows. First, the dynamic model used for the class of continuous structures described by the generalised wave equation is introduced. Based on our previous work, the design of a distributed Vibration Clamping Absorber (VCA) [9] and a Distributed Active Shock Absorber (DASA) [10] is then presented. The numerical simulation work demonstrates that the VCA is effective for primary resonance control but not effective for free vibration control. The same results hold for the DASA; that is, it is effective for free vibration control but not effective for resonant vibration control. Therefore, a hybrid active vibration control system is proposed. The design is based on cooperative use of the VCA in combination with the DASA. The principle of the hybrid system is illustrated and the effectiveness of the integrated system is shown through experiments under different excitation conditions. Finally, some concluding remarks are given.

System Dynamic Model

Consider the class of flexible systems described by the generalised wave equation:

$$\mathbf{m}(x)\ddot{\mathbf{w}}(x,t) + 2\zeta\,\Lambda^{1/2}\dot{\mathbf{w}}(x,t) + \Lambda\mathbf{w}(x,t) = \mathbf{F}(x,t)\,, \quad (1)$$

which relates the displacement $\mathbf{w}(x,t)$ of the equilibrium position of a body, Ω , in N- dimensional space to the applied force distribution $\mathbf{F}(x,t)$. The operator Λ is a time-invariant symmetric, non-negative differential operator with a square root $\Lambda^{1/2}$, and its domain $D(\Lambda)$ is dense in the Hilbert space $H = L^2(\Omega)$. The mass density $\mathbf{m}(x)$ is a positive function of the location, x, on the body. Without changing the properties of the above system, (1) can be normalised by the change of variables $\mathbf{w}(x,t) \to \mathbf{w}(x,t)/\mathbf{m}(x)^{1/2}$. Here, for simplicity, we take $\mathbf{m}(x)=\mathbf{1}$ in (1). The non-negative matrix, ζ, is the damping coefficient of the flexible system and depends on the construction materials and methods used.

From the above condition of operator Λ, we know that its spectrum contains only isolated eigenvalues λ_k with corresponding orthogonal eigenfunctions $\phi_k(x)$ in $D(\Lambda)$, such that:

$0 \le \lambda_1 \le \lambda_2 \le \cdots \le \lambda_n$, $\Lambda\phi_k = \lambda_k\phi_k$, and $\Lambda^{1/2}\phi_k = \lambda_k^{1/2}\phi_k$.

The eigenfunction $\phi_k(x)$ is the mode shape of the flexible system, and the mode frequency is $\omega_k = \lambda_k^{1/2}$. According to the nature of Hilbert space, the solutions of (1) can be expressed as:

$$\mathbf{w}(x,t) = \sum_{k=1}^{N} u_k(t)\phi_k(x) , \tag{2}$$

where, in theory, N should be infinite. However, in practice, it is customary to assume that $\mathbf{w}(x,t)$ can be represented with good fidelity by a truncated mode expression of the form (2), where N may be large but finite. Therefore, by substituting (2) into (1), the mode amplitude satisfies:

$$\ddot{\mathbf{u}}(t) + 2\zeta\Delta^{1/2}\dot{\mathbf{u}}(t) + \Delta\mathbf{u}(t) = \mathbf{f}(t) , \tag{3}$$

where $\Delta^{1/2}$ is the $N \times N$ diagonal matrix with diagonal entries $\omega_1, \omega_2, \ldots, \omega_N$, and $\mathbf{u}(t) = [u_1(t), \cdots u_N(t)]^T$.

In the following analysis, the case of primary resonance [11] is considered and the external force is defined by L harmonic excitations. Using the scalar form of (3), any one-mode amplitude $u_k(t)$ can be shown to satisfy:

$$\ddot{u}_k(t) + 2\zeta_k\omega_k\dot{u}_k(t) + \omega_k^2 u_k(t) = \sum_{j=1}^{L} F_j \cos(\Omega_j t) , \tag{4}$$

where F_j and Ω_j are the amplitude and angular frequency of the j[th] external harmonic excitation.

Design of the VCA and DASA

In this section, the active VCA is developed for the structure described in Equation (4). The purpose of using a VCA is to absorb the vibration energy from the structure upon which external forces are imposed. To achieve this objective, a control force component can be added to (4). This control force is intended to follow the external force variations, but with opposite phase. A nonlinear feedback controller is thus designed to construct this force component. The design methodology for the active VCA is summarised below.

The structure with VCA control is described by:

$$\ddot{u}_k(t) + 2\zeta_k\omega_k\dot{u}_k(t) + \omega_k^2 u_k(t) = \sum_{j=1}^{L} (F_j \cos(\Omega_j t) + K_1\omega_k v_j^2) , \tag{5}$$

and the VCA including the dynamics of the PZT actuator is designed as:

$$\ddot{v}_j(t) + 2\zeta_j\omega_j\dot{v}_j(t) + \omega_j^2 v_j(t) = \sum_{k=1}^{N} K_2\omega_j u_k v_j , \tag{6}$$

where v_j represents one of the control outputs of the VCA, ω_j is its natural angular frequency, ζ_j is its damping ratio, and K_1 and K_2 are the feedback gains [9].

For the shock absorber, the right side of (4) becomes zero. The DASA is then designed as a first-order controller. The structure with DASA control is described by:

$$\ddot{u}_k(t) + 2\zeta_k\omega_k\dot{u}_k(t) + \omega_k^2 u_k(t) = K\omega_k^2 v_j , \tag{7}$$

and the DASA including the dynamics of the PZT actuator is described by:

$$\xi_j\dot{v}_j(t) + v_j(t) = u_k , \tag{8}$$

where v_j represents one of the responses of the DASA, ξ_j is the design parameter and K is the feedback gain [10].

Simulation Results

A simple cantilever beam system is selected as a research vehicle to implement the above VCA and DASA designs. The cantilever beam system may effectively represent a simple model for various transport-vehicle structures, such as an aircraft wing, a helicopter blade, a solar panel of a solar vehicle, etc. A schematic diagram illustrating the closed-loop active structural vibration control is shown in Figure 1. The strain gauges represent the measurement device that monitors the performance of the active control system by providing signals representative of the residual vibration in the structure. The digital controller processes the data obtained from the sensor and provides a control signal to the PZT actuators. The actuators, in turn, drive the structure in such a way that unwanted vibration caused by the excitation is attenuated. Here, a dSpace DS1104 was used as the controller with a sampling frequency of 10K and a low pass filter.

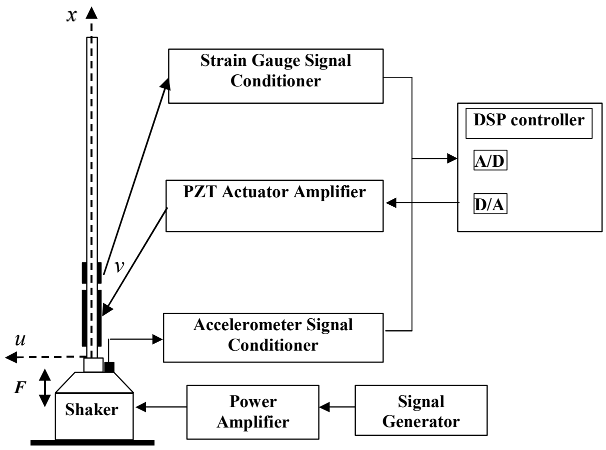

Figure 1. Schematic diagram of a distributed active vibration control system for a cantilever beam.

Suppose that only the VCA is applied on the cantilever beam that is subjected to a first harmonic excitation. Once the vibration develops into a steady-state condition at the excitation frequency of 11.4 Hz, the VCA controller is switched on at the dimensionless time 30. Figure 2 shows the structural time response for the first-mode of the beam while the damping ratio is 0.003. The first resonance has been successfully suppressed by the VCA.

In next case, suppose that the beam is subjected to a shock impulse that causes a 0.4 initial deflection of the beam. Under VCA control, the shock vibration is suppressed. However, the suppression time shown in Figure 3 is the same as that under no control. This result shows the inefficiency of the VCA when it is applied as a shock absorber.

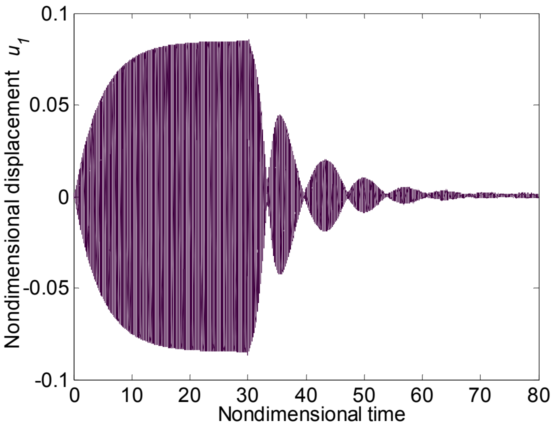

Figure 2. Numerical simulation of the first-mode forced vibration response under VCA control.

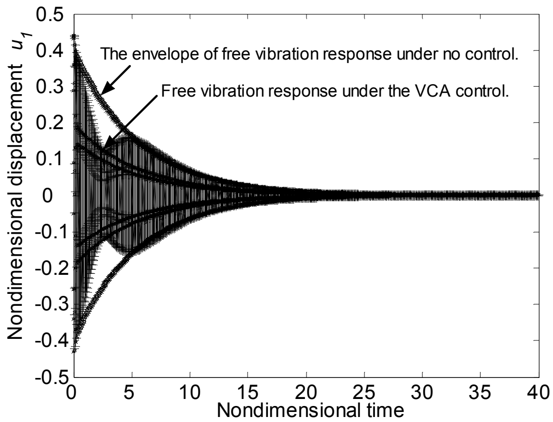

Figure 3. Numerical simulation of the first-mode shock vibration response under VCA control.

Now suppose that only the DASA is applied to the beam that is subjected to a shock impulse. The transient time response of the beam after the shock is shown in Figure 4. Compared with the response in Figure 3, it can be seen that the DASA is much more effective than the VCA under shock vibration. However, when the DASA is applied to resonant control, it is not as effective as the VCA. This can been seen by comparing Figure 2 with Figure 5, which shows the time response of the beam when only the DASA is applied under the same first harmonic excitation.

Hybrid Control System

Both resonant vibration and shock vibration can occur in actual transport-vehicle structures. Therefore, it is necessary to combine the VCA and the DASA controllers into one hybrid control system. Actually, these two absorbers can be implemented in a unified system that shares the same set of sensors, actuators, and digital control system but they can have their own individual control software.

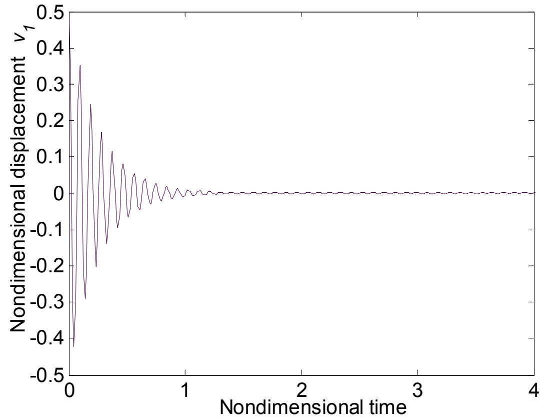

Figure 4. Numerical simulation of the first-mode shock vibration response under DASA control.

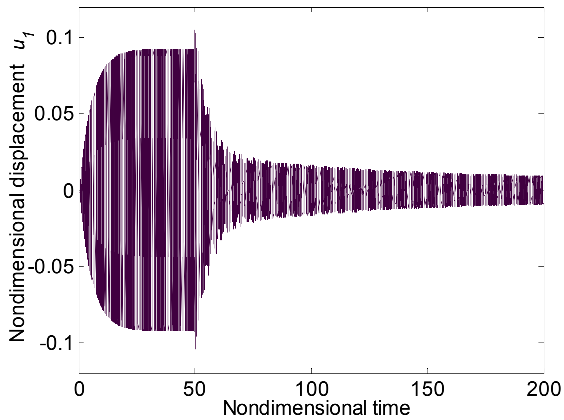

Figure 5. Numerical simulation of the first-mode forced vibration response under DASA control.

To make the VCA and the DASA work harmoniously in one system, it is necessary to design a control arbitrator to determine which controller should be put into action according to the status of external excitation. The principle of this control arbitrator is illustrated in Figure 6. The assumption for this design is that external resonant vibration and free vibration will not occur at the same time, otherwise a more sophisticated arbitrator design is needed and is a subject currently under investigation.

The control arbitrator will decide the switches' positions according to the output of the accelerometer that is attached on the head of the shaker (as shown in Figure 1). When the shaker is turned on, there is a constant amplitude cyclic force applied longitudinally to the beam. Therefore, the accelerometer, which measures longitudinal (vertical) vibration, has a non-trivial output. The control arbitrator switches the strain gauge and PZT actuator to the VCA controller. When the shaker is turned off, or after an impulse shock applied on the shaker, there is no longitudinal force applied to the beam. However, the beam will still have horizontal free vibrations. In this case, the output of accelerometer (which only measures vertical vibrations) is trivial and the control arbitrator will switch the strain gauge and PZT actuator to the DASA controller. The default position of the switches is for the DASA controller; that is, the structure is normally under free or shock vibration control.

If in the application, a primary excitation occurs in the vertical direction, the magnitude of acceleration is greater than zero and the arbitrator will select the VCA controller. Therefore, the structure is under VCA control and the DASA is disabled to output.

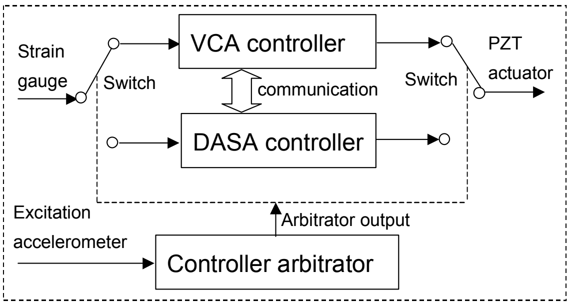

Figure 6. Schematic diagram of a hybrid control system with an arbitrator for the VCA and the DASA.

If the resonant excitation is shut down and the beam's horizontal displacement is not trivial, free vibration will take place. The arbitrator will immediately switch the strain gauge and PZT actuator to the DASA controller upon detecting a trivial output of the accelerometer which is detecting the vertical vibration. There is a synchronised communication between the VCA and the DASA to keep the output of controllers consistent so that the output is smoothly switched when the position of the switches changes.

Experimental Results

To test the design of control arbitrator, one experiment was performed as described below. To show the switching process between the VCA and the DASA, the experiment included two stages. At the first stage, only the first resonance was present and the VCA was put into action by the control arbitrator. Under VCA control, the vibration had been suppressed. To let the first resonance appear in the structure again, at the second stage, the VCA was turn off by manually resetting the control arbitrator. As the external excitation was kept constantly, the first resonance appeared again, and then the external excitation was shut down so there was a change of excitation status.

During the experiment, the cantilever beam was first excited at its first mode frequency 11.4Hz. To let the resonant vibration in the structure fully develop, the arbitrator was designed to have a 20 second delay before it turned on the VCA controller. Then the arbitrator was manually reset at the 80 second time mark while the external excitation was kept on until around the 102 second time mark. The time response of the structure is shown in Figure 7.

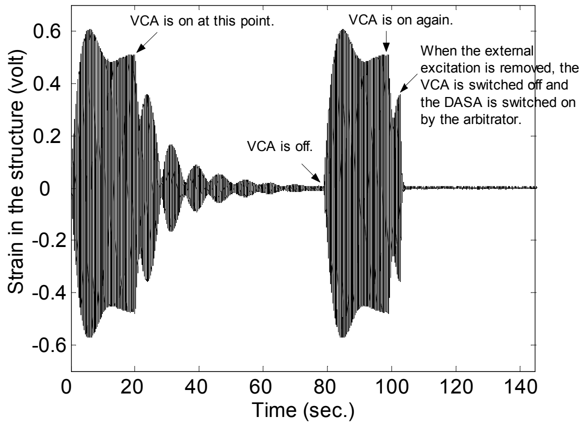

Figure 7. Experimental result for the hybrid control system under the external excitation change.

It can be seen that at the beginning, when the arbitrator detects the external excitation, the VCA is switch on after a 20 second delay. The resonance is subsequently suppressed. At the 80 second mark, as the arbitrator is reset, the VCA control output is actually turned off because there is a 20 second delay for the VCA. Therefore, the first resonance is developed again in the structure under unchanged external excitation. Then, the VCA is switched on by the arbitrator after a 20 second delay. The first resonance is therefore under VCA control. Before the first resonance was completely suppressed, at around the 102 second mark, the external excitation is removed. Upon detecting this change, the arbitrator immediately switches to the DASA. Subsequently, the free vibration is quickly suppressed by the DASA.

The enlarged part of the response in Figure 7 around this turning point is shown in Figure 8. It is shown that the beam's response is smooth during the switch between the VCA and DASA. The free vibration starts at the 102.7 second and is suppressed in less than 1 second. It would have lasted longer if the DASA were not switched on and only the VCA were left to deal with the free vibration as shown in Figure 3 earlier.

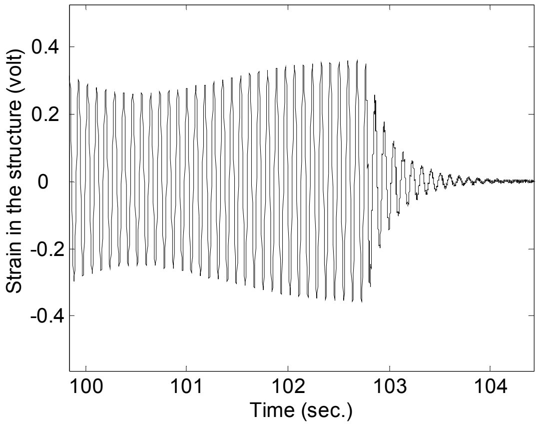

Figure 8. Enlarged show of Figure 6 around the 102 second mark.

Conclusions

In this paper, a novel hybrid control system for controlling both a resonant vibration absorber and a shock absorber used in distributed structures, including transport vehicles, is proposed. Based on our previous studies on the Vibration Clamping Absorber (VCA) and the Distributed Active Shock Absorber (DASA), the hybrid control system is constructed on the basis of cooperative use of the VCA and the DASA through a control arbitrator. The hybrid control system makes the two absorber types (VCA and DASA) share the same hardware system and increases the system's function but reduces the hardware redundancy. This is a significant advantage compared with lumped-parameter absorbers.

As shown in this paper, the VCA design aims to reduce forced vibration by exerting a controlled force equal and opposite to the external disturbing force, but it provides little additional damping to the system. The DASA design, however, aims to increase the system's damping ratio, thus it is suitable for shock attenuation but not as effective as the VCA for eliminating forced vibration. Therefore, it is natural to combine these two designs to maximise their advantages and minimize their disadvantages. The study is evident that the hybrid control system is the synergetic integration of the VCA and the DASA. Numerical simulation and experimental results show that the proposed control strategy leads to effective control of both forced resonant and shock vibration.

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