Vibration Suppression by Variable-Stiffness Members

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An active vibration suppression concept, by varying the stiffness of a new type of variable-stiffness structural member, is proposed and investigated. The characteristics of this type of variable-stiffness system is shown to be different from previously studied types. The active vibration suppression with this type of variable-stiffness member is shown to be always stable. A theoretical investigation on a single-degree-of-freedom system shows potential high efficiency for this type of variable-stiffness system due to its unique hysteretic characteristics. Two different types of control logic are proposed for realistic multi-degree-of-freedom structures with multiple variable-stiffness members. Numerical simulations demonstrate the effectiveness of the proposed strategy. Active vibration suppression experiments of truss structures are performed by using a variable-stiffness member and the proposed control logic, demonstrating the effectiveness of the proposed technique in actual structures.

Introduction

VIBRATION suppression of large space structures is a difficult and important problem because in many cases their damping is expected to be low while the shape accuracy requirement is stringent. One of the most attractive approaches for this problem is the active vibration suppression. Numerous works have been published on the subject, with various types of actuators.

For the active vibration suppression of truss structures, active truss members have been proposed and studied in several papers.[1-4] In most of these papers the active members elongate and contract, generating internal forces in the process. Particularly, the variable-length active members that are composed of piezoelectric actuators have been actively studied[1,2] because of their simplicity, light weight, and minimum consumption of source power. However, their effective application may be limited to small vibration because of their extremely small stroke.

Recently the present authors have proposed another new type of active truss members, whose axial stiffness is drastically varied by a small stroke of piezoelectric actuators. The theoretical investigation on the single-degree-of-freedom system and numerical simulations of a multi-degree-of-freedom structure have suggested the high effectiveness of the variable-stiffness members for the vibration suppression.[5]

The purpose of this paper is to confirm the effectiveness of the active vibration suppression by using a new type of variable-stiffness member in realistic structures. Two types of control logic are proposed, and their effectiveness is demonstrated in numerical simulations for multi-degree-of-freedom structures with multiple variable-stiffness members. Furthermore, a cantilevered truss beam with an active variable-stiffness member was fabricated, and experiments for vibration suppression were performed, demonstrating the effectiveness of this approach.

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Example of Variable-Stiffness Active Member

Figure 1 schematically shows the construction of a possible variable-stiffness active member. A piezoelectric actuator is installed inside the inner element of the telescopic variable-length member. When the compressively preloaded piezoelectric actuator contracts, the clamp between the outer and inner elements is released, reducing the axial stiffness of the member to zero. When the piezoelectric actuator elongates to the original size, the elements are clamped to each other, recovering the stiffness. As a result, the axial stiffness of the member can be varied stepwise by a very small on/off stroke of the piezoelectric actuator unless the force on the member exceeds the frictional force of clamped condition. In this paper the load is assumed not to exceed the limit. It should be noted that the discussion in this paper is valid not only for this type of device but also for all type II variable-stiffness structures whose characteristics are described later.

Characteristics of the Structure with Variable-Stiffness Members

It is known that vibration can be controlled by varying the stiffness of structures. Chen[6] has shown that the vibration of strings can be controlled by varying the tension. Because the lateral stiffness of strings is proportional to the tension, his work showed that the vibration can be suppressed by varying the stiffness. A single-degree-of-freedom model of the dynamics equivalent to the tension-controlled string is shown in Fig. 2a, where the stiffness $\bar{K}$ is variable. In this paper this type of variable-stiffness system is referred to as type I. On the other hand a single-degree-of-freedom model of the structural dynamics using the new type of variable-stiffness member (such as Fig. 1) is shown in Fig. 2b and will be referred to as type II. The stiffness $K\text{-}\Delta K$ in the figure stands for the stiffness of passive members in the structure. To clarify the characteristics of the type II system, both the type I and type II systems will be discussed in the following paragraphs.

Figure 3 schematically shows the load on the variable-stiffness structure of Fig. 2, f, against the displacement x. For the

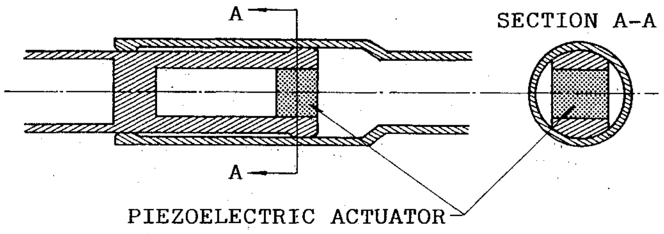

Fig. 1 Example of a variable-stiffness active member.

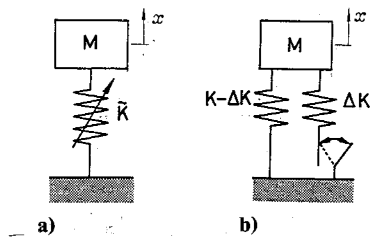

Fig. 2 Single-degree-of-freedom models of structures with variable-stiffness structural elements: a) type I; b) type II.

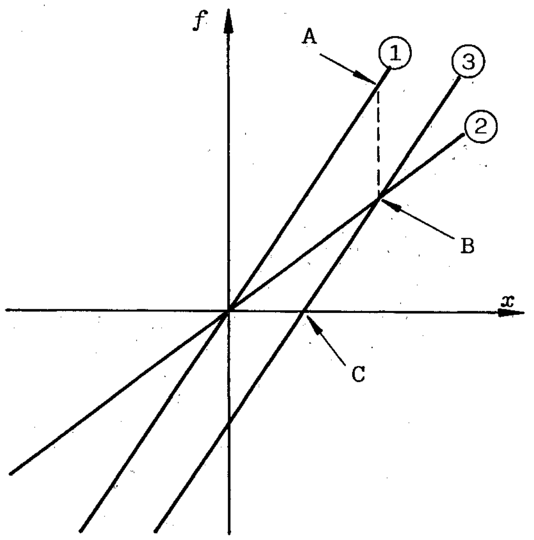

Fig. 3 Characteristics of variable-stiffness structures.

type I system the state moves along lines 1 and 2 when the stiffness is high and low, respectively. The state jumps from point A to point B when the stiffness suddenly reduces and, inversely, jumps from point B to point A as the stiffness suddenly recovers. When the state jumps from point B to A, the energy of the system increases. The equation of motion of the type I system is

$$M\ddot{x} + \tilde{K}x = f \tag{1}$$

where M is the mass, $\tilde{K}$ is the variable stiffness, x is the displacement, and f is the external force.

In the case of type II the state similarly jumps from point A to point B in Fig. 3 as the stiffness suddenly reduces, but remains at point B when the stiffness suddenly recovers and then moves along line 3. This difference plays an important role in the vibration suppression and will be discussed in later sections. It should also be noted that, unlike the case of the type I, the system will not receive any energy by the reduction/recovery action of the type II variable-stiffness member. This fact guarantees the stability of the closed-loop type II system, as will be shown later. The equation of motion of type II system with full stiffness is

$$M\ddot{x} + Kx - \Delta K x_o = f \tag{2}$$

whereas that of the reduced stiffness is

$$M\ddot{x} + (K - \Delta K)x = f \tag{3}$$

where x_o is the displacement at the moment of the final stiffness recovery, and K and ΔK are shown in Fig. 2b. The term x_o characterizes the type II system.

Vibration Suppression of Single-Degree-of-Freedom Systems

The solution of Eq. (1) for the case $f = 0$ while $\tilde{K}$ is kept constant holds:

$$x^2 + (M/\tilde{K})\,\dot{x}^2 = a^2 \tag{4}$$

where a is a constant to be determined by the initial condition. Equation (4) indicates that the solution draws an elliptic locus moving clockwise around the origin of the phase plane, which plots the velocity against the displacement. The locus is a vertical ellipse when the stiffness $\tilde{K}$ is large and a horizontal ellipse when $\tilde{K}$ is small. Therefore, in order to suppress the vibration (i.e., to make the state point close to the origin) as much as possible per cycle, the value of $\tilde{K}$ should be a maximum in the first and third quadrants and a minimum in the second and fourth quadrants of the phase plane. To comparatively study the two systems shown in Fig. 2, let us assume that the stiffness variation range of the system shown in Fig. 2a is limited to $K - \Delta K \leq \tilde{K} \leq K$. Then the vibration of the system shown in Fig. 2a can be most effectively suppressed per cycle by reducing the stiffness of type I variable-stiffness structure to $K - \Delta K$ when $x\dot{x}$ is negative and recovering it to K when $x\dot{x}$ is positive, as shown in Fig. 4a. This logic is called logic A in this paper. From an investigation of Eq. (4) and the value of $\tilde{K}$, it can be seen that the amplitude of vibration reduces by the factor of $1 - (\Delta K/K)$ per cycle by this logic.

In the case of the type II system the solution of Eq. (2) holds:

$$[x - (\Delta K/K)\,x_o]^2 + (M/K)\,\dot{x}^2 = b^2 \tag{5}$$

where b is a constant to be determined by the initial condition. In order to investigate the vibration suppression of the type II system, let us assume that the initial state is point A of Fig. 4b and that $x_o = 0$ and $\Delta K/K \geq 0.5$. If the full stiffness is kept, the state point draws a circular arc and crosses the axis of the abscissa at point B. At this moment, let us reduce the stiffness for a short duration. Because the duration is short, the state point moves slightly during it. Therefore, the state at the moment of stiffness recovery can be approximated by point B. Because $x_o = a$ after the stiffness recovery, Eq. (5) indicates that the subsequent locus is a circle whose center is point C. This circle passes through the axis of abscissa at point D, which indicates that the amplitude of vibration has been reduced by the factor of $1 - 2\,(\Delta K/K)$ in a half-cycle. From these investigations it can be seen that the vibration is suppressed by a factor of $\{1 - 2(\Delta K/K)\}^2$ per cycle by reducing the stiffness of the type II variable-stiffness structure for a short duration at the moment of $\dot{x} = 0$. This reduction rate is much better than that of logic A with the type I system. When $\Delta K/K \leq 0.5$, more effective vibration suppression can be performed by shifting the time of initial stiffness reduction to the moment when

$$x = a\sqrt{K/(2\Delta K)} \tag{6}$$

as shown in Fig. 4c. The subsequent locus is a circle that passes through the origin. If the stiffness is reduced for a short duration at the origin ($\dot{x} = 0$) according to the previously mentioned logic, the values of x, $\dot{x}$, and x_0 are approximately all zero at the moment of stiffness recovery. As a result, the vibration can be almost completely suppressed within a cycle. This type of control logic for the type II single-degree-of-freedom system is referred to as logic B. The preceding investigation suggests the potential high effectiveness of vibration suppression by the type II variable-stiffness members.

In the single-degree-of-freedom model discussed earlier, the mass of the variable-stiffness spring is assumed to be zero. Therefore, the energy stored in the spring just before the stiffness reduction appears to disappear completely at the moment of stiffness reduction. In actuality, the variable-stiffness spring has mass. The energy stored in the spring before the stiffness reduction is converted into vibration energy of the spring. In the preceding discussion it is assumed that this local vibration quickly damps during the short duration of stiffness reduction. Because the frequency of this local vibration is expected to be extremely high, the vibration can be expected to damp quickly. The friction can also be expected to contribute to the damping. Even if the local vibration does not damp completely before the stiffness recovery, it should be noted that the vibration of the main mass can be suppressed to some extent.

As previously mentioned, the reduction/recovery actions of stiffness of the type II variable-stiffness structure never supply any energy. The load on the variable-stiffness structure is a reaction of the motion of the main mass. This fact indicates that active vibration suppression by using the type II variable-stiffness structure will never make the system unstable even if the control logic is improper. This is an important advantage of the present approach.

Modal Equations of Motion of Multi-Degree-of-Freedom System with Multiple Variable-Stiffness Members

In the modal coordinates the equation of motion of structures with multiple variable-stiffness members is

$$\ddot{q} + \eta\Omega\dot{q} + \Omega q = g_o + \eta G\dot{q} + Gq + \Phi^T f \tag{7}$$

where

$$\Omega = \text{diag}\,(\omega_1^2,\ldots,\omega_m^2) \tag{8}$$

$$g_o = \sum_{j=1}^{na} s_j \Phi^T \Delta K_j \Phi q_{oj} \tag{9}$$

$$G = \sum_{j=1}^{na} (1 - s_j)\Phi^T \Delta K_j \Phi \tag{10}$$

$$\Phi = (\phi_1,\ldots,\phi_m) \tag{11}$$

$$s_j = \begin{cases} 1 & \text{when stiffness of } j\text{th member is full} \\ 0 & \text{when stiffness of } j\text{th member is low} \end{cases} \tag{12}$$

and q is the displacement vector in the modal coordinate, η is the damping matrix/stiffness matrix ratio, ω_i is the angular frequency of the ith mode of the full-stiffness system, ΔK_j is the decrement of stiffness matrix in the physical coordinates due to stiffness reduction of the jth active member, ϕ_i is the ith mode shape with full stiffness, q_{oj} is the displacement vector at the final stiffness recovery of the jth active member, and n_a is the number of active members.

The q_{oj} term, which characterizes type II active members, appears in the equation because the free length of the active member (therefore, the equilibrium point) varies according to the displacements at the moment of stiffness recovery. Elimination of this term results in the equation for the type I system. The present approach is to suppress the vibration by switching s_j.

Vibration Suppression of Multi-Degree-of-Freedom Systems with Multiple Active Members

The logic B for the single-degree-of-freedom system may be effective in suppressing a specified vibration mode of the multi-degree-of-freedom system to a limited extent. But it seems to be incapable of suppressing multiple modes because the active system can respond to only the specified single mode in logic B.

a)

b)

c)

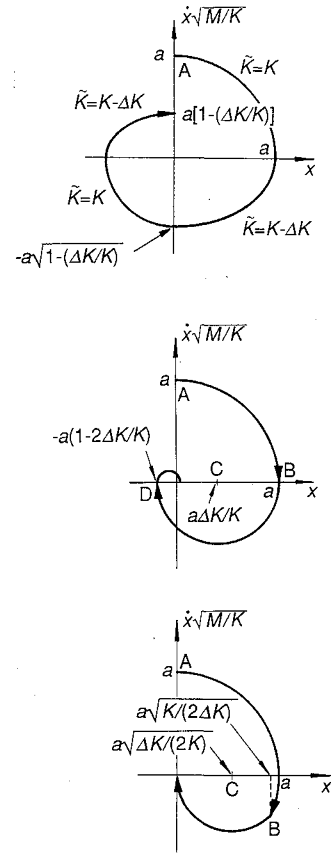

Fig. 4 Vibration suppression of single-degree-of-freedom systems: a) logic A (type I); b) logic B (type II, $\Delta K/K \leq 1/2$); c) logic B (type II, $\Delta K/K \geq 1/2$)

When the stiffness of loaded type II active elements suddenly decreases, the strain energy stored in the active members and neighboring structure is released and converted to energy for higher modes, such as axial vibration of the truss members. This load-release strategy is effective for vibration suppression since the vibration at higher modes damps sooner, and because the higher vibration modes have less effect on the shape accuracy of the structure. Higher performance can be expected if the stiffness of the active member is reduced when the load on it is at a maximum. This strategy can be implemented by decreasing the stiffness of the jth type II variable-stiffness member for the duration of t_f when the sign of $\dot{f}_{aj}f_{aj}$ changes from positive to negative if $|f_{aj}| \geq \epsilon$, and $t - t_{oj} > t_c$, where f_{aj} is the load of the jth variable-stiffness member; t_{oj} is the time of final stiffness recovery of the jth variable-stiffness member; and t_f, t_c, and ϵ are constants. The duration t_c is introduced to avoid chattering. This logic is referred to as logic C and is an extension of the proposed logic by the authors in Ref. 5 for structures with multiple variable-stiffness members. An advantage of this logic is its simplicity. Logic C releases the strain energy stored in and around each variable-stiffness member when it is at a maximum.

Another possible logic is to reduce the stiffness when it produces the maximum benefit. When the damping term and external force are neglected, the equation of motion of the ith mode of full-stiffness system is

$$q_i + \omega_i^2 q_i = g_{oi} \tag{13}$$

where g_{oi} is the ith element of vector g_o. If the values of q_i and $\dot{q}_i$ are known to be $q_{\tau i}$ and $\dot{q}_{\tau i}$, respectively, at time $t = \tau$, then the value of the q_i can be estimated from Eq. (13) as a_{oi} when $\dot{q} = 0$ for the first time, giving

$$a_{oi} = g_{oi}/\omega_i^2 + [q_{\tau i} - (g_{oi}/\omega_i^2)] \cos\omega_i\tau_{oi} + (\dot{q}_{\tau i}/\omega_i) \sin\omega_i\tau_{oi} \quad (14)$$

where

$$\tau_{oi} = \frac{1}{\omega_i} \tan^{-1} \frac{\dot{q}_{\tau i}}{\omega_i[q_{\tau i} - (g_{oi}/\omega_i^2)]} \quad (15)$$

If the stiffness of the jth variable-stiffness member is reduced at $t = \tau$ for a short duration, the equation of the subsequent motion can be approximated by

$$q_i + \omega_i^2 q_i = g_{ji} \quad (16)$$

where g_{ji} is the ith element of the vector

$$\boldsymbol{g}_j = \boldsymbol{g}_o + \Phi^T \Delta K_j \Phi \, (\boldsymbol{q}_\tau - \boldsymbol{q}_{oj}) \quad (17)$$

In Eq. (17) the displacement at the moment of stiffness recovery is approximated by $\boldsymbol{q}_\tau$, which is the displacement vector at $t = \tau$, because the duration of stiffness reduction is short. Then the value of q_i can be estimated from Eq. (16) as a_{ji} when $\dot{q}_i$ becomes zero for the first time, giving

$$a_{ji} = g_{ji}/\omega_i^2 + [q_{\tau i} - (g_{ji}/\omega_i^2)] \cos\omega_i\tau_{ji} + (\dot{q}_{\tau i}/\omega_i) \sin\omega_i\tau_{ji} \quad (18)$$

where

$$\tau_{ji} = \frac{1}{\omega_i} \tan^{-1} \frac{\dot{q}_{\tau i}}{\omega_i\, [q_{\tau i} - (g_{ji}/\omega_i^2)]} \quad (19)$$

The value of a_{ji} approximately represents the amplitude of vibration of the ith mode after the stiffness reduction/recovery action of the jth active member. Since the amplitude is

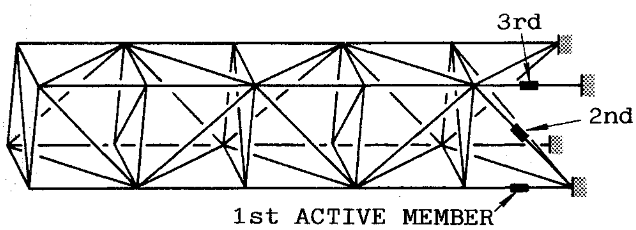

Fig. 5 Cantilevered truss beam for numerical simulation.

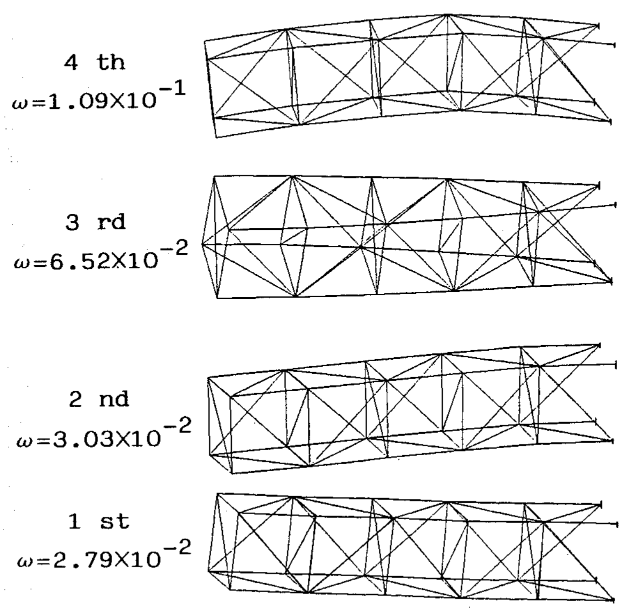

Fig. 6 Lowest four vibration modes.

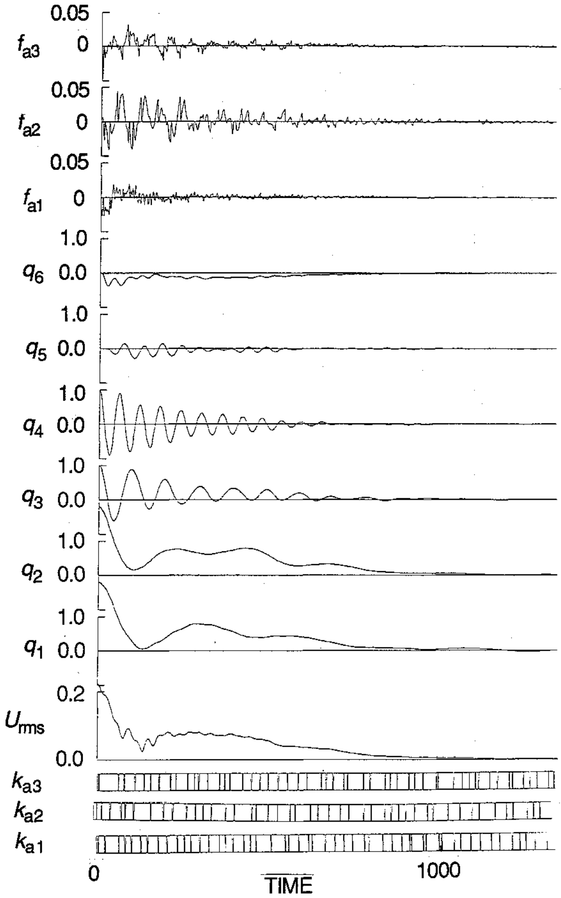

Fig. 7 Numerical simulation results of vibration suppression by logic C.

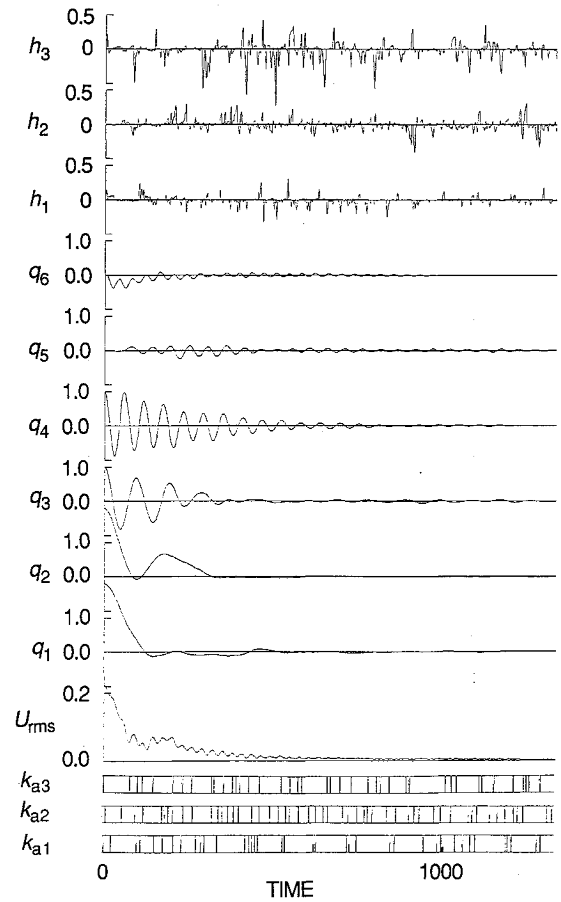

Fig. 8 Numerical simulation results of vibration suppression by logic D.

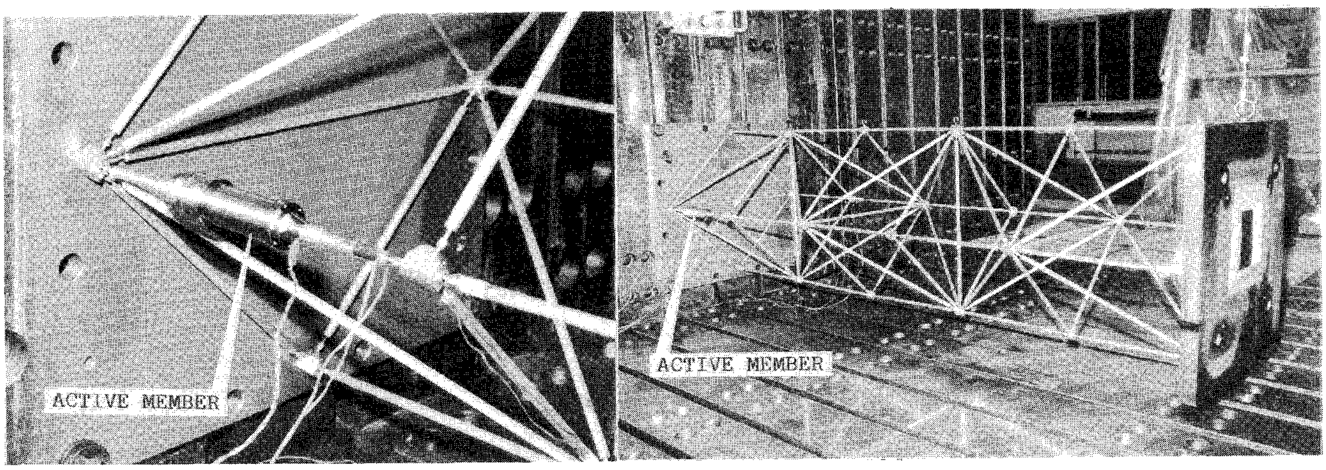

Fig. 9 Cantilevered truss beam for experiments with a variable-stiffness member.

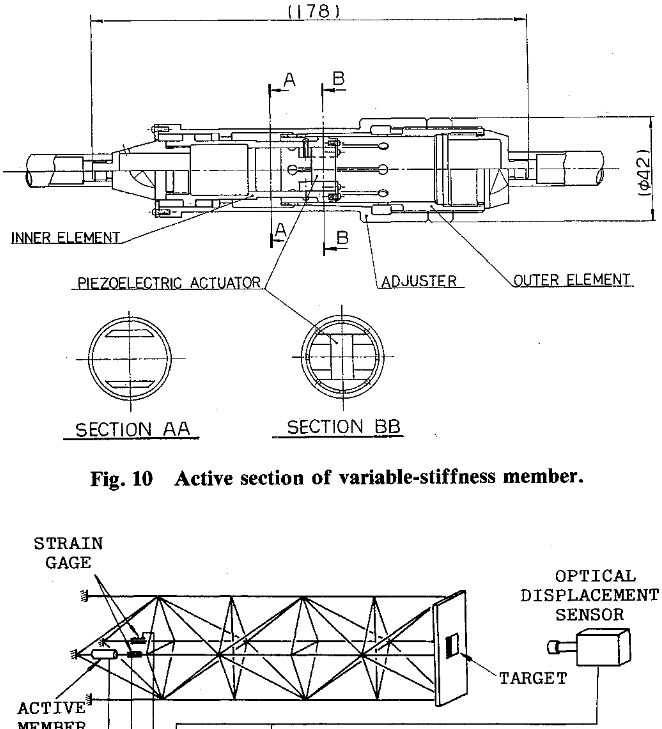

Fig. 10 Active section of variable-stiffness member.

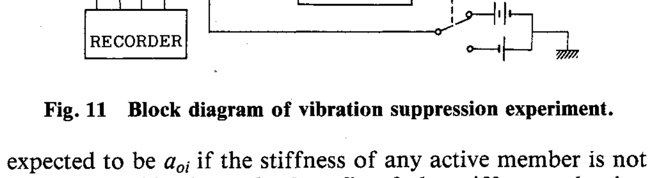

Fig. 11 Block diagram of vibration suppression experiment.

expected to be a_{oi} if the stiffness of any active member is not reduced at this time, the benefit of the stiffness-reduction action for the jth active member can be evaluated by comparing a_{ji} with a_{oi}. The present strategy for vibration suppression is to reduce the stiffness of the jth active member when its expected benefit is at a maximum. A possible implementation of this strategy is to reduce the stiffness of the jth active member for duration t_f, when the sign of $\dot{h}_j h_j$ changes from positive to negative if $h_j \geq \epsilon_j$ and $t - t_{oj} > t_c$, where

$$h_j = (\sqrt{p_o} - \sqrt{p_j})/\sqrt{p_o} \tag{20}$$

$$p_j = \sum_{i=1}^{m} w_i a_{ji}^2 \qquad (j = 0,\ldots,n_a) \tag{21}$$

and w_i is the weighting factor for the ith mode, and t_{oj} is the time of the final stiffness recovery of the jth variable-stiffness member. This control logic is referred to as logic D in this paper.

Numerical Examples

As an example, the vibration control of a three-dimensional truss shown in Fig. 5 is investigated. The length of the members are unity except for the diagonal ones. The stiffness and mass per unit length of the members are unity, and η is 1%. Three variable-stiffness members are installed so tht all of the vibration modes can be suppressed. It should be noted that the truss is stable even when the stiffness of all the active members is zero, because of the additional member in the bottom bay. Figure 6 shows the lowest four modes of the truss with full stiffness. The damping ratios of modes 1–4 are 1.4×10^{-4}, 1.5×10^{-4}, 3.3×10^{-4}, and 5.5×10^{-4}, respectively.

Figure 7 shows an example of the transient time history of the controlled structure according to logic C. In the figure, k_{aj} is the stiffness of the jth active member, and u_{rms} denotes the rms value of displacement in the physical coordinates. In the simulation all 60 vibration modes are kept in the mathematical model. The initial values are $q_1 = q_2 = 2$, $q_3 = q_4 = 1$, and all the other q are zero. The value of ϵ_j is

$$\epsilon_j = |f_{a0j}| \exp\{(t_{oj} - t)/\tau_d\} \tag{22}$$

where f_{a0j} is the value of f_{aj} at the moment of the final stiffness reduction of the jth active member. The other parameter values are $t_f = 2$, $t_c = 2$, and $\tau_d = 50$. Figure 7 shows that all of the modes damp out rapidly. The action of the active member excites (or transfer the energy to) the higher modes. However, the time history of u_{rms} (which does not show very high frequency components) suggests that the vibration of much higher modes is not large enough to substantially affect the shape accuracy.

Figure 8 shows a simulation example of the application of logic D, where $t_f = 2$, $t_c = 2$, $\epsilon_j = 0.01$, and

$$w_i = \begin{cases} 1 & \text{for} \quad i = 1,\ldots,20 \\ 0 & \text{for} \quad i = 20,\ldots,60 \end{cases} \tag{23}$$

The values of ϵ are set constant in this case because the h are normalized. The figure shows rapid damping of all of the modes.

The previously mentioned numerical examples demonstrate the effectiveness of the proposed approach for the multi-degree-of-freedom system with multiple type II variable-stiffness members.

Vibration Suppression Experiments

To verify the proposed concepts of vibration suppression by varying the stiffness of the type II variable-stiffness member,

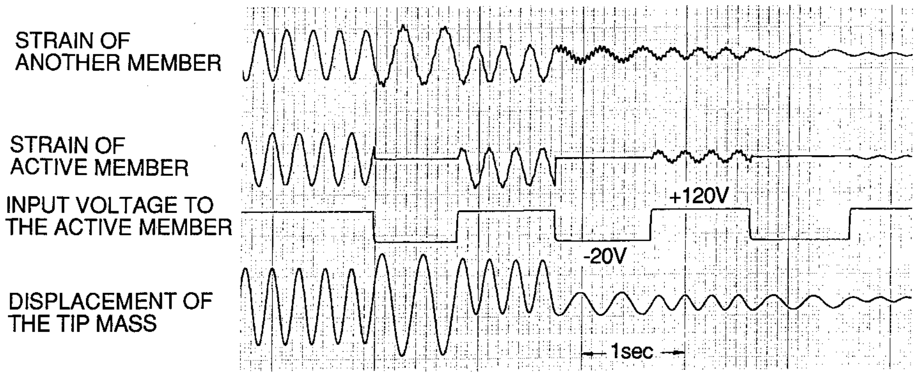

Fig. 12 Response of vibrating structure to stiffness variation.

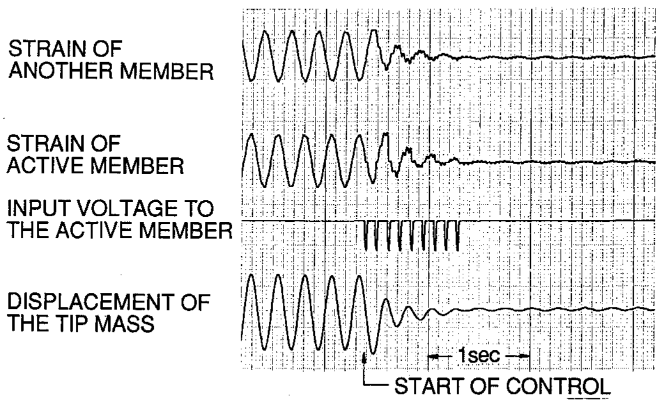

Fig. 13 Results of active vibration suppression experiment by logic B.

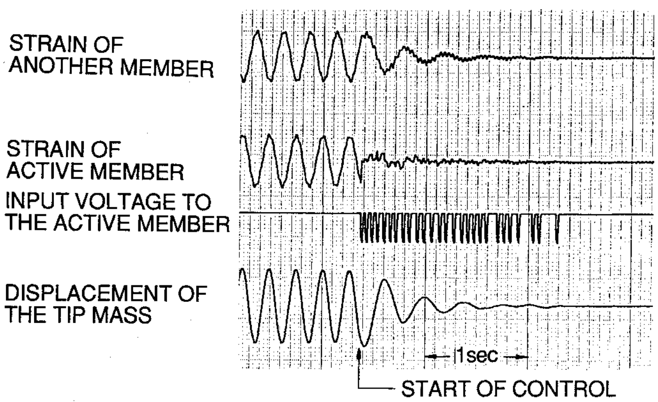

Fig. 14 Results of active vibration suppression experiment by logic C.

a cantilevered truss beam with an active member was fabricated as shown in Fig. 9, and experiments for vibration suppression were performed. All of the truss members are aluminum tubes whose outer diameter is 10 mm and with a thickness of 1 mm. The length is 330 mm except for the diagonal ones. At the tip of the beam, a mass of 100 kg is attached. Since the tip mass is hung by a wire to compensate for gravity, the motion of the tip mass is limited to the horizontal plane. Figure 10 shows the active section of the variable-stiffness member. Because the object of the experiment is to verify the proposed vibration suppression concept, serious weight saving of the members is not intended. The static load capability of the member in the locked condition is between 50 and 60 N, whereas the frictional force in the released condition is almost zero.

Two kinds of control logic were applied. Figure 11 shows the block diagram of the control loop. The input voltage to the active member was normally 120 V. The voltage was reduced to -20 V for 20 ms when $\dot{y}=0$ and $|y| > \epsilon$, reducing the stiffness of the active member, where y is the input signal to the control system. When switch A is tuned right, the input signal to the control logic is the displacement of the tip mass. Therefore, the control system approximately implements the logic B. On the other hand, when the switch is turned left, y is the strain on the active member, which is proportional to the load on the active member. Therefore, the control system implements the logic C control. In the experiment of vibration suppression the structure was excited by an initial displacement of the tip mass, and subsequently the control was started during the free vibration.

Before the active vibration suppression was performed, the stiffness of the active member was manually varied during the free vibration. Figure 12 shows some examples of time histories of tip-mass displacement, input voltage to the active member, strain of the active member, and the strain of another member. The figure shows that the first natural frequency is reduced from 3.8 Hz to 2.5 Hz by reducing the stiffness of the active member to zero (i.e., by reducing the input voltage to the active member to -20 V). While the stiffness is zero, the strain (i.e., the load) of the active member is zero. The strain time histories show that the higher vibration modes can be excited by the action of the on/off action of the active member. However, they damp out relatively rapidly. It is seen that the displacement amplitude can be increased by the reduction of the stiffness. It is also seen that the center of vibration for the displacement time history depends on the displacement at the moment when the stiffness is recovered.

Figure 13 shows the time histories of the active vibration suppression by logic B, when the displacement signal was fed back. The figure shows that the first vibration mode damped out rapidly when the active vibration control started. After nine actions of stiffness reduction, the vibration amplitude was reduced to the level of ϵ, and the control system ceased to activate the active member. The higher modes were not excited very much in this example. The first action of the stiffness reduction was triggered by the control start signal rather than a peak in the signal y. Therefore, it was not effective due to the improper timing. In this experiment the time delay of the control system due to the computational time of the processor was about 10 ms.

An example of the active vibration control by logic C is shown in Fig. 14, where the strain of the active member was fed back and the stiffness is reduced at the peak of the absolute value of the strain signal. It can be seen that the vibration is suppressed rapidly, although the number of stiffness reduction actions was larger than that of Fig. 13. It should be noted that this control logic can be implemented locally because it uses only the strain signal of each active member. This is an important advantage of this logic.

The aforementioned experiments have demonstrated that the vibration of actual truss structures can be suppressed by the stiffness control of type II variable-stiffness members.

Although only the engaging/disengaging type of variable-stiffness member is used in these experiments, the experiment suggests that any kind of type II variable-stiffness active member can be effectively used for the active vibration suppression.

Conclusions

A new approach for the active vibration suppression of a space truss structure by using variable-stiffness active members has been proposed and investigated. Because the active member do not supply any energy to the structure, the stability of the closed loop is guaranteed. Two types of control logic have been proposed for multi-degree-of-freedom structures with multiple variable-stiffness members, and their effectiveness has been demonstrated in numerical simulations. Furthermore, a cantilevered truss beam with an active variable-stiffness member was fabricated, and experiments for vibration suppression were performed, demonstrating the effectiveness of the approach. By these numerical investigations and experiments, the effectiveness of the proposed concept has been confirmed for realistic structures.

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