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Recent Developments in Japan Relevant to Structural Vibration Control

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ABSTRACT

This paper reports the recent trends in active vibration control in Japan, especially, based on papers selected in the Proceedings of First International Conference on Motion and Vibration Control (1st MOVIC) held at Yokohama, Japan on Sept.7-11, 1992. Firstly, it classifies vibration control methods and vibration controllers, especially active dynamic absorbers which are widely used in mechanical and civil engineering. Secondly, it covers basic problems in the control of vibration of flexible structures such as formulating a reduced-order model required for designing vibration controllers, proper arranging of sensors and actuators, and preventing of spillover instability. Finally, the practical use of control theories such as LQ control theory, H^{∞} control theory, neural network theory, and other topics are discussed.

INTRODUCTION

Vibration control is to prevent resonance and unstable vibration, as well as quick suppression of transient vibration by complementing the lack of internal damping from the outside or generating internally a force which cancels external forces. As conventional vibration control method, the passive type vibration control has been the main current which does not complement any energy feeding from the outside. The first active vibration control device introduced in Japan was composed of a sensor and a controller embodied in the form of a servo damper. The control method was proposed in 1970 by Tominari et al.[1] in order to prevent chattering vibration of a machine tool caused by the lack of internal damping of the machine tool structure. The servo damper was followed by a torsion servo damper devised by Nakada.[2] Then,Tanaka et al.[3] tried to establish a control system design method for a servo damper incorporating modern control theory. This method proved to be excellent for vibration-controlling performance, but was not adopted for practical application for lack of acknowledgement and reliability.

In order to maintain and enhance the capability, performance, and dwelling properties of a wide scope of engineering fields, active vibration control have recently become a key technology.[4],[5] For example, with industrial robots, lighter weight, energy saving, higher speed and higher accuracy are in increasing demand. One obstacle to achieving such properties is flexible arms, creating the need to obtain a practical method for controlling vibration.

Although lighter weight bodies are needed to increase of energy saving in automobiles, it is also necessary to control the vibration of elastic automobile bodies with the lighter weight. Increasingly lighter and taller buildings have been constructed to meet demand for lower cost. Such buildings involve the problem of swaying caused by strong winds. This can easily be verified by the appearance of the papers in the past ten years pertaining to the passive and active vibration control methods which have been contained in the Transactions of the Japanese Society of Mechanical Engineers (Series-C), see Fig. 1.

The background of this activation of researches may be referred to the fact that the modern control theory considered before hard-to-understand and not likely to be placed on practical application has become serviceable to the control system design easily owing to subsequent researches and progress of computer application techniques. The dynamic behaviors of object to be vibration controlled have become analytical in detail as well experimentally as theoretically,[6] and that introduction of an FFT analyzer has enabled easy examination of the vibration control results i.e. remarkable advance both on hardware and software, which can be itemizes as follows:

(1) Development of the software for control system design;
(2) Provision of computer with capability of high speed operation and introduction of DSP etc.;
(3) Development of new active vibration control device and actuator;
(4) Advance in the computer aided vibration analyzing method and experimental modal analyzer;
(5) Appearance of a new control system design method.

In order to exchange the most advanced ideas internationally, The First International Conference on motion and Vibration Control was held at Yokohama, Japan on Sept. 7-12, 1992. The conference Proceedings contain 191 papers received from 17 countries. The technical sessions cover many fields of motion and vibration control as listed in Table 1. Many papers with up-to-date information have gathered.

In this paper, the control devices whose design methods have been developed in Japan are presented into active ones. Current trends and complex problems of the active vibration control are also described. A procedure for making a reduced-order model for an flexible structure is presented. Finally the results from applications of newly released and recently tried control system design methods including in improved version of LQ control theory, H ∞ control theory, sliding mode control, and neural network, are presented.

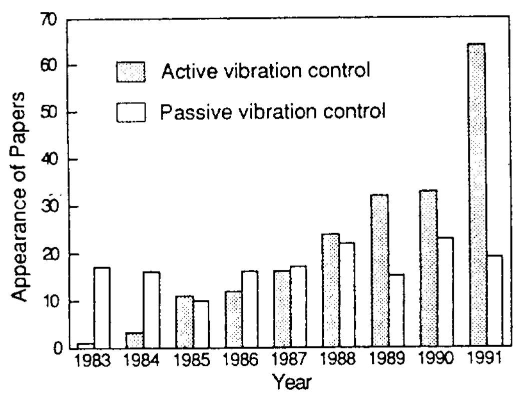


Fig. 1 Appearance of papers in past 9 years pertaining to vibration control which have been contained in the Trans. of JSME

Topics	papers
Structural control	27
Vibration control and isolation	24
Sensor and actuator	9
Motion control	10
Information instruments	4
Robotics	19
Control of aerospace structures	6
Control of rotor	4
Control of magnetic force	15
Vehicle control	16
Signal processing and devices	10
Modeling and identification	7
Application of control theory and control method	16
Control of sound	14
Control of flow-induced vibration	10

Table 1. Topics developed in 1st MOVIC and numbers of paper

ACTIVE VIBRATION CONTROLLER

Classification of Vibration Controller

The main purpose of vibration control is to stabilize an controlled object, and a high reliability is required in a vibration control device. This is the reason why a passive vibration control device has been used conventionally. They do not require energy feed and are therefore free from risk of generating unstable state. However this type of vibration control device have no sensors and can not respond to variation in parameters of the controlled object or controlling device. This has resulted in the evolution of a new type of vibration control device, which is equipped with sensors, an actuator, and a controller which exerts vibration control from externally feed energy. This is called active vibration controller.

In order to generate a vibration control force by the actuator, it is necessary to receive a reaction force with any way, since action is followed by reaction. According to the way to receive the reaction force, vibration control method is also classified[7] as follows:

(1) A method using the reaction of a fixed point
(2) A method using the reaction of auxiliary mass
(3) A method using the reaction of auxiliary structure

Figure 2 indicates a schematic view of the three methods. The controlled objects are simplified into one-degree-of-freedom system, and these state space representation by matrixes **A, B** and **c** and vector **X**.

Method (1) permits the most simple construction of a control system of three methods, provided there is fixed point near the actuators mounting point. For instance, an active vibration isolation system and an active suspension system for cars belong to this method. If there is a fixed point to mount the actuator this method is applicable. The precision vibration control technique to isolate environmental vibrations based on this method will be applicable in many fields, particularly in a processing facility for extra-precision measurement using laser device and STM[8], active suspension of vehicles[9], microgravity environments[10] etc.

However, if a fixed point is not readily available for mounting the actuator then another method will be required to receive the reaction

Method (2) uses the reaction inertia force of the auxiliary mass to produce the vibration control force in an actuator. Several types of dynamic absorber which use this method are widely used in the field of engineering, since it has the advantage of permitting the actuator to be mounted at any point. therefore, this dynamic absorber is explained in more detail in the following section.

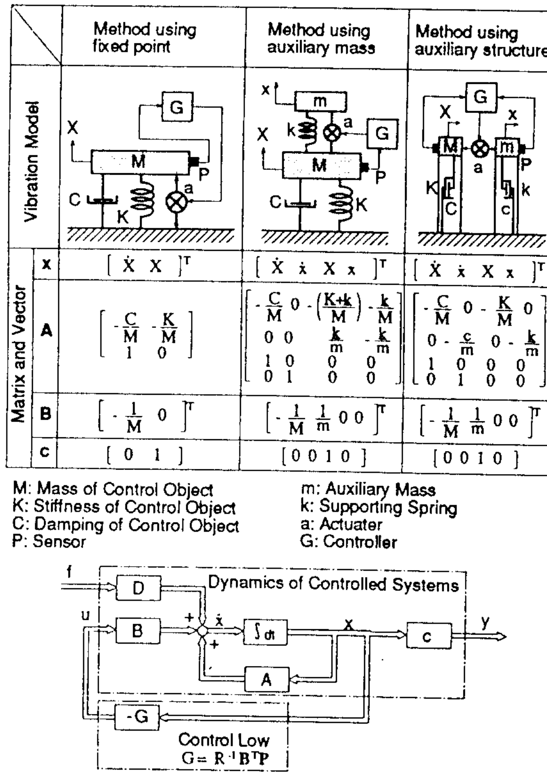


Fig. 2 Classification of vibration control method and these schematic view

Method (3) generates a control force with an actuator mounted on a structure located parallel to the flexible structure, to control the vibration of that structure. An application of this system is shown in Fig. 3.[11] The control device can be easily constructed, because it only requires insertion of an actuator between the two structures.

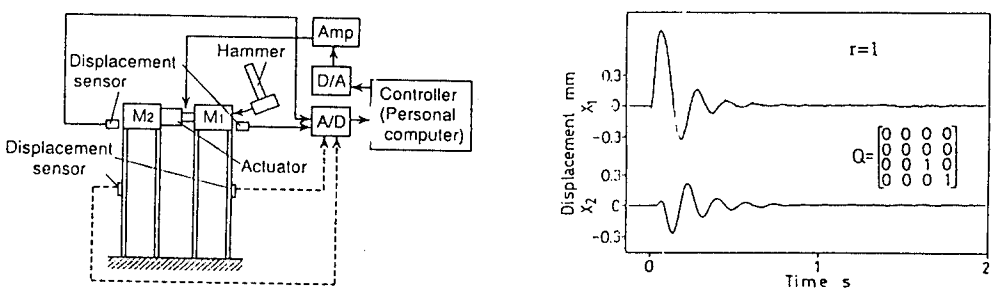


Fig.3 Active vibration control of structures arrengged in parallel and these vibration control effect

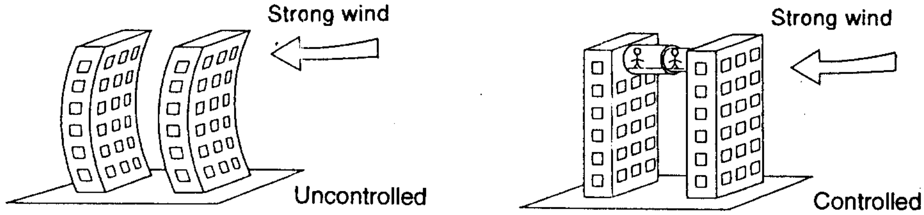


Fig. 4 Concept of the vibration control in two buildings using the method (3)

Figure 4(a) illustrates adjacent, but independently constructed multistory buildings. Although this design is architecturally preferred, it is structurally weak because of its susceptibility to strong winds. By placing an active vibration control device in the upper portion of the two buildings the vibration generated by the wind will be eliminated.

Construction of Active Dynamic Absorber

Dynamic absorbers are classified into the passive type, semi-active type, active type and hybrid type as shown in Fig. 5. The passive type is generally called a dynamic absorber, and an excellent controlling effect is obtained by keeping the condition of three elements consisting of the mass, spring and damper in optimum adjustment. Recently, the theory for controlling vibration in a multi-degree-of-freedom system using the dynamic absorber has been established[12], and its scope of application has been expanded for controlling vibration in structures[13],[14].

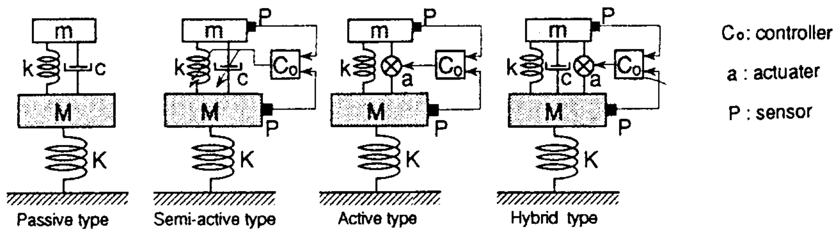


Fig.5 Classification of dynamic absorbers

However, since the dynamic absorber does not have its own sensors, it can not adapt to changes of characteristics, if any, in the control object or in itself, and it often becomes difficult to control vibration effectively. This problem is particularly serious where the mass ratio is small. There are two ways to solve this problem. One way is to change the characteristics of the spring or the damper of the passive dynamic absorber, insuring the optimum adjustment condition is maintained. This method is a semi-active dynamic absorber, although it belongs in passive type fundamentally. The other way is to use an active dynamic absorber with sensors and actuators. Advantages of the active dynamic absorber are that its vibration control effect is not only excellent and robust, but it is able to control the vibration of many-degree-of-freedom systems alone.[15],[16] The active dynamic absorber is formed by substituting the damper of the dynamic absorber with an actuator which is controlled by a controller and sensor. Because the auxiliary mass is driven by an active device, it is called an active mass.

In recent year, a number of plans to construct skyscraper have been released. Although active vibration control techniques have been actively sought as countermeasures against strong winds, some problems remain unsolved about their reliability and energy saving. A practical solution to this problem is the hybrid vibration control method, which combines both passive and active vibration control mechanisms into a single control unit. For example, the Yokohama Landmark Tower(296m tall building shown in Fig.6 opened at July 16, 1993) has adopted this hybrid method. The hybrid dynamic absorber arranges an actuator and a damper in parallel. Because vibration energy is absorbed by the damper, some reliability is ensured to a

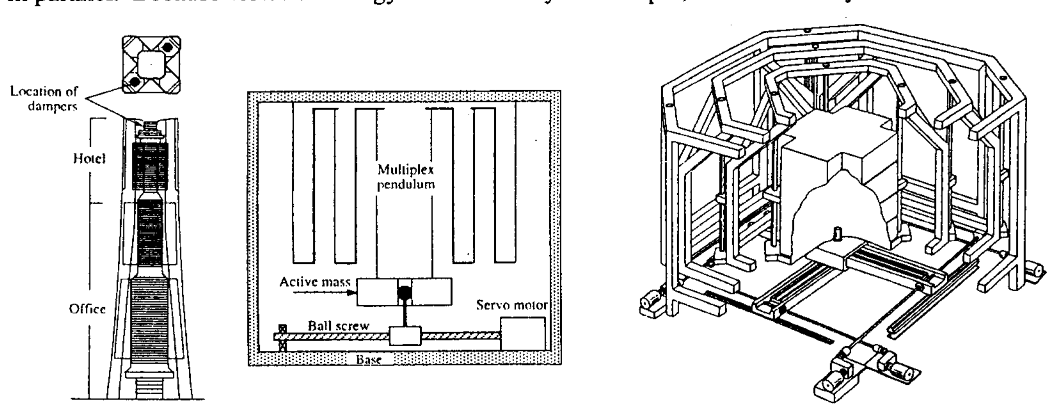


Fig. 7 Yokohama Landmark Tower and the multi-stage pendulum typetuned mass damper installed in it

certain degree even though a failure is generated by the active system. Three recent applications are presented by way of example.

Figure 7 shows the outline of the vibration control device [18] incorporated in the Yokohama Landmark Tower. A hanging pendulum is driven by a motor operating linearly and independently in two horizontal directions. It is possible to turn the natural frequency of the pendulum to that of the building. A important feature of the arrangement is a compact multi-step pendulum so as to turn the natural frequencies of the tower.˙ The counter-electromotive force produced by the rotation of the motor serves as a damper.

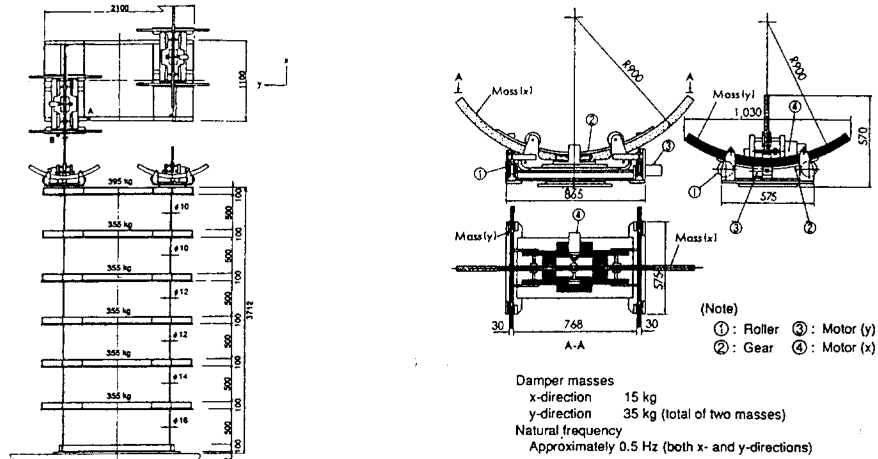


Fig. 8 Arrangement of 2-axis hybrid mass damper installed in a six-stage structure 3.7 m high

Figure 8 illustrates a 2-axis hybrid mass damper with an active mass, which is formed by cutting the arc of a circle and driven by a servo motor.[19] Because the motions of an arc of a circle have constant period, the natural frequency of the mass can be set to any value through proper selection of the radius. The application of damping in the direction of motion forms a hybrid vibration control device which consists of the motor drives and damping forces. The counter-electromotive force of the motor serves as a damper. In addition, passive operation is provided in the event of power failure.

Figure 9 shows the construction of a hybrid dynamic absorber applied to a small tower structure, a space structure, and ordinary machinery.[20] A magnetic damping force acts in a direction opposite to the motion of a conductor contained within a magnetic field. An auxiliary mass is made of a Cu conductor around which a drive coil is wired thus producing an active drive force. This auxiliary mass is called the hybrid mass. It is supported by a parallel plate spring, and a pair of such springs are provided symmetrically to the left and right. This type of hybrid absorber consists of both passive and active systems and permits independent optimal design. The effectiveness of vibration control using the hybrid dynamic absorber designed optimally by passive and active systems is demonstrated in Fig. 10.

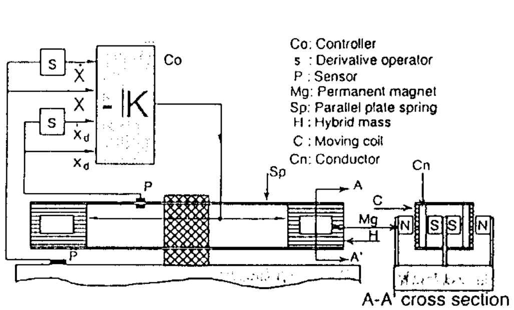


Fig.9 Hybrid dynamic absorber and its control system

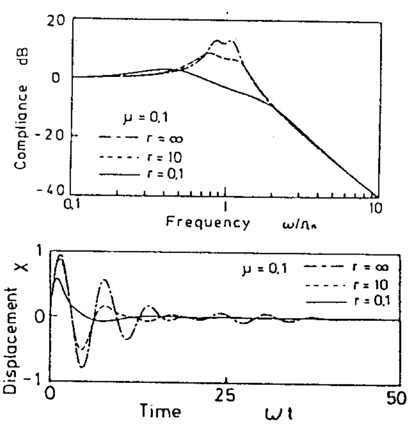


Fig. 10 Controlled frequency and time responses

CURRENT TRENDS OF ACTIVE VIBRATION CONTROL

Examples of active vibration control in structures which are of interest in recent years include:

(1) Vibration control of flexible robot manipulators.[21],[22]
(2) Active vibration control of elastic automobile bodies.[23]
(3) Swing control of high-rise buildings and bridge towers.[24]-[26]
(4) Control of flexible rotors with magnetic bearings.[27],[28]
(5) Vibration control of large space structures.[29]

There are common problems in the vibration control of flexible structure. With industrial robots, lighter weight, energy saving, higher speed and higher accuracy are in increasing demand. One obstacle to achieving such properties is flexible arms, creating the need to obtain a practical method for controlling vibration. Although lighter weight bodies are needed to increase of energy saving in automobiles, it is also necessary to control the vibration of elastic automobile bodies with the lighter weight. Increasingly lighter and taller buildings have been constructed to meet demand for lower cost. Such buildings involve the problem of swaying caused by strong winds. When active vibration control of an flexible structure is to be achieved, the following three items must be individually considered:

(1) Preparing reduced order model.
(2) Placement of sensors and actuators.
(3) Countermeasures for spillover.

Because flexible structures exhibit vibration charActeristics of a distributed parameter system, their natural frequency is of infinite order. When modern control theory is employed to control the vibration of flexible structure, it is necessary to make a reduced-order model of the structure, because of restrictions on controller design. Once a proper reduced-order model is prepared, the corresponding control system can be easily designed. An excellent software package for designing a control system is now available. However, by ignoring higher modes order reduction the model may invite vibration instability called spillover indicated by Balas.[30].

Proper arrangement of sensor and actuator is also an important subject. Figure 11 shows the relationship between the vibration mode shapes of a flexible structure, the location of sensor y and actuator u, the controllability and observability, as well as the uncontrollability and unobservability of the control system.[31] If the sensor is placed on the node of a particular vibrational mode, unobservability occurs, since this mode is not sensed by the sensor. If the actuator lies on a node, uncontrollability results. Accordingly neither the sensor nor the actuator should be located on a node of the vibration mode to be controlled. In particular, the actuator is best placed in the neighborhood of an anti-node (where the mode amplitude maximizes) of the mode in question.[32] This is because the equivalent mass of the mode in question minimizes at its anti-node, which minimizes in turn the energy required to active control. effectively.

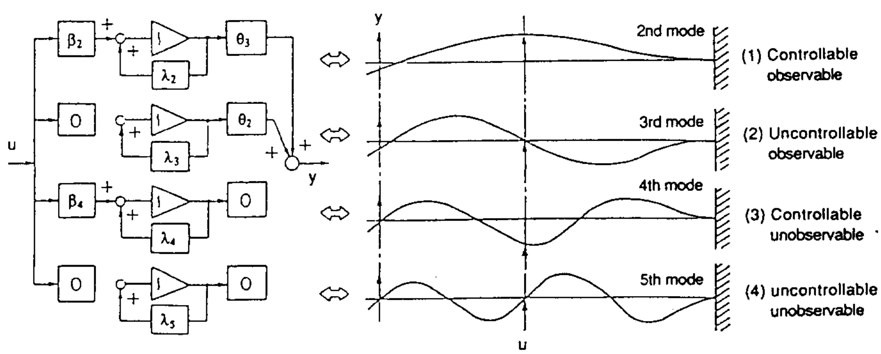


Fig. 11 Relationship between controllability/observability and locations of sensors and actuators

Lastly, spillover countermeasures are closely related to the preparation of a reduced-order model and the location of the sensor and actuator. Spillover can be prevented structurally by making a reduced-order model with sensors and actuators located at the nodes of a higher order modes to be ignored through active utilization of the uncontrollability and unobservability. Recent research has introduced filters for reducing the influence of the residual higher order modes, $H\infty$ control theory for suppressing the influence of the residual higher modes, and the modified feedback control law for taking into account the higher mode affection. In addition, there is a method with which a structure less sensitive to the control system is provided by giving a damping to the ignored higher order mode.

Figure 12 is a schematic diagram of this concept, where B_1 and $G_1(s)$ are transfer functions of the control object with reduced-order, $H_1(s)$ is the state feedback control law for the reduced-order control object, and $G_2(s)$ is the transfer function of the residual higher order modes. Figure 12(a) shows a method for preventing spillover structurally by eliminating influence of the residual higher order modes[33]. Fig. 12(b) indicates a method to accomplish the prevention through introduction of a low pass or notch filter.[34] The $H\infty$ control theory belong to this category of filters because it provides a means by which the influence of high frequency ranges upon the control system are reduced.[35] In Fig. 12(c) the control law delivered from the transfer function Hc(s) correcting the influence of higher order modes is introduced.[36]

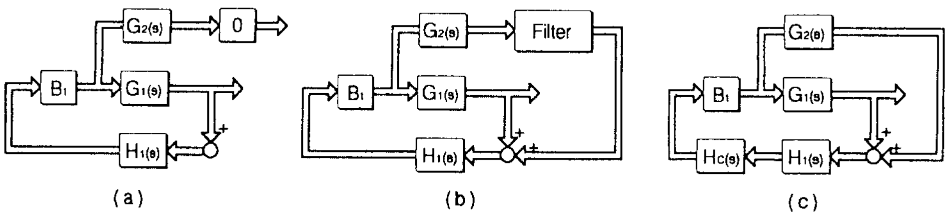


Fig. 12 Schematic diagram of the concept for preventing spillover

A METHOD FOR MAKING REDUCED-ORDER MODEL [33]

To control the vibration of a flexible structure by using modern control theory, it is necessary to make a reduced-order model of the structure and represent it in a state equation. As applicable methods for modeling complex structures, there will be finite element and transfer matrix methods. They divide the structure into many sections or elements and make models of them. They can be useful modeling techniques when combined with a modal analysis method.

The authors presented a method for making a reduced-order model with lumped masses at designated points of a flexible structure.[33] This method is based on the occurrence of uncontrollability and unobservability realized at the nodes of the vibration modes of structures. After determining the strategically selected points at the nodes of the lowest vibration mode to be neglected for making reduced-order model, the lumped mass model is constructed at these points. An advantage of this method is the construction of a state feedback control system using feedback signals from sensors in the absence of an observer. This is possible because the control value is determined by signal received directly from sensors located at the lumped mass points.

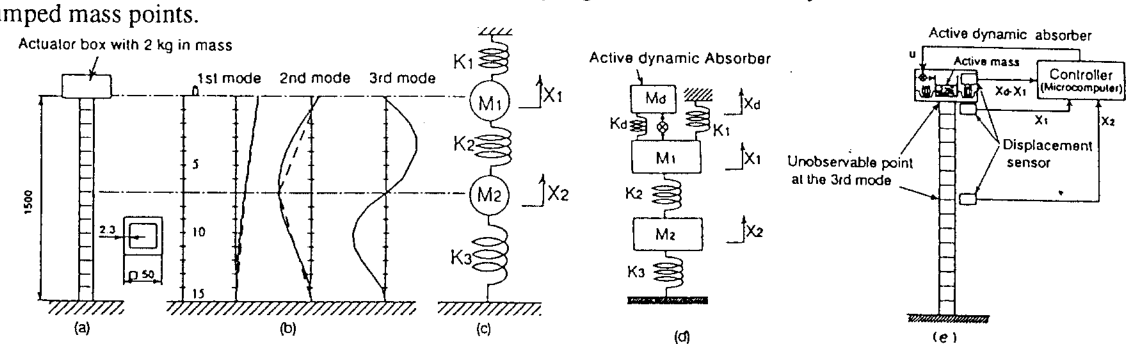


Fig. 13 Modeling procedure into a reduced-order system and vibration control system; (a) Tower-like structure to be vibration controlled, (b) Vibration mode shapes, (c) Lumped mass model represented by 2 DOF model, (d) 2 DOF system with an active dynamic absorber, (e) Control system construction.

The effectiveness of this method is demonstrated by applying vibration control to a flexible tower-like structure. Figure 13(a) illustrates the shape of the tower-like structure. Fig. 13(b) indicates the first three vibrational mode shapes of this structure by a solid line. In order to make a two degree-of-freedom system, two nodes of the 3rd vibration mode are selected as the lumped mass points. Figure 13(c) exhibits the reduced order lumped model represented by the two degrees-of-freedom system with determined physical values. Because the sensor location is identical to that of the mass points, the sensor signal and the variable of state feedback are in good agreement. That is, the sensor signal can be state fed back directly. This type of model, in which the state variable and the actual physical amount are in well correspondence, is called a physical model. No observer is required for this model, and response lags generated by an observer after external turbulence are eliminated. This is an important matter in the vibration control when a random exciting force acts on structures.

CONTROL THEORY AND ITS APPLICATION IN STRUCTURAL CONTROL

Theory for Active Vibration Control

There are several control theories which should be mentioned: classic control theory completed in 1950's, modern control theory which arose early in the 1960's and has attained the stage of practical application: and new wave robust control theory which focuses around $H\infty$ control theory and learning control centering on neural network.

Classical control theory represented by PID control continues to provide great utility in practical applications for conventional control systems, but does not suit the vibration control of many-degree-of-freedom systems like flexible structures, since it is difficult to deal with multi-variable systems.

Modern control theory which is based upon optimum control theory was at first very hard to apply for user because of its mathematical unintelligibility. Assisted by the progress of the computer utilization techniques, it has become widely used in the field of vibration control. Optimum control theory deals with linear state-variable feedback control system. It is emphasized that state-variable feedback is the most effective way to maintain a desired process balance by achieving desired closed-loop-system eigenvalues. In this theory, the state feedback gain is determined to minimize the performance index which is a weighted sum of squared errors and controlling inputs. Because the performance index is expressed in a linear quadratic form, this is called LQ control theory. In order to obtain stabilizing system, However, the LQ control theory has a weak point not to be robust against a controlled system with a neglected residual higher modes. To solve the problem, a robust control theory represented by $H\infty$ control arose in the 1980's.

Optimum Control Theory

To design the vibration control system, mainly the LQ or LQG control theory is employed. In the background, the development of the computer usage technology for realizing the LQ theory can not be ignored. In particular, it is very significant that the state feedback control law can be easily solved by a personal computer. To control the vibration of structures, the state equation is expressed as follows;

$$\dot{X} = A \cdot X + b \cdot u + d \cdot v \tag{1}$$

where X, u, v are state variable vector, control value, distarbance variable, and A, b, d are coefficient matrixes of the control object, respectively. According to the LQ control theory, the control value u is determinded by

$$u = -r^{-1} b^T P \cdot X = -K \cdot X \tag{2}$$

where P is the solution of Riccati equation.

$$PA + A^T P - Pbr^{-1} b^T P + Q = 0 \tag{3}$$

The design parameters are the weighting matrix Q and the weighting factor r represented by the following quadratic performance index J.

$$J = \int_0^\infty [X^T Q X + ru^2]\, dt \tag{4}$$

When the above control value u is obtained, the performance index is minimized. The state feedback gain vector K can be determined automatically using a package program which solves the Riccati equation. Even if the control system is unstable, the state feedback ensures stabilization regarding to the control system epressed by Eq.(1).

Points to be considered when the LQ control theory is applied are as follows:

(1) Making exactly mathematical model of the object to be controlled.
(2) Locating the positions of sensors which allow state feedback.
(3) Giving significance to the weighting factor as a design parameter.
(4) Modifying the feedback control law when the residual higher modes influence.

To achive points (1) and (2), a method is described by preparing a lumped physical model in the previous section. Point (3) is yet unsolved, and the current situation is that the weighting factor is determined through simulation on the cut-and-try basis. To solve the problem presented by points(3), a process has been developed by which the influence of the design parameters on control effectiveness can be measured. The following procedure is employed[37]:

(1) Express the state equation as a non-dimensional system.
(2) Determine the similarity law which serves as a bridge between the non-dimensional and dimensional systems.
(3) Clarify the role of the weighting factors Q and r by the non-dimensional system.
(4) Determine the weighting factors of the dimensional system using the similarity law,

The active dynamic absorber indicated in Fig. 14 was designed by this procedure[37]. One example of the time responses under a random exciting force is demonstrated in this figure.

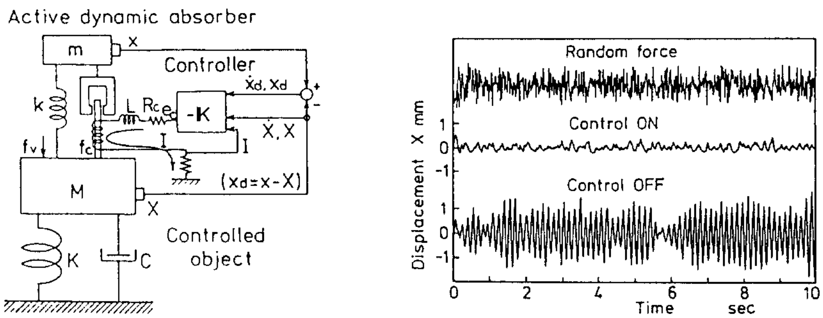


Fig.14 Control system of an active dynamic absorber and its vibration control result

It is necessary to develop reduced-order control method where the spillover caused by the residual higher modes should be conquered. To achieve point (4), Yoshida[29] proposed a spillover considered optimum control method using the minimum norm method and a robust control method using a filter,. According to the former method, the state feedback gain vector K_c which takes account of the residual high order modes is represented by Eq.(5) using the optimum gain vector K indicated in Eq(2) and the matrix M_c which expresses the restriction of control mechanisms.

$$K_c = KM_c^T(M_cM_c^T) \tag{5}$$

Then, the control value is as follows,

$$u = -K_c^T M_c X \tag{6}$$

With attention focused upon the fact that instability caused by spillover occues in high order modes, control which uses a low-pass filter to eliminate high order elements from control value is performed. Based on the method, the feedback gain vector K_e and control value u are found as Eqs. (7) and (8).

$$K_e = r^{-1}b_e^TP \tag{7}$$

$$u = -K_e^TX_e \tag{8}$$

where X_e is the state vector which includes the state vector of the low-pass filter. In order to confirm the usefulness of the present control methods, experiments using a 8-story model structure shown in Fig. 15 were carried out for the 4th order reduced control[38]. Figure 16 compares the test results of frequency response characteristics in no control and displacement criterion controlled cases together with the theoretical ones. It reveals that even when there is a restriction of control mechanism, the active dynamic absorber effectively controls multi-mode vibrations. It confirms that taking account of spillover provides relatively large damping effects up to high order modes, and that including a filter obtains even larger stable damping effect at the vibration control target mode.

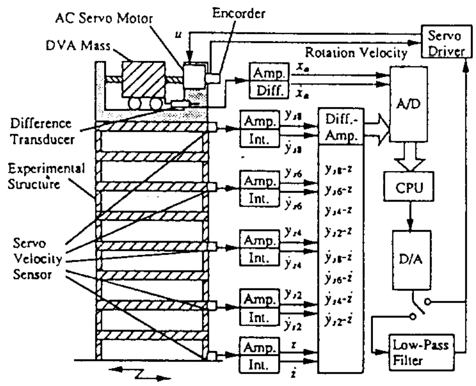


Fig. 15 8-story model structure

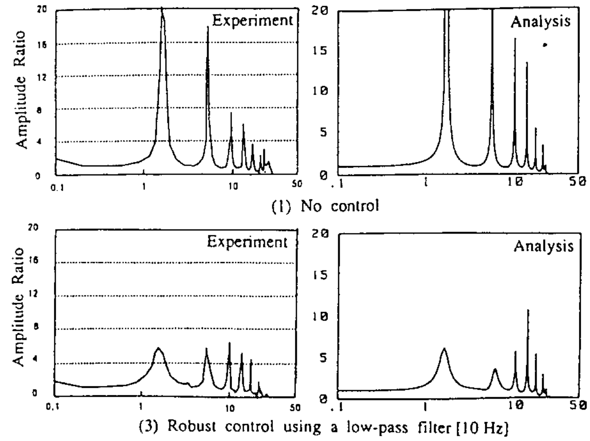


Fig. 16 Compared frequency responses

H$^\infty$ Control Theory [39]

The H$^\infty$ optimal control is the control method that a cost function defined in frequency domain is getting to the smallest. This cost function is called as written as H$^\infty$ norm and written as follows;

$$\|G(s)\|_\infty = sup\ \bar{\sigma}\{G(j\omega)\} \tag{9}$$

where the $\bar{\sigma}\{G(j\omega)\}$ is the singular value of the $G(j\omega)$, and that is the root of the greatest eigenvalues of $G^*(j\omega)G(j\omega)$, ($G^*(j\omega)$ is the complex conjugate transfer function of $G(j\omega)$). The design specification of the control system is to design the dynamic compensator $H(s)$ based on the reduced order model and to suppress the spillover and to control the vibration of the lower modes. This belongs to the H$^\infty$ mixed sensitivity problem.

The block diagram of the H mixed sensitivity problem is shown in Fig. 17. Where, A_r, B_r, D_r and C_r mean the reduced order control system. u is the control force, w is the disturbance, y is the actual measured value, and z_1 and z_2 are the control variables. $W_1(s)$ and $W_2(s)$ are the weighing function matrixes, and $G(s)$ is called as augmented plant. $H(s)$ is the H$^\infty$ controller which is given as follows;

$$H(s) = C_H(sI - A_H)^{-1}B_H \tag{10}$$

When the additive uncertainly is shown in Fig. 18, the robust stability condition is defined based on the small gain theorem as follows;

$$\|\Delta P_a(s)N(s)\|_\infty < 1 \tag{11}$$

$N(s)$ is the following transfer function from the disturbance w to the control force u,

$$N(s) = H(s)[I - P_{br}(s)H(s)]^{-1}P_{dr}(s) \tag{12}$$

where,

$$P_{br}(s) = C_r(sI - A_r)^{-1}B_r \tag{13}$$

$$P_{dr}(s) = C_r(sI - A_r)^{-1}D_r \tag{14}$$

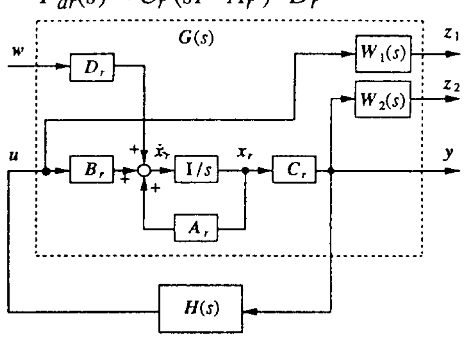


Fig. 17 Block diagram of closed loop system with H$^\infty$ Control

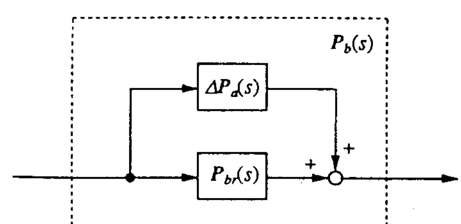


Fig. 18 System with additive uncertainty

When some selected weighting function $W_1(s)$ covers the upper limit of the uncertainty in whole frequency, we have the following expression,

$$\bar{\sigma}\{\Delta P_a(j\omega)\} \leq \bar{\sigma}\{W_1(j\omega)\} \quad (15)$$

Equation (11) becomes as follows;

$$\| W_1(s)N(s) \|_\infty < 1 \quad (16)$$

If Eq.(16) is satisfied, the closed loop system of the actual control object $Pb(s)$ can keep robust stable. As the next step, the following transfer function from the disturbance w to the measurement output y is considered,

$$M(s) = [I - P_{br}(s)H(s)]^{-1}P_{dr}(s) \quad (17)$$

$M(s)$ is called the settling function. When the gain of it is small in the lower frequency domain, the regulator with better performance can be designed. Using the weighting function $W_2(s)$ with high gain in the lower frequency domain, we can write Eq. (18).

$$\| W_2(s)\, M(s) \|_\infty < 1 \quad (18)$$

The cost function of the mixed sensitivity problem in H$^\infty$ control is as follows;

$$\left\| \begin{matrix} W_1(s)\, N(s) \\ W_2(s)\, M(s) \end{matrix} \right\|_\infty < 1 \quad (19)$$

If Eq.(19) is satisfied, Eq.(18) should be satisfied simultaneously. The controller $H(s)$ is designed based on H$^\infty$ mixed sensitivity problem.

Above mentioned design procedure of $H(s)$ will be realized by a control system design tool using MATLAB, and DSP. In particular, an interesting experiment has verified its effectiveness in the prevention of spillover. Figure 19 shows the schematic diagram of the experimental apparatus constructed to examine this effectiveness[16].

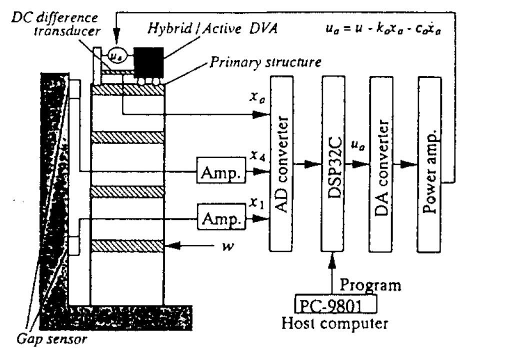


Fig. 19 Schematic diagram of experimental apparatus

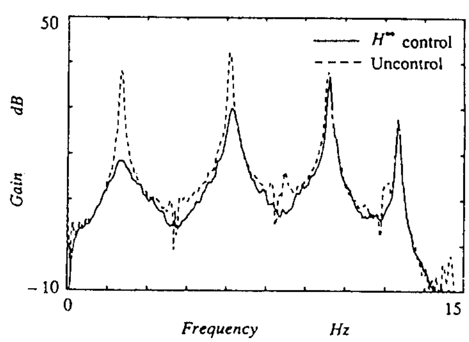


Fig. 20 Experimental results of frequency response

Figure 20 shows one experimental result which demonstrate the robustness of the H$^\infty$ controlled hybrid dynamic absorber, although it is designed using a reduced-order model with two-degrees-of-freedom. Much further progress is expected..

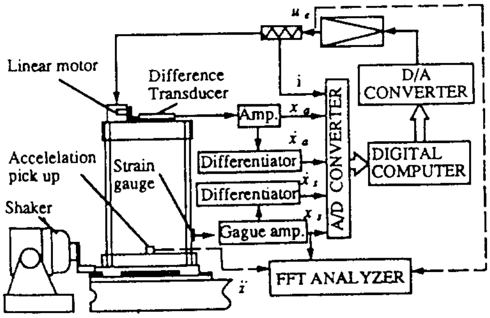
Neural Network

Recently, an active control method of a dynamic absorber is presented by applying neural network[40]. The control object is a single-degree-of-freedom system and its dynamic characteristics were identified in the neural network by using a white noise excitation. After carrying out the identification, the synthesis of controller neural network was performed by the back propagation method.And the usefulness of the method was verified experimentally.

Other Topics

Variable structure control theory, i.e. sliding mode control theory, has been researched in Russia where it was shown to be applicable not only to linear systems but also non-linear systems with time deformation.

With recent dissemination of digital control, this theory is attracting new interest. It is being applied to robot arms, with special attention paid to robust control system design. The appearance of variation, non-



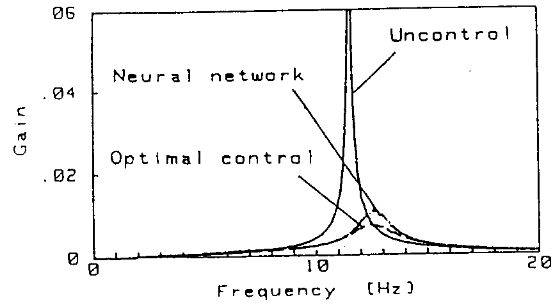


Fig. 21 Neural network controller for controlling vibration of structures and controlled results

linear terms, and unknown parameters characteristic of the sliding condition increases when the state variable is confined to a changeover surface.

In those cases where the nature of the external turbulence is known in advance or easily anticipated. a combination of feedforwad control used to offset anticipated turbulence, state feedback provides an excellent means of vibration control which can not be obtained from feedback control alone. Many studies worthy of attention are also reported in this field[41],[42].

CONCLUSION

In the past vibration control was employed solely as a post design and countermeasure. More recently vibration control devices have been included in structural designs from outset. The active control device built into the Yokohama Landmark Tower described in this paper is an important example of this trend. With this concept of designing incorporated, the active control can demonstrate its effect, and this is considered to be a very favorable trend.

This paper has outlined in some detail the general trend toward the active control of vibration. It has farther considered particular recently developed vibration control devices according to the selection of the control system employed. The idea of preparing a reduced order physical model for application in the vibration control of flexible structures was also examined. This method was shown to provide optimal performance for both the structure and the control system. Traditionally, weighting factors have been selected for control system with active dynamic absorber using LQ control theory. In this paper a process was demonstrated by which the weighting factor is optimally selected.

Recently such a problem occurs that the desired performance of controlled objects has not achieved because of obstacle of vibrations generated by motion control. One urgent area of research to be under taken is the simultaneous control of motion and vibration. Certainly, if the materials reported in this paper have provided some reference to further research in the rapidly accelerating field active vibration control, then we have achieved our purpose.

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