Active vibration control in machinery

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This paper reports the recent tendency in the active vibration control of flexible structures which are fundamental components in machinery. In the first half of the paper, it covers basic problems such as making a reduced-order model of the structure required for designing a vibration controller and its state equation representation, proper arrangements of sensors and actuators, and a method of prevention of the spillover instability occurred by ignored residual modes. In the last half, it deals with classification of vibration control methods and a vibration controller design using LQ control theory. In particular, the design method of the active dynamic absorber which is widely known in mechanical engineering and civil engineering is explained in detail.

Keywords: Vibration control, Flexible structure, Modeling, Spillover, LQ control, Active dynamic absorber

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1. INTRODUCTION

Recently, interest has been growing in the necessity for controlling vibration of flexible structures, inviting earnest practical studies in various fields of mechanical engineering, aerospace engineering, civil engineering and so on. With industrial robots, lighter weight, energy saving, higher speed and higher accuracy are in increasing demand. One obstacle to achieving such properties is elastic vibration arms, creating the need to obtain a practical method for controlling such vibrations. Although lighter weight bodies are needed to increase of energy saving in cars, it is also necessary to control the vibration of elastic car bodies with the lighter weight.

Manipulators used in outer space must handle large payloads with light-weight flexible structures and this naturally involves the problem of elastic vibration. Increasingly lighter and taller buildings have been constructed to meet demand for lower costs. Such buildings involve the problem of swaying caused by strong winds and the active control of vibration is becoming a pressing question.

Being distributed-parameter systems, flexible structures involve characteristics not found in the

active control of rigid structure. To active control the vibration of a flexible structure, consideration must be given to the following:

- (1) Approximate modeling
- (2) Arrangement of actuators and sensors
- (3) Problem of spillover instability

Problem (1) arises from reducing the order, due to restrictions on control system design, of the controlled objects having infinite-order vibration characteristics. If the approximate model is not suitable, no control system design is possible. Also, order reduction may invite the problem of unstable vibration control related to problem (3). Thus, the quality of modeling is most essential in solving problems of active vibration control.

Problem (2) is related to controllability and observability in modern control theory. The facility of control is determined by the location of actuators and sensors. Also, modeling is done in accordance with the locations of such units. Thus, the selection of locations is important.

A feedback controller is developed for a finite number of modes of the flexible structure and the controllability and observability conditions necessary for successful operation are displayed. If the controller is designed by a reduced-order model, the

control system is led to instabilities, called control and observation spillover,¹⁾ by the effect of the residual (uncontrolled) modes. Therefore, how to prevent the effect of the residual modes is important for actively controlling vibration in the flexible structures.

This paper deals with the active control of vibration of flexible structures which are fundamental components under many fields of engineering. It covers basic problems such as making a reduced-order modeling and state equation representation of the flexible structure, proper arrangements of sensors and actuators, a method of presentation of the spillover, classification of vibration control methods, and vibration controller design.

2. MODELING AND STATE EQUATION

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. Among the methods for modeling flexible structures, the one used most often is the mode development method.²⁾ It obtains the N -order vibration model of the bending vibration equation for a beam having a uniform cross-section. However, the method may be difficult to apply to a structure with cross-sections of different shapes.

On the other hand, as applicable methods for modeling cross-section of different shapes, 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.

Figure 1 shows one example of a frequency re-

Fig. 1 Frequency response of a flexible structure.

sponse of a displacement, when an exciting force acts on point 1 and the corresponding displacement is obtained at point 3, of a structure analyzed by the transfer matrix method. There are the first three resonance peaks in the range of 20 Hz and an approximate model should be made in a three-degree-of-freedom system. According to the modal analysis method, the response in Fig. 1 can be divided into 3 responses of a one-degree-of-freedom system as shown in Fig. 2. Thus, the transfer function between the exciting force F_3 and the displacement x_1 at point 1 can be represented by the product of the transfer function as follows:

$$\frac{X_1}{F_3} = \sum_{i=1}^3 \frac{1}{K_i} G_i(\omega) \quad (1)$$

where, $G_i(\omega)$ is nondimensional transfer function of the i -order mode, K_i is equivalent stiffness in i -order mode, and

$$G_i(\omega) = \frac{1}{1 - (\omega/\Omega_i)^2 + j2\zeta_i(\omega/\Omega_i)} \quad (2)$$

where Ω_i and ζ_i are the natural frequency and damping ratio in i -order mode. In general, the relation between force F_a acting on point a and response X_b of point b can be given by³⁾

$$\frac{X_b}{F_a}(\omega) = \sum_{i=1}^N \frac{\Phi_{ia}\Phi_{ib}}{K_i} G_i(\omega) \quad (3)$$

This can be represented in a block diagram as shown in Fig. 3. The Φ_{ia} and Φ_{ib} are eigen mode elements of row a and column b, K_i is modal stiffness in i -order mode, and N is a modal number for modeling.

On the other hand, the general form of transfer function treated in control theory can be given by

Fig. 2 Division of frequency response in Fig. 1 into three one-degree-of-freedom systems.

Fig. 3 Block diagram representing relation in Eq. (3).

$$\frac{X_b}{F_a}(s) = G(s) = \frac{b_m s^m + b_{m-1} s^{m-1} + \dots + b_1 s + b_0}{s^n + a_{n-1} s^{n-1} + \dots + a_1 s + a_0} \quad (4)$$

The transfer function obtained by modal analysis can be converted into the transfer function dealt with in control theory by relating both equations together by replacing s with $j\omega$ and $2n$ with N .

If an approximate model can be made of a flexible structure by using the transfer function $G(s)$ in Eq. (4) as a response of a 1-input 1-output system, the state equation here can be represented by

$$\dot{X} = AX + bu + dv \quad (5)$$

$$y = cX \quad (6)$$

where

$$A = \begin{bmatrix} 0 & 1 & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \\ -a_0 & -a_1 & -a_{n-2} & -a_{n-1} \end{bmatrix}$$

$$b = [0 \ 0 \ 0 \ \dots 1]^T$$

$$c = [b_0 \ b_1 \ \dots b_{m-1} \ b_m]$$

The relations between the coefficients in $G(s)$ and those of matrix A , b , and c can be represented in companion form.

3. LOCATIONS OF SENSORS AND ACTUATORS

3.1 Meanings of Controllability and Observability with Flexible Structures

By carrying out state feedback as follows:

$$u = -KX \quad (7)$$

on the assumption that all state variables are directly observable, Eq. (5) can be changed into the following state equation for a closed loop system.

$$\dot{X} = (A - bK)X \quad (8)$$

where K is state feedback gain vector. The root of Eq. (8) can be obtained by solving the following characteristic equation:

$$|sI - A + bK| = 0 \quad (9)$$

If the control system is controllable, the root of the closed loop system can be determined by determining K suitably. Thus, if controllable and observable, any unstable system can in principle be stabilized. Also, a method for proving controllability and observability using matrix A , b and c has been formulated.

In practice, however, an attempt to control the vibration of a flexible structure by using a state feedback technique will encounter practical problems such as where actuators and sensors should be located. Since the locations of actuators and sensors are related with controllability and observability respectively, it is important to understand the physical meanings of controllability and observability completely.

By using coordinate transformation matrix T for a canonical form corresponding to modal analysis, we can carry out a primary conversion.

$$x = TZ \quad (10)$$

where Z is the state vector of the canonical form. Putting this into Eqs. (5) and (6), we have the following state equations:

$$\dot{Z} = \tilde{A}Z + \tilde{b}u \quad (11)$$

$$y = \tilde{c}Z \quad (12)$$

where $\tilde{A}$, $\tilde{b}$ and $\tilde{c}$ are represented by the following canonical form:

$$\tilde{A} = T^{-1}AT = \begin{bmatrix} \lambda_1 & 0 \\ & \lambda_j \\ 0 & \lambda_n \end{bmatrix}$$

Fig. 4 Canonical-form state equations represented in block diagram.

Fig. 5 Relation between controllability/observability and locations of sensor and actuator.

$$\tilde{\mathbf{b}} = \mathbf{T}^{-1}\mathbf{b} = \{\beta_1 \ \beta_2 \dots \beta_n\}^T$$

$$\tilde{\mathbf{c}} = \mathbf{c}\mathbf{T} = \{\theta_1 \ \theta_2 \dots \theta_n\}$$

where λ_i is an eigen value. This canonical state equation can be represented by a block diagram as shown in Fig. 4. A comparison of this with Fig. 3 proves that eigen mode elements ϕ_{ia} and ϕ_{ib} in modal analysis correspond to β_i and θ_i in canonical form and that the transfer function in i -order mode corresponds to the i -order closed loop system based on eigen value λ_i . According to modern control theory, the system is controllable if $\beta_i \neq 0$ or observable if $\theta_i \neq 0$ and $\beta_i = 0$ and $\theta_i = 0$ are signs of uncontrollability and unobservability respectively.

Figure 5 shows cases: (1) controllable and observable, (2) uncontrollable and observable, (3) controllable and unobservable and (4) uncontrollable and unobservable together with corresponding vibration modes.⁴⁾ The u denotes the location of an actuator and y denotes that of a sensor. Thus, the mode with u on a node of vibration is uncontrollable. In a mode with y on a node, no movement can be detected (unobservable).

Now, it is understood that the meanings of controllability and observability are able to explain clearly the vibration modes and that when selecting location of actuators and sensors, vibration modes must be examined.

3.2 Prevention of Spillover by Utilizing Nodes of Vibration Modes

As described in introduction, spillover is the phenomenon of a prank played on the control system by the residual system excluded when the

Fig. 6 Control and observation spillover.

order of the model of the flexible structure is reduced to design a control system. Figure 6 shows a spillover structure in a block diagram drawn by Balas. This is shown clearly that the sensor outputs are contaminated by the residual modes (denote A_R , B_R and C_R) through the term $C_R u_R$, which called observation spillover, and the feedback control excites the residual modes through the term $B_R f(t)$, which called control spillover.

To prevent spillover instability, such control system design methods as a modal filter system, IMSC⁵⁾ and DVFC⁶⁾ have been presented. Here we present a practical preventive method utilizing nodes of the vibration modes. Utilizing the phenomenon of uncontrollability and unobservability occurring at nodes of vibration modes it places sensors and actuators at nodes of excluded vibration modes. As shown in Fig. 5, if a sensor is located at

Fig. 7 A method of making reduced-order model without spillover.

a node, no signals of the mode are observed and in principle there will be no observation spillovers. Also, there will be no control spillovers when the actuator is located at the node of the mode.

Figure 7 is a block diagram showing that with the order reduced by the primary mode and with a sensor and actuator located at nodes of the secondary and tertiary modes respectively, state feedback is applied only to the primary mode. By utilizing nodes in this way, ϕ_{2u} and ϕ_{3y} can be reduced to zero with the secondary mode having no effect on the primary and with control force not affecting the tertiary mode, and observation and control spillover can be prevented.⁷⁾ For active control of flexible structures, it is necessary to examine vibration mode shapes and determine suitable locations for actuators and sensors. This is also useful for making a reduced-order model of the flexible structure without spillover instability.⁷⁾

4. VIBRATION CONTROLLER

4.1 Classification of Vibration Controller

The active vibration controller is roughly classified into the hydraulic type, pneumatic type and electromagnetic type, according to the power source used in the actuator. 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 by the author⁸⁾ as follows:

- (1) A method based on the use of a fixed point
- (2) A method based on the reaction of auxiliary mass
- (3) A method based on the reaction of auxiliary structure

Figure 8 indicates a schematic view of the three

	Type using fixed point	Type using auxiliary mass	Type using supporting structure
Vibration model			
Matrix and vector	$x = \begin{Bmatrix} \dot{x} \\ x \end{Bmatrix}^T$ $A = \begin{bmatrix} -\frac{C}{M} & -\frac{K}{M} \\ 1 & 0 \end{bmatrix}$ $B = \begin{bmatrix} -\frac{1}{M} & 0 \end{bmatrix}^T$ $c = \begin{bmatrix} 0 & 1 \end{bmatrix}$	$x = \begin{Bmatrix} \dot{x} \\ \dot{x} \\ x \\ x \end{Bmatrix}^T$ $A = \begin{bmatrix} -\frac{C}{M} & 0 & -\frac{K+k}{M} & \frac{k}{M} \\ 0 & 0 & k/m & -k/m \\ 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 \end{bmatrix}$ $B = \begin{bmatrix} -\frac{1}{M} & \frac{1}{m} & 0 & 0 \end{bmatrix}^T$ $c = \begin{bmatrix} 0 & 0 & 1 & 0 \end{bmatrix}$	$x = \begin{Bmatrix} \dot{x} \\ \dot{x} \\ x \\ x \end{Bmatrix}^T$ $A = \begin{bmatrix} -\frac{C}{M} & 0 & -\frac{K}{M} & 0 \\ 0 & 0 & 0 & -k/m \\ 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 \end{bmatrix}$ $B = \begin{bmatrix} -\frac{1}{M} & \frac{1}{m} & 0 & 0 \end{bmatrix}^T$ $c = \begin{bmatrix} 0 & 0 & 1 & 0 \end{bmatrix}$

M: mass of controlled object
K: stiffness of controlled object
C: damping of controlled object
P: sensor
m: auxiliary mass
k: supporting spring
a: actuator
G: controller

Fig. 8 Classification of vibration control method and these schematic view.

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

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.

Method (3) generates a control force with an

P : sensor a : actuator Co : controller

Fig. 9 Active vibration Control which belongs to method (3).

actuator mounted on a structure located parallel to the flexible structure, to control the vibration of that structure. Two types of this method are shown in Fig. 9. The method is that both flexible structures are controlled actively at the same time with a single actuator placed between them. Therefore, auxiliary mass are not necessary so that the vibration control equipment can be simple and light, this method is becoming popular in the field of civil engineering.⁹⁾

4.2 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. 10. The passive type is generally called as 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, and hence it is widely employed. Recently the theory for controlling vibration in a multi-degree-of-freedom system using the dynamic absorber has been established,¹⁰⁾ and its scope of application has been expanded.¹¹⁾

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

P : sensor a : actuator Co : controller

Fig. 10 Classification of dynamic absorber.

Fig. 11 Application examples of hydraulic type active dynamic absorbers.

to use an active device with sensors and actuators.

Recently this type of active dynamic absorber has become more popular. Also, known as active mass damper, it will be widely employed in the vibration control of high-rise buildings. In principle, the displacements and velocities of the control object and the active mass is measured by the sensors, and its physical quantity is fed back by state variables, and therefore it is robust against characteristic changes. In order to obtain energy saving and reliability to control effect, the hybrid dynamic absorber which has combined advantages of the passive and the active types has recently been developed.^{12,13)}

The active dynamic absorbers proposed so far may be roughly classified into the hydraulic type and electromagnetic type, by the difference of the power source to be injected into the actuator. The hydraulic type active dynamic absorbers are applicable to vibration control of the control object which requires a powerful and compact vibration controller. Two application examples of this type are shown in Fig. 11. In order to eliminate self-excit-

Fig. 12 A variety of method for supporting an active mass.

Fig. 13 Construction of electromagnetic type active dynamic absorber.

ing chatter vibration occurring in machine tools, an active dynamic absorber called the servodamper shown in Fig. 11 (a) was proposed.¹⁴⁾ For reducing the impact vibration of an air-hammer, an active controller using the servodamper (b) has been developed.¹⁵⁾

Based on the driving method of the auxiliary mass, the linear drive system and motor drive system are available. The former is the method of directly moving the auxiliary mass, and the latter is to indirectly move the auxiliary mass by rotating the ball screw with servo motor.¹⁶⁾ Besides, by the supporting method of the auxiliary mass, the spring support, guide support and pendulum support are known. The schematic view of supporting method is indicated in Fig. 12. Since this auxiliary mass is actively controlled, it is also known as active mass.

The authors already proposed two types of spring-supported active dynamic absorbers linearly driven by electromagnetic force,^{17,18)} but in the present study, considering the generality, a standard actuator of active dynamic absorber as shown in Fig. 13 was composed. It consists of the active mass possessing a ring-shaped magnetic circuit, auxiliary springs for supporting it from both sides, moving coil, main body having a mounting part to the control object, and displacement sensor for detecting the relative displacement between the main body and active mass. Since the active mass is supported by the linear guide surface, this active dynamic absorber

Fig. 14 Control system of active dynamic absorber.

is a guide-supported electromagnetic linear actuator type.

Figure 14 shows the structure of an active dynamic absorber mounted on a control object of one-degree-of-freedom. This absorber comprises the drive unit and controller shown in Fig. 13. In the controller, from the absolute displacement X and relative displacement x_d of the control object detected by two displacement sensors, velocity signals, are produced by differentiating these displacements, and also control current I is detected, and these five signals are multiplied by feedback gain to produce the control value u of state feedback.

5. CONTROL SYSTEM DESIGN USING LQ CONTROL THEORY

To design the control system, mainly the LQ or LQG control theory is employed. In the background of the promotion of research and development of the active dynamic absorber, the development of the computer usage technology for realizing the LQ theory can not be ignored. In particular, it is very significant that the Riccati equation for determining the state feedback control law can be easily solved by a personal computer. This paper is accordingly intended to unveil the vibration control performance when controlled by using the LQ control theory by composing an active dynamic absorber shown in Fig. 14.

5.1 Representation of Nondimensional State Equation

Using the notations expressed in Fig. 14 and the relative displacement, the equation of motion is expressed as follow

$$\ddot{X} = -\frac{C}{M}\dot{X} - \frac{K}{M}X + \frac{k}{M}x_d + \frac{K_c}{M}u + \frac{K_v}{M}v \quad (13)$$

$$\ddot{x}_d = \frac{C}{M} \dot{X} + \frac{K}{M} X - \left(1 + \frac{m}{M}\right) \frac{k}{m} x_d - \left(\frac{1}{M} + \frac{1}{m}\right) K_c u - \frac{K_v}{M} \dot{v} \quad (14)$$

However, if the control system is designed directly in this state equation, the general property of the active dynamic absorber is lacking, and reference materials for control system design with a generality are not obtained. To solve this problem, the state equation is made dimensionless using the following quantities;

$$\mu = m/M, \quad \gamma = \omega_n/\Omega_n, \quad \zeta = C/(2\sqrt{KM}) \quad (15)$$

where $\omega_n = \sqrt{k/m}$, $\Omega_n = \sqrt{K/M}$ are natural frequencies of the dynamic absorber and control object and, respectively. When scale conversion factors are introduced,¹⁸⁾ the following nondimensional state equation with superscript is obtained,

$$\dot{\bar{X}} = \bar{A}\bar{X} + \bar{b}\bar{u} + \bar{d}\bar{v} \quad (16)$$

where;

$$\bar{X} = \{\bar{X}_1 \quad \bar{X}_2 \quad \bar{X}_3 \quad \bar{X}_4\}^T \quad (17)$$

$$\bar{A} = \begin{bmatrix} -2\zeta & 0 & -1 & \mu\gamma^2 \\ 2\zeta & 0 & 1 & -(1+\mu)\gamma^2 \\ 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 \end{bmatrix} \quad (18)$$

$$\bar{b} = [1 \quad -(1+1/\mu) \quad 0 \quad 0]^T \quad (19)$$

$$\bar{d} = [1 \quad -1 \quad 0 \quad 0]^T \quad (20)$$

The control value and state feedback gain vector are as follows.

$$\bar{u} = -\bar{K}\bar{X} \quad (21)$$

$$\bar{K} = \{\bar{K}_1 \quad \bar{K}_2 \quad \bar{K}_3 \quad \bar{K}_4\}^T \quad (22)$$

5.2 Control System Design

For control system design, the LQ control theory is employed. In this case, the design parameters are the weighting matrix Q and the weighting factor R given in the following quadratic performance index J .

$$J = \int_1^\infty [X^T Q X + u^2 R] dt \quad (23)$$

According to the LQ control theory, the control

value u for minimizing this performance index is formulated as follows.

$$u = -R^{-1}b^T P X = -K X \quad (24)$$

where P is the solution of Riccati equation.

$$P A + A^T P - P b R^{-1} b^T P + Q = 0 \quad (25)$$

The package program for solving the Riccati equation has been prepared, and the value of K is determined automatically. By this state feedback, even if the control system is unstable, it is known that the stabilization is guaranteed, where $\bar{Q}$ is a nondimensional weighting matrix of Q as follows.

$$\bar{Q} = \begin{bmatrix} q_1 & 0 & 0 & 0 \\ 0 & q_2 & 0 & 0 \\ 0 & 0 & q_3 & 0 \\ 0 & 0 & 0 & q_4 \end{bmatrix} \quad (26)$$

where,

- q_1 : nondimensional weighting factor applied to velocity of control object
- q_2 : nondimensional weighting factor applied to relative velocity
- q_3 : nondimensional weighting factor applied to displacement of control object
- q_4 : nondimensional weighting factor applied to relative displacement

5.3 Vibration Control Effect

For demonstrating the vibration control effect of the active dynamic absorber, the relationship between the frequency response and impulse response are shown in Figs. 15 to 17.

Figure 15 is a nondimensional frequency response diagram of the control object when only the dis-

Fig. 15 Nondimensional frequency response when only displacement of control object is weighted.

Fig. 16 Nondimensional frequency response when only velocity of control object is weighted.

(a)

(b)

Fig. 17 Comparison of the vibration control effect by nondimensional impulse responses when weighting parameters change.

placement of the control object is weighted, and the weighting factor R is changed.

Figure 16 shows also a nondimensional frequency responses when only the velocity of the control object is weighted. Anyway, as the weighting factor decreased, the resonance peak was suppressed, and the vibration control effect was enhanced, but the lowering of resonance peak is more obvious in Fig. 15. In this case, meanwhile, the two peaks tend to be separated to both sides, but they tend to converge into one in Fig. 16.

Figure 17 (a) is an impulse response corresponding

to Fig. 15, and (b) corresponds to Fig. 16. In both cases, along with the decrease of R , the vibration control effect was improved, but the convergence was faster in the former case. Understanding of the correspondence of these responses will be very convenient for the design of control system.

6. CONCLUSION

Now, we hope that the reader has understood that vibration control in the flexible structure is an abstruse problem requiring wide knowledge from vibration analysis to modeling, from the locations of sensors and actuators to control system design. Studies on vibration control in the flexible structures have just begun. We hope that this report will be of some help in promoting research and development in this area.

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