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A NEW SENSING TECHNOLOGY USING PIEZOELECTRIC FILM SENSORS FOR CONTROLLING VIBRATION OF MULTI-MODES IN FLEXIBLE STRUCTURES

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Abstract

This paper deals with motion and vibration control of multi-modes of a flexible transportation system. In order to reduce the settling time from one position to another one, the vibration control is remarkably important in the transportation system. Since transportation system moves long distance, no limit in the measuring range of the sensor is required. For the purpose, piezoelectric films pasted on flexible structures are used as the point sensor to detect vibrations. To control the first two vibration modes, the reduced-order physical modeling method proposed by authors [1] is applied and the control system is designed by LQI control approach. Control experiments demonstrate that the settling time is well shortening by controlling many-modes of vibration in addition to motion.

INTRODUCTION

Motion and vibration control of flexible transport systems for moving equipments and loads from one place to another are considered. In order to reduce settling time, higher speeds and higher positioning accuracy are increasingly demanded for transportation system. As well, in recent years strong demands for energy savings have also been made. To meet these demands, transport systems are constructed with a light structure, and become relatively flexible [2]. As a result, the transport system suffers a decline in its natural frequency because of its increased flexibility. Then, increasing acceleration under the operation invites vibration, mainly reducing positioning accuracy. The influence of the vibration occurred in positioning is not negligible on the lose time in settling. In addition, the transport system needs a robust control for considering up-and-down motion of the elevator with an unknown load. Although the input shaping method to suppress the vibration of flexible structures has been proposed [3], [4], it is not effective on such a time varying system.

In order to reduce the settling time, we have studied basically to control motion and many-modes of vibration for the flexible transportation system [5]. In this study, a structural filtering technique was used to suppress spillover invited by neglected higher modes. A structural modification method was also applied to avoid an uncontrolled symmetric mode called drumming mode [6]. In addition, it has been demonstrated that the robust control under changing the position of the load was obtained using LQ control [7]. However, these studies were done by the use of laser sensors with a limitation on the measuring range.

The next problem is how to measure multi-modes of vibration of the flexible tower structure instead of the laser sensor. Since transportation system moves long distance, no limit in the measuring range of the sensor is required for vibration detection. For the purpose, piezoelectric films pasted on flexible structures are used as the point sensor of vibration detection. To control the first two modes, a two DOF (degree-of-freedom) lumped mass model is constructed using the reduced-order physical modeling method. Furthermore, a mounting technique of the piezoelectric film to detect the same dynamic behaviour as with the laser sensor is shown. It is reported that the scaled transportation system is well controlled on the motion and the first two modes of vibration using two piezoelectric film sensors.

A COMPOSED TRANSPORTATION SYSTEM AND THE VIBRATION QUALITY

The transport system considered in this paper is shown in Fig.1. The transport system is composed of a transport table driven by a servomotor through a feed drive system, a flexible tower structure equipped with an elevator on the transport table and a control system with sensor and amplifier. The flexible tower structure is made of two flat plate structures with different thickness arranged in parallel and is 1000 mm in height, 180 mm in width and 1.5 mm and 2.6 mm in thickness. In order to move a carrier on the elevator from one position to another position, two driving systems are installed in the horizontal and vertical directions.

When two flat plates with the same shape and stiffness are used, the vibration mode called “drumming mode” appears. The drumming mode takes so that two flat plates vibrate symmetrically at opposite phase with the same frequency. Thus it is impossible to control this vibration mode by using only one actuator. So in this research two flat plates with different thickness are used in order to avoid the drumming mode. The flexible tower structure is moved in the transverse direction by the feed drive system consisted of a PWM amp., an AC servomotor and a feed drive mechanism. In the case of Fig.1, the vibration of the structure is measured by displacement sensors like laser sensors.

The displacement mode shapes in vibration of the flexible tower structure from 1^{st} to 4^{th} mode, analyzed by use of a ME Scoop are shown on the right hand side of Fig.1. The natural frequencies of the vibration modes are 2.6 Hz, 9.6 Hz, 16.5 Hz, and 26.3 Hz, respectively. In this paper, the vibration control of the flexible tower structure is in respect to the 1st and 2nd modes, because the higher modes have only slightly influence on the settling time. The strategic points for making a vibration control model with 2DOF are the two points of the node of the 3rd vibration mode as shown in Fig.1.

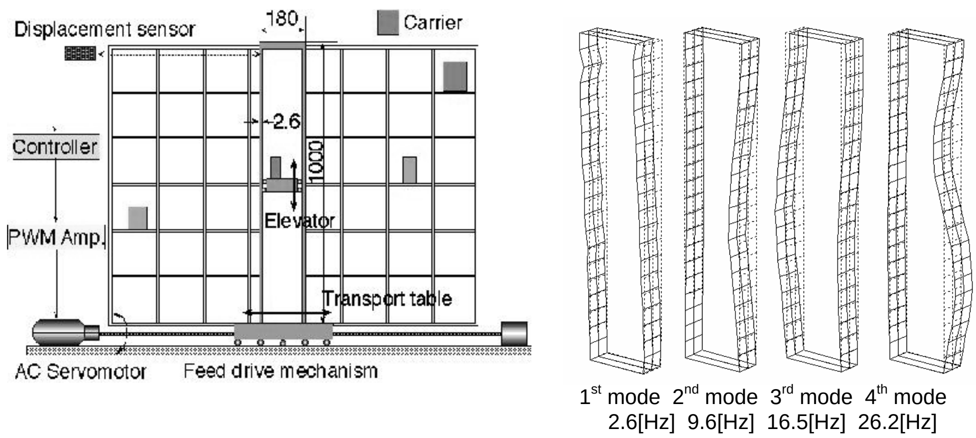


Fig.1 Schematic view of transportation system and vibration mode shapes of tower structure

CONTROLLED SYSTEM MODEL

The first part for making a control model is to construct a lumped mass model from the flexible tower structure for controlling many modes of vibration. The second part of the procedure is to make a model from the PWM amp to the transport table. The third part of the procedure is to interlace the two models to make a control model for motion and vibration control of the structure.

Reduced Order Modelling Method

For constructing the lumped mass model of the flexible tower structure, a method presented by one of authors is schematically. Generally, the mass matrix $\boldsymbol{M}$ and the stiffness matrix $\boldsymbol{K}$ on the physical domain are shown in the following equations.

$$\boldsymbol{M} = \left(\boldsymbol{\Phi}\boldsymbol{\Phi}^{\boldsymbol{T}}\right)^{-1} \tag{1}$$

$$\boldsymbol{K} = \left(\boldsymbol{\Phi}^{\boldsymbol{T}}\right)^{-1} \boldsymbol{\Omega}^2 \boldsymbol{\Phi}^{-1} \tag{2}$$

Here, $\boldsymbol{\Phi}$ is a normalized modal matrix, and $\boldsymbol{\Omega}$ is a diagonal matrix of the natural frequencies of each mode. Because a lumped mass model is not obtained at this stage, the exact value of $\boldsymbol{\Phi}$ is an unknown quantity. Therefore, a temporary normalized modal matrix is constructed from mode shapes.

$$\boldsymbol{\Phi} = \begin{bmatrix} \phi_{11} & \phi_{12} \\ \phi_{21} & \phi_{22} \end{bmatrix} \tag{3}$$

Since these elements are given from the mode shapes on the distributed parameter system, the temporary normalized modal matrix constructed from the mode shapes is not guaranteed to satisfy the equation of the mass matrix. Therefore, by using a modification vector $\delta\boldsymbol{\Phi}$ and iterating the error function, an non-diagonal elements tends toward zero. So, the expected normalized modal matrix $\boldsymbol{\Phi}$ is obtained.

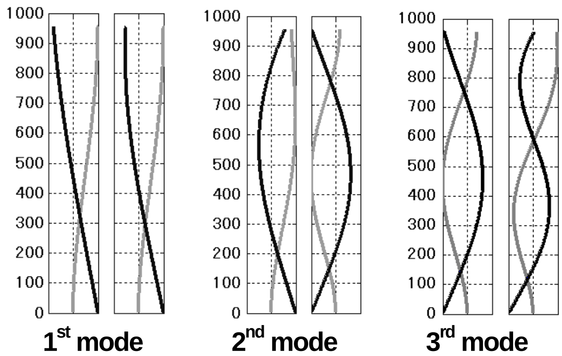
$$\boldsymbol{\Phi} + \delta\boldsymbol{\Phi} \rightarrow \boldsymbol{\Phi} \tag{4}$$

Finally, the mass matrix $\boldsymbol{M}$ and Stiffness matrix $\boldsymbol{K}$ in the physical domain are then determined.

Relationship between Displacement and Moment Mode Shapes in Vibration

In this research, piezoelectric film sensors to detect the strain at pasted points are used as a point sensor for sensing vibration. Therefore, it is necessary to investigate the displacement mode and moment mode corresponding to the strain on the flexible structures [8]. Fig.2 shows the relationship between displacement and moment modes of the left and right plates of the flexible tower structure, where the dark and thin lines indicate the moment and displacement modes, respectively. These displacement modes from 1st to 3rd are the same to them in Fig.1. It is found that nodal points of the 3rd moment modes is located at 201 mm from bottom of left plate and 156 mm of right one. If these points are selected as sensing points pasted by piezoelectric film sensors, the 3rd mode is unobservable. We call this as a structural filtering.

Since the sensing points have been determined, the next problem is to search modelling points corresponded to the sensing points. An algorithm to search them is constructed to find points on the plates having the same frequency and time responses between them. Using the algorithm, the corresponding modelling points are found at 699mm from bottom of the left plate and at 516 mm of the right plate.



2.6[Hz] 9.6[Hz] 16.5[Hz]

Fig.2 Relationship between displacement modes and moment ones

Making Reduced Order Model expressed by 2 DOF System

By the above mention procedure, sensing points indicated by black circles and corresponding modeling points indicated by white circles are determined on the 3rd vibration modes as shown in Fig. 3. Fig.4 illustrates the reduced order 2 DOF model expressed by two modeling points. These two points are called as mass 1 and mass 2, at the upper side and lower height, respectively.

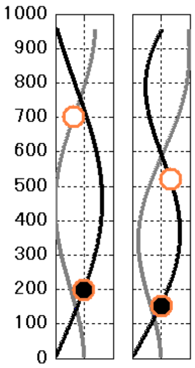


Fig.3 Modeling and sensing points

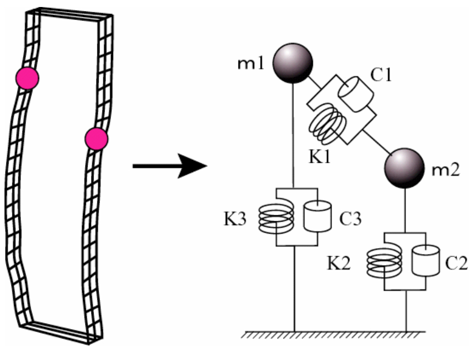


Fig.4 Reduced order 2DOF model

First, the temporary normalized modal matrix is constructed from displacement mode shapes shown in Fig.2 as follows,

$$\boldsymbol{\Phi} = \begin{bmatrix} 0.53911 & -0.2111 \\ 0.43141 & 1.1018 \end{bmatrix} \tag{5}$$

After modifying the normalized modal matrix using above mentioned procedure, the parameter of the mass and spring elements of a control model with 2DOF is obtained by Eqs. (1) and (2). Then, the vibration control model of the undamped system with 2DOF is made. Next, we add damping elements to this model. The damping parameters are obtained by comparing the resonance peaks of the frequency response between the actual system and the simulated one. The damping parameters become c1 = 1.3499 Ns/m, c2 = 0.0 Ns/m, c3 = 3.2274 Ns/m. Finally, the parameters shown in Fig.4 are determined as follows,

$$m1 = 2.6141 \text{ [kg]},\ m2 = 0.7071 \text{ [kg]},$$
$$k1 = 2.0108\times10^{3} \text{ [N/m]},\ k2 = 0.3549\times10^{3} \text{ [N/m]},\ k3 = 0.4520\times10^{3} \text{ [N/m]}$$

Making a Model to Express Motion

The control model of the rigid body part that expresses translation motion of the transport table and the part of the flexible tower structure is expressed by 2DOF. Thus, at the next stage, a model to express motion of the transport table is needed. The model of motion is made by an approximation of the measured transfer function. An approximate transfer function is expressed as follows,

$$G = \frac{k\omega_n^2}{s(s^2 + 2\varsigma\omega_n + \omega_n^2)} \tag{6}$$

Then, each parameter in Eq.(6) is as follows,

$$\zeta = 0.23[-],\quad \omega_n = 2\pi \times 15[rad/s],\quad k = 0.53[m/V]$$

CONTROL SYSTEM DESIGN

This purpose of research is to control the motion of the transport table and the vibration of the 1st and 2nd mode of the flexible tower structure. The control model is expressed as a state equation with the velocity and displacement of each mass, and the acceleration, velocity and displacement of the table. In the state vector indicated in Eq. (7), x_1, x_2 and x_t are the displacement of mass1, mass2. and the table, respectively.

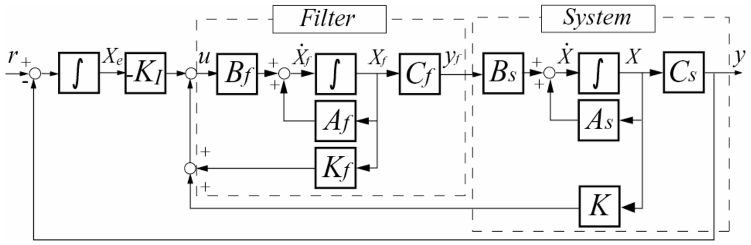
$$\boldsymbol{X} = \{\dot{x}_1 \quad \dot{x}_2 \quad x_1 \quad x_2 \quad \ddot{x}_t \quad \dot{x}_t \quad x_t\}^T \tag{7}$$

$$\begin{aligned} \dot{\boldsymbol{X}} &= \boldsymbol{AX} + \boldsymbol{Bu} \\ \boldsymbol{Y} &= \boldsymbol{CX} \end{aligned} \tag{8}$$

These matrices $\boldsymbol{A}$, $\boldsymbol{B}$ and $\boldsymbol{C}$ are obtained by Fig.4 and Eq.(7). Although LQ control theory is effectively used for active vibration control of flexible structures, there is some positioning error for tracking control. In this research, the LQI control theory that enables LQ control theory to cope with servo problem is applied. Figure 6 shows a block diagram of the LQI control system. The feedback gain vectors $\boldsymbol{K}$ and $\boldsymbol{K_I}$ are simultaneously obtained by applying LQI control theory to the augmented system expressed by the following state variable.

$$\boldsymbol{X}_c = \{\dot{x}_{f1} \quad x_{f1} \quad \dot{x}_{f2} \quad x_{f2} \quad \dot{x}_1 \quad \dot{x}_2 \quad x_1 \quad x_2 \quad \ddot{x}_t \quad \dot{x}_t \quad x_t\}^T \qquad \text{'(9)}$$

To avoid spillover, a low pass filter is appended to the control system to cut off the control signal at high frequency [9]. Fig.5 shows the relationship of the control object and the low pass filter with state variables x_{f1} and x_{f2}.



.Fig.5 Block diagram representation of LQI control system

EXPERIMENT

Experimental Setup

Figure 6 shows a schematic view of a constricted experimental setup. Dimensions of the transportation system were already described in Fig.1.

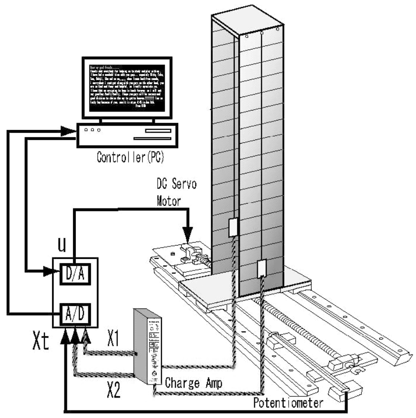


Fig.6 Experimental setup

The potentiometer is used to measure the displacement of the table. The velocity value of the table is desired from time differential calculations of the displacement obtained from the potentiometer. Two piezoelectric films pasted at the nodes of 3rd moment modes measure the displacements of two mass points. The velocity values of two mass points are obtained also by time differential calculations of the displacements obtained from piezoelectric films. Measured displacement signals are send to a personal computer through A/D converter, and velocity signals are calculated in the computer. The low pass filters are also constructed in the computer; therefore the state feedback from the filter is easily realized. According to such a way, an unstable condition owing to phase lag of the filter is not happen, even if 4th order low pass filter is used.

Experimental result

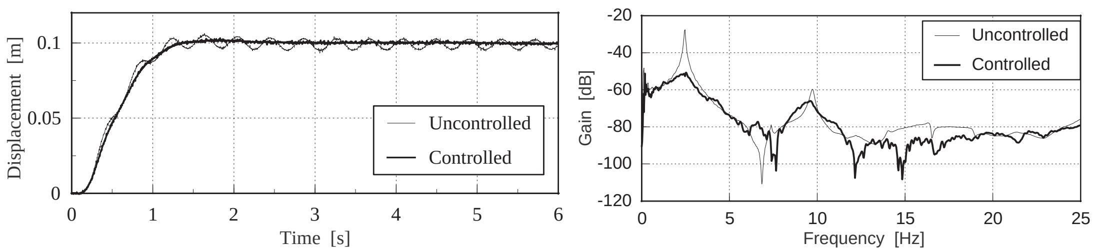


Fig.7 Experimental result of frequency and step responses

In order to confirm a reduce effect of a settling time from one position to another one, an experimental study has been carried out. The target displacement on the step input is

0.1 m. Figure 7 shows the experimental results of step responses with LQI control and without control. The experiment results of frequency responses with LQI control and without control are shown too. These results are obtained by the piezoelectric film corresponded to measurements at mass1. Control experiments has been demonstrated in Fig.7 that the settling time was well shortening by controlling many-modes of vibration and resonance peaks of 1st and 2nd vibration modes were well suppressed.

CONCLUSIONS

Because actual transportation systems demand to realize a high-speed motion with short settling time and long moving distance, this paper has dealt with a new approach to bring these demands to a conclusion. As the results, the following knowledge has been gained in this paper.

Vibration problems caused by lightweight structures are solved by vibration control of many modes using the proposed modelling and controlling methods.

Control experiments demonstrated that the settling time was well shortening by controlling many-modes of vibration in addition to the motion.

In order to obtain the long moving distance for the transportation system, piezoelectric films pasted on flexible structures make it possible to detect many modes of vibration as a point sensor.

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