DEVELOPMENT OF HYBRID MASS DAMPER USING LINEAR-INDUCTION-SERVOMOTOR FOR VIBRATION CONTROL OF TALL BUILDINGS

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ABSTRACT

Hybrid mass dampers using AC-servomotor with convertible active and passive mode for obtaining good habitability of tall or slender buildings against wind or earthquake excitations have been already developed by the authors. This type of mass damper has been applied to two actual buildings, and has been verified to be effective. In this study, a new hybrid mass damper using cylindrical linear-induction-servomotor, which is often used to elevators in the building, just has been developed to make the mechanical system simpler and to obtain larger vibration control capacity. In this paper, the mechanical characteristics and analytical model of the current mass damper including the motor and driver are described. And also an active-passive mode switching rule which has been already proposed by the authors, is improved in order to expand vibration control performance against wider range excitations. Through two shaking table tests using a five-story model building equipped with a small-scale mass damper and a large-scale mass damper model on the table, it is verified that the mass damper has more satisfying performance and the proposed analytical or design model, mode switching rule are valid. In addition, by analysis for a hypothetical 7-story building, the vibration control performances of the mass damper against some excitations is predicted accurately.

KEYWORDS

Vibration control; Hybrid mass damper; Linear-induction-servomotor; Tall building
Active-passive mode switching; Shaking table test; Simulation analysis ; Wind and earthquake load

INTRODUCTION

In Japan, some types of active or hybrid mass damper have been developed and applied to actual buildings. These mass dampers are designed to improve habitability of tall buildings during strong winds or weak to moderate earthquakes. Fujita et al. (1993) have proposed a hybrid mass damper with convertible active and passive modes, that works as an active mass damper providing higher vibration control performance during winds or weak earthquakes which may occur frequently, and also works as a passive mass damper during extreme strong winds or medium to severe earthquakes. They also carried out practical application researches using a hydraulic actuator(Fujita et al. 1991) and a AC-servomotor (Fujita and Yonezawa et al. 1994 , Fujita and Shimazaki et al. 1994 and Ohyama et al. 1994).

In case of the mass damper using AC-servomotor, some devices such as rack-pinion, bawl-screw and reduction gear are necessary to transmit rotation of the motor to linear movement of the mass. These transmission devices would decrease whole control performance of the mass damper. Meanwhile, when a linear-induction-servomotor, which is introduced in this paper, is used as an actuator for the damper, no transmission device is necessary and no friction induced by the device generate. Resultantly, more compact and noiseless system with easy maintenance and high vibration control performance can be realized. Also as the mode switching from active to passive mode can be performed only by turning the power of motor on or off, proposed converting mechanism could act with sufficient smoothness. On the other hand, as the ready-made linear-servomotor is used, it is necessary to take the fact, which its motor power is restricted and the electric efficiency is not so larger than another motors, into consideration in designing the mass damper system.

In this paper, the mechanical characteristics and analytical model of the current mass damper including the motor and driver are described. An active-passive mode switching rule, which has been already proposed by the authors and is aimed to expand vibration control performance in wide range, is proposed. Through two shaking table tests using a five-story model building equipped with a small-scale mass damper and a large-scale mass damper model on the table, it is examined that the mass damper has more satisfying performance and the proposed analytical model, mode switching rule are valid. In addition, through analysis for a hypothetical 7-story building, the vibration control performances of the mass damper against some excitations will be estimated quantitatively.

HYBRID MASS DAMPER USING LINEAR-INDUCTION-SERVOMOTOR

Developed mass damper has a hybrid structure in which active and passive mechanism part are connected in parallel. As an actuator, cylindrical linear-induction-servomotor, which is often used to elevators in the building as shown in Figure 1, is used. In order to verify the efficiency of developed mass damper and to examine mechanical characteristics of full scale model, two experiments were carried out. One experiment was a fundamentally executed shaking table test on a five-story steel-framed model building equipped with a small-scale mass damper (Type-1). The another was a shaking table test on a large-scale mass damper (Type-2), which was supposed to be installed on a hypothetical 7-story full-scale building.

Figure 2 shows the structure of the cylindrical linear-induction-servomotor introduced in this paper. The stator is consisted of four steel blocks, and the outside of the stator is wrapped in coil. The rotor is consisted of a steel rod covered with aluminum to work as a secondary conductor. The most important characteristic of the linear-servomotor is that as the motor has a cylindrical shape, the structure of the motor can be made simple to maintain necessary gap between the stator and the rotor, by the reason that a rotor levitates in a stator. Consequently, no friction induced by the structure and more compact and noiseless system can be obtained easily.

Figure 3 shows a drawing of small-scale mass damper (Type-1), and Table 1 lists its specifications. Structural components of the passive mechanism part are moving mass, coil springs and a viscous damper. The moving mass of 80 kg in weight is consisted of a liner-motor rod and a steel plate. Coil springs and a viscous damper are installed between the rod and the stator. The natural period of the mass damper is aimed to coincide with the first natural period of the five-story model building. As the motor whose capacity is 7.6 kW is too large as an actuator for the mass damper model, it is expected that the limit of capacity of motor will not occur.

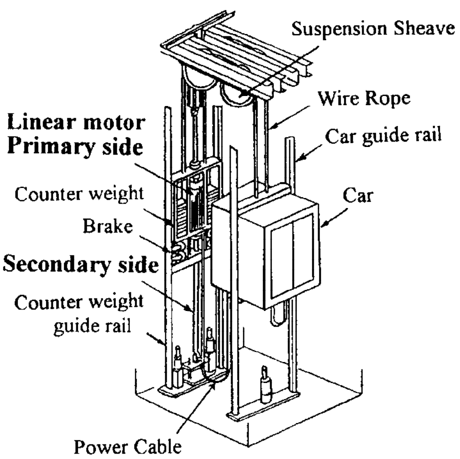


Fig.1 Construction of linear motor elevator

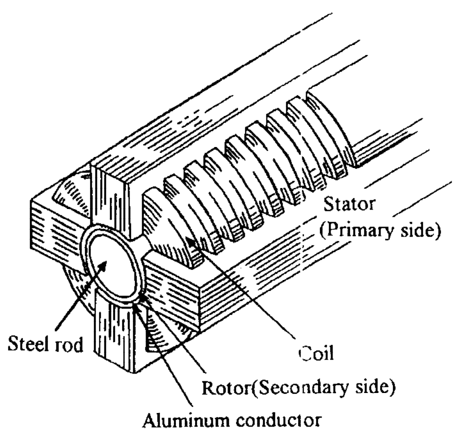


Fig.2 Drawing of cylindrical linear-induction-servomotor

Figure 4 shows a model building which has 3.4 m height and 5,800 kg weight and a small-scale mass damper. Its natural frequency and damping ratio are 2.41 Hz and 0.8 % for the first mode, and 6.44 Hz and 2.1 % for the second mode, respectively. In the hybrid mass damper, the mode switching can be performed only by turning the power off and on. By this means, the mode switching mechanism could work with sufficient smoothness.

Figure 5 shows a drawing of a large-scale mass damper (Type-2), and Table 2 lists its specifications. Structural components of the passive mechanism part are moving mass, coil springs and an oil damper, the damping force of which is proportional to the square of the velocity in order to control the moving mass within a range of permissible stroke for extreme earthquake excitations. The stator of linear-motor itself worked as a part of the moving mass of about 4,000 kg in weight, which is supported by one-direction linear guides. The linear-servomotor is the same as motor of Type-1. As the driving part is consisted of only two 7.6 kW linear-motors mounted on the moving mass for easy maintenance, the height of mass damper is only 880 mm in height and a compact system can be realized. The mode switching is the same as that of the Type-1. The hypothetical building to be controlled by this mass damper has seven stories, whose specifications are shown in the later section and Figure 6 shows a view of a shaking table test on the mass damper.

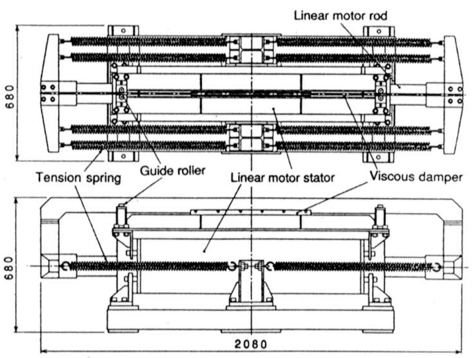


Fig.3 Drawing of smale-scale mass damper (Type-1)

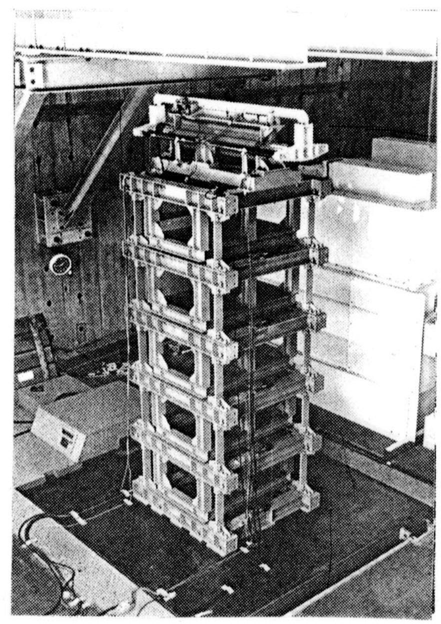


Fig.4 View of five-story model building and Type-1 mass damper [Test-1]

Table 1 Specifications of Type-1

Moving mass	85.8 kgf
Natural frequency	2.38 Hz
Damping ratio	9 %
Stroke	±20 cm
Motor power	7.6 kW (rated) × 1
Control force	444 kgf (max.)

Table 2 Specifications of Type-2

Moving mass	4290 kgf
Natural frequency	0.96 Hz
Damping coefficient	0.184 kgf$(s/cm)^2$
Stroke	±40 cm
Friction coefficient	0.008
Motor power	7.6 kW (rated) × 2
Control force	1500 kgf (max.)

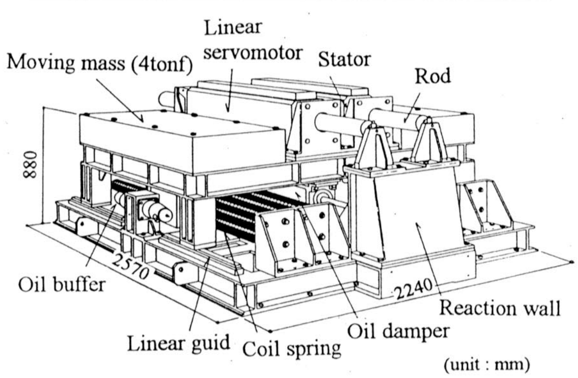


Fig.5 Drawing of large-scale mass damper (Type-2)

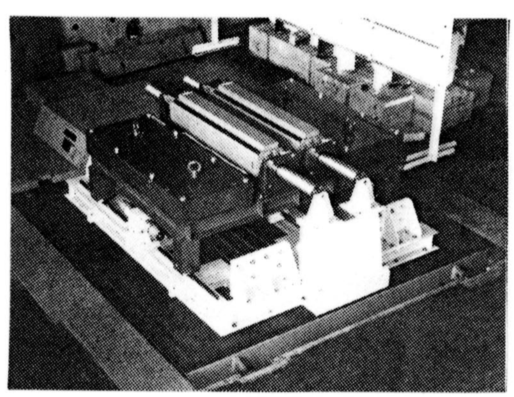


Fig.6 View of large-scale mass damper [Test-2]

ANALYTICAL MODEL OF MASS DAMPER

Dynamic characteristics of mass damper

In the current system, velocity of the moving mass is controlled. In order to acquire dynamic characteristics of the mass damper, transfer function(G) of input voltage to the motor-driver(u) and response velocity of the moving mass($\dot{x}_d$), were evaluated as Eq. (1) through frequency response tests.

$$G(s) = \dot{x}_d(s) / u(s) \qquad \text{--- (1)}$$

where s is laplacian operator. Figure 7 shows an example of the test results for the mass damper of Type-2, compared with identified model. This system can be identified as a second order system. In this figure, the result of an analytical model shown in Eq.(6) is also shown. Similar characteristics were also obtained for the mass damper of Type-1.

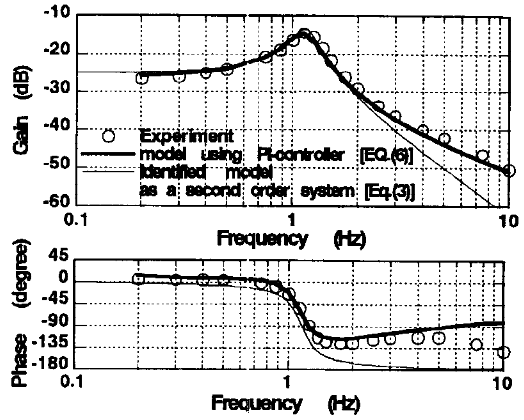


Fig.7 Dynamic characteristics of the mass damper (Type-2)

Equation of motion for building with mass damper

In the building system, it is assumed that the mass damper is located on the roof floor and building has n-degree-of-freedom. Equation of motion for the building system is expressed as Eq.(2). But equation of motion for mass damper system should be changed depending on the passive or active state, as shown in the following section.

$$[M]\{\ddot{x}\} + [C]\{\dot{x}\} + [K]\{x\} = \{s\}(-m_d(\ddot{x}_d + \ddot{x}_n + \ddot{z})) - [M]\{1\}\ddot{z} + \{w\} \qquad \text{--- (2)}$$

where $[M],[C]$ and $[K](n \times n)$ are the building's mass, damping and stiffness matrices, respectively; $\{x\}(n \times 1)$ is relative displacement vector of the building to the ground; $\{s\} = \{0,\cdots 0,1\}^T (n \times 1)$ is a vector representing the story of the building where mass damper is installed; $\ddot{z}$ is seismic ground acceleration; $\{w\}(n \times 1)$ is wind force vector; m_d is mass of the moving mass; x_n is relative displacement of the roof floor to the ground; x_d is relative displacement of the moving mass to the roof floor of the building.

Mass damper system in active mode As the relation between the input voltage and the velocity of the mass is expressed by a second order system, the equation of motion for the mass damper system can be expressed as follows.

$$\ddot{x}_d + 2\zeta_a\omega_a\dot{x}_d + \omega_a{}^2\dot{x}_d = K_a\omega_a{}^2 u \qquad \text{--- (3)}$$

where ζ_a, ω_a and K_a are the damping ratio, natural frequency and gain constant, respectively. These constants in the case of Type-2 mass damper, ζ_a, ω_a and K_a, are 0.13, 7.3 radian/sec and 5.5 [cm/sec]/volt, respectively. However, Eq.(3) cannot express the non-linearity of damping characteristics and the saturation of the motor-force. Adopting Proportional and Integration(PI)-operation of actual control inside motor driver, the force of motor(F_a) against input voltage to motor(u) is expressed as follows;

$$\dot{e} = u - K_{fv}\cdot\dot{x}_d - K_{fd}\cdot x_d \qquad \text{--- (4)}$$

$$F_a = K_e\cdot(K_p\cdot\dot{e} + [K_p/T_i]\cdot e) \qquad \text{--- (5)}$$

where e is deviation voltage of output from input. K_{fv} and K_{fd} are feedback gain constant of velocity and displacement of mass damper. K_p and T_i are gain constant and time constant of PI-controller. K_e is gain constant of output force to input. Resultantly, the equation of motion of mass damper is expressed as follows:

$$m_d\ddot{x}_d + c_d|\dot{x}_d|\dot{x}_d + k_d x_d + \mu_d m_d g\,\mathrm{sgn}(\dot{x}_d) = F_a \qquad \text{--- (6)}$$

where c_d, k_d and μ_d are damping, stiffness and frictional coefficient of the mass damper, respectively. In addition, the frequency response of analytical model using Eqs.(4) to (6) is shown in Figure 7 as a thick solid line, and it is clear that the result using Eqs.(4) to (6) is better than that using Eq.(3).

Mass damper system in passive mode In the passive mode, the force of motor is zero. The system has two kinds of state in which the mass is moving or not. If the resultant force acting on the moving mass is smaller than static frictional force, the mass cannot move. This condition should be considered (Fujita and Shimazaki et al. 1994).

Analytical model of motor-driver system In general, a motor-driver is imposed by some limitations of capacity. This motor-driver also works under limitations, which are motor's and driver's temperature. In order to evaluate these limiting indices, the following model for identification is adopted.

The driver current (I) can be expressed as Eq.(7). The power of motor (P) can be obtained from the motion of the mass damper as Eq.(8). As the relation between the temperature change (δ) and the square value of the driver current can be expressed as a first order system such as Eq.(9), the motor's or driver's temperature (θ) can be obtained by Eq.(10).

$$I = \sqrt{\alpha F_a^{\,2} + I_0^{\,2}} \qquad \text{--- (7)}$$

$$P = F_a \cdot \dot{x}_d / \eta \qquad \text{--- (8)}$$

$$T_t \dot{\delta} + \delta = K_t I^2 \qquad \text{--- (9)}$$

$$\theta = \theta_0 + \delta \qquad \text{--- (10)}$$

where α and I_0 are current coefficient and inverter exciting current, respectively; η is transmission efficiency; θ_0, δ, T_t and K_t are room temperature, temperature change, time and gain constant of temperature change, respectively.

CONTROL RULES OF ACTIVE MODE

To simplify the control operation in the active mode, modal-filter is applied, through which only necessary modal coordinates are filtered. When Eq.(2) of the active mode system are converted into the equation of motion for respective modal coordinates, the following equation is obtained for i-th mode:

$$M_i \ddot{q}_i + C_i \dot{q}_i + K_i q_i = \phi_{in}\left\{-m_d(\ddot{x}_n + \ddot{x}_d + \ddot{z})\right\} - \beta_i M_i \ddot{z} + w_i \qquad \text{--(11)}$$

where $q_i, M_i, C_i, K_i, \beta_i$ and w_i are i-th mode displacement, generalized mass, damping and stiffness, mode participation factor and generalized wind force, respectively; ϕ_{in} is component at the roof floor of i-th mode of modal matrix. Here, the state valuable is defined as Eq.(12), finally the state equation, Eq.(13), is obtained.

$$\{X\} = \{q_1, q_2 \cdots q_k, \dot{q}_1, \dot{q}_2 \cdots \dot{q}_k, x_d, \dot{x}_d\}^T \qquad \text{--(12)}$$

$$\{\dot{X}\} = [A]\{X\} + \{B\}u + \{D\}\ddot{z} + \{w\} \qquad \text{--(13)}$$

where $\ddot{x}_n \approx \sum_{j=1}^{k} \phi_{jn} \ddot{q}_j$; $[A], \{B\}$ and $\{D\}$ are a constant matrix and vectors, respectively.

The control system is designed using the optimal regulator. In order to decrease the relative displacements and velocities of the building, the performance index(J) is defined as follows:

$$J = \int_0^\infty \left\{ \sum_{j=1}^{k} (\lambda_j q_j^{\,2} + \mu_j \dot{q}_j^{\,2}) + \gamma\, u^2 \right\} dt \qquad \text{--(14)}$$

The obtained results through the shaking table test of a five-story model building equipped with Type-1 mass damper in Figure 4 are shown as follows. The excitation input was stationary random excitation, whose power spectrum is constant between 0.5 and 20 Hz. And the first and the second modes were selected as control objective. Figure 8 shows comparison between response roof floor accelerations in non-controlled, passive and active mode. Comparing the response of the active mode with that of non-controlled, the ratio of response is about 1/20 for the first mode and about 1/10 for the second mode. Consequently, the sufficient vibration control performance of this type of mass damper and the efficiency of modal-filter in order to avoid a spill-over phenomenon is verified.

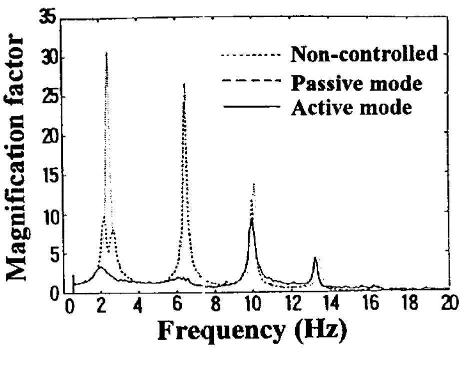


Fig.8 Transfer function of response acceleration at roof floor [Test-1]

ACTIVE-PASSIVE MODE SWITCHING RULES

In this system, when the vibration control is kept in the active state exceeding the rated capacity of the motor, the protection system of motor and driver will be worked. When this protection system is actuated and then the motor stops, so, the system cannot re-start automatically. This protecting action of motor driver itself is called "overload stopping", caused by overheat of motor and driver. Therefore, controller of the mass damper always monitors the temperatures of motor and driver, switches the mode to passive just before the protection system works to avoid overloading if necessary . When the overloading state is returned to normal state, the mode is switched back to active by the controller. In case of the linear-servomotor, limiting temperatures are set at 140℃ in the motor and 80℃ in the driver, respectively.

The active mode is switched to the passive mode at the time when the motor temperature has risen to 130℃, or when the driver temperature has risen to 70℃. When the motor temperature has dropped to 100℃ and when the driver temperature has dropped to 50℃, the controller starts to calculate the root-mean-square value of input voltage during twice of the first natural period of the building. When the root-mean-square value is below a certain value, u^*, and when an instantaneous value of input voltage is below an another certain value, u', the controller turn on the motor and the mode is switched from the passive to the active mode. These conditions are considered for the purpose of smooth switching without any shocks.

In order to verify the validity of the active-passive mode switching algorithm and analytical model, the shaking table tests of the large-scale mass damper was carried out shown in Figure 6. The excitation inputs were the first modal response acceleration at the roof floor of the hypothetical 7-story building subjected to winds or earthquakes. Outline of the test are shown in Figure 9. The specifications of the building and the wind external force are shown in the following section. Control was performed only against the first mode, and the designated damping ratio of the building were 20% for winds and 15% for earthquakes, respectively.

Figure 10 shows the typical test results of the mass damper compared with the analytical results, in the case when the building response (this duration time is about 1200 seconds) to a 100-year return period wind in Tokyo. In order to confirm the motion of switching mode from active to passive, also inversely, temporary switching condition was set as following. When the motor temperature has risen to 45℃, the mode is switched from active to passive. After switching, when the motor temperature has dropped to 40℃, the mode is switched from passive to active again. From this test result, the motor temperature starts from 25℃, at the time of about 550 seconds, the motor temperature has risen to 45℃, the mode is switched from active to passive. While in the passive mode, the motor temperature drops. After a while, as the motor is sufficiently cooled, the mode is switched from passive to active. Analytical results well reproduce not only the motion of the mass damper, but also the state of the motor and driver, such as power and temperature of motor. Furthermore, the timing of the mode switching are also in good agreement. In another case when the building response to an earthquake, similar results are obtained. These results show the validity of the mode switching algorithm and the analytical model as mentioned before.

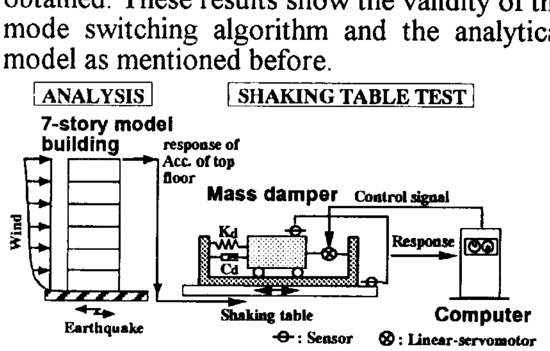


Fig.9 Outline of shaking table test on a large-scale mass damper [Test-2]

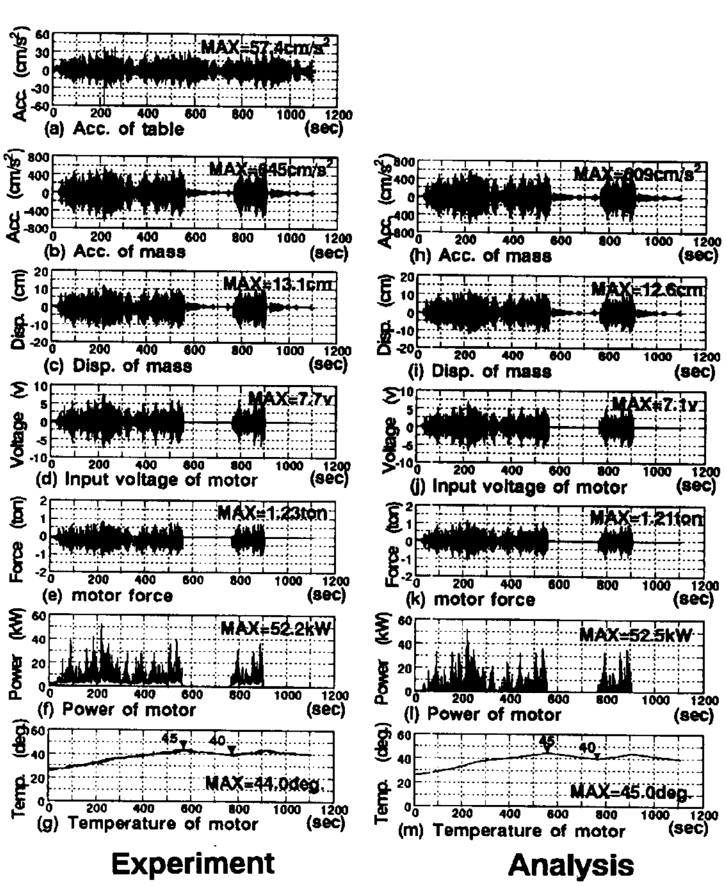


Fig.10 Results of test for the building response to a 100-year return period wind compared with analytical results [Test-2]

PREDICTION ANALYSIS OF VIBRATION CONTROL PERFORMANCE

Outline of prediction analysis

Using the analytical model which consists of Eqs.(2) to (10), prediction analysis of vibration control performance against winds or earthquakes is carried out. Figure 11 shows the specifications of the hypothetical building and mass damper. Its natural frequency and damping ratio are 0.97 Hz, 0.5 % for the first mode and 2.7 Hz, 1.4 % for the second mode, respectively. The first and the second modes are selected as control objectives and the designated damping ratio of the building were set at 20% for the first mode and 10% for the second mode against winds, and 10% for the first mode and 5% for the second mode against earthquakes, respectively.

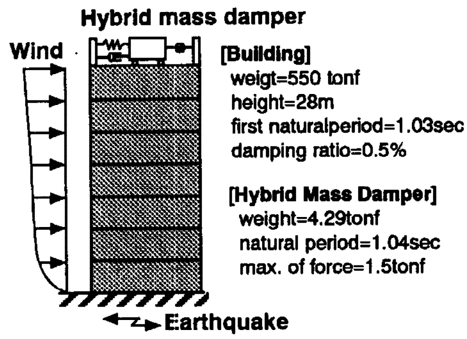


Fig.11 Specifications of the hypothetical building and mass damper

Vibration control performance against wind

The wind external force applied in the analysis was determined according to the "Recommendation for Loads on Buildings (Architectural Institute of Japan, 1993)" stipulated by the Architectural Institute of Japan. The time history of wind force is stochastically simulated from power spectrum of the first modal generalized wind force on wind direction.

Figure 12 shows the comparisons between roof floor accelerations in the non-controlled, the passive and active mode against winds. The horizontal axis represents the average wind velocities at the top of the building and corresponding return period too. When the average wind velocity is below 17 m/s, the vibration control performance of the passive mode is not demonstrated because the mass damper does not work due to friction. In the active mode, both the maximum value and the root-mean-square value of acceleration at the roof floor is significantly reduced, and satisfactory vibration control performance is demonstrated. In the passive mode when the average velocity exceeds 17 m/s, because the frictional coefficient of the current mass damper using linear-induction-servomotor is smaller than that using AC-servomotor, the vibration control performance of the current mass damper against weak wind has been larger than AC-servomotor-type.(Fujita and Shimazaki et al. 1994).

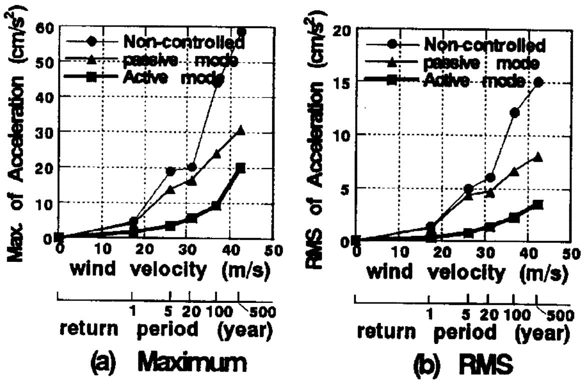
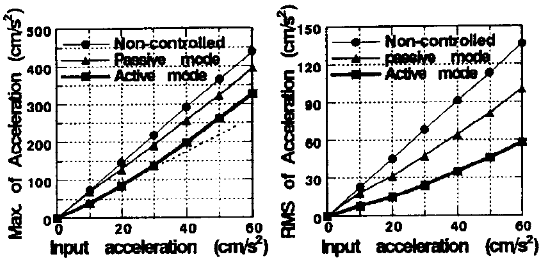


Fig.12 Predicted vibration control performance against winds

In this analysis, it is assumed that the initial temperature is 15℃ and the duration time of analysis is 20 minutes. Even though the average wind velocity is set at 42 m/s, correspond to 500-year return period, the motor temperature remains still 126℃ less than 130℃ and so the mode switching from active to passive does not work. In this case, the force saturation of the motor is generated, it can be said that the vibration control performance in the active mode does not so deteriorates.

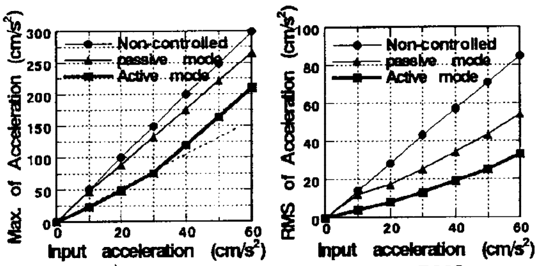
Vibration control performance against earthquake

Figure 13 and 14 show the comparisons of vibration control performance between against Hachinohe-EW and El Centro-NS earthquake excitations, respectively. Figure 15 shows response time histories in the case of non-controlled and the active mode against El Centro-NS of a 40 cm/s^2 ground acceleration. As the results of analysis against both earthquakes, the ratio of the maximum value of response acceleration in the active mode to that in non-controlled is 2/3, and the ratio of the root-mean-square value of response acceleration in the active mode to that in non-controlled is between 1/2 and 1/3. The overload stopping does not occur in this analysis. It is considered that the duration time is too short to rise the motor temperature. Meanwhile, the vibration control performance starts to drop at ground acceleration of 30 cm/s^2 due to the saturation of motor force. From this result, in case of the active vibration control against earthquake, it seems that the overload stopping could not occur even during large excitations, but the vibration control performance might drop due to the saturation of the motor force. Consequently, it will be necessary to develop the method to evaluate the vibration control performance exactly in the state of motor saturation.



(a) Maximum (b) RMS

Fig.13 Predicted vibration control performance against Hachinohe-EW earthquake



(a) Maximum (b) RMS

Fig.14 Predicted vibration control performance against El centro-NS earthquake

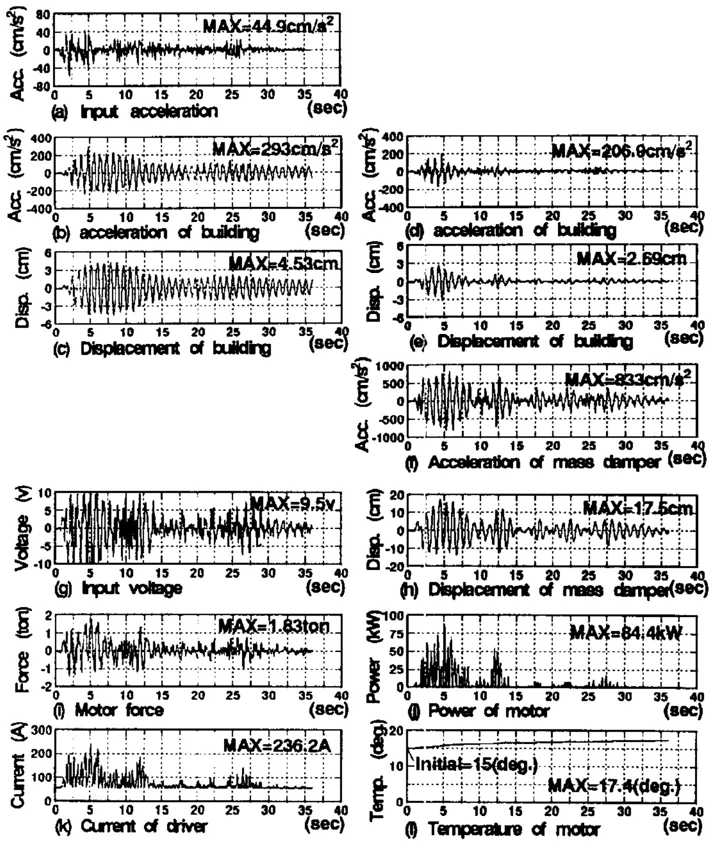


Fig.15 Response to El centro-NS earthquake of a 40cm/s² in case of non-controlled and the active mode

(b),(c) : Non-controlled mode

(d)~(l) : Active mode

CONCLUSIONS

In this paper, a new-type hybrid mass damper using linear-induction-servomotor, which can be designed as a more compact, powerful type, is outlined. Through vibration tests and simulation analysis, it is verified that this mass damper can work with sufficient smoothness both in active and passive mode and also simulation results are well agreed with the test results. Furthermore, through the prediction analysis for a hypothetical seven-story building, the quantitative estimation of accurate vibration control performance is obtained. Consequently, it is shown that the necessary information for actual vibration control design, such as a design model of the mass damper in both active control and mode switching control and a criteria and a strategy for judging condition of mode switching, are obtained.

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