FEASIBILITY STUDY OF ACTIVE MICRO-VIBRATION CONTROL SYSTEM USING PIEZOELECTRIC ACTUATORS FOR FLOOR STRUCTURE OF PRECISION MANUFACTURING FACILITIES

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ABSTRACT :

As the semiconductor processing technology keeps evolving, the demand for vibration-free environment of semiconductor manufacturing facilities keeps becoming severe. We develop an active micro-vibration control system by using piezoelectric actuators to suppress the vibration of the floor in precision manufacturing facilities. The system is a smart structure that includes the actuators put on the beams of the floor, acceleration sensors and controllers. The main purpose is to suppress the vertical vibration of the floor where the machines are installed. The smart structure actuator is suitable for micro-vibration control except earthquakes for the limit of the capacity. Feasibility tests were executed by using a large-scale experimental model, which is a steel frame of two-dimensional truss to represent as the floor structure of the manufacturing facilities. The weight of the model is 1800kg and its span is 6m. The size of the piezoelectric actuator is 25 x 25 x 36mm. The test result shows that the exception vibration control performance is achieved and expresses the feasibility of the control system for the real structure.

KEYWORDS: Active Vibration Control, Piezoelectric Actuator, Floor Structure

1. INTRODUCTION

As the semiconductor processing technology keeps evolving and the design rule keeps becoming small rapidly, the demand for vibration-free environment of semiconductor manufacturing facilities keeps becoming severe. Besides, the semiconductor manufacturing equipment improves enlargement for large-size wafers and its operational speed becomes fast to raise the production capacity during the recent years, hence force that is generated by the manufacturing equipment grows and the equipment and the installation floor are excited by the force. The production capacity of the equipment and the adjacent machines has been decreased by the influence of the vibration.

We developed an active micro-vibration control system by using piezoelectric actuators to suppress the vertical vibration of the floor in precision manufacturing facilities. The system is a smart structure (**FUJITA**) that includes the actuators put on the beams of the floor, acceleration sensors and controllers. The main purpose is to suppress the vertical vibration of the floor where the machines are installed. The smart structure actuator is suitable for micro- vibration control except earthquakes for the limit of the capacity.

Participation factor to the structure of the actuator, which both ends are supported on the structure, was considered using a simple beam model analytically. We discuss where the both ends of the actuator are supported on the structure is very important for the participation factor.

Feasibility tests were executed by using a large-scale experimental model, which is a steel frame of two-dimensional truss to represent as the floor structure of the manufacturing facilities. The weight of the model is 1800kg and its span is 6m. The size of the piezoelectric actuator is 25 x 25 x 36mm. The test result shows that the exception vibration control performance is achieved and expresses the feasibility of the control system for the real structure.

2. OUTLINES OF LARGE-SCALED EXPERIMENTAL MODEL

Figure 1 shows a large-scaled experimental model, which is a steel frame structure of two-dimensional truss to represent as the floor structure of the semiconductor manufacturing facilities. Table 1 shows specifications of the experimental model. Total weight of the model is 1800kg and its span is 6m and its height is 1.7m. Figure 2 shows the installation of a piezoelectric actuator on the upper chord of the model. Two brackets are installed on the upper chord. The piezoelectric actuator is held between the two brackets with a rod and an adjustable bolt. The adjustable bolt pushes the piezoelectric actuator and generates pre-compression force between the two brackets. The rod is supported by two linear bearings. Figure 3 shows the piezoelectric actuator and table 2 summarizes its specifications.

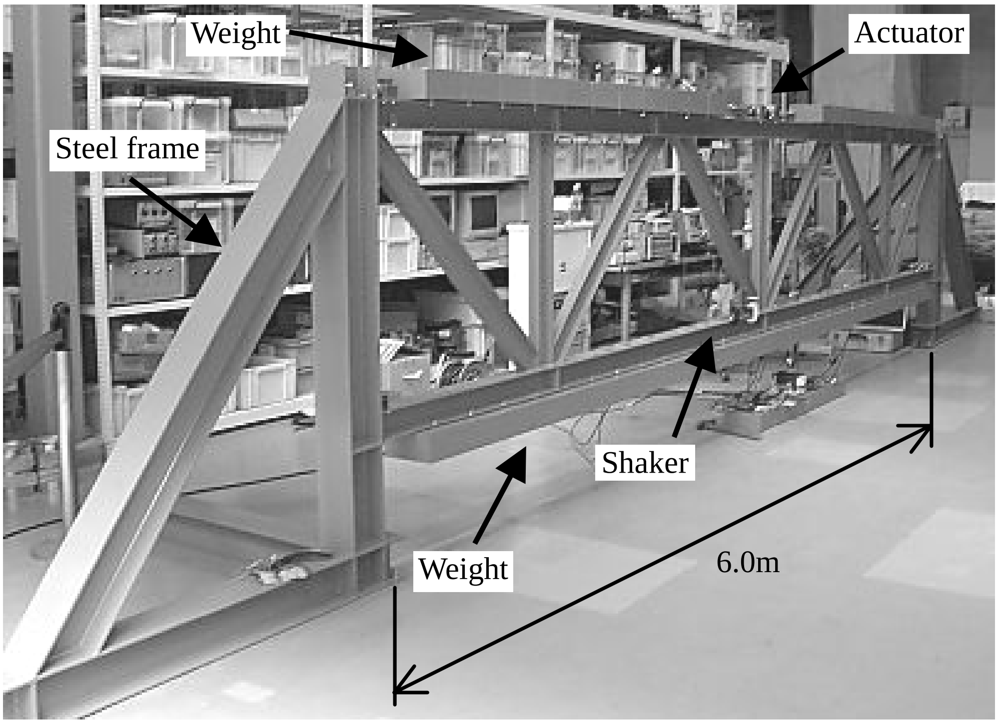

Figure 1 Large-scaled experimental model

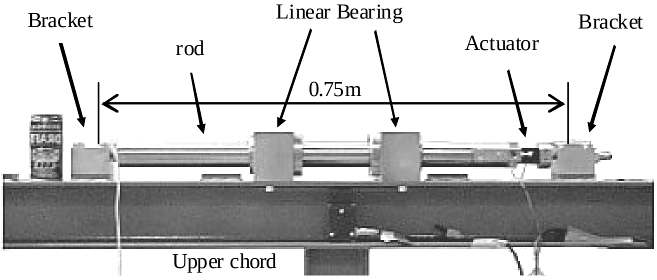

Figure 2 Installation of actuator on upper chord

Table 1 Specifications of the experimental model

Span x Height	6.0m x 1.7m
Member	H100x100x6x8, H100x50x5x7
Total weights	1800kg
1st mode	26.3Hz, 0.23%
2nd mode	69.2Hz, 0.38%
3rd mode	83.3Hz, 0.53%

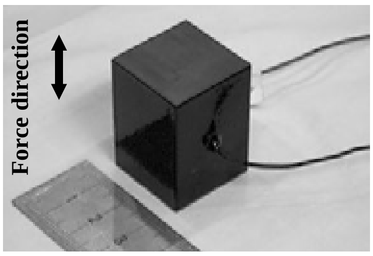

Figure 3 Piezoelectric actuator

Table 2 Specifications of the piezoelectric actuator

Outer size	25x 25x 36Hmm
Rated voltage	100V
Capacitance	80μF
Max. displacement	30μm
Max. force	21kN

3. ANALYTICAL CONSIDERATION

3.1. Consideration Fixed Straight Beam with Uniform Section

To simplify the problem, an analytical model is treated as a fixed straight beam with uniform section modulus. Figure 4 shows the analytical model for the simple beam with piezoelectric actuator. Vertical vibration of the continuous system is expressed as modal coordinates,

$$\ddot{q}_i + 2\zeta_i\omega_i\dot{q}_i + \omega_i^2 q_i = -\frac{1}{m^*_i}\left\{M_a\frac{d\phi_i(x_1)}{dx} - M_a\frac{d\phi_i(x_2)}{dx}\right\} \tag{3.1}$$

where q_i, ζ_i, ω_i and m^*_i are modal displacement, modal damping ratio, modal frequency and modal mass of i-th mode of the beam, correspondingly. x is position of length direction of the beam. $\phi_i(x)$ is a modal shape function as follow,

$$\phi_i(x) = \cosh k_i x - \cos k_i x - \frac{\cos k_i l - \cosh k_i l}{\sin k_i l - \sinh k_i l}(\sinh k_i x - \sin k_i x), \quad \cos k_i l \cosh k_i l = 1 \tag{3.2}$$

where k_i is a solution of the frequency equation. l is a length of the beam. x_1 and x_2 are points of the two brackets. M_a is concentrated moment at x_1, $-M_a$ is concentrated moment at x_2, which are caused by the control force of the actuator.

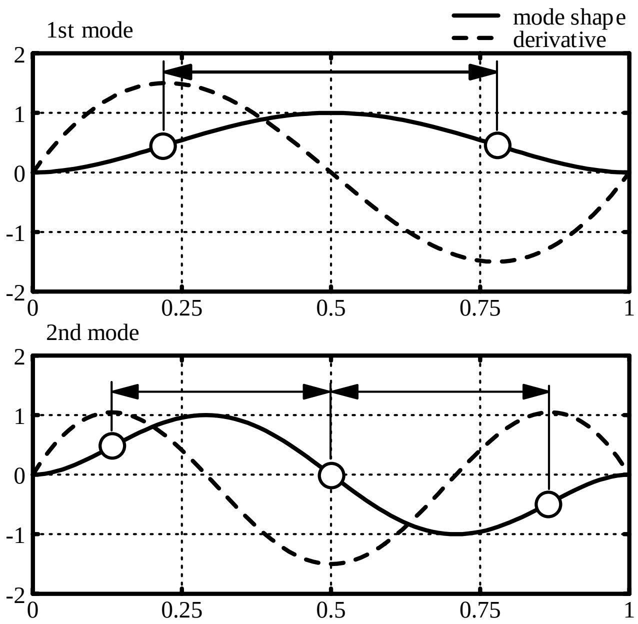

Figure 5 Mode shape functions and their derivative

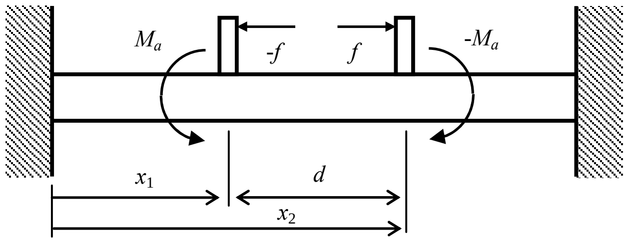

Figure 4 Analytical model of simple beam

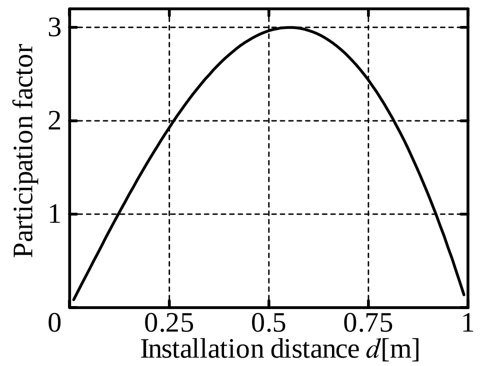

Figure 6 Relation between installation distance and participation factor

Now, we consider carefully that the right term of the Eqn. 3.1. means a participation factor for the *i*-th mode of the equipped actuator on the beam. The magnitude of the participation factor depends on the difference from x_1 to x_2 of the derivative of the mode shape function $\phi_i(x)$ with respect to x. Hence, it is very important where x_1 and x_2 are. Figure 5 shows the mode shape functions and their derivative functions when i = 1, 2. The participation factor is maximized where x_1 is the maximum of the derivative and x_2 is the minimum of the derivative. The white circles indicate a pair of the points for the maximum of the participation factor in the 1st mode, two pair of the points in the 2nd mode. However, a distance of the pair of the points, which the participation factor is maximized, is so long that it can be difficult to realize in a practical use. Here, the distance and mean of x_1 and x_2 are defined as d and x_a Hence Eqn. 3.1. is transformed

$$\ddot{q}_i + 2\zeta_i\omega_i\dot{q}_i + \omega_i^2 q_i = -\frac{M_a}{m^*_i}\left\{\frac{d\phi_i(x_a - d/2)}{dx} - \frac{d\phi_i(x_a + d/2)}{dx}\right\} \tag{3.3}$$

Figure 6 shows the relation between the distance d and the participation factor in a case of the beam length

l=1.0m, x_a= l/2 and for the 1st mode. The participation factor becomes the maximum where the distance d is at 55% of the beam length l. As the participation factor is proportionally increasing where the distance d is less than 40% of the beam length l, it is preferable to lengthen the distance d as far as possible in the practical use.

3.2. Analytical Model for Complicated Structure

An analytical model for complicated structure is also expressed as modal coordinates as follows,

$$\ddot{q}_i + 2\zeta_i\omega_i\dot{q}_i + \omega_i^{\,2} q_i = \frac{\phi_{Ai} f - \phi_{Bi} f}{m^*_{\ i}} \tag{3.4}$$

where φ_{Ai} and φ_{Bi} are participation factors of force f direction, where the force f and $-f$ act. The modal parameters for the complicated structure are obtained from finite element method (FEM). Figure 7 shows the modal analysis results of FEM for the experimental model, which was composed of two dimensional beam elements. Harmonic excitation test was executed by the piezoelectric actuator. Figure 8 shows the outlines of the harmonic excitation test by the actuator. Figure 9 shows the excitation test results in the cases of the distance d were 350, 750 and 1050mm. The vertical axis represents vertical response acceleration on the center of the upper chord of the model. It was confirmed that the tendency, which the response acceleration grows with increasing the distance d, was similar to the simple beam participation factor growing. It was also confirmed that the analytical results are in good agreement with the experimental results.

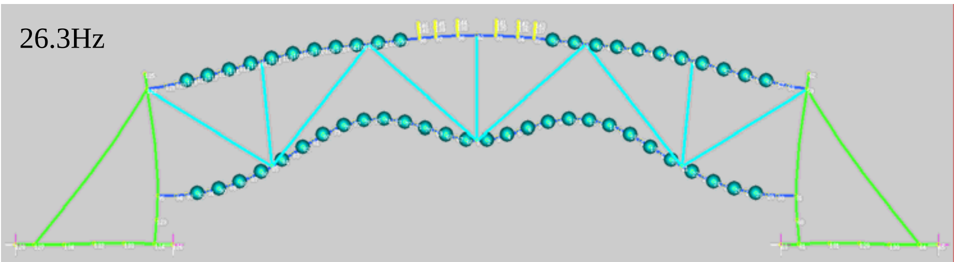

Figure 7 Modal analysis of FEM for the experimental model

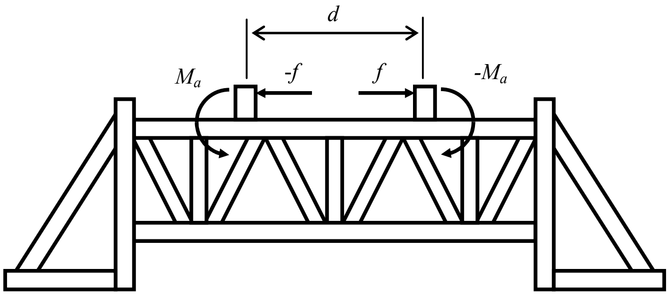

Figure 8 Outlines of the harmonic excitation test by the actuator

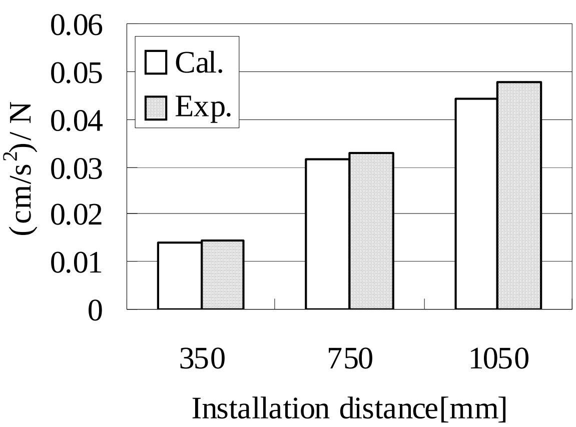

Figure 9 Results of the harmonic excitation by the actuator compared with the analysis

4. VIBRATION CONTROL TEST

4.1. Outlines of Control System

Figure 10 shows control system for the experimental model. The control system consists of three vertical acceleration sensors and controller. The controller was designed by a model-matching method (**FUJITA**) to reduce a response of the 1st mode of the experimental model. A modal filter (**MEIROVITCH**) was also included and calculated the 1st mode and the 3rd mode acceleration. The 3rd mode acceleration, which obstructed a performance of the 1st mode controller, was eliminated. The modal filer was obtained from the responses of the three sensors experimentally when the 1st mode and the 3rd mode were excited by the actuator. Eqn. 4.1. shows the modal matrix ϕ, which was obtained experimentally. An inverse matrix of the modal matrix became the modal filter.

$$\begin{Bmatrix} a_A \\ a_B \\ a_C \end{Bmatrix} = [\phi]\begin{Bmatrix} \ddot{q}_1 \\ \ddot{q}_3 \end{Bmatrix} = \begin{bmatrix} 0.77 & -2.02 \\ 1.24 & 2.35 \\ 0.83 & -1.56 \end{bmatrix}\begin{Bmatrix} \ddot{q}_1 \\ \ddot{q}_3 \end{Bmatrix} \tag{4.1}$$

where a_A, a_B and a_C are the responses of the three sensors, q_1 and q_3 are modal displacements of the first mode and third mode.

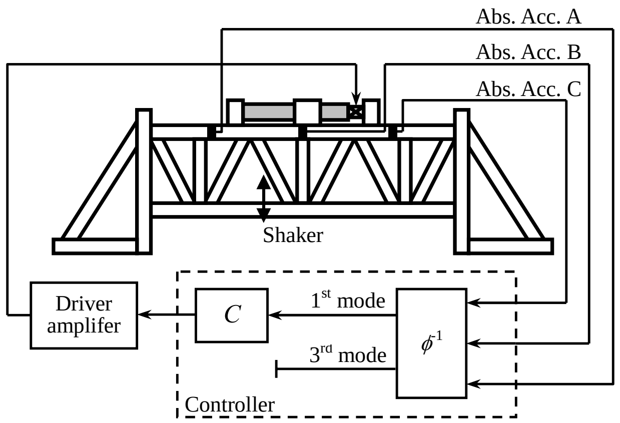

Figure 10 Control system for the experimental model

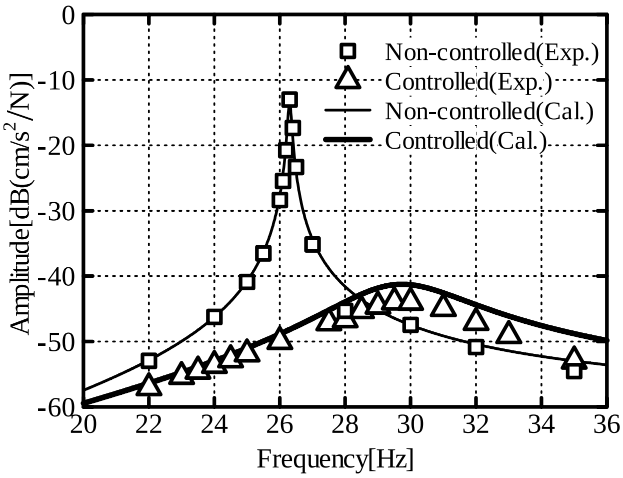

Figure 11 Vibration control performance

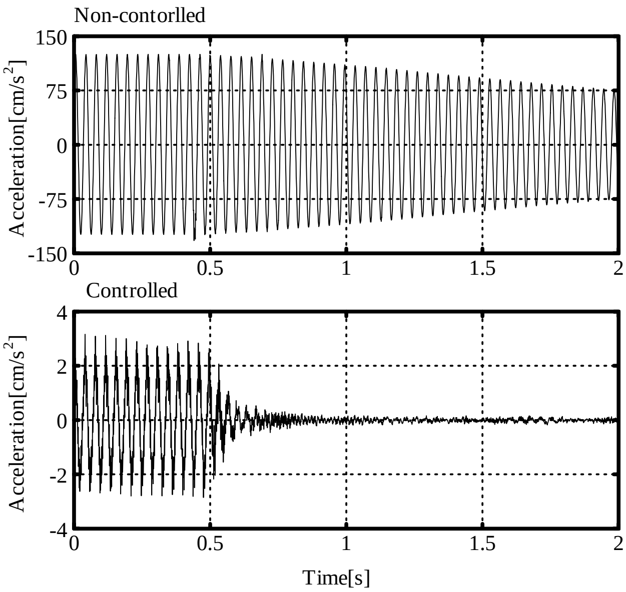

Figure 12 Free vibration test results

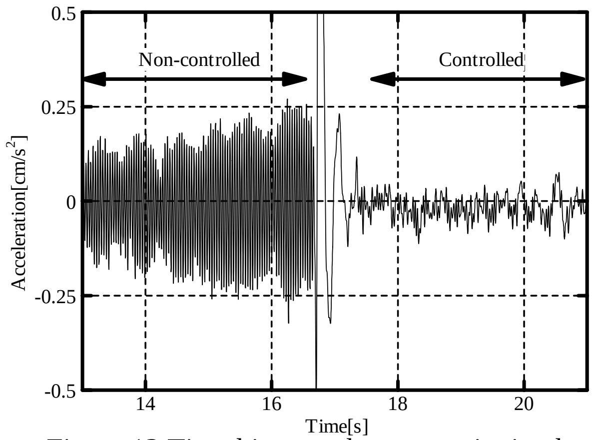

Figure 13 Time history when no excitation but ambient ground vibration was affected

4.2. Control Test Results

Excitation test was executed by a shaker, which was installed on the lower chord of the experimental model. Figure 11 shows a comparison of non-controlled and controlled against a harmonic excitation of the shaker, which force was 3N. The horizontal axis represents frequency and the vertical axis represents the vertical response acceleration on the center of the upper chord of the model. The controlled response acceleration was reduced to 1/32 of the non-controlled in the harmonic excitation of the shaker. Figure 12 shows the time histories of the response acceleration on the center of the upper chord of free vibration test in cases of the non-controlled and the controlled. The damping ratio of the 1st mode was calculated from the time histories, it became from 0.23% to 7.6%. It is shown that the proposed system demonstrates good vibration control performance. Figure 13 shows the time history of the response acceleration on the center of the upper chord in a case when no excitation but ambient ground vibration was affected. It was also confirmed an effective reduction against small vibration.

4.3. Verification of Analytical Model

Table 3 summarizes the first mode parameters of the large scaled experimental model, which are obtained from the modal analysis of the FEM model. The displacement of the actuator (x_a) is obtained using Eqn. 4.2.

$$x_a = f / k_m \tag{4.2}$$

where k_m is the stiffness of the mechanical part which the piezoelectric actuator is installed. The stiffness k_m is identified from an actual measurement of the mechanical part of the experimental model. Figure 11 also shows a comparison between the test results and the analytical results of the vertical acceleration of the chord in the cases of non-controlled and controlled. In this figure, the lines indicate the analytical results of non-controlled and controlled. Figure 14 and 15 show the comparisons between the test results and the analytical results of the force and the displacement of the piezoelectric actuator in frequency domain. Figure 16 shows time history responses against random excitation compared with the analytical results. High and low frequency components of the time histories in this figure were eliminated by band-pass-filter. It is confirmed that the analytical results are in good agreement with the test results of the vertical acceleration, the actuator force and the actuator displacement in both frequency and time domain. Validity of the analytical model for the proposed system is confirmed.

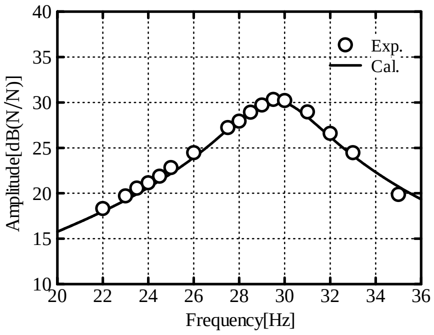

Figure 14 Actuator force response

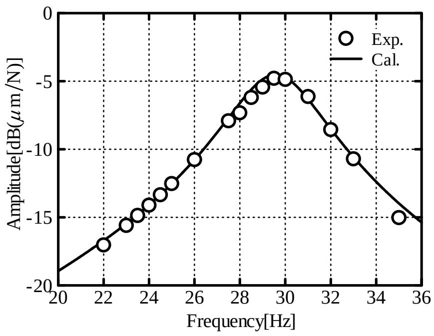

Figure 15 Actuator disp. response

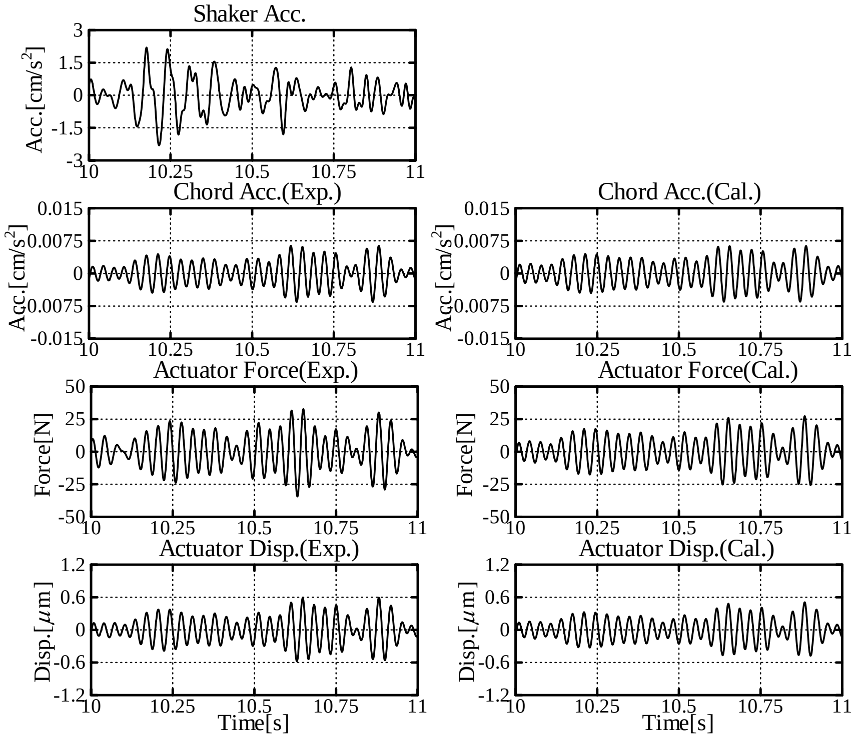

Figure 16 Time histories of random excitation

Table 3 Fist mode parameters of the experimental model (d=750mm)

Modal mass	719.5kg
Natural freq.	26.3Hz
Damping ratio	0.23%
Participation factor A	0.2826
Participation factor B	-0.2826
Participation factor of shaker	0.7319

5. ESTIMATION OF THE SYSTEM FOR REAL STRUCTURE

In order to estimate the system for real structure, we suppose a two-dimensional truss structure in figure 17 as the real floor structure of precision manufacturing facilities. Total weight of the structure is 142ton and span is 48m. Its natural frequency of the first mode is 4.7Hz and damping ratio is supposed 1%. The installation distance *d* is designed 2m. The stiffness of the mechanical part which the actuator is installed is estimated from the experimental model by using a geometrical similarity. Figure 15 shows the time history responses when a certain random vertical excitation is affected to the center of the structure. The response acceleration is reduced from 2.6cm/s^2 to 1.0cm/s^2. The maximum values of the actuator force and displacement are 5.4kN and 51μm. The estimated specifications of piezoelectric actuator for the real structure are summarized in Table 4. The proposed system for the real structure can be realized using the piezoelectric actuator on the market.

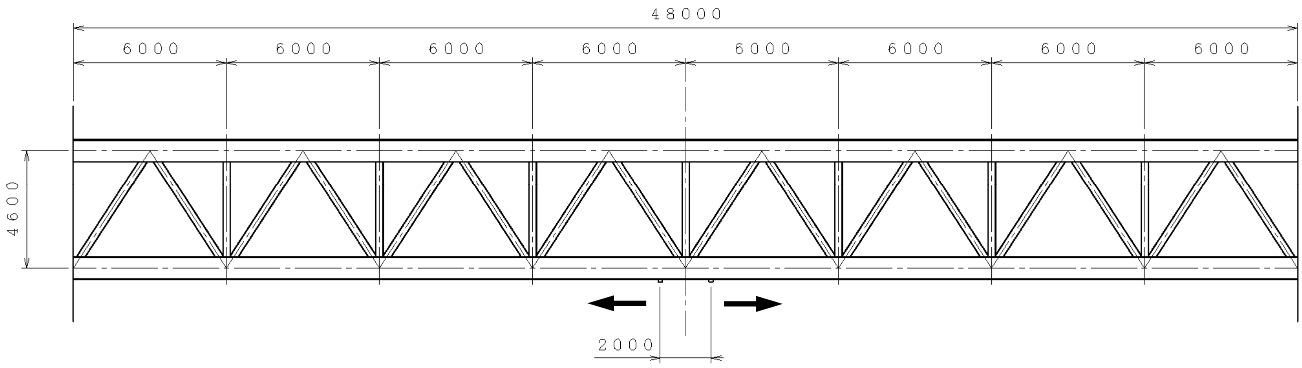

Figure 17 Supposed floor structure for the estimation of the proposed system

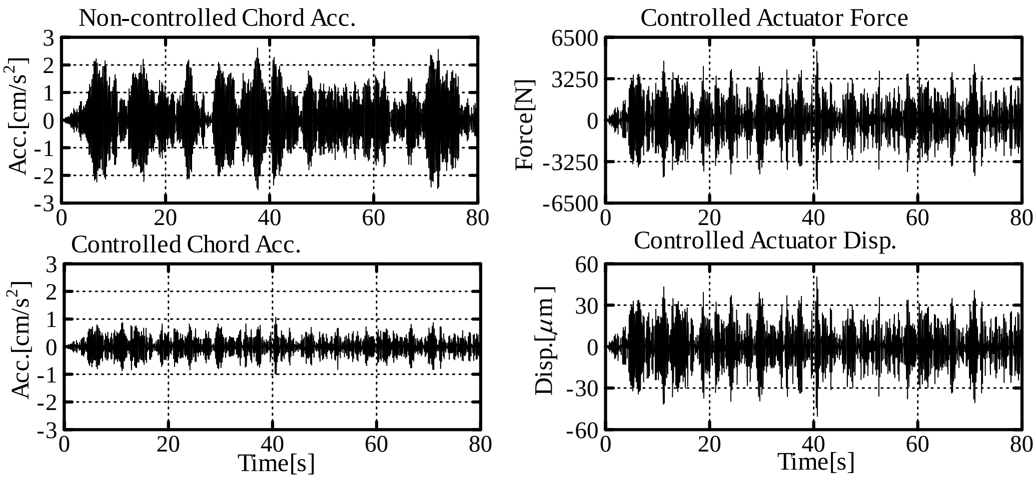

Figure 18 Estimated vibration control performance and actuator responses

Table 4 Specifications of the estimated actuator

Max. force	7kN
Max. displacement	80μm
Installation number	
Number of parallel	3
Number of series	3

6. CONCLUSIONS

In this paper, the active micro-vibration control system by using the piezoelectric actuators to suppress the vertical vibration of the floor in the precision manufacturing facilities was discussed. The participation factor of the actuator, which both ends are supported on the structure, was considered. It became clear where the best points for the both ends of the actuator were. Through the tests using the large-scaled experimental model, the excepted vibration control performance was achieved and the validity of the analytical model for the system was confirmed. The estimation of the system for the real structure was shown. These expressed the large feasibility of the proposed system.

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