Model-based adaptive feedforward compensation for disturbance caused by overturning moment in 2-dimensional shaking table systems

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Abstract
This paper presents a disturbance compensation approach using a model-based feedforward control with an adaptive algorithm for the two-dimensional (2D) shaking table systems. In the system, the overturning moment due to a specimen largely deteriorates the motion performance of the table, thereby decreasing the reproducibility of the desired earthquake acceleration. To solve the problem, first, as one of the disturbances, the overturning moment is modeled for the target shaking table system. In the modeling process, a physical model is derived based on the geometrical arrangement and the equation of motion, wherein the moment is modeled as a disturbance acting on each actuator. Based on this model, the feedforward compensators based on the mathematical disturbance model are adopted to cancel the disturbance. In addition, an adaptive algorithm is employed to reduce the effect of the modeling error and/or parameter variation. The designed compensators are validated by conducting experiments using a 2D laboratory prototype of the shaking table system.

Key words: Shaking table systems, Disturbance compensation, Modeling, Feed forward compensation, Recursive least-squares algorithm, Acceleration reproducibility

1. Introduction

Shaking table systems are widely used test facilities that evaluate the seismic capacity of structures; the system generally comprises an electro-hydraulic servo system including servo valves, actuators with cylinders and pistons, tables, and sensors (Nowak et al., 2000)(Ogawa et al., 2000). Because the purpose of the shaking table system is to determine the earthquake acceleration precisely of a target object (specimen) placed on a table, the control system design of the actuator is one of the key technologies for conducting accurate seismic tests. One of the main issues of the controller design of the shaking table system is the disturbance compensation because of the reaction force, which is due to the resonant vibrations and/or overturning moment of the specimen during the excitation. Hence, many control methodologies have been proposed to improve the reproducibility of the acceleration waveform.

In general, a displacement feedback control system with a proportional-integral compensator for the displacement reference obtained by double integrating the target acceleration waveform (Seki et al., 2008b)(Hironaka et al., 2010) or three variable controller (TVC) comprising TVC feedback and feedforward components (Tagawa and Kajiwara, 2007) has been designed as a basic control system. In addition, additional compensators are practically implemented using the basic control system to improve the disturbance compensation capability for dealing with various types of shaking tests. The reference generation through the iterative tests is one of the major approaches to compensate for the many uncertainties in the system; the approach is advantageous for specimens showing linearity in the test (Yasuda et al., 1991)(Tang et al., 2014). The reaction force feedback detected using the force sensor or disturbance observer can compensate for the reaction force because the resonant vibration of the specimen and/or uncertainties is in the low frequency range (Dozono et al., 2004)(Iwasaki et al., 2005). The feedforward compensation with an adaptive identifier is proposed to compensate for the
2. System configuration of plant and actuator control system

2.1. Configuration of target shaking table system

Figure 1 (a) shows an overview of the 2D laboratory prototype of the shaking table system. The table and specimen can be shaken both horizontally and vertically using the actuators. The system has a three-degree-of-freedom for the motion of the table.

reaction force as another approach, wherein the vibration model of the specimen is identified using an adaptive algorithm in real time (Seki et al., 2008a). The loop-shaping filter has an important role in suppressing the disturbance having a specific frequency, wherein the disturbance frequency can be automatically detected using a simple adaptive algorithm (Seki et al., 2009). The minimal control synthesis algorithm has the advantages of coping with the internal parameter variation, external disturbances, and nonlinear dynamics (Stoten and Shimizu, 2007). These additional controllers should be appropriately selected depending on the requirement and scale of the seismic test, e.g., the resonant frequency of a specimen, linear or nonlinear specimen, whether the test is a collapse test, and single or multi-axis shaking. In other words, various types of compensators should be prepared to consider the variation in the test conditions.

This paper presents a disturbance compensation approach using a model-based feedforward control with an adaptive algorithm for the multi-axis shaking table systems. The overturning moment due to the specimen is considered as one of the disturbance problems. Although various types of disturbances, such as the reaction force due to the resonant vibration of the specimen, simultaneously act on the actual system during the shaking test, this study considers only the overturning moment of a rigid specimen in the multi-axis shaking table system to simplify the problem and verify the effectiveness of the control approach. In other words, this control approach targets the seismic test without conducting repetitive tests. As a basic examination, a laboratory 2D (Y-axis: horizontal direction, Z-axis: vertical direction) shaking table system with a rigid specimen is used as the target system. Based on the system, this paper describes the modeling of the overturning moment and design the feedforward compensator for the moment. In the modeling process, a physical model is derived based on the geometrical arrangement and the equation of motion, wherein the moment is modeled as a disturbance acting on each actuator. In the controller design, because high stability of closed-loop control system is required for shaking table system, the feedforward compensators based on the mathematical disturbance model are adopted to cancel the disturbance. In addition, an adaptive algorithm is applied to reduce the effect of the modeling error and/or parameter variation. Because the proposed compensator is considered under the add-on feedforward scheme, the other problems (e.g., resonant vibration of the specimen) are solved to combine with the additional compensators such as the reaction force compensator for the resonant vibration of the specimen. The effectiveness of the proposed control approach is verified by conducting experiments using the laboratory 2D shaking table system.

2. System configuration of plant and actuator control system

2.1. Configuration of target shaking table system

Figure 1 (a) shows an overview of the 2D laboratory prototype of the shaking table system. Figure 1 (b) shows a schematic configuration of the system. The mechanism as a plant system comprises the shaking table with a specimen
Table 1  Physical parameters of 2D shaking table. These parameters are derived from dimensional data and material.

| Parameter | Value  |
|-----------|--------|
| $M_t$ [kg] | 7.10   |
| $M_s$ [kg] | 9.93   |
| $l_{th}$ [m] | 0.30   |
| $l_{tv}$ [m] | 0.02   |
| $l_{sh}$ [m] | 0.10   |
| $l_{sv}$ [m] | 0.10   |
| $l_b$ [m] | 0.20   |

Fig. 2  System configuration of uni-axis shaking table. A piston as the actuator is driven by a power amplifier through a servo valve. The actuator is controlled based on the feedback signals of the piston displacement and the pressure. The feedforward integrators are applied to the acceleration reference for a displacement control system.

coupled onto an actuator in the Y-axis (horizontal direction) and two actuators in the Z-axis (vertical direction) via the link mechanisms. The table is shaken both horizontally and vertically based on the target acceleration reference. Because the motion in the X-axis direction is mechanistically constrained, the system has a three-degree-of-freedom for the motion of the table. Table 1 lists the physical parameters of the mechanism, where $M_t$: mass of table, $M_s$: mass of specimen including beams, $l_{th}$: width of table, $l_{tv}$: height of table, $l_{sh}$: width of specimen, $l_{sv}$: height of specimen, and $l_b$: height of beams. Here, because the natural frequency of the specimen is higher than the frequency of the control bandwidth, the table and specimen can be modeled as rigid bodies.

2.2. Configuration of actuator system

Figure 2 shows an illustrative system configuration of an actuator system. The cylinder as an actuator is driven by a servo valve through a power amplifier based on the control signal. The actuator is conventionally controlled by a state feedback compensator using the measured displacement and the pressure of the cylinder. In the shaking table system, because the acceleration response of the specimen should be accurately reproduced, the acceleration is selected as the target signal. Hence, the feedforward integrators are applied to the acceleration reference, because the control system is designed as a displacement control system (Seki et al., 2009).

2.3. Modeling of actuator dynamics

The servo valve can be approximated by a constant gain within the control bandwidth as:

$$q_m = K_s u,$$  \hspace{1cm} (1)

where $q_m$: output flow rate of servo valve, $u$: control input, and $K_s$: gain of power amplifier and servo valve.

By considering the motion of the piston near its center, a mathematical model of the cylinder is given as:

$$q_m = A_d \frac{dy_d}{dt} + K_i \frac{dp_m}{dt} + C_{al} p_m,$$  \hspace{1cm} (2)

where $y_d$: displacement of piston, $p_m$: differential pressure in cylinder, $A_d$: piston area, $K_i$: internal stiffness of cylinder, and $C_{al}$: leakage coefficient in cylinder.
Fig. 3  Block diagram of actuator. Based on the physical model of an actuator, common block diagram of the piston displacement for the control input can be represented.

![Block diagram of actuator](image)

The driving force $f_a$ generated by the actuator is calculated using the following equation.

$$f_a = A_a p_m$$  (3)

The characteristic of the link coupling can be assumed as a rigid body, because the natural frequency exists in the higher frequency range. Hence, the relationship between the piston displacement $y_d$ and the table acceleration $y_a$ is given as follows.

$$y_a = \frac{d^2 y_d}{dt^2}$$  (4)

The table is driven by the force of the actuator through the coupling, wherein the effect of the friction acting on the piston can be neglected, as the frictional force is negligible compared to the driving force. Hence, the mechanical motion can be expressed using the following kinetic equations:

$$M y_a = f_a - d,$$

where $M$ is mass of piston, $M_t$ mass of table, $M_s$ mass of specimen, and $d$: disturbance caused by rotational motion.

Figure 3 shows a block diagram of the actuator, which is based on the aforementioned mathematical models. In the figure, $s$ represents Laplace operator. The model parameters for each actuator are identified by curve-fitting based on the responses of the measurement frequencies. Figure 4 shows the plant characteristics of the actuator in the Y-axis as an example, where the broken and solid lines indicate experimental result and the model characteristic, respectively. Table 2 lists the identified mechanical parameters.

### Table 2: Identified parameters of each actuator

| Actuator | $K_a$ [m³/s/√V] | $A_a$ [m²] | $C_{al}$ [m³/s/Pa] | $K_a'$ [m³/Pa] | $M_a$ [kg] |
|----------|-----------------|-----------|-------------------|----------------|----------|
| Y-actuator | $4.7 \times 10^{-4}$ | $5.3 \times 10^{-4}$ | $5.3 \times 10^{-4}$ | $5.3 \times 10^{-4}$ | $7 \times 10^{-2}$ |
| Z-actuator-1 | $2.8 \times 10^{-7}$ | $2.8 \times 10^{-7}$ | $2.8 \times 10^{-7}$ | $2.8 \times 10^{-7}$ | $3 \times 10^{-2}$ |
| Z-actuator-2 | $1.8 \times 10^{-8}$ | $2.5 \times 10^{-8}$ | $2.5 \times 10^{-8}$ | $2.5 \times 10^{-8}$ | $4 \times 10^{-2}$ |

2.4. Basic control system of actuator

Figure 5 shows a model matching the two-degree-of-freedom control system for the actuator, where $G_p(s)$: plant characteristic shown in Fig. 3, $K_p$: pressure feedback gain, $G_a(s)$: augmented plant including pressure minor feedback loop, $C(s)$: PI (Proportional-Integral) feedback compensator, $F(s)$: feedforward filter, $C_c(s)$: integral compensator, $r_a$:
acceleration reference, $r_d$: displacement reference, $u_{fb}$: output of $C(s)$, and $u_{ff}$: output of feedforward compensator. The feedback gains in $K_p$ and $C(s)$ are designed to ensure sufficient system stability. The gain-crossover frequency is set as 5 Hz in the system.

$F(s)$ is designed as a third order low-pass filter to allow the feedforward compensator to be proper. The filter parameters are set to satisfy the servo characteristic up to 10 Hz. The reference $r_d$ is generated by $r_a$ through $C_c(s)$, which is directly specified as the seismic waveform (Seki et al., 2009). Figure 6 shows the servo characteristics of the acceleration response for reference $r_a$, where the broken and solid lines indicate the experimental result and an approximate model as a dead time component, respectively, where the dead time corresponds to 11 sample delays (i.e. $L = 11T_s$, $T_s = 0.2$ ms). The figure shows that the required servo characteristic can be ensured up to 10 Hz.

### 3. Modeling of 2D shaking table considering overturning moment

#### 3.1. Modeling of disturbance due to overturning moment

From Fig. 1 (b), because the center of gravity integrated with the table and specimen does not correspond to an action axis of actuator, the moment is generated by the motion of the actuator. Hence, the tracking performance of the actuator deteriorates, because the moment acts on the control system as a disturbance. To compensate for the disturbance, the mathematical disturbance model is derived based on the geometrical arrangement and the equation of motion. Here, because the movement of the table is negligible, the effect of the link coupling can be ignored to simplify the model.

By excluding the inertial force of the piston, $f_i$ acting on the table and specimen from each actuator can be expressed as follows, respectively:

$$f_{ry} = \frac{M_{tg} + M_{by}}{M_b} f_{uy}, \quad (6)$$

$$f_{rz} = \frac{M_{cz} + M_{szi}}{M_z} f_{uzi}, \quad (7)$$
Fig. 7 Block diagram of disturbance due to moment. The disturbances for each actuator are derived based on the driving force signals of each actuator.

where subscripts \( y \) and \( z \) represent Y and Z axes, and \( i \) is number of the actuator in the Z-axis. Here, \( M_{z1} \) and \( M_{z2} \) indicate the mass driven by the actuators in the Z-axis, wherein each mass is set as half of the total mass.

The moment of an object at a point is derived by exterior product between position and force vectors at the point. In the 2D shaking table, the coordinate point and force can be defined as follows:

- The center of gravity integrated with the table and specimen is set as the origin.
- The coordinates of the action point obtained from the actuators are set as \( P_{y}(y_{y}, z_{y}) \) and \( P_{z}(y_{zi}, z_{zi}) \).
- The forces generated by the actuator are represented as \( f_{ry} \) and \( f_{rzi} \).

The moment acting about the X-axis \( m_{mx} \) can be formulated as follows.

\[
m_{mx} = -z_{y} f_{ry} + \sum_{i}^{2} y_{zi} f_{rzi} \quad (8)
\]

If the table and specimen rotate with an angle \( \phi \) about the X-axis, an equation of motion about the X-axis can be given as follows:

\[
I_{x} \frac{d^2 \phi}{dt^2} = m_{mx}, \quad (9)
\]

where \( I_{x} \): moment of inertia around X-axis. Assuming that \( \phi_{i} \) is small, the displacements of the actuators in the Y and Z axes, \( y_{dy} \) and \( y_{dzi} \), respectively, are expressed in the following equations, when the table and specimen rotate at an angle \( \phi \) about the center of gravity.

\[
y_{dy} = -z_{y} \phi_{x} \quad (10)
\]

\[
y_{dzi} = y_{zi} \phi_{x} \quad (11)
\]

Based on equations (8)-(11), the disturbances \( f_{my} \) and \( f_{mzj} \) due to the moment can be expressed as follows:

\[
M_{t1y} \frac{d^2 a_{y}}{dt^2} = f_{my}, \quad (12)
\]

\[
M_{t2z} \frac{d^2 a_{zi}}{dt^2} = f_{mzj}, \quad (13)
\]

where \( M_{tij} \): mass of table and specimen for each actuator. Based on equations (6)-(13), Fig. 7 shows a block diagram of the disturbance due to the moment. The parameters in the block diagram can be calculated using the dimensions and material of each part with the help of the CAD data.

Figure 8 shows a block diagram of the 2D shaking table combined with the block diagrams of Fig. 5 and Fig. 7, where the “Moment Calculation” indicates the block diagram of Fig. 7.

3.2. Validation of mathematical model

The mathematical model of the 2D shaking table in Fig. 8 is evaluated by comparing numerical simulations and experiments using the laboratory prototype. The following are the conditions of the experiments:

- The acceleration reference to the actuator in the Y-axis is given as a pseudo-earthquake waveform shown in Fig. 9, where the frequency component includes from 1 Hz to 10 Hz.
- The acceleration references to the actuators in the Z-axis are set as zero.

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Fig. 8 Block diagram of 2D shaking table including the effect of disturbance due to the moment. Moment calculation part is shown in Fig. 7.

Fig. 9 Acceleration reference and its spectrum for the actuator in the Y-axis, where upper figure shows acceleration waveform in time domain and bottom figure shows frequency analysis result. The frequency component includes from 1 Hz to 10 Hz.

- The gain-cross frequency is set as 5 Hz, considering the sufficient system stability.
- The compensators are implemented using a Digital Signal Processor (DSP) with a sampling period of 0.2 ms.

Figure 10 shows the comparative results between simulations and experiments, where Fig. 10 (a) indicates acceleration responses (upper) and their frequency spectrums (lower) of the actuator in the Y-axis, and Figs. 10 (b) and (c) indicate the ones of the actuators in the Z-axis. In the figure, chain, broken, and solid lines represent references, experimental results, and simulation results, respectively. In the results of the actuator in the Y-axis and the actuator-1 in the Z-axis, simulation results coincide with experimental results, although the actuator-2 in the Z-axis in Fig. 10 (c) includes a not small error due to modeling errors for nominal value, backlash of link coupling, misalignment of the mass, etc. In addition, the acceleration waveforms of the actuators in the Z-axis are deteriorated by the moment, although the acceleration reference is set as zero.

4. Adaptive feedforward compensation based on disturbance model

4.1. Disturbance model-based feedforward compensation

Based on the proposed disturbance model, a feedforward compensator is designed to cancel the effects of the disturbance. Here, a compensator design for the disturbance, which acts on actuator-1 in the Z-axis by the motion of the actuator in the Y-axis, is presented as an example.

Figure 11 shows a block diagram of the control system including the disturbance feedforward compensator, where
Fig. 10 Comparative results of acceleration waveform and frequency analysis between simulation (solid lines) and experiment (broken lines), where upper figures show acceleration references and responses and bottom figures show the results of frequency analyses. The simulation results include the error for experimental results due to the parametric error in the disturbance model.

$G_d$: transfer function of disturbance from Y-axis to Z-axis (actuator-1), and $C_d(s)$: feedforward compensator. From Fig. 7, $G_d$ can be formulated as follows.

$$\frac{f_{m1}}{f_{uy}} = G_d = \frac{(M_{y} + M_{sy})M_{sz1}z_{y1}f_{z1}}{M_{y}I_{x}}$$ (14)

To cancel the effects of the disturbance, the compensation signal $u_d$ can be calculated as follows.

- The acceleration response $f_{uy}$ is estimated using the acceleration reference $r_{uy}$ through the uni-axis shaking table model $e^{-Lt}$, because the servo characteristic of $y_{uy}$ for $r_{uy}$ can be expressed using the dead time component ($L$ corresponds to the 11 sample delays) as shown in Fig. 6.
- The estimation value of the driving force $f_{uy}$ is calculated by multiplying $M_{y}$ by piston acceleration.
- The estimation value of the moment force $f_{m1}$ is calculated using $G_d$.
- The compensation signal $u_d$ is generated through the compensator $C_d(s)$ to cancel the disturbance.

Using the actuator model from $u_{z1}$ to $f_{az1}$ in Fig. 11, $C_d(s)$ can be formulated as follows.

$$C_d(s) = G_{pc1}^{-1}(s)H(s)$$ (15)

$$G_{pc1}(s) = \frac{f_{ac1}}{u_{c1}} = \frac{K_wA_a}{K_{a}s + C_{a} + K_{pc1}K_w}$$ (16)

The filter $H(s)$ is designed as a second-order low-pass filter to allow $C_d(s)$ to be proper.

$$H(s) = \frac{\omega_n^2}{(s + \omega_n)^2}$$ (17)

$$\omega_n = 2\pi \times 100$$
By designing the compensators in the other actuator control systems similar to Fig. 11, the disturbances due to each moment can be suppressed.

4.2. Adaptive feedforward compensation

The disturbance model-based feedforward compensator includes the parameter errors in the disturbance model. To reduce the parameter errors between the disturbance model and the actual one, an adaptive algorithm is applied to identify the parameters of the disturbance model in real time as shown in the identification part of Fig. 11. In the part, $G_{h1}(s)$ is a transfer function of the output of $A_0$ for the control input $u_z$. The recursive least-square (RLS) method is used as an identification algorithm.

Although the control system of actuator-2 in the Z-axis is omitted in Fig. 11, the disturbance acting on actuator-1 in the Z-axis $f_{mz_1}$ is given using the following equation from Fig. 7.

$$f_{mz_1}(i) = k_{yy}f_{ay}(i) + k_{yz_1}f_{az_1}(i) + k_{yz_2}f_{az_2}(i)$$  \hspace{1cm} (18)

These parameters ($k_{yy}$, $k_{yz_1}$, $k_{yz_2}$) are estimated using the RLS in real time. To identify the parameters $k_{yy}$, $k_{yz_1}$, and $k_{yz_2}$ in Eq. (18), the following equation is defined:

$$f_{mc1}(i) = \hat{\theta}^T \xi(i) \quad (i = 0, 1, 2, \ldots)$$  \hspace{1cm} (19)

where

$$\hat{\theta} = [k_{yy} \quad k_{yz_1} \quad k_{yz_2}]^T, \quad \xi(i) = [f_{ay}(i) \quad f_{az_1}(i) \quad f_{az_2}(i)]^T.$$  

$i$ represents sampled-data. The estimate value of output signal $f_{mc1}(i)$ can be given as follows.

$$f_{mc1}(i) = \hat{\theta}^T \hat{\xi}(i) \quad (i = 0, 1, 2, \ldots)$$  \hspace{1cm} (20)

$\hat{\theta}^T$ is the estimate value of identification parameters $\theta^T$. As the cost function for the estimate value, the following equation expressed by sum of square errors is introduced.

$$J(\hat{\theta}) = \sum_{i=0}^{k}[\hat{\theta}^T \xi(i) - f_{mc1}(i)]^2$$  \hspace{1cm} (21)
To minimize the cost function of Eq. (21) for $\hat{\theta}$, following equation should be satisfied.

$$ \frac{\partial J}{\partial \theta} = 0 $$  \hspace{1cm} (22)

By solving Eq. (22), $\hat{\theta}^T$ can be derived by

$$ \hat{\theta} = \left( \sum_{i=0}^{k} \xi(i)\xi^T(i) \right)^{-1} \left( \sum_{i=0}^{k} \xi(i)f_{mc1}(i) \right). $$  \hspace{1cm} (23)

Here, $\Gamma(k)$ and $z(k)$ is defined as follows.

$$ \Gamma(k) = \left( \sum_{i=0}^{k} \xi(i)\xi^T(i) \right)^{-1} \hspace{1cm} (24) $$

$$ z(k) = \sum_{i=0}^{k} \xi(i)f_{mc1}(i) \hspace{1cm} (25) $$

By expressing $\hat{\theta}$ as $\hat{\theta}(k)$, Eq. (23) can be expressed as follows.

$$ \hat{\theta}(k) = \Gamma(k)z(k) \hspace{1cm} (26) $$

From Eqs. (24) and (25), $\Gamma^{-1}(k)$ and $z(k)$ are transformed to

$$ \Gamma^{-1}(k) = \sum_{i=0}^{k-1} \xi(i)\xi^T(i) + \xi(k)\xi^T(k) = \Gamma^{-1}(k-1) + \xi(k)\xi^T(k) \hspace{1cm} (27) $$

$$ z(k) = \sum_{i=0}^{k-1} \xi(i)f_{mc1}(i) + \xi(k)f_{mc1}(k) = z(k-1) + \xi(k)f_{mc1}(k) \hspace{1cm} (28) $$

By multiplying both sides of Eq. (27) by $\Gamma(k)$ from left side and $\Gamma(k-1)$ from right side, the following equation is derived.

$$ \Gamma(k-1) = \Gamma(k) + \Gamma(k)\xi(k)\xi^T(k)\Gamma(k-1) \hspace{1cm} (29) $$

Moreover, the following equation is obtained by multiplying both sides of Eq. (29) by $\xi(k)$ from right side.

$$ \Gamma(k)\xi(k) = \frac{\Gamma(k-1)\xi(k)}{1 + \xi^T(k)\Gamma(k-1)\xi(k)} \hspace{1cm} (30) $$

By substituting Eq. (30) into Eq. (29), $\Gamma(k)$ is derived as follows.

$$ \Gamma(k) = \Gamma(k-1) - \frac{\Gamma(k-1)\xi(k)\xi^T(k)\Gamma(k-1)}{1 + \xi^T(k)\Gamma(k-1)\xi(k)} \hspace{1cm} (31) $$

Based on Eqs. (26), (27), (28), and (31), $\hat{\theta}(k)$ can be calculated by the following equation.

$$ \hat{\theta}(k) = \hat{\theta}(k-1) - \frac{\Gamma(k-1)\xi(k)}{1 + \xi^T(k)\Gamma(k-1)\xi(k)}[\xi^T(k)\hat{\theta}(k-1) - f_{mc1}(k)] \hspace{1cm} (32) $$

The identification parameters are calculated using Eqs. (31) and (32). The estimation signals ($f_{mc1}, f_{ag}, f_{ac1}, f_{ac2}$) are used in the identification algorithm because these signals cannot be measured.

5. Experimental verifications

The effectiveness of the proposed feedforward compensation using the disturbance model is verified by conducting experiments using the laboratory setup. The following are the conditions of the experiments:

- The acceleration reference to the actuator in the Y-axis is set as a pseudo-earthquake waveform as shown in Fig. 9, where the frequency component includes the frequencies from 1 Hz to 10 Hz.
- The acceleration references to the actuators in the Z-axis are set as zero.
- The compensators are implemented using a DSP with a sampling period of 0.2 ms.

Although a real earthquake waveform is used to evaluate the performance, the pseudo-earthquake waveform including the frequency component from 1 Hz to 10 Hz is applied from the viewpoint of the limitation of the piston stroke and control bandwidth of actuator. Figure 12 shows the experimental results of the acceleration responses of the table and their frequency analyses. Figure 12 (a) shows the acceleration responses of the actuator in the Y-axis. Figures 12 (b)
Section 6: Conclusions

This paper presents a disturbance compensation approach using a model-based feedforward control with an adaptive
algorithm for the multi-axis shaking table systems. The overturning moment due to the specimen was considered as one of the disturbances. As a basic examination, a laboratory 2D shaking table system with a rigid specimen is used as the target system. For the target system, a physical model is derived based on the geometrical arrangement and the equation of motion, wherein the moment is modeled as a disturbance acting on each actuator. To compensate for the disturbance, feedforward compensators based on the mathematical disturbance model were adopted. In addition, the adaptive algorithm was applied to reduce the modeling error. The proposed control approach was validated by conducting experiments using the laboratory prototype. As a basic examination, this paper showed the effectiveness of the designed control system in compensating the overturning moment due to the specimen. Because the other deterioration factors, such as the reaction force due to the resonant vibration of the specimen, act on the actual system, the system should be combined with the other additional compensators and the effectiveness needs to be verified in the next step. In addition, it is necessary to extend this approach to simultaneously excite the 3D shaking table system. Finally, the proposed method in this paper is required to implement and evaluate using an actual system that is excited by a real earthquake waveform.

References

Dozono, Y., Konno, T., Horiuchi, T. and Katsumata, H., Shaking-Table Control by Real-Time Compensation of the Reaction Force Caused by a Nonlinear Specimen, Trans. on ASME, Journal of Pressure Vessel Technology, Vol.126, No.1 (2004), pp.122-127.

Hironaka, K., Miura, J., Asano, Y., Yasuda, K., Kouda, M. and Toyooka, A., Shaking-Table Control Taking Account of Reaction Force (Controller Design of Observer-Based Force Compensation), Transactions of the Japan Society of Mechanical Engineers, Series C, Vol.76, No.764 (2010), pp.842-850.(in Japanese)

Iwasaki, M., Itoh, K., Kawafuku, M., Hirai, H., Dozono, Y. and Kurosaki, K., Disturbance Observer-Based Practical Control of Shaking Tables with Nonlinear Specimen, Proceedings of the 16th IFAC World Congress (2005), 6pages.

Nowak, R. F., Kusner, D. A., Larson, R. L. and Thoen, B. K., Utilizing Modern Digital Signal Processing for Improvement of Large Scale Shaking Table Performance, Proceedings of 12th World Conference on Earthquake Engineering, No.2035 (2000).

Ogawa, N., Nagasaki, T., Sato, E., Ohtani, K. and Katayama, T., Development of Core Technology for 3-D 1200 Tone Large Shaking Table, Proceedings of 12th World Conference on Earthquake Engineering, No.2156 (2000).

Seki, K., Iwasaki, M., Kawafuku, M., Hirai, H. and Yasuda, K., Reaction Force Compensation Using Adaptive Identifier in Shaking Table Systems, Transactions of the Japan Society of Mechanical Engineers, Series (C), Vol.74, No.744 (2008a), pp.2052-2058.(in Japanese)

Seki, K., Kikuchi, M., Kawafuku, M., Iwasaki, M., Hirai, H. and Yasuda, K., Controller Design for Reaction Force of
Specimen Considering Disturbance Suppression in Shaking Table Systems, Transactions of the Japan Society of Mechanical Engineers, Series C, Vol.74, No.745 (2008b), pp.2206-2213.(in Japanese)
Seki, K., Iwasaki, M., Kawafuku, M., Hirai, H. and Yasuda, K., Adaptive Compensation for Reaction Force with Frequency Variation in Shaking Table Systems, IEEE Transactions on Industrial Electronics, Vol.56, No.10 (2009), pp.3864-3871.
Stoten, D. P. and Shimizu, N., The feedforward minimal control synthesis algorithm and its application to the control of shaking-tables, Proceedings of the Institution of Mechanical Engineers, Part I: Journal of Systems and Control Engineering, Vol.221, No.3 (2007), pp.423-444.
Tagawa, Y. and Kajiwara, K., Controller development for the E-Defense shaking table, Proceedings of the Institution of Mechanical Engineers, Part I: Journal of Systems and Control Engineering, Vol.221, No.2 (2007), pp.171-181.
Tang, Y., Shen, G., Zhu, Z.-C., Li, X. and Yang, C.-F., Time waveform replication for electro-hydraulic shaking table incorporating off-line iterative learning control and modified internal model control, Proceedings of the Institution of Mechanical Engineers, Part I: Journal of Systems and Control Engineering, Vol.228, No.9 (2014), pp.722-733.
Yasuda, C., Fujita, K., Kawamoto, K. and Araki, Y., Acceleration Waveform Control of Electric-Hydraulic Seismic Simulator (2nd Report, Application of Digital Iterative Equalization Method), Transactions of the Japan Society of Mechanical Engineers, Series C, Vol.57, No.536 (1991), pp.1221-1227.(in Japanese)