Development of Full Active Seismic Isolator via Mechanical Control

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Abstract
Reducing collapse damages of buildings and constructions by earthquakes is an important issue for human lives. Hydraulic active vibration suppression is made by mobilizing cylinders with the most suitable command signals through servo valves after computing data. These data are collected by various measuring instruments. However unforeseeable contingency takes place in natural disasters such as electric power failure or disconnection by sharp vibration.

If any of such disaster should occurs, the system does not function. The excess power would rather cause out of control. The system itself contains potential for harm. This study is to propose the simple and reliable control system for hydraulic cylinder using robust mechanical structure instead of electronic measurement and control. The results of model experiments and simulations presented in our previous works show good results. But it is necessary to raise the precision of simulation to put it into practical use. For the purpose of improving accuracy, we made experiments and simulations for various parameters such as inertial load mass, link ratio for control sensibility, and coefficient of oil switching flow and so on. Then, the control performance is evaluated.

Keywords: Active seismic isolator, Vibration control, Mechanical controller, Hydraulic spool, Link feedback, Earthquake

1. Introduction
Huge earthquakes in various parts of the world like the Great East Japan Earthquake cause enormous damages. To prevent building collapsing on such earthquakes, those techniques using hydraulic actuators for seismic isolation and vibration suppression are being proposed and put to practical uses. These implements prevent building quaking by driving actuators after scanning and computing data of seismic waves through sensors. Those techniques have a number of advantages to be able to set up control laws freely in computer. While life of building is over 50 years long, hardware and software of computers and peripherals remodeling come up every several years. Renewals and maintenances of equipment are now big problems. In case monitoring data for control exceed the limits of sensitivity or software contains bugs, there are possibilities of unexpected and confused behavior. Additionally, there could be dysfunction of the system itself due to disconnection of electricity. The importance of assured performance by seismic isolation and vibration suppression when earthquakes occur should be recognized again. The much simpler active seismic isolation and vibration suppression system should be needed. Prior to the present electronic control systems, mechanical linkages, pendulums and spool valves had been used. Those systems are much more reliable in terms of performance certainty or ease of maintenance.

In this study, a novel hydraulic active seismic control system consisting of simple mechanism instead of existing electronic active control systems is proposed. Seismic vibration entry and movement of object for control are connected with hydraulic switching valve. They are mechanical linkage and hydraulic directions are switched by driving actuators without using electronic control. This study is to develop such steady and stable seismic suppression control system.
2. Proposed mechanism

Figure 1 shows the concept of the mechanism proposed in this study. It is composed of hydraulic actuator to control force, spool valve to control actuator and linkage mechanism to transfer relative displacement between ground surface and building to spool valve. The inside of spool is divided into 3 parts by cylinder rod of actuator. Hydraulic power unit is connected with central room of spool. The link and spool framed by dashed lines behave as mechanism to control actuator. When ground vibrates relative displacement between ground surface and building is transferred to cylinder rod first. Accordingly spool valve is opened and flow passage from hydraulic power unit arises to put reverse pressure to the building against earthquake. Since the degree of spool valve opening controls the flow rate, the bigger displacement to spool is, and the stronger pressure from actuator to building is.

Shaking table  
Controller  
Structure model

Fig. 1 Schema of mechanism

\[ z \text{ [m]} : \text{Displacement of ground} \]
\[ x_1 \text{ [m]} : \text{Relative displacement of building} \]
\[ x_2 \text{ [m]} : \text{Relative displacement of cylinder rod} \]
\[ x_3 \text{ [m]} : \text{Relative displacement of spool} \]
\[ m_1 \text{ [kg]} : \text{Mass of building} \]
\[ m_2 \text{ [kg]} : \text{Mass of cylinder rod} \]
\[ m_3 \text{ [kg]} : \text{Mass of spool} \]
\[ k_1 \text{ [N/m]} : \text{Spring rate 1} \]
\[ k_2 \text{ [N/m]} : \text{Spring rate 2} \]
\[ c_1 \text{ [N-s/m]} : \text{Damping coefficient 1} \]
\[ c_2 \text{ [N-s/m]} : \text{Damping coefficient 2} \]

3. Simulation model
3.1 Modeling the proposed system

Parameters used for simulation are shown in Fig. 2.

In case left side pressure of actuator is \( p_1 \) and right side \( p_2 \), deferential pressure \( p \) is shown as Eq. (1).

\[ p = p_2 - p_1 \]  \hspace{1cm} (1)
In case left side pressure of actuator is $p_1$ and right side $p_2$, differential pressure $p$ is shown as Eq. (1).

$$p = p_2 - p_1$$  \hspace{1cm} (1)

Oil volume flowed from spool is proportional to valve opening and closing. Proportional constant of flow volume against spool opening and closing defines $q \text{ [m}^3\text{/s]}$. When difference between pressure to spool and pressure in actuator where oil is flowed becomes smaller, flow volume decreases, too. That indicates that relationship between pressure difference $p$ and flow volume $Q$ can be expressed as proportional equation. Constant proportion $r$ is defined as relationship between pressure difference in actuator $p$ and flow volume $Q$. Oil flow volume to actuator from spool is defined as $Q$. $Q$ is indicated positive when flowed from left to right. If volume in actuator is $V$ and oil and oil compression ratio is $b$, oil volume $Q$ flowed to actuator is expressed as Eq. (3).

$$Q = q(x_2 - x_3) - rp = d(x_1 + x_2) + \frac{bV}{2}\dot{p}$$  \hspace{1cm} (2)

where, $x_3$ is shown as Eq. (3) from $x_1$ and link ratio $h$.

$$x_3 = -hx_1$$  \hspace{1cm} (3)

From Eq. (2) and (3), force from motion table to structure model is shown as Eq. (4).

$$f_{ac} = ap$$  \hspace{1cm} (4)

Eq. (5) to (7) are simultaneous equations driven by each element parameter of proposed system. However, link weight and moment inertia are set to 0 in simulation.

$$m_1(\ddot{x}_1 + \ddot{z}) = -k_1x_1 - c_1\dot{x}_1 + f + f_{11} + ap$$  \hspace{1cm} (5)

$$m_2(\ddot{x}_2 + \ddot{z}) = -k_2x_2 - c_2\dot{x}_2 - ap$$  \hspace{1cm} (6)

$$m_3(\ddot{x}_3 + \ddot{z}) = f_{12}$$  \hspace{1cm} (7)

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**Hydraulic power unit**

- $p_1 \text{ [Pa]}$: Pressure of left side
- $p_2 \text{ [Pa]}$: Pressure of right side
- $V \text{ [m}^3\text{]}$: Volume of actuator
- $a \text{ [m}^2\text{]}$: Area of cross section of actuator
- $q \text{ [m}^3\text{/s]}$: Coefficient of spool rod displacement for velocity of flowing fluid
- $r \text{ [m}^3\text{(Pa} \cdot \text{s)}\text{]}$: Coefficient of pressure difference for velocity of flowing fluid

Fig. 2 Parameter of spool and actuator
where, inertia moment of link is 0 here. In case \( h = l_2 / l_1 \), following Eq. (8) is obtained.

\[
f_{11} = hf_{12} \tag{8}
\]

Eq. (9) is obtained by substituting Eq. (7) into Eq. (5) using Eq. (8).

\[
m_1(\ddot{x}_1 + \ddot{z}) = -k_1x_1 - c_1\dot{x}_1 + f + hm_2(\ddot{x}_3 + \ddot{z}) + ap + \frac{bV}{2} \dot{p} \tag{9}
\]

Eq. (10) is calculated by substituting \( x_3 = -hx_1 \) into Eq. (9) from relationship of link restraint.

\[
x_1(2m_1 + h^2m_3) + \ddot{z}(m_1 - hm_3) = -k_1x_1 - c_1\dot{x}_1 + f + ap + \frac{bV}{2} \dot{p} \tag{10}
\]

Oil volume flow from spool valve is proportional to valve opening. In case proportional constant is \( q \) [m\(^2\)/s] and dimension of hydraulic cylinder inner end face, oil volume flowing into hydraulic cylinder is obtained by Eq. (11).

\[
q(hx_1 + x_2) = a(\dot{x}_1 - \dot{x}_2) \tag{11}
\]

when if pressured difference between right and left cylinders is \( p = p_2 - p_1 \) [Pa] (\( p_1 \): right cylinder pressure, \( p_2 \): left), oil compressibility is \( b \) [Pa\(^{-1}\)], hydraulic cylinder content is \( V \) [m\(^3\)] and oil compressed when flowing into cylinder is taken accounts Eq. (12) is calculated.

\[
q(hx_1 + x_2) - \frac{bV}{2} \dot{p} = a(\dot{x}_1 - \dot{x}_2) \tag{12}
\]

where, in case oil pressure to be flowed is much higher, inflow volume decreases. Eq. (12) is calculated when deduction coefficient against oil pressure is \( r \) [m\(^3\)/Pa\cdot s]

\[
q(hx_1 + x_2) - \frac{bV}{2} \dot{p} = a(\dot{x}_1 - \dot{x}_2) + rp \tag{13}
\]

As the result, from Eq. (6), (10) and (13), Eq. (14) is obtained.

\[
\begin{bmatrix}
\ddot{x}_1 \\
\ddot{x}_2 \\
\ddot{z} \\
\dot{p}
\end{bmatrix} = 
\begin{bmatrix}
0 & 1 & 0 & 0 & a \\
-\frac{k_1}{m_1 + h^2m_3} & -\frac{c_1}{m_1 + h^2m_3} & 0 & 0 & \frac{m_1 + h^2m_3}{a} \\
0 & 0 & -\frac{k_2}{m_2} & -\frac{c_2}{m_2} & 0 \\
\frac{2hq}{bV} & 2a & \frac{2q}{bV} & \frac{2a}{bV} & \frac{2r}{bV}
\end{bmatrix}
\begin{bmatrix}
x_1 \\
x_2 \\
\dot{x}_1 \\
\dot{x}_2 \\
p
\end{bmatrix}
\tag{14}
\]


3.2 Model parameters setting

From Eq. (14), transfer characteristic between ground surface and building acceleration was simulated. At the same time another model without control parts was tested for comparing. That was composed of only mechanical seismic isolation system that is what we call passive earthquake-proof mechanism. Parameters for simulating are shown as Table 2. In this case, parameters were set to correspond to experimental installation of this study explained in Chapter 5. Because simulation was conducted using altered values of \( m_1 \) and \( h \), they are represented plural.

<table>
<thead>
<tr>
<th>( m_1 )</th>
<th>( m_2 )</th>
<th>( m_3 )</th>
<th>( k_1 )</th>
<th>( k_2 )</th>
<th>( c_1 )</th>
<th>( c_2 )</th>
</tr>
</thead>
<tbody>
<tr>
<td>260 kg</td>
<td>3 kg</td>
<td>7 kg</td>
<td>41058 N/m</td>
<td>36204 N/m</td>
<td>598 N/s/m</td>
<td>1000 N/s/m</td>
</tr>
<tr>
<td>386 kg</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>706 kg</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>( h )</th>
<th>( a )</th>
<th>( b )</th>
<th>( y )</th>
<th>( q )</th>
<th>( r )</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.0</td>
<td>( 4.1 \times 10^{-4} ) m²</td>
<td>( 7.14 \times 10^{-10} ) l/Pa</td>
<td>( 3.1 \times 10^{-5} ) m³</td>
<td>( 1.288 \times 10^{-2} ) m²/s</td>
<td>( 1.083 \times 10^{-9} ) m/(Pa·s)</td>
</tr>
</tbody>
</table>

4. Simulations

Parameter values proposed in the previous chapter are used for explanation about simulation conditions. In this study system development to make ground shaking insensitive to structure has been carried out. Structure acceleration peak divided by shaking table acceleration peak is defined as acceleration amplitude damping ratio. This ratio is used as performance evaluation index. This is obtained by Eq. (14) in simulation.

4.1 Effects of changes in Link ratio

In the past studies reports were made paying attention to influence to acceleration amplitude damping ratio by change of link ratio \( h \). Where, acceleration amplitude damping ratio was adjusted by link ratio change. If link ratio increases by 1.0, 1.2 or 2.0, acceleration amplitude damping increases. Results are shown in Fig.3. In this experiment \( m_1 \) was 260 kg. From Fig. 3, gain peak and natural frequency were confirmed to decrease with increasing link ratio. Acceleration amplitude damping ratio is shown in Table 2. In the case of digital control link ratio functions as servo amplifier gain.

![Fig. 3 Effectiveness of the rate of \( l_1 \) and \( l_2 \) (Bode plot)](image-url)
Table 2  Model parameters setting

<table>
<thead>
<tr>
<th>Link ratio h</th>
<th>Acceleration amplitude damping ratio</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.0</td>
<td>0.181</td>
</tr>
<tr>
<td>1.5</td>
<td>0.126</td>
</tr>
<tr>
<td>2.0</td>
<td>0.091</td>
</tr>
</tbody>
</table>

From above results simulation and experiment of lower acceleration amplitude ratio (link ratio \( h=2.0 \)) were followed.

4.2 Effects of changes in Mass weight

In this section change of \( m_1 \), control object, was simulated for validation. Comparing \( m_1 \) mass change influence with simulations and experiments, obtaining the relationship is the purpose. If the change to parameters by \( m_1 \) mass change can be clarified, it can be applied in case the object is the real building. Damping coefficient \( c_1 \) of ground and object is used as changeable parameters. It is difficult to measure damping coefficient \( c_1 \) of real building and ground. So, change of structure model mass was simulated.

Fig. 4 shows simulation results providing \( m_1=260 \) kg, 386 kg, 706 kg and \( h=2.0 \). Obviously natural frequency decreases with increase of mass. It is the same with real building. No alteration was added to damping coefficient \( c_1 \) in this simulation. Since the same spool and actuator were used in this experiment, equal specification was used for simulation. Practically it is as a matter of course that specification of actuator must be graded up with increase of mass. Simulation results show that damping effect comes down with mass volume. If experiment and simulation results become equal, damping characteristic can be analogized by simulations without producing real spool and actuator for experiments. Experiments based on Fig.4 results.

![Simulation Results](image)

**Fig. 4** Effectiveness of the mass of \( m_1 \)

Table 3  Model parameters setting

<table>
<thead>
<tr>
<th>( m_1 ) [kg]</th>
<th>Acceleration amplitude damping ratio</th>
</tr>
</thead>
<tbody>
<tr>
<td>260</td>
<td>0.178</td>
</tr>
<tr>
<td>386</td>
<td>0.139</td>
</tr>
<tr>
<td>706</td>
<td>0.081</td>
</tr>
</tbody>
</table>
5. Experiment and discussion
5.1 Experiment system

This section confirms the proposed system function actually by the proposed reproductions model. Fig. 5 shows the small scale system. Blue dashed portion is building, Motion table yellow is ground, and red is spool valve and link parts. The system is put on the rail. Table as ground is vibrated by hydraulic vibration unit. Vibration is controlled by frequency stabilizing displacement control. Constantly pressured oil into spool in control parts is provided from oil pump of hydraulic power unit. To reduce negative influence by friction in control parts, small radial ball bearing are inserted into hinge part of link. Both shaft centers of spool are fixed. Dry bearing bushes are set in for matching strictly spool shaft and case. Packing is not used at all for spool because of friction. Performance of spool hydraulic circuit switching was improved by remaking spool width overlapping a little.

![Experimental apparatus](image)

Fig. 5 Experimental apparatus

5.2 Experimental method

Vibration experiments: Input displacement amplitude 4.5 mm, Proportional sweep sine curve 0~60 seconds 0.5~5 Hz and 60~120 seconds 5~0.5Hz. Link ratio was 2.0. For comparison, the same experiments were conducted as passive earthquake-proof system after dismounting damping mechanism. Sweep sine curve was made from Agilent’s Multi-function generator. Hydraulic cylinder was used actuator. Hydraulic pressure was 10 MPa for actuator and 4 MPa for spool. To compare with simulations structure model mass was varied. Experiments were conducted under conditions of 260 kg, 386 kg and 706 kg.

5.3 Experimental results
5.3.1 Mass weight 260 kg

Fig. 6 shows displacement amount measured in experiments and Fig. 7 shows BD gained from those results. Damping around resonance point is effectual when frequency increases, but performance in low frequency zone comes down. As coordinate here is displacement, amplitude is high. But acceleration is small in low frequency zone.

The reasons of damping characteristic difference between to and from have not been proved yet.
5.3.2 Mass weight 386 kg

Fig. 8 shows amount of displacement measured in experiment. Fig. 9 shows BD from those results. Although big difference between simulation and experiment results had been predicted, changes of $c_1$ did not cause such effect.
5.3.3 Mass weight 706 kg

Such big mass of this experiment system had not been planned at first. Intensity of guidetrails and wheels were too low over specification limits in 706 kg case to conduct satisfactory experiments. Abnormal random power of friction seemed to defy measurement in numerical terms. Experiment data were gained but it did not go far enough to analyze them. New experiment system is required for bigger mass for future. Not only system intensity but enough power of actuator worth mass will be needed for clear isolation effects. More oil quantity for actuator will be needed. Eventually spool bore must be widened and oil pressure must be boosted.

6. Effects of other parameters

Among parameters shown in Table 2, such parameters relating to control object as building mass $m_1$, quake-proof gum spring constant $k_1$ and damping constant $c_1$ cannot be changed. Types of hydraulic oil are limited and compression ratio cannot be changed either. Among remaining parameters, characteristic change when link ratio $h$
and spool mass $m_1$ are changed has been confirmed in the past studies. Since flow volume reduction $r$ against pressure varies by oil volume proportional constant $q$ against spool opening, $r$ is treated together with $q$. Therefore changeable parameters are cylinder rod mass $m_2$, quake-proof gum spring constant $k_2$, damping constant $c_2$, actuator cross area $a$, cylinder volume $V$ and proportional constant $q$ against spool opening.

<table>
<thead>
<tr>
<th>Table 4</th>
<th>Change of parameter</th>
</tr>
</thead>
<tbody>
<tr>
<td>Possible to change</td>
<td>Impossible to change</td>
</tr>
<tr>
<td>$q$</td>
<td>$m_1$</td>
</tr>
<tr>
<td>$m_2$</td>
<td>$k_1$</td>
</tr>
<tr>
<td>$k_2$</td>
<td>$c_1$</td>
</tr>
<tr>
<td>$c_2$</td>
<td>$b$</td>
</tr>
<tr>
<td>$a$</td>
<td></td>
</tr>
<tr>
<td>$V$</td>
<td></td>
</tr>
</tbody>
</table>

Focusing attention on actuator cross area $a$ and flowing volume proportional constant $q$ against spool opening is important. Those are related much with power given to building model from actuator.

Actuator cross-section area $a$ is varied by proportional constant $q$. To explore those relationships, changing both parameters were simulated. Gain peaks change plots by simulations are shown in Fig. 10. A fixed actuator cross-area $a$ and change proportional constant $q$ shows a line from blue to red, and adversely a fixed $q$ and changeable $a$ shows a line in black.

![Fig. 10 Effectiveness of $a$ and $q$ (Peak of gain)](image)

From Fig.10 it was confirmed that the most appropriate isolation performance can be obtained with increasing both $a$ and $q$. Using curve lines shown in Fig.10 enables to select spool and actuator of this system.
7. Conclusion

The vibration control system proposed in this study has been demonstrated that it is much reliable because computerized control is unnecessary. It was also turned out that this system can also reduce natural frequency and acceleration comparing with ordinary damper. The purpose of this study was to enhance accuracy by numerous changes of parameters for practical use. In this study two kinds of factors such as structure model mass, 206 kg, 386 kg and 706 kg, and link ratio, 1.0, 1.5 and 2.0 were simulated and model experiments were conducted based on the results of simulations. It was found that link ratio performs the same function with servo amplifier gain in the case computerized control.

For this reason there are possibilities that the system vibrates itself by making gain too much. Much attention has to be paid for gain adjustment. So it can be said that simulations fill very important role. Packing was not used for spool and the system ran very smoothly with oil friction only. Friction of this part was so small that experiments results could have been predicted by simulations. In the case of the heaviest structure model, 706 kg, analysis could not be made by strength poverty of experiment system. Experiment results of Structure model 260 kg and 386 kg showed they fit approximately with simulations. In other words if it is within tolerable range, it may be possible to design experiment system based on simulations.

8. Future works

Combination of link, spool and actuator is applicable to mass damper system. Mechanism proposed in this study can be applied to such areas that need vibration reduction as architecture, large-scale construction, heavy machinery, automobile and rail.

It can be used for seismic isolated building against earthquake or strong wind. It can also be applied to container crane for wind, cargo transportation for vibration, ordinary building for earthquake or heavy machinery for shaking. If simulation can improve reliability after this, application of this system should be considered.

In order to realize full scale model experiments, it is necessary to carry out tests with intensity system that can be sustainable to intermediate level mass between present system and full scale model. Assumed specification of intermediate level experiment system will be mass 50,000 kg, actuator 100 kN and laminated rubber as support.

References

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