THE ENERGY DISSIPATION BEHAVIOR OF DISPLACEMENT DEPENDENT SEMI-ACTIVE HYDRAULIC DAMPER

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The functionalities and energy dissipation capability of Displacement dependent semi-active hydraulic damper, DSHD are analyzed and discussed with complete experiment which involves the factors to affect time-delay of DSHD, and the reasons to cause sliding of the oil cylinder under the dynamic state. The test results show that 1) Time-delay occurs in unloading period because of the hysteresis phenomenon. Herein, Kelvin solid is approved to simulate this process and shows that stiffness of the brace and damping coefficient of oil will lead time-delay; 2) It's important to avoid sliding of the cylinder, caused by insufficient oil pressure and residual air in the oil cylinder and pipe, for achieving excellent performance of DSHD. These phenomena can be improved by applying appropriate pre-pressure to the oil tank.

Key Words : sliding of the oil cylinder, hydraulic mechanism, time-delay, directional control valve, check valve, relief valve, energy dissipation, insufficient saturation degree, hysteresis

1. INTRODUCTION

In the past years, several external excitations with strong magnitude occurred over the worldwide and devastated some countries. It collapsed many structures and destroyed priceless properties, even killed people. Therefore, it is always an important subject for engineering research to improve the capability of seismic resistance in the structure to reduce the damage caused by disasters. Based on demand for power supply and the sort of control force, the techniques of vibration reduction are classified as three types: active control, passive control and semi-active control[1)–4)]. The main aim of these techniques is to enhance the energy dissipation capability and consequently improve the seismic performance of structures[4)]. The hydraulic cylinder of Displacement Dependent Semi-Active Hydraulic Damper (DSHD) shown in **Photo 1** can be imagined as a damper connector with characters of loosing and closing switch via a flexible directional valve to connect the structure and enhanced components (e.g., the brace). Therefore, the elastic deformation of the enhanced components, but not viscosity of fluid in the cylinder, is the resource to yield control force acting on the structures.

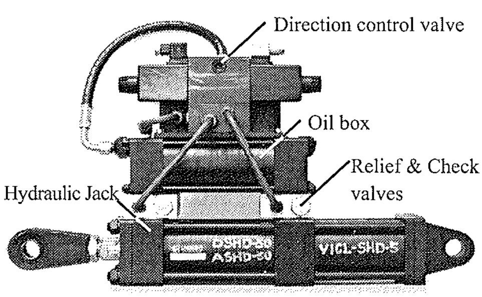


Photo 1 Sideview of DSHD

The research and development of hydraulic mechanism is widely applied in the field of mechanical engineering because it preserves the advantages of small volume, high power, low vibration, easy automation, and good durability. Meanwhile, it is also used to develop semi-active hydraulic dampers for civil engineering structures, e.g., Taylor Device[5)], Semi-active hydraulic damper, SHD[6)–7)], Magnetorheological Damper, MRD[8)], Electrorheologic Damper, ERD[9)], to achieve shock absorption and energy dissipation. Professor Kobori proposed AVS (active variable stiffness)[10)] that implement an off-on switch valve to enable oil pressure components as the joint of controllable switch. AVS cooperates with the switch algorithm as control laws to acquire perfect efficiency of shock absorption in the real structure. Further-

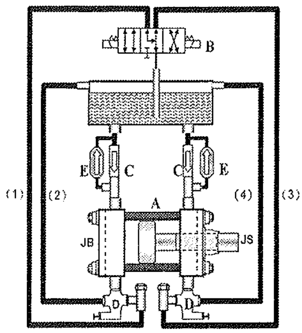


Fig.1 The organization of DSHD

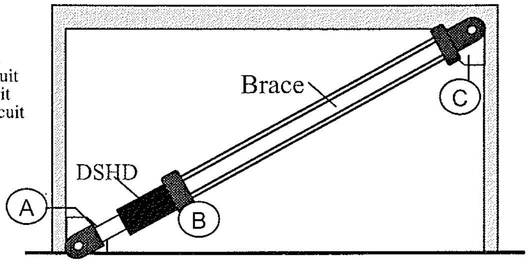


Fig.2 Concept of DSHD application

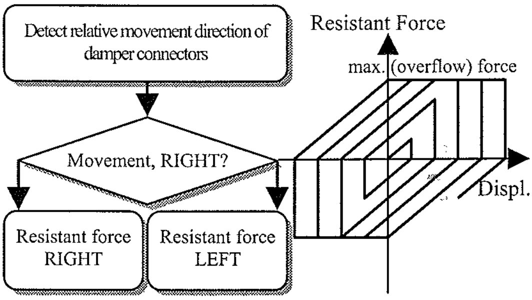


Fig.3 DSHD control algorithm and typical hysteretic loops

more, Yang[11] improved the control law of AVS and suggested resetting semi-active damper that is better than switching semi-active damper.

Shih and Sung[12] proposed a device, Displacement Dependent Semi-Active Hydraulic Damper (DSHD) to improve the energy dissipation efficiency. This preliminary research showed the excellent capability of its energy dissipation through experimental data and numerical studies. But, the energy dissipation performance of DSHD is mainly affected by the time delay and slide of oil cylinder. Therefore, this paper extends previous research in achievement of DSHD[12] to investigate the factors that affect time-delay of DSHD and reasons that cause slide of oil cylinder according to experimental data. Particularly, the Directional Valve and the Check Valve of DSHD are combined with special design of oil pipes to simplify off-on switch control. DSHD cooperates with both energy dissipation characteristics and performance of above mentioned valves to be an energy dissipation device with high efficiency. Therefore, the modified DSHD is recommended for promoting the hysteresis energy dissipation behavior under the dynamical status.

2. STRUCTURE AND ENERGY DISSIPATION BEHAVIOR OF DSHD

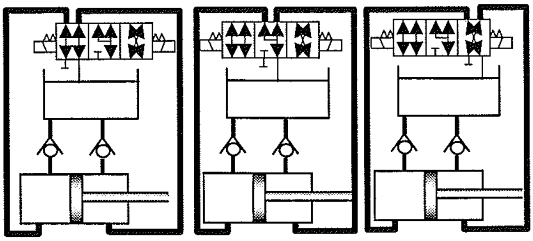
(1) Principle and organization of DSHD

Herein, DSHD device shown in **Photo 1**, and the system of DSHD is composed of the hydraulic jack, directional valve, check valve, relief valve, and oil cylinder shown in **Fig.1**.

Particular current is input through directional valves and DSHD will be switched among functions of "tensile restraint," "compressive restraint," and "free restraint". As the scheme shown in **Fig.2**, DSHD (section AB) is connected with the brace (section BC, the resistant component) and the structure.

Fig.3 presents the force-displacement relationship of the brace action. The detailed operation process in **Fig.2** is explained by four stages as followings. Stage I: the structure moves to the right, the length of AC increases but AB is restricted to elongate. The resistant force (tension) is then produced by elastic deformation of the brace BC. Stage II: as the structure yields the longest displacement of brace to the right, the length of AC approaches to the maximum while the largest tension exists in the brace BC. If the status of AB function changes from "tensile restraint" through "free restraint" to "compressive restraint", then the restriction on AB is released, i.e., AB begins to deform, which is transferred from BC in Stage I, and the force in BC is freed. Stage III: as onset of the structure oscillates to the left, the length of AC decreases. The brace BC is forced to deform with compression because AB is unable to shorten. Stage IV: AC will perform the shortest length with maximum compression existing in the brace BC while the structure yields the longest displacement to the left. By reverse process of Stage II, if the status of AB function changes from "tensile restraint" through "free restraint" to "compressive restraint", then the restriction on AB is released. i.e. AB begins to deflect again, which is transferred from BC in stage III, and the force in BC is freed. The relationship between bracing force and displacement is shown in **Fig.3**.

It reveals that the vibration energy is exhausted while resetting the brace before yielding, because the movement of fluid oil in the damper works while adjusting the switch of directional valve. **Fig.4** shows the variety of connection states of DSHD device.



(a) free elongated (b) free restraint (c) free shortened

Fig.4 Different connection state of DSHD device

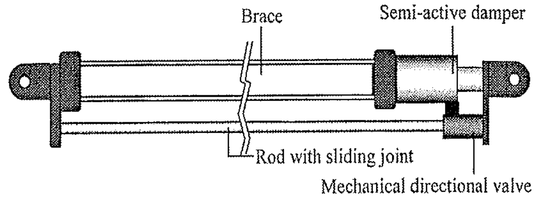


Fig.5 DSHD brace with integrated motion sensing mechanism

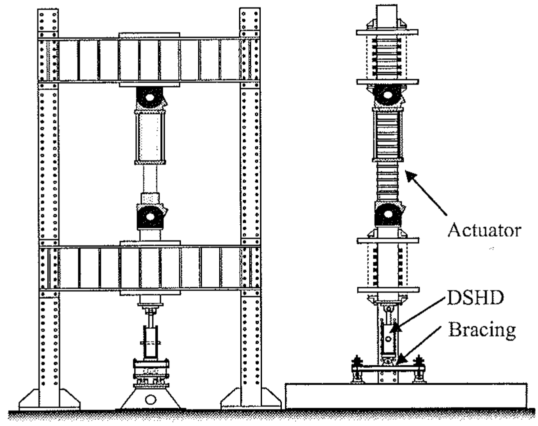


Fig.6 Experimental Setup

(2) The function of primary components of DSHD

Usage of the check valve is to control fluidity of the left and right circulating oil pipes. The directional control valve will control three DSHD situations, "tensile restraint," "compressive restraint," and "free restraint," while restricting the direction of fluid flow. Because of "one-way release" on the restraints, DSHD system can halve the control procedure. It is helpful to simplify the device design. The extremely little discharge of the throat valve is set to guarantee that DSHD can be auto homing without manual operation. The purpose of installing relief valve is to limit the maximum output force of oil cylinder. Furthermore, it can provide the passive energy dissipation device while the directional control valve is out of control.

The directional control valve – the core of DSHD – can receive the signal of direct voltage, auxiliary hydraulic source, and mechanical output to alter the position of the valve, as well as changing the direction of the resistant force in the oil cylinder. It is quite simple to determine the control laws of the optimal direction of resistant force. The direction of relative motion corresponding to both apexes of the damper position is the key to change the direction of resistant force in the damper for yielding the maximum negative work. The procedure of control rules and typical hysteresis loop of DSHD are shown in **Fig.3**.

The advantages of DSHD are summarized as follows:

- Simplicity: Only the signal reflecting the direction of relative motion is required for direct output-feedback. The directional control valve can take the signal of mechanical output without computer. This device can be performed without power supply for the entire system. **Fig.5** reveals that the DSHD brace structure integrates with the mechanical directional control valve, brace, and motion sensing mechanism.
- Autonomy: DSHD installed in the structure is independently operated to present the function of maximum energy dissipation.
- Low Power Requirement: External power input can be ignored.
- Safety Protection: Associated with the passive control model, this damper still works for maintaining structural safety when the control mechanism is damaged.

Comparing with ADAS (Added Damping And Stiffness)[13] and viscous damper which are controlled under the same conditions of vibration force, stiffness and amplitude, the previous research[12] of DSHD components is observed to provide following characteristics: 1) the maximum ratio of energy dissipation in variety of stiffness, 2) the minimum bracing stiffness with identical energy dissipation, 3) the certain ratio of energy dissipation approaching to 1.0 even under strong excitation, 4) the steady hesteresis energy dissipation while magnifying the vibration amplitude, and 5) the optimal ratio of energy dissipation for any critical excitation force. Herein, the experiment is setup to investigate the energy dissipation behavior of DSHD and verify detailed functions as well as related problems.

3. INSTALLATION OF TEST

In practical, time-delay of the component function and sliding of the oil cylinder will have an effect on capability and behavior of energy dissipation. Therefore, an experiment architecture shown in **Fig.6** is designed to study the reasons of time-delay, sliding of the oil cylinder, and hysteresis behavior.

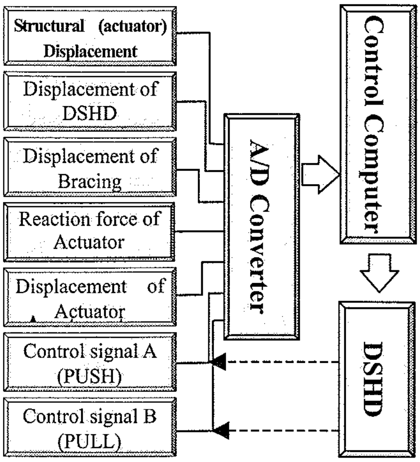
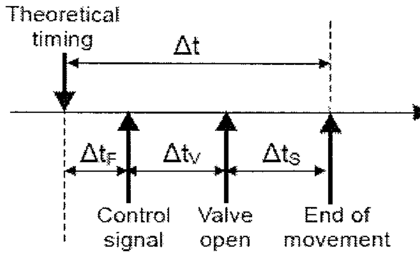
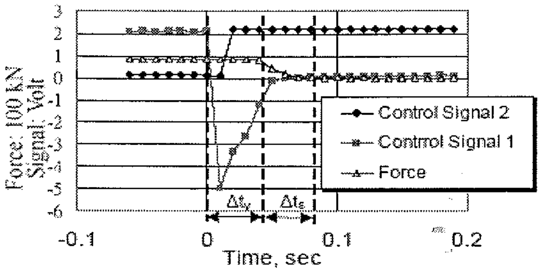


Fig.7 Flow of response and control signals



(a) Definition of time delay



(b) Experimental data

Fig.8 Time delay (a) Definition (b) Measured signal

The entire test system contains a MTS-1MN actuator, which includes Temposonic displacement gauge and Load Cell, a MTS407 controller, a control computer, a signal receiver, a direct current power supply, a manual cutoff switch of electromagnetism valve, a UPS (unceasing power system) device, three displacement gauges, a test frame, and DSHD, etc. The fiowchart of signal measurement, measured the potential difference on electromagnetic valve, is expressed as **Fig.7**.

DSHD connected with the brace and main structure is designed to constantly provide resistant force to structural vibration. The elastic deformation in the brace, which is substituted by a simply supported beam to match actual mechanical deflection, leads the force to ignite the dissipation of structure energy. Three displacement gauges, from top to bottom, are installed in this test to measure the deformation of the structure, sliding displacement of the oil cylinder and deformation of the brace. Especially, the measure values of upper displacement gauge and actuator should be identical. Sliding of the oil cylinder and deformation of the brace are directly measured by the gage to prevent errors induced by the gap of connection. Meanwhile, the signal of electromagnetism switch is obtained by directional control valve to study timing inaccuracy due to magnetic hysteresis induced by electromagnetism valve.

4. ANALYSIS OF TEST RESULTS

The factors of time-delay and sliding of the oil cylinder, discussed as follows, are related to the whole structure.

(1) The factors of time-delay

The more time-delay is occurred, the worse control is served. Therefore, a test is setup for DSHD to catch on the factors of this matter. The whole procedure can be scheduled as following three variables: Δt_F – time-delay when the filter is processing the signal of vibration; Δt_V – time-delay of magnetic hysteresis as the directional control valve is changing the switch; and, Δt_S – required duration while the brace is pushing or pulling the oil cylinder to the neutral position after switching the valve. The total time-delay Δt shown in **Fig.8** can be defined as following.

$$\Delta t = \Delta t_F + \Delta t_V + \Delta t_S \tag{1}$$

a) Time-delay when the filter is processing the signal of vibration

The seismic sensor can detect vibration and transfer it into digital signal through wires. The analog or digital filter that filters noise in transformation algorithm may cause time-delay. For example, if the sample of signal is 1 000 Hz, the norm of 20 points can be selected to filter the electronic noise for more than 50 Hz. In fact the real frequency of vibration is usually dominated by natural frequency of the structure that about 0.5 to 1 Hz. Therefore, the response is assumed as linear if the period is less than 0.02 second and the signal filtered by normalization should occur 0.01 second ago. So, Δt_F is estimated as 0.01 second.

b) Time-delay of magnetic hysteresis as the directional control valve is changing the switch

When certain voltage charges the electric magnet in the electromagnetism valve, the magnitude of magnetic field will increase as raising the current. Because of electromagnetic induction, however, between the current and magnetism, the complete switching of the valve must not be counted until the magnetic field in the electric magnetic is under stable condition. To calculate above time-delay, a section of signal wire is simultaneously connected to the signal wire of the electromagnetism valve and a control computer is installed to collect information via signal retrieving system. **Fig.8** shows the typical signal of the electromagnetism valve and the acting history of internal force in DSHD. It explains why the value of internal force in DSHD begins to change while the valve finishes the switching, and the result satisfies the assumption in **Fig.8** (a).

c) Time-delay while regressing the internal force of the brace

The brace is usually considered as weightless with large stiffness corresponding to the entire structure. The released elastic brace can be connected with damping mechanism in DSHD and be simulated by Kelvin Solid[14]. As following is the first order Ordinary Differential Equation (ODE) derived to formulate relationship of bracing force and displacement at the instant moment while the brace is relaxed.

$$q_1\dot{\varepsilon} + q_0\varepsilon = \sigma \tag{2}$$

where,

q_1 is total damping coefficient in the oil system;

q_0 is bracing stiffness;

ε is bracing displacement;

σ is external force applied on DSHD.

Substitute Eq.(2) into initial displacement ε_0 ($= \sigma_0/q_0$) and let the external force on DSHD be 0. Herein, σ_0 is the internal force before relaxing the brace. Then, Eq.(2) is solved as

$$\varepsilon = \varepsilon_0 e^{-\frac{q_0}{q_1}t} \tag{3}$$

Let's define that the regressing process stops as the deformation is one-tenth of initial displacement and time t equal to Δt_S, then we have

$$0.1\varepsilon_0 = \varepsilon_0 e^{-\frac{q_0}{q_1}\Delta t_S} \tag{4}$$

i.e.,

$$\Delta t_S = \frac{q_1}{q_0}\ln 10 \tag{5}$$

Defined the relaxation ratio as follows:

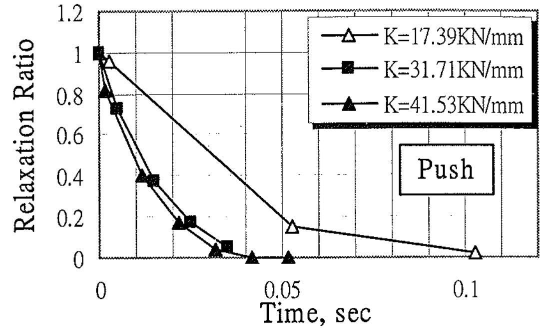
$$\theta(t) = \frac{F(t)}{F_0} \tag{6}$$

Where,

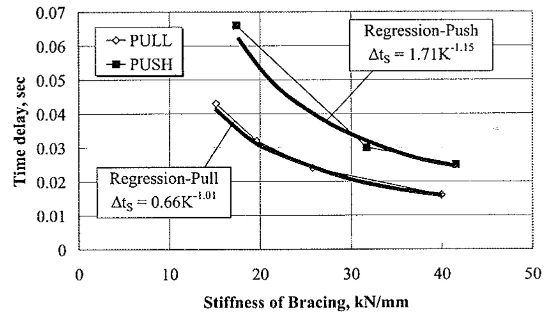
F_0 is the bracing force before the movement of brace;

$F(t)$ is bracing force after t seconds movement of bracing.

It is observed that Δt_S is in inverse proportion to the bracing stiffness but in direct proportion to damping coefficient of the oil system. **Fig.9** demonstrates the relationship between time and relaxation ratio in variety of bracing stiffness.



(a) Relaxation time history



(b) Regression of time delay with respect to stiffness

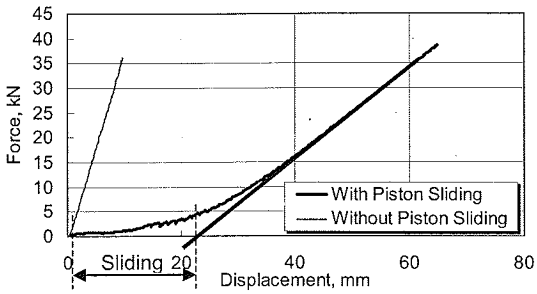
Fig.9 The influence of stiffness of bracing on time delay due to relaxation

Table 1 Time delay induced by relaxation of bracing force

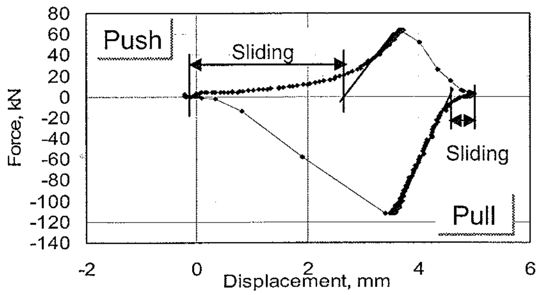
Bracing Stiffness (kN/mm, push/pull)	Δt_S (sec)	Regression Curve ($\varepsilon(t)/\varepsilon_0 = e^{-\alpha t}$, α)
59.43/39.94	0.036/0.016	N.A/145
41.53/25.78	0.025/0.024	93/93
31.71/19.62	0.030/0.032	78/71
17.39/15.12	0.066/0.043	35/54

The unbalanced piston set (large and small areas for push and pull direction, respectively) is installed in the oil cylinder to yield different damping coefficient and spring stiffness, both are also large and small in push and pull direction, respectively. Therefore, the following discussion referred in **Fig.9** involves push and pull parts.

Table 1 lists the experimental data regarding time-delay Δt_S and relaxation ratio. Substituting the experimental data into the exponential decay function for recursive statistics, the relationship of bracing stiffness and time-delay is shown in **Fig.9**. It is experiential that the time-delay corresponding to the tensile side with strong stiffness (K = 59.43 kN/mm) is pretty long. The reason is the residual air in the cylinder induces the spring stiffness but much less than the stiffness of the brace and oil pressure. Therefore, the duration is much longer than other cases for relaxation ratio that is approaching to 1/10. Furthermore, according to **Fig.9**, Δt_S is inversely proportional to the

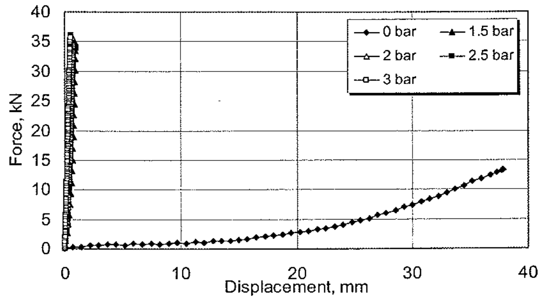


(a) sliding of oil cylinder

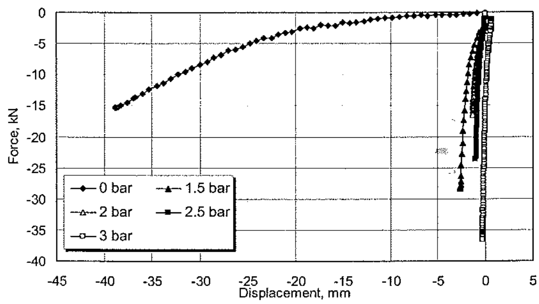


(b) residual air in the cylinder

Fig.10 The Influence of piston sliding on the hysteretic behavior of DSHD



(a) push



(b) pull

Fig.11 Sliding of piston rod

bracing stiffness to 1.01 and 1.15 power for pull and push cases, respectively. This result is quite close to the assumption of Kelvin Solid, i.e., Δt_S is in inverse proportion to the bracing stiffness.

Theoretically, the pre-pressure in the oil cylinder will affect the compressive capability of oil as residual air, particularly, exists in the cylinder. It means that the pre-pressure may change the total stiffness of oil and induces time-delay. In this test, oil pressure will be incrementally pre-pressured from zero to three atm. However, this factor can be neglected since Δt_S is altered less than 0.01 second. Similarly, the location of piston, which is another possible factor to affect the compressive capability of oil, can also be ignored due to the test.

(2) Sliding of the oil cylinder

Sliding of the oil cylinder is the displacement, which occurs as the piston of the oil cylinder moves without action of external force in loading process. The relationship of force-displacement due to sliding becomes two curves while the oil cylinder works in the beginning of loading process. It causes the loss of DSHD power even then failure of capability.

Fig.10 shows the influence of piston sliding with respect to the hysteretic loop of DSHD.

The sliding path of cylinder will become a stable curve gradually while the displacement of DSHD is increasing. Thus the sliding increment is defined as the intercept of the displacement axis intercrossed at the point that is extended from asymptote of this stable curve. Quite a little sliding of the oil cylinder occurs on the side that bears the push loading. It is about 2.8mm and may reduce the control force more than 67%. The sliding on this side is effected by a large amount of residual air in the cylinder. Even though the sliding also happens on the pull side because of the gap in the check valve, it is just about 0.7 mm that is too few to change the control force.

a) Influence by pre-pressure in the oil box

The oil box is sealed and pre-pressured to prevent sliding of the oil cylinder because of insufficient saturation degree. Experimental results shown in **Fig.11** indicate that the sliding increments on both of the pull and push sides will be decreased by ascension of the pre-pressure while the oil cylinder is gradually stressed from 0 to 3 atm of pre-pressure.

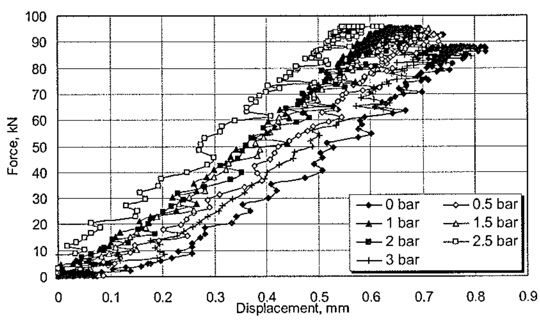
This phenomenon proves that insufficient saturation degree will obviously induce sliding and increasing pre-pressure of the oil box to 1.5 atm can effectively solve it.

b) Influence by residual air

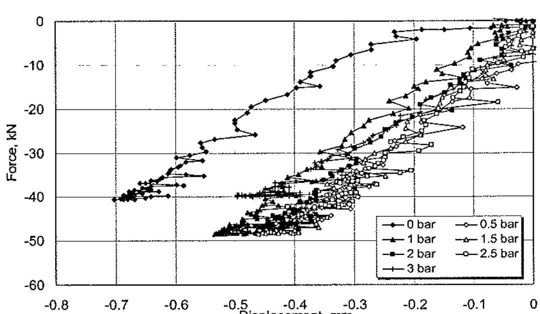
The sliding displacement of the oil cylinder is still about 3–5 mm even though it has been initially vanished by pressuring the oil box for insufficient saturation degree. Therefore, it is necessary to adjust pre-pressure existing in the oil box gradually under perfect oil supply condition. **Fig.12** implies that sliding displacement of the oil cylinder without pre-pressure is larger than that with pre-pressure because of residual air in the cylinder. Consequently, 1–3 atm of pre-pressure is suggested to get rid of residual air and avoid vacuum phenomenon in the check valve.

c) Influence by spring stiffness of the check valve

There is a spring in the check valve to jostle the piston of valve. If oil flows to the spring, the fluid may pass the valve by overcoming the compression in the spring. Inversely, the fluid will stop if the oil flows away from the spring. DSHD has to absorb oil via the check valve so that the vacuum phenomenon will take place to induce sliding of the oil cylinder if pre-pressure in the spring is too large. Nevertheless, if the pre-pressure in the spring is too low, the check valve might fail to stop sliding of the oil cylinder without sufficient oil as DSHD begins to move backward. Herein, pre-pressure of the check valve is setup in two cases, 0 and 5 kg, to eliminate this defect. **Fig.13** indicates that the pre-pressure can truly reform this shortcoming since the sliding displacement for 0 kg case is a little bit larger than 5 kg case



(a) push



(b) pull

Fig.12 Influence of pre-pressure of oil box

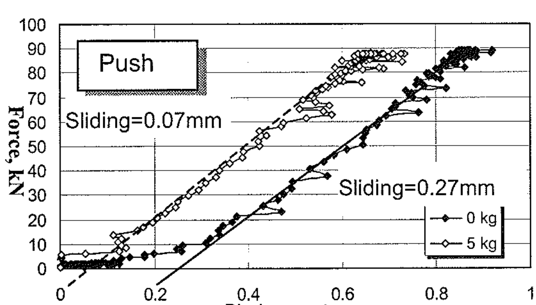


Fig.13 Influence of check valve spring force

5. VERIFICATION OF HYSTERETIC BEHAVIOR

The bracing stiffness, vibration amplitude and the maximum resistant force of DSHD are the primal parameters to control the energy dissipation performance of DSHD. By varying these three parameters, the theoretical and experimental methods are used to acquire the ratio of energy dissipation under the conditions of semi-active and passive hysteresis behavior with overflow. Based on the result shown in **Table 2**, the experimental results for semi-active and passive dissipation energy are highly close to the theoretical solution. In the other word, the actual performance of energy dissipation matches the predication.

The ratio of energy dissipation is defined as follows:

$$E.R. = \frac{QED}{UPLH \times UPLD} \tag{7}$$

Where,

QED is quantity of energy dissipation per each cycle of hysteresis loop;

$UPLH$ is the difference of the upper and low limit, devoted force of hysteresis loop;

$UPLD$ is the difference of the upper and low limit displacement.

Furthermore, **Table 2** denotes that the overflow behavior of semi-active energy dissipation follows the increment of overflow pressure. However, the ratio of energy dissipation for this model is still higher than that for the passive model. Contrarily, the low overflow pressure represents the large ductility ratio while the ratio of energy dissipation is almost the same. Therefore, this emergent passive model still appears

Table 2 Energy ratio of experimental and theoretical result

Semi-Active Mode Amplitude=10 mm, Stiffness=30/16 kN/mm		
Overflow F_{max} Push/Pull, kN	Energy Ratio, E.R Experiment	Energy Ratio, E.R Theory
26/32	0.962	0.964
54/64	0.924	0.928
75/94	0.879	0.895
100/133	0.846	0.851
137/189	0.794	0.789
Correlation Factor ρ		0.994

Passive Mode Amplitude=10 mm, Stiffness=30/16 kN/mm		
Overflow F_{max} Push/Pull, kN	Energy Ratio, E.R Experiment	Energy Ratio, E.R Theory
26/32	0.925	0.937
54/64	0.846*	0.884
75/94	0.716*	0.826
100/133	0.681*	0.768
137/189	0.679	0.713
Correlation Factor ρ		0.939

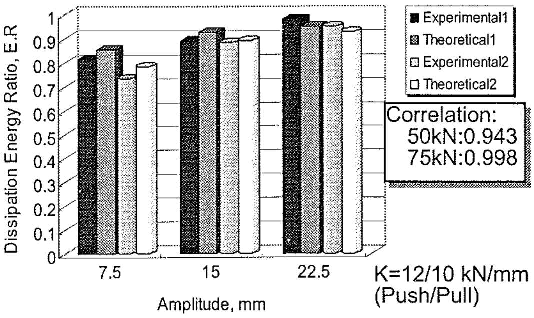


Fig.14 Dissipation energy ratio for various amplitude

good control effects under the strong shock. **Table 2** draws an inference that the more scope of earthquake will lead the larger ratio of energy dissipation.

In advanced, **Fig.14** indicates that the ratio of energy dissipation follows the increment of amplitude and approaches to 1 under the overflow condition of semi-active model. Meanwhile, the correlation factor of both analyses tends to 1 with two sets of experimental and theoretical models. Therefore, DSHD performs superior energy dissipation capability, which, especially, can be promoted under the large earthquake.

6. THE EFFECTS OF VIBRATION CONTROL WITH DSHD

(1) The influence of time-delay of DSHD

Time-delay problem causes the loss of energy dissipation capacity of DSHD. The loss quantity is related to natural frequency of structure. Considering periodic simple harmonic motion of structure, the loss ratio of time delay to energy dissipation capacity can be expressed as follows:

$$\eta = \frac{\Delta E}{E} = 1 - \cos^2(2\beta\pi), \quad \beta \leq 0.25 \qquad (8)$$

Where,

ΔE is loss of energy dissipation capacity in one cycle;

E is the optimal maximum energy dissipation quantity in one cycle;

β is the ratio of time delay to period of the structural free vibration ($\beta = \Delta t / T$, Δt is time delay and is period of the structural free vibration).

When quantity of time delay reaches quarter period of structural free vibration ($\beta = 0.25$), the loss ratio achieves 100%. That is, DSHD is loss of the energy dissipation function. Given the experiment in this research as example, the time delay quantity is about 0.05 seconds. Therefore, the loss ratio of energy dissipation capability is 90.5%, 34.5% and 9.5% for 0.25 seconds, 0.5 seconds and 1.0 seconds period of structure, equipped with DSHD respectively. This shows that DSHD, proposed in this research, is suitable for structures with long period.

(2) The influence of sliding of oil cylinder

Considering periodic simple harmonic motion of structure again, the influence of sliding of oil cylinder to loss ratio can be expressed as follows:

$$\eta = \frac{\Delta E}{E} = \gamma - \frac{\gamma^2}{4}, \quad \gamma \leq 2 \qquad (9)$$

Where,

γ is the ratio of sliding displacement of oil cylinder to structural amplitude of vibration.

This equation shows that the influence of sliding of oil cylinder on energy dissipation effect is deserved to be heeded. The relative story drift of high-rise building is comparatively small. If the sliding displacement of oil cylinder is set to be constant, the loss ratio of energy dissipation of the proposed damper is relatively large. To diminish this influence, structure, equipped with DSHD, should install displacement amplification mechanism, such as elbow mechanism, to enlarge structural displacement quantity. In addition to improve the problem of slide of oil cylinder, this method can also enhance the effect of damping force.

7. CONCLUSION REMARK

Several conclusions are summarized from experimental analyses and remarked as follows:

1. The time-delay of Kelvin Solid model can simulate the instant behavior as switching the directional control valve. It proves that the bracing stiffness is in inversely proportional to the time-delay.
2. The more bracing stiffness and less area of piston of oil cylinder cause less required flow rate by switching of DSHD, lead to shorter time-delay. Moreover, the higher pressure of oil and the larger flow rate of valves result in reducing quantity of time-delay. Practically, relative to mass, bracing stiffness is very large that causes reaction frequency to be large, too. Therefore, the time-delay would not be serious for DSHD.
3. The area of piston and oil pressure will affect the output force capacity of DSHD. Consequently, higher oil pressure and less area of piston should be adopted to design oil cylinder. Secondly, discharge of valve should be paid heed to use larger flow rate to reduce viscous force when changing the direction of resistant force.
4. Residual air in the oil cylinder will reduce the total stiffness of DSHD and incorrectly magnify the time-delay.

5. Residual air in the oil cylinder will cause sliding of the cylinder. It must be vanished by fulfilling oil carefully as well as keeping away from incoming air.
6. The sliding displacement of the oil cylinder will be controlled by spring stiffness of the check valve. The optimal solution should be determined by more tests.
7. The sliding of oil cylinder has relations with residual air, saturated degree of oil and transformation of oil pipe. The simple and effective method of solving these problems is to raise pre-pressure (standard pressure) in the procedure of oil filler. The pre-pressure in the oil box should be added with 1–3 atm to minimize the influence of residual air.
8. The gap between the cylinder axis and components will cause sliding of the oil cylinder and affect the control effect.
9. Improper installation of the damper will increase residual air existing in the cylinder as well as change the sliding displacement, and also reduce capability of energy dissipation very much.
10. The hydraulic damper can avoid energy loss due to friction and low-cycle fatigue. The ductility of DSHD is infinite, i.e., there's no restriction on fatigue and ductile problems to satisfy the requirement of seismic reduction for any type of excitation.

The damping force in DSHD is dependent with the displacement. It is quite suitable for the long-period structures.

ACKNOWLEDGMENT: The National Science Council of Taiwan, R.O.C. supported this research through grant No.NSC-92-2625-z-327-003. This support is gratefully acknowledged.

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(Received October 3, 2003)