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Seismic control of multi-storied buildings by means of accelerated liquid mass damper

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ABSTRACT: Resisting force characteristics of an accelerated liquid mass damper of open circulation type, a kind of passive vibration control devices, is discussed basing on pressure gradient in discharge tubes measured by sinusoidal excitation tests. Empirical resistance rules are derived, which is valid in weakly turbulent region of pipe flow. Also "hanging wall system" is proposed, by which resisting force of the damper is enabled to be exerted directly onto the top of high buildings within the scope of conventional building construction technique.

1 INTRODUCTION

The author has been developing "accelerated liquid mass dampers", a kind of devices for passive control of structural vibration (Kawamata, 1987 and 1989). The basic concept was originally presented by the author (Kawamata, 1973).

The device is consisted of a pair of piston-cylinders which is installed in a structure so as to conduct reciprocal movement of pistons in accordance with structural vibration. There are two different basic types of the device.

In the "direct type" of the damper, single conduit tube of small diameter connects directly both the cylinders and the cavity if filled with liquid such as water, oil etc. When the reciprocal movement is forced to the pistons, it causes the liquid in the conduit tube oscillatory flow of high velocity, thus exerting inertial as well as viscous resistance to the vibrating system. Incorporation of this type of damper leads to elongation of natural period, reduction of effective external and inertial forces of excitation and restriction of resonant peak amplitude of response. Though the device shows powerful effect to vibration induced by external forces like wind or excitation of machines, it has a drawback in the case of random excitation of earthquake that transmissibility of the damped system does not converge to zero towards high frequency domain.

In contrast, in the "open circulation type" of the damper, the cavity of the cylinders is joined to a liquid reservoir via a check valve which is closed by positive pressure in the cylinder. Liquid squeezed out from the cylinder is lead back to the reservoir through a long discharge tube. The movement of the pistons gives rise to intermittent one-way flow in the discharge tubes. Its effect of controlling structural responce to excitation of earthquake waves was demonstrated by model experiments.

One of the objects of the present paper is to discuss the results of pressure measurement in the vibration tests of the open circulation damper and to examine the characteristics of pressure gradient in the intermittent sinusoidal pipe flow. Also, empirical resistance rules and equivalent viscous damping evaluation are presented.

On the other hand, the demand for application of vibration control techniques is very high in the field of multi-storied high buildings and the efficiency of the applied energy dissipating or active control devices largely depends upon the scheme by which the devices are incorporated in the building skeletons. The fundamental vibration mode of high buildings can be most efficiently controlled by exerting controlling force directly onto the top of the buildings. "Tendon systems" were proposed by Abdel-Rohman and Leipholtz (1978) and by Kawase et al. (1990) for the same purpose. However, They seem to bring difficulties: in the view point of site execution for the former and of design flexiblity for the latter.

The second object of this paper is to propose an alternative scheme of the damper installation which enables us to transmit the resisting force to the building top by means of "hanging wall system", which can be realized within the scope of conventional building construction technique.

A test result demonstrating the effectiveness of the hanging wall system is also presented.

2 RESISTING FORCE CHARACTERISTICS OF AN OPEN CIRCULATION DAMPER

To investigate the resisting force characteristics of the damper of open circulation type, a model damper was subjected to harmonic motion and pressure measurement was conducted.

2.1 *Out line of tests*

An acrylic model damper, shown in Fig.1, 36cm wide and 15.5cm high, was composed of cylinders of 4.4cm inner diameter and a hollow piston sealed by wound silk thread bands. Check valves were provided on the piston heads.

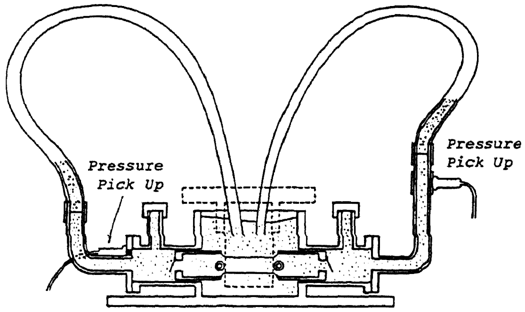


Fig.1 Tested model damper

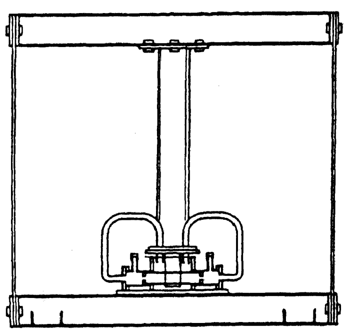


Fig.2 Test frame with the damper

The damper was installed in a steel portal frame, 1.0m wide and high, in a manner shown in Fig.2. Columns of the frame were formed by 9mm thick flat bars and the top beam was connected to the damper by a central rigid member. The frame was subjected to harmonic excitation on a shaking table, testing frequency being 2, 3, 4 and 5 Hz.

Using strain transducers, inner pressure was measured at an outer end of a cylinder and at a point of a discharge tube about 15cm apart from the cylinder head.

Three different tubes were used being filled with water and water solution of glycerin. The parameters of the tested damper are listed in Table 1.

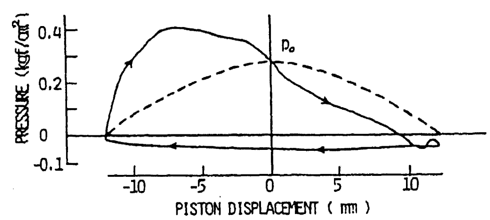
Table 1 Test parameters of damper

Inner diameter (mm)	6.0	9.0	12.0
Length (cm)	85.5	162.0	187.5
Liquid	water	40%G	60%G
Kinematic viscosity at 20°C (cm^2/sec)	0.0100	0.0314	0.0833

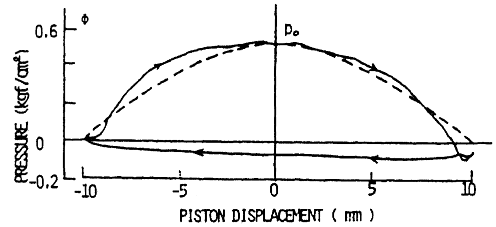
G: glycerin

2.2 *Property of displacement-pressure loops and energy dissipation*

Fig.3 shows typical examples of loops representing relatoion between piston displacement and pressure in the discharge tubes. Also, dotted lines represent cosine curves corresponding to fictititous cases of viscous damping.

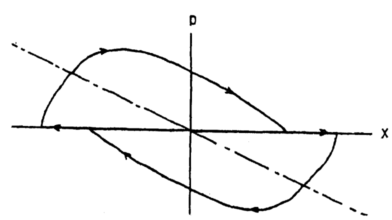


(a) 9mmΦ tube, l=154cm, 60% glycerin: 2Hz excitation



(b) 9mmΦ tube, l=154cm, 60% glycerin: 5Hz excitation

Fig.3 Pressure-piston displacement loops



12mmΦ tube, water: 2Hz excitation

Fig.4 Loop of positive pressure combined for both cylinders

As seen in Fig.3(a), when driven in lower frequencies, the pressure loops take extremely asymmetric shape, suggesting that the inertial resistance of the liquid flow is added in accelerating phase and is subtracted in decelerating phase to and from the viscous resistance. For the oscillation of higher frequencies, however, the asymmetricity diminishes.

Though the liquid goes to freely discharging flow of zero pressure when the check valve is opened, low negative pressure caused by saction in the cylinder occurred. The maximum negative pressure was almost the same for all the testing cases.

When the positive pressure loops for both the cylinders are combined, a closed loop is formed with an axis inclined against the coordinate axes, as shown in Fig.4. It means that the damper provides the structure with an additional inertia which leads to elongation of natural period of vibration. However, this inertial effect seemed to be almost negligible in the frequency range over 3 Hz.

The pressure drop within the cylinder by flow contraction can be estimated by comparing the pressures measured in the tube and at the cylinder head. The pressure drop was negligibly small for the tubes of 12mm and 9mm diameter. In the case of 6mm diameter tube, some pressure drop within the cylinder was observed and it was almost proportional to the one along the discharge tube.

Dissipated work by the positive pressure evaluated by the area enclosed by the loop is greater than the one of the viscous resistance represented by the dotted lines. In the "*weakly turbulent region*" *to be mentioned* in the next paragraph, the ratios of the former to the latter were approximately 1.0, 1.1 and 1.2 for the tubes filled with water, 40% glycerin and 60% glycelin solutions, respectively.

2.3 Pressure gradient-velocity relation in the pipe flow

By the test results, relation between sectional mean velocity of the liquid flow and pressure gradient along the discharge tube is investigated with reference to the inner pressre at zero point of piston displacement, i.e. p_o in Fig.3.

Fig.5 shows an example of relation between p_o and the amplitude of piston displacement, $\overline{x}$. The p_o-$\overline{x}$ diagrams start in small displacement with very low pitched slope and soon go into straight lines, transitting to nonlinearly increasing portion for larger displacement. These three different phases are supposed to correspond to laminar, weakly turbulent and finally properly turbulent regimes of flow. A schematic diagram of the phase transition is shown in Fig.6.

Transition from the weakly turbulent to turbulent phase seems to take place when Reynolds number defined based on thickness of boundary layer, R_δ, reches 500. R_δ is defined as

$$R_\delta = \overline{u}_m \delta / \nu \qquad (1)$$

where $\overline{u}_m$: amplitude of mean velocity
ν : kinematic viscosity
δ : thickness of boundary layer

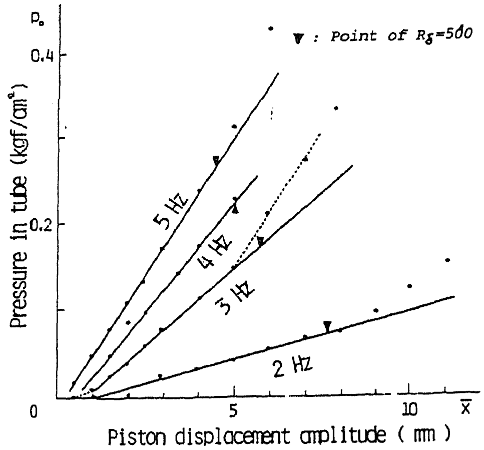


Fig.5 Pressure-displacement amplitude relation

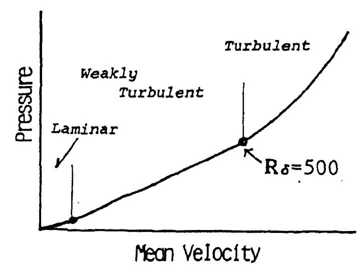


Fig.6 Phase transition

ω : cicular frequency

$$\delta = \sqrt{2\nu / \omega} \qquad (2)$$

For simplicity, p_o-$\overline{x}$ relations can be reinterpreted without much loss of accuracy that they are represented by straight lines connecting the coordinate origin to the point of R_δ=500.

Now, the dependence of pressure gradient on the mean velocity amplitude is examined by comparing it with the one of steady laminar flow in circular pipe. According to the laminar solution of the Poiseuille flow, pressure drop along the pipe axis is given by

$$\Delta p = 32 \nu \rho \ell \overline{u}_m / D^2 \qquad (3)$$

where Δp : pressure drop
ρ : density
ℓ : length of pipe
D : diameter of pipe

In the tests, the measured pressure in the tube corresponds to the pressure drop for the length from the pick up point to the free end of the discharge tube.

Compared to the prediction by Eq.(3), the pressure obtained in the tests took higher values in general and its ratio seemed to be governed by Stokes' parameter, the ratio of pipe radius to thickness of boundary layer. Stokes' parameter is defined by

$$\lambda = R / \delta = \sqrt{\omega / 2\nu} \cdot R \qquad (4)$$

where λ : Stokes' parameter
R : radius of pipe

Fig.7 shows $\eta - \lambda$ elations, were η is the ratio of the observed pressure drop to the one estimated by eq.(3). Disregarding some scattering of datas, the relations can be represented by linear functions for the respective tubes, i.e.

$$\left.\begin{array}{ll} \text{for 6mm}\Phi\text{ tube,} & \eta = 0.11\lambda + 0.4 \\ \text{for 9mm}\Phi\text{ tube,} & \eta = 0.68\lambda - 1.8 \\ \text{for 12mm}\Phi\text{ tube,} & \eta = 0.094\lambda - 2.8 \end{array}\right\} \quad (5)$$

In experiments of sinusoidally oscillating air flow in circular pipes, Hino and Sawamoto (1976) observed the occurrence of weakly turbulent flow, where the velocity profile of laminar flow was baisically maintained, only very small time fluctuation of velocity being overlaid. Though they did not refer to pressure gradient, it is presumably far greater than the one of the laminar flow.

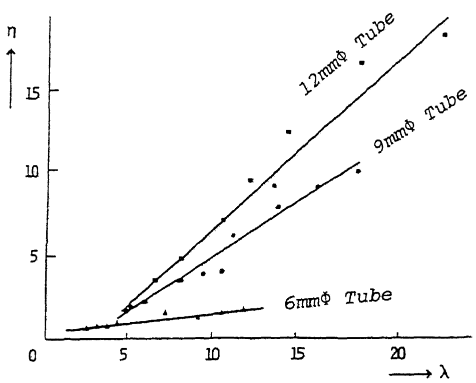


Fig.7 $\eta - \lambda$ relations

2.4 Equivalent viscous damping

Considering the linear dependence of the damper resistance on velocity in the weakly turbulent region, responce analysis of damped vibration system can be conducted by a linearized theory, and equivalent viscous damping force is evaluated in the following.

Let β be the ratio of sectional area of the cylinder to that of the discharge tube, i.e.

$$\beta = A / a \qquad (6)$$

where A : sectional area of cylinder
a : sectional area of tube

When sinusoidal velocity of

$$\dot{x} = \bar{\dot{x}} \cos\omega t \qquad (7)$$

is forced to the pistons, the mean velocity in the tube becomes

$$u_m = \bar{u}_m \cos\omega t = \beta \bar{\dot{x}} \cos\omega t \qquad (8)$$

Also, the resisting force acting to the piston at $x=0$ is given by

$$R_0 = A p_0 = \beta a p_0 \qquad (9)$$

where R_0: resisting force to piston
P_0: pressure in the end of tube

Eqs.(3) and (4) are used to express p_0, and also the energy dissipation factors and the effect of flow contraction mentioned in 2.2 are taken into account.

As the result, the equivalent viscous resistance to the piston movement can be represented by the followings.

$$R = R_0 \cos\omega t = C_{eq} \dot{x} \qquad (10)$$

$$C_{eq} = 32 \xi \eta \nu \beta^2 m_\ell / D^2 \qquad (11)$$

$$m_\ell = \rho a(\ell + \Delta\ell) \qquad (12)$$

C_{eq}: equivalent viscous damping coefficient

m_ℓ : mass of liquid in the tube
ℓ : total length of discharge tube
$\Delta\ell$: fictitious length of tube for considering the flow contraction effect
ξ : energy dissipation factor

When the damper is incorporated in a single mass vibration system of natural frequency of ω_0, the equivalent damping ratio referring to the critical is evaluated by a simple formula:

$$h_{eq} = \frac{C_{eq}}{2m\omega_0} = \frac{16\xi\eta\nu}{\omega_0 D^2} \cdot \frac{\beta^2 m_\ell}{m} \qquad (13)$$

h_{eq} : equivalent viscous damping ratio

m_ℓ : mass of vibrating system

ω_0: natural frequency of the system

It muat be noted that the damping ratio is dependent on responding frequency because the governing parameter λ in Eq.(5) is frequency dependent.

2.5 Steady vibration response of a single mass system

To examine the applicability of the resistance rule obtained, the result of theoretical analysis is compared to resonance curves obtained by shaking table tests of a damped frame.

The portal frame shown in Fig.2 with a weight attatched to the top beam was subjected to harmonic excitation. The tests ware carried out on the frame with the same damper as shown in fig.1, with liquids filled and also without liquid (empty damper) as well for the purpose of comparison. Parameters of the frame and the dampers are listed in Tables 2 and 3.

Table 2 Parameters of test frame

Spring constant	k=48.9 kgf/cm
Weight of lumped mass	W=75.9 kgf
Natural frequency	n_0=4.0 Hz
Inherent damping ratio	h =0.2 %

Table 3 Parameters of dampers

Sectional area of cylinder A=15.2 cm^2

* G: glycerin	Case(a)	Case(b)
Tube diameter (mm)	9	6
Tube length (cm)	166	90
Liquid	water	60% G *
Temperature	21° C	20° C
Table acceleration	0.05g	0.05g

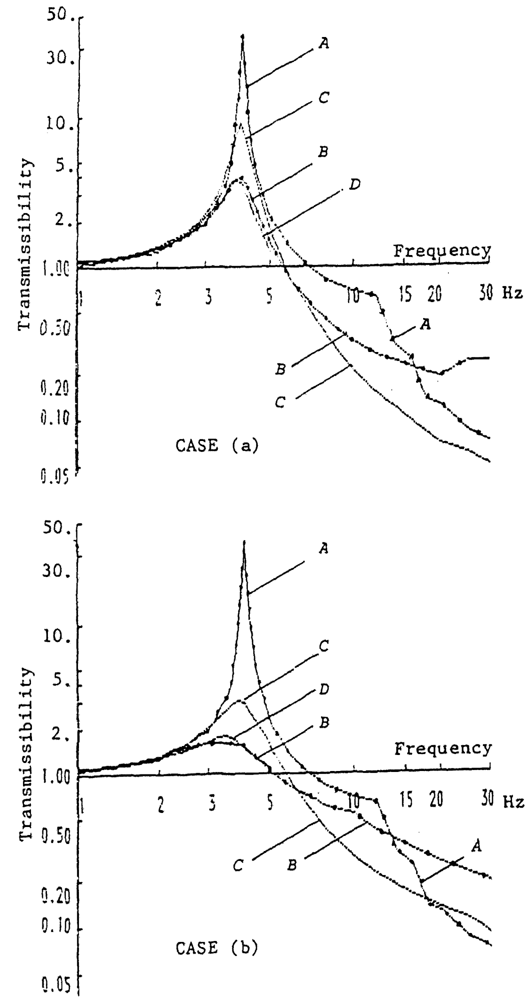


Fig.8 Comparison of transmissibility: thery and experiment

Fig.8 shows test results and theoretical prediction of transmissibility for the top beam. Notation of the curves is as follows.

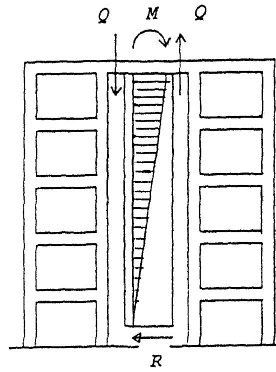
A: test result for the case of empty damper
B: test result for the case of filled damper
C: prediction based on damping ratio of Eq.(13)
D: prediction modified considering friction damping

Theoretical analysis bsaed on Eq.(13) gave fairly larger transmissibility than the test results (Curve *C*). Hence, a constant friction damping force of 1.8kgf, corresponding to friction of the piston sealing and to the negative pressure in the cylinders, was taken into analyses. The curves *D* are blank for the frequency region over 5Hz, because iteration procedure to allow for the friction did not converge as displacements were so small.

Though the influence of the friction damping is considerable as the responding displacement was small, the agreement of the theoretical prediction with the test results indicate the applicability of the derived resistance rule of the damper in the weakly turbulent region.

3 METHOD OF DAMPER INSTALLATION IN MULTI-STORIED HIGH BUILDINGS

High buildings have a tendency of being dominantly affected by the fundamental mode of vibration and the most efficient way of restricting its amplitude is to exert damping or controlling force diredtly onto the top of the building.



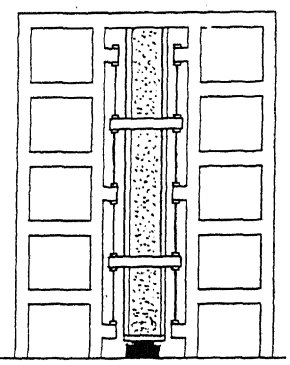


Fig.9 Continuous wall Fig.10 Hanging wall

Fig.9 represents installation of a damper in a building, where it is directly connected to the top beam by an isolated cotinuous wall. In such a case, the wall is subject to large bending moment and reacting shearing forces in the top beam cannot be resisted in high buildings. To overcome this difficulty, the author propose a new installation scheme by the use of "hanging wall".

In the hanging wall system, a continuous wall, fixed to the top beam and connected to a damper at the bottom end, is supported vertically by rollers provided at several points in intermediate height of the neighboring columns as shown in Fig.11(a).

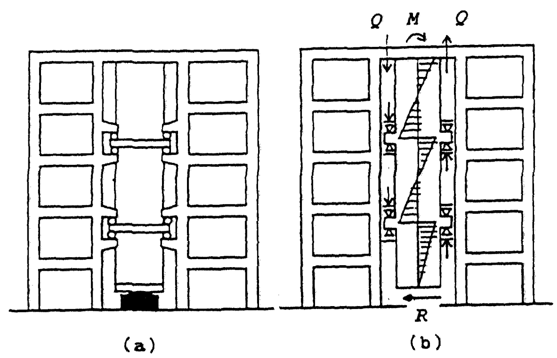


Fig.11 Principle of hanging wall system

In this scheme, bending moment of the wall is restricted being cancelled by oposite moments of reacting forces from the roller suports as seen in Fig.11(b).

Actually, it is recommended to realize the condition of a roller support by suspending the intermediate points of the wall by long steel rods whose ends are fixed in the column. Fig.10 shows an example of such construction.

To demonstrate the effectiveness of the proposed scheme, a three storied steel model frame of 1.9m height, shown in Fig.12, was tested on the shaking table. Here, a truss, instead of the wall, was suspended by steel strings of 2.5mm diameter whose ends were fixed to the neighboring beams. The same damper as the Case(b) in Table 3 but filled with water as well as the empty damper was incorporated in the frame.

Fig.13 shows the comparison of the transmissibility for the both cases. It is apparent that the hanging truss system worked effectively to control vibration not only of the fundamental mode but also of the higher modes.

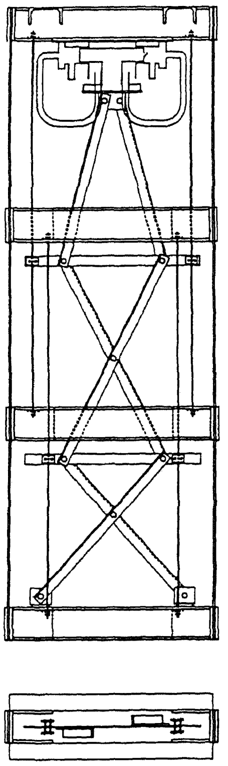


Fig.12 Three storied test frame

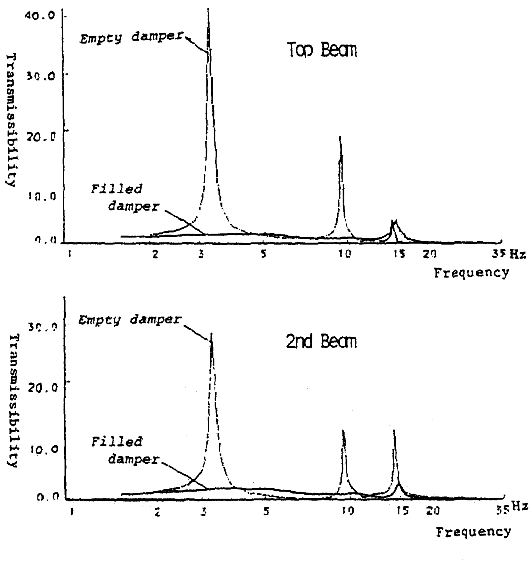


Fig.13 Test results: transmissibility

4 CONCLUSION

The property of flow resistance in the discharge tubes of an accelerated liquid mass damper of open circulation type was investigated by examining the results of excitation tests. As the result, empirical rules were presented for evaluating the resisting forces of the damper in the weakly turbulent region of the pipe flow.

To facilitate the application of the dampers to multi-storied high buildings, a new scheme of damper installation, the hanging wall system was proposed,

For further advancement towards application, damper resistance in the strongly turbulent region must be investigated.

REFERENCES

Abdel-Rohman, M. and Leipholtz, H.H.E. 1987. Active control of flexible structure, *ASCE* 104 ST8:

Hino, M and Sawamoto, M. 1976. Experiments on transition to turbulence in an oscillatory pipe flow, *J. Fluid Mech.* 75 Part 2: 193-207

Kawamata, S. 1973. Development of a vibration control system of structures by means of "mass pump", *Bul. Earthq. Resist. Struct. Resear. Center,* 7, Inst. Indust. Science, Univ. Tokyo

Kawamata, S. 1987. Accelerated liquid mass damper and principles of structural vibration control, *Trans. 9th SMiRT Conf. Lausanne,* K1: 737-742

Kawamata, S. 1989. Accelerated liquid mass damper as tools of structural vibration control, *Proc. 9th WCEE, Tokyo-Kyoto, VIII, 421-426*

Kawase, H. et al. 1990. A new approach for structiral vibration control: damped wire system, *Summaries of Tech. Papers of Ann. Meet.,*Architec. Inst. Japan