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PASSIVE CONTROL OF BUILDING FRAMES BY MEANS OF LIQUID DAMPERS SEALED BY VISCOELASTIC MATERIAL

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SUMMARY

In the purpose of installing in passive vibration control system of building frames, a liquid damper of new type has been developed. In the liquid damper, narrow gaps between a cylinder and a pair of pistons is packed with viscoelastic polymer to form soft rings which, permitting relative movement of the pistons and the cylinder, serve to seal pressurized liquid. This device enables very simple form of liquid damper and its easy production.

In the present paper, resisting force characteristics of the new damper is reported, where the resistance is of two components: viscoelastic resistance of the sealing ring and inner pressure of the liquid flowing through a narrow orifice. An analytical model of the resisting force constructed based on theory of viscoelasticity and turbulent flow rule of liquid is presented. The results of shaking table tests of portal frame specimen provided with the dampers and their numerical simulation are discussed.

INTRODUCTION

Rapid growth of base isolated building construction in Japan since disastrous earthquake in Kobe in 1995 has been remarkable: 625 planned base isolated buildings were approved by the government by July 1998 [1]. In parallel with the development of base isolation, increasing attention has been paid to R&D of vibration control technique. Among variety of vibration control systems, passive control by the use of inter-storey dampers seems to have potential of wide application to seismic design of building frames including the one for retrofitting [2]. It is advantage of this technique that it can be practised by only small extension of conventional construction technique. Examples of applications of steel and viscoelastic dampers to large buildings [3] indicate that these techniques are becoming mature to be used in common practice of seismic design of buildings.

Typical devices of energy absorption for the use of inter-storey installation are hysteretic dampers and viscous or visco-elastic dampers. Among the viscous dampers, which enables more effective vibration control than the hysteretic one, oil dampers seem to be attractive for design purpose as it provides very high resisting capacity within compact form. However, the oil dampers are relatively expensive as they need precision machining in order to incorporate shaft bearings and pressure sealings. Take, Kawamata and Funaki [4] proposed a new type of liquid damper to eliminate this difficulty. In the liquid damper, small gap between a cylinder and a pair of pistons was packed with visco-elastic polymer to form soft rings which, permitting relative movement of the pistons, serve as sealants of inner pressure. The sealing rings, being consisted by energy absorbing material, gave rise to additional resisting force to inner pressure resistance caused by liquid flow through an orifice. This scheme of inner pressure sealing enabled very simple and compact composition of liquid damper. With regard to the new damper, resisting force characteristics were investigated by dynamic excitation tests of damper specimens and an analytical model of the damper resisting force was constructed. Also, to demonstrate the applicability and efficiency of the damper, 1/3-scaled steel portal frames having the dampers were subjected to shaking table tests. Time integration program for response analysis was developed where the resisting force rule

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of the damper was implemented. The vibrational response of the damped frame specimen simulated by the analysis program was compared with the test results. Funaki presented in his doctoral thesis [5] the constructed model of the damper resistance and response analyses of the tested damped frame specimen. This paper presents outlines of the development of the vibration control system by the use of the new damper.

LIQUID DAMPER SEALED BY VISCOELASTIC MATERIAL

Fig.1 shows the scheme of the proposed liquid damper. The damper is composed of a cylinder and a pair of pistons, the latter being linked together by outer framework so as the simultaneous same movement is forced. Inside of the cylinder is divided by a diaphragm to form upper and lower cavities which are to be filled with liquid such as oil. Through diaphragm a bolt is installed in which an axial hole of small diameter is bored to form an orifice for liquid flow. As a characteristic feature of the damper, the gap between the cylinder and the pistons are packed with soft viscoelastic polymer, thus rings serving to seal pressurised liquid being formed. When relative displacement of the pistons is forced, the sealing ring is subjected to dynamic shearing deformation which gives rise to viscoelastic resisting force. At the same time, the contained liquid moves reciprocally between the upper and the lower cavities flowing through the narrow orifice, in which strong turbulence occurs. The turbulent liquid flow in the orifice brings high inner pressure which increases nonlinearly with increase of piston velocity. In consequence, the damper exerts resisting force of two components against piston movement: viscoelastic resistance caused by shearing deformation of the sealing material which is almost linearly proportional to piston velocity and axial force corresponding to the inner pressure of the liquid which increases nonlinearly with regard to piston velocity.

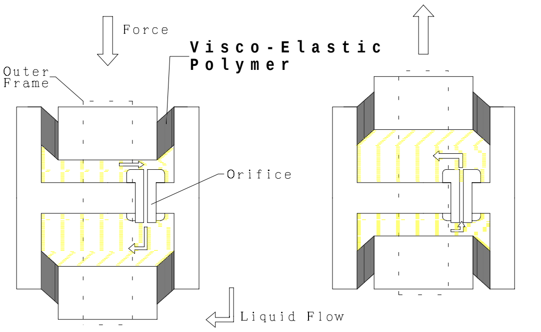


Figure 1 Scheme of Liquid Damper

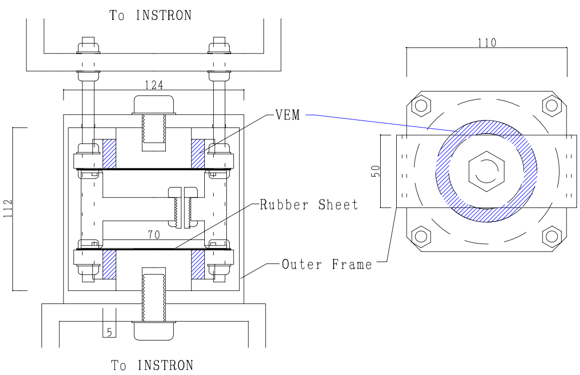
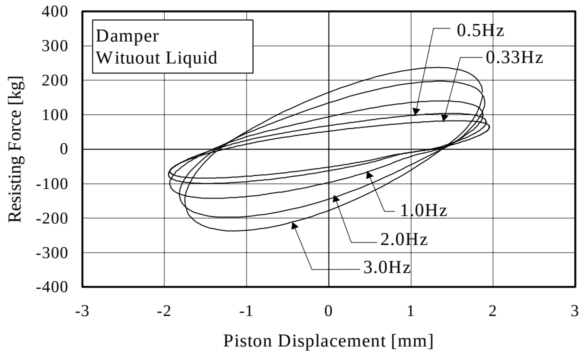


Figure 2 Damper Specimen (unit: mm)

Fig.2 shows a specimen of the liquid damper used in harmonic excitation tests by the use of a numerically controlled dynamic actuator, INSTRON 8500. Inner diameter of the cylinder was 70mm. Visco-elastic material of acrylic polymer, VEM of Sumitomo-3M Co., was used to form the sealing rings having section of 5.0mm× 20mm. The pistons were subjected to harmonic excitation of frequency of 0.33Hz-5.0Hz and displacement amplitude up to 5.0mm. The followings are typical examples of the tests results in which damper oil (ν =0.20cm^2/sec, at 20ºC) was used. Fig.3 shows hysteresis loops of resisting force versus piston displacement for the case of the damper without oil, where visco-elastic resistance of the sealing material is the only component of the resisting force. The loops took the forms of ellipse, the inclination of its axis becoming steeper for the excitation of higher frequency as is common to many kinds of viscoelastic polymer. Fig.4 shows hysteresis loops of the difference of inner pressure between the upper and lower cavities versus piston displacement for the case of damper filled with oil having orifice of 1.5mm diameter and length of 24mm. The shape of the loops apparently differs from ellipse, indicating inner pressure nonlinearly increasing with flow velocity. It takes the maximum pressure with some phase-lag from the point of zero displacement where the piston velocity takes the maximum value. This point is discussed in the next section.



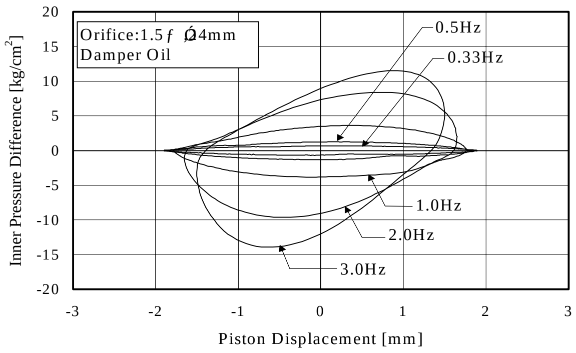
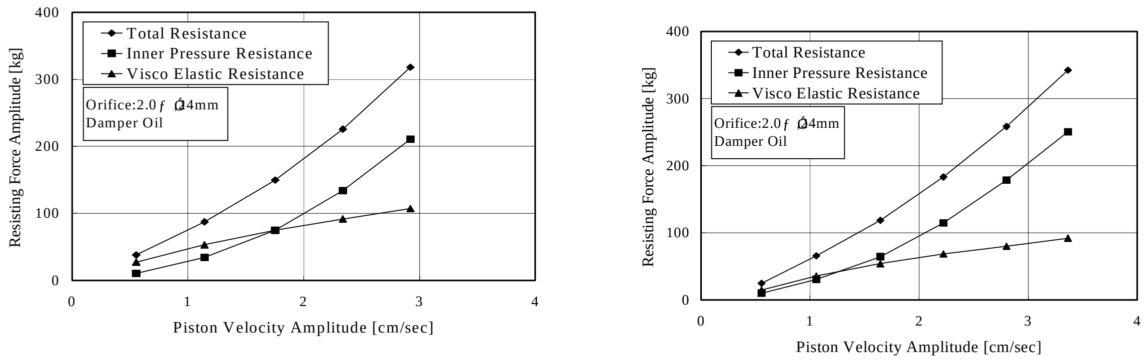


Figure 3 Hysteresis Loop of Visco-Elastic Resistance **Figure 4 Hysteresis Loop of Inner Pressure**



(a) 1.0Hz Excitation **(b) 2.0Hz Excitation**

Figure 5 Relation of Resisting Force Amplitude to Piston Velocity Amplitude

Fig.5 shows the relation of total resisting force amplitude accompanied by constituent visco-elastic resisting force and inner pressure force amplitudes plotted against piston velocity amplitude. While the visco-elastic resistance increases almost linearly with the piston velocity, increase of inner pressure is nolinear. Therefor, inner pressure surpasses the viscoelastic resistance in some point of piston velocity, the surpassing point of piston velocity becomes the smaller for the higher frequency of excitation. According to this observations, it can be concluded that inner pressure of liquid produces strong resistance for the structural response of large amplitude and high frequency while viscoelastic resistance dominates for the response of low frequency and small amplitudes. The different characteristic of both the components of resistance fortifies each other in the weak points of respective property. The maximum inner pressure of the contained liquid was about 30kg/cm^2 in the extent of the test series, no damage to the sealing material having been experienced.

THEORETICAL MODEL OF DAMPER RESISTANCE

Viscoelastic Resistance

The hysteresis loops of the sealing material of the damper represent viscoelastic resistance whose property has strong dependence on frequency of excitation. Though the simple viscoelastic model of Maxwell liquid unit is able to represent the elliptical loops having inclination of the axises becoming steeper towards higher frequency, its dependence on frequency seems too strong. Therefor, to realise more moderate dependence on frequency, a 4-parameter viscoelastic model which is composed of parallel combination of a Maxwell fluid and a Kelvin solid was adopted, as the latter, having constant inclination of loops for any frequency, might moderate the dependence. Fig.6 shows the 4-parameter model, P and δ representing force and total displacement. When the model is subjected to harmonic excitation, $P-\delta$ relation can be written by the following complex form:

$$P_1 + iP_2 = (K_1 + iK_2)(\delta_1 + i\delta_2) \quad (1)$$

where $K_1 + iK_2$ is a complex spring constant. In the present case of 4-parameter model, the constant is given by

$$K_1 + iK_2 = \left(\frac{k_M c_M^2 \omega^2}{k_M^2 + c_M^2 \omega^2} + k_K \right) + i\left(\frac{k_M^2 c_M \omega}{k_M^2 + c_M^2 \omega^2} + c_K \omega \right) \tag{2}$$

where ω is frequency of excitation. Constants K_1 and K_2 in Eq.(2) can be found in observed hysteresis loops as shown in Fig.7. As it is difficult to determine the four parameters simultaneously from observed data, an alternative method was proposed [6], in which the unknown parameters were identified successively in the following steps:

1) k_K for the Kelvin spring is determined using hysteresis data corresponding to effectively static loading,

2) k_M and c_M for the Maxwell unit are obtained from a set of hysteresis data for excitations in two different frequencies, and

3) finally c_K is calculated by introducing the obtained K_2 data and k_M and c_M determined already into the relation of K_2 and the parameters.

The parameters actually identified with regard to the test specimen shown in Fig.2 is given in Table 1. In Fig.8 hysteresis loop of the damper force theoretically predicted using parameters in Table 1 are compared to the test results.

Table 1 Viscoelastic Parameters for Test Specimen

k_M	c_M	k_K	c_K
804 kg/cm	126 kg sec/cm	150 kg/cm	32.8 kg sec/cm

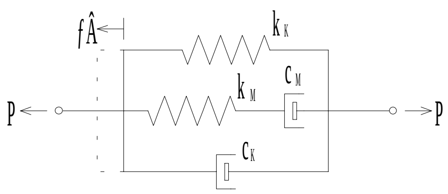


Figure 6 4-parameter Model

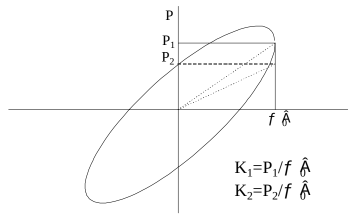
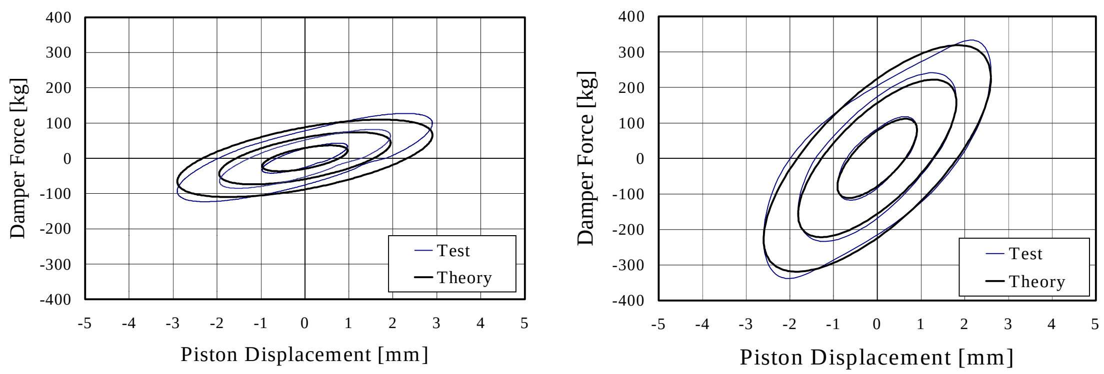


Figure 7 Force-Displacement hysteresis Loop



(a) 0.33Hz Excitation **(b) 3.0Hz Excitation**

Figure 8 Comparison of Predicted Viscoelastic Resistance with Test Results

Inner Pressure Resistance

As seen in the test results, hysteresis loops of inner pressure difference between the upper and lower cavities show phase-lag in the point of the maximum value. Fig.9 shows a schematic representation of the loop. As is well known, Maxwell fluid model of viscoelasticity provides hysteresis ellipse in which the maximum force occurs with some phase-lag from the point of zero displacement. This is caused by the deformation of the spring element which is linked with dashpot in series and subjected to the same force as the dashpot. In the case of the present damper, the viscoelastic sealing rings are supposed to be subject to outward deflection by inner pressure as shown in Fig.10. Therefor, it was presumed that the phase-lag of inner pressure resistance could be simulated by idealizing the mechanism of pressurization by a Maxwell fluid model shown in Fig.11, where the spring element (k_P) corresponds to the elastic deformation of the sealing rings and the dashpot (c_P), to flow resistance in the orifice.

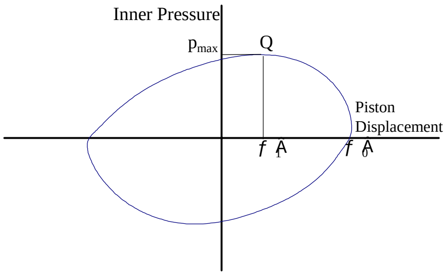


Figure 9 Schematic Diagram of Hysteresis Loop of Inner Pressure Resistance

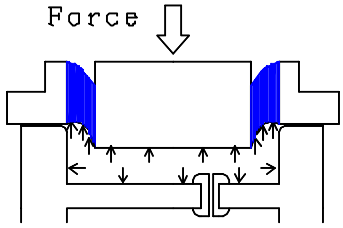


Figure 10 Elastic Deformation of Sealing Ring by Inner Pressure

In the following, the methodology of parameter identification for the Maxwell fluid model is briefly outlined. Referring to Fig.9, at the point Q, where the maximum inner pressure occurs, displacement of the dashpot, δ'' is zero. Therefor, the piston displacement δ_1 represents solely the deformation of the spring element, δ' and $p_{\max}-\delta_1$ relations plotted for the hysteresis loops of various piston displacement amplitude correspond to $p-\delta'$ curve of the spring element. Fig.12 shows $p_{\max}-\delta_1$ relations obtained from the results of a test case. The diagram shows almost linear relations and the spring constant k_P can be determined for the respective cases of excitation frequency.

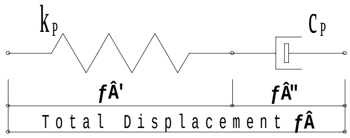


Figure 11 Maxwell Fluid Model for Pressure Resistance

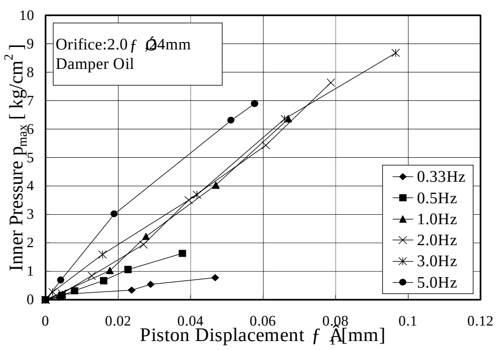


Figure 12 $p_{\max}-\delta_1$ Relations

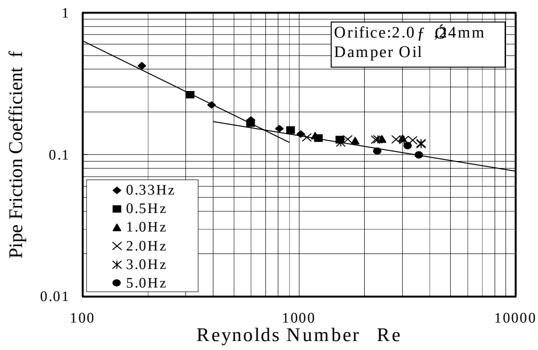


Figure 13 Relation of Pipe Friction to Reynolds Number

In the next step, viscous parameter of the dashpot, c_K, has to be searched for. As the dashpot corresponds to resistance of turbulent liquid flow in the orifice, c_K is not a constant parameter, the pressure difference being non-linear with regard to velocity of the liquid flow. As the spring element takes the maximum stress $p_{\max}$ at the point Q in Fig.9, its velocity is zero. Hence, velocity at the point Q is equal to the maximum velocity of the dashpot, $\dot{\delta}''_{\max}$, corresponding to $p_{\max}$. Deriving correlation of the maximum dashpot velocity to the maximum pressure difference from the excitation test data and following the methodology presented by Kawamata et al. [7], flow rule of liquid through the orifice can be constructed. In the flow rule, the pressure difference is given by

$$p = \frac{\rho L}{2D} {u_m}^2 f \tag{3}$$

where ρ : density of liquid, L : length of orifice, D : diameter of orifice, u_m : mean velocity of orifice flow, f : pipe friction coefficient. The friction coefficient is to be correlated to Reynolds number defined by

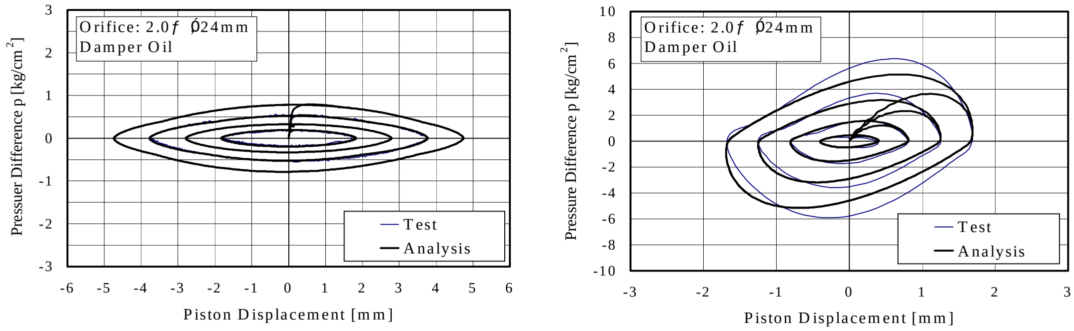
$$\mathrm{Re} = u_m D/\nu \tag{4}$$

where ν : kinematic viscosity of liquid. Fig.13 shows an example of f -Re relation obtained from the results of excitation test. For this case, the pipe friction coefficients can be represented as

$$f_1 = 20.0\,\mathrm{Re}^{-0.75} \quad for \;\; \mathrm{Re} \le 700 : (Transition\ region), \tag{5}$$

$$f_2 = 0.765\,\mathrm{Re}^{-0.25} \quad for \;\; \mathrm{Re} > 700 : (Turbulent\ region). \tag{6}$$

Eq.(6) means that pressure difference in turbulent region is proportional to 1.75th power of the flow velocity as in the case of steady liquid flow in a smooth pipe. In Fig.14, hysteresis loops of pressure difference obtained by the test are compared with those simulated by time step analysis based on the derived flow rule and the spring constant.



k_P=150kg/cm^3 and f_s in Eq.(5) and (6) were applied

(a) 0.33Hz Excitation **(b) 3.0Hz Excitation**

Figure 14 Comparison of Pressure Difference: Test and Analysis

EXCITATION TESTS OF A STEEL FRAME MODELWITH PASSIVE VIBRATION CONTROL SYSTEM

Test Specimen

To demonstrate the effect of passive control of vibration by the use of the proposed liquid damper, a 1/3-scaled steel frame was subjected to excitation tests. A pair of portal frames were linked together by transverse beams to form a table shape on which steel weight of 4.0 ton was placed as shown in Fig.15. In a vibration controlled specimen, a liquid damper was installed between V-shaped bracing and a basement beam in each frame as shown in Fig.16. The used dampers were the same as shown in Fig.2 and the damper oil (ν =0.20cm^2/sec, at 20ºC) was filled. Orifice of 2.0mmϕ and 24mm long was incorporated. In this damped specimen, the web of main beams was perforated to form an opening in order to make it more flexible. For the purpose of comparison, a specimen without dampers and bracings was also tested, where the main beams were replaced by full web one without opening. Static loading tests of horizontal force and free vibration tests were conducted. The obtained fundamental parameters of the specimens are listed in Table 2.

Figure 15 Test Frame with Damper

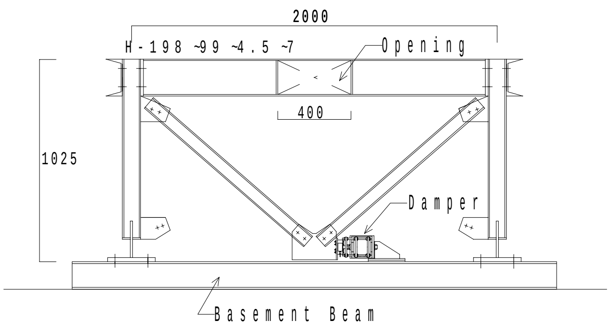


Figure 16 Test Frame with Damper (unit: mm)

Table 2 Fundamental Parameters of Test Frames *case of dampers incorporated

Specimen	Horizontal Stiffness [10^3 kg/cm]	Natural Frequency [Hz]	Effective Mass [kg sec^2/cm]	Damping Ratio [%]
Undamped (Full Web Beam)	6.69	5.88	4.90	0.53
Damped (Open Web Beam)	1.94	3.24 (4.08*)	4.66	0.91 (20.1*)

Response Analysis of Frames with Dampers

The test frame specimen with the dampers can be idealised by a single mass system shown in Fig.17 (a). Action of the damper is represented by a parallel combination of a 4-parameter model for viscoelastic resistance and a Maxwell fluid model for pressure resistance. In this case, the inner pressure which is evaluated in a manner described in the previous section has to be transformed into resultant force being multiplied by effective sectional area of the pistons. As the spring and dashpot of the Kelvin solid (k_K and c_K) are subjected to identical displacement to the mass, these parameters can be included in those of the structure as shown in Fig.17 (b), the pair of Maxwell fluid models remaining for evaluating the additional resisting force to the vibration system. A time integration program for the response analysis of the single mass system was developed using Runge-Kutta method. In the discretized time step equation of motion, the additional terms of resisting force by the two Maxwell fluid models were included for which evaluation procedure proposed by Hatada, Kobori et al. [8] was adopted.

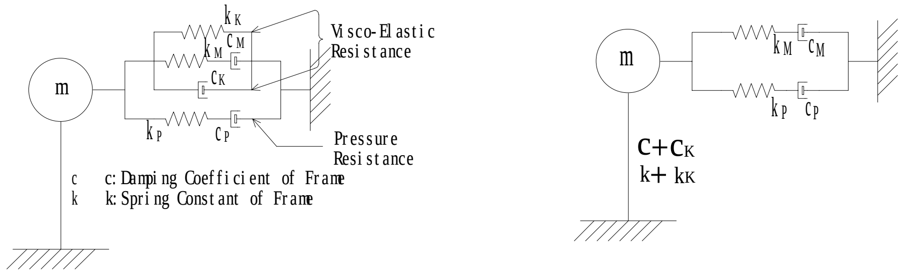


Figure 17 Single Mass System for Damped Frame

Results of Test and Analysis

Fig.18 shows resonance curves obtained both from harmonic excitation test of the damped specimen and the corresponding analysis. In Fig.19 results of the test and the analysis are compared with respect to hysteresis loop of resisting force of the damper. Fairly good agreement can be seen between tests and analysis. Fig.20 shows input and response of the damped specimen to an earthquake input, Hachinohe EW with time axis compressed into 1/2. While the maximum response acceleration of the undamped frame to the same input was 1222gal, the one of the damped case was reduced to 454gal. Also, maximum relative displacement of 8.9mm of the undamped specimen was reduced to 3.7mm for the damped one. The results of numerical simulation of the responding acceleration and the hysteresis of damper resistance are shown in Fig.20 (b) and (c) providing good coincidence with the test results.

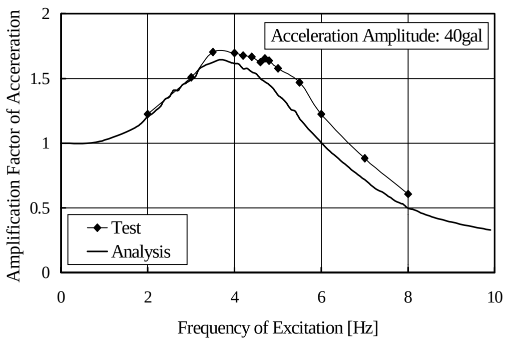


Figure 18 Resonance Curve to Harmonic Excitation

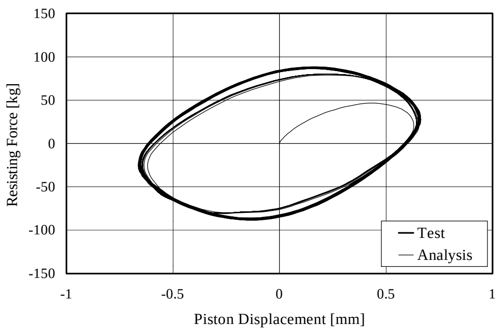
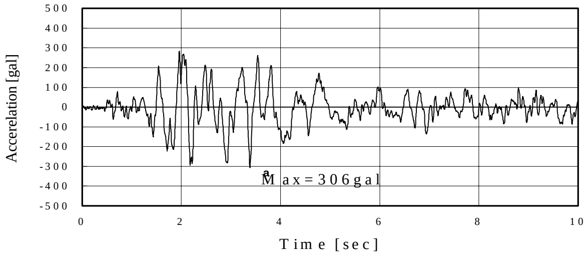
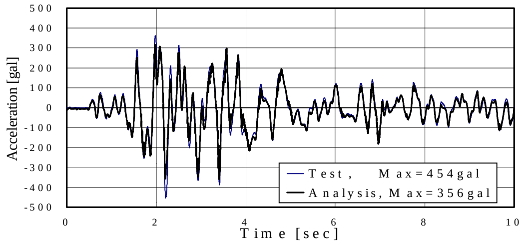


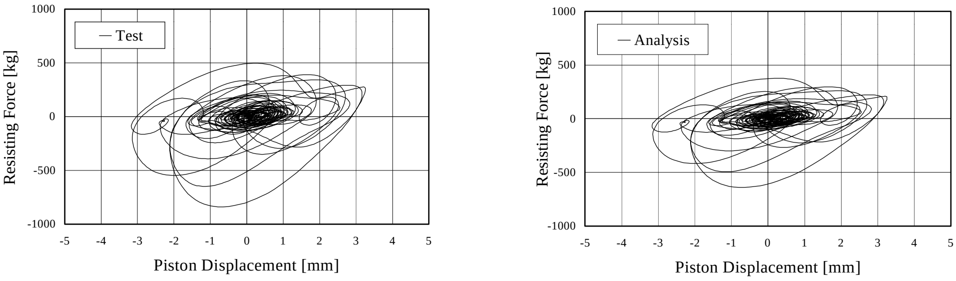
Figure 19 Hysteresis Loop of Damper Resistance for 4.0Hz Excitation



(a) Acceleration of Shaking Table



(b) Acceleration of Frame



(c) Hysteresis Loop of Damper

Figure 20 Response of Damped Frame to Excitation of Earthquake, Hachinohe EW, 1/2 Time Axis

CONCLUSION

In order to apply as inter-storey dampers in passive vibration control system for building frames, a liquid damper of the simplest form was developed. In the new damper, soft rings of viscoelastic polymer packed in the gap between a cylinder and pistons took the role of sealing the liquid under high inner pressure.

Dynamic excitation tests of the damper revealed the characteristics of resisting force of the damper which was consisted of two components: visco-elastic resistance of the sealing rings and inner pressure resistance of the liquid flowing in narrow conduit of orifice. It was verified that the analytical model for the resisting force constructed on the basis of theory of viscoelasticity and turbulent flow rule represented hysteresis of the damper resisting force with enough accuracy.

Shaking table tests conducted on 1/3 scaled portal frames provided with the new dampers demonstrated remarkable reduction of vibration response and proved the applicability and the validity of response analysis program developed for the passive control system.

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All references but [7] are written in Japanese

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