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JSSI MANUAL FOR BUILDING PASSIVE CONTROL TECHNOLOGY PART-11 TIME-HISTORY ANALYSIS MODEL FOR VISCOUS DAMPERS

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SUMMARY

Viscous damper is known to have high damping performance, and it is being used not only in buildings industries but also in others as passive control device. Viscous material is typically considered to have little but considerable elastic stiffness and it depends on condition. The elastic stiffness stems from the compressibility of the viscous material, which is much more significant than deformation of the steel component. This paper presents an overview of time history analysis model for viscous damper considering the influence of elastic stiffness and corresponding elastic force that is set proportional to α times power of velocity, where α shall take the value between 0 and 1.

Analogous to Maxwell body, it consists of nonlinear viscous element and elastic spring element in series. Given the deformation history, the constitutive rule of this model becomes nonlinear differential equation which must be solved numerically in order to obtain the force output. The differential equation is solved step by step by applying the Runge-Kutta method, a move-forward type simple numerical integration scheme. The results of the proposal analytical method accord with experimental ones. This paper also shows some points to maintain accuracy of the model. The study was conducted by the members of the Device Analysis Working Group of JSSI Response Control Committee, for the purpose of promoting reliable time history analysis of the buildings with viscous dampers [1].

INTRODUCTION

This paper targets viscous dampers with cylindrical body, and which generates damping force by flow resistance of filling material. These types of viscous dampers control excessive increase of damping force with filling material, and have been applied to many buildings as passive control devices. It is noted that they have little dependency on temperature or amplitude of input deformation and are easy to be modeled for time history analysis.Damping force of viscous damper is nonlinear which is proportional to exponent α times power of velocity. Viscous dampers, whose exponent α is near 0, are popular due to the power transmitted to the structure may not become excessive.

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It is desirable that the time history analysis model of passively controlled buildings with viscous dampers can be simulated as precise as possible. Since the parameters for the analysis model which are shown by manufacturers of viscous damper may be determined with performance of small dampers, it is expected to verify the performance of viscous dampers which are applied to the building actually. It is to be noted that oil damper under input deformation, whose differential wave with respect to time differs with the cosine wave with its 90 degrees phase difference, cannot offer supposed performance, because the performance is examined under sinusoidal excitations. Analogous to the mechanism of oil damper, cylindrical viscous damper may have reduction of damping performance under extremely small excitation.

Fig. 1 shows the schematic diagram for added component which consists of viscous damper and brace connected in serial. Since this paper only targets damper, damping force F_d and deformation u_m of damper are considered. K_m and C_m represent equivalent stiffness and equivalent viscous damping coefficient of damper, respectively.

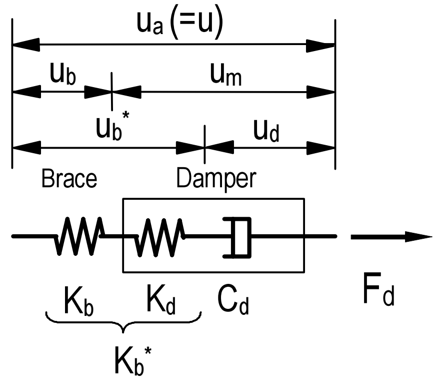
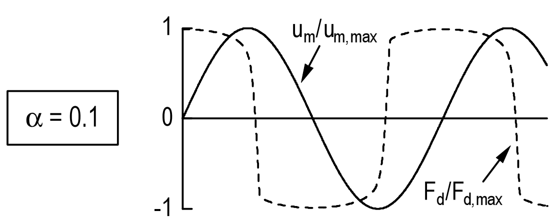
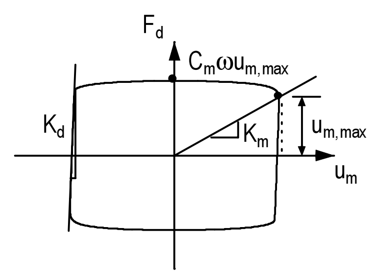


Fig. 1 Schematic Diagram for Added Component

BASIC HYSTERESIS CHARACTERISTICS OF VISCOUS DAMPER

Fig. 2 (a) shows time history of deformation and damping force of viscous damper without relief valve under sinusoidal excitation with circular frequency ω and amplitude $u_{m,max}$. And the hysteresis loop shows swelled parallelogram as in Fig. 2 (b).





(a) Time History **(b) Hysteresis Loop**

Fig. 2 Behavior of Viscous Damper under Sinusoidal Excitation (Exponent α: 0.1)

The stiffness of the hysteresis loop results from internal stiffness of viscous damper by compression stiffness of filling material. Fig. 3 represents an example of relationship between internal stiffness K_d and standard damping force F_d, and they are in a proportional relation. In this example, they are in about a relation $K_d = 4F_d$. Therefore, it is inappropriate to consider the internal stiffness as infinite, and it should be regarded correctly in analysis model.

To accommodate the hysteresis loop of viscous damper to one of analysis model, one of considerable points is to install pins at the both ends of damper to avoid bending moment. Since there is no experimental data about behavior of viscous damper under extremely small excitation, the study about it based on the constitutive rule of viscous damper will be shown later.

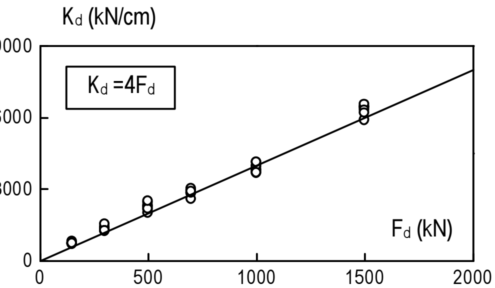
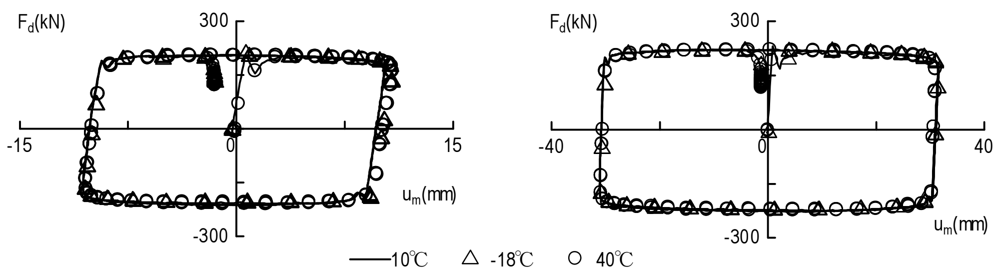


Fig. 3 Example of Relationship between Internal Stiffness K_d and Standard Damping Force F_d

Figs. 4 compare hysteresis loops of viscous damper under temperature -18°C, 10°C and 40°C. These are the results of dynamic excitation tests for a single damper under various temperatures. As the hysteresis loops match well, viscous dampers shown here have little dependency on temperature. However, dependency of viscous damper on temperature is peculiar to the property of the filling material, it is necessary to consider the dependency of the damper which is applied to the building actually.



(a) Amplitude: 10 mm **(b) Amplitude: 30 mm**

Fig. 4 Study about Dependency on Temperature (Damper Capacity: 200 kN, Frequency: 0.2 Hz)

SUMMARY OF TIME HISTORY ANALYSIS MODEL OF VISCOUS DAMPER

The time history model of viscous damper is represented as Maxwell body shown in Fig. 1. Unlike in the case of oil damper, internal viscosity is modeled as a dashpot element which generates damping force proportional to α times power of velocity. Spring element, connected in serial with dashpot element, has internal stiffness which is linear to deformation.

As shown in Fig. 5, damping force of viscous damper F_d is proportional to α times power of velocity $\dot{u}_d$ and the exponent α takes 0 to 1 in general. If the exponent α is 1, damping force and velocity have linear relation and the hysteresis loop is ellipse. As α gets closer to 0, the increment of damping force by

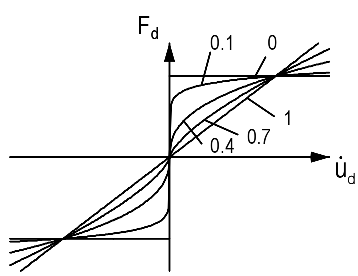


Fig. 5 Relationship between Damping Force F_d and Velocity $\dot{u}_d$

velocity becomes gradual as shown in Fig. 5. If α is 0, damping force is constant such as friction damper. That is to say, dashpot element behaves as velocity-dependent if α is 1, as displacement-dependent if α is 0 and takes a middle position of them if α is between 0 and 1.

With condition of compatibility of deformation and velocity, and constitutive rule of nonlinear dashpot element and linear spring element, constitutive rule of whole viscous damper becomes first-order differential equation as Eq. (1).

$$\frac{\dot{F}_d(t)}{K_d} + \mathrm{sign}(F_d(t))\left(\frac{|F_d(t)|}{C_d}\right)^{\frac{1}{\alpha}} = \dot{u}_m(t) \tag{1}$$

In time history analysis, damping force is calculated by applying Runge-Kutta method to Eq. (1).

Fig. 6 represents relationship between velocity and equivalent viscous coefficient, which is obtained by damping force being divided by velocity, with respect to exponent α. If α is 1, equivalent to oil damper without relief valve, equivalent viscous coefficient is constant since damping force is proportional to velocity. On the other hand, as α gets closer to 0, it becomes very large under small velocity. This represents that the distribution of damping force between dashpot element and spring element of Maxwell body varies with velocity. (Kasai [2], [3]) Especially under extremely small velocity, deformation of dashpot element might not arise. Therefore, it is desirable to consider the possibility of reduction of damping force under extremely small amplitude for viscous damper, which is applied to the building actually.

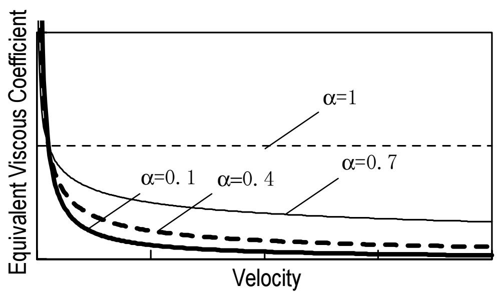
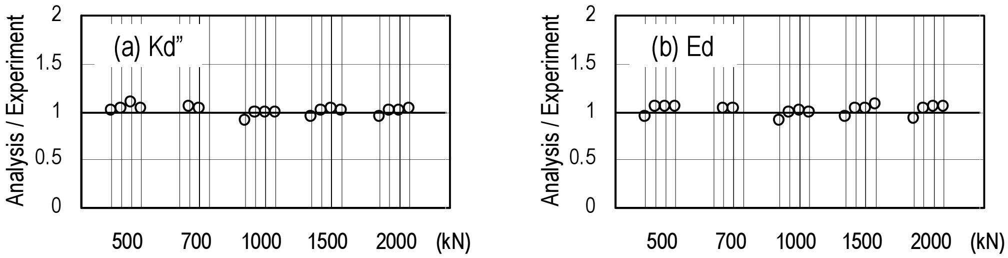


Fig. 6 Alteration of Equivalent Viscous Coefficient by Exponent α

RESULTS OF TIME HISTORY ANALYSIS AND THEIR VALIDITY

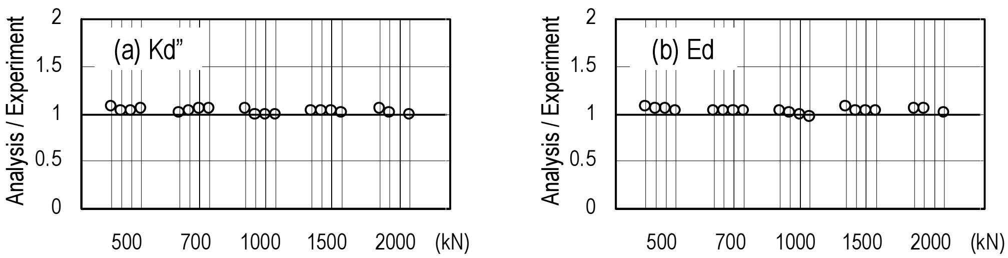
Figs. 7 and 8 represent the results of time history analysis of viscous dampers, whose exponent α is 0.1, under sinusoidal excitation divided by those of experiment for loss stiffness K_d" and absorbed energy in a stable loop E_d. Loss stiffness K_d" is calculated by dividing maximum damper force of hysteresis loop by maximum deformation. Loss stiffness K_d" and absorbed energy E_d was obtained in the third loop of sinusoidal excitation. Those graphs take the capacities of dampers for each horizontal axis and show the dependency of viscous damper on frequency and amplitude of excitation. For the errors take 10 % at a maximum, the analysis model is well accurate independently on frequency or amplitude of excitation.

Fig. 9 represents hysteresis loop of viscous dampers, whose capacity varies 500, 1,000 and 2,000 kN, under sinusoidal excitation, and Fig. 10 represents those under seismic excitation. These show that the time history analysis model of viscous damper is well accurate under sinusoidal or seismic excitation.



(a) Loss Stiffness K_d" **(b) Absorbed Energy in a Loop E_d**

Fig. 7 Accuracy of Analysis Model (Exponent α: 0.1, Amplitude: 10mm)
(Frequency: 0.01, 0.2, 0.5, 1.5 Hz left to right for each damper capacity)



(a) Loss Stiffness K_d" **(b) Absorbed Energy in a Loop E_d**

Fig. 8 Accuracy of Analysis Model (Exponent α: 0.1, Amplitude: 50mm)
(Frequency: 5, 10, 20, 50 Hz left to right for each damper capacity)

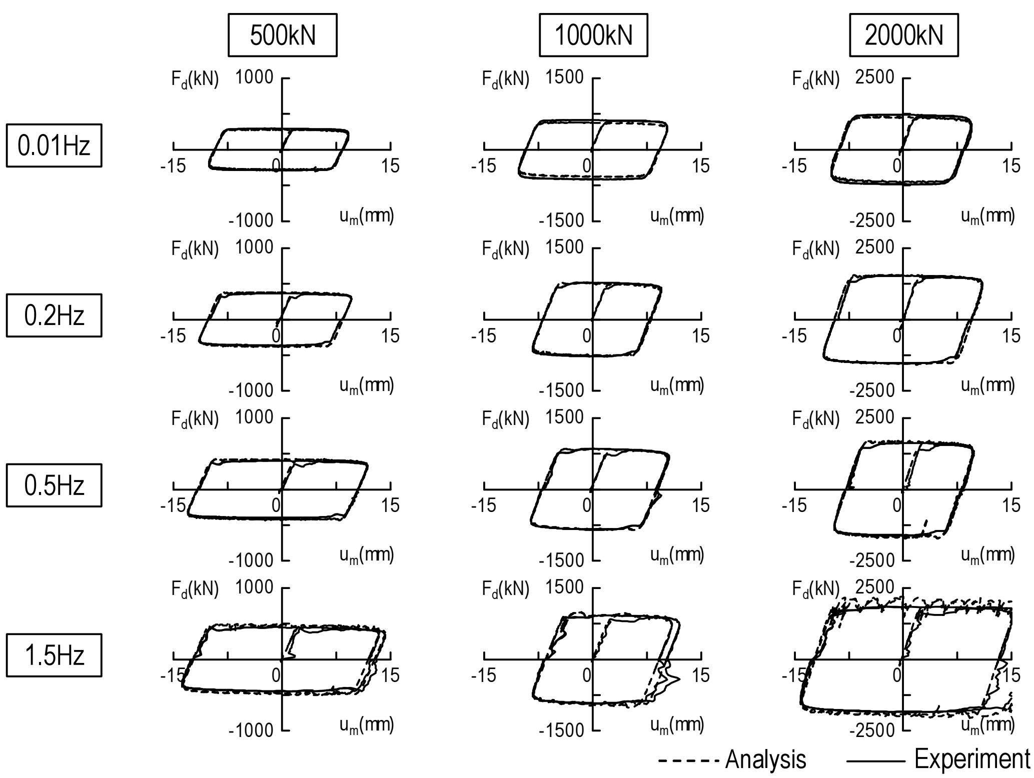


Fig. 9 Sinusoidal Responses with Respect to Frequency of Excitation
(Damper Capacity: 1,000, 1,500, 2,000 kN)

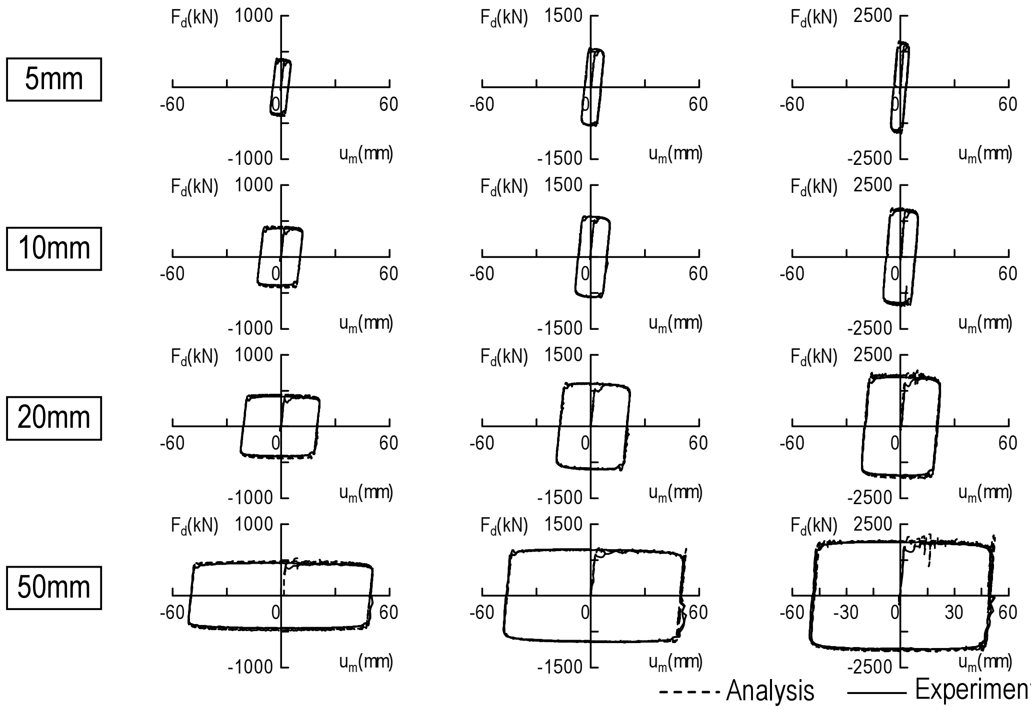
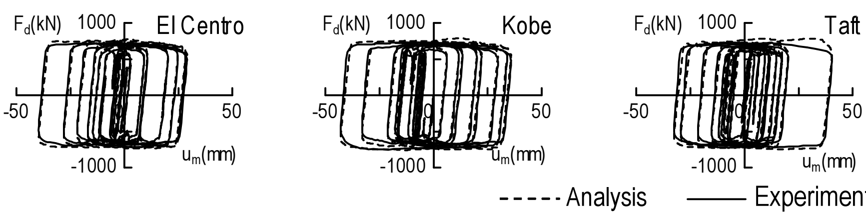


Fig. 10 Sinusoidal Responses with Respect to Amplitude of Excitation (Damper Capacity: 1,000, 1,500, 2,000 kN)



(a) El Centro NS **(b) JMA Kobe** **(c) Taft EW**

Fig. 11 Seismic Responses (Damper Capacity: 700 kN, First Eigen Period of Building: 0.35 sec)

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