



JSSI MANUAL FOR BUILDING PASSIVE CONTROL TECHNOLOGY PART-10 TIME-HISTORY ANALYSIS MODEL FOR NONLINEAR OIL DAMPERS

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SUMMARY

Oil dampers have been applied to many buildings in Japan, due to their advantages as follows. 1) The hysteresis loop is oval that shows they have high damping performance. 2) The damping performance is independent with temperature from -10°C to +80°C. 3) Time history analysis model is represented as concise Maxwell body and simulates behavior of actual oil damper well. This paper introduces an overview of time history analysis model of oil damper and points to notice to apply the model to time history analysis.

For the damping performance varies with the stiffness of oil, clevises and cylinder rod, oil damper is represented by Maxwell body, a serial connection of an elastic spring and a linear or a bilinear dashpot. For the given increment of the model deformation, the force is calculated by solving equilibrium equation of the spring and the dashpot with an assumption of a constant velocity during each time step. When time step interval is small enough, numerical result matches very well with one of experiment. 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 oil dampers or viscous dampers [1].

INTRODUCTION

With passively controlled buildings with oil dampers, dampers are generally installed with V type braces, or connected to braces in series, and their damping force is controlled with their relief valves so that the force transmitted to the structure may not be excessive. On the other hand, in seismic isolation buildings, oil dampers which have small damping coefficient and no relief valve are used in general.

It is desirable that the time history analysis model of passively controlled buildings with oil dampers can be simulated as precise as possible. Since the parameters for the analysis model which are shown by manufacturers of oil damper may be determined by performance of small dampers, it is expected to verify the performance of oil dampers which are applied to the building actually. It is to be noted that oil damper

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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. Furthermore, damping performance of response under extremely small excitation (less than 0.2 mm) of oil damper appears to be worse than the standard. (Ooki [2])

Fig. 1 shows the schematic diagram for added component which consists of oil 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.

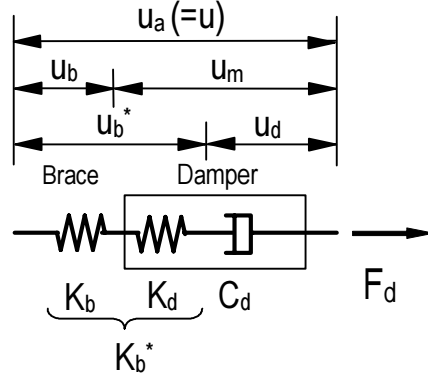
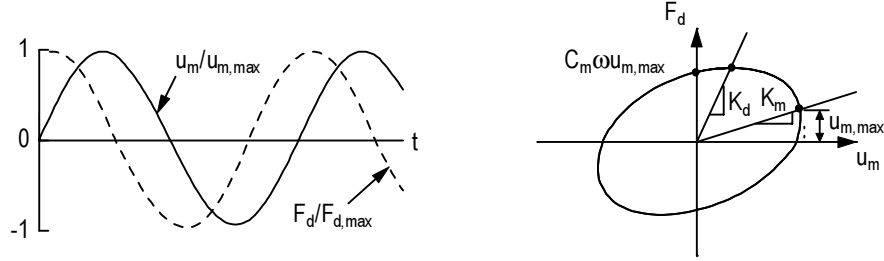


Fig. 1 Schematic Diagram for Added Component

BASIC HYSTERESIS CHARACTERISTICS OF OIL DAMPER

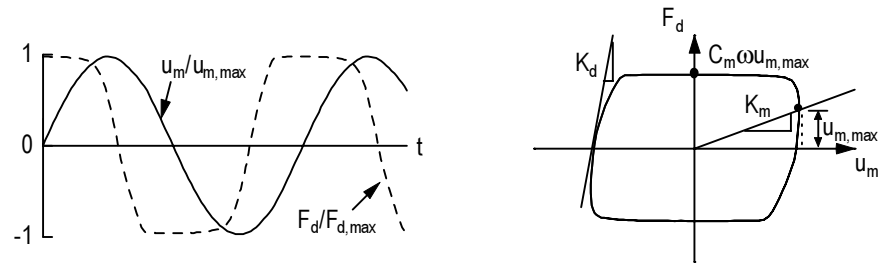
The basic principle of oil damper is expressed as a piston structure. When oil damper receives a disturbance, oil will flow by increase of internal pressure and generate damping force. Fig. 2 (a) shows time history of deformation and damping force of oil damper without relief valve under sinusoidal



(a) Time History

(b) Hysteresis Loop

Fig. 2 Behavior of Oil Damper under Sinusoidal Excitation (Without Relief Valve)



(a) Time History

(b) Hysteresis Loop

Fig. 3 Behavior of Oil Damper under Sinusoidal Excitation (With Relief Valve)

excitation with circular frequency ω and amplitude $u_{m,max}$. And the hysteresis loop shows a declined ellipse as in Fig. 2 (b). It is noted that equivalent stiffness K_m and equivalent viscous damping coefficient C_m of oil damper in this figure differ from internal stiffness K_d and internal viscous damping coefficient C_d shown in Fig. 1. On the other hand, hysteresis loop of oil damper with relief valve shows swelled parallelogram since the relief valve controls excessive damping force as shown in Fig. 3 (b).

The stiffness of the hysteresis loop results from internal stiffness of oil damper by compression stiffness of oil, and it becomes prominent under excitation under high frequency. Fig. 4 represents the internal stiffness K_d and the internal viscous damping coefficient C_d of oil damper. (Kasai [3]) They are in about a relationship $K_d = 10C_d$, and the internal stiffness is much less than one of clevises or cylinder rod. Therefore, it is inappropriate to consider the internal stiffness as infinite, and it should be regarded correctly in analysis model.

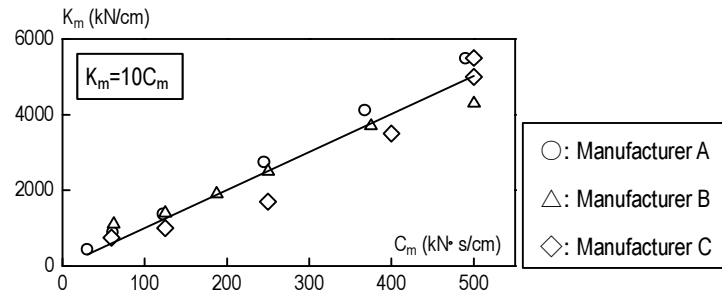


Fig. 4 Relationship between Internal Stiffness K_d and Internal Viscous Damping Coefficient C_d (Damper Capacity: 125 to 2,000 kN)

To accommodate the hysteresis loop of oil damper to one of analysis model, the considerable points are as follows.

- 1) To install pins at the both ends of damper to avoid bending moment.
- 2) To keep the input velocity below each own capacity.
- 3) To mention the reduction of the damping performance under extremely small amplitude.

About 2), the linearity between velocity and damping force of oil damper has its bound, over which the damping force will proportional to the second power of the velocity. The reduction mentioned at 3) results from the air bubble in oil which cannot be removed utterly and the sliding friction of damper. Under extremely small amplitude, the form of the hysteresis loop maintains declined ellipse, but the damping performance becomes less than the standards.

SUMMARY OF TIME HISTORY ANALYSIS MODEL OF OIL DAMPER

In general, time history analysis model of oil damper is modelled as Maxwell body shown in Fig.1. Maxwell body consists of dashpot, which has internal viscous damping coefficient C_d , and elastic spring, which has internal stiffness K_d , connected in serial. Fig. 5 represents the dependency on frequency of Maxwell body while relief valve does not work. This accommodates one of actual oil damper.

On the other hand, when relief valve works, internal viscous damping coefficient will reduce and the damping force will be controlled. To simulate this behavior, relationship of damping force and velocity of dashpot of Maxwell body is modeled as bilinear shown in Fig. 6 (b), where p represents the reduction factor of viscous damping coefficient. The values of these parameters, C_d , K_d , p and relief velocity, are shown by manufacturers.

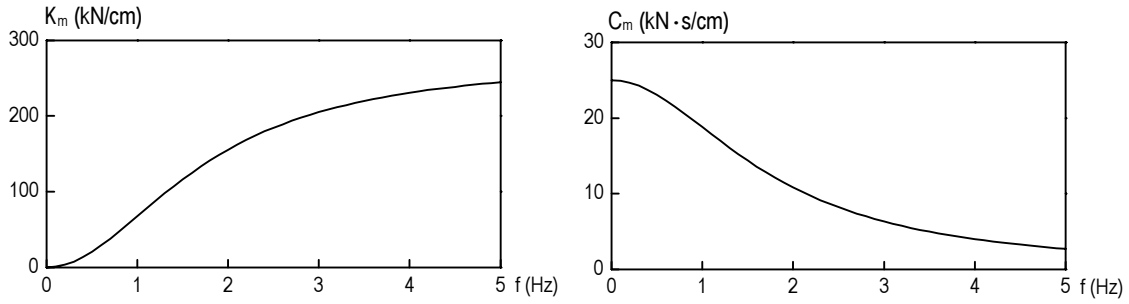
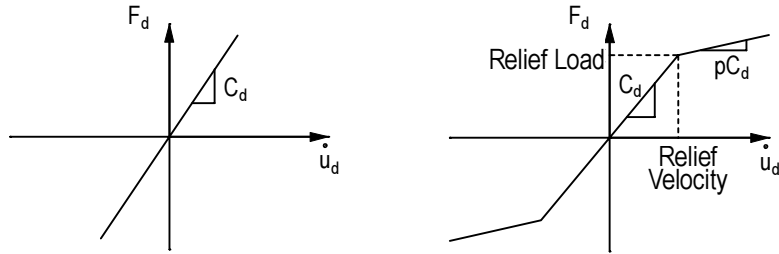


Fig. 5 Dependency on Frequency of Maxwell Body
(where $K_d = 275$ kN/cm and $C_d = 25$ kN·s/cm)



(a) Without Relief Valve

(b) With Relief Valve

Fig. 6 Relationship between Damping Force F_d and Velocity \dot{u}_d

RESULTS OF TIME HISTORY ANALYSIS AND THEIR VALIDITY

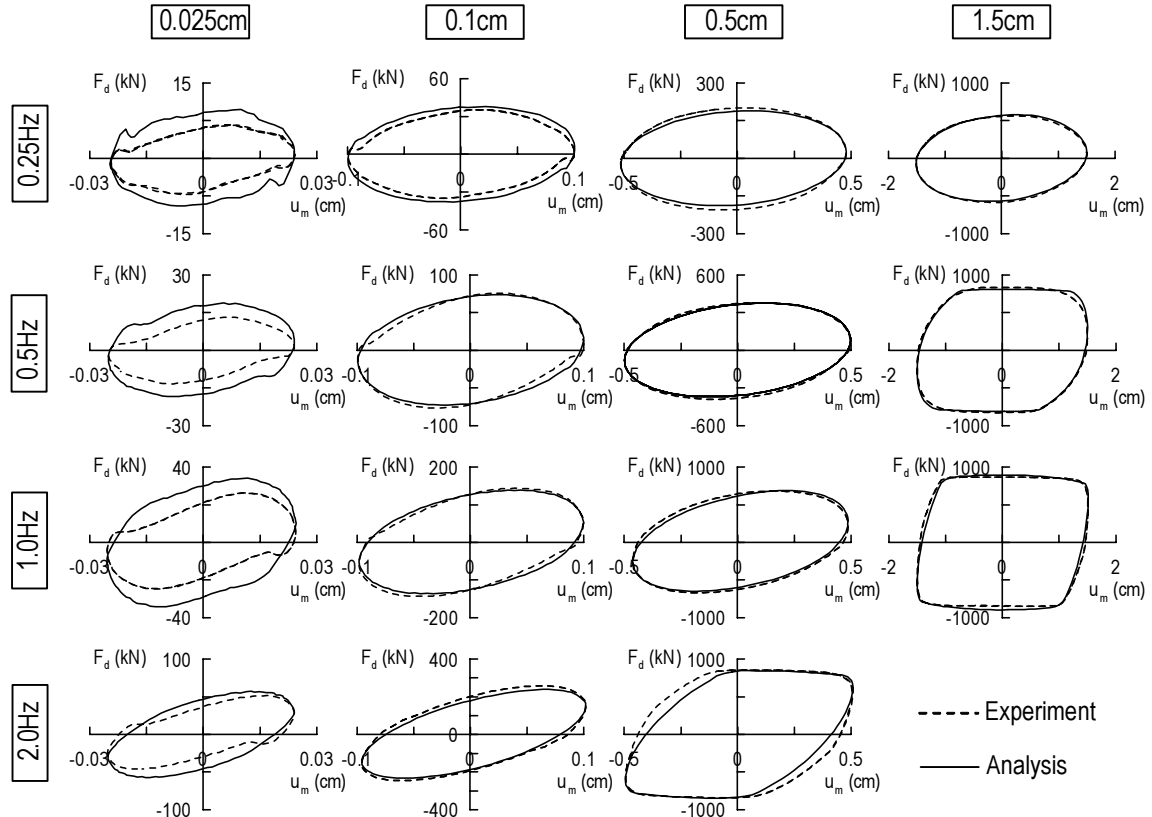


Fig. 7 Sinusoidal Responses (Damper Capacity: 1,000 kN)

Fig. 7 compares hysteresis loops of experiment and analysis of oil damper under sinusoidal excitation. The damper capacity of the damper is 1,000 kN and its relief velocity is 32 cm/sec. The responses under amplitude from 0.1 to 1.5 cm match very well, and it means that the analysis model described above is valid. But the responses under amplitude 0.025 cm don't match for the reduction of damping performance as mentioned above.

Fig. 8 compares hysteresis loops of experiment and analysis of oil damper under seismic wave. The input wave is two times differentiated of the acceleration of JMA Kobe, whose maximum value is modified for the capacity of the testing machine. The damper capacity is 500 kN and its relief velocity is 32 cm/sec. The responses match very well as ones under sinusoidal excitation, and show the validity of the analysis model.

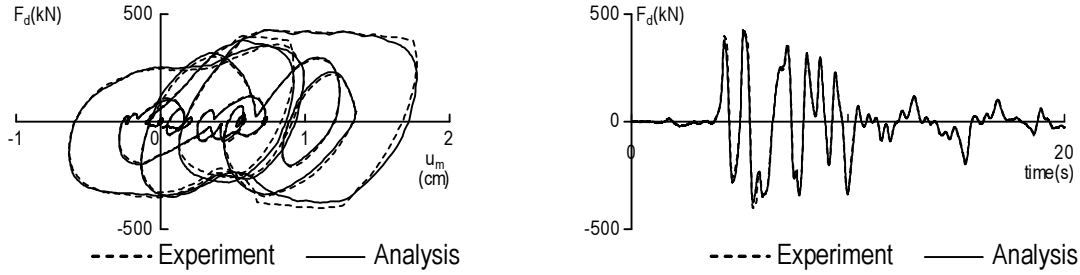


Fig. 8 Hysteresis Loop and Time History under JMA Kobe (Damper Capacity: 500 kN)

Fig. 9 represents the values of results of analysis divided by ones of experiment about maximum damping force F_{max} , absorbed energy by loop E_d , equivalent stiffness K_d and equivalent damping factor h_{eq} . The equivalent stiffness K_d was obtained by hysteresis loop with least-square method for the results under amplitude 0.025 to 0.1 cm, and it was obtained as the equivalent stiffness at the maximum deformation for the other results. The equivalent damping factor h_{eq} was obtained by the expression shown as Eq. 1 where u_{max} represents the maximum deformation.

$$h_{eq} = \frac{E_d}{2\pi \cdot K_d \cdot u_{max}^2} \quad (1)$$

Errors of the results for amplitude 0.025 cm are large because of errors of hysteresis loops, but the other results show the accuracy of the analysis model.

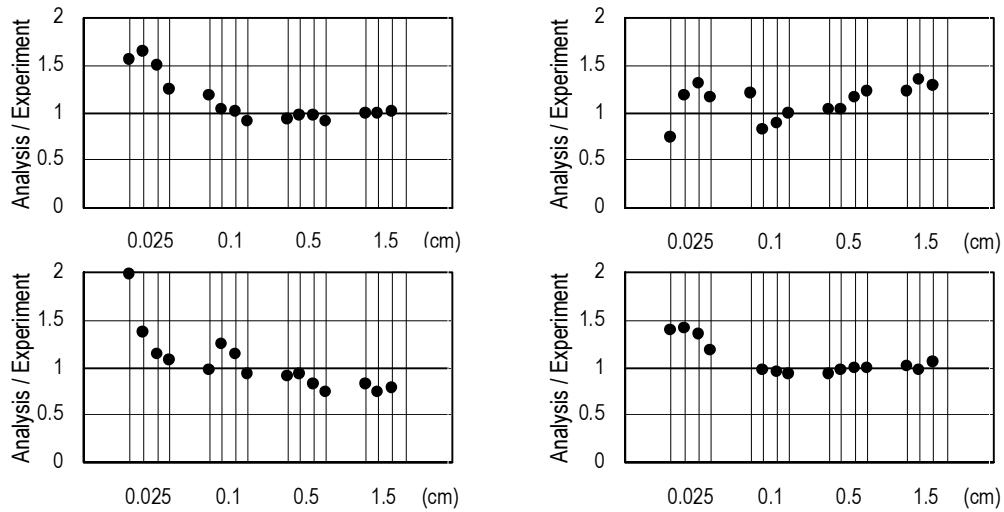


Fig. 9 Accuracy of Analysis under Sinusoidal Excitation (Damper Capacity: 1,000 kN)

Fig. 10 shows equivalent viscous damping coefficient C_m , obtained by experiment, normalized by the standard value at 20 °C $C_m(20^\circ\text{C})$ for -30 to +80 °C. The values of three capacities of oil damper are plotted for each temperature. Errors with respect to the standards are below 3% for -10 to +80 °C, but they are increased by 5 to 10% for lower temperature. Without proper oil for low temperature, the fluctuation of viscous damping coefficient needs to be considered in time history analysis model of oil damper.

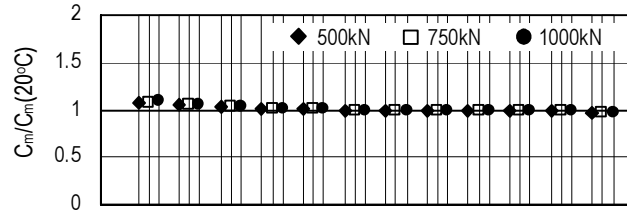


Fig. 10 Dependency on Temperature of Equivalent Viscous Damping Coefficient C_m

Fig. 11 compares hysteresis loop of results of time history analysis and the theoretical solutions under sinusoidal excitation with 5 Hz frequency, with respect to time interval of time history analysis. With the model of the analysis, velocity of dashpot is assumed as average velocity between couple of each time steps. If the time interval is not short enough, this assumption will not be materialized and the accuracy of result will be worse as shown in Fig. 11 (c).

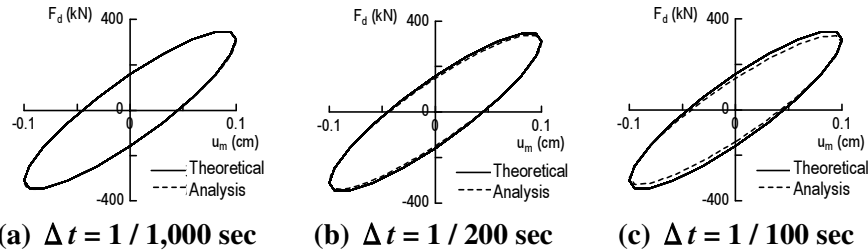


Fig. 11 Comparison with Theoretical Solution with Respect to Time Interval Δt (Sinusoidal Excitation 5 Hz)

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