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Development of the large capacity high damping oil damper

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ABSTRACT: This paper deals with the development of a large capacity high damping oil damper which is an aseismatic device for the piping system, equipment, etc in nuclear power plants. The oil damper was tested to verify that it allowed thermal expansion of the piping system and its damping force was proportional to the velocity of the seismic vibration and free from temperature dependency. In this development, a small size oil damper with simple construction was also fabricated and tested to investigate the basic hydrodynamic mechanism of oil dampers. The large capacity high damping oil damper was developed bearing in mind these test results. The rated load of this oil damper was 2000kN.

1. INTRODUCTION

Various dampers have been studied as aseismatic devices for piping systems, equipment, etc. in nuclear power plants. The dampers utilizing plastic deformation of metallic materials or utilizing extrusion of lead are included among them. The dampers used in nuclear power plants should necessarily follow the thermal expansion of piping system and equipment. The oil dampers used widely in various industrial fields allow such thermal expansion without unnecessary force.

Oil dampers can be classified into two typical types. One is the viscosity-resistance damper utilizing high viscosity oil in a laminar flow. The damping force of this type is proportional to the velocity of the vibrating object. The second is the constant orifice damper which utilizes oil in a turbulant flow. The damping force of this type is proportional to the square of the velocity of the vibrating object. However, these oil dampers have a damping force which varies considerably due to the ambient temperature and to the heat generated in oil during vibration. Also, in the case of the latter type of dampers, estimating the seismic response of piping system is complicated.

The authors have developed a velocity proportional type oil damper by paying attention to the design of the orifice and have verified experimentally that the damper allows free thermal expansion of the piping system, and the damping force is independent of the temperature. Also, in order to investigate the basic hydrodynamic mechanism of viscous damping, a test was performed by using a small size oil damper with simple construction.

2. DESIGN OF THE OIL DAMPER

When designing an oil damper, the shape of the small tube or fine slot is an important factor. A circular tube and an orifice have already been designed from simple formulas[1],[2]. Also, Sekiguchi and Asami have studied a cylindrical-orifice oil damper and of an oil-film damper based on theoretical analysis and experiments[3],[4]. However, there seems to be few reports on the design formulas for the velocity proportional type damper under the turbulent flow, which is the fundamental function of this study.

Theoretical formulas have been derived based on the following assumptions:

1. The fluid is in the form of a fully developed turbulent flow.
2. The fluid is incompressive Newtonian fluid.
3. The influence of mass transfer of the fluid and of the piston is negligibly small.
4. Estimation of the damping force is made as a steady flow.

As shown in Fig.1, there is a sudden repetitive contraction and expansion of the flow in the damper. The energy loss generated during conversion of pressure energy to velocity energy contributes to the damping effect. It is generally considered that the energy loss is large in the case of sudden expansion[5]. Therefore the structure of the damper was devised to make the damper a turbulent condition at the suddenly expanding portion. An analysis of the damping force accentuated on the suddenly expanding portions was attempted in the following.

The head of loss caused by sudden expansion "h" can be expressed by the Bernoulli's formula as follows:

$$q_1{}^2/2g + p_1/\rho g = q_2{}^2/2g + p_2/\rho g + h \qquad (1)$$

where (q_1, q_2) and (p_1, p_2) are the velocities and the pressures of flow at the suddenly contracted portion and the fully expanded portion; g, r and h are the gravitational acceleration, the density of the fluid and the head of loss respectively. When the pressure of the sudden expanding portion is represented with P_x , the law of momentum is applied:

$$p_1 a + p_x (A-a) - p_2 A = \rho(q_2{}^2 A - q_1{}^2 a) \qquad (2)$$

where A is the area of the piston and a is the area of the orifice. Then, since

$$q_1 a = q_2 A \text{ (equation of continuity) and } p_x \simeq p_1 \qquad (3)$$

By substituting Eqs.(2) and (3) into (1), we obtain

$$h = (1 - a/A)^2 q_1{}^2 \; / \; 2g \qquad (4)$$

By taking the fact that $p_x = p_1$, $a/A = 0$ into consideration and introducing into Eq.(4) the notation $\zeta = (1-a/A)$, we obtain

$$h = \zeta^2 q_1{}^2 \; / \; 2g \qquad (5)$$

On the other hand, under the assumption that there is no energy loss at the contraction side, the Energy Equation for a steady flow becomes:

$$q_0{}^2/2g + p_0/\rho g + H = q_1{}^2/2g + p_1/\rho g \qquad (6)$$

where q_0 and p_0 are the velocity and the pressure at the sufficiently far portion from the orifice. In Eq.(6), considering that q_0 is small ,introducing the notation C_v (Coefficient of Velocity)[2] ,head H becomes:

$$H = q_1^2 \;/\; (C_v^2 \cdot 2g) \tag{7}$$

In Eqs.(5) and (7), if $\zeta = 1/Cv$, then the Eq.(5) may be the same as Eq.(7). This means if the damping occurs by the head of loss at the sudden expanding portion, the design of a damper may be achieved on the basis of dynamic pressure resistance which is led from Eq.(7). Based on this idea, the damping strength was evaluated by the dynamic pressure resistance. The basic equation for the force of dynamic pressure is given as:

$$F = \rho A^3 v^2 \;/\; 2Cd^2 a^2 \tag{8}$$

where F is the force of dynamic pressure, v is the the velocity of the piston and Cd is the flow coefficient.

Paying heed to "a" orifice area in Eq.(9), and when the shape of the pressure control valve could be determined such that:

$$a = c_1 (y)^{1/2} \tag{9}$$

Then, $$F = ky = pA_U \tag{10}$$

where c_1 is the coefficient of the valve lift, y is the displacement from the valve seat, k is the spring constant of the spring of the pressure control valve and Av is the effective area of the pressure control valve respectively. Based on the above equations , the damping force and the damping coefficient respectively become :

$$F = (A^2/C_d c_1) \cdot (\rho k/2A_v)^{1/2} \cdot v \tag{11}$$

$$C = (A^2/C_d c_1) \cdot (\rho k/2A_v)^{1/2} \tag{12}$$

The design of the damping force in this study was accomplished using Eqs.(11) and (12).

Figure 2 shows the outline of the large capacity oil damper developed on the basis of the design method mentioned above.Silicone oil, having high temperature resistance was selected as a hydraulic fluid. Both metal rings and synthetic rubber "o" rings both were used in the large capacity oil damper for oil sealing. Figure 3 shows the hydraulic pressure circuit of the damper. The construction of the small size oil damper was identical to Fig.1. This damper had no oil reservoirs and check valves.

3. TEST

The investigation concerning the basic damping characteristics of the oil damper was performed by using the small size oil damper. Both the pressure chambers, the tension side and the compression side, are connected to each other only through the orifice set up in the piston. The relationship between the damping force and the state of the flow of the oil in the orifice were investigated for each of the orifices with different shapes shown in Table 1. The damping force was measured under the steady flow by the movement of the piston with constant velocities.

A test of the large capacity oil damper was also performed. The force generated during the constant low running speed of the piston was measured. This test was performed to evaluate the reaction force against the thermal expansion of the piping system.

The evaluation of damping was made on the basis of reaction forces generated when sinusoidal vibrations were applied to the damper. Fig.4 shows the measuring methods. A forced sinusoidal displacement was applied to the damper, and the reaction force was measured. The test was performed in a frequency range of 0.5Hz - 10Hz. The damper was designed to exert the maximum resistance of 2000kN at the maximum capacity of the testing machine which has the following specifications and conditions : Frequency : 3.18Hz, Velocity : 40kine, 2000kN.

In order to evaluate the damping characteristics for the variation of the oil temperature and durability of the damper against repeated vibrations caused by earthquakes, the endurance test was made with 600 cycles under the maximum capacity of the testing machine on the damper.

4. TEST RESULTS

The test results of the small size oil damper are shown in Fig.5. From the results, no effect on the orifice length from the damping force was obserbed. The factor which infuluences the damping force drastically is the diameter of the orifice. Also, the shapes at the endurance portion and the exit portion of the orifice varies the damping force.

Figure 6 shows the result of the low running speed test. The damping force is less than 1% of the rated load of the damper.

Hysteresis loops (displacement-force) at 0.5, 1.0, 2.0, and 3.0Hz under the displacement of approximately 20mm are shown in Fig.7. Fig.8 shows hysteresis loops in the case of displacement of 5, 10, 15, and 20mm displacement levels at a frequency of 3.18Hz respectively.

Figure 9 shows the maximum velocity on the axis of abscissa and the generated force on the axis of ordinate on a basis of Fig.7 and Fig. 8. According to Fig.9,it is confirmed that the damper is a type of $F = Cv$.

There was no abnormality observed during the vibration test with a load close to the rated load of 2000kN repeated for 600 cycles. Fig.10 shows the hysteresis loops at the beginning of the test(room temperature)and the 370 cycles (140 ^{0}C) respectively. The damping force and the dissipated energy maintained constant values regardless of cycles and oil temperature.

5. CONCLUSIONS

The following conclusions were reached through this study:

1) It is confirmed that the developed oil damper is $F=Cv$ type damper which is free from the influence of the temperature and friction force of the damper is less than 1% of the rated load.

2) In the light of protection against earthquakes, the durability of the damper is sufficient.

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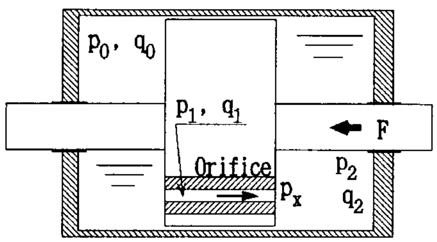


Fig.1 Pressure-flow condition

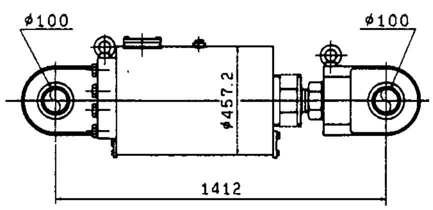


Fig.2 Outline of oil damper

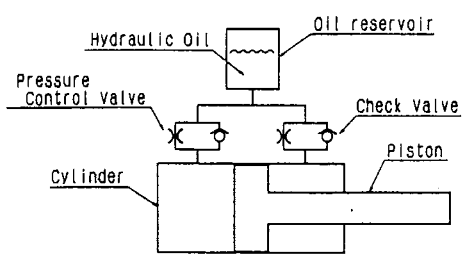


Fig.3 Oil circuit of oil damper

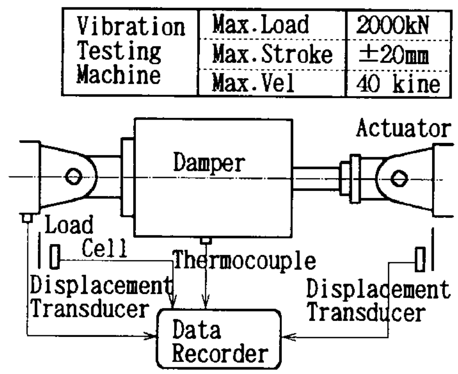
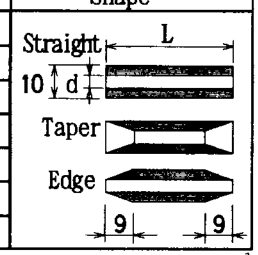


Fig.4 Schematic Diagram of Dynamic Test

Table 1 Specification of orifice

Orifice	d	L	Shape
Straight	4	10	
Straight	4	20	
Straight	4	40	
Straight	3	40	
Straight	5	40	
Taper	4	40	
Edge	4	40	

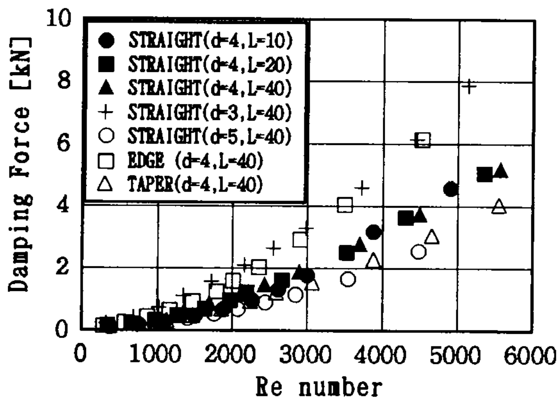


Fig.5 Damping Force
(Small Oil Damper)

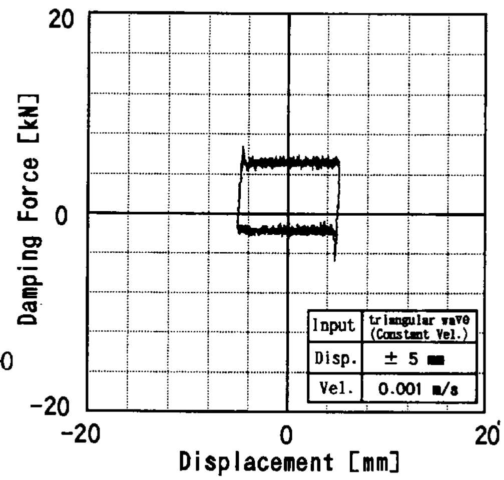


Fig.6 Damping force in low running speed test

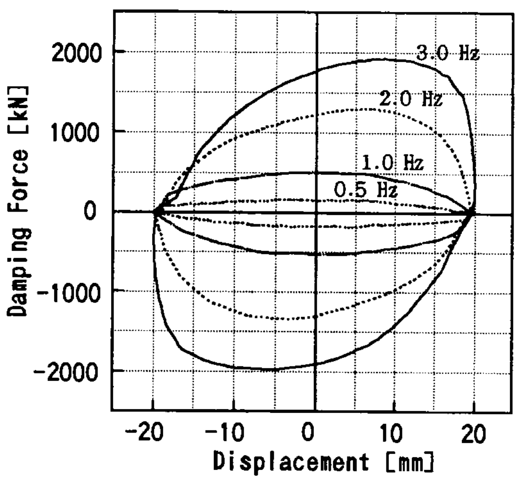


Fig.7 Hysteresis Loop in Vibration Test (Constant Displacement)

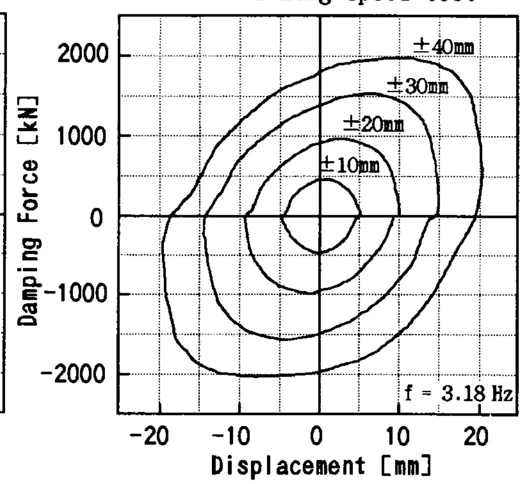


Fig.8 Hysteresis Loop in Vibration Test (Constant Frequency)

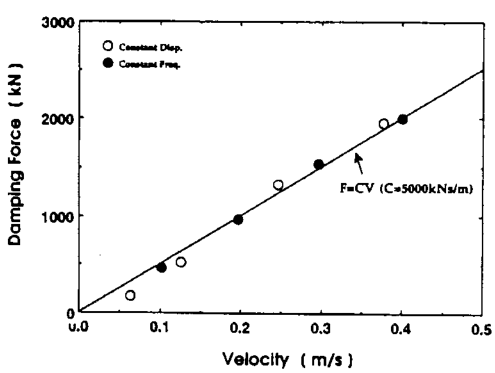


Fig.9 Damping force
(Large damper)

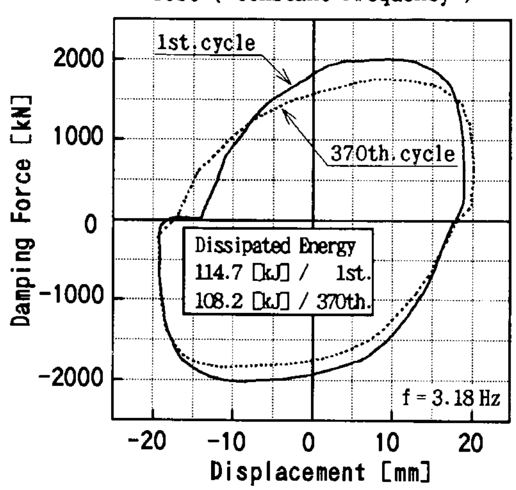


Fig.10 Hysteresis Loops in Endurance Test