621.042.7:621-752

Application of the Oil Damper to Spherical Tank in Earthquake-Resistant Design*

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Spherical tanks at present are designed so as not to break down or collapse even when they are exposed to a typical earthquake like the El Centro. However, if the oil damper is applied to the tanks, their resistance to earthquakes will be increased even more. This paper deals with a theoretical analysis on the suitable damping coefficient of a double acting damper fitted horizontally between the bottom of the tank and its base. In the theory, the optimum damping was selected to minimize the transmissibility resonance curve in the predominant frequency range of earthquakes. A scheme for the application of the damper was proposed, with which forced and free vibration tests were conducted for an actual 200,000 m³ gas tank with and without a damper. As a result of the studies it was proved that the application of a damper to spherical tanks is very effective in increasing their resistivity to earthquakes.

1. Introduction

The number and scale of chemical plants at present are far larger than those of the day of the last Kanto Earthquake of 1923. Earthquake disaster of chemical plants, therefore, is widely discussed in these days.

If facilities in chemical plants, particularly storage tanks for dangerous article, were broken down by earthquake, a large calamity would be caused. Severe design criteria, therefore, are commonly used for the design of storage tanks, and they are so designed as not to break down or collapse when they are exposed to strong earthquake.

An increasing number of spherical storage tanks are now being used, which are usually supported in a manner shown in Fig. 1 and the support is composed of columns and tie-rods which are connected to adjacent columns. Spherical tanks are designed to avoid plastic deformation due to horizontal earthquake acceleration of 0.3g.

According to elasto-plastic analysis carried out recently, the currently used spherical tanks will withstand the El Centro earthquake or similar one in magnitude, although their supports may partly be plastically deformed.

The vertical component of earthquake acceleration usually is less intense as compared to the horizontal component and is only a fraction of gravitational ac-

celeration. Hence, only in special circumstances, vertical acceleration should be taken into consideration for gravity loaded structures.

As mentioned above, the currently used spherical tanks are sufficiently designed to withstand earthquakes. If, however, some damping device can be applied to the support of the tank, the relative displacement between the tank and its base can be remarkably decreased, and the tank will be duly safer in even larger earthquake.

The most fundamental damping equipment is the oil damper which is widely used in automobiles. Large oil dampers are manufactured today to be used for highway bridges to prevent earthquake damage. Since oil dampers used for the above purpose are sufficiently large, they can readily be applied for use in large spherical tanks. Oil dampers can be easily attached to the spherical tank between the bottom of the tank and its base in horizontal direction.

Theoretical analysis on the suitable damping coefficient of the double acting dampers attached to the tank was carried out. A vibration field test on the actual large spherical tank with dampers was also carried out.

The results of theoretical analysis and experiment are presented herein.

This paper only deals with the spherical tanks for gas. In comparison with the weight of the tank, the vibration characteristic of the tank is hardly influenced by the weight of gas. On the other hand not only the weight but also the sloshing effect must be considered in case the content is liquid. Vibration behaviour of the spherical tank for liquid is not referred to in this paper, although the application of dampers for improving antiseismal

* Received 22nd August, 1973

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performance against earthquake can be confirmed through almost the same theoretical analysis as that for the gas tank.

2. Response spectrum and aseismic design

2.1 Response spectrum curves

The use of the so-called spectrum curves is customary to describe response of structures to earthquakes. The curve shows the relationship between the maximum velocity or acceleration of a system having a single degree of freedom during an earthquake and the natural period, taking the fraction of critical damping as parameter.

A typical acceleration spectrum curve is shown in Fig. 2. In this figure, the maximum acceleration during earthquake is taken as ordinate and the natural period of the structure is taken as abscissa.

These curves in Fig. 2 are the average of the two horizontal component of the four strong ground motions recorded in the U.S.A. Before averaging, each spectrum was normalized so that there was a unit area under the zero-damped curve. The ordinate of the average spectrum curves of Fig. 2 only indicates the normalized value. The average acceleration curves in Fig. 2 are indicative of the characteristics of the ground motion to be expected within 70 km or so of the center of a large earthquake. The peak of the curves in Fig. 2 shifts to the right as longer distance.

Recently, many strong ground motions have been recorded in Japan, from which an average response spectrum curves for earthquakes in Japan are obtained. The following statements can be made from the response spectrum curves:

- (1) Large values in response spectra can be expected for structures having a natural period between 0.1 and 1 second.
- (2) When damping is increased, the response spectrum of the structure is remarkably decreased.

2.2 Aseismic design

As indicated from the spectrum curves, one should note that aseismically designed structures should not have natural frequency within the range of 1 and 10 Hz. and also have a large damping coefficient. The weight of a structure must be minimal and at the same time the structure must have a high rigidity. These are required conditions to keep the natural frequency of a structure above 10 Hz. On the other hand from the requirements to keep the natural frequency less than 1 Hz., the response spectra for a large distance from epicentre must be taken into account. Also resistance to wind is another factor not to be neglected in design of light weight structures. Neither an extremely rigid nor a flexible structure is economical despite the technical feasibility. Therefore, from a practical view point, the goal for aseismic design should be approached by finding a method to increase the damping coefficient of a structure.

In dynamic analysis of aseismic design of structures, the following methods are customarily used:

- (1) Modal analysis using response spectrum curves for ground motion.
- (2) Time history modal analysis using the recorded ground motion.
- (3) Time history response analysis by integrating the differential equation of motion.

The methods (1) and (2) are adequate to analyze a spherical tank without dampers. But they are inadequate to determine the suitable damping coefficient of oil dampers attached to a tank, because the damping force of the dampers is so large that the vibration modes of tanks with and without dampers are rather different. The above mentioned method (3) gives accurate solutions for the response of a spherical tank with or without dampers. However,

Fig. 1 Construction of spherical tank

Fig. 2 Average acceleration spectrum curve for strong ground motion

solutions can be obtained for only limited ground motions and they lack in generality.

Authors, therefore, derived resonance curves for spherical tanks subjected to sinusoidal base movement. Damping coefficient of dampers is so selected that the transmissibility of force or displacement is not so large in the predominant frequency range of earthquake, 1 to 10 Hz.

3. Spherical tank as vibration system.

A spherical tank is shown schematically in Fig. 1 when an external horizontal force is applied to the spherical tank; it generally produces not only horizontal displacement but also rotation of the tank. For example, if a horizontal force is applied at the top of the tank towards right in Fig. 1, the tank makes both horizontal displacement toward right and also clockwise rotation. If a horizontal force is applied at the bottom of the tank in the same direction, the tank makes horizontal displacement in the same direction but anti-clockwise rotation. Therefore, if the force is applied at a certain point of the tank, only the horizontal displacement will be produced without any coupled rotation. Such a point is the elastic center of the tank.

The elastic center of a tank can be accurately determined by experiment, but it is reasonable to consider that this point is in the vicinity of the upper intersecting point of the tie-rod and column.

The torsional spring constant about the horizontal axis through elastic center E in Fig. 1 is

$$K_{\phi} = \frac{AE}{H} \sum_{i=1}^n e_i^2 + 2 \frac{aE}{\ell} \sin^2 \psi \sum_{i=1}^{n/2} e_i^2 \quad \dots (1)$$

where A and a are the cross-sections of a column and a tie-rod respectively, E is the modulus of elasticity, ℓ is the length of tie-rod, n is the number of columns, and ψ and e_i are the values given in Fig. 1.

The first term on the righthand side in Eq. (1) is the torsional spring constant of the columns and the second term is that of the tie-rods. Columns resist both tensile and compressive force, whereas tie-rods only resist tensile force. Therefore in the first term of Eq. (1), summation is done for all the columns n, but in the second term summation is done only for half the number of tie-rods, n/2. The spring constant at the elastic center in horizontal direction K_y is,

Fig. 3 Equivalent system with damper

$$K_y = n \frac{3EI}{H^3} + \frac{aE}{\ell} \cos^2 \psi \sum_{i=1}^n \cos^2 \theta_i \quad \dots (2)$$

Where EI is the bending stiffness of the column, and θ is the angle indicated in Fig. 1. In Eq. (2), the columns are assumed to be pin joints since they are not so tightly fixed to the foundation.

An equivalent vibration system for the spherical tank with oil dampers is shown in Fig. 3, where C is the damping coefficient of the dampers, E is the elastic center, K_{ϕ} and K_y are spring constants about point E, m is the mass of the tank, and i is the radius of gyration around the horizontal axis through the center of mass. In Fig. 3 y denotes the horizontal displacement of the center-of-mass and ϕ is the rotation angle, positive directions of y and ϕ are shown in Fig. 3.

In the following section, a two-degree-of-freedom system is analysed to obtain the transmitted force through the support K_y and the oil damper C.

The object of attaching dampers to the tank is to decrease the transmitted force through K_y . But the force transmitted through C should not increase. The important thing is to balance these two forces.

4. Equation of motion

If the base of the tank oscillates with given displacement y_0 in horizontal direction, the equation of motion is given in terms of y, the relative displacement of the center of gravity with respect to the base. The equations of motion are

$$\left. \begin{aligned} m\ddot{y} + K_y y + C\dot{y} - K_y s\phi - C(s+h)\dot{\phi} &= -m\ddot{y}_0 \\ mI^2\ddot{\phi} + (K_{\phi} + K_y s^2)\phi + C(s+h)\dot{\phi} - K_y s y - C(s+h)\dot{y} &= 0 \end{aligned} \right\} \quad \dots (3)$$

To obtain the frequency response of the structure, the forced displacement is assumed to be sinusoidal, $y_0 = a_0 \sin \omega t$. Then the response of this system is,

$$\left. \begin{aligned} Y &= Y_s \sin \omega t + Y_c \cos \omega t \\ \phi &= \phi_s \sin \omega t + \phi_c \cos \omega t \end{aligned} \right\} \quad \dots (4)$$

And the notations are defined as follows

$$\left. \begin{aligned} Y_s/a_0 &= T_s, Y_c/a_0 = T_c, K_{\phi}/K_y = e^2 \\ s\phi_s/a_0 &= T_{\phi s}, s\phi_c/a_0 = T_{\phi c}, K_y/m = v^2 \\ \omega/v &= \eta, c/(2\sqrt{mK_y}) = \zeta \end{aligned} \right\} \quad \dots (5)$$

Substituting Eq. (4) into Eq. (3), identifying the component of $\sin \omega t$ and $\cos \omega t$, and solving the simultaneous equations for T_s , T_c , $T_{\phi s}$ and $T_{\phi c}$ will give solutions. Calculating $T = \sqrt{T_s^2 + T_c^2}$ we obtain the resonance curve for the ratio of a_{rel} to a_0 , where a_{rel} is the displacement amplitude of the mass-center of the tank and a_0 is that of the base.

Forces transmitted through the supporting structure and dampers of the tank are given by the terms having K_y and C in Eq. (3). Using F_y and F_c for the amplitude of these, we obtain

$$\left. \begin{aligned} F_y/(a_0 K_y) &= \sqrt{(T_s - T_{\phi s})^2 + (T_c - T_{\phi c})^2} \\ F_c/(a_0 K_y) &= 2\zeta\eta \\ &\times \sqrt{T_s - (1 + \frac{h}{s})T_{\phi s}^2 + T_c - (1 + \frac{h}{s})T_{\phi c}^2} \end{aligned} \right\} \dots\dots\dots (6)$$

5. Resonance curves and optimum damping coefficient

Numerical values used for calculating for the spherical tank of 200,000 m³ are as follows

$$\begin{aligned} mg &= 1200 \text{ t}, mI^2 = 23.42 \times 10^8 \text{ kg cm s}^2 \\ K_y &= 72.5 \text{ t/cm}, e = \sqrt{K_\phi/K_y} = 78.4 \times 10^2 \text{ cm} \\ \nu/2\pi &= (1/2\pi) \sqrt{K_y/m} = 1.32 \text{ Hz} \\ s &= 398 \text{ cm}, h = 1502 \text{ cm} \end{aligned}$$

Resonance curves thus obtained are shown in Figs. 4 and 5. In Fig. 4, resonance at $\eta \approx 1$ is the transverse vibration mode and resonance at $\eta \approx 6$ is the rolling mode, the vibration center of which is slightly above the center of gravity. Resonance vibration mode at $\eta \approx 3.5$ with a large fraction of critical damping is the rolling mode, the center of which is the point where oil dampers are attached.

It is clear from this figure that in the case of light damping two peaks appear at $\eta \approx 1$ and $\eta \approx 6$ in the resonance curve. As damping increases, the peak at $\eta \approx 6$ will disappear and the peak at $\eta \approx 1$ will only decrease in value. The more the damping

Fig. 4 Resonance curves of relative displacement amplitude at center of gravity

Fig. 5 Resonance curves of transmitted force amplitude through damper

is increased, the lower is the value of this peak.

It should be noted from Fig. 4 that all resonance curves pass through the points P and Q independent of the degree of damping. Consequently at a certain degree of damping, the peak value coincides with that of point P and becomes a minimum. If damping becomes larger than this particular value, the peak of the resonance curve occurs at larger η , and approaches $\eta \approx 3.5$ and the ordinate of the peak is increased.

Judging from Fig. 4, it seems reasonable to determine ζ such that the point of resonance coincides with P in order to minimize resonance. This resonance curve is given when $\zeta \approx 0.8$. However this will be shown to be incorrect.

The resonance curves for force transmitted through the dampers are shown in Fig. 5. It is shown in this figure that if such a large fraction of critical damping as $\zeta = 0.8$ is assumed, the force transmitted through dampers becomes so large that the problem will arise whether the tank is sufficiently strong or not.

Resonance curves for the force transmitted through the support of the tank show almost the same tendency as that of Fig. 4. The optimum damping coefficient, therefore, is to be determined by considering the properties given in Fig. 4 and Fig. 5. The force transmitted through the support of the tank is distributed to the circumference of the spherical tank, whereas the force transmitted through dampers is concentrated at the bottom of the tank, so the latter must be kept smaller than the former. The characteristics of the frequency component for the displacement amplitude of the ground motion should also be considered. The choice of $\zeta = 0.10 \sim 0.15$ seems adequate.

6. A scheme for application of the damper

An earthquake will generate a horizontal force in arbitrary direction, thus the dampers must be installed to act equally effective to horizontal force in arbitrary direction. Therefore, three double acting dampers are installed at the bottom of the tank and its base at an angle of 120 degrees as shown in Fig. 6. There are two cylindrical brackets, one attached to the bottom of the tank and the other to its base. These brackets face each other. One end of the damper is connected to one bracket and the other end is connected to the other bracket with pins as shown in Fig. 6.

Damping coefficient of the oil dampers attached to the spherical tank of 200,000 m³ is 1.33 t/cm/sec per one damper. Damping coefficient of three dampers combined is about 2 t/cm/sec. This value satisfies the above mentioned requirement $\zeta = 0.1$.

Dimensions of the oil damper are the length 1,280 mm, the outer diameter of cylinder 218 mm, the diameter of piston 180 mm and the stroke ± 100 mm. Damping

force of the damper with respect to velocity of the piston is saturated as the velocity increases. The maximum force is 33 tons per one damper (50 tons for three dampers).

7. Experimental results

The forced and free vibration tests of an actual 200,000 m³ spherical tank were performed with and without dampers in November 1972. This tank is located in Minato-ku Tokyo, Japan, its diameter being 36 m, the weight 1,190 ton, the designed capacity of gas 170 tons and gas pressure 8.5 kg/cm². Tests were carried out with the tank in empty condition.

(1) Damping characteristic of the damper

The characteristic curve of the damper used in this test is shown in Fig. 7. This damper has a nonlinear characteristic as shown in Fig. 7. The average damping coefficient and the saturated damping force of the oil damper show the expected value. But within a range of small piston velocities, the damping force yielded smaller value than the expected value. And the vibration test was performed in this range.

In case of large earthquakes, the earthquake response amplitude of the tank will be so large that a larger damping force is expected as shown in Fig. 7, and the effect of oil dampers will be more pronounced than that of this vibration test at small piston velocity.

Fig. 6 Oil damper installation

Fig. 7 Damping force characteristics of damper

(2) Forced vibration test

Sinusoidal excitation force is applied to the bottom of the spherical tank by an electro-hydraulic actuator fixed between the bottom of the tank and its foundation. The excitation frequency was 0.5 to 10 Hz. and the maximum excitation force was 10 tons. Resonance curves for the amplitude at the top of the tank with and without dampers are shown in Fig. 8. The resonance at 1 ~ 1.5 Hz. is horizontal vibration, and the resonance at about 8 Hz. is rolling mode vibration as expected from theoretical analyses. Small resonances of tie-rods were also observed.

To calculate the spring constant K_y , it is assumed that the tie-rods can only hold tensile force. But in practice, tie-rods will hold both tensile and compressive force when displacement of the tank is not so large. Therefore, resonance frequencies observed in vibration tests were slightly

Fig. 8 Resonance curves of displacement amplitude (top of tank)

Fig. 9 Relation between exciting force and resonance peak amplitude

higher than those obtained through calculation.

Relation between the exciting force and the resulting amplitude at the bottom of the tank is shown in Fig. 9. The fractions of critical damping of the tank with and without dampers are 10.5% and 6.7% respectively. By attaching dampers, resonance amplitude of the tank is decreased by 64%.

(3) Free vibration test

A tension rod is connected to the bottom of the tank through a small piece. The piece has a notch in the center so that it will break at a given load and the tank will start damped natural vibration in horizontal direction. The waveforms of the damped natural vibrations of the tank with and without dampers are shown in Fig. 10. Fractions of critical damping of the tank with and without dampers are 12% and 6.7% respectively and are approximately equal to that obtained by forced vibration test.

(4) Experimental results

The application of oil-dampers to the spherical tank is very effective to insulate the structure from earthquake motion. Displacement amplitude is rather small to assure effectiveness of dampers in this vibration test. However, for the larger amplitude, the larger damping will be expected and consequently resonance amplitude of the tank with dampers will become about half or so of that of the tank without dampers. From these experimental results, we can say that in order

Fig. 10 Waveforms of damped natural vibrations

to use an oil damper more effectively, points to which special attention should be paid are as follows:

- (1) Damping force should be 30% larger
- (2) If there is some gap between damper and bracket, the damping force decreases.

8. Conclusions

The application of oil dampers to the spherical tank is theoretically very effective for increasing their resistance to the earthquake. This effectiveness is confirmed experimentally by the forced and free vibration tests of an actual 200,000 m³ spherical tank.

To attach oil dampers to the tank is so easy that this aseismic device can be applied to the tank already constructed. Moreover, the attaching and detaching of dampers are so simple that checking and maintenance are very easy. For preventing damage from earthquakes, installation of a set of oil dampers is found to be far more economical as well as easier than strengthening the columns or other structural parts.

Being applicable to aseismic design for facilities in chemical plants, this method is practical and is expected to be widely used. If a rational equivalent system is chosen, optimum damping coefficient can be defined by calculation for a vibration system with a few degrees of freedom.

Acknowledgement is made of Tokyo Gas Co. for their cooperation in the experiment. The assistance of the staff of the Ishikawajima-Harima Heavy Industries Co. is greatly appreciated.

Reference

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