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〈**Original**〉

Experimental Study on the Damping Characteristics of a Cylindrical Structure Containing Oil and Bearing Balls

윤활유와 베어링 볼을 내장한 원통형 구조물의 감쇠특성에 관한 실험적 연구

Bong Jo Ryu* and Seon Ho Song**

류 봉 조 · 송 선 호

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Key Words : Critical Damping(임계감쇠), Damping Ratio, Fraction of Critical Damping(감쇠율), Equivalent Viscous Damping(등가 점성 감쇠), Damped Natural Frequency(감쇠 고유 진동수), Oil Damper(오일 감쇠기), Oil-Ball Damper(오일볼 감쇠기).

ABSTRACT

The damping characterisitcs of a cylindrical structure containing oil and bearing balls is investigated for external bending forces. The experimental data obtained through the use of bearing balls with viscous oil in a column is given and analyzed. The viscous action of the oil and inertia effects of the balls on the inside of column create a drag force. The drag force dampens the vibration of the column.

This study aims to search for an optimum combination of oil and balls which would produce maximum damping. Machining oils of various viscosities along with ball bearings of various sizes place inside cantilevered aluminium tubes of various diameters to create a rig on which the damping properties of the oil and balls can be studied. The cantileved tubes are studied in both horizontal and vertical positions in order to gauge the effect of gravity on the system.

The actions of the ball in the column and damping characteristics are investigated according to the dimensionless terms. The Buckingham theorem is used to reduce the variables and to predict the damping of an oil ball column. Though the damping ratio remains fairly constant in the horizontal position of column, the damping ratio begins to increase as the ratio of the number of balls and column length rise above 0.28 in the vertical position of oil ball column. The ratio of the ball diameter to column diameter influences the damping ratio with an optimum diameter ratio. Slenderness ratio and gravity effects on the damping ratio are investigated.

요 약

윤활유와 베어링 볼을 내장한 원통형 구조물이 외부로부터의 굽힘 하중을 받을 때의 진동감쇠 특성을 조사하였다. 실험대상인 원통형 구조물의 내부에는 윤활유의 점성력과 베어링 볼의 관성력에 의한 저항력이 발생하며 이 저항력에 의하여 원통형 구조물의 진동이 감쇠된다.

*Member, Taejon National University of Technology

**Korea Institute of Nuclear Safety

윤활유와 베어링 볼의 감쇠력이 최대가 되는 최적 혼합체를 찾기 위하여 점도가 다른 윤활유와 재질 및 직경이 다른 베어링 볼을 알루미늄 관에 넣고 감쇠력을 측정하였다. 실험 결과를 무차원 변수를 사용하여 분석하였으며 수평 원통과 수직 원통의 감쇠율을 비교하였다. 수평 원통의 경우에 감쇠율의 변화가 거의 없었으나 수직 원통의 경우 원통 길이에 대한 베어링 볼 직경의 비율이 약 0.28 이상부터 감쇠율이 증가하기 시작하였다.

볼의 직경과 원통 직경 비에 대한 감쇠율의 변화와 세장비 및 중력 효과에 대한 감쇠율의 영향도 조사하였다.

Nomenclature

c_c : Critical damping coefficient, $c_c = 2\sqrt{km} = 2m\omega_n$
c, c_1, c_2 : Damping coefficients
C_D : Drag coefficient
D_B : Ball diameter
D_C : Cylinder diameter
F_d : Drag force between oil ball mixture and piston
F_r : Froude number, $Fr = v/\sqrt{gh}$
g : Gravitational acceleration
$|G_{ff}|$, $|G_{vv}|$, $|G_{aa}|$: Auto spectral density function of the force, velocity and acceleration, respectively.
h : Oil ball mixture height
l : Length of the cantilevered cylinder
Re : Reynolds number, $Re = \rho_0 v D_B / \mu$
RDF : Ratio of oil ball weight (gravity force-buoyancy force) to the oil viscous force, defined in Eq. (3)
St : Strouhal number, $St = v/(\omega D_B) = y(t)/D_B$
$v(t)$, $y(t)$: Vibrational velocity and displacement of the cylinder
$\Delta\rho$: Density difference between ball and oil, $= \rho_B - \rho_O$
ζ : Damping ratio, fraction of critical damping, $= c/c_c$
μ : Dynamic viscosity of oil
ν : Kinematic viscosity of oil, $= \mu/\rho_O$
ρ_B : Ball density
ρ_O : Oil density
ω : Angular frequency, $= 2\pi f$ [rad/sec]
ω_d : Damped natural frequency, $= \omega_n\sqrt{1-\zeta^2}$
ω_n : Natural frequency of the oil ball damper

1. Introduction

Many vibration absorbers use a viscous damping mechanism to reduce vibration amplitudes. The literature related to viscous dampers is extensive, however, no investigators have studied the oil ball damper examined here. Generally, the related work focussed on a particular aspect of viscous or structural damping in a fluid, porous or a granular type material.

Asami[1] presented the dynamic characteristics of an oil damper and applied these results to the variable oil damper, where the damping coefficient is regulated continuously by using a variable damping mechanism. In the fluid damper, air or gas can be used to create a frictional damping force instead of oil and there are many investigations and applications of pneumatic dampers[2,3,4]. Granular type damping material has been used to give a large reduction of impact force, noise and resonant displacement[5,6]. A combined type of damping treatment was suggested by using porous material to increase the oil viscosity[7].

Among the various damping treatments using visous fluid, granular damping materials, etc., the choice of a damping treatment for a particular machine component depends on many factors such as its effectiveness, the cost and the hospitality of the surrounding environment. For practicality and effectiveness, a damping mechanism should provide a required level of damping, and preferably have easily adjustable characteristics. Fluid dampers and granular type damping treatnments have advantages of high damping values but the magnitude of damping cannot be changed in a simple manner. This study aims to search for an optimum combination of oil and balls which would produce maximum damping. Machining oils of various viscosities along with ball bearings of various sizes place inside cantilevered aluminium tubes of various diameters to create a rig on which the damping properties of the oil and balls can be studied. The cantilevered tubes are studied in both horizontal and vertical positions in order to gauge the effect of gravity on the system.

The viscous action of the oil and inertia effects of the balls on the inside of column create a drag force. The drag force dampens the vibration of the column. The Buckingham theorem is used to reduce the variables and to predict the damping of an oil ball column.

2. Theoretical Modelling

The oil ball damper exhibits a viscous force just as the case of the oil damper, the damping force is proportional to the vibrational velocity. The velocity profile for an oil damper can be determined by its geometry. It is very difficult, however, for the case of an oil ball damper to calculate either the velocity profile or the amount of damping by analytic methods because of the complex geometry that exists. In this motion, some disturbances are generated by unsteadiness in the flow or fluctuation of the solid boundaries. Fig. 1 shows the simplified model of a column with oil and balls.

2.1 Reduction of Variables

The damping ratio of an oil ball damper is dependent upon many parameters, for example, oil ball mixture hight, oil viscosity, ball diameter, ball material density, inner diameter of the column, gap between the ball and column, gravity, velocity of vibration and frequency of vibration, etc. To derive a certain functional relationships without recourse to complex theory, the classical technique, dimensional analysis, was applied to the experiement.

Using the Buckingham, π theorem[8] the damping ratio ζ of the damper can be found as a function of the following dimensionless terms.

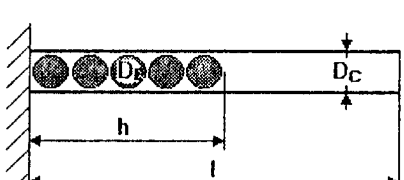

Fig. 1 Simplified model of a column containing oil and ball shot.

$$\zeta = f\left(\frac{\mu}{\rho_O v D_B},\ \frac{F_d}{\rho_O v^2 D_B^2},\ \frac{\sqrt{gh}}{v},\ \frac{\omega_n D_B}{v},\ \frac{\Delta\rho}{\rho_o},\ \frac{D_B}{D_C},\ \frac{D_C}{l},\ \frac{h}{l}\right) \tag{1}$$

where the first term is known as the inverse of the Reynolds number, Re, the second term is the drag coefficient, C_D, the third is the inverse of the Froude number, Fr, the fourth is the inverse of the Strouhal number, St, and is a ratio of ball diameter to transverse movement of column, and the remaining terms are an oil ball density ratio, and length ratios in that order. These dimensionless terms can be divided into two groups, force related terms and material related terms. The familiar Reynolds number of fluid dynamics, given by $Re = \rho_O v D_B / \mu$, for a sphere moving steadily through a viscous fluid, expresses the ratio of an inertia force to a viscous force. The Froude number, $Fr = v/\sqrt{gh}$, is the ratio of a convective inertia force to gravitational force.

The Strouhal number, given by $St = v/(\omega D_B)$, characterizes the relative importance of purely time -dependent inertia terms with respect to convective terms, however, in this case it is a ratio of cylinder movement to ball diameter, $y(t)/D_B$.

Form experiments, the Reynolds number was found to be less than 1 for movement of the column containing the oil ball mixture[9]. Thus, the inertia forces are much less than the visous forces. Also, it is noted that greater damping exists for a greater height of oil ball mixture. Thus gravity forces are deemed to be important. The ratio of Reynolds number to Froude number squared is

$$\frac{Re}{Fr^2} = \frac{\rho_O v D_B}{\mu}\frac{gh}{v^2} = \frac{\rho_O g D_B h}{\mu v} = \frac{\rho_O g}{\mu v / D_B h} \tag{2}$$

Equation (2) is a ratio of gravity force to viscous force. By using the density difference between ball and oil, $\rho_B - \rho_o$, insted of the oil density, ρ_o, a dimensionless term, RDF, is obtained

$$RDF = \frac{(\rho_B - \rho_o)g}{\mu v / D_B h} = \frac{\Delta\rho g}{\mu v / D_B h} = \frac{\text{gravity force} - \text{buoyancy force}}{\text{viscous force}} \tag{3}$$

The RDF number is a ratio of oil ball weight (gravity force − buoyancy force) to the oil viscous force.

2.2 Calculation of the Damping Coefficient

The damping coefficient, c, can be calculated from experimental results by two different methods. The first method uses the measured auto spectral density functions of force and velocity and the second method consists of multiplying the measured damping ratio, ζ, by the critical damping value, c_c. Damping coefficient c_1 is calculated from[10]

$$c_1 = \left|\frac{G_{ff}}{G_{vv}}\right| = \left|\frac{G_{ff}}{G_{aa}/\omega^2}\right| \tag{4}$$

where G_{ff} is the auto spectral density function of the force, G_{vv} is the auto spectral density function of the velocity, and G_{aa} is the auto spectral density function of the acceleration. The critical damping value is $c_c = 2\sqrt{km} = 2m\omega_n$ and the damping coefficient c_2 is

$$c_2 = \zeta c_c = 2\zeta\sqrt{km} = 2\zeta m\omega_n \tag{5}$$

3. Experiments

3.1 Experimental Rig

The experimental set-ups involving the aluminium tubes are shown in Fig. 2. Prior to carrying out tests on the tubes, the diameters of all the balls to be used were measured and also the viscosities of four different oils were measured using a viscometer. The density of the oils were measured as well by weighing a known volume of oil. Table 1 shows the viscosity and density values of these oils.

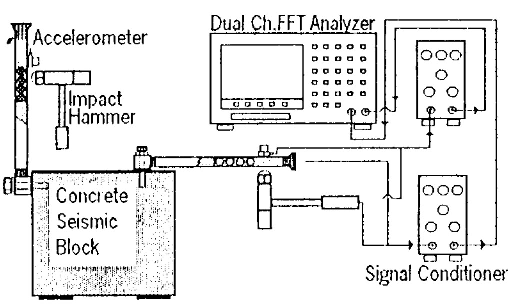

Fig. 2 Schematic diagram of experimental rig.

Table 1 Measured oil viscosity

Oil	Density (Kg/m^3)	Viscosity (centi Stokes, 10^{-6} Ns/m)
1	852.7	59
2	868	122
3	901.6	156.5
4	879.2	467.5

Two different sizes of tube with rubber stoppers on the end (8mm I.D., 25.7mm I.D.) were taken and the impact tests were performed on them to find the amount of damping present in each. Placing the accelerometer at 0.81, which is at the nodal point of the second mode of vibration, eliminates the effects of the second mode.

All experiments were undertaken using this concrete block which was resiliently mounted on a concrete base block. By taking such precautions, it was possible to measure low levels of beam structural damping, which were subtracted from damping values of the oil ball damper.

The outputs from transducers were amplified by signal conditioning amplifiers. The signals from which were analyzed by a dual channel FFT analyzer to estimate the frequency response function between force and acceleration.

3.2 Measurement of the Damping Ratio

Prior to measuring the damping ratio from the accelerance function, the frequencies of bending modes of the cantilever beam were measured and compared with classical beam theory. The percentage differences between the measured and predicted values of the frequency were within $\pm 5\%$ and it was shown that the tube and seismic block behaved as if they were rigidly terminated, at least in the modes of vibration considered here. From a baseband measurement with an analysis bandwidth of 1 kHz, the measured accelerance function was curve fitted for three modes using a single degree of freedom (SDOF) and a multi degree of freedom (MDOF) polynomial curve fit.

The resulting damping values (percentages of critical damping) from SDOF and MDOF fits had very similar values so that this experimental rig contained almost no coupling between the modes. As the value of damping ratio of the first mode was more than 4 to 10 times that of the second and third modes respectively, only the first mode was considered for analysis. The first modal frequency of the different tubes analyzed ranged between 10 and 50 Hz. Thus the baseband measurement of a frequency span of 50Hz, without zooming, ensured sufficient frequency lines within the 3dB bandwidth of the resonance, and the auto spectral density functions of the force and the acceleration could be used for calculating a damping coefficient of good quality.

By varying the height of the oil ball mixture, ball size, ball diameter, oil density, and cylinder inner diameter, each damping ratio was calculated from a SDOF curve fit from 16 averaged raw accelerance functions. The damping force is the sum of the forces due to the oil ball mixture and that due to the structural damping of aluminium tube. The damping due to the latter force was subtracted from the results of the measurements.

4. Damping of the Oil Ball Column

4.1 Damping of the Oil Ball Damper

It can be seen from Fig. 3 and Fig. 4 that the effect of the ratio h/l is minimal in the horizontal position. It should be noted that in the horizontal position just filling the tube with oil alone is enough to produce a damping ratio of approximately 0.58%, while adding enough copper ball bearings to give an h/l value of approximately 0.5, only raises the damping ratio to approx. 0.65%. However, when the tube was tested in the vertical position, Fig. 3 shows that the damping ratio is strongly influenced by the ratio h/l.

Fig. 3 also shows that for an h/l ratio of less than approximately 0.28 the damping ratio remains reasonably constant. Above an h/l value of approximately 0.3 the curve begins to rise sharply.

It should be noted that by putting the tube in the vertical position and just filling it with oil produced a damping ratio of 0.95%, indicating that gravity has a significant effect on the damping ratio in the vertical position. The damping ratio is small below an h/l value of 0.28, probably because at the fixed end of the tube, the tube displacement is very small, therefore there is not much opportunity for shearing of the oil and hence the damping is small.

As the ratio increases beyond 0.28 the tube displacement increases and therefore more shearing of the oil can take place thereby increasing the damping ratio. For Figures 3 and 4, the experiments were carried out with the tube full of oil, regardless of the number of balls in the tube. It was than decided to study the effect of only putting enough oil in the tube to just cover the top of the balls. This effect was only studied with the tube in the vertical position

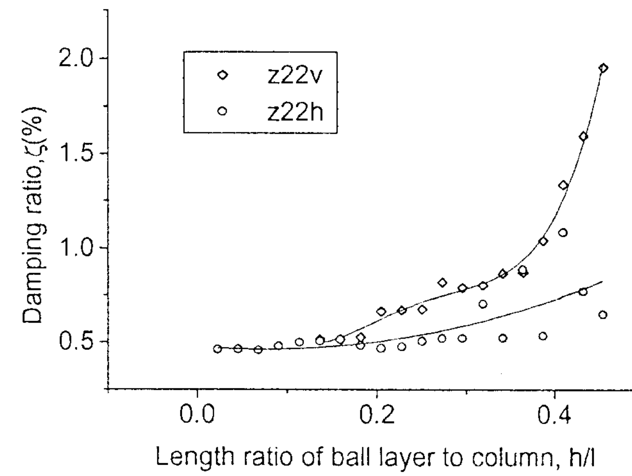

Fig. 3 Experimentally determined damping ratios ζ % *vs* length ratio of ball layer to column h/l; 22.23mm steel ball in 25.4mm inner diameter column.

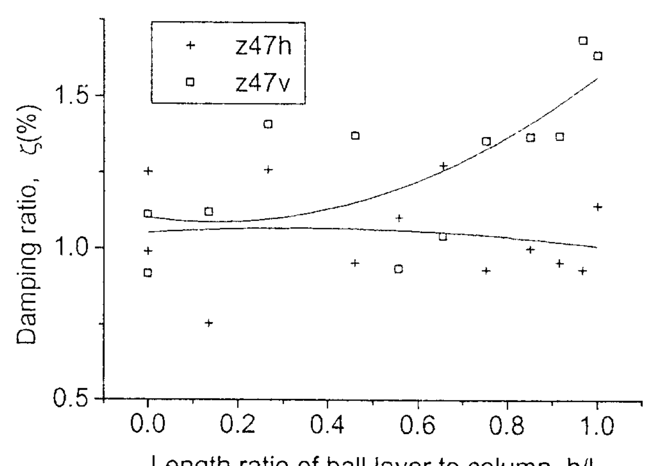

Fig. 4 Experimentally determined damping ratios ζ % *vs* length ratio of ball layer to column h/l; 3.1mm steel ball in 4.7mm inner diameter column.

and the results are shown in Fig. 5, which shows that the damping ratio is substantially higher when the tube is full of oil.

4.2 Influence of Diameter Ratio

In order to examine the effect of the ratio, D_B/D_C, four different ball sizes were used with the tube in the vertical position and an h/l ratio of 0.3. The oil viscosity used was 156.5 centi Stokes. The tube was placed in the vertical position because it was thought that this would show more clearly the effect of D_B/D_C than the horizontal position. The ball sizes used were : 25.4mm, 22.23mm, 18.0mm, and 3.1mm.

It can be seen from Fig. 5 that the optimum ball dia. to tube dia. ratio is approximately 0.54. This result is supported by Fig. 6 which shows the curves for different ball sizes with diameter ratios ranging from 0.99 to 0.70. The diameter ratio of 0.54 probably gives the highest damping ratio because it allows the greatest amount of relative motion between the tube and the ball thus creating the opportunity for a large amount of shearing to take place between the oil and balls, and, the oil and tube wall. When the diameter ratio drops below 0.5 there would be a tendency for the system to lock up and therefore very little shearing of oil could take place. As the diameter ratio increases above 0.54 there is less relative motion between the tube and the balls because there is less clearance and therefore there is less shearing of the oil, leading to lower damping ratios.

4.3 Slenderness Ratio Effect

It should be noted that the curves in Fig. 4 showing the damping ratios produced with the tube in the vertical position show no upward trend as was the case in Fig. 3 or Fig. 6. A possible explanation for this phenomena may be that a gravity effect is only significant for certain slenderness ratios. Where the slenderness ratio is defined as of tube length to tube diameter, D_B/l. Therefore it may be tentatively proposed that slender tubes will not show a gravity effect and therefore placing them in a vertical position will not enhance the damping properties of the oil, ball and tube system.

4.4 Gravity Effect

As gravity seems to play an important part in damping when the tube is in the vertical position, it was included in the dimensional analysis. All figures show a strong dependance on the parameter h, and diameter ratio of the ball and tube. The effect that gravity has on the system may be related to the fact that when the tube is vertical and full of balls, the force acting on the balls towards the bottom of the tube is much greater than when the tube is vertical and full of balls, the force acting on the balls towards the bottom of the tube is much greater than

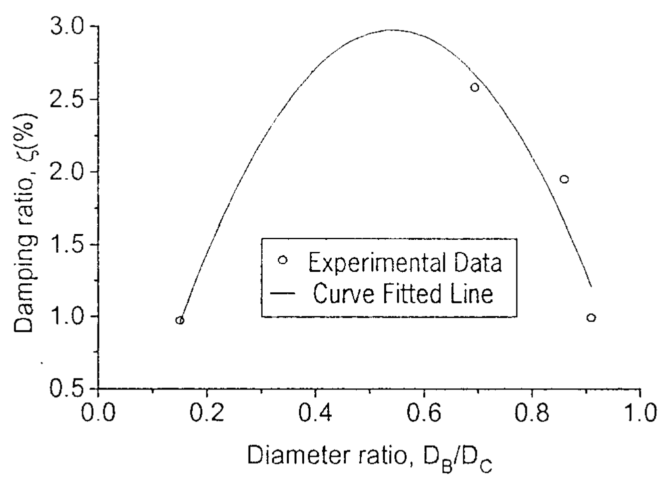

Fig. 5 Dynamic effect on damping ratio

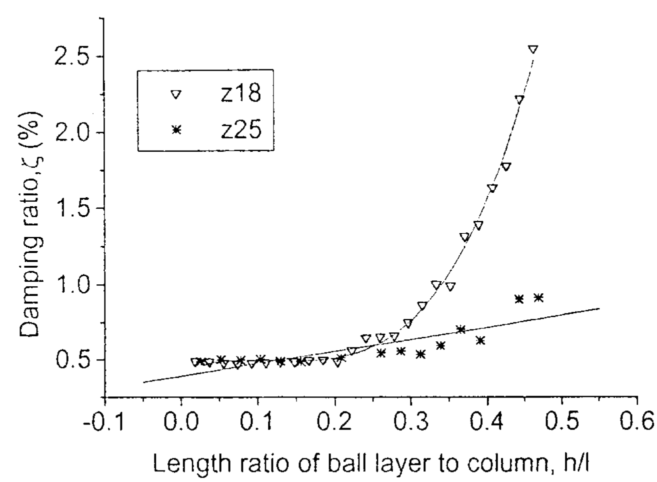

Fig. 6 Experimentally determined damping ratios ζ % *vs* length ratio of ball layer to column h/l ; 18mm steel ball and 25.4mm copper ball in 25.4mm inner diameter column.

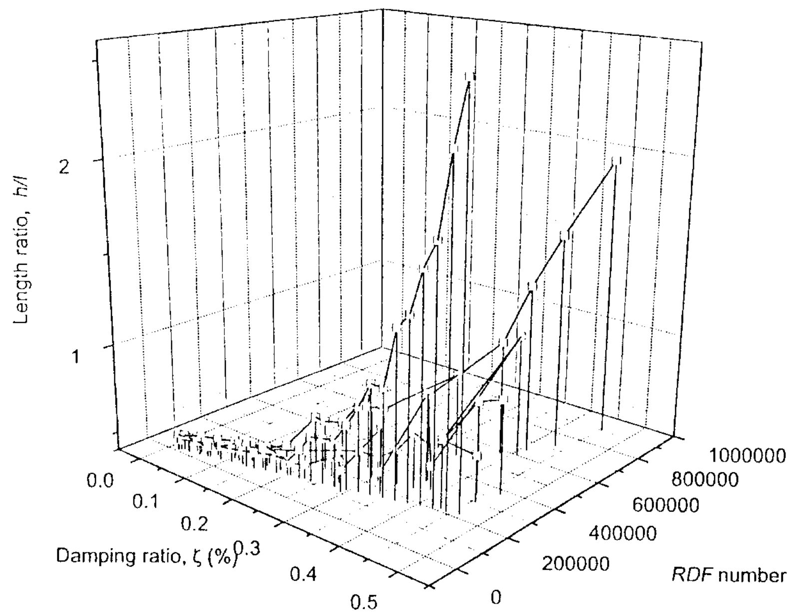

Fig. 7 Experimentally determined damping ratios ζ % *vs* *RDF* number with length ratio h/l.

when the tube is horizontal. Therefore the force between adjacent balls is large and this produces a large shear stress as the balls move relative to one another, thereby producing a large damping ratio.

4.5 Damping with RDF Number

The damping ratio relation to the density ratio is investigated by using the combined forms, i.e., *RDF* number. This dimensionless variable was formed to present the damping of an oil ball damper by dimensionless terms which possess physical meaning. The *RDF* number is defined by equation (3) and is a ratio of Reynolds number to Froude number multiplied by the density ratio of ball to oil. Shown in Fig. 7 is the experimentally determined damping ratios versus the *RDF* number and length ratio. Though the amount of variance is too large to obtain a linear dependence, it can be used to investigate the damping characteristics of an oil damper. It is shown from this figure that the damping ratio of the oil ball damper has a dependence on *RDF* number. As the *RDF* number has a physical interpretation of the ratio of gravity force to viscous force, the higher the oil viscosity the smaller the *RDF* number and the larger the damping ratio, the larger the ball diameter the larger the *RDF* number and the smaller the damping ratio as shown in this figure.

5. Concluding Remarks

It was found that when the tube was placed in the horizontal position the damping ratio remained fairly constant. However, when the tube was placed in the vertical position it was found that the damping ratio began to increase as the h/l ratio rose above 0.28. The ratio of the ball dia. to tube dia. was also found to influence the damping ratio with an optimum diameter ratio between 0.55 and 0.6. For tubes with a slenderness ratio greater than 70 (i.e. $l/D_C >$ 70), no increase in damping was observed when they were placed in the verical position. Because the damping ratio increased for tubes with a slenderness ratio less than 40 in the vertical position, gravity must play an important part in determining the damping ratio.

As one of the objectives of this report was to find an optimum combination of oil and balls to give maximum damping we would recommend that a tube with a slenderness ratio less than or equal to 40 be used, a ball dia. to tube dia. ratio between 0.55 and 0.6 be used, the tube should be in the vertical position, an h/l ratio equal to approximately 0.6 could be used to reduce the cost of ball bearings, a light oil with a viscosity of approximately 156 c.S. could be used, and the tube should be full of oil.

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References

(1) Asami, T. and Sekiguchi, H., 1988, "Study on an Oil Damper with a Variable Damping Mechanism," *JSME International Journal Serices* Ⅲ, Vol. 31, No. 2, pp. 370～379.

(2) Hilyard, N.C., 1974, "The Influence of Pneumatic Damping on the Performance of a Flexible Foam Vibration Isolator," *Journal of Sound and Vibration* 32, pp. 71～76.

(3) Fox, M.J.H. and Whitton, P.N., 1980. "The Damping of Structural Vibration by Thin Gas Films", *Journal of Sound and Vibration* 73(2), pp. 279～295.

(4) Chow, L.C. and Pinnington, R.J., 1987, "Practical Industrial Method of Increasing Structural Damping in Machinery," *Journal of Sound and Vibration* 118(1), pp. 123～139.

(5) Shen, R.Y., Chow, L.C. and Pinnington, R.J., 1988. "On the Prediction of Dynamic Characteristics of a Long Sand-Filled Structure," *Proceedings of Inter-Noise 88 at Avignon, France* Vol. 1, The Source of Noise, pp. 543～546.

(6) Popplewell, N. and Semercigil, S.E., 1989, "Performance of the Bean Bag Impact Damper for a Sinusoidal External Force," *Journal of Sound and Vibration* 133, pp. 193～223.

(7) Chow, L.C. and Pinnington, R.J., 1986. "On the Prediction of Oil Layer Damping on Plates," *ISVR Technical Report* No. 137, *University of Southampton, England*.

(8) Imrie, B.W., 1968. "The Dimensionless Efficiency of Teaching Dimensional Analysis," *Bull. Mech. Engng. Educ.*, Vol. 7, pp. 227～235.

(9) Song, S.H. and Koss, L.L., 1993, "Dynamic Characteristics of an Oil Ball Damper", *Inter. Journ. of Analytical and Experimental Modal Analysis*, Vol. 8, No. 4, pp. 265～285.

(10) Bendat, J.S. and Piersol, A.G., 1981, Engineering Applications of Correlation and Spectral Analysis. *N.Y. : John Wiley & Sons, Inc.*