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PRINCIPLES AND CHARACTERISTICS OF VISCOUS DAMPING DEVICES (GYRO-DAMPER), THE DAMPING FORCES WHICH ARE HIGHLY AMPLIFIED BY CONVERTING THE AXIAL MOVEMENT TO ROTARY ONE

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SUMMARY

Viscous devices (Gyro-dampers) , the damping forces of the viscous material are highly amplified by converting the axial movement to a rotary one with a ball screw nut, namely the velocity working to the viscous material is much amplified and in the course of re-conversion to axial direction the damping forces are further amplified, have been newly developed. The gyro-damper devices are compact and can use silicone which is less temperature dependent. The principles and characteristics are theoretically explained taking the non-linear characteristics of the viscous material, lead angle, inertia and friction factors into consideration. Tests have been executed and the test results show the correctness of the theory.

INTRODUCTION

There are four ways to gain increased viscous damping force; (1) raise the viscosity of viscous material ; (2) enlarge the area containing the viscous material ; (3) increase the velocity applied to the viscous material ; (4) amplify the damping force of the viscous material. The first method (increase the viscosity of viscous material) is limited by the material's viscous capacity. Furthermore, as the material's viscosity increases, temperature dependency of the viscous damping force is intensified. The left of Figure 1 shows the temperature dependency of 90,000 poise polyisobutylene which we can obtain as the highest viscosity. 90,000 poise at 30ºC is shown to be 219,000 and 534,000 poise at 20ºC and 10ºC respectively, which may require heat-resisting material. The

Second method (enlarge the area containing the viscous material) involves excessive device cost and requires larger installation space for the device, which may hinder efficient use of the space. The right of Figure 1 shows the relationship between damping force and velocity of the above mentioned polyisobutylene whose shear area is 1 m^2 and clearance is 0.5mm at 20ºC.As the velocity increases, the damping coefficient decreases. This tendency is intensified in case of higher viscosity. In this case if the relative velocity is 10kine, ordinary relative velocity of buildings, the damping force is 7.5ton while the material has 12.5ton at 50 kine. Therefore if the gained velocity can be increased, it is possible to use higher damping capacity of polyisobutylene at high velocity. The authors adopted the function of screw and nut as a method to increase velocity and to satisfy (4)amplify the damping force at the viscous material. A nut rolls one revolution per pitch (or lead). Therefore the external edge velocity of the nut is increased $\pi D/p$ times (where D is the outer diameter of the nut) and the increased velocity is applied to the viscous material (hereafter $\pi D/p$ is called amplification factor: φ). The gained damping force at the viscous portion is further φ times amplified in the course of re-conversion to axial direction. For effective function of this device, it is necessary to minimize the friction between the screw and nut to convert linear motion to rotary one. Although lubricant can be used to lessen the friction, the authors adopted a ball bearing screw which uses ball bearings to gain stable performance. Ball bearings are also used to minimize the friction between rotary nut and fixed portion (it is called support bearing or thrust bearing). The left of Figure 2 shows the concept of disk type whose diameter of nut is enlarged to make the velocity faster. The right of Figure 2 shows the concept of cylinder type which has a tube whose diameter is limited for the purpose of installing

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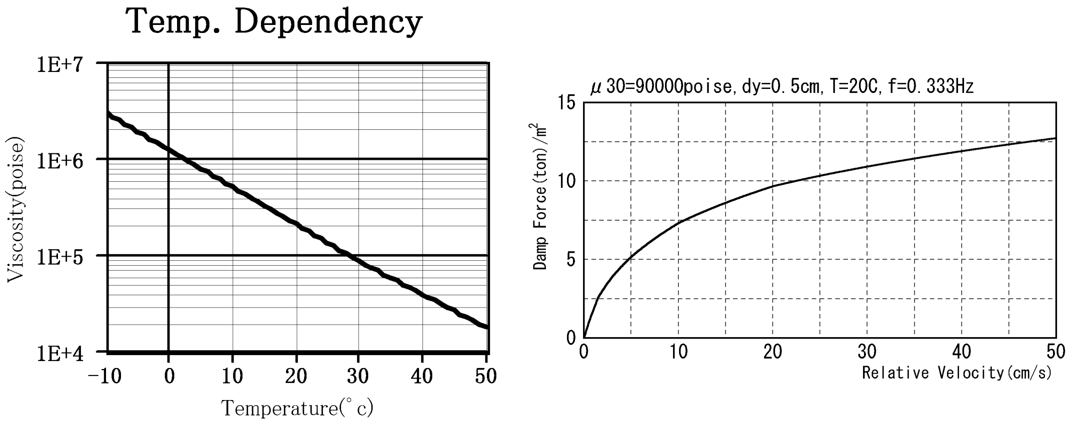
inside walls. These devices do not require high viscosity but can use silicone, which is less temperature dependent.



olyisobutylene

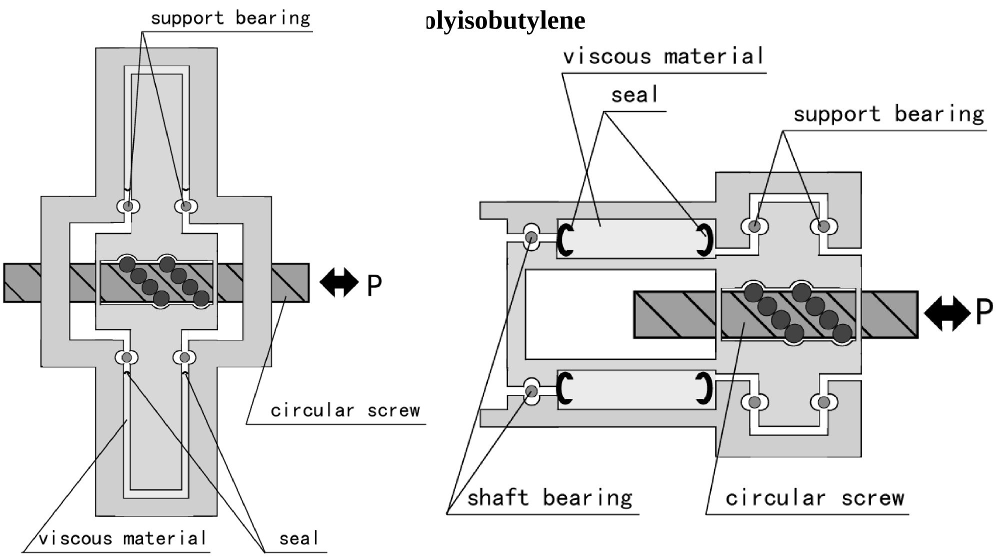


Figure 2. Concept of Gyro-damper (Disk type: left, Cylinder type: right)

THEORETICAL FORMULA

Resisting force of Gyro-damper consist of (1) viscous damping force by viscous material *Qd* , (2) inertia developed by rotary movement of rotor *Ip* and (3) friction force *F* of ball bearing screw *Qf1*, support bearing *Qf2*, seal *Qf3* and shaft bearing *Qf4*, as shown in equation (1) and Figure 3 .

$$Q = Qd + Ip + F \quad (F = Qf1 + Qf2 + Qf3 + Qf4) \tag{1}$$

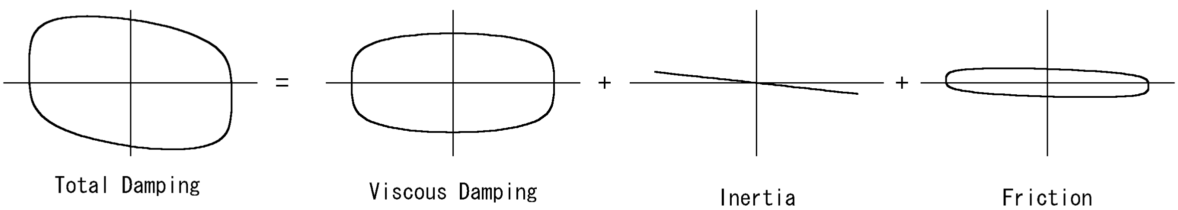


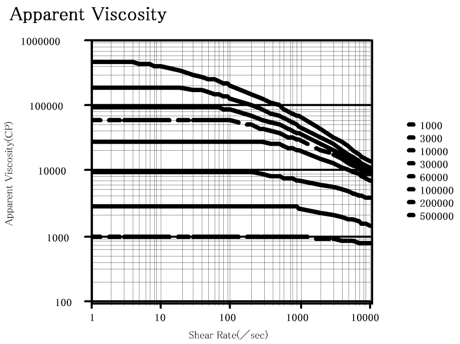
Figure 3. Resisting force of Gyro-damper

Viscous damping force

Axial velocity V_0 is increased $\pi D/p$ times (where D is diameter of rotary body in case of cylinder type and 2/3 of disk diameter in case of disk type). Therefore the shear rate Vs of viscous material is $V_0 \pi Dp.dy = \varphi V_0/dy$ (where dy is the clearance of the viscous material). When viscosity is low or shear rate is small, viscous resistance of viscous material at the given shear rate is obtained by multiplying the viscosity by the shear rate. However when the shear rate increases, the viscous resistance becomes lower than the multiplied value of the viscosity and shear rate. The drops of viscous resistance according to the increase of the shear rate are treated as a drop of the viscosity itself, (apparent viscosity = $\mu(Vs)$) .The portion of viscosity's dropping is assumed as quadratic curve ($Log\mu(Vs) = a + b\ LogVs + c\ (LogVs)^2$) provided that the viscosity at 1 sec^{-1} is base viscosity and the value at 10^3 sec-1 and 10^4 sec^{-1} is the actual value obtained at experiment. Therefore, the shear rate is the value of 1 sec^{-1} or $10^{(-b/c)}$, which is larger. Figure 4 shows the relationship between apparent viscosity and shear rare of representative silicone (from 1,000cst to 500,000cst at 25ºC) and the value of a, b and c. The apparent viscosity $\mu(Vs)$ at a shear rate can be obtained by using this equation. The viscous damping force at the viscous material is $\mu(Vs) \cdot Vs \cdot A$ and it is increased φ times in the course of re-conversion . Therefore the viscous resistance Qd is

$$Qd = \phi \cdot \mu(Vs) \cdot Vs \cdot A \tag{2}$$

equation (2).



Viscosity	a	b	c
1,000	3.0000	0.0735	-0.0245
3,000	3.4771	0.1818	-0.0635
10,000	4.0000	0.1206	-0.0550
30,000	4.4771	0.2408	-0.0980
60,000	4.7782	0.2209	-0.1067
100,000	5.0000	0.1921	-0.1098
2,000,000	5.3010	0.1478	-0.1150
5,000,000	5.6990	0.0155	-0.1003

Figure 4. Apparent viscosity of silicone

Temperature dependency of viscous material, silicone

Basic viscosity of silicone is the value at 25ºC, μ_{25}. The viscosity μt at temperature, t , is calculated by equation

$$\mu_t = \mu_{25} \times 1.02^{(25-t)} \tag{3}$$

(3) .

Inertia

The angle acceleration of rotary body is $2\pi \cdot Aa/p$ (where Aa is axial acceleration). Therefore the torque is $2\pi \cdot I \cdot Aa/p$(where I is inertia moment). The axial resistance by this torque, Ip, is $(2\pi/p)^2 \cdot I \cdot Aa$. In case of cylinder type Ip is calculated by equation (4), where W , D, and d is the self-weight, outer-diameter and inner-diameter of

$$Ip = W\left(D^2 + d^2\right) \cdot \left(2\pi / p\right)^2 \cdot Aa / 8g \tag{4}$$

the rotor, and g is gravity acceleration. In case of disk type d =0.

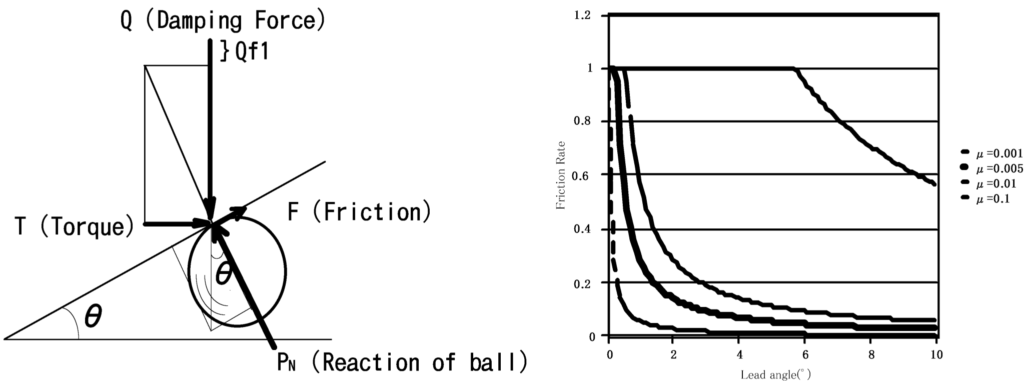
Friction force

Friction of ball bearing screw

The friction force on helix , *F1*, is $Q{\cdot}\mu 1{\cdot}\cos\theta$ where *Q* is total damping force , $\mu 1$ is friction factor, and θ is lead angle($\tan\theta = p/\pi d_0$, d_0 :diameter of screw) , as shown on the left of Figure 5. As the axial force due to this friction, *Qf1*, is $F1/\sin\theta$, equation (5) is gained.

$$Qf1 = F \,/\, \sin\theta = Q\cdot\mu 1\cdot\cos\theta = Q\cdot\mu 1 \,/\, \tan\theta \tag{5}$$

The right of Figure 5 shows the relationship between friction rate,$\mu 1/\tan\theta$, and lead angle, θ,depend on friction factor. As the friction factor of ball bearing screw is 0.005, the lead angle should be at least more than 3 degrees for the smooth conversion from axial movement to rotary one.



Friction & lead angle **Friction rate & lead angle**

Figure 5. Ball bearing screw

Friction of support bearing

As the support bearing stands under axial force, *Q*, the torque is $Q{\cdot}\mu 2{\cdot}R1$ where $\mu 2$ is friction factor and *R1* is radius of support bearing. The axial force due to this friction, *Qf2*, is Equation (6).

$$Qf2 = Q\cdot\mu 2\cdot 2\pi R1 \,/\, p = Q\cdot\mu 2\cdot\phi 1 \tag{6}$$

Friction of seal

Friction force of seal at seal portion is $f{\cdot}L$ where *f* is friction force per unit and *L* is the total length of seal. The axial force due to this friction ,*Qf3*, is $f{\cdot}L{\cdot}\varphi 2$,where$\varphi 2$ is amplification factor, = $2\pi R2/p$ (*R2* is the distance of seal from the axis of screw thread), as shown in Equation (7).

$$Qf3 = f\cdot L\cdot\phi 2 \tag{7}$$

Friction of shaft bearing

Friction force of shaft bearing is $P{\cdot}\mu 3$ where *P* is the force working to shaft bearing and $\mu 3$ is friction factor of shaft bearing. The axial force due to this friction , *Qf4*, is $P{\cdot}\mu 3{\cdot}\varphi 3$,where $\varphi 3$ is amplification factor, = $2\pi R3/p$(*R3* is the distance of shaft bearing from the axis of screw thread), as shown in Equation (8).

$$Qf4 = P\cdot\mu 3\cdot\phi 3 \tag{8}$$

Total resisting force *Q* is obtained by placing the equation (2),(4),(5),(6),(7) and (8) into the equation (1), as shown in Equation (9).

$$Q = \frac{\phi \cdot \mu(Vs) \cdot Vs \cdot A + W\left(D^2 + d^2\right) \cdot (2\pi / p)^2 \cdot Aa / 8g + f \cdot L \cdot \phi 2 + P \cdot \mu 3 \cdot \phi 3}{\left(1 - \mu 1 / \tan\theta - \mu 2 \cdot \phi 1\right)} \quad (9)$$

In case the force due to inertia can be neglected or the influence can be taken into the consideration of the structural stiffness, Equation (9) can be expressed as Equation (10), where α is $1/(1 - \mu 1/\tan\theta - \mu 2 \cdot \varphi 1)$, and C is the friction value due to seal and shaft bearings. The value of C shall be defined through experiment. The value of $\mu 1$ and $\mu 2$ is usually 0.005.

$$Q = \alpha \cdot \mu(Vs) \cdot Vs \cdot A + C \quad (10)$$

STUDIES

Four devices (three are cylinder type and one is disk type, D-14) are tested. One,SL-9, among three cylinder type is used for long stroke (±300mm) application as damping device of base –isolation. The rest, S-2 and S-150, is for short stroke. Design strength of D-14, SL-9, S-2 and S-150 is 14 ton, 9 ton, 2ton and 150ton respectively. In case of S-2 and D-14 , two different types of viscosity materials are tested (5,000 and 10,000cst are for D-14 and 10,000 and 100,000cst for S-2). The specifications of four devices are shown on Table 1. The loading was done by using three dynamic actuators(5, 30 and 300ton) depending on the strength of each device with 5 cycles of stationary waves at each amplitude. The test temperature is room temperature except S-150test. For S-150 , sine wave test is performed under room temperature and artificially raised temperatures up to 38.3ºC. From each test, the average value between +load and –load (when hysteresis loop crosses over 0 deformation) at second cycle are considered to be damping force at the velocity of 2πf. The theoretical value is calculated by using Equation (10) and Equation (3), the viscosity of each viscous material under the test temperature. The relationship between damping force and velocity in the test data and theoretical values are indicated in Figure 6 for D-14, Figure 7 for S-2,Figure 7 for SL-9 and S-150 respectively. The theoretical value of D-14 , shown in Figure 6, is calculated by increasing the α value, because the viscosity of viscous material is increased by the pressure due to centrifugal force of the material.

Table 1. Specification of Specimen

SPECIFICATION of SPECIMEN						
TYPE	D14		SL-9	S-2		S150
STRENGTH	14	14	9	2	2	150
Diameter of Screw (mm)	36	36	50	20	20	120
Lead (mm)	10	10	50	20	20	25
Clearance (mm)	3	3	3	2.5	2.5	3
Outer Diameter (mm)	390	390	160	80	80	290
Outer D. of Rotor (mm)	300	300	110	50	50	219
Effective L of Rotor (mm)	-	-	1200	270	270	1250
Viscosity	5,000	10,000	10,000	10,000	100,000	100,000
Lead angle (º)	5.05	5.05	17.67	17.67	17.67	3.80
Amplification F. (φ)	17.44	17.44	6.91	7.85	7.85	27.51
Test Temp. (ºC)	21.3~22.8	21.2~22.1	24.0~27.9	17.0~21.0	20.0~22.0	7.9~38.3
Frequency (Hz)	0.1~2.0	0.1~2.0	0.1~2.0	0.1~2.0	0.1~2.0	0.1~1.0
Amplitude (mm)	1.3~30.3	1.3~2.96	2.3~390.4	1.6~29.2	1.7~29.7	1.0~45.3
Friction factor μ_1&μ_2	0.005	0.005	0.005	0.005	0.005	0.005
Factor α	2.583	2.583	1.053	1.077	1.077	1.191
Friction Force (Ton)	0.34	0.34	0.03	0.06	0.06	5.30

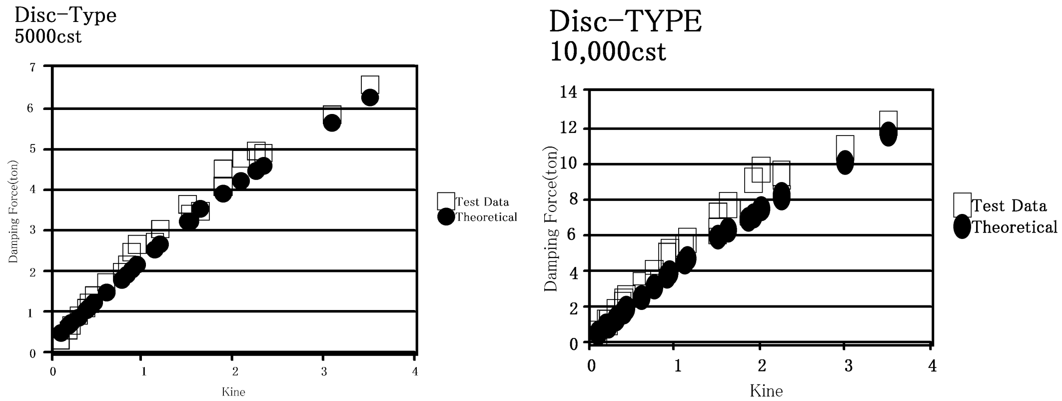


Figure 6. Comparison between test and theoretical value of D-14

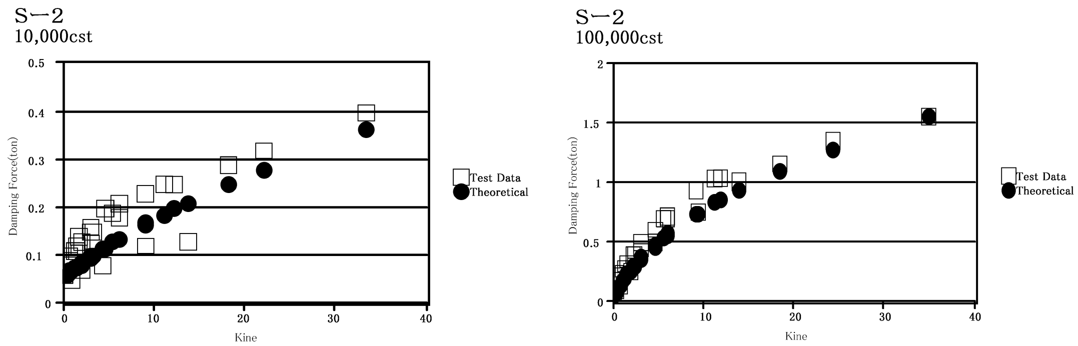


Figure 7. Comparison between test and theoretical value of S-2

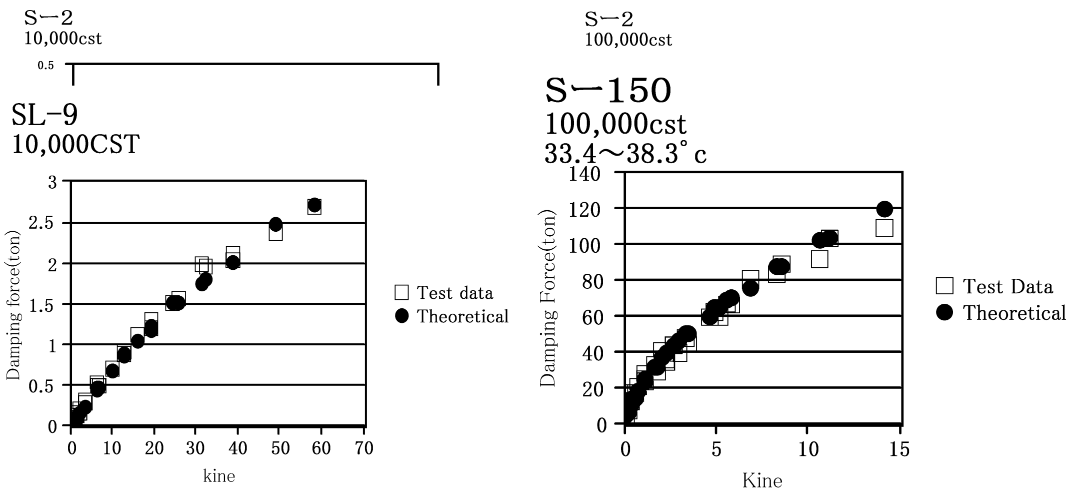


Figure 8. Comparison between test and theoretical value of SL-9 and S-1

CONCLUSIONS

The performance of Gyro-damper can be evaluated by using Equation (10) and the damping force and damping coefficient which engineer wants can be obtained to change the screw pitch, diameter of rotor, area of rotor and viscosity of viscous material. Silicone which is less temperature-dependent can be used. This device is useful not only for large amplitude but also for micro-amplitude. Therefore it is also useful as the device against bending deformation. Although the α value for disk type was taken from the test data, increase of apparent viscosity under pressure shall be examined.

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