

ER Fluid Dampers and Their Application in Shock Mitigation¹

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Abstract. The objective of this paper is to investigate the application of ER fluid dampers for shock protection of shipboard equipment. Several ER fluid dampers have been considered for this purpose. Dynamic models of these ER fluid dampers have been derived and important design parameters have been identified. A preliminary parametric study has been performed to compare the performance of these ER fluid dampers. A simple two degree of freedom system has been used to investigate the effect of ER fluid dampers on shock/vibration mitigation and their size and power requirement.

1. Introduction.

An ER (electro-rheological) fluid damper makes use of the rheological property of the ER fluid to control its damping force to the motion to be damped. The damping force of an ER fluid damper consists of a passive force due to the viscosity of the fluid and an active ER damping force which is controllable by the application of an electric field on the fluid. Damping force due to viscosity is present at all times and is also called the zero-field damping force. A good ER fluid damper is expected to provide a wide range of damping effect, this usually requires it to have a low zero-field damping force and a high controllable damping force due to the ER effect.

In this paper we will investigate the application of ER dampening devices for shock/vibration attenuation via analysis and comparison of several typical ER fluid dampers. In the following section, we first give the dynamic models of these ER fluid dampers. Then a preliminary parametric study is performed in Section 3 to compare the performance of these ER fluid damper. In Section 4, a simple two degree of freedom system is used to investigate the effect of ER fluid dampers on shock/vibration mitigation and their size and power requirement.

2. ER Fluid Dampers

The six ER fluid dampers we are going to investigated are: 1) COTS (commercial-of-the shelf) damper (Figure 1 a)), 2) sliding piston ER damper (Figure 1 b)), 3) sliding cylinder ER damper (Figure 1 c)), 4) concentric cylinder ER damper (Figure 1 d)), 5) rotary disk ER damper (Figure 1 e)), and 6) rotary cylinder ER damper. The construction of a rotary cylinder damper is similar to a concentric cylinder damper.

To model the above ER fluid dampers, we assume that the ER fluid behaves like a Bingham fluid. More specifically, the shear stress τ and the shear rate $\dot{\gamma}$ are related by

$$\tau = \mu\dot{\gamma} + \tau_y(E)\text{sgn}(\dot{\gamma}) \quad (1)$$

where μ is the Newtonian viscosity coefficient of the fluid and $\tau_y(E)$ is the yield stress.

The zero field damping force F_v (or torque T_v) and ER fluid damping force F_{ER} (or torque T_{ER}) of each ER fluid damper are given as follows

1) COTS ER fluid damper

$$F_v = \frac{12\mu A_p^2 L}{bh^3} V, F_{ER} = \frac{2L}{h} A_p \tau_y(E) \quad (2)$$

where A_p is the cross section area of the piston, L the length of the plate, b the width of ER valve plate, h the distance between the two electrodes (gap size), and V the velocity of the piston.

2) Sliding piston ER fluid damper

$$F_v = (\alpha_1 + \alpha_2 + \alpha_3)V, F_{ER} = \tau_y(E)\pi DL \quad (3)$$

$$\alpha_1 = \frac{(\frac{D^2}{4} - \frac{nS^2}{4} + \frac{Dh}{2})(\frac{D^2}{4} - \frac{nS^2}{4})}{\frac{Dh^3}{12\mu L} + \frac{nS^4}{128\mu L}} \quad (4)$$

$$\alpha_2 = \mu \frac{V}{h} \pi DL \left(1 + 16 \frac{h^2}{S^2} \left(\frac{D^2}{nS^2} - 1 \right) \right) \quad (5)$$

$$\alpha_3 = \frac{(D^2 - nS^2 + 2Dh)}{\frac{4Dh^3}{3\mu L} + \frac{nS^4}{8\mu L}} n\pi S^2 \quad (6)$$

where V is piston velocity, L the piston length, D the diameter of the piston, S the diameter of the holes, n the number of the holes, and h the gap size.

3) Sliding cylinder ER fluid damper

It is assumed that all the stationary cylinders have the same thickness d and all the moving cylinders have the same thickness e . The gap sizes between stationary cylinders and moving cylinders are assumed to be equal. Suppose there are N stationary cylinders, there will be N moving cylinders and $2N+1$ sliding surfaces. Let the radius of the central cylinder be r_0 and the gap size be h . Then the radius of the sliding surfaces can be determined using the following equations.

$$r_1 = r_0, r_{2i} = r_{2i-1} + e + 2h, i = 1, \dots, N. \quad (7)$$

$$r_{2i+1} = r_{2i} + d, i = 1, \dots, N, \frac{D}{2} = r_0 + Nd + Ne + 2Nh \quad (8)$$

where r_i is the radius of the i th sliding surface and D is the diameter of the piston.

$$F_v = F_{v1} + F_{v2} \quad (9)$$

$$F_{v1} = \Delta P_v A_p = \frac{12\mu L A_p^2 V}{bh^3} \left(1 + \frac{bh}{2A_p} \right) \quad (10)$$

$$F_{v2} = \mu \frac{VbL}{h} \left(\frac{6A_p}{bh} \left(1 + \frac{bh}{2A_p} \right) + 1 \right) \quad (11)$$

$$F_{ER} = \frac{2\tau_y L}{h} A_p + \tau_y bL \quad (12)$$

¹ This project is supported by Navy Sea Systems Command.

$$b = \sum_{i=1}^{2N} 2\pi r_i (21), A_p = \pi r_1^2 + \sum_{i=1}^N \pi(r_{2i+1}^2 - r_{2i}^2) \quad (13)$$

where L is the length of the piston, h is the gap size, and V is the velocity of the piston.

4) Concentric cylinder ER fluid damper

The thickness of all the cylinders are assumed equal and the gap sizes between all adjoining cylinders are assumed equal as well. There will be n gaps for the ER fluid to flow through the piston if there are n concentric cylinders. Suppose the radius of the central rod is r_0 and the gap sizes between adjoining cylinders are much smaller than the piston diameter, then the radius of the gaps can be calculated from the following formulas:

$$r_1 = r_0, r_i = r_{i-1} + d + h, i = 2, \dots, n. \quad (14)$$

$$\frac{D}{2} = r_0 + n(d + h) \quad (15)$$

where r_i ($i=1, \dots, n$) is the radius of the i th gap, d is the thickness of the cylinder, h is the gap size, and D is the piston diameter.

$$F_v = F_{v1} + F_{v2} \quad (16)$$

$$F_{v1} = \frac{12\mu L A_p^2 V}{bh^3}, F_{v2} = \frac{6\mu A_p V}{bh^2} A_c \quad (17)$$

$$F_{ER} = \frac{2\tau_y L}{h} A_p + \tau_y A_c \quad (18)$$

$$A_p = \pi r_1^2 + \sum_{i=2}^n \pi(r_i^2 - (r_i - d)^2) + \pi\left(\frac{D^2}{4} - \left(\frac{D}{2} - d\right)^2\right) \quad (19)$$

$$A_c = \sum_{i=1}^n 2\pi L(2r_i + h), b = \sum_{i=1}^n 2\pi r_i \quad (20)$$

where L is the length of the piston and V is the velocity of the piston.

5) Rotary disk ER fluid damper

Suppose all the rotating disks have the same thickness d and all the stationary disks have the same thickness e , then the maximum number N of rotating disks can be calculated by the following formula:

$$L_0 = Nd + (N+1)e + 2Nh \quad (21)$$

where L_0 is the inner length of the cylinder case.

$$T_v = \frac{\mu \omega \pi}{2h} (R_2^4 - R_1^4) \quad (22)$$

$$T_{ER}(E) = \frac{2n\pi\tau_y(E)}{3} (R_2^3 - R_1^3) \quad (23)$$

where ω is the angular velocity of the rotational axis, h is the gap size between the disks, R_2 is the outer radius, R_1 is the hub radius, and n is the number of the sides of rotating disks.

6) Rotary cylinder ER damper

It is assumed that all the stationary cylinders have the same thickness and all the moving cylinders pieces have the same thickness as well. The gape sizes between stationary cylinders and moving cylinders are assumed to be equal. Then, suppose there are N stationary cylinders, there will be $N+1$ moving cylinders and $2N+1$ sliding surfaces.

Let the radius of the central cylinder be r_0 , the thickness of other moving cylinders be d , the thickness of the stationary cylinder be e , and the gap size be h . Then the radius of the sliding surfaces can be calculated using the same equations for concentric cylinder ER fluid dampers.

$$T_v = \frac{2\pi L \mu \omega}{h} \sum_{i=1}^{2N+1} r_i^3 \quad (24)$$

$$T_{ER} = 2\pi\tau_y L \sum_{i=1}^{2N+1} r_i^2 \quad (25)$$

where L is the length of the piston, r_i is the radius of the i th rotating cylinder, ω is the angular velocity of the rotating axis, and h is the gap between a rotating cylinder and a stationary cylinder.

3. Performance Comparison

As we have seen in Section 2, both zero-field and ER damping forces are proportional to the piston length or the valve length. The zero-field damping force is proportional to the velocity of piston and the viscosity of the ER fluid. While the ER damping force is independent of viscosity coefficient and the velocity of the piston, it is dependent on the yield stress of the ER fluid.

For the general design of an ER damper system, the main constraints can be categorized as structural constraints, constraints from power supply, and constraints from available ER fluids. Structural constraints are mainly allowable minimum gap size and the size of the damper system (diameter and length of the outer cylinder, size and weight of the power supply system, etc.). The primary constraints from power supply are maximum voltage and maximum power available. The minimum viscosity and maximum yield stress of available ER fluids will limit the achievable performance of an ER damper system. There are also other kinds of constraints on ER damper systems encountered in practical applications, for instance, stroke length, desired damping characteristics, etc.

Since a good ER damper should provide a wide range of damping, we will investigate in this section, assuming similar geometric parameters, the maximum ER damping force, the zero-field damping force, and the ratio of ER damping force over zero-field damping force. This ratio may be used to measure the controllable range of damping forces.

The constraints on the dampers are assumed to be: 1) inner diameter of cylinder: 80 mm, 2) inner length of the cylinder: 180 mm, 3) minimum gap size: 1.0 mm; 4) viscosity of the ER fluid: 0.1 Pa.s (100 cp). Without loss of generality, we assume that the velocity of the piston is 1 m/s.

For translational dampers, since the piston side-wall serves as an electrode, its length cannot assume the whole inside length of the outer cylinder in order to guarantee a required stroke length.

Suppose the length of the piston $L = \gamma L_0$, where L_0 (here we assume it to be 180mm) is the inside length of the outer cylinder and γ is a constant, then the stroke of the piston is $0.5(1 - \gamma)L_0$. In the following, we assume that $\gamma = 0.3$. In order to compare the performance of the dampers discussed in the above sections, for gap size $h=1$ mm and the yield stress $\tau_y=5$ kPa. In Table 1 and Table 2, we listed the zero-field damping force (torque), ER damping force (torque), the ratio E_{ff} of ER damping force (torque)

over zero-field damping force (torque) for translational dampers and rotary dampers, respectively.

The values in the tables are obtained based on the following assumptions: 1) for sliding cylinder ER dampers, all the moving cylinders between the fixed cylinders have the same thickness 2.0mm and all the fixed cylinders have the same thickness 2.0mm; 2) for concentric cylinder ER dampers, all the cylinders have the same thickness of 2.0mm; 3) for rotary disk ER dampers, all the rotating and stationary disks have the same thickness 2.0mm and $R_1 / R_2 = 0.25$; 4) for rotary cylinder ER dampers, all cylinders have the same thickness of 2.0mm and $R_1 / R_2 = 0.25$.

It can be seen from Table 1 that the COTS damper has the largest ER damping force but at the same time has a huge zero-field damping force. The ratio Eff of the ER damping force over the zero-field damping force is very small comparing to other damper designs. In order to give a large range of adjustable damping, a large gap size or a short valve length has to be used. However, increasing the gap size or reducing valve length will result in a decrease in the ER damping efficiency. It is difficult to make a trade-off between zero-field and ER damping forces for this type of dampers. From Table 2, it can be seen that the performance of the two rotary dampers are similar, both have high ratio of ER damping torque over zero-field damping torque.

The comparison among the proposed ER fluid damper designs reveals that the sliding cylinder damper design, the concentric cylinder damper design, and the two rotary damper designs are more promising than the other designs. In particular, the two rotary dampers have the potential to provide a very wide range of damping if an appropriate device is provided to transfer a translational motion to a rotary motion.

Table 1. Comparison of translational ER fluid dampers

	COTS	Sliding piston	Sliding cylinder	Concentric cylinder
Zero_field damping force(N)	21715	20.4	317	574
ER damping force(N)	9048	226	1561	2470
Ratio Eff	0.42	11.1	4.9	4.3

Table2. Comparison of rotary ER fluid dampers

	Rotary disk	Rotary cylinder
Zero_field damping torque(Nm)	0.6	0.73
ER damping torque(Nm)	39.6	45.9
Ratio Eff	66.0	63.2

4. Shock Suppression Using ER Fluid Damper

In this section, a simple system shown in Fig.2 is used to demonstrate effectiveness of ER fluid damper in shock impact reduction on the supported mass m_1 which may represent a shipboard equipment.

Assume that the displacements of the mass m_1 and the supporting plane are small, the equation of motion of the demo system can be written as

$$\ddot{x}_1 + 2\zeta_1\omega_1\dot{x}_1 + \omega_1^2x_1 - \omega_1^2x_2 = f_{ER} \quad (26)$$

$$\ddot{x}_2 + (\omega_2^2 + \omega_3^2)x_2 - \omega_3^2x_1 = f \quad (27)$$

$$\omega_1^2 = \frac{k_1}{m_1}, \omega_2^2 = \frac{k_2l_1^2}{J_s}, \omega_3^2 = \frac{k_1l_1^2}{J_s} \quad (28)$$

$$\zeta_1 = \frac{c_1}{2m_1\omega_1}, f = \frac{Fl_1}{J_s}, f_{ER} = \frac{F_{ER}}{m_1} \quad (29)$$

where x_1 and x_2 are the displacements of mass m_1 and the supporting plane at the supporting point of spring K2, m_1 is the mass of mass m_1 , J_s is the moment of inertia of the supporting plane about the hinge, c_1 is the zero-field damping coefficient of the ER fluid damper, k_1 is the spring constant of spring K1, k_2 is the spring constant of the spring K2, F_{ER} is the equivalent ER damping force on mass m_1 , l_1 is the distance between the hinge joint and the supporting point of the spring K2, l is the distance between the hinge joint and the point the force F exerted on the supporting plane.

Since an ER damper cannot generate power, the following semi-active sky-hook control logic is used here

$$F_{ER} = \begin{cases} \text{sat}(2(\zeta_d - \zeta_1)\omega_1m_1\dot{x}_1), & \text{if } (\dot{x}_1 - \dot{x}_2)\dot{x}_1 > 0; \\ 0, & \text{otherwise.} \end{cases} \quad (30)$$

Since the available ER damping force is always limited, it can be shown that when ζ_d is very small, the acceleration of mass m_1 is very large, and when ζ_d is too large, the acceleration of mass m_1 is also large due to the saturation effect. Therefore, an appropriate damping ratio must be chosen to achieve an optimal attenuation of the "g" effect on mass m_1 .

In computer simulation, we assume that $\omega_1 = 62.83$ rad/s (10Hz), $\omega_2 = 125.66$ rad/s (20Hz), $\omega_3 = 88.87$ rad/s (14.1Hz), $\zeta_1 = 0.01$, and $\zeta_2 = 0.005$. The shock is a square pulse with a magnitude such that without applying ER damping, mass m_1 will experience a 10g acceleration.

The relationship between the maximum acceleration of mass m_1 under the shock response and the desired damping ratio is given in Figure 3. It can be seen that when ζ_d is less than 2.6, the magnitude of acceleration of mass m_1 decreases as ζ_d increases.

In the range $\zeta_d > 2.6$, the curve in Figure 3 is not very smooth. The reason for this is that saturation function introduces nonlinearity into the system. The optimal desired damping ratio is 2.6. At this damping ratio, a 34% reduction is achieved from the undamped response. The time response of accelerations of m_1 is given in Figure 4.

As we know, there are always volume and power limitations for a dampening system in practical applications. Therefore, it is desirable to estimate a prior the size and power requirement of an ER damper for a specific application to determine its feasibility. Here we give an estimation on the size and power requirements of rotary disk dampers for shock dampening of shipboard equipment.

According to the military specifications on the shock test of shipboard equipment (MIL-S 901D (Navy), 1989), shock tests are divided into three categories: 1) light weight test; 2) medium weight test; 3) heavy weight test. In light weight test, the weight of the equipment should not exceed 550 pounds. In medium weight test, the maximum weight of the equipment is 7400 pounds. For heavy weight test, the weight of equipment is not specified.

Let us assume that a ball screw is used to transfer the translational motion of the supported equipment to the rotational motion of the rotary disk damper. Then the ER damper force can be written as

$$F_{ER} = \frac{2N\pi\tau_y(E)}{3\eta D_h \tan \lambda} (D_2^3 - D_1^3) \quad (31)$$

where η is the ball screw efficiency, D_h is the pitch diameter, λ is the lead angle, D_2 and D_1 are the inner and outer diameters of the rotating disk, and N is the maximum number of rotating disks that can be installed in the cylinder case, which is a function of the inner length of the cylinder case.

Suppose four dampers are used to support a equipment and the acceleration reduction in Fig.4 is to be achieved, we have

$$14.25m_1 = \frac{2N\pi\tau_y}{3\eta D_h \tan \lambda} (D_2^3 - D_1^3) \quad (32)$$

The length and diameters of the ER dampers for different weight of equipment to be supported is given in Figure 5 assuming: $\tau_y = 5000$ Pa, $D_1/D_2 = 0.20$, $\eta = 0.90$, $D_h = 1.5$ in, $\lambda = 17.5^\circ$, thickness of rotating and stationary disks: 1.0 mm, gap size: 1.0 mm.

The power required for an ER dampers consists of two parts: capacitive and resistive power. The capacitive power varies with the working frequency of the ER dampers, the larger the maximum working frequency, the larger the capacitive power. The resistive electric power mainly depends on the current density of the ER fluid in the damper. For a preliminary study, the equivalent circuit for an ER damper is considered as a RC circuit as shown in Figure 6 (Sturk, et al, 1995; Lou, et al, 1994):

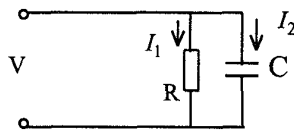


Figure 6. Equivalent circuit for an ER damper.

Assume a sinusoidal voltage input, i.e. $V = V_m \sin(2\pi ft)$, then the power W can be written as

$$P = I_d A_c V_m \sin^2(2\pi ft) + \pi f C V_m^2 \sin(4\pi ft) \quad (33)$$

where f is the frequency of the signal, I_d is the current density, A_c is the charged area, and C is the capacity of the ER damper. The peak power requirement is derived as

$$P_p = 0.5 I_d V_m + 0.5 V_m \sqrt{(2\pi f C V_m)^2 + (I_d A_c)^2} \quad (34)$$

The peak power requirements for different weight of the equipment are given in Figure 7 based on the following assumptions: 1) $f = 50$ Hz; 2) $V_m = 5000$ volt, 3) $I_e = 1 \mu A/cm^2$, 4) dielectric constant: 3.0. It is noted that a large part of the power is used for the capacitor charge. It can be seen that the power required for an ER dampers is fairly small comparing to the weight of the equipment to be supported.

5. Conclusions

This research has demonstrated the feasibility and potential of ER dampening systems to the shock dampening of shipboard equipment. The study is based on damper device construction, ER fluid development, and electrical power supplies. It has been shown that a significant reduction of shock impact on shipboard equipment can be achieved via ER damping devices. An

investigation on the size and power requirement has been conducted, it shows that ER dampening devices are quite suitable for shock protection of light weight and medium weight shipboard equipment according to military shock test specifications.

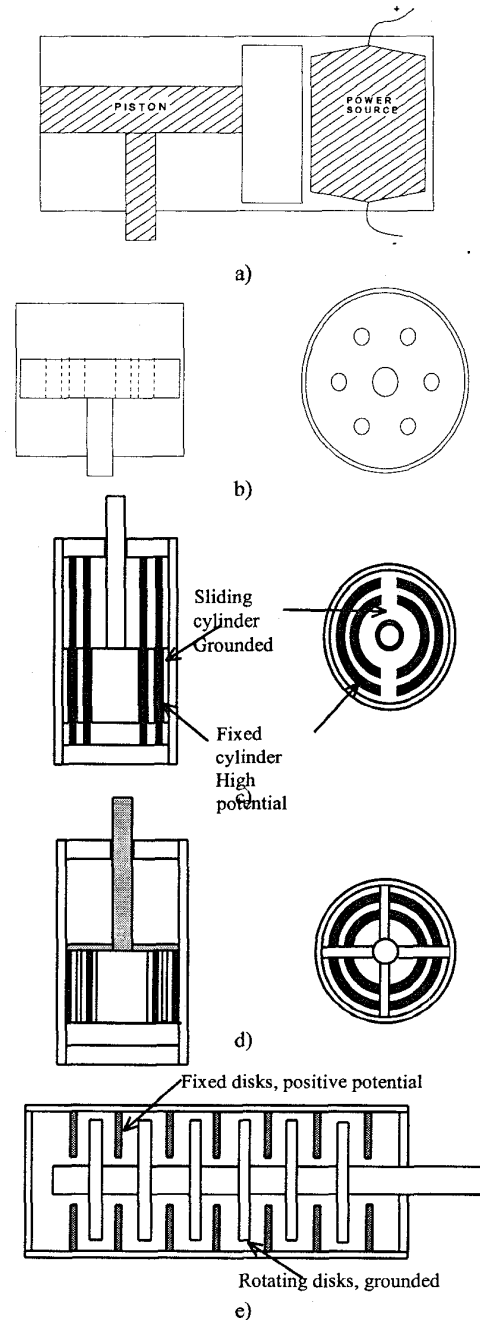


Fig.1. Schematic of ER fluid dampers

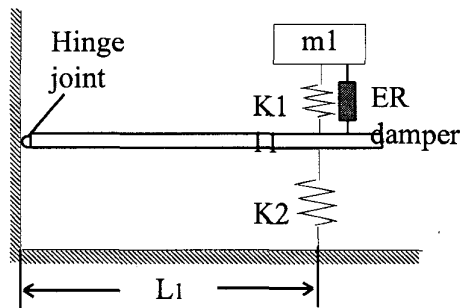


Fig.2. A two degree of freedom spring-mass system.

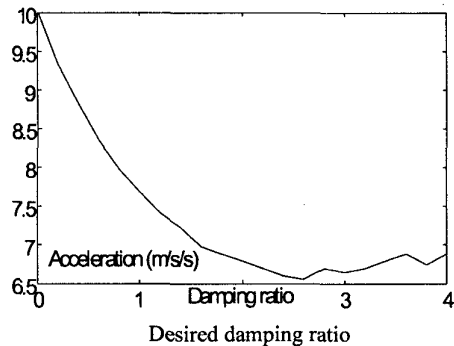


Fig.3. Max. acceleration of mass m1 vs. desired damping ratio.

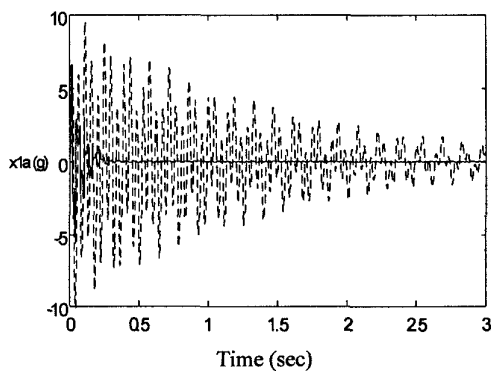


Fig.4. Time response of the acceleration of mass m1, dashed line: without ER damping, solid line: with ER damping.

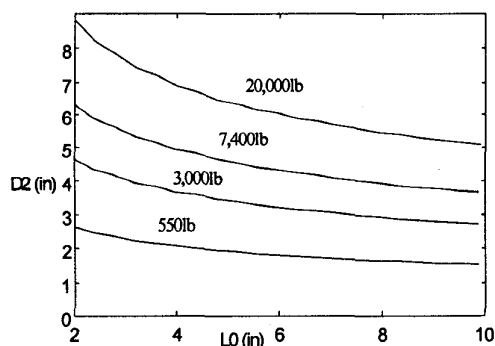


Fig.5. ER damper size vs. equipment weight.

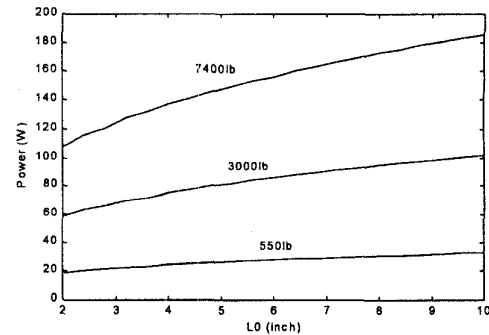


Fig.7. Power requirement vs. equipment weight.

References

- Blackwood, G.H. and von Flotow, A.H., Recent Advances in Active Control of Sound and Vibration, 1993.
- H. Block and J.P. Kelly, Journal of Physics, Part D: Applied Physics, vol.21, pp.1661-1677, 1988.
- M.J. Brenann, M.J. Day, R.J. Randal, Smart Materials and Structures, vol.4, pp.83-92, 1995.
- W.A. Bullough, D.J. Peel, J.L. Spruston, Developments in Electrorheological Flows and Measurement Uncertainty, Fed-Vol.205/AMD-Vol.190, ASME 1994.
- Crede, C.E., John Wiley and Sons, Inc., 1951.
- F.E. Fillisko and L.H. Radzilowski, Journal of Rheology, vol.34, pp.539-552, 1990.
- Von Flotow, A.H., Proceedings of 27th Conference on Decision and Control, 1988.
- D.L. Hartsock, R.F. Novak, and G.J. Chaundy, Journal of Rheology, vol.35, pp.1305-1327, 1991.
- Herzog, R., Journal of Dynamic Systems, Measurements, and Control, September 1994, Vol. 116, pp. 367-371.
- Hyde, T.T. and Anderson, E.H., SPIE, Vol. 2193, 1994, pp. 36-46.
- R.E. Jordan and M.T. Shaw, IEEE Trans. Elec. Insulation, vol.24, no.5, pp.849-879, 1989.
- D. Karnopp, M.J. Crosby, and R.A. Harwood, Journal of Engineering for Industry, pp.619-626, 1974.
- Z. Lou, R.D. Ervin, and F.E. Fillisko, Journal of Fluids Engineering, vol.116, pp.571-576.
- Military Specification, Mil-S-901D (Navy), 1989.
- Peel, D. J. and Bullough, W. A, Proc. Instn. Mech. Engrs., Vol. 208, p. 253-266.
- S. Rakheja and S. Sankar, Journal of Vibration, Acoustics, Stress, and Reliability in Design, vol.107, pp.398-403, 1985.
- Stanway, R., Sproston, J., and Firoozian, R.; J. of Dynamics, Measurement, and Control, Vol. 111, March, 1989, p. 91-102.
- J.H. Spurr and R. Munzing, Proceeding of Developments in Electrorheological flows and Measurement Uncertainties, pp.51-56, ASME 1994.
- M. Sturk, X.M. Wu, and J.Y. Wong, Vehicle System Dynamics, vol.24, pp.101-121, 1995.
- Thompson, A.G., Vehicle System Dynamics, Vol. 5, 1976, pp. 187-203.
- Tupper, T. and Crawley, E.F., Proceedings of American Control Conference, 1995.
- K.W. Wang, Y.S. Kim, and D.B. Shea, Journal of Sound and Vibration, vol.177, no.2, pp.227-237, 1994.