STUDY OF ELECTRORHEOLOGICAL DAMPER FOR CONTROL OF ROTOR VIBRATION

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Abstract

Rotating machines have a wide range of applications. Rotors must be designed carefully to avoid natural frequencies within the range of operating speeds. Vibrations occur because of existence of imbalance in the system and rotational speed is close to the critical speed. Torsional vibrations are not easily detectable and difficult to control. For safety of rotating machines, various methods are used to reduce their torsional vibrations. Use of semi-active dynamic damper with electrorheological (ER) fluid is one option.
Smart material is a mixture of micron-sized semiconducting or polarizable material particles in a non-conducting fluid like silicone oil. ER fluids are smart materials and their rheological properties can be modified quickly and continuously under the application of electrical field. The fundamental modes of operation of ER damper are flow mode, shear mode and squeeze mode. This paper aims studying a theoretical model of a torsional vibration damper with modifications in the bob geometry to include *end effect* in the study. The effect of ER fluid on the torsional vibration control is examined here using a single degree of freedom system model. The effects of end geometry of the bob on torsional damping and stiffness are studied here.
Keywords: torsional vibrations, ER dampers, vibration control, smart fluids, rotor vibrations.

1. Introduction

It is important to study torsional vibration in many mechanical systems with rotating mechanical components. The rotory machine parts like shafts, axles etc. are rotating with high speed and faces torsional vibrations due to various reasons like imbalance and uneven wearing of parts. During the resonance condition the torsional vibration effect is unpredictable and dangerous. Torsional vibration level of a machine decides the feasibility and efficiency of rotating machines. Hence it is desirable to reduce the vibration level to the acceptable limit.

Gordaninejad et al. [1] designed single and multi-electrode cylindrical ER dampers. They prepared one, two, three and fifteen electrode ER dampers in a small range. Forced longitudinal vibration controls of these cylindrical dampers using ER fluid were studied by using dampers individually or by connecting in series. Performance of these fluid dampers was examined experimentally and it was concluded that with larger electrode surface and smaller electrode gap, higher damping coefficient can be achieved. Choi and Werely [2] demonstrated the effectiveness and feasibility of ER and MR fluid based landing gear systems to control longitudinal vibrations due to landing impact. The theoretical model for ER/MR shock struts was developed. Telescopic type of helicopter landing gear system in sliding mode was constructed and experimental data was obtained at different excitation velocities and electric fields. They demonstrated that acceleration and displacement were significantly reduced with this controller. Choi et al. [3] designed and manufactured cylindrical seat damper with Arabic gum based ER fluid. During experimentation, acceleration from the accelerometer mounted on the seat cushion and the vertical displacement of the seat is measured from the

LVDT. A full car model with seat suspension is established and feasibility of this model is demonstrated.

Y. Sun and M.Thomas [4] carried out the modeling of ER fluids for steady state and oscillatory dynamic conditions in shear mode to control torsional vibrations with the use of diatomite based ER fluid. They used the Bingham rheological model to explain the behavior of ER fluid at high shear rates. The Bingham rheological model and developed two models namely Quasi-Bingham and General Quasi-Bingham models. These models account for shear thinning effect and ER fluid behavior at low shear rate. They verified their models with experimental results.

The resistance offered by the fluid at top and bottom of a cylindrical bob is known as *end effects*. Many times the equations are derived without considering the *end effects*. If the end effect is accounted, it may give rise to desirable effects in dampers. This study aims the modification in bob design to external conical geometry to quantify the change in geometry.

2. Modeling for modified geometry

In rotational rheometer, an immersed bob or spindle is rotated in the fluid (Refer Fig. 1) whose viscosity is to be determined. The torque required to turn the bob is a function of viscosity of the fluid.

Considering the rheometer with a co-axial bob of external radius R with a conical end and Rb is the internal radius of the cup. H is a height of the bob. The semi cone angle of the cone be α. Initially, keeping the bob fixed, cup is moved with angular velocity ω. C is damping coefficient when zero electric field, C_{ER}, & K_{ER} are equivalent damping coefficient and equivalent stiffness when an electric field is non-zero.

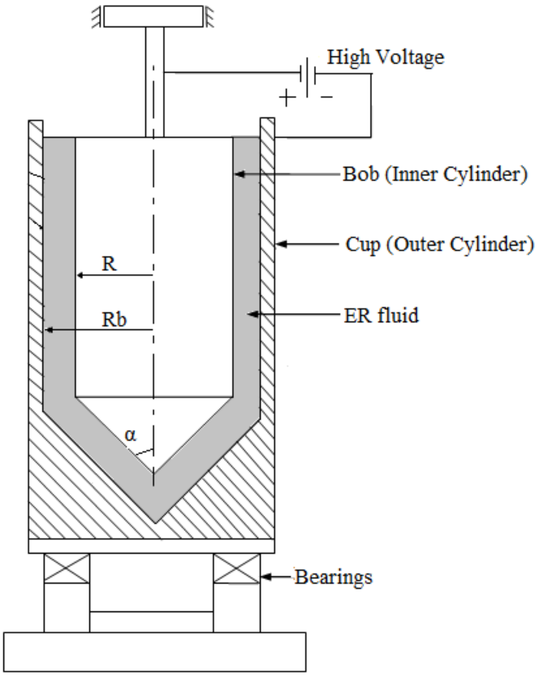


Figure 1: Conceptual setup showing bob with conical end.

θ and $\dot{\theta}$ is the angular displacement and angular velocity of a cup respectively. Two cylinders of inertia J_1 and J_2 serves as electrodes for an ER effect. Shearing of the ER fluid takes place in the annular gap of bob and cup and also with the conical bottom of the bob and cup.

Total torque, M_t = M + M_e

where, 'M' is the torque at the bob surface.

$$M_t = \left(2\pi R^2 H + \frac{2\pi R^3}{3\sin\alpha}\right)\tau$$

With this modified definitions the following modified equations are obtained in the way of [4].

or $$M_t = 2\pi R^2 \left(H + \frac{R}{3\sin\alpha}\right)\tau = K_{ER}\theta + \mathrm{C}\,\dot{\theta} + C_{ER}\dot{\theta} \qquad (R_b = R) \tag{1}$$

Shear stress of the fluid below the yield stress under dynamic oscillations is [4],

$$\tau = \left[G_0\,\gamma\left(1 - e^{-(\gamma_y/|\gamma|)}\right) + \eta_\infty\dot{\gamma} + (\eta_0 - \eta_\infty)\,\frac{\dot{\gamma}}{1 + (t\dot{\gamma})^n}\right]$$

and $$M_t = 2\pi R^2\left[H + \frac{R}{3\sin\alpha}\right] \times \left[G_0\,\gamma\left(1 - e^{-(\gamma_y/|\gamma|)}\right) + \eta_\infty\dot{\gamma} + (\eta_0 - \eta_\infty)\,\frac{\dot{\gamma}}{1+(t\dot{\gamma})^n}\right] \tag{2}$$

$$\dot{\gamma} = \frac{R}{e}\,\dot{\theta}$$

$$= 2\pi R^2\left[H + \frac{R}{3\sin\alpha}\right] \times G_0\,\gamma\left(1 - e^{-(\gamma_y/|\gamma|)}\right) + 2\pi R^2\left[H + \frac{R}{3\sin\alpha}\right] \times \eta_\infty\,\frac{R}{e}\,\dot{\theta} + 2\pi R^2\left[H + \frac{R}{3\sin\alpha}\right] \times (\eta_0 - \eta_\infty)\,\frac{R\,\dot{\theta}}{e[1+(t\dot{\gamma})^n]} \tag{3}$$

$$= K_{ER}\theta + \mathrm{C}\,\dot{\theta} + C_{ER}\dot{\theta}$$

$$C = \frac{2\pi R^3}{e}\left[H + \frac{R}{3\sin\alpha}\right]\eta_\infty \qquad \text{when, E= 0} \tag{4}$$

The damping and stiffness parameters of ER fluids considering the shearing at bottom of the bob,

$$C_{ER} = \frac{2\pi R^3\left[H+\frac{R}{3\sin\alpha}\right](\eta_0 - \eta_\infty)}{e[1+(t\dot{\gamma})^n]} \tag{5}$$

The nonlinear complex stiffness of the damper is,

$$K_{ER} = K_{ERO}\left(1 - e^{-(\gamma_y/|\gamma|)}\right) \tag{6}$$

where, $K_{ERO} = \dfrac{2\pi R^2\left[H + \frac{R}{3\sin\alpha}\right] \times G_0\,\gamma}{\theta}$

or $$K_{ERO} = \frac{2\pi R^3\left[H+\frac{R}{3\sin\alpha}\right]\times G_0}{e} \qquad \text{where, } \gamma = \frac{R}{e}\,\theta \tag{7}$$

Considering the above equations for the damping coefficient and torsional stiffness, non- dimensional equation for amplitude is formed.

Undamped natural frequencies of the primary system ω_{11} and of the absorber ω_{22} can be written

as, $$\omega_{11} = \sqrt{\frac{K_1}{J_1}} \text{ and } \omega_{22} = \sqrt{\frac{K_{EReq}}{J_2}} = \sqrt{\frac{K_{ERO}\left(1 - e^{-(\gamma_y/|\gamma|)}\right)}{J_2}} \tag{8}$$

The amplitude of vibration of the primary inertia J_1 can be written with dimensionless entities as follows [4],

$$\frac{\theta K_1}{M_0} = \sqrt{\frac{(2\zeta r)^2 + (r^2-\beta^2)^2}{(2\zeta r)^2\;(r^2-1+\mu r^2)^2 + [\mu r^2\beta^2-(r^2-1)(r^2-\beta^2)]^2}} \tag{9}$$

$$\mu = \frac{J_2}{J_1}, r = \frac{\omega}{\omega_{11}}, \quad \beta = \frac{\omega_{22}}{\omega_{11}}, \zeta = \frac{C+C_{eq}}{2J_2\omega_{11}} \tag{10}$$

3. Results and Discussions

Equations 5 and 6 give the damping of ER fluids in the absence and presence of electric field respectively. Equation 7 gives the nonlinear damping stiffness. Considering R= 0.01356 m,

H=0.0375 m, e=0.0014 m, $\alpha = 60^0$torsional stiffness is calculated. Table 1 represents the calculated torsional stiffness K_{ER} with conical bob end using the equations given.

Table 1: Parameters with modified geometry bob

Sr. No.	E^*	$G_O{}^*$	γ_y^*	K_{ER}
1.	0	0	0	0
2.	250	0	0	0
3.	500	300	0.0077	0.060
4.	750	500	0.0051	0.072
5.	1000	1700	0.0074	0.334
6.	1250	2100	0.0130	0.614
7.	1500	3900	0.0130	1.141
8.	1750	4000	0.0160	1.324
9.	2000	4400	0.0230	1.742
11.	2250	6000	0.0200	2.231
12.	2500	6100	0.0280	2.600
13.	2750	8400	0.0230	3.326
14.	3000	11000	0.0220	4.273
15.	3250	11900	0.0250	4.866
16.	3500	12200	0.0280	5.208

*values taken from [4]

Table 2 shows the calculated values of damping coefficient C at zero electric field, damping coefficient with varying electric field C_{ER} and damping ratio ζ for the bob with conical end at constant shear rate.

Table 2: Parameters with modified geometry bob

Sr. No.	E*	n*	t^*	$(\eta_o - \eta_\infty)^*$	$\dot{\gamma}^*$	C	C_{ER}	ζ
1.	0	1	0.0116	0	300	1E-04	0	9.25E-04
2.	250	1	0.0214	0.0071	300	1E-04	4.87E-7	9.30E-04
3.	500	1	0.7350	8.7946	300	1E-04	2.00E-5	1.11E-03
4.	750	1	0.7000	19.9601	300	1E-04	4.73E-5	1.36E-03
5.	1000	1	0.6640	27.2440	300	1E-04	6.84E-5	1.55E-03
6.	1250	1	0.5900	31.6035	300	1E-04	8.98E-5	1.75E-03
7.	1500	1	0.6040	38.8105	300	1E-04	1.09E-4	1.93E-03
8.	1750	1	0.5720	53.7906	300	1E-04	1.56E-4	2.37E-03
9.	2000	1	0.5450	57.8874	300	1E-04	1.76E-4	2.55E-03
10.	2250	1	0.5050	72.3545	300	1E-04	2.42E-4	3.16E-03
11.	2500	1	0.4840	80.3222	300	1E-04	2.80E-4	3.51E-03
12.	2750	1	0.4330	86.7181	300	1E-04	3.36E-4	4.03E-03
13.	3000	1	0.4300	93.2866	300	1E-04	3.61E-4	4.26E-03
14.	3250	1	0.4250	102.175	300	1E-04	3.96E-4	4.59E-03
15.	3500	1	0.4240	104.454	300	1E-04	4.13E-4	4.75E-03

*values taken from [4]

Fig. 2 shows comparison between torsional stiffness with electric field for proposed modification with the results given in [4].

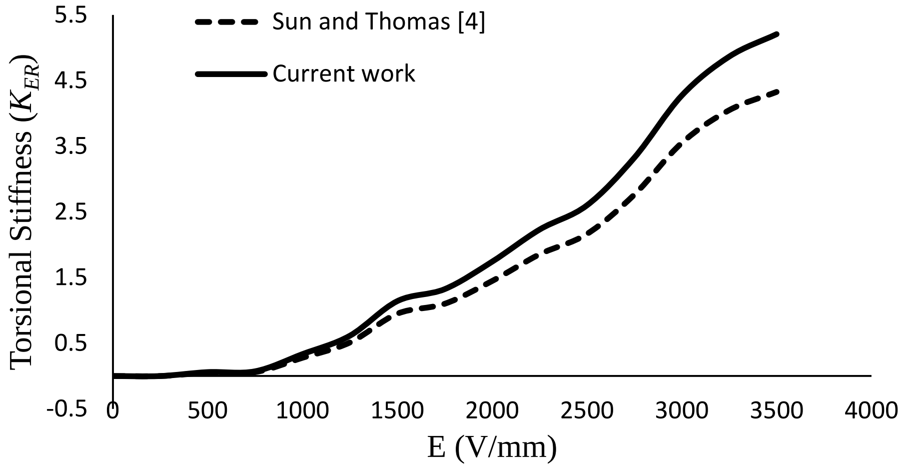


Figure 2: Variation of torsional stiffness with electric field

The presented values of torsional stiffness K_{ER} in Fig. 2 are calculated using Eq. (6) for strain of 0.015. It can be clearly seen that the torsional stiffness values calculated with the bob geometry modification are on the higher side in comparison with the values of the earlier geometry [4]. It rises to 5.2 Nm/rad from 4.3 Nm/rad. Clearly, the stiffness increases with the increase in the electric field making it possible for the damper to take more torsional loads.

Fig. 3 shows the plot of damping ratio with electric field obtained using Eq. (10). It can be seen that the damping ratio increases with the increase in electric field for both modified and original geometry of [4]. With the modification in the bob geometry at the end, the surface area increases hence the damping capacity goes up. This is reflected through the increased value of damping ratio from 0.0039 to 0.0047.

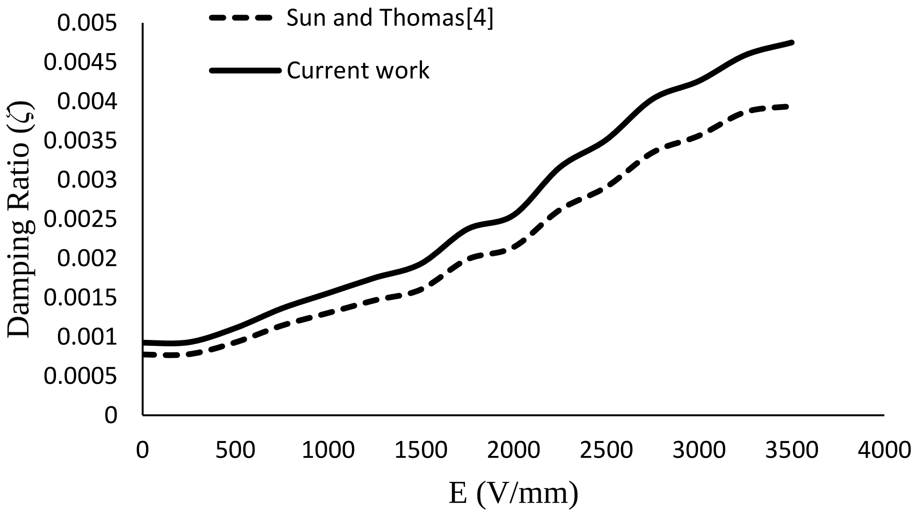
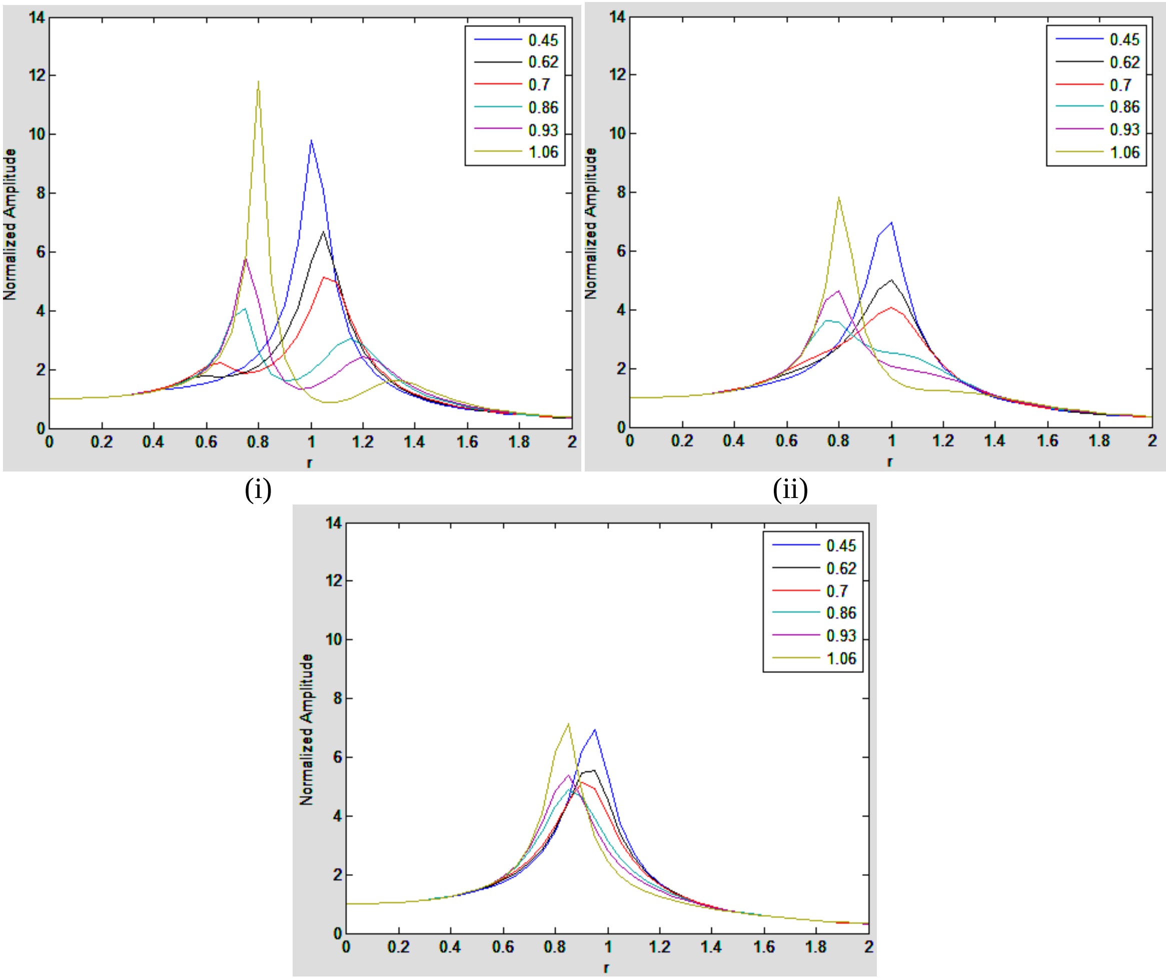


Figure 3: Variation of damping ratio with an electric field.

Increase in damping ratio dampens the system vibrations. This is due to the fact that as viscosity of the ER fluid increases with the increase in electric field, more energy is required to overcome the resistance offered by the ER fluid. The vibrational energy is used for this purpose causing the amplitude of vibration to reduce.

Fig. 4 (i), (ii) and (iii) show the plot of normalised amplitude with frequency ratio for different values of β using Eq. (9) for different damping ratios ζ.



(i) (ii)

(iii)

Figure 4: Normalise amplitude of primary system for the bob with conical end for (*i*) ζ=0.14, (*ii*) ζ=0.25 and (*iii*) ζ= 0.42 .

Amplitude of vibration decreases when the damping ratio ζ changes viz. 0.14, 0.25 and 0.42. For ζ= 0.25, the amplitude of vibration of the primary system reduces considerably as compared to the other two values of ζ. It can be observed in Fig. 4 (ii) that the normalized amplitude is least when the frequency ratio β is 0.86. Thus, the optimum value of β is said to be 0.86.

4. Conclusions

On the basis of the modification in the geometry of a bob, the following conclusion can be drawn.

ER fluids can be used for damping torsional vibrations of a system. The torsional stiffness and the equivalent damping of the ER fluid increase with increase in electric field thus making the damper capable of bearing larger torque and more effective in absorbing the vibrations. Thus different bob geometry modifications, leading to increase in area of contact of the bob with the ER fluid will enhance torsional capacity and damping properties of a torsional damper.

References

1. Gordaninejad F., Ray A., and Wang H. Control of Forced Vibration using Multi-Electrode Electro-Rheological Fluid Dampers, Journal of Vibration and Acoustics, 119, 527-531, (1997).

2. Choi Young-Tai and Wereley Norman M. Vibration Control of a Landing Gear System Featuring Electrorheological /Magnetorheological Fluids, Journal of Aircraft, 40 (3), 432-439, (2003).
3. Choi S. B., Choi J. H., Lee Y. S., and Han M. S, Vibration Control of an ER seat Suspension for a Commercial vehicle, Transactions of the ASME, 125, 60-68, (2003).
4. Sun Y and Thomas M. Control of Torsional Rotor Vibrations using an Electrorheological Fluid Dynamic Absorber, Journal of Vibration and Control, 17 (8), 1253-1264, (2010).
5. Vliet VanTon. Rheology and Fracture Mechanics of Foods, CRC Press, Taylor and Francis Group, (2014).