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Compromising vehicle handling and passenger ride comfort using ER-damper

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1. Introduction

Electrorheological (ER) fluids are fluids which exhibit fast and reversible changes in their rheological properties under the influence of external electrical fields. ER fluids commonly are composed of polarisable solid particles dispersed in non conducting oil. Upon the imposition of external electric field, the particles are polarized and form a chainlike structure along the direction of the field. This structure is responsible for the rheological properties alteration of the ER fluid from a fluid like state to a solid like state which exhibits a yield stress. The reversible and dramatic change in the rheological properties of ER fluids coupled with the instant response offers wide potential in industrial application. One of the promised applications is on the vibration control area, especially on semi active suspension.

Semi active suspension system has attracted a great deal of attention since this system combine the best feature of both passive and active control system. This system can achieve the optimum compromise between ride comfort and road handling for several road conditions. To achieve this condition, semi active system uses a controllable damper, a damper the damping level of which can be adjusted. There are two ways to adjust the damping level, whether controlling the fluid properties or modifying the orifice. The latter has a slower response time since it is using electro mechanical valve mechanism [1]. This method also leads to the inclusion of more moving parts inside of the damper, which could potentially lead to decreased reliability and a shortened lifetime.

2. Parallel plates method

Basically, the flow mode ER-damper consists of a piston rod and a piston head in concentric cylinder tubes which also act as electrodes (Fig. 1). If a certain level of forces given to the piston rod, there will be a pressure drop between upper section and lower section inside the annular gap and the fluid will move with a certain flux or velocity.

In the case where electrode gap D is much less than the inner electrode diameter $2R_1$, the solution for the Poiselle flow between concentric cylinder collapses to that of the flow between parallel plates [2]. For the fluid between parallel plates, the force equilibrium is [3]

$$\frac{d\tau}{dr} = \frac{\Delta P}{L} \tag{1}$$

where τ is shear stress; L is electrode length; ΔP is pressure drop.

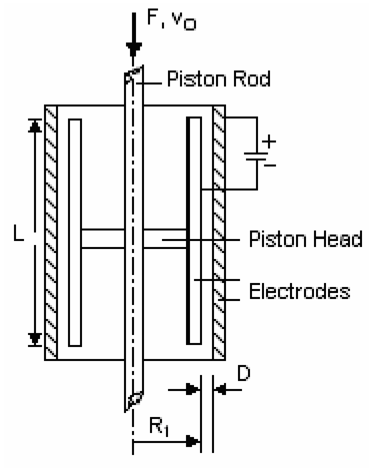

Fig. 1 Schematic diagram of flow-mode ER-damper

In zero field condition, the particle phase of ER fluid is randomly distributed in dispersed phase. So that, ER fluid will act as a Newtonian fluid the shear of which stress is proportional to the velocity profile gradient through the gap as

$$\tau = \mu_0 \frac{du}{dr} \tag{2}$$

where μ_0 is viscosity and u is velocity.

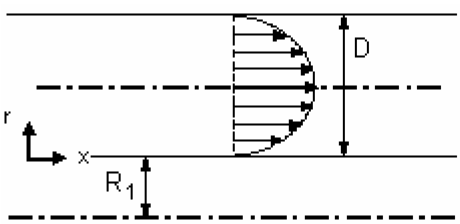

Fig. 2 Newtonian fluid velocity profile

The equation for the velocity profile (Fig. 2) can be obtained by substituting equation (2) to (1) and integrating with respect to r, as

$$u(r) = \frac{\Delta P}{2\mu_0}\left(r^2 - Dr\right) \tag{3}$$

By inserting the non-slip condition at the boundary such that $u(0) = u(D) = 0$, yields

$$\tau\left(r\right)=\frac{\Delta P}{2L}\left(2r-D\right) \tag{4}$$

The fluid volume flux through the electrodes (Q_N) is obtained via integrating the velocity profile over the annular electrode gap, which is

$$Q_N=-\frac{\pi R_1 D^3 \Delta P}{6\mu_0 L} \tag{5}$$

The fluid volume flux through annulus (Q_N) is equal to the fluid volume flux displaced by the piston head (Q_p). Since $Q_p = A_p v_0$ and $\Delta P = -F / A_p$, yields

$$F=\frac{A_p^{\,2} 6L\mu_0}{\pi R_1 D^3} v_0 \tag{6}$$

by noting that $F = C_N v_0$, gives

$$C_N=\frac{A_P^{\,2} 6L\mu_0}{\pi R_1 D^3} \tag{7}$$

in the above equations, A_p, C_N and v_0 are area of piston head, zero field damping constant and piston head velocity respectively.

If a certain amount of electric field strength is applied to ER fluid, the particle phase of the fluid will polarize and form a chain like structure between the electrodes. The fluid will only flow if the stress given is greater than a critical point called yield stress τ_y. This condition can be modeled by Bingham plastic behavior as

$$\tau=\tau_y sgn\left(\frac{du}{dr}\right)+\mu_p\frac{du}{dr} \tag{8}$$

The higher electric field strength (E) is applied, the higher τ_y will be produced. The relationship between E and τ_y is [4]

$$\tau_y\left(E\right)=\alpha E^2+\beta E+\chi \tag{9}$$

where α, β and χ are the characteristic value for particular ER fluid which is obtained via experiment.

The velocity profile of Bingham flow can be divided into three regions (Fig. 3). Region *1* and *3* are called in a post yield condition while region *2* is called as pre yield condition. On region *1* and *3*, τ is higher than τ_y so that the material is sheared. On the other hand, τ in region *2* is lower than τ_y so that the material will flow as a plug.

Substituting equation (8) to equation (1), integrating the result with respect to *r* and inserting boundary condition such as shown in Fig. 3, yields velocity profile equations for each region as

$$u_1(r)=\frac{\Delta P}{2\mu_P L}\left(r^2-2r_{p1}r\right) \tag{10}$$

$$u_2(r)=-\frac{\Delta P}{2\mu_P L}r_{p1}^{\;2} \tag{11}$$

$$u_3(r)=\frac{\Delta P}{2\mu_P L}\left(r^2-2r_{p2}r+2r_{p2}D-D^2\right) \tag{12}$$

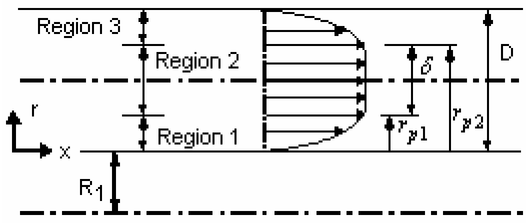

Fig. 3 Bingham fluid velocity profile

From the integration of equation (1), shear stress in region *2* is

$$\tau_2=\frac{\Delta P}{L}r+C_1 \tag{13}$$

by inserting boundary condition such that $\tau\left(r_{p1}\right)=\tau_y$ and $\tau\left(r_{p2}\right)=-\tau_y$ yields

$$\tau_y=\frac{\Delta P}{L}r_{p1}+C_1\,;\;\; -\tau_y=\frac{\Delta P}{L}r_{p2}+C_1 \tag{14}$$

Subtraction of these equations yields equation for plug thickness, δ, as

$$r_{p2}-r_{p1}=\delta=\frac{\tau_y A_p 2L}{\left|F\right|} \tag{15}$$

by introducing non dimensional plug thickness $\bar{\delta}=\frac{\delta}{D}$ and noting that $r_{p2}-r_{p1}=\delta$ and $r_{p1}-r_{p2}=D$ gives equations

$$r_{p1}=\frac{D\left(1-\bar{\delta}\right)}{2} \tag{16}$$

$$r_{p2}=\frac{D\left(1+\bar{\delta}\right)}{2} \tag{17}$$

By integrating fluid velocity profile for the whole regions with respect to *r* and inserting the value from equation (16) and (17), yields the equation for volume flux of Bingham flow, Q_B, as

$$Q_B = \frac{\pi R_1 \Delta P D^3}{12\mu_P L}\left(1-\bar{\delta}\right)^2\left(2+\bar{\delta}\right) \quad (18)$$

since $\Delta P = -F / A_p$, gives

$$F_B = \frac{12\mu_P L A_p^{\ 2}}{\pi R_1 D^3}\left(\left(1-\bar{\delta}\right)^2\left(2+\bar{\delta}\right)\right)^{-1} v_0 \quad (19)$$

so that the damping constant is

$$C_B = \frac{12\mu_P L A_p^{\ 2}}{\pi R_1 D^3}\left(\left(1-\bar{\delta}\right)^2\left(2+\bar{\delta}\right)\right)^{-1} \quad (20)$$

For Newtonian condition ($E = 0$ and $\bar{\delta} = 0$), equation (20) is similar to equation (7). Here, μ_P is plastic viscosity.

3. Result and discussion

The nominal damper design for the simulation is: L is 101.6 mm, R_1 is 2.54 cm, piston head diameter is 5.08 cm, and electrode gap is 1.5 mm. The damper then is filled with commercially available ER fluid made by Smart Technology Ltd, LID 3354. Dynamic viscosity of the fluid is 68.2 MPa s and the relation between yield stress with electric field strength is $\tau_y = 0.26E^2$ kPa [5]. By inserting all these values to the equations (7), (8) and (20), gives result which is plotted in Fig. 4.

Using this plot, the damping coefficient C for each electric field strength was obtained from the slope of the best linear curve fitting each line. The relationship between electric field strength applied to the damping constant is shown in Fig. 5.

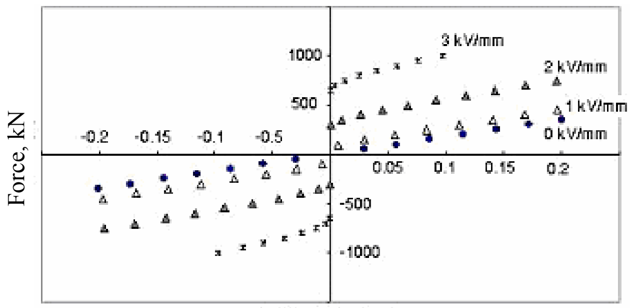

Fig. 4 Force versus velocity diagram

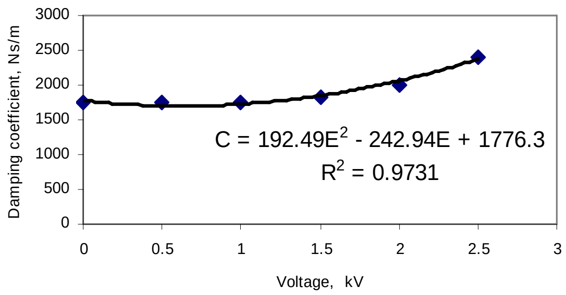

Fig. 5 Damping coefficient versus applied electric field

To examine the ride comfort and road handling, the suspension system is simplified as quarter car model with one degree of freedom (Fig. 6). This model still retains many of essential characteristic of a more complex system in its response to excitation [6].

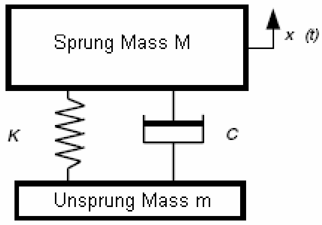

Fig. 6 Quarter car model

Ride comfort can be evaluated by transmissibility, defined as the ratio of the transmitted force to the excitation force, while road holding is characterized by the amplification ratio, related to the ratio of the resulting wheel hoop amplitude to the excitation amplitude of the motion. The equation of movement for the system above is written in the following form

$$M\ddot{x}(t) + C\dot{x}(t) + Kx(t) = f(t) \quad (21)$$

where M, C, K sprung mass, damping coefficient and spring stiffness respectively while $x(t)$ is the displacement and $f(t)$ is the road excitation. By assuming that a quarter of the weight of a car is 375 kg and the stiffness of the spring is 25000 N/m, the natural frequency of the system will be $f_n = \frac{\sqrt{K/M}}{2\pi} = 1.3$ Hz and the critical damping is $C_c = 2M\sqrt{K/M} = 6123.72$ Ns/m.

When a car implementing ER damper is being driven over a road with a certain surface profile, the degree of comfort and stability is highly determined by the damping coefficient as well as the road contour. For simplification, the road contour is approximated by a sine wave having various wavelengths L. Having through the road, the system will response and the contour of the road act as a support harmonic action on the system with the excitation frequency $\omega_{ex} = 2\pi v/L$. Here, v is the car velocity.

The transmissibility and the amplification ratio can be determined by the following formula [7]

$$Tr = \frac{\sqrt{1+(2\zeta r)^2}}{\sqrt{\left(1-r^2\right)^2+(2\zeta r)^2}} \quad (22)$$

$$AR = \frac{1}{\sqrt{\left(1-r^2\right)^2+(2\zeta r)^2}} \quad (23)$$

where r is the ratio between excitation frequency to the natural frequency and ζ is the ratio between actual damping coefficient and the critical damping. Fig. 6 shows the transmissibility and the amplification ratio of the system when the car with a certain velocity through over the road having certain wavelength so that the excitation frequency is as high as 2 Hz.

From the figures above (Figs. 7 and 8), it is clearly observed that the increasing of damping level will increase the transmissibility and decrease the amplification ratio. It means that the suspension with high damping level will provides good vehicle handling and stability but gives low comfort, since most of the road input is transmitted to the car. On the other hand, the soft damping level will yields good comfort but reduce the wheel contact to the road surface, which in turns reduce the vehicle handling. The best trade off between comfort and stability is obtained with the damping value at the intersection between normalized amplification ratio curve and the normalized transmissibility curve.

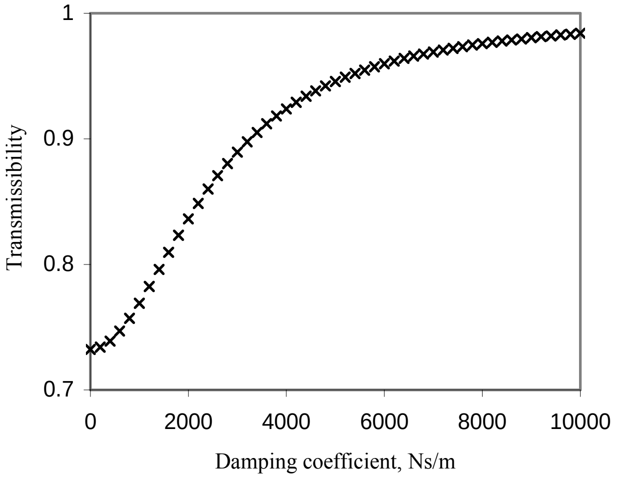

Fig. 7 Transmissibility versus damping constant for an excitation frequency at 2 Hz

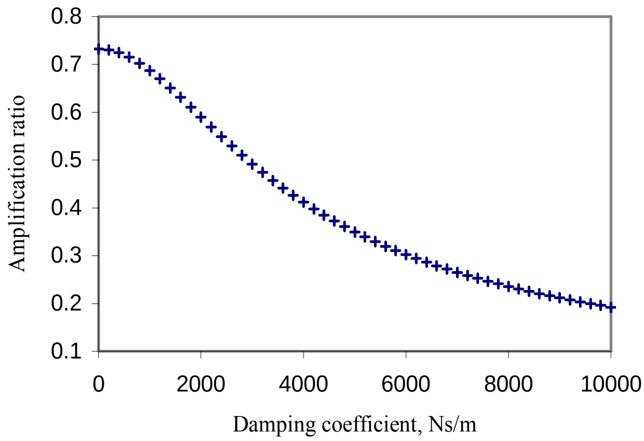

Fig. 8 Amplification ratio versus damping constant for an excitation frequency at 2 Hz

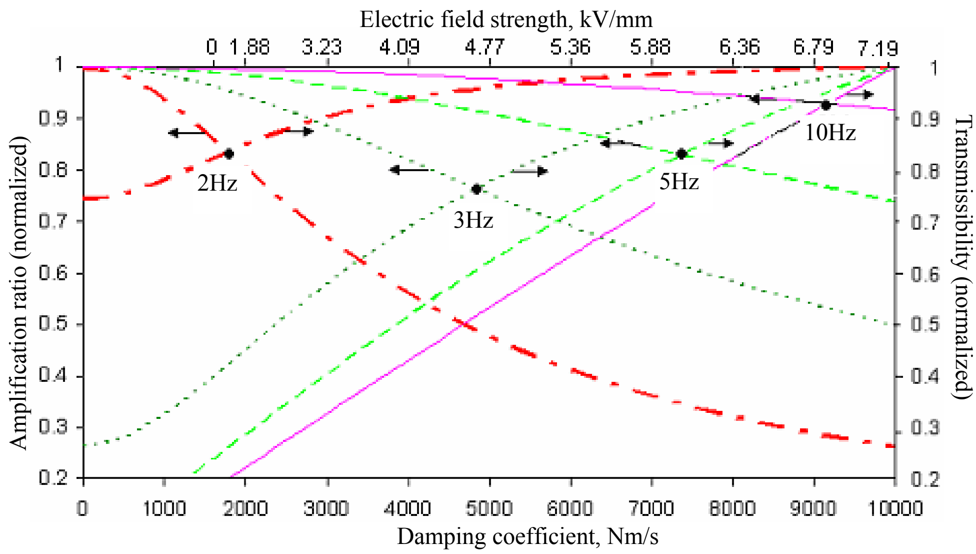

Fig. 9 Normalized transmissibility and amplification ratio versus electric field strength needed and damping coefficient for various excitation frequencies

The differences in vehicle velocity and the road profile yields a variation on the excitations frequency, which in turn require different damping level. Fig. 8 shows the optimum damping level for various excitation frequencies. At the excitation frequency as high as 2, 3 and 5 Hz, the optimum damping level is about 1786 Ns/m, 4854 Ns/m and 7242 Ns/m. Recalling the relationship between damping coefficient and the electric field strength (Fig. 5), the field strength needed to achieve the best comfort and stability compromise at the excitation frequency as high as 2, 3 and 5 Hz is 1.3, 4.7 and 6 kV/mm respectively (Fig. 9.

4. Conclusion

The performance of suspension system is determined by the compromise between ride comfort and road handling which can be evaluated by transmissibility and amplification ratio. The implementation of ER damper allows the system to adjust its damping level due to the road profile by controlling the electric field applied. For the simplified road profile and the nominal damper design, the field strength needed to achieve the best comfort and stability compromise at the excitation frequency as high as 2, 2.5 and 3 Hz is 1.3, 3.8 and 4.7 kV/mm respectively.

5. Acknowledgment

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6. References

1. **Suh, M.S., and Yeo, M.S.** Development of semi-active suspension systems using ER fluids for wheeled vehicle.-J. of Intelligent Material Systems and Structures, 1999, v.10(7), p.738-742.
2. **Nicos, M., Scott, A.B. and Douglas, P.T.** Electrorheological damper with annular ducts for seismic protection applications.-Smart Materials and Structures, 1996, v.5, p.551-564.
3. **William F. Hughes,** An Introduction to Viscous Flow.-London: McGraw-Hill Book Company, 1979.
4. **Kamath, G. M., Hurt, M. K., Wereley, N.M.** Analysis and testing of Bingham plastic behavior in semi-active electrorheological fluid dampers.-Smart Materials and Structures, 1996, v.5, p.576-590.
5. http://www.smarttec.co.uk, (15th January 2005)
6. **Sassi, S., Cherif, K., and Thomas, M.** On the design and testing of a smart car damper based on electro-rheological technology.-Smart Materials and Structures, 2003, v.12, p.873-880.
7. **Thomson, W.T.** Theory of Vibration with Applications, 2nd Edition.-New Jersey: Prentice-Hall Inc, 1981.

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TRANSPORTO PRIEMONĖS VALDYMO IR KELEIVIO VAŽIAVIMO PATOGUMO DERINIMAS NAUDOJANT ER SLOPINTUVUS

R e z i u m ė

Straipsnyje parodyti elektroreologinių slopintuvų pranašumai valdant nepageidautinus virpesius, kurių priežastis yra kelio paviršiaus nelygumai. Iš pradžių sudarytas matematinis modelis, aprašantis srauto tarp lygiagrečių plokščių būsenos įtaką, pagrįsta lygiagrečiu plokščių priartėjimu. Nustačius apkrovimo sąlygas ir slopintuvo slopinimo lygį, pakaba imituojama, esant skirtingoms kelio paviršiaus sąlygoms, kurios traktuojamos kaip sinusinė banga. Važiavimo patogumas nusakomas jėga, nuo kelio perduodama keleiviui ir išreiškiama stiprinimo santykiu. Gauti rezultatai rodo, kad elektroreologinis slopintuvas leidžia pasiekti geriausią transporto priemonės valdymo kelyje ir keleivio važiavimo patogumo santykį, reguliuojant ER slopintuvų slopinimo lygį pagal kelio sąlygas. Norint sukonstruoti geriausiai suderintą slopintuvą, elektrinio lauko stiprumas turi pasiekti 1.3, 4.7 ir 6 kV/mm, kai žadinimo dažnis didesnis kaip 2, 3 ir 5 Hz.

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COMPROMISING VEHICLE HANDLING AND PASSENGER RIDE COMFORT USING ER-DAMPER

S u m m a r y

This paper presents the superiority of electrorheological damper in controlling the unwanted vibration caused by road surface irregularities. First, a mathematical model to describe the flow behavior on the damper is developed based on parallel plates approach. After determining the load condition and the damping level of the damper, the suspension then is simulated in several road conditions, which are simplified as sine waves. The ride comfort is evaluated by the force transmitted from the road to the passenger, while the road handling is evaluated by amplification ratio. The result shows that electrorheological damper can achieve the best compromise between vehicle road handling and passenger ride comfort by adjusting its damping level due to the road condition. For the nominal damper design, the electric field strength needed to achieved the best compromise at the excitations frequency as high as 2, 3 and 5 Hz is 1.3, 4.7 and 6 kV/mm respectively.

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СОГЛАСОВАНИЕ УПРАВЛЕНИЯ ТРАНСПОРТНЫМИ СРЕДСТВАМИ И КОМФОРТОМ ЕЗДЫ ИСПОЛЬЗУЯ ER ГАСИТЕЛИ

Р е з ю м е

В статье представлены преимущества электрореологических гасителей управляя нежелательные вибрации, причиной которых являются неровности поверхности дороги. Создана математическая модель, которая описывает влияние состояния потока между параллельными пластинами. Определив условия нагружения и уровень демпфирования в гасителе, подвеска симулируется при разных условиях поверхности дороги, которые оценивается как синусоидальная волна. Комфорт езды оценивается силой передаваемой от дороги к пассажиру оцениваемой функцией усиления. Полученные результаты показывают, что электрореологическим гасителем можно достигнуть наилучшего согласования между управлением транспортного средства в дороге и комфортом езды пассажира, регулируя уровень гашения ER по условиям дороги. Для конструирования оптимального гасителя, при наилучшем согласовании величина интенсивности электрического поля должна достигнуть 1.3, 4.7 и 6 кВ/мм при частотах возмущения не выше 2, 3 и 5 Гц.

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