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THE USE OF RHEOLOGICAL FLUID TO CONTROL VIBRATION IN MECHANICAL SYSTEMS

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Abstract

Successful application of rheological fluids changed classical damper characteristics with uncontrollable viscosity fluid by introduction of controllable viscosity fluids. Changes in viscosity effect damping constant, therefore, behavior of mechanical system can be controlled by variable damping force or variable damping torque. Internal combustion engines and other machines use torsional vibration damper for limiting amplitudes of vibration at critical velocities. The design of this type of absorber consists of flywheel with friction rings that are free to rotate on the shaft and springs under tension to generate friction between flywheel and friction rings. When amplitude of torsional vibration increases, the flywheel does not follow the shaft oscillations and energy is dissipated by friction due to relative motion. The springs tension becomes a critical parameter in effective energy dissipation process. The second design is known as untuned viscous vibration damper. In this damper a torque results from the viscosity of fluid within the flywheel cavity and an relative motion. This particular design can find a wider range of applications by utilization of rheological fluid with a controllable apparent viscosity. In this paper discrete models of mechanical systems performing translation and torsional motion with controlled damping are discussed. The modeling indicates that rheological fluids can be effectively applied to control unwanted vibrations in mechanical systems.

INTRODUCTION

The vibrations in a mechanical system are controlled basically by the use of a linear and torsional dampers or suitable vibration absorbers. Vibration absorbers are also vibrating systems tuned to the frequency of the exciting force or torque. The absorbing system reduces the vibrations of the main system on the way of use of the damping force due to the viscosity of the fluid or suitable friction forces. The friction type torsional vibration damper commonly known as the Lanchester damper is used in torsional vibrating systems, such as gas and diesel engines to decrease the amplitude of torsional vibrations at critical speeds. The damper consists of two flywheels which are free to rotate on the shaft and driven only by means of the friction related to the normal pressure maintained by the tension of the springs. Properly adjusted, the flywheels rotate with the shaft for small oscillations. However, when the torsional oscillations of the shaft tend to become large, the flywheels do not follow the shaft motion because of their inertia, and energy is dissipated by friction due to relative velocity. The dissipation of energy limits the amplitude of vibration and preventing high shear stress in the shaft. The same property of damping unvanted torsional vibration can be achieved by replacing flywheels under spring tension by the structure with rheological fluid (RF) and applying proper (magnetic or electrical) external field. The second design - untuned viscous torsional damper is effective over relatively wide frequency range. It consists of a free rotating disc within a cylindrical cavity filled with viscous fluid. Often it is incorporated into the end pulley of a crankshaft that drives the cooling fan belt and is known as a Houdaille damper. This damper can be relatively simple converted to controllable untuned viscous vibration damper by the use of RF as a damping fluid. RF will change the character of damping torque, which will be based on viscous and Coulomb property of friction. The use of RF will improve damping characteristics, providing continuously variable damping torque and opportunity to work with critical damping over wide frequency range.

EQUIVALENT DAMPING OF RHEOLOGICAL FLUIDS

The response of rheological fluid RF under applied external field (RF) results from the changes in apparent viscosity of the suspension. In the absence of an applied external field rheological fluid often exhibit Newtonian-like behavior associated mostly with base fluid physical properties. Applied external field changes this behavior and RF shows a variable yield stress which depends on strength of that field. The Bingham plastic model of viscosity is often used to describe that phenomenon.

$$\tau_{RF} = \tau_{\circ}(RF) + \eta \frac{\partial \dot{x}}{\partial h} \tag{1}$$

when $\tau \geq \tau_{\circ}$

Where:

$\tau_{\circ}(RF) =$ Yield Stress as a function of external field.

$\eta \frac{\partial \dot{x}}{\partial h}$ =Newtonian Shear Stress proportional to dynamic viscosity of the base fluid η and velocity gradient $\frac{\partial \dot{x}}{\partial h}$

Below the yield stress the RF behaves viscoelastically. *Figure 1* shows the behaviour of the shear stress of RF under external field.

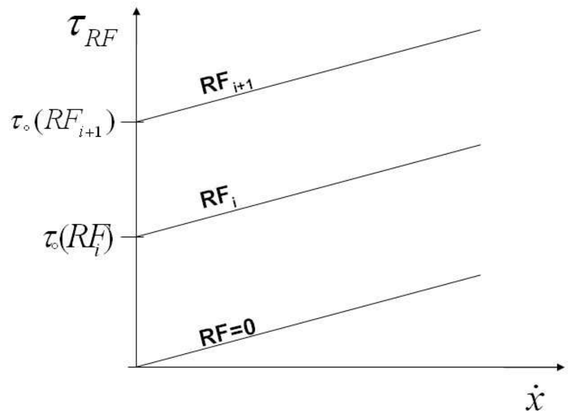

Figure 1: Shear stress of rheological fluid versus relative velocity and applied external field

According to Figure 1 the shear stress of RF can be expressed as:

$$\tau_{RF} = \tau_\circ(RF) + \frac{\partial \tau_{RE}}{\partial \dot{x}} \tag{2}$$

and damping factor C_{RF} is:

$$C_{RF} = [\{\tau_\circ(RF) + \frac{\partial \dot{x}}{\partial h}\dot{x}\}A]\frac{1}{\dot{x}} \tag{3}$$

where A is chosen oblique area (m^2)
The damping force F_d is

$$F_d = \tau_{RF} A \tag{4}$$

and can be expressed as:

$$F_d = \begin{cases} \tau_\circ(RF)A + \frac{\partial \tau_{RF}}{\partial \dot{x}} A\dot{x} & \dot{x} > 0 \\ 0 & \dot{x} = 0 \\ -\tau_\circ(RF)A - \frac{\partial \tau_{RF}}{\partial \dot{x}} A\dot{x} & \dot{x} > 0 \end{cases} \tag{5}$$

considering that:

$$\tau_\circ(RF)A = F_{d\circ}(RF) \tag{6}$$

represents yield cotrolled by external field damping force
and:

$$\frac{\partial \tau_{RF}}{\partial \dot{x}} A\dot{x} = F_{d\eta} \tag{7}$$

represents damping force proportional to the velocity $\dot{x}$.
The damping force is:

$$F_d = F_{do}(RF) + F_{d\eta} \tag{8}$$

Equivalent Damping

The 1 DOF vibrating system response under harmonic excitation force $F_\circ sin(\omega t)$ is:

$$m\ddot{x} + F_{d\circ}(RF)sgn(\dot{x}) + c_\eta \dot{x} + kx = F_\circ sin(\omega t) \tag{9}$$

where damping force has two components. One of them is a Newtonian type ad is proportional to the velocity $\dot{x}$, and a second one, which depends on strength of external field and direction of motion expressed by $sgn(\dot{x})$.

The energy dissipated, $\triangle E_\eta$ in a viscously damped system per one cycle with viscous damping coefficient c_η is:

$$\triangle E_\eta = \oint F_{d\eta} dx = \int_0^{\frac{2\pi}{\omega}} c_\eta \dot{x} \frac{dx}{dt} dt = \int_0^{\frac{2\pi}{\omega}} c_\eta \dot{x}^2 dt \tag{10}$$

Substituting: $x = Xsin(\omega t)$ and $\dot{x} = \omega X cos(\omega t)$ into above equation

$$\triangle E_\eta = c_\eta \int_0^{\frac{2\pi}{\omega}} (\omega^2 X^2 \omega cos^2(\omega t)) dt \tag{11}$$

We obtain

$$\triangle E_\eta = c_\eta \pi \omega X^2 \tag{12}$$

The second damping component represented by force $F_{d\circ}(RF)$ in Equation 8 yields following expression for dissipated energy:

$$\triangle E(RF) = F_{d\circ}(RF) \int_0^{\frac{2\pi}{\omega}} [sgn(\dot{x})\dot{x}] dt \tag{13}$$

and

$$\triangle E(RF) = F_{d\circ}(RF) X [\int_0^{\frac{\pi}{2}} cos(\omega t) d(\omega t) - \int_{\frac{\pi}{2}}^{\frac{3\pi}{2}} cos(\omega t) d(\omega t) + \int_{\frac{3\pi}{2}}^{2\pi} cos(\omega t) d(\omega t)] \tag{14}$$

Solving the integration yields that the energy dissipated by controllable damping force $F_{d\circ}(RF)$ is:

$$\triangle E(RF) = 4F_{d\circ}(RF)X \tag{15}$$

To create a viscously damped system of equivalent energy loss, we obtain:

$$\pi C_{eq}\omega X^2 = 4F_{d\circ}(RF)X + c_\eta \pi\omega X^2 \tag{16}$$

Thus the equivalent viscous-damping coefficient C_{eq} yields

$$C_{eq} = \frac{4F_{d\circ}(RF)X + c_\eta \pi\omega X^2}{\pi\omega X^2} \tag{17}$$

In terms of equivelant damping ratio ξ_{eq}:

$$C_{eq} = 2\xi_{eq}\omega_n m \tag{18}$$

and :

$$\xi_{eq} = \frac{4F_{d\circ}(RF)X + c_\eta \pi\omega X^2}{2\pi\omega\omega_n X^2} \tag{19}$$

The 1 DOF system with equivalent damping C_{eq} which will dissipate as much as energy as system described by Equation 8 is:

$$\ddot{x} + 2\xi_{eq}\omega_n\dot{x} + \omega_\eta^2 X = f_\circ sin(\omega t) \tag{20}$$

and this is also an approximation of the Equation 8

UNTUNED VIBRATION DAMPER WITH FIELD CONTROLABLE FLUID

The untuned viscous damper can be represented as a two - degree - of - freedom system. It is attached to the shaft, as being fixed at one end and has two dampers: Newtonian and a Coulomb type combined together at the other and. The damper is excited by harmonic torque $M_\circ e^{i\omega t}$ See Figure 2. Two equations of motion (21,22) are:

$$J\ddot{\theta} + k\theta + C_{eq}(\dot{\theta} - \dot{\theta}_d) = M_\circ e^{j\omega t} \tag{21}$$

$$J_d\ddot{\theta} - C_{eq}(\dot{\theta} - \dot{\theta}_d) = 0 \tag{22}$$

Where:

$$\theta = \theta_\circ e^{j\omega t} \tag{23}$$

and

$$\theta_d = \theta_{d\circ} e^{j\omega t} \tag{24}$$

After substitution of complex amplitudes (23,24) into equations of motion we have:

$$[(\frac{k}{J} - \omega^2) + j\frac{C_{eq}\omega}{J}]\theta_\circ - j\frac{C_{eq}\omega}{J}\theta_{d\circ} = \frac{M_\circ}{J} \tag{25}$$

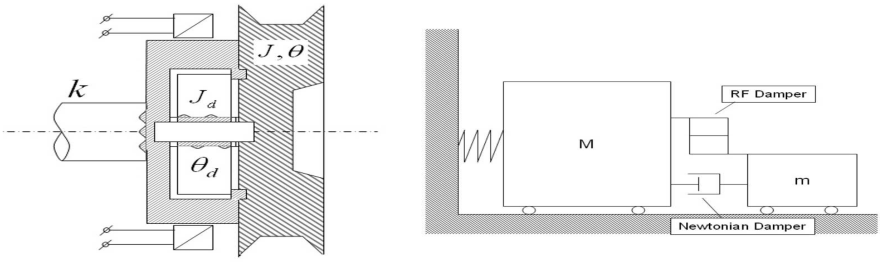

Figure 2: Schematic of untuned viscous vibration damper and its linear equivalent

$$(-\omega^2 + j\frac{C_{eq}\omega}{J_d})\theta_{d\circ} = \frac{jC_{eq}\omega}{J_d}\theta_\circ \tag{26}$$

Knowing that $\frac{k}{J} = \omega_n^2$ and $\frac{J_d}{J} = \mu$ we obtain following eguation as a function of ξ_{eq}, μ, ω, and ω_n. The graphical interpretation is shown in Figure 3.

$$|\frac{k\theta_\circ}{M_\circ}| = \left\{\frac{\mu^2(\frac{\omega}{\omega_n})^2 + 4\xi_{eq}}{\mu^2(\frac{\omega}{\omega_n})^2(1-\frac{\omega^2}{\omega_n^2})^2 + 4\xi_{eq}^2[\mu(\frac{\omega}{\omega_n})^2 - (1-\frac{\omega^2}{\omega_n^2})]}\right\}^{\frac{1}{2}} \tag{27}$$

Considering property of critical damping of vibrating system, the optimum equivalent damping ratio is:

$$\xi_{eq_{opt}} = \frac{\mu}{[2(1+\mu)(2+\mu)]^{\frac{1}{2}}} \tag{28}$$

The optimum equivalent damping constant for vibration damper with FR is:

$$C_{eq_{opt}} = 2\xi_{eq_{opt}}J\omega_n \tag{29}$$

This equivalent optimum damping constant can be tuned and controlled by intensity of applied external field.

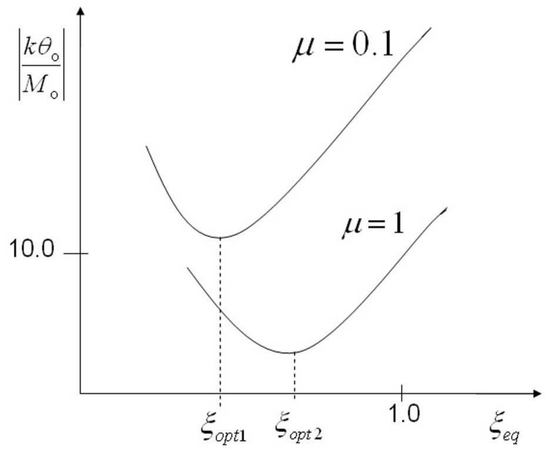

Figure 3: Graphical expression of amplitude ratio $|\frac{k\theta_\circ}{M_\circ}|$ *versus equivalent damping ratio* ξ_{eq} *for constant* μ

CONCLUSIONS

The new generation of untuned but controllable vibration dampers can be design by utilization of rheological fluids as a damping media. These dampers posses both Newtonian and Coulomb friction properties and can be tuned by the use of an external field to work with critical damping in the wide range of frequencies. The damping torque can be speed independent and continuously variable.

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