MAGNETORHEOLOGICAL FLUIDS IN VISCOS FRICTION TORSIONAL VIBRATION DAMPERS

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Abstract- Nowadays the torsional vibration damping of medium and slow-speed engines is achieved through untuned viscous friction dampers which are optimized for one particular severe order of vibration depending on the type of engine. Silicone fluids are presently used for its suitability to medium and slow-speed engines to suppress torsional vibrations of crankshaft. The challenge encountered in the design of these kinds of dampers is the desire of a damping fluid with sufficient elasticity to provide optimum tuning simultaneously with optimum damping while at the same time possessing acceptable stability over the operating temperatures and shear rates over the range of service conditions. Hence, silicone fluids are presently used for its suitability to this application. Since the viscous friction damper can be tuned for only one mode of vibration, relatively high vibratory amplitudes and shear stresses are induced in the crankshaft at other modes of vibration for which it is not optimized. Also, as the temperature of the fluid increases its viscosity falls down leading to further reduction in effectiveness of the damper. Seizing of damper can take place if the rise in temperature of the casing is above 60°C according to tests conducted by the B.I.C.E.R.A laboratories, irrespective of the initial temperature of the casing. Magnetorheological fluids can prove to be a successful surrogate for silicone fluids.

Magnetorheological (MR) fluids are smart fluids whose apparent viscosity changes when subjected to a magnetic field, to the point of becoming a viscoelastic solid. The yield stress of the fluid in its active state can be controlled very accurately by varying the magnetic field intensity and hence its ability to transmit force can be controlled.

I. INTRODUCTION

Viscous Friction Dampers are used in heavy duty applications such as medium-speed compression ignition engines and highly powered slow-speed marine propulsion engines to suppress torsional vibrations of crankshaft. The challenge encountered in the design of these kinds of dampers is the desire of a damping fluid with sufficient elasticity to provide optimum tuning simultaneously with optimum damping while at the same time possessing acceptable stability over the operating temperatures and shear rates over the range of service conditions. Hence, silicone fluids are presently used for its suitability to this application.

Since the viscous friction damper can be tuned for only one mode of vibration, relatively high vibratory amplitudes and shear stresses are induced in the crankshaft at other modes of vibration for which it is not optimized. Also, as the temperature of the fluid increases its viscosity falls down leading to further reduction in effectiveness of the damper. Seizing of damper can take place if the rise in temperature of the casing is above 60°C according to tests conducted by the B.I.C.E.R.A laboratories, irrespective of the initial temperature of the casing. Magnetorheological fluids can prove to be a successful surrogate for silicone fluids.

Magnetorheological (MR) fluids are smart fluids whose apparent viscosity changes when subjected to a magnetic field, to the point of becoming a viscoelastic solid. The yield stress of the fluid in its active state can be controlled very accurately by varying the magnetic field intensity and hence its ability to transmit force can be controlled.

II. THEORETICAL BACKGROUND

List of Notations

- \( J_d \) = mass polar moment of inertia of damper inertia member, kg.m\(^2\)
- \( J_e \) = mass polar moment of inertia of equivalent torsional pendulum, kg.m\(^2\)
- \( C_e \) = torsional rigidity of equivalent torsional pendulum, N.m/rad
- \( F \) = excitation frequency, \( F=60.0 \times 2\pi \), cycles per min.
- \( F_m \) = natural frequency of main system, \( F_m=60.0 \omega_m/2\pi \), cycles per min.
- \( S_d \) = specific damping torque of damper assembly, N.m/rad per sec
- \( (S_d)_o \) = optimum value of \( S_d \)
- \( K_d \) = non-dimensional damping coefficient of damper assembly, \( K_d = S_d/J_d \omega_m \)
- \( (K_d)_o \) = optimum value of \( K_d \)
- \( N \) = revolutions per minute
- \( n \) = order number of the excitation
- \( Q_E \) = resultant excitation torque, N.m
- \( Q_m \) = torque in main shaft, N.m
- \( U_E \) = equivalent mass ratio, \( U_E=J_d/J_m \)
- \( \theta_m \) = vibratory amplitude at \( J_m \) (\( J_E \)), radian
- \( \theta_i \) = relative vibratory amplitude at \( J_d \) and \( J_m \) (\( J_E \)), radian
- \( \theta_s \) = static deflection of original system, \( \theta_s=Q_d/C_E \), radian

A detailed torsional analysis performed by designers or engineers in practice involves the following steps,

- Calculation of torsional natural frequencies.
- Calculation of the torsional mode shapes.
- Development of the interference diagram.
- Definition of the coincidence of the excitation frequencies with the torsional natural frequencies.
• Calculation of the dynamic torsional oscillations at all the masses in the system based on the expected dynamic torque modulations, stress concentration factors and amplification factors.
• Calculation of the dynamic torsional stresses in all the shafts based on the expected dynamic torque modulations, stress concentration factors and amplification factors.
• Comparison of the calculated results with the applicable codes or specifications to determine compliance with regard to separation margin, stresses, gear loading and coupling dynamic torque.
• Parametric analysis to determine possible coupling modifications when the separation margin, stress levels and coupling torque are not acceptable.

A full torsional analysis may not be necessary for some simple systems. The degree of detail to identify any potential problems depends on the system and its operating conditions. Consider the approach followed by designers as given in Example 29.2 of for Fig.1(a). The tangential effort per cylinder is 0.0287 N.m and it is required to limit the maximum 4th order vibratory stress in the shaft to ±22407920.68 N/m².

\[ U_E = J_d / J_E \equiv 2/(\theta_m / \theta_i - 1) = 0.429 \]

The mass of the damper \( J_d \) from the above equation comes out to be 0.0339 kg.m². The viscosity of the fluid to be used is found out by first calculating for optimum damping coefficient and optimum specific damping torque.

\[ (K_d)_o \equiv \sqrt{[2/[1+U_E(2+U_E)]]} \equiv 0.76 \]
\[ (S_d)_o \equiv (K_d)_o \cdot J_d \cdot \omega_m \equiv 28.585 \text{ N.m per rad/s} \]

Since the specific damping torque is a function of the damper’s dimensions and the viscosity of the fluid, it will remain constant over the entire range of engine operation. When the engine operates at some other resonant frequency for which it is not optimised, the quantity which changes is the damping coefficient as the specific damping torque and mass of the damper remains constant (see equation 4). Our analysis consists of the following steps:-

a) Finding out all the natural frequencies of the system.

b) Studying the effect of varying damping coefficient on vibration amplitude for different mass ratios.

For finding out all the natural frequencies of the system, the following script is formed and executed in MATLAB.

```matlab
C1 = input('enter value of C1:');
C2 = input('enter value of C2:');
C3 = input('enter value of C3:');
C4 = input('enter value of C4:');
C5 = input('enter value of C5:');
J1 = input('enter value of J1:');
J2 = input('enter value of J2:');
J3 = input('enter value of J3:');
J4 = input('enter value of J4:');
J5 = input('enter value of J5:');
J6 = input('enter value of J6:');
omega_m = 1225000 rad/sec; %
Q_m = 176.48 N.m; %
\( \theta_m \) = ±0.0103 radian;
N = [1 0 0 0 0 0;... 0 1 0 0 0 0;... 0 0 1 0 0 0;... 0 0 0 1 0 0;... 0 0 0 0 1 0 0;... 0 0 0 0 0 1];
A = [J1+C1 C1 0 0 0 0;... J2+C1+C2 C2 0 0 0 0;... 0 0 0 0 0 0;... 0 0 0 0 0 0;... 0 0 0 0 0 0];
solve(det(A),y)
```

**Fig.1** – Model for torsional vibration analysis: (a) Original system; (b) Equivalent torsional pendulum.
Inserting the values \( J_1=J_2=J_4=J_5=0.0226 \text{ kg.m}^2 \), \( J_6=1.80775 \text{ kg.m}^2 \) & \( C_1=668305 \), \( C_2=C_3=C_4=463690 \text{ N.m/rad} \), \( C_5=210717 \text{ N.m/rad} \) from Example 29.2 of Reference, we get the values of natural frequency as

\[
\begin{align*}
  t_1 \left( \omega_{m1}^2 \right) &= 0 \\
  t_2 \left( \omega_{m2}^2 \right) &= 1222785.76; \omega_{m2} = 1106 \text{ rad/s} \\
  t_3 \left( \omega_{m3}^2 \right) &= 11747584.56; \omega_{m3} = 3427.47 \text{ rad/s} \\
  t_4 \left( \omega_{m4}^2 \right) &= 34515464.83; \omega_{m4} = 5875 \text{ rad/s} \\
  t_5 \left( \omega_{m5}^2 \right) &= 62337707.12; \omega_{m5} = 7895.42 \text{ rad/s} \\
  t_6 \left( \omega_{m6}^2 \right) &= 81862357.14; \omega_{m6} = 9047.78 \text{ rad/s}
\end{align*}
\]

Since the system is optimised for \( \omega_{m2} \) (1222785.76≈1225000), we will find out the damping coefficient for \( \omega_{m3} \) (\( \omega_{m4}, \omega_{m5} \) and \( \omega_{m6} \) are way beyond engine operating range). Considering 6th order of vibration the critical speed can occur at 5455 rpm corresponding to \( \omega_{m3} \).

The value of damping coefficient for \( \omega_{m3} \) is obtained from equation 4, \( K_d = 0.25 \). The damping coefficient drops from 0.76 to 0.25 and there is a substantial increase in the magnification factor. The effect of different values of damping coefficient on peak values of the magnification factor for different mass ratios is depicted below.

Though the vibratory amplitude is not greatly affected by departures from optimum value of damping coefficient, we can attain optimum value at all frequencies and orders with the use of Magneto-rheological fluids.

As mentioned earlier these fluids change their apparent viscosity when an external magnetic field is applied. By varying the viscosity we can achieve optimum specific damping torque and optimum damping coefficient at all orders and frequencies. We now shift our discussion towards MR fluid technology & the various factors associated with it.
The behavior of MR fluid under applied external field can be described by the Bingham plastic model.

The chains of particles resist to a certain level of shear stress and break when the shear stress exceeds a critical value called apparent yield stress of the material. The yield stress is important in many applications and depends on volume fraction of magnetic particles, particle distribution and applied magnetic field.

IV. MR FLUID COMPONENTS & MODES OF OPERATION

MR fluids are basically non colloidal suspensions of micro sized magnetisable particles in an inert base fluid along with some additives. Thus there are basically three components in an MR fluid:-

A. Base fluid
B. Metal particles
C. Additives

A. Base fluid
The base fluid is an inert or non magnetic carrier fluid in which the metal particles are suspended. The base fluid should have natural lubrication and damping features. For better implementation the base fluid should have low viscosity and it should not vary with temperature. Commonly used base fluids are hydrocarbon oils, mineral oils and silicon oils.

B. Metal particles
Metal particles are used for easy and quick magnetization. Commonly used metal particles are carbonyl iron, powder iron and iron cobalt alloys with an approximate size of the order of 1µm to 10µm. Metal particles of these materials have the property to achieve high magnetic saturation due to which they are able to form a strong magnetizing chain. The concentration of magnetic particles in base fluid can go up to 50%.

C. Additives
A frequent problem in MR fluids is the tendency of the active magnetic particles to aggregate and settle down, disturbing the homogeneity of the MR fluid. Surfactants are used to eliminate or decrease this sedimentation effect. Stabilizers are also used as additives for controlling the viscosity of the fluid, maintaining friction between metal particles and to reduce the rate of thickening of the fluid for prolonged usage.

The mode of operation of MR fluids is based according to the type of fluid flow. These correspond to the various applications for which they are being currently used. Following are the three modes of MR fluid operation:-

A. Valve mode
B. Shear mode
C. Squeeze mode

A. Valve mode
In this mode of MR fluid operation, fluid flows through the two fixed surfaces and magnetic field is applied perpendicular to the direction of flow. This mode of MR fluid technology is used in various types of dampers and shock absorbers in the automobile industry.

B. Shear mode
In this mode, fluid flows between surfaces when one plate is stationary and the other moves relative to it as shown in Fig.6. Shear mode is used in various types of brakes and clutches of vehicles.

C. Squeeze mode
This mode is used for low motion and high force applications. In this mode of MR fluid technology externally applied force is absorbed with the help of MR fluid. The yield stress developed through this mode is approximately ten times of the stress developed in either valve or shear mode.
V. MATHEMATICAL MODELS OF MR FLUIDS

Mathematical models are represented by a mathematical function whose coefficients are determined rheologically, i.e. the parameter values are adjusted until the quantitative results of the model closely match the experimental data.

Thus, the dynamic response of MR fluid devices is reproduced by a semi-empirical relationship. The two widely accepted models of MR fluid behavior are as follows:

- **Bingham model**
  
  An ideal Bingham body behaves as a solid until a minimum yield stress $\tau_0$ is exceeded and then exhibits a linear relation between the stress and rate of shear or deformation. Accordingly the shear stress developed in the fluid is given by
  \[
  \tau = \tau_0 + \mu \gamma 
  \]
  where $\tau_0$ is yield stress, $\mu$ is the viscosity and $\gamma$ is the shear rate. The onset of flow does not occur until the shear stress exceeds the yield stress (i.e., for $\tau < \tau_0 = \gamma = 0$).

- **Bouc-Wen model**
  
  The Bouc-Wen model is also an effective tool in characterizing the behavior of MR fluids. It is supposed to reproduce the response of hysteretic systems to random excitations.

VI. SUPER STRONG MR FLUIDS

Since the discovery of MR fluids, an investigation of the maximum yield stress obtainable through different possible concentrations and magnetic field is ongoing. However, the efforts have achieved limited results. For example, under a magnetic field of $H=398 \text{ kA} \text{m}^{-1}$ (or $B = 0.5 \text{ T}$), a new iron-cobalt MR fluid at 25% volume fraction produces a yield shear stress of 80 kPa, while a carbonyl iron MR fluid at 25% volume fraction has a yield shear stress of 60 kPa. Even after taking magnetic saturation of iron (2.1T) into account, an obtained performance curves as represented in Fig.8 are compared with the theoretical model.

An alternative to the Bingham model is the Herschel-Bulkley model which accounts for the post-yield shear thinning behavior of MR fluids. The Herschel-Bulkley model can be expressed as
  \[
  \tau = \tau_0 + K \gamma^{1/m} 
  \]
  where $K$ and $m$ are fluid parameters. For $m > 1$ the above equation represents shear thinning fluid while shear thickening fluids are described by $m < 1$. Note that for $m = 1$ the Herschel-Bulkley model reduces to Bingham plastic model.
CONCLUSION

Magneto-rheological fluid technology has a wide scope in the coming era. This technology is very useful in those places where controlled fluid with varying viscosity is required. The pivotal motive of this paper is to demonstrate the benefits of replacing silicon fluids by magneto-rheological fluids in torsional vibration dampers. The working and modeling of magneto-rheological fluid technology is also explained to support the claim. The main features of MR fluid technology are fast response, simple interface between electrical input and mechanical output and intelligent controllability. Hence, it can be interfaced with the vehicle’s electronic control unit for optimum damping over the entire range of operation. The main drawbacks of MR fluid technology are its high cost and fluid thickening due to prolonged use. There is a vast scope of research in MR fluid technology. An improvement in the life span of MR fluid devices along with sensor and feedback system would make it the smart technology of future.

REFERENCES