Dynamic characteristics of Non Newtonian fluid Squeeze film damper

C.P. Palaksha¹, S.Shivaprakash², H.P.Jagadish³

¹Research Scholar and PG Student, Department of Mechanical Engineering, New Horizon College of Engineering, Bangalore-560103, India Email: palakshacp@gmail.com
²Assistant Professor, Department of Mechanical Engineering, New Horizon college of College of Engineering, Bangalore – 560103, India Email: shivaprakashgttc@gmail.com
³Associate Professor, Department of Mechanical Engineering, BMS Evening College of Engineering, Bangalore –560 019. India, Email: jagadishhawaldar@yahoo.com

Email: jagadishhawaldar@yahoo.com

Abstract. The fluids which do not follow linear relationship between rate of strain and shear stress are termed as non-Newtonian fluid. The non-Newtonian fluids are usually categorized as those in which shear stress depends on the rates of shear only, fluids for which relation between shear stress and rate of shear depends on time and the visco inelastic fluids which possess both elastic and viscous properties. It is quite difficult to provide a single constitutive relation that can be used to define a non-Newtonian fluid due to a great diversity found in its physical structure. Non-Newtonian fluids can present a complex rheological behaviour involving shear-thinning, viscoelastic or thixotropic effects. The rheological characterization of complex fluids is an important issue in many areas. The paper analyses the damping and stiffness characteristics of non-Newtonian fluids (waxy crude oil) used in squeeze film dampers using the available literature for viscosity characterization. Damping and stiffness characteristic will be evaluated as a function of shear strain rate, temperature and percentage wax concentration etc.

Keywords: Squeeze film damper, non-Newtonian fluid, Temperature effect, viscosity, Damping, Stiffness characteristics.

1. Introduction

A fluid can be defined as a material that distort continually under the application of an external force. A key factor that distinguishes the performance of fluids is viscosity because it relates the neighbouring stresses in a moving fluid to the rate of deformation of the fluid constituent [1, 2]. When a fluid is sheared, it begins to move at a rate of deformation inversely proportional to viscosity [3, 4]. There are a certain class of fluids, called non-Newtonian fluids, in which the viscosity differs with the rate of shear. There are various types of non-Newtonian fluids. Pseudo plastic fluids are those fluids for which viscosity decreases with increasing shear rate and hence are often mentioned to as shear-thinning fluids. When the viscosity increases with rate of shear, the fluids are referred to as shear-thickening fluids. These fluids are less common than with pseudo plastic fluids. For non-Newtonian fluids, more detailed constitutive equations, comprising several material considerations, are needed to define the reaction of these fluids to intricate, time-dependent flows. Currently successful theories are either restricted to very definite, simple flows, especially generalizations of simple shear flow or extensional flow, for which rheological data can be used to develop practical models. Availability of literature is also very less and several researchers are interested only on the rheology of non-Newtonian fluids, and there is a limited coverage of this fluid as
used in squeeze film damper [5, 6, 7, 8, 9, 10, 11, 12]. Most of the researchers interested on the effects of temperature, rate of shear, wax concentration and pressure on the viscosity. However, it is normally found that the effect of temperature is much more obvious on the fluid viscosity. Attempt is made to use the properties of non-Newtonian fluids in squeeze film damper. The paper analyses the damping and stiffness characteristics of non-Newtonian fluids (waxy crude oil) used in squeeze film dampers using the available literature[3] for viscosity characterization and will be evaluated as a function of shear strain rate, temperature and percentage of wax concentration.

2. Viscosity model for waxy oil

A universal model appropriate for defining the non-Newtonian performance of waxy crude oil has been developed [3]. This model predicts viscosity as a function of rate of shear, temperature and percentage of wax concentration.

\[
\mu = \frac{B_1}{\gamma} \left[\left(\frac{\gamma + A_1}{A_2}\right)^n - 1\right]^\frac{1}{n} e^{\left(\frac{C}{\gamma + D - W}\right)}
\]  

(1)

Where \(\gamma\) is the shear rate, \(T\) is the temperature, \(w\) is the percentage of wax and \(\mu\) is the viscosity, \(n_1\) is the model parameter. When \(n_1=1\) it describes the Newtonian behaviour and when \(n_1=2\), it describes hyperbolic fluid behaviour. \(A, B, C\) and \(D\) are the model parameters.

For arriving at Equation (1), a general model which relates the shear stress to the rate of shear is as shown in equation (2) is used.

\[
\tau = B\left(\left(\frac{\gamma + A}{A}\right)^n - 1\right)\frac{1}{n}
\]

(2)

Relating the effect of temperature and the rate of shear Equation (2) gets the following form

\[
\mu = \frac{B_1}{\gamma} \left[\left(\frac{\gamma + A_1}{A_2}\right)^n - 1\right]^\frac{1}{n} e^{\left(\frac{C}{\gamma + D - W}\right)}
\]

(3)

Correlating the effects of the rate of shear, temperature and percentage of wax concentrations into Equation (2), Equation (1) is obtained. The model parameters i.e \(A, B, C, D\) and \(n\) can be calculated by least square nonlinear regression analysis. This equation is used for analysing the damper characteristics by keeping rate of shear, temperature, wax concentration and combination of these parameters constant.

3. Introduction to squeeze film dampers with orbital motion

Squeeze film dampers (Figure 1) find application in the air craft engine rotors and are usually mounted on rolling element bearings, with little or no inherent damping to fight the unbalance response, especially at critical speeds and to eliminate rotor dynamic instabilities. The coordinates of the orbit are presented in Equation (4) [13].

![Figure 1. Squeeze film dampers with orbital motion](image)
The radial and tangential forces acting on the system by neglecting stiffness terms are

\[ F_r = C_{yy} V_y + C_{yr} V_t = C_{yr} \omega e \]
\[ F_t = C_{ty} V_y + C_{tt} V_t = C_{tt} \omega e \] (5)

The direct damping coefficient is

\[ C_{tt} = \frac{F_t}{\omega e} \] (6)

And the cross-coupled damping coefficient is

\[ C_{rt} = \frac{F_r}{\omega e} \] (7)

The radial component of the force, \( F_r \) is considered as a stiffness force defined as the direct stiffness coefficient given by

\[ K_{rr} = \frac{F_r}{\omega e} = C_{rt} \omega \] (8)

The squeeze film velocity is given by

\[ \frac{\partial h}{\partial t} = e \omega \sin \theta \] (9)

For the present case with zero spin velocity, Reynolds equation reduces to

\[ \frac{\partial}{\partial x} \left[ h^3 \frac{\partial p}{\partial x} \right] + \frac{\partial}{\partial z} \left[ h^3 \frac{\partial p}{\partial z} \right] = 6 \eta U \frac{\partial h}{\partial x} + 12 \mu \frac{\partial h}{\partial t} \] (10)

The governing Reynolds's equation for the long bearing case with zero spin velocity is

\[ \frac{1}{r^2 \sin \theta} \left[ h^3 \frac{\partial p}{\partial \theta} \right] + h^3 \frac{\partial^2 p}{\partial x^2} = 6 \mu \omega \frac{\partial h}{\partial \theta} + 12 \mu \frac{\partial h}{\partial t} \] (11)

The long bearing approximation is applicable in cases in which the end seals are absent. Considering the long bearing approximation in which the journal axis parallel to the bearing for the steady state operating condition, the change in pressure in the circumferential direction is negligible in relation to the pressure change in axial direction.

Using the film thickness Equation and ignoring the pressure distribution in the diverging wedge, that is, using the half Somerfield condition, one gets

\[ P_\theta - P_0 = \frac{6 \mu \omega \rho^2}{C^2} + \frac{n(2 + \cos \theta) \sin \theta}{(2 + n^2)(1 + \cos \theta)^2} = 0 \quad 0 \leq \theta \leq \pi, \pi \leq \theta \leq 2\pi \] (12)

Considering the equilibrium of the journal, the load capacity can be obtained from the integration of the following equations.
\[
\int_0^\pi (P_\theta \cos \theta)Lr d\theta + W \cos \alpha = 0 \quad (13)
\]
\[
\int_0^\pi (P_\theta \sin \theta)Lr d\theta - W \sin \alpha = 0 \quad (14)
\]

Where
\[
W \cos \alpha = \frac{12\mu_0 L r^3 n^3}{C^3(2+n^2)\sqrt{1-n^2}} \quad (15)
\]
\[
W \sin \alpha = \frac{6\mu_0 L r^3 n\pi}{C^2(2+n^2)\sqrt{1-n^2}} \quad (16)
\]

Capacity of load carrying is given by, \( W \) is
\[
W = \frac{6\mu_0 L r^3 n\pi}{C^2(2+n^2)\sqrt{1-n^2}} \quad (17)
\]

The radial and the tangential force generated in the damper are components of the load capacity and are defined as
\[
F_r = W \cos \alpha = \frac{24\mu_0 L r^3 n e}{C^3(2+n^2)(1-n^2)} \quad (18)
\]
\[
F_t = W \sin \alpha = \frac{12\mu_0 L r^3 n\pi}{C^2(2+n^2)(\sqrt{1-n^2})} \quad (19)
\]

The Direct stiffness, direct damping and cross-coupled damping coefficients are respectively
\[
K_{dl} = \frac{24\omega\mu_0 L n r^3}{C^3(2+n^2)(1-n^2)} \quad (20)
\]
\[
C_{dl} = \frac{12\mu_0 L \pi r^3}{C^3(2+n^2)\sqrt{1-n^2}} \quad (21)
\]
\[
C_{rdl} = \frac{24\mu_0 L n r^3}{C^3(2+n^2)(1-n^2)} \quad (22)
\]

### 3.1 Mathematical analysis of long squeeze film damper

Dynamic characteristics of long squeeze film damper such as direct stiffness coefficient, direct damping coefficient and cross coupled damping coefficient are obtained using Equations 20, 21 and 22 respectively for different damper configurations as given in Table 1 and using the viscosity model (Equation 1) and Table 2 respectively.

#### Table 1. Damper specifications

| Description       | Parameters |
|-------------------|------------|
| Clearance c, mm   | 0.1        | 0.2       |
| Length, L, mm     | 30         | 40        |
| Diameter D mm     | 100        | 100       |
| L/D ratio         | 0.3        | 0.4       |
| Excitation Frequency \( \omega \) rad/sec | 100        | 100       |
| Eccentricity Ratio, n | 0.1        | 0.1       |
Table 2. Predicted parameters of waxy oil [3]

| Parameters          | Values          |
|---------------------|-----------------|
| A (s⁻¹)             | 8.9041*10⁴      |
| B (Pa)              | 2.6288*10⁻⁷     |
| C (K)               | 6720.71         |
| D (wt %)⁻¹          | 6.75*10⁻²       |
| n₁                  | 1.353           |

4. Results and discussions

The Dynamic characteristics of long squeeze film damper using non-Newtonian fluid as damping fluid are presented in this section for the damper specifications presented in Table 1 and fluid specifications presented in Table 2. Direct damping, direct stiffness, and cross coupled damping coefficients are evaluated for the following conditions:

1. Variable shear strain rate with Wax concentration percentage and Temperature constant
2. Variable wax concentration percentage with shear strain rate and Temperature constant
3. Variable temperature with shear strain rate and wax concentration percentage constant

The effect of shear strain rate (T=24°C, w=1%) on the viscosity of the non-Newtonian fluid is illustrated in Figure 2. The viscosity decreases with the rate of shear as it is a shear thinning fluid and is prominent in the initial stage and gradually smoothens off at higher rate of shear and more or less remains constant with rate of shear.

![Figure 2. Apparent Viscosity, \(\mu_{app}\) (Pas) vs. shear strain rate (s⁻¹)](image)

The Direct Stiffness coefficient (\(K_{d1}\)), damping coefficient and cross coupled damping coefficients (\(C_{ct}\)) (T=24°C, w=1%) decrease with the shear rate as is evident from Figure 2 in proportion to the decrease in viscosity of the fluid (Figure 2). The fluid particles break down at high shear rates and consequently the viscosity decreases leading to a corresponding effect on these coefficients.
The viscosity variation with wax concentration \((T=24^\circ C, \dot{\gamma}=10 \text{ s}^{-1})\) follows an increasing trend as the wax particles bind the fluid particles together resulting in increased viscosity at higher concentrations as outlined in Figure 4.
The opposite effect is observed on the Direct Stiffness coefficient ($K_{dl}$), damping coefficient and cross coupled damping coefficients ($C_{rtl}$) and increases when the wax concentration ($T=24^\circ C$, $\dot{\gamma}= 10\ s^{-1}$) increases in the fluid as this increases the viscosity (Figure 4) and consequently these coefficients increase in proportion to this increase in viscosity and is illustrated in Figure 5.

![Figure 4. Apparent Viscosity, $\mu_{app}$ (Pas) vs. Wax percentage (%)](image)

![Figure 5. (a) Direct Stiffness coefficient ($K_{dl}$) vs. Wax percentage (%)
(b) Direct damping coefficient ($C_{dl}$) vs. Wax percentage (%)
(c) Cross coupled damping coefficient ($C_{rtl}$) vs. Wax percentage (%).](image)
The effect of temperature on viscosity ($\omega=1\%$, $\dot{\gamma}=10 \text{ s}^{-1}$) is highlighted in Figure 6 and follows the normal pattern of exponential decrease as the temperature breaks the bond between the fluid particles and consequently decreases the viscosity.

Figure 6. Apparent Viscosity, $\mu_{\text{app}}$ (Pas) vs. Temperature ($^\circ$C)

The Direct Stiffness coefficient ($K_{\text{dl}}$), damping coefficient and cross coupled damping coefficients ($C_{\text{rtl}}$) ($T=24^\circ$C, $\omega=1\%$) decrease with the temperature (Figure 7) in accordance with the viscosity decrease observed in Figure F6.

Figure 7. (a) Direct Stiffness coefficient ($K_{\text{dl}}$) vs. Temperature ($^\circ$C)  
(b) Direct damping coefficient ($C_{\text{dl}}$) vs. Temperature ($^\circ$C)  
(c) Cross coupled damping coefficient ($C_{\text{rtl}}$) vs. Wax percentage (%)
These coefficients increase with decrease in the clearance and L/D ratio and increase in eccentricity ratio for all cases. Thus the damper with specifications, c=0.1 mm, L/D=0.40, n=0.10 achieves the best performance and c=0.2 mm, L/D=0.3, n=0.05 records least performance as it is evident from the plots. It is also evident that the direct damping coefficient is unaffected by small changes in eccentricity ratios as illustrated in Figures 3 (b), 5 (b) and 7 (b).

5. Conclusions
1. The effect of rate of shear, percentage of wax concentration and Temperature on Dynamic characteristics of non-Newtonian fluid long squeeze film damper is presented in this paper.
2. These coefficients reduce with the shear rate and temperature as the fluid particles are broken down at high shear rates and temperatures leading to reduction in viscosity. However, these characteristics increase with increase in wax concentration as the fluid binds together firmly at higher concentrations resulting in enhanced viscosity. For a particular damper configuration, these coefficients increase with decrease in clearance, increase in L/D ratio and eccentricity ratio.
3. The direct damping coefficient is found to be unaffected by small changes in eccentricity ratios and the damper with same clearance and L/D ratio shows identical performance irrespective of the eccentricity ratio of the damper.
4. Non-Newtonian fluids are found to be greatly useful in damping the synchronous vibrations of shaft as these fluids obviates the need of end seals in the damper and can be modelled as per the requirement of the industry.
5. The optimum amount of damping can be obtained by controlling the parameters used in the study.

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