DYNAMIC CHARACTERISTICS OF MR FLUID SHORT SQUEEZE FILM DAMPER IN TERMS OF REYNOLDS NUMBER

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Abstract

Magnetorheological (MR) fluids are suspensions of fine micron-sized particles suspended in an appropriate carrier medium. Their rheological properties can be controlled by the application of an appropriate magnetic field and can be used in a variety of applications where damping and stiffness characteristics need to be controlled at a particular excitation frequency, based on the requirements. This cannot be accomplished with conventional fluids and thus MR fluids find application in squeeze film dampers of aircraft jet engines to provide variable damping in accordance with the magnetic field, at a particular excitation frequency. Limited information is available on the stiffness and damping characteristics of magnetorheological fluid squeeze film dampers in the literature. This paper provides information on the stiffness and damping characteristics of MR fluids used as external damping medium in Squeeze film Dampers in terms of the Reynolds number of the squeeze film. The paper calibrates the stiffness and damping characteristics of particular magnetorheological fluid squeeze film damper theoretically in terms of Reynolds number of the squeeze film for two clearance and L/D ratios operating at low eccentricity ratios, using a constant field viscosity model. The stiffness and damping characteristics are found to decrease with the increase in Reynolds number. The Reynolds number of the squeeze film is very low, highlighting the fact that the flow in the film has ceased and has solidified in accordance with the literature. The dynamic coefficients are presented in the form of empirical equations based on the theoretical investigations for different clearance, L/D ratio and eccentricity ratio, in terms of Reynolds number of the damper and enables its easy evaluation for a particular damper configuration by a mere knowledge of the Reynolds number of the squeeze film. The results assist the designer in obtaining the stiffness and damping characteristics of the squeeze film damper, based on the Reynolds number. Alternatively, the stiffness and damping characteristics of the squeeze film damper are calibrated in terms of Reynolds number for particular damper configurations. However, the research focuses on the stiffness and damping coefficients for a limited number of damper configurations under a constant excitation frequency whilst a large number of clearances, L/D ratios and excitation frequencies have not been taken into account .This research is thus intended only to showcase the method of approach by selecting finite number of damper configurations. Future work should be focused towards investigations that present the empirical equations for the entire range of clearance, L/D ratio and eccentricity ratio.

Keywords: *squeeze film damper, magnetorheological fluid, Reynolds Number, short bearing*

Nomenclature

A = direct stiffness constant, *GN/m*

B = direct damping constant, *GNs/m*

c = clearance in the damper, *m*

C_{rrs} = short bearing radial damping coefficient, *GNs/m*

C_{rts} = short bearing cross-coupled damping coefficient, *GNs/m*

C_{trs} = short bearing cross-coupled damping coefficient, *GNs/m*

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Manuscript received on 13 Jan 2010; Paper reviewed and accepted as a Full Length Contributed Paper on 29 Mar 2010

C_{tts}=C_{ds} = short bearing tangential or direct damping coefficient, *GNs/m*
D = diameter of the damper, *m*
e = eccentricity or whirl radius, *m*
E = cross coupled damping constant, *GNs/m*
F_r = radial force in the damper, *N*
F_t = tangential force, *N*
H = magnitude of the applied magnetic field, Tesla
K = fluid parameters and K > 0
K_{ds} = short bearing direct stiffness coefficient, *GN/m*
L = length of the damper, *m*
m = fluid parameters and $m > 0$
n = eccentricity ratio = *e/c*
p = pressure at angular location, θ^o
P = viscosity model constants
Q = viscosity model constants
r = damper radius, *m*
$\dot{r}$ = whirl velocity, *m/s*
Re = Reynolds number of the squeeze film
sgn $(\dot{\gamma})$ = signum function
t = time, *s*
U = surface velocity, *m/s*
V_r = radial whirl velocity, *m/s*
V_t = tangential whirl velocity, *m/s*
W = load Capacity of the damper, *N*
x,y,z = coordinate axes
τ = total shear stress, N/m^2 (*Pa*)
υ = kinematic viscosity, m^2/s
η = field independent plastic viscosity, Ns/m^2
ω = excitation frequency, rad/s
ρ = density, Kg/m^3
α = attitude angle, degree
θ = angular coordinate of the damper, degrees
τ_o = yield stress caused by the applied field, N/m^2 (*Pa*)
$\dot{\gamma}$ = shear strain rate, s^{-1}
η_e = effective viscosity, Ns/m^2
η_{app} = apparent viscosity due to the field, Ns/m^2

Introduction

The most commonly encountered problems in rotor dynamics are the unnecessary steady state synchronous vibration levels and sub synchronous rotor instabilities. The first problem may be minimized by improved balancing, or by introducing simplifications into the rotor-bearing system to move the system critical speeds out of the operating range, or by introducing external damping to limit peak amplitudes at traversed critical speeds. Sub-synchronous rotor instabilities may be avoided by eliminating the instability mechanism, by increasing the natural frequency of the rotor-bearing system as much as feasible, or by introducing damping to raise the onset speed of instability. Lightweight, high performance engines exhibit a trend towards increased flexibility leading to high sensitivity to unbalance with large vibration levels and reduced reliability. Squeeze film dampers (SFDs) are essential components of high-speed turbo machinery devices as they offer the distinctive advantages of dissipation of vibration energy and isolation of structural components, as well as the capability to improve the dynamic stability characteristics of naturally unstable rotor-bearing systems. SFDs are used primarily in aircraft jet engines to provide viscous damping to anti-friction bearings which themselves have little or negligible inherent damping. The basic limitation of a conventional squeeze film damper is that the damping is a function of the rotor speed, which, in turn, depends on the viscosity of the damping medium. In order to satisfy the variable damping requirement at a particular excitation frequency, the viscosity of the damper fluid has to be variable [1].This requirement cannot be met by conventional oils and hence, Magnetorheological Fluids find applications in situations which require variable damping. These fluids are suspensions of fine micron size particles dispersed in a suitable carrier medium whose rheological properties can be varied in conjunction with the magnetic field. The particles undergo dipole magnetization under the influence of the field and gets oriented parallel to the direction of the field. Thus, these materials act as solids as long as the shear stress is lower than a threshold value which depends on the field strength and as quasi-Newtonian fluids if the shear stress is higher. The change induced in the field shear stress produces a variation in their apparent viscosity and is successfully used in applications involving valves, dampers and clutches and adaptive structures [2, 3], but industrial applications are limited and require considerable attention. Research should thus be focused towards their applications in rotor dynamics. Very limited information is available regarding the use of MR fluids and their stiffness and damping characteristics in active journal bearings and squeeze film dampers. While general aspects and applications of MR fluids have been dealt with to some extent [4, 5], sharing their promising characteristics, it was only in the last decade that some papers have appeared on Squeeze film Dampers [6, 7]. No information is available regarding the stiffness and damping characteristics of MR fluids used in squeeze film dampers and its relation to the Reynolds number of the squeeze film.

This paper relates the stiffness and damping characteristics of the squeeze film damper in terms of Reynolds number of the squeeze film.The flow characteristics of the fluid are thus specified in terms of the stiffness and damping characteristics of the squeeze film damper. The investigation focuses on the effect of Reynolds number of the squeeze film on the stiffness and damping characteristics of a short Squeeze film damper based on the short bearing approach and a constant field viscosity model [8]. Alternatively, the stiffness and damping characteristics of the squeeze film operating at low eccentricity ratios are presented in terms of Reynolds number of the squeeze film, for two clearance and L/D ratios, as outlined in Table-1.

Magnetorheological Fluid

M R Fluids

Magneto-rheological fluids are a class of controllable fluids, in that, their rheological properties; especially viscosity can be controlled in conjunction with the magnetic field. These are suspensions of micron-sized, magnetizable particles in an appropriate carrier liquid. The important feature of MR fluids is their ability to change from free-flowing, linear viscous liquids in milliseconds to semi-solids having controllable yield strength when exposed to an external magnetic field.

MR Fluid Model

MR fluids are represented by a simple Bingham visco-plasticity model as depicted in Fig.1, describing the essential field-dependent fluid characteristics. This model gives the total shear stress τ

$$\tau = \tau_o (H)\, Sgn (\dot{\gamma}) + \eta \dot{\gamma} \tag{1}$$

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, ω (rad/sec)	100	100
Eccentricity Ratio, n	0.1	0.1
Range of Shear Strain Rate, $\dot{\gamma}$ (s^{-1})	2 to 10	
Constant Magnetic field intensity, H (Tesla)	0.4	

where τ_o is the yield stress caused by the applied field, H is the Magnetic field intensity, $Sgn (\dot{\gamma})$ = Signum function, η is the field independent plastic viscosity and $\dot{\gamma}$ is the shear strain rate. The field independent plastic viscosity, η is the slope of the post-yield shear stress versus shear strain rate. The fluid post-yield viscosity is assumed to be a constant in the Bingham model. As MR fluids demonstrate shear thinning effect as illustrated in Fig.1, the Herschel-Bulkley visco-plasticity model can be employed to accommodate this effect. The constant post-yield plastic viscosity in the Bingham model is replaced with a power law model dependent on shear strain rate. Thus,

$$\tau = \left(\tau_o (H) + K |\dot{\gamma}|^{\frac{1}{m}} \right) Sgn\, \dot{\gamma} \tag{2}$$

It can be inferred by comparing eqn. (2) and (1) that the plastic viscosity of the Herschel Bulkley model is

$$\eta_e = K |\dot{\gamma}|^{\frac{1}{m} - 1} \tag{3}$$

This indicates that the equivalent plastic viscosity, η_e decreases as the shear strain rate, $\dot{\gamma}$ increases when m > 1 (Shear thinning).This model can also be used to describe the fluid shear thickness effect when m < 1. The Herschel-Bulkley model reduces to the Bingham model when m = 1, therefore, η_e=K.

The apparent viscosity, η_{app} of the MR Fluid in flow is the instantaneous change in viscosity, due to the magnetic field and is defined as the ratio of shear stress and the shear strain rate. Referring to equation [1], the apparent viscosity has been defined in the literature [8, 9] as detailed below

$$\eta_{app} = \frac{\tau}{\dot{\gamma}} = \frac{\tau_y}{\dot{\gamma}} + \eta_o \tag{4}$$

The apparent viscosity is a function of the strain rate and the applied magnetic field. Accordingly, a viscosity model based on constant magnetic field can be proposed for the particular MR Fluid. This paper investigates the Squeeze Film Damper based on constant magnetic field viscosity model.

Constant Field Viscosity Model

A constant field viscosity model can be developed of the form

$$\eta_{app} = P\,\dot{\gamma}^{-Q} \tag{5}$$

where *P* and *Q* are constants for a particular fluid. The plot of apparent viscosity versus strain rate $\dot{\gamma}$ is the basis for the above model [8]. For a constant magnetic field intensity of 0.4 Tesla, an equation of the form outlined below is fitted to the data of MR fluid (MRF 132LD).

$$\eta_{app} = 10000\,\dot{\gamma}^{-0.9856} \tag{6}$$

Thus P=10000 and Q = -0.9856 for the fluid under consideration. The negative exponent in eqn. (5) indicates that the viscosity decreases with increase in the strain rate and is a characteristic feature of the shear thinning fluids. This equation forms the basis for the subsequent evaluation of the dynamic coefficients of the squeeze film damper.

Reynolds Number

Reynolds Number is defined as the ratio of the inertia forces to the viscous forces, that is

$$\mathrm{Re} = \frac{UL\rho}{\eta_{app}} = \frac{UL}{\upsilon}\,. \tag{7}$$

where U, L, ρ, υ and η are characteristic values of the velocity, length, density, kinematic viscosity and dynamic viscosity of the fluid respectively. If Re is small, the viscosity forces will be predominant and the effect of viscosity will be felt in the whole flow field. On the contrary, if Re is large, the inertial forces will be predominant and in such a case the effect of viscosity can be considered to be restricted in a thin layer, adjacent to a solid boundary. However, if Re is very large the flow ceases to be laminar and becomes turbulent. The Reynolds number at which the transition from laminar to turbulent occurs is known as critical Reynolds number. The paper evaluates the dynamic characteristics based on low Reynolds number and inertia forces are thus neglected in this case.

The Reynolds number for the squeeze film is given by

$$\mathrm{Re} = \frac{\rho\,\omega\,c^2}{\eta_{app}} \tag{8}$$

Squeeze Film Damper with Orbital Motion

MR Squeeze film dampers are relatively new devices that utilize MR fluids to mitigate small amplitude, large force rotor dynamic vibrations. They find applications in aircraft engine rotors and are usually supported on roller bearings which have very little inherent damping. External dampers in the form of squeeze film dampers are incorporated in order to provide damping at the critical speeds and to eliminate rotor dynamic instabilities (Fig.2). A Squeeze Film Damper (SFD) consists of an inner non-rotating journal and a stationary outer bearing with a small clearance for the damping fluid. Fig.2 shows an idealized schematic of this type of fluid film bearing. A journal is mounted on the external race of an anti-friction bearing and prevented from rotation with loose pins or a squirrel cage that provides a centering elastic mechanism. The annular thin film between the journal and housing is filled with a lubricant supplied as a splash from the rolling bearing elements lubrication system or by a pressurized delivery. The journal moves due to dynamic forces acting on the system, the fluid is displaced to accommodate these motions. As a result, hydrodynamic squeeze film pressures are developed in the clearance space which exert reaction forces on the journal and provide for a mechanism to minimize transmitted forces and the rotor amplitude of motion. The orbit of the damper is nearly circular for small amplitudes of motion of the damper. The majority of Squeeze films Dampers include the arrangement where the journal (and rolling element bearing) is spring-supported, and the bearing is firmly attached to the engine frame. The (soft) spring support and squeeze film damper experience the same deflections though the dynamic loads divide unequally between them. Dampers in jet engines function at low pressure levels (2 or 3 bars) to avoid extreme load and volume in the lubrication system. Most aircraft engines do not use any kind of hydrodynamic journal bearings to prevent the risk of fluid film bearing induced instabilities, particularly due to the cross coupled stiffness coefficients. The end seals are used to avoid end leakage of the damping fluid.

The dynamic characteristics of the Squeeze film damper are the stiffness and damping coefficients. The principal or direct stiffness, K_{ds} is extremely important with respect to the machine's vibration performance and its magnitude, with respect to the shaft stiffness, governs the location and amplitude of the rotor's critical speeds and is equally important for stability purposes. It is a function of the excitation frequency and is sometimes referred to as the "frequency dependent stiffness".

The off-diagonal stiffness coefficients are the cross-coupled stiffness coefficients. Almost all structures have such cross-coupled stiffness terms but most are symmetric in nature. Rotor systems are unique in that this symmetry usually does not exist. Fundamentally, their presence and their asymmetry results from the various fluids rotating within the machine, such as oil in bearings and gas in labyrinth seals. Instead of opposing the rotors whirling motion like the direct damping, the cross-coupled stiffness combine to create a force pointing in the whirl direction, promoting the shaft vibration. Fortunately, these are absent in Squeeze film dampers as the oil film does not rotate and is the reason for the high stability exhibited by these dampers.

The damping characteristics in a Squeeze film damper are the direct or principal damping co-efficient C_{ds} (C_{tts}) and the cross-coupled damping co-efficient C_{rts}. Like their direct stiffness counterparts, the principal damping terms state much about the machine's unbalance response and stability and are often the main source of damping in the entire machine. However, their effectiveness in reducing the critical speed amplification factors and preventing sub-synchronous instabilities is determined also by the bearing's direct stiffness coefficients as well as the shaft stiffness. For the direct damping to be effective, the bearing cannot be too stiff because the damping force depends on journal motion. Also, contrary to one's initial instincts, too much damping can actually be detrimental. The direct stiffness coefficient, K_{ds}, direct damping coefficient, C_{ds}= C_{tts} and cross-coupled coefficient, C_{rts} indicate the stability aspects of the damper and their evaluation assumes significance in this context and are derived from a solution of the Reynolds equation (Eqn.8) for a non-rotating journal bearing assuming laminar flow, circular whirl orbit, π-film and an inertia less oil film, that is, low Reynolds number (10).

$$\frac{1}{r^2}\frac{\partial}{\partial\theta}\left[h^3\frac{\partial p}{\partial\theta}\right]+h^3\frac{\partial^2 p}{\partial x^2}=6\,\eta_{app}\,\omega\,\frac{\partial h}{\partial\theta}+12\,\eta_{app}\,\frac{\partial h}{\partial t} \tag{9}$$

The co-ordinates of the orbit in this case,

$$z = e\cos\,\omega t\,, \qquad y = e\sin\,\omega t \tag{10}$$

$$r = e\,, \qquad \dot{r} = V_r = 0\,, \quad V_t = e\,\omega \tag{11}$$

The instantaneous clearance 'h' at any angle θ from the vertical is given as

$$h = c\,(\,1 - n\cos\theta\,) \tag{12}$$

The radial and tangential forces for the short bearing approximation where the flow is unrestricted without the end seals are given by

$$F_r = C_{rrs}\,V_r + C_{rts}\,V_t = C_{rts}\,\omega e \tag{13}$$

$$F_t = C_{trs}\,V_r + C_{tts}\,V_t = C_{tts}\,\omega e = C_{ds}\,\omega e \tag{14}$$

The radial damping coefficient, C_{rrs} is defined as the ratio of the radial force generated due to the squeeze film action to the radial velocity, V_r of the film. Thus,

$$C_{rrs} = \frac{F_r}{V_r} \tag{15}$$

The direct damping coefficient, C_{ds} (C_{tts}) is defined as the ratio of the tangential force generated due to the squeeze film action to the tangential velocity, V_t of the film. Thus,

$$C_{ds} = C_{tts} = \frac{F_t}{\omega\,e} \tag{16}$$

The cross coupled damping coefficient, C_{rts} is defined as the ratio of the Radial force generated in the squeeze film to the tangential velocity, V_t of the film. Thus,

$$C_{rts} = \frac{F_r}{\omega\,e} \tag{17}$$

The cross coupled damping coefficient, C_{trs} is defined as the ratio of the tangential force generated in the squeeze film to the radial velocity, V_r of the film. Thus,

$$C_{trs} = \frac{F_t}{V_r} \tag{18}$$

The Direct stiffness coefficient, K_{ds} is derived from the radial force, F_r, and is defined as

$$K_{ds} = \frac{F_r}{e} = C_{rts}\,\omega \tag{19}$$

Thus the direct stiffness coefficient is a function of the excitation frequency ω of the damper. It is clear from the eqns. (14) and (17) that the radial damping coefficient, C_{rrs} and the cross coupled damping coefficient, C_{trs} do not exist in Squeeze film Dampers as the radial velocity V_r is zero. Hence these are not evaluated in this paper.

The Squeeze velocity of the film with whirl frequency ω is

$$\frac{\partial h}{\partial t} = e\,\omega \sin\theta \tag{20}$$

The governing Reynolds equation for this case with zero spin velocity from eqn. (8) is

$$\frac{1}{r^2}\frac{\partial}{\partial\theta}\left[h^3\frac{\partial p}{\partial\theta}\right] + h^3\frac{\partial^2 p}{\partial x^2} = 12\,\eta_{app}\, e\,\omega\, \sin\theta \tag{21}$$

The short bearing approximation is applicable in cases in which the end seals are absent. Considering the short 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. The Reynolds equation modifies to

$$h^3\frac{d^2 p}{d x^2} = 12\,\eta_{app}\, e\,\omega\, \sin\theta \tag{22}$$

The pressure distribution in the film is obtained by integration as

$$p = \frac{6\,\eta_{app}\,\omega}{h^3}\,\frac{dh}{d\theta}\,\frac{x^2}{2} + \frac{A}{h^3}x + B \tag{23}$$

The constants in the above equation are evaluated from the boundary conditions,

$$p = 0 \text{ at } x = \pm\frac{L}{2}$$

where the co-ordinate system of Fig.2 is reckoned midway along the length L of the bearing (perpendicular to the plane of the paper). The two constants are evaluated to give the pressure p as

$$p = \frac{3\,\eta_{app}\,\omega}{h^3}\,\frac{dh}{d\theta}\left(x^2 - \frac{L^2}{4}\right) \tag{24}$$

Using the film thickness eqn.(11) and ignoring the pressure distribution in the diverging wedge, that is, using the half Sommerfeld condition, we have

$$p = \frac{3\,\eta_{app}\,\omega n}{c^2}\left(x^2 - \frac{L^2}{4}\right)\frac{\sin\theta}{(1 - n\cos\theta)^3} \quad For\ 0 \le \theta \le \pi \tag{25a}$$

$$= 0 \text{ for } \pi < \theta < 2\pi \tag{25b}$$

Considering the equilibrium of the journal, the load capacity can be obtained from the integration of the following equations

$$2\int_0^{\pi}\int_0^{L/2} pr\,\cos\theta\; dx\, d\theta = -W\cos\alpha \tag{26a}$$

$$2\int_0^{\pi}\int_0^{L/2} pr\,\sin\theta\; dx\, d\theta = -W\sin\alpha \tag{26b}$$

where

$$W\sin\alpha = \frac{\pi\,\eta_{app}\,\omega\, D\, n\, L^3}{4\,c^2(1-n^2)^{3/2}} \tag{27}$$

$$W\cos\alpha = \frac{\eta_{app}\,\omega\, D\, n^2 L^3}{c^2(1-n^2)^2} \tag{28}$$

The load capacity W is

$$W = \frac{\eta_{app}\,\omega\, D\, n\, L^3}{4\,c^2(1-n^2)^2}\sqrt{16\,n^2 + \pi^2(1-n^2)} \tag{29}$$

The radial and the tangential force generated in the damper are components of the load capacity and are defined as

$$F_r = W\cos\alpha = \frac{\eta_{app}\,\omega\, D\, n^2 L^3}{c^2(1-n^2)^2} \tag{30}$$

$$F_t = W\sin\alpha = \frac{\pi\,\eta_{app}\,\omega\, D\, n\, L^3}{4\,c^2(1-n^2)^{3/2}} \tag{31}$$

The direct stiffness, direct damping and direct cross coupled damping coefficients are respectively

$$K_{ds} = \frac{\eta_{app}\,\omega\,D\,n\,L^3}{c^3\,(1-n^2)^2} \tag{32}$$

$$C_{ds} = \frac{\pi\,\eta_{app}\,D\,L^3}{4\,c^3\,(1-n^2)^{3/2}} \tag{33}$$

$$C_{rts} = \frac{\eta_{app}\,D\,n\,L^3}{c^3\,(1-n^2)^2} \tag{34}$$

These equations can be given in terms of the Reynolds number by eliminating the apparent viscosity term from eqn.(8)

$$\eta_{app} = \frac{\rho\,\omega\,c^2}{\mathrm{Re}}$$

Simplification yields

$$K_{ds} = \frac{\rho\omega^2\,D\,n\,L^3}{c\,(1-n^2)^2\,\mathrm{Re}} = \frac{A}{\mathrm{Re}} \tag{35}$$

$$C_{ds} = \frac{\pi\,\rho\,\omega\,D\,L^3}{4\,c\,(1-n^2)^{3/2}\,\mathrm{Re}} = \frac{B}{\mathrm{Re}} \tag{36}$$

$$C_{rts} = \frac{\rho\omega\,D\,n\,L^3}{c^3\,(1-n^2)^2\,\mathrm{Re}} = \frac{E}{\mathrm{Re}} \tag{37}$$

where the constants A, B and E are

$$A = \frac{\rho\omega^2\,D\,n\,L^3}{c\,(1-n^2)^2} \tag{38}$$

$$B = \frac{\pi\,\rho\,\omega\,D\,L^3}{4\,c\,(1-n^2)^{3/2}} \tag{39}$$

$$E = \frac{\rho\,\omega\,D\,n\,L^3}{c\,(1-n^2)^2} \tag{40}$$

The constant A in eqn. (35) represents the direct stiffness coefficient of the damper operating at unit Reynolds number and is a function of the excitation frequency, damper configuration and the damper fluid used.

The constant B in eqn. (36) represents the direct damping coefficient of the damper operating at unit Reynolds number and is a function of the excitation frequency, damper configuration and the damper fluid used.

The constant E in eqn. (37) represents the cross coupled damping coefficient of the damper operating at unit Reynolds number and is a function of the excitation frequency, damper configuration and the damper fluid used.

The theoretical squeeze film damping and stiffness characteristics for varying strain rates can be obtained using eqns.(35, 36 and 37) in conjunction with the constant field viscosity model developed in eqn. (6).The shear strain rate is varied in the range of 2 to 10 s^{-1} with the magnetic field constant at 0.4 Tesla as specified in Table-1. The MR fluid data and damper specifications with reference to Fig.2 used in the analysis are presented in Tables-1 and 2.

Results and Discussions

This section presents the analysis of the MR Fluid short squeeze film damper based on the specifications presented in Tables-1 and 2. Figs. 3 and 4 illustrate the variation of direct stiffness coefficient, K_{ds} as a function of the Reynolds number for constant field intensity of 0.4 Tesla. The shear strain rate is varied in the range of 2 to10 s^{-1} in the entire analysis. The stiffness variation is found to be hyperbolic, diminishing with increasing Reynolds number. It is evident from Figs. (3 and 4) that the maximum and minimum values of the stiffness coefficients are attained respectively with damper configurations c = 0.1 mm, L/D = 0.4, n = 0.2 and c = 0.2 mm, L/D = 0.3, n = 0.1. This indicates that the stiffness coefficient is enhanced by smaller clearance, higher L/D ratio and eccentricity ratio.

Equation (35) predicts infinite stiffness at zero Reynolds number and zero stiffness at infinite Reynolds number. Thus, the damper cannot be used in the lower range of Reynolds number as the rotor becomes too rigid

Table-2 : MR Fluid Specifications

Type	Manufacturer	Fluid Parameters		Density, ρ (Kg/m^3)
		m	K	
MRF 132 LD	Lord Corporation	2.86	4.202	3055

and is incapable of generating the required damping forces as the damper relies on the rotor movement to generate the damping forces. Similarly, the damper cannot be operated at higher Reynolds number as it yields a highly flexible rotor, unable to generate the required pressure forces to support the load. The damper has to be used at an optimal Reynolds number for its satisfactory performance, based on experience.

From Figs. 5 and 6, it is seen that the variation of direct damping coefficient, C_{ds} (C_{tts}) as a function of the Reynolds number for constant field intensity of 0.4 Tesla is again hyperbolic as is the case with the stiffness coefficient, decreasing with increasing Reynolds number . It is apparent from the Figs. (5 and 6) that the maximum and minimum values of the damping coefficient are obtained respectively with damper configurations c = 0.1 mm, L/D=0.4, n = 0.2 and c = 0.2 mm, L/D = 0.3, n = 0.1. The damping coefficient is enhanced by smaller clearance, higher L/D ratio and eccentricity ratio.

The direct damping coefficient is not influenced by eccentricity ratio for similar clearance and L/D ratio as is evident from Fig. (5), and the curves corresponding to the damper configuration c = 0.1mm, L/D = 0.3, n = 0.1 and c = 0.1mm, L/D = 0.3, n = 0.2; approximately show identical values. Similarly, the curves corresponding to damper configuration c = 0.1mm, L/D = 0.4, n = 0.1 and c = 0.1 mm, L/D = 0.4, n = 0.2 show identical values. The same result occurs in Fig.(6), in which the damper configurations corresponding to c = 0.2 mm, L/D = 0.3, n = 0.1 and c = 0.2 mm, L/D = 0.3, n = 0.2 and the damper configurations corresponding to c = 0.2 mm, L/D = 0.4, n = 0.1 and c = 0.2 mm, L/D = 0.4, n = 0.2 indicate nearly identical values. This is evident from Table-3 in which the constant B corresponding to these damper configurations indicate identical values. Eqn.(36) predicts infinite damping at zero Reynolds number, which is the case of "solidified" MR fluid under the influence of the field. Under these conditions, the damper acts as a rigid bearing, transmitting the vibratory forces to the mounting, rendering the damper ineffective. On the contrary, for very high Reynolds number, the damping coefficient decreases considerably, and the film is incapable of dissipating the vibration energy. Hence, care should be exercised in selecting the suitable magnetic field to be applied to the damper for its satisfactory performance.

Figures 7 and 8 depict the variation of cross coupled damping coefficient, C_{rts} as a function of the Reynolds number for constant field intensity of 0.4 Tesla. The cross coupled damping coefficient variation is found to be hyperbolic according to the eqn.(37), diminishing with increasing Reynolds number. It is evident from the Figs.(7 and 8) that the maximum and minimum values of the cross coupled damping coefficient are attained respectively with damper configurations c = 0.1 mm, L/D = 0.4, n = 0.2 and c = 0.2 mm, L/D = 0.3, n = 0.1. The cross coupled damping coefficient is thus enhanced by smaller clearance, higher L/D ratio and eccentricity ratio.

Figure 9 shows the variation of the apparent viscosity as a function of the shear strain rate in the range of 2 to 10 s^{-1} for a constant field intensity of 0.4 Tesla, as predicted by eqn.(6). The viscosity decreases significantly in the range from 5.052 kPas to 1.034 kPas with increase in the strain rate in accordance with the constant field viscosity model, behaving as a shear thinning fluid [8,9].

Table-3 : Values of Constants, A, B and E (A gives the direct stiffness coefficients in GN/m, B and E give the direct and cross coupled damping coefficients in terms of GNs/m)

Clearance, c (mm)	Ecc. Ratio, (n)	Length, L (mm)	Diameter, D (mm)	L/D Ratio	A	B	E
0.1	0.1	30	100	0.3	8.416E-05	6.570E-06	8.416E-07
	0.1	40	100	0.4	1.995E-04	1.559E-05	1.995E-06
	0.2	30	100	0.3	1.790E-04	6.888E-06	1.790E-06
	0.2	40	100	0.4	4.243E-04	1.633E-05	4.243E-06
0.1	0.1	30	100	0.3	4.208E-05	3.288E-06	4.208E-07
	0.1	40	100	0.4	9.975E-05	7.795E-06	9.975E-07
	0.2	30	100	0.3	8.950E-05	3.444E-06	8.950E-07
	0.2	40	100	0.4	2.122E-04	8.163E-06	2.122E-06

The values of the constants A, B and E for different configurations of the damper (specified in Table-1), are presented in Table-3, which gives the stiffness and damping coefficients in terms of GN/m and GNs/m respectively.

Conclusions

The investigations reveal that Magnetorheological Fluids can be effectively used in Squeeze film dampers to provide variable stiffness and damping, by varying the strain rate, in accordance with the requirement of the rotor dynamic system. These coefficients can be varied at a particular excitation frequency using Magnetorheological Fluids, which cannot be obtained with conventional oils. This is the advantage that Magnetorheological Fluids offer over conventional fluids. This property can be utilized in MR active squeeze film dampers in which the damping varies in accordance with the feedback signal received from the rotor movement. This signal, which is proportional to the deviation from the prescribed value, energizes the coils of the magnet proportionately to set up the required magnetic field to generate the required damping forces. This damper utilizes the unique advantages of the synergic effects of a controllable squeeze film damper and a magnetic damper. The Magnetorheological Fluid undergoes a drastic change in its apparent viscosity under the influence of strain rate, at a constant magnetic field. This change in viscosity, in turn, influences the stiffness, damping coefficients and the Reynolds number of the squeeze film. The Reynolds number of the squeeze film, consequently, is very low, indicating that the flow in the film has ceased and the film has solidified under the influence of the magnetic field [8,9]. The stiffness and damping coefficients increases with smaller clearance, higher L/D ratio and eccentricity ratios. The paper links the flow characteristics of the fluid to the damping and stiffness characteristics of the Magnetorheological fluid squeeze film damper and the designer can gain an insight into the theoretical dynamic characteristics possible with a particular damper configuration and vary the same in accordance with the requirement by controlling the Reynolds number of the squeeze film. The dynamic coefficients are presented in the form of empirical equations based on the theoretical investigations for different clearance, L/D ratio and eccentricity ratio, in terms of Reynolds number of the damper and enables its easy evaluation for a particular damper configuration by a mere knowledge of the Reynolds number of the squeeze film. The research focuses on only a finite number of damper configurations to showcase the method of approach and thus future work should be focused towards investigations that present the empirical equations for the entire range of clearance, L/D ratio and eccentricity ratio.

Acknowledgement

The authors would like to acknowledge the financial support received from Propulsion Panel, Aeronautics Research and Development Board (AR&DB), DRDO, New Delhi for the Research and Development sponsored Project, vide DRDO File No. DARO / 08 / 1041391 / M / I 28th Dec 2006.

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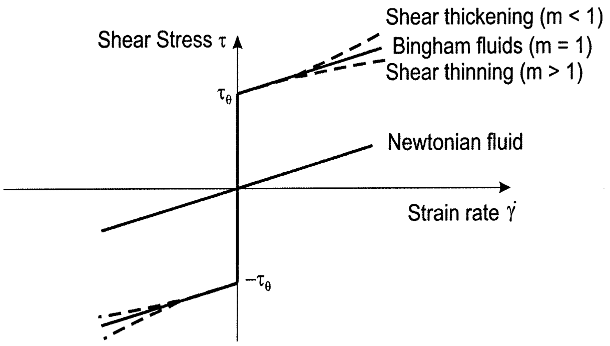

Fig.1 Viscoplasticity Model for MR Fluids

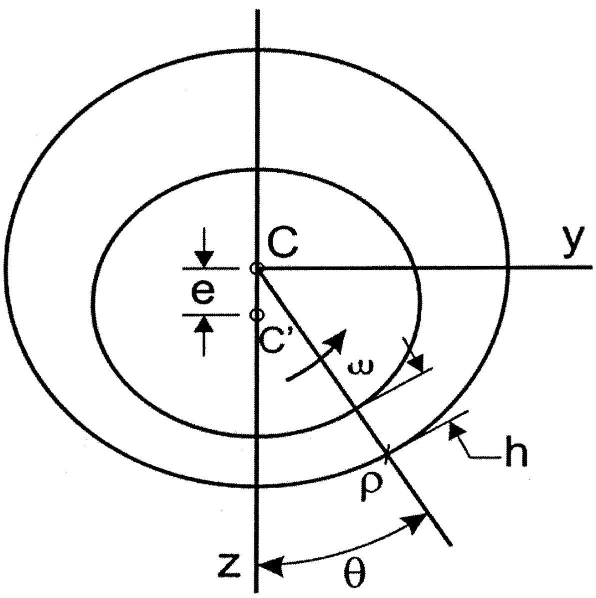

Fig.2 Squeeze Film Damper with Orbital Motion

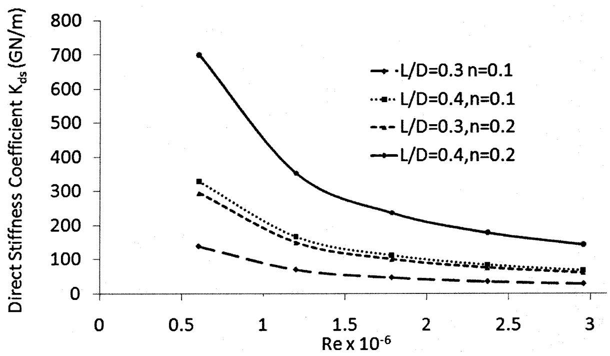

Fig.3 Stiffness Coefficient (GNs/m) as a Function of Reynolds Number for Clearance c = 0.1 mm

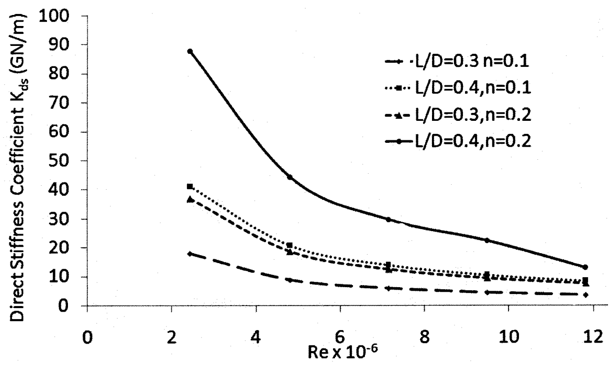

Fig.4 Stiffness Coefficient (GNs/m) as a Function of Reynolds Number for Clearance c = 0.2 mm

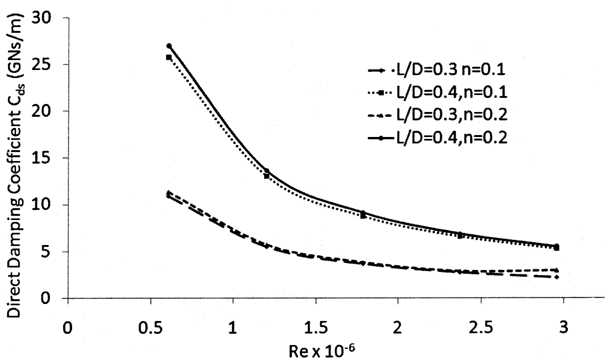

Fig.5 Damping Coefficient (GNs/m) as a Function of Reynolds Number for Clearance c = 0.1 mm

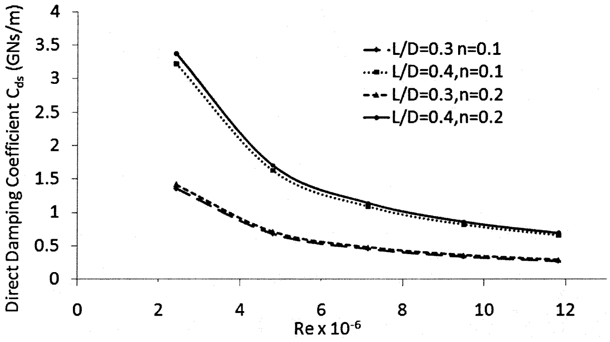

Fig.6 Damping Coefficient (GNs/m) as a Function of Reynolds Number for Clearance c = 0.2 mm

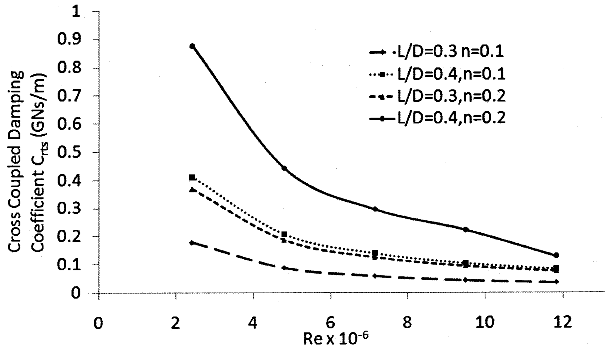

Fig.8 Cross Coupled Damping Coefficient (GNs/m) as a Function of Reynolds Number for Clearance c = 0.2 mm

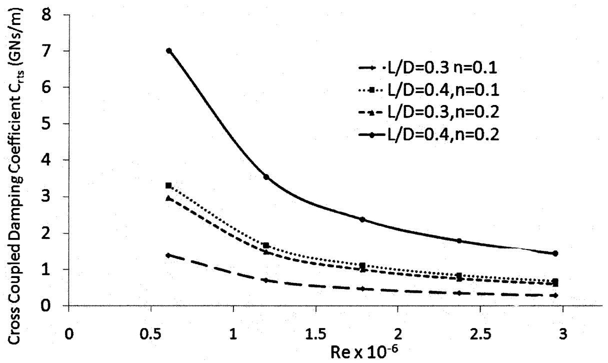

Fig.7 Cross Coupled Damping Coefficient (GNs/m) as a Function of Reynolds Number for Clearance c = 0.1 mm

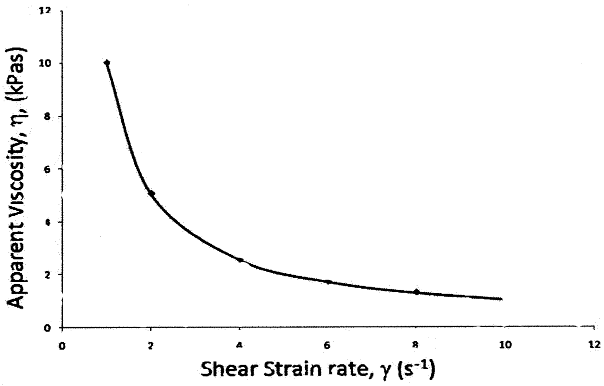

Fig.9 Apparent Viscosity (Pas) Vs Shear Strain Rate (s^{-1}) as Predicted by Equation (6)