무한 소폭 전기유변 스퀴즈 필름 댐퍼의 해석

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I. INTRODUCTION

Since Winslow (1) has reported an electro - rheological (ER) effect which features remarkable and reversible changes in the properties of the fluid due to an imposed external electric field, numerous applications of ER fluids in mechanical devices, such as clutches, control valves, active dampers, and etc. have been proposed to improve dramatically their performances (2,3).

When the external electric field is imposed to the ER fluid, it behaves as a Bingham fluid, displaying a field - dependent yield shear stress which is widely variable. Without the electric field, the ER fluid has a reversible and constant viscosity so that it flows as a Newtonian fluid. Another salient feature of the ER fluid is that the time required for the variation is very short (< 0.001 sec) (4-6). These attractive characteristics of the ER fluid provide the possibility of the appearance of new engineering technology, for instance, an active vibration control system. Recently, the application of the ER fluid to rotor-bearing systems has been also initiated.

Tichy (7) developed explicit forms of Reynolds' equation for ER fluids to analyze one-dimensional bearings such as a long squeeze film damper and a long journal bearing. The damping force of the long squeeze film damper is nearly proportional to the yield shear stresses so that the ER squeeze film damper can be quite effective for reducing the vibration of the associated rotor-bearing system. Morishita and Mitsui (8) experimentally verified that the rotor vibration can be substantially reduced in a wide range of rotating speeds by employing the ER fluid on the squeeze film damper instead of using conventional Newtonian fluids. Dimarogonas and Kollias (9) investigated the stability properties of the ER fluid journal bearings. They found that the force coefficients of ER journal bearings can be considerably changed by employing the electric field, and a significant extension of the stability region can be obtained. Nikolajsen and Hoque (10) had built and tested a new type of the ER damper consisting of several moving thin disks for a rotor-bearing system. They reported a possibility of eliminations of critical speeds and instability problems by controlling the external electric field imposed on ER fluid domains.

As mentioned the above, the applications of ER fluids to the dámpers and bearings are very effective for reducing the vibration of rotor-bearing systems and controlling the properties of bearings. However, researche on the applications of ER fluids to rotor-bearing systems are still to be further explored.

In the present paper, the lubrication equation for short squeeze film dampers operating with ER fluids is developed and solved to investigate the effects of yield shear stresses of ER fluids on the damping capability of the ER short squeeze film dampers. The Bingham lubrication theory develped by Wada *et al.* (11-14) is adopted herein to analyze the ER short squeeze film dampers. However, Wada *et al.* analyzed several types of one dimensional bearings lubricated with greases, but not a squeeze film damper. Tichy (7) also presented only a long bearing solution for the ER squeeze film damper employing Bingham lubrication theory similar to Wada's. As the results of the ER short squeeze film damper analysis, the dimensionless damping coefficients, fluid film pressure, and velocity profiles with the variation of yield shear stresses of ER fluids are presented.

II. ANALYSIS

Fig. 1 shows the geometry and coordinate

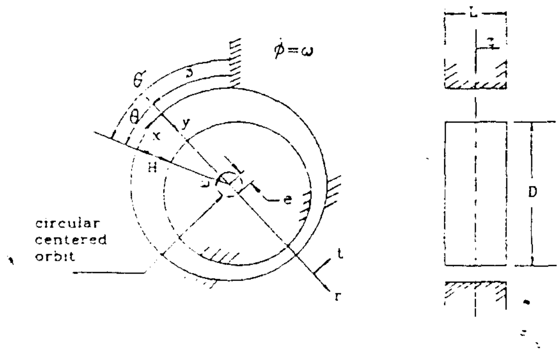


Fig. 1 Squeeze film damper geometry system for a short squeeze film damper executing a circular centered orbit. The journal is whirling with amplitude (e) and constant angular frquency (ω), but not spinning.

Considering the normal assumptions of laminar, incompressive, and isoviscous flow in the narrow annular region, the continuity and momentum equations describing two dimensional steady flow (neglecting the circumferential direction flow) are given in dimensionless form as follows:

$$\frac{\partial v}{\partial \eta} + \frac{\partial w}{\partial \xi} = 0 \qquad [1]$$

$$\frac{\partial p}{\partial \xi} = \frac{\partial \tau}{\partial \eta} \qquad [2]$$

Since ER fluids are assumed to be modeled as Bingham fluid of which the yield shear stress can be varied with respect to the electric field, the dimensional flow equations for the ER fluids are given by

$$\mu \dot{\gamma} = T \mp T_o \quad (|T| \geq T_o) \qquad [3a]$$

$$\mu \dot{\gamma} = 0 \quad (|T| < T_o) \qquad [3b]$$

where μ, $\dot{\gamma}$, T_o are the viscosity, the shear rate, and the yield shear stress of the ER fluids, respectively. According to the experimental results reported in (4-6), the relation between the yield shear stress and the electric field strength is given by the dimensional equation as follows:

$$T_o = \alpha \left(\frac{E}{H} \right)^{\beta} \qquad [4]$$

where E and H are the applied voltage and a gap which is the film thickness of the squeeze film damper, respectively. Both parameters α and β are the experimental constants of which the range of the exponent (β) is 1 to 2.4. The ER fluid equations, Eqs. [3] and [4], can be expressed in the following dimensionless form.

$$\frac{\partial w}{\partial \eta} = \tau \mp \tau_o \quad (|\tau| \geq \tau_o) \qquad [5a]$$

$$\frac{\partial w}{\partial \eta} = 0 \quad (|\tau| < \tau_o) \qquad [5b]$$

where

$$\tau_o = T_o \frac{C}{\mu \omega R} = \frac{\tau_{o,avg}}{h^{\beta}} \qquad [6a]$$

$$\tau_{o,avg} = \alpha \left(\frac{E}{C} \right)^{\beta} \frac{C}{\mu \omega R} \qquad [6b]$$

Since the flow of ER fluids in the squeezed film region is Poiseuille flow, a typical profile of the fluid film velocity has a rigid core in the center plane of the film as shown in Fig. 2. Thus the boundary conditions for the dimensionless velocity of ER fluids are given by

$$w = 0 \quad v = 0 \quad (\eta = 0) \qquad [7a]$$

$$w = w_c \quad (\eta = h_1) \qquad [7b]$$

$$w = w_c \quad (\eta = h_2 = h_1 + h_c) \qquad [7c]$$

$$w = 0 \quad v = \frac{\partial h}{\partial t} \qquad (\eta = h) \qquad [7d]$$

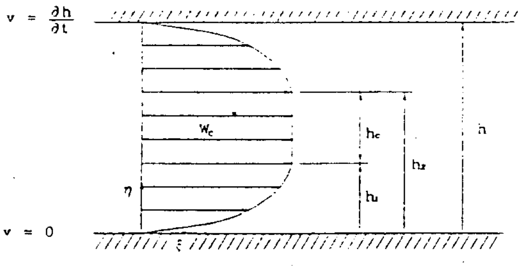


Fig. 2 Velocity profile of ER fluid

where w_c and h_c are the dimensionless velocity and thickness of the core moving as a solid, respectively.

After integrating Eqs. [2] and [5] with the above boundary conditions, Eqs. [7], the following velocity expressions are obtained in the dimensionless form.

$$w_1 = \frac{1}{2} p'(\eta^2 - h_1 \eta) + \frac{w_c}{h_1} \eta$$

$$(0 \leq \eta \leq h_1) \qquad [8a]$$

$$w_2 = w_c \qquad (h_1 \leq \eta \leq h_2) \qquad [8b]$$

$$w_3 = \frac{1}{2} p'(\eta - h)(\eta - h_2) - \frac{w_c}{h-h_2} (\eta - h)$$

$$(h_2 \leq \eta \leq h) \qquad [8c]$$

where p' is the dimensionless pressure gradient in the axial direction.

From the above velocity equations and the continuity equation, Eq. [1], a modified Reynolds' equation for the ER short squeeze film damper is obtained in the following dimensionless form.

$$\frac{d}{d\xi} \left\{ \frac{1}{6} p' h_1^3 + w_c (h - h_1) \right\} + \frac{\partial h}{\partial t} = 0 \qquad [9]$$

From the fact that the shear stresses at the film thickness, h_1 and h_2, have the same values as the yield shear stress of ER fluids, the following velocity equation of the core moving as a rigid body is obtained.

$$w_c = -\frac{1}{2} p' h_1^2 \qquad [10]$$

Also, in the squeezed film region, the shear stress is symmetric with respect to the center plane of the film so that the core will start to form at the film thickness, h_1, given by

$$h_1 = \frac{h}{2} \pm \frac{\tau_o}{p'} \qquad [11]$$

where the positive and negative signs are used with the negative and positive pressure gradient, respectively. As the magnitude of the axial pressure gradient decreases, the core of the ER fluid film expands to fill the squeeze film region (that is, h_1 approaches zero), and the velocity of the core becomes zero as described in Eq. [10]. Finally, there exists no flow in the squeezed film region. The axial pressure gradient at this point has the minimum value of the pressure gradient required to initiate flow. This critical pressure gradient is expressed as follows (4);

$$p'_c = \mp \frac{2\tau_o}{h} \qquad [12]$$

Substituting Eqs. [10] and [11] into the modified Reynolds' equation, Eq. [9], and expressing the squeezed film term of the modified Reynolds' equation in a coordinate system rotating with the same frequency and direction as the whirling motion, the final modified Reynolds' equation can be obtained in the following dimensionless form.

$$\frac{d}{d\xi}\left\{ \frac{(h\,p' \pm 2\tau_o)^2 (h\,p' \mp \tau_o)}{12\,p'^2} \right\} = -\frac{dh}{d\theta} \qquad [13]$$

Note that in the case of a Newtonian fluid ($\tau_o = 0$) the above equation becomes the conventional Reynolds' equation for the short squeeze film damper.

According to the solution of the conventional short squeeze film damper operating with a Newtonian fluid, the axial pressure profile is parabolic so that the axial pressure gradient at the center of the damper ($\xi = 0$) is zero. But in the case of the short squeeze film damper operating with ER fluids, an axial pressure gradient greater than the critical pressure gradient shown in Eq. [12] is required to initiate flow at the center of the ER damper. Therefore, the boundary conditions for the pressure of the ER short squeeze film damper could be described as follows;

$$\frac{dp}{d\xi} = p_c' \qquad (\xi = 0) \qquad [14a]$$

$$p = 0 \qquad (\xi = \pm L/D) \qquad [14b]$$

In order to calculate the ER fluid film pressure, the following cubic equation in terms of the axial pressure gradient is derived by integrating the modified Reynolds' equation, Eq. [13], with respect to the axial direction, and applying the boundary condition, Eq. [14a].

$$p'^3 - 3\left\{ \frac{p_c'}{2} - \frac{4}{h^3}\frac{dh}{d\theta}\xi \right\} p'^2 + \frac{p_c'^3}{2} = 0 \qquad [15]$$

Among the three roots of the above cubic equation, finding the axial pressure gradient greater than the critical pressure gradient at each specific location, the final ER fluid film pressure profile satisfying the boundary condition, Eq.

[14b], is determined by integrating the axial pressure gradient.

Dimensionless radial and tangential film forces are obtained by integration of the pressure field over the journal surface, that is:

$$f_r = \frac{F_r}{C_f} = \frac{D}{L}\int_0^{L/D}\int_{\psi}^{2\pi} p\cos\theta\, d\theta\, d\xi \qquad [16a]$$

$$f_t = \frac{F_t}{C_f} = \frac{D}{L}\int_0^{L/D}\int_{\psi}^{2\pi} p\sin\theta\, d\theta\, d\xi \qquad [16b]$$

where

$\psi = 0$ for uncavitated full film solution

$\psi = \pi$ for cavitated half film solution

In the above equations, C_f is a conversion factor and defines dimensionless film forces. The above ER fluid film forces can be expressed in terms of the force coefficients which are very useful to analyze the vibration of rotating machinery supported on squeeze film dampers. Since the film forces of the squeeze film damper executing a circular centered orbit are produced due to the tangential velocity of the journal center, the following dimensionless damping coefficients only exist without the stiffness coefficients (15).

$$C_{tt} = -\frac{f_t}{v_t} = -\frac{f_t}{\varepsilon} \qquad [17a]$$

$$C_{rt} = -\frac{f_r}{v_t} = -\frac{f_r}{\varepsilon} \qquad [17b]$$

where v_t ($= \omega e / \omega C$) is the dimensionless tangential velocity of the journal center.

III. RESULTS AND DISCUSSION

To represent the results of analysis of the ER short squeeze film damper, dimensionless fluid film pressure and velocity profiles, and dimensionless damping coefficients are calculated for varying values of averaged yield shear stresses of ER fluids. It is assumed that the ER short squeeze film damper is cavitated (π - film). Its L/D ratio is 0.25, and the exponent of the ER fluid, β =1, is used in the following calculations.

Figures 3(a) and 3(b) show the profiles of dimensionless ER fluid film pressure for the eccentricity ratios ε = 0.5 and 0.8, respectively. As the magnitude of averaged yield shear stresses, $\tau_{o,avg}$, increases (that is, the strength of the applied electrical field increases), the pressure substantially increases when compared with the pressure produced in the conventional squeeze film damper operating with a Newtonian fluid ($\tau_{o,avg}$ = 0). The increment of the pressure at the minimum film thickness is very large and the magnitude

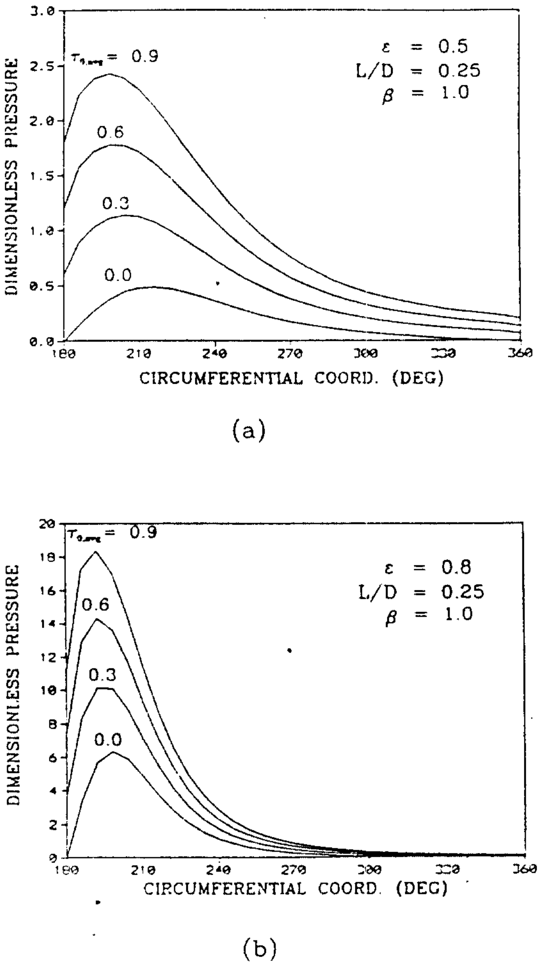


Fig. 3 Dimensionless pressure profiles
(a) $\varepsilon = 0.5$ (b) $\varepsilon = 0.8$

of the pressure at the maximum film thickness is also greater than zero, because of the critical axial pressure gradient required to initiate a flow of ER fluids.

Figure 4 shows the effects of the yield shear stresses on the dimensionless axial pressure field. The pressure profile of short squeeze film dampers is parabolic in the case of a Newtonian fluid ($\tau_{o,avg} = 0$), but it is not true in the case of ER fluids ($\tau_{o,avg} > 0$) as shown in Figure 4. The increment of the yield shear stress causes an increment of the axial pressure gradient which raises the ER fluid film pressure. There exists a discontinuity in the pressure profiles at the center of the ER squeeze film damper ($\xi = 0$) where the axial pressure gradient is equal to the critical axial pressure gradient. This result satisfying the boundary condition,

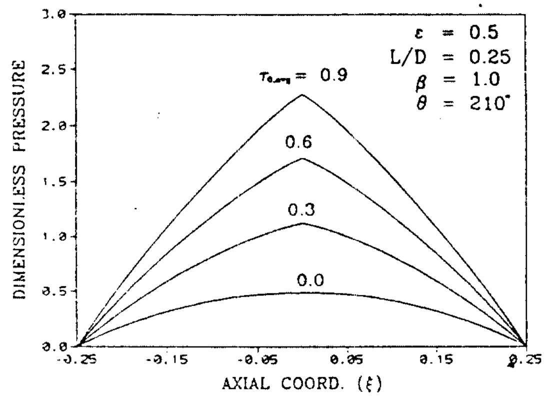
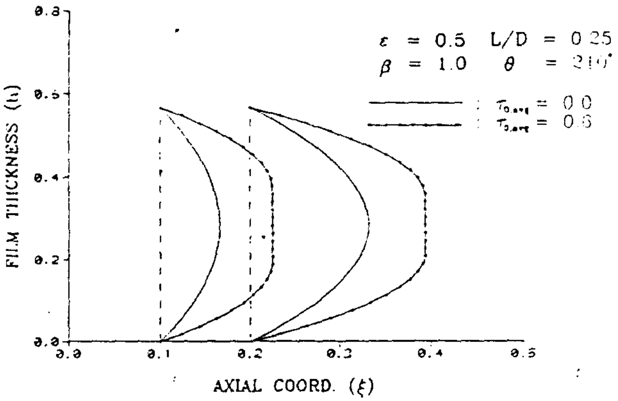


Fig. 4 Dimensionless pressure profiles
at $\theta = 210°$, $\varepsilon = 0.5$

Eq. [14a], can be easily obtained by substituting $\xi = 0$ into the governing cubic equation, Eq. [15].

Figures 5(a) and 5(b) show dimensionless velocity profiles of the ER fluid ($\tau_{o,avg} = 0.6$) and a Newtonian fluid ($\tau_{o,avg} = 0$) at the axial locations $\xi = 0.1$ and 0.2 for



(a)

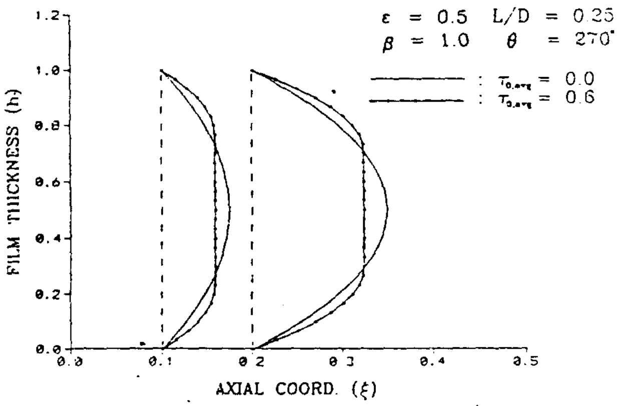


Fig. 5 Velocity profiles for ε =0.5, $\tau_{0,avg}$ =0.0 and 0.6 (a) θ=210° (b) θ=270°

circumferential locations, θ = 210° and 270°, respectively. There are floating rigid cores which play an important role in increasing ER fluid film pressure produced in the narrow squeezed film.

From the above pressure fields presented, the ER fluid film forces and corresponding force coefficients are calculated. Since the squeeze film damper executing a circular centered orbit cannot generate any fluid film forces without the velocity of the journal center, it has only damping coefficients without stiffness coefficients. The dimensionless direct and cross coupled damping coefficients, C_{tt} and C_{rt}, are shown in Figures 6 and 7, respectively. These damping coefficients are calculated using the assumptions that the ER fluid film is cavitated (π film) and its experimental exponent is β = 1. Both the direct and cross coupled damping coefficients are considerably increased as averaged yield shear stresses increase. In

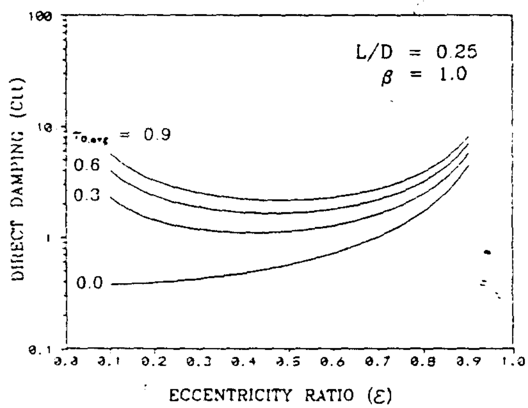


Fig. 6 Direct damping coefficients

a rotor system supported on squeeze film dampers, the direct damping coefficient, C_{tt}, plays a role as pure damping which reduces the vibration of the rotor system. But the cross coupled damping coefficient,

C_{rt}, plays a part in the stiffness which directly affects the critical speeds of the rotor system (16,17). Morishita and Mitsui (8) reported also that the 1st and 2nd critical speeds of a flexible rotor system incorperating the ER squeeze film damper have been increased and its damping ratio has also been considerably increased with increasing the strength of the electric field.

In order to investigate the effect of the experimental exponent value on the damping coefficients, the increment rates of the damping coefficients with β = 2 relative to those with β = 1 are represented in Table I. Both of the damping coefficients, C_{tt} and C_{rt} ,

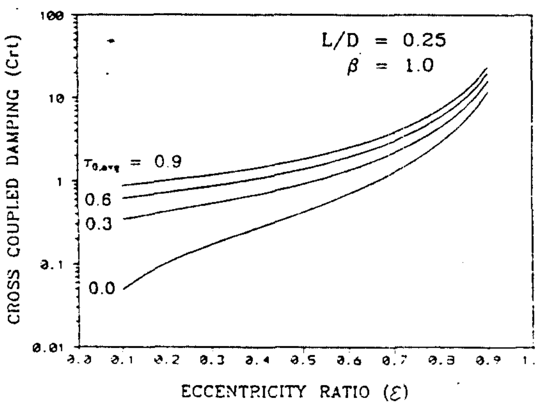


Fig. 7 Cross coupled damping coefficients

increase with increasing value of the exponent of ER fluids. In particular,, the percentage increase rate in the damping coefficients becomes larger as both the yield shear stresses and the eccentricity ratio increase because of the higher effectiveness of the yield shear stress at the narrow film gap.

In the process of numerical calculation for the ER long squeeze film damper (7), it might be difficult to get a converged solution because the governing equation (similar to the cubic equation, Eq. [15]) is highly nonlinear near the unknown circumferential location at which a rigid core fills the fluid film gap. However, in the case of the ER short squeeze film damper, there is no difficulty in the numerical procedure since the location that a rigid core fills the gap is always the center for a the short squeeze film damper.

VI. CONCLUSIONS

The lubrication analysis of short squeeze film dampers operating with ER fluids having the characteristics of Bingham fluids has been carried out. The governing equation for the ER short squeeze film damper was developed and solved to investigate the possibility of a great improvement in the damping capability of the squeeze film damper. It was found that a substantial increase in both the direct

Table I. Increment rates of dimensionless damping coefficients with $\beta = 1$ to those with $\beta = 2$

ε	Ctt			Crt		
	$\tau_{o,avg}$			$\tau_{o,avg}$		
	0.3	0.6	0.9	0.3	0.6	0.9
0.2	2.8 %	3.2 %	3.4 %	38.8 %	43.8 %	46.1 %
0.4	10.0 %	12.4 %	30.5 %	43.3 %	53.0 %	57.6 %
0.6	22.7 %	30.8 %	35.0 %	58.6 %	77.1 %	86.8 %
0.8	50.5 %	74.4 %	88.8 %	98.9 %	142.0	167.3 %

and cross coupled damping coefficients was obtained by using ER fluids instead of Newtonian fluids. It is anticipated that the ER short squeeze film damper could be very effective for actively reducing the vibration and controlling the critical speeds of a rotor-bearing system by tailoring the strength of the electric field.

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