PERFORMANCE OF THE TWO PARALLEL PLATES BEARING OF INFINITE LENGTH SQUEEZED TOGETHER LUBRICATED WITH ELECTRO-RFLUIDHEOLOGICAL

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ABSTRACT

A hydrodynamic squeeze film damper (SFD) is essentially a bearing within a bearing. In general, a SFD is often considered to be a bearing system that accommodates both advantages of journal and ball bearing . In spite of significant advancements in the lubrication technology and the development of meticulous damper design procedures; the bearings do fail in practice with serious consequences, particularly in large installations such as power plants, rolling mills, etc., according to many tribologists and practicing lubrication experts around the world are now a days involved in the design of the control technology for journal bearing systems using an entirely new design approach based on the electro-rheological (ER) fluids to suit the requirements of high speed and heavy load operations. A very reason method for doing this is the design of a newer squeeze film damper lubricated with electro-rheological (ER) fluids. If such a fluid is used to lubricate a squeeze film damper system, and an electric field between the rotor and the stator is applied, there will be alteration in the dynamic properties (stiffness and damping) and makes possible to control the damper system at faster rate.

Keywords: squeeze film damper, journal and ball bearing, electro-rheological (ER) fluids and dynamic properties

INTRODUCTION

In squeeze film damper rotor shaft rotates with the inner race of a supporting ball bearing, whereas its stationary outer race, which acts as the journal of SFD, whirls within the housing of the SFD. Whereas, the outer race is usually prevented from rotating and the squeezing action from the oil action from the cavity gives rise to damping forces. A squeeze film damper (SFD) installed at an outer race of a ball bearing support shaft adds externally additional damping to the flexible rotor system for improving damping capacity, vibration and stability**. [1]**

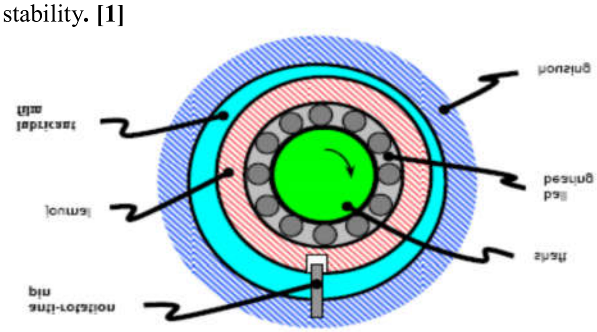

Fig.1 A Typical Squeeze Film Damper

Electro-rheological Fluids

Electro-rheological (ER) fluids – Intelligent materials whose rheological properties (viscosity, yield stress, shear modulus, etc.) readily controlled by using external electric field. Electro-rheological fluids are often referred as ER- fluids are suspensions of extremely fine conducting particles (up to 50 micrometers diameter) in an non conducting fluid. When external electric field is applied, the particles are polarized and form a chain like structure along the direction of the field as shown in the figure-2(a,b). This structure is responsible for the rheological properties alteration of the fluid. Thus the remarkable feature of ER fluid is that, it can instantly behave from a fluid like state to a solid like state (solidify in to jelly-like state) which exhibits a yield stress within a fraction of a second. When the field is removed, the material reverts back to a liquid state within milliseconds. These chains resist shear along a direction vertical to the field, the fluid reacts like a solid. Thus the apparent viscosity of these fluids changes reversibly by an order of 100,000 in response to electric field. Hence these fluids are now known as ER fluid rather than by the older term electro viscous fluids. This change is gradual, reversible and proportional to the applied electric field. This requires a very high voltage and low current. They are also known as one of highly functional fluids whose apparent viscosity can be varied by externally applied electric field strength **[2,3,4]**

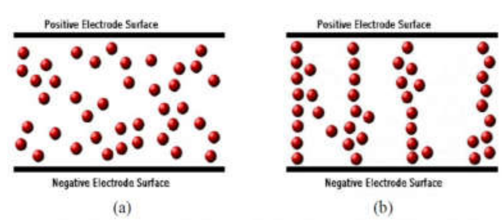

Fig-2(a) ER fluid structure without applied Electric field
(b) ER fluid structure with applied Electric field

The performance of a parallel plate bearing lubricated with electro-rheological Fluids which was modelled using dimensional analysis is presented. The typical part of a parallel plate bearing is shown in figure 3. The modeling and analysis of rotor-dynamic system using parallel plate bearings, where the dynamic co-efficient (stiffness and damping co-efficient) for the lubricant film were obtained from the Reynolds's equation.

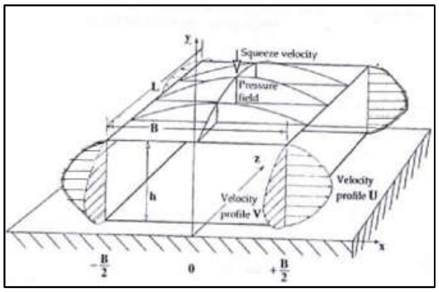

Fig. 3 Squeeze film between two parallel plates

When the top plate is pressed downwards, the fluid flow pattern i,e., the velocity profile of the fluid is shown in figure 3 . The Gwidon and Andrew **[5]** analyzed the flow pattern using Reynolds approach. They expressed the dependence of pressure on the space co-ordinates and the temporal co-ordinates as given below;

$$\frac{\partial}{\partial x}\left(\frac{h^3}{\mu}\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial z}\left(\frac{h^3}{\mu}\frac{\partial p}{\partial z}\right) = 6U\frac{\partial h}{\partial x} + 12\frac{\partial h}{\partial t} \quad \text{---1}$$

Tichy **[6]** assumed iso-viscous lubricant and zero entering velocity (U= 0) and simplified the above equation and the simplified equation is;

$$\frac{\partial}{\partial x}\left(h^3\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial z}\left(h^3\frac{\partial p}{\partial z}\right) = 12\mu\frac{\partial h}{\partial t} \quad \text{---2}$$

Where 'h' is the fluid film thickness in equation (2) which is replaced with symbol 'c' which is the general symbol for the clearance in bearing analysis. Replacing 'h' with 'c' in equation which becomes;

$$\frac{\partial}{\partial x}\left(c^3\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial z}\left(c^3\frac{\partial p}{\partial z}\right) = 12\mu\frac{\partial c}{\partial t} \quad \text{----3}$$

Re-arranging the Equation

$$\frac{\partial}{\partial x}\left(\frac{\partial p}{\partial x}\right) + \frac{\partial}{\partial z}\left(\frac{\partial p}{\partial z}\right) = \frac{12\mu}{c^3}\frac{\partial c}{\partial t} \quad \text{----4}$$

Pressure developed in the film:

For two parallel plates of infinite length or the length in the z direction is infinitely long (long bearing approximation). Assuming no end leakage in the z direction i.e. V=0, hence $\frac{\partial p}{\partial z}=0$ which simplifies the equation 4 to;

$$\frac{\partial}{\partial x}\left(\frac{\partial p}{\partial x}\right) = \frac{12\mu}{c^3}\frac{\partial c}{\partial t} \quad \text{----5}$$

Reducing the above equation

$$\frac{\partial^2 p}{\partial x^2} = \frac{12\mu}{c^3}\frac{\partial c}{\partial t}$$

Integrating once,

$$\frac{\partial p}{\partial x} = \frac{12\mu}{c^3}\frac{\partial c}{\partial t}x + c_1 \quad \text{----6}$$

Integrating again,

$$p = \frac{6\mu}{c^3}\frac{\partial c}{\partial t}x^2 + c_1 x + c_2 \quad \text{----7}$$

Applying boundary conditions from figure-3 (at no load and no pressure) i.e. P=0 at $x = \pm\frac{B}{2}$, and the pressure gradient $\frac{\partial p}{\partial x} = 0$ at x=0. Substituting these boundary conditions into equations (7) yields,

$$c_1 = 0$$

substituting $c_1 = 0$ in equation (6) which yields

$$c_2 = -\frac{6\mu B^2}{4c^3}\frac{\partial c}{\partial t}$$

Substituting for c_1 and c_2 in the equation (6) which gives the pressure distribution as a function of $\frac{\partial c}{\partial t}$

$$p = \frac{6\mu}{c^3}\frac{\partial c}{\partial t}\left[x^2 - \frac{B^2}{4}\right] \quad \text{----8}$$

Where the term $\frac{\partial c}{\partial t}$ is negative for a positive squeeze direction as shown in figure 3.

Static Characteristics of Parallel Plate Bearing-Load Carrying Capacity:

The squeeze load carrying capacity i.e., W can be obtained by integrating the pressure distribution on the bearing surface area 'A' i.e.

$$W = \int_A p \,.\, dA$$

$$W = \int_0^L \int_{-\frac{B}{2}}^{+\frac{B}{2}} p\,dx\,dy$$

$$W = L\int_{-\frac{B}{2}}^{+\frac{B}{2}} p\,dx$$

Where L= length of the fluid in z direction.

Substituting value of p from equation (8)

$$W = L\int_{-\frac{B}{2}}^{+\frac{B}{2}}\left[\frac{6\mu}{c^3}\frac{\partial c}{\partial t}\left(x^2-\frac{B^2}{4}\right)\right]dx$$

integrating above equation yields

$$W = -\frac{\mu B^3 L}{c^3}\frac{\partial c}{\partial t} \quad \text{or}$$

$$W = -\frac{\mu L B^3}{c^3}V \quad \text{----9}$$

Dynamic Characteristics of Parallel Plate Bearing-Load Carrying Capacity:

Dynamic characteristic refers to stiffness and damping characteristics. Stiffness and damping co-efficient are important because stability or otherwise of the system depends on these parameters.

Stiffness -K_d:

The stiffness may be calculated in the following way

The load capacity W is;

$$W = \frac{\mu B^3 L}{c^3}V$$

Stiffness K_d may be defined as the load capacity per unit clearance or stiffness is the first differential of the load capacity with respect to clearance.

$$K_d = -\frac{\partial W}{\partial c} \quad \text{or ;}$$

$$K_d = -\frac{\partial}{\partial c}(W)$$

Substituting value of W in above equation, the equation reduces to;

$$K_d = -\frac{\partial}{\partial c}\left(\frac{\mu B^3 L}{c^3}\frac{\partial c}{\partial t}\right) \quad \text{----10}$$

Differentiating the equation yields

$$K_d = \frac{3\mu B^3 L}{c^4}\frac{\partial c}{\partial t} \quad \text{or}$$

$$K_d = \frac{3}{c}\left(\frac{\mu B^3 L}{c^3}\frac{\partial c}{\partial t}\right)$$

$$K_d = \frac{3W}{c}\ \text{N-m} \quad \text{----11}$$

The stiffness is found to be directly proportional with the load and inversely proportional with clearance.

Damping Co-efficient -C_d:

The damping co-efficient Cd may be defined as the damping force per unit velocity

We have load capacity $W = \frac{\mu B^3 L}{c^3}V$

$$\frac{\partial w}{\partial v} = \frac{\mu B^3 L}{c^3} \quad \text{N-s/m} \quad \text{----12}$$

$$C_d = \frac{\mu B^3 L}{c^3} \quad \text{N-s/mm} \quad \text{----13}$$

Where $C_d = \frac{\partial w}{\partial v}$

$$K_d = \frac{3\mu(E)(1+E^2)B^3 L}{c^4}\frac{\partial c}{\partial t} \quad \text{----14}$$

$$C_d = \frac{\mu(E)(1+E^2)B^3 L}{c^3} \quad \text{N-s/mm} \quad \text{----15}$$

The values K_d and C_d for different voltage applied were estimated using the equations which are given in 14and 15. K_d and C_d are found to be dependent on applied electric field 'E' ,bearing breadth 'B', bearing length 'L' and bearing clearance 'c'. K_d in addition to above four parameters also depends on $\frac{\partial c}{\partial t}$ i.e., the velocity of squeezing.

MODELLING OF ELECRRO-RHEOLOGICAL FLUID RESPONSE – DIMENSIONAL ANALYSIS APPROACH

A dimensional approach in the present study was made use for obtaining modeling of electro-rheological fluids. In addition a phenomenon of shear thinning and thickening was also found to influence the behavior of the electro-rheological fluids. The Herschel and Bulkley model which modeled shear thinning was also incorporated in the final model of the theoretical model developed **[7]**. Assuming μ_e is largely depends on $\dot{\gamma}$ and E, it is possible to obtain a relation between the viscosity, shear strain rate and the intensity of the applied electric field using Rayleigh's method of dimensional analysis.

Using the equation of the form;

$$\mu_e(E) = K|\dot{\gamma}|^p(E)^q \quad \text{----161}$$

Introducing the corresponding MLT units,

$$ML^{-1}T^{-1} = K\left(T^{-1}\right)^p\left(M^{\frac{1}{2}}L^{-\frac{1}{2}}T^{-1}\right)^q \quad \text{---- 17}$$

Equating the coefficients of M, L, and T on both sides and simplifying, the values of p and q are;

p = -1, q = 2.

Introducing these values of p and q in equation (16) which simplified to;

$$\mu_e(E) = K|\dot{\gamma}|^{-1}(E)^2 \quad \text{----18}$$

This is the change in viscosity due to the applied electric field. In practice the electro-rheological fluids were found to exhibit shear thinning. The shear thinning of electro-rheological fluids was modelled by Herschel-Bulkley and is given as;

$$\mu_e = K|\dot{\gamma}|^{\frac{1}{m}-1} \quad \text{----19}$$

Taking into the effect of shear thinning as modelled by Herschel-Bulkley and incorporating in the model developed by the dimensional analysis, the total viscosity μt is given below;

$$\mu_t = K|\dot{\gamma}|^{-1}E^2 + K|\dot{\gamma}|^{\frac{1}{m}-1} \quad \text{----20}$$

Where 'm' in the equation (20) takes the value which is larger compared to unity when shear thinning occurs. When 'm' is large i.e., a case of shear thinning which is practically observed in electro-rheological fluids, then equation (20) can be written as;

$$\mu_t = K|\dot{\gamma}|^{\frac{1}{m}-1}E^2 + K|\dot{\gamma}|^{\frac{1}{m}-1} \quad \text{----21}$$

$$\mu_t = \left[K|\dot{\gamma}|^{\frac{1}{m}-1}\right](1+E^2) \quad \text{----22}$$

Equation (22) is applicable for electro-rheological fluids which undergo large degree of shear thinning.

$$\mu_t = \mu(E)\,(1+E_2)$$

$$\text{where } \mu(E) = \left[K|\dot{\gamma}|^{\frac{1}{m}-1}\right] \quad \text{-----23}$$

The equation (23) represents the total viscosity estimated by the dimensional analysis and taking into effect of shear thinning of the electro-rheological fluid under the action of the electric field. Total viscosity predicted in (23) is proportional to the square of the electric field intensity and hence the yield stress τ_y is proportional to E_2.

The numerical values of μ_t can be estimated using μ(0), where μ(0) co –efficient of friction for zero viscosity. Literature review indicated authors quoting the values of μ(0). Sharana Basavaraja etal **[8]** are one such authors who had taken as 1.100x10-7 Pa-s for μ(0) and 0 to 4 Kv/mm as range for applied electric field 'E'. Equation (23) is used to estimate μ_t taking the values of μ(0) and E. The estimated value of μ_t (E) is plotted as a function of E and is shown in the figure 4.

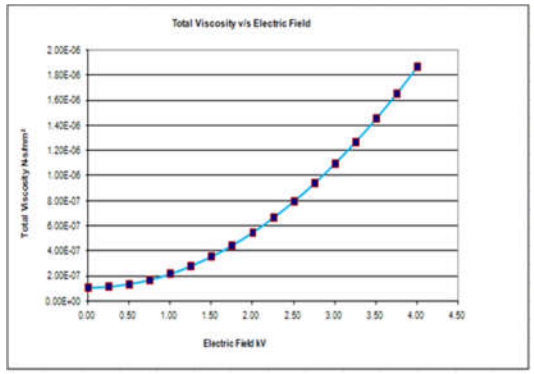

Fig:4. Dependency of total viscosity with electric field

The plot shows that μ_t is exponential function of applied electric field E. the μ_t start with a value μ(0) which is 1.100x10-7 Pa-s in the absence of applied electric field. The exponent of E in the function of μ_t is 2.

Results and Discussions:

Dependency of the load capacity W, the stiffness K_d and the damping co-efficient C_d for combinations of two arbitrary values of c i.e., c= 0.1 and c= 0.2; two values of B i.e., B= 0.1 and B= 0.2 and taking $\frac{\partial c}{\partial t}$ as 500m/s were calculated and plotted in figures 5, 6 and 7.

The dependence of the load capacity 'W' on applied electric field 'E' for different combinations of clearance 'c' and breadth 'B' with the squeeze velocity of 500 m/s is shown in fig. 5

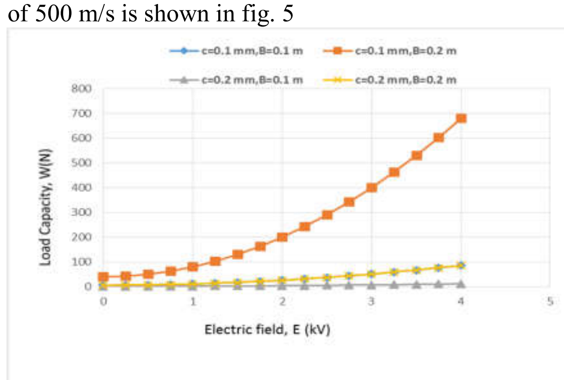

Fig 5 Dependency of Load capacity (W) with Electric field (E)

Curves with symbols (square) stands for the load capacity 'W' as a function of applied voltage 'E' when c=0. 1 mm, B=0.2 m and V=500m/s. The minimum value of W, is 40 N, the maximum value of W, is 680 N, The load capacity 'W' is found to be an exponential function of applied electric field 'E' for the above geometry of bearing and speed of rotation of the shaft. The exponent of applied electric field 'E' is 2. The exponential dependency of load capacity 'W' with applied electric field 'E' is in concurrent with the response of electro-rheological fluids with applied electric field ‘E’.

Curves with symbols (x) stands for the load capacity ‘W’ as a function of applied voltage E when c=0.2 , B= 0.2 and V=500m/s. Curves with symbols (rhombus) stands for the load capacity ‘W’ as a function of applied voltage E when c=0.1 , B= 0.1 and V=500m/s. The numerical values of the load capacity ‘W’ is found to be identical for the above geometry of bearings and its speed i.e., the above combinations of c, B and velocity V. The minimum value of W, is 5 N, the maximum value of W, is 85N, The load capacity ‘W’ is found to be function of the geometry particularly the clearance c and applied electric field ’E’. The positive effect of applied electric field ‘E’ on load capacity ‘W’ is negated by the increase in clearance c leading to not steep increase of W with E.

Curves with symbols (triangle) stands for the load capacity ‘W’ as a function of applied voltage E when c=0.2, B= 0.1 and V=500m/s. The minimum value of W, is 0.625N, the maximum value of W, is 10.625N, The load capacity ‘W’ is found to be function of the geometry particularly the clearance c and applied electric field ’E’. The positive effect of applied electric field ‘E’ on load capacity ‘W’ is negated by the increase in clearance c leading to decrease of W with E.

The load capacity ‘W’ for a given level of applied electric field is found to increase when c decreased from 0.2 to 0.1 and breadth “B” decreased from 0.2m to 0.1m. This quantum of change in W was found to be more at higher applied electric field E.

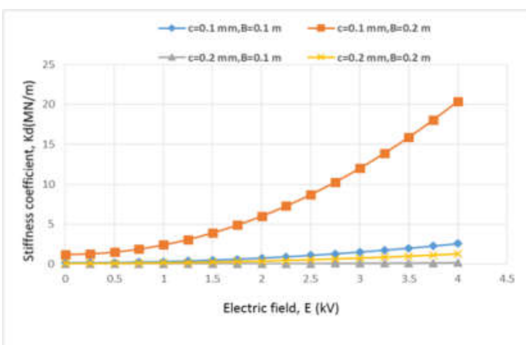

Fig6: Dependency of Stiffness co-efficient (K_d) with Electric field (E)

The dependence of the stiffness K_d on applied electric field ‘E’ for different combinations of clearance ‘c’ and breadth ‘B’ with the squeeze velocity of 500 m/s is shown in figure 6

Curves with symbols (square) stands for the stiffness K_d as a function of applied voltage ‘E’ when c=0. 1 mm, B=0.2 m and V=500m/s. The minimum value of the stiffness K_d, is 1.2MN/m, the maximum value of the stiffness K_d is 20.4MN/m .The stiffness K_d is found to be an exponential function of applied electric field ‘E’ for the above geometry of bearing and speed of rotation of the shaft. The exponent of applied electric field ‘E’ is 2. The exponential dependency of the stiffness K_d with applied electric field ‘E’ is in concurrent with the response of electro-rheological fluids with applied electric field ‘E’.

Curves with symbols (rhombus) stands for the stiffness K_d as a function of applied voltage E when c=0.1, B= 0.1 and V=500m/s. The minimum value of stiffness K_d , is 0.15MN/m, the maximum value of stiffness K_d , is 2.5MN/m, The stiffness K_d is found to be function of the geometry particularly the clearance c and applied electric field ’E’. The positive effect of applied electric field ‘E’ is negated by the increase in clearance c leading to not steep increase of K_d with E.

Curves with symbols (x) stands for the stiffness K_d as a function of applied voltage E when c=0.2, B= 0.2 and V=500m/s. The minimum value of the stiffness K_d , is 0.075MN/m, the maximum value of the stiffness K_d , is 1.275MN/m , The stiffness K_d is found to be function of the geometry particularly the clearance c and applied electric field ’E’. The positive effect of applied electric field ‘E’ on the stiffness K_d is negated by the increase in clearance c leading to not steep increase of K_d with E.

Curves with symbols (triangle) stands for the stiffness K_d as a function of applied voltage E when c=0.2, B= 0.1 and V=500m/s. The minimum value of K_d , is 0.009375 MN/m the maximum value of K_d, is 0.159375MN/m The stiffness K_d is found to be function of the geometry particularly the clearance c and applied electric field ’E’. The positive effect of applied electric field ‘E’ on stiffness K_d is negated by the increase in clearance c leading to decrease of K_d with E.

The stiffness K_d for a given level of applied electric field is found to increase when c decreased from 0.2 to 0.1 and breadth “B” decreased from 0.2m to 0.1m. This quantum of change stiffness K_d in was found to be more at higher applied electric field E.

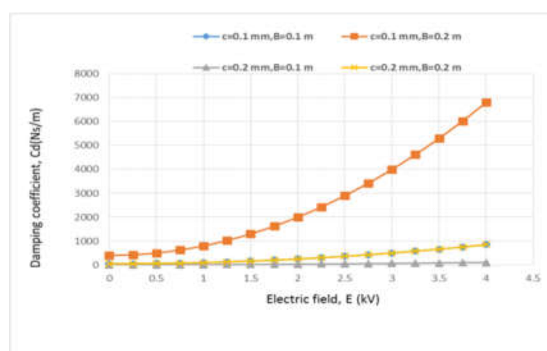

Fig7: Dependency of Damping co-efficient (C_d) with Electric field (E)

The dependence of the damping co-efficient C_d on applied electric field ‘E’ for different combinations of clearance 'c' and breadth 'B' with the squeeze velocity of 500 m/s is shown in figure 7

Curves with symbols (square) stands for the damping co-efficient C_d ' as a function of applied voltage 'E' when c=0. 1 mm, B=0.2 m and V=500m/s. The minimum value of damping co-efficient C_d , is 400 Ns/m, the maximum value of damping co-efficient C_d, is 6800 Ns/m. The damping co-efficient C_d is found to be an exponential function of applied electric field 'E' for the above geometry of bearing and speed of rotation of the shaft. The exponent of applied electric field 'E' is 2. The exponential dependency of damping co-efficient C_d' with applied electric field 'E' is in concurrent with the response of electro-rheological fluids with applied electric field 'E'.

Curves with symbols (x) stands for the damping co-efficient C_d as a function of applied voltage E when c=0.2, B= 0.2 and V=500m/s. Curves with symbols (rhombus) stands for the damping co-efficient C_d as a function of applied voltage E when c=0.1, B= 0.1 and V=500m/s. The numerical values of the damping co-efficient C_d is found to be identical for the above geometry of bearings and its speed i.e., the above combinations of c, B and velocity V. The minimum value of damping co-efficient C_d is 50 Ns/m, the maximum value of damping co-efficient C_d , is 850 Ns/m. The damping co-efficient C_d is found to be function of the geometry particularly the clearance c and applied electric field 'E'. The positive effect of applied electric field 'E' on damping co-efficient C_d is negated by the increase in clearance c leading to not steep increase of C_d with E.

Curves with symbols (triangle) stands for the damping co-efficient C_d as a function of applied voltage E when c=0.2, B= 0.1 and V=500m/s. The minimum value of damping co-efficient C_d is 6.25 Ns/m, the maximum value of damping co-efficient $C_d$is 106.25 Ns/m. The damping co-efficient C_d is found to be function of the geometry particularly the clearance c and applied electric field 'E'. The positive effect of applied electric field 'E' on damping co-efficient C_d is negated by the increase in clearance c leading to decrease of damping co-efficient C_d with E.

The damping co-efficient C_d for a given level of applied electric field is found to increase when c decreased from 0.2 to 0.1 and breadth "B" decreased from 0.2m to 0.1m. This quantum of change in damping co-efficient C_d was found to be more at higher applied electric field E.

Fig 5 – The value of load capacity W is found to increase with applied electric field 'E'. The bearing was lubricated with Electro-rheological fluid whose co-efficient of friction μ_t was found to increase exponentially with the applied electric field 'E' as found out by the model using dimensional analysis given in the equation (23).

Fig 6 – The stiffness K_d is dependent on $\frac{B^3}{C^3}$ apart from μ and V. As the geometrical parameter comes down, stiffness K_d will be more dependent on geometry rather than lubricant characteristics.

Fig 7 – The damping co-efficient C_d is dependent on $\frac{B^3}{C^3}$ apart from μ and V. As the geometrical parameters comes down, damping co-efficient C_d will be more dependent on geometry rather than lubricant characteristics.

CONCLUSIONS:

The evaluations of characteristics like the load capacity W, stiffness co-efficient K_d and damping co-efficient of the bearing are functions of bearing geometries i.e, the clearance c and breadth B, applied electric field E and speed of the rotor. The full potential of electro-rheological fluid is made use under the conditions of geometry of bearing where B/c ratio is maximized. The dynamic characteristics indicate that the stiffness and damping characteristics can be controlled by controlling the electric field intensity where B/c ratio is maximized. Thus the electro-rheological fluid bearing or damper can be controlled for static and dynamic characteristics based on the system requirement.

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