Dynamic Modeling of a Liquid Bearing in the Head-Disk Interface

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Abstract --- Viscous liquid bearings could become a commercially valuable technology for the information storage industry. In the least, the research knowledge gained may be important for modeling lubricants in conventional air bearing drives. Several non-Newtonian empirical models are given to relate the stress to the strain for the bearing liquid. The load bearing capacity, the amount of normal force the fluid can sustain, is calculated for several models by introducing an order of magnitude analysis. Similarly, the power consumption due to frictional forces on the surface of the slider due to the fluid motion is computed.

I. INTRODUCTION

Higher storage recording densities, reliable operation, and lower production costs are the key factors in developing future disk drive systems. Two types of efforts, modifying current technology and introducing a new technology, may be developed to establish dynamic stability during intermittent contacts between the head and disk. The current trend of modifying the existing technology focuses at system miniaturization, and has been exhaustively studied during the past five years [1].

The second option, which is the focus of this paper, is to introduce a liquid (lubricant) bearing, commercially known as VISqUS [2], instead of an air-bearing. This novel concept is an especially attractive alternative since it can avoid tribomechanical problems at the head-disk interface. The viscosity and relaxation time of the ultra-thin liquid film becomes orders of magnitude greater than the bulk liquid, and as a consequence it acts as a solid lubricant and slider disk collisions are avoided. Therefore, in principle, one could eliminate the expensive carbon overcoating process. However, due to the problems associated with contamination and single side operation, this option is somewhat questionable for manufacturers, even though some claim that this technology could operate a high performance single disk with 500 megabytes of storage at less than 1 W power consumption [3]. To obtain a feasibility study of liquid bearing systems, it is critical to relate the design parameters to the microscopic rheological properties of the thin liquid film. Even though VISqUS technology has been proven to be unfeasible for drive manufacturers, the knowledge we have learned may provide insight towards a hybrid approach with an air and liquid bearing mixture which will be potentially important for future drive systems.

II. PERFORMANCE VARIABLES FOR LIQUID BEARINGS

For liquid bearing modeling, accurate incorporation of thin film non-Newtonian rheology is essential for quantitative information on drive performance. Two key performance properties for liquid bearing drives are the load

bearing capacity and the power consumption due to frictional forces on the surface of the slider. The load bearing capacity, W , for a fluid can be defined as,

$$W \equiv \iint (\pi_{zz} - P_{amb}) dA \quad (1)$$

where π_{zz} is the normal component of the total stress tensor and P_{amb} is the ambient pressure. W is a measure of the normal force the slider can support at a given film thickness, velocity, and fluid properties. This value is critical for estimating drive operating parameters for the desired performance.

The power consumption due to fluid frictional forces, P_c , can similarly be expressed as

$$P_c \equiv \iint [\bar{\tau}(\bar{\gamma}) \cdot \bar{v}] \cdot d\bar{A} \quad (2)$$

where $\bar{\tau}$ is the deviatoric stress (i.e., $\bar{\pi} = P\bar{\delta} + \bar{\tau}$, where P is the isotropic pressure and $\bar{\delta}$ is the identity tensor), $\bar{\gamma}$ is the shear rate of the flow, and $\bar{v}$ is the fluid velocity. P_c is especially critical for liquid bearing drives, and is directly related to the amount of energy required to sustain operation. Since there is a large market for portable lap-top or hand held drives, conserving energy through the reduction of P_c becomes increasingly important.

In order to obtain accurate expressions for W and P_c in terms of fluid microrheological parameters, it is necessary to solve for the velocity profile and normal stress distribution of the bearing. Velocity profiles can be obtained by solving the equation of motion using the lubrication approximation. Assuming that inertial effects are negligible compared to viscous and pressure forces, the appropriate steady state equations for non-Newtonian fluids can be expressed as,

$$\frac{\partial P}{\partial x} = \frac{\partial \tau_{xz}}{\partial z} + \frac{\partial \tau_{xx}}{\partial x}; \quad \frac{\partial P}{\partial y} = \frac{\partial \tau_{yz}}{\partial z} + \frac{\partial \tau_{yy}}{\partial y}; \quad \frac{\partial P}{\partial z} = 0 \quad (3)$$

where P is the isotropic pressure distribution and τ_{ij} is the i,j th deviatoric stress tensor component. The coordinate system is shown in Figure 1 which is a simplified schematic of a liquid bearing. Rigorous treatment of temperature

effects involves the simultaneous solution of the momentum and energy balance equations. Any changes in the fluid temperature may be approximately handled, however, by neglecting the coupling between the momentum and energy equations and parameterizing the apparent viscosity as a function of temperature. Although this is not studied here, a similar analysis as presented can be performed for temperature effects.

Figure 1. Schematic of a liquid bearing.

Constitutive equations for τ_{ij} must be incorporated with Eq. (3) to solve for the velocity profile in terms of pressure gradients. To solve for the pressure distribution, the equation of continuity for incompressible fluids is integrated across the bearing,

$$\frac{\partial}{\partial x} \int_0^h v_x dx + \frac{\partial}{\partial y} \int_0^h v_y dy = 0 \quad (4)$$

and a Reynolds type of equation is formed where v_x and v_y are the x and y components of the relative disk velocity, respectively. For non-Newtonian fluids, the resulting equation is a nonlinear partial differential equation which can be solved using either approximate analytic or numerical techniques.

III. RHEOLOGICAL MODELS

In order to solve Eqs. (3) and (4) a constitutive relationship is needed between the stress, $\bar{\tau}$, and the rate of strain, $\dot{\bar{\gamma}}$. There are numerous rheological equations of state which relate the stress to the strain. The fluids used in liquid bearing drives (and lubricants for conventional air bearings) are non-Newtonian shear-thinning in nature. Constitutive equations (generalized Newtonian fluid models) which incorporate non-linear effects are usually in one of two general forms:

$$\tau = f_1(\dot{\gamma}) \quad \text{or} \quad \dot{\gamma} = f_2(\tau) \quad (5,6)$$

where f_1 and f_2 denote different functional forms which depend on the model chosen, $\dot{\gamma} = (\dot{\bar{\gamma}} : \dot{\bar{\gamma}})^{1/2}$ and $\tau = (\bar{\tau} : \bar{\tau})^{1/2}$. It is often possible to change from one form to the other for a particular model to make computations easier.

There are several models in the form of Eq. (5). Some of these are listed in Table 1 for reference. The simplest model is the well known power law model of Ostwald and de Waele [4], where μ and n are fluid dependent parameters (the Newtonian fluid model is recovered if $n=1$).

Models of the form of Eq. (6) are often referred to as nonlinear viscous models. One of the first models of this type was proposed by Eyring [5], where τ_E is referred to as

Rheological Model	Constitutive Expression
Newtonian (Newt)	$\tau = \mu \dot{\gamma}$ (7)
Power Law (PL)	$\tau = \mu \dot{\gamma} \dot{\gamma} ^{n-1}$ (8)
Eyring (EY)	$\dot{\gamma} = \frac{\tau_E}{\mu} \sinh\left(\frac{\tau}{\tau_E}\right)$ (9)
Bair and Winer (BW)	$\dot{\gamma} = \frac{\tau_L}{\mu} \ln\left(1 - \frac{\tau}{\tau_L}\right)^{-1}$ (10)
Gecim and Winer (GW)	$\dot{\gamma} = \frac{\tau_L}{\mu} \tanh^{-1}(\tau / \tau_L)$ (11)
Iivonen and Hamrock (IH)	$\dot{\gamma} = \frac{\tau_L}{\mu} [(1 - \tau / \tau_L)^{-1} - 1]$ (12)

Table 1. Several forms of constitutive rheological equations of state.

the Eyring stress and μ is a viscosity. Bair and Winer [6] proposed another form where a limiting shear strength τ_L has been incorporated. Gecim and Winer [7] attempted to simplify this expression and proposed a similar form. Iivonen and Hamrock [8] proposed a relatively simple, but highly non-linear form, once again with a limiting shear strength. For these forms, the stress is less than or equal to the limiting shear strength (i.e., $\tau \leq \tau_L$). Equations (9-12) are similar forms, especially at the high shear rate limit. Figure 2 shows a comparison of the dimensionless shear

Figure 2. Comparison of several non-linear viscous constitutive relations.

stress, τ/τ_L , as a function of the dimensionless strain rate, $\mu\dot{\gamma}/\tau_L$, for these models (for the Eyring model it was assumed that $\tau_E=\tau_L/3.5$).

IV. ORDER OF MAGNITUDE ANALYSIS

A. Load Bearing Capacity

In order to obtain qualitative information on both the load bearing capacity and power consumption, an order of magnitude analysis has been generalized from a previous treatment of air bearings [9]. The pressure difference in Eq. (1) can be replaced by an integrated pressure derivative, which can be related to the deviatoric stress via Eq. (3). The resulting form,

$$W = \iint \left[\int_0^x \frac{\partial P}{\partial x} dx \right] dA = \iint \left[\int_0^x \frac{\partial \tau_{xz}}{\partial z} dz \right] dA \quad (13)$$

where the only component of the stress tensor incorporated is the shear stress τ_{xz} (normal stress effects are neglected), can be investigated using an order of magnitude or dimensional analysis estimation. Using approximations for the integrations in Eq. (13), W can be expressed as

$$W \propto L^2 B R_f^3 \frac{d\tau_{xz}}{dz} \quad (14)$$

where L and B are the length and width of the bearing surface area, and R_f is defined as a percentage reduction of a known surface area. Depending on the form of the rheological equation used (either Eq. (5) or Eq. (6)), the above expression results in

$$W \propto L^2 B R_f^3 \left(\frac{df_1}{d\dot{\gamma}} \right) \left(\frac{d\dot{\gamma}}{dz} \right) \text{ or } W \propto L^2 B R_f^3 \left(\frac{df_2}{d\tau} \right)^{-1} \left(\frac{d\dot{\gamma}}{dz} \right) \quad (15,16)$$

For slider bearing flow, $\dot{\gamma} = dv_x / dz$, and

$$\frac{d\dot{\gamma}}{dz} = \frac{d^2 v_x}{dz^2} = \frac{V_x}{\Delta h^2} \quad (17)$$

where V_x is the linear velocity of the rotating disk, and Δh^2 refers to the change from the inlet to the outlet of the square of the separation height h .

B. Power Consumption

Similarly, an order of magnitude analysis of Eq. (2) yields an approximate expression for the power consumption due to frictional forces.

$$P_c \propto L B R_f^2 V_x \tau_{xz} \quad (18)$$

This expression reduces to either,

$$P_c \propto L B R_f^2 V_x f_1(\dot{\gamma}) \text{ or } P_c \propto L B R_f^2 V_x f_2^{-1}(\dot{\gamma}) \quad (19, 20)$$

depending on whether the equation of state is in the form of Eq. (5) or Eq. (6), where f_2^{-1} indicates the inverse of the function f_2 .

V. RESULTS AND DISCUSSION

Table 2 shows the resulting expressions for the load bearing capacity and power consumption for each of the non-Newtonian models described. To illustrate the general trends of the equations, Figure 3 shows the ratio of the load bearing capacity of each model to a Newtonian fluid. All models show a decrease in W with increasing shear (increase in V_x or a decrease in h) which is to be expected for shear thinning fluids. The differing slopes of each curve show that the model has a pronounced effect on the calculated load, and this emphasizes the importance of obtaining an accurate rheological equation to predict drive parameters.

Figure 3. Load bearing capacity for various rheological models normalized to a Newtonian fluid. (The power law exponent is 0.3, and for the Eyring model it was assumed that $\tau_E=\tau_L/3.5$).

A comparison of the power consumption for the different non-linear viscous models is illustrated in Figure 4. There is a decrease in power consumption with increasing shear, which is again attributed to shear thinning behavior.

Figure 4. Power consumption for various rheological models normalized to a Newtonian fluid ($n=0.3$, and $\tau_E=\tau_L/3.5$).

Typical bearings operate with separations of less than 20 nm and disk velocities on the order of 10 m/s. This confined geometry produces extremely high shear rates, and

Model	Load Bearing Capacity, W	Power Consumption, P_c
Newt	$L^2 BR_f^3 \mu \frac{V_x}{\Delta h^2}$	$LBR_f^2 \mu V_x \frac{V_x}{\Delta h}$
PL	$L^2 BR_f^3 \mu n \left(\frac{V_x}{\Delta h} \right)^{n-1} \frac{V_x}{\Delta h^2}$	$LBR_f^2 \mu n V_x \left(\frac{V_x}{\Delta h} \right)^n$
EY	$L^2 BR_f^3 \mu \left[\left(\frac{\mu V_x}{\tau_E \Delta h} \right)^2 + 1 \right]^{-1/2} \frac{V_x}{\Delta h^2}$	$LBR_f^2 \tau_E V_x \sinh^{-1} \left(\frac{\mu V_x}{\tau_E \Delta h} \right)$
BW	$L^2 BR_f^3 \mu e^{-\frac{\mu V_x}{\tau_L \Delta h}} \frac{V_x}{\Delta h^2}$	$LBR_f^2 \tau_L V_x \left[1 - e^{-\frac{\mu V_x}{\tau_L \Delta h}} \right]$
GW	$L^2 BR_f^3 \mu \operatorname{sech}^2 \left(\frac{\mu V_x}{\tau_L \Delta h} \right) \frac{V_x}{\Delta h^2}$	$LBR_f^2 \tau_L V_x \tanh \left(\frac{\mu V_x}{\tau_L \Delta h} \right)$
IH	$L^2 BR_f^3 \mu \left(\frac{\mu V_x}{\tau_L \Delta h} + 1 \right)^{-2} \frac{V_x}{\Delta h^2}$	$LBR_f^2 \tau_L V_x \left[1 - \frac{1}{\frac{\mu V_x}{\tau_L \Delta h} + 1} \right]$

Table 2. W and P_c for the non-Newtonian models given in Table 1. Results shown are exact to within a proportionality constant.

the bulk viscosity models described above may not accurately describe the behavior of the fluid. Ultra-thin fluid films may have viscosities which are orders of magnitude greater than in the bulk [10]. Therefore, Eqs. (13) and (18) based on bulk rheological data should be modified to account for confinement (or ultra-thin film) effects. In general, the apparent viscosity of a thin polymeric film is given as $\eta(\dot{\gamma}, h)$. If η is separable, i.e.,

$\eta = \eta^b(\dot{\gamma}) f(h)$ where the superscript b denotes bulk values, one can extract $f(h)$ from the rheological data for the thin liquid film, and W and P_c are trivially modified as

$$\frac{W}{W^b} = \frac{P_c}{P_c^b} = f(h) \quad (21)$$

If η is non-separable, then the simple scaling relationships will fail, and a complete analysis using Eqs. (3) and (4) must be performed.

VI. ACKNOWLEDGMENTS

This work was supported in part by the National Science Foundation under grant ECD-8907068. The financial

support from Hewlett-Packard Corporation is also greatly appreciated (PRP and MSJ).

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