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In-plane friction behaviour of a ferrofluid bearing☆

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ABSTRACT

Ferrofluid bearings have been demonstrated to be very interesting for precision positioning systems. The friction of these bearings is free of stick-slip which results in an increase of precision. More knowledge on the friction behaviour of these bearings is important for there application in precision positioning systems. This paper demonstrates that the friction of a ferrofluid bearing can be modelled by a viscous damper model and provides a basic model to predict the friction behaviour of a bearing design. The model consists of a summation of a Couette flow with a Poiseuille flow such that there is no net fluid transport under the bearing pads. The model is experimentally validated on a six degrees of freedom stage using ferrofluid bearings. A stiffness in the form of a closed-loop control gain is introduced in the system to create a resonance peak at the desired frequency. The damping coefficient can be identified from the peak height of the resonance, since the peak height is the ratio of total energy to dissipated energy in the system. The results show that the newly derived model can be used to make an estimate of the damping coefficient for small($\sim$ 1 mm) stroke translations. Furthermore, the model shows that the load capacity of a ferrofluid pocket bearing is affected during sliding.

1. Introduction

The repeatability of precision positioning systems can be improved by reducing the effects of stick-slip in system [1]. Stick-slip is the result of a spontaneous jerking motion which is introduced when overcoming the static friction coefficient between two sliding contacts. Bearing concepts like magnetic bearings, fluids bearings and flexures don't have this stick-slip effect but have other drawbacks like complexity, cost, or the storage of energy while moving.

Ferrofluid bearings, first proposed by Rosensweig et al. [2], provide a cost-effective alternative to these more conventional bearing systems. The bearing consist of a magnet and a ferrofluid that are attracted to each other forming a thin layer of ferrofluid inbetween the permanent magnet and the opposing bearing surface (Fig. 1).

The permanent magnet makes it a natural candidate for combination with Lorentz actuators, as demonstrated in various systems [3–15]. The result is a bearing that has distinct advantages for precision positioning systems, such as inherent stability, viscous friction, linear actuation, absence of external equipment, and no discernible stick slip effects. Furthermore, the carrier fluid can be chosen to suit the operating environment and the design allows for a compact, lightweight and cost effective solution.

Ferrofluid bearings have been successfully incorporated in precision positioning systems. Cafe [9,10] has built a six degrees of freedom(DoF) stage with nanometer accuracy, demonstrating that the bearing can be used in high precision positioning systems. Mok [13], Habib [11] and van Moorsel [15] have successfully implemented ferrofluid bearings in combination with low-cost sensor solutions, to capitalize on the cost-effectiveness.

Ferrofluid bearings can be divided into pressure bearings and pocket bearings. The load capacity of a ferrofluid pressure bearing is solely developed by the pressure in the fluid developed by the magnetic bodyforce [16]. The load capacity and stiffness behaviour of ferrofluid pocket bearings have recently been described in Refs. [17–19]. Though, this previous work does not yet include the effect of translating the bearing, nor does it describe the friction of the bearing. Due to this uncertainty that is introduced in the model, Café [9] and Habib [11] have put a large safety factor on the friction forces during the design of the system, resulting in a situation where the friction forces are dominating the disturbance forces.

This paper describes and experimentally validates a basic model of the in-plane friction behaviour of a ferrofluid bearing. It will do so by deriving a model describing the viscous damping forces of a ferrofluid bearing. The model will be experimentally validated on a demonstrator stage.

2. Theoretical bearing model

The forces that act on a ferrofluid bearing are found by deriving the flow field between two surfaces from the general Navier-Stokes

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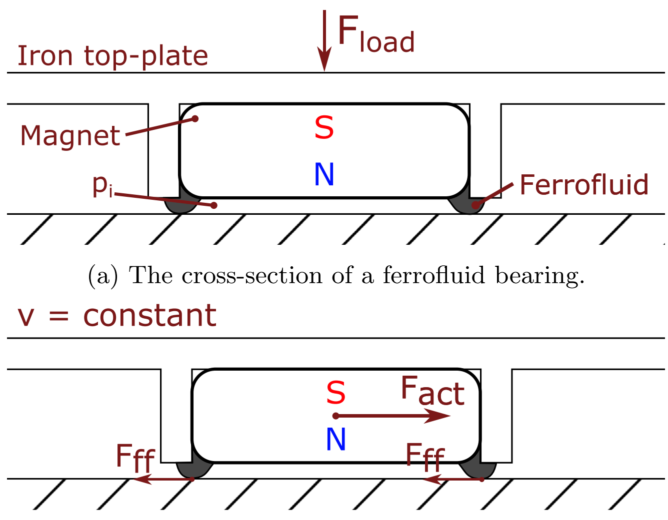

(a) The cross-section of a ferrofluid bearing.

(b) Actuator forces and friction forces.

Fig. 1. a shows how a load bearing ring is created by the permanent magnet and ferrofluid, while the iron top-plate increases the magnetic field intensity a the underside. b shows the actuation force and counteracting friction forces for a constant speed.

equation. The flow field is then used to determine the shear stresses in the system which can be related to the friction forces. An analysis of the viscosity of ferrofluids is added to verify the used viscosity model.

2.1. Viscosity

The viscosity of a ferrofluid changes when subjected to a magnetic field [20]. This happens due to two different effects: rotational viscosity and particle chain formation. The following section discusses the impact of these effects on the rheology of the fluid.

2.1.1. Rotational viscosity

The effect of rotational viscosity is caused by the alignment of the particles to the magnetic field. This results in a larger effective viscosity when the vorticity is perpendicular to the magnetic field. The viscosity of the fluid using spherical particles can be modelled with the following relation that uses η_c for carrier viscosity, ϕ for volumetric concentration, β for the angle between magnetic field and vorticity, μ_0 for magnetic permeability of vacuum, m for magnetic moment of a ferrofluid particle, H for magnetic field intensity, k for Boltzmann constant and T for temperature [21].

$$\eta = \eta_c\left(1 + \tfrac{5}{2}\phi + \tfrac{3}{2}\phi\tfrac{\alpha - \tanh\alpha}{\alpha + \tanh\alpha}\sin^2\beta\right)$$
$$\alpha = \tfrac{\mu_0 mH}{kT} \tag{1}$$

The first term of this equation presents the viscosity of the carrier fluid, the second term presents the increase in viscosity due to the suspension of particles and the third term presents the change in viscosity due to the magnetic field. For large values of α this relation has a maximum value of:

$$\eta_{max} = \eta_c\left(1 + \frac{5}{2}\phi + \frac{3}{2}\phi\right) = \eta_c(1 + 4\phi) \tag{2}$$

The viscosity of a ferrofluid is often given in the absence of a magnetic field, the relation for the viscosity then reduces to the Einstein formula [22]:

$$\eta_0 = \eta_c\left(1 + \frac{5}{2}\phi\right) \tag{3}$$

A typical value for the increase in viscosity caused by the effect of rotational viscosity can be calculated by combining relation (2) and (3) and assuming a typical concentration of about $\phi = 8\%vol$.

$$\frac{\eta_{max}}{\eta_0} = \frac{1 + 4\phi}{1 + \frac{5}{2}\phi} = \frac{1 + 4 \times 0.08}{1 + \frac{5}{2}0.08} = 1.1 \tag{4}$$

This relation shows that the increase in viscosity due to the magnetic attraction is in the order of 10%.

2.1.2. Particle chain formation

The particle chain formation, often referred to as the magneto-viscous effect [23], is the formation of chain like structures in the fluid due to the magnetic interaction between the particles. These structures are more difficult to rotate in the fluid resulting in a larger resistance to shear which results in an increase in effective viscosity [24]. Applying a magnetic field on the fluid increases the resistance to rotation even more resulting in an even further increase in viscosity. Shear forces in fluid might break the chains in the fluid resulting in a shear thinning effect. The formation of chains can be investigated by analysing the dipolar interaction parameter λ which is given with the following relation that uses M_0 for particle magnetization strength and V for particle volume.

$$\lambda = \frac{\mu_0 m^2}{4\pi kTd^3} = \frac{\mu_0 M_0^2 V}{24kT} \propto d^3 \tag{5}$$

Chain like structures will develop in the fluid when this parameter becomes larger than one. Increasing this parameter results in longer chains in the fluid [25]. The formula shows that λ increases with the diameter d of the particles resulting in only the larger particles contributing to the formation of chains. It has been shown that even a small concentration of large particles in the fluid can cause a high increase of viscosity [26]. For the models presented in this paper, it is key to choose a ferrofluid at which the dipolar interaction parameter is lower than one for all suspended magnetic particles.

2.2. Flow field

The geometry of the ferrofluid seal consists of a thin layer of fluid which is held fixed on the magnet against a moving counter surface (see Figs. 1 and 2). The derivation of the flow field starts with the general Navier-Stokes equations for incompressible Newtonian fluids, with an additional term ($\mu_0 M_s \nabla H$) describing the magnetic body forces. The assumption of an Newtonian fluid is reasonable for magnetic fluids with a small effect of rotational viscosity and a small dipolar interaction parameter λ. The relation uses $\vec{u}$ for fluid velocity, p for pressure, η for viscosity and $\vec{f}$ for body forces.

$$\rho\left(\frac{\delta\vec{u}}{\delta t} + \vec{u}\cdot\nabla\vec{u}\right) = -\nabla p + \eta\nabla^2\vec{u} + \mu_0 M_s\nabla H + \vec{f}$$
$$\nabla\cdot\vec{u} = 0 \tag{6}$$

For a typical bearing application, the Reynolds number in the flow can be shown to be small as is done in the following relation that uses L for the length of the bearing, U for its speed and ρ is the density of the ferrofluid.

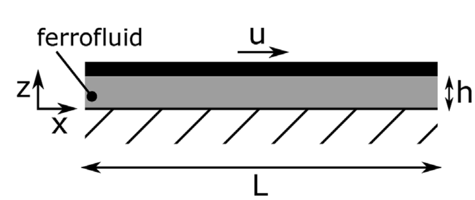

Fig. 2. Two large plates($L \gg h$) moving with respect to each other with velocity u and separated with a ferrofluid film with height h.

$$\frac{\partial p}{\partial x} = \eta\frac{\partial^2 u_x}{\partial z^2} + \mu_0 M_s\frac{\partial H}{\partial x} \tag{9}$$