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LONGITUDINAL SURFACE ROUGHNESS EFFECTS IN MAGNETIC FLUID LUBRICATED JOURNAL BEARINGS

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Key words: surface roughness, magnetic fluids, journal bearings, lubrication characteristics.

ABSTRACT

According to the magnetic fluid model incorporating with the stochastic model, the performance characteristics of a magnetic fluid lubricated short journal bearing with rough surfaces are investigated. A stochastic magnetic fluid based Reynolds equation has been derived. Analytical expressions for the bearing performances are also obtained. From the results obtained, the effects of longitudinal roughness patterns result in an increased load capacity as well as a reduced friction parameter and attitude angle. The amounts of increased load and reduced friction parameter are more pronounced for the bearing operating at higher values of the volume concentration, magnetic parameter, roughness parameter and eccentricity ratio.

I. INTRODUCTION

Journal bearings play an important role in many engineering applications such as turbo-machinery, machine tools, heat exchangers, and internal combustion engines. The classical analysis focused on the study of load capacity and friction parameter for the bearing operating at different eccentricity ratios, for example, the works by Barrett et al. (1980) and Capone et al. (1994). Owing to the development of modern engineering, the increasing use of magnetic fluids as lubricants has drawn great attention. Magnetic fluids are composed of three parts, namely, a base fluid, single domain of ferromagnetic nano-particles and a surfactant as Shliomis (1974). Magnetic fluids can be applied to various fields of engineering and biomedical problems, for example, power generation by Yamaguchia et al. (2008), inductive transducers by Popa et al. (1997), ceramic ball grinding by Childs et al. (1995), microchannel flows by Chang et al. (2007), ferrofluidic seals by Ochonski (1989) and drug targeting of diseases by Xu et al. (2005). In addition, magnetic fluids have also been applied to the lubrication of hydrodynamic journal bearings, such as the contributions by Motazeri (2008), Patel et al. (2010), and Lin et al. (2013). From their results, magnetic fluid lubricated journal bearings provide increased load carrying capacity as compared to conventional ones. However, these studies assume that the sliding surfaces of journal bearings are completely smooth. In practice, bearing surfaces are roughened as a result of the manufacturing process. Therefore, the effects of surface roughness should be included in the analysis of bearing performances. Applying the stochastic theory, a general Reynolds-type equation taking account of the effects of surface roughness in bearings has been derived (Christensen, 1969-1970). Accordingly, many scholars have applied this stochastic model to investigate the bearing characteristics of hydrodynamic journal bearings, for example, the studies by Christensen and Tonder (1973), Turaga et al. (1999) and Gururajana and Prakash (1999, 2002). According to their results, the effects of surface roughness provide significant influences on the lubrication performances characteristics as compared to the journal bearing with smooth surfaces. In the present study, the present study is mainly concerned with the surface roughness effects in a magnetic fluid lubricated journal bearing. In accordance with the magnetic fluid model of Shliomis (1974) incorporating with the stochastic model of Christensen (1969-1970), the influences of longitudinal surface structures on the lubrication characteristics of magnetic fluid based short journal bearings are presented in terms of the film pressure, the modified Sommerfeld number, the friction parameter and the attitude angle.

II. ANALYSIS

Fig. 1 describes the physical geometry of a magnetic fluid lubricated journal bearing with rough surfaces under an applied magnetic field $\vec{B}=(0,B_0,0)$. The inner journal with radius r_j is rotating with angular velocity ω within the outer bearing with length L. On the ground of the magnetic fluid model of Shliomis (1974), the magnetic fluid-based Reynolds equation governing the local film pressure p_L and the equation of local

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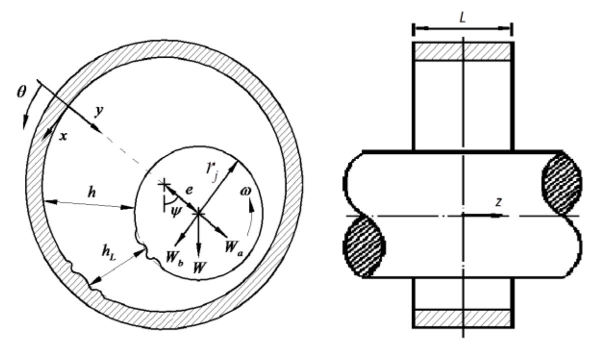

Fig. 1. Geometry of a magnetic fluid lubricated rough journal bearing.

velocity component u_L for a short journal bearing are given by Lin et al. (2013).

$$\frac{\partial}{\partial z}\left\{\frac{h_L{}^3}{\eta}\frac{\partial p_L}{\partial z}\right\}=6\omega(1+\beta)\frac{\partial h_L}{\partial\theta} \tag{1}$$

$$u_L=\frac{1}{2\eta(1+\beta)r_j}\frac{\partial p_L}{\partial\theta}(y^2-h_L y)+\frac{r_j\omega}{h_L}y \tag{2}$$

where h_L denotes the local film thickness, $\theta = x / r_j$ is the circumferential coordinate, η is the viscosity of the suspension, and

$$\beta=\frac{3}{2}C\frac{M-\tanh M}{M+\tanh M} \tag{3}$$

In this equation, C represents the volume concentration of ferromagnetic particles, M is the magnetic Langevin parameter describing the strength of external magnetic field B_0.

$$M=\frac{\alpha\mu_p B_0}{B_c T} \tag{4}$$

where α depicts the magnetic moment of a ferromagnetic particle, μ_p is the free space permeability, B_c is the Boltzmann constant and T describes the absolute temperature. For the present study, the bearing surfaces with one-dimensional longitudinal roughness structures are considered. In this situation, the roughness pattern is assumed to possess the form of narrow valleys and ridges running in the direction of rotation. Then, the local film thickness h_L can be represented by two parts described in the following.

$$h_L = h+\delta(z,\xi) \tag{5}$$

where h denotes the nominal smooth height of the film thickness, δ describes the random height measured from the smooth height, and ξ is a random variable representing the surface structure for longitudinal roughness. In accordance with the Christensen (1969-1970) stochastic theory of rough surfaces, a stochastic Reynolds equation for magnetic fluid based short journal bearings including the effects of longitudinal rough surfaces can be described as:

$$\frac{\partial}{\partial z}\left\{\frac{1}{E\left(h_L{}^{-3}\right)}\frac{1}{\eta}\frac{\partial p}{\partial z}\right\}=6\omega(1+\tau)\frac{\partial E(h_L)}{\partial\theta} \tag{6}$$

where $p = E(p_L)$ represents the mean film pressure, and the expectancy operation $E(h_L)$ is defined by

$$E(h_L)=\int_{\delta=-\infty}^{\infty} h_L\cdot g(\delta)d\delta \tag{7}$$

where $g(\delta)$ describes the density function for the stochastic height variable δ. Since most sliding surfaces display Gaussian roughness distribution, a polynomial form of the probability density distribution is selected as Christensen (1969-1970).

$$g(\delta)=\begin{cases}35(\gamma^2-\delta^2)^3/(32\gamma^7), & -\gamma\le\delta\le\gamma\\ 0, & elsewhere\end{cases} \tag{8}$$

where γ represents the half total range of the random height variable. Substituting equations (5), (7) and (8) into equation (6), and performing the integration, one can derive a stochastic magnetic fluid based Reynolds equation for short journal bearings including the surface roughness effects of longitudinal patterns.

$$\frac{\partial}{\partial z}\left\{(h^3-\frac{2}{3}\gamma^2 h)\frac{1}{\eta}\frac{\partial p}{\partial z}\right\}=6\omega(1+\beta)\frac{\partial h}{\partial\theta} \tag{9}$$

According to the study of Batchelor (1977), the viscosity of suspension increases with the volume fraction of ferromagnetic nano-particles, $\eta = \Phi\cdot\eta_m$.

$$\Phi = 1+2.5C+6.2C^2 \tag{10}$$

where η_m denotes the viscosity of main liquid. Introduce non-dimensional variables and parameters as follows.

$$z^* = \frac{z}{L},\ \lambda = \frac{L}{2r_j},\ p^* = \frac{pc_r^{\,2}}{\eta_m \omega r_j^{\,2}},\ \varepsilon = \frac{e}{c_r} \tag{11}$$

$$h^* = \frac{h}{c_r} = 1 + \varepsilon\cos\theta,\ R = \frac{\gamma}{c_r} \tag{12}$$

where c_r is the clearance between the bearing and the journal, and R denotes the surface roughness parameter. Then the stochastic magnetic fluid based Reynolds equation for a rough journal bearing can be written in a non-dimensional form.

$$\frac{\partial}{\partial z^*}\left\{\left(h^{*3} - \frac{2}{3}R^2 h^*\right)\frac{\partial p^*}{\partial z^*}\right\} = 24\lambda^2\Phi(1+\beta)(-\varepsilon\sin\theta) \tag{13}$$

III. BEARING PERFORMANCES

The boundary conditions for non-dimensional mean film pressure are: $p^* = 0$ at $z^* = \pm 1/2$ and $\partial p^* / \partial z^* = 0$ at $z^* = 0$. The non-dimensional mean pressure can be obtained after integrating the Reynolds equation subject to the boundary conditions.

$$p^* = \frac{12\lambda^2\Phi(1+\beta)(-\varepsilon\sin\theta)}{h^{*3} - 2R^2h^*/3}\cdot\left(z^{*2} - \frac{1}{4}\right) \tag{14}$$

The mean load capacity $W = (W_a + W_b)^{1/2}$ is obtained by integrating the mean film pressure. The mean load components along the line of center and perpendicular to the line of center are, respectively,

$$W_a = 2r_j\int_{z=0}^{L/2}\int_{\theta=0}^{\pi} p\cos\theta d\theta dz \tag{15}$$

$$W_b = 2r_j\int_{z=0}^{L/2}\int_{\theta=0}^{\pi} p\sin\theta d\theta dz \tag{16}$$

Using non-dimensional quantities, $W_a^{\ *} = W_a c_r^{\,2}/\eta_m\omega L r_j^{\,3}$ and $W_b^{\ *} = W_b c_r^{\,2}/\eta_m\omega L r_j^{\,3}$, one can achieve

$$W_a^{\ *} = 2\varepsilon\Phi(1+\tau)S_m\cdot\int_{\theta=0}^{\pi}\frac{\sin\theta\cos\theta d\theta}{h^{*3} - 2R^2h^*/3} \tag{17}$$

$$W_b^{\ *} = 2\varepsilon\Phi(1+\tau)S_m\cdot\int_{\theta=0}^{\pi}\frac{\sin^2\theta d\theta}{h^{*3} - 2R^2h^*/3} \tag{18}$$

where S_m represents the mean modified Sommerfeld number.

$$S_m = \frac{\eta_m\omega L^3 r_j}{4Wc_r^{\,2}} \tag{19}$$

The mean attitude angle ψ can be calculated from the components of mean load capacity.

$$\psi = \tan^{-1}\left(\frac{W_b^{\ *}}{W_a^{\ *}}\right) \tag{20}$$

The local shear stress on the moving surface in the sliding direction is

$$\tau_{Lh_L} = \eta\frac{\partial u_L}{\partial y}\bigg|_{y=h_L} = \frac{h_L}{2(1+\beta)r_j}\frac{\partial p_L}{\partial\theta} + \Phi\eta_m\frac{\omega r_j}{h_L} \tag{21}$$

Taking the expected value gives the mean shear stress considering the surface roughness effects for longitudinal patterns [13].

$$\tau = E[\tau_{Lh_L}] = \frac{E[h_L]}{2(1+\beta)r_j}\frac{\partial p}{\partial\theta} + \Phi\eta_m\omega r_j E\left[\frac{1}{h_L}\right] \tag{22}$$

The mean friction force F is calculated by integrating the mean shear stress around the journal surface. Thereafter, the mean friction parameter f_p can be obtained.

$$F = 2r_j\int_{z=0}^{L/2}\int_{\theta=0}^{2\pi}\tau d\theta dz \tag{23}$$

$$f_p = \frac{F}{W}\cdot\frac{r_j}{c_r} \tag{24}$$

Performing the integration and arranging the equation, one can derive the expression for the mean friction parameter.

$$f_p = \Phi S_m\left\{\begin{array}{l}\dfrac{2\pi}{\lambda^2\sqrt{1-\varepsilon^2}} + \dfrac{R^2}{9\lambda^2}\displaystyle\int_{\theta=0}^{2\pi}\frac{d\theta}{h^{*3}} \\ + \varepsilon^2\displaystyle\int_{\theta=0}^{2\pi}\frac{\sin^2\theta d\theta}{h^{*3} - 2R^2h^*/3}\end{array}\right\} \tag{25}$$

IV. RESULTS AND DISCUSSION

The effects of longitudinal surface roughness on the performance characteristics of short journal bearings lubricated with magnetic fluids under an applied magnetic field are investigated in the present study. To analyze the lubrication performances, parameters are illustrated as follows:

Length-to-diameter ratio: $\lambda = L/2r_j = 0.5$;

Volume concentration parameter: $C = 0\text{-}0.24$;

Langevin parameter: $M = \alpha\mu_p B_0 / B_c T = 0\text{-}10$;

Surface roughness parameter: $R = \gamma / c_r = 0\text{-}0.25$.

Fig. 2 shows the film pressure versus circumferential coordinate (deg.) for different values of the volume concentration C, the magnetic parameter M and the roughness parameter R at the eccentricity ratio $\varepsilon = 0.5$. Comparing with the case of smooth surfaces with a non-magnetic fluid, the bearing lubricated with a magnetic fluid ($C = 0.12$) under an applied magnetic field ($M = 5$) yields a higher film pressure. Increasing values of the volume concentration ($C = 0.24$) and the magnetic field ($M = 10$) enlarges the increments of the pressure. When the influences of surface roughness are included, the roughness structures of longitudinal patterns ($R = 0.1$) results in an increased pressure. Furthermore, larger increments of the film pressure are obtained for the bearing with a higher roughness parameter ($R = 0.2$). Fig. 3 displays the film pressure versus circumferential coordinate (deg.) for different values of C, M and R at the eccentricity ratio $\varepsilon = 0.6$ and 0.7. It is observed that the effects of surface roughness ($R = 0.1$, 0.2) provide higher values of the film pressure for the magnetic fluid lubricated bearings with a magnetic field ($C = 0.24$, $M = 10$) as compared to the case of smooth surfaces ($R = 0$). In addition, the influences of longitudinal surface roughness on the film pressure are more pronounced for the bearing operating at a higher value of the eccentricity ratio ($\varepsilon = 0.7$).

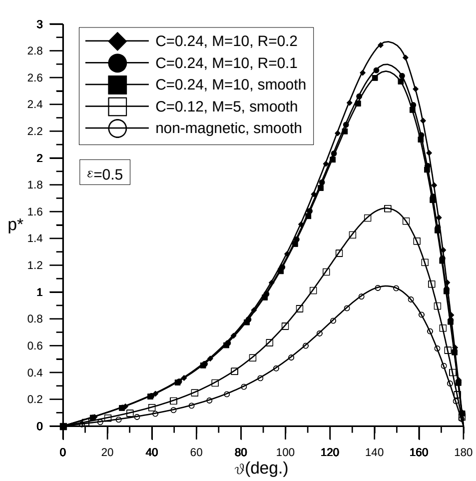

Fig. 2. Film pressure versus circumferential coordinate for different C, M and R at $\varepsilon = 0.5$.

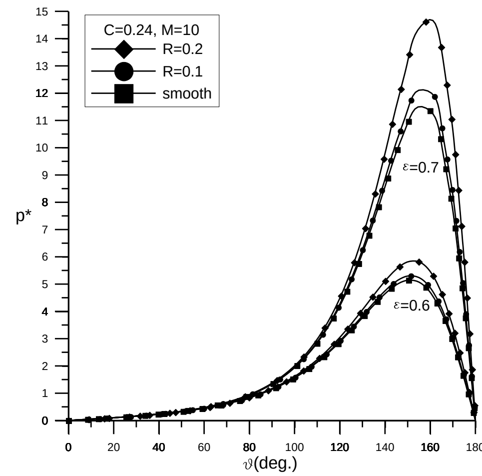

Fig. 3. Film pressure versus circumferential coordinate for different C, M and R at $\varepsilon = 0.6$ and 0.7.

Fig. 4 presents the modified Sommerfeld number versus roughness parameter for different values of the volume concentration C and the magnetic Langevin parameter M at the eccentricity ratio $\varepsilon = 0.6$. The journal bearings with magnetic fluids with applied magnetic fields ($C = 0.06$, $M = 5$; $C = 0.12$, $M = 5$; $C = 0.12$, $M = 10$; $C = 0.18$, $M = 10$; $C = 0.24$, $M = 10$) are seen to provide decreased values of S_m as compared to the bearing lubricated with non-magnetic fluid. It is also observed that the magnetic fluid lubricated rough ($R > 0$) bearings result in a smaller value of the modified Sommerfeld number when comparing with the case with smooth surfaces ($R = 0$). In addition, increasing the value of the roughness parameter R increases the surface roughness effects of longitudinal patterns on the values of S_m. In other words, the influences of longitudinal surface roughness provide higher values of the load capacity for the journal bearings lubricated with magnetic fluids under an applied magnetic field.

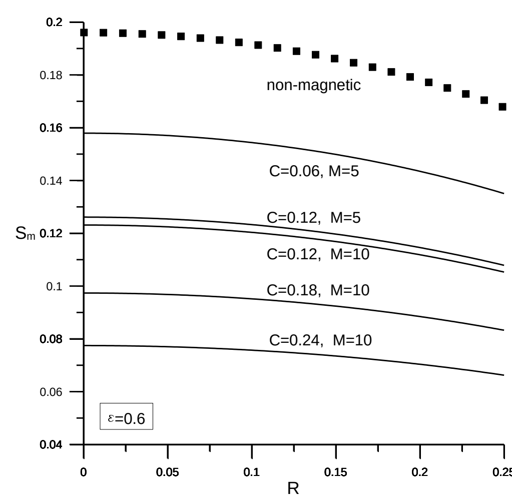

Fig. 4. Modified Sommerfeld number versus roughness parameter for different C and M at $\varepsilon = 0.6$.

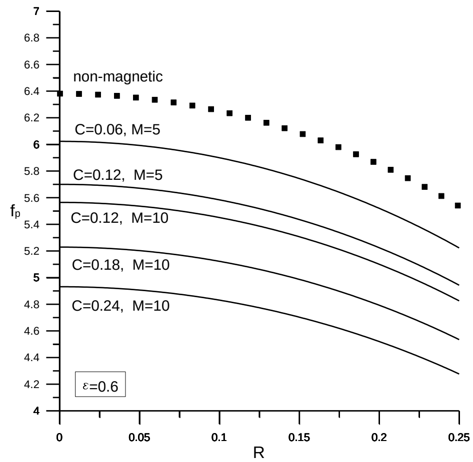

Fig. 5. Friction parameter versus roughness parameter for different C and M at $\varepsilon = 0.6$.

Fig. 5 describes the friction parameter versus roughness parameter for different values of C and M at $\varepsilon = 0.6$. It shows that the friction parameter decreases with increasing values of the roughness parameter. Comparing with the non-magnetic fluid lubricated case, the effects of surface roughness on the values of f_p are more emphasized for the bearing operating with a larger value of the volume concentration and the magnetic parameter ($C = 0.24$, $M = 10$).

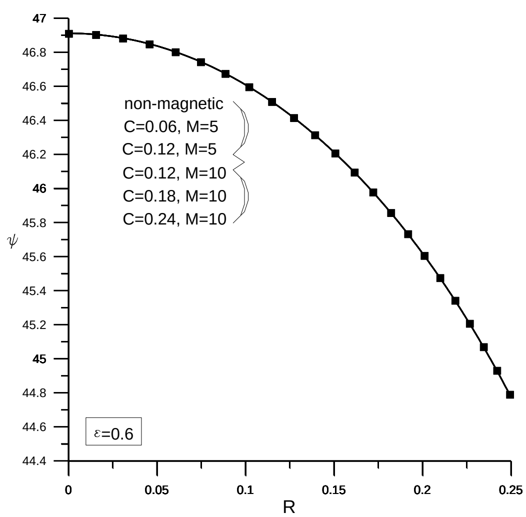

Fig. 6. Attitude angle (deg.) versus roughness parameter for different C and M at $\varepsilon = 0.6$.

Fig. 6 depicts the attitude angle (deg.) versus roughness parameter for different values of C and M at the eccentricity ratio $\varepsilon = 0.6$. It shows that the magnetic fluid lubricated bearings do not alter the values of ψ as compared to the non-magnetic fluid lubricated bearing. However, the roughness effects of longitudinal surface structures decrease the attitude angle especially for the bearing with a larger roughness parameter.

Table 1. Illustrative data for the bearing system.

Volume of ferromagnetic particles including surfactants	V_p	$1.3636 \times 10^{-4} m^3$
Volume of the main liquid	V_m	$1.00 \times 10^{-3} m^3$
Magnetic moment of a ferromagnetic particle	m	$2 \times 10^{-19}\ A \cdot m^2$
Free space permeability,	μ_p	$4\pi \times 10^{-7}$ $kg \cdot m \cdot s^{-2} \cdot A^{-2}$
Applied magnetic field	B_0	$1.63627 \times 10^5\ A \cdot m^{-1}$
Boltzmann constant	B_c	1.38×10^{-23} $kg \cdot m^2 \cdot s^{-2} \cdot K^{-1}$
Absolute temperature	T	$298\,K$
Journal radius	r_j	$0.04\ m$
Bearing length	L	$0.04\ m$
Radial clearance	c_r	$1.00 \times 10^{-4}\ m$
Half random roughness	γ	$2.00 \times 10^{-5}\ m$

To show the values of C, M, and R in the real situation, Table 1 lists the illustrative data for the bearing system. From these data, one can obtain the parameters.

Volume concentration parameter: $C = V_p/(V_p + V_p) = 0.12$;

Langevin parameter: $M = \alpha\mu_p B_0 / B_c T = 10$;

Surface roughness parameter: $R = \gamma / c_r = 0.2$.

The readers are also suggested to refer typical values as illustrated in a previous study on magnetic fluid-lubricated circular squeeze films Lin (2012). Tables 2-3 present numerical values of the load capacity and the friction parameter of magnetic fluid lubricated rough journal bearings for different values of C, M and R at the eccentricity ratio $\varepsilon = 0.5$ and 0.7, respectively. It is shown that the quantitative influences of longitudinal surface roughness on short journal bearings lubricated with magnetic fluids in the presence of applied magnetic fields are further emphasized for the bearing operating at higher values of the volume concentration, the magnetic parameter, the roughness parameter and the eccentricity ratio.

Table 2. Load capacity of magnetic fluid lubricated rough journal bearings.

Load capacity W^*						
	Smooth		R =0.1		R =0.2	
C	ε =0.5	ε =0.7	ε =0.5	ε =0.7	ε =0.5	ε =0.7
M =0						
0	.754	2.368	.766	2.461	.804	2.812
0.06	.884	2.776	.898	2.885	.943	3.297

0.12	1.048	3.290	1.064	3.419	1.116	4.259
M =5						
0	.754	2.368	.766	2.461	.804	2.812
0.06	.937	2.943	.952	3.058	1.000	3.495
0.12	1.174	3.685	1.192	3.830	1.252	4.376
M =10						
0	.754	2.368	.766	2.461	.804	2.812
0.06	.949	2.981	.964	3.098	1.012	3.510
0.12	1.202	3.775	1.221	3.923	1.282	4.482

Table 3. Friction parameter of magnetic fluid lubricated rough journal bearings.

Friction parameter f_p						
	Smooth		R =0.1		R =0.2	
C	ε =0.5	ε =0.7	ε =0.5	ε =0.7	ε =0.5	ε =0.7
M =0						
0	9.820	3.939	9.682	3.806	9.259	3.374
0.06	9.820	3.939	9.682	3.806	9.259	3.374
0.12	9.820	3.939	9.682	3.806	9.259	3.374
M =5						
0	9.820	3.939	9.682	3.806	9.259	3.374
0.06	9.264	3.716	9.134	3.590	8.735	3.183
0.12	8.768	3.517	8.644	3.398	8.267	3.013
M =10						
0	9.820	3.939	9.682	3.806	9.259	3.374
0.06	9.147	3.669	9.018	3.545	8.624	3.143
0.12	8.560	3.433	8.439	3.317	8.071	2.941

V. CONCLUSIONS

A stochastic magnetic fluid based Reynolds equation for a short journal bearing with rough surfaces of longitudinal structures has been derived in accordance with the magnetic fluid model of Shliomis (1974) incorporating with the stochastic model of Christensen (1969-1970). Analytical expressions for the bearing performances are also included. Based upon the results obtained, the effects of longitudinal roughness patterns signify an increase in the load capacity as well as a reduction in the friction parameter and the attitude angle. Comparing with the non-magnetic smooth-surface case, the amounts of the increased load and the reduced friction parameter are more pronounced for the journal bearing operating at a higher value of the volume concentration, the magnetic parameter, the roughness parameter and the eccentricity ratio. For example from the results in Tables 2 and 3, the increased amounts of the non-dimensional load are 0.198 (at $C=$ 0.06, $M=$ 5, $R=$ 0.1, $\varepsilon=$ 0.5) and 4.482-0.754=3.728 (at $C=$ 0.12, $M=$10, $R=$0.2, $\varepsilon=$0.7); and, the reduced amounts of the friction parameter are 0.686 (at $C=$ 0.06, $M=$ 5, $R=$0.1, $\varepsilon=$0.5) and 6.879 (at $C=$0.12, $M=$10, $R=$0.2, $\varepsilon=$ 0.7) as compared to the non-magnetic smooth-surface situation.

VI. ACKKNOWLEDGEMENT

This study is extended from the support by the Ministry of Science and Technology of Republic of China: MOST 103-2221-E-253-001-.

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