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Stability analysis of rough journal bearings under TEHL with non-Newtonian lubricants

Mongkol Mongkolwongrojn *, Chatchai Aiumpronsin

Department of Mechanical Engineering, Faculty of Engineering, King Mongkut's Institute of Technology Ladkrabang, Bangkok 10520, Thailand

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A B S T R A C T

The combined effects of surface roughness and lubricants rheology on stability of a rigid rotor supported on finite journal bearing under thermal elastohydrodynamic lubrication have been investigated using the transient method. The newly derived time dependent modified Reynolds and the adiabatic energy equations were formulated using a non-Newtonian Carreau viscosity model. The simultaneous systems of modified Reynolds equation, elasticity equation, energy equation, and the rotor motion equation with initial conditions were solved numerically using multigrid multi-level method with full approximation technique. From the characteristic equation, the instability threshold is then obtained with various surface roughness parameters and the elastic modulus of the bearing liner materials. The results show that stability of the bearing system deteriorates with decreasing both the power law exponent and the elastic modulus of bearing liner material. The rough surface journal bearing with transverse pattern under TEHL regime exhibits better stability when compared with the rough surface journal bearing with longitudinal pattern.

1. Introduction

In the design of high speed journal bearings operating under heavy load, the consideration of journal bearing with rough surface in TEHL and flow rheology is significant to predict the stability accurately especially when the film thickness is in the same order as the surface roughness. Few theoretical studies of the journal bearing under elastohydrodynamic lubrication at steady state with Newtonian lubricants are available. Lahmar et al. [1], Mokhimer et al. [2], Taylor and O'Callaghan [3], Oh and Huebner [4], Jain et al. [5] investigated the significant effects of bearing deformation on the bearing performance characteristics with rigid bearing. Thermal effects in thermoelastohydrodynamic (TEHD) lubrication are important due to the strong dependence of viscosity on temperature. Fillon et al. [6,7] and Bouchoule et al. [8] analyzed the TEHD behavior of tilting-pad journal bearings experimentally and theoretically. The comparisons between the theoretical results with experimental data showed that both thermal and the elastic deformation are very important to predict bearing performance. In 2004, Bouyer and Fillon [9] presented the incompressible thermoelastohydrodynamic analysis due to the film pressure which leads to the significant effect on the static characteristics of a plain journal bearing. Transient problems for journal bearing have been studied by numerous investigators. Hashimoto and Mongkolwongrojn [10] studied the dynamic characteristic of turbulent partial journal bearing under hydrodynamic lubrication. Oh and Goenka [11], Van der Tempel et al. [12], Majumdar et al. [13], Prabhakaran Nair et al. [14] and Piffeteau et al. [15] studied the effect of elastic distortion on dynamically loaded journal bearings.

It is now well accepted that the surface roughness and surface pattern has a significant effect on the stability characteristics of hydrodynamic journal bearing. Majumdar and Ghosh [16] analyzed the stability of rough journal bearing using the perturbation technique. Their results shown that the stability improves significantly for a very rough surface bearing with smooth journal surface. Ramesh and Majumdar [17] used the non-linear transient method to show the effects of surface roughness on stability of hydrodynamic journal bearing. Turaga et al. [18], and Gururajan and Prakash [19] studied the influence of the surface roughness on bearing characteristics. Zhang and Cheng [20], and Raghunandana and Majumdar [21] presented the effect of shear thinning fluid on THD of dynamically loaded journal bearing. Kim and Cho [22] developed the 3D average flow model by considering the elastic deformation of pad. And recently, Meng et al. [23] studied the effects of elastic deformation, inter-asperity cavitation and temperature on flow factors.

Modern lubricants contain polymeric additives with shear thinning property. The present study is to incorporate the effects of the shear thinning behavior of lubricant, elastic deformation of the bearing liner, the temperature and pressure dependent

* Corresponding author.
E-mail addresses: kmmongko@kmitl.ac.th (M. Mongkolwongrojn), chataium@gmail.com (C. Aiumpornsin).

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Nomenclature

A_h	amplitude roughness, m
B	dimensionless damping coefficient
c	bearing radial clearance, m
c_f	specific heat of lubricant, J/(kg K)
e	eccentricity of journal bearing, m
h	oil film thickness, m, $\overline{h}=h/c$
I	second invariant of the strain rate tensor, $\mathrm{I}^*=((u_2-u_1)^2+(v_2-v_1)^2)/h^2$
K	dimensionless spring coefficient
m	mass of journal, kg, $M=c\omega^2 m/W$
n	the power law exponent
p	fluid film pressure, Pa
r	journal radius, m
t	time, s, $\tau=\omega_p t$
u	velocity of lubricant in x-direction, m/s
v	velocity of lubricant in y-direction, m/s
x	coordinate in circumferential direction
y	coordinate in axial direction
z	coordinate across fluid film direction
E	elastic modulus of bearing liner, Pa
F_Φ	fluid force component perpendicular to line of center, N
F_R	fluid force component along line of center, N
L	journal bearing length along the axial axis, m
N	rotational speed, rpm
T_m	mean temperature of lubricant across the film, K
T_0	inlet fluid temperature, K
W	resultant fluid force, N
β	coefficient of thermal expansion, 1/K
γ	viscosity–temperature coefficient, 1/K
ρ	density of the lubricant, kg/m^3
ρ_0	density of lubricant at ambient pressure and inlet temperature, kg/m^3
υ	Poisson ratio
Ω	whirl ratio, $\Omega=\omega_p/\omega$
ω	angular velocity of journal, rad/s
ω_p	angular velocity of whirl, rad/s
ε	eccentricity ratio of journal bearing
μ	viscosity, Pa s
μ_0	limiting low shear viscosity, Pa s
μ_∞	limiting high shear viscosity, Pa s
λ	a characteristic relaxation time constant, s
θ	circumferential angle measured from the vertical axis, rad
Φ	attitude angle, rad
λ_l	wave length of roughness, m

characteristics of lubricant viscosity, and the surface roughness of bearing liner in order to analyze the stability of thermal elastohydrodynamic journal bearing. The stability of rough journal bearing was examined using both the perturbation technique and non-linear transient method.

2. Theory

In the present lubrication analysis a full circular journal bearing configuration is considered. The assumptions in the theory of thermal elastohydrodynamic lubrication are:

(1) inertia and body forces of the lubricant are neglected;
(2) both bearing and rotor are rigid;
(3) no slip condition exists;
(4) the fluid flow is laminar;
(5) pressure variation across the film thickness is neglected.

2.1. Reynolds equation

In this research the non-Newtonian lubricant can be approximated using a Carreau viscosity model as

$$\mu=\mu_\infty+(\mu_0-\mu_\infty)(1+\lambda^2 I)^{(n-1)/2} \tag{1}$$

where

$$I=\left(\frac{\partial u}{\partial z}\right)^2+\left(\frac{\partial v}{\partial z}\right)^2 \tag{2}$$

Adopting the perturbation method described by Dien [24], the time dependent modified Reynolds equation with Carreau viscosity model can be formulated as

$$\frac{\partial}{\partial x}\left(\frac{\rho h^3}{\eta}\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial y}\left(\frac{\rho h^3}{\mu^*}\frac{\partial p}{\partial y}\right)=6\omega r\left(1-\frac{2}{\omega}\dot{\Phi}\right)\frac{\partial \rho h}{\partial x}+12\rho\dot{e}\cos(\theta-\Phi) \tag{3}$$

where

$$\eta=\mu^*+2\left(\frac{u_2-u_1}{h}\right)^2(\mu_0-\mu_\infty)\left(\frac{n-1}{2}\right)\left(1+\lambda^2\left(\frac{u_2-u_1}{h}\right)^2\right)^{(n-3)/2}\lambda^2 \tag{4}$$

$$\mu^*=\mu(I^*)=\mu_\infty+(\mu_0-\mu_\infty)(1+\lambda^2 I^*)^{(n-1)/2} \tag{5}$$

where $x=r\theta$ and boundary conditions are:

$$p(\theta=0)=0,\quad p(y=0)=p(y=L)=0\quad\text{and}\quad p(\theta^*)=\frac{\partial p}{\partial\theta}\big|_{\theta^*}=0$$

2.2. Energy equation

The mean film temperature of the non-Newtonian Carreau lubricant can be obtained by integrating the energy equation over the film thickness. Assuming adiabatic conditions, the energy equation using Carreau viscosity model can be newly formulated as

$$\begin{aligned}&\left(\rho c_f\frac{u_2}{2}h-\rho c_f\frac{h^3}{12\eta}\frac{\partial p}{\partial x}\right)\frac{\partial T_m}{\partial x}+\left(-\rho c_f\frac{h^3}{12\mu^*}\frac{\partial p}{\partial y}\right)\frac{\partial T_m}{\partial y}\\&\quad=\mu^*\left(\frac{u_2^2}{h}\right)+\left(\beta h T_m\frac{u_2}{2}\right)\frac{\partial p}{\partial x}+\left(\frac{h^3}{12\eta}-\beta T_m\frac{h^3}{12\eta}\right)\left(\frac{\partial p}{\partial x}\right)^2\\&\qquad+\left(\frac{h^3}{12\mu^*}-\beta T_m\frac{h^3}{12\mu^*}\right)\left(\frac{\partial p}{\partial y}\right)^2+\beta h T_m\frac{\partial p}{\partial t}-\rho c_f h\frac{\partial T_m}{\partial t}\end{aligned} \tag{6}$$

2.3. Viscosity–pressure–temperature relationship

A general relation proposed by Roelands [25] was used in this study and the viscosity μ^* can be expressed as

$$\mu^*=\mu_0^*\exp\{(\ln\mu_0+9.67)(-1+(1+5.1\times10^{-9}p)^z)-\gamma(T-T_0)\} \tag{7}$$

where μ_0^* is the viscosity at reference temperature and at atmospheric pressure.