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The effect of lubrication film thickness on thermoelastic instability under fluid lubricating condition

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ABSTRACT

An idealized model consisting of a thermal conductor and a thermal insulator separated by a thin layer of lubricating fluid is developed to investigate thermoelastic instability with fluid lubrication. The governing equations are solved for the critical speed. A new dimensionless parameter H_0 is defined to predict the critical speed. Furthermore, the effects of various materials and the wavelength of perturbations on thermoelastic instability are discussed. It has been found that the migration speed of hot spots is nonzero, but typically very slow compared with the sliding speed and the relation between the critical speed and the fluid film thickness is non-linear. In addition, a material with low elastic modulus, low thermal expansion coefficient and high thermal conductivity will experience a high critical speed.

1. Introduction

Various kinds of mechanical frictional components with lubrication, such as sealing parts, bearings and clutches, are impressionable to one type of the local high temperature on the surface named hot spots which may cause a surface damage. Hot spots are caused by a type of instability during the slipping and friction process when the sliding speed is in excess of a threshold value that relies on the wavelength of the perturbation in the system. This instability, which is caused by frictional heat, elastic deformation and cooling effect, is named thermoelastic instability or TEI. This phenomenon was first introduced by Barber [1]. A theoretical model composed of two solid surfaces sliding with each other was developed to investigate TEI in frictional systems. A perturbation of the nominally uniform contact pressure on the interface of the rubbing pair will lead to a nonuniform distribution of frictional heat and a nonuniform temperature field. The subsequent thermal expansion will transform the distribution of the contact pressure in turn. This process that moves in a loop causes hot spots in some cases. A multitude of researches have been done for TEI in dry frictional systems [2–5].

For frictional parts with lubrication, the heat generation that has a significant influence on TEI makes a conspicuous difference from the non-lubricated friction. During the working process, the heat in wet frictional parts is produced totally or partly by film shearing. Though TEI may be alleviated because of not only the less quantity of heat compared with dry fiction but also the cooling effect of fluid, hot spots can still appear on friction surfaces under certain conditions. Evidence of hot spots from an experiment for wet multi-disk clutches is shown in Fig. 1, where the dark areas represent the regions that have experienced high local temperature, i.e., hot spots.

TEI in frictional systems with lubrication was first examined by Banerjee and Burton [6] who developed a theoretical model composed of a thermal conductor and an insulator sliding along the interface in the presence of a liquid lubricating film. An equation for calculating the critical speed of lubricative face type seals was proposed by Banerjee as $V_c = h_0\lambda\sqrt{K/\mu\alpha}$, where V_c, h_0, μ, λ, K, and α represent the critical speed, nominal fluid film thickness, fluid viscosity, wavenumber of the perturbation, thermal conductivity and thermal expansion coefficient of metal, respectively. Then Jang and Khonsari [7–9] extend Banerjee's work by considering surface roughness.

According to their work, we may make a prediction of the critical speed for wet frictional mechanical components during fluid lubrication period. However their estimation still shows several deficiencies, e.g., their results are mainly based upon stationary wave solutions, i.e., the perturbation is presumed to be non-moving relative to the conductor. In reality, the perturbation moves at a relatively low speed with respect to the good thermal conductor, but not zero. Under an assumption of stationary perturbation, the fluid pressure and convection effects become small, therefore this assumption, non-moving perturbation, may lead to neglecting some significant impact factors on TEI and the

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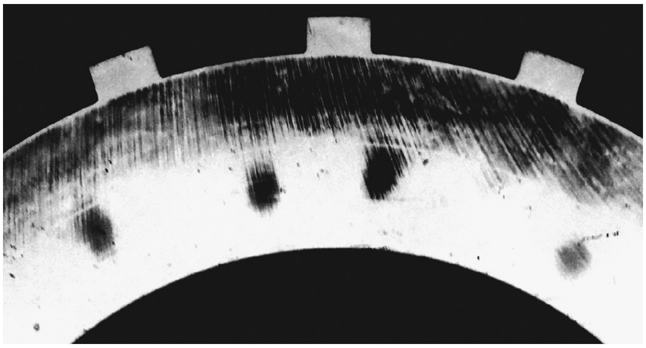

Fig. 1. Evidence of hot spots in a wet multi-disk clutch.

solution can be viewed as an approximate prediction for the critical speed.

In the current paper, we will examine the influence of lubricant film thickness on the critical speed of TEI on the basis of the moving solutions and predict a reasonable critical speed. For the purpose of evaluating the effect of finite disk thickness on TEI in the dry fiction system, Lee and Barber [3] introduced a model consisting of two semi-infinite planes and a layer with a finite thickness. Their results, which have been confirmed by Yi et al. [10], indicate that with the layer geometry there is a preferred wavelength of TEI. In an attempt to examine the effect of lubricating fluid film thickness on TEI, we introduce a fluid with finite thickness between the friction surfaces. In Lee's work, there is a minimum critical speed when $\lambda a = 0.2$, where a represents the half layer thickness. In this paper, we will demonstrate there is also a minimum point, which is being driven by the film thickness, on the critical speed curve.

2. Model

Taking the automotive wet clutch as our research object, the friction disk is often composed of a type of paper based material while the mating disk is made of steel. Since the thermal conductivity of the paper based material is about 10 times smaller than that of steel, we can make a hypothesis that the wet clutch system can be simplified as a thermal conductor–insulator system. This assumption has been adopted by Banerjee [6] and Jang [8] to examine the critical speed of the facing type seal.

A simplified schematic model is shown in Fig. 2. Following Lee's work [3], we introduce a spatially sinusoidal perturbation, moving at a speed c to the positive x-direction, in a fluid pressure, which grows exponentially with time. More general patterns of perturbations can be obtained by means of superposition, which is equivalent to the Fourier transform. The pressure perturbation will lead to other perturbations in the system, e.g., stresses, temperature fields, fluid thickness, therefore all perturbations will have a similar expression. If we introduce a coordinate system (x,y) moving with the perturbation field, the total fluid pressure can be represented as

$$p_0 + \Re\{p_1 \exp(bt + \imath\lambda x)\} \tag{1}$$

where p_0, p_1, b, and t represent the unperturbed pressure, perturbed pressure, growth rate of the perturbation and time respectively, $\imath = \sqrt{-1}$.

We assume that the sinusoidal surface (conductor) moves at a speed V to the positive x-direction while the plane surface (insulator) is stationary, as shown in Fig. 2. With respect to the two disks, the relative velocities of the perturbation are, separately, $c_1 = c$ and $c_2 = c - V$ to the positive x-direction. In order to simplify the expression of perturbations, we introduce two local

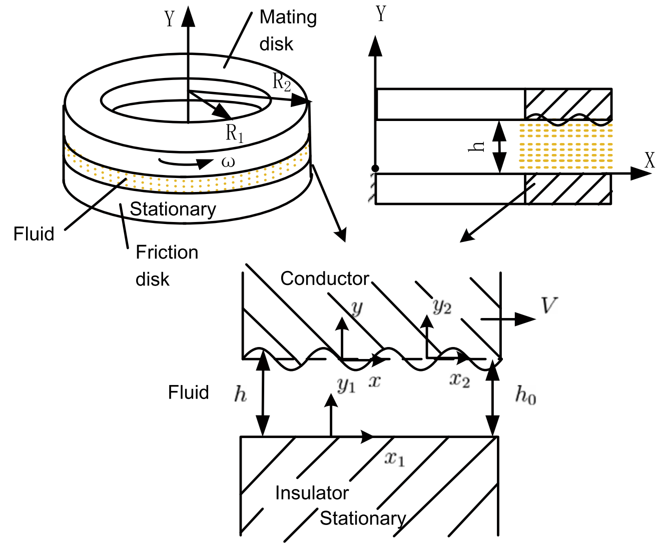

Fig. 2. Schematic diagram.

coordinate systems: (x_1,y_1) and (x_2,y_2) that are standing with disk 1 and disk 2 separately, as shown in Fig. 2. The relations of different coordinate systems are

$$x_1 = x + c_1 t, \quad x_2 = x + c_2 t \tag{2}$$

$$y_1 = y + h_0, \quad y_2 = y \tag{3}$$

$$h = h_0 + \Re\{h_1 \exp(bt + \imath\lambda x)\} \tag{4}$$

where h, h_0, and h_1 represent, respectively, the entire fluid film thickness, the nominal fluid film thickness and the perturbation of fluid film thickness.

The gap between these two surfaces is full of lubricating fluid with constant viscosity. Our research is based on the fluid lubrication condition and the influence of surface roughness is neglected. To satisfy the assumption, the fluid thickness needs to be at least three times greater than the surface roughness, i.e., $h_0 \geq 3Ra$ [8], where Ra represents the surface roughness.

Some assumptions for fluid have been made: 1, the flow is laminar; 2, the gravity and inertia forces can be ignored; 3, the compressibility of fluid is negligible; 4, the fluid is Newtonian; 5, the fluid pressure is constant across the film thickness; and 6, there is no slip between the fluid and the surfaces of disks.

2.1. Fluid pressure and stress

In the research on fluid, we will use the local coordinate system (x_1,y_1) during calculation.

2.1.1. Equilibrium

On the basis of previous assumptions, the Navier–Stokes equation can be simplified to a two-dimensional equation, as

$$\frac{\partial^2 v_x}{\partial y_1^2} = \frac{1}{\mu}\frac{\partial p}{\partial x_1}, \tag{5}$$

where v_x, μ, and p are, respectively, the fluid velocity in the x-direction, the viscosity, and the pressure of fluid. According to Eq. (1), the pressure perturbation can be expressed as $\Re\{p_1 \exp(bt + \imath\lambda(x_1 - c_1 t))\}$. Hence the simplified Navier–Stokes equation (5) can be rewritten as

$$\frac{\partial^2 v_x}{\partial y_1^2} = \Re\left\{\frac{\imath\lambda p_1}{\mu} \exp(bt + \imath\lambda(x_1 - c_1 t))\right\}. \tag{6}$$