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Finite element modeling on local hot banding of sealing ring

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ABSTRACT

A finite element method was presented to investigate local hot banding for sealing rings in a wet clutch. A discrete model of the sealing pair was established, and the critical speed of hot banding was obtained by numerical computation. The relationships between the critical speed and the frictional coefficient, elastic modulus, thermal conductivity, thermal expansion coefficient, and sealing ring size were determined. Experimental investigations were performed using a seal test rig, with comparisons of the calculated and experimental critical speed results made to enable an assessment of the proposed method's accuracy. The numerical and experimental results indicate the effectiveness of the finite element discrete model in simulating the hot banding of sealing rings.

1. Introduction

Seals are basic and essential parts in mechanical equipment, with relatively simple structures. In automobiles, sealing rings are applied in the rotating connections in automatic automotive transmissions, such as double clutch transmissions (DCT), automated transmissions (AT), continuously variable transmissions (CVT) and so on. These rotating connections utilize sealing rings to sustain the oil pressure of the transmission. Such sealing rings in the transmission can maintain the pressure supply of the rotating elements and prevent oil leakage. Sealing performance is an important evaluating indicator of overall transmission performance [1].

When developing new types of transmissions, we find that local thermal failures occur on the sealing surface, which leads to locally severe wear and a high leak rate. The failure of a sealing ring results in the loss of supplying pressure and ultimately causes transmission failure. Therefore, it is necessary to study the formation of local high temperatures on the sealing surface and relevant influencing factors.

Non-uniform temperature distributions and local high temperatures are easily generated on the sealing surface under high pressures and speeds. According to previous studies, the instability phenomenon of a sliding contact is the most important cause contributing to these effects [2,3]. Frictional heat generation during sealing pair friction causes thermoelastic distortion, which leads to a deflection of the sealing ring and, in turn, affects the contact pressure distribution. Consequently, this phenomenon tends to vary heat generation and alter temperature distributions. If the sliding speed is sufficiently high, the thermal and mechanical feedback becomes unstable, and a small perturbation in the sealing system intensifies the thermoelastic stress and deformation. This phenomenon, which was first demonstrated by Barber [4], is known as thermoelastic instability (TEI). An area of localized high temperature is known as a “hot spot” or “hot banding”, and the sliding speed above which the system becomes unstable is referred to as the critical speed.

To predict the critical speed and investigate the TEI phenomenon, analytical [5,6], numerical [7,8], and experimental [9,10] approaches have been widely applied. For example, Decuzzi et al. [11] proposed a two dimensional analytical model for predicting the critical speed of hot spot in brakes and clutches, and found that the number of hot spots was independent of the radial thickness ratio. Lee et al. [12] investigated the TEI of functionally graded material with finite thickness, sliding against two half-plane frictional materials at certain speeds. Analytical methods have not been found to be particularly useful for determining stable boundaries for bodies with more general shapes. Instead, an appropriate numerical approach is preferred for simulating thermal instability behaviors. Zagrodzki [13] used the finite element spatial discretization to investigate the transient thermoelastic process of frictional systems. Yi [14] and Du [15] used the finite element methods for determining the critical speeds in brakes and clutches, by performing a preliminary Fourier decomposition of the resulting matrices to improve conditioning of the eigenvalue problem. Simultaneously, experimental research is important for observing thermal instability behaviors. Cristol-Bulthé et al. [16] conducted a laboratory test on a braking tribometer to research interactions between transient thermal phenomena and friction physical mechanisms. Kasem et al. [17] carried out braking experiments to investigate the hot spots during stop-braking from thermal and tribological perspectives.

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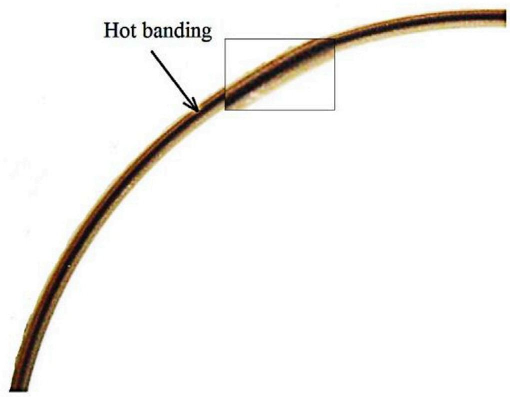

Fig. 1. Hot banding on a sealing ring.

TEI studies have been widely used to determine the characteristics of hot spots and corresponding critical speeds. In practice, hot spots more commonly appear in transient sliding contact parts, such as brakes and clutches. For local high temperatures, hot banding occurs more readily than hot spots. Fig. 1 shows an example of hot banding on a sealing ring. In simulation models, there are distinct differences between hot spots and hot bandings. Hot spots are local high temperature points in a periodic temperature field along the circumferential direction. Hot bandings are local high temperature rings at certain radii in a nonhomogeneous temperature field along the radial direction. Hot banding on a sealing surface is more harmful because it can lead to a locally high pressure and thus local wear, ultimately resulting in increased leakage. However, the issue of hot banding on a seal has received little attention.

Generally, the formation of hot banding during the sliding contact of a sealing pair causes localized high temperatures and stresses, leading to failure of the sealing material. Therefore, it is necessary to perform an analysis to predict the critical speed of hot banding emergence. In this study, we conducted a numerical simulation of coupled transient thermal-mechanical properties based on TEI theory. The finite element method was used to simulate the transient thermoelastic behavior of a sealing pair during instability. We investigated the effects of both the sealing material properties and the seal size on the critical speed. The finite element model was applied to an actual sealing pair, and the results were compared with experimental findings.

2. Basic methodology

The sealing ring examined in this study is located in the groove between the housing and the rotating shaft in a wet clutch. The sealing principle is shown in Fig. 2. The clearance between the housing and the shaft is 0.2 mm. The sealing ring expands under pressure and its own elasticity when functioning. The sealing ring employed in this study has a joint, which causes considerably more leakage than the remaining parts of the sealing ring. Most heat is generated through friction when the sealing ring and the shaft contact each other and experience relative rotation. Thus, heat flow through the oil is ignored in the model. The sealing ring maintains the oil pressure p_0 and prevents oil leakage from the clearance between the housing and the shaft. "p_a" represents the ambient pressure. The sealing principle indicates that the end surface "BC" is the primary sealing face, while the cylindrical surface "AB" is the auxiliary sealing face. Under normal operating conditions, the shaft rotates, while the sealing ring remains at rest; that is, the sealing ring does not rotate with the shaft. Thus, we ignore heat flow through the housing in the model.

For hot bandings, a periodic or non-uniform temperature distribution appears only along the radial direction, whereas the temperature distribution around the circumference remains uniform, which is clearly different from the distribution observed for hot spots. We used a cylindrical polar coordinate system (r, θ, z) for the sealing system. As the temperature distribution remains uniform along the circumferential direction θ, the coordinate system r, z is built as shown in Fig. 2.

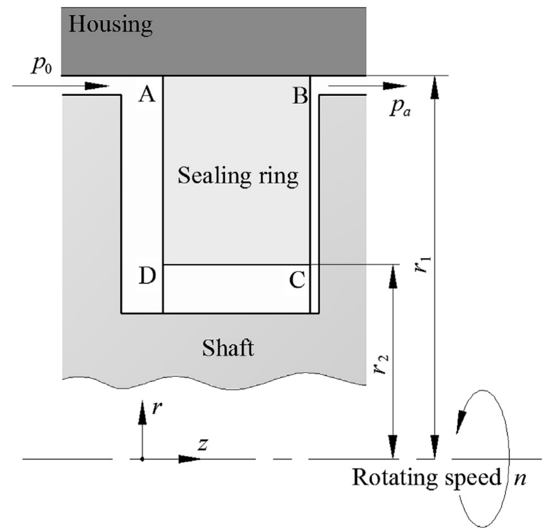

Fig. 2. The sealing principle in the transmission.

The sealing ring is mechanically fixed along r and free along z. The shaft is mechanically fixed along r and z. The heat source Q at the contact surface comes from the frictional heating. At all other nodes, there is no heat source ($Q = 0$). One assumption made by the model is that the sealing material is linear elastic.

The temperature field $T(t)$ of a sealing pair is governed by the transient heat conduction equation:

$$\frac{\partial^2 T}{\partial r^2} + \frac{1}{r}\frac{\partial T}{\partial r} + \frac{\partial^2 T}{\partial z^2} = \frac{1}{K}\frac{\partial T}{\partial t} \tag{1}$$

$$K = \frac{\lambda}{\rho C_P} \tag{2}$$

where λ is the thermal conductivity, ρ is the density, and C_P is the specific heat of the material.

When the sealing ring functions, several factors can produce a perturbation in the contact pressure and the temperature field, such as installation errors or vibrations transmitted from the road to the clutch. We consider a temperature field perturbation form:

$$T(r,z) = T_0(r,z) + \exp(bt)\Theta(r,z) \tag{3}$$

where T_0 is the temperature field in steady state, and b is the exponential growth rate.

The perturbed temperature Θ can be expressed in a discrete form:

$$\Theta(r,z) = \sum_{i=1}^{N} \Theta_i H_i(r,z) \tag{4}$$

where N is the number of elements, and H_i (r, z) is the shape function.

Substituting Eq. (3) into Eq. (1), we can obtain the following form:

$$\frac{\partial^2 \Theta}{\partial r^2} + \frac{1}{r}\frac{\partial \Theta}{\partial r} + \frac{\partial^2 \Theta}{\partial z^2} - \frac{b}{K}\Theta = 0 \tag{5}$$

To obtain a valid finite element formulation, we multiply Eq. (5) by an arbitrary trial function, i.e., a weight function H_j (r, z), and integrate over the domain Ω_n:

$$\iint_{\Omega_n} \left\{ K\left(\frac{\partial^2 \Theta}{\partial r^2} + \frac{1}{r}\frac{\partial \Theta}{\partial r} + \frac{\partial^2 \Theta}{\partial z^2}\right) - b\Theta \right\} H_j \mathrm{d}\Omega_n = 0 \tag{6}$$