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Influence of wear on thermoelastic instabilities in automotive brakes

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ABSTRACT

In typical automotive disk brakes, two pads of organic material slide against both surfaces of a gray cast iron disk. The two materials have contrary properties with regard to elasticity, temperature and wear. One aim of this is to avoid thermoelastic instabilities (TEI) that lead to material disruption and vibrations. TEI results from local thermal expansion at regions of elevated temperature, which increases local contact pressure and destabilizes the temperature field. Common TEI models scarcely account for wear. To investigate TEI, a multi-field-problem (temperature and elastic field) must be solved. Wear can be included via boundary conditions, in terms of time-dependent contact topography and load-distribution. The basic interaction mechanisms between wear, temperature and elasticity can be explained by a minimal model that allows a nonlinear investigation and a multi-DOF-model. The inclusion of wear results in a periodic movement of hot bands that influence the effective braking radius. A nonlinear analysis shows that this periodicity is possible, although a linear model computes purely real eigenvalues. A comparison with experiments indicates that the correct stability threshold and the correct eigenform are found. The presented approach shows how wear influences the dynamics of TEI, and that a periodic motion of TEI is possible if wear is taken into account. This can only be partially described by linear models because it is highly nonlinear (local loss of contact). The approach focusses on automotive brakes, but can similarly be applied to clutches or disk brakes with multiple disks, e.g. in aircraft.

1. Introduction

The contact in automotive disk brakes is typically of a brake pad with more than ten ingredients sliding against a gray cast iron disk. The mechanical energy dissipated in the boundary layer between pad and disk is transformed into heat and plastic deformation, which results in wear of the contacting bodies. The generated heat is conducted into both bodies: pad and disk. It has been observed in many experiments that the rise in temperature is not always homogenously distributed over the contact surface [4,1,9]. At high sliding velocities, the sliding bodies show local material disruptions, which arise from local temperature maxima [2]. The resulting classical patterns are known as “hot bands” and “hot spots”. Hot bands are rings of elevated temperature on the brake disk. The width of this ring is considerably less than the width of the pad. Hot spots describe the maxima of a temperature field that periodically occur around the circumference of the disk. Investigations that show thermal measurements of both phenomena are published in [19].

The occurrence of local thermal maxima can be mathematically understood as instabilities of the temperature field. A disturbance that locally increases in the temperature field leads to a local rise in material volume due to thermal expansion. At such a position, the surface area slightly bulges and carries a greater frictional load in the contact between pad and disk. This mechanism has been described first in [4] and is known as a “thermoelastic instability” (TEI). Many models are available today that describe different aspects of TEI by finite element models e.g. [23,22] or analytical approximations, e.g. [14,13,8].

One phenomenon frequently observed in experiments, but still lacking a deeper and fundamental model-based investigation, is the occurrence of migrating hot bands [15,10]. These hot bands are not fixed, but slowly propagate from the inner radius to the outer radius and back again. Furthermore, the splitting and unification of hot bands have been observed [6]. Fundamental experimental investigation of the migration of hot bands was carried out by [21,12]. Experiments have shown a clear link between the time-dependent radial position of a hot band and the time-dependent braking torque. Because the hot band carries the majority of the global frictional load, the average frictional radius is nearly identical with the position of the hot band. To avoid periodic braking torques, the motion of hot bands must be suppressed.

Only a minority of the models for hot bands are capable of describing their movements. Few publications suggest models for the motion of hot bands. Dow and Burton [7] studied a linear model of a wear-exposed blade sliding over a body. The model shows an unstable solution with a periodic component. Because infinite amplitudes and traction forces between the sliding bodies are allowed, no steady-state solution is found. Publications on TEI that allows for a local loss of contact typically involve a steady-state solution of the

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disturbed temperature fields, but those models do not include wear [24,20]. Kao and Richmond [11] investigated a two-dimensional model of a disk brake by FEM simulation in time domain. By adding wear, they found a time dependency of the position of the temperature maximum, but the computation time is too short to cover a full hot band migration cycle. For basic investigations, a minimal model that finds a link between wear and the motion of hot bands was suggested in [16].

The present work covers a systematic investigation of a minimal model for hot band motion. This minimal model is discussed linearly and nonlinearly by including a possible loss of contact between pad and disk. Additionally, the model is expanded to a multi-DOF model that is compared with experiments.

2. Minimal model

2.1. Basic equations

The model under investigation is shown in Fig. 1. Inspired by [16,18], it consists of two rods with length h and quadratic surface area A. These rods represent two neighboring regions in the sliding contact area. The two rods both have the same materials properties: Young's modulus E, thermal expansion α, thermal conductivity λ, heat capacity c and density ρ. While one end of each rod stands on a rigid, nonconductive foundation, the other end is pressed with global normal force N_{glob} against a rigid, nonconductive counterface, which slides over the surface with velocity v and coefficient of friction μ. Although many investigations, e.g. [17], show that this coefficient is highly dynamic, the present investigations apply constant values for μ as an approximation. In the sliding interface, time- and load-dependent topography heights u_1 and u_2 are introduced to consider the influence of wear. The normal contact forces are represented by N_1 and N_2, while a discrete temperature value for each rod is applied: T_1 and T_2.

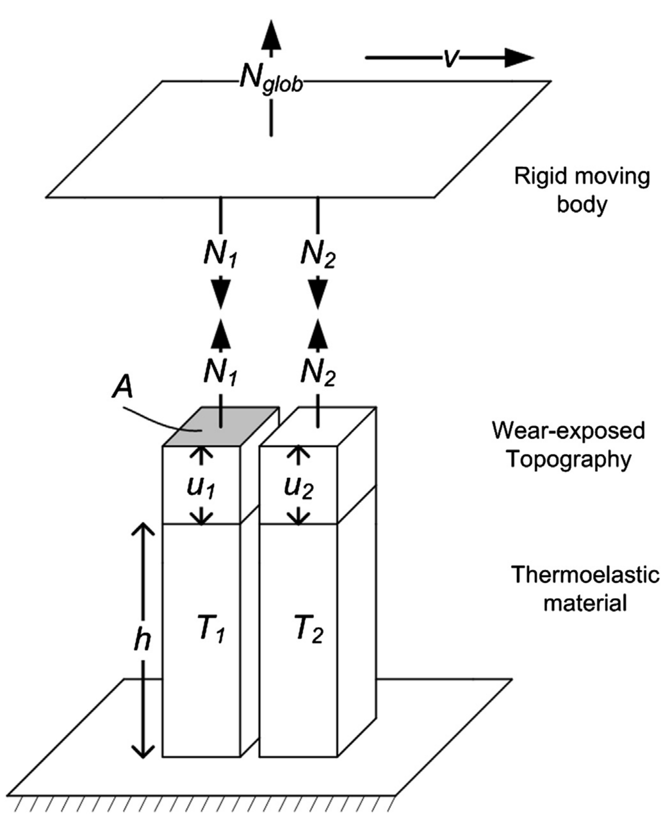

Fig. 1. Minimal model of two rods sliding against a rigid counterface.

In comparison to a similar model suggested by Barber [5], elasticity is included here.

The temperature dynamics of the rods can be expressed by

$$\dot{T}_1 = \frac{1}{hA\rho c}(-\mu v N_1 - \lambda h(T_1 - T_2)),$$
$$\dot{T}_2 = \frac{1}{hA\rho c}(-\mu v N_2 - \lambda h(T_2 - T_1)) \quad (1)$$

It is assumed that the dissipated mechanical energy is fully transformed into heat and conducted into the rods. To ensure compliance with continuous models, a compressive pressure force is associated with a negative numerical value of N_1 and N_2. Furthermore, (1) takes into account the thermal conduction between the two contacting rods. The wear-exposed height coordinate of the topography can be formulated as

$$\dot{u}_1 = vw\mu\frac{N_1}{A},$$
$$\dot{u}_2 = vw\mu\frac{N_2}{A}, \quad (2)$$

which state a linear dependency of material loss to normal pressure and can be described by a wear coefficient $w > 0$. Furthermore, a balance of the normal forces

$$N_{glob} = N_1 + N_2 < 0 \quad (3)$$

must be satisfied.

Two cases must be distinguished if a local loss of contact is allowed in the model.

1. If both rods are in contact with the counterface, both forces are nonzero.

$$N_1 < 0$$
$$N_2 < 0 \quad (4)$$

The normal forces can be determined by evaluating the kinematic constraint in the normal direction, which enforces equal length of both rods, elongated by thermal expansion and elastic properties.

$$u_1 + \alpha h T_1 + \frac{h}{EA}N_1 = u_2 + \alpha h T_2 + \frac{h}{EA}N_2 \quad (5)$$

2. If only one of the rods, index n, is in contact, the other, index m, does not carry any load. In this case the for distribution is directly known.

$$N_n = N_{glob}$$
$$N_m = 0 \quad (6)$$

here, the unloaded rod m is shorter than its neighbor n.

$$u_n + \alpha h T_n + \frac{h}{EA}N_n > u_m + \alpha h T_m \quad (7)$$

The system is fully described by the set of Eqs. (1)–(7).

3. Nondimensional formulation

The only relevant solutions are those where the disturbances have opposite signs for both rods [18]. The number of degrees of freedom of the systems can therefore be reduced by introducing a constant reference solution.

$$T_1 = T_{ref} + T$$
$$T_2 = T_{ref} - T$$
$$u_1 = u_{ref} + u$$
$$u_2 = u_{ref} - u$$