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Modelling of heat transfer between two rollers in dry friction

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ABSTRACT

An analysis of heat transfer between two rollers in dry friction is presented in this paper. The contact is peripheral and is assumed to be imperfect. The heat transfer at the interface is modelled by a thermal contact resistance. The heat flux is generated by dry friction at the interface. The two rollers are cooled by convection. A numerical model has been developed to determine the steady state temperature in rollers. Taking into account the transport phenomenon due to motion, the mesh is correlated with the velocity. The accuracy of the mesh is validated by comparison with an available analytical solution developed for a single roller in rotation. The thermal behaviour is analysed with respect to: (i) the velocity, (ii) the heat convection coefficient, and (iii) the thermal contact resistance. The evolutions of the temperature and the partition coefficient of frictional heat are presented and discussed.

1. Introduction

Heat transfer between solids in friction occurs in a large number of technological applications, such as mechanical guiding systems, ball bearings, gearing systems, braking systems, forging and hot rolling. Frictions can generate high intensity heat, which can cause severe damages to mechanical systems. Frictional heat generation is difficult to model since it involves a number of phenomena, which are not well known. Constraints due to mechanical and thermal loadings lead to damages to the interface as well as to the solid matter and, therefore, to a decrease of the mechanical resistance of the solids.

It is difficult to determine experimentally the heat flux distribution in the solids under friction since taking measurements in the vicinity of the interface is very complicated. Theoretical approaches allow an estimation of the heat flux and the conduct of a sensitivity analysis of this parameter to other physical quantities of the problem under study. Initial analytical approaches on the evaluation of the partition coefficient between two semi-infinite materials in perfect contact have been proposed by Jaeger [1] and Blok [2]. The authors have based their approach on the theory of a single mobile source scanning the surface of a semi-infinite medium. This theory provided a simple solution for very small and very large relative velocities, but had difficulties dealing with moderate velocities. Recently, Laraqi [3] developed an exact analytical solution for this problem valid for any relative velocity. Other analytical models have been developed for single or multiple moving heat sources on an homogenous medium [4–6] or multilayer [7].

In practice, solids in friction exchange heat between them at the interface. The determination of the temperatures and the flux partition requires, therefore, the taking into account of the thermal interfacial coupling. A number of authors carried out their studies while considering the hypothesis of a perfect contact between the solids [8,9]. Experimental studies [10,11] have shown that there is exist an important sudden change in temperature in the vicinity of the interface, which demonstrate the imperfect character of the contact. In order to take into account such phenomenon, imperfect sliding contact models have been proposed [11,12]. Such models introduce two contact thermal parameters: (i) the intrinsic partition coefficient and (ii) the thermal contact resistance.

These parameters depend on micro-geometry of the interface, the mechanical and thermal properties of the materials, the relative velocity, the load and other physical factors. An experimental method based on the inverse problem techniques has been developed in order to identify the thermal parameters of the sliding contact [14].

Heat transfer at the interface of two solids is closely linked to the phenomena of thermal constriction, which take place near the contact. Such phenomena have been widely studied for static contacts [15–18]. Only a few studies have been conducted for mobile contacts [19,20].

The work presented in this paper aims at determining the temperature and the partition coefficient in a frictional system made of two rollers. The imperfect contact characteristic is taken into

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Nomenclature

a	radius	m
K	thermal conductivity	$\mathrm{W\,m^{-1}\,K^{-1}}$
p	heat partition coefficient	q_1/q_g
q	heat flux density	$\mathrm{W\,m^{-2}}$
R_c	thermal contact resistance	$\mathrm{m^2\,K\,W^{-1}}$
u	temperature rise	K
V	linear velocity	$\mathrm{m\,s^{-1}}$

Greek symbols

α	heat convection coefficient	$\mathrm{W\,m^{-2}\,K^{-1}}$
β	half-angle of contact	
κ	thermal diffusivity	$\mathrm{m^2\,s^{-1}}$
ρ	radial coordinate	m
φ	angular coordinate	rad
ω	rotational velocity	$\mathrm{rad\,s^{-1}}$

Superscripts

a	ambient
g	generated by friction
i	index of body ($i = 1, 2$)

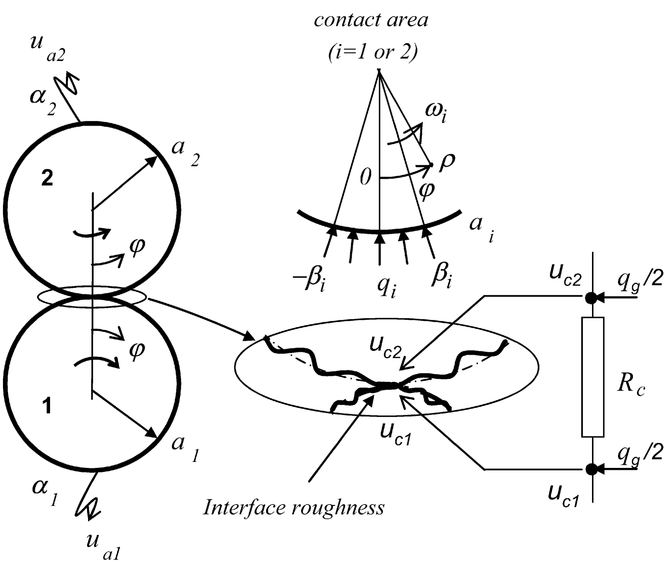

Fig. 1. Simplified scheme of rollers and their interface.

account by adopting the sliding contact models [12,13]. A refined mesh numerical model has been developed in order to derive and highlight the thermal gradients in the solids. The validation of the mesh has been conducted by comparing of the numerical results with an available analytical solution for a single roller [21]. The coupled transfers model presented in this paper has been developed in order to analyse the influence of the velocity, the thermal contact resistance and the convective exchanges coefficients on the temperature and the flux partition.

2. Description of the device

The system under study is constituted from 2 cylindrical rollers having a radius a_i ($i = 1, 2$) and in peripheral contact as shown in Fig. 1. The two rollers are in motion with respect to the contact at an angular velocity ω_i and are cooled by convection (heat convection coefficient α_i and ambient temperature u_{ai}). A heat flux q_g is generated by friction at the interface. In order to take into account the irregularities of the interface the contact between the two solids is considered to be imperfect (Fig. 1). It is represented by a thermal contact resistance, R_c, receiving at each of its ends half the flux generated by friction (the intrinsic partition coefficient is taken equal to 0.5). The surface area of the contact can be evaluated with the Hertz theory. Note that β_i is the semi-angle of the contact opening.

3. The model

The heat transfer in the rollers is assumed to be 2D. The temperature is denoted as $u(\rho, \varphi)$. The physical parameters of the problem are summarised in Fig. 1. According to the physical conditions described previously, the governing equations can be written as

$$\frac{\partial^2 u_i}{\partial \rho^2} + \frac{1}{\rho}\frac{\partial u_i}{\partial \rho} + \frac{1}{\rho^2}\frac{\partial^2 u_i}{\partial \varphi^2} - \frac{\omega_i}{\kappa_i}\frac{\partial u_i}{\partial \varphi} = 0 \quad (i = 1, 2) \tag{1}$$

The last term of the left-hand side of Eq. (1) represents the relative motion of the rollers with respect to the contact area. It is the convection term whose importance depends on the values of the angular velocity ω and the thermal diffusivity κ.

The boundary conditions with respect to the radial direction ρ are given by:

$$K_1\frac{\partial u_1}{\partial \rho} = 0.5q_g + \frac{u_2(a_2,\varphi) - u_1(a_1,\varphi)}{R_c}$$
$$(\rho = a_1,\ -\beta_1 \leqslant \varphi \leqslant \beta_1)$$
$$K_2\frac{\partial u_2}{\partial \rho} = 0.5q_g + \frac{u_1(a_1,\varphi) - u_2(a_2,\varphi)}{R_c}$$
$$(\rho = a_2,\ -\beta_2 \leqslant \varphi \leqslant \beta_2)$$
$$K_i\frac{\partial u_i}{\partial \rho} = -\alpha_i\big(u(\rho,\varphi) - u_{ai}\big)$$
$$(\rho = a_i,\ -\pi \leqslant \varphi \leqslant -\beta_i \text{ and } \beta_i \leqslant \varphi \leqslant \pi) \tag{2}$$
$$\frac{\partial u_i}{\partial \rho} = 0 \quad (\rho = 0,\ -\pi \leqslant \varphi \leqslant \pi) \tag{3}$$

The first and the second parts of Eq. (2) are the heat flux entering through the interface ($-\beta_i \leqslant \varphi \leqslant \beta_i$). The third part of Eq. (2) is the external convection cooling ($-\pi \leqslant \varphi \leqslant -\beta_i, \beta_i \leqslant \varphi \leqslant \pi$). The heat convection coefficient is α_i and the ambient temperature u_{ai}.

The heat transfer is periodic along the angular direction. The periodicity conditions can be written as

$$u_i(\rho, -\pi) = u_i(\rho, \pi) \tag{4}$$
$$\frac{\partial u_i(\rho,\varphi)}{\partial \varphi} = \frac{\partial u_i(\rho,\pi)}{\partial \varphi} \tag{5}$$

4. Results and discussion

The governing equations are solved numerically by using the finite volume method. The iterative method (Successive Overall Relaxation, SOR) has been adopted to solve the discrete equations. Taking into account the rotation of the roller, the relaxation coefficient for the SOR method is less than 1 (an optimal value is 0.95). The optimal mesh has been fixed to 60 cells for the radial direction and 120 for the azimuth direction. The iterative process has been initialised from 1D analytical model by writing the thermal

Table 1
Numerical data for the comparison with DesRuisseaux et al. [21] results.

K ($\mathrm{W\,m^{-1}\,K^{-1}}$)	κ ($\mathrm{m^2\,s^{-1}}$)	a (m)	α ($\mathrm{W\,m^{-2}\,K^{-1}}$)	q ($\mathrm{W\,m^{-2}}$)	β (rad)
10	2.67×10^{-6}	10^{-2}	100	10^5	0.16

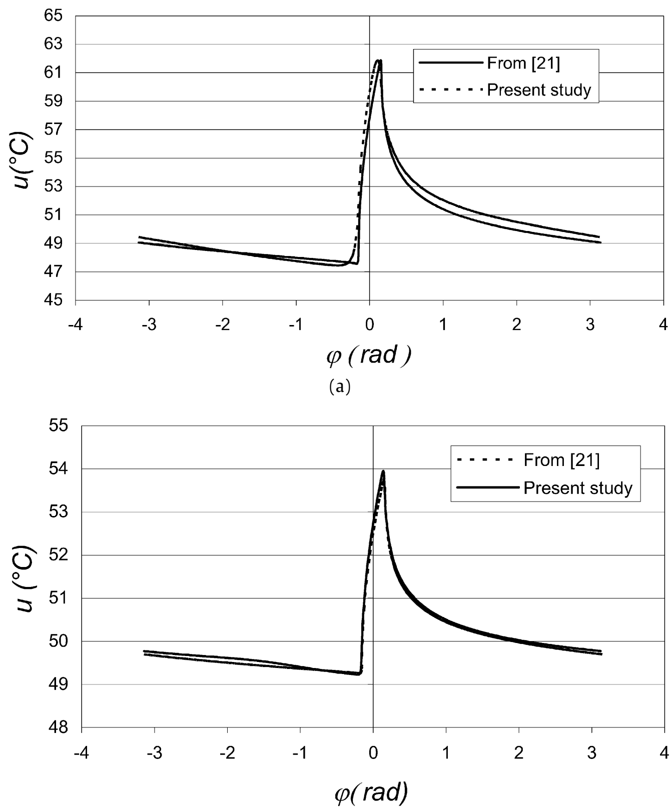

Fig. 2. Comparison of surface temperatures: (a) $\omega = 0.5\ \mathrm{rad\,s^{-1}}$, (b) $\omega = 5\ \mathrm{rad\,s^{-1}}$.

balance. It is important to choose carefully the mesh in the radial direction because the thermal gradients with respect to the radial direction decrease as the velocity increases. It is imperative to correlate the radial size of the mesh to the velocity, particularly in the vicinity of the heated region. The size of the radial grid is inversely proportional to $\sqrt{\omega}$. To ensure a good accuracy of the results with relatively short computing times, we adopted a mesh refined in the vicinity of the contact and large beyond this zone.

The numerical model is validated by comparing our results with those calculated by an analytical solution developed by DesRuisseaux et al. [21]. The authors considered a single roller, rotating at a constant speed and receiving heat flux on a portion of its outer periphery, the remaining surface being cooled by convection. We have carried out comparisons for different speeds and heat convection coefficients. Numerical data used for this comparison are compiled in Table 1. Fig. 2 shows the comparison of surface temperatures for velocities $\omega = 0.5\ \mathrm{rad\,s^{-1}}$ (Fig. 2a) and $\omega = 5\ \mathrm{rad\,s^{-1}}$ (Fig. 2b). The representative profiles are in good agreement. The temperature rise in passing the contact is around 14.4 °C for the case $\omega = 0.5\ \mathrm{rad\,s^{-1}}$ and about 4.7 °C for $\omega = 5\ \mathrm{rad\,s^{-1}}$. The ratio of these temperature increments is about $14.4/4.7 = 3.06$, which essentially corresponds to the square root of the inverse ratio of speeds: $\sqrt{5/0.5} = 3.16$. Above a certain speed (or Peclet number) the temperature rise in the contact is inversely proportional to the velocity.

The heat flux entering each of the rollers through the contact is in reality not known. It depends on the thermal properties of the materials and on the operating conditions. Various operating conditions based on different velocities and heat convection coefficients were considered. In order to reduce the number of varying parameters for the analysis, the rollers were taken as identical, both in dimensions and materials. The effect of the thermal contact resistance on the thermal behaviour of the rollers is analysed by varying its value.

Table 2
Numerical data for materials.

	K ($\mathrm{W\,m^{-1}\,K^{-1}}$)	κ ($\mathrm{m^2\,s^{-1}}$)	a (m)	β (rad)
Rollers 1 and 2	10	2.67×10^{-6}	2×10^{-2}	2.5×10^{-3}

Table 3
Numerical data for operating conditions.

V ($\mathrm{m\,s^{-1}}$)	α_1 ($\mathrm{W\,m^{-2}\,K^{-1}}$)	α_2 ($\mathrm{W\,m^{-2}\,K^{-1}}$)	q_g $\mathrm{W\,m^{-2}}$	R_c ($\mathrm{m^2\,K\,W^{-1}}$)
0.2, 2 and 10	20, 100 and 500	20	5×10^6	10^{-3}, 10^{-4} and 10^{-5}

Table 4
Heat partition coefficient $p_1 = q_1/q_g$.

α ($\mathrm{W\,m^{-2}\,K^{-1}}$)		V ($\mathrm{m\,s^{-1}}$)								
		0.2			2			10		
		R_c ($\mathrm{m^2\,K\,W^{-1}}$)			R_c ($\mathrm{m^2\,K\,W^{-1}}$)			R_c ($\mathrm{m^2\,K\,W^{-1}}$)		
α_1	α_2	10^{-3}	10^{-4}	10^{-5}	10^{-3}	10^{-4}	10^{-5}	10^{-3}	10^{-4}	10^{-5}
100	20	0.515	0.603	0.686	0.515	0.606	0.740	0.515	0.607	0.759
500	20	0.518	0.629	0.751	0.518	0.633	0.824	0.518	0.634	0.849

The numerical data used for the two rollers are shown in Table 2. The parametric analysis has been conducted with the parameters' values as set out in Table 3.

Table 4 shows the heat partition coefficient $p_1 = q_1/q_g$ calculated for 27 operating points. When the two rollers' partition coefficients are identical the logical value of 0.5 is found. Increasing the heat convection coefficient of one of the rollers (here roller 1) leads to an increase of the partition coefficient. For large values of the thermal contact resistance, for example $R_c = 10^{-3}\ \mathrm{m^2\,K\,W^{-1}}$, the increase in the partition coefficient is negligible. This result implies that if the surfaces are of a bad quality (poor contact), the cooling of one of the two solids does not lead to a change in the heat partition between the two solids.

For small values of the thermal contact resistances ($R_c \leqslant 10^{-5}\ \mathrm{m^2\,K\,W^{-1}}$) the sensitivity of the partition coefficient to velocity and heat convection coefficient variations becomes much more important:

(i) For $R_c = 10^{-5}\ \mathrm{m^2\,K\,W^{-1}}$, when α_1 increases from 20 to 500, the partition coefficient increases by about 50% in comparison with the symmetrical case ($\alpha_1 = \alpha_2 = 20$). This effect is further reinforced with an increase in rollers' velocity.

(ii) For the largest value of α_1 (here $\alpha_1 = 500$) the increase in the partition coefficient is 50% for $V = 0.2$ and reaches 70% for $V = 10$.

5. Conclusion

A numerical model was developed for the analysis of heat transfer between two rollers, which takes into account the thermal coupling via a thermal contact resistance. This approach is much more realistic than that adopted by many authors, which consider the contact as perfect. The right selection of the mesh is very important for this type of problems. The mesh was validated by comparing the results from this work with those of an analytical model of a single roller available in the literature.

The numerical results obtained from the developed model highlighted the inter-related roles of three parameters considered in the present work: the velocity, the heat convection coefficient and the thermal contact resistance. The numerical results show particularly that the thermal contact resistance controls the influence of the two other parameters on the partition coefficient. Indeed, it is only when the thermal contact resistance becomes small enough, typically $R_c \leqslant 10^{-5}\ \mathrm{m^2\,K\,W^{-1}}$, that both the velocity and the heat convection coefficient begin to have an noticeable effect on the partition coefficient.

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