THERMOELASTIC CONTACT ANALYSIS OF VEHICLE BRAKE SYSTEM

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ABSTRACT

This paper focuses the studies on the thermoelastic contact analysis of vehicle brake system based on contact, heat and thermoelastic parameters. The conditions of the contact were given for the movable interface. The kinetic behaviors of the thermal and contact parameters have been taken into account while calculating temperature fields and stresses in the friction zone. To calculate temperature fields, heat models of the friction contact were elaborated to make allowance for redistribution of heat flows at friction. Based on numerical methods, surface and mean bulk temperatures in the friction pair were predicted. It has been established that due to heat generation during friction the actual contact area in disc brakes contracts and becomes about 30% of the nominal one. This brings about inhomogeneity of temperature fields and considerable rise of surface temperatures and thermal stress in the rubbing bodies. The proposed calculation method can be used to forecast service characteristics of brakes and to optimize brake design for given materials of the friction pair.

NOMENCLATURE

n – coordinate along outer normal to the surface, m
q, q_n, q_d – heat flow intensity, Wt/m^2
p - pressure on the contact area, Pa
u, v – displacement vector components, m; T – temperature, ^0C
T_t, T_l – surface temperature of counter-body and lining, correspondingly, °C
k – heat conduction of the contact, Wt/(s-degree-m)
t – time, s
t_T – braking time
c – coefficient of specific heat, J/(kg-K)
λ - heat conduction, Wt/(m-K)
ρ - density, kg/m3;
x, y, z –coordinates, m
r – radius vector, m
R_{in}, R_{out} – internal and outer radii of the friction lining, m
ε_{rr}, ε, ε_{zz}, ε_{rz} – strain tensor components;
σ_{rr}, σ, σ_{zz}, σ_{rz}, σ_r, σ_z - stress tensor components, MPa
G – shear modulus, MPa

INTRODUCTION

One of the chief factors conditioning stress state of the frictional material in a brake lining is the friction temperature gradient. The decisive effect on the frictional and wear characteristics of the brake system is exerted by temperature generated at friction. So far to calculate stress at operation it is necessary to determine temperature fields in the rubbing bodies [1, 2].

In the present work, a system of interrelated problems (i.e. contact, heat and thermoelastic parameters) for vehicle brake system has been considered. The conditions of the contact were given for the movable interface. The kinetic behaviors of the thermal and contact parameters were taken into account at calculating temperature fields and stresses in the friction zone. To calculate temperature fields, heat models of the friction contact were elaborated to make allowance for redistribution of heat flows at friction. Based on numerical methods surface and mean bulk temperatures in the friction pair were calculated. The proposed calculation method can be used to forecast service characteristics of brakes and to optimize brake design for given materials of the friction pair.

MODELING AND ANALYSIS

Calculation of thermal stresses is based on solution of the dynamic problem of thermoelasticity. When solving the problem on contact pressure distribution $\rho(r, t)$ as well as temperature field $T(r, t)$ and thermoelastic displacements $u(r, t)$ and $w(r, t)$ in the rubbing bodies the following assumptions were taken:

- Interaction proceeds over the nominal contact area that varies with time
- Heat is generated on the contact surface by a plane source.

Heat problem at friction can be reduced to calculation of temperature field at boundary conditions of the second order

$$\lambda = \frac{\partial T}{\partial n} + q_n = 0 \qquad \ldots(1)$$

The distribution of the heat flow intensity q in radial direction is complex and is dependent on time of braking:

$$q(r,t) = f_{fr} V(r,t) p(r,t), \qquad R_{in} \le r \le R_{\max}, \qquad 0 \le t \le t_p \qquad \ldots(2)$$

while the sliding velocity changes linearly

$$V(t) = V_0 \left(1 - \frac{t}{t_p} \right) \qquad \ldots(3)$$

Non-stationary temperature field $T(x, y, z)$ brings about stress state changing with time. The problem is solved in the axisymmetric statement since the model describing a real object has the rotation symmetry relative to Z-axis (Fig.1). That’s why the components of displacement in circumferential direction and stresses $\sigma_{r\varphi}$. and $\sigma_{z\varphi}$ wouldn’t depend on φ. The main equations take the form:

$$\frac{\partial \sigma_{rr}}{\sigma r} + \frac{\partial \sigma_{rz}}{\sigma r} + \frac{1}{r}\left(\sigma_{rr} - \sigma_{\varphi\varphi}\right) = \rho \frac{\partial^2 u}{\partial t^2}, \qquad \frac{\partial \sigma_{rz}}{\sigma r} + \frac{\partial \sigma_{zz}}{\sigma r} + \frac{\sigma_{rz}}{r} = \rho \frac{\partial^2 v}{\partial t^2} \qquad (4)$$

$$\varepsilon_{rr} = \frac{\partial u}{\partial r}, \varepsilon_{\varphi\varphi} = \frac{u}{r}, \varepsilon_{zz} = \frac{\partial v}{\partial z}, \varepsilon_{rz} = \frac{1}{2}\left(\frac{\partial u}{\partial z} + \frac{\partial v}{\partial r} \right), e = \frac{\partial u}{\partial r} + \frac{u}{r} + \frac{\partial v}{\partial z} \qquad (5)$$

$$\begin{aligned} &\Delta u - \frac{u}{r^2} + \frac{1}{1-2\mu} \mathrm{X} \frac{\partial e}{\partial r} - \frac{\rho}{G} \mathrm{X} \frac{\partial^2 u}{\partial t^2} = \frac{2(1+\mu)}{1-2\mu} \mathrm{X} \frac{\partial (\alpha T)}{\partial r}, \\ &\Delta v + \frac{1}{1-2\mu} \mathrm{X} \frac{\partial e}{\partial z} - \frac{\rho}{G} \mathrm{X} \frac{\partial^2 v}{\partial t^2} = \frac{2(1+\mu)}{1-2\mu} \mathrm{X} \frac{\partial (\alpha T)}{\partial r} \end{aligned} \qquad (6)$$

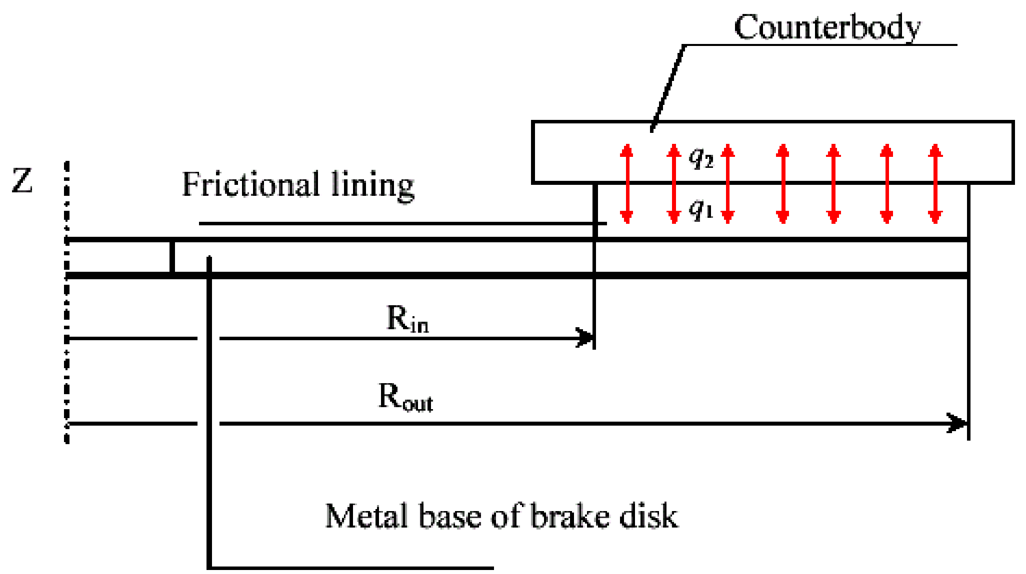


Fig.1 Schematic representation frictional contact model

Laplacian operator in this case will be

$$\Delta = \frac{\partial^2}{\partial r^2} + \frac{1}{r}\frac{\partial}{\partial r} + \frac{\partial^2}{\partial z^2} = \frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial}{\partial r}\right) + \frac{\partial^2}{\partial z^2} \qquad \ldots(7)$$

To solve the problems of thermal contacts the following algorithm has been used. Let thermoelastic displacements $u(r_i, t)$, $v(r, t)$ and temperature $T(r_i, t)$ in the sites of the finite element net be known. Then, contact pressure at the passage to further time interval $[t_{n+1}, t_{n+2}]$ will be determined using the found new values of thermoelastic strains depending on temperature regimes of the following time t_{n+2} and its value changes. Variation of $\rho(r, t)$ will, in its turn, affect the function of heat generation intensity that will lead to redistribution of temperature field and then of thermoelastic displacements.

The general block diagram for solving the thermoelastic contact problem is shown in Fig. 2. In the given statement, main attention is paid to the presence of the moving boundary and interrelation between the kinetics of temperature regime, thermoelastic deformations and contact parameters that determine heat generation. The employed model of frictional interaction of two bodies makes it possible to consider thermal interaction between the composite lining and counter-body at their contact and to estimate heat emission q_d of the counter-body into the lining during frictional interaction.

As far as metal counter-body heats faster due to its higher thermophysical characteristics than the frictional lining, so an additional heat flow q_d from the counter-body into the lining emerges

$$Q_d = \begin{cases} k(T_t - T_1) - \text{at contact interface between bodies} \\ 0 - \text{at contact absence} \end{cases}$$

At calculations, k is taken as 30 Wt/s-K-m.

Temperature fields and stresses were calculated based on the following initial data:

- Material of the counter-body is cast iron for which c = 540 J/(kg-K); λ = 30 Wt/(m-K) and ρ = 7300kg/m3
- for the frictional material ρ = 2600 kg/m3.

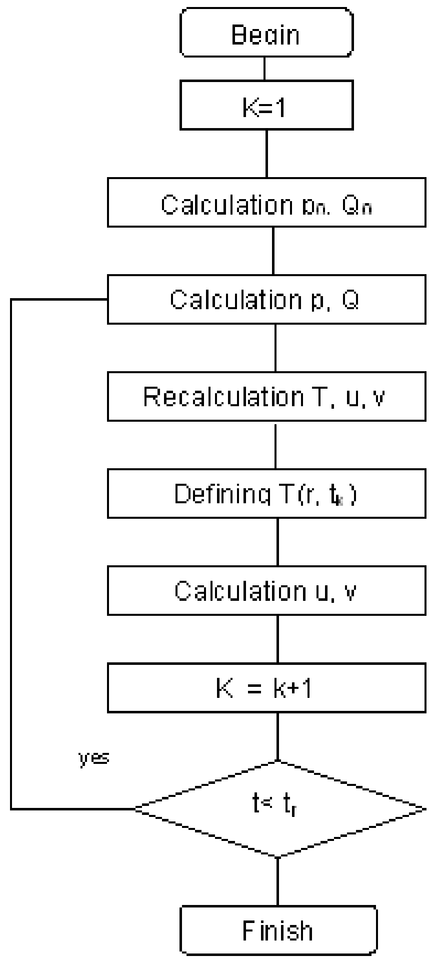


Fig.2 Flowchart for solving thermoelastic contact problem

Nonlinearity of thermophysical characteristics of the frictional material was used to calculate temperature fields and heat flows were given by temperature dependencies.

RESULTS AND DISCUSSION

Based on the developed algorithm for the thermal contact problem, the stress state of the frictional lining material of a disc brake has been calculated at a single braking ($0 \leq t \leq t_T$, t_T =3.5s). In Fig.3 contact pressure distribution is shown across the friction lining width. Fig.4 presents temperatures; Fig.5 reflects the intensity of heat flows. Width of the annular lining (L) is about 0.035 m.

As it is seen from Fig.3, at incipient braking t = 0, that is, in the absence of temperature effect, distribution of contact pressures across lining width varies but negligibly (0.77 to 1 MPa). Heat flows and temperature are distributed evenly 0.3s after the moment of braking contact pressure changes and increases heat flow power in response to reducing contact area and surface temperature reaches 100 °C (Fig.4, 5). Upon t = 0.5s temperature induced strain in the counter-body leads to considerable reduction of the contact area of the rubbing bodies. The maximum value of the heat flow is reached at t =1s which is then lowers. Owing to localization of the heat source and due to temperature strains, temperature on the lining surface continues to rise and reaches by 450 °C t = 2s. At t = 1s the contact area is finally formed which constitutes about 30% of the contour one and then varies but slightly. The reduction of the contact area leads to growing contact pressure. The analysis of contact pressure and temperature variation across the friction lining width has proved that their variation dynamics is dependent on the counter-body toughness. Temperature-induced deformation of the counter-body affects heat regime of the brake as a whole.

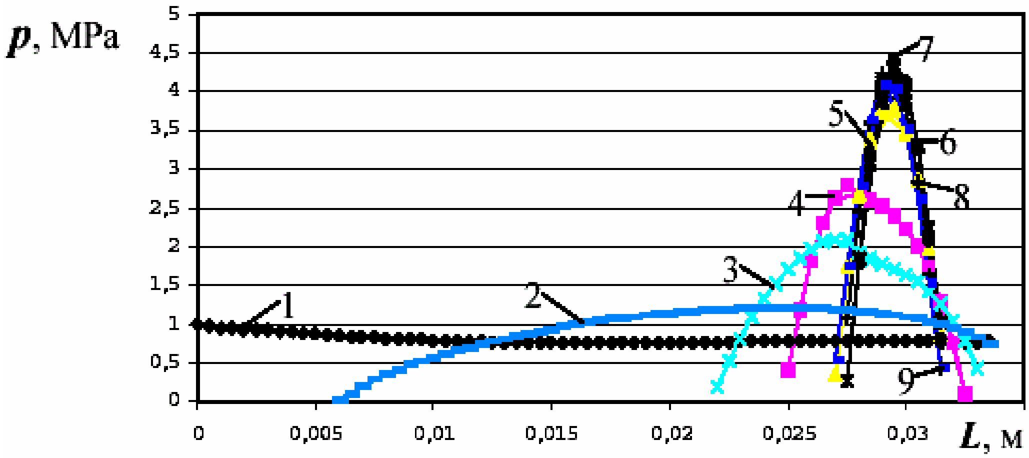


Fig. 3 Distribution of contact pressure across lining width at various braking times.

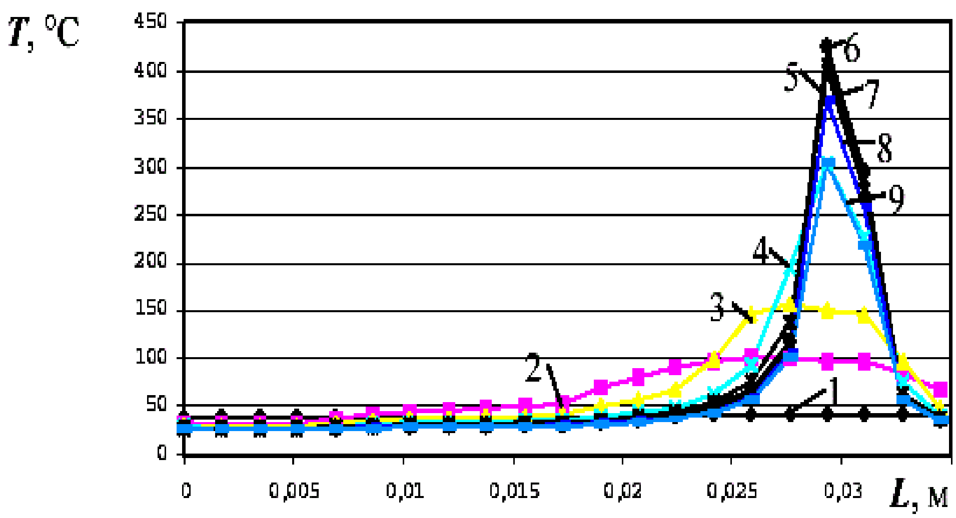
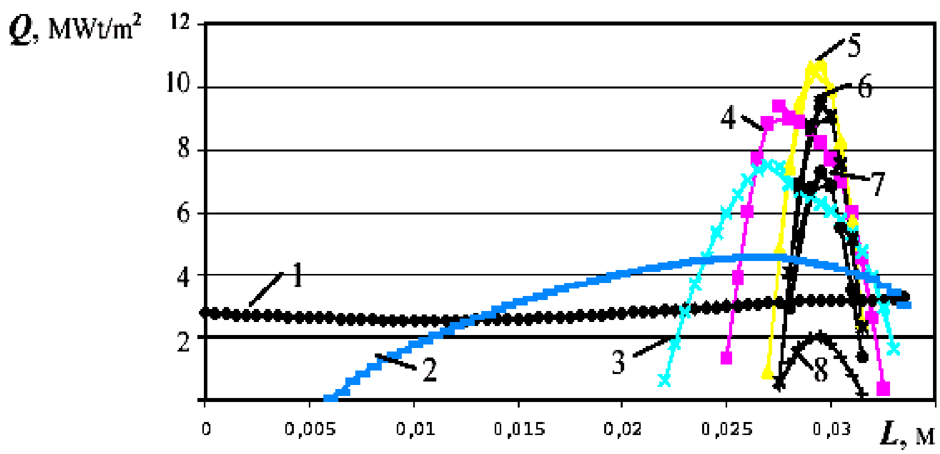


Fig. 4 Distribution of temperature across lining width at various braking time.



In Fig.3-5 braking time is respectively
1 – t =0.02s; 2 – t = 0.1s; 3 – t = 0.3s;
4 – t = 0.5s; 5 – t = 1.0s; 6 – t = 1.5s; 7 – t = 2.0s; 8 – t =3.0s; 9 – t = 4.0s.

Design properties of the braking system as well as thermoelastic strain of the friction pair members result in localization of temperatures and contact pressures within the vicinity of the outer edge of the lining. This intensifies wear of the material in this region. As a result, contact surfaces of the friction pair become overheated while thermal stresses reach 96 MPa at t = 2s and 53 MPa at t = 3.5s. Axial stresses σ_z on the lining surface are the compressive stresses (14 MPa) and radial ones are the tensile stresses (39 MPa). A considerable stress gradient might be the reason of crack formation in the lining material.

Therefore, discrete character of the contact and small actual contact area of the rubbing bodies contribute into uneven distribution of the temperature field and temperature stresses, and spur nucleation of cracks, warping and shrinkage of brake members.

CONCLUSIONS

It has been found out that as a result of intense heat generation during friction the actual contact area of the friction contact in disc brakes contracts and equals about 30% of the nominal one. This becomes the reason of increased inhomogeneity of temperature fields and perceptible rise of surface temperatures and temperature stresses in the rubbing bodies. Increased toughness of the counter-body was found to reduce inhomogeneity of contact pressures over the frictional lining width, stabilize actual contact area at high enough friction temperature and finally alleviate heat loading on the brake unit.

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