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Temperature in the railway disc brake at a repetitive short-term mode of braking

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ABSTRACT

The finite element (FE) model to determine the transient temperature field in the ventilated disc brake of the traction diesel multiple unit (DMU) has been proposed. The advantage of the developed numerical model is the representation of mutual motion of the stationary pad and the rotating disc, by a heat source of arbitrary shape moving over the stationary disc. Computations were carried out for the pad and the disc separately introducing the heat partition ratio. Both the single and the multiple modes of braking were examined. The calculated distributions in contact temperature were compared with the corresponding results obtained from analytical solutions of the boundary-value thermal problem of friction, and with experimental data determined by the method of thermocouples. It was demonstrated that the calculated mean temperature on the friction surfaces of the brake components and the bulk temperature of the disc during multiple brake application agree well with the corresponding results, obtained by methods mentioned above.

1. Introduction

Deceleration of a vehicle by means of a friction brake proceeds with transformation of kinetic energy into heat on the friction surfaces. As a result temperature of contacting bodies increases significantly even in normal operating conditions. In view of the growing demands on the braking performance, the increasing velocity of railway passenger vehicles, and consequently an amount of energy dissipated during braking, in modern vehicles the traditional tread brake is replaced with disc brake system [1–3]. Thermal loads, which friction pair will be subject to in operation, are a key issue in the design process of braking systems as well as for the selection of materials. In the course of the design of railway vehicles, proper selection of friction materials is verified by testing on a full-scale dynamometer. Test rigs of this kind simulate actual operating conditions of a brake to analyse certain frictional characteristics.

Due to the high cost of conducting experimental studies to determine the temperature, it is advisable to use simulation methods first and only after positive verification of initial assumptions proceed to the prototype production phase. During brake application, particularly continuous, emergency or multiple brake application with short cooling periods, brake disc is subject to considerable heat load. There are two types of effects of thermal load on brakes: i) bulk thermal effects such as deformation of a brake disc (coning) and ii) local effects, e.g. thermal cracks [4].

Mechanical and thermal stresses as well as local overheating in a brake disc resulting from frictional heating may lead to changes in material structure, cracks and other damage shortening its lifespan in service. To counteract such negative effects correct temperature estimation is therefore essential.

Calculations of temperature and its gradients during braking, using the finite element method (FEM) were carried out by many researchers [5]. In the paper [6] a three-dimensional model of a locomotive wheel-mounted brake disc, was used to analyse changes in temperature on the friction surfaces of a disc and surfaces of flow channels during a single emergency brake application. The model included cooling of the disc by air with the efficiency varying through the braking process. The calculated maximum temperature of the disc surface was approximately 15% lower than the temperature measured during experimental investigations. Numerical calculations of the temperature distribution and thermal stresses in the brake disc during continuous brake application and braking to standstill were presented in the article [7].

Comparison of the outcomes of calculations of temperature on the surface of the railway disc made under the assumption of non-uniform and uniform pressure of pads on the disc was carried out using the two-dimensional [8] and the three-dimensional [9] FE model. In the paper [8] it was established that at the considered angular velocity of the brake disc, temperature gradient along circumference of the disc can be

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Nomenclature

A	contact surface area, (m^2)
A^{vent}	surface area of cooling, (m^2)
b	thickness of heat-absorbing layer;
c	specific heat capacity, (J/(kg °C))
f	coefficient of friction;
Fo	Fourier number;
G	mass, (kg);
$G_{p,d}$	masses of spatial basic regions, (kg);
$G_{p,d}^{(i)}$	masses of additional regions absorbing heat, (kg);
h	heat transfer coefficient, (W/(m^2 °C))
K	thermal conductivity, (W/(m °C))
k	thermal conductivity, (W/(m °C))
n	number of braking
p	current contact pressure, (MPa)
p_0	nominal value of the contact pressure, (MPa)
q	current specific power of friction, (W/m^2)
q_0	initial specific power of friction, (W/m^2)
r_{eff}	effective radius of friction for the contact area, (m)
r_m	effective radius at very small values of θ_0, (m)
R, r	outer and inner radius, (m)
T	temperature, (°C)
T_a	ambient/initial temperature, (°C)
T_{V_n}	bulk temperature, (°C)
$\overline{T}$	mean temperature, (°C)
t	time, (s)
t_s	braking time, (s)
t_c	cooling time after stop, (s)
V	current translational speed at effective radius, (m/s)
V_0	initial translational speed at effective radius, (m/s)
W_0	work done during one braking, (J)
W^{vent}	heat dissipated, (J)
w	specific work done during single braking, (J)

Greek symbols

γ	heat partition ratio, dimensionless
Γ	contact region
δ	thickness, (m)
δ^{eff}	effective depth of penetration of heat, (m)
ε	coefficient, dimensionless
θ_0	semi-angle of the contact area, (deg)
ρ	density, (kg/m^3)
ψ	coefficient, dimensionless
ω	angular velocity, (rad/s)

Subscripts

d	indicates disc
p	indicates pad

neglected, therefore two-dimensional model was applied [8]. The calculated maximum temperature on the friction surfaces of the disc was significantly higher at nonuniform distribution of the pressure on the contact surface of the pad and the disc. The result was validated by the measurements of temperature on the contact surfaces of the disc, performed during the full-scale dynamometer test and is consistent with the phenomena described in the articles [10,11].

Use of FEM to calculate temperature gradient in the process of optimization of the brake pad for obtaining the most uniform distribution of pressure on the brake disc was presented in the article [12]. In the quoted paper it was assumed that optimum pad is characterised by the minimum temperature gradient. The computations were carried out studying influence of grooves in the pad, mechanical properties of the friction material, Young's modulus of the intermediate layer and thickness of the reinforcement. Experimental research on the dynamometer confirmed that increasing in number of grooves in the pad reduces the temperature gradient in the disc. By contrast, in [1] railway brake disc was optimized using FEM. For this purpose, a two-dimensional model of the segment of the disc was examined to determine the temperature changes in the course of a single continuous brake application and braking to a stop.

The impact of design modifications of the brake disc, i.e. number of cooling fins, thickness of the friction ring and width of the cooling fins on temperature was verified. It was assumed that total heat is absorbed by the disc, distribution of the heat flux on the friction surface is uniform and temperature gradient in the disc is negligibly small both in radial and circumferential direction.

Numerical calculations (using FEM) of temperature distribution and thermal stresses performed in the design process of the wheel-mounted disc for the power cars of the high-speed trains TGV were presented in the paper [2]. Selection of the optimal design of the brake disc was based on the results of these calculations. The resulting disc design was tested on dynamometer, underwent the approval procedure and was successfully brought to operation on TGV trains.

The diversity of aims of the quoted papers, i.e. optimization of the geometry of the brake disc, optimization of the shape of the brake pad, determination of mechanical and thermal stresses, an explanation of the mechanism of heat localisation or cracking of the brake disc shows great potential of FEM in the design and evaluation of braking systems for railway vehicles. An overview of works on the modelling of temperature distribution in the friction systems using numerical methods is included in the article [5].

Experimental studies are often used to verify numerical models. In the case of railway braking systems full-scale dynamometer tests are a very good data source [2,6–8,12]. Temperature of the friction couple during the dynamometer test is measured using: i) thermocouple (e.g. K type) located under the friction surface of the disc [6–8] or brake pads [12] ii) sliding thermocouples [8], or iii) IR pyrometer [8]. The disadvantage of these methods of temperature measurement is that the data are restricted only to the selected points in friction couple. To study the phenomena associated with the localisation or uneven distribution of the heat flux on the surface of the disc it is necessary to use IR thermography [8,13]. One of the techniques is to capture images with the high speed camera only on a narrow area of the disc with a specified angular position. Series of images allows to recreate the entire image of the disc for the given revolution [8]. A difficulty in using the thermal imaging is to ensure a uniform emissivity of an object of interest [8,13].

During braking, in particular high power brake application, transfer of friction material to the brake disc surface is possible, which can significantly disturb the temperature measurement by thermographic camera.

In the design of the friction couple analytical models are used alongside experimental and numerical modelling to estimate the temperature distribution. A review of thermal problems of friction taking into account different working and material input parameters during braking, are included in the articles [14,15].

This paper is a continuation of studies on frictional heating and temperature distribution in the sliding components of railway braking systems from the article [3]. The main purpose is to conduct a comparative analysis of the temperature evolution in the disc brake of the diesel multiple unit during a single and repeated brake application, calculated on the basis of: