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Thermal analysis of locomotive wheel-mounted brake disc

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HIGHLIGHTS

► At each time step the local HTC was calculated, and used for disc thermal analysis.
► Numerical results compare well with experimental data.
► Lagging effect renders no cooling at the beginning of the braking.
► Disc surface temperatures increased with increasing braking time, and then decreased.

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ABSTRACT

In recent decades the improvement of the braking performances are required due to high speed of trains. The generated frictional heat, during braking operation causes several negative effects on the brake system such as brake fade, premature wear, thermal cracks and disc thickness variation. It is then important to determine the temperature field of the brake disc. In the present work, thermal analysis of the wheel-mounted brake disc R920K for the ER24PC locomotive is investigated. The brake disc and fluid zone are simulated as a 3D model with a thermal coupling boundary condition. The braking process is simulated in laboratory and the experimental data are used to verify the simulation results. During the braking, the maximum temperature was observed in the middle of braking process instead of the braking end point. Moreover, a large lagging was observed for fins temperature which renders no cooling at the beginning of the braking.

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1. Introduction

As the speed restriction of trains continues to expand, overheating and thermal deformation on brake systems are going to be critical for emergency braking. Even if dynamic braking systems are used in normal service braking, their performances are not sufficient to ensure an emergency braking at high speed. So friction braking systems have crucial role in emergency braking. For several years, brake discs increasingly became more popular than shoes brakes in many different types of vehicles. Several advantages of brake discs over shoes brakes are reported, including better stopping performance (disc cooled readily), easy-to-control (not self-applying) and less susceptible to brake fade, which largely contributed to their popularity [1]. The thermal analysis of brake discs is a primary stage in the study of the brake systems; because the temperature determines thermo-mechanical behavior of the structure. In the braking surface, high temperatures and thermal gradients are produced. This generates stress and deformations in which the consequences are manifested by the appearance of cracks [2]. During the normal braking, influence of the radiation heat transfer on the total amount of dissipated energy to the surround is insignificant [3–7], and conduction and convection modes of heat transfer play a crucial role in contribution of heat exchange of the brake system. The problem of the fluid flow between the fins of the ventilated type of brake discs is often analyzed in individual studies based on the CFD method [8–11]. Nevertheless often an average, constant value of the heat transfer coefficient is used at temperature calculations in brake discs [5,12,13].

As it is reported in the experimental results, Complex structure of the ventilated type brake disc causes a great change in the local heat transfer coefficient distribution. Therefore, assuming a constant value for heat transfer coefficient does not seem to be logical.

In this paper, thermal analysis of the wheel-mounted brake disc R920K for the ER24PC locomotive which is manufactured in MAPNA Locomotive Engineering and Manufacturing Company (MLC) in cooperation with SIEMENS AG is investigated. Driving force of ER24PC Loco is supplied using a diesel-electric engine with the maximum speed of 160 km/h. This locomotive is used to pull the passenger wagons. For calculating the heat transfer coefficient,

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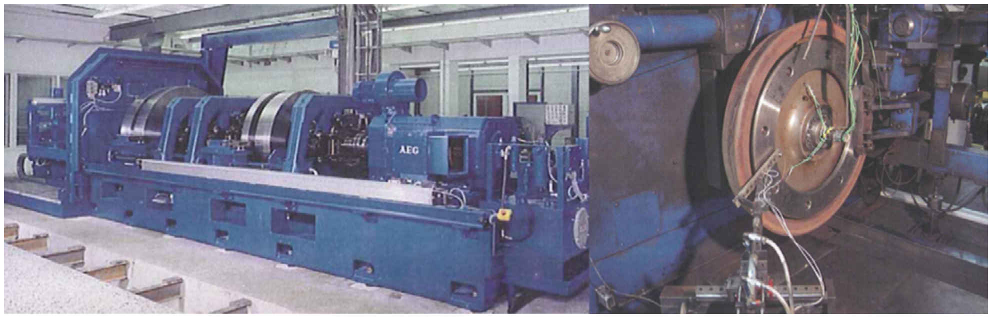

Fig. 1. ZF-dynamometer with coupleable flywheel masses and assembly of loco wheel.

the fluid flow within the channel was modeled using the FLUENT CFD software. At each time step, due to the locomotive speed and temperature distribution in the brake disc, the local heat transfer coefficient of fins was calculated and was applied as a boundary condition for the brake disc thermal analysis. An experimental data verified the modeling results.

2. Experimental set up

A railway brake disc system is tested on the ZF-Dynamometer (Fig. 1) in the Faiveley Transport Company and results were reported to MLC [14]. ZF-Dynamometer is able to run with specific mission profiles in dry and wet conditions. The dynamometer has an electric motor of 536 kW and up to four coupleable flywheel masses to simulate various weights and loads of vehicles. The measurement of temperature is a very important step in the test procedure. For this purpose, a K Type Thermocouple in 1.5 mm thickness was used. For modeling the braking phenomena, locomotive wheel was accelerated with the constant value of 0.8 m/s^2 and reached to the desired velocity, then the braking started and caused constant deceleration with the rate of 1.117 m/s^2. During the braking, braking surface and fins wall temperatures were recorded and used for validating the numerical results.

3. Modeling

The ER24PC locomotive consists of two bogies. Each bogie has four wheels with one set of wheel-mounted brake discs which consists of two brake discs arranged on both sides of a wheel and are bolted together through the wheel web (Fig. 2).

3.1. Thermal modeling

Regarding to the uniform pressure or the constant wear boundary condition at the contact surface, two methods are available for calculating the braking heat generation rate. Uniform pressure distribution in the contact region is often valid when the pad is new. However after braking for several times, assumption of uniform wear is more pragmatic. In this study, the pad was used several times and uniform wear between pad and brake disc is stabilized, hence the heat flux is just a function of time and it is independence of the spatial variables [5].

For a vehicle which is decelerating on a level surface from a higher velocity V_1 to a lower velocity V_2 the braking energy E_b can be written as

$$E_b = \frac{1}{2}m\left(V_1^2 - V_2^2\right) + \frac{1}{2}I\left(\omega_1^2 - \omega_2^2\right) \tag{1}$$

where I is related to the mass moment of inertia of the rotating parts, m locomotive mass and ω is the angular velocity of rotating parts. If the locomotive stops completely ($V_2 = \omega_2 = 0$) then all the rotating parts will be expressed relative to the revolutions of the wheel. Eq. (1) can be rewritten as follows

$$E_b = \frac{1}{2}m\left(1 + \frac{I}{R_w^2 m}\right) V_1^2 = \frac{1}{2}k_{cf} m V_1^2 \tag{2}$$

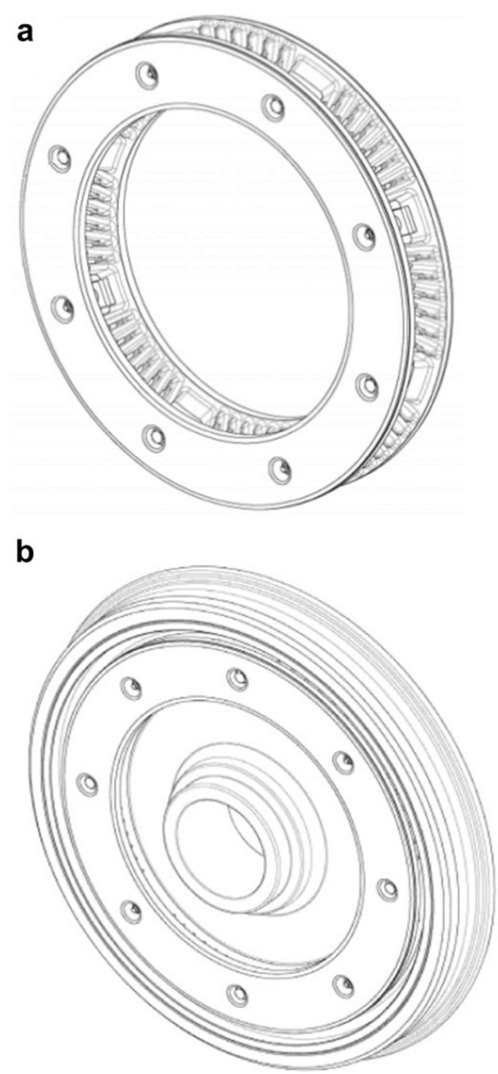

Fig. 2. (a) Wheel-mounted brake disc, (b) wheel-mounted brake disc R920K, set, mounted.

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