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A NUMERICAL STUDY OF TEMPERATURE AND STRESS DISTRIBUTIONS IN THE BRAKING DISK

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Abstract: *Peak temperatures in the brake system have been identified numerically using calculation procedures. These calculation approaches were validated on the simple rotating disk involving the heat transfer by a convection, conduction and radiation. Furthermore the stress calculation was carried out in order to identify the stress peaks. Calculations have been performed on the real train path including braking, accelerating and waiting at the train stations. Generally, the temperature peaks have been identified immediately after several braking periods within a short time interval. These temperature maxima, however, drop quickly due to the heat conduction and simultaneously the higher stresses in the disk material have been observed.*

Keywords: Temperature distributions, brake disc, simulations, braking systems, brake cycles.

1. Introduction

The objective of the paper is to investigate a temperature distribution on the rotating disk and indirectly on the surface of brake elements. Results of the temperature distribution in time and space were used to identify temperature peaks at surfaces of the rotating disk. Furthermore, the stress analysis was performed. Peak temperatures determined for the particular train path including braking, waiting at stations and acceleration can predict service conditions of the braking system and help to find appropriate materials for the brake elements in regarding to the temperature performance.

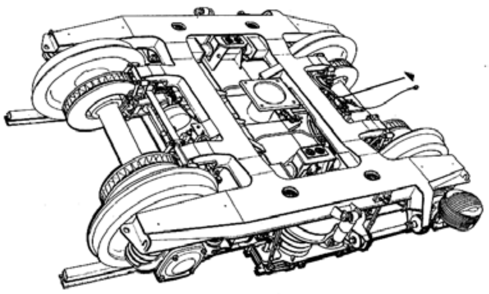

Fig. 1: The example of disc brakes for rail vehicles - GP 200 S chassis for wagon Amee and Bmee without axle generator [Reference 5]

Calculation procedures including settings of the boundary conditions were verified by the experimentally provided data. Temperature distributions obtained numerically were used for the structural analysis. The cast iron material commonly denoted as EN-GJL-250 (CSN 42 2425) was proposed by designer as the main material of brake discs with material properties as follows: the tensile modulus $E = 110000\,\mathrm{MPa}$; the Poisson number $\mu = 0.26$; the density $\rho = 7200\,\mathrm{kg \cdot m^{-3}}$; the specific heat capacity $c_p = 460\,\mathrm{J \cdot kg^{-1} \cdot K^{-1}}$; the thermal expansion coefficient $\alpha_1 = 11.7 \cdot 10^{-6}\,\mathrm{K^{-1}}$ and the thermal conductivity $\lambda_t = 48.5\,\mathrm{W \cdot m^{-1} \cdot K^{-1}}$.

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2. Methods

To identify temperature peaks at the surface brake elements, the numerical analysis of the heat flux balance was performed. The heat was generated because of a friction during the braking phases and the cooling of the brake disk was controlled by the convection and the heat transfer to surroundings. This heat transfer to surroundings was carried out by a force or natural convection, heat conduction and radiations. The intensity of the heat transfer for different heat transfers is given by parameters such as a speed of the rotating disk, the coefficient of the heat conductivity, size of surfaces, temperature differences etc. Details about fundamental aspects of the heat transfer can be found e.g. in the reference 1. The commonly used non-dimensional parameter for convection is the Nusselt number which takes a form as

$$N_U = \alpha \cdot \frac{l_{ch}}{\lambda} \tag{1}$$

where α is the heat transfer coefficient for a convection, l_{ch} is characteristic length and λ is the heat transfer coefficient for a conduction. From experiments of the simple rotating disk, the Nusselt number varies between values from tens to thousands depending on the Reynolds number (references 2 and references 3). The Reynolds number is defined as

$$R_e = \omega \cdot \frac{R^2}{\upsilon}. \tag{2}$$

where ω is the angular velocity, R radius and υ is a kinematic viscosity. The Nusselt number for different Reynolds numbers is shown in Figure 2.

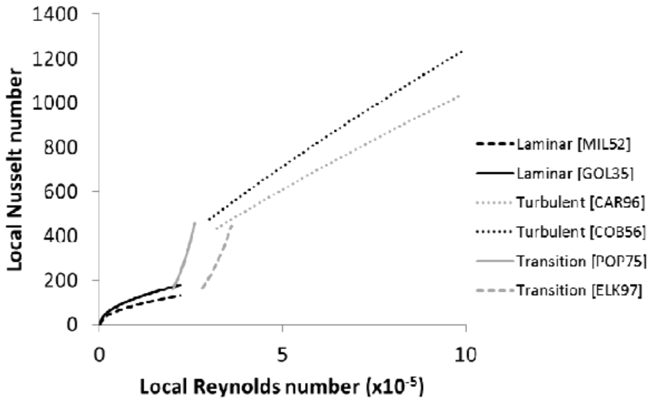

Fig. 2: Local Nusselt numbers over a rotating disc in quiescent air for isothermal surface

In our study, The Reynolds number is lower than $3 \cdot 10^5$ so that the laminar flow conditions were considered. the Nusselt number took values from 10 to 200. Taking into account the heat transfer by a radiation, the estimated heat flux of the rotating disk can be calculated. The combination of the theoretical calculation and experimental results provided the heat flux value which can be used further as the boundary conditions for simulations. However, the averaged surface temperature field, the constant surrounding conditions and many other simplifications can consequently reduce the accuracy of the heat flux prescription. In our study the heat transfer balance was simulated for the service condition including braking, acceleration and waiting with short time steps.

Because of the short time step, the influence of the radiation in respect to the small surface and the lower temperature can be neglected. Therefore, only natural and force convection can be considered for calculations. In this particular case, only the heat convection was taken into account and the heat coefficient for convection was directly prescribed at boundary conditions. The solver calculates automatically the amount of the heat transfer due to convection based on the actual local temperate difference between disk surfaces and surroundings. Results were validated by experiments which described declines of the heat at the disk surface during rotations or breaks. The figure 3 shows the drop of the surface temperature calculated numerically and validated by experiments. For the particular heat transfer coefficient prescribed in the simulation, the satisfied match could be found between experimentally and numerically provided surface temperature of the brake disk.

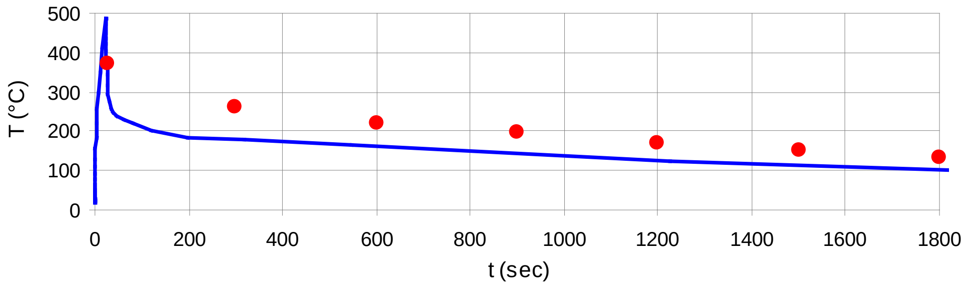

Fig. 3: The temperature drop in time during braking. The blue colour indicates the heat convection and red points express measured values.

Measured surface temperatures on the disk at the beginning and at the end phase helped to precise the value of the heat convection coefficient (see Fig. 3). Besides the convection, the heat accumulation was considered for calculations as well. However, the heat accumulation effect was significant only in particular service regimes mostly at the beginning. The amount of the heat accumulated into materials was estimated from the proportion of temperature changes provided by experiments.

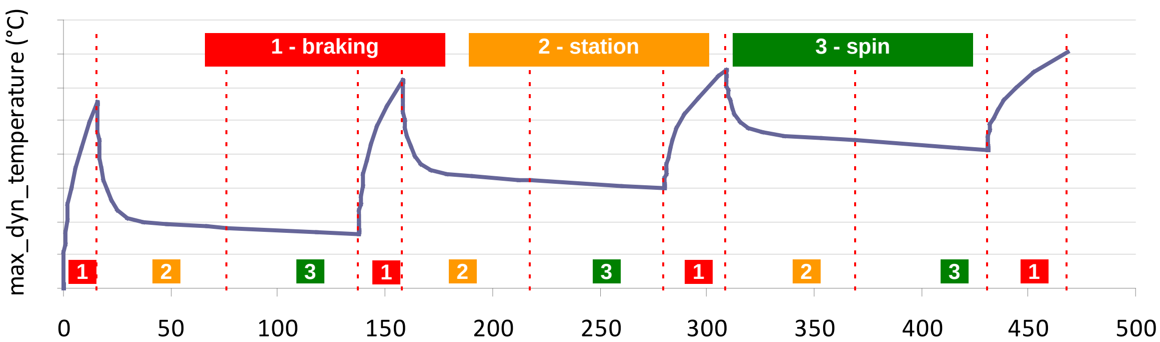

Fig. 4: The development of the maximum temperature on the model versus time (simulation of experiment)

3. Simulation of driving cycle - calculation of surface temperature on the brake disc

For setting of boundary conditions, data about service train path e.g. speed limits in time etc., were used.. Furthermore, the simulation study considered different slopes of the train track expressed in ‰ which took the positive or negative value based on the climbing or descending of the train. Moreover the braking by the engine was taken into account in the study which finally led to a reduction of the loading of the braking system. Results such as peak surface temperatures etc. were plotted in time of the train movement. Other effects such as the heat conduction in the brake elements were not considered in calculations because of the complexity. Nevertheless, experiments demonstrated obviously that the amount of the heat transported towards brake elements accounted approximately 3% of the total generated heat due to braking. Therefore, this part of conduction was neglected for further calculations.

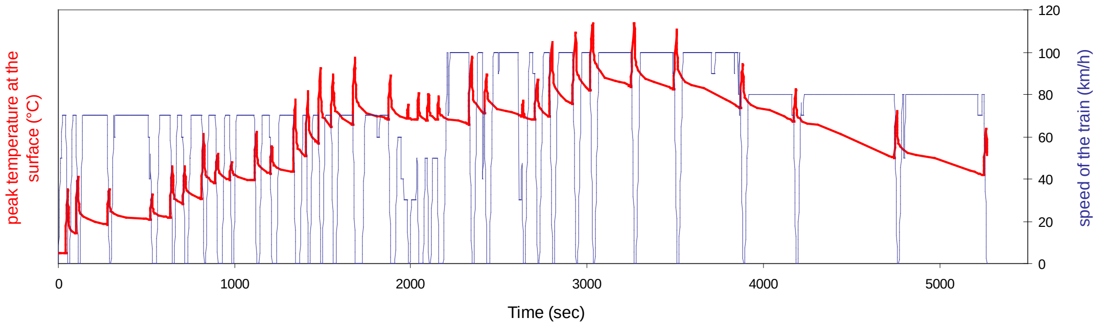

Fig. 5: Graph of maximum instantaneous surface temperature at the contact surface of the brake disc with the brake pads on the time (simulations of the driving cycle)

4. Stress analysis on the brake disc

Following calculations deal with the stress identification on brake discs for stop mode of train during one braking period, from the speed of 100 km/h to stall. The simulated model consisted of three components - brake disc, hub and ground cloth. Because of the complexity, all three parts of the model were considered as a one solid body without any gaps between them. Simultaneously, the pre-stressed bolt connections were neglected. The great influence on the stress results had a fixed storage of the brake disc to the hub through screw connections. The stiffness of the hub did not allow deformations because of temperatures and the thermal expansion coefficient of the material of the disc. For materials of the brake disc the ultimate tensile strength (R_{mt} = 250 MPa) was lower than the limit of the compressive strength (R_{md} = 950 MPa). The classical hypothesis HMH (Huber-Mises-Hencky) was applied for tough materials and it was inappropriate for our calculations. For our particular problems the basic hypothesis of strength for brittle materials, like e.g. Mohr hypothesis (basic linear variation also known as Mohr-Coulomb) was suitable and therefore it was preferred.

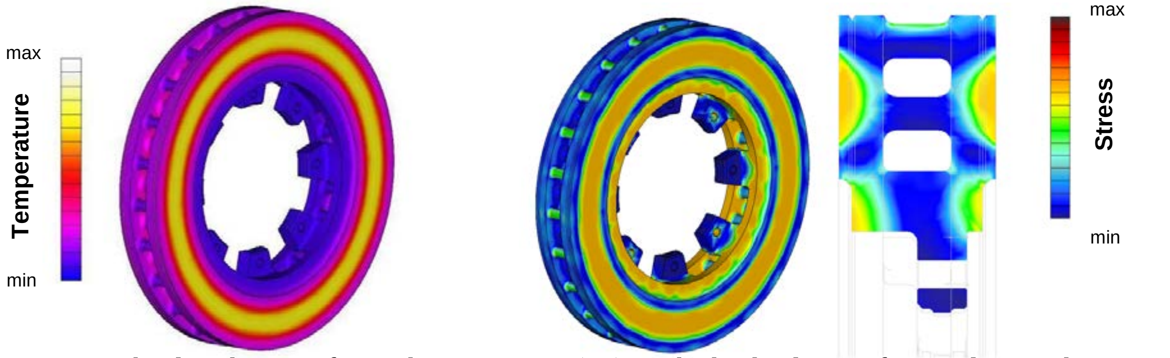

Fig. 6: The distribution of actual temperature (°C) on the brake disc surface and equivalent stress (von Mises) (MPa) on the brake disc at the end of the braking phase

5. Conclusions

The paper presented possibilities to numerically predict temperature and stress distributions in time of the train path. Thermodynamics knowledge and experiments were used to determine values of the natural or force convection coefficients. The obtained results e.g. peak surface temperatures determined service conditions under which the brake element materials must survive. The train path contained several braking, acceleration and waiting periods. For particular train path, temperature peaks were identified in time. The temperature maxima appeared immediately after the braking. In several seconds after the braking, this peak surface temperature rapidly dropped due to intensive heat conduction in the brake disc. However, even the short exposition by the higher temperature can be crucial for several materials used commonly for braking elements. Furthermore, stress analyses were performed and the peak stress was located. This numerical study demonstrated a feasibility to predict several operating conditions and limits which can be take into account for a design of the brake system.

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References

Incropera, F.P. et al. (2007) Introduction to Heat Transfer, John Wiley&Sons.

Harmand, S., Pelle, J., Poncet, S. & Shevchuk, I.V. (2007) Review of fluid flow and convective heat transfer within rotating disk cavities with impinging jet.

Shevchuk, I.V. (2009) Convective Heat and Mass Transfer in Rotating Disk Systems. Springer Verlag, Berlin, Heidelberg, pp.1-239.

Brabec, P., Fraňa, K. & Voženílek, R. (2015) Calculation of the braking discs, technical reports, Technical university of Liberec.

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