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Computer Simulations on the Thermal Behaviors of a Friction Pad in High-Speed Train Disk Brakes

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Abstract : The thermal behaviors of disk-pad braking models has been analyzed for a high-speed train brake system using the coupled thermal-mechanical analysis technique. The temperature distribution, thermal distortion, and contact stress in the disk-pads contact model have been investigated as functions of the convective heat transfer rate. The FEM results indicate that multiple spot type pads show more stabilized thermal characteristics compared with those of the flat type pads for the increased convective heat transfer rate. The maximum contact stress for a friction pad loaded against a rubbing disk was occurred on the edge of the pad at the disk-pad interface.

Key words : brake, flat pad, multiple spot pad, thermal behavior, braking time, FEM

Introduction

A contact type brake system must satisfy a certain set of requirements such as a stabilized friction characteristics, non-even dynamic behaviors due to unstable temperature distributions, no or minimum vibration and noise, and low wear rates for a disk and pads. All these technical goals have to be achieved simultaneously at a reasonable cost with a safety, comfort, reliability and durability.

The running speed of 300 km/h in a high-speed disk brake needs updated and improved safety standards of a braking system which are affected by many factors. Typically, the designing of disk-pads brake has been dependent on many experiences. The disk-pad contact problems were generally solved by trial and error, or by testing the prototype until the right result was obtained.

The thermal problems on the contact surface between the disk and the pads, which mainly depends on the dissipating mechanism of the braking energy are analyzed in order to better understand the design parameters which influence the thermal behaviors of the disk and the pads.

The thermal aspect of disk-pad brake behavior has been the focus of most work because of the need to control the temperatures to the acceptable levels [1-3]. The temperature level should be kept low to escape coned wear and thermal deformations of the disk-pads brake.

In actual running conditions, the temperature of a disk plate is influenced by wind generated in the vicinity of the disk-pad structure. Therefore, the convective heat transfer effects must be added to get more accurate informations in the disk-pad contact mode with a coupled thermal-mechanical analysis [4,5].

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In the exact analytical approach, it is usually complex and difficult to include disk rotor edge effects and heat conduction to the pads, hat, and mounting flange. Therefore, the finite element method is used to obtain approximate solutions to more geometrically exact models [6]. The computer simulation technique gives good overall brake temperature induced informations if one is interested in computing many sequential braking cycles during braking periods.

This paper presents the thermal behaviors of the friction pads for a high speed disk type brake system. A two-dimensional numerical analysis calculating the temperature distributions, thermal behavior distortions, and contact stresses between disk and pads has been presented using a coupled thermal-mechanical analysis technique.

Numerical Computations

Braking energy

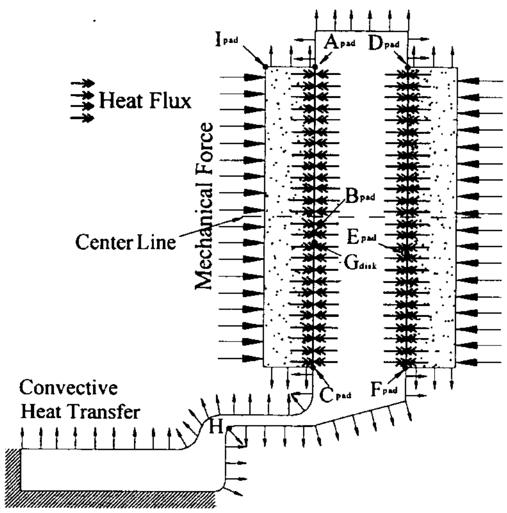
The thermal analysis of brake requires an accurate determination of both the total energy absorbed by the brakes and how this energy is distributed between the disk and the pads. The distribution of braking energy between the rubbing surfaces of the brakes is related directly to the thermal resistance associated with both sides of the interface where the heat is generated.

For short braking periods, the total heat generation at the interface equals the heat absorbed by the disk and pads. It becomes convenient to express the relative braking energy γ absorbed by the disk or pads in terms of the material properties as

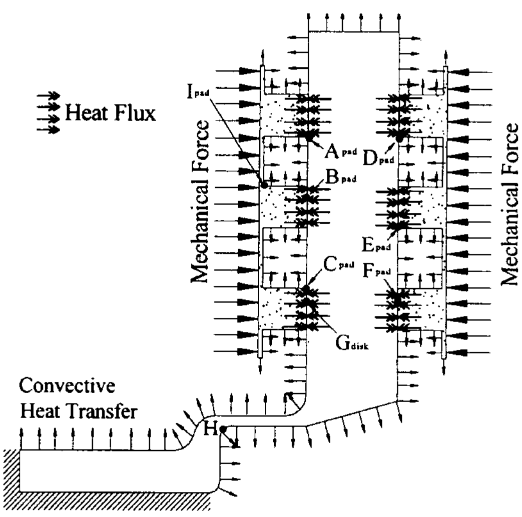
$$\gamma = \frac{1}{1 + \sqrt{\frac{\rho_p c_p k_p}{\rho_d c_d k_d}}}$$

where
c_d = disk specific heat, Nm/kg-K
c_p = pad specific heat, Nm/kg-K
k_d = disk thermal conductivity, Nm/mh-K
k_p = pad thermal conductivity, Nm/mh-K
ρ_d = disk density, kg/m^3
ρ_p = pad density, kg/m^3

The braking energy is naturally partitioned between the disk and the pads depending on the temperature field in pads and disk surfaces. The partition of the braking energy depends on the thermal characteristics of the friction pad materials and on the boundary conditions such as convection and radiation. Based on the above equation, the absorbed brake energy by the disk is approximately 81% for a single brake-to-stop event if rolling and air resistance are neglected in this study. The remained portion of the generated heat energy at the interface will go to the pads.



(a) Flat pad - disk contact model



(b) Multiple spot pad - disk contact model

Fig. 1. Friction pad-disk contact model and boundary conditions.

FEM models and boundary conditions

In applying the disk-pad contact models, it is assumed that:
① The braking energy is uniformly distributed over the entire swept area of both rubbing surfaces.
② The rotor geometry could be approximated by a semi-infinite slab wherein a heat flux varying linearly with time is supplied to the slab at each of its two surfaces.
③ The disk-pad rotor is homogeneous.
④ A constant uniform pressure distribution between the disk and the pads is assumed.
⑤ Constant cooling occurs at the rubbing surface of the rim, hat and mounting flange simultaneously.
⑥ No thermal radiation takes place.
⑦ The deceleration rate during braking is constant.
⑧ Conduction to and cooling from the hub is neglected.
⑨ Thermal properties of disk-pad materials are invariant with temperature.

A disk-pad sector for two contact models as shown in Fig. 1 was analyzed for determining disk-pad brake temperature distributions, thermal deformation in axial direction, and contact stress for a friction pad. The boundary conditions and the important positions of the pads are shown in Fig. 1.

The thermal analyses for two disk-pad models were performed using a non-linear FEM program MARC [7]. Four-node isoparametric quadrilateral ring and four-node heat transfer axisymmetric ring elements are simultaneously used in the finite element analysis. The finite element model was subdivided into 1,507 elements and 1,694 nodes for a disk-flat pad contact model and 1,319 elements and 1,682 nodes for a disk-spot pad contact model. The computer simulation data and material properties for the FEM analysis are given in Table 1 and Table 2, respectively.

To analyze the mechanical and thermal effects for two disk-pad contact models, Fig. 2 describes three deceleration curves for a brake application such as
① From 300 km/h to 230 km/h: -0.005143V+0.892483 m/s^2
② From 230 km/h to 70 km/h: -0.010062V+1.20675 m/s^2
③ From 70 km/h to 0 km/h: 1.011 m/s^2

In this deceleration curve, a mass per disk is 17,000 kg and

Table 1. Simulation data for a braking system.

Simulation Conditions	Values
Vehicle axle load, kg	17,000
Wheel diameter, mm	920
Number of disk per a axle	4
Area of a disk, m^2	0.26
*Area of a pad, m^2	0.04
Thickness of a disk, mm	45
*Thickness of a pad, mm	24
Initial velocity of braking, km/h	300
Convection coefficient, W/m^2°C	0, 50, 100, 150, 200
Atmospheric temperature, °C	35

*UIC code 541-3 for a pad

Table 2. Material properties for a disk and pads.

Specifications	Disk	Pads
Elastic modulus, N/mm^2	2.15×10^{11}	1.05×10^{9}
Poisson's ratio	0.3	0.3
Mass density, kg/m^3	7,850	2,500
Coefficient of thermal expansion, mm/mm · K	12×10^{-6}	7×10^{-6}
Thermal conductivity, W/m · K	45	4.6
Specific heat, J/kg · K	460	800

the braking operation lasts 118.7 seconds.

The coefficient of friction is considered as constant during the whole brake application for a simple computation. A constant convection coefficient was applied to all contact surfaces exposed to air. The heat transfer into the disk and pads was considered to be axisymmetric in this study. Initially the heat is assumed to be uniformly distributed on the disk-pad contact surface of the whole ring.

Result and Discussions

The computer simulations on the thermal contact problems of a disk-pad brake has been made to obtain temperature distributions, thermal behaviors, and contact stresses in the disk-pad type brake during braking time. The convective heat transfer rate of 200 W/m^{2o}C was applied to all contact surfaces of the disk-pad brake exposed to air as boundary conditions.

Temperature distribution

To obtain the temperature distribution in the disk-pad interface during braking, Fourier's two dimensional equation for non-steady heat conduction which dominates the temperature condition in the pads was solved using the finite element method.

Based on two contact models in Fig. 1 and the above assumptions, the temperature distributions on the rubbing

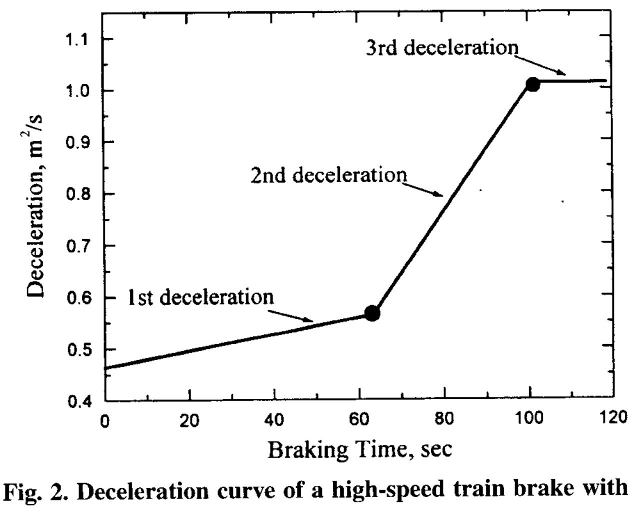


Fig. 2. Deceleration curve of a high-speed train brake with a maximum speed of 300 km/h.

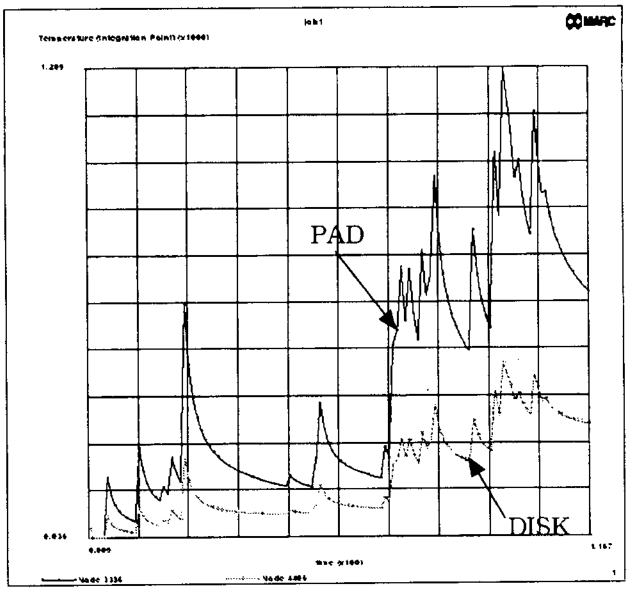


Fig. 3. Temperature distribution of the disk-flat pad contact model with a constant coefficient of the convective heat transfer rate, 200 W/m^{2o}C.

surface between the disk and the pads are shown in Figs. 3 and 4 as a function of braking time. At the start of each brake application the surface temperature of a disk-pad rotor rapidly increases to a maximum value and then decays due to conduction of heat into internal portions of a disk and pads, to a smaller extent, the hat and mounting flange. In Figs. 3 and 4, the temperature of the pads shows high compared with that of the disk with a convection coefficients of 200 W/m^{2o}C.

As indicated in Fig. 1(a), the maximum temperature of the flat pads occurs at the position Bpad, which is a little lower part of the center line of the flat pads. In Fig. 1(b), the maximum temperature of the spot pads is reached near the center of the disk-spot pad contact, Bpad.

In Fig. 5, the maximum temperature distributions of the disk-pad contact models are presented as a function of the convective heat transfer rate from zero to 200 W/m^{2o}C. The maximum temperature of the disk-spot pad contact model decreases as the convective heat transfer rate increases. But the maximum temperature in the disk-flat pad interface does not influenced by the convection and radiation. The calculated results indicate that the maximum temperature of the multiple spot pads is a little low compared with that of the flat pads due to the increased convection effect, 200 W/m^{2o}C as shown in Fig. 5.

At the end of one braking stop, the maximum temperature on the rubbing surface of the disk-flat pad has nearly reached a constant asymptotic value, over 315°C for a disk and 650°C for the flat pads as shown in Fig. 5. The maximum temperatures of the disk-spot pads during braking are 360°C for a disk and 630°C for spot pads with a convection coefficients of 200 W/m2°C. In these results, the maximum temperature of the pads is nearly two times high compared with that of the disk rubbing surface. This is primarily due to a low thermal conductivity and thermal capacity of the friction materials.

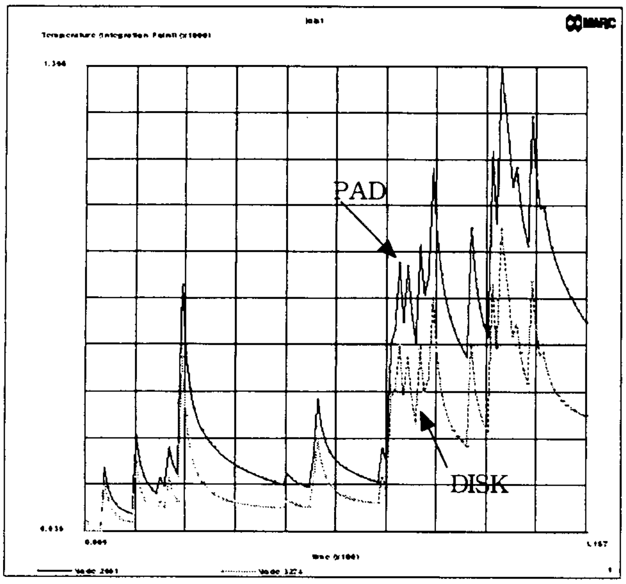


Fig. 4. Temperature distribution of the disk-spot pad contact model with a constant coefficient of the convective heat transfer rate, 200 W/m²°C.

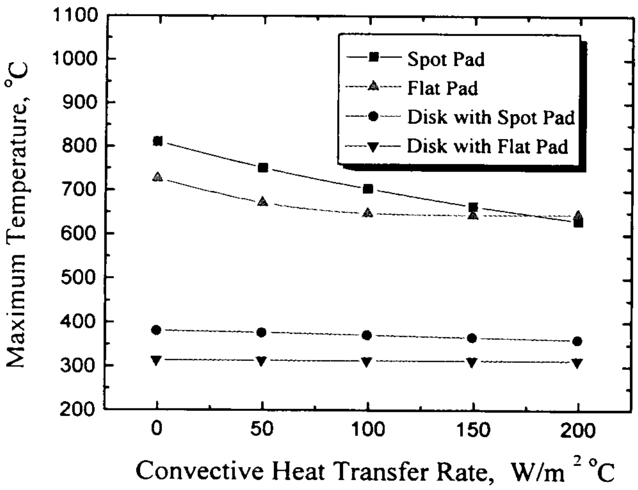


Fig. 5. Maximum temperature distribution of two disk-pad contact models as a function of the convective heat transfer rate.

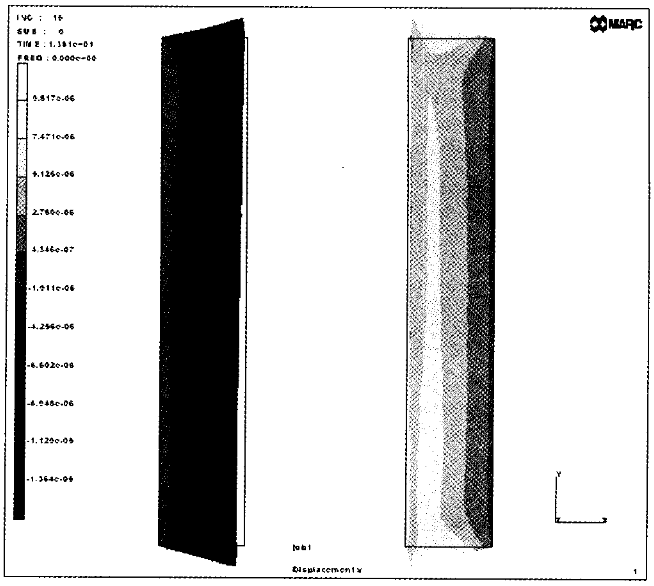


Fig. 6. Deformed profiles of the flat pads.

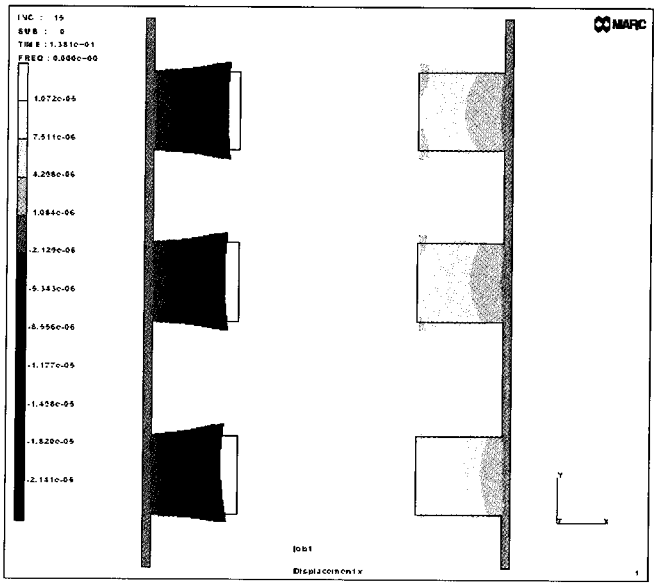


Fig. 7. Deformed profiles of the multiple spot pads.

Thermal behaviors

During braking, the disk plate will be deflected due to non-uniform temperature distributions transported from the rubbing surfaces. On cooling, the deformed contact surface between the disk and the pads returns to a normal position. The repeated thermal loads due to braking and cooling cycling may lead to coned wear of the disk-pad interface, vibration and low energy squeal. Therefore, it is necessary to so design the disk-pad type brakes, that where possible, zero, or minimum thermal deformation takes place. The thermal behavior in axial direction of a disk brake is important if we investigate the thermally caused deformation of a disk-pad interface, microscopically.

Fig. 6 shows the deformed geometry of the flat pads in axial direction after 13.81 seconds of the brake time. The deformed shape of the pads shows a concave. After 118.7 seconds, the maximum deformation in axial direction occurs at the position Bpad, right face of left flat pad in Figs. 1(a) and 6. The disk plate was deflected to the left side due to thermal effects on the hat surface, macroscopically. Therefore, the maximum displacement of the flat pad occurs at the right face of left flat pad.

Fig. 7 shows the axial displacement the spot pads after 13.81 seconds of the brake time. After 118.7 seconds, the maximum displacement in axial direction occurs at the position Bpad, right face of the second spot pad, which is located near the upper edge of spot pads before braking.

Basically, the general phenomenon of the disk-spot pads is very similar to the disk-flat pad behaviors. The disk, which is rubbed against the spot pads, deflected to the left side due to

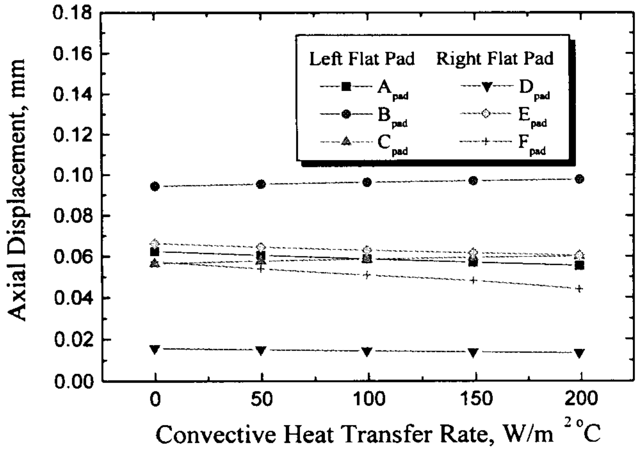


Fig. 8. Axial displacement of the flat pads at the contact position against the disk.

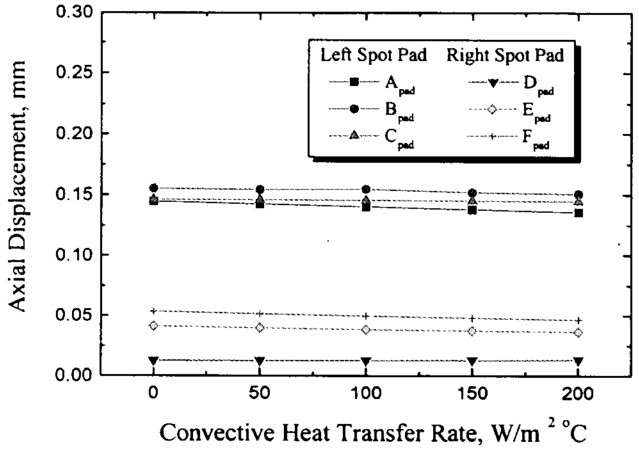


Fig. 9. Axial displacement of the spot pads at the contact position against the disk.

thermal effects on the hat surface.

In Figs. 8 and 9, the maximum axial displacements are presented in terms of the convective heat transfer coefficients for the flat and spot pads, respectively. In Figs. 1(a) and 8, the axial displacement of the flat pads is slightly increasing or decreasing depending on the thermal behaviors of the hat area and the rubbing contact position. In Fig. 9, the FEM results show that the maximum axial displacement of the multiple spot pads is slightly decreasing as the convective heat transfer coefficient increases. The axial displacements of left spot pads is about three times high compared with those of the right spot pads. This is primarily due to thermal effects of of the hat, which is transported from the rubbing surfaces.
From the calculated results, the thermal behavior of the disk-spot pad contact model shows more stable compared with that of the disk-flat pad if we consider good friction characteristics between the disk and the pads.

Contact stress

The contact stress distributions of the pads after 118.7 seconds of the braking cycle are shown in Figs. 10 and 11. For a flat

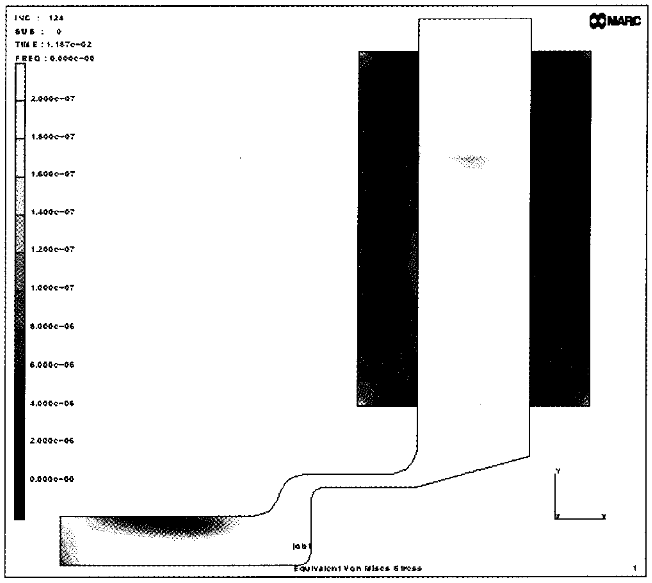


Fig. 10. Contact stress distributions of the flat pads.

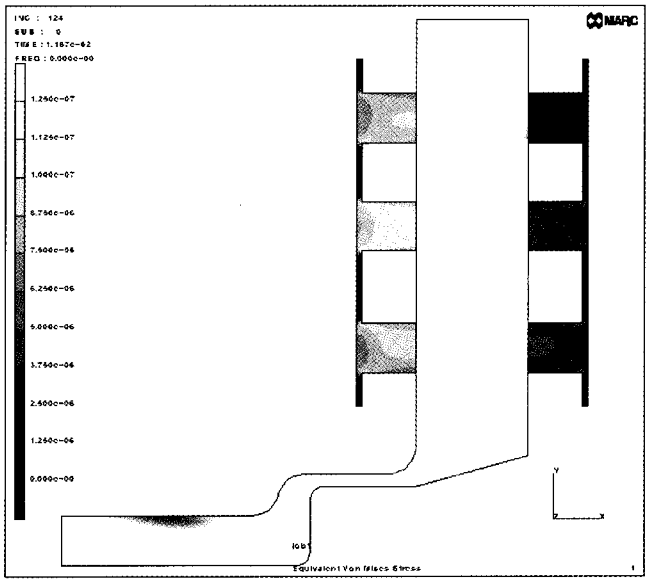


Fig. 11. Contact stress distributions of the multiple spot pads.

pad, the highest thermomechanical contact stresses occur on the edge part of the pad at the disk-pad interface. It can be seen that the largest stress concentration occurs at the corners of Ipad as shown in Figs. 1(a) and 10. The magnitude of these corner edge stresses decreases linearly as the convection coefficient and the distance from the backplate increases. The highest contact stresses are due primarily to the differential expansion between the disk and the pads.

As shown in Figs. 1(b) and 11, the maximum contact stress occurs at the bottom corner edge Ipad of the center spot of the pad. This may be influenced by the differential expansion of the spot pad-backplate interface and the local expansions of the disk surface due to concentrated thermal heatings. This differential expansion arises from the strong temperature gradients through the pad materials, particularly across the

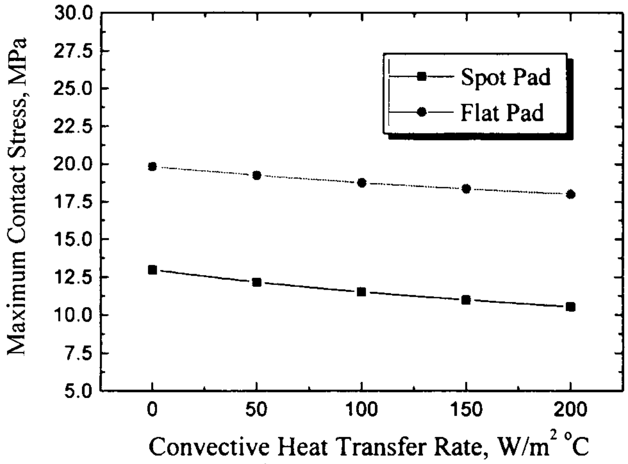


Fig. 12. Maximum contact stress for two pad models against the disk as a function of the convective heat transfer coefficient.

underlayer which has a low thermal conductivity, and from the different expansion coefficients of the pad materials.

In Fig. 12, the maximum contact stress between two pad models was presented in terms of the convective heat transfer rate from 0 to 200 W/m^2°C. In Fig. 12, the calculated FEM results indicate that the maximum contact stresses are decreasing for the increased convection coefficient. The maximum stresses of the spot pad are low compared with those of the flat pad. This is due to a distance of the corner edge from the backplate and the local dilation of the disk against the spot pad.

Fig. 12. Maximum contact stress for two pad models against the disk as a function of the convective heat transfer coefficient.

Conclusions

A finite element analysis of the two-dimensional heat conduction equation has been carried out so as to obtain the temperature distributions, the thermal behaviors, and contact stresses for two pad models.

The calculated results indicate that the maximum temperature of multiple spot pads is a little low compared with that of flat pads for the increased convection effects.

The thermal contact behavior of the disk-spot pad contact model shows more stable compared with that of the disk-flat pad model as the convective heat transfer rate increases.

For a flat pad, the highest thermomechanical contact stresses occur on the corner edge part of the pad at the disk-pad interface. For a multiple spot pad, the maximum contact stress occurs at the corner edge of the center spot of the pad. This may be influenced by the differential expansion of the spot pad-backplate interface and the local expansions of the disk surface due to concentrated thermal heatings.

Based on the results, multiple spot type pad in the disk-pad contact model may be recommended for the high-speed trains.

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