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Finite Element Analysis of Temperature Distribution and Thermally Caused Deformation in Ventilated Disk Brakes

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Abstract—In order to analyze the thermal effects of the rotor models, the finite element technique was used in this study. The length of the hat was investigated as a design parameter. At the start of each brake application the disk surface temperature rapidly increases to a maximum value and then decays due to external cooling and thermal conduction to the hat. The calculated results indicate that the long length of the hat shows the minimum deformation in axial direction, which is related to the thermal problems, coned wear, vibration and noise.

Key words : Disk Brake, Thermal Deformation, Thermal Behavior, FEM, Braking Time

1. Introduction

A brake system must satisfy a certain set of requirements such as high and stable friction, noneven dynamic behaviors due to unstable temperature distributions, no or minimum vibration and noise, and low wear rates for the rotor and friction materials. All these technical goals have to be achieved simultaneously at a reasonable cost with a safety, comfort and durability.

The temperature distribution of a disk brake due to the frictional heating is analyzed in order to better understand the design parameters which influence the thermomechanical behavior of the rim, hat and mounting flange.

The thermal aspect of disk brake behavior has been the focus of most work because of the need to control the temperatures to the acceptable levels [1-4]. The temperature level should be kept low to escape coned wear and thermal deformations of the rotor.

Typically, the designing of disk brakes has been dependent on experience. The disk brake problems were generally solved by trial and error, or by testing the prototype until the right result was obtained, making it difficult to make timely improvements of disk brake performance.

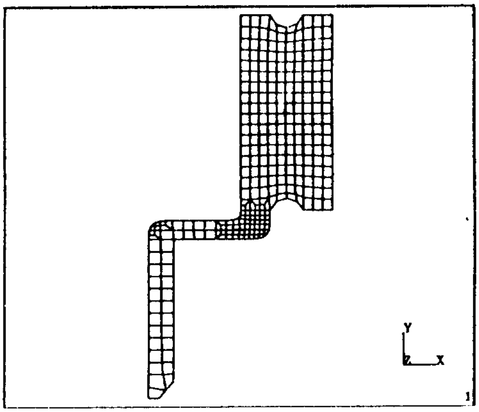
In actual running conditions, the temperature of a disk rotor is influenced by wind generated in the vicinity of the disk rotor. Therefore, parameters which consider this influence must be added to the finite element analysis. But, these are very few cases in which such wind influences have been considered. In this study, the influence of wind velocity is assumed to be constant due to lack of any data for quantification of the influence of any factor on the wind generated.

In the exact analytical approach, it is usually difficult to include rotor edge effects and heat conduction to the pads, hat, and mounting flange. Therefore, finite element method is used to obtain approximate solutions to more geometrically exact models [5,6]. This technique gives good overall brake temperature informations if one is interested in computing many sequential braking cyclings-a fade and recovery series during braking.

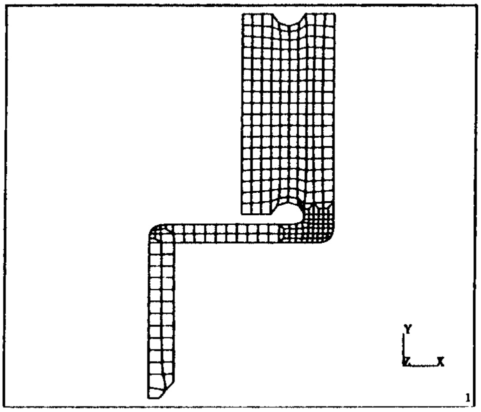
The purpose of this paper is to analyze the temperature distributions and the thermal deformations for two models. In these models, the length of the hat was investigated as a design parameter.

2. FE Analysis

A FEM model of an entire rotor may be desirable if circumferential temperature variations are of interest as the disk rotates during braking. Small sector models which are shown in Fig. 1 can, however, be utilized to accurately predict the temperature distribution and thermal behavior. The rotor model of Fig. 1(a) shows the short length of the hat compared with that of Fig. 1(b). But the total length from the left side of mounting flange to the right face of the rim is same for two models. The different hat length between two models as a design parameter may



(a) Model I



(b) Model II

Fig. 1. FEM models for two rotors.

present an important role for the thermally caused deformation in axial direction.

In applying the sector model with a FE mesh results to a full rotor, 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 rotor is homogeneous.
4. Constant cooling occurs at the rubbing surface of the rim, hat and mounting flange simultaneously.
5. No thermal radiation takes place.
6. The deceleration rate during braking is constant.

Table 1. Simulation data for the finite element analysis

Parameters	Values
• Total car weight, kg	1,325
• Deceleration rate, m/s^2	5.89
• Initial velocity, km/h	97
• Mass moment of inertia, g	470
• Heat transfer coef., W/m^{2o}C	60.74
• Atmosphere temperature, °C	35
• Braking effectiveness on front wheels, %	80

Table 2. Material properties for a pad and rotor

Materials / Parameters	Pad (Asbestos free friction material)	Disk (Cast iron)
• Elastic modulus, N/mm^2	820	125
• Poisson's ratio	0.25	0.25
• Density, kg/m^3	3,660	7,100
• Coef. of thermal expansion, mm/mmK	20×10^{-6}	12×10^{-6}
• Thermal conductivity, W/mK	1.01	54
• Specific heat, J/kgK	1,034	585

7. Conduction to and cooling from the hub is neglected.
8. Thermal properties of disk material are invariant with temperature.

Using the above assumptions, a rotor sector for two models in Fig. 1 was analyzed for determining disk brake temperature distributions and thermal deformations in axial direction.

The thermal analyses for two rotor models were performed using non-linear FEM program MARC [7]. 4-node isoparametric quadrilateral ring and 4-node heat transfer axisymmetric ring elements are simultaneously used in the finite element analysis. The finite element model was subdivided into 340 elements and 280 nodes for Model I and 360 elements and 294 nodes for Model II as shown in Fig. 1. The simulation data and material properties for the FEM analysis are given in Table 1 and Table 2, respectively.

3. Temperature Distributions

Analysis of heat conduction of disk rotor using

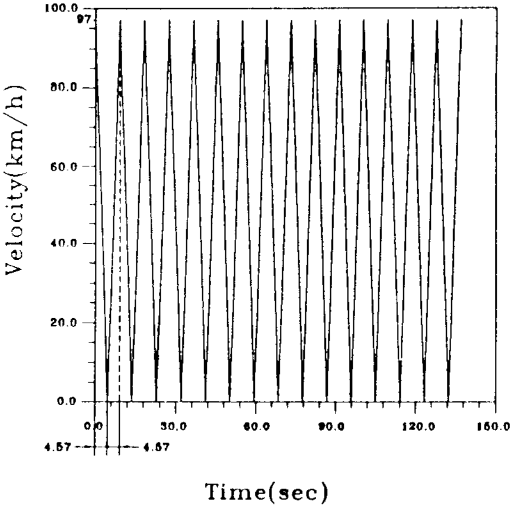


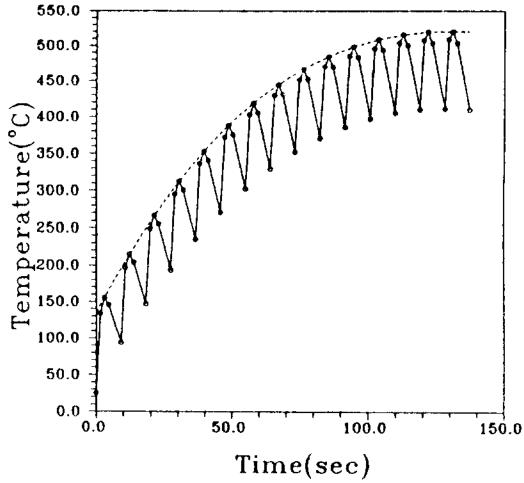
Fig. 2. Braking cycle schedule for two models.

finite element method has been made to obtain temperature distributions and thermal behaviors in the disk rotor relative to braking time. The calculation method is that the thermal flux to the disk rotor and the surface heat transfer rate of each part of the disk rotor were calculated as the boundary conditions. Then, to obtain the transitional temperature distribution in the disk rotor during braking, Fourier's two dimensional equation for non-steady heat conduction which dominates the temperature condition in the rotor was solved using finite element method.

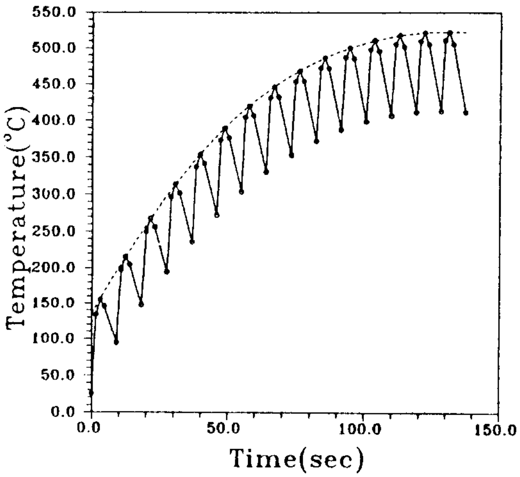
To analyze the thermal effects for two disk rotors, a braking schedule of fifteen 97 km/h fade stops was assumed as shown in Fig. 2. Each braking was characterized by a 5.89 m/s^2 deceleration from 97 km/h to zero velocity, accelerating back to 97 km/h for a braking cycle.

The thermal load applied was described power flux varying with time[1]. A constant convection (cooling) coefficient was applied to the external surfaces of the disk rotor. The heat transfer into the disk was considered to be axisymmetric in this study.

Based on two models in Fig. 2 and the above assumptions, maximum values of rim temperature on the rubbing surface are shown in Fig. 3 as a function of braking time. At the start of each brake application the surface temperature of a rotor rapidly increases to a maximum value and then decays due to external cooling (convection) and conduction of heat into internal portions of the rim, to a smaller extent, the hat and mounting flange. This decay continues until commencement of the next brake application. By the end of the fifteenth stop, the maximum temperature on the rubbing surface of the rotor has nearly reached a constant asymptotic value, over 500°C.



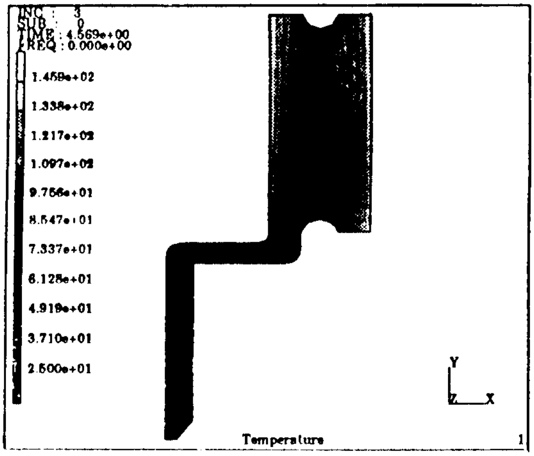
(a) Model I



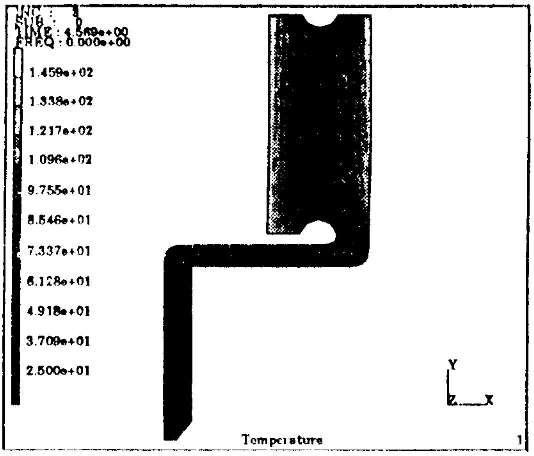
(b) Model II

Fig. 3. Maximum temperature on the braking surface as a function of time.

The calculated results indicate that the surface temperature of the rim reaches a peak at 85% of braking time, approximately. The maximum rim temperature for two rotors shows no difference between

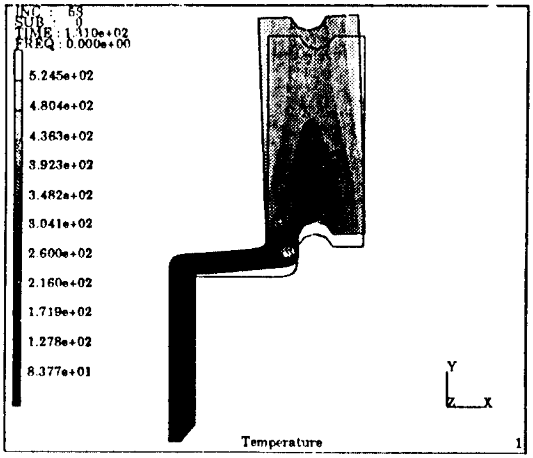


(a) Model I

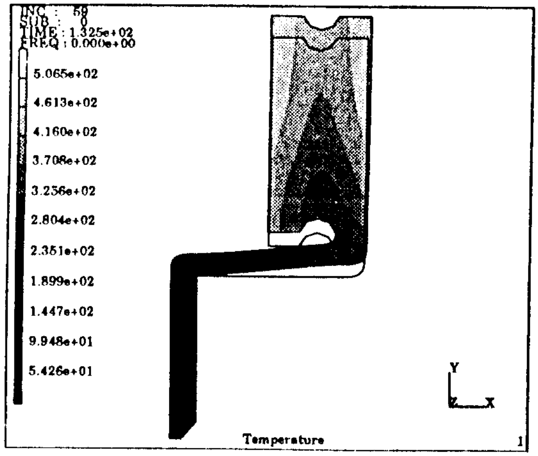


(b) Model II

Fig. 4. Temperature distribution on the deformed geometry at the first brake period.



(a) Model I



(b) Model II

Fig. 5. Temperature distribution on the deformed geometry at the fifteenth brake period.

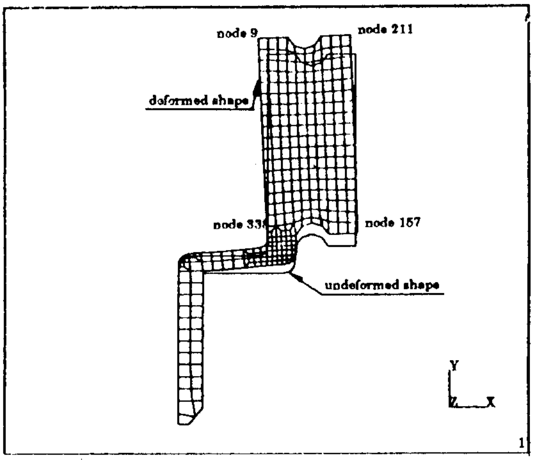
two models because the conduction of heat to the hat and mounting flange is considerably small as shown in Figs. 4 and 5.

The temperature distributions for two models at the end of the first brake stop are shown in Fig. 4. The temperature profiles shown are almost same at the first brake stop. But the temperature distributions at the end of the fifteenth brake stop show different profiles as shown in Fig. 5. This may lead to a thermal behavior depending on the dissipation rate to the hat and mounting flange.

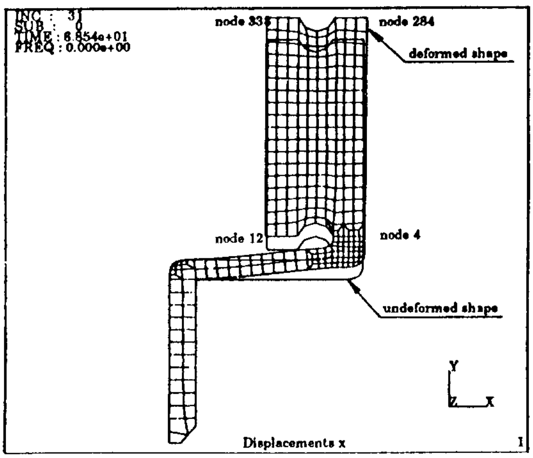
4. Thermal Behaviors

Brake rotor thermal deformation has been a problem as long as disk brakes have been existed, chiefly in the form of rotor coning.

During braking, the disk rotor will be deflected due to non-uniform temperature distributions transported from the rubbing surfaces. On cooling (running), the deformed rotor returns to a normal position. The repeated thermal load due to braking and cooling cycling may lead to coned wear of the rotor,

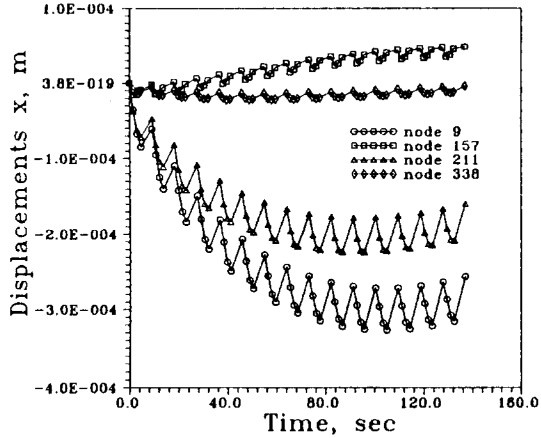


(a) Model I

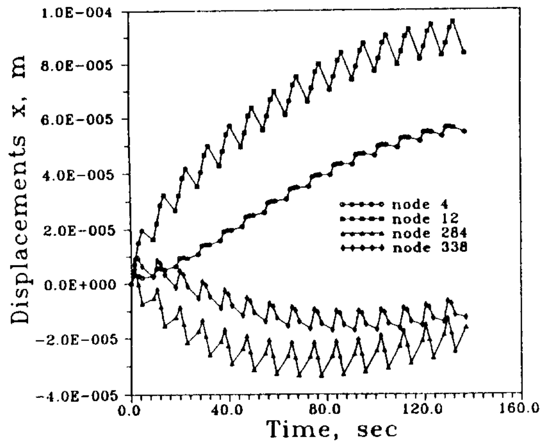


(b) Model II

Fig. 6. Node numbers on the rim.



(a) Model I



(b) Model II

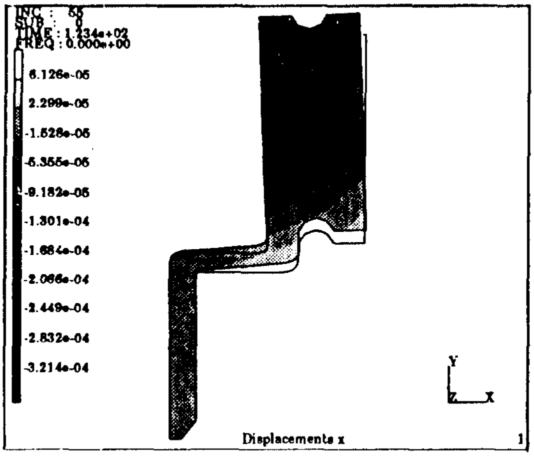
Fig. 7. Maximum deformation on the braking surface as a function of time.

vibration and low energy squeal. Therefore, it is necessary to so design the rotor, that where possible, zero, or minimum deformation takes place. The axial deformation of a disk rotor is important if we investigate the thermally caused deformation of a disk rotor.

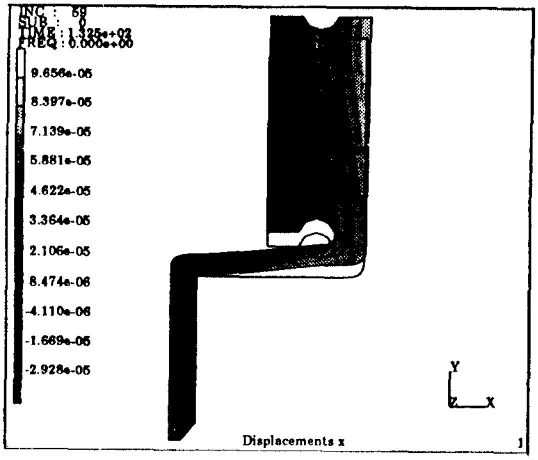
In Fig. 6, four numbers listed at each location on the rim represent the node number for two models. The node number can predict the thermal behaviors for the repeated thermal cycling load.

In Fig. 7, the axial displacement at 4 locations on the rotor rim are presented for fifteen brake stops as a function of braking time. In Fig. 7(a), the maximum deformation in axial direction occurs at the node 9, left face of the rim. And the deformed position of the node 157 moves to the positive x direction. The upper part of the rotor with the short length of the hat try to deform in the negative x direction.

In Fig. 7(b), the maximum deformation in the positive x direction occurs at the node 12, left lower part of the rim. At the upper part of the node 338 and 284 on the rotor, the deformed profiles show that the maximum deflection occurs at the tenth brake stop. And then the deformed rim profiles of the rotor try to recover from the maximum deformed shape to the original one until the fifteenth



(a) Model I



(b) Model II

Fig. 8. Deformed geometry at the fifteenth brake period.

stop. This phenomena may radically reduce the axial deformation compared with the short length of the hat as shown in Fig. 1(a). This means that most of the thermal energy dissipates to the radial direction compared with the axial direction in case of the long length of the hat. The maximum deformation in axial direction is 300 μm for the short hat of Model I and 90 μm for the long hat of Model II as shown in Fig. 7. As is obvious from the calculated results, Model II of the brake rotor shows good thermal behaviors compared with Model I. Obviously, the minimum deformation of the rotor in axial direction may reduce coned wear of the rotor, macro-vibration and low energy squeal in the running condition.

Fig. 8 shows the deformed geometry for two models at the fifteenth brake period. In this figure, two models of the rotor show the different deformed profiles.

5. Conclusions

A finite element analysis of the two-dimensional heat conduction equation has been carried out so as to obtain the temperature distributions and the thermal behaviors for two rotor models. The fifteen braking schedules were employed to investigate the thermal effects on the surface of the rotor.

The surface temperature of the rotor rim is a rising saw-toothed profiles that tends to approach a periodic behavior. The axial deformation of Model II is very small compared with one of Model I. This means that the long length of the hat may present good braking performance compared with the short length of the hat if we consider the rotor coning effect, macro-vibration and noise.

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