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Temperature Field Study of Brake Disc in a Belt Conveyor Brake

Hou Youfu, Wang Daoming, and Meng Qingrui

***Abstract*—**To reveal the temperature field distribution of disc brake in downward belt conveyor, mathematical models of heat transfer for disc brake were established combined with heat transfer theory. Then, the simulation process was stated in detail and the temperature field of disc brake under conditions of dynamic speed and dynamic braking torque was numerically simulated by using ANSYS software. Finally the distribution and variation laws of temperature field in the braking process were analyzed. Results indicate that the maximum surface temperature occurs at a time before the brake end and there exist large temperature gradients in both radial and axial directions, while it is relatively small in the circumferential direction.

***Keywords*—**Downward belt conveyor, Disc brake, Temperature field, Numerical simulation.

I. INTRODUCTION

THE braking process of downward belt conveyor is an energy conversion process, in which the energy converted from kinetic energy of both conveyor and material is mainly absorbed by disc brake. The temperature and temperature gradient on brake disc will lead to large thermal stress and fatigue failure, and thus, debase the braking force and reduce the braking safety. Therefore, deeply study on the temperature field distribution of friction pair is necessary. It could provide preconditions for studying on thermal-elastic instability problems due to frictional heat, and offer a theoretical basis for the design of high-performance disc brake; furthermore it has a good theoretical guidance and engineering applications [1], [2].

Barber [3] experimentally studied on the train brake and firstly described the generation mechanism of thermal-elastic instability of the sliding system; Zagrodzki [4] researched on thermal-elastic problems with different models in the case of constant sliding speed, but it did not conform to the actual braking conditions. Zhou [5] put forward a thermodynamic model for several cycle brakes condition, and then adopted finite difference method for thermodynamics calculation of a real vehicle. Huang [6] numerically simulated of the braking process of disc brake, but only considered the impact of a single friction plate on the temperature field. Fu [7] simulated on the temperature field of disc brake for belt conveyor with ANSYS software, however, the influence of ribs were not taken into consideration.

In this paper, nonlinear finite element method was utilized to simulate the temperature field of brake disc under conditions of dynamic speed and dynamic braking torque. Meanwhile, the influence of ribs on temperature field distribution was taken into consideration. Finally the distribution and variation laws of temperature field in braking process were analyzed.

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II. MATHEMATICAL MODEL OF HEAT TRANSFER

In heat transfer, the basic refrigeration approach can be classified into three types: convection, conduction and radiation [8]. In braking process, friction force forms in two friction pairs with relative motion, and heat continuously produces and accumulates in contact surface. An instant in time, the region between two friction pairs is not exposed to air. The temperature in contact surface is higher than that in the internal of friction pairs, thus heat conduction is the only refrigeration way; whereas, in friction and non-contact surfaces, convection, conduction and radiation exist simultaneously. The thermal conductivity of brake disc is far larger than that of friction plate, so the brake disc has an apparent better heat conduction effect.

In the modeling process of temperature field, the following assumptions were made

- Friction coefficient between brake disc and friction plate remains unchanged;
- Friction pair materials are isotropic and their thermal parameters do not change with temperature;
- All friction work converted into heat without considering the impact of material wear;
- Friction plate is in full contact with brake disc, and in contact surface, instantaneous temperatures at corresponding points of both friction plate and brake disc are equal;
- Temperature field is symmetry with respect to the central plane of the brake disc.

Three-dimensional model of disc brake is shown in Fig. 1. The input heat fluxes of brake disc and friction plate are respectively

$$q_d = F_d \cdot F_f \cdot f \cdot v(x,y,t) \tag{1}$$

$$q_p = (1-F_d) \cdot F_f \cdot f \cdot v(x,y,t) \tag{2}$$

where, F_d is the weight value of heat flux input to brake disc $\left(0 \le F_d \le 1\right)$; F_f is the weight value of friction power converted into heat flux, F_f=1; f is the friction force between brake disc and friction plate; $v(x,y,t)$ is the relative speed of brake disc and friction plate.

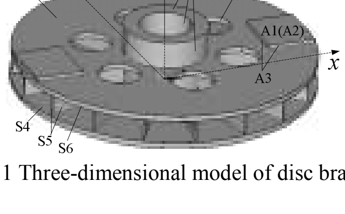

Fig. 1 Three-dimensional model of disc brake

Heat transfer equations for surfaces of brake disc and friction plate are as follows

● In the surface of brake disc (S1) in contact with friction plate, there exist natural heat conduction and input heat flow, whereas there exist air convection and thermal radiation in non-contact surface, then

$$-\lambda_d \frac{\partial T_d}{\partial z} = (1-\zeta)h_d(T_d - T_{sur}) - \zeta q_d + (1-\zeta)\varepsilon\sigma(T_d^4 - T_{sur}^4) - \zeta\lambda_c(T_P - T_d) \quad (3)$$

where, in contact surface, $\zeta = 1$; in non-contact surface, $\zeta = 0$.

Instantaneous temperatures at corresponding points in contact surface are equal, and then (3) can be changed as

$$-\lambda_d \frac{\partial T_d}{\partial z} = (1-\zeta)h_d(T_d - T_{sur}) + (1-\zeta)\varepsilon\sigma(T_d^4 - T_{sur}^4) - \zeta q_d \quad (4)$$

● In the outer surface of brake disc (S6), the convection hole surface (S3), the surface of rib (S5) and the side surface of friction plate (A3), there exist air convection and thermal radiation, then

$$\lambda_d \frac{\partial T_d}{\partial x} n_x + \lambda_d \frac{\partial T_d}{\partial y} n_y = -h_d(T_d - T_{sur}) - \varepsilon\sigma(T_d^4 - T_{sur}^4) \quad (5)$$

$$\lambda_p \frac{\partial T_p}{\partial x} n_x + \lambda_p \frac{\partial T_p}{\partial y} n_y = -h_p(T_p - T_{sur}) - \varepsilon\sigma(T_d^4 - T_{sur}^4) \quad (6)$$

● In the friction surface of friction plate (A2), there exist natural heat conduction and input heat flux, and then

$$\lambda_p \frac{\partial T_p}{\partial z} = q_p \quad (7)$$

● There exist air convection and thermal radiation in the back surface of brake disc, then

$$\lambda_p \frac{\partial T_p}{\partial z} = -h_d(T_d - T_{sur}) - \varepsilon\sigma(T_d^4 - T_{sur}^4) \quad (8)$$

● Internal torus (S2) is far away from the heat input area and the braking time is short, thus it can be regarded as adiabatic. The back surface of friction plate (A1) is also adiabatic.

In the foregoing equations, h_d , h_p are respectively convection coefficients of brake disc and friction plate, $\mathrm{W/(m^2 \cdot K)}$; λ_d , λ_p are respectively thermal conductivities of brake disc and friction plate, $\mathrm{W/(m \cdot K)}$; T_d , T_p are surface temperatures of brake disc and friction plate, K; $n_i (i = x, y, z)$ is the unit outer normal direction of boundary surface; σ represents for Stefan-Boltzmann constant, $\sigma = 5.67\times10^{-8}\,\mathrm{W/(m^2 \cdot K^4)}$; and ε absorption rate; T_{sur} ambient temperature, K.

III. Temperature Field Simulation

A. Characteristic parameters of materials

The selected disc brake as research object is shown in Fig. 2. Materials for friction pairs are respectively 16Mn and environment friendly non-asbestos. Their characteristic parameters are shown in Table I.

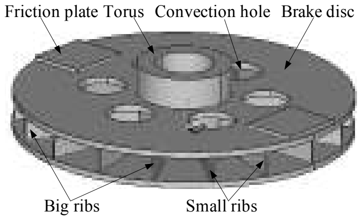

Fig. 2 Disc brake device

TABLE 1
CHARACTERISTIC PARAMETERS OF MATERIALS FOR FRICTION PAIRS [9]

Characteristic parameters	Symbol	Brake disc	Friction plate
Thermal conductivity ($\mathrm{W/(m \cdot K)}$)	λ	53.2	0.295
Density ($\mathrm{kg/m^3}$)	ρ	7866	2206
Specific heat capacity ($\mathrm{J/(kg \cdot K)}$)	c	473	2530
Thermal expansion coefficient ($\mathrm{K^{-1}}$)	α	11×10^{-6}	30×10^{-6}
Elastic modulus (GPa)	E	165	1.5
Poisson's ratio	μ_0	0.28	0.25

B. Simplification and meshing

The structure and loading mode of friction pairs are symmetric, and half disc is taken for analysis. The torus in is far away from the friction area, and therefore it can be omitted in the modeling process.

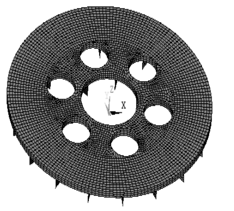

Fig. 3 Finite element model

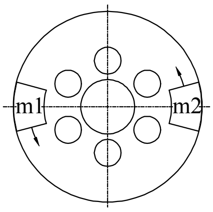

Fig. 4 Simulation principle

Fig. 3 shows the finite element model of brake disc. In the modeling process of temperature field, three-dimensional thermal solid element SOLID70 was selected to create the brake disc, and hot shell element SHELL57 for the establishment of the ribs.

C. Infliction of thermal parameters

The simulation principle of temperature field is shown in Fig. 4. Moving heat source was applied during the simulation process. Heat flux was imposed on instantaneous contact region; while on non-contact area thermal radiation coefficient and convection coefficient were exerted. All these thermal parameters were input by the command program of ANSYS software up to the brake end.

In full contact state, the contact surface is a rectangular area. However, it is difficult to select an ever-changing rectangular area in ANSYS software, thus the contact area was assumed to be a fan-shaped region with an angle of 30° , just as areas m1 and m2 shown in Fig. 4.

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The formula for the rotational speed of brake disc is

$$\omega(t) = v(t) / r_0 \tag{9}$$

where, $v(t)$ is belt speed, m/s; r_0 is drive roller radius, m.

The pressure formula of friction plate on brake disc is

$$p(t) = \frac{M(t)}{n \cdot A_m \cdot r_0} \tag{10}$$

where, n is the number of disc brake; $M(t)$ stands for the braking torque, and A_m the effective friction area.

In the simulation procedure, the rotating angle of two friction plates for each movement is $\pi / 90\text{rad}$; Thus, the rotating time of heat source for each movement is $\pi / 90 / \omega(t)$.

IV. Surface Temperature Field Analysis

The braking torque is 85500 N·m, the initial rotational speed of brake disc is 67r/min and the braking time is 28s.

Fig. 5 is the surface temperature fields of brake disc at different times. In Fig. 5, the dotted lines indicate positions of friction plate and arrows denote the moving direction of heat source. The friction plate rotates counterclockwise with respect to brake disc.

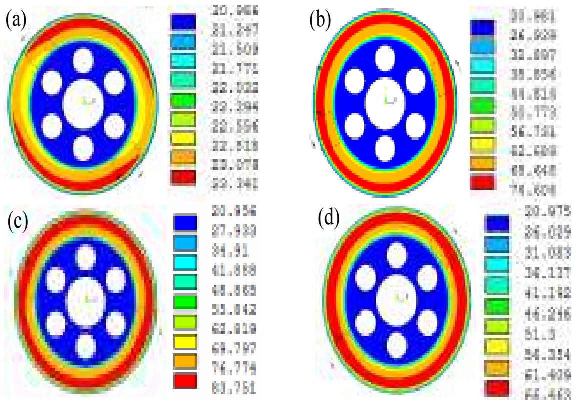

Fig. 5 Surface temperature fields of brake disc at different time
(a) t=1s; (b) t=10s; (c) t=20s; (d) t=27.937s

In braking process, the temperature field changes with the input heat flux and heat exchange conditions. The input heat flux is mainly relevant to friction coefficient, pressure on brake disc, the angular velocity of brake disc and friction radius; while heat exchange is connected with the physical properties of friction pair materials and external environmental factors.

During braking, the surface temperature field of brake disc gradually approaches axial symmetric figure, which due to the comprehensive effect of heat generation and refrigeration. At the beginning, the heat dissipation of surface temperature to the internal body and the air is slight, and the surface temperature field is mainly concerned with the input heat flux. In this process, the input of heat flux and the heat dissipation of surface temperature are carried out simultaneously; the input heat flux gradually reduces while heat conduction tends to dominate, so the temperature field goes close to axial symmetric distribution before the brake end.

Fig. 6 shows temperature field distributions of friction surface, the back surface and the axial section of brake disc at the brake end. All dimensions in Fig. 6 are given in millimeter. The region from z=0mm to z=60mm represents the axial position of ribs, while from z=60mm to z=90mm, it stands for axial position of brake disc; the region from r=224mm to r=540mm stands for non-friction area and r=540mm to r=800mm for friction area on brake disc.

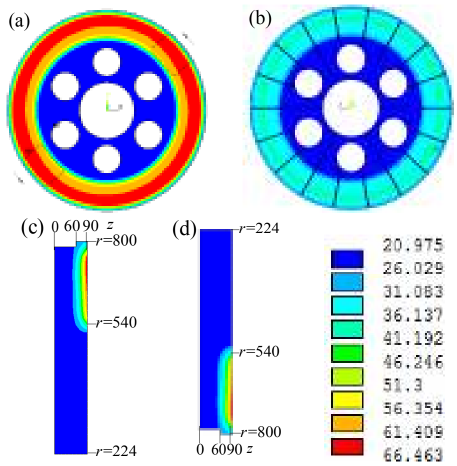

Fig. 6 Temperature field distributions of brake disc at the brake end
(a) Friction surface (b) Back surface of brake disc
(c) Axial section at $\theta = 0^\circ$ (d) Axial section at $\theta = 180^\circ$

In Fig. 6(b), the temperature field of the back surface of brake disc is not continuous, which shows massive shape with almost equal size. The position between each massive shape is just the vertical position of each rib. This phenomenon is more clearly at the brake end, which is mainly due to that the brake disc is an excellent heat conductor and the refrigeration pattern is heat conduction; whereas in non-contact area, the main form is thermal convection, the heat conducted to ribs is greater than that convected to air.

In Fig. 6(c) and Fig. 6(d), the heat on friction surface conducts to radial and axial directions of brake disc; however, as the braking time is very short, the influence of heat generated by friction on both radial and axial directions of brake disc is slight.

V. Time History Analysis of Temperature Field

Fig. 7(a) shows the temperature vibration with time at different axial coordinates (z=30mm, z=50mm, z=70mm, z=90mm) at the friction radius r=675mm and circumferential angle $\theta = 0^\circ$.

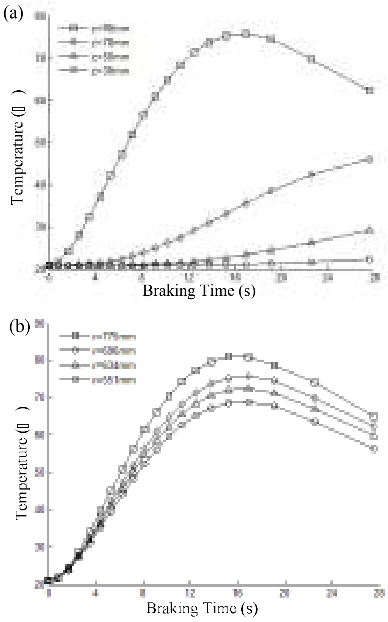

Fig. 7 Time history curves of temperature field
(a) For different axial coordinates
(b) For different radial coordinates

It can be seen from Fig. 7(a), the generated friction heat is much higher than that transferred to the internal body. It makes the surface temperature higher than the internal temperature. Therefore, there is a lager temperature gradient in axial direction, which induces the heat to transmit inside constantly. In z=70mm, there exists a significant temperature rise at about 6s, while at z=50mm the time for temperature rise is at about 15s. It indicates that in the heat conduction process, the farther it is away from the friction zone, the longer it is affected and the less heat it attains.

Fig. 7(b) shows the temperature vibration with time at different friction radiuses (r=551mm, r=634mm, r=696mm, r=775mm) on the friction surface with circumferential angle $\theta = 0^{\circ}$.

It can be concluded from Fig. 7(b) that:

● The larger the friction radius is, the higher the temperature is, which is mainly due to that the input heat flux is proportional to the friction radius.

● In the braking procedure, the surface temperature has gone through the process of temperature rise to temperature drop. At the initial braking stage, from 0s to 10s, the brake torque increased rapidly, the heat flux input to the friction surface is great. At this time, the surface temperature is low and the temperature gradient is relatively small, which results in little heat dissipation, and thus the surface temperature increases greatly; At the mid-braking stage, the input heat flux continuously reduces while the axial temperature gradient becomes large, the generated friction heat is lost by conduction, and thus the surface temperature rose slowly and reaches the maximum value of $86^{\circ}C$; At the later stage, at about 17s, the effect of heat dissipation is greater than that of input heat flux, the surface temperature gradually decreases.

VI. Conclusion

● In the whole braking process, the surface temperature undergoes a dynamic process of first increases and then falls, the maximum temperature occurs at a time before the brake end, not appears at the end time.

● There exist large temperature gradients in both radial and axial directions, while it is relatively small in circumferential direction, and the radial temperature increases with the friction radius.

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