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THE TEMPERATURE DISTRIBUTION IN THE WHEEL/BRAKE SHOES CONTACT

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Abstract

An analytical solution for the temperature in wheel-rail and wheel-brake shoe was obtained. The friction heat sources appear simultaneously. The steady-state temperature distribution in a wheel of locomotive or wagon undergoing uniform heating in rail and four brake shoes and nonuniform cooling is presented. The conduction of heat from the sliding contact zones in the axial direction of the brake shoes and the radial direction of the wheel, as well as the heat loss by convection from the sides and periphery of the disc, are taken into account. The analysis shows that the temperature friction can be well evaluated by knowledge of the rolling heat for wheel/rail contact and sliding wheel/brake shoe contacts.

Key words: Wheel/brake shoe contact; Temperature friction; Heat partition.

1. INTRODUCTION

The thermal aspect around the contact region can be studied by examining the heat transfer between stationary break shoe and a moving body (relative to the heat source). The slip between wheel and braking system causes friction heating of both bodies. When a brake is working, the transformation of kinetic energy of moving masses into thermal energy takes place.

This kinetic energy is dissipated between two bodies and appreciably raises their temperature at the area of the sliding contact. Two aspects are special importance in this analysis - the nature and distribution of heat partition into each body at the interface and the resultant temperature fields in the two bodies both at the interface and with respect to dept

While the time for reaching the quasi-steady-state conditions can be very short for a moving body, it can be relatively long for a stationary body.

Thus, it may require a long time to arrive at steady-state conditions for sliding. In this case, the heat partition fractions for the two bodies may also vary for a long time before reaching steady-state value.

Many analytical investigations have been conducted in the past for prediction of the flash temperature and the heat partition [1,2,3].

While railway wheels are heating by friction in the contact patch and in the contact brakes, there is also heat loss due to conduction through the contact patch into the rail and into brake shoe. A literature survey revealed that considerable attention has been devoted to the cooling and heating of rolling elements [4,5,6].

In this paper, an analytical solution is proposed, for the temperature distribution in the brake shoes, when wheel is heating by rolling in rail contact and subject to four surfaces heating by sliding in brake shoes.

2. THE FRICTION HEAT TRANSFER MODEL

The wheel is considered that a short rotating cylinder. This wheel, subjected to heating by rolling and sliding friction and nonuniform convective cooling at arbitrary angles (β_c) along the circumference, is shown in Fig. 1.

The relative positions of the rail and the brake shoes will be analyzed by the angular location. Heating is assumed to be provided by means of rolling friction in the angular (γ_a) and means of sliding friction in the four intervals γ_{21}, γ_{22}, γ_{31} and γ_{32} (Fig. 1).

External cooling is assumed to be provided by means of airflow convective heat transfer (convection coefficient h) in the interval (C_1, C_2) and (C_3, C_4) and by forced convection heat transfer (convection coefficient $h+h_o$) in the interval (C_6, C_7), (C_8, C_9) and (C_{10}, 0).

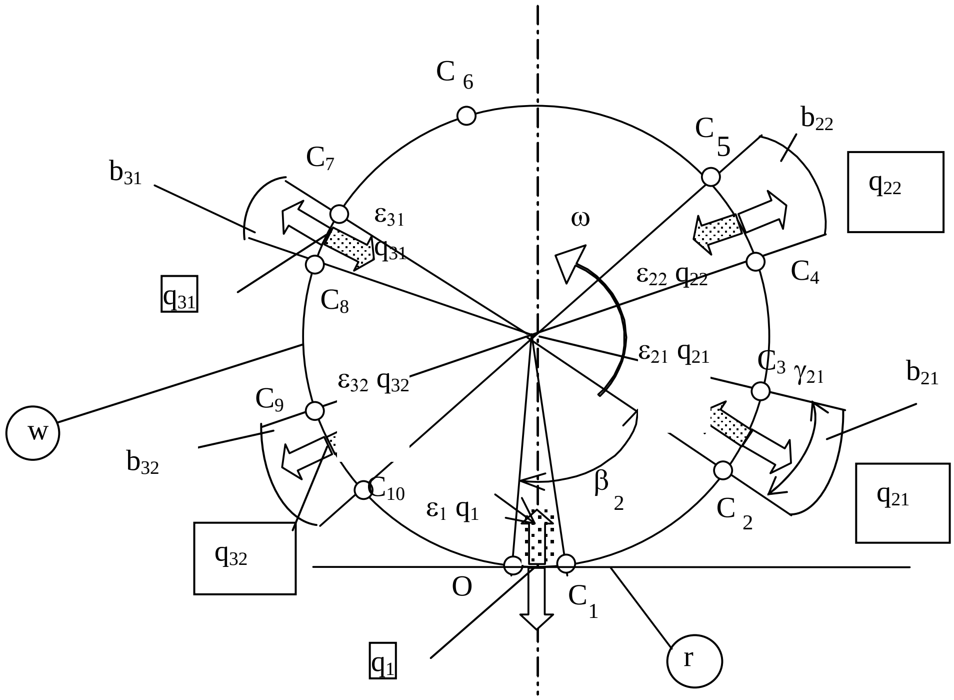


Fig. 1.

We seek to determine the cyclically steady-state temperature distribution in the wheel and brake shoes based on the following assumptions:

1. The temperature of wheel varies along its radius and circumference; the wheel is considered as the disc with constant radius and thickness.

2. The heat conduction in the axial direction is neglected, because the thickness of disc is small compared with its radius.

3. The thermal properties of the wheel and the brake shoes are independent of temperature and remained uniform.

4. The heat loss by convection occurs from the exposed periphery and from the sides of the wheel; there is also heat loss by convection from side of the brake shoes.

5. The entire surface of the brake shoes is in contact with the wheel.

6. The convention coefficients are constant and Biot's numbers are ≤ 1.

7. The temperature of the wheel is equal to that of the rail in the contact zone.

8. The temperature of the wheel is equal to that of the brake shoes in the contact zone.

9. There is no phase change occurring in the brake shoe or the wheel material at the friction surface.

10. The periodic heating at the wheel periphery causes a steady oscillation in temperature in thin annular part of the wheel.

Adopting a fixed Eulerian reference frame, the steady state heat transfer equation in cylindrical coordinates for the wheel is

$$\frac{\partial^2 T}{\partial r^2} + \frac{1}{r}\frac{\partial T}{\partial r} + \frac{1}{r^2}\frac{\partial^2 T}{\partial p^2} = \frac{h'}{2\lambda\delta}T$$

(1)

where T is the temperature rise and (r, ϕ) the coordinate system fixed to the wheel.

The boundary conditions and the solution are obtained in our papers [3, 4, 5].

3. HEAT SLIDING FRICTION DISTRIBUTION COEFFICIENT IN THE BRAKE SHOE

Considering the heat transfer by conduction along the length of the brake shoe and by convection from the periphery (Fig. 2), the differential equation for the temperature distribution at any axis distance z is given

$$\frac{\partial^2 T}{\partial z^2} - \frac{\alpha_b p_b}{\lambda_b A_b}(T - T_o)$$

(36)

where λ_b is thermal conductivity of the brake shoe material, T the temperature in the brake shoe at any axial distance z, T_o the ambient temperature and, α_b the heat transfer coefficient for the brake shoe, p_b, A_b the perimeter and cross-sectional area of the brake shoe.

Substituting $T_b=T-T_o$ and $m_b=(\alpha_b p_b/\lambda_b A_b)^{1/2}$, the general solution of equation (35) is given by $T_b = Ae^{mbz} + Be^{-mbz}$, A and B are the constants which are defined by boundary conditions.

With the boundary conditions

$$T_b = 0, \quad z = g_b$$

(37)

$$\lambda_b \left.\frac{\partial T_b}{\partial z}\right|_{z=0} = -q_{2b} = -(1-\varepsilon_2)q_2$$

(38)

and $T_b = T_{s2}, \quad z = 0$

(39)

the solution of equation (36) becomes:

$$T_b(z) = \frac{T_s \sinh\{m_b(g_b - z)\}}{\sinh(m_b g_b)}$$

(40)

The sliding frictional power dissipation rate in the contact patch of wheel and brake shoe is proportional to the pressure

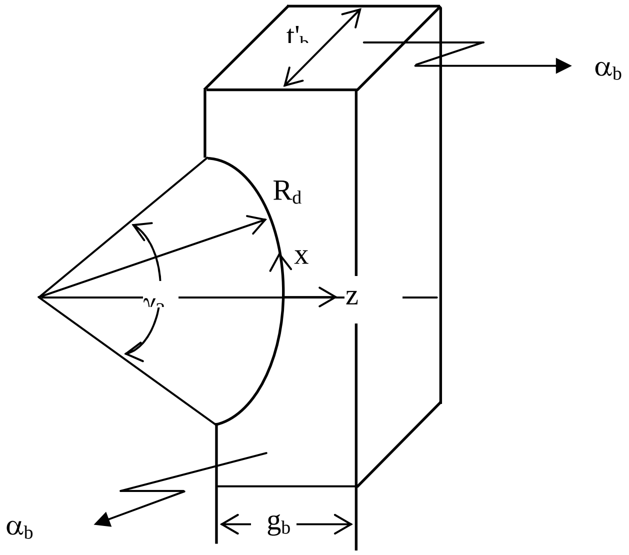


Fig.2.

$$q_2 = \mu_b v_o p_b(x).$$

We assume that friction coefficient (μ_b) and pressure $p_b(x)$ are constants and that all the frictional power dissipation is transformed in heat.

Using equations (40) and (38) can be calculated the surface brake shoe temperature

$$T_{s2} = \frac{(1-\varepsilon_2)q_2}{\lambda_b}\frac{1}{m_b}\tanh(m_b g_b)$$

(41)

Dimensionless temperature of brake shoe surface

$$\theta_{s2} = \frac{T_{s2}\lambda_d}{q_1 R} = (1-\varepsilon_2)\frac{q_2}{q_1}\frac{\lambda_d}{\lambda_b}\frac{1}{m_b R}\tanh(m_b g_b) =$$

$$= (1-\varepsilon_2)\frac{q_{a2}}{q_1}\frac{1}{m_b R}\tanh(m_b g_b) = C_{b2} - \varepsilon_2 C_{b2}$$

(42)

with

$$C_{b2} = \frac{q_{a2}}{\lambda_{a2}}\frac{1}{m_b R}\tanh(m_b g_b) \quad \text{and}$$

$$q_{a2} = \frac{q_1}{q_2}, \quad \lambda_{a2} = \frac{\lambda_b}{\lambda_d}$$

The partition coefficient ε_2 will be calculated with equations (21) and (42)

$$\theta\left(1, \beta_2 + \frac{\gamma_2}{2}\right) = C_{b2} - \varepsilon_2 C_{b2} \qquad (43)$$

It is generally assumed that every wheel has four brake shoes. The partition heat coefficient for every brake shoe (ε_{21}, ε_{22}, ε_{31} and ε_{32}) are calculated by conditions that the temperature for the wheel and rail, left brake shoe and right brake shoe are respectively equal [3,4].

4. SURFACE BRAKE - SHOES TEMPERATURE

The surface brake–shoes dimensionless temperature can be obtained by numerical solution. Thus, for example, the Figures 3, 4, 5 and 6 show the dimensionless temperature for the b21, b22, b31, b32 brake- shoes (UIC 542) respectively.

The curve 1 is obtained for following parameters: - relative heat flux: $q_{a21} = q_{a22} = q_{a31} = q_{a32} = 0.4$;

- convection heat transfer coefficient ,

$h = 100$ Wm^{-2} ^{o}C^{-1};

The curve 2 is obtained for following parameters: - relative heat flux: $q_{a21} = q_{a22} = q_{a31} = q_{a32} = 0.2$,

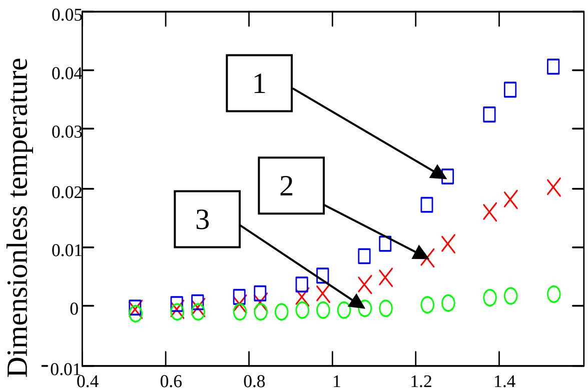
- convection heat transfer coefficient,

$h = 100$ Wm^{-2} ^{o}C^{-1};

The curve 3 is obtained for following parameters: - relative heat flux: $q_{a21} = q_{a22} = 0.024$ and $q_{a31} = q_{a32} = 0.4$ (inequal efficiency of brale-shoes);

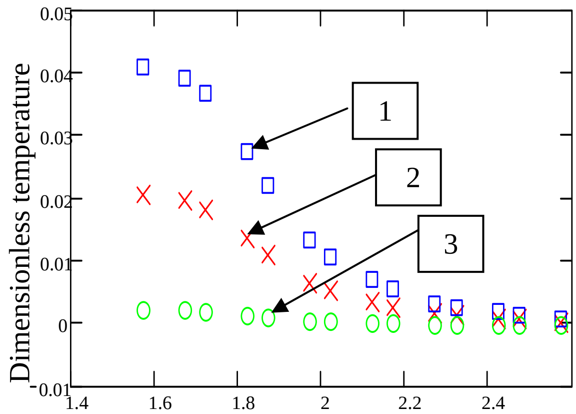
- convection heat transfer coefficient,

$h = 100$ Wm^{-2} ^{o}C^{-1}.



Angular position of b21 brake- shoe

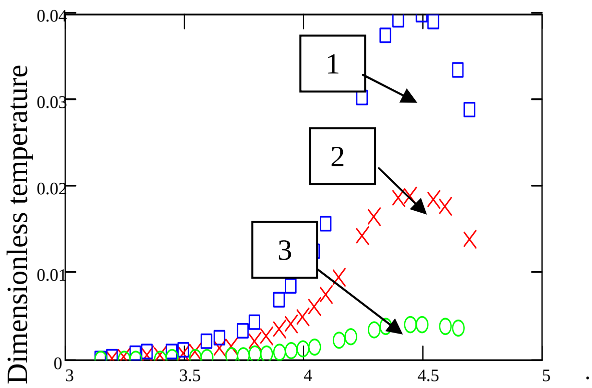
Fig.3.



Angular position of b22 brake- shoe

Fig. 4.

The maximum temperatures predicted for the rail and left and right brake shoes are summarized in Table 1. These temperatures are calculated by the normal conditions for wheel/rail and brake/shoe contact [6,7] ($q_1 = 5\text{x}10^6$ W/m^2, $q_{a21} = q_{a22} = q_{a31} = q_{a32} = 2\text{x}10^6$ W/m^2, $h_o = 100$ W/m^{2}K for high speed, and $h_o = 20$ W/m^{2}K for low speed).



Angular position of b31brake- shoe

Fig.5.

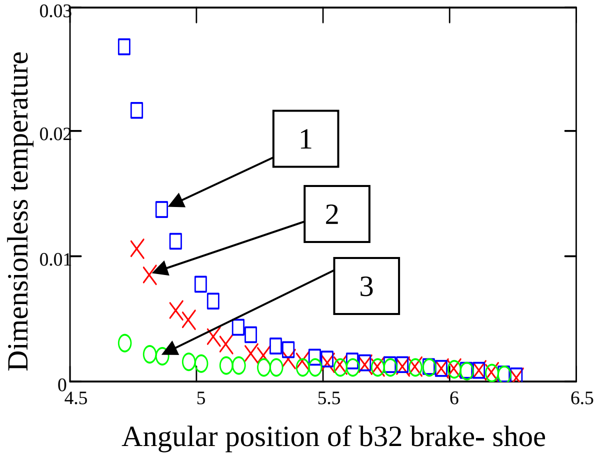


Fig.6.

5. CONCLUSIONS

An analytical solution is presented for the temperature distribution of brake–shoes in contact with rotating wheel (cylinder).

The maximum temperature occurs towards nearly exit of heating contact for the left or right brake shoe.

It is shown that the maximum dimensionless contact temperatures of wheel and brake-shoes are induced in the brake shoe heating contact.

When the brake shoes haven't similary efficiency (inequal friction), the temperature field of the all brake-shoes is modified.

Table 1.

Temperature [°C]	Low Speed (30 m/s)	High Speed (90 m/s)
Maximum		
-Left b31 brake shoe	327	675
-Left b32 brake shoe	363	720
-Right b21 brake shoe	87	183
-Right b22 brake shoe	293	640

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7. NOMENCLATURE

A_b = cross- sectional area of brake shoe, m^2
Bi = Biot number = hR/κ, dimensionless
Bi_o = Biot number of in both supplementary cooling zones = h_oR/k
Bi_s = Biot number of in lateral cooling zones = $h'R^2/(2\delta k)$
g_b = brake shoe thickness, m
h = convection heat transfer coefficient in main frontal zone, $Wm^{-2}\,{}^oC^{-1}$
h_o = convection heat transfer coefficient in supplementary cooling, $Wm^{-2}\,{}^oC^{-1}$
h' = convection heat transfer coefficient in lateral zones, $Wm^{-2}\,{}^oC^{-1}$
k = thermal conductivity, $Wm^{-1}\,{}^oC^{-1}$
P_b = brake shoe perimeter, m
Pe =Peclet number = $a_H v/(2\kappa)$, dimensionless
$q_{1,\,21,22,\,31,32}$ = heat source of strength 1, 2, and 3, Wm^{-2}
$q_{a\ 21,\ 22,\ 31,\ 32}$ = relative heat source of strength 21, 22, 31 and 32 = $q_{21,22,31,32}/q_1$, dimensionless
r = radial coordinate

R = mean radius, m
t = time, s
t'_b = width of brake shoe, m
T = temperature rise in excess of ambient temperature, ^oC
T_w = surface wheel temperature, ^oC
z = normal to surface coordinate dimensionless
β = polar coordinate for heat source or cooling zone rad
β_c = polar coordinate for beginning supplementary cooling zone (C_6 point, Fig.1), rad
$\gamma_{1, 21,22, 31,32}$ = angular length of heat source 1, 2 respectively 3, rad
δ = half of wheel length, m
$\varepsilon_{1, 21,22\ 31,32}$ = heat partitioning factor in regions 1,2, and 3
κ = thermal diffusivity,$m^2\ s^{-1}$
μ = friction coefficient
ψ = angular location, rad
θ = dimensionless temperature = $kT/(q_1R)$
θ_w = dimensionless surface wheel temperature = $kT_w/(q_1R)$