A. TUDOR, C. RADULESCU, I. PETRE

RESEARCH

Thermal Effect of the Brake Shoes Friction on the Wheel /Rail Contact

An analytical solution for the temperature distribution in wheel/ brake shoe contact is presented. The wheel of locomotive or wagon is simultaneously heated by the friction due to wheel/ contact and two brake shoe contacts.The analysis was applied to calculate the partition heat coefficients between wheel and rail and between wheel and left or right brake shoe.

These coefficients can be used to calculate the temperatures for wheel/rail contact wheel/ right brake shoe and wheel/left brake shoe. 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 are induced in the brake shoe heating contact. Since the thermal penetration depth is very small, thermally induced plastic deformations for wheel are restricted to a very thin surface layer. When the brake shoes haven't similarly efficiency (unequal friction), the temperature field of the wheel is modified and the temperature of the best efficiently brake shoe increases.

Keywords: *Wheel rail contact; Brake shoe friction; Frictional heating; Temperature*

1. INTRODUCTION

The frictional heat is generated when two bodies slide against each other with a relative speed and a contact pressure. This frictional heat is generated at sliding or rolling interface of wheel/brake shoe contact and wheel/rail contact.

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 of 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.

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.

Using a quasi-steady-state approach, Ling (1970) expressed the temperature solution for a cylinder subject to cooling and heating. Patula (1981) provided the solution of the temperature for cylinder subject to cooled and heated on parts of its surface but insulated on the rest Ulyse and Khonsari (1993) used an analytical solution for the temperature distribution in a cylinder subject to surface heating and no uniform cooling Ertz and Knothe (2002) analyzed a comparison of analytical and numerical methods for the calculation of temperatures in wheel/rail contact. The friction heat distribution between a stationary pin and a rotating disc was studied by Yevtushenko et. al. (1996) and Kar and Bahadur (1981). Tudor and Radulescu (2003) analyzed the heat friction partition coefficients of wheel contact with rail and two brake shoes.

A. TUDOR, *Professor, "Politehnica" University of Bucharest, Romania*

C. RADULESCU, *Ph.D. Student, Wagon Depot of Craiova, Romania*

I. PETRE, *Assist. Professor, "Valahia" University of Targoviste, Romania*

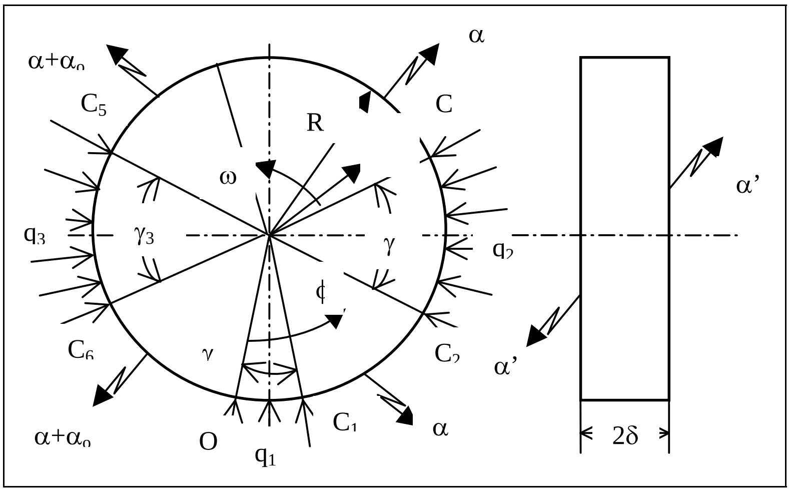

Figure 1. Geometry and boundary conditions

In this paper, an analytical solution for the friction heat temperature distribution in a wheel subject to surface heating by rolling in rail contact and subject to two surfaces heating by sliding in brake shoes is presented. This also nonuniforming cooling is shown. The friction temperature distribution is obtained by a Fourier transform technique.

2. THE FRICTION HEAT TRANSFER MODEL OF WHEEL

The wheel is considered that a short rotating cylinder. This wheel, subjected to heating by rolling and sliding friction and no uniform convective cooling at arbitrary angles (β_c) along the circumference, is shown in Fig. 1. In this paper, we used the model presented in [9].

The relative positions of the rail and the bra-ke shoes will be analyzed by the angular location. Heating is assumed be provided by means of rol-ling friction in the angular (γ_a) and means of sli-ding friction in the two intervals γ_2 and γ_3 (Fig. 1).

External cooling is assumed to be provided by means of airflow convective heat transfer (convection coefficient α) in the interval (C_1, C_2) and (C_3, C_4) and by forced convection heat transfer (convection coefficient $\alpha+\alpha_o$) in the interval (C_4, C_5) and (C_6, 0) (fig. 1).

It is used the following dimensionless variables

$$\theta = \frac{\lambda T}{q_1 R}; \quad \rho = \frac{r}{R}; \quad \rho_1 = \frac{R_1}{R}; B_i = \frac{\alpha R}{\lambda};$$

$$B_{io} = \frac{\alpha_o R}{\lambda}; \quad B_{is} = \frac{\alpha' R^2}{\lambda \cdot 2 \cdot \delta}; \quad q_{a2} = \frac{q_2}{q_1}; \quad q_3 = \frac{q_3}{q_1}$$

The heat conduction equation in dimensionless variables is

$$\frac{\partial^2 \theta}{\partial \rho^2} + \frac{1}{\rho}\frac{\partial \theta}{\partial \rho} + \frac{1}{\rho^2}\frac{\partial^2 \theta}{\partial \phi^2} = B_{is}\theta \qquad (1)$$

The solution of differential equation to accepct the boundary conditions will be writing as [9]:

$$\theta(\rho, \psi) = \theta_{o1}\varepsilon_1 + \theta_{o2}\varepsilon_2 + \theta_{o3}\varepsilon_3 \qquad (2)$$

The dimensionless parameter θ_{o1}, θ_{o2} and θ_{o3} are defined in [9]. In order to calculate the temperature rise of the disc with the dimensionless radius $\rho = \rho_1$ and $\rho = 1$ using equation (1), the heat distribution coefficients (ε_1, ε_2, ε_3) are evaluated by Tudor and Radulescu [9].

3. HEAT ROLLING FRICTION TEMPERATURE

The heat rolling friction distribution coefficient (ε1) is the ratio of the amount of heat energy flowing into the wheel, as a disc to the total heat generated at the rolling interface wheel/rail contact.

When wheel and rail are brought into contact under the action of the static wheel load, the area of contact and the pressure distribution are usually calculated with the Hertz's theory. If a tangential force T is transmitted between wheel and rail, there is always a mean relative velocity in the contact point. Sliding occurs within the whole contact area and the tangential force is T = μNw (Nw – wheel load).

The contact patch moves with respect to the wheel surface and the friction heating within the contact patch is a time dependent heat source (Fig. 2).

With the typical values for wheel/rail contact, aH ≅ 5 mm, ar = 14,2·10-6 m2/s and vo=26 m/s, one gets L = 4584. In this case, the longitudinal and lateral heat conduction (x – and y – direction), can, therefore, be neglected and heat conduction equation is [8]

$$a_r \frac{\partial^2 T_r}{\partial z^2} = \frac{\partial T_r}{\partial t} \tag{3}$$

Thus, T_r represents the temperature rail rise due to the heat supply within the contact patch.

Since the heat flew is one-dimensional, this problem is similar to a semi-infinite solid with an arbitrarily distributed heat source $q_1(t)$ applied to the surface z = 0 at $t \geq 0$. The solution $T_r(z,t)$ has to fulfil the differential equation (3), the initial condition

$$T_r(z, t = 0) = 0 \tag{4}$$

and the boundary condition

$$-\lambda_r \frac{\partial T_r}{\partial z}(z = 0, t) = q_{1r}(t) = (1 - \varepsilon_1) q_1(t) \tag{5}$$

The solution of this problem can be found in the book of Carslaw and Jaeger,

$$T_r(z,t) = \frac{\sqrt{a_r}}{\lambda_r \sqrt{\pi}} \int_0^t q_{1r}(t - t') \exp\left(-\frac{z^2}{4 a_r t'}\right) \frac{dt'}{\sqrt{t'}} \tag{6}$$

The points on the surface of wheel and rail pass through the contact patch at different speeds due to the sliding velocity.

The largest heat flux and the highest temperatures recur the major hertzian axis which is parallel to the rolling direction at y=0. It is meaningful to substitute the time t elapsed since entering the contact patch with the current position x in a coordinate system fixed to the contact patch (Fig. 2)

$$x = vt - a_H \tag{7}$$

With the dimensionless coordinates

$$x_a = \frac{x}{a_H} \quad \text{and} \quad z_a = \frac{z}{z_\delta},$$

the temperature of rail (T_r) can be calculated as

$$T_r(z_a, x_a) = \frac{\sqrt{a_r}}{\lambda_r} \sqrt{\frac{a_H}{\pi v_o}} \int_{-1}^{x_a} q_{1r}(x_a') \exp\left(-\frac{dx_a'}{2(x_a - x_a')}\right) \frac{dx_a'}{\sqrt{x_a - x_a'}} \tag{8}$$

The analytical solution of the integral in equation (8) is quite simple if we assume a constant heat flow rate q_{1r} at the rail surface within the contact patch.

The frictional power dissipation rate in rolling contact patch is proportional to the pressures, the coefficient of friction μ and the sliding velocity as can be considered constant values:

$$q_1(x_a) = \mu v_s p_z(x_a) = \mu v_s p_o \sqrt{1 - x_a^2} \tag{9}$$

The average heat flow at the surface

$$q_1 = \frac{1}{2} \int_{-1}^{1} q_1(x_a) dx_a = \frac{\pi}{4} \mu v_s p_o \tag{10}$$

It is generally assumed that the frictional power dissipation is transformed in heat. With the heat partition factor ε_1, this can be written as

$$q_{1r}(x_a) = (1 - \varepsilon_1) q_1(x_a) \quad \text{for rail}$$

$$\text{and} \quad q_{1w}(x_a) = \varepsilon_1 q_1(x_a) \quad \text{for wheel} \tag{11}$$

The maximum temperature occurs at the trailing edge of the contact patch.

For this case, dimensionless temperature of disc surface is

$$\theta_r = \frac{T_r(0,1)\lambda_d}{qR} = C_r - \varepsilon_1 C_r \tag{12}$$

with $C_r = \sqrt{\frac{8 a_H a_r}{\pi R^2 v_s}} \frac{\lambda_d}{\lambda_r}$.

The part of the rolling frictional heating (ε_1) that flows into wheel can be defined with equations (2) and (12):

$$\theta(1, \beta_1) = C_r - \varepsilon_1 C_r \tag{13}$$

The angle $\beta_1 = \gamma_1 = 2 \operatorname{asin}(a_H/R)$.

For example, the normal conditions of wheel/rail contact are wheel load Fn= 100 kN, wheel radius R= 0,5 m, vehicle speed vo= 26 m/s, sliding velocity (longitudinal), vs= 1 m/s, semi-axis of the contact ellipse in rolling direction a_H = 5,88 mm. In this case $\beta_1 = \gamma_1$ = 0,024 rad.

4. HEAT SLIDING FRICTION TEMPERATURE DISTRIBUTION

Considering the heat transfer by condition along the length of the brake shoe and by convection from the periphery (Fig. 3), the differential equation for the temperature distribu-tion at any axis distance z is given

$$\frac{\partial^2 T}{\partial z^2} - \frac{\alpha_b p_b}{\lambda_b A_b}(T - T_o) \qquad (14)$$

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 (14) is given by $T_b = Ae^{m_b z} + Be^{-m_b z}$, A and B are the constants which are defined by boundary conditions.

With the boundary conditions

$$T_b = 0, \quad z = g_b \qquad (15)$$

$$\lambda_b \left.\frac{\partial T_b}{\partial z}\right|_{z=0} = -q_{2b} = -(1 - \varepsilon_2) q_2 \qquad (16)$$

and

$$T_b = T_{s2}, \quad z = 0 \qquad (17)$$

the solution of equation (14) becomes:

$$T_b(z) = \frac{T_s \sinh\{m_b (g_b - z)\}}{\sinh(m_b g_b)} \qquad (18)$$

The sliding frictional power dissipation rate in the contact patch of wheel and brake shoe is proportional to the pressure $q_2 = \mu_b v_o p_b(x)$.

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 (18) and (16) can be calculated the surface brake shoe temperature

$$T_{s2} = \frac{(1 - \varepsilon_2) q_2}{\lambda_b} \frac{1}{m} \tanh(m_b g_b) \qquad (41)$$

Dimensionless temperature of brake shoe surface

$$\theta_{s2} = \frac{T_{s2} \lambda_d}{q_1 R} = C_{b2} - \varepsilon_2 C_{b2} \qquad (19)$$

with

$$C_{b2} = \frac{q_{a2}}{\lambda_{a2}} \frac{1}{m_b R} \tanh(m_b g_b) \quad \text{and} \quad q_{a2} = \frac{q_1}{q_2}, \quad \lambda_{a2} = \frac{\lambda_b}{\lambda_d}.$$

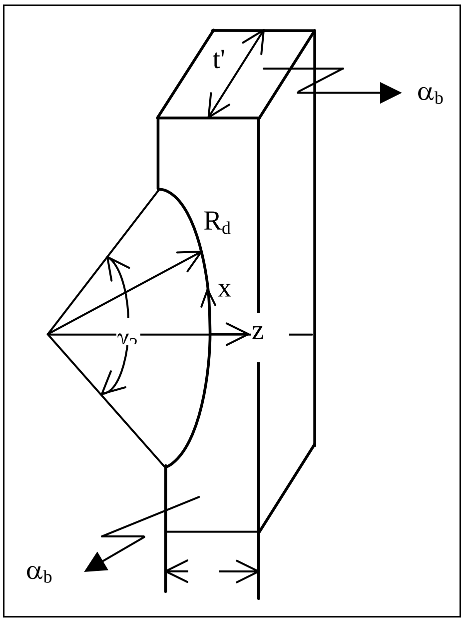

Figure 3. Co-ordinate system for temperature calculation in wheel/brake shoe contact

The partition coefficient ε_2 will be calculated with equations (19) and (2)

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

It is generally assumed that every wheel has two brake shoes. The relative position is defined by the angles β_2, β_3, γ_2 and γ_3 for the left (b_2) and right brake shoe (b_3) (Fig. 4).

The partition heat coefficient for every brake shoe (ε_2 and ε_3) are calculated by conditions that the temperature for the wheel and rail, left brake shoe and right brake shoe are respectively equal.

5. STEADY-STATE WHEEL TEMPERATURE

The bulk temperature of the wheel increases with time due to continuous frictional heating on its rolling surface.

Therefore, the temperatures of wheel and rail are different when a point on the surface of the wheel comes into in the area of contact again. This gives rise to a considerable heat flow from the hot wheel into the cold rail due to conduction through the contact patch.

Outside the area of contract, frictional heat flows from the wheel into ambient air by convection at the free surfaces.

During the very short time period that every point on the surface is in rolling contact, he thermal penetration depth is very small compared to the size of the contact patch. With the boundary conditions, the wheel it is considered as a central disc and a thin annulus. This thin annulus of the disc is affected by temperature oscillation.

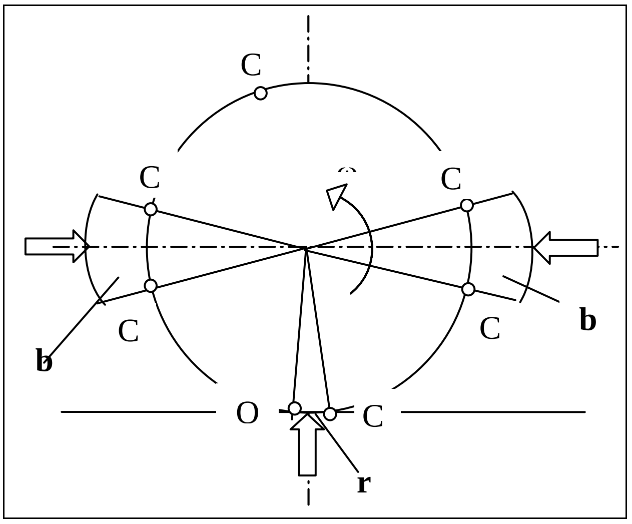

Figure 4. Relative position of heat sources

The dimensionless temperature of the central disc

$$\theta_c = \frac{T_c \lambda_d}{q_1 R} =$$

$$\frac{\varepsilon_1\gamma_1 + \varepsilon_2\gamma_2 q_{a2} + \varepsilon_3\gamma_3 q_{a3}}{(\beta_c - \gamma_1 - \gamma_2)\alpha + (2\pi - \beta_c - \gamma_3)\alpha_o + \pi\frac{R}{\delta}\alpha'} \frac{\lambda_d}{R} \tag{21}$$

6. RESULTS AND DISCUSSION

The dimensionless wheel surface tempe-rature $\theta(1,\psi)$ as the angular location is shown in the fig. 5 for different values of Biot's number (0.2,1,5).

The local temperature rise around the heat sources decreases with increase of the parameter B_{io}.

On notices that the maximum temperature in Fig. 5, when the two brake shoes have equals friction efficiently, is between wheel and the left brake shoes. This can be explained by examining the friction thermal history of a material element. A point uniformly heated by the heat sources increases its temperature significantly to reach a maximum at the end of the heating wits.

The cooling and heating less affect points inside the wheel. For this cyclically steady-state problem, due to the length of time needed for the heat to be conducted into the inner layers, the maximum temperatures at interior locations are always shifted in the direction of rotation.

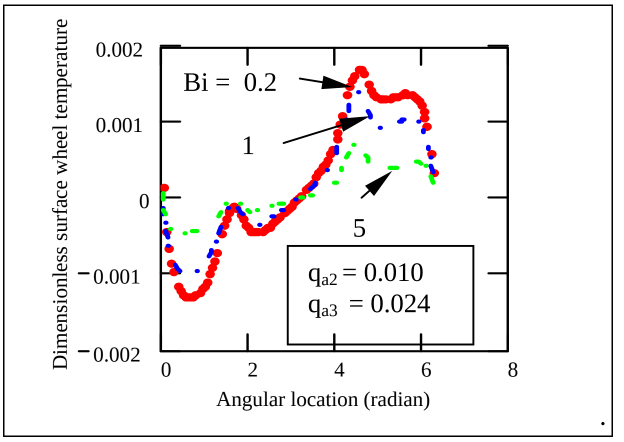

Figure 5. Surface wheel temperature

This point is then cooled as the results of convective cooling as well as circumferential and radial heat conduction.

Figure 6 presents the temperature profile for the case where the brake shoes haven't similarly efficiency. The negative values of the dimensi-onless temperatures show that cooling by convec-tion is higher than the rolling friction heating.

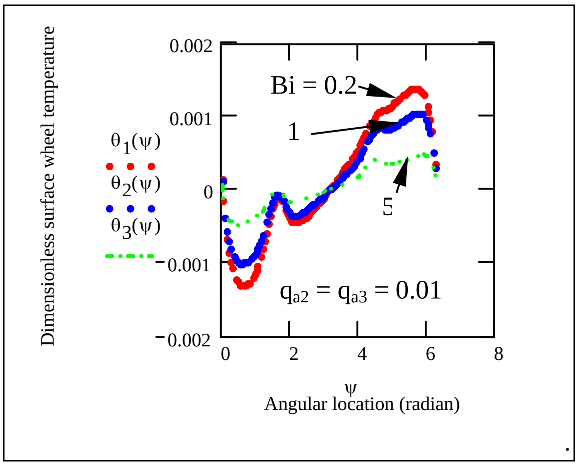

Figure 6. Surface wheel temperature

The data in Table 1 are usual operating conditions of European locomotives and wagons at the low and high speed or carriages at the low speed [8]. The maximum temperatures are evaluated for these conditions.

7. CONCLUSIONS

An analytical solution is presented for the temperature distribution of rotating wheel (cylinder) subjected to no uniform cooling and three uniform friction heating.

Table 1.

	Units	*Low speed*	*High speed*
Normal load [8]	F_n (kN)	100	100
Vehicle speed [8]	v_o (m/s)	30	90
Sliding velocity (longitudinal) [8]	v_s (m/s)	1	3
Coeficient of friction -in wheel / rail contact -in brake shoe	μ	 0.3 0.15	 0.1 0.1
Normal load on brake shoe	F_{bs} (N)	2230	2230
Frictional rolling power dissipation [8]	$P_{friction}$ (kW)	30	30
Frictional sliding power in brake shoe	P_{fb} (kW)	10	20
Geometry of brake shoe	- transversal section area (m^2) - perimeter (m) - wheel contact area (m^2)	0.0048 0.656 0.02	0.0048 0.656 0.02
Maximum temperature for: -left brake shoe -right brake shoe -rolling rail contact	^oC	 445 383 158	 703 656 271

The analysis was applied to calculate the partition heat coefficients between wheel and rail and between wheel and left or right brake shoe.

These coefficients can be used to calculate the temperatures for wheel/rail contact wheel/ right brake shoe and wheel/left brake shoe.

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 are induced in the brake shoe heating contact. Since the thermal penetration depth is very small, thermally induced plastic deformations for wheel are restricted to a very thin surface layer.

When the brake shoes haven't similary efficiency (inequal friction), the temperature field of the wheel is modified and the temperature of the best efficiently brake shoe increases.

REFERENCES

[1] M.K. Kar and S. Bahadur, Heat Transfer Analysis for a Pin-and-Disc Sliding System, WEAR, Vol. 67(1), 1981, pp. 71.

[2] M. El-Sherbing and T.P. Newcomb, The Temperature Distribution due to Frictional Heat Generated between a Stationary Cylinder and a Rotating Cylinder, WEAR, Vol. 42(1), 1977, pp. 23.

[3] F.F. Ling, Surface Mechanics, Wiley Interscience, New York, 1970.

[4] E.J. Patula, Steady-State Temperature Distribution in a Rotating Roll Subject to Surface Heat Fluxes and Convective Cooling, J. Heat Transfer, Vol. 103, 1981, pp. 36-41.

[5] P. Ulyse and M.M. Khonsari, Thermal Response of Rolling Components under Mixed Boundary Conditions: An Analytical Approach, J. Heat Transfer, Vol. 115, 1993, pp. 857-865.

[6] Yeotushenko, O. Ukhanska, R. Chapovska, Friction Heat Distribution between a Stationary Pin and a Rotating Disc, WEAR, Vol. 196, 1996, pp. 219.

[7] K. Knothe and S. Liebelt, Determination of Temperatures for Sliding Contact with Applications for Wheel-Rail Systems, WEAR, Vol. 189, 1995, pp. 91-99.

[8] M. Ertz and K. Knothe, A Comparasion of Analytical and Numerical Methods for the Calculation of Temperatures in Wheel / Rail Contact, WEAR, Vol. 253, 2002, pp. 498-508.

[9] Tudor and C. Radulescu, Analysis of heat friction partition in wheel/rail contact and whell/brake shoe contact. Univ. „Politehnica" Bucharest, Sc. Bull. Series D. Mech. Eng- in press.