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Three-Dimensional Finite Element Analysis on Thermal Stress in a Brake Drum of Heavy Commercial Truck

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تحليل الإجهادات الحرارية لأسطوانة الفرامل للشاحنات التجارية باستخدام نظرية العناصر المحدودة

الملخص

يولد استخدام الفرامل في الشاحنة التجارية الثقيلة عند السرعات العالية حرارة مفرطة على السطح عند الاحتكاك مع اسطوانة الفرامل. قد تسبب هذه الحالة آثار غير مرغوب فيها على اسطوانة الفرامل التي تؤدي في نهاية المطاف إلى بداية ظهور في الشقوق الحرارية. ومن أجل تصميم أمثل لأسطوانة الفرامل تكمن أهمية فحص الإجهاد الحراري نتيجة لدرجات الحرارة المفرطة الناتجة عن احتكاك سطح الأسطوانة بالفرامل عند الكبح. تقوم هذه الدراسة بتحليل الإجهاد الحراري على اسطوانة الفرامل للشاحنات التجارية الثقيلة باستخدام نظرية العناصر محدودة ثلاثي الأبعاد كمرحلة الأولى وذلك لتحليل الصدع الناتج عن الحرارة المفرطة نتيجة الاحتكاك على اسطوانة الفرامل. تقوم الدراسة على تحليل تأثير اثنين من دواسات الفرامل على اسطوانة الفرامل، ونتيجة التسخين والتبريد الذي يحدث تعاقبياً على اسطوانة الفرامل أثناء استعمال الفرامل. تم تطبيق الدراسة عند على حالة استخدام شدة تأثير الفرامل عند التباطؤ بقيمة تساوي نصف تسارع الجاذبية الأرضية. بينت نتائج الدراسة أن أكبر قيمة مسجلة للحرارة الناتجة عن الاحتكاك عند منتصف المسافة من بداية الكبح. بينما تصل أعلى قيمة للإجهاد الحراري عند ربع وقت حصول الكبح.

Summary

Brake application in a heavy commercial truck running with high speed will generate an excessive thermal environment on the frictional surface of brake drum. This condition may cause undesirable effects on the material of the brake drum that eventually lead to the initiation of heat cracks. Therefore, for optimized design of a brake drum, it appears to be very important to examine the thermal stress due to the excessive temperature in the rubbing surface of the brake drum in the course of braking. This paper reports a 3-D finite element analysis on thermal stress in a brake drum of heavy commercial truck as the first stage of investigating the initiation of heat crack in the brake drum. The braking is done by two brake shoes, and the heating and cooling occur alternatively in the brake drum during the brake application. A severe brake application with deceleration 0.5*g* is taken into account. The history of the transient temperature and thermal stress in the course of braking are plotted and discussed. It is found that the maximum temperature occurs when the braking time is reaching a half-way, whilst the maximum thermal stress occurs when it is reaching at around one-fourth of braking time.

Key-words:

Brake drum, 3-D finite element analysis, transient temperature, thermal stress.

INTRODUCTION

Friction brakes are widely used to provide an inexpensive, consistent and reliable means of retardation, and conveniently, generate large frictional forces to give high rates of deceleration. A severe thermal environment is created at such a friction interface and satisfactory braking performance largely depends on the effective dissipation of heat energy from this interface.

Up to now, the drum cracking of heavy and large commercial vehicle has become a widespread problem caused by thermal damage due to excessively high surface temperature in the course of brake application. The brake drums are prone to overheat on long downgrades because they must be braked more frequently under high speed braking and heavier load.

Nowadays, the predominant trend in the development of large vehicles is toward higher speeds and heavier loads, which increases considerably the energy required to be dissipated by the brake drum, and, consequently, it will generate excessive temperature that produce high thermal stresses. This condition may cause undesirable effects on the material of the brake drum that eventually lead to the initiation of heat cracks.

The investigation of transient temperatures, thermal stresses, and thermal distortions in brake drum have been extensively performed by the brake drum designers and researchers attending to this field. Most thermal analyses assume that the heat generated during braking is uniformly distributed over the nominal area of contact between lining and drum. In practice, manufacturing tolerance and drum distortion give rise to non-uniform contact and the frictional heat is generated over discrete areas forming a band or bands whose total width is less than the total band width of the lining material.

The finite element analysis is a powerful technique for the solution of many engineering problems, and has been extensively used for thermal analysis applied to the brake components. Three-dimensional analysis on temperature distribution in a brake drum using finite element method has been reported by Khairul Fuad et al. 2008. Some other papers analyzing the thermal analyses in brake drum in two dimensional analyses have been published, either in axisymmetric analysis (Masashi Daimaruya, 1977; Khairul Fuad, 1994) or in Cartesian coordinates (Ramachandra Rao et al. 1985 and 1988; Day. A.J, 1985; Ashworth et al. 1977). Ramachandra Rao et al. proposed a clock mechanism as a model to identify the heating and cooling that alternatively occur in the brake drum during braking.

This study reports the three-dimensional numerical analysis of thermal stresses in a brake drum of heavy commercial truck. The braking is done by two brake shoes and the heating and cooling occur alternatively in the brake drum during the brake application. A severe brake application is taken into account for the wheel load of 40 kN, and the braking is applied from a speed of 100 km/h until it stops with a deceleration of 0.5g.

FRICTIONAL HEAT FLUX

A leading and trailing shoe brake model is analyzed in this study. Heating and cooling that occur alternatively in the brake drum during braking is prescribed as a clock mechanism (Fig.1). The first heating occurs when the drum absorbs the energy input, or the frictional energy, for the time interval during which the drum moves from one end of each brake shoe to the other respective end with its varying velocity (say from point 1 to point 2 and from point 3 to point 4 which is depicted in Fig.1). At this period, cooling through convection takes place in the other parts of brake drum. In this analysis, the effect of thermal radiation during the cooling is neglected because it is too small compared to the convection effects. All parts of the brake drum are subjected to cooling at this period. As soon as this time lapses, the appropriate heat energy, the second heating, is given again, noting that the drum speed falls at the rate of $0.5g$ deceleration. Thus, heating and cooling occurs alternatively until the braking is fully terminated.

Investigation of brake thermal problems has so far been limited by a number of simplifying assumptions concerning the distribution of frictional heat generation over the friction interface and the proportions which are transferred to each part of the friction pair. In this analysis, the braking condition is considered under assumption of perfect contact between the brake shoes and the brake drum, and equal amount of generated frictional heat is distributed over the rubbing surface. The brake drum is assumed to absorb 95% of a major share of heat energy and the remainder goes into the brake shoes (Ashworth et al. 1977; Newcomb T.P., 1958-59 and 1961).

Considering about 5% of heat energy going to brake shoes, the heat flux resulting from the transformation of instantaneous kinetic energy can be written as follows

$$q = 0.95 \frac{\Delta E_k}{\Delta t\, A_b} \tag{1}$$

where q is the heat flux generated on the frictional surface, ΔE_k is the alteration of kinetic energy from one edge to the other edge of brake shoe, Δt is the time interval during heating, and A_b is the area of the two brake shoes surface contacted to the drum.

TRANSIENT TEMPERATURE ANALYSIS

The three-dimensional unsteady state heat conduction equation for the transient temperature analysis in the brake drum is governed by the following equation

$$\frac{d^2T}{dx^2} + \frac{d^2T}{dy^2} + \frac{d^2T}{dz^2} = \frac{\rho c}{k} \frac{dT}{dt} \tag{2}$$

Neglecting the thermal radiation effects, the boundary surface condition can be written in the general form as

$$q = -k\frac{dT}{d\eta} - h\left(T - T_o\right) \tag{3}$$

where

ρ : Density
c : specific heat
k : thermal conductivity.
T : brake drum temperature at x,y,z,t
T_o : surrounding air temperature
t : Time
x,y,z : Cartesian coordinate
η : direction of heat flow
h : heat transfer coefficient

FINITE ELEMENT MODEL

A heavy and large brake drum of a commercial truck is considered for analysis. The temperature distribution is found by solving Eq.(3) using numerical analysis. ANSYS finite element based software is used to solve the problem. Fig.2 shows the finite element model of the brake drum with a circumferential fin near its open end.

SOLID90 is chosen as the element type since the simulation is done in three-dimensional model and temperature is the only degree of freedom. The model is meshed by using wedged hexahedral element shape with the total number of nodes is 47736. Appropriate element length is identified using trial and error method until it produces best and most consistent element shape. This process is important since the mesh pattern contributes significantly to the accuracy of the simulation result. It is finally come out with length element of 0.004 m and this value give best mesh for the model volume.

THERMAL STRESS ANALYSIS

Thermo-elasticity describes the behavior of elastic bodies under the influence of non-uniform temperature fields. It represents, therefore, a generalization of the theory of elasticity. The thermal stress may arise in a heated body either because of a non-uniform temperature distribution, or external constraints, or a combination of these causes.

The total strains at each point of the heated body are, thus, made up of two parts. The first part is a uniform expansion proportional to the temperature rise ΔT. Since this expansion is the same in all directions for an isotropic body, only normal strains and no shearing strains arise in this manner. If the coefficient of linear thermal expansion is denoted by α, this normal strain in any direction is equal to $\alpha\Delta T$.

The second part comprises the strains required to maintain the continuity of the body as well as those arising because of external loads. These strains are related to the stresses by means of the usual Hooke's law of linear isothermal elasticity. Thus, the strains are the sum of the two components and are, therefore, related to the stresses and temperature in any general form of orthogonal coordinate system as

$$\varepsilon_{ij} = \frac{\sigma_{ij}}{2\mu} + \frac{(1-2\nu)}{E}\delta_{ij}\sigma_{kk} - \delta_{ij}\alpha\Delta T \tag{4}$$

where

σ_{ij} : deviatoric stress components
σ_{kk} : hydrostatic stress components
μ : Lame's constant
ν : Poisson's ratio
E : modulus of elasticity
δ_{ij} : Kronecker deltas
α : linear thermal expansion
ΔT : temperature change

The strains are related to the displacements in the same manner as in isothermal elasticity since purely geometrical considerations are involved; in a rectangular Cartesian coordinate system the pertinent equations are as follows:

$$\varepsilon_{ij} = \begin{Bmatrix} \varepsilon_x \\ \varepsilon_y \\ \varepsilon_z \\ \gamma_{xy} \\ \gamma_{yz} \\ \gamma_{zx} \end{Bmatrix} = \begin{Bmatrix} \dfrac{\partial u}{\partial x} \\ \dfrac{\partial v}{\partial y} \\ \dfrac{\partial w}{\partial z} \\ \dfrac{\partial u}{\partial y} + \dfrac{\partial v}{\partial x} \\ \dfrac{\partial v}{\partial z} + \dfrac{\partial w}{\partial y} \\ \dfrac{\partial w}{\partial x} + \dfrac{\partial u}{\partial z} \end{Bmatrix} \tag{5}$$

where u, v, and w are the components of the displacement vector in the x, y, and z direction respectively.

RESULT AND DISCUSSION

The 3-D temperature distribution in the brake drum has been analyzed using finite element method. The analysis is performed with a number of assumptions. The drum material is made of grey cast iron A48 Class 40 that is homogenous and isotropic. The thermal properties of this material are as follows:

- Thermal conductivity, k = 51 W/m.K
- Specific heat, C_p = 550 J/kg.K
- Density, ρ = 7196 kg/m^3
- Thermal diffusivity, α = 12.89 x 10^{-6} m^2/s

The heat transfer coefficient along the interface between the surface of brake drum and surrounding air is also constant at 50 W/m^2.K. The main data for the brake drum dimensions are as follows

- Total length of brake drum = 270 mm
- Outer diameter of brake drum = 450 mm
- Inner diameter = 420 mm
- Thickness of brake drum = 15 mm
- Thickness of circumferential fin = 10 mm
- Length of circumferential fin = 20 mm
- Shoe contact angle = 100^o
- Shoe width = 200 mm

The wheel load of 40 kN is taking into account and the braking is applied from a speed of 100 km/h to complete stop with a deceleration of 0.5*g*, and it takes 5.663 second. The initial temperature of the brake drum and surrounding air are set to be 30°C.

Fig.3 shows the profile of temperature rise of nodes *A*, *B*, *C*, and *D* during the first brake application. The effect of cyclic heating and cooling fluctuates the temperature rise of node *A* and reaches the peak temperature at the mid-time of brake application, when the time lapse is 2.85 s, and decreases afterward. The peak temperature of point *A* is about 248°C. This phenomenon is predominantly caused by the decreasing dissipated kinetic energy input with respect to time due to the decreasing of truck's speed. Furthermore, it can be considered that the heat convection into the surrounding air is greater than the dissipated kinetic energy input. On the other hand, nodes *B*, *C*, and *D* continuously increase until the end of braking. This means that the heat conduction of these nodes still advances until the braking is fully terminated.

The temperature history of the inner surface of brake drum in the course of braking is shown in Fig.4. It can be seen that the temperature rise in the frictional contact area is evenly distributed. The temperature rise at the flank of brake drum increases gradually, due to the poor heat conduction flowing to this area.

Fig.5 depicts the temperature history of the *a-a* cross-section. The steep or higher temperature gradient occurring in the thickness layer of 1.0 cm from the inner surface will cause higher thermal stress; and the maximum thermal stress occurs at the rubbing surface. Small gradient temperature is shown when the heat flows to the region of the circumferential fin.

The history of von Mises thermal stress for nodes *A, B, C*, and *D* is shown in Fig.6. The 95-MPa maximum thermal stress occurs at node *A* when the time of brake application is reaching $t = 1.32$ sec. Similar with the temperature profile, the profile of thermal stress is also fluctuating due to heating and cooling in the course of brake application. Considering the history of thermal stress, we can conclude that the highest thermal stress occurs at around one-fourth of braking time.

Fig.7 shows the von Mises thermal stress history in *a-a* cross-section in the course of braking. The circumferential fin has an effect to reduce the thermal stress and creates a significant thermal stress gradient that occurs in the region of rubbing surface having a layer thickness of 0.4 mm. It shows a good agreement with the results reported by Masashi Daimaruya et al. that the function of circumferential fin is not only to reinforce the brake drum but also to reduce the thermal stress and the effect of bell-mouthing due to thermal distortion.

CONCLUSION

The thermal stress distributions under a severe brake application have been discussed as the first step in investigating the initiation mechanism of heat cracks in the brake drum, the thermal stress distributions under a severe brake application have been discussed. A heavy and large brake drum of a commercial truck is considered and analyzed by using three-dimensional finite element method. It was found that the severe thermal environment occurred in the inner surface of brake drum will create a high thermal stress in the rubbing surface. The effect of cyclic heating and cooling fluctuates the temperature rise in the rubbing surface and reaches the peak temperature at the mid-time of brake application, and decreases afterward. The profile of thermal stress created in the brake drum also fluctuates and reaches the maximum value when the braking time is around one-fourth of the fully stopped brake application.

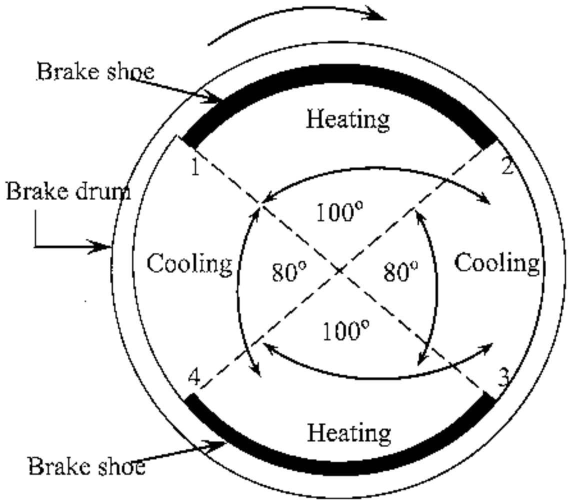

Fig.1 Clock mechanism of heating and cooling.

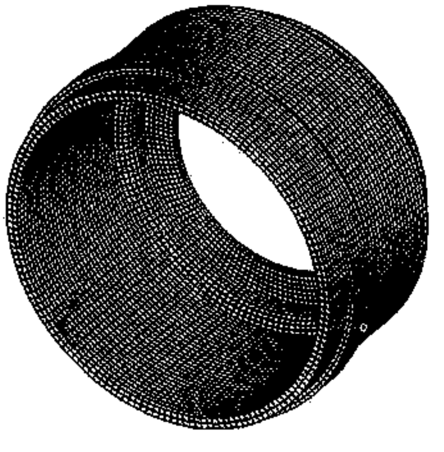

Fig.2 Meshing of finite element model.

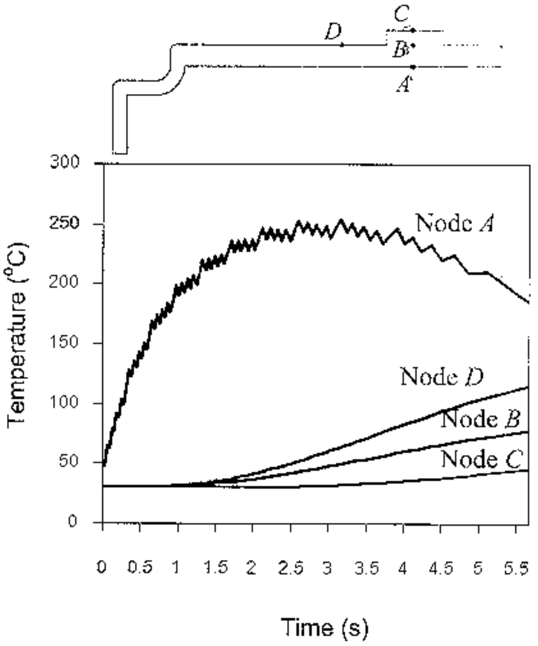

Fig.3 Temperature history of nodes *A*, *B*, *C*, and *D* at brake application.

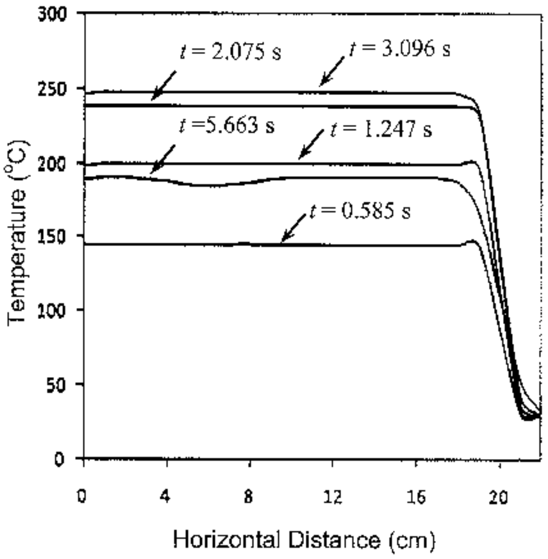

Fig.4 Temperature history of inner surface.

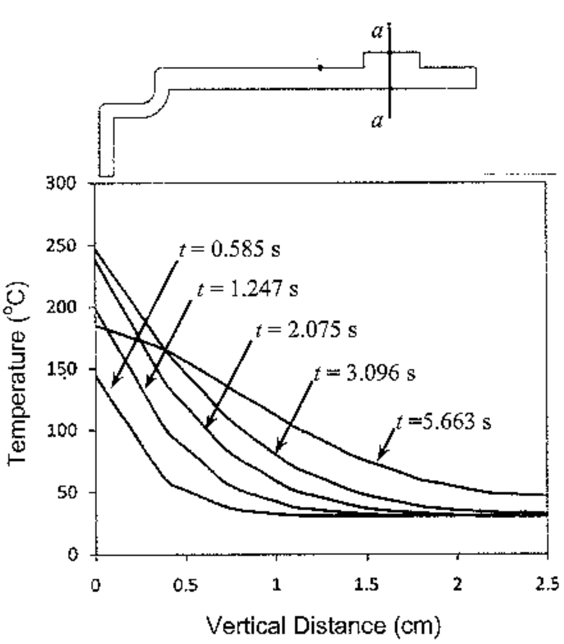

Fig.5 Temperature history of *a-a* cross-section.

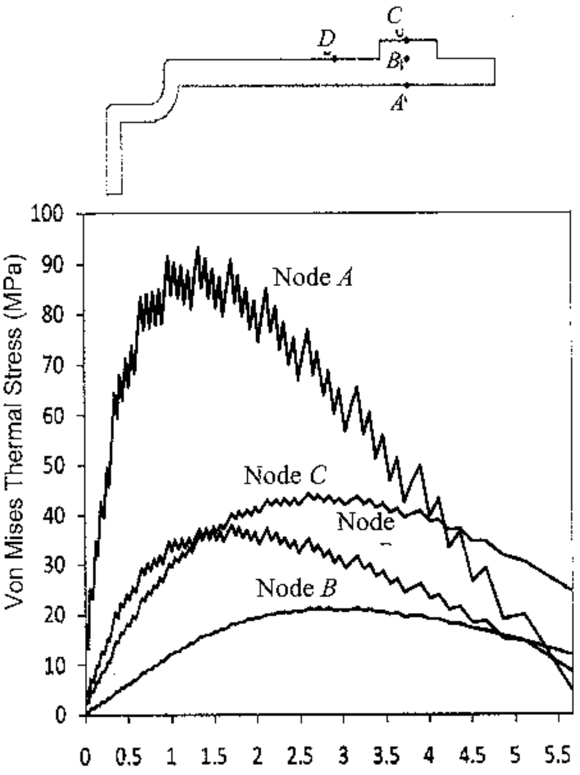

Fig.6 Thermal stress history of nodes *A*, *B*, *C*, and *D* at brake application.

Time (s)

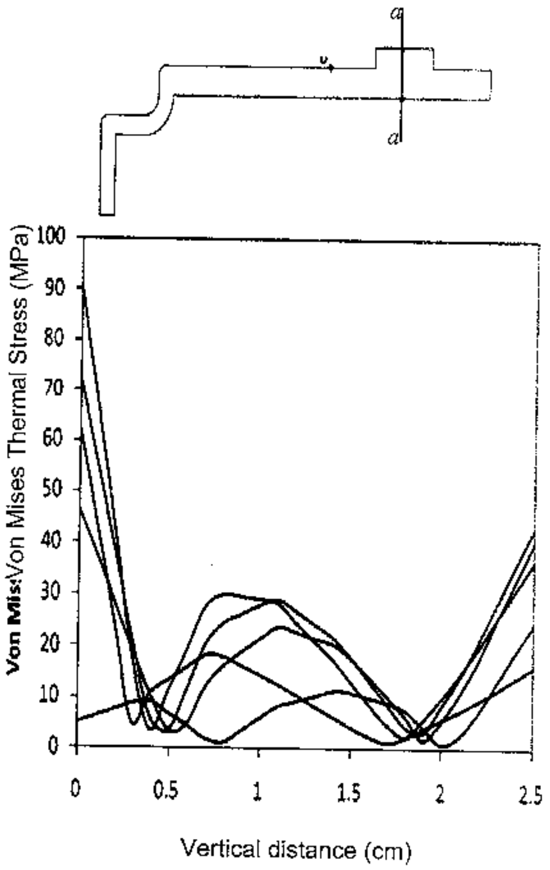

Fig.7 Thermal stress history in *a-a* cross-section.

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