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Investigation of Heat Transfer Phenomena in a Ventilated Disk Brake Rotor with Straight Radial Rounded Vanes

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Abstract: In this research, two major models are used for calculation of frictional heat generation: namely macroscopic and microscopic model. In the macroscopic model, the law of conservation of energy or First Law of Thermodynamics is taken into account. And for the microscopic model, parameters such as the duration of braking, velocity of the vehicle, dimensions and geometry of the brake system, materials of the disk brake rotor and the pad are taken into account. For calculation of prescribed heat flux boundary condition in this model two kinds of pressure distribution is considered: uniform wear and uniform pressure. In high demand braking applications, vented disks consisting of two rubbing surfaces separated by straight radial vanes are normally employed as they utilize a greater surface area to dissipate heat. Within this paper the conduction heat transfer into a high performance passenger car front brake disk has been investigated using Finite Element Method.

Key words: Disk brake, temperature rise, frictional heat generation, vented disk, pad, finite element method

INTRODUCTION

Disk brakes in recent decades have been widely used in light vehicles. Proper performance of a vehicle brake system is one of its advantages. Long repetitive braking leads to temperature rise of various component of the vehicle brake system that reduces the performance of the brake system.

Braking performance of a vehicle can be significantly affected by the temperature rise in the brake components. High temperatures during braking may cause brake fade, premature wear, brake fluid vaporization, bearing failure, thermal cracks and thermally-excited vibration. Therefore, it is important to predict the temperature rise of a given brake system and assess its thermal performance in the early design stage (Valvano and Lee, 2000).

Disk brakes have great diffusion, because permit a strong and modulating braking. Today they are coupled with mechatronic systems as A.B.S. During the motion the vehicle has kinetic energy. The brakes have the purpose to diminish or destroy this energy by dissipating it in the form of heat. Because the temperature of 400-500°C are obtained during a demanding braking the materials have to be able to support the high mechanical and thermal stresses. The disk brake and the pads have to own a high thermal conductibility, a high surface-volume ratio and a high mechanical resistance to wear. The disk must have limited mass in order to diminish the inertia forces and non-suspending mass. Moreover it cannot endure high strain in the braking zone in order to maintain uniform contact with the pads (Gotowicki *et al*., 2005).

Disc brakes are exposed to large thermal stresses during routine braking and extraordinary thermal stresses during hard braking. High-g decelerations typical of passenger vehicles are known to generate temperatures as high as 900°C in a fraction of a second. These large temperature excursions have two possible outcomes: thermal shock that generates surface cracks and/or large amounts of plastic deformation in the brake rotor. In the absence of thermal shock, a relatively small number of high-g braking cycles are found to generate macroscopic cracks running through the rotor thickness and along the radius of the disc brake (Mackin *et al*., 2002).

Formation of hot spots as well as non-uniform distribution of the contact pressure is an unwanted effect emerging in disc brakes in the course of braking or during engagement of a transmission clutch. If the sliding velocity is high enough, this effect can become unstable and can result in disc material damage, frictional vibration, wear, etc. (Voldøich, 2006).

Frictional heat generation in the sliding contact of two bodies influences friction and wear characteristics of brake systems. We note that numerous experimental evidences suggest that the contact area is generally circular, e.g., tread broke railway wheels exhibit circular thermally affected zones on the surface (Yevtushenko and Chapovska, 1997).

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Long repetitive braking, such as one which occurs during a mountain descent, will result in a brake fluid temperature rise and may cause brake fluid vaporization. This may be a concern particularly for the passenger cars equipped with aluminum calipers and with a limited air flow to the wheel brake systems (Lee, 1999).

According to Ostermeyer (2001), the contact area in brake systems shows characteristic structures. With respect to wear, equilibrium of flow of growing and destruction of hard patches is to be found on the contact surface. These patches modulate the friction coefficient of the brake system. Dealing with this principal wear mechanism of brake pads the dynamic friction coefficient describes the stationary and transient friction behavior of brake pads. For instance the fading effect is result of a temporary higher destruction rate than the grow rate, when normal force and velocity get high, up to that point, where the equilibrium of flow of power is reached again on a lower level.

Gao and Lin (2002) have presented an analytical model for the determination of the contact temperature distribution on the working surface of a brake. To consider the effects of the moving heat source (the pad) with relative sliding speed variation, a transient finite element technique is used to characterize the temperature fields of the solid rotor with appropriate thermal boundary conditions. Numerical results shows that the operating characteristics of the brake exert an essentially influence on the surface temperature distribution and the maximal contact temperature.

Dufrénoy (2004) proposed a macrostructural model of the thermomechanical behavior of the disc brake, taking into account the real three-dimensional geometry of the disc-pad couple. Contact surface variations, distortions and wear are taken into account. Real body geometry and thermoelastoplastic modeling of the disc material are specially introduced. Such a model aims to give predictions of the thermal gradients varying with time and of the thermomechanical response of the components. Predictions of the temperature distributions are compared with experimental measurements obtained by thermographs and thermocouples. Such a model seems to be a suitable base for the study of the thermal dissipation and the thermomechanical behavior and for the introduction of local friction effects.

The primary function of a brake rotor is to act as a friction surface, generating an opposing torque to a shaft. The automotive and aerospace industries rely on these devices to provide vehicle deceleration. In this application, the brake rotor commonly forms part of the wheel assembly, transferring the opposing torque to the tires of the vehicle. During braking, energy is transferred to the rotor in the form of heat. As a result, the brake rotor must also serve as an efficient energy dissipation and storage device. In most cases, air must be circulated through the rotor to provide adequate cooling. The passages, formed by the radial fins between the braking surfaces, act as a centrifugal fan, facilitating the required air flow for cooling (McPhee and Johnson, 2007).

Recently metallurgical study of disk brakes has been carried out in many studies (Mosleh *et al*., 2004; Mutlu *et al*., 2005; Hecht *et al*., 1999; Gudmand-Høyer *et al*., 1999; Uyyuru *et al*., 2007; Cho *et al*., 2005; Boz and Kurt, 2007; Blau and McLaughlin, 2003).

In this research analysis of the conduction heat flow with true representation of heat generation during braking action within the rotors and the pad using Finite Element Method (FEM) is investigated.

MATERIALS AND METHODS

Governing equations in disk brake system: Figure 1 shows the disk brake system of a car and pad that is separated from wheel assembly to better show the disk and the pad in sliding contact. As it can be seen, typical disk brake system and caliper assembly of a solid disk brake rotor is completely noticeable. Figure 2 shows schematic form of the disk and the pad in sliding contact.

Modes of heat transfer is disk brakes: As it can be seen from Fig. 3, prediction of the heat transfer in disk brake system is quite complex. The heat flux will vary with time; a linearly decreasing heat flux is often assumed. The heat transfer coefficient (h) will rarely be constant, although an average value is usually assumed.

In order to accurately determine brake temperatures at a particular time (t) in the braking phase, knowledge of the heat transfer coefficient is required. The heat transfer

Fig. 1: Disk brake rotor, pad and caliper assembly

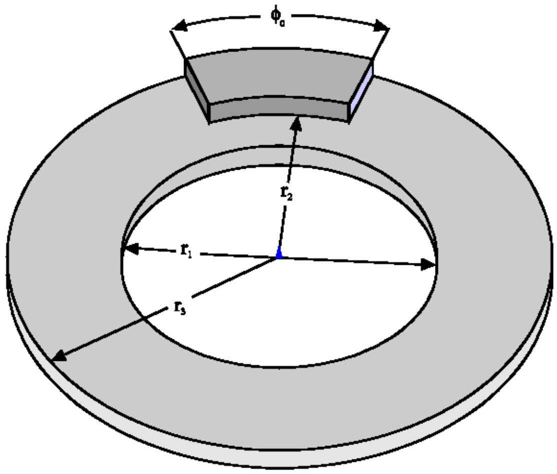

Fig. 2: Schematic form of the disk and the pad in sliding contact

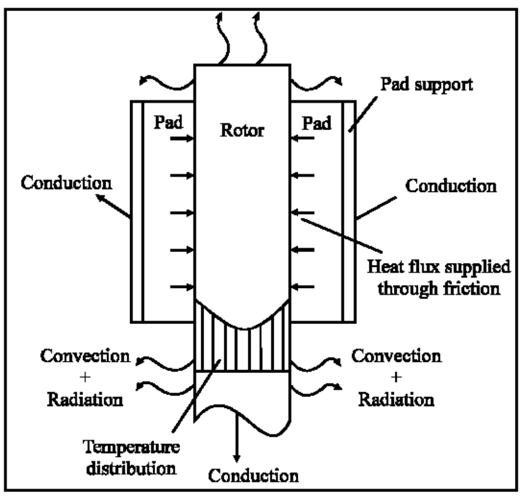

Fig. 3: Heat transfer mechanism in a disk brake system

coefficient (h) will depend on the airflow in the region of the brake rotor and the vehicle speed. It will therefore vary constantly throughout the braking process. It is generally considered extremely difficult to obtain accurate values of the heat transfer coefficient.

Heat dissipation from the brake disk will occur via conduction through the brake assembly and hub, radiation to nearby components and convection to the atmosphere. At high temperatures heat may create chemical reactions in the friction material, which may dissipate some of the braking energy. While conduction is an effective mode of heat transfer, it can have adverse effects on nearby components. Such effects include damaged seals, brake fluid vaporization, as well as wheel bearing damage. Radiation heat transfer from the rotor will have its greatest effect at higher temperatures but must be controlled to prevent beading of the tire. It is estimated that the amount of heat dissipation through radiation under normal braking conditions is negligible. Convection to the atmosphere must then be the primary means of heat dissipation from the brake rotor. Convection is governed by the expression:

$$Q = hA(T_s - T_\infty) \quad (1)$$

Also known as Newton's law of cooling.

Where:
Q = Rate of heat transfer (W)
h = Convection heat transfer coefficient
A = Surface area of the rotor m^2
T_s = Surface temperature of the brake rotor (°C)
T_∞ = Ambient air temperature (°C)

It can be seen from this expression that in order to maximize heat transfer from the rotor (increase Q) and keep the rotor temperature (T_s) to a minimum value, the value of heat transfer coefficient (h), or the surface area (A_s) needs to be increased. As it is required to keep T_s to a minimum value, improvements must be made through increasing the heat transfer coefficient (h) and/or the surface area (A_s) of the rotor. The amount by which the surface area can increase is confined by the diameter of the wheel and the requirements of minimizing unsprung mass, so improvements in cooling can best be made through increased values of the heat transfer coefficient. The use of an internally ventilated rotor will increase both surface area (extra internal area exposed to the atmosphere) and the heat transfer coefficient, due to forced convection created by the internal airflow, with negligible influence on unsprung mass. The material selection and the physical dimensions of the rotor will also have a direct bearing on cooling ability.

SRV-R rotors: In this study, the ventilated brake disk used is a SRV rounded rotor (Fig. 4) with 39 vanes. Recently many investigations have been carried out on disk brake rotor to study and improve ventilation through rotors (Wallis *et al*., 2002; Johnson *et al*., 2003; McPhee and Johnson, 2007).

Johnson *et al.* (2003) used Particle Image Velocimetry (PIV) to measure air velocities through a high solidity radial flow fan utilized as an automotive vented brake rotor. For three typical rotational speeds, the flow characteristics were captured at the inlet and exit of the rotor, as well as internally through the cooling passages. Inlet measurements showed a swirling entry flow condition with significant misalignment of flow onto the

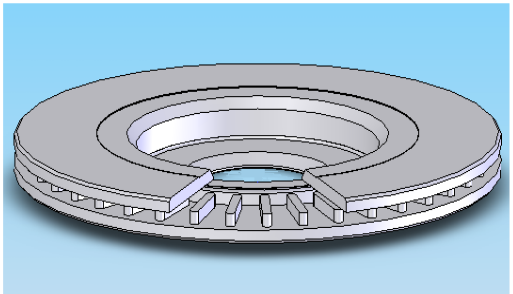

Fig. 4: Ventilated disk brake rotor with straight radial rounded vanes

vanes. As a result large regions of flow separation were found in the internal vane-to-vane passages on the suction side surfaces, which would lead to poor heat transfer conditions.

McPhee and Johnson (2007) employed experimental and analytical methods for better understanding of convection through the fins of a brake rotor. The experimental approach involved two aspects, assessment of both heat transfer and fluid motion. A transient experiment was conducted to quantify the internal (fin) convection and external (rotor surface) convection terms for three nominal speeds. For the given experiment, conduction and radiation were determined to be negligible. Rotor rotational speeds of 342, 684 and 1025 rpm yielded fin convection heat transfer coefficients of 27.0, 52.7, 78.3 $Wm^{-2} K^{-1}$, respectively, indicating a linear relationship. At the slowest speed, the internal convection represented 45.5% of the total heat transfer, increasing to 55.4% at 1025 rpm. The flow aspect of the experiment involved the determination of the velocity field through the internal passages formed by the radial fins. Utilizing PIV, the phase-averaged velocity field was determined. A number of detrimental flow patterns were observed, notably entrance effects and the presence of recirculation on the suction side of the fins.

Empirical correlations versus TASCflow results (Wallis *et al.*, 2002) shows that the model suggested by the following correlations for description of heat convection through SRV-R rotors best describes the heat flow through this type of rotors. So we use these correlations for obtaining the heat transfer coefficient used in this study.

$$V_{ave} = (-0.020 + 0.908d - 0.202d^2)^{0.5}\omega(D/2) \tag{2}$$

$$Re_{D/2} = (D/2)\,{}^{V_{ave}}/_{\nu} \tag{3}$$

$$Nu_{D/2} = 0.045\,Re^{0.8}(D/2)^{0.2}[1 + 6.6(D/2)^{0.8}] \tag{4}$$

Models of heat dissipation in disk brakes

Microscopic model: Rate of frictional heat generation is equal to friction power. Some of this frictional heat is absorbed by the disk and the rest is absorbed by the pad. According to Majcherczak *et al.* (2005) two kinds of thermal contact descriptions are usually used in analyses:

Perfect contact: Considering equal surface temperature of the disk and the pad.

Imperfect contact: Considering a heat resistance between the disk and the pad due to the formation of third body constituted by detached particles.

Since contact surfaces of the disk and the pad is not the same, they don't have perfect contact. Considering the imperfect contact, the heat partition coefficient is given as:

$$\sigma = \frac{\xi_d S_d}{\xi_d S_d + \xi_p S_p} \tag{5}$$

where, ξ_p, ξ_d are the thermal effusivity of the pad and the disk and S_p, S_d are the frictional surfaces of the pad and the disk, respectively. Thermal effusivity is obtained from the following equation:

$$\zeta = \sqrt{k\rho c} \tag{6}$$

To calculate the frictional heat generation at the contact zone of the two components of the brake system, the friction coefficient between two sliding components, relative sliding velocity, geometry of the disk and the pad and the pressure distribution at the sliding surfaces is required. Here we consider two assumptions for pressure distribution:

Pressure distribution

- Uniform pressure

$$p = p_{max} \tag{7}$$

- Uniform wear

$$\delta = kpr = const \Rightarrow p = p_{max}\frac{r_2}{r} \tag{8}$$

where, δ is defined as wear, p_{max} is the maximum distributed pressure in the pad and p is the pressure at radial position.

Heat generation: Figure 5 shows the contact surface element of the pad and the disk. The rates of heat

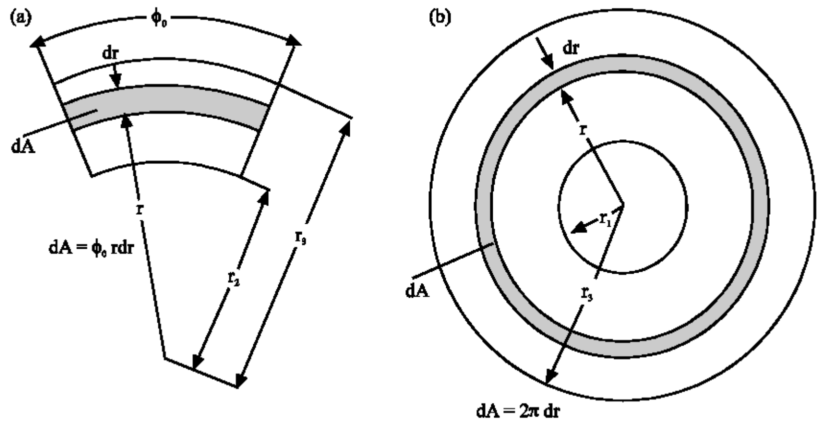

Fig. 5: Contact surface element of two components (a) the pad and (b) the disk

generated due to friction between these surfaces are formulated as follows.

$$d\dot{E} = dP = VdF_f = r\omega\mu p\phi_0 rdr \tag{9}$$

$$d\dot{E} = d\dot{E}_p + d\dot{E}_d \tag{10}$$

$$d\dot{E}_p = (1-\sigma)dP = (1-\sigma)\mu p\omega\phi_0 r^2 dr \tag{11}$$

$$d\dot{E}_d = \sigma dP = \sigma\mu p\omega\phi_0 r^2 dr \tag{12}$$

where, $d\dot{E}$ is the rate of heat generated due to friction between two sliding components and dF_f is the friction force. $D\dot{E}_p$ and $D\dot{E}_d$ are the amount of absorbed heat by the pad and the disk, respectively.

Heat flux: To obtain the prescribed heat flux boundary condition at the surfaces of two components of the brake system, rate of thermal energy is divided by the surface contact area of each component.

- Pad

$$q_1(r,t) = \frac{d\dot{E}_p}{dS_p} = (1-\sigma)\mu pr\omega(t) \tag{13}$$

$$q_{0_1}(r) = q_1(r,0) = (1-\sigma)\mu pr\omega_0 \tag{14}$$

- Disk

$$q_2(r,t) = \frac{d\dot{E}_d}{dS_d} = \frac{\phi_0}{2\pi}\sigma\mu pr\omega(t) \tag{15}$$

$$q_{0_2}(r) = q_2(r,0) = \frac{\phi_0}{2\pi}\mu\sigma pr\omega_0 \tag{16}$$

Heat flux for uniform pressure distribution is a function of time and space variable r; the angular velocity decreases with time during braking action and the work done by friction force grows as radial space variable increases. This phenomenon is quite often when the pad is new. However after several braking, assumption of uniform wear is more realistic. Heat flux obtained for the uniform wear is just a function of time and it is independent of the space variable; the work done by friction force is the same at radial direction.

Macroscopic model: Brakes are essentially a mechanism to change the energy types. When a car is moving with speed, it has kinetic energy. Applying the brakes, the pads or shoes that press against the brake drum or rotor convert this type of energy into thermal energy. The cooling of the brake dissipates the heat and the vehicle slows down. This is all to do with the first law of thermodynamics, sometimes known as the law of conservation of energy which states that energy cannot be created nor destroyed; it can only be converted from one form to another. In the case of brakes, it is converted from the kinetic energy to the thermal energy.

$$E_c = \tfrac{1}{2}MV_0^2 \tag{17}$$

where, M is the total mass of the vehicle and V_0 is the initial speed of the vehicle. To obtain the amount of heat dissipated by each of the front brake disks we should know the weight distribution of the car. Therefore the amount of heat dissipated by each of the disks is:

Table 1: Geometry and material propertied of the disk and the pad

Item	Disk	Pad
k ($W\ m^{-1}\ K^{-1}$)	45.0	12.0
ρ ($kg\ m^{-3}$)	7850.0	2500.0
c ($J\ kg^{-1}\ K^{-1}$)	460.0	900.0
Inner diameter (mm)	66.5	88.0
Outer diameter (mm)	133.0	133.0
Thickness (mm)	20.4	18.8

Table 2: Vehicle data

Vehicle: Samand 1,8i EP2	Tyre pressure: 2.1 (bar)	Total piston area: 22.90 (am^2)
Test mass: 130 kg	Master cylinder: φ20.64×16/15 (mm)	Disk/Drum: φ266×20.4 (mm)
Axle load FA/RA: 772/535	Brake pedal ratio: I = 3.75	Effective radius: 108 (mm)
Deceleration: 8 ($m\ sec^{-2}$)	Lining material: Textar 4005	Pad angle: 40 (°C)
Rolling radius: 0.30 m	Brake: TT/54/13/20	Efficiency: 100%
Tyre size: 185/65 R 15 GY	Piston: φ54 (mm)	Road and weather condition: Dry
Initial vehicle speed: 151.4 ($km\ h^{-1}$)	Cylinder pressure: 90 (bar)	

$$E = 0.5 \times \tfrac{1}{2} m V_0^2 = 0.25 m V_0^2 \tag{18}$$

where, m is the amount of mass on the front axle of the vehicle. Velocity of the vehicle slows down with the assumption of constant deceleration according to following relation:

$$V = V_0 (1 - t/t_b) \Rightarrow \omega = \omega_0 (1 - t/t_b) \tag{19}$$

Automotive brake application: Consider the automotive disk brake issued from Samand vehicle. The disk material is steel, while the pad material is made of an organic matrix composite. Their physical properties at room temperature are detailed in Table 1. The details of the dimensions and operating condition at the initial vehicle speed of V_0 = 151.4 ($km\ h^{-1}$) are given in Table 2.

RESULTS AND DISCUSSION

Maximum contact surface temperatures of the pad and the disk for two types of pressure distributions are shown in Fig. 6. Maximum temperature obtained for uniform pressure distribution is higher than that for uniform wear. The reason is that with the assumption of uniform pressure, the work done by the friction force grows as the radius increases. Meanwhile the work done by the friction force with the assumption of uniform wear does not vary with radius. Moreover the mean values of the surface temperature for the two types of pressure distribution are the same. As it can be seen, the difference between the surface temperature of the disk and the pad is relatively high. This is due to the thermal resistance between the disk and the pad, constituted by the accumulation of wear particles that form a thin layer (usually called third body). Therefore, this thermal resistance causes a heat partition between the disk and the pad.

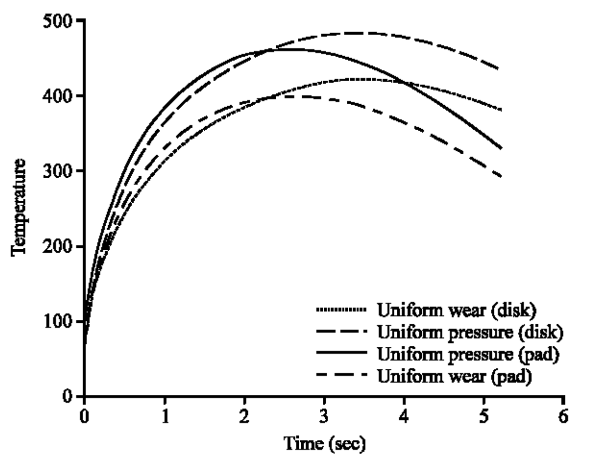

Fig. 6: Maximum disk and pad surface temperature versus time

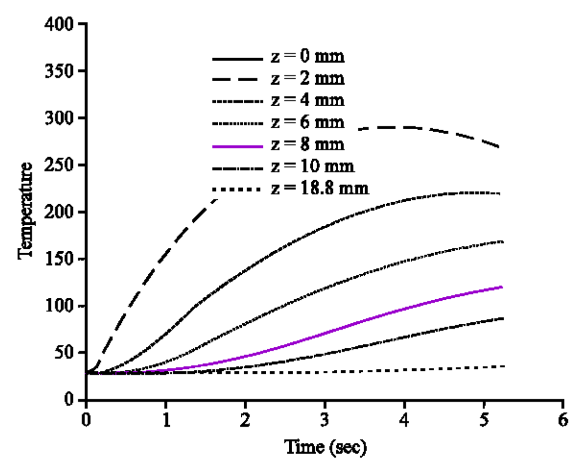

Fig. 7: Pad temperature difference in z direction at r = 108 (mm) versus time (uniform wear)

In Fig. 7 pad temperature versus time at the position of mean sliding radius (r = 108 mm) and different z

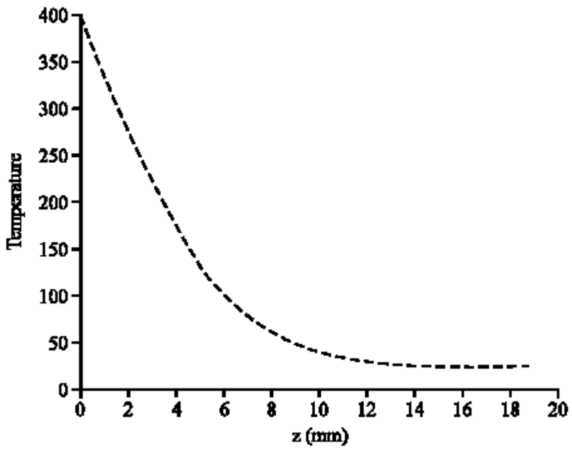

Fig. 8: Pad temperature in axial direction (uniform wear)

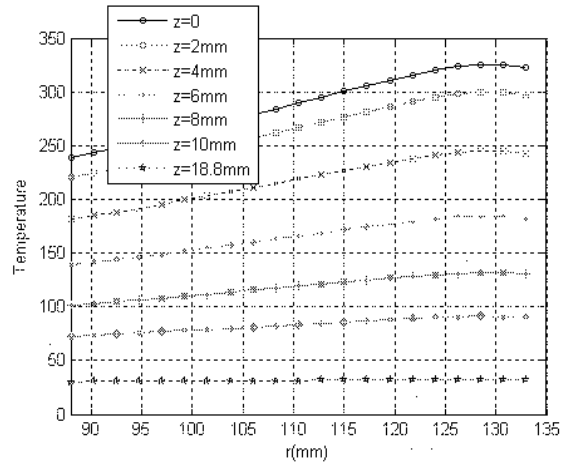

Fig. 9: Pad temperature in radial distances at different axial position (uniform pressure)

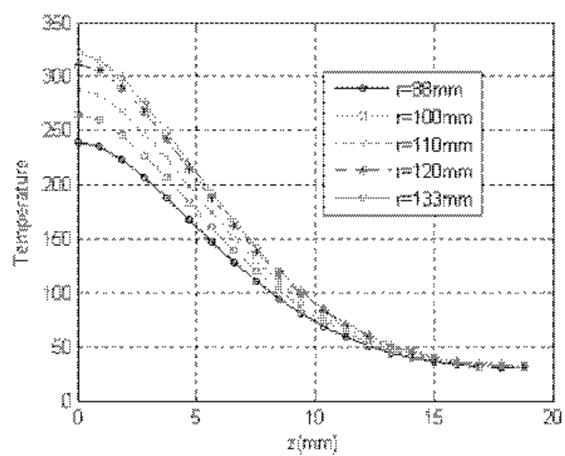

Fig. 10: Pad temperature in axial direction at different radial distances (uniform pressure)

locations with the assumption of uniform wear is illustrated. As it can be seen, pad temperature at contact surface of the disk and the pad (z = 0) has the highest value and it gradually decreases at different z location up to z = 18.8 mm. Furthermore the slope $\frac{\partial T}{\partial t}\big|_{t=0}$ at the position of z = 0 has the highest value. So that temperature increases with time, rapidly. The slope decreases with time until the time t = 2.63 sec that the slope is zero. This time is the time which maximum temperature occurs and after that the slope is negative and temperature decreases with time. This behavior is dominant for z = 2 mm that maximum temperature is occurred at time t = 3.86 sec and for z = 4 mm at time t = 4.97 sec. However at different z position such as z = 6, 8, 10, 18.8 mm this phenomenon isn't dominant and the slope $\frac{\partial T}{\partial t}\big|_{t=0}$ decreases as z increases. At this position the slope increases with time which means that the temperature is increasing with time. If the braking action is repeated e.g., during mountain decent, this may be a concern, especially for brake systems equipped by aluminum calipers that may cause brake fluid vaporization. On the other hand the highest brake fluid temperature is often observed during the so-called heat soaking period or when the car is parked after long repetitive braking such as one which may be encountered in a high mountain descent.

In Fig. 8, pad temperature in axial direction at time of t = 2.63 sec with the assumption of uniform wear is illustrated. As it can be seen, the value of temperature differences in axial direction is very high (about 373.33°C). However this temperature distribution is dominant for the time period that the pads have sliding contact with the disk. After this period especially when the car is parked, according to Newton's law of cooling, the temperature at the position of z = 0 decreases. Meanwhile the temperature at z = 18.8 mm increases rapidly, because of low convection coefficient (natural convection) and heat soaking phenomena.

As it can be seen from Fig. 9 temperatures in radial distances have almost a linear behavior. Since, as mentioned earlier the work done by friction force for the assumption of uniform pressure is proportional to radial distance.

In Fig. 10 pad temperature in axial direction at different radial distances and at the end of the braking time is illustrated. As it can be seen, the value of temperature difference in axial direction is relatively high. Furthermore, because of the assumption of uniform pressure, the temperature grows as the radial distance increases.

Figure 11 shows contact surface temperature variation of the disk along radial direction with the

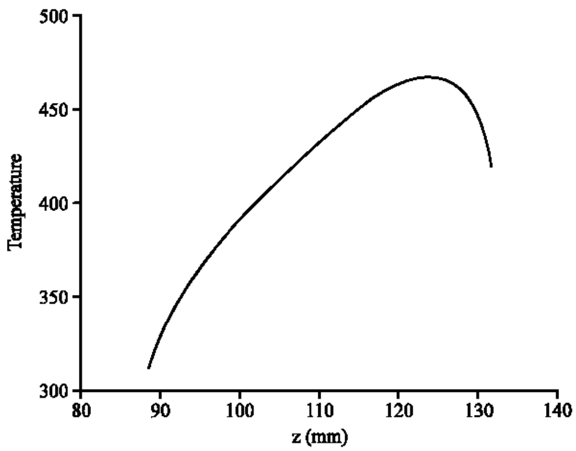

Fig. 11: Disk surface temperature along radial direction (uniform pressure)

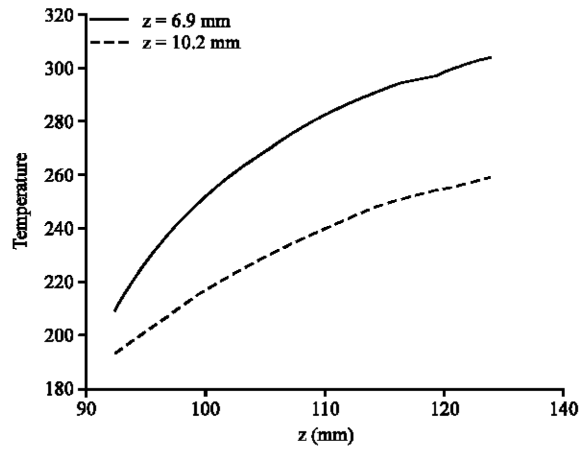

Fig. 13: Fin temperature variation in radial distances (uniform pressure)

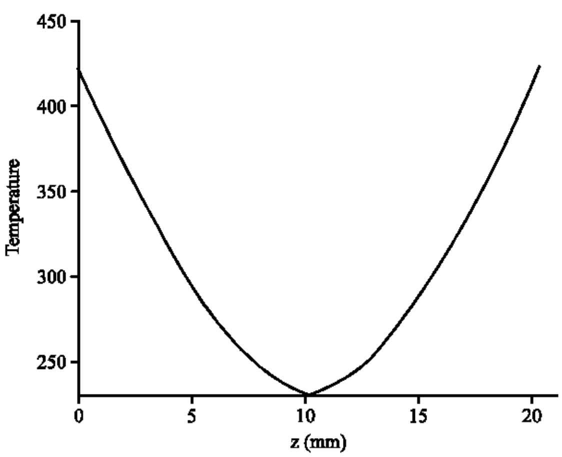

Fig. 12: Disk temperature variation along axial direction

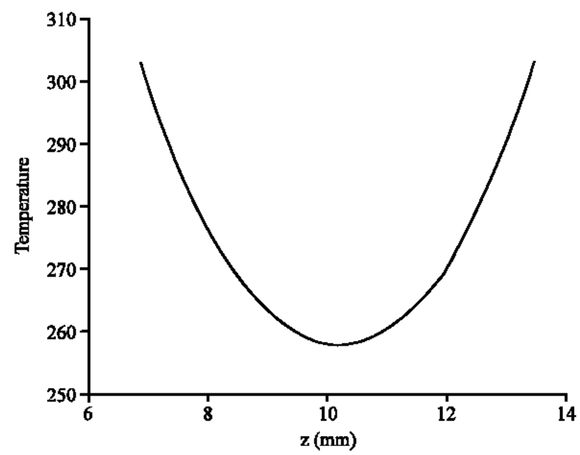

Fig. 14: Fin temperature in z direction

assumption of uniform pressure. As it can be seen, temperature increases along r direction up to r = 124 mm because the work done by friction force is higher in radial direction and after that value of temperature falls because of convective heat dissipation to the environment.

Figure 12 shows disk temperature variation along axial direction at radial position of maximum temperature. As it can be seen, the value of disk contact surface temperature (z = 0, z = 20.4 mm) is higher than the other parts and because of symmetrical behavior, disk temperature must have a minimum value at the plane of symmetry which locates at the position of z = 10.2 mm.

Figure 13 shows the value temperature variation of fins along radial distances, at two different axial locations with the assumption of uniform pressure. As it can be seen, temperature grows as radial distance increases and at the vicinity of contact surface; z = 6.9 mm, the value of temperature is more than that at the plane of symmetry which locates at the position of z = 10.2 mm.

Figure 14 shows the temperature distribution of a fin along axial direction. As it can be seen, the value of fin temperature near contact surfaces; z = 6.9 and z = 13.5 mm, is higher than other parts. Because of symmetrical behavior, the value of fin temperature must have a minimum value at the plane of symmetry i.e., z = 10.2 mm.

Figure 15 shows pad and caliper temperature variation versus time during braking action (t_b = 5.26 sec) and after braking action (after 40 sec). At the beginning of braking action, the value of temperature at sliding contacts increases rapidly with time whereas the temperature for position of caliper assembly remains constant. After this period that the pad don't have any sliding contact with the disk, according to Newton's law of cooling the pad contact surface temperature at z = 0

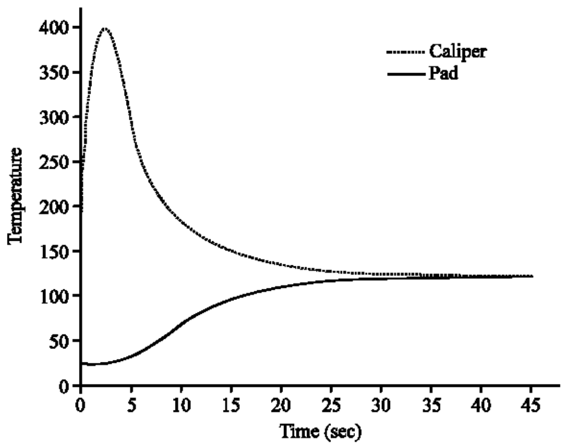

Fig. 15: The value of pad and caliper temperature versus time during and after braking action

Table 3: Calculated and measured data

Vehicle speed (km h^{-1})	71.50	101.00	151.40
Braking time (sec)	2.73	3.65	5.26
Total heat generated (kJ)	76.10	151.90	341.40
Average vane velocity (m sec^{-1})	3.21	4.54	6.81
Convection coefficient (W m^{-2} k^{-1})	48.40	63.80	88.20
Maximum temperature (°C) (measured data)*	187.00	270.00	395.00
Maximum temperature (°C) (numerical value)	144.50	252.50	419.00

*These data is reported by IKCo for Samand vehicle

decreases. Meanwhile caliper temperature increases rapidly. As it can be seen it takes 40 sec for two diagrams to merge. This may be a concern especially for disk brakes equipped by aluminum calipers that may cause brake fluid vaporization because of heat soaking phenomena.

Calculations of total heat generated due to friction, average vane velocity, convection coefficient and maximum contact temperature of the rotor for three different vehicle speeds is shown in Table 3. Calculation of flow properties has been conducted using standard air at 25°C. As it can be seen the vane velocity versus vehicle speed has a linear relation. However the heat convection coefficient versus the vehicle speed can be stated by the following relation:

$$h = CV^{0.8} \tag{20}$$

where, V is the vehicle speed and C is a constant. Furthermore measured experimental data and numerical values of the maximum contact surface temperature of the rotor for three different vehicle speed, is compared with each other. As it can be seen at higher vehicle speeds more heat is generated and consequently surface temperature increases. However the difference between measured data and numerical values of surface temperature shows that the assumed constant cylinder pressure is more realistic for higher speeds.

In many studies (Dufrénoy, 2004; Majcherczak *et al*., 2005; Gotowicki *et al*., 2005), the heat generated due to friction was calculated using macroscopic model. Whereas in this study true representation of heat generation during braking action within the rotors and the pad, is investigated using macroscopic and microscopic model. The amounts of heat generated due to friction by these two models are in a good agreement. Using microscopic model help us to better understand the effect of parameters such as the duration of braking, velocity of the vehicle and it's variation with time, dimensions and geometry of the brake system, material of the brake rotor and the pad and pressure distribution in the contact zone. Furthermore the heat flux calculated in these studies is considered only as a function of time. However, in this paper the heat flux calculated as a function of time and space variable.

CONCLUSION

As mentioned earlier, the brake rotor must serve as an efficient energy dissipation and storage device. In order to achieve this purpose, air must be circulated through the rotor to provide adequate cooling. The passages, formed by the radial fins between the braking surfaces, act as a centrifugal fan, facilitating the required air flow for cooling.

The use of an internally ventilated rotor will increase both surface area (extra internal area exposed to the atmosphere) and the heat transfer coefficient, due to forced convection created by the internal airflow, with negligible influence on unsprung mass.

A common problem that may happen to disk brake system is that after several braking when the car is parked, there's no air flow in the wheel assembly and convection heat flow to the environment is via natural convection. But as mentioned earlier while conduction is an effective mode of heat transfer it can have adverse effects on nearby components. One of these problems is brake fluid vaporization that happened because of heat soaking phenomena and is intensified when the vehicle is parked. So it is recommended that a brake fluid with appropriate DOT rating is used.

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