THREE-DIMENSIONAL SIMULATION OF THERMAL STRESSES IN DISCS DURING AN AUTOMOTIVE BRAKING CYCLE

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Abstract: *In this study, a 3-Dimensional finite element simulation of a braking cycle including braking and cooling steps is presented. In order to induce thermal stresses, thermo-mechanical material behaviour and interactions between pad and discs are considered in the simulation. The results reveal that compressive and tensile stresses are happening in the braking and cooling steps respectively. Cyclic tensile stresses in the disc of heavy trucks would lead to the initiation of superficial radial cracks in them. The occurrence of hot spot phenomenon in the discs are also observed and discussed thoroughly. The proposed model could be utilized to estimate fatigue life of the braking discs in the automobiles and heavy vehicles..*

Keywords: AUTOMOTIVE BRAKING, FINITE ELEMENT METHOD, THERMAL STRESS

1. Introduction

The components of duty vehicles such as trucks, due to their heavy weights endure large mechanical loads during their life-span, hence a proper maintenance and repair of them is essential. Brake disks, for instance, goes thorough very large braking loads which consequently shorten its life-time. Due to the friction on the contact surfaces of the pad and disc, kinetic energy of vehicle transforms into heat[1]. A great deal of the heat is transferred to the brake system components. Therefore, the components of brake system would be exposed to high temperatures, leading to less favorable conditions for brake system such as reduction of friction coefficient, plastic deformation of disc, and thermal cracking in disc surface which subsequently weakens braking performance. Moreover, variations of stresses and strains during the cyclic heating and cooling cause the initiation and growth of cracks on the surface of brake discs[2]. Hence, calculations of temperatures and thermal stresses during a braking cycle are very important for design and improve of brake system[3].

Due to the overheating in the friction surfaces, deformation occurs in the discs and results in the change of contact state, so it leads to a local temperature increase (which is known as hot spots and hot bands in the literature)[4]. Therefore, in order to have more accurate predictions, the temperature and stress fields must be considered simultaneously. Fig. 1 shows a sample of thermal cracks on the brake disc surface after several hard braking cycles.

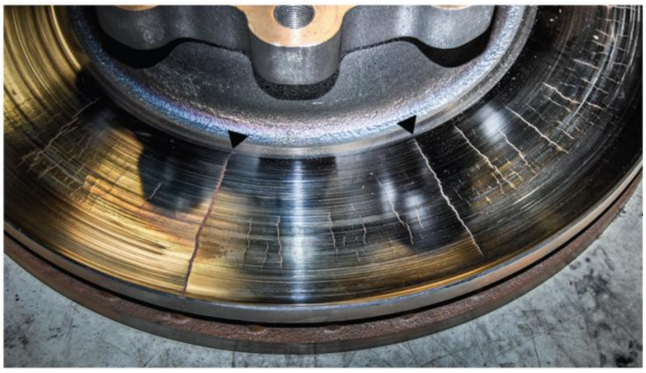


Fig. 1 *Radial thermal cracks due to hard braking in heavy vehicles[1].*

In order to solve the coupled field problems like temperature and stress in disc brakes, generally there are two ways: sequential coupling and direct coupling. Most of previous studies considered friction heat as a prior known heat flow over the contact surface, so as to save on computational costs and avoid complexity in the simulation. For example, Gigan[5] studied a non-rotational heavy vehicle brake disc with a non-uniform heat flow distribution over the surfaces. However, their results were slightly different from the experimental observations. Some researchers are also using three-dimensional models to simulate uncoupled temperature and stress fields, resulting in less accurate solutions due to the nonlinearity and large displacements during hard braking[6].

The explicit coupled temperature-displacement algorithm is used in this paper, as the closest module to the real situation in hard braking.

2. Simulation

Geometry: a straight vaned type of commercial heavy vehicle brake disc is chosen to be studied here which consists of a hat, a neck, and two friction surfaces. Simplified cross-section model of a commercial truck brake disc is shown in Fig. 2. It has 36 straight vanes, each acquire an angle of 3 degrees, and spaced every 7 degrees in a 360 degrees cycle. Total thickness of discs is 48mm which includes two frictional surfaces along with a 18mm space for ventilation. Outer radius and inner radius of discs are respectively 218mm and 128mm. Each radial vane has a 77mm length. Overall dimensions of the disc and pad are summarized in Table 1.

Table 1: *Overall dimensions of brake disc and pad*

Item	*Disc[mm]*	*Pad[mm]*
Height	45	15
Inner radius	128	134.5
Outer radius	218	211.5

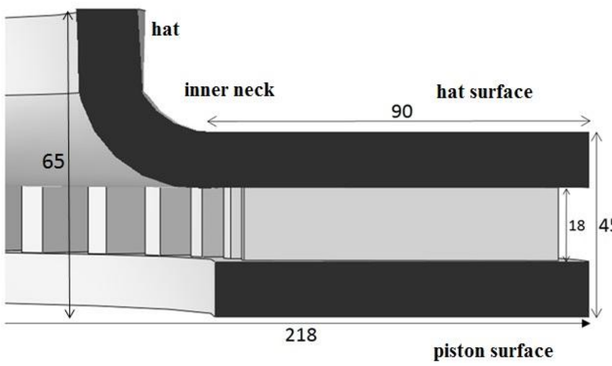


Fig. 2 *Cross-section of a brake disc.*

A longitudinal cross-section of brake pad and its dimensions are shown in Fig. 3. The pad is covering 60 degree of disc surface. Modelling of brake disc associated with brake pad is carried out in ABAQUS finite element commercial software.

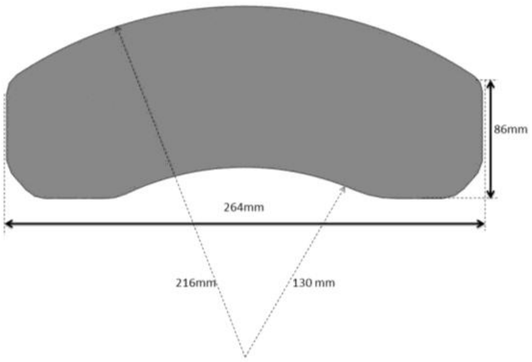


Fig. 3 *Brake pad geometries*

Mesh: A sample of mesh generation for the assembled brake disc and pads is shown in Fig. 4. The utilized element for the mesh is C3D8T type with a size of 3-4mm.

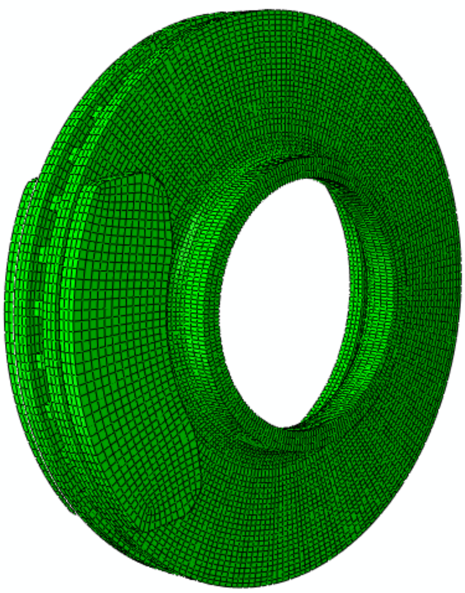


Fig. 4 *A sample of mesh generation for brake disc and pads*

Material model: Thermal properties of the pad, made from organic friction materials, is given in Table 2[3].

Table 2: *Brake pad material properties*[3]

Item	*Brake pad*
Young's modulus(Gpa)	28
Poisson's ratio	0.29
Density(Kg/m^3)	2700
Conductivity	2.36
Specific heat	4000

Thermal diffusivity and the thermal conductivity are important material parameters for heat transfer in brake discs. Brake rotors in trucks, compared to passenger cars, are subjected to larger amounts of stresses, so they require materials with high thermal fatigue strength. Besides, high strength and durability is necessary to withstand high torque loads during braking. On the other hand, high thermal conductivity helps quick transportation of frictional heat away from the frictional surfaces. The studied brake disc is made of grey cast iron with a pearlitic microstructure. Grey cast iron is widely used as truck brake disc material because of its superior thermal fatigue strength, low squeal and good wear resistance which is important for a safe performance during its life time. Table 3 shows temperature-dependent parameters of brake disc material[4].

Table 3: *temperature-dependent parameters of brake disc material*[4]

Parameter	25°C	100°C	200°C	300°C	400°C	500°C
Young's modulus[GPa]	101	98.5	96.8	96.3	81.3	80.3
Poisson's ratio	0.3	0.3	0.3	0.3	0.3	0.3
Density[Kg/m^3]	7293	7272	7243	7213	7182	7152
Conductivity [W/m.°C]	53.2	51.3	47.1	42.9	39.1	36.2
Specific heat [J]	488	532	563	599	631	669
Expansion [1/°C]	1.22e-5	1.26e-5	1.31e-5	1.37e-5	1.39e-5	1.4e-5

Plastic hardening model used in this paper is a non-linear kinematic, isotropic one. Table 4 shows material constants for the prediction of plastic hardening in the brake disc material.

Table 4: *Material constants of plastic hardening model for brake disc material*

Temperature	C_1[GPa]	$Gamma_1$	C_2[GPa]	$Gamma_2$
25	98.5	627	368	9220
300	95.5	607	361	9390
400	79.5	546	341	9690
500	60.2	474	319	101000
600	41.5	417	270	111000

Boundary Conditions: Different types of braking are defined, three types of which are explained here. In “stop braking”, the initial angular velocity of the brake disc is a predefined value and would be reduced to zero through a constant or non-constant deceleration. Another type is the ordinary braking, known as “snub braking”, where speed is decreased to a certain value (for safety reasons, e.g. speed control limits). The last one, “drag braking”, also known as “downhill braking”, is the use of brakes for preserving a specific speed while moving downwards a hill.

The third type is the one used in this paper due to its frequent repetition in heavy trucks. Moreover, initial temperature of the brake disc and pads is assumed to be 25°C. The speed of vehicle is assumed as 80 km/h which results in a constant angular velocity of 425r/min (44.46 rad/s) on brake discs during 10 seconds of braking cycle. A brake cycle is composed of two parts: A 10 seconds of uniform braking, and a cooling phase assumed to be 300 seconds (5 minutes). The following assumptions are made in the modelling of brake rotor in the finite element simulation:

1- Brake pressure is uniformly distributed all over the back side of the pads.
2- Friction coefficient remains constant during a braking cycle.

Table 5 illustrates braking parameters used in this study. A schematics of prescribed boundary conditions are also depicted in Fig. 5.

Table 5: *Braking parameters*

Braking time (s)	10
Braking pressure (kPa)	1000
Disc brake rotational velocity (rad/s)	44.46

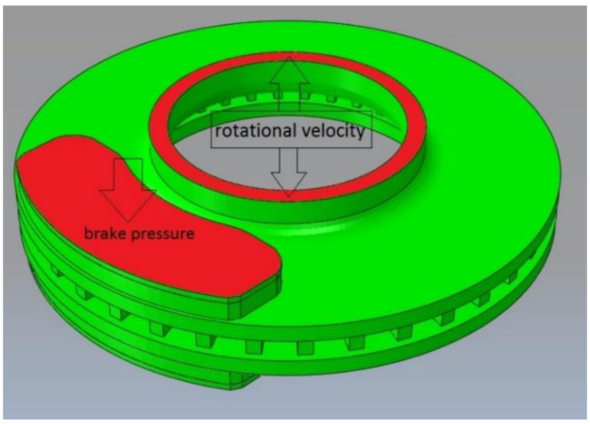


Fig. 5 *Schematics of prescribed boundary conditions*

In order to simulate a more accurate solution for the cooling process of the brake system components, both convection and radiation are accounted. Convection coefficient is computed in [7] for various brake disc velocities. Fig. 6 illustrates heat transfer coefficient at brake disc surfaces in terms of temperature, while the vehicle has an 80 km/h speed. As can be seen, convection coefficient increases gradually by the rise of temperature.

Since temperature is different on various sides of disc surface, for gaining a more reliable solution, the surface of disc is partitioned to three sub-region; each has a different value of convection coefficient (see Fig. 7). In order to use a constant coefficient value on both sides of the disc, Bio number must be less than 0.1.

Bio number is determined as $Bio=h*L/k$, where h is the heat transfer coefficient, L is the characteristic length (or thickness of frictional layer), and k is the thermal conductivity of the material.

Using the thickness of each piece of brake disc (L=13.5mm) and temperature-dependent values of h and k, it would be determined that for a temperature range of [25°C-800°C], Bio number would be less than 0.1, thus a constant value of convection heat transfer coefficient can be used for solving the cooling phase of brake cycle.

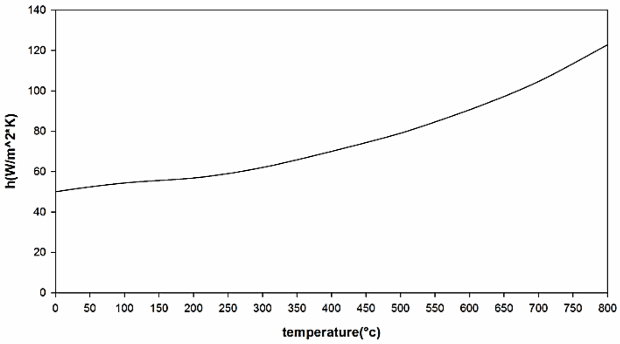


Fig. 6 *The variation of convection heat transfer coefficient in terms of temperature at 80km/h*

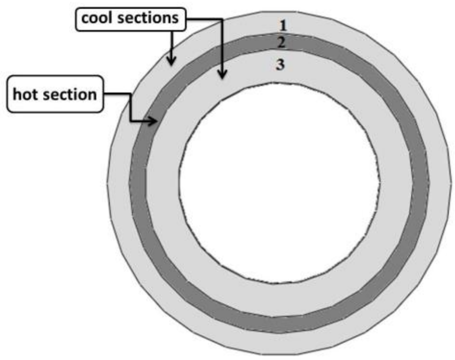
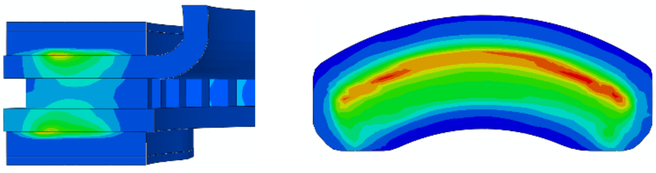


Fig. 7 *Three defined regions of brake disc surface for convection*

3. Results

Variation of stresses in braking and cooling phases of a brake disc would be presented and discussed in this section. Fig. 8 (a) shows the temperature distribution of front and back surfaces of brake disc during braking cycle. As can be seen, the temperature field is not symmetrical on both sides of the disc. Maximum temperature of the surfaces appears to happen in the inner ring of the frictional region, while maximum temperature of the back frictional surface occurs in the outer ring. Due to the cross-displacement of the neck and hub, known as coning, the contact conditions on both sides of the rotor would not be similar to each other. In addition, non-friction region almost has no temperature elevation. As can be seen in Figure 8 (b), it is clear that temperature distribution on the surface is non-constant, so that a non-uniform wear occurs on the pad and leads to an overheat on the pads.



(a) (b)

Fig. 8 *Temperature distribution on a) a cross section of brake disc and pads at the end of braking (10s) and b) the surface of pad at the end of braking (t=10s)*

Fig. 9 shows a cross-section of disc in addition to a series of points in radial direction, named as path-1. Temperature diagram over path-1 at different moments, i.e. 2, 4, 6, 8, and 10s, is shown in Fig. 10. As can be seen, initially the temperature in all points is equal to 25°C (room temperature); during the braking, it rises non-uniformly. Hence, the temperature on a specific radius of the brake disc surface would be maximum and is called a hot band in which it is most likely for fatigue cracks initiation over the surface of brake discs. Schematics of hot bands formation according to the pass of time is shown in Fig. 11.

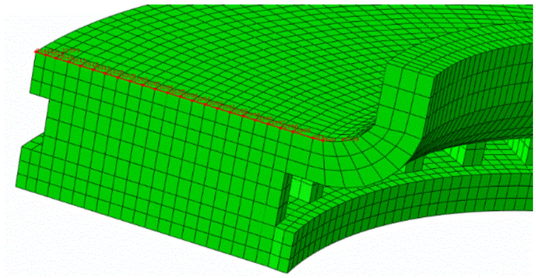


Fig. 9 *Cross-section of brake disc and the definition of path-1 (red dots)*

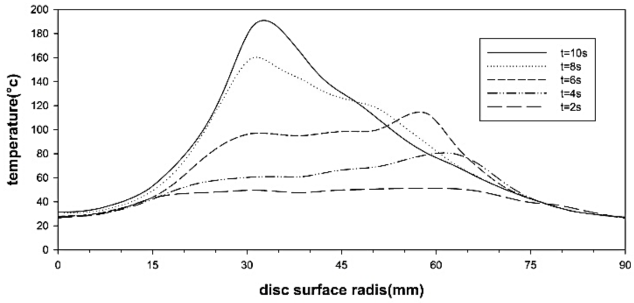


Fig. 10 *Temperature diagram over the surface path-1 at t=2,4,6,8,10s*

Circumferential stresses, also called hoop stresses, are the main component of developed thermal stresses, the variations of which during braking and cooling cycles would be the major cause in the initiation and propagation of surface radial cracks. As can be seen in Fig. 12, at the first couple of seconds in braking cycle, the distribution of circumferential stresses is relatively uniform, due to the uniform distribution of braking pressure and temperature over the frictional surfaces. However, over the time, coning on the hat of the disc leads to a non-uniform distribution, creating hot bands and hotspots on the surface. After 6s of braking, a hot band of frictional heat causes an intense change in the hoop stresses.

Distribution of circumferential stresses on the upper surface of brake discs is shown in Fig. 13. As can be seen, while at 10s the highest compressive stresses have been occurred in the location of hot band, proceeding to the cooling cycle (t=300s), the same place undergoes the highest tensions. This phenomena results in the initiation of cracks in the hot bands.

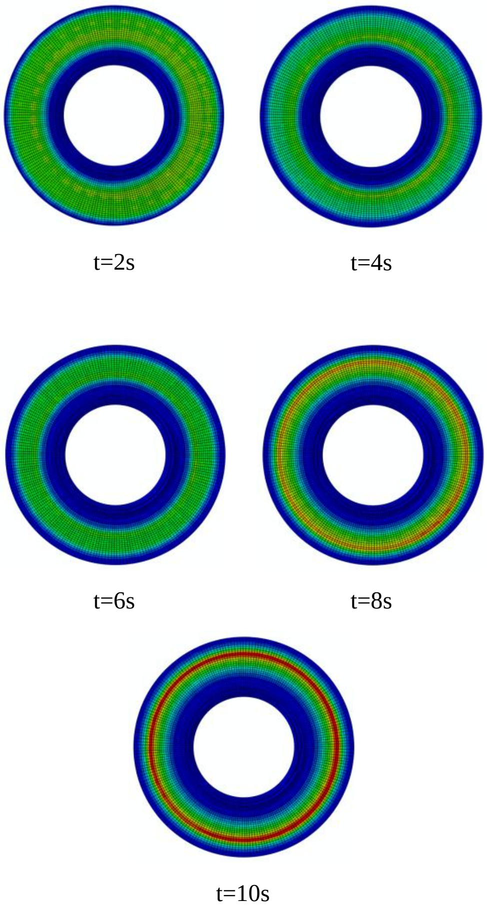


Fig. 11 *Schematics of hot bands formation by passing the time (hot bands are depicted in red)*

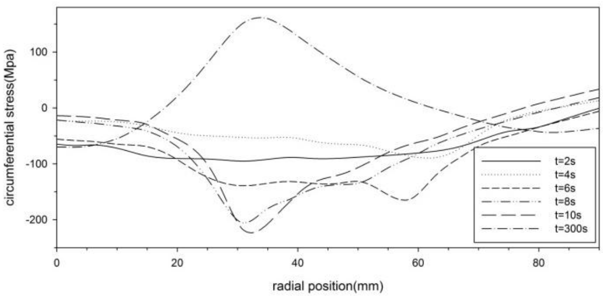


Fig. 12 *Circumferential (hoop) stresses over the path-1 at t=2,4,6,8,10s*

4. Conclusion

In this paper, a three-dimensional simulation of a braking cycle (including braking and cooling) carried out which causes high circumferential cyclic thermal stresses in brake discs. The simulation identified that the interaction of frictional heat generation, thermal distortion, and effect of coning in brake disc would create a “hot band”, which occurs at the working-surface of the disk. Moreover, the results show that critical stresses also occurred in the hot band. Therefore, the model used in this paper can be further used for examination of the geometry and material developments as well as fatigue analysis.

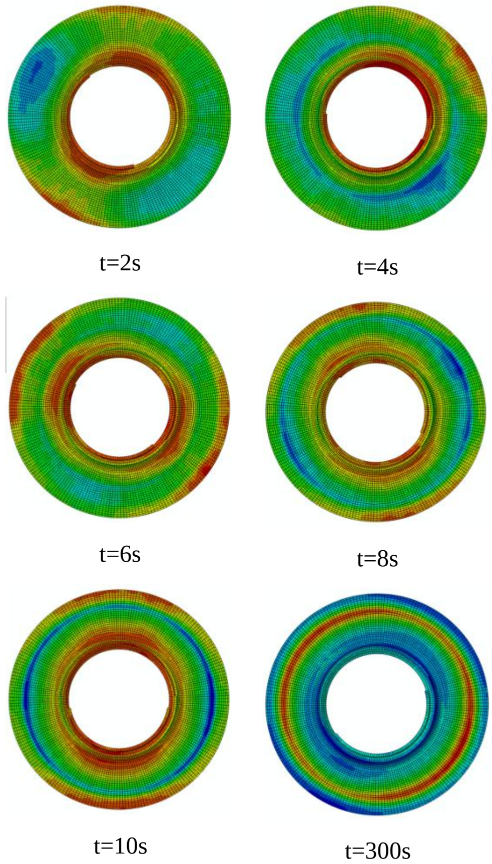


Fig. 13 *Circumferential stresses for brake disc surface at t=4, 6, 8, 10, 300s*

5. Referencs

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