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A Numerical Study on the Thermo-Mechanical Instabilities of Heavy Vehicle Brake Disc

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A B S T R A C T

The primary objective of a brake disc is to absorb frictional heat during braking and dissipated it immediately by convection and radiation. However, during hard and repetitive brakings, thermal coning on brake disc generates surface hot spots which are responsible for the undesired accumulation of compressive stresses on the surface of the brake disc. These stresses would lead to disc cracking and finally failure of it. In the current paper, a coupled transient thermo-mechanical FE analysis of a heavy vehicle braking system is carried out in a way that thermal coning of the disc and surface hot spots and bands are recognizable. Braking condition is chosen from a standard for hard braking in trucks. Moreover, five additional braking actions with different severities are investigated to study the effects of braking severity on thermo-mechanical instability of brake discs. Comparison of numerical results of transient temperature during braking and cooling phases with experiment reveal a high accuracy of thermal prediction of this model. Also, the results show that thermal coning of brake disc is varied between 0.05 to 0.7 mm depending on braking severity and tangential location of the disc. Additionally, surface hot spots experience higher temperature gradients in higher decelerations. Finally, results show that circumferential compressive stresses during braking are the major component of thermal stresses and should be taken into account for life estimation analysis.

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1. Introduction

Brake system is a vital component of any vehicle. Due to the frictional force on the contact surfaces of pads and disc, the speed of the vehicle can decrease or reach to zero. During this process, virtually all of the kinetic energy of the vehicle transforms into heat.[1] A great deal of this heat is carried to the brake disc.[2] Therefore, this part of the brake system would be exposed to high temperatures, leading to less favorable conditions for brake system such as friction coefficient reduction, premature wear, thermoelastic instability and thermal cracking in the surface of the disc which finally weakens braking performance and leads to failure of the disc.[3] Understanding these problems is essential since the

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braking system is directly related to the safety of passengers.

Thermal coning is a displacement happening in the brake disc which deforms the natural constant and uniform contact between disc and pads, making an uneven pressure distribution along the radial and circumferential directions. This non-uniformity in braking pressure would lead in thermo-mechanical instability on the surface of brake disc including non-uniform temperature and stress distributions.[4] If the thermal coning emerges non-uniformly at the contact surface, a concentration of pressure is observed at the area with higher deformations. Higher pressure generates high temperature in the surface in which increases the thermal expansion in the area.[5] Consequently, it is necessary to accurately understand this phenomenon in order to improve the designing quality and reduce the risks of unanticipated failure. An image of temperature distribution on a heavy vehicle brake disc surface, taken by a thermal camera during hard braking is shown in Figure 1 [6]. Apparently, a few areas of surface namely hot spots, gain in relatively higher temperatures that cause damage to the material of brake disc and the performance of the braking system. In what follows, Numerical and experimental efforts in literature to study the thermo-mechanical instability of brake disc are discussed.

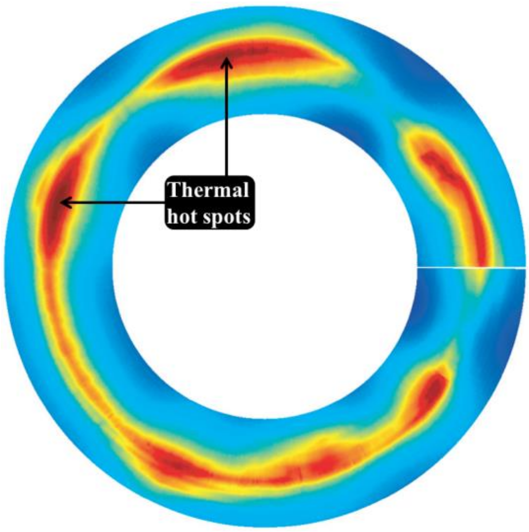

Figure 1. Surface temperature image of a heavy vehicle brake disc, taken by a thermal camera during a hard braking action[6]

It has been declared by Lee[7] that variable distribution of frictional heat on the surface of brake disc would lead to an unfavorable deformation on it. This deformation also causes friction loss and cracking in the brake disc material. Consequently, thermo-mechanical fatigue on the surface of brake disc is the primary failure mode in the brake system.[8]

Valvano and Lee[9] proposed a technique to evaluate the distortion of the temperature of the brake disc. In this uncoupled method, thermal parameters were calculated at first and then, were imported to FE software as input to determine thermal stress distribution. Using this method, they were capable of including both heat input and cooling aspects of a given brake system. Koetniyom [10] concluded from a thermal analysis of a brake disc under hard braking actions that geometrical characteristics of the disc determine the thermal features of the brake system including the maximum temperature and overall efficiency. Akhtar et al. [11] studied The effects of sliding speed on the brake disc on braking pressure and temperature distribution and also on the generated heat flux on the frictional surfaces. Tiwari et al.[12] conducted a transient analysis on brake disc in order to determine the thermal stress and displacement variations across the radial direction of the frictional surface. Manjunath and Suresh [13] examined the performance of the disc for solid and ventilated shapes by a thermal and structural analysis in the FE method. They evaluated thermal stress and deformation of the disc and concluded that ventilated discs hold greater overall performance and have a higher working life. Parab et al. [14] carried out thermal and structural analysis on cast iron, stainless steel, and carbon-carbon composite brake discs. The structural analysis aimed to assess the strength of the disc brake, and thermal analysis was carried to examine the effects of excessive temperature on the physical behavior of brake disc.

Distribution of normal pressure has a significant role on temperature and thermal stress and generally thermoelastic instability of brake disc. Geometric dimensions of the disc, number of pistons and the shape of the cross-section of the disc are some of the factors affecting thermo-mechanical behavior of the brake system. Day et al. [15] analyzed the effects of braking pressure distribution on temperature distribution, wear, banding, hot spots and surface damage of brake disc which are representative to thermoelastic instability. They took advantage of a two dimensional axis-symmetric model of a brake disc and found that in order to minimize these effects, pressure distribution should be as uniform as possible. Cho et al. [16, 17] carried out numerical and experimental studies on thermo-mechanical instability of disc to find the optimum design of disc-pad assembly in means of the thickness of the

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disc. They achieved that, both thermal and structural aspects of the brake disc should be considered to get the optimum design of the brake system. They also found that the dimensions and shape of the pad and the type of pressurization had no notable effects on structural or thermal behaviors of the disc. Zagrodzki [18] studied the temperature distribution, thermal fatigue, and distortion on repeated braking cycles for a train brake disc and concluded that non-uniform distribution of temperatures would generate coning of brake disc which is dependent to temperature. Okamura and Yumoto [19] concluded that thermal disc coning in ventilated discs could be reduced by increasing the bending stiffness of it. That is achieved by increasing the number of vanes and reducing the bending stiffness of the hat of the disc.

In this paper, a three dimensional model of a heavy vehicle ventilated brake system consisted of a brake disc, two brake pads, and two brake pad protectors are numerically analyzed using an explicit coupled thermo-mechanical method. Geometries and materials properties of disc and pads are chosen from literature, corresponding to what is currently being used in commercial trucks. Using appropriate mechanical and thermal boundary conditions, this model is capable of predicting the generating hot spots and hot bands during brakings, which are essential to consider if precise thermo-mechanical results are needed.

2. Theoretical Background

2.1. Transient thermal-structural coupling

The temperature of the object in transient heat transfer is varying over time to approach in equilibrium. In the case of the brake disc and brake pad, the contact friction heating of the brake is equal to a transient variation of the moving heat source. Accordingly, the temperature filed can be solved through the transient thermal analysis method. The transient thermal method is expressed as follows[20]:

$$\rho c\left(\frac{\partial T}{\partial t}+\{v\}^T\{L\}T\right)=\{L\}^T[D]\{L\}T+q \qquad (1)$$

where:

$$\begin{cases}\{L\}^T=\left[\frac{\partial}{\partial x},\frac{\partial}{\partial y},\frac{\partial}{\partial z}\right]\\ \{v\}^T=\left[v_x,v_y,v_z\right]\\ [D]=\begin{bmatrix}K_{xx}&0&0\\0&K_{yy}&0\\0&0&K_{zz}\end{bmatrix}\end{cases}$$

$\{L\}^T$ is vector operator, $\{V\}^T$ is heat velocity vector, and K_{xx} , K_{yy} and K_{zz} represent heat conduction coefficient respectively in directions x, y and z. {q} is heat flux vector per volume unit and q is heat generation rate, equal to zero in this problem. By elaborating Equation 1, transient heat conduction is computed as:

$$\rho C_p\left(\frac{\partial T}{\partial t}+v_x\frac{\partial T}{\partial x}+v_y\frac{\partial T}{\partial y}+v_z\frac{\partial T}{\partial z}\right)= \frac{\partial}{\partial x}\left(k_x\frac{\partial T}{\partial x}\right)+\frac{\partial}{\partial y}\left(k_y\frac{\partial T}{\partial y}\right)+\frac{\partial}{\partial z}(k_z\frac{\partial T}{\partial z}) \qquad (2)$$

During the braking, due to the unsteady essence of the process, temperature and stress equations are interdependent. A variation in temperature leads to structural deformations that in turn, this deformation changes the thermal boundary conditions of the problem. Coupling method solves the thermal-structural equations of the material simultaneously. In the transient method, heat transfer equations are combined by forward difference time integration rule:

$$T^x_{(i+1)}=T^x_{(i)}+\Delta t_{(i+1)}\dot{T}^x_{(i)} \qquad (3)$$

where T^x is the temperature at node x and the subscript, "i" refers to the increment number of analysis. In the explicit forward difference time integration, the current temperatures are obtained using known values of lumped capacitance matrix $\dot{T}^x_{(i)}$ from the previous increment. In the coupled method, calculation of the temperature and thermal stresses fields are iterative. In the coupled thermal-structural analysis, the energy equation is[20]:

$$\int_V\left\{\rho\left(Q-\frac{\partial U}{\partial t}\right)+\sigma_{ij}\frac{\partial V_i}{x_j}\right\}dV = \int_S HdS \qquad (4)$$

which V is the volume of continuous medium, U is internal energy; H is boundary heat flux density, and σ_{ij} is Cauchy stress component.

The equation of the transient structural temperature and thermal stress and strain field used in FE analysis is as[20]:

$$\dot{u}^T(K_u\dot{u}(t)+M_T\dot{T}(t)-F(t))=0 \qquad (5)$$

$$T^T\left(C_u\dot{T}(t)+M_u\dot{u}(t)-D-Q-K_TT(t)\right)=0 \qquad (6)$$

that C_u is heat capacity matrix, K_T is heat conduction matrix, M_u is thermo-mechanical

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coupling matrix, D is dissipative vector and Q is thermal load vector.

3. Simulation

Numerical analysis of a brake system is carried out in ABAQUS software using a 3D transient coupled thermo-mechanical analysis. Since temperature and thermal stress distributions vary along the radial and circumferential directions, all the thermal-structural results should be mapped on all parts of the disc.

3.1. Geometry and boundary conditions

A commercial heavy vehicle brake system including a brake disc and two pads are chosen to be studied. Brake disc consists of a hat, a neck, and two frictional surfaces. Ventilation area connects the frictional surfaces using 36 straight vanes; each of them covers an angle of 3 degrees of the surface. The outer and inner radiuses of discs are respectively 218 mm and 128 mm. A cross-section of the brake disc and pads along with their dimensions are shown in Figure 2. The pad covers 60 degrees of the surface of the disc in each side.

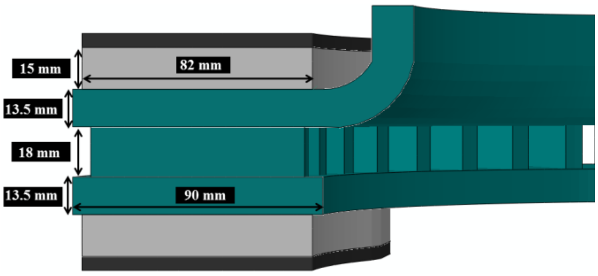

Figure 2. A cross-section of brake disc and pads along with necessary geometric dimensions

A hard stop braking is defined using the European standard in braking for the heavy vehicles[21]. In "stop braking", the initial velocity of the disc is a predefined value, and the vehicle should entirely stop in a defined time (t_s). Initial velocity of the vehicle is assumed to be 60 km/h and braking time is 6.8s with a constant deceleration of 5.20 rad/s^2 for the disc. Braking cycle consists of a braking phase which disc and pads are sliding on each other and frictional heat is generated, and a 200 s cooling that disc is assumed to be motionless and convection heat transfer are cooling the disc down. Additionally, 5 braking conditions with 25% and 10% higher and 10%, 25% and 50% lower decelerations are defined and analyzed to study the effects of braking severity on thermo-mechanical behavior of the brake disc. In all braking conditions, initial velocity assumed to be the same, but braking pressure and braking time varied in each case. It is shown to be an inverse linear relationship between deceleration value and braking pressure in brake discs.

$$P \approx \ddot{\theta} \quad (7)$$

Table 1 demonstrates the conditions of these braking cases.

Table 1. Braking conditions for different cases

Braking case	Braking Time (t_s)	Normal Pressure (P)
25% faster	5.44s	1.25 MPa
10% faster	6.18s	1.1 MPa
main case	6.8s	1 MPa
10% slower	7.56s	0.90 MPa
25% slower	9.07s	0.75 MPa
50% slower	13.61s	0.50 MPa

The following assumptions are also considered in this model:

1- Brake pressure is applied uniformly to a solid shim on the back side of the pad and is assumed to be 1 MPa for the main case.

2- Coefficient of friction remains constant during a braking cycle.

3- The initial temperature of the brake disc and pads is assumed to be 80°C to simulate the real working condition of a brake system.

4- Kinetic contact method is used to simulate the frictional contact between disc and pads.

Furthermore, explicit thermal-displacement meshes with a size of 3 mm for disc and 4 mm for brake pad is utilized. Figure 4 illustrates the assembly of the brake system and mechanical boundary conditions of brake disc and pads. According to Figure 3, the neck part of the disc can only rotate on the z-axis, and pads can move on this direction.

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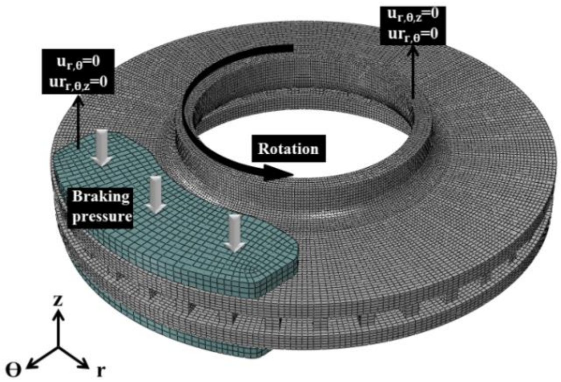

Figure 3. Meshed view of brake system and mechanical boundary conditions on the disc and pads

To obtain a more accurate thermal and structural result for the model, convection heat transfer in brake disc is taken into account. The coefficient of convection for brake disc is being computed in [22] for a variety of rotational velocities and surface temperatures. All of the external surfaces of the brake disc are considered to have a convection heat transfer with the surrounding air, and the values of heat transfer coefficient were used to calculate the emitted heat flux from surfaces. Figure 4 shows temperature and velocity dependent values of coefficient of convection, extracted from[22].

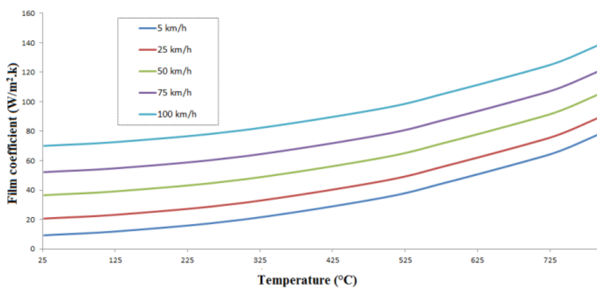

Figure 4. Coefficient of convection heat transfer used for cooling prediction of brake disc.[22]

In order to apply the same coefficient of convection value on both sides of the disc, The Biot number should be less than 0.1. Biot number is defined as:

$$Biot=h*L/k \tag{8}$$

which "h" is the coefficient of convection heat transfer, "L" is the characteristic length that is the thickness of the frictional layer in this model, and "k" is the thermal conductivity of the material (grey cast iron in this work). Considering the thickness of frictional parts to be 13.5mm and "k" and "h" values for the grey cast iron, it is determined that for a temperature range of 25-500 °C, Biot number remains lower than 0.1.

3.2. Material

Thermal conductivity and diffusivity are determining parameters of a material for a good heat transfer. These parameters are essential for commercial vehicle brake discs. Brake discs in trucks, compared to passenger cars, are exposed to the more substantial amounts of thermal stresses, making them require materials with high thermal strength. Durability and High strength are also parameters needed for brake rotors since these parts should withstand high torque loads (due to the mass of vehicle) during the braking process. Grey cast iron is extensively used as the material for manufacturing truck brake disc. Superior thermal strength, low squeal and excellent wear resistance along with low costs of production make it an appropriate choice of mass production. Grey cast iron grade GG-25 is a favorite alloy for its high strength and thermal conductivity and is used as the material of brake disc for the current study. Table 2 presents the temperature-dependent parameters of GG-25 grey cast iron.

Table 2. Temperature-dependent thermal parameters of GG-25 grey cast iron[23]

Parameter	25°C	100°C	200°C	300°C	400°C	500°C
E[GPa]	101	98.5	96.8	96.3	81.3	80.3
ρ[kg/m³]	7293	7272	7243	7213	7182	7152
k[W/m.°C]	53.2	51.3	47.1	42.9	39.1	36.2
c[J]	488	532	563	599	631	669
α_V[1/°C]*10^6	12.2	12.6	13.1	13.7	13.9	14
υ	0.3	0.3	0.28	0.28	0.28	0.28

Cast irons exhibit non-isotropic behavior during tension and compression, in a way that ultimate strength is relatively higher under compressive loadings. Consequently, the plasticity model of cast iron should be taken into account to obtain higher accuracy in thermal stress results. Stress-strain behavior of GG-25 cast iron under tensile and compressive loadings is shown in Table 2. These parameters are used in the FE model to calculate the response of brake disc for variable thermal loadings during braking and cooling phases.

Table 3. Mechanical behavior of grey cast iron in compression and tensile loading[24]

Strain (%)	Compressive stress (MPa)	Tensile stress (MPa)

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0.1	102.2	102.2
0.2	207	153.1
0.3	280	188.5
0.4	348	208
0.5	387	224
0.6	422	237
1	487	269
1.5	536	282
2	566	296

Brake pad is made of organic frictional materials and its thermal parameters are imported form literature[23], as shown in Table 4.

Table 4. Material properties for brake pad[23]

Parameter	**E [GPa]**	**υ**	**ρ[kg/m^3]**	**k[W/m.°C]**	**c[J]**
value	28	0.29	2700	2.36	4000

4. Results

In this section, results from analysis consisted of coning value, temperature, and thermal stress distributions are studied, and the effects of severity in braking are being investigated.

4.1. Disc coning

As mentioned earlier, brake coning is a longitudinal deformation on brake disc stem from rotational velocity of the disc and uneven distribution of temperatures on the disc surface. Figure 5 illustrates the longitudinal displacement amount on the brake disc at the middle of braking for the main case. According to this figure, maximum displacement happens at the side without brake pad, and instead, in the brake pad side, the displacement is almost zero. That is because the braking pressure presses the disc from both sides. This non-equal amount of deformation on different sides of the surface makes an abnormal contact between disc and pads that generates hot spots and hot bands on the surface of the brake disc.

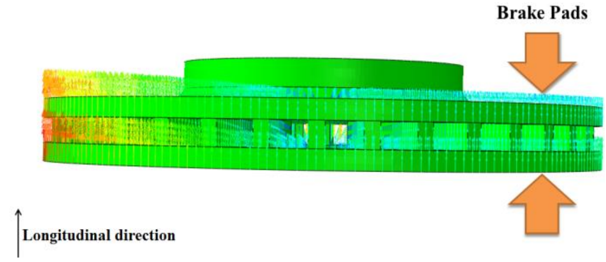

Figure 5. Longitudinal deformation of brake disc surface (disc coning). The side with pads has lower displacement due to brake pressure, instead the other side owns highest vertical deformation

Longitudinal deformation values for all the points around the outer edge of the disc for the different braking cases are shown in Figure 6 at the middle of the braking process. Clearly, in the case with 25% more deceleration, the highest displacement happens at the side without pads equal to 0.6 mm, while this deformation is around 0.2 mm in the brake pad side. In all cases, deformation on the pad side is almost 1/5 of the other side. Also, the deformation amount decreases non-linearly by decreasing the deceleration.

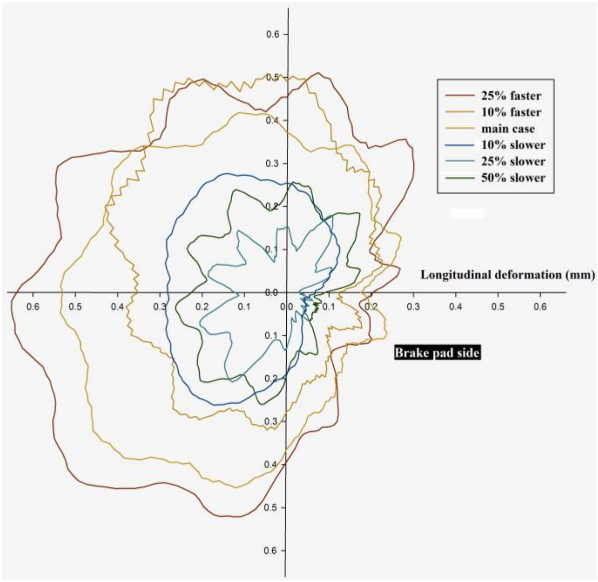

Figure 6. Thermal coning of brake disc in angular direction for different deceleration values at the middle of braking

This deformation determines the amount of pressure intensity at the surface of the disc and as a result, the quality of temperature and thermal stress distribution. In Figure 7 the maximum amount of longitudinal deformation is specified for each braking case. A non-linear correlation can be seen between the deceleration and displacement value. For example, in the 50% slower case, displacement of the disc is equal to 0.378 mm, while in the case with 25% lower deceleration this value is equal to 0.407 mm. This phenomenon stems from the fact

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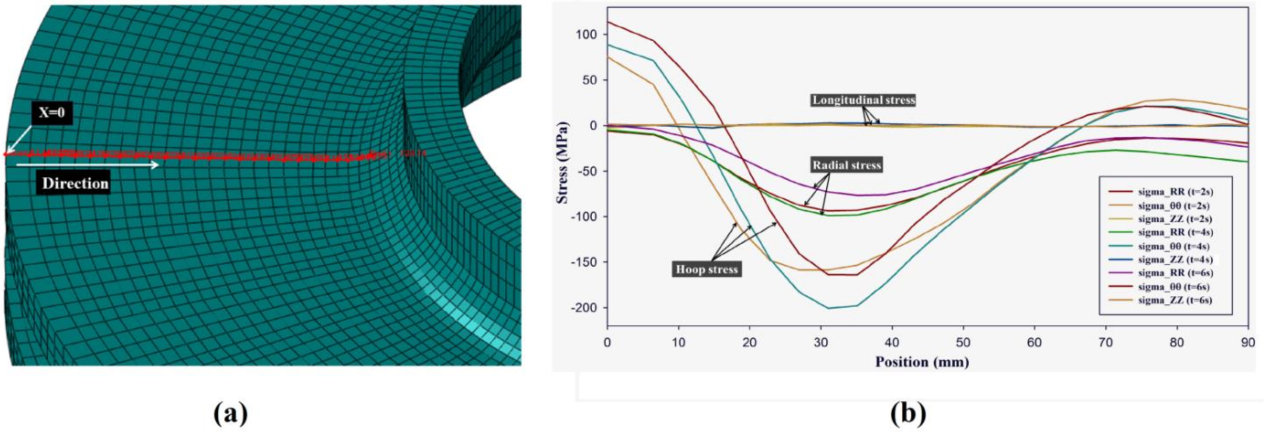

Figure 8. A 90 mm path consisted of surface nodes (a) and thermal stress distribution along the path on a cylindrical coordinate system (b) at t= 2, 4, 6 s

that however in the 50% slower case, the velocity gradient is relatively lower and as a result; the heat flux is lower since in this case, the braking time is longer, total heat applied to the brake disc is more too. Consequently, the deformation of the brake disc is almost the same is these cases.

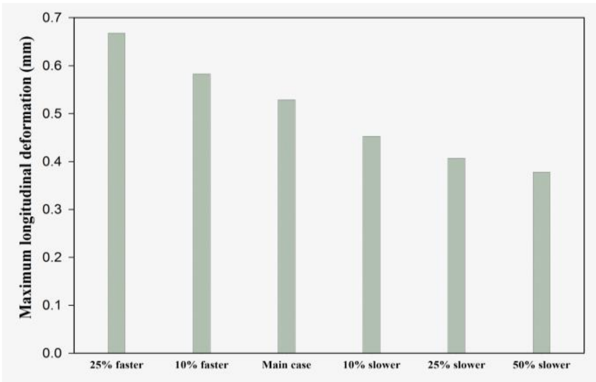

Figure 7. Maximum longitudinal deformation of braking cases

4.2. Temperature and thermal stresses

Results of FE analysis consists of thermal-structural changes during the braking process for brake disc and pads. Temperature diagram of a 2 mm depth point of the disc through the braking and cooling phases was generated from the analysis and compared to experimental results from Collignon et al. [24]. Results revealed a good accuracy of the current model on predicting the temperature distribution. The maximum temperature of the numerical model was 292°C compared to 286°C of experiments. Then, generated stresses were investigated on a cylindrical coordinate system to find the most critical component of thermal stresses. Figure 8 shows a surface path along the radial direction of the disc. Thermal stress components on the nodes of this surface are extracted and can be shown in Figure 8-b for times t= 2,4 and 6s during the braking action. It can be seen that in each time, compressive circumferential stress component are relatively higher than the radial or z-direction stresses. Longitudinal stress is less than 10 MPa and is stem from normal braking pressure applied to the frictional surfaces. Consequently, considering the circumferential stress during braking and cooling phases is essential to evaluate the thermo-mechanical instability of the brake disc.

Figure 9 demonstrates a set of points on the brake disc that are used to create circumferential stress diagrams during the braking cycle. Node 1 is at the surface, and the rest are in the depth of the frictional layer with 2 mm distance with each other. Using the thermal stress plot in these points, the progression of cracks can be anticipated.

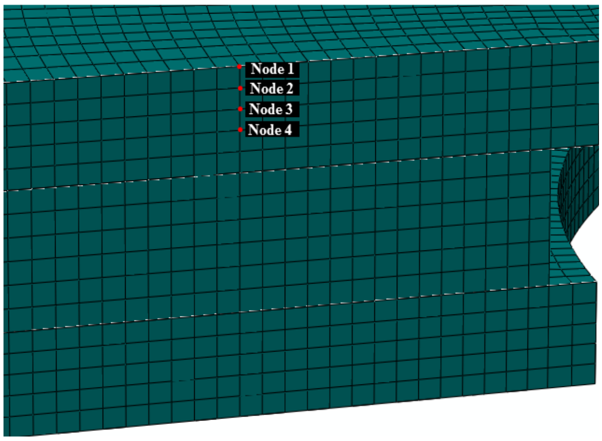

Figure 9. A surface point and 3 points through the thickness used to estimate the circumferential changes in z direction

Circumferential stress diagrams along the braking and cooling phases are illustrated in Figure 10 for a set of nodes (see Figure 9) in each braking case. In figure 10-a, highest circumferential stress

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happens for 25% faster case with a compressive stress of -242 MPa, and by increasing the braking time, stress magnitude decreases, in a way that for 50% slower case, surface circumferential stress don not exceed -91 MPa. In node-2, maximum circumferential stress decreases by almost 30% in all cases and also, the gradient of changes in thermal stress decreases in both braking and cooling phases. In each point, highest stresses belong to the braking with the shortest braking time. So, despite longer barking time for low deceleration processes, lower normal pressure makes smaller amounts of frictional heat that generates circumferential stresses with lower amplitudes.

As mentioned earlier, thermal coning on the brake disc leads to thermo-mechanical instabilities on the brake disc. An important aspect of this instability is the concentration of temperature on the surface that can change the properties of the material of brake disc. Moreover, this concentration of frictional temperature can generate thermal microcracks that lead to the failure of the brake system. Figure 11 illustrates temperature contours in the middle of the braking phase for each braking case with different decelerations. Surface hot spots with higher temperatures compared to other areas can be seen in all braking cases. The number of hot spots is dependent on the geometric of the disc and braking velocity, however usually 4 or 5 oval-

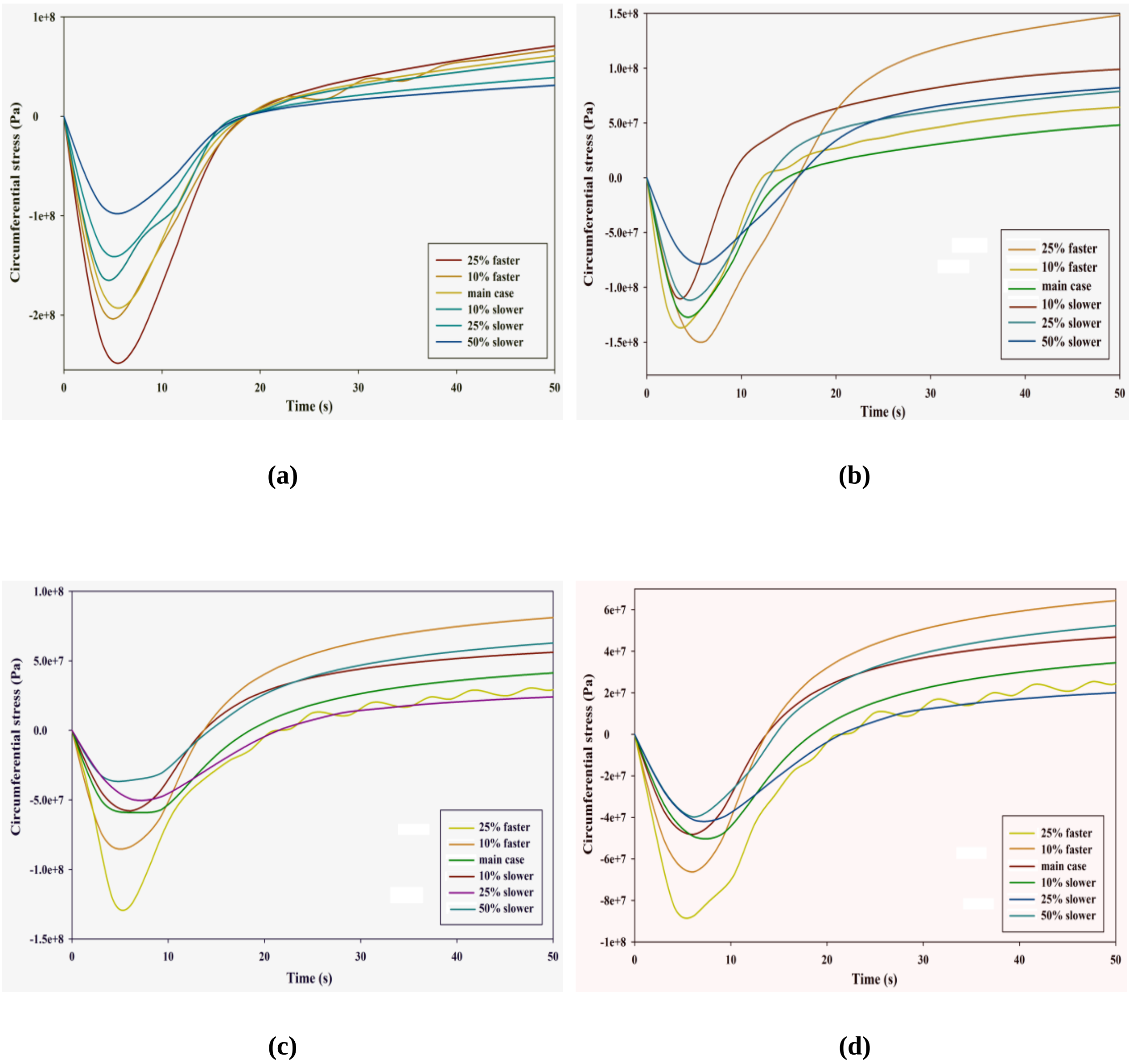

Figure 10. Circumferential stresses for each braking mode during the braking and cooling phases for the Node-1 (a), Node-2 (b), Node-3 (c) and Node-4 (d)

shape hot spots are recognizable in every braking condition. Increasing the braking severity increases

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the maximum temperature generated on the surface. For instance, the maximum temperature in 50% faster case is 341 °C, while in the main case is not more than 280 °C. Maximum temperature is almost the same in 10% and 25% slower cases. Despite having lower braking pressure, since braking time is higher in 25% faster case, the surface is exposed to heat for a longer time and as a consequence; surface temperature reaches up to 270 °C. This variation in temperatures along the radial and circumferential directions leads to non-uniform thermal expansion of the grey cast iron and as a result, non-uniform surface thermal stresses.

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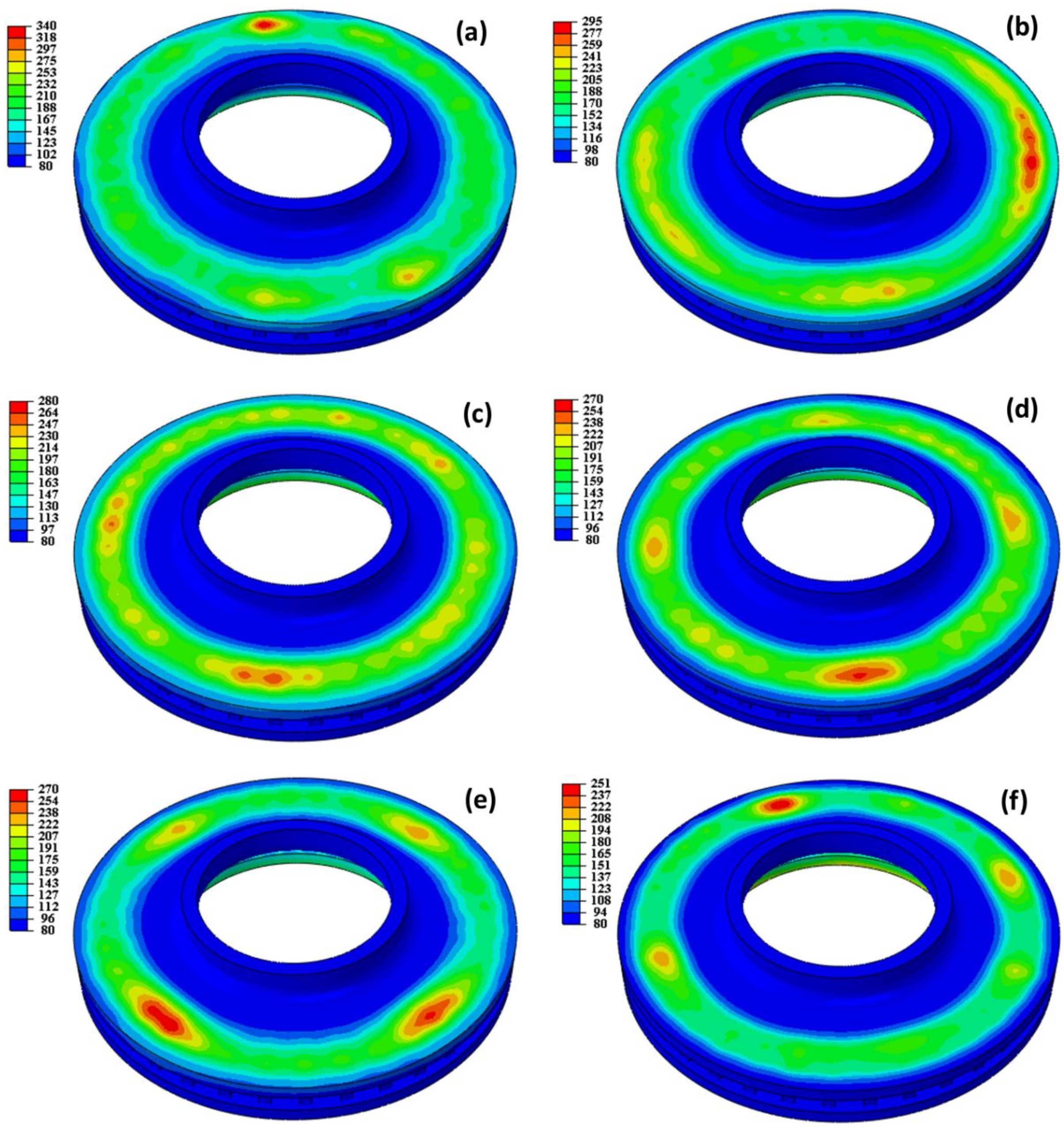

Figure 11. Surface temperature and hot spots for 25% faster (a), 10% faster (b), main (c), 10% slower (d), 25% slower (e) and 50% slower (f) braking cases at the middle of braking

Thermal circumferential stress on the upper surface of the brake disc can be seen in Figure 12 for all braking cases at the middle of braking phase. During the braking, frictional heat makes expansion in the material of the disc, and since there is constrain in the tangential direction, thermal stresses generate. Clearly, in all cases some areas in the surface reach a higher compressive stress than other parts of the surface. The locations of these compressive stresses are the same as hot spots that were generated during hard brakings. Stress value has a direct connection with the severity of the braking. In the 25% faster case, highest compressive stress is -207 MPa, while for 10% faster case, thermal stress do not exceed -185 MPa. Apparently, in the

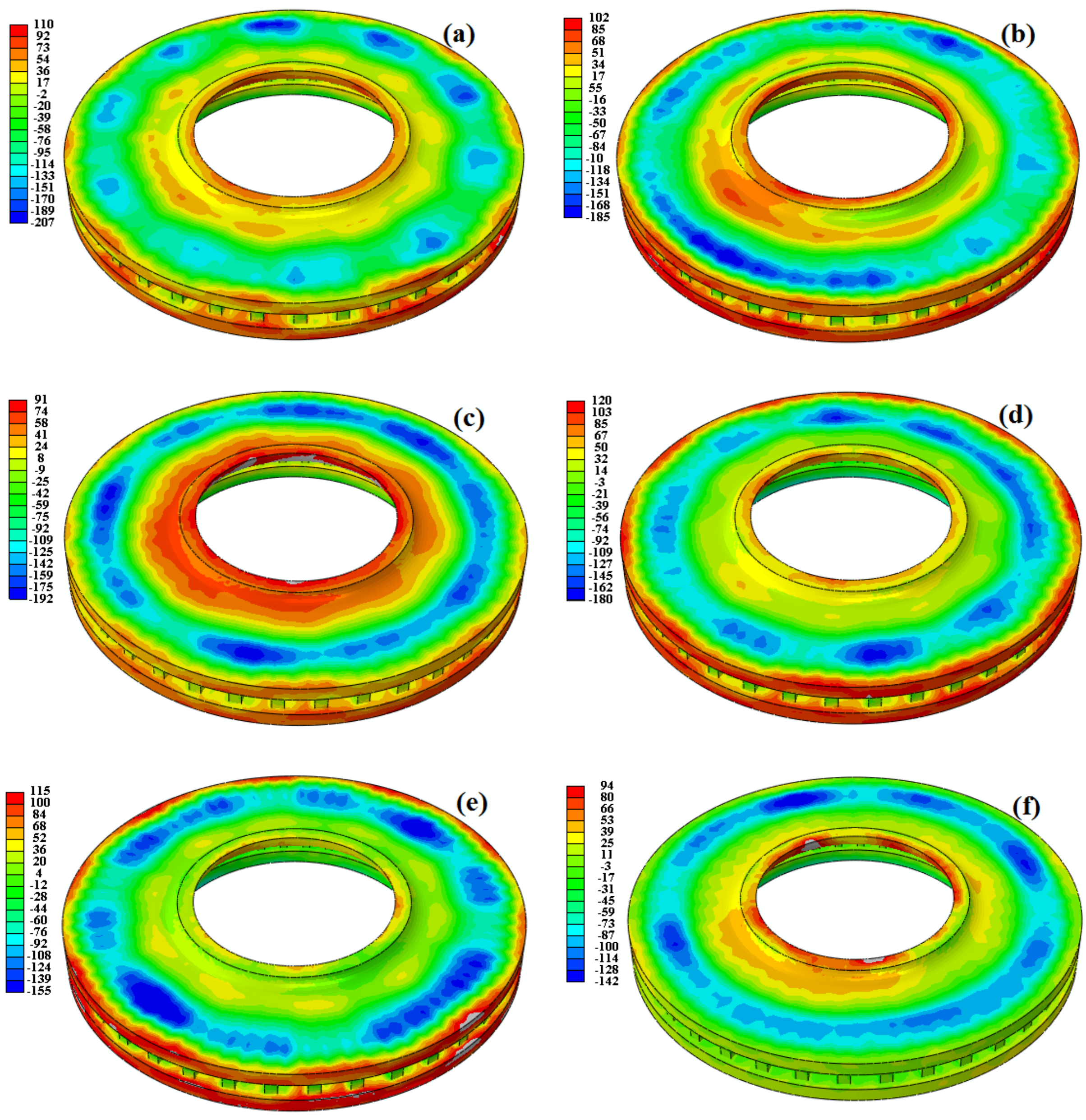

Figure 12. Circumferential stress at the middle of braking for 25% faster (a), 10% faster (b), main (c), 10% slower (d), 25% slower (e) and 50% slower (f) braking cases

main case, maximum circumferential stress is higher than 10% faster case. This is due to the fact that even though in main case maximum temperature is lower than the other case, braking time is longer and as a result, there is more time for cast iron to expand and create compressive stresses. Considering circumferential is vital for design and life calculation purposes and by lowering this component of generated stresses, working life cycle of this part can increase significantly.

5. Conclusions

Thermo-structural effects of hard brakings are studied in this paper on a three dimensional model of brake disc and pads using coupled thermo-mechanical analysis method. Utilizing appropriate thermal and mechanical boundary

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conditions on brake disc and pads makes it possible to observe surface hot spots generating during the braking. These hot spots are stem from a deformation on the brake disc known as thermal coning that makes the pressure distribution non-uniform along the radial and circumferential directions. Thermal coning is directly related to the severity of braking action. As for braking with $\ddot{\theta}$=-6.5 rad/s^2 maximum deformation is 0.6 mm, while for the braking with $\ddot{\theta}$equal to -3.9 rad/s^2, longitudinal deformation does not exceed 0.3. It was revealed that in all braking cases, highest compressive stress happens in the surface of the disc and by moving to the depth, stress magnitude reduces dramatically. Maximum compressive stress reaches up to -207 MPa in 25% faster case and by reduction of braking severity; maximum stress reduces in a non-linear manner. Finally, for new brake disc design purposes, it is necessary to choose the material and geometrical shape wisely in order to reduce the variation of circumferential stress.

Declaration of Conflicting Interests

The author(s) declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

List of symbols

h	Convection coefficient
E	Modulus of elasticity
k	Thermal conductivity
c	Specific heat
α_V	Thermal expansion
υ	Poisson's ratio
p	braking pressure
$\ddot{\theta}$	Deceleration of brake disc
T	Temperature
t	Time
ρ	Density

References

[1]. Mackin, T.J., S.C. Noe, K. Ball, B. Bedell, D. Bim-Merle, M. Bingaman, D. Bomleny, G. Chemlir, D. Clayton, and H. Evans, *Thermal cracking in disc brakes.* Engineering Failure Analysis, 2002. **9**(1): p. 63-76.

[2]. Belhocine, A. and M. Bouchetara, *Investigation of temperature and thermal stress in ventilated disc brake based on 3D thermomechanical coupling model.* Ain Shams Engineering Journal, 2013. **4**(3): p. 475-483.

[3]. Ouyang, H., A.R. Abu-Bakar, and L. Li, *A combined analysis of heat conduction, contact pressure and transient vibration of a disk brake.* International Journal of Vehicle Design, 2009. **51**(1-2): p. 190-206.

[4]. Akop MZ, K.R., Mansor MR, Rosli MAM, *Thermal stress analysis of heavy truck brake disc rotor.* Journal of mech eng Technology, 2009. **1**(1): p. 43-52.

[5]. Qi, H.S. and A.J. Day, *Investigation of disc/pad interface temperatures in friction braking.* Wear, 2007. **262**(5-6): p. 505-513.

[6]. Le Gigan, G., T. Vernersson, R. Lundén, and P. Skoglund, *Disc brakes for heavy vehicles: an experimental study of temperatures and cracks.* Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering, 2015. **229**(6): p. 684-707.

[7]. Lee, K. and J. Barber, *An experimental investigation of frictionally-excited thermoelastic instability in automotive disk brakes under a drag brake application.* Journal of Tribology, 1994. **116**(3): p. 409-414.

[8]. Bagnoli, F., F. Dolce, and M. Bernabei, *Thermal fatigue cracks of fire fighting vehicles gray iron brake discs.* Engineering Failure Analysis, 2009. **16**(1): p. 152-163.

[DOI: 10.22068/ase.2020.496]

[9]. Valvano, T. and K. Lee, *An analytical method to predict thermal distortion of a brake rotor*. 2000, SAE Technical Paper.

[10]. Koetniyom, S., *Temperature analysis of automotive brake discs.* The Journal of King Mongkut's University of Technology North Bangkok, 2003. **13**: p. 36-42.

[11]. Akhtar, M.M.J., O.I. Abdullah, and J. Schlattmann, *Transient thermoelastic analysis of dry clutch system.* Machine Design, 2013. **5**(4): p. 141-150.

[12]. Tiwari, A.K., P. Yadav, H. Yadav, and S. Lal, *Finite element analysis of disc brake by ANSYS workbench.* International Journal of Research in Engineering & Advanced Technology, 2014. **2**(2): p. 1-6.

[13]. Manjunath, T. and P. Suresh, *Structural and thermal analysis of rotor disc of disc brake.* International journal of innovative research in science, Engineering and Technology, 2013. **2**(12): p. 7741-7742.

[14]. Parab, V., K. Naik, and A. Dhale, *Structural and thermal analysis of brake disc.* International Journal of Engineering Development and Research, 2014. **2**(2): p. 1398-1403.

[15]. Day, A., M. Tirovic, and T. Newcomb, *Thermal effects and pressure distributions in brakes.* Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering, 1991. **205**(3): p. 199-205.

[16]. Cho, H., C. Cho, and C.-B. Kim, *Thermal and mechanical performance analysis in accordance with disk stiffness changes in automotive disk brake*. 2007, SAE Technical Paper.

[17]. Choi, J.-H. and I. Lee, *Finite element analysis of transient thermoelastic behaviors in disk brakes.* Wear, 2004. **257**(1-2): p. 47-58.

[18]. Zagrodzki, P., *Analysis of thermomechanical phenomena in multidisc clutches and brakes.* Wear, 1990. **140**(2): p. 291-308.

[19]. Okamura, T. and H. Yumoto, *Fundamental study on thermal behavior of brake discs*. 2006, SAE Technical Paper.

[20]. Jian, Q. and Y. Shui, *Numerical and experimental analysis of transient temperature field of ventilated disc brake under the condition of hard braking.* International Journal of Thermal Sciences, 2017. **122**: p. 115-123.

[21]. Commission, E., *Commission Directive 98/12/EC of 27 January 1998 adapting to technical progress Council Directive 71/320/EEC on the approximation of the laws of the Member States relating to the braking devices of certain categories of motor vehicles and their trailers.* 1998/01/27.

[22]. Pevec, M., I. Potrc, G. Bombek, and D. Vranesevic, *Prediction of the cooling factors of a vehicle brake disc and its influence on the results of a thermal numerical simulation.* International Journal of Automotive Technology, 2012. **13**(5): p. 725-733.

[23]. Gigan, G., V. Norman, J. Ahlström, and T. Vernersson, *Thermomechanical fatigue of grey cast iron brake discs for heavy vehicles.* Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering, 2015: p. 0954407017739723.

[24]. Collignon, M., A.-L. Cristol, P. Dufrénoy, Y. Desplanques, and D. Balloy, *Failure of truck brake discs: A coupled numerical–experimental approach to identifying critical thermomechanical loadings.* Tribology International, 2013. **59**: p. 114-120.