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# STUDY ON THE SIMULATION OF THERMAL-STRUCTURE COUPLING ON RETARDING DISK AND ANALYSIS OF BRAKE COMPONENT MODEL

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**Abstract.** Brake system is an important part of automobile safety performance; brake is its core element, directly affecting the safety of driver. Based on the principle of disc brake friction, the characteristic of thermal-structure coupling has a great influence on such parameters as the thermal fatigue life and resonance of shouting. This paper suggest a proper simplification of the car's front disc brake. The thermal-structural coupling of the brake disc on the emergency braking condition has been simulated by means of finite element software. The thermal fatigue life is predicted, and the modal analysis of brake is finally carried out.

**Keywords:** disc brake; thermal-structural coupling; fatigue life.

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**Problem statement.** The automobile braking system plays an important role in deceleration safely [1–3]. If the brake of car fail in the running, the unbearable consequences will break out. The principle of braking system is as follows. When braking happens in a flash, the friction between the brake disc and friction resistance is generated. Then most of the kinetic energy is converted into heat, and braking is achieved effectively. However, when the car runs at a high speed, "hot spots" are found in the local area of the brake disc [4]. They have a great influence on the local change of contact pressure and temperature, which leads to the phenomenon of thermal elastic instability (TEI) [5]. In the process of thermal-structural coupling [6], the TEI not only have an adverse effect on the performance of a normal brake, but also cause the heat recession of the brake disc. The crack or rupture happens in the surface of the brake disc after multiple braking, which substantially reduces the service life of the brake widely.

**Latest research and publication analysis.** Zhang Lijun [7] of Tongji University has carried out a bench test during the research on the rules of temperature rise and thermal deformation of the brake disc. The results of experiment show that the rate of temperature rise has a close relationship with the brake pressure. At the same time, temperature rise cause deformation of the brake disc, which is related to the obvious phenomenon of thermal-structural coupling.

Friction leads not only to energy loss, but also to wear of the parts and reduction of the service life [8]. The friction heat mechanism mainly involves the aspects listed below [9, 10].

1. The surface of the friction pair is not smooth, and the micro-bulges are different in various degrees. When there is relative movement between the friction pairs, the micro-bulges of surface shear to generate friction heat.

2. The surface of the friction pair between the local adhesion points make adhesion, tensile and fracture during the friction. Plastic deformation occurs to the contact area and the surrounding material, which gives off a lot of heat.

3. In the general atmospheric environment, friction material also releases a considerable amount of heat by thermal degradation.

**Basic material.**

***1. Modeling and simulation analysis of the disc brake***

At present, the disc brake is one of the most widespread types of brakes [11]. It has a number of apparent advantages. Under the same braking torque output, the size and quality of the drum brake structure is also relatively small, simple, and easy for maintenance and preservation. Meanwhile, it also has the stable brake torque, good thermal stability and so on. The structure of the disc brake is shown in Fig. 1.

Braking is an essential part in the process of driving. When the driver presses the brake, pedal effort is amplified through the booster, and oil pressure is pressed into brake cylinder. The piston drive friction plate is pressed onto the brake disc under the hydraulic of oil delivery. After that, friction moment is generated between the friction plate and the brake disc, and the kinetic energy of the car is converted to friction heat. Afterwards, the brake disc is forced to stop. Finally, the scrolling wheel stops moving, as well as the brake disc, because the brake disc is connected with the wheels.

***1.1. Establishment and simplification of the three-dimensional model***

Parameters of the vehicle parts and the friction pair are shown in Tables 1 and 2, respectively.

In the process of braking, friction parts include the brake disc and the friction plate, whose friction heat is almost absorbed. Therefore, their temperature rises higher, especially that of the brake disc made of a metal. It is the direct cause of hot fatigue failure.

The disc brake structure has the character of relativity; thermal load on both sides of the friction interface is symmetrical. Considering resource saving, the model is established through removing the tiny structures affecting characteristics of thermal-structural coupling to improve meshing, such as holes, small convex platform, irrelevant chamfering, etc. [12]. In addition, the hub used to connect the brake disc and the vehicle and the part of the friction plate used to install the friction lining board are simplified. The three-dimensional model of the main part of the brake disc is simplified as shown in Fig. 2.

***1.2. Physical parameters of model material***

Material characteristics of the friction pair are shown in Tables 3 and 4 [12].

***1.3. Establishment of the finite element model***

The three-dimensional model is established using Pro/E. Firstly, simplified data in the form of stp is imported into ABAQUS software. The C3D8RT thermal coupling unit with eight nodes and hexahedron is chosen in this study. Rotation of the brake disc around the axis is controlled by rotational degree of freedom applied in the central reference point of the brake disc. In this paper, the whole model is divided into 960 units, and the friction plate and brake disc are subdivided into 168 and 792 units, respectively. The grid model of the brake disc and the friction plate are calculated.

***1.4. Displacement and load boundary conditions***

Before loading, the reasonable array parameters and constraints should be determined to get accurate results. The brake disc and the friction plate are defined as surface-to-surface contact, and the brake disc is only applied to the reference center point to rotate freely, so as to restrict the freedom of the friction plate except the axial displacement. The initial temperature and speed of the vehicle emergency are considered for braking at 100 km/h to maintain uniform deceleration, applying the initial angular velocity in the reference center of the brake disc, 3.17 MPa of the friction brake pressure, brake time, and variation of angular velocity as shown in the following formula:

$$\omega = 87.63 - 28.16t \ (0 \le t \le 3.1s)$$

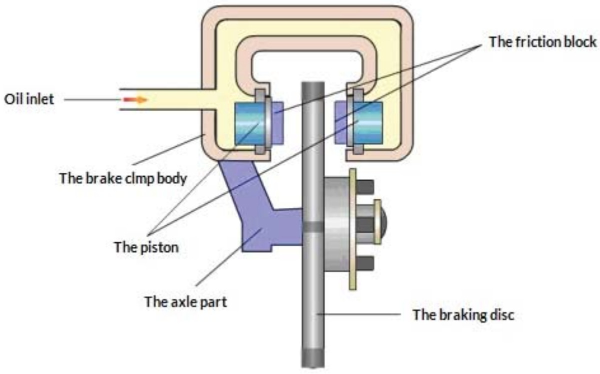


**Fig. 1.** Structure of disc brake

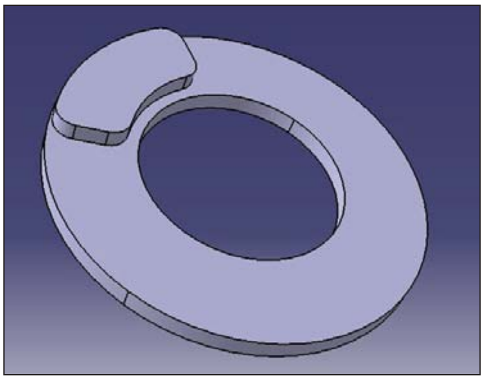

**Fig. 2.** Simplified three-dimensional model of the disc brake

**Table 1.** Parameters of the parts of the entire vehicle

| Full quality | Wheelbase $L$, mm | Height of centroid $H$, mm | Tire rolling radius $R$, mm | The distance to the rear axle center $L_2$, mm |
|---|---|---|---|---|
| 1798 | 2715 | 500 | 317 | 1285 |

**Table 2.** Size parameters of the friction pair

| Wrap angle | Inner radius $d_i$, mm | Outer radius $d_0$, mm | Thickness δ, mm | Wrap angle θ, ° |
|---|---|---|---|---|
| Brake disc | 75 | 138 | 28 | 360 |
| Friction plate | 84 | 136 | 13 | 60 |

**Table 3.** Material parameters of the friction pair

| Wrap angle | Material | Density, kg·gm$^{-3}$ | Friction factor, μ |
|---|---|---|---|
| Brake disc | Resin matrix composites | 2595 | 0.38 |
| Friction plate | ZG1Gr13 | 7228 | 0.38 |

**Table 4.** Thermal physical properties of the friction pair

| Wrap angle | Specific heat $c$, J/(kg·K) | Coefficient of heat conduction $K$, W/(m·K) | Coefficient of thermal expansion α, K$^{-1}$ | Modulus of elasticity $E$, GPa | Poisson ratio, μ |
|---|---|---|---|---|---|
| Brake disc | 1465 | 1.212 | $1.165\cdot10^{-5}$ | 95 | 0.3 |
| Friction plate | 460 | 48.46 | $4.39\cdot10^{-6}$ | 145 | 0.3 |

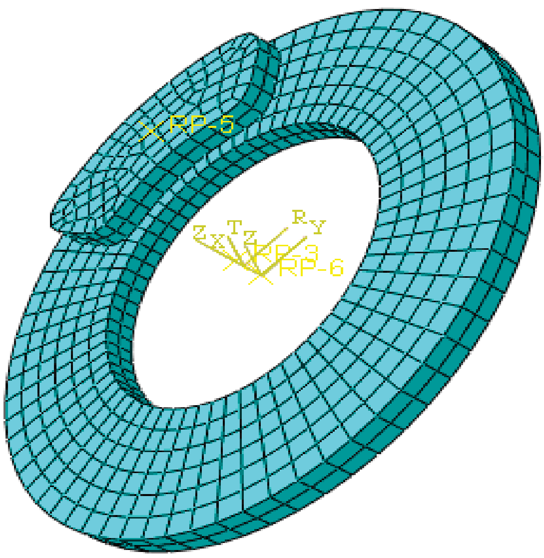


**Fig. 3.** Mesh thermal-structure coupling model

### 1.5. Convection heat transfer boundary conditions

The relationship between the heat transfer coefficient and the time simulated by using the MATLAB software is obtained as follows:

$$h = 135.96 - 40.79t \ (0 \le t \le 3.1s)$$

The fitted curve is shown in Fig. 4.

As the friction plate is fixed in the clamp, only four sides of the working surface of the brake disc are perpendicular to the surface, which is subject to heat convection with the air. Based on the heat transfer coefficient for natural convection in finite space, the surface heat transfer coefficient is determined [13]. Empirical value h in this paper is 5.3 w/m$^2$. The inner ring of the brake disc is insulated.

### 2. Finite element analysis of the brake disc

The brake model is introduced into the ABAQUS to carry out the finite element simulation; its initial speed is set 100 km/h for braking imminently. The distribution of temperature field, stress field and the change in temperature rise and stress are obtained by calculation and analysis. The thermal fatigue life of the brake disc is estimated with the help of the prediction model of thermal fatigue life [14].

### 2.1. Temperature field analysis of the brake disc

The rule of temperature distribution from beginning to end is shown in Fig. 5.

Fig. 5 shows that at the beginning of braking, the brake rotates very fast. The frictional heat flux is stable and the velocity of heat input is much faster than the speed of heat conduction in the brake disc. Brake disc surface temperature rises sharply; the hot red zone of in Fig. 5 is gradually expanding to the maximum temperature at 2 s.

With the decrease of rotational speed, the intensity of heat flux decreases gradually, with the heat flux between surface heat and air convection heat dissipation. The role of the internal thermal conduction is stronger, so that the surface temperature of the disk begins to decline. During the process of braking, the temperature of the brake disc increases first and then decreases, and the distribution of temperature field is not symmetrical.

### 2.2. Rule of temperature distribution of circumference on the brake disc surface

In the previous research on the temperature field of the disc brake, there is no difference of temperature between the circumferential nodes on the same radius of the brake disc.

The temperature characteristics of 10 different nodes with different phase angles are selected to study at the 117 mm radius of the brake disc. The temperature distribution curve is shown in Fig. 6.

It can be seen from Fig. 6 that the temperature gradient of the circumferential nodes is relatively small, the curve is close to the whole, and the whole is the first to rise and then decrease. However, circumference temperature is not equal everywhere. At 2.2 s, the temperature of node 2 in the region of friction reaches 230 °C, then the temperature of node 7 exceeds 190 °C. The node's temperature of the friction region is slightly higher than that of the non-frictional contact area.

### 3. Stress field analysis of the brake disc

It can be concluded from the analysis of temperature field of the brake disc that the effect of the intermittent impact of the surface of the brake disc is the heat input and heat dissipation by moving the heat source, resulting in a periodic change in temperature. However, according to the mechanism of stress generation, the increase of the brake disc temperature generates thermal stress. Therefore, it is necessary to study the distribution of stress field of the brake disc. Under the condition of emergency braking, the equivalent stress field of the brake disc at different moments of time is shown in Fig. 7.

As can be seen from Fig. 7, the distribution of stress field is asymmetric, similar to the temperature distribution of the brake disc. The red part is mainly concentrated in the friction radius, where stress concentration occurs due to the friction heat flow. Temperature gradient of axial and radial of the brake disc is generated due to friction heat in the area of friction. The equivalent stress makes up 227.9 MPa at 2 s. In the later stages of braking, the value of stress decline with the decrease of speed. At the end of braking, there is still a lot of residual stress.

### 4. Fatigue life prediction for the brake disc

According to the previous analysis, the temperature and stress distribution in different parts of the brake disc is different. The higher the temperature of the brake disc is, the higher the thermal stress is. Consequently, it is more likely to generate cracks. The most of cracks are formed under the effect of constant cyclic load. The cracks extend along the radial direction until the fracture.

### 4.1.1. Life prediction for the brake disc

As can be seen, the highest temperature of the brake disc is acquired near the friction radius, where thermal fatigue crack likely occur. According to the mechanism of thermal fatigue crack, the highest stress of the brake disc is 229.8 MPa, achieved in 1.952 s. The average equivalent stress of 6 nodes selected on the friction area at the moment is σeq = 227.95 MPa, the total strain is $\varepsilon_a = \sigma_{eq}/E = 0.001572$. The fatigue performance parameters of the brake disc are shown in Table 5.

The parameters of Table 4 are substituted into the formulas above to result in the new formula as follows:

The strain-life curve of the brake disc built using MATLAB software is shown in Fig. 8.

Fig. 8 shows the brake disc failure after pressure cycling for 9593 times when the initial speed reaches 100 km/h and the brake pressure is 3.17 MPa. That is, the service life of the brake disc is 9593 times under normal conditions.

**CONCLUSIONS.** In this study, the model of thermal-structural coupling between the brake disc and the friction plate is established using finite element analysis software ABAQUS. The model reflects distribution of temperature and stress fields under emergency braking conditions. The conclusion is summarized as follows:

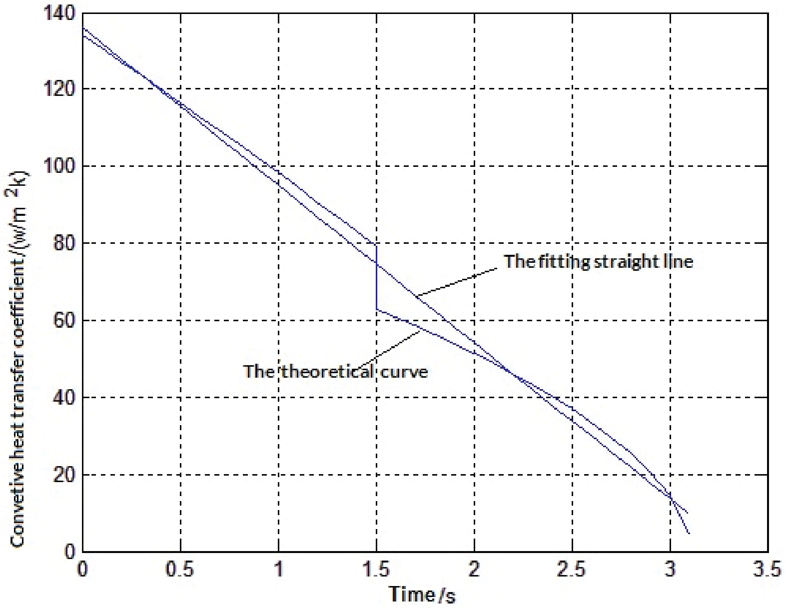


**Fig. 4.** Convective heat transfer coefficient with braking time of the brake disc's radiating surface

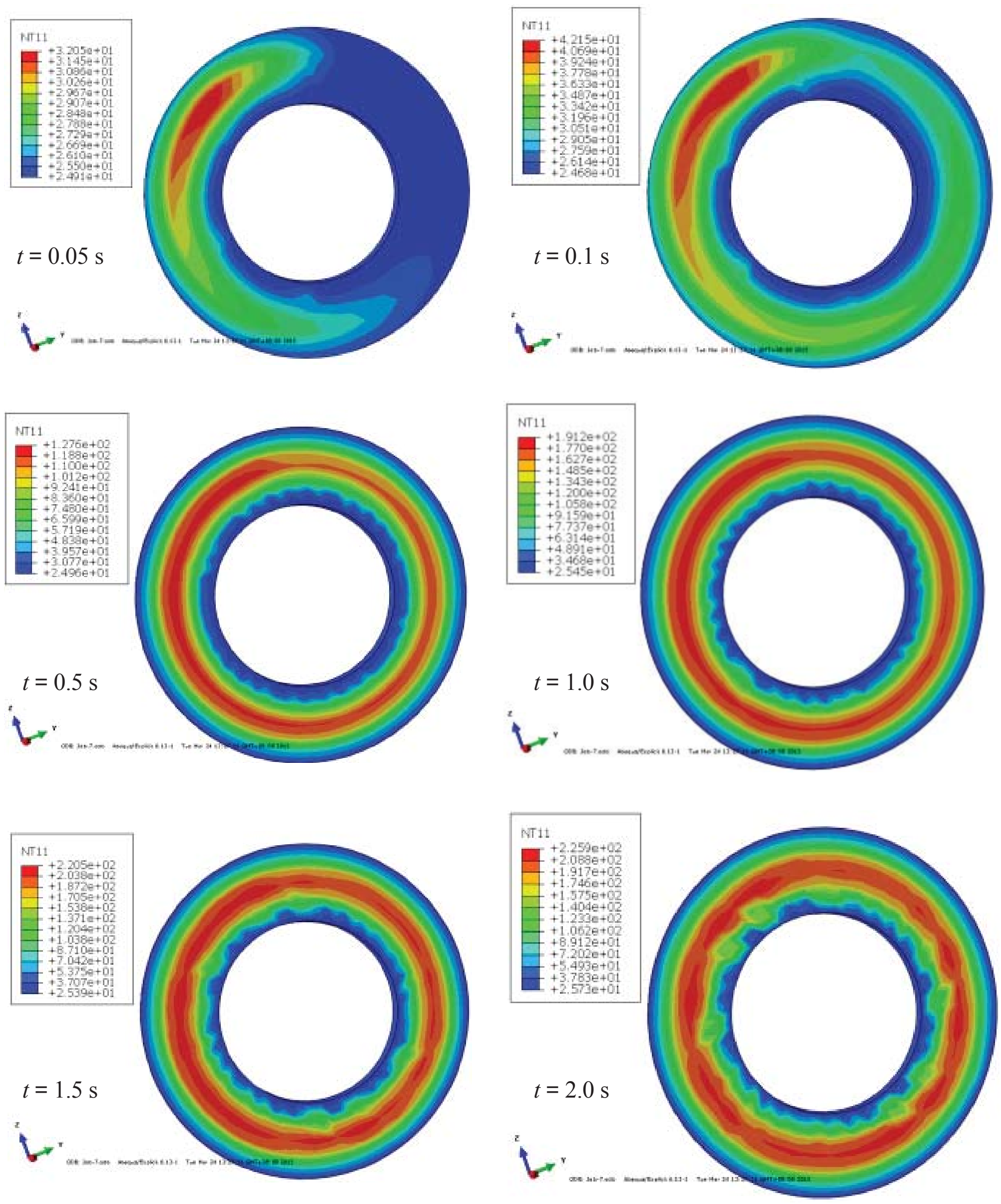


**Fig. 5.** Temperature field distribution of the brake disc at different moments of time

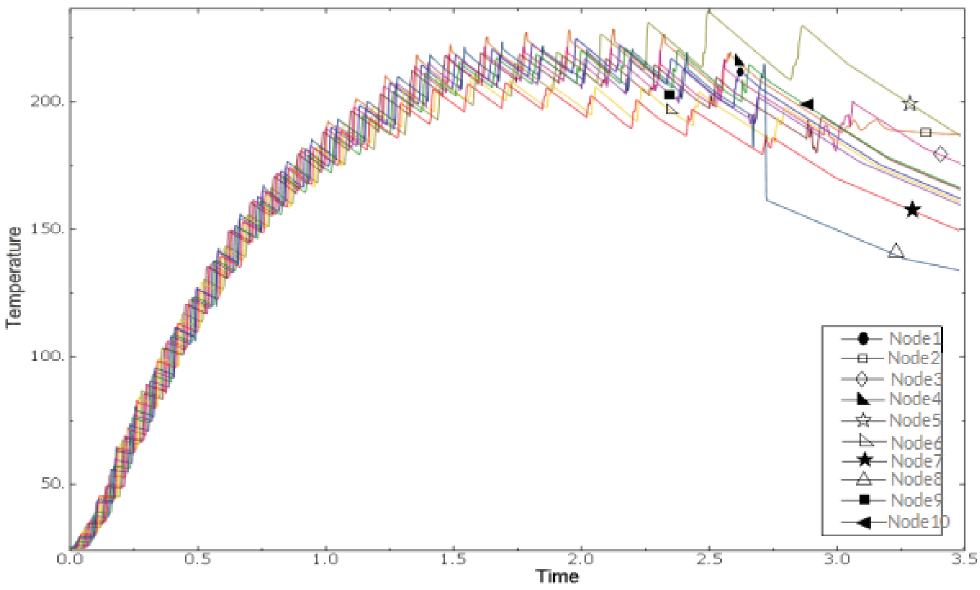


**Fig. 6.** Temperature of circumferential nodes in the same radius on the surface of the brake disc

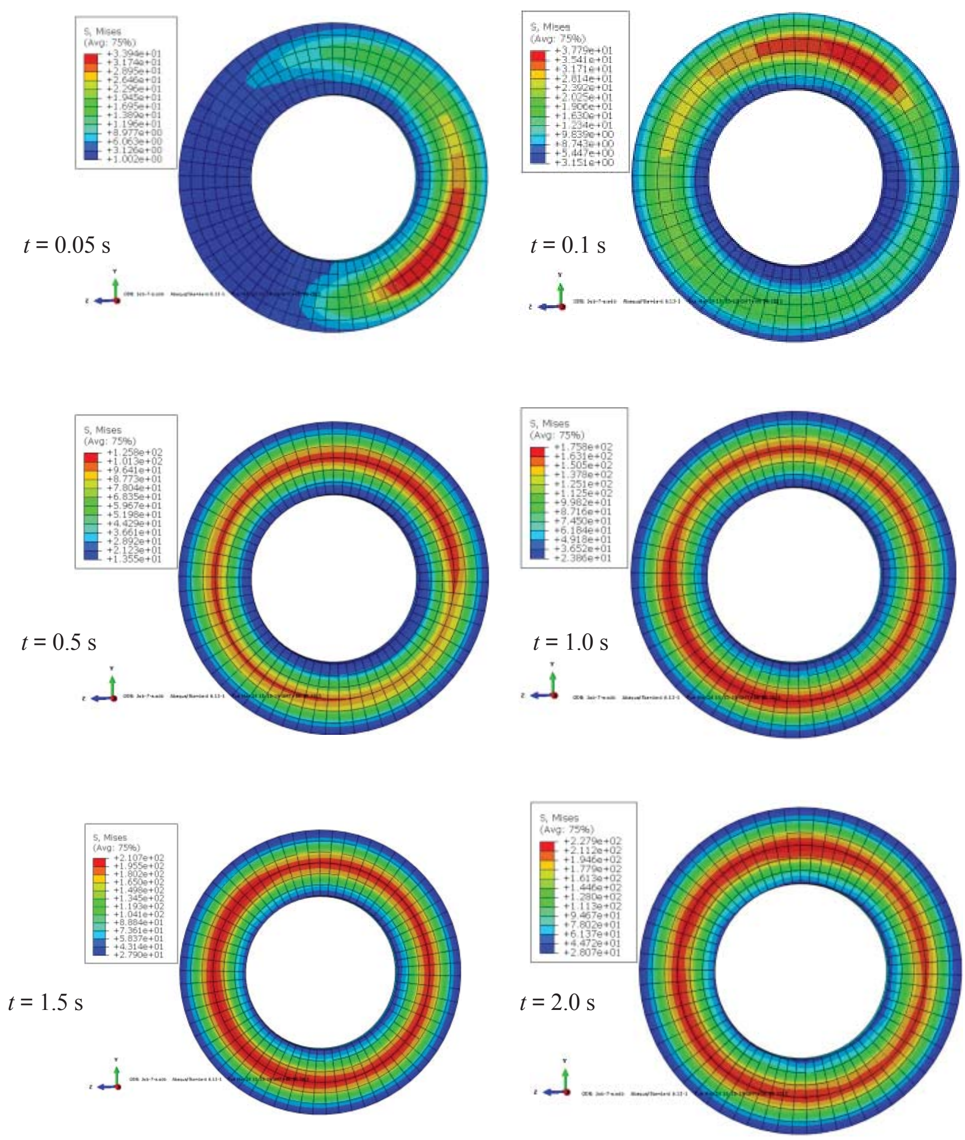


**Fig. 7.** Stress field distribution of the brake disc at different moments time

**Table 5.** Fatigue performance parameters of the brake disc

| Fatigue strength coefficient $\sigma_{eq}$, MPa | Fatigue strength index, $b$ | Fatigue ductility coefficient $\varepsilon_f$, % | Fatigue toughness index, $C$ |
|---|---|---|---|
| 650 | –0.12 | 7.4 | –0.6 |

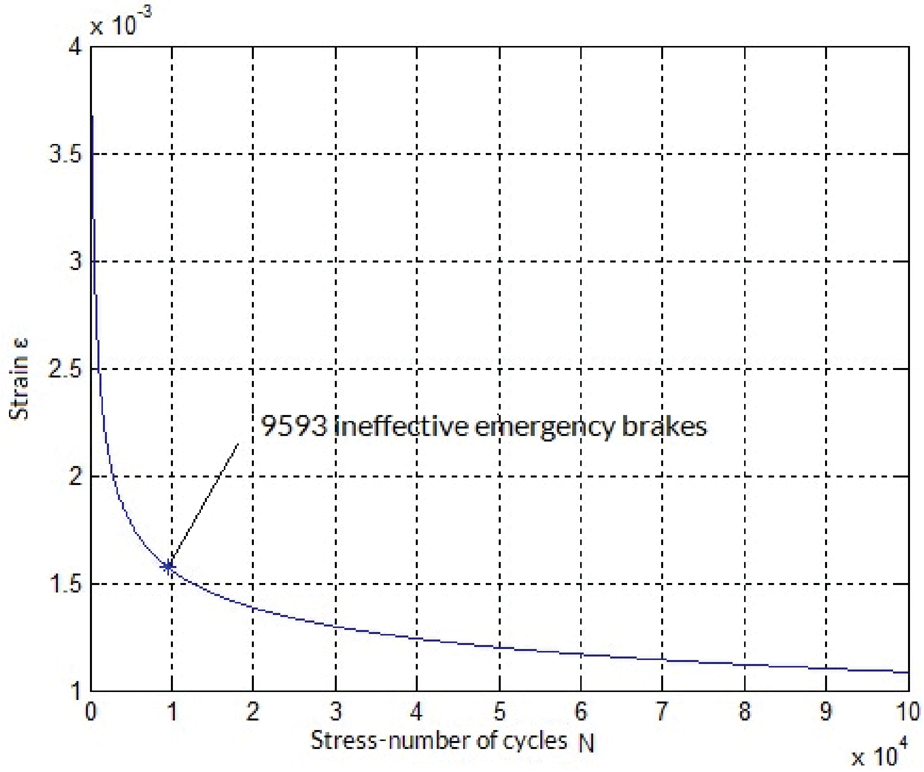


**Fig. 8.** Temperature of circumferential nodes in the same radius on the surface of the brake disc

1. At the beginning of braking, the temperature rises comparatively fast. The temperature peak is in the middle of braking. At the end of braking, the curve appears as smooth. Temperature field distributes asymmetrically, the temperature gradient of axis and radial changes significantly, whereas the change of circumferential gradient is quite small.

2. The highest temperature of the brake disc is acquired near the friction radius. The farther the distance to friction area is, the lower the temperature is.

3. According to the analysis of triple stress court, the gradient of circumferential stress is minimum, which is the main stress component of the brake disc. Yet, the stress of axial nodes is far less than that of axial and radial.

4. The change of the stress is caused by temperature changes, and the thermal deformation, in turn, affects the contact pressure. Therefore, temperature field, stress field and contact pressure are highly coupled.

5. Thermal fatigue life of the brake disc under the condition of braking imminently can be calculated by the formula of Manson-coffin.



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