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Simulation Analysis and Verification of Temperature and Stress of Wheel-Mounted Brake Disc of a High-Speed Train

Junsheng Qu¹, Wenjing Wang^{1*} , Ziyu Dong¹ and Wei Shan²

Abstract

During the braking process, a large amount of heat energy is generated at the friction surfaces between the brake disc and pads and rapidly dissipates into the disc volume. In this paper, a three-dimensional thermo-mechanical coupling model of high-speed wheel-mounted brake discs containing bolted joints and contact relationships is established. The direct coupling method is used to analyze the temperature and stress of the brake discs during an emergency braking event with an initial speed of 300 km/h. A full-scale bench test is also conducted to monitor the temperatures of the friction ring and bolted joints. The simulation result shows that the surface temperature of the friction ring reaches its peak value of 414 °C after 102 s of braking, which agrees well with the bench test result. The maximum alternating thermal stress occurs in the bolt hole where the maximum circumferential compressive stress is -658 MPa and the maximum circumferential tensile stress is 134 MPa. During the braking process, the out-of-plane deformation of the middle part of the friction ring is larger than that of the edge, which increases the axial tensile load of the connecting bolt. This work provides support for the design of brake discs and connecting bolts.

Keywords: High speed train, Wheel-mounted brake disc, Temperature field, Stress field, Thermo-mechanical coupling model, Bolt

1 Introduction

Compared with other braking methods, disc brakes have the advantages of simple structure and reliability, which plays an important role in the operational safety of high-speed trains. In the process of train braking at high initial speed, the heat generated by friction causes the brake disc to produce a sharp temperature gradient, which leads to high thermal stress [1, 2]. Thermal fatigue is the main cause of brake disc failure, which directly affects the reliability of the brake disc and train operation safety [3–6]. At present, the finite element method is widely used to analyze the temperature fields and stress fields in the braking process, and the main method is the

thermo-mechanical coupling method [7–9]. In the actual braking process, the temperature field distribution of the brake disc directly affects the distribution of the thermal stress field. The deformation of the brake disc also has a certain impact on the distribution of the temperature field [10].

To optimize the structure of the brake disc and improve its service performance, scholars have carried out extensive research by simulation and experiment. Jean-Gabriel et al. Ref. [11], established an analytical model for the multi-point temperature calculation on the disk surface of high-speed trains, which can accurately calculate the three-dimensional temperature inside the disk in a short time. Adam et al. Ref. [12] studied the effect of cooling on the temperature distribution of the brake disc during cyclic braking by simulation and the results show that convective heat transfer cannot reduce the maximum temperature of the brake disc in a single braking process.

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Hwang and Wu [13] analyzed the thermo-mechanical coupling behavior of a single brake of a ventilated brake disc by using a three-dimensional finite element model and concluded that the uneven distribution of the contact pressure led to an uneven temperature field and the temperature field affected the change in the contact pressure. Benhassine et al. Ref. [14] compared and analyzed the temperature field, stress field and contact pressure of three kinds of axle-mounted brake disc materials and summarized the influence of the temperature field on the contact pressure and stress field. Zhou et al. Ref. [15] studied the fracture of brake disc bolts under a section downhill of the Lanzhou-Wulumuqi railway line. The finite element model of bolt thread with hexahedral meshes was established to obtain the stress states, temperature and the change of bolt performance during continuous braking. Wang et al. Ref. [16] analyzed the load evolution of axle-mounted brake disc bolts in the process of long slope continuous braking and studied the influence of the preload, thermal expansion coefficient and other factors on the load of brake disc connecting bolts. Wang et al. Ref. [17] studied the temperature trends of train brake discs in the process of high-speed braking and concluded that the change in temperature causes fluctuation in the instantaneous friction coefficient and a distribution of thermal stress on the brake disc, which implied thermal damage. Ghadimi et al. Ref. [18] conducted a three-dimensional simulation analysis on the brake temperature field of the wheel-mounted brake disc of an ER24PC locomotive and compared the results with experimental results. This research shows that the convection design of the brake disc has little effect on the maximum temperature during the braking process and the maximum temperature value depends on the braking deceleration. Dae-Jin et al. Ref. [19] studied the influence of thermal stress on the contact surface of the brake disc under different contact pressure distributions by using a three-dimensional finite element model. It was found that under the condition of variable pressure distribution, the area where the maximum thermal stress appears is similar to the fatigue crack area of the actual disc brake.

The experimental results show that crack initiation usually comes from the bolt hole near the middle of the brake disc, and the bolt hole on the disc surface is the dangerous part of brake disc fatigue [20]. Figure 1 shows the fatigue cracks on the bolt holes of a wheel-mounted brake disc in practice.

Most of the simulation work on temperature and stress field of brake disc ignore the influence from the complex contact nonlinearity and material nonlinearity and some studies lack the support of actual test data. To reveal the causes of thermal fatigue cracks in brake discs, a three-dimensional thermo-mechanical coupling

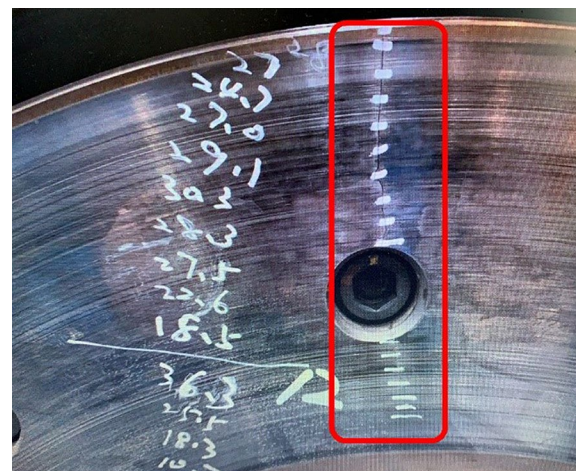


Figure 1 Fatigue crack at the bolt hole edge of the wheel-mounted brake disc

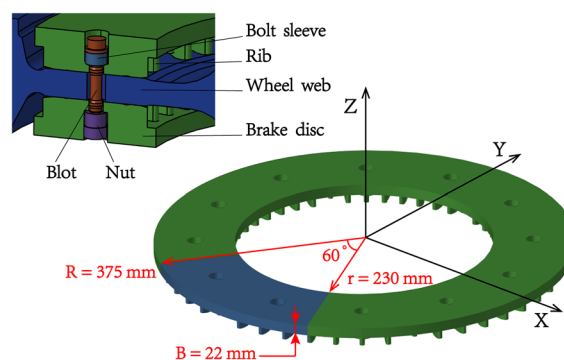


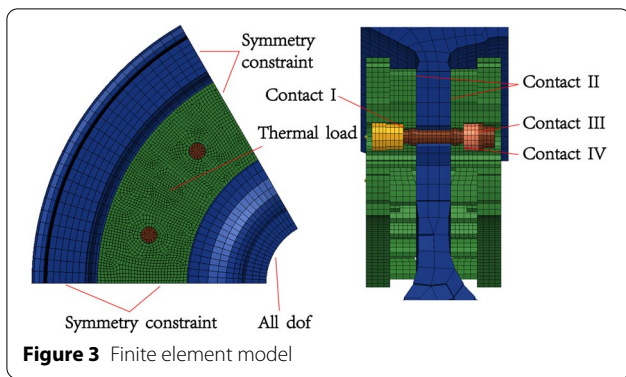
Figure 2 Geometric model of wheel-mounted brake disc

model of wheel-mounted brake discs of high-speed trains with bolts with the contact relations of relative components is established. The temperature field, stress field, displacement field and bolt temperature of brake discs under 300 km/h emergency braking conditions are simulated and analyzed by the direct coupling method, and a full-scale bench test is carried out. The research results provide theoretical support for brake disc design, bolt selection and application of fault analysis.

2 Finite Element Simulation of Wheel-Mounted Brake Disc

2.1 Finite Element Model and Material Properties

A typical ventilated-type wheel-mounted brake assembly for high-speed train is studied in this paper and its geometry is illustrated in Figure 2. The brake disc is connected to the wheel by twelve high strength bolts. Based on the



symmetry of the structure and the distribution of the heat dissipation ribs, only one-sixth of the brake assembly was employed for thermal-mechanical analysis and the finite element model is shown in Figure 3. The model is meshed with SOLID5 element for multiple physical field analysis. The total number of nodes and elements are 44330 and 33198 respectively. The internal surface of the wheel hub is fully-constrained, the periodically symmetry constraint is imposed to the symmetrical surface, and the frictional heating flux is applied to the friction surface of the brake disc.

The frictionally generated heat flows through the brake disc volume and enters into other components through the contact surfaces between different parts. The contact surfaces in the finite element model (by Conta174 and Targe 170 elements) are shown in Figure 3, which include the nut and the hole of brake disc body (Contact I), rib

and wheel web (Contact II), bolt and sleeve (Contact III), and sleeve and the hole of brake disc body (Contact IV) respectively. A hard contact was used for the normal contact interaction while an isotropic Coulomb friction formulation for the tangential contact interaction. The friction coefficient was chosen to be 0.4. Considering the rotation and separation may occur between the contact surfaces, the finite sliding formulation was used.

In addition to the mechanical parameters, the boundary conditions need to consider the convective heat transfer coefficient, heat radiation and heat flux. The brake disc body is made of cast steel, and the connecting bolt is made of high-strength alloy steel. The material parameters and thermo-physical properties are listed in Tables 1 and 2 [20], respectively, and the materials are isotropic.

2.2 Calculation of Thermal Load and Relevant Parameters

In the process of emergency braking, the brake disc converts the kinetic energy of the train into heat energy through friction. The heat flux is imposed on the friction surface of the brake disc to simulate the friction heat input load between the brake disc and the brake pad.

Based on the theoretical basis of the brake friction and thermal analysis, the following assumptions are made before calculating the heat flux: (1) The brake process is a uniform deceleration process, and the brake pad pressure is constant during the braking process. (2) The friction coefficient between the brake disc and the pad is constant during the braking process. (3) The friction heat formed in the process of braking is generated on the contact

Table 1 Brake disc material properties

Temperature (°C)	Tensile strength (MPa)	Yield strength (MPa)	Thermal expansion coefficient (10 ⁻⁶ .K ⁻¹)	Elastic modulus (GPa)	Poisson's ratio	Thermal conductivity (W.m ⁻¹ .K ⁻¹)	Specific heat capacity (J.kg ⁻¹ .K ⁻¹)
20	1235	1137	11.6	211	0.28	45.2	473
200	1130	1030	11.6	202	0.28	43.6	523
400	1050	945	13.6	187	0.29	43.9	607
600	565	420	14.2	168	0.3	42.3	754
700	191	100	14.2	152	0.31	43.8	805

Table 2 Bolt material properties

Temperature (°C)	Tensile strength (MPa)	Yield strength (MPa)	Thermal expansion coefficient (10 ⁻⁶ .K ⁻¹)	Elastic modulus (GPa)	Poisson's ratio	Thermal conductivity (W.m ⁻¹ .K ⁻¹)	Specific heat capacity (J.kg ⁻¹ .K ⁻¹)
20	1160	755	16.8	199	0.3	38.3	524
200	968	714	16.8	199	0.3	36.4	575
400	919	651	17.4	199	0.3	32.7	607
600	880	620	17.8	157	0.3	29.1	752

surface, and it is uniformly input to the friction surface of the brake disc via heat flux. The heat flux is calculated by the energy conversion method, and the formula is as follows [21, 22]:

$$q(t) = \frac{-\eta M a v(t)}{2nA}, \tag{1}$$

where η is the thermal absorption rate of the brake disc ($=0.9$), M is the mass of the whole vehicle ($=64400$ kg), n is the number of the brake discs of the whole vehicle ($=8$), A is the friction surface area of the brake disc ($=0.28$ m²), a is the braking deceleration, $v(t)$ is the vehicle speed.

The thermal contact conductivity is the reciprocal of the contact thermal resistance, which is related to the surface roughness, contact pressure, contact temperature and other parameters of the contact surface [23]. The higher the contact pressure and the greater the temperature difference between the two contact surfaces are, the higher the thermal contact conductivity is, and the thermal contact conductivity decreases with increasing contact surface roughness [24]. The thermal contact conductivity is also related to the materials of the two contacting objects. The calculation formula of the thermal contact conductivity of the same material is as follows [25]:

$$h_c = 10^5 \left[c_1 \lambda \left(\frac{p}{Hv} \right)^{2/3} + \frac{c_2}{Ra_t^n} \right], \tag{2}$$

$$Ra_t = Ra_1 + Ra_2,$$

$$c_1 = 0.06 \in [0.055, 0.065],$$

$$c_2 = 0.09 \in [0.085, 0.095],$$

$$m = 0.8 \in [0.8, 0.9],$$

$$n = 0.7 \in [0.7, 0.8],$$

where p (MPa) is the contact pressure, λ (W/mK) is the thermal conductivity of the material, Hv is the Vickers hardness of the material ($=247$ kg/mm²). Ra_1 and Ra_2 are the surface roughness of the two contact surfaces ($=1.6$ μ m). c_1, c_2, m and n are coefficients of the equation.

In the process of train braking, the brake disc conducts forced convection heat exchange with the air. When the train is stationary, the brake disc and air conduct natural convection heat exchange, which transfers heat from the brake disc to the air. Convective heat transfer is related to the fluid flow state, fluid physical properties, temperature

and geometry. There is also radiant heat exchange between brake disc components. According to the theory of heat transfer, the convective heat transfer coefficient [26] and the heat radiation coefficient [20] are obtained:

$$h_t = 0.664 \left(\frac{L\mu}{\gamma} \right)^{1/2} Pr^{1/3} \frac{\lambda_a}{L}, \tag{3}$$

$$h_r = \varepsilon \sigma (T^2 + T_a^2)(T + T_a), \tag{4}$$

where Pr is Prandtl number ($=0.7$), λ_a is air thermal conductivity coefficient ($=2.6 \times 10^{-2}$ W/mK), L is wall length ($=2\pi r$), μ is air flow velocity (m/s), γ is air kinematic viscosity ($=14.8 \times 10^{-6}$ m²/s), ε is emission coefficient ($=0.5$), σ is Boltzmann constant ($=5.7 \times 10^{-8}$ W/m²K⁴), T_a is far-field temperature.

In the process of train braking, the convective heat transfer coefficient of the brake disc surface at different positions and at different times is different. In other words, the convective heat transfer coefficient varies with space and time [27–30].

2.3 Thermo-Elastic Stress Field

The transient temperature field is taken as the load condition into the following formula to obtain the thermal stress field of the brake disc. If the strain tensor ε and Cauchy stress tensor σ are proportional to the temperature variation [20], then

$$\{\varepsilon_0\} = \alpha(T - IT_0) = \alpha \Delta T, \tag{5}$$

$$\{\varepsilon\} = \nabla_s \mathbf{u}, \tag{6}$$

$$\{\sigma\} = D\alpha \Delta T, \tag{7}$$

where I and D are the unit vector and the elastic matrix respectively, and α is the thermal expansion coefficient of the material of brake disc. Then the governing equations and the corresponding boundary conditions of the thermo-elastic problem in the single problem domain Ω are obtained,

$$\nabla \cdot \sigma + \mathbf{b} = 0, \tag{8}$$

$$\mathbf{u} = \mathbf{u}_0, \tag{9}$$

where \mathbf{b} and \mathbf{u}_0 are volume force ($=0$) and symmetrical boundary displacement condition ($=0$).

3 Full Scale Bench Test

An emergency brake test was carried out on a full-scale brake dynamic test bed (ID281975) at high speed. Thermocouples were arranged 3 mm below the middle of the friction ring of the brake disc, every 120° and at the bolt



Figure 4 Distribution diagram of the thermocouple of the brake disc friction ring

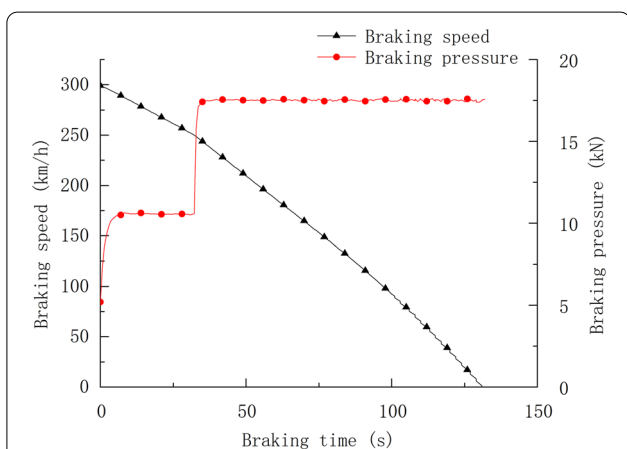


Figure 5 Change of braking pressure with speed at initial speed of 300 km/h

root, as shown in Figure 4. The temperature of the brake disc friction ring and bolt was recorded in real time.

According to the pressure curve of the emergency braking of a high-speed train, the pressure in emergency braking should be applied in stages during the test. If the initial braking speed is 250 km/h and below, the braking pressure is 17.5 kN; otherwise, the braking pressure is 10.5 kN. The relationship between brake pressure and speed is shown in Figure 5.

Before the test, 20 emergency braking events with an initial speed of 120 km/h were carried out to reduce the influence of uneven contact on the temperature field

distribution of the brake discs. The temperature increases of the brake disc and connecting bolt under emergency braking conditions with initial speeds of 160–350 km/h were measured in the test, as shown in Table 3.

4 Calculation Results

The temperature field, displacement field, stress field and bolt temperature of the wheel-mounted brake disc under 300 km/h emergency braking conditions are simulated and analyzed by using the established three-dimensional thermo-mechanical coupling model of the wheel-mounted brake disc with bolts and considering the contact relationship of each component.

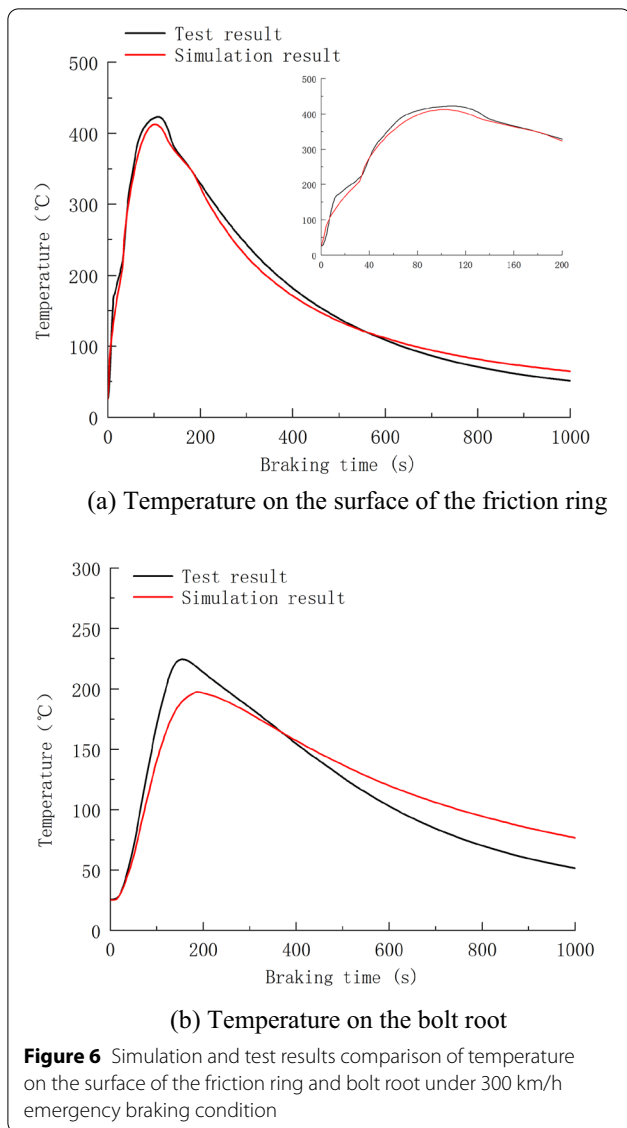
4.1 Temperature Field Distribution

The comparison between the calculated value and the test value of the surface temperature of the friction ring during emergency braking with an initial speed of 300 km/h is shown in Figure 6(a), in which the braking time is 130 s. At the beginning of braking, the surface temperature of the friction ring increases rapidly and reaches the maximum temperature before the end of braking. The highest temperature does not occur at the end of braking, and the temperature decrease starts from the balance between the heat input and dissipation. After the brake, the natural convection heat transfer begins, and the rate of the decrease in temperature tends to be gentle with increasing time. In the first 32 s, the rate of increase of the surface temperature of the friction ring increases because the speed decreases to 250 km/h, the braking pressure increases from 10.5 kN to 17.5 kN, and the heat flux input increases. The calculated value of the highest temperature of the surface of the friction ring is 414 °C (102 s after the start of braking), and the test value is 423 °C (109 s after the start of braking), with an error of 2%.

The comparison between the calculated value and the test value with time is shown in Figure 6(b). The increase in the temperature of the bolt is mainly caused by the conduction of heat on the surface of the friction ring through the brake disc body, so the rate of the increase in temperature is lower than that of the surface of the friction ring of the brake disc and the highest temperature occurs after the end of braking, later than that of the surface of the friction ring. The calculated

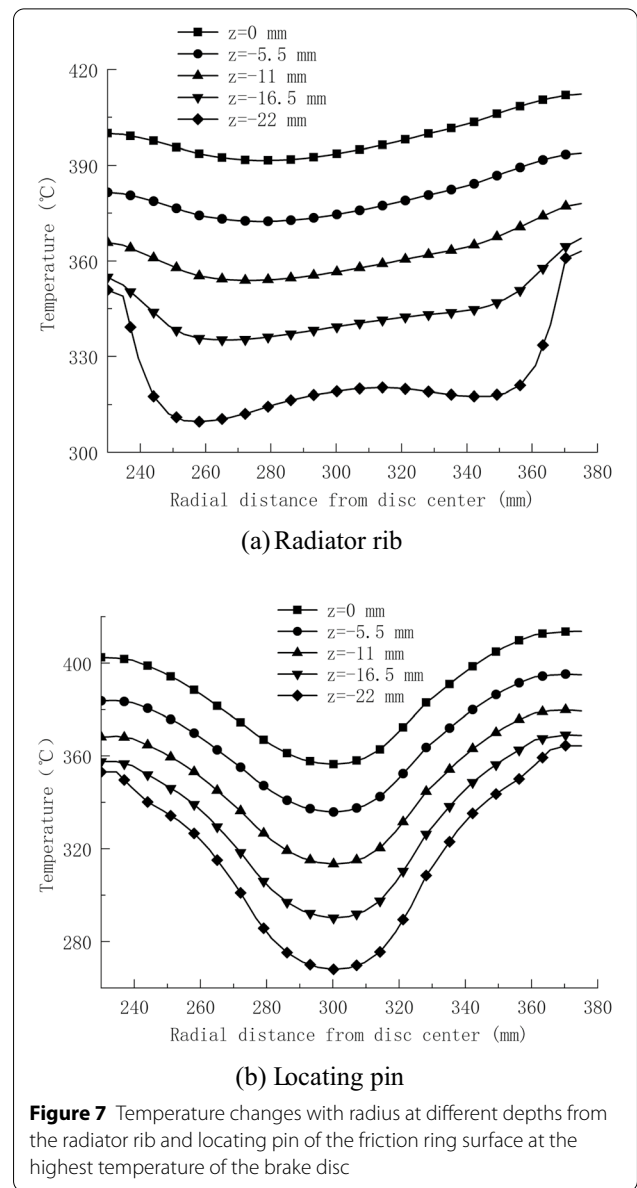
Table 3 Maximum temperature rise of brake disc friction ring and bolt in emergency braking with different initial speeds

Initial velocity (km/h)	160	200	250	280	300	330	350
Braking time (s)	43	59	83	104	130	151	146
Maximum temperature rise of friction ring surface (°C)	217.2	303.6	357.0	404.4	397.5	455.4	554.3
Maximum temperature rise of bolt (°C)	57.4	90.8	155.6	184.5	199.0	227.8	263.4



value of the maximum temperature of the bolt is 197 °C (185 s after the start of braking), and the test value is 225 °C (154 s after the start of braking), with an error of 12%. The error mainly comes from the material parameters (such as the change of thermal conductivity and specific heat of the material), contact state (there are complicated contacts in the whole assembly, and the contact relationship leads to the errors between FE simulation and test result), thermocouple installation (the contact state of the thermocouple and the bolt will change with the rotation of the brake disc) and other factors.

Figure 7(a), (b) shows the change in temperature with radius at different depth positions of the radiator rib and locating pin of the friction ring at the highest



temperature of the brake disc, where $z=0$ represents the surface of the friction ring. The temperature of the outer diameter is higher than that of the inner diameter, and the temperature of the middle part is lower than those of the inner and outer sides due to the effect of the radiator rib. Compared with the position of the radiator rib, the temperature difference between the center of the surface of the friction ring and the inside and outside diameter of the locating pin is larger, which is related to the larger volume and higher thermal capacity of the locating pin.

The temperature contour of the brake disc at the time of brake pressure change (32 s), the time of the highest

temperature on the surface of the friction ring (102 s), the end time of braking (130 s) and the time of the highest temperature at the end of the bolt (185 s) are shown in Figure 8. It can be seen that at the beginning of braking, the temperature changes uniformly. With the progress of braking, due to the influence of the geometry structure, heat conduction and heat dissipation, a larger temperature gradient appears on the surface of the friction ring, and a higher temperature and temperature gradient still exist at 55 s after braking.

4.2 Stress Field Distribution

The radial stress and circumferential stress at the bolt hole of the brake disc friction ring are shown in Figure 9. Figure 10 shows the circumferential stress contour of the brake disc at different time. Due to the influence of the bolt connection and temperature gradient, the circumferential stress of the brake disc is concentrated on the bolt hole and its corresponding edge of the friction ring during the braking process. The maximum circumferential compressive stress and tensile stress at the bolt hole are 658 MPa and 134 MPa. High alternating thermal stress range occurs in the braking process, and thermal fatigue cracks are more likely to be generated at the bolt hole of the wheel-mounted brake disc under the action of multiple emergency braking cycles.

4.3 Distribution of Thermal Deformation Field

The axial (normal) deformation contour of the brake disc at the moment of the maximum circumferential

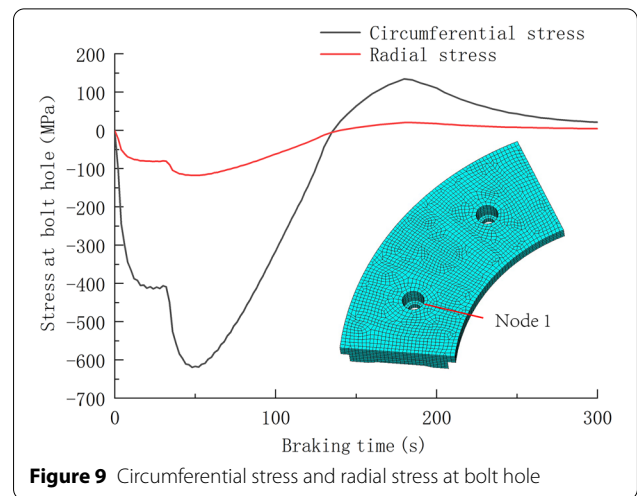


Figure 9 Circumferential stress and radial stress at bolt hole

compressive stress (48 s) on the friction ring surface is shown in Figure 11. Because of the bolt connection, the deformation of the bolt hole and locating pin affected by the temperature and mechanical constraints is less than that of the radiator rib, so it produces a higher compressive stress. In addition, Figure 12 indicates that the axial deformation of the middle part of the friction ring (radial $R=318$ mm) is larger than that of the edge. Specifically, during the braking process, the central part of the whole disc body bulges outward, which not only increases the axial tensile load of the connecting bolt but also reduces the contact pressure between the radiator rib of the disc

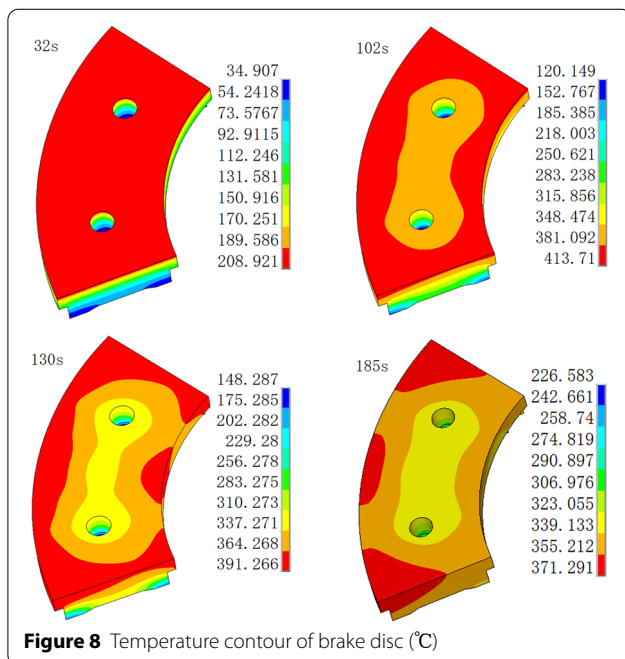


Figure 8 Temperature contour of brake disc (°C)

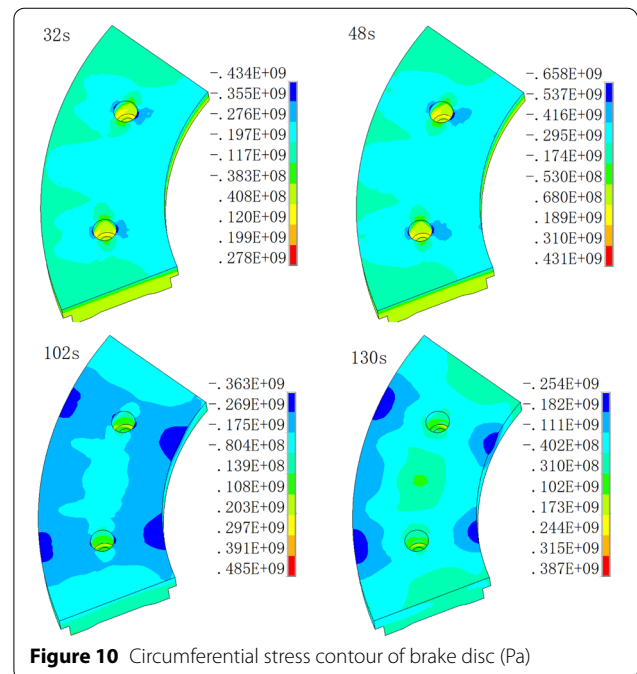


Figure 10 Circumferential stress contour of brake disc (Pa)

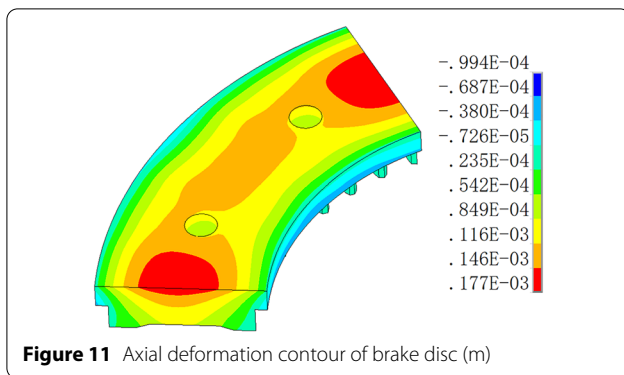


Figure 11 Axial deformation contour of brake disc (m)

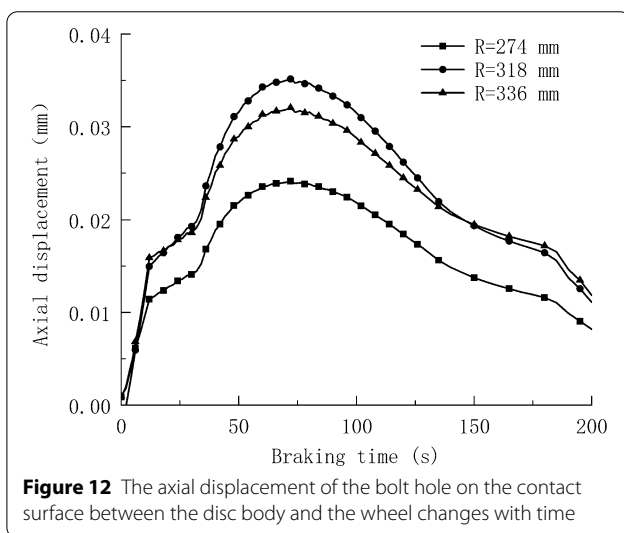


Figure 12 The axial displacement of the bolt hole on the contact surface between the disc body and the wheel changes with time

body and the wheel and reduces the contact friction. The stress state of the connecting bolts deteriorates continuously under the load of the wheel rail, so there is a risk of looseness and fracture.

5 Conclusions

To reveal the cause of the fatigue cracking on the edge of the bolt hole on the surface of the brake disc of a high-speed train, a 3D thermo-mechanical coupling model of the wheel-mounted brake disc of the high-speed train is established in this paper, which includes the bolt and considers the contact relationship between the components. The temperature field and the stress field of the brake disc under the emergency braking condition of the initial speed of 300 km/h are simulated and analyzed and verified by a full-scale bench brake test. The main conclusions are as follows.

- (1) In the process of braking, there are two inflexions in the temperature increase curve due to the variable braking pressure. The calculated value of the high-

est temperature on the surface of the friction ring is 414 °C, the test value is 423 °C, and the error is 2%. This verifies the accuracy of the calculation model and method.

- (2) The distribution of the temperature field of the brake disc is greatly affected by the geometry, and there are temperature gradients in the radial, circumferential and axial directions. In the process of braking, the main thermal stress is circumferential stress, the maximum alternating thermal stress is located at the edge of the bolt hole, the maximum circumferential compressive stress is 658 MPa and the maximum circumferential tensile stress is 134 MPa. That is, the fatigue crack is more likely to initiate from the hole edge after multiple braking events.
- (3) During the braking process, the axial deformation of the middle part of the friction ring is larger than that of the edge, and the thermal deformation of the disc body increases the axial tensile load of the connecting bolt and reduces the contact pressure between the radiator of the disc body and the wheel. The superposition effect of this working condition and wheel rail vertical impact load should be considered in bolt selection and design to prevent bolt loosening and fracture.

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Author Contributions

Wenjing Wang was responsible for the paper programming and test. Junsheng Qu carried out the simulation, test and drafted the manuscript. Ziyu Dong performed the test data processing. Wei Shan provided the test condition. All authors read and approved the final manuscript.

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Declarations

Competing Interests

The authors declare no competing financial interests.

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