Thermomechanical Coupling Simulation and Analysis of Wet Multi-Disc Brakes During Emergency Braking

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Abstract. The main reason for the thermal failure of wet multi-disc brakes during emergency braking is thermoelastic instability due to thermomechanical coupling. To analyse the phenomena, a three-dimensional finite element model of a wet multi-disc brake friction pair is built and numerical simulation is carried out with thermal-structural coupling method, based on ANSYS software. The contact pressure distribution, temperature and stress field during emergency braking are obtained. The results show that high temperature is located at the outer diameter of friction linings during emergency braking, large temperature gradient and stress gradient are generated at edges of radial grooves, which lead to the friction disc warp. Through simulation and analysis, this paper can provide a basis for engineering application and design research of wet multi-disc brakes.

1. Introduction
Wet multi-disc brakes are increasingly being used in heavy-load construction machinery due to the advantages of large braking torque, long service life, and strong resistance to heat fade, etc [1]. Thermal failure is the main failure form of wet multi-disc brakes. A large number of studies [2] reveal the coupling relationship between temperature, stress, and contact pressure during braking, and when the thermal-structural coupling exceeds the critical condition, it will lead to thermoelastic instability (TEI) of brake discs, resulting in great temperature gradient and stress gradient in the contacting components, which is the origin of various thermal failure problems of friction discs.

Numerical methods are generally used to analyse the thermomechanical phenomenon in brakes. A transient analysis of the thermoelastic contact problem of a disc brake was performed using the finite element method by Choi [3]. Belhocine [4] obtained contact pressure on the surface of the brake disc by building a three-dimensional thermomechanical coupling model. The effects of sliding speed, braking pressure and initial temperature on brake friction performance under different braking conditions were investigated in the article by Zhang [5]. The distribution of stress and strain caused by the thermoelastic instability in wet multi-disc clutches was analysed and tested by Sun [6]. Sun [7] also carried out a thermal reliability analysis of wet brake under the coupling of temperature field and stress field during emergency braking and continuous braking. Limited by computational resources, some scholars simulated frictional heat generation by inputting the heat flux density function calculated from the initial contact pressure, and some scholars used the sequential coupling method. Both of these models cannot accurately describe the coupling phenomenon during the braking process. Some scholars also ignore the radial grooves on the friction disc in their simulations.

During emergency braking, the frictional heat caused by the large sliding speed and brake pressure is much higher than the heat conduction to the inside of discs and the heat exchange by convection,
which easily leads to TEI and cause serious safety problems. This paper establishes a three-dimensional finite element (FE) model of the brake disc friction pair and carries out the simulation with thermal-structural coupling method during emergency braking based on ANSYS software. The causes of the thermal ablation and friction discs warp are revealed through analysis of contact pressure distributions, temperature field and stress field, providing a basis for the engineering application and design research of wet multi-disc brakes.

2. Structure and operating characteristics of the wet multi-disc brake

Figure 1 shows the structure of the wet multi-disc brake applied to an electric deck crane. The main components include the brake case, the brake piston, the return springs, the spline shaft, and the friction pairs that consist of multiple friction discs and steel discs. The friction discs are connected to the brake case, and steel discs rotate with the shaft through splines. A friction disc consists of a core board and friction linings of copper-based powder metallurgical material, and radial grooves are provided on both sides of the friction disc.

When braking, the hydraulic oil pressure is unloaded and the spring pushes the piston to compress friction discs and steel discs so that the rotating shaft is slowed down by the friction force provided by friction pairs. With this structure, the wet multi-disc brake can still provide a normal braking force in the event of a hydraulic system failure, ensuring the safety of construction machinery during operation.

The electric deck crane has the following two main operating conditions. One is the normal operating condition where the motor first reduces the shaft speed to around 20rpm and then the brake works to complete the park. The other is the emergency operating condition, a full-speed brake under a loss of power release condition with the initial speed of 1650rpm, and the emergency brake is completed only by the friction torque provided by the brake.

3. Thermomechanical coupling model

3.1. Mathematical model
Frictional heat flux density $q(x, y, t)$ generated by the contact surface during sliding friction is

$$q(x, y, t) = f v(x, y, t) p(x, y, t)$$

(1)
Where $f$ is the friction factor of the dynamic and static friction discs, $v(x,y,t)$ is the relative sliding velocity of the dynamic and static friction discs, $p(x,y,t)$ is the contact pressure of the dynamic and static friction discs.

Most of the frictional heat flux generated on the contact surfaces will be absorbed by the dynamic and static friction discs, while some will be absorbed by the lubricating film, particulate matter, etc. The proportional coefficient is $\delta$.

$$q(x,y,t) \times \delta = \lambda_s \frac{\partial T_s}{\partial y} - \lambda_f \frac{\partial T_f}{\partial y}$$  \hspace{1cm} (2)

where $\lambda_s$, $\lambda_f$ are the thermal conductivity of the dynamic and static friction discs respectively.

The boundary condition on the contact surface is that the temperature at the actual contact point on the friction pair is continuous.

$$T_s(x,y,t) = T_f(x,y,t)$$  \hspace{1cm} (3)

where $T_s$, $T_f$ are the contact point temperatures of the dynamic and static friction discs respectively.

During braking, there is not only thermal but also mechanical strain on the friction disc, the total strain $\varepsilon$ on the friction disc is a superposition of thermal and mechanical strain.

$$\varepsilon = \varepsilon_E + \varepsilon_T$$  \hspace{1cm} (4)

Total stress $\sigma$ in the friction disc is:

$$\sigma = D \cdot (\varepsilon_E + \varepsilon_T)$$  \hspace{1cm} (5)

where $\varepsilon_E$ is the mechanical strain, $\varepsilon_T$ is the thermal strain, and $D$ is the elastic matrix of the material.

### 3.2. FE model and boundary conditions

The transient thermal-structural coupling simulation of the friction pair during emergency braking is carried out in the ANSYS transient structural module. In the FE model, the spline structure that has less impact on the contact is simplified, instead, radial supports are applied to the outer circular surface of the friction disc and the inner circular surface of the steel disc, a joint with rotational velocity is added to the steel disc, and brake pressure is applied to the surface where the piston acts. The frictional contact is defined and thermal-structural coupling is directly achieved through contact elements CONTA174 and TARGE170, which contain temperature and displacement degrees of freedom. It is assumed that the friction factor $f$ is a constant and the heat transfer coefficient of the contact section is defined according to empirical data as 30000W/(m$^2$K). Table 1 shows the properties of each material of the brake disc, Table 2 shows the emergency braking parameters and Figure 2 shows the emergency braking characteristics.

#### Table 1. Material properties used in the simulation.

| Material properties                  | Steel disc | Core board | Friction lining |
|--------------------------------------|------------|------------|-----------------|
| Density, $\rho$ (kg/m$^3$)           | 7850       | 7820       | 6700            |
| Elastic modulus, $E$ (GPa)           | 206        | 211        | 5               |
| Poisson’s ratio, $\mu$               | 0.3        | 0.3        | 0.27            |
| Thermal Conductivity, $k$ (W/(m$^2$K)) | 60.5      | 50         | 43              |
| Specific heat, $c$ (J/(kg$^\circ$K)) | 434        | 477        | 460             |
| Thermal expansion coefficient, $\alpha$  \hspace{1cm} (10$^{-5}$/K) | 1.2        | 1.2        | 1.71            |

#### Table 2. Emergency braking parameters.

| Friction factor $f$ | Max. brake pressure $p_{max}$ (MPa) | Initial velocity $\omega_0$ (rad/s) | Braking time $t$ (s) | Initial temperature $T_0$ ($^\circ$C) |
|---------------------|-------------------------------------|-------------------------------------|---------------------|--------------------------------------|
| 0.1                 | 1.49                                | 172.79                              | 0.5                 | 30                                   |
4. Results and discussion

4.1. Temperature distribution

Figure 3 shows the maximum and minimum temperature of the frictional pair at various moments of simulation. The maximum temperature rises rapidly at the beginning of braking and reaches its maximum value at \( t=0.385 \) s. During 0.38–0.5 s, the generated frictional heat flux decreases with the reduction of sliding speed, and the maximum temperature increases slowly and gradually starts to decline under the effect of heat conduction, while the minimum temperature continues to rise. During 0.5–2 s, the braking is complete, with the transfer of heat, the temperature difference of the frictional pair decreases, and the temperature distribution gradually becomes uniform. When \( t=2 \) s, the temperature varies from 51 to 56°C.

Figure 4 presents the temperature field of the contact surfaces when the temperature reaches a maximum during emergency braking (\( t=0.385 \) s). The temperature is continuous in the circumferential direction due to the rotation but shows a tendency to rise and then fall in the radial direction. The highest temperature appears in the 52–58.6 mm radius area. As the radial grooves divide the friction contact surfaces, the temperature in the radial grooves of the friction disc is much lower than that on contact surfaces, thus, a large temperature gradient occurs at the edges of the radial grooves, resulting in higher thermal stress.
4.2. Von-mises stress distribution

The stress in the brake disc increases continuously during the braking and reaches its maximum value at \( t = 0.5 \text{s} \). Figure 5 shows the von-mises stress distribution of the brake disc at this moment. The maximum value is located at the inner diameter of the radial grooves on the contact surface of the friction disc, approximately 139.47 MPa, with a large stress gradient, caused by the great temperature gradient at this location. The maximum von-mises stress value on the steel disc is located at the inner diameter, which is mainly caused by the radial supports applied. It is obvious that the stress concentration generates at the edge of the radial groove on the contact surface. It is also noted that the circumferential stress of the friction disc is significantly higher than the radial and axial stress, with a maximum value of approximately 163.21 MPa, which is likely to cause radial cracks in the friction disc.

4.3. Contact pressure distribution

Figure 6 presents the contact pressure distribution at different moments. During the emergency braking, the large frictional heat flux generated at the outer diameter of the lining causes uneven thermal expansion of the material, resulting in an increase of contact pressure with a maximum value of 3MPa, significantly higher than the applied braking pressure of 1.49MPa. The contact pressure distribution gradually becomes uniform as heat transfer proceeds after the braking (\( t = 1 \text{s} \)), and the maximum value declines to around 2.8MPa.

The simulation results clearly reveal the thermomechanical coupling phenomenon during emergency braking. The uneven temperature rise on contact surfaces caused by radially varying sliding speed directly affects the contact pressure distribution, while the change in the contact pressure distribution will in turn strengthen the unevenness of the frictional heat flux generated. Under these
circumstances, once the temperature exceeds the allowable value, thermal ablation damage will occur in the high-temperature area on the steel disc.

![Contact pressure distribution](image)

**Figure 6.** Contact pressure distribution (a) $t=0.25s$, (b) $t=1s$.

### 4.4. Warp characteristics of friction disc

Due to uneven frictional heat flux on the friction contact surface, the thermal expansion at the outer diameter of the lining is higher than that at the inner diameter, and the thermal expansion of the contact surface is higher than that of the core plate and the radial grooves. The thermal stress, brake pressure, and mechanical restraint together lead the friction disc to warp.

As shown in Figure 7(a)(c), a group of equidistant nodes are taken in the radial and circumferential direction of the friction disc, and the axial deformation of them is recorded in Figure 7(b)(d) respectively. With the increase of radius, the axial deformation shows a trend of increasing and then decreasing, which is consistent with the temperature and contact pressure distribution. The axial deformation of nodes in the radial direction also rises with the frictional heat generation and transfer during the braking process. At all times, the axial deformation at the edges of the linings is slightly larger than that in the middle, and the axial deformation of the lining is significantly larger than that of grooves, causing a large deformation gradient during the braking process. After braking, the stress gradient and axial deformation gradient gradually decreases as the temperature field of the friction disc becomes uniform, but the overall deformation still shows an increasing trend as the thermal expansion area continues to expand.
5. Conclusion

In this paper, a FE model of the wet multi-disc brake friction pair is built in ANSYS software, and the thermal-structural coupling simulation of the friction pair during emergency braking is carried out to provide a basis for the engineering application and design research of wet multi-disc brakes. The main conclusions of this paper are as follows.

- During the emergency braking, the large frictional heat flux and the contact pressure distribution on friction contact surfaces are coupled and mutually reinforcing, and high temperature is located at the outer diameter of friction linings, which can easily lead to thermal ablation.
- The friction disc warps during the emergency braking, and great temperature gradient and stress gradient are generated at the edges of radial grooves. The high circumferential stress tends to cause radial cracks in the friction disc.

Acknowledgments

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Reference

[1] Zhao W Q and Wang C S 2003 A review of relevant studies on wet multiple brake Acta Armamentarii 24 111–4
[2] Yevtushenko A A, Grzes P and Adamowicz A 2015 Numerical Analysis of Thermal Stresses in Disk Brakes and Clutches (A Review) Numerical Heat Transfer 67 170–88
[3] Choi J H and Lee I 2004 Finite element analysis of transient thermoelastic behaviours in disk brakes Wear 257 47–58
[4] Belhocine A and Bouchetara M 2012 Simulation of fully coupled thermomechanical analysis of automotive brake discs SIMULATION 88 921–35
[5] Zhang C, Zhang T, Li X, Lv Y and Zhong B 2019 Effect on friction performance of mining wet brake under different working conditions Mechanics of Advanced Materials & Structures 1–16
[6] Sun D Y, Hu F B, Deng T and Luo Y 2010 Simulation and experiment for warp characteristic of wet multiple disc clutches Journal of Chongqing University 33 1–6
[7] Sun D Y, Liu S, Hao Y Z, Luo Y, Wang Y and Qin D T 2019 Thermal reliability analysis of in wheel wet brake assembly under high-intensity braking with multi-field coupling Automotive Engineering 41 161–9