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Contact Stress Analysis of Disc Cutter Peg-hole with Interference Fit

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Abstract

In order to study the relation of interference fit between cutter body and cutter ring, theoretical design of interference fit of the cutter peg-hole is carried out according to the theory of thick walled cylinder (TCT). The finite element method is used to analyze the mapping relationship between the contact stress of cutter ring and cutter body. The thermal expansion assembly of the cutter ring is simulated, and the temperature of the thermal expansion assembly of the cutter ring is determined. The result shows the contact length between the cutter body and the cutter ring has little influence on the contact stress of the cutter ring, but with the increase of the contact length the stress concentration of the retaining ring will increase. The contact stress of the cutter ring is basically the same at the same amount of interference and temperature, but the contact stress of the ring decreases with the increase of the contact length at high temperature, and when the contact length is the same, the contact stress and the radial force of the mating surface increase with the increase of the amount of interference. The reasonable temperature of the thermal expansion assembly of the cutter ring is 160°C.

Key words : Tunnel boring machine Disc cutter Interference fit Contact stress Thermal expansion assembly

1 Introduction

With the development of underground constructions such as subway construction, water diversion, urban comprehensive pipeline gallery and mineral exploitation, the application of TBM (Tunnel boring machine) is becoming more and more extensive. The front end of the tunnel boring machine is installed with a cutterhead, which mainly undertakes the functions of cutting rock soil and stabilizing the palm surface, the rock breaking is mainly completed by the disc cutter installed on the cutterhead, so the disc cutter directly affects the excavation efficiency and tunnelling performance of the TBM (Tan et al. 2013). The disc cutter is one of the key parts of the TBM, and it runs mainly depending on the rolling of the cutter ring and rock during the tunnelling process. The cutter ring is subjected to the action of cutting force, radial force and axial force during the rock breaking process. It is subjected to great torque and impact. The wear of the cutter ring is serious and the consumption is huge, and the cost of its loss is about 1/3 of the total cost of the process (Entacher et al. 2012; 2013). Due to the force characteristics of the disc cutter in construction, the interference fit connection between the cutter ring and the cutter body is adopted. The connection mode of this peg-hole interference fit connection with other peg-holes has the characteristics of simple structure, high positioning precision, strong load bearing capacity and reliable working in vibration and shock environment. In the past, the disc cutter mainly depended on the import. In recent years, the domestic disc cutter manufacturers have actively developed the cutter, and the performance of the disc cutter has been improved gradually. However, because of the improper design or assembly process between

the cutter body and the cutter ring, the performance degradation or even failure of the disc cutter often occurs. For example, the fracture of the cutter ring, the cracking of the weld seam and the cutter ring falling off, and so on. The contact stress of the cutter ring is large if the interference amount of the peg-hole between the cutter body and the cutter ring is large. Under this condition, the cutter ring is easily cracked when it is subjected to a large impact load during the rock breaking. On the other hand, the peg-hole interference between the cutter body and the cutter ring is small, so the contact stress of the cutter ring is small. In the process of rock breaking, the contact disc cutter is caused by the friction between the cutter ring and the rock, and the cutter ring and the cutter body is easy to slip. These two cases all can cause the failure of the disc cutter. Scholars at home and abroad have made extensive researches on the corresponding problems of interference fit connection of peg-holes, and have achieved certain results. [Zhang et al.\(2000\)](#) used the finite element method to analyze the stress in the interference connection of the peg-hole([Doležel et al. 2014](#) and [Wang et al. 2017](#)). This method has the quantitative advantage compared with the traditional design method. The research provides the quantitative guidance for the design of the interference fit of the peg-hole. [McMillan et al.\(2016\)](#) found that the interface pressure of the rolls used for the hot rolling of steel sections is not increased to maintain the torque, and the friction coefficient between the shaft and the hub is an important factor responsible for maintaining the torque increase through the neutron diffraction experiment and the crack flexibility test. It provides the basis for the disc cutter peg-hole design of interference fit. [Chakherlou et al. \(2010\)](#) tested the fatigue life of aluminum alloy and steel shaft under different interference amount through fatigue test, and simulated the stress distribution of interference fit with three-dimensional finite element. The reason for improving fatigue life of disc aluminum alloy parts is analyzed. It provides a method for stress analysis of interference fit of disc cutter peg-hole. [Luo et al. \(2013\)](#) developed a precision assembly and testing system, designed a compliant clamping mechanism, realized the interference fit assembly of multi ring parts, overcame the assembly clamping resistance caused by the radial orientation deviation between the assembled parts and the die assembly holes, and put forward the assembly force/stiffness position method. However, this assembling method can be used to fit the peg-hole with small interference. It is not suitable for assembling the peg-hole with large interference of the disc cutter. [Shi et al. \(2017\)](#) proposed a deterministic model based method to analyze the contact behavior of each scale on the rough surface, established the deterministic contact model of each scale, and analyzed the evolution of contact area with the normal contact pressure on the metal surface at various scales. It provides reference for the analysis of contact stress between disc cutter ring and cutter body. The pressure of loading of bearing interference fit in the hydraulic torque converter is simulated and tested by [Butugeqi \(2017\)](#) , and the thermal assembly deformation of the guide wheel of the torque converter is analyzed, it provides technical support for thermal assembly process parameters between disc cutter ring and cutter body. However, there are few reports on the design and contact stress analysis of the interference fit between the cutter body and the cutter ring of the disc cutter. Therefore, the interference fit of the disc cutter peg-hole is studied in this paper. The influence on the interference fit of the disc cutter from geological factors is considered. At the same time, the contact stress of the disc cutter is analyzed by the finite element method, and the thermal assembly process of the cutter ring is simulated, and the mapping between the interference and contact

stress between the cutter body and the cutter ring is revealed. It is of great significance to improve the performance of disc cutter and for the development of major national tunnel projects.

2 Structure composition and force model of disc cutter

2.1 The structure of disc cutter

The structure diagram of the disc cutter is shown in Fig. 1, which is mainly composed of the cutter shaft, the cover, metal floating oil seal, retaining rings, cutter ring, cutter body, O type sealing strip and tapered roller bearings. The cutter is mounted on the cutter seat through the cutter shaft, and the cutter seat is fixed on the cutterhead. During the construction of the TBM, the cutterhead is rotating while advancing in both positive and negative directions while advancing. Thus the cutter mounted on the cutterhead is rotated with the rotation of the cutterhead, and revolve on its own axis under the action of the cutter ring and rock friction. Rock breaking is carried out by rolling cutter ring on the rock. The sketch map of the disc cutter installation is shown as Fig. 2.

2.2 The force model of disc cutter

The schematic diagram of the force model of disc cutter at work is shown in Fig. 3 (Balci 2009). The cutter is affected by vertical counter force F_N , side force F_S and rolling force F_R from rock face. Because of the constant change of the rock, the larger radial force and axial force will be received by the disc cutter rolling and breaking rock, with larger impact load accompanied. Therefore, the method of interference assembly between the cutter ring and the cutter body of the disc cutter can be adopted to ensure the cutter can work stable and reliable.

Fig.1 The structure of disc cutter

Fig.2 The sketch map of disc cutter installation

Fig.3 The schematic diagram of the force of disc cutter

3 Interference fit design of disc cutter peg-hole

As shown in the force model of disc cutter, the cutter body and the cutter ring are in interference with the peg-hole because of the complicated force in the rock breaking process and the convenience of the cutter ring and the cutter body assembly. In the process of disc cutter rock breaking, the cutter ring is rotated with the rotation of the cutterhead under the action of rock friction. The long time work the disc cutter will be heated by friction. Therefore, the interference between the cutter body and the cutter ring is not only to ensure that the cutter ring does not rotate in relative rotation and the radial and axial movement, but also produces the proper contact stress to ensure the performance of the disc cutter.

This paper takes the 19 inch disc cutter as an example to design the interference fit between the cutter body and the cutter ring. The diameter of the 19 inch disc cutter body is 324mm, the contact length between the cutter body and the cutter ring is among the 84mm~89mm(Acaroglu 2011). When the disc cutter breaks rock, the uniaxial compressive strength of the rock is generally within 250MPa. At this time, the theoretical torque value of the cutter ring is obtained according to the CSM model of the disc cutter breaking rock. The CSM model is suitable for the engineering calculation of the disc cutter size of 5 to 19 inches and the rock compression strength range from 30 to 250MPa. It is based

on the mathematical model and has been verified by a large number of experiments with the linear cutting experimental platform. It has high accuracy and can be used to estimate the resultant force in the working process of the disc cutter (Rostami 2013 and Li et al. 2014). The specific force model is shown as follow:

$$F_t = \frac{P^0 \phi RT}{1 + \psi} \quad (1)$$

In the formula, F_t is the force of the disc cutter (kN), R is the radius of the disc cutter (mm), the T is the width of the disc cutter blade (mm), ψ is the pressure distribution coefficient of the blade tip, which decreases with the increase of the blade width, which is -0.2 to 0.2, generally 0.1, and ϕ the contact angle of the cutter blade with the rock.

$$\phi = \arccos\left(\frac{R - p}{R}\right) \quad (2)$$

In the formula, p is a disc cutter penetration (mm/rev), P^0 is the pressure of the rock crushing zone, which is related to the rock strength, the geometry of the disc cutter, the penetration and the spacing of the disc cutter, and the pressure of the rock breaking zone P^0 can be expressed as:

$$P^0 = C_3 \sqrt{\frac{S}{\phi \sqrt{RT}}} \sigma_c^2 \sigma_t \quad (3)$$

In the formula, C is a constant similar to the contact angle of rock, with a value of 2.12, S is the spacing of the cuts, σ_c is the uniaxial compressive strength of rock and σ_t is the tensile strength of rock. In either case, to estimate the normal and rolling forces, the following formulas can be used (Rostami et al. 1996).

$$F_N = F_t \cos\left(\frac{\phi}{2}\right); \quad F_R = F_t \sin\left(\frac{\phi}{2}\right) \quad (4)$$

Where: F_N is the Normal force, F_R is the Rolling force.

The diameter of the 19 inch standard type disc cutter is 482.6mm, the width of the cutter ring tip is 19mm, the rock compressive strength is 250MPa, the rock tensile strength is 25MPa, the cutter space is 80mm. According to the use experience of disc cutter in the hard rock construction, the penetration is 3mm. The rock breaking force of the disc cutter is calculated to be 316.59kN, and the torque on the cutter ring is calculated, the theoretical value is $6.02 \times 10^6 \text{ N} \cdot \text{mm}$. From the structure of the disc cutter in Figure 1, it is known that the shaft shoulder and the retaining ring are fixed in the axial direction of the cutter ring and the cutter body. The axial movement of the cutter ring is not very likely. It is mainly the relative rotation of the cutter ring and the cutter body. By calculation, the minimum pressure of the cutter ring and the cutter body does not rotate is (Song et al. 2018):

$$p_{\min} = \frac{2M}{\pi D^2 h f} \quad (5)$$

In the formula, M is the maximum torque of the interference connection. The value is $6.02 \times 10^6 \text{ N} \cdot \text{mm}$, D is the diameter of interference fit surface, the value is 324mm, h is the interference fit length, the value is 84mm, f is the friction coefficient, and the value is 0.11. After substituting the data for calculation, it can be obtained the $P_{\min} = 3.95 \text{ MPa}$.

According to the thick walled cylinder theory (TCT) (Wang et al. 2017), the minimum effective interference value of the cutter ring and cutter body without sliding is calculated when the disc cutter is working normally (Feng et al. 2017; Nie 2014; Martin et al. 2014). As follow:

$$\bar{\delta}_1 = \frac{P_{\min} D}{E_1} \left(\frac{d_1^2 + D^2}{d_1^2 - D^2} + \mu_1 \right) + \frac{P_{\min} D}{E_2} \left(\frac{D^2 + d_2^2}{D^2 - d_2^2} - \mu_2 \right) \quad (6)$$

In the formula, d_1 and d_2 are respectively the diameter of the containment and the contained parts. The values are 365 mm and 273mm respectively. E_1 and E_2 are respectively the modulus of elasticity of the containment and the contained parts. The values are $2.1 \times 10^5 \text{ MPa}$. The μ_1 and μ_2 are Poisson's ratio of containment and containment respectively, the value is 0.3. After substituting the data for calculation, it comes to the result $\bar{\delta}_1 = 0.122 \text{ mm}$.

After the assembly of the cutter ring and the cutter body, the change of the roughness of the mating surface reduces the actual interference^[19].

$$\bar{\delta}_2 = K_1 R_{a1} + K_2 R_{a2} \quad (7)$$

In the formula, R_{a1} and R_{a2} are respectively the surface roughness of the containment and the contained parts. The roughness coefficient is K_1 and K_2 , and $K_1 = K_2 = 5$ when $R_{a2} > 1.25 \mu\text{m}$. After substituting the data for calculation, it can be obtained the $\bar{\delta}_2 = 0.016 \text{ mm}$.

In addition, considering the heat generated by the friction between the cutter ring and the rock when the disc cutter is working, it is necessary to ensure the reliable connection between the cutter ring and the cutter body under the condition of temperature rise. According to the feedback information from the site construction, the maximum working temperature of the disc cutter is generally around 70°C , so the cutter ring is set at 70°C . The expansion of the cutter ring is affected by temperature as follow.

$$\bar{\delta}_3 = D \times (t_1 - 20) \times \alpha_1 \quad (8)$$

In the formula, t_1 is the working temperature of the cutter ring, and the value is 70°C . The α_1 is coefficient of thermal expansion of the material, and the value is $8.6 \times 10^{-6} / \text{K}$. After substituting the data for calculation, it can be obtained the $\bar{\delta}_3 = 0.139 \text{ mm}$.

Therefore, under the normal working conditions, the minimum effective interference between the cutter ring and the cutter body is as follow.

$$\bar{\delta} = \bar{\delta}_1 + \bar{\delta}_2 + \bar{\delta}_3 \quad (9)$$

After substituting the data for calculation, it can be obtained the $\bar{\delta} = 0.277 \text{ mm}$. According to the

minimum effective interference between the cutter ring and the cutter body, the design coordination relationship is $R7/s7$, the minimum interference is 0.277mm, and the maximum interference is 0.391mm.

4 Contact stress analysis of interference connection between cutter body and cutter ring

The contact length of the 19 inch disc cutter body and the cutter ring is generally 84mm~89mm. In order to compare the performance of the interference fit connection between the cutter body and the cutter ring at different contact lengths, we adopt the simulation and contact stress analysis on the interference length of the peg-hole with the contact length of 84mm and 89mm, respectively.

Firstly, the axisymmetric model is set up by the cutter ring, the cutter body and the retaining ring used the software ABAQUS. Secondly, the elastic body material is used for the cutter ring, the cutter body and the retaining ring. The modulus of elasticity is $2.1 \times 10^5 \text{MPa}$ and the Poisson's ratio is 0.3. The thermal expansion coefficient of the cutter ring and the cutter body is $9.1 \times 10^{-6}/\text{K}$ and $12.43 \times 10^{-6}/\text{K}$, respectively. Finally, the grid is divided by the CAX41 axisymmetric unit and the CAX3 axisymmetric unit, the total number of nodes is 5147, total number of elements is 4909, which the linear quadrilateral elements of type CAX41 is 4784 and the linear triangular elements of type CAX3 is 125. The boundary condition is that the cutter ring and the cutter body are fixed along the contact line and can move radially, and the cutter ring and the cutter body can not rotate each other. The contact between the cutter ring and the cutter body is set as follows: the friction coefficient is defined as 0.11 in the tangential direction of the cutter body and the cutter ring contact surface, the surface is elastic sliding, the sliding coefficient is set to 0.005, and the radial direction is set to hard contact. A finite element model for the simulation of interference fit of cutter body and cutter ring is established.

4.1 Contact stress analysis of different contact lengths at normal temperature

According to theoretical interference volume of interference fit of cutter peg-hole, the stress and contact stress curves of the cutter body and the cutter ring with a contact length of 84mm at the temperature of 25°C are shown in Fig. 4.

Fig. 4 Stress and contact stress curves of 84mm contact length at room temperature

From Fig. 4, it can be seen that the maximum equivalent stress point of the cutter ring appears at the transition circle of the inner diameter of the cutter ring when the interference amount of the cutter ring and cutter body is theoretical interference and the contact length is 84mm at 25°C. The maximum equivalent stress value is 117MPa and the average equivalent stress of the interference contact surface is about 88MPa. The contact stress at both ends of the contact surface of the cutter ring and the cutter body is more fluctuant, and the contact stress of the center contact is more stable. The maximum contact stress is 143MPa which appears at the transition round angle of the inner diameter of the cutter ring, and the average contact stress in the center area of the cooperating surface is about 15MPa.

Under the condition of 25°C, the interference amount of the cutter body and the cutter ring is the theoretical interference amount, the stress cloud chart and the contact stress curve of the cutter body

and the cutter ring with the contact length of 89mm, as shown in Fig. 5.

Fig. 5 Stress and contact stress curves of 89mm contact length at room temperature

It can be seen from Fig. 5 that the maximum equivalent stress point of the cutter ring appears at the transition circle of the inner diameter of the cutter ring when the interference amount of the cutter ring and cutter body is theoretical interference and the contact length is 89mm at 25°C. The maximum equivalent stress value is 118MPa, the average equivalent stress of the interference surface is about 88MPa, and the stress is basically the same as that of the contact length 84mm. The contact stress fluctuates at both ends of the contact surface of the cutter ring and the cutter body. The maximum contact stress appears at the transition round angle of the inner diameter of the cutter ring, the size is 124MPa, and the average contact pressure of the center area of the cooperating surface is about 13MPa. Under the corresponding interference amount, the contact stress value of the center area of the contact surface of the cutter ring is higher than that of the theoretical calculation. The contact stress of the 84mm cutter ring is higher than the corresponding contact stress value of the 89mm cutter ring, and the contact stress fluctuation is larger than that of the 89mm cutter ring. It is caused by the influence of thermal expansion and the smaller contact length.

4.2 Contact stress analysis of different contact lengths at high temperature

Considering the temperature rise of disc cutter working, the stress ring of the cutter ring at the theoretical interference level of 84mm and 89mm at 70°C is shown in Fig. 6.

Fig. 6 The stress of different contact length cutter ring at high temperature (Unit: MPa)

Fig. 6 shows that the maximum equivalent stress point of the disc cutter structure with a contact length of 84mm is located on the retaining ring at a temperature of 70°C. The maximum equivalent stress of retaining ring is 158MPa, and the maximum equivalent stress of the cutter ring is about 106MPa. The maximum equivalent stress point in the disc cutter structure with the contact length of 89mm is also located on the retaining ring, the maximum equivalent stress is 157MPa, the maximum equivalent stress of the cutter ring is about 105MPa, which is basically the same as that of the cutter ring with the contact length of 84mm. However, it can be seen through the stress analysis that the contact length of 89mm cutter ring is better than the contact length of 84mm cutter ring in the stress distribution uniformity, so the retaining ring of the 84mm cutter ring is more prone to stress concentration than that of the 89mm cutter ring, and the disc cutter with the 84mm ring is higher in the material and welding strength of the retaining ring. In summary, the contact length of the cutter ring and the cutter body has a certain influence on the interference performance and the distribution of the stress distribution of the retaining ring. The longer contact surface should be adopted in the case of the structure allowed, otherwise the better performance of the retaining ring and welding material should be selected.

4.3 Contact stress analysis of different interference amount of cutter ring in high temperature

According to the previous theoretical calculation, the relationship between the cutter body and the cutter ring is R_7/s_7 , and the minimum interference is 0.277mm. In order to analyze the optimum interference amount of the cutter ring under the different interference volume of the disc cutter, the

contact condition between the cutter body and the cutter ring with 0.277mm, 0.334mm and 0.391mm interference rings. Fig. 7 is a cloud map of the contact stress distribution between the cutter ring, the cutter body and the retaining ring when the cutter ring is different interference in the high temperature condition. Among them, the Fig. 7a, Fig. 7b and Fig. 7c graphs are the contact stress distributions of the interference volume of 0.277mm, 0.334mm and 0.391mm respectively. The Fig. 8 is a contact stress curve between the cutter ring and the cutter body at high temperature.

Fig. 7 Stress distribution of interference fit at high temperature (Unit: MPa)

Fig. 8 Contact stress contrast curve of cutter ring

From Fig. 7 it can be known that the thermal expansion of the cutter ring of the three matching relations occurs under the action of temperature load. The maximum equivalent stress points all appear on the contact surface of the cutter ring and the retaining ring of the cutter. The maximum equivalent stress is about 97MPa, and the different interference amounts have little influence on the stress of the components.

From the contact stress contrast curve in Fig. 8, we can see that the maximum contact stress of the structure appears on the transition circle at both ends of the cutter ring under three different interference amounts. The maximum contact stress is 60.7MPa when the interference is 0.277mm, and the average contact stress in the middle of the cutter ring is 5~7MPa. The maximum contact stress is 84.5MPa when the interference is 0.334mm, and the average contact stress in the middle of the cutter ring is 8~11.5MPa. The maximum contact stress is 107.3MPa when the interference is 0.391mm, and the average contact stress in the middle of the cutter ring is 12~16MPa.

The change curve of the radial force between the cutter ring and the cutter body is shown in Fig. 9.

Fig. 9 Radial force change curve of mating surface

According to the radial force variation curve in Fig. 9, the maximum radial force at three different interference levels is 1.872kN, 2.273kN and 2.666kN at room temperature respectively. However, the temperature increases the radial force decreases gradually. The maximum torque values of the three kinds of interference fit relations are $7.76\text{kN} \cdot \text{M}$, $12.66\text{kN} \cdot \text{M}$ and $20.05\text{kN} \cdot \text{M}$ respectively when the temperature rises to the working temperature of 70°C .

5 Simulation and analysis of the cutter ring heat assembly

It can be seen from the above that the amount of interference of the cutter ring and cutter body can meet the construction needs of the disc cutter. The performance of the disc cutter should be ensured by the assembly process after the interference amount is determined. Because the material of the cutter ring is generally H13 steel, according to the material and size of the cutter ring, the assembly process of the cutter ring and the cutter body is determined to be a thermal expansion assembly. The cutter ring is assembled on the cutter body after the cutter ring is heated to a certain temperature by a heating furnace. According to the thermal expansion effect of the cutter ring material, the inner diameter of the cutter ring is larger than that of the cutter body outer diameter and interference volume, after the cutter ring is cooled, the cutter ring and cutter body produce a predetermined interference volume.

5.1 Theoretical calculation of the assembly temperature of the heat expansion of the cutter ring

The maximum interference ratio of the cutter ring and the cutter body is 0.391mm, and the basic diameter of the cutter circle is 324mm through theoretical calculation. Refer to the design manual to confirm that when $D=324\text{mm}$, the maximum gap between the cutter body and the cutter body is 0.111mm (Nie 2014). According to the theory of metal linear expansion and the thermal expansion coefficient of the cutter ring material (H13 hot die steel), the thermal expansion temperature of the tool ring is calculated as the follow,

$$t = \frac{Y_{\max} + \Delta}{\alpha D \times 10^3} + t_0 \quad (10)$$

In the formula, $Y_{\max}$ is the maximum interference amount, the value is 0.391mm, Δ is the maximum assembly gap, the value is 0.111mm, the α is the thermal expansion coefficient, the thermal expansion coefficient of H13 steel is $10 \times 10^{-6}/\text{K}(100^\circ\text{C})$, t_0 is room temperature, the value is 25°C .

It can be calculated from the data to formula (10) that the installation temperature of the cutter ring thermal expansion is $t=153^\circ\text{C}$. In practice, the cutter ring is heated to more than 160°C and heat preservation.

5.2 Simulation and analysis of the heat assembly of the cutter ring interference

Simulation of thermal expansion of cutter ring by applying temperature load to disc cutter ring was done. The contact pressure change of the cutter ring and the cutter body is analyzed to verify the correctness of the installation temperature of the cutter ring thermal expansion theory.

Firstly, the interference fit model of the cutter ring and the cutter body is established. Considering the assembly clearance and the interference fit between the cutter ring and the cutter body, the total interference is set at 0.502mm. The temperature load of 180°C is applied to the ring of cutter. In order to show the change of interference with temperature, the whole analysis process is divided into 10 incremental steps. The temperature of the ring is reflected by the current incremental step.

The Fig. 10 shows the contact stress distribution of cutter ring and cutter body in different incremental step. Where the diagram in Fig. 10a is 0 s, the interference margin of the cutter ring and the cutter body has been set up to 0.502mm, and the temperature of the cutter ring is 25°C . As shown, the maximum contact stress of the cutter ring and the cutter body is 205MPa, and the average contact stress in the middle of the contact surface is about 34MPa.

The chart in Fig. 10b shows the contact stress distribution map of the cutter ring and the cutter body at 0.7seconds. From the graph, the maximum contact stress of the cutter ring and the cutter body is 20MPa when the cutter ring temperature is 140°C . The average contact stress in the middle of the contact surface is about 3.2MPa. In this situation, the amount of interference still exists between the cutter ring and the cutter body. The contact stress distribution map of cutter ring and cutter body is shown in Fig. 10c in 0.8seconds. It can be seen from the graph that the contact pressure between the cutter ring and the cutter body is 0MPa when the cutter ring temperature is 160°C . At this point, the interference volume between the cutter ring and the cutter body has completely disappeared.

The above analysis shows that the minimum temperature of the cutter tool ring installed is 160°C by the finite element simulation method. It is close to the installation temperature of 153°C of theoretical calculation. The temperature of the hot assembly is far below the minimum temperature of 225°C at the low temperature of the cutter ring, which reduces the influence of the heating cutter ring

on the hardness of the cutter ring during the thermal assembly. Finally the cutter ring thermal expansion installation temperature was determined to 160°C. It provides a scientific theoretical basis for the thermal expansion assembly of the cutter ring.

Fig. 10 Contact stress between cutter ring and cutter body in different time history (Unit: MPa)

6 Conclusion

(1) The peg-hole interference fit design between the cutter ring and the cutter body of the 19-inch disc cutter is completed by using the force prediction model of the disc cutter and the thick-wall cylinder theory. The interference fit between the peg-hole is R7/s6, the minimum amount of interference is 0.277mm, the maximum amount of interference was 0.391mm.

(2) The contact stress analysis of the interference fit between the cutter body and the cutter ring of the disc cutter is carried out. The contact stress curve on the cutter ring is similar to the bathtub curve when the cutter body and the cutter ring are different in contact length. The maximum contact stress points of the cutter ring appear on the transition fillet at both ends of the cutter ring. The interference between the cutter ring and the cutter body is the main factor affecting the contact stress, and the temperature has little effect on the contact stress. However, the contact stress decreases with increasing contact length at high temperature. When the contact length between the cutter ring and the cutter body is the same, the contact stress and the radial force of the mating face increase correspondingly with the increase of the interference volume.

(3) The thermal expansion assembly of the cutter ring and the cutter body is simulated and analyzed. The results show that the contact stress between the cutter ring and the cutter body is 0MPa when the installation temperature of the cutter ring is 160°C, which is basically consistent with the theoretical calculation of heating temperature 153°C. Therefore, it is very important to control the temperature in the hot assembly of the cutter ring and the cutter body in order to effectively ensure the metallographic structure and performance of the cutter ring. The research results in this paper are of great guiding significance and reference value for the design and assembly of interference fit of disc cutter.

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Figure

- 1 cutter shaft 2 end cover 3 floating oil seal
 4 cutter body 5 cutter ring 6 retaining rings
 7 O type sealing strip 8 tapered roller bearings

Fig.1 The structure of disc cutter

Fig.2 The sketch map of disc cutter installation

Fig.3 The schematic diagram of the force of disc cutter

Fig. 4 Stress and contact stress curves of 84mm contact length at room temperature

Fig. 5 Stress and contact stress curves of 89mm contact length at room temperature

(a) 84 mm cutter ring

(b) 89 mm cutter ring

Fig. 6 The stress of different contact length cutter ring at high temperature (Unit: MPa)

(a)

(b)

(c)

Fig. 7 Stress distribution of interference fit at high temperature (Unit: MPa)

Fig. 8 Contact stress contrast curve of cutter ring

Fig. 9 Radial force change curve of mating surface

Fig. 10 Contact stress between cutter ring and cutter body in different time history (Unit: MPa)