

# A fluid-solid coupling simulation method for convection heat transfer coefficient considering the under-vehicle condition

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**Abstract.** The convection heat transfer coefficient is one of the evaluation indexes of the brake disc performance. The method used in this paper to calculate the convection heat transfer coefficient is a fluid-solid coupling simulation method, because the calculation results through the empirical formula method have great differences. The model, including a brake disc, a car body, a bogie and flow field, was built, meshed and simulated in the software FLUENT. The calculation models were K-epsilon Standard model and Energy model. The working condition of the brake disc was considered. The coefficient of various parts can be obtained through the method in this paper. The simulation result shows that, under 160 km/h speed, the radiating ribs have the maximum convection heat transfer coefficient and the value is  $129.6W/(m^2 \cdot K)$ , the average coefficient of the whole disc is  $100.4W/(m^2 \cdot K)$ , the windward of ribs is positive-pressure area and the leeward of ribs is negative-pressure area, the maximum pressure is 2663.53Pa.

## 1. Introduction

With the increase of the train speed, more kinetic energy should be transferred when applying the brake. Most kinetic energy is transferred by the contact friction of brake disc and pad, which are important parts of foundation brake rigging. Hence the temperature-rise effect of brake disc would have an influence on train safety. Heat dissipation of the brake disc is in three forms: heat conduction, heat convection and heat radiation. 20%-30% of heat is released by the means of heat convection and heat radiation when applying the brake, of which the quantity of heat convection is accounted for 80% [1-2]. Thus the convective heat transfer coefficient is an important index in evaluating the heat dissipation performance of the brake disc. How to calculate the convective heat transfer coefficients of different heat dissipation surfaces is a key link when designing a brake disc.

Nowadays, the methodology of convective heat transfer coefficient includes the empirical formula method, testing method and numerical simulation. Voller [1] concluded equations of convective heat transfer that suggested by researchers both in China and overseas. Yanlei ZHANG [3] measured temperatures of some fixed-points and derived the convective heat transfer coefficient of target structure, according to the known condition and testing methods. Xing JIN [4] obtained the changing law on local and global Nu number of brake disc surface through the method of naphthalene sulphonamide. The changing law on inside average Nu number was also obtained by technology of naphthalene leaching. Yan HE [5] used thermal-mass analogy method in wind tunnel, which studied the convective heat transfer coefficient changing law of general rotating round-discs by wind blowing on a round disc. Yan HE also applied this testing result to tire, based on similarity principle. Zhigang





YANG [6] calculated the convective heat transfer coefficient by using CFD software FLUENT to do the numerical simulation on a single brake disc.

Testing method requires large quantity investment of testing equipment. Also, testing result is limited by the measurement location limitation. Compared with the testing method, the empirical formula method, namely, calculating by empirical formula is much easier. However, calculation results can differ greatly from different empirical formulas. When calculating the convective heat transfer coefficient based on various equations that concluded by Voller in literature [1], results show that: Under rotation speed of 800r/min, coefficient of disc surface varies from  $24W/(m^2 \cdot K)$  to  $90W/(m^2 \cdot K)$ ; the coefficient of radiating rib varies from  $20W/(m^2 \cdot K)$  to  $86W/(m^2 \cdot K)$ . Thus, it can be concluded that different empirical equations will cause large deviation of calculation results. With the rapid development of computer technology, CFD software already has capability to get precise simulation results. Therefore, a numerical simulation method is introduced in this paper: Based on under-vehicle conditions, building the model of the brake disc and simulating its fluid field by FLUENT.

## 2. Geometry modelling

### 2.1. The brake disc modelling

In this paper, a brake disc of 160km/h train-set is as an example to be analysed. The geometry model of this brake disc and the radiating rib structure are shown in **Figure 1**.



**Figure 1.** The brake disc model, the left one shows the whole model, and the right one shows the radiating rib structure.

The external diameter of the brake disc is 640mm; the internal diameter is 200mm. The thickness of whole disc is 110mm; the thickness of disc surface is 15-25mm and the thickness of wearing surface is 10mm. The material of brake disc is cast steel and its properties are shown in **Table 1**.

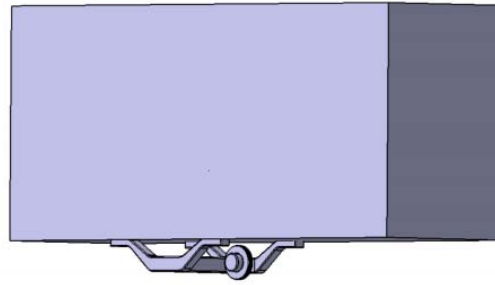
**Table 1.** The brake disc material properties.

Name	Unit	Value
Specific heat capacity, $C_d$	$J \cdot g^{-1} \cdot K^{-1}$	0.419
Density, $\rho$	$kg \cdot m^{-3}$	7228
Thermal expansion coefficient, $\alpha$	$10^{-6} \cdot K^{-1}$	10.9
Thermal conductivity, $\lambda$	$W \cdot m^{-1} \cdot k^{-1}$	48.46

### 2.2. The under-vehicle condition modelling

Taking the work condition of brake disc into consideration, a geometry model, including a car body, a bogie and a brake disc, is built. It is shown in **Figure 2**. Symmetry planes of the brake disc and the bogie are coincident. The horizontal distance between centre of the bogie and centre of the brake disc is 695mm. The vertical dimension between the car-bottom and centre of the brake disc is 713mm. The other dimensions of the model are shown in **Table 2**.





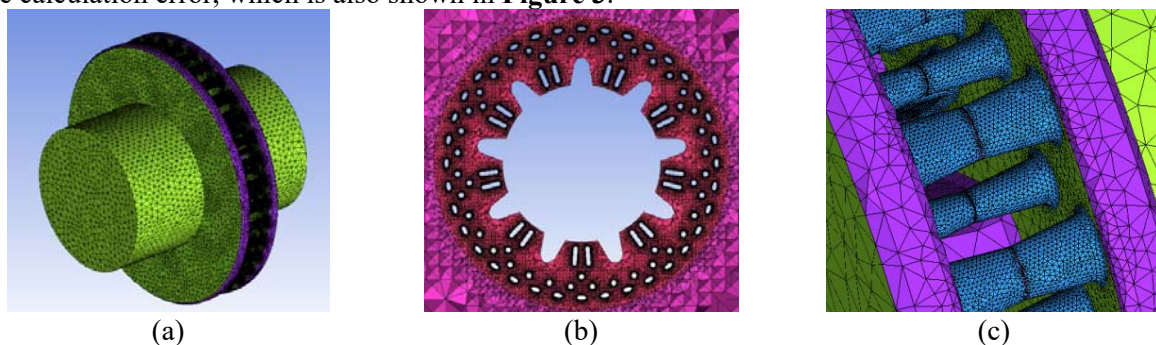
**Figure 2.** The under-vehicle condition model.

**Table 2.** The under-vehicle condition geometry dimensions.

Dimension name	Value (mm)
Car body (length $\times$ width $\times$ height)	25500 $\times$ 3105 $\times$ 4433
Bogie (length $\times$ width $\times$ height)	3260 $\times$ 2690 $\times$ 710
Bogie wheelbase	2600
Horizontal distance between centre of the bogie and centre of the brake disc	695
Vertical dimension between the car-bottom and centre of the brake disc	713
Wheel diameter	915

### 3. The CFD procedure

The geometry model is transferred to the software ICEM and meshed with Tetra-mixed elements as shown in **Figure 3**. Meshes at small dimension structure and acute angles are refined well to decrease the calculation error, which is also shown in **Figure 3**.



**Figure 3.** Mesh of brake disc.

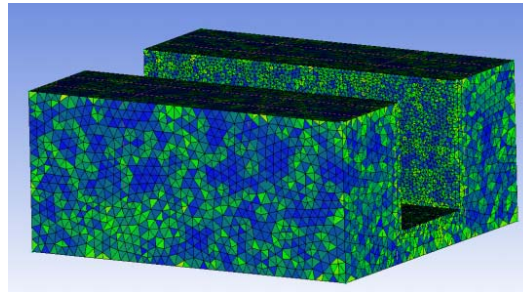
The dimension of the flow field is set up to make sure that the wind can go back to the restorative flowing stage. Using the brake disc diameter as characteristic length, the length in wake direction is set as 20 times of the characteristic length, the width and the height of the field are both set as 5 times of the characteristic length [7]. Besides, the mesh of the brake disc is set as rotation dynamic mesh while the mesh of the flow field is set as static mesh, which can simulate the working state of disc. The mesh size is 1-300mm, which is shown in **Table 3**. The mesh of the flow field is shown in **Figure 4**. The model mesh is obtained having a total of 7 million elements.

**Table 3.** The mesh sizes.

Part	Size (mm)
Axle	20
Car body	200
Disc surface	20



Radiating rib	1-3
Other parts of disc	16
Interface	32
Flow field	300



**Figure 4.** Mesh of flow field.

In the next step, the mesh model is transferred to the software Fluent and the calculation setting is made. The calculation models are Viscous- and Energy-model, because of the turbulent flow and the heat convection. The calculation equation of Viscous-model is standard k-epsilon equation, because the flow is not a complicated flow. The disc material properties are adapted as given in **Table 1**, and the gas material properties are adapted consulting standard atmosphere.

After application of these material properties in the model, the boundary condition is applied to the model. The boundary condition types of the field inlet, outlet and disc surface are velocity-inlet, pressure-outlet and no slip, respectively.

Afterwards the wind velocity of inlet, the rotation angular velocity of disc and the input thermal power on frictional surface are applied through user-define function. The initial velocity is 160km/h and the deceleration is 1m/s<sup>2</sup>, thus the wind velocity of inlet has the following form

$$v = 44.44 - 1.0 \times t \quad (1)$$

The brake disc and the axle are coaxial structure, thus the angular velocity of disc has the following form

$$\omega = \frac{v}{R} \quad (2)$$

where

$v$  is the wind velocity of inlet in equation (1);

$R$  is the wheel diameter, which has been shown in **Table 2**.

Based on the law of conservation of energy, the input thermal power on frictional surface is transferred from the kinetic energy when applying the brake. And this part is accounted for about 90% of the kinetic energy. Thus the input thermal power on frictional surface has the following form

$$P = \frac{\eta \Delta E}{\Delta t} = \frac{\eta(1/2 \times mv_0^2 - 1/2 \times mv_t^2)}{\Delta t} \quad (3)$$

where

$\eta$  is equal to 90%;

$v_0$ , which is the initial velocity, is equal to 44.44m/s;

$v_t$  is the wind velocity of inlet in equation (1).

## 4. Results and discussions

Final the model is run and post-processed successfully to visualize convective heat transfer coefficient and pressure.

### 4.1. Convective heat transfer coefficient

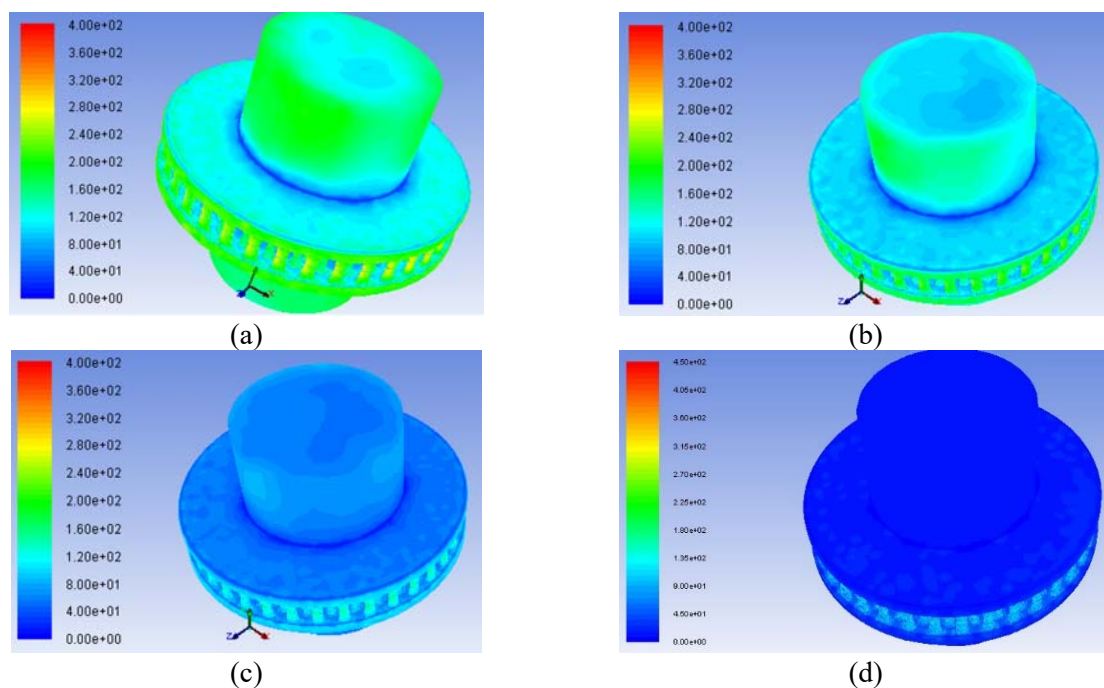


The main heat dissipation surfaces of the disc are the disc surface and radiating ribs, the coefficient results of them are shown in **Table 4**. The contours of convective heat transfer coefficient are shown in **Figure 5**. The following conclusions can be made through **Figure 5** and **Table 4**:

- the coefficient has a decreasing trend during the braking period, which is shown in **Figure 6**;
- the maximum coefficient is about  $129.6 W/(m^2 \cdot K)$ , which is found on the ribs;
- the average coefficient of the whole disc is about  $100.4 W/(m^2 \cdot K)$ .

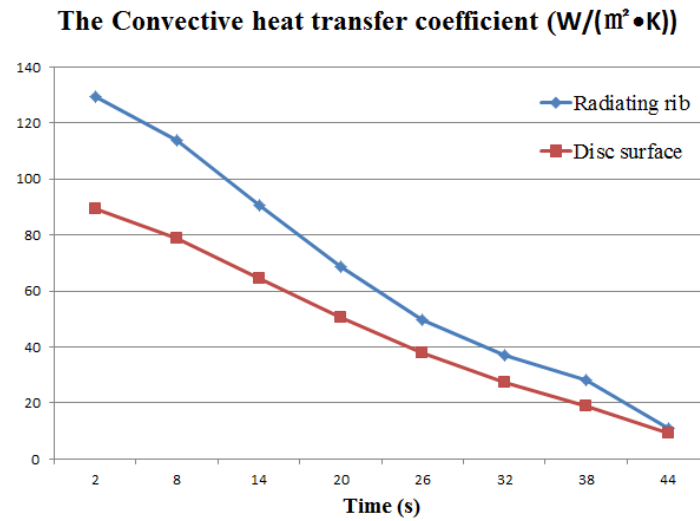
**Table 4.** The convective heat transfer coefficient results.

Time	Train speed (km/h)	Radiating rib ( $W/(m^2 \cdot K)$ )	Disc surface ( $W/(m^2 \cdot K)$ )
2	152.8	129.56	89.46
8	131.2	113.61	78.69
14	109.6	90.51	64.31
20	88.0	68.63	50.37
26	66.4	49.64	37.86
32	44.8	37.24	27.29
38	23.2	28.03	19.10
44	1.6	10.92	9.12



**Figure 5.** Contours of the convective heat transfer coefficient (time from 11s to 44s).





**Figure 6.** The chart of the convective heat transfer coefficient.

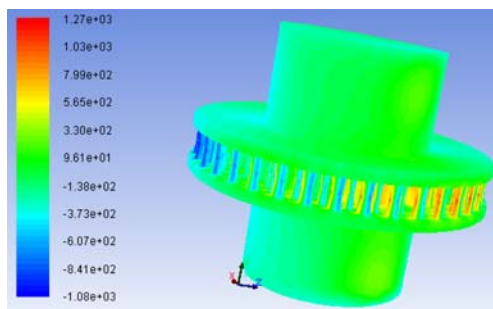
#### 4.2. Pressure

The maximum positive- and negative-pressure are shown in **Table 5** and the contours of pressure are shown in **Figure 7**. The following conclusions can be made through **Figure 7** and **Table 5**:

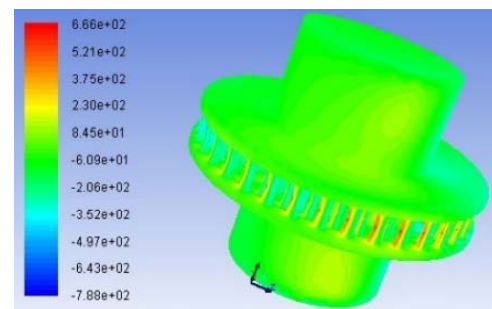
- the windward of ribs is positive-pressure area;
- the leeward of ribs is negative-pressure area;
- the positive- and negative-pressure value both decrease with the decreasing of the speed, the pressure approaches zero when the wind velocity approaches zero;
- the decreasing speed of positive-pressure is fast than that of negative-pressure.

**Table 5.** The maximum positive- and negative-pressure results

Time	Train speed (km/h)	Maximum positive-pressure (Pa)	Maximum negative-pressure (Pa)
2	152.8	2663.53	-2002.89
8	131.2	1784.682	-1376.39
14	109.6	1236.46	-1024.30
20	88.0	666.31	-788.14
26	66.4	324.71	-472.91
32	44.8	134.22	-276.14
38	23.2	25.27	-87.64
44	1.6	0.22	-7.46

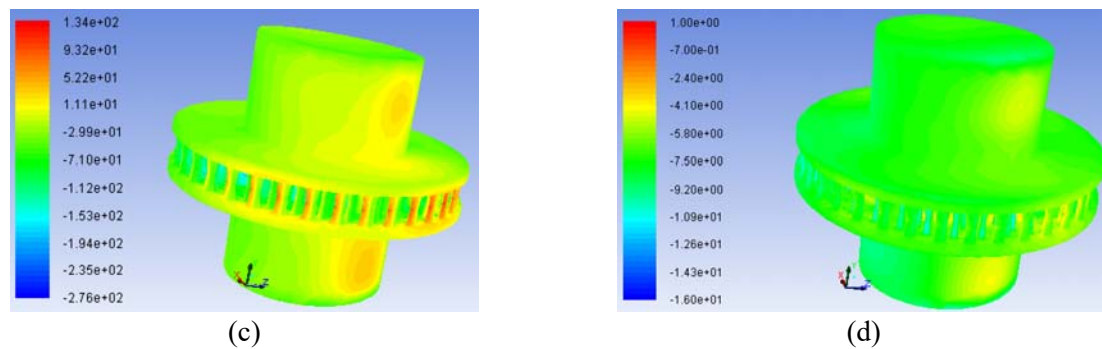


(a)



(b)





**Figure 7.** Contours of pressure (time from 11s to 44s).

## 5. Conclusions

In this paper, a numerical simulation method, which is based on under-vehicle conditions, is successfully carried out. The convective heat transfer coefficients of the brake disc surface, radiating ribs and other parts are calculated through this method. The following conclusions can be made through this paper:

- calculation results by the means of empirical formula method can differ greatly from different empirical formulas, thus the empirical equations cannot describe the heat dissipation performance of the brake disc accurately;
- through the numerical simulation method in this paper, the under-vehicle condition is taken into consideration, that is more tally with the actual situation;
- during the whole braking period, the convective heat transfer coefficient of rib is greater than that of disc surface, the maximum value is  $129.6W/(m^2 \cdot K)$ , thus the radiating rib is the main heat dissipation part of the brake disc;
- the maximum pressure is  $2663.53Pa$ , and the positive- and negative-pressure of the brake disc both decrease with the decreasing of train speed.

## Acknowledgments

The authors are grateful to the members in the Brake Technology Laboratory, Institute of Railway and Urban Rail Transit, and Tongji University for a lot of good suggestions and discussions.

Thanks the support of the National Natural Science Foundation of China (Grant No.11502169) and the Project of National Science and Technology Supporting Plan (2015BAG12B01-20).

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