Numerical Simulation Study on Air-side of Diesel Locomotive Finned Tube Double Channel Radiator

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Abstract. Based on the heat transfer and flow resistance of diesel locomotive radiator, the influence was studied about the cooling air inlet speed on heat transfer coefficient and pressure loss of finned tube double channel radiator with louver by using CFD method. Temperature field, speed field and pressure field distribution of radiator calculation results were analyzed. The radiator was tested at different cooling air inlet speed, compared test results with simulation results, the greatest relative error of the heat transfer coefficient and pressure loss were 7.21% and 7.72%, respectively.

Keywords: Finned tube double channels radiator; CFD; Heat transfer; Flow resistance.

1. Introduction
Radiator is one of the most important components of the diesel locomotive cooling system which has a significant impact on the engine's power, economy and reliability[1-2]. With the development of diesel locomotive towards high power, a series of new problems must be addressed to ensure that high-power internal combustion engine is working properly, one of the most prominent problems is the internal combustion engine cooling part[3-4]. The thinner boundary layer plays an important role in improving the heat transfer performance of the radiator[5-6].

2. Calculation Model
2.1. Physical Model
Locomotive diesel engine has two sets of cooling water system, high temperature cooling water system is used to cool cylinder liner, cylinder cap and supercharger and the working water temperature is about 65°C~85°C; low temperature cooling water system is used to cool intercooler and oil heat exchanger. Radiator core consists of a set of crossed tubes and parallel fins mounted on the tube and the crossed louveres are arranged on the radiator fins. There are two types of medium through the radiator: cooling water and cooling air and the two loops are independent of each other. Cooling water flows through the copper pipe, the heat absorbed by the cooling water in the copper pipe is transmitted to outside through heat conduction of pipe wall and fins. Fan takes away heat from outer wall of the tube by pumping cooling air flows through the louver and the channel between fins. The materials of radiator fins and cooling tube are copper.
2. Grid Model and Boundary Conditions
Setting speed value in calculation area entrance and the direction is perpendicular to the entrance boundary, the cooling air inlet temperature is set to 40°C, the calculation area outlet is set to pressure outlet and the ambient pressure is set to a standard atmospheric pressure. The wall temperature of high and low temperature cooling tube is set to 85°C and 68°C, respectively. Calculation shows that the Re is 6978 in the flow region and the turbulence model used k-ε model, the medium temperature is not high so do not consider the impact of radiation and gravity. After calculation of the model, the residual curve and the monitoring point curve are convergent and the import and export both meet conservation of mass.

3. Result Analysis
3.1. The Distribution of Temperature Field, Pressure Field and Velocity Field
Keeping louver angle of 16°, high temperature part fin spacing 2.33mm, low temperature part fin spacing 2.00mm unchanged, doing calculation when inlet wind speed is 11m/s, 12m/s, 13m/s, 14m/s, 15m/s, respectively; keeping louver angle of 16°, inlet wind speed 11m/s unchanged, doing calculation when fin spacing set following five cases respectively:
1. Fin spacing of low temperature part is 1.56mm, fin spacing of high temperature part is 1.75mm (low 1.56 high 1.75);
2. Fin spacing of low temperature part is 1.75mm, fin spacing of high temperature part is 2.00mm (low 1.75 high 2.00);
3. Fin spacing of low temperature part is 2.00mm, fin spacing of high temperature part is 2.33mm (low 2.00 high 2.33);
4. Fin spacing of low temperature part is 2.33mm, fin spacing of high temperature part is 2.80mm (low 2.33 high 2.80);
5. Fin spacing of low temperature part is 2.80mm, fin spacing of high temperature part is 3.50mm (low 2.80 high 3.50).
Taking following conditions as an example to analyze temperature field, pressure field and velocity field: inlet wind speed 11m/s, louver angle of 16°, low temperature fin spacing 2.00mm, high temperature fin spacing 2.33mm.
Temperature field distribution shown in Figure 3, on the whole, temperature gradually increased with cooling air through the cooling tube to the radiator inside, and the cooling effect of front cooling tube (near the cooling air inlet) is better than rear cooling tube; the cooling effect on both sides of the tube is better than back of the tube. The cooling air speed dropped to zero when the air passes around the tube to the top of the arc and the formation of the rear stagnation point is not conducive to heat dissipation.

Pressure field distribution shown in Figure 4, on the whole, the pressure appeared downward trend along the air flow direction, the maximum pressure appeared in the head of tube, the tube head would form the front stagnation point when the air flows through tube, at this position, the dynamic pressure head is converted to static pressure head and the pressure is greatest. The crossed cooling tube leads to a circular distribution of pressure which is conducive to enhancing the disturbance of air and improved cooling capacity; the two sides of the tube formed a low pressure area. When the air flows along both sides of the cooling tube, the flow rate gradually increased and pressure gradually reduced as the flow cross-sectional area decreases, the static pressure head is converted to dynamic pressure head and the pressure is minimized at the maximum position of the pipe diameter.

Speed field distribution shown in Figure 5, when the cooling air flows through the cooling tube, the convective heat transfer between air and water tube increased because of flow cross-sectional area decreased and flow rate increased; because the kinetic energy loss of cooling air at the rear of the tube is basically exhausted at the near wall, the stagnation occurs under the influence of front and rear pressure differences, so the heat transfer time increased. The cooling air can flow from the lower flow area through the louvers created on the radiator, and the cooling capacity increased by enhancing air disturbances. Due to the hindrance of the front cooling tube, the air flow through the middle of the louver is less than the air flow through both sides, as shown in Figure 6, the cooling air passing through the louvers increased as the inlet flow rate increases. Taking one of the louvers as an example, the volume flow through the louvers at 11m/s is $1.65 \times 10^{-4}$ m$^3$/s and the volume flow through the louvers at 15m/s is $2.33 \times 10^{-4}$ m$^3$/s, the volume flow increased by 44.2% while overall flow resistance increased by 73.4%.

Figure 3. Temperature distribution of the radiator in air-side.

Figure 4. Pressure distribution of the radiator in air-side.
3.2. Influence of Cooling Air Inlet Speed on Resistance and Heat Transfer Coefficient

As shown in Figure 7 and Figure 8, the heat transfer coefficient and the pressure loss increased with increase of the cooling air inlet speed. The heat transfer coefficient grows faster when cooling air inlet speed is low and the growth rate is relatively slow when cooling air inlet speed is high, this is caused by its own core structure and heat dissipation conditions. In order to predict the heat transfer coefficient and pressure loss at any inlet speed, fitted the calculation results into function expressions, the fitting function of the heat transfer coefficient and the cooling air inlet speed is:

\[ K_v = 23.84 \times V^{0.5543} \]  

(1)

The fitting function of the pressure loss and the cooling air inlet speed is:

\[ \Delta P_v = 12 \times V^{1.651} \]  

(2)

For the same type of radiator, it can uses formula (1) and formula (2) to predict heat transfer coefficient and pressure loss under the premise without changing the structure.

3.3. The Influence of Fin Spacing on Heat Dissipation of Radiator

The effect of the fin spacing on the heat dissipation performance of the radiator is shown in Figure 9,
on the whole, the total pressure loss showed a downward trend as the fin spacing of high and low temperature cooling part increases; the total heat transfer coefficient increased as fin spacing increases and the total heat release decreased as fin spacing increases, although the heat transfer coefficient increased with the fin spacing, the heat dissipation area decreased with the increase of the fin spacing, and the total heat release is reduced. In the case of the finned tube double channel radiator with louver, the ratio of heat release of the high temperature part to total heat release in the five schemes above was 42.8%, 44.7%, 44.9%, 46.7% and 45.7% respectively, the overall proportion rose slightly and the amplitude of variation is not significant; the ratio of heat dissipation area of the high temperature part to total heat dissipation area was 42.2%, 41.7%, 41.1%, 40.1% and 38.5% respectively, it showed a clear downward trend. That is, the increase of fin spacing would reduce the heat release of high and low temperature parts, heat release of high temperature cooling part accounted for a slight increase in the proportion of total heat release and the variation range is not large, as shown in Figure 10.

**Figure 9.** Influence of radiator fin spacing on heat dissipation.

**Figure 10.** Variation law of the ratio of heat dissipation area to heat dissipation.

### 3.4. Comparison of Simulation Results and Test Results

Keeping the louver angle of 16°, low temperature fin spacing 2.00mm, high temperature fin spacing 2.33mm and other conditions unchanged, carried out radiator bench test at different inlet speed. Compared radiator test results and simulation results at different inlet wind speed and it can be seen from table 2 that the change law of test results and the simulation results are basically the same, the maximum relative error of the heat transfer coefficient is 7.21% and the maximum relative error of the pressure loss is 7.72%. Therefore, it can be seen that the computational model established in this paper is correct and the simulation calculation is feasible. The reason for the error is mainly due to the simplification of the simulation model and the limitation of the test conditions.

**Table 2.** Comparison of heat transfer coefficient.

| Entrance wind speed /m·s⁻¹ | Air temperature in front of the radiator t₁a/℃ | Air temperature behind the radiator t₂a/℃ | Water temperature in front of the radiator t₁w/℃ | Water temperature behind the radiator t₂w/℃ | Heat transfer coefficient of simulation calculation /w·(m²·K)⁻¹ | Heat transfer coefficient of test results /W·(m²·K)⁻¹ | Relative error /% |
|----------------------------|-----------------------------------------------|-------------------------------------------|-----------------------------------------------|-----------------------------------------------|-------------------------------------------------|-------------------------------------------------|---------------|
| 11                         | 38.06                                         | 74.92                                     | 87.97                                         | 82.50                                         | 90.00                                            | 97.00                                            | 7.21          |
| 12                         | 38.04                                         | 73.47                                     | 88.00                                         | 82.22                                         | 95.30                                            | 99.00                                            | 3.74          |
| 13                         | 38.18                                         | 72.55                                     | 88.15                                         | 82.03                                         | 98.70                                            | 102.00                                           | 3.24          |
| 14                         | 38.06                                         | 71.65                                     | 88.12                                         | 81.70                                         | 102.80                                           | 105.00                                           | 2.09          |
| 15                         | 38.47                                         | 71.53                                     | 87.10                                         | 81.38                                         | 107.10                                           | 111.00                                           | 3.51          |

### 4. Conclusion

1. The maximum relative error of heat transfer coefficient is 7.21% and the maximum relative error of pressure loss is 7.72% compared with the test results, it is shown that the simulation model can be applied to the radiator performance research.
2. The heat transfer coefficient and the pressure loss increased with the increase of the cooling air inlet speed, the heat transfer coefficient grows faster when the cooling air inlet speed is low, the growth rate is relatively slow when the cooling air inlet speed is high.
3. High and low temperature cooling part heat release, total heat release and pressure loss are increased with the decrease of the fin spacing. When the fin spacing decreases, the amount of heat dissipation increases steadily and the pressure loss increases sharply.

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