Research on Self-adaption Model of Cooling Intensity for Construction Vehicle Based on CFD and $\varepsilon$-NTU

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Abstract To improve the stability of power take-off and energy conservation of cooling system, it is essential to investigate a control model of cooling system for the working engine under varied ambient temperature. A Double-drum roller from a manufacturer in China was employed as the research basis for a self-adaption model of cooling intensity. At first, a numerical investigation was performed on its engine cabin, and the results were experimentally validated. Secondly, a self-adaption model based on $\varepsilon$-NTU method was solved with the combination of Ferrari and Shengjin methods, and finally presented in terms of heat transfer rate, fan rotation speed and ambient temperature. At last, under a constant heat flux of the radiator, the numerical results and predicted outcomes were compared at -30°C and 20°C to confirm the accuracy of that model. The results stated that the maximum error between experimental data and simulation results is 7.89%, which validates the correctness of the implemented numerical analysis. The Ferrari and Shengjin methods are eligible to solve the derived model, which is shown with heat transfer rate, fan rotation speed and ambient temperature. The deviation between numerical results and estimated values are 0.8°C at 20°C and 0.75°C at -30°C, respectively. The conclusions of this work could provide a theory basis and reference for the relative research on the self-adaption model of cooling intensity for construction vehicles.

1. Introduction

Double-drum roller belongs to the road machinery category in the classification of construction machinery, which is widely used in some large projects, such as the fill and compaction operations of highways, railways, airport runways, dams and stadiums. The temperature of cylinder jacket is too high or too low is not conducive to the stability of engine power output, therefore, the preventions of both overheating and super cooling phenomenon are necessary, which requires the designers to predict the heat transfer rate, thus to control the outlet temperature of the thermal fluid better.

With the development of CFD method, many researches which conducted by experiment formerly can be achieved on computers nowadays. Gullberg and Lofdahl [1-3] used CFD method to investigate the fan static, MRF, sliding mesh, MRF error and continuity of the cooling fan respectively. Itsuhei Kohri [4] compared the CFD simulation results and experimental data of different types of cooling fan to verify the accuracy of the simulation. Anusonti-Inthra [5] studied the aerodynamic performance of the high load cooling fan with CFD method. Zhan [6] numerically investigated the flow field characteristics of the cooling fan under different working conditions by CFD method. Zhou [7] re-designed the CPU cooling fan by the methods of theoretical design and CFD verification.
Lockwood [8] combined the intelligent control and CFD analysis to achieve the real-time fan speed control. Wagner [9] developed and verified an electric control model of water pump and thermostat. Stephens [10] studied the performance of the fan on the basis of the interaction between the radiator and the fan. Thorat [11] characterized the radiator by the porous medium, then the flow field of the vehicle and the CFD analysis results were validated. Lu [12] optimized the compression ratio and fan speed of the compressor.

On the basis of relevant research, this paper took a double-drum vibratory roller as the research object, established a function model including heat dissipation value, ambient temperature and fan speed based on ε-NTU method. By limiting the heat dissipation value, the fan speed and shaft power under different working conditions were estimated and the accuracy of the model was verified.

2. CFD Simulation and Experimental Validation

2.1. Vehicle for investigation
This paper takes the double-drum vibratory roller as research object, it weighs 13000kg, diameter of vibrating wheel is 1300mm and the rated power of engine can reach 98kw. The roller is shown in Figure 1.

![Figure 1. Vehicle for investigation](image)

2.2. Geometric model
According to the design drawings provided by manufacturer, the 3D model of the engine cabin was established and simplified. The model was shown in Figure 2.

![Figure 2. Geometric model of the engine cabin](image)

2.3. Computational grid and boundary condition
The tetrahedral and hexahedral grid were used to mesh the model, the rotating area of cooling fan, the inlet and outlet of the engine cabin were space locally refined. The coolant radiator, CAC, hydraulic oil radiator, engine, air filter and muffler were surface locally refined. The grid was increased by the scale factor at the far end of the wind tunnel in order to reduce the amount of computation due to the smaller effect of air flow. The total grid number of the model is about 12.77 million and the model grid is shown in Figure 3.

The boundary conditions are shown as follows, the fan speed is 2300r/min, the ambient temperature is 45 ℃, and the temperature of pressurized air, cooling fluid, hydraulic oil entrance are
186°C, 100°C and 80°C, respectively. In the phase of defining physical properties of air, the heat capacity was defined as 1.0082 kJ/(kg·℃), the thermal conductivity was defined as 0.0282 W/(m·℃), the dynamic viscosity was defined as 2×10⁻⁵ Pa·s. The iteration is considered to be converged when the scaled residual keeps fluctuating slightly within several different iterations that was set according to the requirements of simulation. The physical properties of the thermal fluid is shown in reference [13], the detail boundary conditions and simulation results are shown in reference [14-19].

![Figure 3. Grid of the virtual tunnel](image)

2.4. Experimental test
The experiment was conducted in the test field of an enterprise engineering center in China. Experiments were selected in August when the weather was hot, and the vehicle kept working for 2 hours under high loads. The PT100 and PTX7517 sensors were installed on the inlet and outlet of the radiator to measure the data, the arrangement of corresponding sensor was shown in Figure 4. The entire system was deemed as equilibrium when the data measured by the sensor kept fluctuating slightly.

![Figure 4. Sensor arrangements for the test points in the experiment](image)

2.5. Experimental results and validation
The temperature of pressurized air and coolant were measured directly by the sensor at the exit, infrared thermometer gun was used at the import and export of metal fittings to collect the temperature of hydraulic oil, comparison between simulation results and experimental data was shown in
The information from the table showed that the minimum error was 3.57% and the maximum error was 7.89%, within an acceptable tolerance.

### Table 1. Comparison between simulation results and experimental data

| Radiator       | Simulation (°C) | Experiment (°C) | Deviation (%) |
|----------------|-----------------|-----------------|---------------|
| CAC            | 55.79           | 60.57           | 7.89          |
| Coolant        | 93.21           | 96.89           | 3.80          |
| Hydraulic oil  | 66.64           | 69.11           | 3.57          |

3. Model Development

3.1. The core formula

Based on the \( \varepsilon \)-NTU method, heat flux calculation formula is defined as:

\[
q = \varepsilon \cdot C_{\text{min}} \cdot (T_{\text{in-h}} - T_{\text{in-c}})
\]

Where \( q \) is the local heat flux, \( \varepsilon \) is the ratio of actual heat transfer effect to the maximum heat transfer effect, \( C_{\text{min}} \) is the minimum heat capacity, \( T_{\text{in-h}} \) and \( T_{\text{in-c}} \) are the inflow temperature of hot fluid and cold fluid respectively.

3.2. Decomposition of formula

1) Heat flux \( q \)

The heat flux is employed to control the outlet temperature of the thermal fluid inside the radiator, the heat flux, deemed as a qualitative condition, hold the outlet temperature in the proper range.

2) Radiator efficiency \( \varepsilon \)

Radiator efficiency is the ratio of the actual heat transfer effect to the maximum heat transfer effect. Consideration should be given to the fact that the efficiency is affected by the air condition when the fluid state on the thermal fluid side has been determined. The fin element unit was numerically investigated under different boundary conditions and the \( \varepsilon \)-NTU method was applied to fit the efficiency function.

3) The minimum heat capacity \( C_{\text{min}} \)

In this paper, the minimum heat capacity is the cold air, and further split into:

\[
C_{\text{min}} = C_{\text{pc}} \cdot m_c = C_{\text{pc}} \cdot \rho_c \cdot Q
\]

where \( C_{\text{pc}} \) is specific heat capacity at constant pressure, \( m_c \) is mass flow rate of cold air, \( \rho_c \) is density of cold air, \( Q \) is air volume flow rate through the radiator.

According to reference[20], the relationship between the speed of fan inside engine cabin and flow rate was expressed as a function, and flow fields of the engine cabin under different speeds were investigated by CFD method, thus to obtain flow rate of the engine cabin. The ratio of flow rate to the flow rate at free eddy current was calculated, and the average was here to simplify the calculation.

To determine the amount of air into the radiator, flow rate is defined by introducing the intake ratio coefficient of engine cabin [18], which is defined as:

\[
\begin{align*}
Q &= f(Q_f) = f(k_d \cdot Q_f) \\
Q_c &= k_c \cdot n
\end{align*}
\]

Where \( Q_f \) is the air volume flow rate at fan free vortex, \( k_d \) is the inlet air ratio coefficient of engine cabin, \( Q_c \) is the actual volume flow rate of air into the engine cabin, \( n \) is fan speed, \( k_c \) is speed ratio coefficient.
In this paper, the influence of temperature was neglected, and the biquadratic polynomial function was used to fit the flow function:

\[ Q = R k_d Q_c^4 + S k_d Q_c^3 + U k_d Q_c^2 + V k_d Q_c \] (4)

\[ C_{pc} = 1003 + 0.02 T_{average-c} + 4 \times 10^{-3} T_{average-c}^2 \] (5)

\[ \rho_c = \frac{1.293 \times 273.15}{T_{average-c} + 273.15} \] (6)

In summary, the calculation formula of heat flux is defined as follows:

\[ q = \frac{353.18295 \varepsilon \cdot Q(1003 + 0.02 T_{in-c} + 0.0004 T_{in-c}^2)}{T_{in-c} + 273.15} \times (T_{in-h} - T_{in-c}) \] (7)

4. Derivation of Model

The two-variable cubic function is used to fit the radiator efficiency function, which is shown as Formula 8, the fitting surface is shown in Figure 5.

\[ a_1 T_{in-c}^3 Q + a_2 T_{in-c}^2 Q + a_3 T_{in-c} Q + a_4 T_{in-c}^2 + a_5 T_{in-c} + a_6 T_{in-c}^3 + a_7 T_{in-c}^2 + a_8 T_{in-c} + a_9 T_{in-c}^3 + a_{10} T_{in-c}^2 \] (8)

The following equation is obtained by substituting equation (8) into equation (7):

\[ a_1 A_{mc}^3 Q + a_2 A_{mc}^2 T_{in-c} Q + a_3 A_{mc} T_{in-c}^2 Q + a_4 A_{mc} T_{in-c} Q + a_5 A_{mc}^2 + a_6 A_{mc} + a_7 A_{mc}^3 T_{in-c}^3 \]

\[ + a_8 T_{in-c}^2 A_{mc} Q^3 + a_9 T_{in-c} A_{mc} Q^4 + a_{10} Q^4 = \frac{A_{mc} q(T_{in-c} + 273.15)}{353.18295(1003 + 0.02 T_{in-c} + 0.0004 T_{in-c}^2)(T_{in-h} - T_{in-c})} \] (9)

Take the formula \( q = C_{mc} m_k (T_{in-h} - T_{out-h}) \) into consideration, \( Q \) is regarded as independent variables and the equation is rearranged, the final expression is:

\[ a_{10} Q^4 + (a_7 T_{in-c} A_{mc} + a_6 A_{mc}) Q^3 + (a_5 A_{mc} T_{in-c} + a_3 A_{mc}^2 + a_8 A_{mc}^2 T_{in-c}) Q^2 \]

\[ + (a_2 A_{mc} T_{in-c} + a_7 A_{mc}^3 T_{in-c}) (C_k T_{in-c} + a_4 A_{mc} T_{in-c}^2 + a_{10} A_{mc}) Q \]

\[ - \frac{A_{mc} C_{mc} m_k (T_{in-h} - T_{out-h})(T_{mc} + 273.15)}{353.18295(1003 + 0.02 T_{in-c} + 0.0004 T_{in-c}^2)(T_{in-h} - T_{in-c})} = 0 \] (10)

5. Solution of Model

The formula (10) is rearranged as a simplified form:
\[
\begin{aligned}
&\left\{\begin{array}{l}
Q^3 + bQ^3 + cQ^2 + dQ + e = 0 \\
 b = \frac{a_5T_{m-c}^2A_{nc} + a_4A_{nc}}{a_{10}} \\
 c = \frac{a_5A_{nc}^2T_{m-c} + a_3A_{nc}^2 + a_2A_{nc}^2T_{m-c}}{a_{10}} \\
 d = \frac{a_2A_{nc}^3T_{m-c} + a_3A_{nc}^3T_{m-c}^3 + a_4A_{nc}^3T_{m-c}^2 + a_5A_{nc}^3}{a_{10}} \\
 e = -\frac{A_{nc}^4C_{ph}/m}{353.18295a_{10}} \times \frac{T_{m-c} + 273.15}{T_{m-h} - T_{m-c}} \\
 &\times \frac{T_{m-h} - T_{out-h}}{1003 + 0.02T_{m-c} + 0.0004T_{m-c}^2}
\end{array}\right.
\]

After observing the equation characteristics, equation (11) was transformed into a set of equations by the Ferrari's method:

\[
2Q^2 + (b + M)Q + 2\left(y + \frac{N}{M}\right) = 0
\]  
(12)
\[
2Q^2 + (b - M)Q + 2\left(y - \frac{N}{M}\right) = 0
\]  
(13)
\[
8y^3 - 4cy^2 - (8e - 2bd)y - e(b^2 - 4c) - d^2 = 0
\]  
(14)

Where \(M\) in the equation (12):

\[
M = \sqrt{8y + b^2 - 4c}
\]  
(15)
\[
N = by - d
\]  
(16)

The value of \(y\) was calculated by using equation (14), then plugged it into equation (12) and (13) to solve the value of \(Q\). As to equation (14), it could be solved by Shengjin method, the expression was as follows:

\[
a(1)y^3 + b(1)y^2 + c(1)y + d(1) = 0
\]  
(17)

Where: \(a(1), b(1), c(1), d(1) \in R, a(1) \neq 0\).

Heavy root discriminant:

\[
A = b(1)^2 - 3a(1)c(1)
\]  
(18)
\[
B = b(1)c(1) - 9a(1)d(1)
\]  
(19)
\[
C = c(1)^2 - 3b(1)d(1)
\]  
(20)

Total discriminant:

\[
\Delta = B^2 - 4AC
\]  
(21)
According to equation (14), coefficients of equation (17) were:

\[ a(1) = 8 \quad b(1) = -4c \quad c(1) = -(8e-2bd) \quad d(1) = -e(b^2-4c) \]

When \( A=B=0 \):

\[ 2e^2 + 24e - 6bd = -32ec - bdc + 9eb^2 + 9d^2 = 0 \]  
\[ (22) \]

The value of \( y \):

\[ y_1 = y_2 = y_3 = \frac{c}{6} = \frac{bd - 4e}{2c} = \frac{-3eb^2 + 12ec - 3d^2}{8e - 2bd} \]  
\[ (23) \]

When \( \Delta = B^2 - 4AC > 0 \), the expression of real solutions was as follows:

\[ y_i = \frac{c}{6} - \frac{1}{24} \sqrt{64c^3 + 2304ec + 288bdc - 864b^2e - 864d^2} \]

\[ -12 \sqrt{5184b^2e^2 + 1152b^2d^2e + 5184d^4 - 27648b^2ce^2 - 3456b^3cde - 27648cd^3e - 3456bcd^3} \]

\[ + 24567c^3e^2 + 15360b^3ce - 192b^2c^3d^2 + 768b^2c^3e - 3072c^4e + 768c^3d^2 \]

\[ - 49152e^3 + 768b^3d^3 + 36864bde^2 + \frac{1}{24} \sqrt{-64c^3 + 2304ec + 288bdc - 864b^2e - 864d^2} \]

\[ + 12 \sqrt{5184b^2e^2 + 1152b^2d^2e + 5184d^4 - 27648b^2ce^2 - 3456b^3cde - 27648cd^3e - 3456bcd^3} \]

\[ + 24567c^3e^2 + 15360b^3ce - 192b^2c^3d^2 + 768b^2c^3e - 3072c^4e + 768c^3d^2 \]

\[ - 49152e^3 + 768b^3d^3 + 36864bde^2 \]  
\[ (24) \]

When \( \Delta = B^2 - 4AC = 0 \), the value of \( y \):

\[ y_1 = \frac{c}{2} + \frac{9eb^2 - 32ec - bdc + 9d^2}{2e^2 + 24e - 6bd} = \frac{c^3 - 20ec - 4bdc + 9eb^2 + 9d^2}{2c^2 + 24e - 6bd} \]  
\[ (25) \]

\[ y_2 = y_3 = \frac{-9eb^2 + 32ec + bdc - 9d^2}{4e^2 + 48e - 12bd} \]  
\[ (26) \]

When \( \Delta = B^2 - 4AC < 0 \), the value of \( y \):

\[ z = \frac{2Ab(1) - 3a(1)B}{2 \sqrt{A^3}} \]

\[ \theta = \arccos z = \arccos\left(\frac{2Ab(1) - 3a(1)B}{2 \sqrt{A^3}}\right), \quad (A > 0, -1 < z < 1) \]

\[ y_1 = \frac{-b(1) - 2 \sqrt{A \cos \frac{\theta}{3}}}{3a(1)} \]
\[
y_{2,3} = \frac{-b(1) + \sqrt{A(\cos \frac{\theta}{3} \pm \sqrt{3} \sin \frac{\theta}{3})}}{3a(1)}
\]
\[
z = \frac{-64c^3 + 2304ec + 288bdc - 864eb^2 - 864d^2}{\sqrt{(16c^2 + 192e - 48bd)^3}}
\]
\[
\theta = \arccos\left(\frac{-64c^3 + 2304ec + 288bdc - 864eb^2 - 864d^2}{\sqrt{(16c^2 + 192e - 48bd)^3}}\right)
\]
\[
y_{1} = \frac{c}{6} \cdot \frac{\sqrt{16c^2 + 192e - 48bd}}{12} \cdot \cos\left\{\frac{1}{3} \arccos\left[\left(\frac{16c^2 + 192e - 48bd}{2}\right)^2\right]\right\}
\]
\[
y_{2,3} = \frac{c}{6} + \frac{\sqrt{16c^2 + 192e - 48bd}}{24} \cdot \cos\left\{\frac{1}{3} \arccos\left[\left(\frac{16c^2 + 192e - 48bd}{2}\right)^2\right]\right\} \pm \sqrt{3} \sin\left\{\frac{1}{3}\right\}
\]

After obtaining the value of \(y\), solve equation (12), (13).

When \(\Delta_1 \geq 0\), the value of \(Q\):
\[
Q_{1,2} = \frac{-\left(b + \sqrt{8y + b^2 - 4c}\right) \pm \sqrt{2b^2 + 2b\sqrt{8y + b^2 - 4c} - 8y - 4c - \frac{16(by - d)}{\sqrt{8y + b^2 - 4c}}}}{4}
\]

When \(\Delta_2 \geq 0\), the value of \(Q\):
\[
Q_{3,4} = \frac{-\left(b - \sqrt{8y + b^2 - 4c}\right) \pm \sqrt{2b^2 - 2b\sqrt{8y + b^2 - 4c} - 8y - 4c + \frac{16(by - d)}{\sqrt{8y + b^2 - 4c}}}}{4}
\]

Next, shaft power was calculated and the relevant formula was as follows:

Fan total pressure \(P_T\):
\[
p_T = p_s + p_d = p_s + \frac{1}{2} \rho_c v^2
\]

Where \(p_s\) is fan static pressure, \(p_d\) is fan dynamic pressure, \(v\) is inlet section velocity of fan.

Useful power at static pressure \(N_{es}\):
\[
N_{es} = p_s \cdot Q_f
\]

Shaft power \(N_s\):
\[
N_s = \frac{N_{es}}{\eta_s}
\]
Where \( \eta_s \) is the static pressure efficiency. When the ambient temperature is \( T_1 \), speed is \( n_1 \), the fan static pressure, volume flow rate, static pressure efficiency, shaft power are \( p_s1, Q_f1, \eta_s1, p_{zhou1} \), respectively. The fan static pressure \( p_s2 \), volume flow rate \( Q_f2 \), static pressure efficiency \( \eta_s2 \) and shaft power \( p_{zhou2} \) of the same fan can be calculated when ambient temperature and speed are \( T_2, n_2 \), respectively. The specific conversion relationship is shown as follows [21]:

\[
\frac{P_{zhou1}}{P_{zhou2}} = \frac{T_2 + 273.15}{T_1 + 273.15} \left( \frac{n_1}{n_2} \right)^3
\]  

(31)

The speed and temperature were transformed and then put into the formula to solve shaft power. Since the flow rate of fan and total pressure were nonlinear characteristics at the same speed, the polynomial was used to suit the static pressure curve, the expression [22] was as follows:

\[ p_{s1} = XQ_{f1}^2 + YQ_{f1} + Z \]

The polynomial is used to fit the static pressure efficiency, the expression is as follows:

\[ \eta_{s1} = c_0Q_{f1}^5 + c_1Q_{f1}^4 + c_2Q_{f1}^3 + c_3Q_{f1}^2 + c_4Q_{f1} + c_5 \]  

(32)

When ambient temperature is \( T_3 \), speed is \( n_3 \), thehaft power of cooling fan is expressed as follows:

\[
P_{zhou2} = P_{zhou1} \cdot \frac{T_1 + 273.15}{T_2 + 273.15} \left( \frac{n_1}{n_2} \right)^3 \times \frac{T_1 + 273.15}{T_2 + 273.15} \left( \frac{n_1}{n_2} \right)^3
\]  

(33)

6. Calculation of Example and Verification of Results

6.1. Calculation of example

The MATLAB was used to establish M file to solve equation (10). Theoretically, there were eight solutions, but only one was consistent with the actual situation, so airflow and fan speed should be selected step by step. The ambient temperature vehicle works could be used as the variation range of equation coefficient, before reaching the maximum ambient temperature, the speed of each point was solved one by one point. The stored data was output after reaching the maximum value and the specific solution process was shown in Figure 6. The car had four working conditions, were calculated respectively, the specific working conditions were shown in Table 2.
Figure 6. Flow chart of solution

Table 2. Working conditions of vehicle

| Condition | Gear | Direction | Speed (km/h) |
|-----------|------|-----------|--------------|
| condition 1 | 1    | forward   | 6            |
| condition 2 | 2    | forward   | 12           |
| condition 3 | 1    | backward  | 6            |
| condition 4 | 2    | backward  | 12           |

According to the engine performance curve, 93.94 °C was selected as design temperature of cooling outlet and the calculation results were shown in the form of curves, as shown in Figure 7. The curve in the graph was analyzed. When ambient temperature rose, the variation of speed under various working conditions gradually increased. The main reason for this phenomenon was that air density decreased as ambient temperature increased, the actual air quality involved in heat exchange process was reduced, while the smaller temperature difference reduced the radiator efficiency. The air volume flow rate must be increased continuously to ensure the outlet temperature. Further analysis, the range of fan speed under condition 2 was 769~2378 r/min, while condition 4 was 702~2170 r/min. At the same ambient temperature, the relationship of the rotational speed under different working conditions was as follows: condition2 > condition1 > condition3 > condition4, which was in accordance with the analytical results from literature [16].
The formula (35) was used to solve the fan shaft power, as shown in Figure 8. From the figure, the variation of shaft power increased gradually as temperature increased. The difference was small from -30°C to 5°C under four working conditions. When ambient temperature was more than 5°C, the difference of shaft power under different operation conditions came to be revealed. When ambient temperature reached 45°C, the difference of fan power was obvious, the relationship was: condition2 > condition1 > condition3 > condition4. Because the experimental data was not continuous, in this paper, the shaft power at 45°C in condition 1 was selected to be compared with the calculated value of 8.29 kW. The experimental value given by the manufacturer was 8.48 kW and the error was about 2.36%.

6.2. Verification of results
The CFD method was used to numerically investigate the engine cabin of a domestic double-drum vibratory roller mentioned in section 1 of the paper and simulation results were put to validate calculation results of the model. The numerical boundaries of simulation were shown in Table 3.
Table 3. Numerical boundaries

| Parameter                           | Value                  |
|-------------------------------------|------------------------|
| Speed ratio coefficient            | 0.00152 m³/(r/min)     |
| Ambient temperature                | -30—45°C               |
| Thermal fluid inlet temperature    | 100°C                  |
| Thermal fluid outlet design temperature | 93.94°C               |

The speed of cooling fan at -30°C and 20°C were chosen as the verification point, the CFD simulation results and one-dimensional theoretical results were shown in Tables 4 and 5. According to the results, the difference between them ranged from 0.13°C to 0.8°C.

At -30°C, ambient temperature was low, heat convection between thermal air and environment rose as vehicle moved forward, in this way, leading to an effect cut of thermal air recirculation on radiator. In addition, heat exchange performance was improved due to the improvement on the radiator intake state. At 20°C, the effect of thermal air recirculation tended to be found as vehicle moved forward. Except that, air temperature increased, the change of heat exchange performance decreased, and the difference of outlet temperature between design and simulation decreased.

Table 4. Comparison between simulation results and theoretical calculation results at -30°C

| Thermal fluid outlet temperature (°C) |
|---------------------------------------|
| Condition 1        | Condition2 | Condition3 | Condition4 |
| Design              |            |            |            |
| Simulation          | 94.59      | 94.69      | 93.70      | 93.29      |
| Difference          | -0.65      | -0.75      | 0.24       | 0.65       |

Table 5. Comparison between simulation results and theoretical calculation results at 20°C

| Thermal fluid outlet temperature (°C) |
|---------------------------------------|
| Condition 1        | Condition2 | Condition3 | Condition4 |
| Design              |            |            |            |
| Simulation          | 94.66      | 94.74      | 93.81      | 93.61      |
| Difference          | -0.72      | -0.80      | 0.13       | 0.33       |

7. Conclusions

In this paper, the CFD method was used to analyze the cooling performance of a domestic double-drum vibratory roller, and the numerical model were experimentally validated. Secondly, a self-adaption model was established based on ε-NTU method, which was derived with the application of Ferrari's method and Shengjin method and expressed in terms of the heat transfer rate, fan rotation speed and ambient temperature. At last, to maintain the heat transfer rate for the radiator of the road roller, the comparison between the numerical simulation results and designed value was made at -30°C and 20°C to verify the accuracy of the self-adaption model. Based on the work done above, the main conclusions could be summarized as following:

(1) The air thermal field, velocity field and pressure field of the vehicle could be acquired within a certain error range by a reasonable CFD numerical simulation.
(2) A function model including heat dissipation, fan speed and ambient temperature could be established by appropriate substitutions and rearrangements, which was solved by the application of Ferrari and Shengjin method, thus to obtain the characteristics of fan speed suitable for various working conditions.

(3) For a given heat dissipation amount, fan speed could be adjusted according to ambient temperature, thereby reducing power consumption while ensuring work efficiency of the engine. Application results showed that as ambient temperature increased, the rotational speed and shaft power increased nonlinearly, working condition 2 was relatively higher and working condition 4 reversed.

(4) The comparison between the numerical simulation results and designed value was made to verify the accuracy of the self-adaptation model, which could be applied to the design and development of cooling system for engineering vehicles within a certain error range.

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