Simulation and experiment of temperature field of different refrigerated trucks

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Abstracts. Road transportation is an important part of cold chain logistics. In order to study how the flow field and the temperature of the cold source influence the temperature field in the fully loaded car, computational fluid dynamics (CFD) technology is used to model and numerically simulate the mechanical refrigerated truck, dry ice refrigerated truck, and cold storage refrigerated truck. It found that there was a positive relationship between the cold source temperature and the air in the compartment. The velocity of air contributed to the uniformity of the air temperature in the truck. The location distribution of the cold source also had a certain effect on the temperature distribution of the air in the compartment and the surface temperature of the frozen product. Due to the difficulty of obtaining cold plate refrigerated trucks, the temperature field in vehicles was verified by experiments with mechanical refrigerated trucks and dry ice refrigerated trucks. The maximum absolute errors of each temperature measurement point are 0.94 °C and 1.15 °C, indicating that the numerical simulation results are reliable.

1. Introduction

In recent years, the total volume of China's agricultural product logistics has been increasing year by year. In 2018, the total value of China's agricultural product logistics reached 3.9 trillion yuan, an increase of 3.5%, accounting for 1.38% of the national total logistics [1]. However, the meat and aquatic products decay rate reached 8%, 10% [2]. In the transportation of fresh and perishable food, refrigerated transportation is an important part of ensuring food quality safety and quality [3]. At present, China's cold chain logistics is not yet perfect and mature, and the State Council also requires accelerated innovation and application of cold chain logistics technology and equipment [4], develop new types of cold storage materials and adopt advanced energy-saving and energy storage equipment [5]. Xie [6] conducted a numerical simulation on a cold plate refrigerator truck equipped with frozen pork and studied the effect of frozen products with tight stacking and interval stacking on the temperature field. Some researchers [7, 8] designed a refrigerated vehicle with a refrigeration unit, a cold storage unit, and an air supply system, so it can storage cold by plugging in at night when electricity was cheaper. It can control the temperature in the vehicle by adjusting the wind speed, solving the problem that the cold...
plate refrigerated vehicle cannot control the temperature. On this basis, Liu [9] increased the wind guide groove and the inner baffle to make it a multi-temperature refrigerated truck, so that it can continuously control the temperature for more than 10 h at -15 ℃ and 3 ℃. Copertaro [10, 11] designed to encapsulate the phase change material (PCM) which had a high melting point and add it to the insulation layer of the wall to reduce the maximum heat load in summer and reduce the difference between the maximum and minimum values, so that the average energy consumption was reduced 4.65%. The cold storage refrigerated truck had the characteristics of energy saving, simple structure, and easy maintenance, and does not need to consume diesel oil. It can be stored by the night valley electricity price to reduce costs, and had good application prospects in energy saving and emission reduction.

This paper analyzes the refrigerated truck loaded with frozen aquatic products through numerical simulation, using the basic k-ε two equation model and SIMPLE algorithm. According to the temperature of the cold source and the air velocity, three different types of refrigerated trucks were selected for comparison, those were mechanical refrigerated truck, dry ice refrigerated truck, and cold plate refrigerated truck. The effects of airflow velocity and cold source temperature on the cargo surface temperature field are discussed in conjunction with experimental validation. It can provide a theoretical basis for the optimal design of the cold storage refrigerated truck in the future. Because cold plate refrigerated trucks are uncommon and difficult to obtain through leasing, the boundary conditions are verified by mechanical refrigerated trucks and dry ice refrigerated trucks, by combining references to other models of low-speed incompressible fluids in other literature and verified dry ice refrigerated truck model to set the calculation model of the cold plate refrigerated truck.

2. Establishment of mathematical model and physical model of refrigerated truck

Using FLUENT 19.0 to numerically simulate the temperature field of the refrigerated truck, the temperature changes of the refrigerated truck in which the refrigeration unit was installed, dry ice was placed, and the cold storage plate was installed were calculated. Taking the car of the CIMC 4.2 m refrigerated truck as the research object, the internal dimensions of the car were 4200 mm (l) × 2000 mm (h), and the rated maximum load was 3000 kg, so 2700 kg vacuum-packed frozen fish meat stored in a carton pack was stacked in the center of the car. According to the specification [12] the storage density of frozen aquatic products was 470 kg/m³, and the stacking size was 3200 mm (l) × 1200 mm (w) × 1500 mm (h), which were placed in the middle of the car close to the front wall of the car. The outer wall of the carriage was 80 mm thick thermal insulation material. In the dry ice refrigerated truck, two open-top cuboid dry ice containers were placed above the cargo, with dimensions of 500 mm (l) × 500 mm (w) × 50 mm (h), filled with dry ice, and placed on the top of the frozen product along the length of the frozen product office. The cold plate refrigerated truck was equipped with six PCM cold storage plates, four on the left and right sides and two on the top, the size was 1000 mm (l) × 50 mm (w) × 1500 mm (h), the total mass was 520 kg. According to previous studies [13, 14], the cold plate was combined with top and side. Since the refrigerated compartments were symmetrical, in order to simplify the calculation, the model of the left half-sided compartment shown in Figure 1 was established with a symmetrical plane.
In order to facilitate the analysis, the following assumptions were made to the physical model:

1. The heat exchange rate between the external environment and the compartment of the refrigerated truck was even and constant;
2. The air in the car was consistent with the boussinesq assumption;
3. The parameters of the cold storage agent remain unchanged;
4. The temperature distribution in the cold storage plate was uniform at the initial moment;
5. Ignore the flow of the phase change cold storage medium in the PCM cold storage plate;
6. The initial temperature of frozen products in the compartment was -18 ℃.

3. Boundary conditions and solutions

3.1. Inlet

The wind speed at the outlet of the refrigeration unit of the mechanical refrigerated vehicle was 6.5 m/s, the temperature was -21 ℃, the turbulence intensity was 5% [15], and the hydraulic diameter was 500 mm. The dry ice container was a plastic box with an open top. In order to simplify the calculation, the opening of the container was set as the speed inlet.

\[ v = \frac{M}{\rho A} \]  (1)
The dry ice volatilization rate was measured according to the experiment $M=5.5 \text{ g/s}$, $\rho_{\text{CO}_2}=2.8 \text{ g/L}$ at $-78.5 \text{ °C}$, $A$ was the container opening size was $0.5 \text{ m}^2$, and the data used to calculate the wind speed was $0.04 \text{ m/s}$.

### 3.2. Boundary conditions
Because it was a frozen refrigerated transport vehicle, the heat load of the refrigerated vehicle was mainly composed of three parts: the heat load from the outside through the heat conduction into the compartment, the heat load caused by the leakage of the compartment, and the heat load inside the compartment caused by solar radiation [16].

1. Heat flux $q_1$ due to heat conduction and heat radiation
   \[ q_1 = K(t_{\text{ext}} - t_{\text{int}}) \]  
   $K$, the heat transfer coefficient of the carriage, $W/(m^2 \cdot \text{°C})$; $t_{\text{ext}}$, the external surface temperature of the carriage, °C; $t_{\text{int}}$, the temperature inside the car, °C.

   The external surface temperature $t_{\text{ext}}$ of the compartment was obtained according to actual measurement, the interior temperature $t_{\text{int}} = -18 \text{ °C}$, the heat transfer coefficient $K$ was calculated by formula (3)
   \[ K = \frac{1}{\frac{1}{h_1} + \frac{\delta_1}{\lambda_1} + \frac{\delta_2}{\lambda_2} + \frac{\delta_3}{\lambda_3} + \frac{1}{h_2}} \]  
   $h$, Convection heat transfer coefficient, $W/(m^2 \cdot \text{°C})$; $\delta$, thickness, m; $\lambda$, Thermal conductivity, $W/(m \cdot \text{°C})$.

   The heat exchange between the inner surface of the compartment and the air was regarded as natural convection, $h_1 = 10W/(m^2 \cdot \text{°C})$, the heat exchange on the outer surface of the compartment was forced convection, $h_2 = 20W/(m^2 \cdot \text{°C})$; the inner and outer face of wall was made of glass fiber reinforced plastic skin, thickness $\delta_1 = \delta_3 = 0.0025 \text{ m}$, thermal conductivity $\lambda_1 = \lambda_3 = 0.4W/(m \cdot \text{°C})$, the insulation layer was made of rigid polyurethane foam insulation board, thickness was $\delta_2 = 0.075 \text{ m}$, thermal conductivity $\lambda_2 = 0.025W/(m \cdot \text{°C})$. Bring each parameter into the formula (2) to obtain the heat transfer coefficient $K = 0.28W/(m^2 \cdot \text{°C})$ of the refrigerated truck compartment.

2. Heat flux $q_2$ due to vehicle air leakage
   \[ q_2 = \beta q_1 \]  
   $\beta$, the additional coefficient of thermal load caused by vehicle leakage, and $\beta = 0.3$ [17].

3. Total heat flux $q$
   \[ q = 1.1(q_1 + q_2) \]  

Combining the above formulas, the calculation results of the boundary conditions of each wall were shown in Table 1.

| Wall       | $t_{\text{ext}}$ (°C) | $q_1$ (W/m$^2$) | $q_2$ (W/m$^2$) | $q$ (W/m$^2$) |
|------------|----------------------|-----------------|-----------------|---------------|
| Top        | 24                   | 13.6            | 4.1             | 19.47         |
| Left and right | 24                   | 13.6            | 4.1             | 19.47         |
| Door       | 24                   | 13.6            | 4.1             | 19.47         |
| Floor      | 19                   | 12              | 3.6             | 17.16         |
| Front      | 19                   | 12              | 3.6             | 17.16         |

(4) Heat load $Q$
   \[ Q = \sum(q \ast A_{\text{out}}) \]  

$A_{\text{out}}$, the surface area of the wall.

The calculation resulted $Q=900 \text{ W}$. 

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3.3. Model
According to the conditions on the day of the experiment, the initial temperature of the air and the wall of the compartment was set at 19 ℃, and the temperature of frozen goods was set at -18 ℃. The mechanical refrigerated vehicle uses k-ε model. The dry ice refrigerated vehicle and the cold plate refrigerated vehicle refered to others' settings for numerical calculation of low-speed incompressible fluids [18], which used k-ε model, enhanced wall treatment, and full buoyancy effects. The gravity term was set to -9.8 m/s². Eutectic salt was used as cold storage material. The initial temperature was -23 ℃, slightly lower than the freezing temperature of -21.2 ℃. The material property parameters used are shown in table 2. Set the time step length to 1 s. When the average air temperature in the compartment stops dropping and the fluctuation was less than 1 ℃, the temperature inside the compartment was stable and the calculation was stopped.

| Material     | Melting point (℃) | Latent heat (kJ/kg) | Specific heat (kJ/kg K) | Density (kg/m³) | Thermal Conductivity (W/m·K) |
|--------------|-------------------|---------------------|-------------------------|-----------------|------------------------------|
| Eutectic salt | -21.2             | 235                 | 0.908                   | 1164            | 2.22                         |
| Air          | -78.5             | 573                 | 1.005                   | 470             | 0.02468                      |
| Dry ice      |                   |                     |                         |                 |                              |
| Fish [10]    |                   |                     |                         |                 |                              |

3.4. Calculation method
The model was drawn by computer-aided design (CAD), and the mesh was constructed by Mesh, which was a part of ANSYS 19.0. In order to prove that the simulation result was independent of the grid number, different size meshes were used for calculation. By referring to similar studies [19, 20], the number of meshes was set to 150000, 204000, 247000 were calculated separately, and found that the latter two grid divisions can have small differences in the three vehicles, the average temperature difference of 0.05 ℃, 0.32 ℃, 0.11 ℃ after 600s. Therefore, it could be considered that when the number of grids is 204,000, the calculation result was independent of the number of grids, and this division method was selected.

4. Analysis of simulation results
The results of the calculation were imported into Tecplot software for post-processing, and 5 sections are taken along the carriage x = 0.8 m, x = 1.6 m, x = 2.4 m, x = 3.2 m, x = 4.0 m. Figure 2 shows these 5 sections? The velocity field of figure 3 was the temperature field. It can be seen from the comparison of the temperature field and the velocity field that the temperature distribution of the air in the refrigerated truck was closely related to the distance from the cold source and the airflow speed.

4.1. Air velocity distribution
The mechanical refrigerated truck had an initial air velocity of more than 4 m/s due to the action of a fan. The airflow quickly decreased after impacting the door. The airflow flowed along the side wall and floor of the car to the front wall of the car, and then returns to the fan along the frozen product. Therefore, the air velocity near the wall surface was slightly higher and can reach 2 m/s, while the rest was less than 1 m/s.

Dry ice refrigerated trucks and cold plate refrigerated trucks used gravity as the driving force due to the density change caused by temperature, and the flow rate was relatively slow. After the sublimation of dry ice, the denser carbon dioxide flows down the surface of the frozen product and gradually mixes with the air. Therefore, the surface speed of the frozen product was higher, which reached to 0.3 m/s. When the cold air reaches the floor, it flows backward to the rear of the carriage, so the velocity was higher near the floor in the behind of the carriage. After the wall surface was heated, the density
decreased, and it flows upward to the top surface of the cabin, and flows to the dry ice container under the action of the pressure difference to cool and mix with low-temperature carbon dioxide.

(a) Mechanical refrigerated vehicle              (b) Dry ice refrigerated vehicle

(c) Cold plate refrigerated vehicle

Figure 2. The velocity fields of refrigerated vehicles.

The air velocity in the cold plate refrigerated truck was less than 0.2 m/s. The air on the surface of the cold plate was cooled by surface convection heat transfer and heat conduction. After the air cools, the density becomes larger and then flows downward. Therefore, the velocity around the cold plate and near the floor was larger. The air mainly flows close to the wall, and the air velocity near the wall was less than 0.1 m/s.

4.2. Temperature distribution

As can be seen from figure 3, the temperature inside the mechanical refrigerated vehicle was the most uniform due to the large airflow velocity. The temperature difference was within 1 °C, and the temperature gradually increased along the airflow path. The temperature in the front half of the vehicle was slightly higher than that in the rear half. The air temperature on the surface of the goods was -
21.5 °C, slightly lower than the temperature of frozen goods, and the temperature of the part attached to the carriage wall was slightly higher. According to figure 2 (a) and figure 3 (a), the temperature distribution was consistent with the airflow velocity distribution, which is consistent with other study [21].

In the dry ice refrigerated vehicle, the sublimation temperature of dry ice was -78.5 °C, and the cold source temperature was the lowest among the three modes of transportation. Therefore, the internal temperature of the vehicle was the lowest, and the temperature difference at each point was the largest among the three refrigerated vehicles. The temperature at the top of the compartment was the highest, and the temperature of the airflow on the surface of the frozen product increases as the height decreases. Frozen goods on both sides of the air temperature becomes more uniform as the height decreases. The position of the container and the frozen product was in front of the compartment. After the cold air reaches the floor, it flows to the rear of the compartment. Therefore, the air temperature in the front half of the compartment was lower than that in the rear half. The air temperature on the upper surface of frozen goods is below -40 °C, and the side temperature varies greatly from -25 °C to -37 °C.

The temperature of the cold plate refrigerator car was relatively uniform, the temperature difference is within 2 °C. The bottom temperature was lower than the top temperature. Because the temperature difference between the cold plate and the air inside the compartment was smaller, the air temperature far away from the cold plate and the frozen goods was slightly higher. The air on the surface of the frozen product was relatively uniform and maintained at -18.5 °C. The cold air cooled by the cold plate at the top of the carriage passed directly over the upper surface of the frozen goods, so the temperature of the upper surface of the frozen goods is slightly lower [6], and the part close to the carriage wall is slightly higher. The phase change phase of the cold plate can absorb latent heat of 122200 kJ, which can be maintained for 37 hours in theory. Besides, the mass of the refrigerant accounts for 15% of the vehicle load. Therefore, if the transportation time is short, the loading amount of the refrigerant can be appropriately reduced to increase the cargo carrying capacity of the refrigerated vehicle without affecting the thermal insulation effect.

(a) mechanical refrigerated vehicle
4.3. Experiment

In order to verify the correctness of the simulation calculation of the temperature field of the refrigerated vehicle and obtain some boundary conditions required for the calculation, the temperature field was measured and verified by renting a refrigerated vehicle (CIMC 4.2 m refrigerated vehicle). The Testo 175 T2 and Testo 176 T2 temperature recorders were used to collect the temperature inside the vehicle, with the accuracy of 0.01 °C. According to the refrigerated vehicle test standard [22], the sensor probe was placed on the inner surface of the compartment door, side wall and top surface. Probes are 10 cm away from the wall surface, as shown in figure 4. Data were collected every 10 s. In the compartment, 24 cartons of 800 mm × 600 mm × 500 mm in size are stacked tightly, and tape was used to wrap and fix them to simulate the stacking of goods. The gap at the junction of the cartons was ignored. At the bottom of each carton, 1 kg of bagged water was placed as the counterweight, and it was put into the cold storage for pre-cooling in advance, in order to prevent the box from moving in the process of vehicle driving and reduce the temperature fluctuation when the carton was loaded. 

Figure 3. The temperature fields of refrigerated vehicles and cargo surface.

(b) dry ice refrigerated vehicle

(c) cold plate refrigerated vehicle
The experiment was conducted on November 21, 2019, which was a cloudy day with weak solar radiation and a temperature of 19 ℃ and a carriage surface temperature of 24 ℃. The experiment was divided into two stages. The first phase was the mechanical refrigerated vehicle experiment: the refrigerated vehicle was equipped with a Thermal Master refrigerating unit, and the onboard temperature sensor was used to monitor the internal temperature with a precision of 0.1 ℃. The vehicle traveled along the highway at a constant speed of 60 km/h in a straight line, with fewer vehicles on the road and fewer intersections. When the temperature in the car reached the preset temperature of -18 ℃, the temperature fluctuation did not exceed 0.5 ℃. It was considered that the temperature in the car had been stable, and the temperature data were collected. At the end of the experiment, opened the compartment door, and when the air temperature in the compartment reached 17 ℃, started the second phase of the dry ice refrigerator car experiment. put 2 boxes of dry ice, each box containing 10 kg cylindrical dry ice particles. Considering the large temperature difference between the inside and outside of the dry ice refrigerated vehicle and the slow response to the temperature fluctuation due to natural convection, when the temperature inside the vehicle no longer dropped and the fluctuation did not exceed 1 ℃, the temperature inside the vehicle was considered to be stable and temperature data could be collected.

![Figure 4. Layout of planar monitoring points.](image)
Figure 5. Photos of the experiment.

Figure 5 (a) and Figure 6 (a) respectively show the comparison between the experimental value and the simulated value of the mechanical refrigerated vehicle and the dry ice refrigerated vehicle. By comparing the test value with the simulation value, the average error between the measures temperature and the calculated value of the mechanical refrigerated vehicle at each point is 0.94 °C, and the average error of the dry ice refrigerated vehicle is 1.15 °C, which verifies the reliability of the simulation results. When the experimental value of the temperature at the compartment door is equal to the simulated value, compare the experimental value with the simulated value at the remaining points, Figure 6(b) and Figure 7(b) show the temperature comparison between the mechanical and dry ice refrigerated vehicle temperature measurement point 3. It can be seen in different times; car is obtained by computing the temperature field of accords with actual temperature field. The simulation value is always slightly lower than the experimental value, which may be caused by the simplification of the model, the change of solar radiation caused by the cloud movement, and the increase of cold wind infiltration caused by the aging of the carriage wall.
Figure 7. Comparison of dry ice refrigerated vehicle.

5. Conclusion
In order to study the influence of air flow velocity and cold source temperature on the transportation environment of frozen goods in refrigerated vehicles, three common and typical refrigerated vehicles were selected in this paper: mechanical refrigerated vehicle, dry ice refrigerated vehicle, and cold plate refrigerated vehicle. These three vehicles were simulated by CFD and experiments were conducted on mechanical refrigerated vehicles and dry ice refrigerated vehicles. The average error between the measured value and the calculated value was 0.94 ℃ and 1.15 ℃, which verified the reliability of the simulation results. Therefore, the following conclusions are drawn:

(1) The temperature in the compartment of the traditional mechanical refrigerated vehicle is the most uniform, but the flow velocity at each point is quite different, and the airflow is rapidly weakened when it hits the door. The wind speed in most places is less than 2.5 m/s, far less than the exit wind speed of 6.5 m/s.

(2) The internal flow rate of the dry ice refrigerator car is relatively uniform, but the temperature difference is large. The air temperature around the frozen goods is lower than -25 ℃, and the temperature of the car is lower than the demand, which increases the heat exchange of the car wall. Since the storage temperature is much lower than the storage temperature, the storage temperature of frozen goods will fluctuate, and the quality of frozen goods will be affected by multiple transportation.

(3) The airflow velocity inside the plate refrigerator is the lowest, but the cold source temperature is close to the air temperature, so the temperature is more uniform. But because of the lower airflow velocity inside the vehicle, more cold plates are needed to get enough heat transfer area, resulting in higher quality refrigerant being loaded, which reduces the load on the vehicle.

Based on the results it can be concluded that: Thus, the air temperature in the refrigerated compartment is related to the temperature of the cold source. It is found that there was a positive relationship between the cold source temperature and the air in the compartment. The location distribution of the cold source has a certain influence on the air temperature distribution in the compartment and the surface temperature of frozen goods. The uniformity of air temperature in the cold compartment is related to the airflow velocity. The greater the flow velocity is, the more uniform the air temperature in the compartment will be. However, due to the space limitation, the maximum wind speed will continue to be increased after the fan outlet speed reaches a certain value, and the effect on improving the average velocity will be weakened.

In the future, some optimization still needs to be carried out, such as increasing the uniformity of temperature field by increasing the power of air flow provided by the fan, improving the position of cold plates to reduce the number of cold plates, developing refrigerant with higher latent heat value to reduce the load of refrigerant carrier, and determining the number of cold plates according to the transport time.
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