CFD-Based Simulation Research and Structural Improvement Design of Ship Intercooler

Li Bingqu, Liu Yongqi, Lu Min, He Xiaozhen

Abstract—**Taking the L6230ZLC-10 high-speed diesel engine produced by Zichai Machinery Co., Ltd. as the original engine, the intercooler is modeled and simulated by CFD, and the fin structure with five thicknesses and spacings is used to study the different structures of the intercooler. The influence of heat transfer characteristics and resistance characteristics. The results show that with the new fin size intercooler, the temperature difference between the inlet and outlet of the pressurized air can be increased by a maximum of 17.4%, the pressure drop can be reduced by a maximum of 39.6%, and the overall performance is increased by an average of 39.333%. Provide a certain theoretical design basis.**

Index Terms—**CFD, intercooler, fin structure, heat transfer characteristics, resistance characteristics**

I. INTRODUCTION

In recent years, the development level of marine diesel engines has rapidly increased, and supercharged intercooling technology is widely used, and has developed into one of the key technologies of marine engines [1]. The high turbocharger strength of the engine causes the problem of excessively high temperature after the turbo, which puts higher requirements on the performance of the intercooler. How to improve the heat dissipation performance of the intercooler while ensuring the larger power of the engine has become a major problem facing marine engines. Most marine engines use turbocharging. The turbocharger performs work on the high-temperature exhaust gas discharged from the marine engine through the turbine blades to supercharge it, and the engine's inflation efficiency is improved. However, the temperature of the high-temperature exhaust gas will increase further after being pressurized. If not handled properly, the high-temperature gas combustion will reduce the output power of the engine, and knocking will also occur in severe cases. In addition, the intake air temperature will affect the NOx content in the emissions. Experiments show that the higher the intake air temperature, the higher the NOx content. Therefore, the heat exchange capacity of the intercooler determines the power, economy and emission performance of the engine to a certain extent, so the research of the intercooler has important practical significance.

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At present, Chinese and foreign scholars have carried out some meaningful exploration work on intercooler simulation research and structural improvement. In terms of calculation methods, Seyed Mohsen Momeni et al. [2] performed multi-objective optimization on the marine diesel engine intercooler through the firefly algorithm. As a result, the air pressure drop was reduced by 2.95%, the water pressure drop was reduced by 12.41%, and the total cost was reduced by 4.03%. Pierre Salmon et al. [3] conducted a comprehensive analysis of the heat exchanger structure, proposed a three-digit computational fluid dynamics method, and theoretically predicted the total heat transfer rate of the fins. The results confirmed that the values extracted from CFD can be used in the F1 engine cooling cabin Design process.

In terms of vehicle radiator structure, Huang Rui et al. [4] found that fin height and peak distance have an important effect on the heat transfer performance of the heat exchanger, reducing the height of the hot side channel, increasing the number of cooling bands or turbulent fins, and reducing The wave pitch and pitch of the fins will enhance the heat dissipation capacity of the fins. In order to achieve higher cooling performance, M. Harada et al. [5] optimized the matrix of fins, tubes and cores so that they have higher performance and lighter quality. Vaisi A [6] showed in the study that the heat transfer performance of the symmetrical louvered fins was improved by 9.3% and the pressure drop was reduced by 18.2% compared with the asymmetrical louvered fins. Based on CFD, Guo Jianzhong [7] performed performance analysis and fin structure optimization on a certain type of automotive tube-belt louver radiator. It was concluded that the window performance was the best when the window angle was 27° and the spacing was 2.4 mm. Xiao Shougao [8] designed and developed a new type of convex tube automobile radiator structure assembly. Compared with the smooth tube radiator, the heat dissipation area increased by more than 10%.

In the combination of experiment and simulation, Huang Pengchao et al. [9] established two-dimensional and three-dimensional models of radiators based on CFD, and obtained the best heat dissipation effect of lateral dissimilar materials by comparing experiments and simulations. Jing et al. [10] established a radiator composed of B-type water pipes and louver fins. Through the combination of CFD numerical simulation and wind tunnel test, it was concluded that the heat transfer capacity of the radiator increased with the increase of the air inlet velocity. The flow resistance and pressure drop also increase. Yuan Yu [11] based on a commercial vehicle's intercooler temperature field and flow field, using a combination of experiments and simulations, studied the relationship between the intercooler front deflector and cooling flow rate utilization.

In summary, most of the existing research focuses on vehicle radiators, and the focus is on tube-type radiators. In this paper, the thickness and spacing of the fins of the tube-fin radiator will be used as the research object to explore the relationship between the fin structure and the heat exchange characteristics and resistance characteristics of the intercooler, which will provide a theoretical basis for the actual production of the intercooler.

II. PHYSICAL MODEL AND EXPERIMENTAL VERIFICATION

The model in this paper is based on the intercooler of L6230ZLC-10 high-speed diesel engine produced by Zichai Machinery Co., Ltd. The engine parameters of this model are shown in Table 1.

Table 1. Parameters of L6230ZLC-10 high-speed diesel engine

model	L6230ZLC-10	
Cylinder and structure	6-cylinder, in-line	
Cylinder diameter	225	mm
Piston stroke	290	mm
Rated power	1213	Kw
Rated speed	1000	r/min
Theoretical compression ratio	14.0	

Considering the computer performance and efficiency issues, the intercooler of the entire engine cannot be calculated in the actual simulation process, and the actual physical model is appropriately simplified without affecting the calculation results.

(1) Considering the computer performance and efficiency issues, the intercooler of the entire engine cannot be calculated in the actual simulation process, and the actual physical model is appropriately simplified without affecting the calculation results.

(2) In the simulation calculation, a steady state process (steady flow) is set, that is, the flow heat transfer process does not change with time.

(3) The arrangement of the cooling water pipes of the original intercooler model is 20×20 and the length is 400mm. Four of them are selected for the simulation calculation and the length is 10mm.

(4) The fins in the original intercooler model are formed by stamping, and the shape is close to a sinusoidal curve, which is approximated to a sinusoidal curve during simulation calculation.

(5) The surface of the intercooler is rough in actual production, and the surface of the intercooler is treated as a smooth surface in the simulation process.

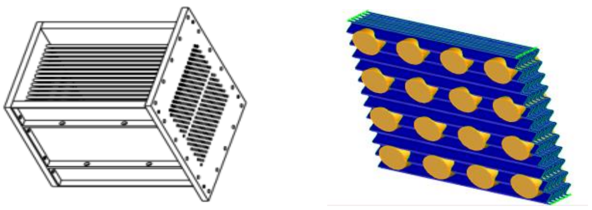

Fig.1 Model simplification

The fluid inlet is defined as the velocity inlet boundary. It can be seen from the experiment that the cooling water inlet velocity is 1.56m/s, the hot gas inlet velocity is 4.6m/s, and the outlets are all set as pressure outlet boundaries. The outer shell is defined as the boundary of the thermal insulation surface, and the inner cooling tube and fins are defined as the boundary of the constant temperature wall.

On the one hand, to ensure the accuracy of the calculation, on the other hand, to improve work efficiency, but also to consider the calculation speed of the computer, we need to deal with the grid. The overall grid structure and local encryption grid structure are shown in Fig. 2.

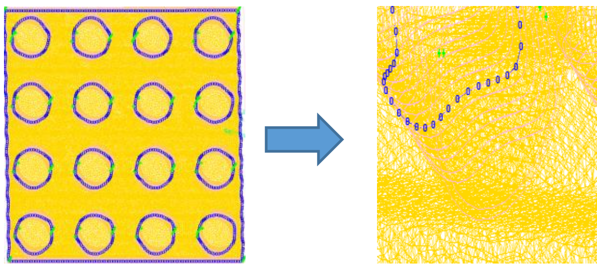

Fig.2 Grid model

In order to improve the reliability of the simulation calculation, four groups of grids are tested when the model is created. Establish the original machine calculation model, calculate the outlet temperature under 100% working conditions, and obtain the simulation results shown in Table 2.

Table 2. Comparison of numerical simulation results of different grid schemes

	Number of grids	Outlet temperature/K	Relative change rate/%
A	1153427	427.254	-
B	1029563	426.778	0.111
C	948136	426.604	0.152
D	824479	424.848	0.563
E	674603	423.692	0.833

It can be seen from Table 2 that the relative change rates of the four schemes B, C, D and E are all within 0.85%. On the premise of ensuring the calculation accuracy, the scheme C with a smaller number of grids is selected as the grid division scheme in this paper.

Mass conservation equation:

The rate of mass increase in the microelement is equal to the static mass flow rate into the microelement, which is the law of conservation of mass [12].In this paper, the fluid is an incompressible fluid, ρ is a constant, steady-state flow, ρ has nothing to do with t, and its equation is:

$$div(\rho U) = 0 \qquad (1)$$

In the formula, ρ represents the density of the fluid; U represents the velocity of th

Momentum conservation equation:

The momentum growth rate of the fluid microelement is equal to the resultant force acting on the fluid microelement, which is the law of conservation of momentum [13]. From this law, the N-S equation can be derived and because it is an incompressible fluid, there are:

$$\frac{\partial u}{\partial t}+div(uU)=div(\eta gradu)-\frac{1}{\rho}\cdot\frac{\partial \rho}{\partial x} \tag{2}$$

$$\frac{\partial v}{\partial t}+div(vU)=div(\eta gradv)-\frac{1}{\rho}\cdot\frac{\partial \rho}{\partial y} \tag{3}$$

$$\frac{\partial w}{\partial t}+div(wU)=div(\eta gradw)-\frac{1}{\rho}\cdot\frac{\partial \rho}{\partial z} \tag{4}$$

The resultant force acting on the fluid microelement, which is the component of u, v, w, U on the x, y, and z axes in the momentum conservation type; η Stands for dynamic viscosity ; t stands for time.

Energy conservation equation:

The energy growth rate of the fluid microelement is equal to the net heat transferred to the fluid microelement and the network acting on the fluid microelement, namely the law of conservation of energy [13]. Because it is an incompressible fluid, Fourier's law of heat conduction is:

$$\frac{\partial T}{\partial t}+div(UT)=div(\frac{\lambda}{\rho\cdot c_p}\cdot gradT)+\frac{S_T}{\rho} \tag{5}$$

In the formula, T stands for fluid temperature; C_p represents for Constant pressure specific heat capacity; S_t stands for Viscous dissipation.

The experiments in this paper were carried out on a bench test bench, and the experiments were carried out under the conditions of 110%, 100%, 90%, 75%, 50%, and 25%, respectively, to obtain the intercooler outlet temperature. Taking the established original machine model as the calculation model, the gas temperature under six working conditions is simulated and calculated. The experimental and simulation results are shown in Fig. 3.

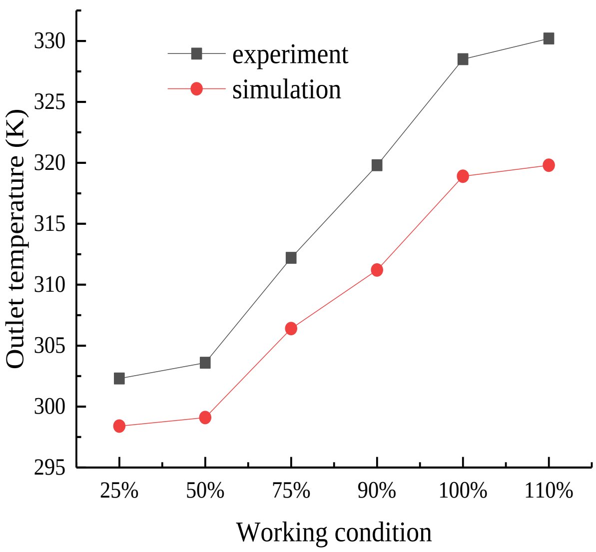

Fig. 3 Comparison of experimental and simulated results

In order to accurately evaluate the usability of the built model, the post-cooling temperature measured by the bench experiment was compared with the simulated intercooler outlet temperature and an error analysis was made. It should be noted that the length of the model used in the simulation calculation is 1/20 of the length of the original intercooler. After calculation, the relative error between the maximum is 3.15% and the minimum is 1.29%. The relative error is within a reasonable range. Considering the reasons of model simplification, the established intercooler model is reasonable and can be calculated based on this model.

III. RESULT

Fig. 4 is a cloud diagram of the inlet and outlet temperature changes when the fin thickness is 0.8mm, 1mm, 1.2mm, 1.4mm, and 1.6mm under 100% engine operating conditions. Under the same operating conditions, as the thickness of the fin increases, the temperature difference between the inlet and outlet of the charge air becomes smaller and smaller. This is because the greater the thickness of the fin, the weaker its turbulence on the charge air and the more the air flow Quickly, the cooling effect becomes worse under the same conditions.

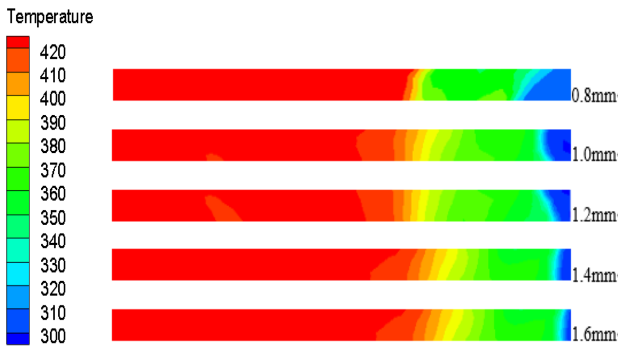

Fig. 4 Cloud diagram of temperature changes at the inlet and outlet of fins with different thicknesses

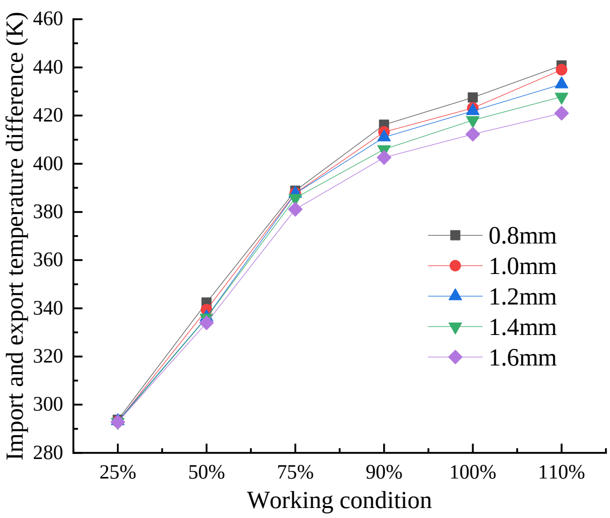

Fig. 5 Effect of fin thickness on inlet and outlet temperature

Fig. 6 is a cloud diagram of pressure changes at the inlet and outlet of compressed air when the fin thickness is 0.8mm, 1.0mm, 1.2mm, 1.4mm and 1.6mm under 100% working condition of the motive. Under the same operating conditions, the thicker the fin thickness, the more obvious the inlet and outlet pressure changes of the charge air. This is because as the thickness of the fin increases, the charge air is subjected to greater resistance by the fin during the flow process. The thickness of the first two groups of fins has a significant effect on the air outlet pressure. This is because when the fin thickness is thinner, the change in fin thickness has a greater effect on the pressure drop. When the thickness of the fin changes from 1.2m to 1.4m, and then increases from 1.4m to 1.6mm, the pressure change of the inlet and outlet of the charge air is not so obvious.

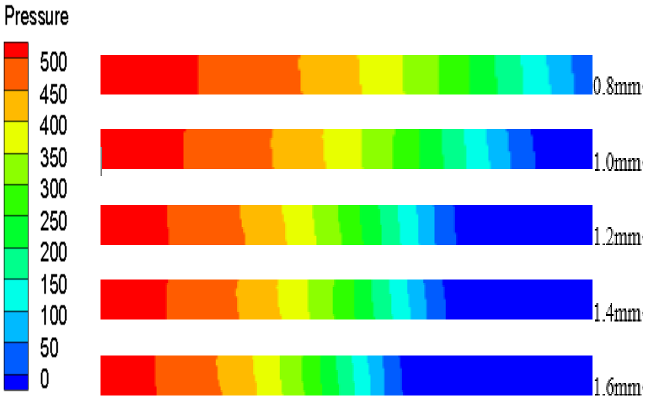

Fig. 6 Cloud diagram of pressure changes at the inlet and outlet of fins with different thicknesses

Fig. 7 shows the effect of fins of different sizes and thicknesses on the pressure drop of the charge air inlet and outlet under different working conditions. Overall, as the engine power increases, the pressure drop of the charge air inlet and outlet increases faster. This shows that the effect of fin thickness on pressure drop is more obvious under higher operating conditions. It can also be seen from the Fig. that at 0.8mm fin thickness, the inlet and outlet pressure drop of the intercooler is the smallest, the resistance of the charged air during the flow is the smallest, and the cooling effect of the intercooler is the best.

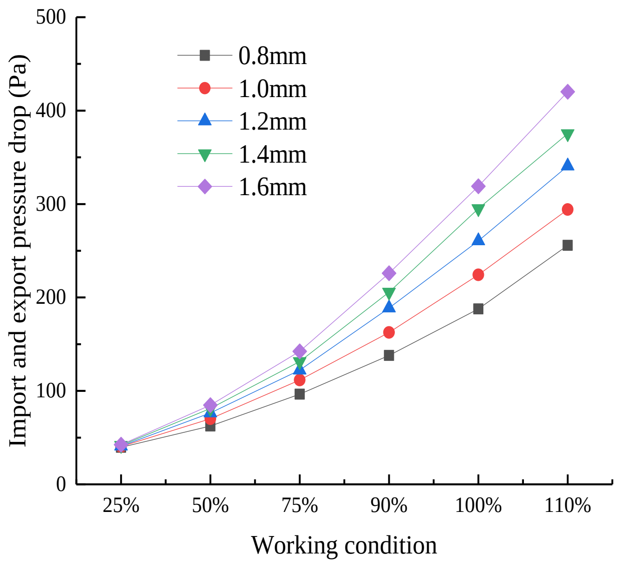

Fig. 7 Influence of fin thickness on inlet and outlet pressure

For the cooling effect of the intercooler, the greater the temperature difference between the inlet and outlet of the charge air, the smaller the pressure drop, the better the cooling effect. When the fin thickness is used as a variable, the temperature difference is used as the numerator, and the pressure drop is used as the denominator. The cooling capacity of the reaction intercooler. As shown in Fig. 8. It can be seen from the Fig. that as the engine operating conditions increase, the comprehensive performance factor first increases and then decreases, reaching a maximum value at 75% operating conditions and a minimum value at 110% operating conditions. It can also be seen from the Fig. that when the fin thickness is 0.8mm, the overall performance factor is much better than other size thickness fins, because the 0.8mm thickness fins have a good turbulence on the charge air It has no obvious hindrance to the hot air, so the fins with a thickness of 0.8mm are finally selected.

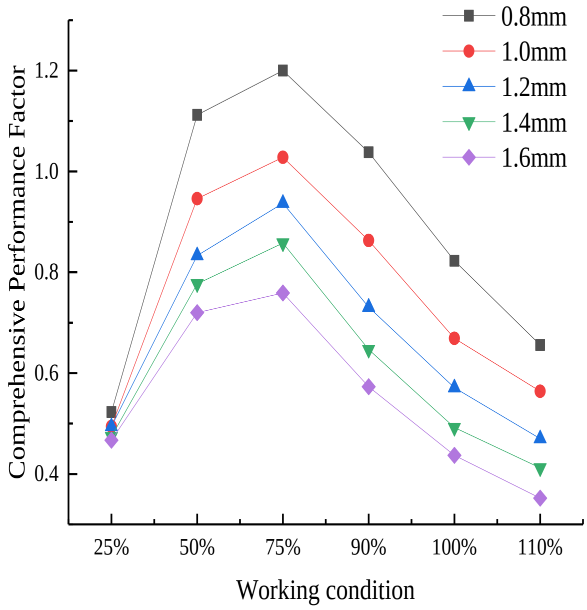

Fig. 8 Influence of fin thickness on comprehensive performance

Fig. 9 is a cloud diagram of the temperature changes at the inlet and outlet when the fin spacing is 0.9mm, 1.2mm, 1.5mm, 1.8mm, 2.1mm under 100% engine operating conditions. Under the same operating conditions, with the increase of the fin spacing, the temperature difference between the inlet and outlet of the pressurized air becomes larger and larger, and the maximum value is reached when the fin spacing is 1.8 mm. This is because the larger the fin spacing, the more the fins increase. The smaller the resistance effect of the compressed air, at the same time it plays a good turbulent effect. When the fin spacing increases to 2.1mm, the temperature difference between the inlet and the outlet decreases. This is because the fin spacing is too large. Although the resistance to the charge air is reduced, the turbulence of the fins is also reduced. The cooling effect of the intercooler is weakened.

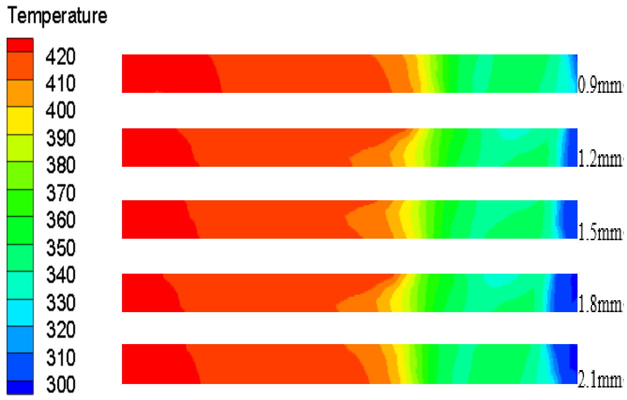

Fig. 9 Cloud diagram of temperature changes at the inlet and outlet of fins with different pitches

Fig. 10 shows the influence of the five pitch fins on the inlet and outlet temperature of the charge air under different working conditions. Compared with the fin thickness, the effect of fin pitch on the heat dissipation capacity of the intercooler is more obvious at low operating conditions, and the relative change rate between the five pitch fin structures is more obvious than the fin thickness. The reason is that the change in fin spacing has a more obvious effect on the air flow. The larger the fin spacing, the smaller its resistance to charge air.

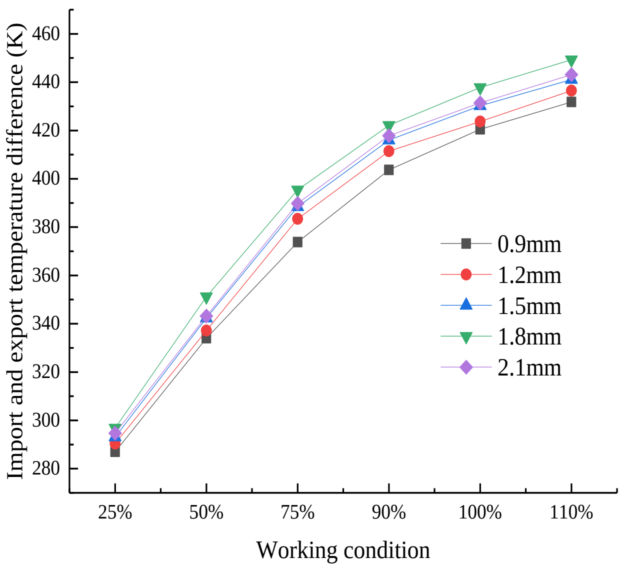

Fig. 10 Effect of fin spacing on inlet and outlet temperature

Fig. 11 is a cloud diagram of inlet and outlet pressure changes when the fin spacing is 0.9mm, 1.2mm, 1.5mm, 1.8mm, and 2.1mm under 100% engine operating conditions. Under the same operating conditions, with the increase of the fin spacing, the pressure drop of the inlet and outlet of pressurized air becomes smaller and smaller, and when the fin spacing is 1.8mm, the pressure drop reaches the minimum value, and the fin spacing continues to increase. The pressure drop at the inlet and outlet increases instead. This is because although the resistance is decreasing, the turbulence of the fins is also weakening, so the best effect is when the fin spacing is 1.8mm.

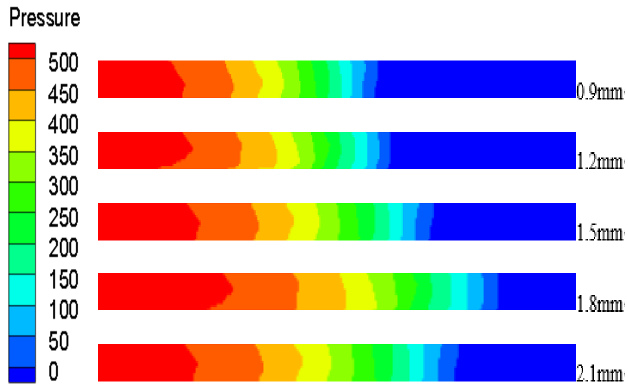

Fig. 11 Cloud diagram of pressure changes at the inlet and outlet of fins with different pitches

Fig. 12 shows the effect of five pitch fins on the pressure drop of the inlet and outlet of pressurized air under different working conditions. With the increase in engine operating conditions, the increase rate of pressure drop at the inlet and outlet of pressurized air is getting higher and higher, and compared with the effect of fin thickness on pressure drop, the effect of fin spacing on the pressure drop of compressed air inlet and outlet is more obvious , Indicating that the fin pitch has a more obvious effect on the pressure drop than the fin thickness. The pressure drops of fins with a pitch of 0.9mm and 1.2mm is much higher than that of the other three groups. This is because the small fin pitch causes a large air flow resistance, which is not conducive to heat dissipation of the radiator.

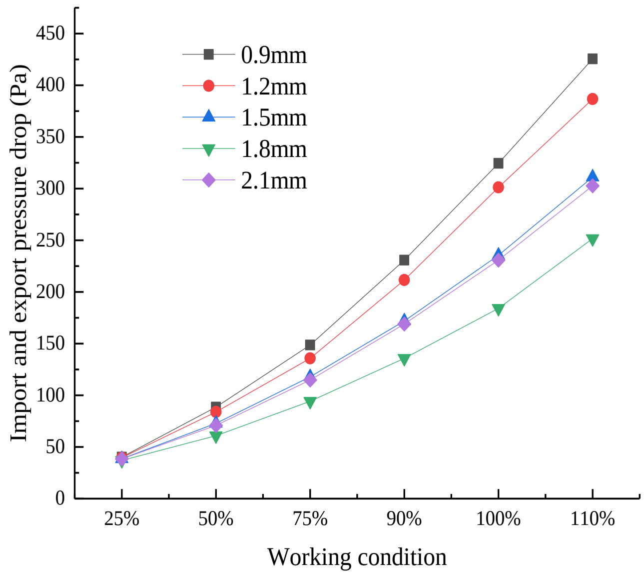

Fig. 12 Influence of fin spacing on inlet and outlet pressure

The variation of the comprehensive performance factor with engine operating conditions when the fin thickness is used as a variable is shown in Fig. 13. Except for the fin spacing of 0.9mm, the other four groups have the maximum comprehensive performance factor at 75%. When the fin spacing is 1.8mm, the cooling effect of the intercooler is optimal, which is used as the fin spacing size of the new intercooler.

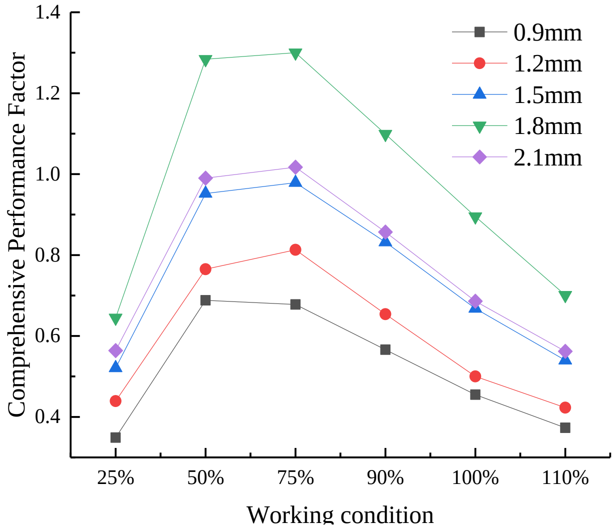

Fig. 13 Influence of fin spacing on comprehensive performance

The fin thickness is 0.8mm and the fin spacing is 1.8mm as the new intercooler fin structure size. A new intercooler model is constructed for simulation, and the temperature difference and pressure drop are compared with the traditional intercooler. The comparison results are shown in the Fig. 14, 15, and 16 are shown. It can be drawn from Fig. 14 that the temperature difference between the import and export of the new fins increased by 17.4%, 15%, 7.3%, 7.7%, 8.2%, and 9.4% under six operating conditions, with an average increase of 10.833%. It can be seen from Fig. 15 that under the six working conditions, the pressure difference between the inlet and the outlet of the new fins decreased by 10.8%, 30%, 31.8%, 27.6%, 26.3%, and 25.5% respectively, with an average decrease of 25.333%. It can be seen from Fig. 16 that the overall performance of the new fins has increased by 31.7%, 28.1%, 33.9%, 48.7%, 46.8%, and 46.8% under six operating conditions, with an average increase of 39.333%.

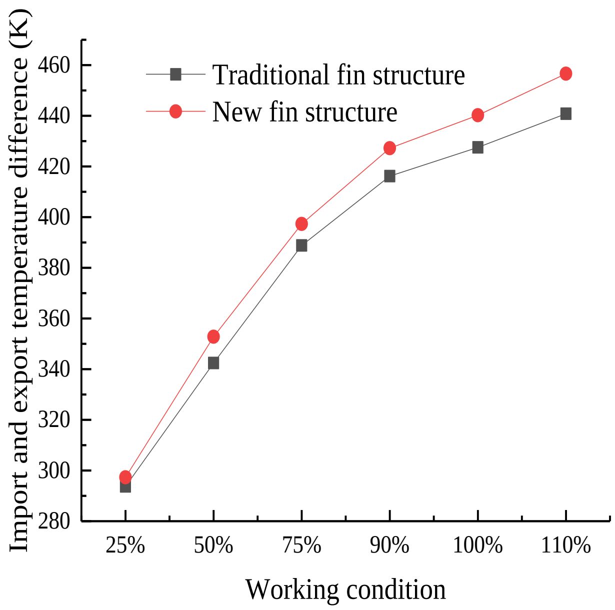

Fig. 14 Comparison of the influence of new and old fin structure on temperature

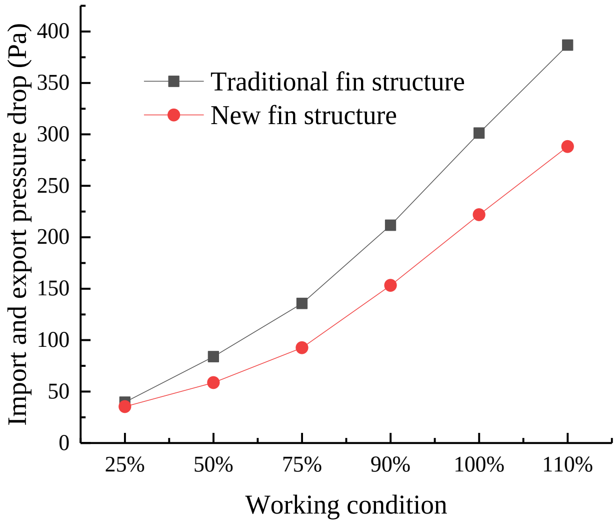

Fig. 15 Comparison of the influence of new and old fin structure on pressure

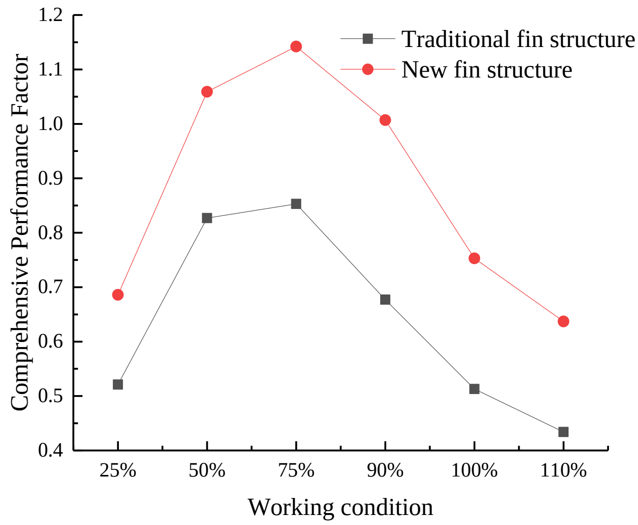

Fig. 16 Comparison of the influence of new and old fin structure on comprehensive performance

IV. Conclusion

For the purpose of improving the cooling performance of the intercooler, using the temperature difference and pressure drop of the inlet and outlet of pressurized air as evaluation indicators, a new intercooler model was constructed and compared with the traditional intercooler model. (1) When the intercooler fin thickness is used as a variable, as the fin thickness increases, the temperature difference between the inlet and outlet of the pressurized air becomes smaller and smaller, and the pressure drop becomes larger and larger. Under 100% operating conditions, the thickness of the fins increased from 0.8mm to 1.6mm, the temperature difference between the inlet and the outlet decreased from 154.5°C to 139.3°C, and the pressure drop increased from 187.8Pa to 319.1Pa. For comprehensive consideration, the new intercooler chooses 0.8mm The thickness of the fins. (2) When the intercooler fin spacing is used as a variable, as the fin spacing increases, the temperature difference between the inlet and outlet of the pressurized air increases first and then decreases, and the pressure drop decreases first and then increases. The inflection points are all 1.8 mm. Under 100% operating conditions, the temperature difference between the inlet and the outlet increased from 147.5°C to 164.8°C and then to 158.4°C, and the pressure drop from 324.5Pa to 184.2Pa and then to 230.8Pa. For comprehensive consideration, the new intercooler uses 1.8mm Spacing fins. (3) Compared with the traditional intercooler, the new intercooler with a fin thickness of 0.8mm and a pitch of 1.5mm is supercharged. The air inlet and outlet temperature differences increased by an average of 10.833%, the pressure drop decreased by an average of 30.833%, and the overall performance increased by an average of 39.333%.

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