Modelling the Cylinder Cooling in an Oil-free Reciprocating Air Compressor

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Abstract. Oil-free air compressors are widely applied in new energy vehicles, pharmaceutical, textile, food and the chemical industry for providing clean air. However, high temperature inside the cylinder due to compression heat and oil-less lubrication may lead to premature failure of the self-lubricating sealing rings and the grease-lubricating bearings. This paper presents the simulation of the cylinder cooling in an oil-free reciprocating air compressor for new energy vehicles. Based on the modelling, the cooling channel outside the cylinder was optimized to enable more effective cooling with limited air flow, obtaining lower temperature level of the cylinder, and therefore higher reliability of the sealing rings and bearings. The results show that the heat transfer model of the cylinder with numerical simulation is effective in predicting the cylinder temperature level as well as the cooling capacity of the air flow. The heat exchange efficiency of the compressor can be improved significantly by applying the optimized cooling duct structure. Compared with the model A, the maximum cylinder temperature was decreased by 13.1°C in the model B, and it also was decreased by 6.3°C in the model C.

Keywords: Oil-free; Air compressor; Heat transfer; Numerical simulation

1. Introduction
Oil-free air compressors are widely applied in new energy vehicles, pharmaceutical, textile, food and the chemical industry for providing clean air. In addition, there are often strict requirements for oil content, sometimes even 100% without oil in many other production and service facilities requiring clean air [1]. The oil-free air compressor can reduce the replacement requirements of oil separator components and the air dryer of the compressor, the cost of consuming lubricating oil and eliminate the pollution of desiccant in the air dryer, and also need not to increase the energy loss caused by the oil separator. In this process and the final product in the air quality is relatively strict requirements, oil-free compression is essential. That's because even the smallest amount of oil pollution can lead to costly production shutdowns, product corruption, product recalls and reputational damage. In order to solve these problems, many companies are turning to oil-free air compressors. As a result, oil-free compressors are becoming more and more common for saving customers production costs.

The global market size of oil-free air compressor is expected to start from 10 billion dollars in 2015 and grow at an annual compound growth rate of 4.2% from 2016 to 2023 [2]. Growth has been driven by demand for key applications of air quality, improved energy efficiency and the adoption of portable technologies. And evidence from industries that have been in use, including the food, beverage and...
pharmaceutical industries, suggests that the use of clean air is becoming increasingly important. Moreover, the oil-free air compressor market is winning out from the food and beverage industry, largely due to a number of regulations governing the safety and health aspects of the industry, and the growing needs to use oil-free air compressors to ensure product quality.

In addition, for the reason that piston rings and guide rings are generally made of solid non-metallic self-lubricating materials [3], their working temperature is much lower than that of metal rings. Therefore, the exhaust temperature of compressor at all levels should be as low as possible, so that piston rings and guide rings have a higher operating life, which can also improve the reliability of rings and bearings.

In this paper, the cooling effect of oil-free air reciprocating compressor were studied from three aspects. Firstly, the model for an oil-free reciprocating air compressor was studied from three aspects. Secondly, the cooling duct structures of the established compressor model were improved, and the heat dissipation effect of the model under different thickness of air duct deflectors is also compared. Finally, the effectiveness of the improved models and different air flow rates was verified and compared by numerical simulation.

2. Model of the double V-type oil-free reciprocating air compressor

2.1. Compressor model A

The geometry of the model A for an oil-free air compressor was simplified before numerical modelling due to its complexity as shown in Figure 1. Bolts, nuts and filleted corner were removed. The simplified geometry model consists mainly of the motor, air deflectors, two crankcases, four cylinders and cylinder heads. Every two cylinders are arranged on each side of the motor of the compressor. The centre lines of the two cylinders on the same side are arranged in a V-shape. The motor and two crankcases have a fan underneath for cooling compressor, respectively.

![Figure 1. Compressor model A.](image)

The cooling duct structures consist of the external profile of the compressor. Each crankcase is enclosed by two air deflectors, which promote efficient use of air flow. Between them, the first deflector wraps around the crankcase cover from the front, and the second deflector wraps around the two sides of the crankcase. Moreover, the plain cooling fins are arranged around the cylinders and cylinder heads of the compressor, which form part of the cooling duct structures.

2.2. Compressor model B

The improved structure of model B for compressor is shown in Figure 2. The type of machine remains unchanged, and the parameters of the compressor are also consistent with those of the original model. Both sides of the motor are mounted with a new deflector for making better use of the air flow of the intermediate fan. In addition, the holes in the deflectors are designed to make room for the gas line. The comparison between model A and model B can prove the effect of improved cooling duct structures.
Compared with the original model, the structure of original deflectors has also improved. The first change is to lengthen the substructure of the front deflector, and two baffles are also added to the outside of the crankcase cover which didn't exist there before as shown in Figure 3. The baffles on the crankcase cover lead the wind from both sides to the middle, making the air flow from the bottom to the top more uniform. The lengthened part beneath the front deflector blocks most of the air flow that leaks out from the bottom, improving the utilization ratio of limited air flow.

2.3. Compressor model C

The improved structure of model C for compressor is also shown in Figure 4. Similarly, the type of machine remains unchanged, and the parameters of the compressor are also consistent with those of the original model. What is different from model A is that the shape of the fins at the cylinders and cylinder heads have changed. The convective heat transfer coefficient and flow loss of corrugated fin and plain fin are different [4]. Therefore, through numerical simulation, we can verify which type of fins works
better for this compressor. In this model, the fins are corrugated, while they are plain in the original model.

![Figure 4. Compressor model C.](image)

3. Numerical simulation study

3.1. Compressor model mesh generation and grid independence verification

There are big differences among the structure and size of each part of the compressor, and the shape of some regions is irregular. Therefore, in this paper, most structures adopted tetrahedral mesh and few regular structures used hexahedral mesh for meshing the model. Mesh refinement was carried out in some key places, such as the outer wall surface of the compressor. Considering the calculation conditions and precision, the appropriate grid size was selected to satisfy the good continuity of data transmission in the whole calculation process.

For the compressor model A, in order to verify the mesh independence, it was divided into 5848924, 6548714, and 8217990 grids, respectively. The grid models of model B and model C were also divided into three types. And the average temperature of the second cylinder and second cylinder head for three models were monitored. The results were shown in Table 1.

| Grids number | Second cylinder | Second cylinder head |
|--------------|----------------|---------------------|
| Model A      |                |                     |
| 5848924      | 358.32         | 366.82              |
| 6548714      | 357.34         | 365.65              |
| 8217990      | 357.95         | 365.50              |
| 5972504      | 351.59         | 359.69              |
| Model B      |                |                     |
| 6666213      | 350.55         | 358.39              |
| 8342044      | 350.83         | 358.22              |
| 5626367      | 352.33         | 360.41              |
| Model C      |                |                     |
| 6060868      | 351.46         | 359.54              |
| 8143534      | 351.02         | 359.14              |

The 6548714 meshes size of compressor model A was used for simulation, and the mesh model is shown in Figure 5. The 6666213 meshes size of compressor model B and the 6060868 meshes size of...
compressor model C were used for simulation respectively, while both of them had a similar mesh model to the original model.

![Image of compressor model C](image)

Figure 5. The original compressor model meshing.

In addition, the grid of fluid part around the compressor is also shown in Figure 6, which is the ambient air environment around the compressor.

![Image of fluid model meshing](image)

(a) The whole part.  (b) The sectional view.

Figure 6. The fluid model meshing.

3.2. Simulation methods

The turbulence model has a significant influence on numerical results. Therefore, it is of great importance to select an appropriate turbulence model for computation. There is a complex three-dimensional turbulent flow in the compressor. And compared with the standard $k-\varepsilon$ model, the main change of the RNG $k-\varepsilon$ model [5] is that the rotation and rotational flow in the mean flow is considered by modifying the turbulent viscosity. Thus, the RNG $k-\varepsilon$ model was adopted.

3.3. Boundary conditions

When calculating the flow field and heat transfer of compressor, the boundary conditions of the calculation area should be set as shown in Figure 7.

(1) Pressure inlet boundary condition and pressure outlet boundary condition were set to simulate the resistance loss of each part of the compressor.

(2) Positions of the fans were treated as internal planes and the fan air flow rate is $445 \text{ m}^3/\text{h}$.

(3) The Fan models were used to analyze heat transfer condition of compressor. Meanwhile, pressure inlet boundary and pressure outlet boundary were set for the fans.

(4) Cylinders, cylinder heads, motor and crankcases were constantly producing heat while compressor was working, therefore, a coupled thermal condition between compressor and air flow was also adopted. The whole simulation of compressor focuses on the overall heat transfer performance rather than the heat transfer inside the compressor casing. Therefore, the model was simplified reasonably. When the compressor works, the cooling part includes the motor, crankcase, cylinder and cylinder head. The heat...
dissipation from the compressor casing to the surrounding air [6] is related to the surface area of the casing, the ambient temperature, the air flow around the casing, the running condition of the compressor and the efficiency of the compressor, which can be shown below.

\[ Q = \alpha F (T_s - T_h) \] (1)

Where \( \alpha = 25 \text{ W/(m}^2\cdot\text{K)} \) [6] is the heat transfer coefficient of forced air on the outer surface of the casing, \( F \) is casing surface area, \( T_s \) is average casing temperature and \( T_h \) is ambient air temperature.

In addition, fans were treated as a performance curve using lumped parameter [7]. It was assumed as an infinite thin plane. The air pressure is going to increase when going through the fan. Besides, the discontinuous pressure rise is defined with the velocity of air. And the fan performance curve [8] is

\[ P = 360.875 + 11.013v - 1.84v^2 + 0.01993v^3 \]

in this paper.

![Figure 7. The schematic diagram of boundary conditions.](image)

4. Results and analysis

4.1. Effects of cooling duct structures on compressor

The influence of the duct structures on the compressor was investigated by comparing the temperature of different models. The improved duct structures can make better use of the limited air flow, thus achieving better cooling effect.

The whole temperature distribution of the original model is similar to that of the improved models as shown in Figure 8, Figure 9 and Figure 10. However, it can also be seen that the local temperature distribution in model B is more uniform due to the increased baffles and air deflectors. And the temperature values of the components in the improved models are lower than those in the original model.

To be precise, the maximum temperature of the model B can be 13.1°C lower than that of the model A, the maximum temperature of the model C can be 6.3°C lower than that of the model A. Therefore, the rationality of the improved models can be verified. Moreover, different thickness of air duct deflectors has been compared in the model B. When the thickness of air duct deflectors increased from 2 mm to 4 mm, the max temperature value changed no more than 0.4°C. In addition, the temperature distribution did not change with the variety of deflectors thickness. Therefore, the thickness of air duct deflectors has little effect on the compressor temperature.
In addition, the temperature distribution of the cylinders and cylinder heads also needs to be noted for compressor. The temperature nephograms of a cylinder and a cylinder head in the model A, B and C are shown in Figure 11, Figure 12 and Figure 13, respectively.

(a) Cylinder head. (b) Cylinder.

Figure 11. The parts of model A.
From above graphs, it can be seen that the temperature of the cylinders and cylinder heads is reduced in the model B and C. In addition, both the cylinders and the cylinder heads are cooler on the side away from the centre line of the cylinder in the model A and C, which is caused by the arrangement of cylinders and crankcases. However, owing to the two baffles added to the crankcase covers as shown in Figure 3, the temperature distribution in model B is more uniform.

4.2. The distribution of velocity
The velocity of cylinders and cylinder heads near the middle side in model B which has a better heat dissipation is greater than that in model A as shown in Figure 14. And the $X = 220\, \text{mm}$ plane in this picture is facing the two cylinders and two cylinder heads on the same side of the motor. In the improved model, the substructure of the front deflector which was different with the original structure was lengthened and two baffles were also added to the outside of the crankcase cover as mentioned above, making the temperature distribution around the cylinder heads and cylinders to be more uniform than the original model. This is because it makes the air flow between the cylinders and the cylinder heads more evenly distributed as shown in this picture, which is also consistent with the temperature results as shown in Figure 11 and Figure 12.
Figure 14. Velocity distribution of plane $X = 220$ mm.

Besides, the shape of compressor fins has a certain influence on the velocity distribution. And the effect can be shown from the velocity distribution in Figure 15(a) and Figure 15(b). The view in the picture is in the direction of the cross section of the cylinder.

Figure 15. Velocity distribution of cylinder cross section.

The outer surface of the cylinders is also a part of the cooling duct structure. Therefore, different fin shapes will form different cooling duct structures. It can be seen from these pictures in Figure 15 that the turbulence of airflow in model B is more severe compared with the model A. Moreover, combined with Figure 11 and Figure 13, it can be confirmed that the corrugated fins can bring better cooling effect than the plain fins.

4.3. Effects of fan air flow rate on compressor

In the original model, the temperature distributions under different fan air flow rates are compared under the condition that other parameters remain unchanged. In the actual situation, too small fan air flow rate [9] will reduce the cooling effect and working efficiency of the compressor unit. But excessive fan air flow rate can also cause a waste of energy.

Therefore, in this paper, the temperature under different air flow rates is compared for the model A in order to find the most economical fan air flow. Since the temperature distribution is similar, the temperature distribution diagram of the model under different conditions will not be listed in this paper. But the Table 2 shows maximum temperature value of the compressor in the model A under different fan air flow rates.
It can be seen from the above data that increasing the air flow of the fan under the same structural arrangement can obviously improve the heat dissipation performance of the compressor due to the increase of ventilation. In the case of small air flow, the increase of air flow has a greater impact on the maximum temperature drop of the compressor. However, with the further increase of air flow, the temperature drop tends to be gentle. At the same time, the power of fans increases. The size, cost, noise and its own calorific value increase correspondingly, and the heat dissipation efficiency decreases. Therefore, it is unreasonable to simply increase the fan power to reduce the surface temperature of the compressor at this time. It needs to start from other aspects for improving the heat dissipation model.

5. Conclusions
The numerical simulation model of the compressor was established, and the influence of different duct structures and air flow rates on the performance of fans was simulated and compared. The main conclusions were drawn as follows.
(1) The reasonable setting of baffles on the compressor casing could make the air flow distribution more uniform, so as to better heat dissipation without changing the fan conditions, which reduced the maximum temperature of the compressor by 13.1°C. Moreover, the thickness of air duct deflectors has little effect on the compressor temperature when it increased from 2 mm to 4 mm.
(2) The heat transfer efficiency of corrugated fins worked better than that of plains fins for this compressor and reduced the maximum temperature of the compressor by 6.3°C
(3) Fan air flow can effectively control the compressor temperature, but when the air flow reaches a certain value, the temperature will not decrease significantly with the increase of the air flow rate. Therefore, considering the heat dissipation effect and economy, the better choice of air flow rate in this compressor is 445 m³/h.

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