Heat Transfer Simulation for Reciprocating Compressor with Experimental Validation

Ruixin Zhou¹, *Bei Guo¹, Xiaole Chen¹, Jinliang Tuo¹, Rui Wu¹, Fabian Fagotti², Yali Zhao², Song Yang², Bo Xu²
¹School of Energy and Power Engineering, Xi’an Jiaotong University;
²Beijing Embraco Snowflake Compressor Company Ltd

*Corresponding Author Email: guobei@mail.xjtu.edu.cn

Abstract: The efficiency of reciprocating compressor can be influenced by heat transfer and the reliability can be also affected by the temperature distribution in compressor. In consideration of the complex relationship of heat transfer, the compressor is divided into six control volumes including the suction muffler, the cylinder, the discharge chamber, the discharge muffler, the discharge line and the compressor shell. The steady state energy balance equations of the open system for each control volume are built up after the crankshaft rotates one cycle. The heat flux of the cylinder is calculated by the existing correlation. The heat transfer coefficient correlations in energy equations are chosen in references and revised by experimental results. Three same type reciprocating compressors used in R290 system installed with thermocouples are tested under some planned conditions in order to ensure accuracy. The simulation results are compared with the experimental results. It shows that the simplified method presented in this paper is effective.

1. Introduction

Compressor is the heart of refrigeration system. The performance of the refrigeration system is greatly dependent on the performance of the compressor and the working conditions. As to the 1-D simulation of refrigeration compressor, Wu Shoufei¹,² simulated a certain refrigeration compressor based on the 4 basic equations of the 1-D flow. Ma Pengcheng¹ simulated the performance of a household refrigeration compressor by building the mathematic model of valve and considering the leakage in the compressor. Bei Guo⁴ studied the valve movement and simulated the thermal performance of reciprocating compressor with the refrigerant R600a experimental validation, but the heat transfer, which can influenced the simulation results, was not considered in that program.

A lot of researchers had been carried out on the heat transfer simulation of refrigeration compressor. Todescat and Fabian Fagotti⁵ studied the heat transfer simulation based on lumped-parameter models
and finite volume method to get the temperature distribution in compressor, and experiments with the refrigerant R134a was also carried out. The heat transfer coefficients of each control volume are determined by experiments. William A. Meyer, Kim, Tiow Ooi and Cesar have adopted classical heat transfer correlations in literatures based on lumped-parameter models and finite volume method. This method that heat transfer coefficients are from experiments is more accurate than the method that coefficients are determined by classical heat transfer correlations in literatures. In order to insure the practicability and precision, This method that heat transfer coefficients are from experiments is adopted. Moreover, R290 is widely used in commercial compressor, such as the drinks refrigerator and wind screen refrigerator which are used in supermarkets, but the refrigeration compressor with R290 was not studied and its experiment was not carried out. In order to optimize the R290 compressor and enlarge the practical scope of simulation, the experiment is carried out on R290 commercial compressors.

In this paper, the heat transfer model is established which calculates the heat flux between different control volumes in the compressor. In consideration of the complex relationship of heat transfer, the compressor is divided into 6 control volumes including the suction muffler, the cylinder, the discharge chamber, the discharge muffler, the discharge line and the compressor shell. The heat transfer coefficient of each control volume is measured by experiment and the heat flux of the cylinder is calculated by Lawton model. The experiments are also carried out and the thermal couple are installed at its exact position shown in following considering the insulation and leakage problems. The experiments results indicate that the heat transfer model of the refrigerator compressor can accurately calculate the thermodynamic performances and the comparable errors are reasonable.

2. Heat transfer simulation

Based on the simulation of performance parameters of reciprocating compressor, numerical simulation method to simulate heat transfer in reciprocating compressor is applied. In order to achieve the heat transfer subroutine, assumptions are shown below.

1) The process of heat transfer is a steady state.
2) The pressure of the suction side is the saturation pressure corresponding to the evaporating temperature, and the pressure of the exhaust side is the saturation pressure corresponding to the condensing temperature.
3) The internal temperature of a control volume is the average temperature of the inlet's and outlet's.
4) For the adjacent control volumes, outlet temperature of one control volume is equal to inlet's of the next control volume.
5) The compressor shell temperature, internal environment of shell and each control volume is average.

2.1 Energy balance equation of control volumes

As shown in Fig 1, I is the suction muffler, II is the cylinder III is the discharge chamber. IV is the discharge muffler. V is the discharge line. VI is the motor. T_{smw}, T_{dcw}, T_{dmw}, T_{cw}, T_{core} and T_{shell} are all the wall temperature of each compressor parts. m_0 is the compressor mass rate and m_1 is the compressor outlet mass rate and the m_1 is the mass rate of gas flow from inner environment to the suction muffler. According to the law of mass conservation, equation (1) and (2) are established.

\[ m_0 = m_{dis} \]  
\[ m_{suc} = m_0 + m_1 \]
The compressor was divided into 6 control volumes, which are the suction muffler, the cylinder, the discharge chamber, the discharge muffler, the discharge line and the compressor shell. The heat transfer simulation was carried out on these 6 control volumes. Each control volume equation was solved simultaneously, and then the wall temperatures and the internal environment temperature were worked out. What must be pointed out is that the equation set should be calculated when the crankshaft rotates 360° angle which is different from the calculation of \( \frac{dQ}{d\alpha} \). The \( \frac{dQ}{d\alpha} \) term need be calculated when the crankshaft rotate \( d\theta \) angle and integrate the heat transfer value. As the same, the compress power \( W_p \), the motor power \( W_c \) and all the mass rate are all integrate among 360° angle.

![Fig 1. The control volume sketch of semi direct intake reciprocating compressor](image1)

![Fig 2. The control volume sketch of indirect intake reciprocating compressor](image2)

The gas intake method of the suction muffler can be divided into 2 types in current refrigeration compressors. One is the semi direct intake method and it is shown in Fig 1. Another is indirect intake method, shown in Fig 2, whose inlet tube of compressor is not aimed to the suction muffler inlet, so the inlet gas would be warmed up by the motor and discharge parts. Both of the gas intake method can be expressed by an intake mixing coefficient \( \xi \) and \( m_1 \) is can calculate by equation (3).

\[
\dot{m}_1 = \dot{m}_{suc} \xi
\]  

(3)

The semi direct intake method of the suction muffler is shown in Fig 1. Its coefficient \( \xi \) is shown in equation (4).

\[
\xi = \frac{\dot{m}_{leak}}{\dot{m}_{suc}}
\]  

(4)

But as to the indirect intake method of the suction muffler, its coefficient \( \xi \) is equal to 1.

Above all the equations on mass rate, the energy balance equation of the suction muffler is shown in equation (5) and equation (6).

\[
\dot{m}_{suc} * h_1 = \dot{m}_{suc} * h_m + (1 - \xi) * \dot{m}_{suc} * h_0
\]  

(5)

\[
\dot{m}_{suc} * (h_2 - h_1) = \alpha_m F_m \left( \frac{T_i + T_s}{2} - T_c \right)
\]  

(6)

According to the research of Fabian[5], Lawton model was adopted in this thesis to calculate the heat transfer value between the gas in cylinder and the cylinder wall. The energy equation of the cylinder is shown in equation (7).
\[
m_{\text{dis}} \cdot h_1 - m_{\text{suc}} \cdot h_5 + m_{\text{leak}} \cdot \text{hie} = - \sum \frac{k(T_\text{t}(t) - T_\text{s})}{D} (0.28 \text{Re}(t)^{0.65} - 0.25 \text{L}(t) \frac{T_\text{n}}{T_\text{c}(t) - T_\text{s}}) \cdot S_c \cdot \frac{d\theta}{dt} + W_p \tag{7}
\]

As shown in Fig. 1, the discharge chamber, the discharge muffler and the discharge line are all treated as control volumes separately and their energy equations are shown in equation (8), equation (9) and equation (10).

\[
m_{\text{dis}} \cdot (h_3 - h_4) = \alpha_{\text{dis}} F_{\text{dis}} \frac{T_3 + T_4}{2} - T_p \tag{8}
\]

\[
m_{\text{dis}} \cdot (h_4 - h_5) = \alpha_{\text{dis}} F_{\text{dis}} \frac{T_4 + T_5}{2} - T_p \tag{9}
\]

\[
m_{\text{dis}} \cdot (h_5 - h_6) = \alpha_{\text{dis}} F_{\text{dis}} \frac{T_5 + T_6}{2} - T_p \tag{10}
\]

The shell temperature is easy to measure and it is also an important value to judge the characteristic of condition case. So the shell temperature is listed in the energy equation of compressor shell. The heat transfer capacity of the inner shell surface \(Q_\text{i}\) and outer shell surface \(Q_\text{o}\) are equal. The energy equation between the internal environment and inner shell surface is shown in equation (11) and the energy equation between the internal environment and outer shell surface is shown in equation (12).

\[
m_{\text{dis}} \cdot h_6 - m_0 \cdot h_6 = \alpha_{\text{i}} F_{\text{i}} (T_\text{i} - T_\text{shell}) + W_p \tag{11}
\]

\[
\alpha_{\text{i}} F_{\text{i}} (T_\text{i} - T_\text{shell}) = \alpha_{\text{o}} F_{\text{o}} (T_\text{o} - T_\text{shell}) \tag{12}
\]

The simultaneous equation set composed by equation (1) to (12) is calculated to get the variables with red color. The \(\alpha F\) term is the product value of heat transfer coefficient and heat transfer area. The mass rate term \(m_0, m_{\text{dis}}, m_{\text{suc}}\), the discharge gas temperature and the compress power \(W_p\) are all calculated by the former simulation program\(^4\) which used to calculate the compressor performance parameters. Motor efficiency curve is tested on the R290 commercial reciprocating compressor, and it is used to calculate the motor power \(W_c\). h terms in the equation set shows that the equation set is nonlinear and the calculation method is to call NIST REFPROP function library and use the Newton’s iterative method to resolve.

2.2 Heat transfer coefficients correlations

Ooi\(^7\) and Joel\(^8\) choose the empirical correlation in references and use them directly without revising and this must cause big errors in simulation. In this thesis, the heat transfer coefficient of each control volume is obtained by combining the theoretical heat transfer correlation with the experimental data. The coefficient correlations of simple control volumes are chosen in references and revised by experimental results. The 3-D flow field simulation of the discharge chamber and discharge muffler, where the gas flow is complicated, is carried out. The corresponding heat transfer correlation type is selected according to the 3-D simulation results. Because the structure of the suction muffler is extremely complex, its heat transfer coefficients correlation is fitted by experimental results. Table 1 shows the revised heat transfer coefficient correlations used in the heat transfer simulation.

| Control volume | Surface | Correlation |
|----------------|---------|-------------|
| Table 1. The revised heat transfer coefficient correlations | | |
Compressor shell

Inner \[ \overline{Nu} = 0.019Re^{0.8}Pr^{0.3} \quad (Pr \geq 0.5) \]

Outer \[ \overline{Nu} = \left( 2 + \left( 0.8 \times Re^2 + 0.23 \times Re^3 \right)Pr^{0.3} \right)^{1/3} \left( \frac{\mu}{\mu} \right) \quad (3.5 < Re < 7.6 \times 10^4, 0.71 < Pr < 380) \]

Inner \[ \overline{Nu} = 0.023Re^{0.8}Pr^{0.3} \]

Discharge line

Outer \[ \overline{Nu} = 0.18 + \frac{0.372Re^{0.8}Pr^{0.3}}{1 + (0.4/Pr)^{0.8}} \left[ 1 + \left( \frac{Re}{282000} \right)^{0.8/5} \right] \quad Re \times Pr > 0.2 \]

Inner \[ \overline{Nu} = 0.023Re^{0.8}Pr^{0.3} \]

Discharge muffler

Outer \[ \overline{Nu} = 0.62Re^{0.8}Pr^{0.3} \left[ 1 + (0.4/Pr)^{0.8} \right] \quad Re \times Pr > 0.2 \]

Inner \[ \overline{Nu} = \left( f / 8 \right) \left( \frac{Re_{e} - 1000}{1 + 12.7 \left( f / 8 \right)^{0.8} \left( Pr^{0.3} - 1 \right)} \right) \quad (Pr \geq 0.5, 3 \times 10^4 \leq Re \leq 5 \times 10^5) \]

Discharge chamber

Inner \[ \overline{Nu} = \left( f / 8 \right) \left( \frac{Re_{e} - 1000}{1 + 12.7 \left( f / 8 \right)^{0.8} \left( Pr^{0.3} - 1 \right)} \right) \quad (Pr \geq 0.5, 3 \times 10^4 \leq Re \leq 5 \times 10^5) \]

Suction muffler

Total \[ \alpha_m F_m = -1.19516 - 5492.97 \ast \dot{m} \]

Cylinder

Inner \[ \overline{Nu} = \frac{q_D}{k(T_e - T_s)} = 0.28 \text{Re}^{0.65} + 0.25L \frac{T_e - T_s}{T_e} \]

3. Experiments

3.1 Arrangements

Fig 3. is the experiment system sketch. According to the requirement of the standard testing condition, the 3m/s wind blow from a fan is adopted.

Fig 3. The experiment system sketch
In order to reach the unusual condensing temperature, the radiation heating installation and water-cooled tube are installed on condenser side. Also in order to reach limiting evaporating temperature, the water-cooled device and small refrigeration device with an extra refrigeration cycle are installed on the evaporator side. The automatic regulating valve is adopted as the throttling valve. Firstly, connect all the solder joints of thermocouples installed in compressor to the temperature acquisition equipment. Then, connect the temperature acquisition equipment to computer. Lastly, open the software to test the thermocouples’ accuracy. Fig 4 shows the compressor installed all thermocouples and shows the solder joints on the shell.

![Image of a compressor with thermocouples installed](image)

**Fig 4.** The compressor sample installed thermocouples

The detailed information of the thermocouples is shown in Table 2. The data acquisition frequency is once per 10 seconds. The temperature of each test point under some condition is the average value of the experimental data which acquired after the system is stable.

| No. | Name | Short name | Position |
|-----|------|------------|----------|
| 1   | T<sub>up</sub> | Temperature of the upper shell | Middle of the upper shell |
| 2   | T<sub>bottom</sub> | Temperature of the bottom shell | Middle of the bottom shell |
| 3   | T<sub>left</sub> | Temperature of the left shell | When facing to socket, middle of the left shell |
| 4   | T<sub>right</sub> | Temperature of the right shell | When facing to socket, middle of the left shell |
| 5   | T<sub>front</sub> | Temperature of the front shell | When facing to socket, middle of the front shell |
| 6   | T<sub>behind</sub> | Temperature of the behind shell | When facing to socket, middle of the behind shell |
| 7   | T<sub>ei1</sub> | The environment temperature NO.1 in compressor | When facing to socket, The right side in compressor and between the shell and motor |
| 8   | T<sub>ei2</sub> | The environment temperature NO.2 in compressor | When facing to socket, The left side in compressor and between the shell and motor |
| 9   | T<sub>0</sub> | Gas temperature at suction pipe | In the middle of the suction pipe |
| 10  | T<sub>1</sub> | Gas temperature at suction muffler inlet | In the middle of the suction muffler inlet |
| 11  | T<sub>2</sub> | Gas temperature at suction muffler outlet | In the middle of the suction muffler outlet |
| 12  | T<sub>3</sub> | Discharge gas temperature of cylinder | In the discharge chamber and facing to the discharge valve |
| 13  | T<sub>4</sub> | Gas temperature at discharge muffler inlet | In the discharge muffler inlet |
| 14  | T<sub>5</sub> | Gas temperature at discharge muffler outlet | In the discharge muffler outlet |
| 15  | T<sub>6</sub> | Gas temperature at the outlet pipe | In the middle of the discharge pipe |
3.2 Experiment results
R290 is used in this experiment. Because the R290 commercial refrigerator is high efficiency when the refrigeration case is in HBP (high back pressure) conditions, the ASHRAREHBP46 case is selected to analyze the heat transfer value of each compressor part.

The detail of the condition: the evaporating temperature = 7.2°C; The condensing temperature = 54.4°C; The suction temperature = 35°C; The ambient temperature = 35°C; The subcooled temperature = 35°C.

Fig 5  Heat transfer value ratio of compressor parts
The heat transfer valve ratio is the ratio between the heat transfer value of each compressor part and the motor power $W_e$. The heat transfer value of discharge line is 3-5 times as large as the value of discharge chamber or discharge muffler. The heat transfer value ratio of motor is about 23%. The inner environment of compressor is heat up by the discharge parts and the motor, but heat transfer of the shell can cool down it. The heat transfer valve ratio of the shell is larger than 50% when the motor power is lower than 220W, because the mass rate is smaller when the refrigeration system is in LBP (low back pressure) cases and the pressure ratio is higher than those in HBP.
Fig 6. The tested temperatures vary with the evaporating temperature

Fig 7. The tested temperatures vary with the condensing temperature

Fig 6 shows the tested temperatures vary with evaporating temperatures when the condensing temperature is 35°C, the suction temperature is 32.2°C and the ambient temperature is 32.2°C. Fig 7 shows the tested temperatures vary with condensing temperatures when the evaporating temperature is 5°C, the suction temperature is 32.2°C and the ambient temperature is 32.2°C. The temperature T_{w} and T_{shell} are all the average value of their testing points.

Fig 8. The tested temperatures vary with the suction gas temperature

Fig 9. The tested temperatures vary with the ambient temperature

Fig 8 shows the tested temperatures vary with suction temperatures when the evaporating temperature is 5°C, the condensing temperature is 54.4°C and the ambient temperature is 32.2°C. Fig 9 shows the tested temperatures vary with ambient temperatures when the evaporating temperature is 5°C, the condensing temperature is 54.4°C and the ambient temperature is 20°C.

4. Comparison

As shown in Table 3, three refrigeration cases are chosen to compare the simulation results and the experimental results. Case No 1. (ASHRARELPB46) can express the LBP conditions. Case No 2. (ASHRAREHBP46) and No 3. can express the HBP conditions.
Experiment value

**Table 3.** Refrigeration cases chosen to compare the simulation and experiment

| Case No. | Evap Temp(℃) | Cond Temp(℃) | Ambi Temp(℃) | Suct Temp(℃) |
|----------|--------------|--------------|--------------|--------------|
| 1        | -23.3        | 54.4         | 32.2         | 32.2         |
| 2        | 7.2          | 54.4         | 35           | 35           |
| 3        | 5            | 5            | 32.2         | 32.2         |

Table 4. shows the results comparison between simulation and experiment, and the error is the calculate using equation (13).

\[
\text{Relative error} = \frac{\text{Experiment value} - \text{Simulation value}}{\text{Experiment value}} \times 100\% 
\]

(13)

**Table 4.** Results comparison between simulation and experiment

| Parameters       | Case No.1 | Case No.2 | Case No.3 |
|------------------|-----------|-----------|-----------|
|                  | Simu | Test | Error (%) | Simu | Test | Error (%) | Simu | Test | Error (%) |
| T0(℃)            | 32.2 | 32.7 | -1.53     | 35.0 | 35.1 | -0.28     | 32.2 | 32.6 | -1.23     |
| T1(℃)            | 73.8 | 90.0 | -18.00    | 60.1 | 65.5 | -8.24     | 62.3 | 68.5 | -9.05     |
| T2(℃)            | 78.8 | 91.6 | -13.97    | 64.3 | 66.6 | -3.45     | 62.3 | 69.5 | -10.36    |
| T3(℃)            | 163.2 | 156.3 | 4.41 | 116.8 | 111.2 | 5.04 | 124.7 | 119.5 | 4.35 |
| T4(℃)            | 157.9 | 148.6 | 6.26 | 111.9 | 108.5 | 3.13 | 119.9 | 116.6 | 2.83 |
| T5(℃)            | 152.1 | 144.4 | 5.33 | 109.4 | 106.4 | 2.82 | 116.1 | 114.4 | 1.49 |
| T6(℃)            | 122.8 | 115.7 | 6.14 | 98.7 | 96.7 | 2.07 | 104.5 | 103.0 | 1.46 |
| Tcw(℃)           | 100.7 | 111.2 | -9.44 | 70.85 | 75.7 | -6.41 | 75.3 | 80.4 | -6.34 |
| T~1~(℃)          | 73.8 | 78.8 | -6.35 | 60.1 | 60.6 | -0.83 | 62.3 | 62.6 | -0.48 |
| Tshell(℃)        | 55.3 | 60.6 | -8.75 | 52.2 | 53.2 | -1.88 | 51.8 | 52.9 | -2.08 |
| Tdcw(℃)          | 131.1 | 126.4 | 3.72 | 89.2 | 88.1 | 1.25 | 96.2 | 94.4 | 1.91 |
| Tdmw(℃)          | 114.5 | 108.8 | 5.24 | 76.3 | 74.4 | 2.55 | 81.0 | 78.9 | 2.66 |
| Wp (W)           | 176.66 | — | — | 317.69 | — | — | 328.20 | — | — |
| Motor efficiency | 75.5% | — | — | 74.9% | — | — | 74.6% | — | — |
| We (W)           | 248.84 | 247.36 | 0.60 | 424.28 | 388.69 | 9.16 | 439.96 | 408.36 | 7.74 |
| m_suc(kg/s)      | 1.11* | — | — | 3.54* | — | — | 3.13* | — | — |
| m_dis (kg/s)     | -9.844 | -9.688 | 1.61 | -3.316 | -3.365 | -1.45 | -2.872 | -2.921 | -1.68 |
| Capacity(W)      | 348.93 | 344.41 | 1.31 | 1022.37 | 1007.5 | 1.48 | 974.27 | 889.52 | -1.71 |
| COP              | 1.402 | 1.392 | 0.71 | 2.410 | 2.592 | -7.02 | 1.987 | 2.178 | -8.77 |

Results shows that the temperature error between simulation and experiment is below ± 20%. The compressor performance parameters error (power, capacity and mass rate) between simulation and experiment is below ± 10%.

**5. Conclusion**

The experiments of R290 refrigeration compressor are carried out. The heat transfer value of discharge line is 3-5 times as large as the value of discharge chamber or discharge muffler. The heat transfer value ratio of motor is about 23%. There are linear relationships between the temperatures in compressor and
the conditions temperature. The heat transfer simulation of reciprocating compressor is studied and the heat transfer coefficients are chosen from reference and revised by experiment data. The relative error of temperatures between simulation and experiment is below ±20% and the relative error of compressor performance parameters is below ±10%. Therefore, the numerical simulation of reciprocating compressor can simulate its working process well.

Reference:
[1] Wu S, Wang Z, Xiong X, et al. Research on System Simulation of Refrigerator Compressor[J]. China Appliance, 455–458, 2012.
[2] Wu Shoufei, Wang Zonghuai, Wang Zhen. Fluid Structure Interaction Based Refrigerator Compressor Exhaust System Simulation [J]. China Appliance, 307–312, 2011.
[3] Ma Pengcheng. The working process simulation and the performance analysis of the R600a refrigeration compressor [D]. Xi an, Xi’an Jiao Tong university, 2011.
[4] Zhou Ruixin, Guo Bei, Lu yang, Chang Yunfeng, Zhao, Yali, Wang Wei, Fabian Fagotti. Numerical simulation for household refrigeration compressor with experimental validation. The 7th International Conference on Compressor and Refrigeration(ICCR2015), 2015.
[5] Todescat, M.L., Fagotti, F., Prata, A.T., Ferreira, R.T.. Thermal energy analysis in reciprocating hermetic compressors. Proceedings International Compressor Engineering Conference at Purdue. West Lafayette, USA, pp. 1419–1428, 1992
[6] Meyer, W.A., Thompson, H.D,. An analytical model of heat transfer to the suction gas in a low-side hermetic refrigeration compressor. International Compressor Engineering Conference at Purdue. West Lafayette, USA, pp. 898–907, 1990.
[7] Ooi, K.T.. Heat transfer study of a hermetic refrigeration compressor. Appl. Therm. Eng. 23, 1931–1945, 2003.
[8] Joel SANVEZZO Jr., Cesar J. DESCHAMPS. A heat transfer model combining differential and integral formulations for thermal analysis of reciprocating compressors. International Compressor Engineering Conference at Purdue. West Lafayette, USA, 2012.
[9] B Lawton, Bsc(Eng), PhD, CEng, IMechE. Effect of compression and expansion on instantaneous heat transfer in reciprocating internal combustion engines. Pro Instn Mech ENgrs Vol 201 No A3(IMechE 1987), 175-186, 1987.
[10] F. Fogotti, M.L. Todescat, R.T.D.S. Ferreira, A.T. Prata, Heat transfer modelling in a reciprocating compressor, in: Proceedings of the 1994 Purdue University Compressor Engineering Conference, pp. 605–610, 1994.