A Numerical Study on the Temperature Field of a R290 Hermetic Reciprocating Compressor with Experimental Validation

Rui Wu
Xi'an Jiaotong University, Xi'an, China, wurui0725@stu.xjtu.edu.cn

Bei Guo
Xi'an Jiaotong University, Xi'an, China, guobei@mail.xjtu.edu.cn

Jinliang Tuo
Xi'an Jiaotong University, Xi'an, China, tuojinliang@stu.xjtu.edu.cn

Yu Zhang
Xi'an Jiaotong University, Xi'an, China, zy466496675@stu.xjtu.edu.cn

Xiaole Chen
School of Energy and Power Engineering, Xi'an Jiaotong University, China, chenxiaole@stu.xjtu.edu.cn

See next page for additional authors

Authors
Rui Wu, Bei Guo, Jinliang Tuo, Yu Zhang, Xiaole Chen, and Ruixin Zhou

A Numerical Study on the Temperature Field of a R290 Hermetic Reciprocating Compressor with Experimental Validation

Bei Guo[1]*, Rui Wu[1], Jinliang Tuo[1], Yu Zhang[1], Xiaole Chen[1], Ruixin Zhou[1]

[1]Xi'an Jiaotong University,
Xi'an, Shaanxi, China
Contact Information (18591783122, guobei@mail.xjtu.edu.cn)

* Corresponding Author

ABSTRACT

A numerical model to predict the temperature field in a R290 (propane) hermetic reciprocating compressor is presented in this work. The control volume method and the lumped parameter method are used in the simulation. The compressor is divided into 6 control volumes, including the suction muffler, the cylinder, the discharge chamber, the discharge muffler, the discharge pipe and the shell. The system of non-linear equations is formed of the energy balance equations of every control column. The temperature field is derived by solving the equations. To validate the numerical model accurately, temperature experiment has been carried out in 3 same-type hermetic reciprocating compressors using R290 as working fluid. The simulation result shows a good agreement compared with the experiment.

1. INTRODUCTION

The R290 as the natural substance, is more environmental friendly compared with other refrigerants. The ODP (Ozone Depletion Potential) is 0, and the GWP (Global Warming Potential) is rather low, only 20. Up to now, many researches in the R290 have been carried out, but most of them focus on the rotary compressors applied in the air conditioners. The researches of the R290 reciprocating compressor used in the refrigerator and the freezer are very few.

The design of the small-type hermetic reciprocating compressors has two indicators: high efficiency and high reliability. The reciprocating compressors have the following major losses: the motor loss, the mechanical loss, the friction loss, the leakage loss and the heat transfer loss. The efficiency and reliability of a hermetic reciprocating refrigerator compressor are affected by the internal temperature. So In recent years, the heat transfer loss has attracted more and more attention. In the study of reducing the heat transfer loss of hermetic refrigeration compressors, more and more scholars have proposed their own research methods.

At present, some scholars use the CFD to simulate the temperature field of the compressors. The advantage of this method is that the calculation is more accurate and the result is more intuitive. But the model and the calculation are too complex and time-consuming. A 3-dimensional simulation of the hermetic reciprocating compressor is still limited by the computer capacity. And it can only be calculated at a certain working condition. It is difficult to generalize to other working conditions of the same model. Besides, if the model is changed, which means all work needs to be repeated.

Other scholars use numerical computation method. Based on the control volume method and the lumped parameter method, the temperature field is derived by solving the system of energy balance equations. The key to the method is to calculate the coefficient of heat transfer. Because the structure of the compressor is relatively complex and the lubricating oil flows irregularly in the compressor, it is not accurate to obtain the temperature field by the numerical simulation. However, the advantage of this method is that it can be applied to different structure sizes, refrigerants, lubricant oils and operating conditions, so it is easy and helpful for designers to design and optimize the compressors at the start.

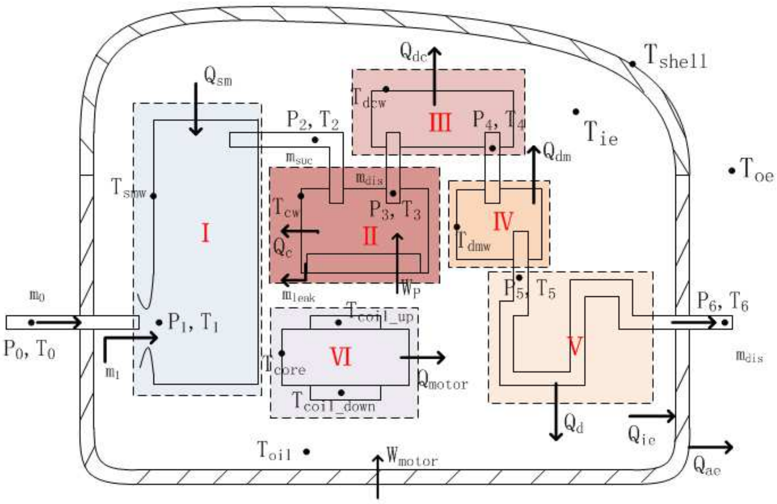
In this paper, the heat transfer model is established which calculates the heat flux between different control volumes, the suction muffler, the cylinder, the discharge chamber, the discharge muffler, the discharge pipe and the shell. The experiments are also carried out. And the experiments results indicate the model is valid.

2. SIMULATION

2.1 Simulation Model

In this simulation, the compressor is divided into 6 control volumes, including the suction muffler, the cylinder, the discharge chamber, the discharge muffler, the discharge pipe and the shell. The compressor uses a semi-direct suction muffle. A schematic view of the compressor is shown in Figure 1.

A steady state condition is assumed for the heat transfer simulation. The suction pressure is the saturation pressure corresponding to the evaporating temperature, and the discharge pressure is the saturation pressure corresponding to the condensing temperature. The internal temperature of a control volume is the average temperature of the inlet and outlet. For the adjacent control volumes, one control volume's outlet temperature is equal to the next control volume's inlet temperature.



Ⅰ-the suction muffler; Ⅱ-the cylinder; Ⅲ-the discharge chamber;
Ⅳ-the discharge muffler; Ⅴ-the discharge pipe and; Ⅵ-the shell
Figure 1: The control volume sketch

2.2 Equations

2.2.1 Mass balance equation:

$$\dot{m}_0 = \dot{m}_{dis} \tag{1}$$

$$\dot{m}_1 = \dot{m}_{leak} \tag{2}$$

$$\dot{m}_{suc} = \dot{m}_0 + \dot{m}_1 \tag{3}$$

2.2.2 The energy balance equation of the suction muffler:

$$\dot{m}_{suc} h_2 - \dot{m}_0 h_0 - \dot{m}_1 h_{ie} = \alpha_{sm} F_{sm} \left(T_{ie} - \frac{T_1 + T_2}{2} \right) \tag{4}$$

2.2.3 The energy balance equation of the cylinder: $-\sum Q_c$ is the sum of heat transfer quantity between the gas of cylinder and the wall of cylinder in a cycle:

$$\dot{m}_{dis} h_3 - \dot{m}_{suc} h_2 + \dot{m}_{leak} h_{ie} = -\sum W_c + W_p \tag{5}$$

2.2.4 The energy balance equation of the discharge chamber:

$$\dot{m}_{dis} (h_3 - h_4) = \alpha_{dc} F_{dc} \left(\frac{T_3 + T_4}{2} - T_{ie} \right) \tag{6}$$

2.2.5 The energy balance equation of the discharge muffler:

$$\dot{m}_{dis}(h_4 - h_5) = \alpha_{dm} F_{dm} (\frac{T_4 + T_5}{2} - T_{ie}) \tag{7}$$

2.2.6 The energy balance equation of the discharge pipe:

$$\dot{m}_{dis}(h_5 - h_6) = \alpha_{dp} F_{dp} (\frac{T_6 + T_5}{2} - T_{ie}) \tag{8}$$

2.2.7 Heat transfer between the inner environment and the compressor shell:

$$\dot{m}_{dis} h_6 - \dot{m}_0 h_0 = \alpha_{si} F_{si} (T_{ie} - T_{shell}) + W_e \tag{9}$$

2.2.8 Heat transfer between the outer environment and the compressor shell:

$$\alpha_{si} F_{si} (T_{ie} - T_{shell}) = \alpha_{so} F_{so} (T_{shell} - T_{oe}) \tag{10}$$

2.3 Solution Method

The above equations form a system of non-linear equations. The energy balance equation of cylinder is calculated by iterative method to get the value of the cylinder temperature (T_{cw}), thereby the values of unknown temperatures (T_{ie}, T_{shell}, T_6, T_5, T_4, T_2 and T_1) are derived. Figure 2 is the flow chart of the solution algorithm.

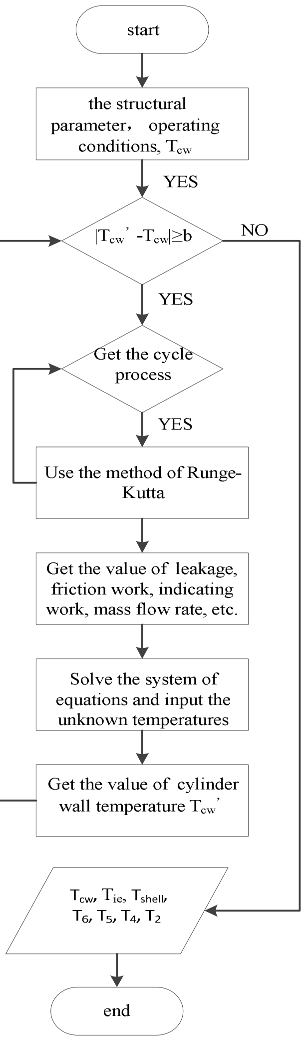


Figure 2: The flow chart of the solution algorithm

3. Experiment Setup

Temperature measurement has been carried out in 3 same-type hermetic reciprocating compressors using R290 as working fluid to validate the simulation model. The R290 hermetic reciprocating compressor used in the experiment has a displacement of 7.55 and the COP (ASHRAEHBP46 test conditions 7.2°C/54.4°C) is approximately 2.74. Twenty-eight thermocouples type T are used, 27 of which are distributed inside the compressor, on the shell and in the gas line; and one of which is in the environment. Moreover, two pressure sensors are placed in the suction and discharge muffler. Figure 1 shows some main temperature points and Table 1 gives the thermocouples measuring points in detail. Figure 3 is The distribution of the thermocouples inside the shell.
Figure 4 shows the schematic overview of the experiment system. This system uses automatic regulating valve. In this system, the evaporating pressure, the condensing pressure, the suction temperature and the environment temperature can be changed.
The experiment connects No.1-No.8 thermocouples to No.1 Agilent 34970A data acquisition instrument and No.9-No.27 thermocouple to No.2 Agilent 34970A data acquisition instrument. No.1 data acquisition instrument and No. 2 data acquisition instrument are connected to different computers, and the configuration of two data acquisition instruments and two data-acquisition boards are same. The GP-IB data transmission line is selected to transmit data from the data acquisition instrument to the computer.
The set-up of the test was based on ASHRAEHBP46 standard operating conditions. The 3m/s wind blow from a fan is adopted. The compressor operates at constant speed of 2,950 rpm. The test conditions of the compressor are shown in Table 2. The overcooling temperature of the No.20 operating condition is 46.1℃, and others are 32.2℃.

Figure 3: The distribution of the thermocouples inside the shell

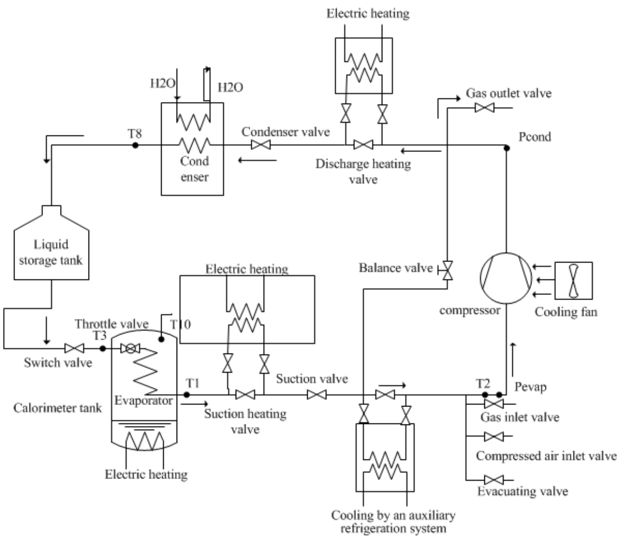


Figure 4: The experiment system sketch

Table 1: Thermocouples measuring points

No.	Name	Implication	Position
1	T_{up}	The temperature of the upside shell	The middle of the upside shell
2	T_{bottom}	The temperature of the downside shell	The middle of the downside shell
3	T_{left}	The temperature of the left shell	In the face of the socket, the middle of the left shell
4	T_{right}	The temperature of the right shell	In the face of the socket, the middle of the right shell
5	T_{front}	The temperature of the front shell	The middle of the shell in the side of the socket
6	T_{back}	The temperature of the back shell	The middle of the shell in the face of the socket
7	T_0	The suction temperature of the compressor	The middle of the suction pipe
8	T_6	The discharge temperature of the compressor	The middle of the discharge pipe
9	T_{ie1}	The right temperature of the shell's inside	In the face of the socket, the right environment between the shell and the core
10	T_{ie2}	The left temperature of the shell's inside	In the face of the socket, the right environment between the shell and the core
11	T_{ie3}	The upside temperature the shell's inside	In the face of the socket, the upside environment between the shell and the cylinder
12	T_{ie4}	The downside temperature of the shell's inside	In the face of the socket, the back environment near the suction pipe
13	T_1	The suction temperature of the suction muffler	In the middle of the inlet of the suction muffler
14	T_2	The discharge temperature of the suction muffler	In the middle of the outlet of the suction muffler, near the suction valve
15	T_3	The discharge temperature of the cylinder	In the discharge chamber, near the discharge vent
16	T_4	The suction temperature of the discharge muffler	In the first cavity of the discharge muffler
17	T_5	The discharge temperature of the discharge muffler	In the second cavity of the discharge muffler
18	T_{oil}	The temperature of the oil sump	In the middle of the oil sump
19	T_{c_d}	The temperature of the downside coil	In the downside of the coil
20	T_{c_u}	The temperature of the upside coil	In the upside of the coil
21	T_{core}	The temperature of the core	In the face of the socket, the right well of the core
22	T_{shaft}	The temperature of the shaft	In the upside of the shaft
23	T_{cw}	The temperature of the cylinder well	In the upside of the cylinder well
24	T_{cover}	The temperature of the cylinder cover	The middle of the outer well of the cylinder cover
25	T_{case}	The temperature of the case	The upside of the case neat the suction pipe
26	T_{smw}	The temperature of the suction muffler well	In the middle of the outer well of the suction muffler
27	T_{dmw}	The temperature of the discharge muffler well	In the middle of the outer well of the discharge muffler
28	T_{oe}	The environment temperature	In the environment

Table 2: Test conditions

No.	Evaporating Temp. /℃	Condensing Temp. /℃	Environment Temp. /℃	Back Temp. /℃
1	-30	35	32.2	32.2
2	-20	35	32.2	32.2
3	-10	35	32.2	32.2
4	0	35	32.2	32.2
5	5	35	32.2	32.2
6	10	35	32.2	32.2
7	-30	45	32.2	32.2
8	-20	45	32.2	32.2
9	-10	45	32.2	32.2
10	0	45	32.2	32.2
11	5	45	32.2	32.2
12	10	45	32.2	32.2
13	-30	55	32.2	32.2
14	-20	55	32.2	32.2
15	-10	55	32.2	32.2
16	0	55	32.2	32.2
17	5	55	32.2	32.2
18	10	55	32.2	32.2
19	-23.3	54.4	32.2	32.2
20	7.2	54.4	35	35
21	5	54.5	27	20
22	5	54.5	37	20
23	5	54.5	32	25
24	5	54.5	32	20
25	5	54.5	32	35
26	5	60	32.2	32.2

4. Results

The experiment tests 6 temperatures of the shell, four temperatures of the shell environment and 2 temperatures of the environment. When processing data, the average temperature is taken. Figure 5 shows the temperature field in ASHRAREHBP46 condition (the evaporating temperature is 7.2℃, the condensing temperature is 54.4℃, the sub cooling temperature is 46.1℃, the suction temperature is 35℃, the ambient temperature is 35℃). The suction temperature T_0 is 35.1℃.The inlet temperature of the suction muffler T_1 is 65.5. The motor temperature is 70.2℃. The cylinder exhaust temperature is 111.2℃.The compressor discharge temperature is 96.7℃，14.5℃ lower than the cylinder discharge temperature. It means the motor and the discharge system heat the suction gas.
The mass flow rate is tested by the calorimeter and the motor power is tested by the power-tongs. The heat

dissipation rate of the control volume is calculated by the Equation (11).

$$\dot{Q} = \dot{m}\Delta h \tag{11}$$

The heat dissipation rate of the motor is derived by the electric effiency. So the ratio of the heat dissipation rate to the motor power for the shell, the dischage chamber, the dischage muffler, the dischage pipe and the motor is derived. The relationship between these ratios and the motor power is shown in the Figure 6. In the discharge system, the heat loss of the discharge pipe is maxium, which is 3-5 times as large as that of the discharge chamber and the discharge muffler. The heat dissipation rate of the motor accounts for about 23%-27% of the motor power. It is approximately equal to the heat loss of the discharge pipe.

Table 3 shows the comparison between the simulation results and the experiment results in No.20, No.26 condition. All of the relative errors between the simulation results and the experiment results is less than 11%. The relative error of No.26 test condition (5℃/60℃) is biggest, up to 11%. Comparing with the experiment results, the relative errors are comparatively small.

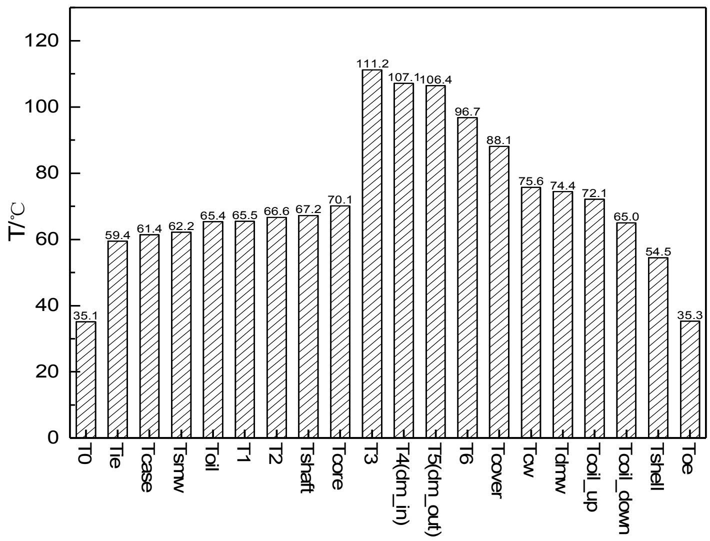


Figure 5: The temperature field in ASHRAEHBP46 condition

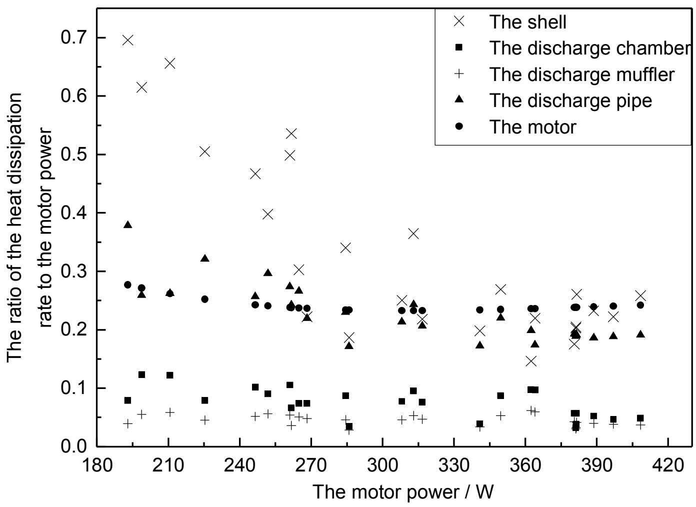


Figure 6: The relationship between the ratio and the motor power

Table 3: The comparasion between simulation results and experiment results

Point /℃	ASHRAEHBP46 No.20（7.2/54.4）			High Back Pressure Test Condition No.26（5/60）		
	Simulation	Experiment	Error /%	Simulation	Experiment	Error /%
T_0	35.0	35.1	-0.28	32.2	32.6	-1.23
T_1	60.1	65.5	-8.24	62.3	68.5	-9.05
T_2	64.3	66.6	-3.45	62.3	69.5	-10.36
T_3	116.8	111.2	5.04	124.7	119.5	4.35
T_4	111.9	108.5	3.13	119.9	116.6	2.83
T_5	109.4	106.4	2.82	116.1	114.4	1.49
T_6	98.7	96.7	2.07	104.5	103.0	1.46
T_{cw}	70.85	75.7	-6.41	75.3	80.4	-6.34
T_{ie}	60.1	60.6	-0.83	62.3	62.6	-0.48
T_{shell}	52.2	53.2	-1.88	51.8	52.9	-2.08
T_{dcw}	89.2	88.1	1.25	96.2	94.4	1.91
T_{dmw}	76.3	74.4	2.55	81.0	78.9	2.66

6. CONCLUSIONS

In this study a thermal model for the investigation of a small-type hermetic reciprocating compressor is shown using mainly numerical methods. The methods combine the control volume method and the lumped parameter method. To validate the thermal model of the hermetic compressor, temperature measurements on several positions in the compressor are carried out. The main conclusions are summarized as follows:

- In ASHRAREHBP46 test condition, the suction temperature is 35.1℃; the inlet temperature of the suction muffler is 65.5; the motor temperature is 70.2℃, the cylinder exhaust temperature is 111.2℃, the compressor discharge temperature is 96.7℃，14.5℃ lower than the cylinder discharge temperature. Both of the motor and the discharge system heat the suction gas.
- In ARSHRAEHBP46 test condition, the heat dissipation rate of the motor accounts for about 23%-27% of the motor power and the heat dissipation rate from the shell to the environment accounts for about 20%-25% of the motor power. The heat dissipation rate of the discharge pipe is more than that of the motor power, 3-5 times as high as the discharge chamber and the discharge muffler. To reduce intake heating, the discharge system is as important as the motor.
- According to the simulation work, the temperature field is derived. Comparing the simulation results with the experiment results, the relative error is less than 11%. In ASHRAEHBP46, the relative error is less than 10%. The results shows that the numerical model has a good agreement with the measurements. The model is valid, which can be used in the start of the design of the same type of the compressor.

NOMENCLATURE

P	pressure	(Pa)
T	temperature	(K)
$\dot{m}$	mass flow rate	(Kg/s)
F	area	(m^2)
$\dot{Q}$	heat flux	(W)
α	heat transfer coefficient	($W/m^2 \cdot K$)
W	power	(W)
h	enthalpy	(J/Kg)

Subscript

0	the inlet of the compressor
1	the inlet of the suction muffler
2	the outlet of the suction muffler
3	the outlet of the cylinder
4	the inlet of the discharge muffler
5	the outlet of the discharge muffler
6	the outlet of the compressor
up	the upside shell
bottom	the downside shell
left	the left shell
right	the right shell
front	the front shell
back	the back shell
ie1	the No.1 point of the shell's inside
ie2	the No.2 point shell's inside
ie3	the No.3 point shell's inside
ie4	the No.4 point shell's inside
ie	the shell's inside
oil	the oil sump
c_d	the downside coil
c_u	the upside coil
core	the core
shaft	the shaft
cw	the cylinder well
cover	the cylinder cover
case	the case
smw	the suction muffler well
dmw	the discharge muffler well
oe	the environment temperature
suc	the suction gas
dis	the discharge gas
leak	the leakage
sm	the suction muffler
dc	the discharge chamber
shell	the compressor shell
dm	the discharge muffler
dp	the discharge pipe

REFERENCES

Ancel C., & Ginies P. (2008). Reciprocating Compressor Program*, International Compressor Engineering Conference at Purdue*, July 14-17, Purdue University, West Lafayette, Indiana, USA, p1-8 (1141).

Almbauer R. A., Burgstaller A., Abidin Z., & Nagy D. (2006). 3-Dimensional Simulation for Obtaining the Heat Transfer Correlations of a Thermal Network Calculation for Hermetic Reciprocating Compressor, *International Compressor Engineering Conference at Purdue*, July 17-20, Purdue University, West Lafayette, Indiana, USA, C079.

Damle R., Rigola J., Perez-Segarra D., & Oliva A. (2008). An Object Oriented Program for the Numerical Simulation of Hermetic Reciprocating Compressor Behavior, *International Compressor Engineering Conference at Purdue*, July 14-17, Purdue University, West Lafayette, Indiana, USA, p1-8 (1402).

Kara, S., & Oguz, E. (2010). Thermal analysis of a small hermetic reciprocating compressor. *Planta Medica, 37*(5), 438-445.

Lohn, S. K., Diniz, M. C., & Deschamps, C. J. (2015). A thermal model for analysis of hermetic reciprocating compressors under the on-off cycling operating condition. *British Food Journal, 90*(1).

Pereira E. L. L., Deschampes C. J., & Ribas F. A. (2008). A Comparative Analysis of Numerical Simulation Approaches for Reciprocating Compressors, *International Compressor Engineering Conference at Purdue*, July 14-17, Purdue University, West Lafayette, Indiana, USA, p1-8 (1303).
Ribas, F. A., Deschamps, C. J., Fagotti, F., Morriesen, A., & Dutra, T. (2008). Thermal analysis of reciprocating compressors - a critical review. *Journal of the Florida Medical Association Florida Medical Association,36*, 290-294.
Takemori, C. K. (2009). Numerical simulation of the fluid-structure interaction applied to the reed valves of a miniaturized compressor under high frequency conditions, *Proc. of International Conference on Compressors and their Systems*.
Zhou, R., Guo, B., Chen, X., Tuo, J., Wu, R., & Fagotti, F., et al. (2017). Heat transfer simulation for reciprocating compressor with experimental validation. , 232(1), 012011.