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Investigation on the Mass Flow Rate of a Refrigerator Compressor Based on the p–V Diagram

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Abstract: The refrigerant mass flow rate of a refrigerator compressor can only be measured by a mass flow meter and heat balance method. This paper focuses on the expansion and compression phase in which the compressor cylinder is closed, and proposes a measurement method of instantaneous mass flow of the refrigerator compressor. The comparison of the experimental pressure variation in the p–V diagram and the theoretical adiabatic process implied that the expansion and compression process of the refrigerator compressor approximated the adiabatic process. Based on the approximations and the experimental p–V diagram, a calculation method for refrigerant mass in the cylinder during the expansion and compression phase is proposed. Subsequently, the mass flow of the refrigerator compressor can be obtained. Furthermore, compared with experimental data and based on the method proposed in this paper, the error of the mass flow rate obtained is less than 3.13%. Based on this calculation method and the experimental p–V diagram, the influence of suction pressure on compressor performance is investigated.

Keywords: p–V diagram; mass flow rate; suction pressure; refrigerator compressor

1. Introduction

The refrigerator compressor determines the mass flow rate of the refrigerator system, which has a significant impact on the refrigeration capacity of the refrigerator [1,2]. Moreover, the flow rate of refrigerant is an important performance parameter that reflects the transport capacity of the compressor [3,4].

Since the flow rate of the reciprocating compressor is unsteady, it is very difficult to measure the instantaneous flow rate directly. The refrigerant flow rate of the compressor is usually measured by a mass flow meter in the refrigeration system or calculated by a mathematical model with many empirical coefficients. To predict the refrigerant mass flow rate in a domestic refrigerator, Fatouha et al. [5] developed a theoretical model of adiabatic capillary tubes. Björk et al. [6–8] proposed a simple technique to measure refrigerant mass flow rate by quick-closing valves and a tank. Based on the technique, the author studied the refrigerant mass charge distribution in both steady-state and transient conditions of the refrigerator.

Researchers have analyzed the influence factors of refrigerant mass flow rate to improve the efficiency of the compressor. Fabrício et al. [9] stated that flow rate loss was linked to the backflow rate through the suction valve. Ribas et al. [10] found that in refrigerator compressors, the vapor superheating in the suction process accounts for 49% of the overall mass flow rate loss. Morriesen et al. [11] revealed that the compressor mass flow rate was strongly linked to the valve motion.

A p–V diagram of the reciprocating compressor is traditionally used to analyze the thermodynamic process in the cylinder [12]. Ma et al. [13] showed that the experimental p–V diagram of a CO_2 compressor and presented the flow losses related to the suction and discharge valves in the diagram. Their research clearly showed that the suction power loss was lower than the discharge flow loss since the amplitude of the pressure loss in the discharge process was much larger than that in the suction process.

Based on the literature mentioned above, few studies have focused on the direct measurement for the flow rate of the refrigerator compressor, which is helpful to reveal the law of mass flow rate variation and to optimize the compressor. In this paper, a refrigerator compressor is modified to facilitate the installation of sensors to record the pressure variation in the compressor cylinder. Based on the measurements and investigation of the expansion and compression phases of the refrigerator compressor, a calculation method for refrigerant mass flow rate is presented. Furthermore, the calculation method is verified by experiments and used to study the performance of compressor.

2. Experimental Setup

2.1. Test Compressor

To install the sensors, an existing WQ153Y refrigerator hermetic compressor (Qianjiang refrigeration, Hangzhou, China) with refrigerant R600a was modified, as shown in the Figure 1. The compressor structural parameters are listed in Table 1. The power of the compressor is rated at 170W and the displacement is 44.53 L·min^{-1}.
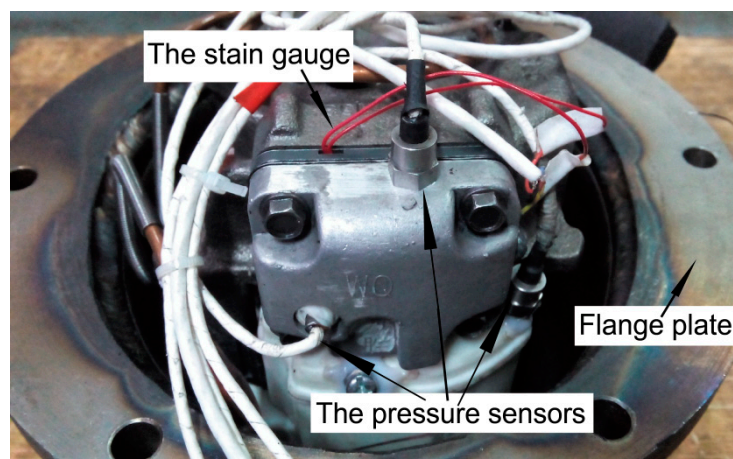

Figure 1. Schematic diagram of modified compressor.

Table 1. The refrigerator compressor structural parameters.

Parameters	Value
Crank Radius	10 mm
Connecting Rod Length	40.5 mm
Cylinder Diameter	31 mm
Rated Speed	2950 r·min^{-1}
Relative Clearance Volume	2.35%

Due to the small internal space of the refrigerator compressor, the sensors to record the compressor p–V diagram were installed with consideration of their dimensions and the actual structure of the compressor. In this paper, the high-precision pressure sensors were embedded in the valve seat, as shown in Figure 2, to record the transient pressure in the cylinder.
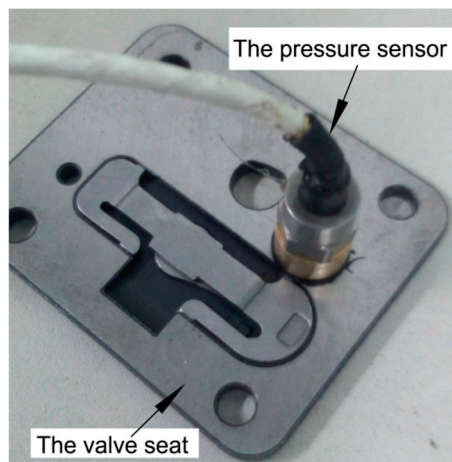
Figure 2. The installation of pressure sensor.

2.2. Data Acquisition System

The data acquisition system for the experiment in this paper is shown in Figure 3. The NI-9205 (National Instruments, Austin, TX, USA) was used for the acquisition of pressure signals from a KULITE pressure sensor (Kulite Semiconductor Products, Inc., Leonia, NJ, USA) with sampling rate set to 50 kHz; measurement accuracy was 0.1% (full scale). The mass flow of the compressor was measured by a Coriolis mass flowmeter (CMF) (Emerson, St. Louis, MO, USA) with a 0.1% reading measurement accuracy.
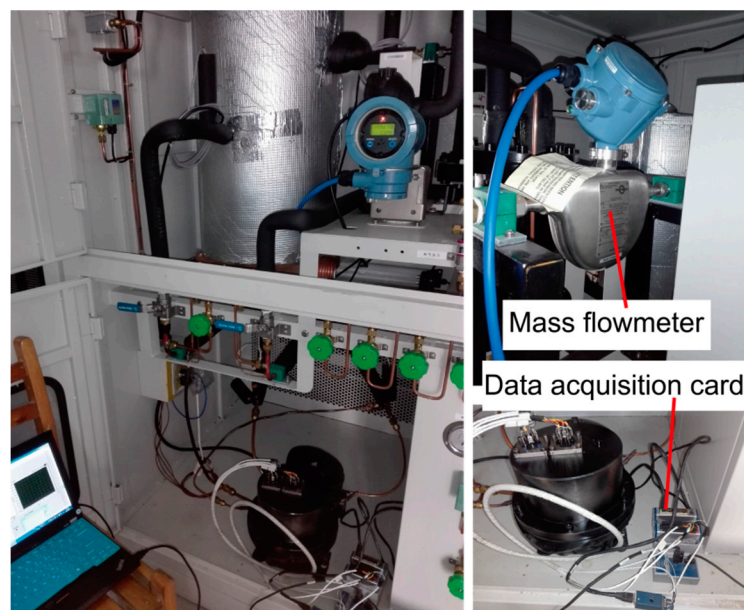

Figure 3. Test facility for the refrigeration system.

3. Method to Calculate the Refrigerant Mass Flow Rate

3.1. Validation of Adiabatic Process Hypothesis in the Expansion Phase and Compression Phase

As shown in Figure 4, the experimental pressure variations in the expansion and compression phases are in good agreement with the theoretical adiabatic pressure variations. The error between the theoretical adiabatic pressure variations and experimental data are listed in Table 2. The maximum error in the compression and expansion phase is −2.50% and 4.64%, respectively. The approximate pressure variation implies that the leakage and heat transfer have little influence on the refrigerator

compressor and can be ignored since the rotation speed of refrigerator compressor is high. Therefore, the compression and expansion phases can be approximately considered as an adiabatic process [14].
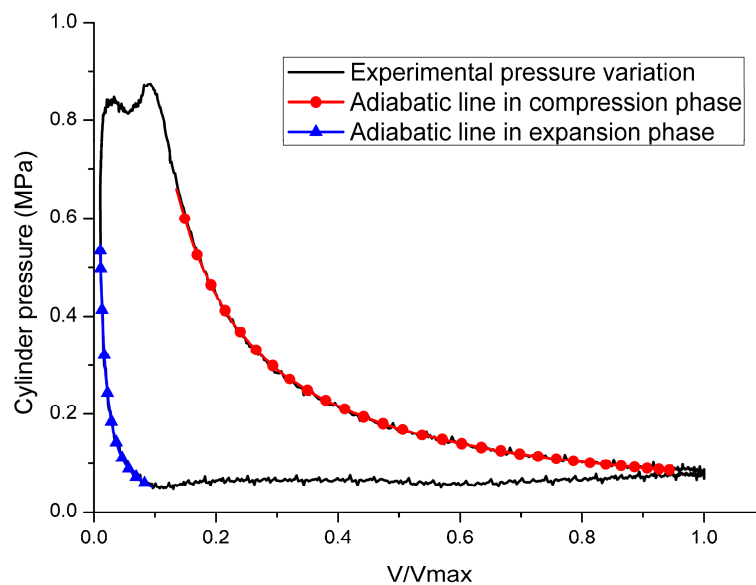

Figure 4. Comparison of theoretical adiabatic pressure variations and experimental data in a p–V diagram.

Table 2. The error between theoretical adiabatic pressure variations and experimental data.

Compression Phase			Expansion Phase		
Experimental Pressure (MPa)	Adiabatic Line (MPa)	Error (%)	Experimental Pressure (MPa)	Adiabatic Line (MPa)	Error (%)
0.088	0.086	1.86	0.529	0.504	4.64
0.089	0.088	1.04	0.473	0.483	−2.16
0.090	0.090	0.24	0.457	0.467	−2.12
0.091	0.092	−1.73	0.449	0.450	−0.23
0.094	0.095	−1.20	0.440	0.432	1.81
0.097	0.098	−1.06	0.425	0.413	2.86
0.099	0.101	−2.10	0.390	0.384	1.53
0.103	0.105	−1.68	0.357	0.366	−2.31
0.109	0.109	−0.81	0.294	0.303	−3.17
0.114	0.114	0.00	0.292	0.287	1.50
0.117	0.120	−1.72	0.268	0.264	1.51
0.126	0.125	0.54	0.246	0.250	−1.68
0.130	0.132	−1.45	0.230	0.236	−2.59
0.137	0.140	−1.63	0.206	0.212	−2.91
0.148	0.148	−0.14	0.202	0.200	0.71
0.154	0.158	−2.50	0.195	0.190	2.65
0.166	0.168	−1.53	0.172	0.170	0.66
0.178	0.181	−1.60	0.159	0.162	−1.35

Table 2. *Cont.*

Compression Phase			Expansion Phase		
Experimental Pressure (MPa)	Adiabatic Line (MPa)	Error (%)	Experimental Pressure (MPa)	Adiabatic Line (MPa)	Error (%)
0.190	0.194	−2.01	0.151	0.153	−1.41
0.205	0.210	−2.37	0.148	0.146	1.33
0.223	0.228	−1.92	0.145	0.139	4.70
0.242	0.248	−2.72	0.121	0.120	0.86
0.268	0.272	−1.31	0.111	0.111	−0.16
0.296	0.299	−1.15	0.106	0.106	0.24
0.325	0.331	−1.74	0.094	0.091	3.97
0.363	0.368	−1.34	0.081	0.081	−0.05
0.412	0.412	0.09	0.075	0.078	−3.75
0.461	0.463	−0.53	0.068	0.066	2.77
0.531	0.525	1.17	0.061	0.063	−2.97

The polytropic exponent is an important indicator to evaluate the reciprocating compressor performance, since faultless and leaking valves show different gradients in the expansion and compression phase. The polytropic exponents in the expansion and compression phase are 1.0968 and 1.0935, respectively, as shown in Figure 5. The approximately equal exponents illustrate that that the modified compressor is faultless and that the heat transfer can be ignored at a higher rotation speed [15].
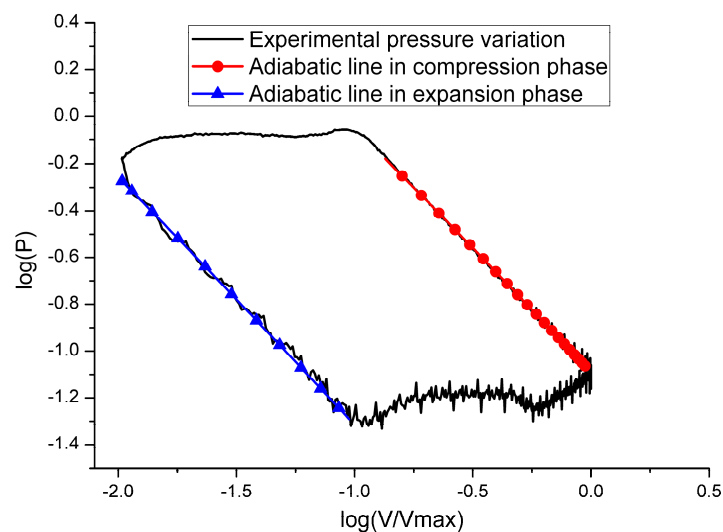

Figure 5. Comparison of experimental data and adiabatic calculation in a log(*p*)–log(*V/Vmax*) diagram

Knowing if the compression and expansion phases are approximate to the theoretical adiabatic process is critical to the correctness of the method for calculating the mass flow rate of the compressor, which is proposed in Section 3.2.

3.2. Calculation Method

Due to the high rotation speed, the refrigerant leakage and the heat exchange between the cylinder wall and the refrigerant can be ignored in the refrigerator compressor [12]. Therefore, in the expansion and compression phases, the valves are closed and the refrigerant mass remains constant [16]. Moreover, the temperature and velocity gradients in compressor cylinder are small enough to neglect the irreversibility [12].

In the compression and expansion phases, the compressor cylinder is regarded as a closed thermodynamic system with masses $\dot{m}_{com}$ and $\dot{m}_{exp}$, respectively. During the compression phase, the refrigerant in the cylinder includes the fresh refrigerant suctioning from the suction line and the refrigerant remaining in the clearance volume. During the expansion phase, the refrigerant in the cylinder only includes the refrigerant remaining in the clearance volume. Thus, the refrigerant mass flow rate of the compressor in one cycle can be calculated:

$$\dot{m} = \dot{m}_{com} - \dot{m}_{exp} \tag{1}$$

where $\dot{m}$ is the refrigerant mass in the chamber, which is assumed to remain constant during the expansion and compression phases.

Following Section 3.1, the compression and expansion phases approximate adiabatic processes. Based on thermodynamics, the two phases follow the relationship:

$$\dot{m}_{exp,com} = V_c \rho \left(p, \frac{C_p}{C_v} \right) \tag{2}$$

where V_c is the cylinder volume at the corresponding time, and p is the refrigerant pressure in the cylinder. The ratio C_p/C_v is usually considered equal to the adiabatic exponent. Moreover, the density ρ of refrigerant in the cylinder is a function of p and C_p/C_v. The physical property data for R600a are derived from NIST PEFPROP [16].

In Equation (2), in fact, C_p/C_v is determined by the corresponding temperature and p. However, the measurement of the instantaneous refrigerant temperature in the cylinder is a difficult problem. In this paper, due to the expansion phase and compression phase being close to the adiabatic process, the adiabatic exponent is approximately equal to the polytropic exponent obtained from the $\log(p)$–$\log(V/Vmax)$ diagram [12]. In addition, p and V_c could be measured directly by experiment.

Based on the Equation (2), a method is proposed to obtain the refrigerant mass in the two phases. A flow chart for calculating the refrigerant mass in the compression phase is shown in Figure 6. The calculation is started by the assumed temperature and the adiabatic exponent obtained experimentally. The pressure, density, and mass in the cylinder can be calculated by the temperature and adiabatic exponent. With the volume variation of the cylinder and adiabatic process, the variation of pressure is obtained. In addition, the convergence condition of the calculation is whether the difference between the experimental and the calculation pressure variations is sufficiently small. Subsequently, the refrigerant mass in the expansion phases can be obtained in the same way.
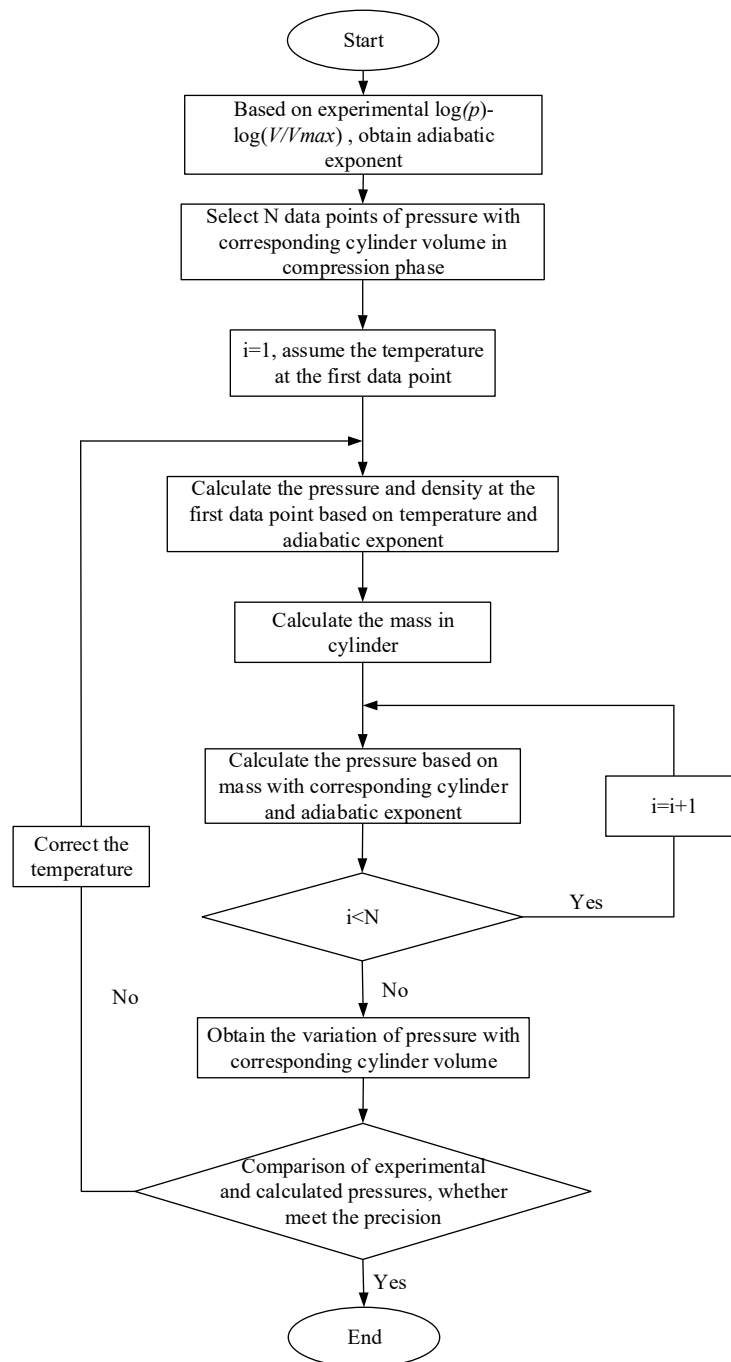
Figure 6. Flow chart for the method of calculating the refrigerant mass in the cylinder

4. Results and Discussion

4.1. Verification of Mass Flow Calculation Method

To verify this method, the mass flow of the compressor is measured in the experiment. The comparison of the experimental and calculated flow rate is shown in Table 3, which implied that the error of the mass flow calculation method proposed in this paper is less than 3.13% and the calculation method meets the engineering requirement.

Table 3. The comparison of the experimental and calculated flow rate.

Suction Pressure (MPa)	Suction Temperature (°C)	Discharge Pressure (MPa)	Flow Rate		Error (%)
			By the p–V Diagram (kg·h^{-1})	By Flow Meter (kg·h^{-1})	
0.11	−5.60	0.59	6.09	5.99	1.64
0.12	−2.50	0.75	6.91	6.72	2.75
0.10	−0.80	0.66	5.17	5.03	2.71
0.11	2.80	0.64	5.45	5.38	1.28
0.08	6.40	0.60	3.51	3.40	3.13

4.2. Effect of Suction Pressure on the Compressor Performance

Volume efficiency and flow loss are two important performance indexes of compressors. Based on the experimental p–V diagram and the mass flow rate calculation method proposed in this paper, the effect of suction pressure on the volume efficiency and the suction flow loss are investigated.

The volumetric efficiency of the refrigerator compressor is calculated as follows:

$$\eta_v = \frac{\dot{m}}{\rho_{Inlet} V_{theo}}$$
(3)

where $\dot{m}$ is the actual mass flow rate of the compressor, V_{theo} is cylinder volume of the refrigerator compressor without clearance volume, and ρ_{Inlet} is the refrigerant density at the suction conditions.

4.2.1. Effect of Suction Pressure on the Volumetric Efficiency

The pressure ratio and clearance volume are usually considered to be the main factors affecting the volumetric efficiency [17]. Based on the calculation method for refrigerant mass in the cylinder, the influence of suction pressure on the volumetric efficiency of the refrigerator compressor is investigated, while the pressure ratio is kept constant. The volumetric efficiency increases from 0.59 to 0.71 when the suction pressure is increased from 76.2 to 112.8 kPa, and the pressure ratio remains 6.45, as shown in the Figure 7. That is a basis for improving the compressor control strategy.
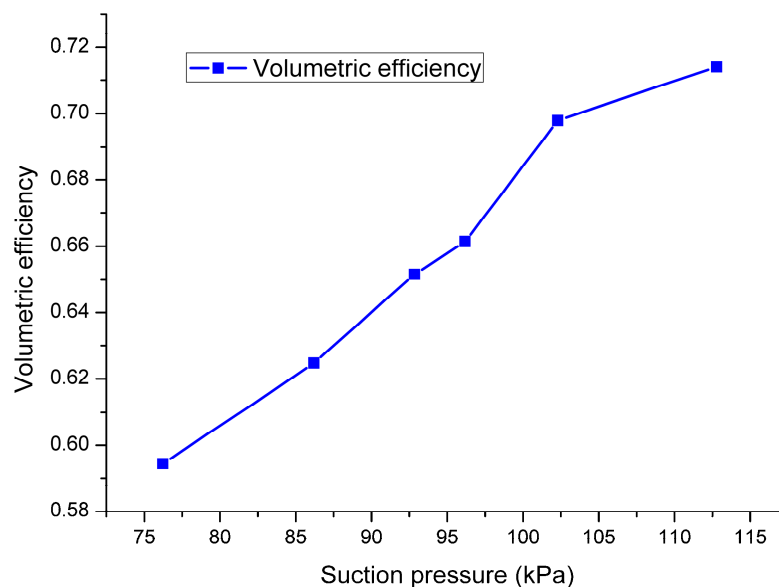
Figure 7. Volumetric efficiencies and suction time under different suction pressures

4.2.2. Effect of Suction Pressure on Suction Flow Loss

Figure 8 shows the experimental p–V indicator diagram, and the flow losses through the valve can be obtained. Since the suction time is longer than the discharge, the suction loss is two to three times the discharge loss and is thus the main form of energy efficiency loss. There is considerable loss in the suction process.
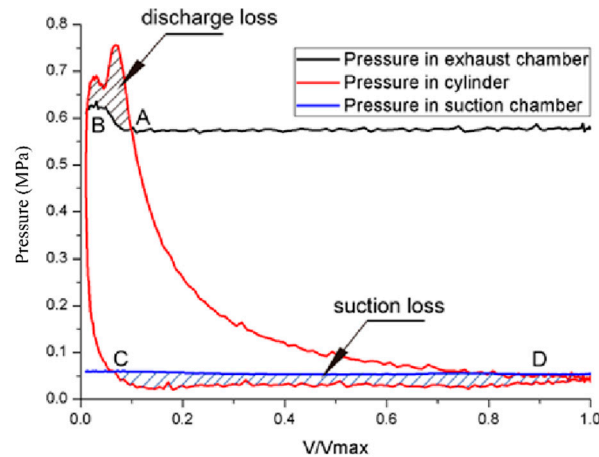

Figure 8. The flow loss through valve in p–V indicator diagram.

The gas force acting on the suction valve decreases as the suction pressure decreases. To open the suction valve, the flow resistance that overcomes the spring force of the valve accounts for a greater proportion of the suction pressure. As shown in Figure 9, the suction loss increases from 12.44% to 20.62%, with the suction pressure varying from 112.78 to 76.2 kPa, and the pressure ratio is maintained at 6.4.
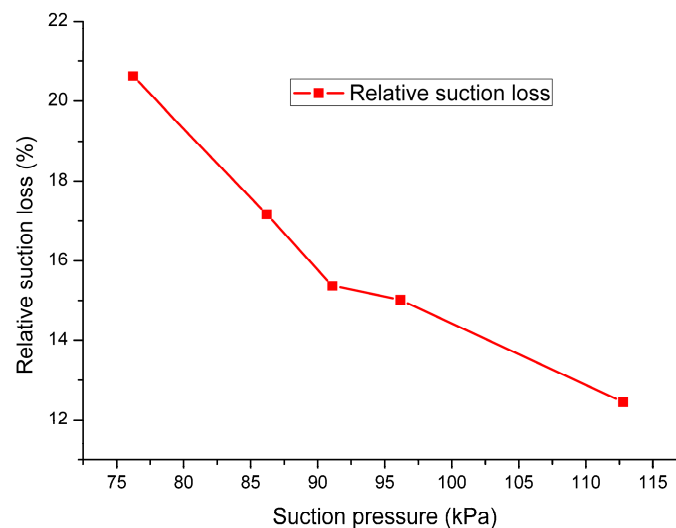

Figure 9. Suction loss under different suction pressures.

5. Conclusions

The refrigerant mass variation in the cylinder is researched experimentally, which shows a significant influence on the mass flow characteristics of a refrigerator compressor. As the first examination of this issue, the study benefits refrigerator compressor optimization. The major conclusions follow:

1. With the compression and expansion phase approximating the adiabatic process being experimentally verified, a calculation method for the mass flow rate of a compressor is proposed, based on the experimental p–V diagram.

2. The accuracy of the calculation method for the mass flow rate of a compressor is directly verified by a mass flow meter. Furthermore, the error of the calculation method is less than 3.13%, which can replace the mass flow meter for most application situations.

3. Based on the calculation method, the influence of suction pressure on compressor performance is investigated. Under a constant pressure ratio, the higher suction pressure leads to a higher volumetric efficiency and less suction loss.

4. As the measuring equipment is expensive and the calculation method is complex, the proposed method at present is mainly suitable for scientific research. In the future, the authors will focus on reducing the complexity of the method, and based on this method, the mass flow meter of compressor will be manufactured.

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Conflicts of Interest: The authors declare no conflict of interest.

Nomenclature

Cp	Specific heat at constant pressure
Cv	Specific heat at constant volume
m	Refrigerator mass in the cylinder
p	Pressure
V_c	The volume of cylinder
V_{theo}	Theoretical volume of the compressor
ρ	Refrigerant density
Subscripts	
com	Compression phase
exp	Expansion phase
$Inlet$	Suction states

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