Development of a High Speed R290 Compressor for Room Air Conditioner

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ABSTRACT

With the implementation of Kigali Amendment, R290, as a natural refrigerant, has become more and more popular in many applications, for example, in room air conditioner. Compared with recently widely used R32 or R410A in this application, R290 has lower per-unit cooling capacity of swept volume. It means the displacement of R290 compressor need to be bigger than that of R32 or R410A compressor, which will increase the cost of compressor significantly. Another effective way to meet the cooing capacity requirement is to improve the rotational speed of R290 compressor. However with the improvement of rotational speed, there are many challenges for compressor performance and reliability related to compressor structure and key components. This paper mainly focuses on the discharge system of R290 compressor. A mathematical model for the compressor's discharge system was established, and its accuracy was verified through experiments. The research results indicate that adopting a dual discharge structure is an effective way to reduce resistance loss at the compressor's discharge port, but it could increase the clearance volume, resulting in a decrease in performance at low speeds. For the R290 compressor studied in this paper, the balance point between the performance of the single and dual discharge structures is approximately 90Hz.

1. INTRODUCTION

The refrigeration industry has been a significant contributor to global warming, primarily due to the widespread use of refrigerants with high greenhouse gas emissions. This has led to an urgent need for the development and adoption of environmentally friendly refrigerant alternatives.

The Kigali Amendment, adopted in 2016 under the Montreal Protocol on Substances that Deplete the Ozone Layer, marks a significant milestone in global efforts to combat climate change. It specifically targets the phase down of hydrofluorocarbons (HFCs), a class of refrigerants widely used in air conditioning and refrigeration systems due to their excellent properties. However, HFCs also possess a high GWP, making them a significant contributor to global warming.

Propane (R290), a hydrocarbon compound, has emerged as a potential solution in this application (Palm, 2008). It has a low GWP compared to HFCs, making it a more environmentally friendly refrigerant choice. Its natural decomposition in the atmosphere does not produce harmful byproducts, further enhancing its sustainability credentials. Additionally, propane is widely available and relatively inexpensive, making it a cost-effective alternative for refrigeration systems.

Despite the environmental advantages of R290, when applied in the field of room air conditioner, its unit volume cooling capacity is relatively low compared to the currently mainstream refrigerants for room air conditioner, such as R32 or R410A. Under typical air conditioning conditions, it only reaches 60% of the cooling capacity of R32 refrigerant. Therefore, to achieve the same cooling capacity, the displacement of the R290 compressor needs to be increased by 160% compared to the R32 compressor, which will lead to an increase in compressor size and consequently, an increase in compressor cost. Additionally, as R290 is a flammable refrigerant, the increased compressor size will also increase the mass of R290 in the compressor, thus posing a higher safety risk.

Another feasible approach, instead of increasing the displacement of the R290 compressor, is to maintain the compressor displacement and increase the operating speed of the compressor to achieve the same cooling capacity. However, increasing the compressor's operating speed may also bring performance and reliability issues that need to be addressed through research.

The rolling piston compressor, with its simple structure, high efficiency, and low cost, is widely used in room air conditioners. Because of the structural characteristics of the rolling piston compressor, when it operates at high speed, the discharge system (including the discharge structure and discharge valve) will become an important factor affecting the compressor's performance and reliability. However, there are currently limited researches in this area (Kang, 2015). Therefore, this article will mainly focus on the study of the discharge system of R290 compressor.

2. MATHEMATICAL MODEL

This article establishes a mathematical model for the discharge system of rolling piston compressors, including a finite element model for the discharge valve and a thermodynamic model for calculating the pressure inside the working chamber (Huang, 2008).

2.1 Finite Element Model for Discharge Valve

The discharge valve of the rolling piston compressor is a reed valve, and its motion characteristics during operation is determined by the gas pressure difference acting on both sides and its own elastic force. To accurately calculate the motion of the valve, it is necessary to select the appropriate type of finite element. In this paper, considering the motion and force characteristics of the valve during operation, as well as the calculation time and accuracy, a variable cross-section beam element is adopted, as shown in Figure 1. Each element has two nodes and four degrees of freedom. The node displacement array q^e and the node force array P^e are as follows:

$$q^e = \begin{bmatrix} v_1 & \theta_1 & v_2 & \theta_2 \end{bmatrix}^T \tag{1}$$

$$p^e = \begin{bmatrix} F_1 & M_1 & F_2 & M_2 \end{bmatrix}^T \tag{2}$$

Wherein，v_1，θ_1，v_2，θ_2 represent the deflection and rotational angle of each node respectively.

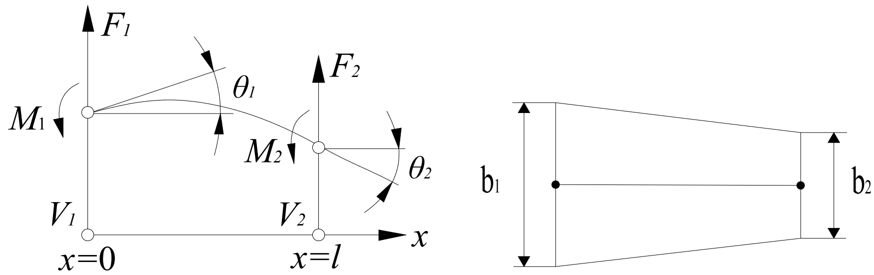


Figure 1: Variable cross-section beam element

Figure 6 is a schematic diagram of the finite element model of the discharge valve established in this paper.

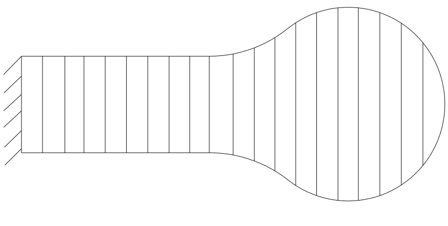


Figure 2: Finite elements model of the valve

After determining the element type, the mass matrix M and stiffness matrix K can be determined, and then the equation of motion for the valve can be obtained:

$$[M]\{\ddot{x}\}+[C]\{\dot{x}\}+[K]\{x\}=\{F\} \tag{3}$$

In the equation, $\ddot{x}$ represents the node acceleration, $\dot{x}$ represents the velocity, x represents the lift, C represents the damping matrix (Qian, 2019), and F represents the load vector.

2.2 Thermodynamic Model

The load on the valve plate during motion mainly comes from the pressure difference between the compressor compression chamber and the back pressure chamber (mainly the chamber formed by the muffler) of the valve. To accurately calculate the pressure changes in these chambers, this paper takes the compression chamber and the back pressure chamber as control volumes respectively, and adopts thermodynamic methods to establish the following energy and mass equations for the control volumes:

$$\frac{dT}{d\theta}=\frac{1}{mC_v}\left\{-T\left(\frac{\partial P}{\partial T}\right)_v\left[\frac{dV}{d\theta}-\frac{v}{\omega}\left(\dot{m}_{in}-\dot{m}_{out}\right)\right]-\sum\frac{\dot{m}_{in}}{\omega}\left(h-h_{in}\right)+\frac{\dot{Q}}{\omega}\right\} \tag{4}$$

$$\frac{dm}{d\theta}=\sum\frac{\dot{m}_{in}}{\omega}-\sum\frac{\dot{m}_{out}}{\omega} \tag{5}$$

Since the motion of the valve is a fluid-structure interaction process, the finite element motion model of the valve and the thermodynamic model of the chambers must be solved simultaneously.

2.3 Impact model between valve and lift limiter

The complexity of the valve motion lies in the fact that it is not only a fluid-structure interaction process, but also involves the impact between the valve and the lift limiter. For the impact process between the valve and the lift limiter, this paper adopts a velocity rebound model. That is, when a certain node on the valve plate impact with the lift limiter, the node will rebound at a lower speed. The rebound velocity is the product of the instantaneous impact velocity and a rebound coefficient, while the displacement of the node is equal to the lift at that position on the corresponding lift limiter profile.

$$\begin{cases} y=H(x) \\ u=-C_r u(x) \end{cases} \tag{6}$$

In the equation, y represents the displacement of the valve after impacting with the lift limiter, H(x) represents the lift at the corresponding position of the lift limiter, u represents the rebound velocity of the valve after impacting with the lift limiter, Cr represents the rebound coefficient, and $u(x)$ represents the instantaneous velocity of the valve when it impacts with the lift limiter.

3. EXPERIMENTAL VERIFICATION

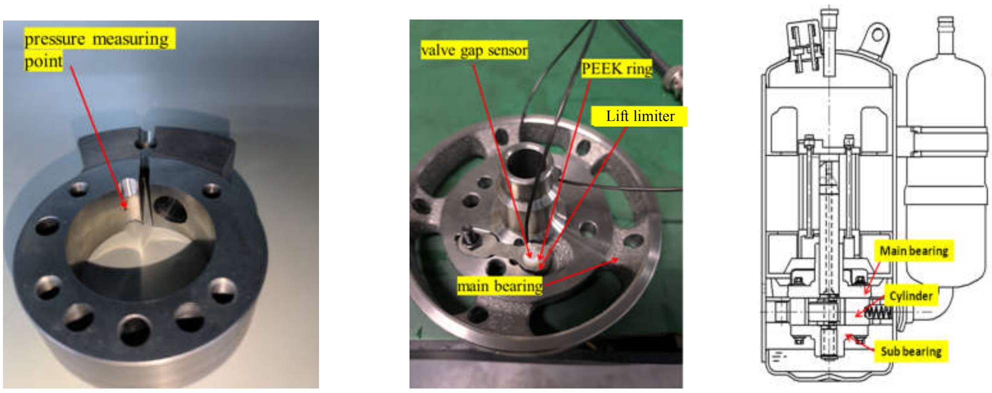
In this paper, a high-speed R290 prototype compressor was developed, whose displacement is the same as that of the R32 compressor currently used in room air conditioners. It's a single-cylinder compressor, since the displacement is small, it has only one discharge valve, which located at the main bearing of the compressor. To achieve the same cooling capacity, the maximum operating frequency of the compressor needs to reach 180 Hz. The basic specifications of the compressor are shown in Table 1. In order to obtain the internal working characteristics of the compressor at high speed, a pressure sensor is installed near the discharge port of the cylinder, and a gap sensor is installed on the valve lift limiter. The specifications of the sensors are shown in Table 2, and the compressor with sensors installed is shown in Figure 3.

Table 1: Basic specification of the prototype

Parameters	Values
Refrigerant	R290
Displacement/ cm^3	10.2
Valve lift/mm	1.77

Table 2: Specification of Sensors

Sensor type	Model	Installation position	Range	Accuracy
Pressure sensor	Kistler 4007c	Compression chamber	0-5MPa	%FSO ≤±0,8
Gap sensor	AEC PU05	Valve lift limiter	0-2mm	Less than ±1%



(a) Cylinder (b) Main bearing (c) Compressor schematic view

Figure 3: Compressor with sensors installed

The above compressor is placed on the R290 compressor performance test bench for testing, and the test conditions are shown in Table 3.

Table 3: Test conditions

Parameters	Values
Suction Pressure/MPa	0.588
Discharge Pressure/MPa	1.883
Suction Temperature/℃	35.0
Frequency/Hz	60-180

Figure 4 shows the trend of compressor isentropic efficiency with rotational speed. At high speeds, as the rotational speed increases, the isentropic efficiency of the compressor decreases significantly. Therefore, improving compressor performance at high speeds is a critical issue of high-speed compressor development.

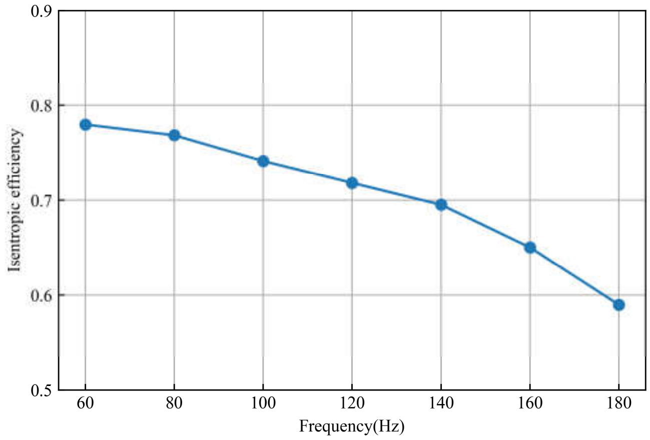


Figure 4: Variation of isentropic efficiency with rotational speed

Figure 5 shows the trend of compression chamber pressure with rotational speed. As the speed increases, over-compression becomes more severe, leading to higher over-compression losses. This is the primary reason for the rapid decline in isentropic efficiency during high speed operation.

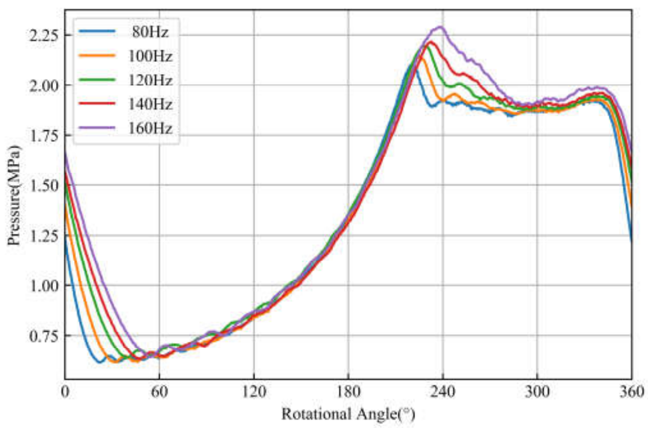


Figure 5: Variation of compression chamber pressure with rotational speed

Figure 6 depicts the trend of discharge valve lift with rotational speed. As the speed increases, the opening angle of the valve will be delayed, and the duration of opening will be longer.

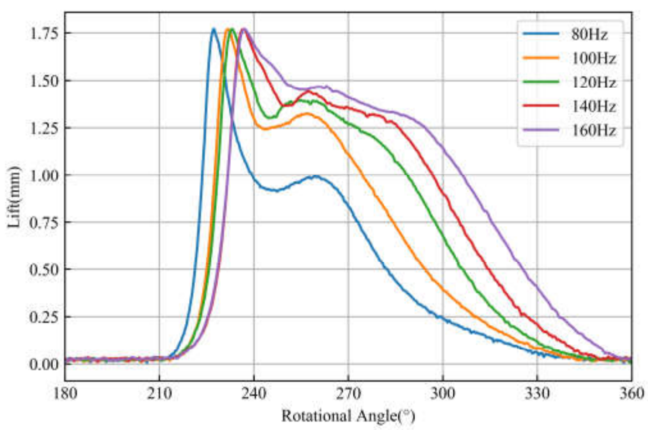


Figure 6: Variation of valve lift with rotational speed

Figure 7 compares the experimentally measured valve lift changes with that calculated by simulation. Based on the comparison in the figure, the valve model established in this paper can accurately simulate valve operation and can be used for optimal design of the discharge system in high-speed compressors.

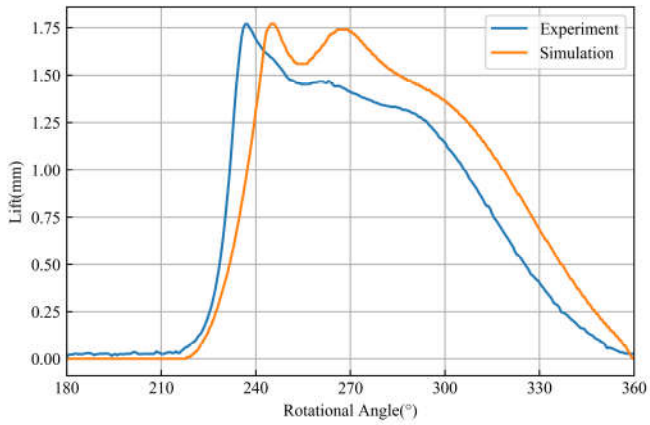


Figure 7: Comparison of valve lift between experiment and simulation at 160Hz

4. OPTIMIZATION OF HIGH SPEED COMPRESSOR

It can be seen from the above analysis that when the R290 compressor maintains the same displacement as the R32 compressor and relies on increasing the speed to increase the cooling capacity, the isentropic efficiency of the compressor will drop significantly at high speeds. This is mainly due to the increase in volumetric flow rate at high speeds, which causes greater resistance loss through the discharge port, leading to increased over-compression loss, reduced compressor efficiency, deteriorated valve operating conditions, and decreased reliability.

Although increasing the size of the discharge port can theoretically reduce discharge resistance loss and improve valve operating conditions, structural limitations arise due to the compressor's small displacement. An alternative solution is to set another discharge structure on the sub bearing of the compressor. Figure 8 compares the pressure changes in the compression chamber at high speeds, calculated using the model presented in this paper, for both the single-discharge scheme and the dual-discharge scheme. Meanwhile, Figure 9 compares the valve lifts variation and figure 10 compares the valve velocity variation. The results indicate that implementing the dual-discharge structure significantly reduces discharge resistance loss and valve impact velocity during discharge process.

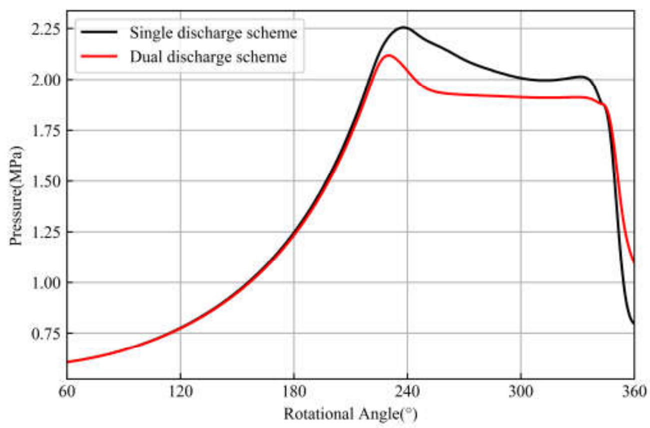


Figure 8: Comparison of pressure between single discharge and dual discharge at 160Hz

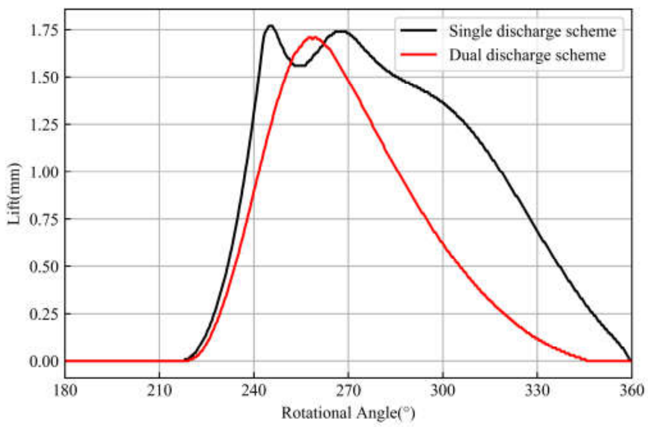


Figure 9: Comparison of valve lift between single discharge and dual discharge at 160Hz

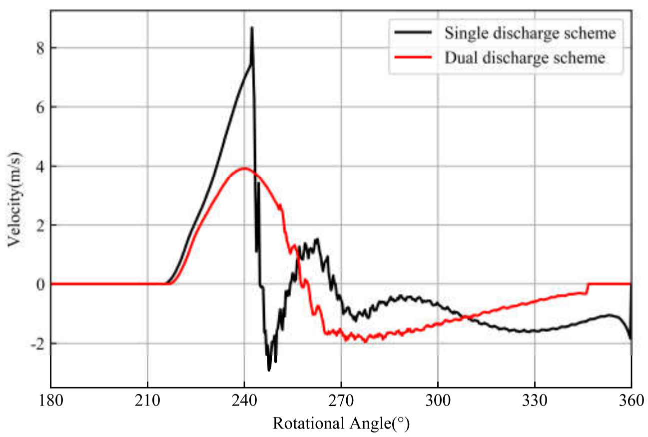


Figure 10: Comparison of valve velocity between single discharge and dual discharge at 160Hz

In this study, a high-speed compressor with a dual-discharge structure was developed and tested under same operating conditions. Its performance was compared to that of the compressor with a single discharge structure. The test results, presented in Figure 11, demonstrate that the dual-discharge design significantly enhances compressor performance at high speeds, aligning with theoretical predictions.

Meanwhile, the dual discharge structure also has the disadvantage of increasing the clearance volume, resulting in a decrease in performance at low speeds. For the R290 compressor studied in this paper, the balance point between the performance of the single and dual discharge structures is approximately 90Hz.

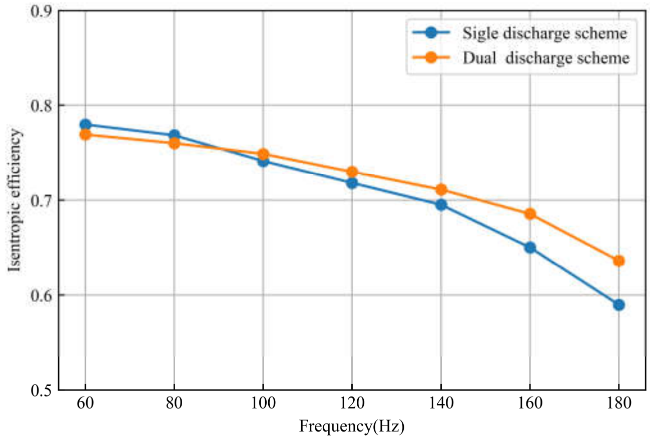


Figure 11: Comparison of isentropic efficiency between single discharge and dual discharge

5. CONCLUSIONS

Under the background of refrigerants substitution, R290 has attracted increasing attention due to its excellent environmental protection characteristics and is expected to become the next generation refrigerant. However, in the field of room air conditioner, compared with R32, R290 has a smaller cooling capacity per unit volume, requiring an increase in refrigerant volume flow rate. This paper aims to achieve this goal by increasing the compressor's rotational speed, but a higher compressor speed poses challenges to the design of the discharge system of the compressor.

This paper establishes a mathematical model for the compressor's discharge system, particularly the discharge valve, and verifies the accuracy of the model through experiments. The research results indicate that for R290 high-speed compressors, the isentropic efficiency of the compressor decreases significantly at high rotational speeds. This is primarily due to the rapid increase in resistance loss at the compressor's discharge port under a larger volume flow rate. Adopting a dual discharge structure is an effective way to solve this problem. It can reduce the discharge resistance loss at high speeds and improve the valve's motion state at the same time. Nevertheless, the dual discharge structure also has the disadvantage of increasing the clearance volume, resulting in a decrease in performance at low speeds. For the R290 compressor studied in this paper, the balance point between the performance of the single and dual discharge structures is approximately 90Hz.

NOMENCLATURE

P	Pressure	(Pa)
T	Temperature	(K)
V	Volume	(m^3)
h	Enthalpy	(J/kg)
$\dot{m}$	Mass flow rate	($kg \cdot s^{-1}$)
θ	Rotational angle	(rad)
ω	Rotational speed	($rad \cdot s^{-1}$)
v	Specific volume	($m^3 \cdot kg^{-1}$)
u	Velocity	(m/s)
F	Force	(N)
x	Displacement	(m)

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