

Article

Experimental Research and Simulation for the Performance of an R290 Heat Pump with Independent Compression

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Abstract

Since the Kigali Amendment entered into force globally, propane (R290) has been regarded as one of the most promising next-generation alternative refrigerants for refrigeration and air conditioning. However, its flammability limits its maximum charge amount and leads to higher flow resistance loss. In this paper, a novel refrigeration cycle with an additional independent compression process was simulated and experimentally tested. The simulation error of capacity was less than 7.1%. The intermediate evaporation temperature was optimized. The results show that the new cycle delivers stable performance advantages over the conventional R290 heat pump in both cooling and heating modes, with average capacity and COP improvements of 4.8% and 7.8% for cooling, and 8.3% and 7.5% for heating. System flow resistance loss decreases by 33.0%, which raises the refrigerant mass flow rate by 12.5% and reduces the required compressor displacement by 6.6% at equivalent cooling capacity on average.

Keywords: R290; heat pump; independent compression; simulation

1. Introduction

Heat pumps play a vital role in achieving energy-efficient heating and cooling while accelerating global carbon neutrality goals, yet traditional heat pump refrigerants have high Global Warming Potential (GWP) and cause strong greenhouse effects upon leakage [1]. As a natural hydrocarbon refrigerant, R290 features negligible GWP and zero Ozone Depletion Potential (ODP) [2], directly addressing the environmental concerns highlighted by the Kigali Amendment to the Montreal Protocol [3]. This shift is not only a regulatory necessity but also a strategic response to the urgent need for sustainable cooling solutions [4]. R290 is a natural hydrocarbon refrigerant [5]. Compared with R134a, it features a higher Coefficient Of Performance (COP), lower refrigerant charge and excellent thermophysical properties [6–8]. It also has negligible GWP, making it an eco-friendly alternative for refrigeration systems [9]. With outstanding thermodynamic characteristics, R290 is regarded as a promising low-GWP substitute refrigerant for heat pumps [10].

As two representative natural working fluids, R290 and CO₂ have been widely investigated in different application scenarios. For CO₂ transcritical systems, previous studies have demonstrated its outstanding performance in high-temperature heat pump drying [11] and simultaneous heating and cooling scenarios [12]. However, for residential room air conditioners operating under conventional temperature ranges, R290 features moderate operating pressure, high cycle efficiency and its system performance can be further improved through cycle structure optimization.



Academic Editor: Zacharie
Tamainot-Telto

Received: 12 June 2026

Revised: 12 July 2026

Accepted: 13 July 2026

Published: 16 July 2026

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R290 presents significant technical challenges in its adoption as a replacement for conventional halogenated refrigerants. The predominant constraint arises from its high flammability [2,13], which has prompted major international safety standards (e.g., IEC 60335-2-40 and EN 378) to impose stringent upper limits on refrigerant charge [14]. Furthermore, because of R290's inherently low volumetric cooling capacity, the mandated charge limitation necessitates compressors with increased displacement volume or larger volumetric flow capability to maintain the cooling load [15]. Structural modifications to conventional compressors are generally required when R290 is applied in heat pump systems [16,17]. When R290 is used as the refrigerant, its high flow velocity inside the system leads to considerable flow resistance loss. This further deteriorates system energy efficiency and requires a compressor with larger displacement. To address the above issues, a novel refrigeration cycle named independent compression was proposed by Guangdong Midea Air-Conditioning Equipment Co., Ltd.; the system improves the Annual Performance Factor (APF) by 6% compared with single-stage compression [18]. Accordingly, this paper focuses on the experimental testing and simulation of independent compression, which is developed based on the structurally modified compressor.

Studies [19] revealed that vapor-injection compressors outperformed two-stage compression units at pressure ratios below 7.5, whereas two-stage compression was more efficient for low-temperature hot water systems below $-20\text{ }^{\circ}\text{C}$. Experimental investigations on two-cylinder independent compression systems with flash separators [20,21] clarified the effects of key operating parameters and determined the optimal pressure for maximum efficiency. Exergy analysis [22] further proved that high compressor discharge pressure paired with moderate evaporation temperature contributed to the optimal energy performance.

Simulation has become an indispensable research method in the field of heat pumps and refrigeration systems, working closely with experiments to carry out relevant investigations [23–28]. On the one hand, researchers established simulation models on mainstream platforms to optimize heat pump configuration, analyze operational characteristics and screen alternative refrigerants [23,24,26,27]. On the other hand, physical experiments were conducted to calibrate and validate these models, and a high consistency between simulation and experimental results was observed [25]. In addition, simulation enables parametric analysis on key operating variables and research on complicated working conditions like hot-gas defrost, overcoming the limitations of pure experimental research [27,28].

Despite the abundant research on R290 thermophysical properties, transcritical CO_2 heat pumps, two-stage/vapor-injection compression structures and system simulation methodologies, several research gaps remain unaddressed in the existing literature. Existing studies rarely propose targeted cycle optimization schemes to resolve the excessive flow resistance loss inherent to R290 systems, with a lack of comprehensive multi-condition experimental data and transient dynamic model verification. Few works have quantified the internal drag reduction mechanism brought by independent compression, nor analyzed its variation across full cooling and heating operating conditions. Prior simulation validations for compression cycles mainly rely on steady-state data, while the transient power matching startup process is seldom calibrated with measured experimental data, which restricts the reliability of simulation models for control optimization.

As an effective solution to the limitations of R290 in heat pump systems, the independent compression concept still lacks adequate experimental investigation and simulation verification to support its practical application. The overall aim of this work is to reveal the mechanism of the novel R290 independent compression, quantify its performance superiority over conventional single-stage compression under wide operating ranges, and establish a validated steady-state and dynamic simulation platform.

This paper investigates the independent compression cycle through three key steps: (1) constructing an experimental platform to measure 39 working conditions; (2) developing and validating a model for steady-state and transient simulations; (3) comparing the performance of independent and single-stage compression. The results demonstrate the superior performance of the independent compression and provide a validated tool for system optimization, laying the groundwork for its practical application in high-efficiency heat pump systems.

2. Experiments and Simulation Validation

2.1. Description of Independent Compression

Conventional rotary compressors are only equipped with a single cylinder. The independent compression forms an auxiliary cylinder by modifying the vane structure of the rotary compressor. This structural design increases the number of compression working chambers without enlarging the overall compressor envelope, thus constructing a dual-working-chamber layout. Figure 1 illustrates the layout of the main and auxiliary cylinders inside the compressor. The two cylinders are arranged in parallel and driven by a single shared motor.

As shown in Figure 2, during cooling mode, the refrigerant gas condenses to a subcooled state 3 within the condenser. It then undergoes primary throttling to a two-phase state 4 before entering the flash evaporator, where it separates into saturated gas state 5 and saturated liquid state 6. The saturated gas refrigerant (5) is replenished into the auxiliary cylinder and compressed into high-temperature gas (8). The saturated liquid refrigerant (6) undergoes secondary throttling to a lower pressure and temperature state (7), entering the evaporator where it absorbs heat and evaporates into gaseous state 1. This then enters the main cylinder, compressed into high-temperature gas state 9. The high-temperature gas 9 from the main cylinder and the high-temperature gas 8 from the auxiliary cylinder mix within the discharge chamber to form state 2. This mixture is then discharged via the compressor's outlet into the condenser, before commencing the next cycle.

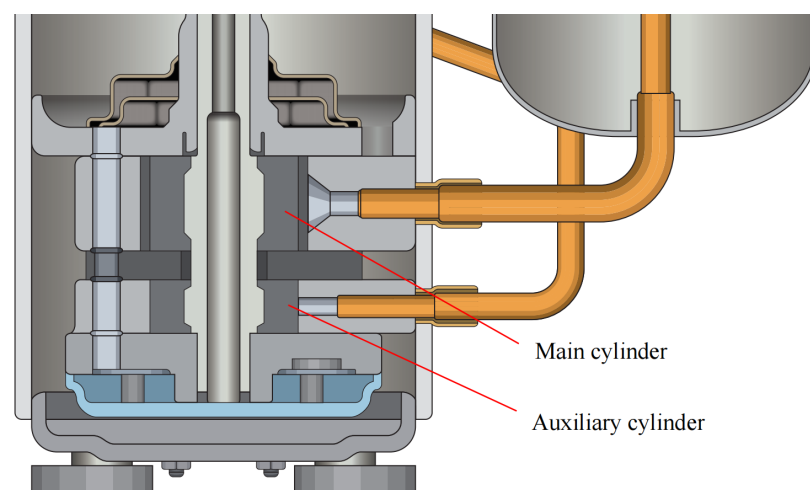


Figure 1. Structural diagram of main and auxiliary cylinders of compressor.

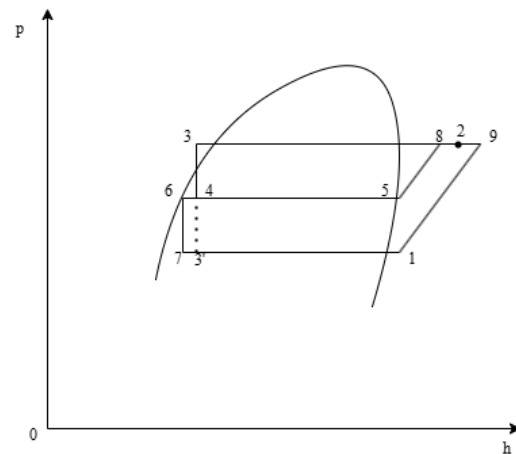


Figure 2. Pressure-enthalpy diagram of independent compression.

2.2. Simulation Model

To balance simulation accuracy and computational efficiency, the model for the independent compression cycle is developed based on the following idealized assumptions:

1. The refrigerant R290 is treated as a pure substance. The refrigerant flow in pipelines is assumed to be one-dimensional and homogeneous, with no phase separation or stratification.
2. The heat transfer coefficients of the evaporator and condenser are assumed to be constant, ignoring the dynamic changes in heat transfer performance caused by fouling, frost formation or uneven air distribution on the heat exchanger surface. The wall material of heat exchangers is assumed to have uniform thermal conductivity.
3. The compressor is modeled with a fixed isentropic efficiency and volumetric efficiency, ignoring the variation in efficiency with operating frequency, pressure ratio and ambient temperature. Mechanical losses (e.g., friction, leakage) in the compressor are neglected, and the compression process is assumed to be adiabatic.
4. Pressure drops in short connecting pipes between system components are assumed to be negligible. The electronic expansion valve is assumed to respond instantaneously to control signals.
5. Ambient temperature and humidity are assumed to be constant during steady-state simulations, ignoring transient fluctuations in the experimental environment. Air flow rates through the evaporator and condenser are assumed to be uniform and stable.

As shown in Figure 3, the simulation model comprises evaporator, condenser, compressor (contains main cylinder and auxiliary cylinder), liquid-vapor separator, electronic expansion valves (EV1 and EV2) and temperature sensors installed at the compressor discharge, as well as at the inlet, middle and outlet of both the condenser and evaporator. The core component of the independent compression system is a twin-cylinder parallel compressor matched with a two-stage throttling assembly, and a gas-liquid separator is arranged between the two throttling stages to realize intermediate suction. Specifically, the refrigerant vapor generated after primary throttling is extracted by the auxiliary cylinder before flowing into the evaporator.

1. Calculation of Compressor Parameters

The volumetric efficiency is used to compute the mass flow rate as follows:

$$dm_2 = \eta_v \cdot \rho_{suc} \cdot N \cdot disp$$

where

η_v : volumetric efficiency

ρ_{suc} : suction density

N : rotary speed of the compressor

$disp$: compressor displacement

The isentropic efficiency is used to compute the enthalpy increase through the compressor. The isentropic efficiency can be expressed as follows:

$$\eta_{is} = \frac{h_{dis} - h_s}{h_d - h_s}$$

where:

η_{is} : isentropic efficiency

h_{dis} : isentropic discharge specific enthalpy

h_s : suction specific enthalpy

h_d : discharge specific enthalpy

The enthalpy increase is then calculated as follows:

$$h_{inc} = h_d - h_s = \frac{h_{dis} - h_s}{\eta_{is}}$$

The enthalpy flow rate increase is calculated and is equal to:

$$dmh_{inc} = dm_2 \cdot h_{inc}$$

As a result, the suction and discharge enthalpy flow rate are equal to:

$$dmh_1 = -dm_2 \cdot h_1(p_1, T_1)$$

$$dmh_2 = -dmh_1 + dm_2 \cdot h_{inc}$$

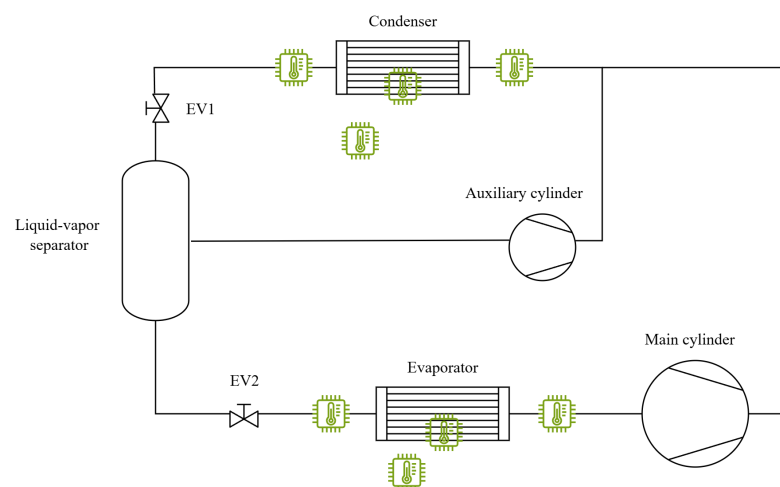


Figure 3. Schematic diagram of simulation model.

2. Calculation of Heat Exchanger Parameters

A set of empirical correlations were adopted to calculate heat transfer coefficients under single-phase flow, condensation and boiling conditions. For fully developed laminar flow in tubes, the Nusselt number takes a fixed theoretical value:

$$Nu_{\text{laminar}} = 3.66 \quad (1)$$

The Gnielinski correlation was used for single-phase flow, which is applicable for $2300 \leq Re \leq 5 \times 10^6$ and $0.5 \leq Pr \leq 2000$. The expression is written as:

$$Nu = \frac{(f/8)(Re - 1000)Pr}{1 + 12.7\sqrt{f/8}(Pr^{2/3} - 1)} \quad (2)$$

where Re is the Reynolds number, Pr is the Prandtl number and f denotes the tube friction factor.

For condensation heat transfer, the Shah correlation combined with the McAdams definition of Nusselt number was selected. The overall Nusselt number for condensation is corrected by a two-phase factor:

$$Nu = Nu_l \cdot F \quad (3)$$

where Nu_l is the liquid-phase Nusselt number calculated by the Gnielinski correlation, and F is the correction factor related to refrigerant vapor quality, mass flux and thermo-physical properties.

The VDI correlation for horizontal tubes was applied to describe the boiling process. The total boiling heat transfer coefficient is calculated as the cubic mean of nucleate boiling and convective boiling components:

$$h = \sqrt[3]{h_{\text{nb}}^3 + h_{\text{cb}}^3} \quad (4)$$

where h_{nb} and h_{cb} represent the heat transfer coefficients of nucleate boiling and convective boiling, respectively. The liquid-phase baseline coefficient for boiling calculation also follows the Gnielinski correlation. The heat transfer gain factor was set to 1.0 in the simulation, indicating no additional modification to the calculated heat transfer coefficients.

The local convective heat transfer coefficient is derived by $h = Nu \cdot k/D$. The overall heat transfer coefficient U is obtained via the thermal resistance superposition method:

$$U = \frac{1}{\frac{1}{h_i} + R_w + \frac{1}{h_o}} \quad (5)$$

where h_i and h_o are the internal and external heat transfer coefficients, and R_w is the thermal resistance of the tube wall. Finally, the heat transfer rate is calculated by the logarithmic mean temperature difference (LMTD) method:

$$Q = U \cdot A \cdot \Delta T_{\text{lm}} \quad (6)$$

in which A is the effective heat transfer area and ΔT_{lm} is the logarithmic mean temperature difference between hot and cold fluids.

3. Calculation of Pressure Drop

The total pressure drop is solved by satisfying the equilibrium between the pressure difference imposed at two ends and the integrated flow resistance along the flow path. The governing equation for mass flow rate calculation is expressed as:

$$\Delta P + \int_0^L \frac{dP}{dz} dz = 0 \quad (7)$$

where ΔP is the pressure difference between the inlet and outlet of the heat exchanger, L is the total flow length and dP/dz is the total static pressure gradient along the flow axis.

The total static pressure gradient is decomposed into three components, namely, frictional pressure drop, accelerational pressure drop and gravity contribution, shown as:

$$\frac{dP}{dz} = \left(\frac{dP}{dz}\right)_F + \left(\frac{dP}{dz}\right)_a + \left(\frac{dP}{dz}\right)_z \quad (8)$$

where $(dP/dz)_F$ denotes the frictional pressure drop gradient, $(dP/dz)_a$ denotes the accelerational pressure drop gradient induced by fluid density variation, and $(dP/dz)_z$ denotes the gravity contribution caused by altitude variation.

For the superheated zone at the evaporator outlet and the subcooled zone at the condenser outlet, the refrigerant presents a single-phase flow state. The regular frictional coefficient is calculated by the Churchill correlation. The expression of the Fanning friction factor f_f is written as:

$$f_f = 8 \cdot \left[\left(\frac{8}{Re}\right)^{12} + \frac{1}{\left([2.457 \cdot \ln\left(\left(\frac{7}{Re}\right)^{0.9} + 0.27 \cdot \frac{\varepsilon}{D}\right)\right]^{16} + \left(\frac{37530}{Re}\right)^{16}\right)^{\frac{3}{2}}} \right]^{\frac{1}{12}} \quad (9)$$

where Re is the Reynolds number of single-phase flow, ε is the absolute roughness of the tube wall and D is the inner diameter of the heat exchange tube.

The frictional pressure drop gradient in the single-phase zone is derived from the Darcy–Weisbach framework:

$$-\left(\frac{dP}{dz}\right)_{F,sp} = \frac{f_f \cdot G^2 \cdot v}{2D} \quad (10)$$

where G is the mass velocity of the refrigerant and v is the specific volume of the single-phase refrigerant.

For the two-phase zone where refrigerant phase change occurs, the homogeneous flow model is adopted in this work, and the McAdams correlation is employed to calculate the two-phase viscosity.

Under the homogeneous assumption, the average specific volume of the two-phase mixture is calculated by:

$$\bar{v} = x \cdot v_v + (1 - x) \cdot v_l \quad (11)$$

where x is the vapor quality and v_v and v_l are the specific volumes of saturated vapor and saturated liquid, respectively.

The two-phase mixture viscosity is defined by the McAdams correlation:

$$\frac{1}{\bar{\mu}} = \frac{x}{\mu_v} + \frac{1 - x}{\mu_l} \quad (12)$$

where μ_v and μ_l are the dynamic viscosities of saturated vapor and saturated liquid, respectively.

Accordingly, the frictional pressure drop gradient of two-phase flow is calculated in the same form as single-phase flow, with fluid properties replaced by homogeneous mixture properties:

$$-\left(\frac{dP}{dz}\right)_{F,tp} = \frac{f_{f,tp} \cdot G^2 \cdot \bar{v}}{2D} \quad (13)$$

where $f_{f,TP}$ is the two-phase friction factor, which is solved by the Churchill correlation using the homogeneous mixture Reynolds number.

The accelerational pressure drop arises from the variation in fluid density along the flow direction. Derived from the momentum conservation equation, the general expression of the accelerational pressure drop gradient is:

$$-\left(\frac{dP}{dz}\right)_a = \frac{1}{A} \cdot \frac{d}{dz}[dm_v \cdot u_v + dm_l \cdot u_l] \quad (14)$$

where A is the cross-sectional area of the flow channel, dm_v and dm_l are the mass flow rates of vapor and liquid phases, and u_v and u_l are the flow velocities of the vapor and liquid phases. Under the homogeneous flow assumption, the velocities of the two phases are equal, and the accelerational term can be simplified with homogeneous mixture properties.

The gravity contribution to the pressure gradient is caused by the altitude variation of the fluid in the flow element, which reflects the conversion between static pressure and gravitational potential energy. The sign of the gravity term depends on the flow direction: positive for upward flow and negative for downward flow.

In the homogeneous flow framework, the gravity pressure drop gradient is expressed as:

$$-\left(\frac{dP}{dz}\right)_z = g \cdot \sin \theta \cdot [x \cdot \rho_v + (1 - x) \cdot \rho_l] \quad (15)$$

where g is the gravitational acceleration, θ is the inclination angle of the tube and ρ_v and ρ_l are the densities of saturated vapor and saturated liquid, respectively.

The singular friction factor of the bend is determined by the center angle of the bend, the curvature radius and the hydraulic diameter. Meanwhile, the regular frictional resistance of the bent section is also calculated according to the developed length of the bend, tube diameter and wall roughness, consistent with the straight tube calculation method. The total pressure drop of the bent tube is the superposition of the regular frictional pressure drop and the singular local pressure drop.

Figure 4 illustrating the calculation procedure of the simulation model.

2.3. Experimental Platform and Model Validation

The experiments were conducted in an enthalpy difference laboratory to precisely evaluate the COP of a room air conditioner employing the independent compression. The laboratory environment allowed for precise control of temperature and humidity, ensuring the accuracy of the test conditions. The geometric parameters of the condenser and evaporator are shown in Table 1.

The experimental setup utilized an R290 variable-frequency air conditioning system with a refrigerant charge of 320 g and a rated cooling capacity of 3.5 kW. Operational data, including compressor speed, power consumption, system voltage and current, were acquired via the air conditioner's electronic control board. System component temperatures were measured using temperature sensors. These data are critical for assessing the energy efficiency of the proposed independent compression system and for validating the accuracy of the air conditioning system model. The schematic diagram of the experimental platform is shown in Figure 5.

This study experimentally investigates the operational characteristics of an R290 air conditioner with independent compression under both steady-state and dynamic conditions. Test conditions were established by combining setting indoor temperature with specific ambient temperatures, as detailed in Table 2. For these tests, the air conditioner operated continuously until reaching a steady state before being shut down, with outdoor temperature maintained at constant levels throughout. In the investigation of the

system's dynamic startup process, the outdoor temperature is held constant. Operational data—including compressor speed, voltage, current, power consumption and temperature and humidity readings from sensors—were recorded at 6 s intervals immediately after startup. After 15 min of operation, the sampling interval was extended to 30 s.

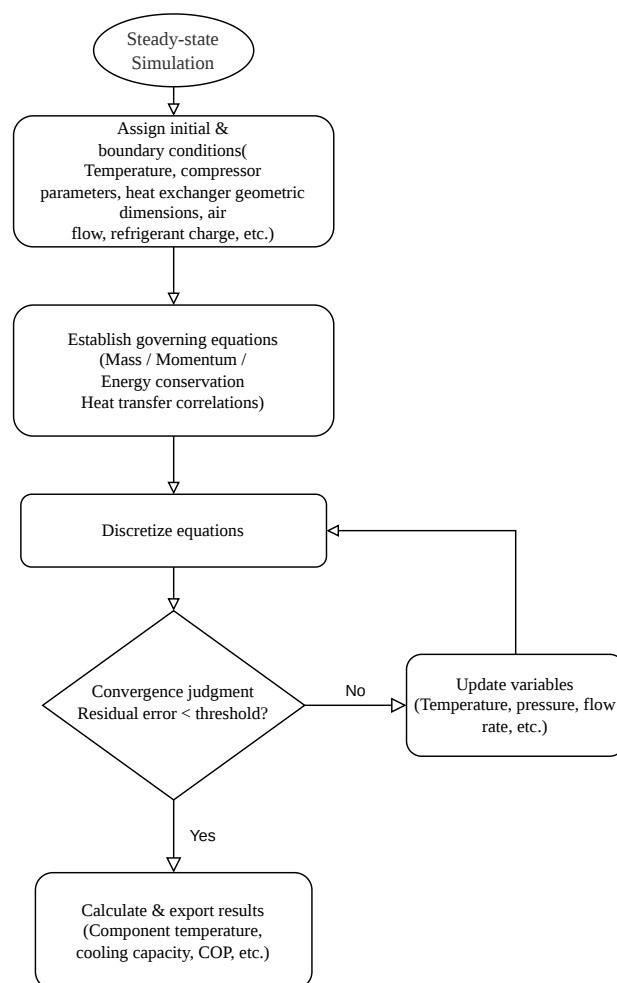


Figure 4. Flowchart of the simulation model.

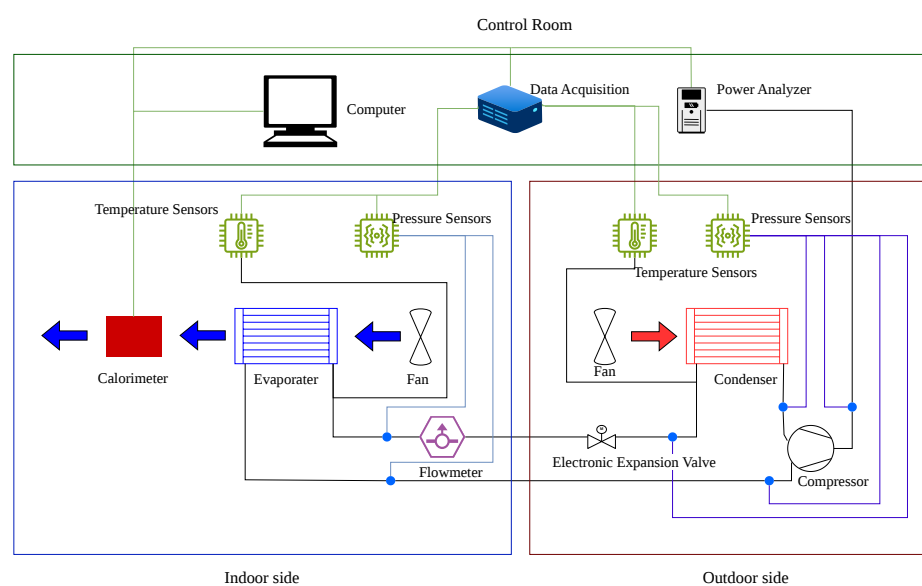


Figure 5. Schematic diagram of experimental platform.

Table 1. Geometric parameters of condenser and evaporator.

	Condenser	Evaporator
Tube material	Copper	Copper
Number of tube row	1	2
Tube length (mm)	870	675
Tube outer diameter (mm)	7	5
Tube inner diameter (mm)	6.54	4.6
Tube Spacing (mm)	21	19.5
Numbers of tubes	24	32

Table 2. Experimental working conditions of system (refrigerant charge: 320 g).

Conditions	Mode	Indoor Temperature DB (°C)/WB (°C)	Outdoor Temperature DB (°C)/WB (°C)	Compressor Frequency (Hz)
1	Cooling	22/15	15/10	49
2	Cooling	22/15	20/15	67
3	Cooling	22/15	25/20	68
4	Cooling	22/15	30/24	75
5	Cooling	22/15	35/24	79
6	Cooling	22/15	40/26	75
7	Cooling	22/15	45/32	56
8	Cooling	22/15	50/32	30
9	Cooling	27/19	50/32	31
10	Cooling	27/19	45/32	57
11	Cooling	27/19	40/26	75
12	Cooling	27/19	35/24	80
13	Cooling	27/19	32/24	79
14	Cooling	27/19	25/20	68
15	Cooling	27/19	20/15	68
16	Cooling	27/19	15/10	50
17	Cooling	32/23	15/10	80
18	Cooling	32/23	20/15	80
19	Cooling	32/23	25/20	80
20	Cooling	32/23	30/25	76
21	Cooling	32/23	35/22	80
22	Cooling	32/23	40/23	75
23	Cooling	32/23	45/25	57
24	Cooling	32/23	50/27	30
25	Heating	27/24	24/15	36
26	Heating	27/24	20/12	44
27	Heating	27/24	15/8	55
28	Heating	27/24	10/6	85
29	Heating	27/24	5/3	32
30	Heating	24/18	25/20	36
31	Heating	24/18	20/15	45
32	Heating	24/18	15/10	55
33	Heating	24/18	10/5	86
34	Heating	24/18	5/4	99
35	Heating	16/10	5/4	100
36	Heating	16/10	10/8	85
37	Heating	16/10	15/10	55
38	Heating	16/10	20/15	45
39	Heating	16/10	25/20	35

To ensure the simulation model accurately replicates the air conditioner's energy consumption under operating conditions, a series of experiments simulating diverse ambient temperatures and system loads were conducted in an enthalpy difference laboratory. The outdoor unit environment temperature was varied across the range of 15 °C to 50 °C.

A total of 39 experimental cases were executed. These tests encompassed both representative operational scenarios and extreme conditions encountered in practical use, thereby imposing a comprehensive spectrum of thermal loads on the system. The acquired dataset provides a robust basis for evaluating performance under realistic operating conditions.

The simulation model is quantitatively validated from three error indicators: mean absolute error (MAE), mean relative error (MRE) and maximum absolute error. The MAE quantifies the average absolute deviation between experimental and simulated values, defined as:

$$\text{MAE} = \frac{1}{n} \sum_{i=1}^n |y_{\text{exp},i} - y_{\text{sim},i}| \quad (16)$$

where:

- $y_{\text{exp},i}$: Experimental data under the i -th working condition,
- $y_{\text{sim},i}$: Simulated data under the i -th working condition,
- n : Total number of working conditions ($n = 24$).

The MRE normalizes the absolute error by the experimental value to reflect the relative deviation, expressed as:

$$\text{MRE} = \frac{1}{n} \sum_{i=1}^n \left| \frac{y_{\text{exp},i} - y_{\text{sim},i}}{y_{\text{exp},i}} \right| \times 100\% \quad (17)$$

The maximum absolute error identifies the largest single deviation between the experimental and simulated results:

$$\text{Max_AE} = \max(|y_{\text{exp},i} - y_{\text{sim},i}|) \quad (18)$$

The error calculation is divided into cooling mode (working conditions 1–24) and heating mode (working conditions 25–39), and the statistical results are shown in Tables 3 and 4.

Table 3. Error analysis of simulation results under cooling mode (Conditions 1–24).

Parameter	MAE	MRE (%)	Maximum Absolute Error
Exhaust temperature	5.48 °C	N/A	9.10 °C
Condenser outlet temperature	3.35 °C	N/A	7.60 °C
Evaporator inlet temperature	2.73 °C	N/A	4.10 °C
Evaporator outlet temperature	3.15 °C	N/A	5.00 °C
Cooling capacity	232.62 W	7.09	771.10 W

Table 4. Error analysis of simulation results under heating mode (Conditions 25–39).

Parameter	MAE	MRE (%)	Maximum Absolute Error
Exhaust temperature	3.91 °C	N/A	5.50 °C
Condenser outlet temperature	2.48 °C	N/A	4.50 °C
Evaporator inlet temperature	2.62 °C	N/A	4.20 °C
Evaporator outlet temperature	3.04 °C	N/A	4.30 °C
Heating capacity	208.55 W	5.76	440.80 W

The accuracy of the established simulation model is validated by comparing the simulated results with experimental data under both cooling and heating modes, as summarized in Figures 6 and 7. Under cooling mode (Conditions 1–24), the MAE of the key temperature parameters, including exhaust temperature, condenser outlet temperature and evaporator inlet/outlet temperatures, is within 5.48 °C, with the maximum absolute error not exceeding 9.10 °C. The cooling capacity prediction shows an MRE of 7.09%, corresponding to

an MAE of 232.62 W and a maximum absolute error of 771.10 W. Under heating mode (Conditions 25–39), the MAE of the temperature parameters is within 3.91 °C, and the MRE of heating capacity is 5.76% (MAE = 208.55 W, maximum absolute error = 440.80 W). The model can accurately predict the steady-state operating characteristics for the independent compression under both cooling and heating modes, with the MRE of cooling/heating capacity below 10%.

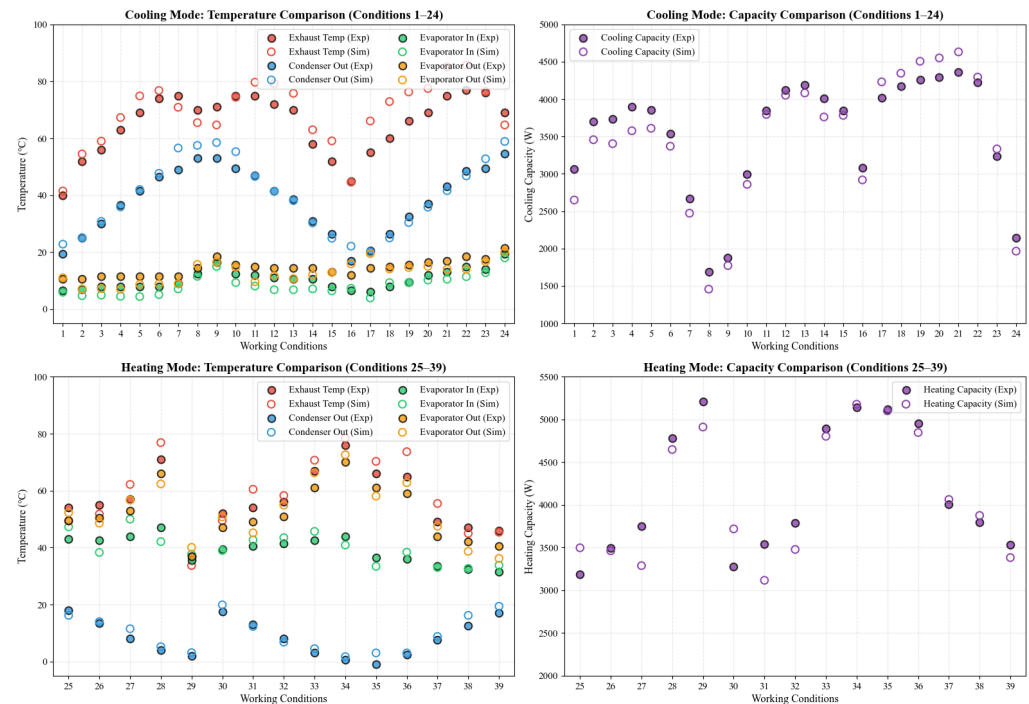


Figure 6. Experimental vs. simulated data comparison.

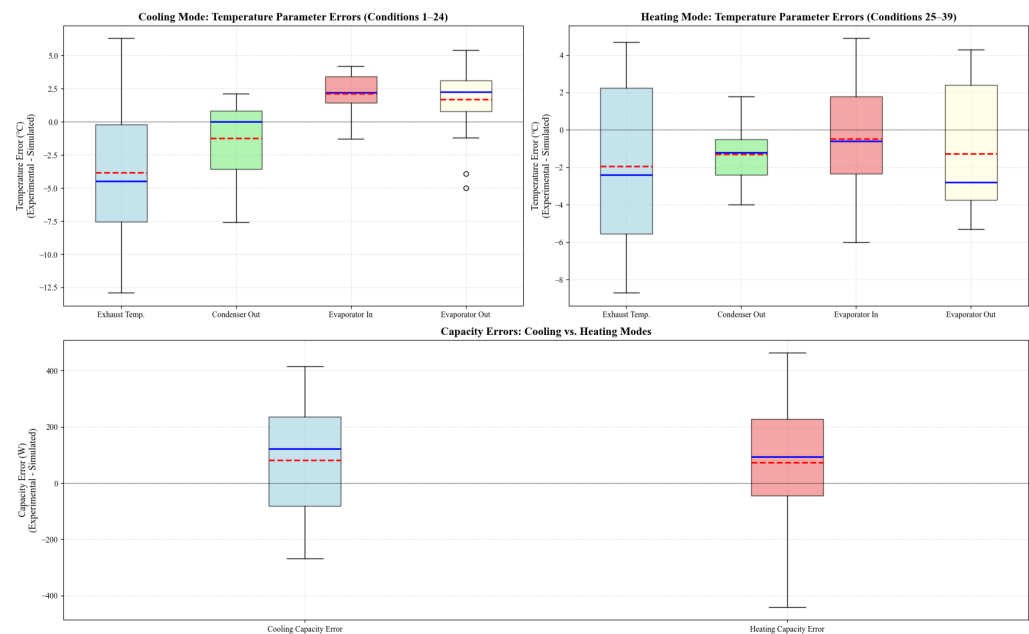


Figure 7. Boxplot of temperature and capacity (The solid blue line denotes the median error, and the red dashed line denotes the mean error).

3. Results and Discussions

3.1. Dynamic Operating Performance

The accuracy of the dynamic simulation was quantified using the following statistical metrics:

- Mean Absolute Error (MAE): $MAE = \frac{1}{n} \sum_{i=1}^n |y_i^{\text{exp}} - y_i^{\text{sim}}|$
- Root Mean Square Error (RMSE): $RMSE = \sqrt{\frac{1}{n} \sum_{i=1}^n (y_i^{\text{exp}} - y_i^{\text{sim}})^2}$
- Mean Absolute Percentage Error (MAPE): $MAPE = \frac{100\%}{n} \sum_{i=1}^n \left| \frac{y_i^{\text{exp}} - y_i^{\text{sim}}}{y_i^{\text{exp}}} \right|$
- Coefficient of Determination (R^2): $R^2 = 1 - \frac{\sum_{i=1}^n (y_i^{\text{exp}} - y_i^{\text{sim}})^2}{\sum_{i=1}^n (y_i^{\text{exp}} - \bar{y}^{\text{exp}})^2}$
- Bias: $Bias = \frac{1}{n} \sum_{i=1}^n (y_i^{\text{sim}} - y_i^{\text{exp}})$

Where y_i^{exp} and y_i^{sim} represent experimental and simulated values at time i , respectively, and n is the number of data points.

Figure 8 illustrates the temporal evolution of simulation accuracy throughout the 600 s operational period. The simulation demonstrates better accuracy during steady-state operation (300–600 s) compared to transient phases (0–300 s), as evidenced by reduced error magnitudes in later time periods.

The dynamic simulation model demonstrates a reasonable ability to predict the operational characteristics, as shown in Table 5. Accuracy data indicate that the model's predicted trends for key system temperature parameters closely match the actual measured values. To further verify the reliability of the dynamic simulation model from the perspective of system energy balance, the system active power during the start-up process is selected as a validation indicator. The comparison between simulation results and experimental measurements is shown in Figure 9.

Table 5. Accuracy metrics for different parameters.

Parameter	MAE	RMSE	MAPE (%)	R^2	Bias
Exhaust Temperature (°C)	3.27	3.84	N/A	0.817	−2.86
Condenser Outlet Temperature (°C)	2.00	2.28	N/A	0.842	+0.60
Evaporator Inlet Temperature (°C)	1.67	2.14	N/A	0.913	−0.95
Evaporator Outlet Temperature (°C)	3.77	4.11	N/A	0.752	+4.18
Power Consumption (W)	202.95	274.10	32.34	0.535	+119.44

The overall change rhythm is basically consistent with the experimental results, but the simulated power starts to rise earlier in the initial stage, as the model simplifies the standby soft-start process of the system. Quantitative accuracy metrics are calculated based on the full 600 s start-up process, as shown in Table 5. The relatively large full-cycle error is mainly concentrated in the initial transient stage (0–40 s): the experimental system maintains a low standby power of 252 W at this stage, while the simulation directly enters the compressor loading process, resulting in a large relative deviation in the low-power interval. When focusing on the steady-state operation stage (300–600 s), the absolute error of steady-state power is controlled within 130 W, with a relative error of 7–9%. In general, the dynamic simulation model can accurately capture the stepwise variation trend of system power and the steady-state power level. The capability to capture the overall dynamic variation patterns of the system is sufficient. This makes the model suitable for system performance trend analysis and optimization design.

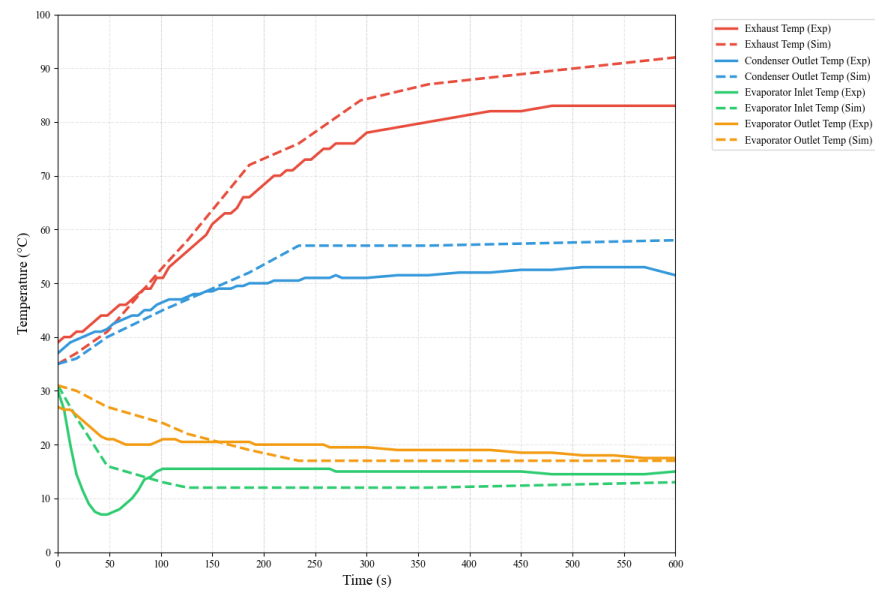


Figure 8. Experimental vs. simulated temperature data comparison of dynamic simulation.

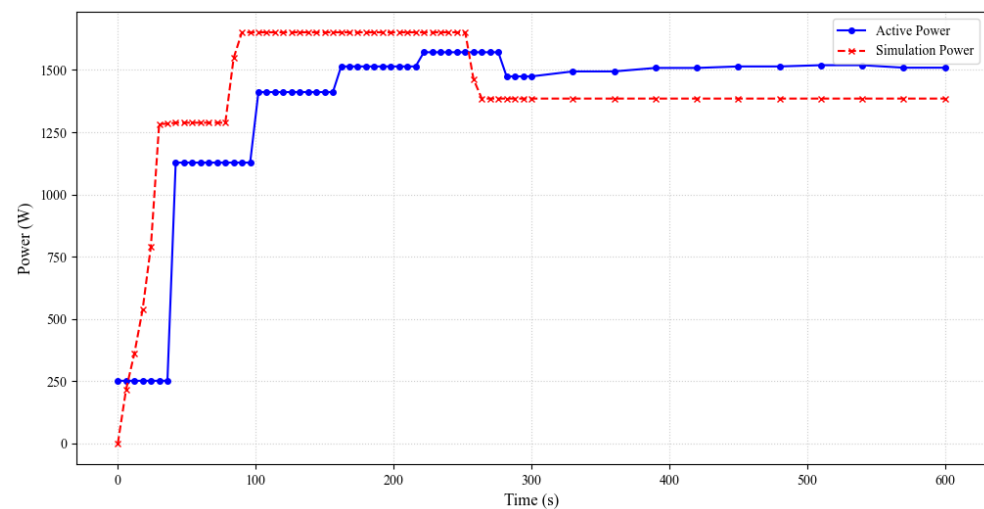


Figure 9. Power consumption comparison of dynamic simulation.

3.2. Influence of Intermediate Temperature on COP

Combined with Figure 10, the variation characteristics of COP under different evaporation temperatures and intermediate temperatures are further analyzed in this section. At each fixed evaporation temperature, the system COP shows a consistent single-peak trend with the rise in intermediate temperature. When the intermediate temperature is lower than the optimal value, the pressure ratio of the low-pressure stage increases, leading to larger compression power consumption and reduced COP. As the intermediate temperature rises to the theoretical optimal value, the system obtains the maximum COP. When the intermediate temperature continues to increase beyond the optimal point, the high-pressure stage bears an excessive pressure ratio, which significantly increases power consumption and causes a gradual decline in COP. Meanwhile, the evaporation temperature exerts a remarkable influence on the overall performance level and optimal matching parameters. With the evaporation temperature increasing from 6 °C to 14 °C, the optimal intermediate temperature shifts obviously to a higher temperature, and the peak COP rises steadily. A low evaporation temperature will result in lower energy efficiency and a more obvious performance attenuation when the intermediate temperature deviates. By contrast, a higher

evaporation temperature effectively improves the operating efficiency, and maintains a relatively high COP within a wider intermediate temperature range. The above results indicate that reasonable matching of intermediate temperature is essential to maintain the efficient operation of the R290 independent compression system.

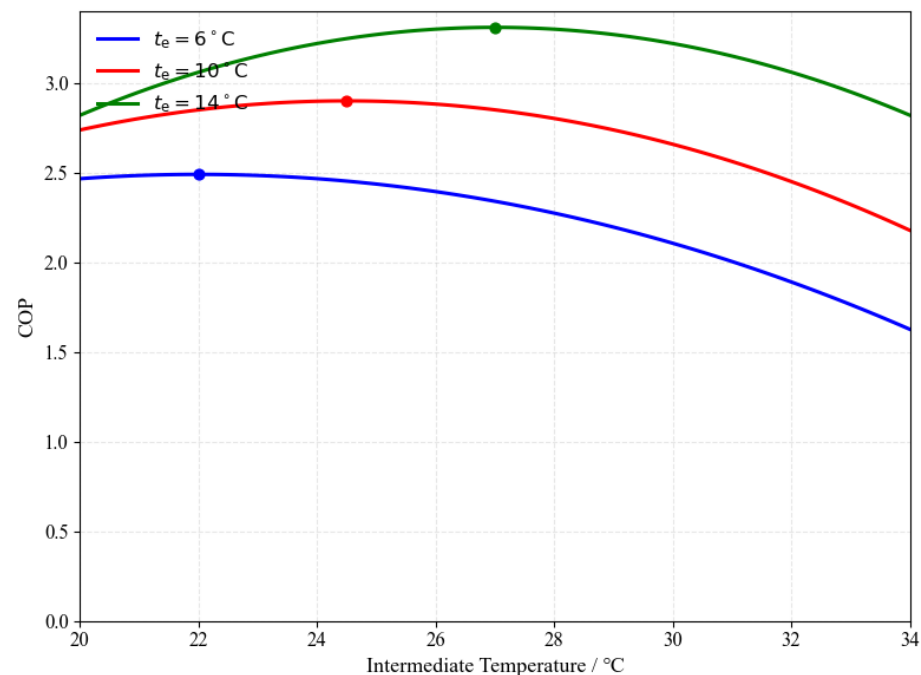


Figure 10. COP variation with intermediate temperature for different evaporation temperatures.

3.3. Comparison Between Independent Compression and Single-Stage Compression

The cooling/heating capacity and COP of independent compression and single-stage compression is systematically compared under 39 groups of full operating conditions, as shown in Figures 11 and 12. The results demonstrate that the independent compression comprehensively outperforms the single-stage compression in both cooling/heating capacity and energy efficiency. Figure 13 compares the performance enhancement of the independent compression system against the single-stage compression system across 39 operating conditions, with separate statistics for cooling mode (24 conditions) and heating mode (15 conditions). In cooling operation, the independent compression system delivers an average cooling capacity improvement of 4.8% and an average COP improvement of 7.8%. The maximum capacity improvement reaches 6.9% at Condition 14, and the maximum COP improvement reaches 11.5% at Condition 8. Even under the least favorable condition, the improvements in capacity and COP remain at 3.5% and 5.0%. In heating operation, the average improvement in heating capacity rises to 8.3%, with an average COP improvement of 7.5%. The maximum heating capacity improvement reaches 16.6% (Condition 32) and the maximum COP improvement reaches 11.1% (Condition 29). These results demonstrate that the independent compression cycle can consistently boost both system capacity and energy efficiency across a wide range of operating conditions.

Figure 14 further reveals the internal mechanism of the performance improvement from the perspective of cycle characteristics, focusing on three typical rated cooling capacity conditions (3000 W, 3500 W and 4000 W). Under identical cooling capacity requirements, the independent compression system consistently delivers a higher refrigerant mass flow rate than the single-stage system. The mass flow rate improvement reaches 15.3%, 11.3% and 11.0% for the 3000 W, 3500 W and 4000 W conditions, respectively, with an average increment of 12.5%. This improvement originates from the suppressed suction pressure

drop in the independent compression architecture: lower flow resistance raises the actual suction pressure and refrigerant density at the compressor inlet, thereby enhancing the effective gas delivery capacity of the compressor. In terms of compressor displacement demand, the independent compression requires a smaller total compressor displacement to achieve the same cooling output. The required displacement is reduced by 9.5%, 5.5% and 4.8% for the three working conditions, respectively, with an average reduction of 6.6%. This indicates that the independent compression configuration enables a more compact compressor design and lower hardware cost while maintaining equivalent cooling performance. Regarding flow resistance loss, the independent compression achieves a significant reduction compared with the single-stage compression, with an average decrease of 33.0% across all tested conditions. The reduction rate rises from 25.0% at 3000 W to 38.5% at 3500 W, and remains at 35.5% at 4000 W. This trend confirms that the split-flow design of the independent compression effectively mitigates the sharp increase in flow velocity inside the evaporator.

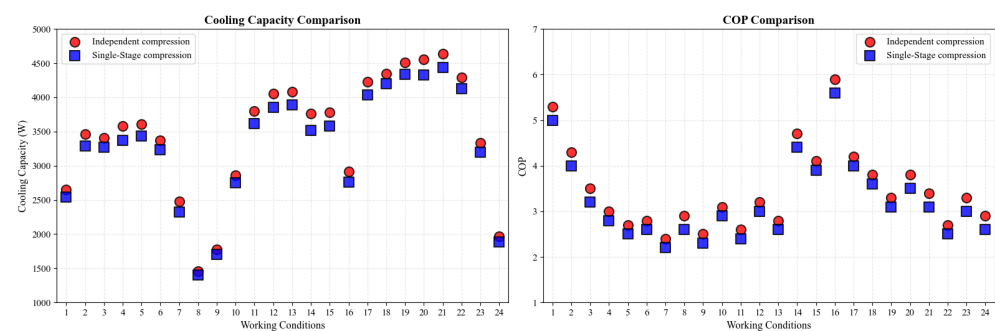


Figure 11. Cooling mode performance comparison.

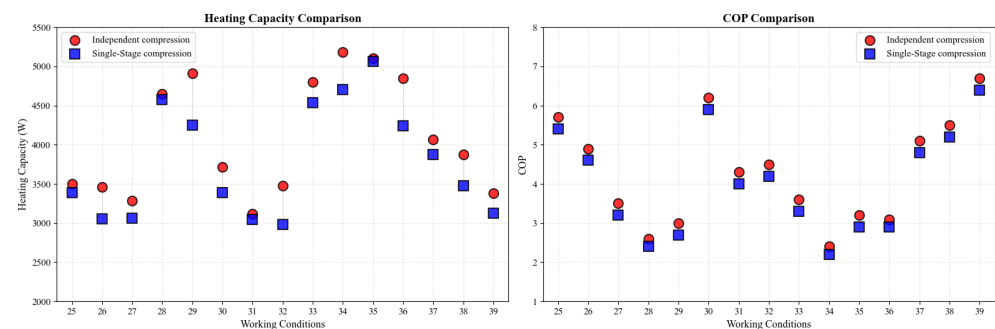


Figure 12. Heating mode performance comparison.

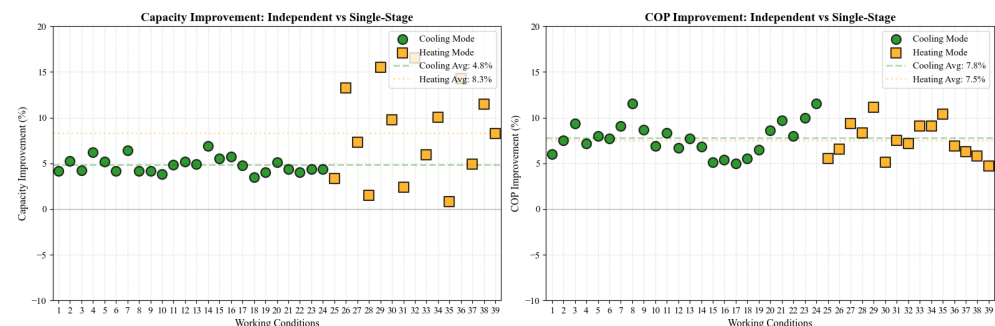


Figure 13. Performance improvement comparison.

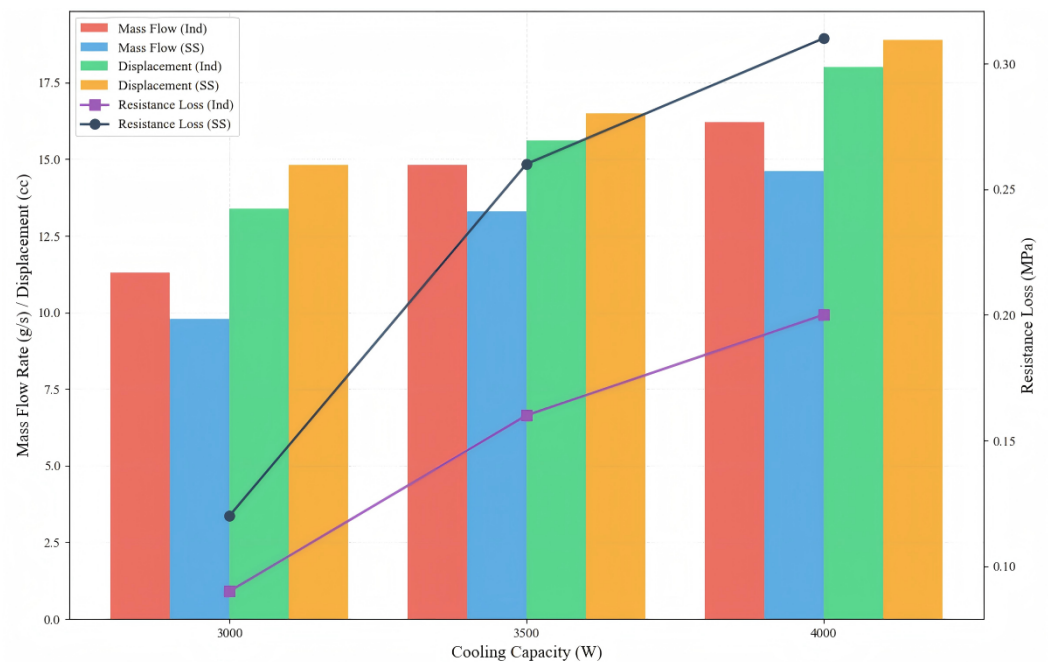


Figure 14. Performance comparison of compression cycles.

3.4. Comparison with Transcritical CO₂ Heat Pump Systems

R290 and CO₂ adopt differentiated technical routes. Combined with existing studies on CO₂ heat pump drying systems and simultaneous heating and cooling systems [11,12], this section provides a targeted comparison to clarify the technical positioning of the proposed R290 independent compression system. The two systems differ significantly in operating pressure and engineering cost. CO₂ transcritical systems usually operate at pressures above 75 bar, requiring high-pressure-resistant components and sealing structures, which brings higher initial investment [11]. In contrast, the R290 system maintains a medium-low operating pressure under conventional air-conditioning conditions, with good compatibility with existing air-conditioning hardware and lower system transformation cost, which is more adaptable to residential room air conditioner applications.

The advantageous application scenarios are complementary. CO₂transcritical systems show prominent performance advantages in high-temperature scenes: relevant studies have verified that CO₂ heat pump dryers can reach a system COP of 2.31, and simultaneous heating and cooling systems maintain a comprehensive COP of around 3.27, which are suitable for industrial drying and commercial combined energy supply scenarios [11,12]. The R290 independent compression system delivers significant energy efficiency gains within the conventional operating temperature range of room air conditioners, with a simpler system structure and good mass production feasibility. In general, the two natural refrigerant systems are suited for different application scenarios. The R290 independent compression proposed in this study provides a feasible optimization path for natural refrigerant replacement in the residential air-conditioning field.

4. Conclusions

This study systematically investigates the performance of an independent compression refrigeration cycle for air-conditioning systems, covering experimental platform construction, steady-state/dynamic simulation and comparative analysis with the conventional single-stage compression cycle. Key conclusions drawn from the research are summarized as follows:

(1) Steady-state simulations were conducted with the simulation results; the MAE of the temperature parameters was below 5.5 °C, the MRE of cooling capacity was 7.09% and the MRE of heating capacity was 5.76%. Transient simulations of the system start-up process revealed that the model accurately captures the dynamic evolution of key parameters from shutdown to steady state. The dynamic MAE of the exhaust temperature, evaporator outlet temperature, condenser outlet temperature, evaporator inlet temperature and evaporator outlet temperature were 3.27 °C, 2.00 °C, 1.67 °C and 3.77 °C.

(2) At each fixed evaporation temperature, the system COP shows a consistent single-peak trend with the rise in intermediate temperature. When the intermediate temperature drops below the optimal value, the compression power consumption increases and the COP decreases accordingly.

(3) The comparative analysis across 39 operating conditions verifies the stable performance advantages of the independent compression over the single-stage cycle in both cooling and heating modes. In cooling mode, the independent compression system achieves average improvements of 4.8% in cooling capacity and 7.8% in COP; in heating mode, the average improvements reaches 8.3% in heating capacity and 7.5% in COP. The performance benefit is essentially attributed to the split-flow, which reduces the system flow resistance loss by 33.0% on average under 3000 W, 3500 W and 4000 W conditions. The independent compression improves the compressor suction density and refrigerant mass flow rate by 12.5% on average, and decreases the required compressor displacement by 6.6% on average under equivalent cooling output. This indicates that the independent compression configuration can simultaneously elevate system energy efficiency and reduce compressor size, presenting favorable engineering application potential.

As an efficient vapor compression refrigeration cycle configuration, the independent compression addresses the core long-standing limitations of conventional single-stage compression, including significant flow resistance, attenuated effective mass flow rate and bottlenecks in improving refrigeration energy efficiency. Nevertheless, the conclusions are drawn based on the specific compressor type and cycle architecture. For future work, detailed two-phase flow models will be introduced to enhance transient simulation performance. The research scope will be extended to more compressor types and cycle configurations for broader conclusions. R290 flammability risk assessment under typical fault conditions will be carried out to support practical engineering applications.

Author Contributions: Conceptualization, T.L.; methodology, J.H.; software, J.H.; validation, J.H.; formal analysis, J.H.; investigation, J.H. and T.L.; resources, T.L.; data curation, J.H. and T.L.; writing—original draft, J.H.; writing—review & editing, J.H. and T.L.; visualization, J.H.; supervision, T.L.; project administration, T.L.; funding acquisition, T.L. All authors have read and agreed to the published version of the manuscript.

Funding: This research was funded by the National Natural Science Foundation of China (NSFC) under Grant No. 52176031 and the Technical Assistance Project under the Multilateral Fund for the Implementation of the Montreal Protocol, Ministry of Environmental Protection of China, under Project No. C/III/S/21/89.

Data Availability Statement: The original contributions presented in this study are included in the article.

Conflicts of Interest: The authors declare no conflicts of interest. The funders had no role in the design of the study; in the collection, analyses, or interpretation of data; in the writing of the manuscript; or in the decision to publish the results.

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