

Article

Experimental Investigation on Refrigerant Charge Optimization of Vapor Compression Refrigeration System Driven by Oil-Free Linear Compressors

Xueliang Fang ^{1,*} and Xinwen Chen ²¹ College of Traffic Engineering, Yangzhou Polytechnic Institute, Yangzhou 225127, China² College of Mechanical Engineering, Yangzhou University, Yangzhou 225127, China; xinwen.chen@yzu.edu.cn

* Correspondence: tcfangxueliang@163.com

Abstract

Vapor compression refrigeration systems account for a substantial share of global electricity consumption in residential and commercial applications, with environmental impacts arising from both energy use and refrigerant leakage. Refrigerant charge optimization offers an economical means of improving system performance without hardware modifications. Oil-free linear compressors mitigate lubricant-induced degradation of heat transfer, yet the combined influence of charge amount on the coefficient of performance (COP) and total equivalent warming impact (TEWI) has not been thoroughly quantified. An experimental investigation was conducted on a vapor compression refrigeration system equipped with an oil-free linear compressor using R134a. The experiments covered refrigerant charges of 220–330 g, piston strokes of 9–12 mm, and pressure ratios of 2.0–3.5. Component-level refrigerant distribution and system performance characteristics were analyzed systematically. The condenser holds 74.7% of the total refrigerant charge at the optimal charge of 280 g. Rising charge reduces superheat and increases subcooling, both of which serve as practical indicators of the charge level. The mass flow rate, cooling capacity, and COP all exhibit characteristic non-monotonic trends. The maximum COP of 4.67 and the maximum cooling capacity of 472.7 W are both achieved at 280 g, which is identified as the optimal operating condition. The oil-free design eliminates lubricant interference and yields a clearly condenser-dominated refrigerant distribution. The TEWI increases by only 3.6% when the charge is raised to 330 g, and this slight environmental drawback is offset by the gain in energy efficiency. A distinct COP reduction is observed at a charge of 220 g. The charge of 280 g achieves the best balance between energy efficiency and lifecycle CO₂ emissions. This work provides quantitative guidance for charge selection in oil-free linear compressor refrigeration systems.



Academic Editor: Nikola Stosic

Received: 20 May 2026

Revised: 20 June 2026

Accepted: 24 June 2026

Published: 27 June 2026

Copyright: © 2026 by the authors.

Licensee MDPI, Basel, Switzerland.

This article is an open access article distributed under the terms and conditions of the [Creative Commons Attribution \(CC BY\)](https://creativecommons.org/licenses/by/4.0/) license.

Keywords: vapor compression refrigeration; refrigerant charge; linear compressor; COP; TEWI

1. Introduction

Vapor compression refrigeration systems account for a considerable share of global electricity consumption in residential and commercial buildings, and their environmental impact extends well beyond energy use to include direct greenhouse gas emissions from refrigerant leakage [1–4]. Among the practical strategies proposed for improving system performance, charge optimization has attracted significant attention because it can

simultaneously enhance energy efficiency and reduce climate impact without requiring major hardware modifications. Unlike component redesign or the adoption of alternative refrigerants, charge adjustment can be applied directly to existing installations, offering a cost-effective and scalable pathway for performance improvement. However, the refrigerant charge exerts strongly nonlinear effects on system operational characteristics, and these effects depend heavily on compressor specifications, heat exchanger geometry, and the prevailing operating conditions [5–7]. Consequently, identifying the optimal charge demands a thorough understanding of how charge variations redistribute the working fluid and alter the thermodynamic state within the system.

The amount of refrigerant circulating in a vapor compression system directly governs the distribution of the working fluid among the condenser, evaporator, and interconnecting piping, which in turn dictates the heat transfer performance, pressure levels, and compressor power draw [8,9]. Experimental investigations have consistently demonstrated that the subcooling at the condenser outlet provides a sensitive and practical indication of the charge level, whereas overcharging leads to condenser flooding and a marked degradation of system performance [10–12]. These findings highlight the value of charge-sensitive parameters for diagnostics, yet they also reveal the need for a more refined, component-level understanding of refrigerant distribution, especially in systems that deviate from conventional compressor architectures.

Notably, the aforementioned experimental studies on refrigerant charge were conducted using conventional oil-lubricated compressors. The presence of lubricating oil affects the experimental results in several ways. The most significant efficiency losses caused by lubricating oil occur in the compressor, with COP reduced by nearly 15% [13]. Li et al. developed numerical models to explore the effects of lubricating oil on flow, heat transfer, and overall system performance [14]. Their findings revealed that, compared with oil-lubricated systems, oil-free systems exhibit a 15–28% higher COP and a 36–60% lower pressure loss [14]. Lubricating oil inevitably affects heat transfer in both the evaporator and condenser, and it also increases the refrigerant charge required by the system. Youbilidrissi et al. reported that under low superheat conditions, more than 20% of the refrigerant can dissolve in the lubricating oil [15]. This dissolved refrigerant has the most significant impact on the total refrigerant charge in the compressor, highlighting the need to minimize the amount of lubricating oil used [16].

Linear compressors differ fundamentally from conventional reciprocating machines. For domestic refrigerators using conventional compressors with R134a as the refrigerant, the COP typically ranges from 1.5 to 3.5 [7,17–19]. Belman-Flores et al. measured a TEWI value above 3150 kgCO₂e for conventional refrigerators in Mexico [20]. In contrast, when a linear compressor is applied in a household refrigerator, the system achieves a higher COP than that of traditional compressor-driven systems, with COP values reaching up to 7.12 depending on the compressor model [21]. By eliminating the crank mechanism and using direct electromagnetic actuation with resonant spring support, oil-free linear compressors avoid the lubricant-related degradation of heat transfer and the solubility problems that compromise traditional systems [15,22]. The removal of oil not only simplifies the refrigerant circuit but also eliminates a major confounding factor in charge optimization studies. In lubricated systems, oil accumulates in heat exchangers and alters the effective charge distribution in a manner that is difficult to separate from the behavior of the refrigerant itself [13]. Previous research on oil-free linear compressor systems has addressed fundamental operational aspects, including dynamic behavior, leakage characterization, and preliminary performance mapping across a narrow range of conditions [8,23]. Building on this foundation, Chen et al. developed a component-wise refrigerant distribution model that explicitly accounts for the two-phase and single-phase regions in the heat exchangers

and connecting lines [9]. Li et al. subsequently proposed a comprehensive multiphysics numerical framework that incorporates the variable-displacement characteristics of the linear compressor, enabling more accurate performance predictions [24].

Although refrigerant charge characteristics in conventional vapor compression systems have been extensively investigated, the charge-induced operational features specific to oil-free linear compressor systems remain insufficiently understood. Only a limited number of parametric experiments have systematically verified the non-monotonic variation in system performance with refrigerant charge. Moreover, even fewer studies have adopted the TEWI index as a metric for evaluating the trade-off between energy efficiency and life-time environmental performance. This gap is particularly relevant because the oil-free architecture modifies both the refrigerant inventory in the heat exchangers and the compressor's sensitivity to pressure ratio, suggesting that charge-optimization guidelines established for conventional systems may not be directly applicable.

In this study, a vapor compression refrigeration system equipped with an oil-free linear compressor and charged with R134a is experimentally investigated. The system performance responses to refrigerant charge are examined under a range of piston strokes and pressure ratios. The work focuses on revealing the component-wise refrigerant mass distribution at different charge levels; quantifying the influence of refrigerant charge on core thermodynamic parameters such as superheat, subcooling, mass flow rate, volumetric efficiency, cooling capacity, and coefficient of performance across multiple operating conditions; and identifying the optimal refrigerant charge that effectively balances energy efficiency with TEWI-based environmental performance. These findings provide a quantitative foundation for the structural design and operational regulation of oil-free refrigeration equipment, and they demonstrate that refined charge optimization can make a meaningful contribution to energy conservation and carbon emission reduction.

2. Experimental Setup

Figure 1a shows the experimental configuration of the vapor compression refrigeration system. The system is composed of the linear compressor, an annular evaporator with a centrally mounted electric heater, and a commercial water-cooled copper coaxial condenser. The entire apparatus is monitored and controlled via a LabVIEW-based platform paired with an NI USB-6341 data acquisition card (National Instruments Corporation, Austin, TX, USA) for real-time data collection. Figure 1b displays the main components of the system: the linear compressor, the evaporator, and the condenser.

The linear compressor integrates a moving magnet linear motor, a flexure spring suspension system, a piston–cylinder assembly, and two reed valves. Its motor contains three rectangular moving magnets and four laminated cores, each wound with a coil. Three rectangular NdFeB permanent magnets are arranged along the motor axis and magnetized with alternating polarities, generating a circulating magnetic flux that links the adjacent laminated cores to form a closed magnetic circuit. This configuration concentrates the magnetic field and provides the reciprocating electromagnetic force needed to drive the piston. The flexure spring suspension system consists of two axially spaced sets of flexure springs, which support the moving piston, permitting free reciprocating motion along the cylinder axis while strongly restraining radial displacement. The springs are fabricated from martensitic stainless steel, a material that offers high radial stiffness combined with sufficient axial compliance, thereby minimizing the piston's radial offset during operation. Suction and discharge are controlled by two reed valves. Because the volumetric flow rate through the suction port exceeds that through the discharge port, the suction reed valve is designed with a correspondingly larger flow area to accommodate the higher flow and reduce flow resistance. The suction valve is 0.3 mm thick, while the discharge valve is

0.2 mm thick. The piston has a diameter of 19 mm and can travel with a maximum stroke of 14 mm.

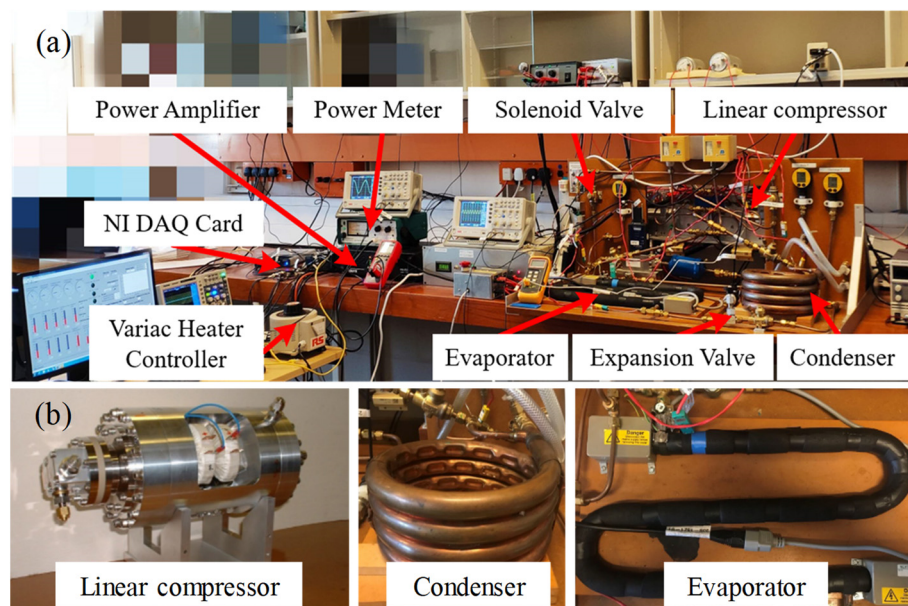


Figure 1. (a) The experimental configuration of the vapor compression refrigeration system; (b) The main components of the system.

The water-cooled coaxial condenser is 2325 mm long, with coolant and refrigerant connection diameters of 12.7 mm and 16 mm, respectively. The evaporator, covered with thermal insulation on its outer surface to suppress unwanted heat exchange with the surroundings, measures 1280 mm in length, 7.9 mm in internal diameter, and 12.7 mm in external diameter. The heat load is precisely controlled by adjusting the electrical power supplied to the central heater, allowing systematic evaluation of compressor performance over a range of operating conditions.

During operation, the linear compressor pressurizes the refrigerant into a high-temperature gas. This gas then enters the condenser, where it condenses into a high-pressure liquid while releasing heat to the cooling water. The high-pressure liquid subsequently passes through the expansion valve, evaporates in the evaporator while absorbing heat, and returns to the compressor as a low-pressure vapor, thereby completing the refrigeration cycle.

To comprehensively evaluate the thermodynamic parameters of the vapor compression refrigeration system, a suite of instruments was deployed to capture the key physical quantities across the experimental setup. The system schematic is shown in Figure 2. Linear variable differential transformer (LVDT) displacement transducers were employed to monitor the piston displacement of the linear compressor, providing real-time positional data essential for assessing its dynamic behavior. DRUCK PMP 1400 pressure transducers were used to acquire multiple pressure measurements at critical points, including the compressor discharge pressure, the evaporator inlet pressure, the compressor suction pressure, and the compressor body pressure. Temperature monitoring was accomplished with K-Type thermocouples installed at strategic locations throughout the system, ensuring that thermal conditions at all relevant positions were recorded. A Hastings HFM-201 main mass flow meter was mounted on the main circuit to measure the refrigerant mass flow rate in the primary loop. A Tylan FM-360 bleed flow meter was installed on the bypass circuit to measure the flow rate of refrigerant returning from the compressor housing to the suction line, enabling control of the piston offset. Both mass flow meters have a

measurement accuracy of $\pm 1\%$. The current in the internal coil of the compressor was measured with a current transducer, thereby capturing the electrical input directly at the motor. All measurement data were acquired through two types of data acquisition cards to accommodate different signal bandwidths and sampling requirements. Low-speed data acquisition cards (LDAQ) were used for the relatively slow-varying signals—namely pressure, temperature, and mass flow—while high-speed data acquisition cards (HDAQ) handled the rapidly fluctuating signals, including pressure, voltage, current, and piston displacement. The detailed specifications and accuracy of all experimental instruments are listed in Table 1.

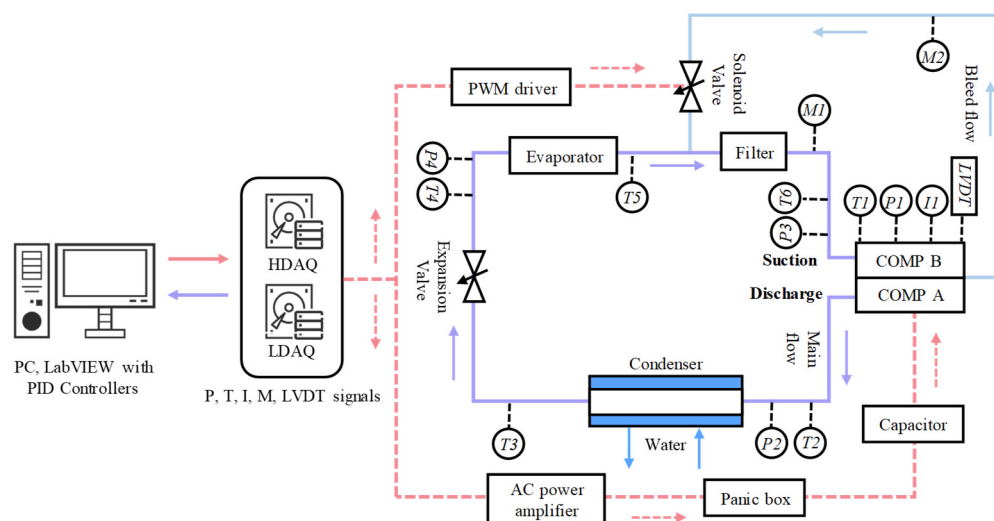


Figure 2. Schematic diagram of experimental setup.

Table 1. The model and accuracy of the experimental instruments.

Instruments	Model	Quantity Accuracy (Refer to Value)
Thermocouple	K-type (WIKA Instrument, Lawrenceville, GA, USA)	$\pm 1.5\text{ }^{\circ}\text{C}$
Pressure transducer	DRUCK PMP1400 (Druck, Leicester, UK)	$\pm 0.15\%$
LVDT	Lucas Schaevitz (Schaevitz Engineering, Pennsauken, NJ, USA)	$\pm 0.025\text{ mm}$
Mass flow meter	Hastings HFM-201 (Teledyne Hastings Instruments, Hampton, VA, USA)	$\pm 1\%$
Bleed flow meter	Tylan FM-360 (Tylan Corporation, Carson, CA, USA)	$\pm 1\%$
Current transducer	LA LEM 25-NP (LEM Holding SA, Geneva, Switzerland)	$\pm 0.5\%$
Voltage attenuator	Fylde 261HVA HV (Fylde Electronic Laboratories Ltd., Preston, Lancashire, UK)	$\pm 0.5\%$

The system performance was experimentally investigated by adjusting the refrigerant charge. Experiments were conducted at five charge levels: 220, 250, 280, 300, and 330 g. The refrigeration system was evacuated by a vacuum pump prior to each charging operation. To achieve thorough evacuation, an electric resistance heater was used to heat the compressor, which vaporized the residual liquid refrigerant inside the compressor housing. Two 1 m long charging tubes were used to connect the refrigerant cylinder, the four-valve manifold, and the refrigeration system. The section of tubing between the refrigerant cylinder and the four-valve manifold was fully filled with liquid refrigerant before formal charging. Minor leakage and residual refrigerant trapped in the tubes were inevitable when disconnecting the charging lines after filling. Based on extensive preliminary tests, an additional 15 g of refrigerant was charged into the system to compensate for such losses.

All experiments were carried out at an ambient temperature of $22\text{ }^{\circ}\text{C}$. Before each test, the compressor shell was heated to $45\text{ }^{\circ}\text{C}$ with an electric resistance heater for 15 min to

evaporate and remove residual liquid refrigerant inside. The system pressure ratio was controlled by adjusting the opening degree of the manual expansion valve. The operating frequency was manually adjusted to achieve the system resonance within 33–37 Hz. The piston stroke was regulated by a LabVIEW PID controller, which increased the stroke from zero in steps of 0.2 mm until reaching the target range of 9–12 mm. The steady state criterion adopted in this study was defined as follows: all parameters, including suction pressure, discharge pressure, mass flow rate, piston stroke, and temperatures, remained stable for a continuous period of 5 min, with pressure fluctuations within $\pm 1\%$ and temperature fluctuations within ± 0.5 °C. Once the steady state was reached, data acquisition was initiated via the front-panel control, which synchronously triggered two NI USB-6341 data acquisition cards. Although the 5 min steady-state duration has been validated via long-term tests for the present system, it is relatively shorter than the stabilization period commonly adopted for large commercial refrigeration systems. Its applicability to larger-scale systems requires further independent verification. Each group of experiments was repeated three times under identical operating conditions, and the final results are presented as average values. The detailed experimental conditions are listed in Table 2.

Table 2. Test conditions for different refrigerant charges.

Parameter	Value
Refrigerant	R134a
Charge (g)	220, 250, 280, 300, 330
Pressure ratio	2.0, 2.5, 3.0, 3.5
Piston stroke (mm)	9, 10, 11, 12
Condenser temperature (°C)	45
Operating frequency (Hz)	33–37
Compressor body temperature (°C)	>35
Ambient temperature (°C)	22

3. Data Analysis and Validation

3.1. Data Analysis

The radial clearance between the linear compressor piston and cylinder allows gas leakage during compressor operation. Liang et al. [25] derived the leakage mass flow rate through the clearance as expressed in Equation (1). The actual mass flow rate is then determined by Equation (2):

$$\dot{m}_{leak} = \frac{\pi D c^3 (P_c^2 - P_b^2)}{24 \mu L_p R_g T_c} \quad (1)$$

$$\dot{m} = \frac{(V_2 - V_1) P_{dis} f}{R_g T_{dis}} - \dot{m}_{leak} \quad (2)$$

where V_2 and V_1 are the cylinder volumes at the beginning and end of the discharge process, respectively; P_c is the cylinder pressure; T_c is the cylinder temperature; P_b is the compressor body pressure; P_{dis} is the discharge pressure; T_{dis} is the discharge temperature; f is the operating frequency; and R_g is the specific gas constant. The radial clearance between the piston and cylinder is 12.5 μm .

The volumetric efficiency, η_V , is given by Equation (3):

$$\eta_V = \frac{\dot{m} R_g T_{suc}}{S A f P_{suc}} \quad (3)$$

where A is the piston area, S is the piston stroke, and P_{suc} is the suction pressure.

The cooling capacity, Q_c , is calculated using Equation (4):

$$Q_c = \dot{m}(h_{\text{evap_out}} - h_{\text{evap_in}}) \quad (4)$$

where $h_{\text{evap_in}}$ is the enthalpy at the evaporator inlet and $h_{\text{evap_out}}$ is the enthalpy at the evaporator outlet.

The compressor power input ($\dot{W}_{in}$) is determined by measuring the voltage and current as follows:

$$\dot{W}_{in} = \frac{1}{t} \int_0^t UI dt \quad (5)$$

where U , I , and t are the voltage, current, and period, respectively.

The COP is obtained by dividing the cooling capacity (Q_c) by the compressor power input ($\dot{W}_{in}$) according to Equation (6):

$$COP = \frac{Q_c}{\dot{W}_{in}} \quad (6)$$

The subcooling and superheat are calculated as:

$$T_{\text{sub}} = T_{\text{sat}} - T_{\text{cond}} \quad (7)$$

$$T_{\text{sup}} = T_{\text{suc}} - T_{\text{sat}} \quad (8)$$

The refrigerant circuit of this vapor compression system includes an evaporator, a condenser, a vapor line, a liquid line, and a filter. Specifically, the vapor line comprises the condenser inlet line and the evaporator outlet line, while the liquid line includes the condenser outlet line. The evaporator inlet line is designed to serve as a two-phase line with a constant void fraction. Chen's extensive analysis provides detailed specifications for the parameters and phase lengths of the linear compressor vapor compression system [9]. The masses of the two-phase and single-phase refrigerant in the heat exchangers are expressed as:

$$m_{tp} = A_{tp} \int_0^{L_{tp}} (\alpha \rho_g + (1 - \alpha) \rho_l) dl \quad (9)$$

$$m = A_{HX} \int_0^{L_{sp}} \rho dl \quad (10)$$

where A_{tp} is the cross-sectional area in the two-phase region; A_{HX} is the cross-sectional area of the heat exchanger; L_{tp} is the length of the two-phase region; L_{sp} is the length of the single-phase region; α is the void fraction; ρ_g is the density of the vapor refrigerant; and ρ_l is the density of the liquid refrigerant.

The refrigerant mass in single-phase form in the filter and pipes is given by Equation (11):

$$m'_{sp} = A_{sp} \int_0^{L_{sp}} \rho dl \quad (11)$$

where A_{sp} is the cross-sectional area of the single-phase line.

Before entering the evaporator, the refrigerant accumulates in the pipes as a two-phase flow. Hence, the refrigerant mass in the pipes is calculated using Equation (12):

$$m'_{tp} = A'_{tp} \int_0^{L'_{tp}} \rho_{tp} dl \quad (12)$$

TEWI was proposed to evaluate the environmental impact of refrigeration systems by combining the direct effects of refrigerant emissions with the indirect effects of fossil fuel consumption for power generation [26]. The TEWI value equals the sum of the direct

and indirect greenhouse gas emissions. Direct emissions are calculated from the annual refrigerant leakage; typical leakage sources include regular operation, accidental loss, and system maintenance [27]. In this work, indirect emissions refer to the greenhouse gases released through the annual power consumption of the refrigeration system. Compared with LCCP and GWP, TEWI is easier to apply and delivers more reliable results. It is widely used for selecting eco-friendly refrigerants [28], and is calculated as follows [29]:

$$TEWI = GWP m_{ref} L_{ann} n + GWP m_{ref} (1 - \alpha_{rec}) + E_{ann} \beta n \quad (13)$$

where m_{ref} stands for the refrigerant charge of the system. L_{ann} is set as the annual leakage rate of 15%. n represents the equipment service life of 15 years. α_{rec} is the refrigerant recovery rate at the end of service life. Its value is 0.95 for China. E_{ann} means the annual energy consumption of the system, with the unit of kWh/year. β is the indirect emission factor of 0.53 kg CO₂e/kWh.

The workflow of data analysis is shown in Figure 3.

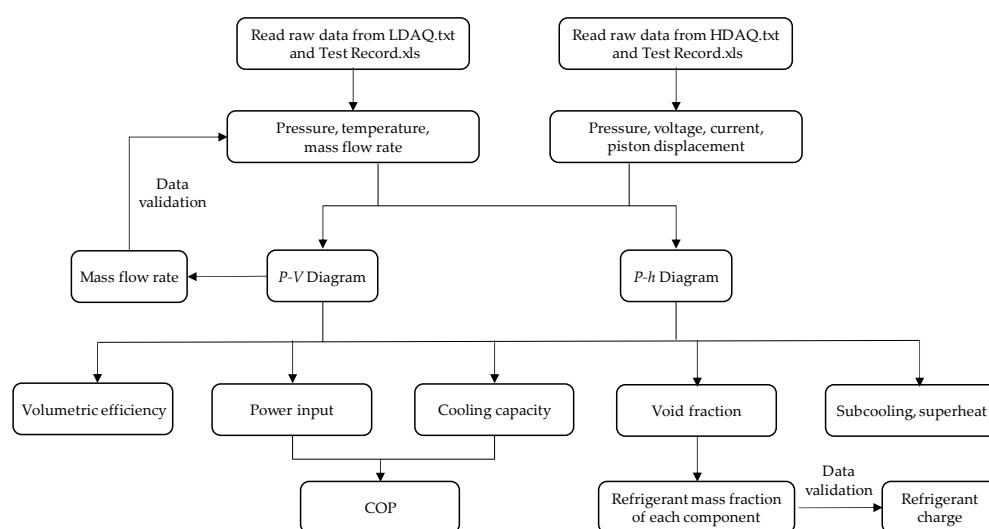


Figure 3. Data analysis workflow.

3.2. Data Validation

The experiments were carried out on a vapor compression refrigeration system equipped with an oil-free linear compressor. The experimental setup was validated by comparing the measured refrigerant charge with predictions from the model proposed by Li et al. [24]. Figure 4a shows that the mean absolute percentage error (MAPE) for refrigerant charge estimation was 4.8%. Most predicted values were lower than the measured values, which is consistent with the observations reported by Li et al. [24]. This discrepancy can be attributed to refrigerant leakage during charging and operation; leakage reduces system pressure and consequently the refrigerant charge. Figure 4b shows good agreement between the measured mass flow rate and the model predictions that include the leakage correction from Equation (1), with a MAPE of 8.2%, confirming that the leakage model is applicable under the operating conditions of this study. Figure 4c presents the refrigerant distribution across system components under the conditions of 280 g refrigerant charge, 12 mm piston stroke and pressure ratio of 3.5. The condenser holds 74.7% of the total refrigerant charge, which is the dominant component for refrigerant storage in the cooling cycle. The evaporator accounts for 9.5% of the total charge, the liquid line 11.4%, and the gas line 2.6%. This distribution characteristic is consistent with the findings of existing experimental studies [30,31].

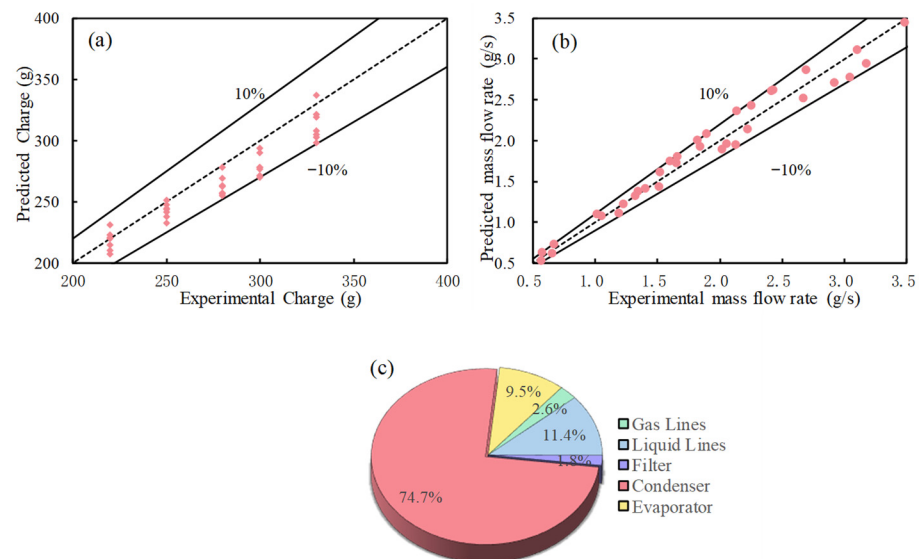


Figure 4. (a) Predicted refrigerant charge; (b) Predicted mass flow rate; (c) Distribution of refrigerant in the components.

3.3. Uncertainty Analysis

The measurement uncertainties of the directly measured parameters are listed in Table 1. For quantities derived from multiple measured parameters, such as power input and volumetric efficiency, an uncertainty propagation analysis was performed to evaluate the combined uncertainty, as expressed by Equation (14).

$$\frac{\Delta N(x_1, x_2, \dots, x_n)}{N(x_1, x_2, \dots, x_n)} = \sqrt{\left(\frac{\partial \ln(N)}{\partial x_1} \Delta x_1\right)^2 + \left(\frac{\partial \ln(N)}{\partial x_2} \Delta x_2\right)^2 + \dots + \left(\frac{\partial \ln(N)}{\partial x_n} \Delta x_n\right)^2} \quad (14)$$

The measurement uncertainties of the power input, volumetric efficiency, cooling capacity, and COP were determined to be $\pm 0.7\%$, $\pm 0.3\%$, $\pm 1.0\%$, and $\pm 1.2\%$, respectively.

4. Results and Discussion

Figure 5 presents the refrigerant mass fraction of each component at different charge levels. The mass fraction in the condenser grows with increasing refrigerant charge, indicating that extra refrigerant mainly accumulates inside the condenser. When the charge was increased from 220 to 330 g, the refrigerant mass in the condenser increased by 91.69 g, corresponding to a 6.4% increase in mass fraction. Overcharging may cause excessive refrigerant accumulation in the condenser, potentially leading to flooding and system performance degradation. These results are consistent with the findings of Ghoubali et al., who tested three different condensers and observed that, in all cases, the condenser exhibited the highest refrigerant mass fraction. However, when a microchannel heat exchanger was used as the condenser, the total charge decreased by 13%, with most of the reduction occurring in the condenser [32]. In contrast, the other components showed minimal variation in refrigerant mass fraction over the entire charge range.

Figure 6 illustrates the variations in superheat, subcooling, suction pressure, and discharge pressure with refrigerant charge at a piston stroke of 12 mm and a pressure ratio of 3.0. As the refrigerant charge increases, the superheat decreases. Specifically, the superheat decreases by 52.1% when the charge rises from 220 g to 330 g. This occurs because a higher suction pressure raises the suction saturation temperature. Notably, excessively low superheat can draw liquid refrigerant into the compressor, potentially causing compressor failure.

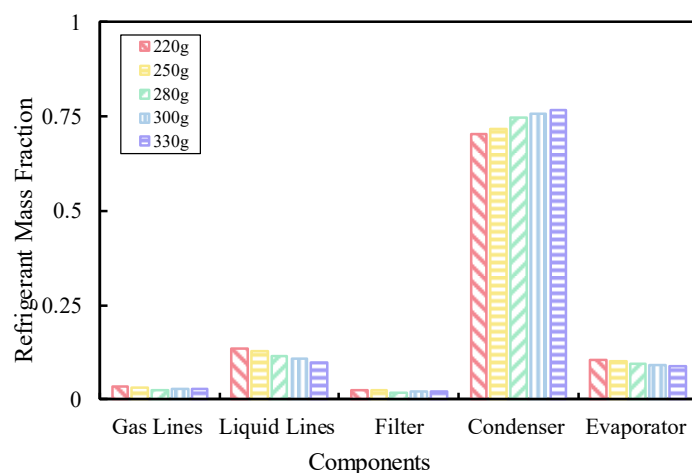


Figure 5. Refrigerant mass fraction of each component for different refrigerant charges.

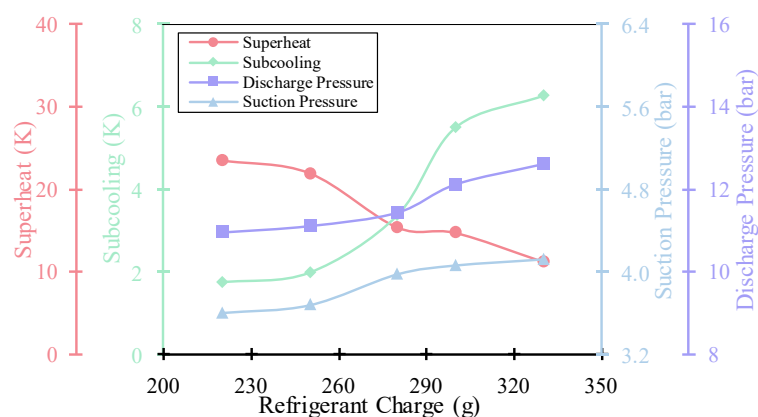


Figure 6. Variation in superheat, subcooling, suction pressure and discharge pressure with refrigerant charge.

The subcooling trend is opposite to that of superheat. As the refrigerant charge increases, refrigerant accumulates in the condenser, raising the condensing pressure and thereby increasing subcooling. When the charge rises from 220 g to 330 g, the subcooling increases by 259.6%. Because of its high sensitivity to charge level, subcooling is widely used as an indicator for determining the appropriate refrigerant charge [12].

Both suction and discharge pressures increase with refrigerant charge. As the charge increases from 220 g to 280 g, the discharge pressure rises slowly from 10.95 bar to 11.43 bar, representing an increase of 0.48 bar, or 4.4%. When the charge further increases from 280 g to 330 g, the discharge pressure shows a faster increase, rising from 11.43 bar to 12.61 bar, which corresponds to an increase of 1.18 bar, or 10.3%. This behavior is attributed to the higher mass flow rate drawn by the compressor at elevated charge levels. In the early stage of charge increase, the system operates under relatively stable conditions, resulting in only a minor increase in discharge pressure. When the charge exceeds a certain threshold, the system approaches saturation while the suction temperature continues to rise, causing the discharge pressure to increase more rapidly.

Compared with the discharge pressure, the effect of increased charge on suction pressure is relatively small. When the charge rises from 220 g to 330 g, the suction pressure increases by only 0.52 bar. Chen et al. indicated that the refrigerant mass fraction in the evaporator shows only a modest variation with charge, which leads to a limited increase in evaporating pressure and thus a minor change in suction pressure [9]. Guzella et al. noted that elevated suction and discharge pressures lead to higher cylinder pressures, which in turn increase axial forces, power input, and copper losses [33]. Therefore, excessive

refrigerant charge is detrimental to vapor compression refrigeration systems, highlighting the necessity of determining an appropriate charge.

Figure 7 presents the variation in linear compressor power input with refrigerant charge at different piston strokes and pressure ratios. Overall, the power input exhibits a clear upward trend as the refrigerant charge increases. At a piston stroke of 12 mm, the power input increases by 11.4% when the charge is raised from 220 to 330 g; under a constant pressure ratio of 3.0, the same increment in charge leads to a 17.5% rise in power input. This behavior is primarily caused by the accumulation of refrigerant mass in the condenser with increasing charge, which raises the condensing pressure and consequently demands more work from the compressor. When the charge is held constant, the power input is observed to increase with piston stroke and decrease with pressure ratio. These changes are driven mainly by variations in the mass flow rate, which directly affect the compressor's workload. For instance, at a fixed charge of 280 g, extending the piston stroke from 9 to 12 mm raises the power input by 53.66 W, whereas elevating the pressure ratio from 2.0 to 3.5 reduces it by 52.48 W.

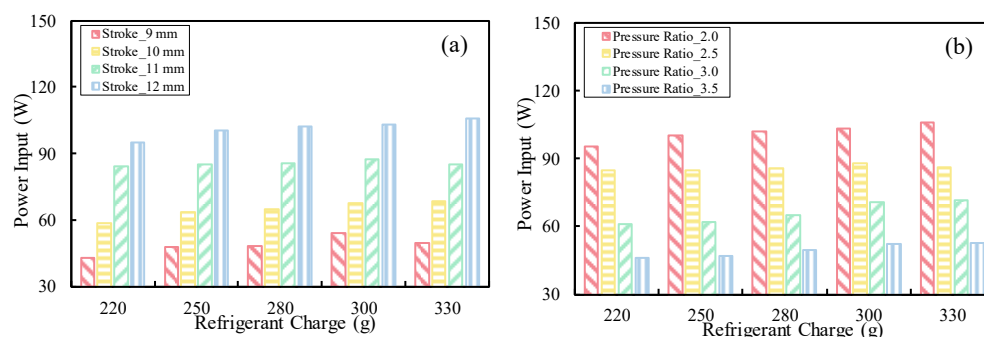


Figure 7. (a) Variation in power input with charge for different piston strokes; (b) Variation in power input with charge for different pressure ratios.

Figure 8 shows the variation in mass flow rate with refrigerant charge at different piston strokes and pressure ratios. The mass flow rate exhibits a non-monotonic trend, initially increasing and then decreasing as the charge increases. At a piston stroke of 12 mm, the mass flow rate rises by 27.5% when the charge is increased from 220 to 280 g, reaching a maximum of 3.45 g/s at 280 g; beyond this optimal point, it gradually declines to 2.87 g/s at 330 g. A similar pattern is observed under a constant pressure ratio: at a pressure ratio of 3.0, the mass flow rate peaks at a charge of 300 g, with a value of 1.32 g/s. In under-charged systems, the mass flow rate increases with charge until the optimal charge is reached, because the added refrigerant enhances the evaporator's effective heat transfer area before flooding occurs. However, as the charge continues to rise, the subcooling zone in the condenser expands, progressively reducing the effective condensing area available for phase change. This contraction in the heat transfer area lowers the condensation efficiency and ultimately reduces the circulating refrigerant mass flow rate.

Piston stroke and pressure ratio also exert a pronounced influence on the mass flow rate in linear-compressor vapor compression refrigeration systems. The mass flow rate increases with piston stroke and decreases with pressure ratio. For a fixed charge of 280 g, raising the piston stroke from 9 to 12 mm increases the mass flow rate by 141.1%, whereas elevating the pressure ratio from 2.0 to 3.5 reduces it by 67.3%. The primary reason for the reduction at higher pressure ratios is the decrease in suction vapor density, which diminishes the mass of refrigerant drawn into the cylinder during each suction stroke. In addition, at higher pressure ratios, the larger pressure differential across the piston–cylinder clearance seal drives greater refrigerant leakage to the compressor body, further reducing the net mass flow rate delivered to the system.

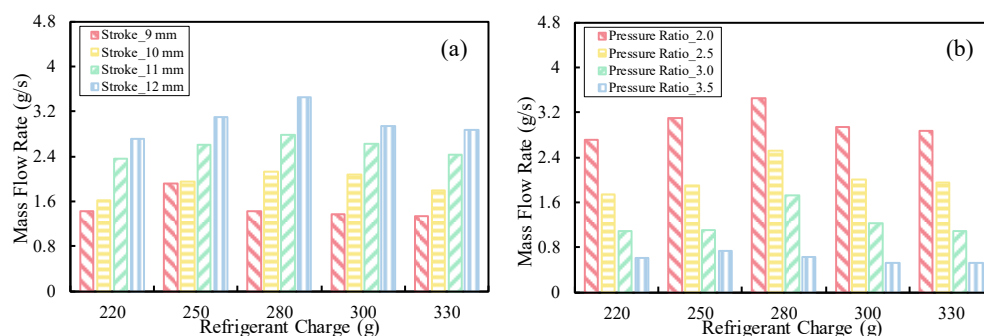


Figure 8. (a) Variation in mass flow rate with charge for different piston strokes; (b) Variation in mass flow rate with charge for different pressure ratios.

Figure 9 shows the variation in volumetric efficiency with refrigerant charge at different piston strokes and pressure ratios. The volumetric efficiency exhibits a trend similar to that of the mass flow rate presented in Figure 8, indicating that the factors governing mass flow also largely dictate the volumetric behavior of the compressor. At a piston stroke of 12 mm, the volumetric efficiency peaks at 59.3% when the charge is 280 g. As the piston stroke becomes smaller or the pressure ratio becomes higher, the peak volumetric efficiency shifts to lower charge levels, which is consistent with the earlier observation that the optimal charge for mass flow rate decreases under such constrained conditions. For a piston stroke of 9 mm and a pressure ratio of 3.5, the maximum volumetric efficiency is reached at a charge of 250 g, with values of 47.5% and 22.7%, respectively.

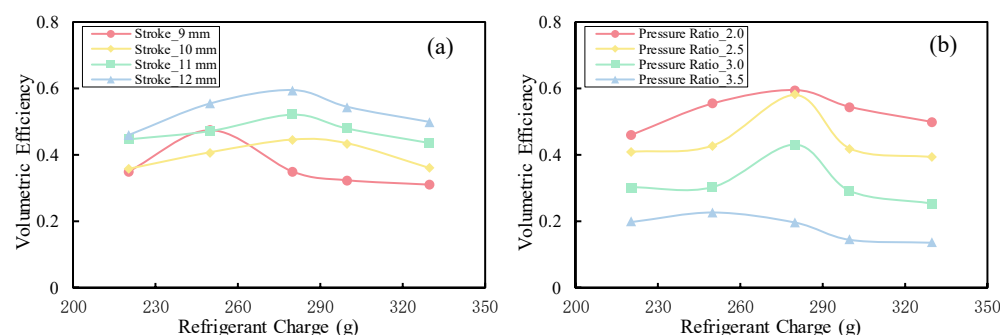


Figure 9. (a) Variation in volumetric efficiency with charge for different piston strokes; (b) Variation in volumetric efficiency with charge for different pressure ratios.

At a charge of 250 g, the volumetric efficiency measured at a piston stroke of 9 mm exceeds that at 10 mm; this result deviates from the overall trend observed in Figure 9. This discrepancy occurs because the mass flow rate is not the sole determinant of volumetric efficiency, and other factors—particularly the suction pressure—can play a decisive role under certain conditions. As shown in Figure 8, when the piston stroke is increased from 9 to 10 mm at a charge of 250 g, the mass flow rate increases by only 1.9%, a modest gain that is insufficient to raise the volumetric efficiency. Under the same conditions, however, the suction pressure rises by 8.3%, which substantially reduces the density of the refrigerant vapor entering the cylinder and impairs the filling process. This elevated suction pressure therefore exerts a dominant negative influence on volumetric efficiency, outweighing the marginal benefit of the slightly higher mass flow rate and causing the overall volumetric efficiency to decrease.

Figure 10 shows the variation in cooling capacity with refrigerant charge at different piston strokes and pressure ratios. The cooling capacity follows a non-monotonic trend, first increasing and then decreasing as the charge rises. At a piston stroke of 12 mm, the cooling capacity increases with refrigerant charge, reaching a maximum of 472.74 W at 280 g, which

is 9.5% higher than that at 220 g; as the charge is further increased beyond this optimum, the capacity declines gradually to 425.52 W at 330 g. A similar pattern is observed under a constant pressure ratio: at a pressure ratio of 3.0, the cooling capacity peaks at a charge of 300 g, achieving a value of 192.57 W. This behavior arises because, before the peak is reached, the mass flow rate through the evaporator is the dominant factor governing the heat transfer process. At low charge levels, the refrigerant mass flow rate in the evaporator is insufficient to fully utilize the available effective heat transfer area; consequently, the evaporator outlet superheat remains high, resulting in a relatively low cooling capacity. Although the refrigerant mass fraction in the evaporator exhibits only minimal variation with charge, as indicated in Figure 5, the mass flow rate passing through the evaporator continues to increase as charge is added. This progressively expands the effective heat transfer area and thus raises the cooling capacity. As the charge approaches the optimum, the evaporation temperature gradually rises, narrowing the temperature difference between the refrigerant and the heat source. At this point, the reduced temperature difference becomes the limiting factor, suppressing any further increase in cooling capacity and eventually causing the trend to reverse.

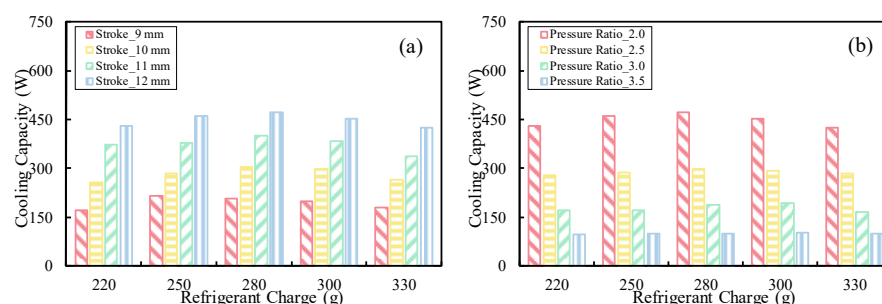


Figure 10. (a) Variation in cooling capacity with charge for different piston strokes; (b) Variation in cooling capacity with charge for different pressure ratios.

Figure 10a demonstrates that cooling capacity increases with piston stroke: at a constant charge of 280 g, raising the piston stroke from 9 to 12 mm leads to a 128.2% increase in cooling capacity. Conversely, Figure 10b shows that cooling capacity decreases markedly with increasing pressure ratio; elevating the pressure ratio from 2.0 to 3.5 reduces the cooling capacity by 79.0%. These variations are directly consistent with the corresponding changes in mass flow rate. As illustrated in Figure 8, for a fixed refrigerant charge, a larger piston stroke and a lower pressure ratio both contribute to a higher refrigerant mass flow rate, which in turn delivers a proportionally greater cooling capacity to the system.

Figure 11 shows the variation in COP with refrigerant charge obtained in the experiments. The COP exhibits a non-monotonic trend, first increasing and then decreasing as the charge increases. At a piston stroke of 10 mm, the COP rises from 4.39 to 4.67 when the charge is raised from 220 to 280 g; further increasing the charge to 330 g causes the COP to decline from 4.67 to 3.87. This behavior reflects the fact that COP is jointly determined by cooling capacity and power input, and the relative rates of change of these two quantities dictate whether COP improves or deteriorates. As shown in Figure 10, at a piston stroke of 10 mm, raising the charge from 220 to 280 g increases the cooling capacity by 18.7%, while Figure 7 indicates that the corresponding power input rises by 11.5%. This is because the additional refrigerant primarily enhances the effective heat transfer area of the evaporator and increases the specific enthalpy difference across the evaporator, whereas the increase in power input mainly arises from the gradual rise in condensing pressure. Over this charge range, both parameters increase, but the rise in cooling capacity is substantially more pronounced, driving the progressive increase in COP toward its peak value. Beyond the optimum, however, the situation reverses: the mass flow rate decreases because the

expansion of the subcooled region in the condenser reduces the effective condensation area. Meanwhile, the power input continues to increase as excess charge accumulates in the condenser. This divergence causes the COP to fall, with the decline accelerating as the charge moves further away from the optimum.

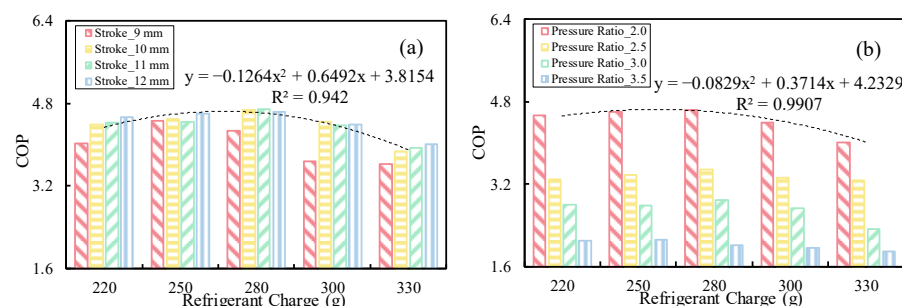


Figure 11. (a) Variation in COP with charge for different piston strokes; (b) Variation in COP with charge for different pressure ratios.

The COP is a key metric for evaluating the overall performance of vapor compression refrigeration systems. Figure 11 indicates that, under typical operating conditions, 280 g represents the optimal refrigerant charge for the linear-compressor vapor compression refrigeration system investigated in this study. At smaller piston strokes and higher pressure ratios, the charge at which the COP reaches its maximum shifts to lower values; specifically, when the piston stroke is 9 mm or the pressure ratio is 3.5, the maximum COP is achieved at a charge of 250 g. This shift occurs because both a smaller piston stroke and a higher pressure ratio reduce volumetric efficiency, which in turn limits the mass flow rate and shifts the optimal charge—where the balance between heat transfer and compressor work is most favorable—to a lower level. In terms of overall system performance, a small piston stroke and an excessively high pressure ratio substantially degrade the COP. At a charge of 250 g, the COP obtained at a 9 mm stroke is 3.2% lower than that at a 12 mm stroke. The effect of pressure ratio is even more pronounced: at the same charge of 250 g, raising the pressure ratio from 2.0 to 3.5 reduces the COP from 4.61 to 2.12, a decrease of 54.0%. These results demonstrate that excessively high pressure ratios and excessively small piston strokes represent atypical and unfavorable operating conditions for linear-compressor refrigeration systems. Under the range of conditions examined, a charge of 280 g maximizes the COP and thereby achieves the highest energy efficiency for the system.

Table 3 compares the COP and TEWI of the system at various refrigerant charges, with all values referenced to the optimal charge of 280 g. TEWI is primarily governed by the refrigerant charge, which determines the magnitude of direct leakage, and by the compressor power input, which drives indirect emissions. As shown in Figure 7, the power input rises with refrigerant charge, mainly because surplus refrigerant accumulates in the condenser, elevating the condensing pressure and forcing the compressor to perform more work. Consequently, an overcharged system suffers from higher direct emissions (due to the larger refrigerant inventory) and higher indirect emissions (owing to the increased power consumption). At a piston stroke of 12 mm and a pressure ratio of 2.5, increasing the charge to 330 g raises CO₂ emissions by 3.6% compared with the optimum, reaching a value of 188.74 kg. The flooding of the condenser caused by the excess charge simultaneously shrinks the effective condensing area. This leads to a decline in cooling capacity while the power input continues to climb, and the concurrent deterioration degrades both COP and TEWI.

Table 3. Comparison of COP and TEWI with optimal charge for different refrigerant charges.

Parameters	Stroke 11 mm Pressure Ratio 2.0				Stroke 12 mm Pressure Ratio 2.5			
	COP	$\Delta\text{COP}/\text{COP}_{280\text{g}}$	TEWI (kgCO ₂ e)	$\Delta\text{TEWI}/\text{TEWI}_{280\text{g}}$	COP	$\Delta\text{COP}/\text{COP}_{280\text{g}}$	TEWI (kgCO ₂ e)	$\Delta\text{TEWI}/\text{TEWI}_{280\text{g}}$
220 g	4.43	−5.7%	5041.98	−4.7%	3.29	−5.5%	5071.01	−4.2%
250 g	4.45	−5.3%	5177.98	−2.1%	3.37	−3.2%	5166.91	−2.4%
280 g	4.70	—	5291.40	—	3.48	—	5293.99	—
300 g	4.38	−6.8%	5425.89	2.5%	3.33	−4.3%	5434.01	2.6%
330 g	3.94	−16.2%	5447.85	3.0%	3.28	−5.7%	5482.73	3.6%

Reducing the refrigerant charge can lower direct CO₂ emissions and, to some extent, reduce the power input because of the smaller mass flow rate. However, as Figure 11 demonstrates, it also depresses the COP. At a piston stroke of 11 mm and a pressure ratio of 2.0, a charge of 220 g reduces CO₂ emissions by 5.5% relative to the optimum, but the corresponding COP penalty is 5.7%, which is considerably larger. This imbalance occurs because, when the system is undercharged, the cooling capacity drops far more steeply than the power input. The restricted refrigerant flow in the evaporator fails to fully exploit the available heat transfer area, and the resulting loss in cooling capacity substantially outweighs any modest reduction in compressor power. Hence, the environmental gain from lower direct emissions is entirely canceled out by a disproportionately severe energy-efficiency penalty. Overcharging produces a lose–lose outcome in which both COP and TEWI deteriorate. The data presented in Table 3 therefore indicate that the COP improvement realized at a charge of 280 g more than compensates for the marginal increase in TEWI relative to lower charge levels, whereas any further increase in charge degrades both performance indicators. Taking into account both COP and lifecycle CO₂ emissions, 280 g emerges as the optimal refrigerant charge for the vapor compression refrigeration system examined in this study.

5. Conclusions

The performance of a vapor compression refrigeration system equipped with an oil-free linear compressor was experimentally investigated. The effects of refrigerant charge, pressure ratio, and piston stroke on system performance were systematically examined. The optimal refrigerant charge was determined by considering both COP and CO₂ emissions. The key findings are summarized as follows:

- The condenser is the component most sensitive to refrigerant charge variation in this vapor compression refrigeration system. The refrigerant mass accumulated in the condenser increases with the charge. At a piston stroke of 12 mm and a pressure ratio of 3.5, the condenser accounts for approximately 74.7% of the total charge at 280 g. The condenser always contains the largest proportion of refrigerant under all tested operating parameters.
- At fixed piston stroke and pressure ratio, the subcooling increases with the refrigerant charge, while the superheat decreases. Both the suction and discharge pressures rise with the refrigerant charge.
- The mass flow rate first increases and then decreases as the refrigerant charge increases. Overcharging expands the subcooling zone, thereby reducing the effective condensing area.
- The cooling capacity and COP follow a trend similar to that of the mass flow rate. Under most operating conditions, the maximum cooling capacity and COP are achieved at a charge of 280 g. At smaller piston strokes and higher pressure ratios, the optimal charge shifts to a lower level.
- The CO₂ emissions increase with the refrigerant charge, primarily owing to the higher power input at larger charges. The results show that, at a charge of 280 g, the benefit of

a significantly higher COP outweighs the penalty of a slight increase in CO₂ emissions. Considering both COP and environmental impact, 280 g is the optimal charge for the vapor compression refrigeration system examined in this study.

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

Funding: This research was funded by the 2025 Jiangsu Higher Education Institutions Basic Research Natural Science Program (Grant No. 25KJD580007) and the 2025 High-Level Talent Program of Yangzhou Polytechnic Institute (Grant No. 2025gzk039).

Data Availability Statement: The data presented in this study are available in the article.

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

Nomenclature

Symbol

A	Piston cross-sectional area [mm ²]
A_{HX}	Cross-sectional area of heat exchanger [mm ²]
A_{sp}	Cross-sectional area of single-phase line [mm ²]
A_{tp}	Cross-sectional area in two-phase region [mm ²]
A'_{tp}	Cross-sectional area of two-phase line [mm ²]
c	piston–cylinder radial clearance [μm]
E_{ann}	Annual energy consumption of the refrigeration system [kWh/year]
f	Operating frequency [Hz]
h_{evap_in}	Specific enthalpy at evaporator inlet [J/g]
h_{evap_out}	Specific enthalpy at evaporator outlet [J/g]
I	Current [A]
L_{ann}	Annual refrigerant leakage rate [%]
L_p	Length of piston–cylinder clearance seal [mm]
L_{sp}	Length of single-phase region [mm]
L_{tp}	Length of two-phase region [mm]
L'_{tp}	Length of two-phase line [mm]
$\dot{m}$	Actual refrigerant mass flow rate [g/s]
$\dot{m}_{leak}$	Leakage mass flow rate through piston–cylinder clearance [g/s]
m_{ref}	Total refrigerant charge [g]
m'_{sp}	Refrigerant mass in single-phase pipes and filter [g]
m_{tp}	Refrigerant mass in two-phase region of heat exchangers [g]
m'_{tp}	Refrigerant mass in two-phase pipes [g]
n	Equipment service life [years]
P_b	Compressor body pressure [bar]
P_c	Cylinder pressure [bar]
P_{dis}	Discharge pressure [bar]
P_{suc}	Suction pressure [bar]
Q_c	Cooling capacity [W]
R_g	Specific gas constant of refrigerant
S	Piston stroke [mm]
t	Time [s]
T_c	Cylinder temperature [°C]
T_{cond}	Condensing temperature [°C]
T_{dis}	Discharge temperature [°C]

T_{sat}	Saturation temperature [°C]
T_{sub}	Subcooling [°C]
T_{suc}	Suction temperature [°C]
T_{sup}	Superheat [°C]
U	Voltage [V]
V_1	Cylinder volume at the beginning of discharge process [mm ³]
V_2	Cylinder volume at the end of discharge process [mm ³]
$\dot{W}_{in}$	Compressor power input [W]
Greek Symbols	
α	Void fraction in two-phase region
α_{rec}	Refrigerant recovery rate at end of service life
β	Indirect CO ₂ emission factor for power generation [kgCO ₂ e/kWh]
η_V	Volumetric efficiency [%]
μ	Dynamic viscosity of refrigerant [Pa·s]
ρ	Density of refrigerant [g/mm ³]
ρ_g	Density of saturated vapor refrigerant [g/mm ³]
ρ_l	Density of saturated liquid refrigerant [g/mm ³]
ρ_{tp}	Density of two-phase refrigerant [g/mm ³]
τ	Integration period for power calculation [s]
Abbreviations	
COP	Coefficient of Performance
DAQ	Data Acquisition
GWP	Global Warming Potential
HDAQ	High-speed Data Acquisition Card
LCCP	Life Cycle Climate Performance
LDAQ	Low-speed Data Acquisition Card
LVDT	Linear Variable Differential Transformer
MAPE	Mean Absolute Percentage Error
PID	Proportional–Integral–Derivative (controller)
TEWI	Total Equivalent Warming Impact

References

1. Zhang, Z.-Y.; Zhang, C.-L. Model-experiment-hybrid optimization of refrigerant charge via iterative calibration for a pilot vapor compression system. *Int. J. Refrig.* **2026**, *186*, 106895. [\[CrossRef\]](#)
2. Qaid, D.S.; Hamza, N.H.; Ashurov, M.; Saadaoui, S.; Elkamchouchi, D.H.; Khan, M.I.; Shewakh, W.M. Energy efficiency and climate sustainability assessment of new HFC-HFO refrigerant blends for air conditioning applications. *Int. Commun. Heat Mass Transf.* **2026**, *175*, 111250. [\[CrossRef\]](#)
3. Al Khiero, H.A.; Boukhanouf, R. Recent Advancements in Design and Energy Performance of Vapour Compression Systems for Air Conditioning in Buildings: A Review. *Energies* **2026**, *19*, 1166. [\[CrossRef\]](#)
4. Wei, Y.; Han, Y.; Zhang, Q.; Cui, H.; Tao, B.; Cui, X. Enhancing cross-individual generalizability of thermal sensation prediction in dynamic environments: Integrating physiological, emotional, and environmental features. *Build. Environ.* **2026**, *291*, 114210. [\[CrossRef\]](#)
5. Li, G.; Ma, J.; Zou, H.; Zhang, R. Experimental investigation of refrigerant charge determination for heating performance in a CO₂ heat pump system for electric vehicles. *Appl. Therm. Eng.* **2023**, *233*, 121147. [\[CrossRef\]](#)
6. Li, K.; Lan, J.; Zhou, G.; Tang, Q.; Cheng, Q.; Fang, Y.; Su, L. Investigation on the Influence of Refrigerant Charge Amount on the Cooling Performance of Air Conditioning Heat Pump System for Electric Vehicles. *J. Therm. Sci.* **2019**, *28*, 294–305. [\[CrossRef\]](#)
7. Cho, W.; Jang, D.S.; Lee, S.H.; Yun, S.; Kim, Y. Refrigerant charge reduction in R600a domestic refrigerator-freezer by optimizing hot-wall condenser geometry. *Int. J. Refrig.* **2020**, *117*, 295–306. [\[CrossRef\]](#)
8. Li, C.; Sun, J.; Zou, H.; Cai, J. Experimental assessment on operation efficiency and output characteristics of an oil-free linear compressor in a refrigeration system under various parameters. *Appl. Therm. Eng.* **2024**, *242*, 122453. [\[CrossRef\]](#)
9. Chen, X.; Li, Z.; Zhao, Y.; Jiang, H.; Liang, K.; Chen, J. Modelling of Refrigerant Distribution in an Oil-Free Refrigeration System using R134a. *Energies* **2019**, *12*, 4792. [\[CrossRef\]](#)
10. Deymi-Dashtebayaz, M.; Farahnak, M.; Moraffa, M.; Ghalami, A.; Mohammadi, N. Experimental evaluation of refrigerant mass charge and ambient air temperature effects on performance of air-conditioning systems. *Heat Mass Transf.* **2017**, *54*, 803–812. [\[CrossRef\]](#)

11. De Rossi, F.; Mauro, A.W.; Musto, M.; Vanoli, G.P. Long-period food storage household vertical freezer: Refrigerant charge influence on working conditions during steady operation. *Int. J. Refrig.* **2011**, *34*, 1305–1314. [[CrossRef](#)]
12. Yun, Y.; Chang, Y.S. Refrigerant Charge Prediction of Vapor Compression Air Conditioner Based on Start-Up Characteristics. *Appl. Sci.* **2021**, *11*, 1780. [[CrossRef](#)]
13. Ossorio, R.; Navarro-Peris, E.; Barta, R.B. Impact of lubricant in the performance of variable speed heat pumps working with R290. *Int. J. Refrig.* **2023**, *145*, 436–445. [[CrossRef](#)]
14. Li, Z.; Shen, H.; Liang, K.; Chen, X.; Zhu, Z. A numerical study on the effect of oil lubricant on the heat transfer and efficiency of a vapour compression refrigeration system. *Int. Commun. Heat Mass Transf.* **2022**, *134*, 106016. [[CrossRef](#)]
15. Youbi-Idrissi, M.; Bonjour, J.; Marvillet, C.; Meunier, F. Impact of refrigerant–oil solubility on an evaporator performances working with R-407C. *Int. J. Refrig.* **2003**, *26*, 284–292. [[CrossRef](#)]
16. Sánchez-Moreno-Giner, L.; Methler, T.; Barceló-Ruescas, F.; González-Maciá, J. Refrigerant charge distribution in brine-to-water heat pump using R290 as refrigerant. *Int. J. Refrig.* **2023**, *145*, 158–167. [[CrossRef](#)]
17. Borikar, S.A.; Gupta, M.M.; Alazwari, M.A.; Malwe, P.D.; Moustafa, E.B.; Panchal, H.; Elsheikh, A. A case study on experimental and statistical analysis of energy consumption of domestic refrigerator. *Case Stud. Therm. Eng.* **2021**, *28*, 101636. [[CrossRef](#)]
18. Bolaji, B.O. Experimental study of R152a and R32 to replace R134a in a domestic refrigerator. *Energy* **2010**, *35*, 3793–3798. [[CrossRef](#)]
19. Rangel-Hernández, V.H.; Belman-Flores, J.M.; Rodríguez-Valderrama, D.A.; Pardo-Cely, D.; Rodríguez-Muñoz, A.P.; Ramírez-Minguela, J.J. Exergoeconomic performance comparison of R1234yf as a drop-in replacement for R134a in a domestic refrigerator. *Int. J. Refrig.* **2019**, *100*, 113–123. [[CrossRef](#)]
20. Belman-Flores, J.M.; Heredia-Aricapa, Y.; García-Pabón, J.J.; Pérez-García, V.; Pérez-Reguera, C.G. Drop-In Replacement of R134a in a Household Refrigerator with Low-GWP Refrigerants R513A, R516A, and R1234ze(E). *Energies* **2023**, *16*, 3422. [[CrossRef](#)]
21. Li, C.; Sun, J.; Zou, H.; Cai, J.; Zhu, T. Characteristic analysis and energy efficiency of an oil-free dual-piston linear compressor for household refrigeration with various conditions. *Energy* **2023**, *270*, 126931. [[CrossRef](#)]
22. Liang, K.; Stone, R.; Hancock, W.; Dadd, M.; Bailey, P. Comparison between a crank-drive reciprocating compressor and a novel oil-free linear compressor. *Int. J. Refrig.* **2014**, *45*, 25–34. [[CrossRef](#)]
23. Li, C.; Zou, H.; Cai, J.; Jiang, Y.; Guo, C. Dynamic behavior analysis of a moving coil oil-free linear compressor in refrigeration system. *Int. J. Refrig.* **2022**, *133*, 235–246. [[CrossRef](#)]
24. Li, Z.; Liang, K.; Chen, X.; Zhu, Z.; Zhu, Z.; Jiang, H. A comprehensive numerical model of a vapour compression refrigeration system equipped with a variable displacement compressor. *Appl. Therm. Eng.* **2022**, *204*, 117970. [[CrossRef](#)]
25. Liang, K. Analysis of oil-free linear compressor operated at high pressure ratios for household refrigeration. *Energy* **2018**, *151*, 324–331. [[CrossRef](#)]
26. Maiorino, A.; Llopis, R.; Duca, M.G.D.; Aprea, C. Environmental impact assessment of R-152a as a drop-in replacement of R-134a in a domestic refrigerator. *Int. J. Refrig.* **2020**, *117*, 132–139. [[CrossRef](#)]
27. Blanco Ojeda, F.W.A.; Almeida Queiroz, M.V.; Marcucci Pico, D.F.; dos Reis Parise, J.A.; Bandarra Filho, E.P. Experimental evaluation of low-GWP refrigerants R513A, R1234yf and R436A as alternatives for R134a in a cascade refrigeration cycle with R744. *Int. J. Refrig.* **2022**, *144*, 175–187. [[CrossRef](#)]
28. Savitha, D.C.; Ranjith, P.K.; Talawar, B.; Rana Pratap Reddy, N. Refrigerants for sustainable environment—A literature review. *Int. J. Sustain. Energy* **2021**, *41*, 235–256. [[CrossRef](#)]
29. Vering, C.; Tanrikulu, A.; Mehrfeld, P.; Müller, D. Simulation-based design optimization of heat pump systems considering building variations. *Energy Build.* **2021**, *251*, 111358. [[CrossRef](#)]
30. Chen, R.; Wu, J.; Duan, J. Performance and refrigerant mass distribution of a R290 split air conditioner with different lubricating oils. *Appl. Therm. Eng.* **2019**, *162*, 114286. [[CrossRef](#)]
31. Li, T.; Lu, J.; Chen, L.; He, D.; Qiu, X.; Li, H.; Liu, Z. Measurement of refrigerant mass distribution within a R290 split air conditioner. *Int. J. Refrig.* **2015**, *57*, 163–172. [[CrossRef](#)]
32. Ghoubali, R.; Byrne, P.; Bazantay, F. Refrigerant charge optimisation for propane heat pump water heaters. *Int. J. Refrig.* **2017**, *76*, 230–244. [[CrossRef](#)]
33. dos Santos Guzella, M.; Cabezas-Gómez, L.; Guimarães, L.G.M. Numerical Simulation of a Single-compartment Household Refrigerator During Start-up Transient and Stationary Cyclic Operation with Different Refrigerant Charges. *Arab. J. Sci. Eng.* **2021**, *46*, 7533–7542. [[CrossRef](#)]

Disclaimer/Publisher’s Note: The statements, opinions and data contained in all publications are solely those of the individual author(s) and contributor(s) and not of MDPI and/or the editor(s). MDPI and/or the editor(s) disclaim responsibility for any injury to people or property resulting from any ideas, methods, instructions or products referred to in the content.