

Performance analysis of R1233zd(E) vapor injection refrigeration system incorporating a flash tank

FAN Haibin^{1,2}, ZHAO Panpan², JIA Lei², HUANG Shifang^{1,3}, ZHANG Xiaosong^{1,3}

(1. School of Energy and Environment, Southeast University, Nanjing 210096, China; 2. Hefei General Machinery Research Institute Co., Ltd., Hefei 230031, China; 3. Engineering Research Center of Building Equipment, Energy, and Environment of Ministry of Education, Southeast University, Nanjing 210096, China)

Abstract: Coupling R1233zd(E) with flash tank vapor injection enhances refrigeration performance and compensates for its inherently low volumetric refrigeration capacity. A thermodynamic mathematical model based on mass and energy conservation was developed for a flash tank vapor injection refrigeration system utilizing R1233zd(E). Key performance parameters, including the coefficient of performance (COP), cooling capacity, and compressor power consumption, were systematically analyzed across an evaporation temperature range of -30 to 5 °C, a condensation temperature range of 20 to 40 °C, and a first-stage compression ratio range of 1.1 to 6.5 . Furthermore, a fitted equation for the optimal first-stage compression ratio was derived based on operating temperatures. Results indicated that the system COP increased with rising evaporation temperature. In particular, at a fixed condensation temperature of 40 °C, the COP increased from ~ 1.86 to 4.53 as the evaporation temperature increased from -30 to 5 °C. Conversely, the COP decreased with increasing condensation temperature; when the evaporation temperature was maintained at -20 °C, the COP decreased from ~ 3.58 to 2.29 as the condensation temperature increased from 20 to 40 °C. An optimal intermediate pressure existed for maximizing the system COP. The deviation between the values calculated by the derived correlation for the optimal first-stage compression ratio and the theoretical model was predominantly within $\pm 3\%$. This provided theoretical guidance for operational parameter setting and intelligent control of the system.

Key words: R1233zd(E); flash tank; vapor injection refrigeration system; coefficient of performance (COP)

DOI: 10.3969/j.issn.1003-7985.2026.03.011

Refrigeration techniques that achieve low temperatures are widely used across various industries, such as gas liquefaction^[1], chemical^[2], food^[3], and construc-

tion^[4] industries. However, “dual carbon goals” have been proposed to solve environmental problems^[5]. GPT efficiency and environmental concerns in refrigeration are becoming increasingly important^[6]. As a result of the Kigali Amendment to the Montreal Protocol, HCFC refrigerants, despite their good performance in the thermal cycle, are being phased out^[7]. Hydrofluoroolefins (HFOs) with a low global warming potential (GWP) are considered the fourth generation of environmentally friendly refrigerants and have become the research hotspot in refrigeration and heat pump replacement^[8].

R1233zd(E) has shown potential in various thermodynamic cycles, including solar-assisted heat pumps, where it demonstrated a flexible solar collector efficiency of 40% across different weather conditions^[9]. In organic Rankine cycles, it has replaced HFC-245fa, achieving higher isentropic efficiency^[10]. In ejector refrigeration, Mwesigye et al.^[11] found that the ejector accounted for 46% to 56% of the exergetic losses, emphasizing the need for optimal generator conditions and performance improvements. Despite its effectiveness, R1233zd(E) has a low volumetric refrigeration capacity, requiring relatively large compressors or high mass flow rates, which can affect system compactness and cost-effectiveness.

A single-stage vapor refrigeration system may not meet the needs of relatively high temperatures or low ambient conditions due to its high compression ratio, which also increases maintenance costs^[12-13]. Techniques such as vapor injection, two-stage compression, and inverter frequency control have been explored to improve system performance^[14-16]. Vapor injection systems, in particular, have been shown to enhance the coefficient of performance (COP) under extreme conditions. Refs. [17-18] demonstrated substantial COP improvements of 16% to 32% in systems utilizing vapor injection. Furthermore, vapor compression cycles with vapor injection can achieve relatively high condensation temperatures or operate efficiently at relatively low ambient temperatures^[19]. Theoretical studies by Qi et al.^[20] reported an ejector-integrated hybrid configuration to improve the COP of air-source heat pumps by 2.8% to 3.1%, while Jung et al.^[21] experimentally determined an optimal in-

Received 2025-10-23, **Revised** 2025-12-05.

Biographies: FAN Haibin (1980—), male, senior engineer; HUANG Shifang (corresponding author), male, doctor, associate professor, shifang.huang@seu.edu.cn.

Foundation items: The National Natural Science Foundation of China (No. 52206005), Anhui Provincial Key Research and Development Project (No. 2022f04020001).

Citation: FAN Haibin, ZHAO Panpan, JIA Lei, et al. Performance analysis of R1233zd(E) vapor injection refrigeration system incorporating a flash tank[J]. Journal of Southeast University (English Edition), 2026, 42(3): 388-396. DOI: 10.3969/j.issn.1003-7985.2026.03.011.

ternal heat exchanger configuration to maximize COP for a vapor injection system. These advancements further highlight the potential for optimizing system performance through innovative configurations. Experimental research has further highlighted the benefits of combining vapor injection with R1233zd(E) in improving thermodynamic performance. However, most studies have focused on optimizing intermediate pressure for specific conditions, with limited guidance on how to select it across varying temperatures^[22-23]. Therefore, developing a general correlation for the first-stage compression ratio based on operating temperatures is vital for system design and optimization.

Building on these insights, this study investigated a vapor injection refrigeration system using R1233zd(E). Performance was assessed in terms of the COP, discharge temperature, cooling capacity, and compressor input, as functions of the evaporation and condensation temperatures and the compression ratio. A correlation for the optimal compression ratio was formulated and experimentally validated, providing guidance for the design and control of refrigeration systems in applications such as industrial cold storage and commercial freezing.

1 System Description

1.1 System architecture and operational mechanism

Fig. 1 presents a schematic diagram of the refrigeration cycle using intermediate vapor injection via a flash tank. This system consists of an evaporator, compressor, condenser, flash tank, two expansion devices, and several auxiliary components that together accomplish phase-change heat transfer of the refrigerant throughout the cycle.

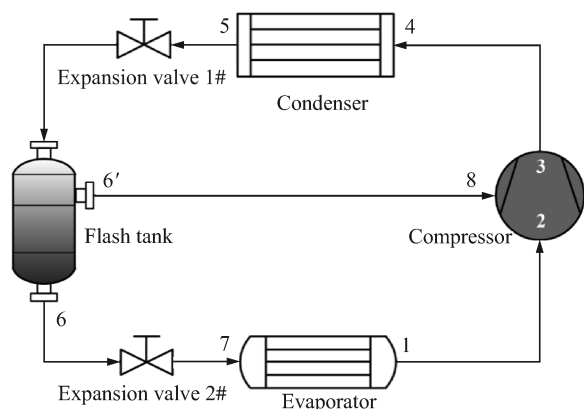


Fig. 1 Schematic diagram of flash tank vapor injection refrigeration system

Fig. 2 shows the P - h diagram of the low-temperature refrigeration cycle with flash tank intermediate vapor injection, clearly illustrating the refrigerant states in the various components of the system. First, the superheated vapor discharged from the compressor to the condensing

pressure (state point 4) entered the condenser to reject heat to the surroundings. During condensation, the refrigerant was converted into liquid (state point 5), which was then expanded through expansion valve 1# into a two-phase mixture with moderate temperature and pressure before entering the flash tank for vapor-liquid separation. Part of the liquid flashed into saturated vapor (state point 6) and was injected into the compressor (state point 8) to provide vapor injection and enthalpy lift, thereby improving the compressor's thermodynamic performance. The remaining liquid refrigerant (state point 6') was expanded through expansion valve 2# (state point 7) and then flowed into the evaporator, where it absorbed heat and evaporated into superheated vapor (state point 1). This vapor was subsequently compressed to the intermediate pressure in the compressor (state point 2), mixed with the vapor separated from the flash tank (state point 3), and further compressed to the condensing pressure, thereby completing the entire refrigeration cycle.

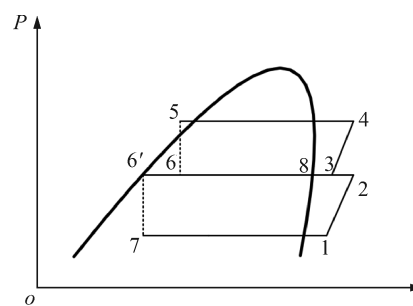


Fig. 2 P - h diagram of vapor injection refrigeration system with flash tank

1.2 Alternative low-GWP refrigerant R1233zd(E)

In the field of refrigeration cycles, R32 and R134a have long been widely used; R32 is favored in small-scale residential air conditioning systems for its high refrigeration capacity and relatively low GWP compared to older refrigerants, while R134a has dominated medium-temperature refrigeration applications due to its moderate thermodynamic properties and nonflammable safety class. However, both refrigerants face increasing substitution demands; R32 exhibits a high saturation pressure that requires strict pressure-resistant system design, and its GWP (675) fails to meet the latest low-carbon regulatory standards; R134a, with a high GWP-100 of 1 300, has been phased out in many regions because of its considerable greenhouse effect.

R1233zd(E) is a type of safe and environmentally friendly refrigerant, belonging to hydrochlorofluorocarbons. Properties of refrigerants R1233zd(E), R32, and R134a are listed in Table 1. Although chlorine is present in the molecular structure of R1233zd(E), this refrigerant's ODP of 3.4×10^{-4} is very close to zero, and its

Table 1 Main characteristics of different refrigerants

Parameter	R1233zd(E)	R32	R134a
Molecular weight/(g·mol ⁻¹)	130.50	52.02	102.03
Critical temperature/°C	166.45	78.11	101.06
Critical pressure/MPa	3.62	5.78	4.06
Normal boiling point (NBP)/°C	18.26	-51.65	-26.07
Triple point temperature/°C	-78.00	-136.81	-103.30
Condensing pressure at 40 °C/MPa	0.22	2.48	1.02
Vapor pressure at 25 °C/MPa	0.018	1.689	0.665
Latent heat of vaporization at -20 °C/(kJ·kg ⁻¹)	212.95	344.03	212.91
Latent heat of condensation at 40 °C/(kJ·kg ⁻¹)	183.06	237.10	163.02
Suction density at -20 °C/(kg·m ⁻³)	1.15	11.16	6.78
Ratio of liquid density to vapor density at 40 °C	105.02	12.18	22.89
Ratio of liquid specific heat to vapor specific heat at 40 °C	1.42	1.08	1.57
Ratio of liquid specific heat to vapor specific heat at -20 °C	1.42	1.80	1.51
ODP	3.4×10 ⁻⁴	0	0
GWP-100	7	675	1 300
ASHRAE 34 safety classification	A1	A2L	A1

GWP is extremely low. As it has excellent performance, is non-combustible, and has low toxicity and high thermal cycle efficiency, R1233zd(E) is recommended for use under high-temperature conditions and is the main alternative refrigerant for next-generation centrifugal water-cooled chillers and high-temperature refrigeration systems.

The saturation absolute pressure at various temperatures is shown in Fig. 3. This figure also presents the saturation pressures of different refrigerants at the working temperatures of the refrigerating system, ranging from -40 to 100 °C. Among the evaluated working fluids, R32 exhibited the sharpest increase in saturation pressure with increasing temperature: its pressure exceeded 5 MPa at 80 °C, reflecting strong temperature-dependent pressure sensitivity that might impose stringent pressure-resistance requirements on system components. R134a showed a

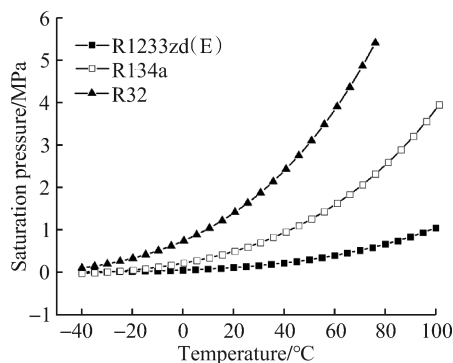


Fig. 3 Saturation pressure under various temperatures of R1233zd(E), R134a, and R32

moderate pressure increase, reaching about 4 MPa at 100 °C. In distinct contrast, R1233zd(E) demonstrated the mildest pressure-temperature correlation, with a saturation pressure remaining below 1 MPa even at 100 °C. This low and stable saturation pressure profile made R1233zd(E) particularly well-suited for refrigeration systems, particularly those operating in relatively high-temperature environments. Therefore, it exhibited very favorable thermodynamic properties for the design of system pipelines and equipment, making it a promising application for refrigeration systems.

2 Thermodynamic Analysis

2.1 Assumptions

To establish a mathematical model suitable for the thermodynamic analysis of the system, the following simplifying assumptions were adopted for the operating conditions: (1) The refrigeration cycle operated under steady-state and steady-flow conditions. (2) Heat exchange between the evaporator, condenser, associated piping, and the surrounding environment was negligible, and the pressure losses in the pipelines were ignored. (3) Changes in kinetic and potential energy of the working fluid during flow were neglected. (4) Adiabatic efficiencies of both the compressor and the throttle valve were set to constant values, which were representative of achievable performance levels in practice and were regarded as independent of operating conditions. (5) Mixing processes at the compressor suction port and inside the flash tank were treated as isobaric processes. (6) The refrigerant leaving the flash tank was assumed to be in a saturated state, with the vapor phase regarded as saturated vapor and the liquid phase regarded as saturated liquid.

The thermophysical properties of the working fluid R1233zd(E) used in this study were obtained from REFPROP 9.0. To cover the typical operating range of refrigeration and low-temperature freezing applications, the evaporating temperature was varied from -30 to 5 °C, while the condensation temperature was set between 20 and 40 °C. The degrees of superheating and subcooling were both fixed at 5 °C. The compressor's isentropic efficiency was maintained at 0.75. Under all operating conditions, the first-stage compression ratio ε ranged from 1.1 to 6.5. Detailed operating conditions are summarized in Table 1. The parameters used for the system's energy analyses are listed in Table 2, which also specifies the boundary conditions of the refrigeration system for subsequent model solution and performance evaluation.

The relative parameters used to analyze the system's energy properties are given in Table 2. Furthermore, the boundary conditions of the refrigerating system were con-

Table 2 Working parameters of the refrigerating system with flash tank

Parameter	Value
Evaporation temperature $T_e/^\circ\text{C}$	-30 to 5
Condensation temperature $T_k/^\circ\text{C}$	20 to 40
Superheating degree $\Delta T_e/^\circ\text{C}$	5
Subcooling degree $\Delta T_k/^\circ\text{C}$	5
Mass flow rate in the first-stage compressor $m_{\text{first-com}}/(\text{kg}\cdot\text{s}^{-1})$	1.0
Atmosphere pressure P_0/kPa	101.325
Atmosphere temperature $T_0/^\circ\text{C}$	25
Compression efficiency η_{com}	0.75
Valve efficiency η_v	0.8
First-stage compression ratio ε	1.1 to 6.5

tained as follows.

2.2 Mathematical modeling

A mathematical model was developed based on the preceding assumptions, with its equations grounded in the fundamental conservation laws of mass, energy, and momentum for the system's primary components. Selected equations are presented in the following section.

As shown in Fig. 1, the compressor in the refrigeration system underwent two compression processes during each cycle. The first was the precompression of the suction vapor, and the second was the polytropic compression after vapor mixing. The mass flow rate of the injected vapor from the flash tank could be calculated by fitting the following equation:

The first-stage compression process before vapor injection could be expressed as follows:

$$P_2 = P_3 = P_8 = \varepsilon P_1 \quad (1)$$

where P_1 , P_2 , P_3 , and P_8 represent the pressures at the corresponding state points, respectively; and ε denotes the first-stage compression ratio.

The energy balance equation in the flash tank could be expressed as follows:

$$(m_{\text{first}} + m_{\text{flash}})h_5 = m_{\text{flash}}h_8 + m_{\text{first}}h_{6'} \quad (2)$$

where m_{first} and m_{flash} represent the mass flow rates of the first-stage compression and the flash tank, respectively; h_5 , $h_{6'}$, and h_8 represent the enthalpies at the corresponding state points, respectively.

The vapor mass flow rate could be calculated as follows:

$$m_{\text{flash}} = \frac{(h_5 - h_{6'})m_{\text{first}}}{h_8 - h_5} \quad (3)$$

The relative injected vapor mass could be calculated as follows:

$$\zeta = \frac{m_{\text{flash}}}{m_{\text{first}}} \times 100\% \quad (4)$$

The total power of the compression process and the enthalpy after compression could be calculated as follows:

$$W_{\text{com}} = (1 + m_{\text{flash}})(h_4 - h_3) + m_{\text{first}}(h_3 - h_1) \quad (5)$$

$$\eta_{\text{com}} = \frac{h_{2,\text{is}} - h_1}{h_2 - h_1} = \frac{h_{4,\text{is}} - h_3}{h_4 - h_3} \quad (6)$$

where W_{com} denotes the total power of the compressor; η_{com} indicates the isentropic efficiency of the compressor; h_3 and h_4 represent the enthalpies at the corresponding state points, respectively; and $h_{2,\text{is}}$ and $h_{4,\text{is}}$ stand for the enthalpies the corresponding state points after an isentropic process of the first-stage and second-stage compressions, respectively.

The total condensing heat capacity could be calculated as follows:

$$Q_{\text{evap}} = m_{\text{first}}(h_1 - h_7) \quad (7)$$

where Q_{evap} denotes the total cooling heat released from the evaporator.

The coefficient of the system could be expressed as follows:

$$\eta = \frac{Q_{\text{evap}}}{W_{\text{com}}} \quad (8)$$

where η represents the COP of the system.

2.3 Modeling verification

Liu^[24] studied the variation in key performance parameters of the flash-tank system for a two-stage rolling piston compressor with a relative intermediate pressure working with R410A. The mathematical model described in this paper was adopted to verify and calculate the theoretical model for the single-stage heat pump system and the vapor injection system. The calculation parameters were selected as follows: condensation temperature of 35 °C and evaporation temperature of 5 °C at relative intermediate pressures of 1.1 P_e , 1.2 P_e , and 1.3 P_e , where P_e denotes the evaporation pressure. The calculated results and the experimental values are compared in Table 3. η_e and η_c represent the experimental and calculated COP values, respectively; the difference denotes the absolute deviation; the percentage indicates the relative deviation of the difference with respect to η_e .

Table 3 Comparison of literature data and model calculation data

Intermediate pressure	η_e	η_c	Difference	Percentage/%
1.1 P_e	3.76	3.747	0.013	0.35
1.2 P_e	3.54	3.536	0.004	0.11
1.3 P_e	3.30	3.381	-0.081	-2.45

Calculation results showed that the difference between the COP of the theoretical circulating system of the refrigeration system calculated by the model described in this paper and the experimental results reported in the literature was less than 0.1; the highest percentage of the difference was only 2.45%, which proved that the pro-

posed model was feasible for a performance analysis of the vapor injection refrigeration system enhanced with a flash tank.

3 Results and Discussion

A performance analysis was conducted on the flash tank vapor injection refrigeration system across a range of first-stage compression ratios from 1.1 to 6.5. Based on the calculation of Eqs. (1) to (4), the vapor mass flow rate supplied by the flash tank under varying first-stage compression ratios (i. e., intermediate pressures) was determined, as shown in Fig. 4.

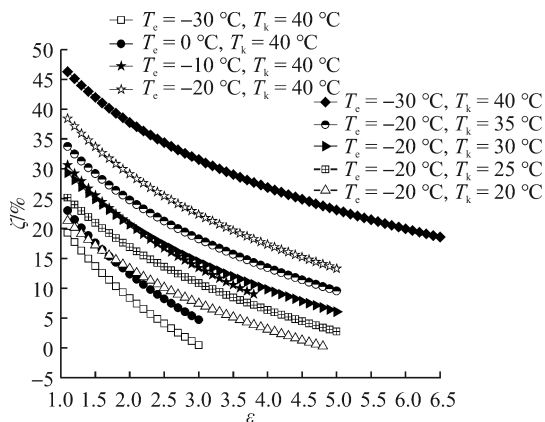


Fig. 4 Variation of relative injected vapor mass from the flash tank with the first-stage compression ratio at different evaporation and condensation temperatures

The relative mass of vapor injected from the flash tank decreased as the first-stage compression ratio increased before injection, suggesting that it decreased with elevated intermediate pressure—a behavior governed by the thermophysical properties of R1233zd (E). Under lower-pressure conditions, the working fluid exhibited a lower dryness fraction and a correspondingly higher evaporation enthalpy, resulting in greater vapor generation for the compressor. When the first-stage compression ratio was held constant, the relative injected vapor mass dropped with rising evaporation temperature. This occurred because an increase in the evaporation temperature correspondingly increased the evaporating pressure. Consequently, both the compressor inlet pressure and the discharge pressure from the first compressor stage increased, resulting in higher intermediate pressure. Hence, the relative mass of vapor supplied by the flash tank decreased. Using the relative injected vapor mass as a key parameter, we computed the system condensation heat, total compressor power consumption, and the system COP based on the parameter's value across various operating conditions.

Fig. 5 presents the variation of COP of the vapor injection refrigeration system with a flash tank. We observed that system performance was significantly influenced by

operating temperature. For example, at a fixed condensation temperature of 40 °C, the COP increased from about 1.86 to about 4.53 as the evaporation temperature increased from −30 to 5 °C. Conversely, the COP reduced with increasing condensation temperature. When the evaporation temperature was maintained at −20 °C, the COP decreased from 3.58 to 2.29 as the condensation temperature increased from 20 to 40 °C. Within a specific cycle, the COP initially increased and then decreased as the first-stage compression ratio increased. Therefore, an optimal intermediate working pressure existed, which made the vapor injection refrigeration system with a flash tank using R1233zd (E) achieve the maximum performance coefficient. The first-stage compression ratio parameter that yields the best COP increases with increasing condensation temperature. The optimal first-stage compression ratio and the corresponding maximum COP for different condensation and evaporation temperatures are listed in Table 4.

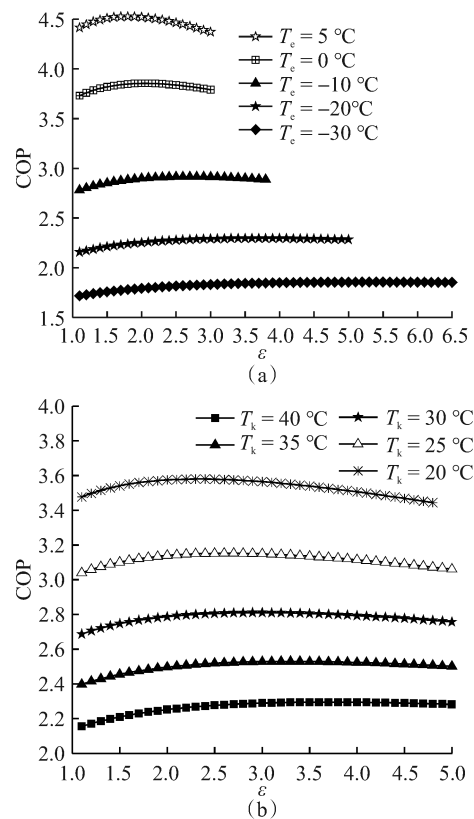


Fig. 5 Variation of performance coefficient with the first-stage compression ratio at different temperatures. (a) Different evaporation temperatures ($T_k = 40\text{ °C}$); (b) Different condensation temperatures ($T_c = -20\text{ °C}$)

As the first-stage compression ratio increased, the system's cooling capacity decreased at a less pronounced rate than the compressor power consumption. Consequently, the COP exhibited an upward trend with increasing compression ratio. Along with the pressure ratio being larger, the system's intermediate pressure increased, gradually decreasing the injected vapor mass flow rate,

Table 4 Optimum first-stage compression ratio with the maximum COP calculated with different methods

$T_e/^\circ\text{C}$	$T_k/^\circ\text{C}$	ε_{om}	ε_{oc}	Difference	Percentage/%
-30	20	3.21	3.173 6	0.036 4	1.13
	25	3.66	3.684 9	-0.024 9	-0.68
	30	4.17	4.196 1	-0.026 1	-0.63
	35	4.67	4.707 4	-0.037 4	-0.80
	40	5.31	5.218 6	0.091 4	1.72
-20	20	2.33	2.327 4	0.002 6	0.11
	25	2.63	2.677 5	-0.047 5	-1.81
	30	2.95	3.027 7	-0.077 7	-2.63
	35	3.31	3.377 9	-0.067 9	-2.05
	40	3.72	3.728 1	-0.008 1	-0.22
-10	20	1.75	1.708 9	0.041 1	2.35
	25	1.95	1.945 1	0.004 9	0.25
	30	2.18	2.181 4	-0.001 4	-0.06
	35	2.43	2.417 7	0.012 3	0.51
	40	2.7	2.653 9	0.046 1	1.71
0	20	1.35	1.318 1	0.031 9	2.36
	25	1.5	1.487 6	0.012 4	0.82
	30	1.67	1.657 1	0.012 9	0.77
	35	1.85	1.826 6	0.023 4	1.26
	40	2.04	1.996 1	0.043 9	2.15
5	20	1.2	1.208 2	-0.008 2	-0.68
	25	1.34	1.362 0	-0.022 0	-1.64
	30	1.49	1.515 8	-0.025 8	-1.73
	35	1.63	1.669 6	-0.039 6	-2.43
	40	1.81	1.823 4	-0.013 4	-0.74

and the cooling capacity increased because of the increase in the vapor-injected mass flow rate. The compressor power decreased gradually due to a decrease in the vapor-injected mass flow rate, which reduced the compressor power's decreasing tendency. Therefore, the COP of the system first increased with increasing first-stage compression ratio, and then showed a decreasing trend.

As shown in Fig. 6, with a constant condensation temperature and a fixed first-stage compression ratio, increasing the evaporation temperature increased the intermediate system pressure. Consequently, the flash gas mass flow rate decreased, which in turn reduced both the system cooling capacity and the compressor power consumption. However, the decrease in cooling capacity was smaller than that in compression power, thereby increasing the performance coefficient of the refrigerating system with increasing evaporation temperature. This was mainly because the refrigerating capacity of the system increased as vapor injection increased, thereby simultaneously increasing the power consumption of the second compressor stage. When the evaporation temperature remained constant, and the condensation temperature

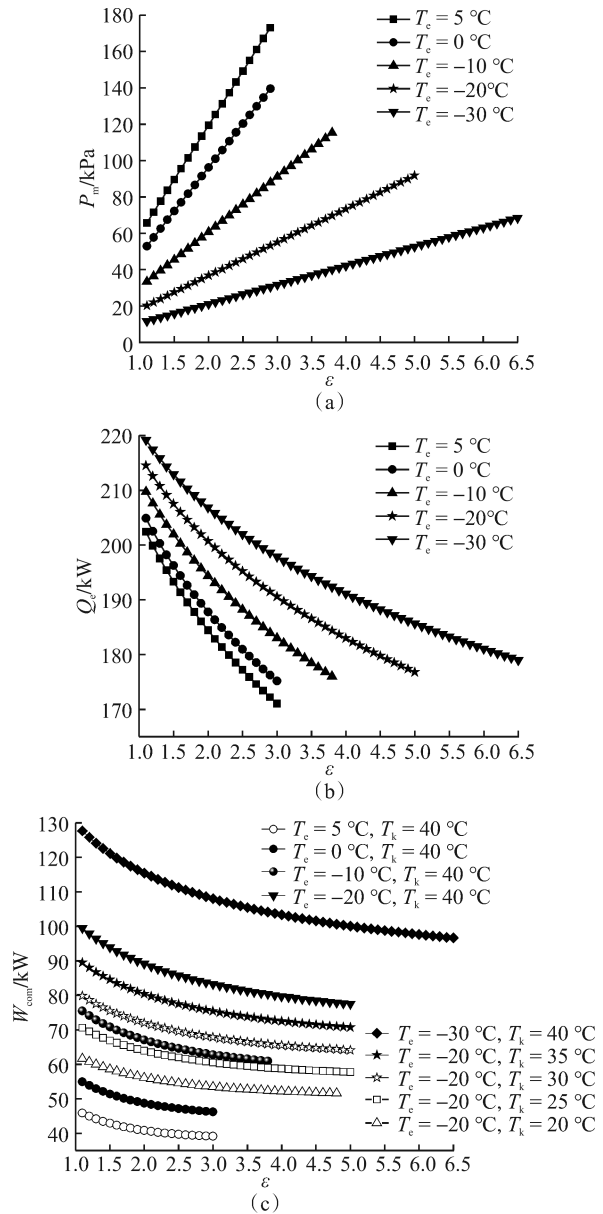


Fig. 6 Variations of system parameters with the first-stage compression ratio under different evaporating and condensation temperatures. (a) Pressure in the flash tank ($T_k = 40^\circ\text{C}$); (b) Total cooling capacity ($T_k = 40^\circ\text{C}$); (c) Compressor power consumption

increased, the mass flow rate of the injected vapor decreased with increasing first-stage compression ratio, leading to a reduction in both heat absorption and compression power. Meanwhile, the rise in condensation temperature increased the power consumption of the second-stage compressor. Therefore, it was necessary to increase the first-stage compression ratio to achieve a relatively low vapor-injected mass flow rate and reduce the power consumption of the second-stage compression, thereby increasing the system's COP.

As shown in Fig. 7, the relationships between the optimum first-stage compression ratio and the maximum COP were calculated using the mathematical model; the

resulting data were fitted with quadratic and linear polynomials. Based on the calculated data and fitting curves, the relationship between the optimum first-stage compression ratio for maximum COP and the system's evaporating and condensation temperatures was obtained; the equation could be expressed as follows:

$$\varepsilon = (0.064013 + 0.0339T_k) - (0.01041 + 8.638 \times 10^{-4}T_k)T_e + (2.0 \times 10^{-4} + 4.7 \times 10^{-5}T_k)T_e^2 \quad (9)$$

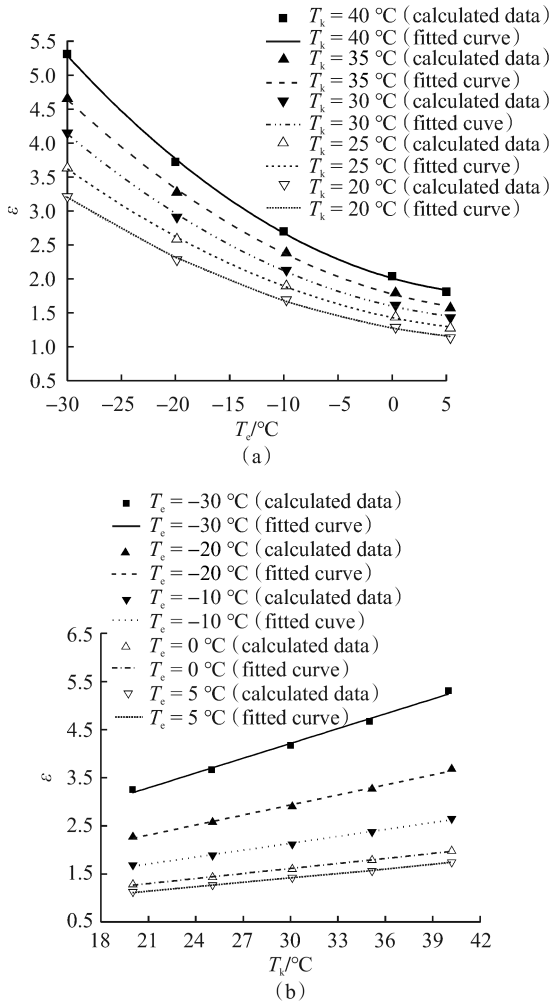


Fig. 7 Variation of first-stage compression ratio of vapor injection refrigeration system with the best COP along with different temperatures at various condensation temperatures. (a) Different condensation temperatures; (b) Different evaporation temperatures

To verify the accuracy of Eq. (9), the optimum first-stage compression ratio corresponding to the maximum COP can be calculated using Eq. (9) and the mathematical model in this work. The results are presented in Table 4. Here, ε_{om} and ε_{oe} represent the optimum first-stage compression ratios, with the maximum COP calculated using the mathematical model and Eq. (9), respectively; the difference denotes the absolute deviation between them, and the percentage indicates the relative deviation of the difference relative to ε_{om} .

As shown in Table 4, the optimum first-stage compression ratio obtained from the fitting formula was very close to that calculated using the mathematical model. The differences between the two sets of results were generally small, with the percentage deviations mostly within $\pm 3\%$ (ranging from -2.63% to 2.36%). In particular, at -30 °C, the maximum positive deviation (1.72%) occurred at 40 °C, while the minimum negative deviation (-0.80%) was observed at 35 °C. For an evaporation temperature of -20 °C, the most significant negative deviation (-2.63%) appeared at 30 °C, and the positive deviation was relatively small (up to 0.11%). At evaporation temperatures of -10 , 0 , and 5 °C, the percentage deviations remained at a low level (mostly within $\pm 2.5\%$), with only individual cases showing a slight increase (e. g., 2.36% at 0 °C evaporation and 20 °C condensation).

Overall, the results from the mathematical model were highly consistent with those obtained using Eq. (9), indicating good agreement between the two methods and verifying the reliability of the calculations. Eq. (9) can be used to guide operating parameter settings for the vapor injection refrigeration system during application, and the thermodynamic model can be useful for intelligent control.

4 Conclusions

(1) The COP calculated using both the mathematical model and the fitted correlation showed a clear positive correlation with the evaporation temperature. As T_e increased from -30 to 5 °C, the COP generally increased; for instance, from about 1.86 at $T_e = -30$ °C and $T_k = 40$ °C to about 4.53 at $T_e = 5$ °C and $T_k = 40$ °C. Conversely, the COP decreased as T_k rose. At a fixed T_e of -20 °C, the COP decreased from 3.58 to 2.29 as T_k increased from 20 to 40 °C.

(2) When the condensation temperature and the first-stage compression ratio were held constant, both the system cooling capacity and the compressor power consumption decreased with increasing evaporation temperature.

(3) For the flash-tank vapor injection system using R1233zd(E), an optimal first-stage compression ratio existed to maximize the COP. Results obtained using the detailed mathematical model and the simplified fitted equation showed good agreement, with relative deviations primarily within $\pm 3\%$, particularly between -2.63% and 2.36% .

(4) The system exhibited the maximum COP at the optimal intermediate pressure. A correlation was fitted to directly calculate the optimal first-stage compression ratio based on the evaporating and condensing temperatures. This equation facilitates the determination of the

optimal operating parameter across different conditions and provides valuable guidance for the intelligent control of refrigeration systems.

References

- [1] ZHANG J R, MEERMAN H, BENDERS R, et al. Comprehensive review of current natural gas liquefaction processes on technical and economic performance [J]. *Applied Thermal Engineering*, 2020, 166: 114736.
- [2] SANTOS A F, GASPAR P D, DE SOUZA H J L, et al. Refrigeration of COVID-19 vaccines: Ideal storage characteristics, energy efficiency and environmental impacts of various vaccine options [J]. *Energies*, 2021, 14 (7): 1849.
- [3] ZHAO S Q, WU J W, GUO Z Q, et al. Magnetic field technology in improving the quality of food refrigeration and freezing: Mechanisms, applications, and challenges [J]. *Journal of Stored Products Research*, 2024, 106: 102254.
- [4] ZHANG T, XIAO W, PAN Y, et al. Space energy efficiency improvement and structure optimization in green and low-carbon renovation of existing buildings II: Models and transformations [J]. *Journal of Southeast University (Natural Science Edition)*, 2025, 55 (6): 1502-1512. (in Chinese)
- [5] GAO M, TANG H J, MA Z P, et al. Data-driven study on key factors for carbon reduction from contaminated soils at regional level: An example of cement kiln co-processing technology [J]. *Journal of Southeast University (Natural Science Edition)*, 2025, 55 (3): 903-911. (in Chinese)
- [6] ZHANG Z, FANG A Z, ZHAO X D, et al. Study on energy-saving effect of a separate radiative cooling-air conditioning hybrid refrigeration system [J]. *Journal of Southeast University (Natural Science Edition)*, 2024, 54(6): 1561-1566. (in Chinese)
- [7] KARTHICK L, SENTHIL MURUGAN V, LEON JOSEPH LEON S, et al. Energy performance of a compression refrigeration cycle using environment-friendly refrigerants [J]. *Materials Today: Proceedings*, 2022, 66: 1519-1525.
- [8] JIANG J T, HU B, WANG R Z, et al. Theoretical performance assessment of low-GWP refrigerant R1233zd (E) applied in high temperature heat pump system [J]. *International Journal of Refrigeration*, 2021, 131: 897-908.
- [9] LEE S J, SHON B H, JUNG C W, et al. A novel type solar assisted heat pump using a low GWP refrigerant (R-1233zd(E)) with the flexible solar collector [J]. *Energy*, 2018, 149: 386-396.
- [10] MOLÉS F, NAVARRO-ESBRÍ J, PERIS B, et al. Experimental evaluation of HCFO-1233zd-E as HFC-245fa replacement in an Organic Rankine Cycle system for low temperature heat sources [J]. *Applied Thermal Engineering*, 2016, 98: 954-961.
- [11] MWESIGYE A, KIAMARI A, DWORKIN S B. Energetic optimization and exergetic performance investigation of an ejector refrigeration system using HCFO-1233zd(E) as a refrigerant [J]. *International Journal of Refrigeration*, 2020, 112: 155-171.
- [12] RAI A, TASSOU S A. Environmental impacts of vapour compression and cryogenic transport refrigeration technologies for temperature controlled food distribution [J]. *Energy Conversion and Management*, 2017, 150: 914-923.
- [13] WALID FARUQUE M, HAFIZ NABIL M, RAIHAN UDDIN M, et al. Thermodynamic assessment of a triple cascade refrigeration system utilizing hydrocarbon refrigerants for ultra-low temperature applications [J]. *Energy Conversion and Management*, 2022, 14: 100207.
- [14] KHAN Y, NAQIB-UL-ISLAM S M, FARUQUE M W, et al. Advanced cascaded recompression absorption system equipped with ejector and vapor-injection enhanced vapor compression refrigeration system: ANN based multi-objective optimization [J]. *Thermal Science and Engineering Progress*, 2024, 49: 102485.
- [15] BAO S, DU K, CAI X, et al. Performance analysis of ammonia-water absorption/compression combined refrigeration cycle [J]. *Journal of Southeast University (English Edition)*, 2014, 30(1): 60-67.
- [16] NABIL M H, KHAN Y, FARUQUE M W, et al. Thermo-economic assessment of advanced triple cascade refrigeration system incorporating a flash tank and suction line heat exchanger [J]. *Energy Conversion and Management*, 2023, 295: 117630.
- [17] D'ANGELO J V H, AUTE V, RADERMACHER R. Performance evaluation of a vapor injection refrigeration system using mixture refrigerant R290/R600a [J]. *International Journal of Refrigeration*, 2016, 65: 194-208.
- [18] WU H G, LIN K L, HUANG H, et al. Research on effects of vapor injection on twin-screw compressor performance [J]. *International Journal of Refrigeration*, 2020, 118: 483-490.
- [19] WANG J J, QV D, YAO Y, et al. The difference between vapor injection cycle with flash tank and intermediate heat exchanger for air source heat pump: An experimental and theoretical study [J]. *Energy*, 2021, 221: 119796.
- [20] QI H J, LIU F Y, YU J L. Performance analysis of a novel hybrid vapor injection cycle with subcooler and flash tank for air-source heat pumps [J]. *International Journal of Refrigeration*, 2017, 74: 540-549.
- [21] JUNG J, JEON Y, CHO W, et al. Effects of injection-port angle and internal heat exchanger length in vapor injection heat pumps for electric vehicles [J]. *Energy*, 2020, 193: 116751.
- [22] LU Z N, XIAO C P, QIU Z J, et al. Thermodynamic analysis of high temperature heat pump vapor system with intermediate vapor injection [J]. *China Heavy Equipment*, 2023(2): 8-11, 54. (in Chinese)
- [23] ZOU P. Experiment on performance of integral two-stage screw compressor unit with economizer vapor injection [J]. *Refrigeration and Air-Conditioning*, 2025, 25(4): 77-81, 86. (in Chinese)
- [24] LIU Q. Experimental study on two-stage compression heat pump system coupled with dual-cylinder rolling piston compressor [D]. Beijing: Beijing University of Technology, 2013.

含闪蒸罐的R1233zd(E)蒸汽注入制冷系统性能分析

樊海彬^{1,2}, 赵盼盼², 贾磊², 黄世芳^{1,3}, 张小松^{1,3}

(1. 东南大学能源与环境学院, 南京 210096; 2. 合肥通用机械研究院有限公司, 合肥 230031; 3. 东南大学低碳型建筑环境设备与系统节能教育部工程研究中心, 南京 210096)

摘要: 将R1233zd(E)与闪蒸罐蒸汽注入技术相结合可以提升制冷性能并弥补其容积制冷量偏低的不足。针对采用R1233zd(E)的闪蒸罐蒸汽注入制冷系统,建立了基于质量与能量守恒的热力学数学模型。当蒸发温度为 $-30\sim 5\text{ }^{\circ}\text{C}$,冷凝温度为 $20\sim 40\text{ }^{\circ}\text{C}$,一级压缩比为 $1.1\sim 6.5$ 时,系统计算并分析了性能系数(COP)、制冷量及压缩机功耗等参数的变化规律,同时构建了基于运行温度的最佳一级压缩比拟合方程。结果表明,系统COP随蒸发温度升高而增大,当冷凝温度为 $40\text{ }^{\circ}\text{C}$,蒸发温度由 $-30\text{ }^{\circ}\text{C}$ 升至 $5\text{ }^{\circ}\text{C}$ 时,COP从约1.86提升至约4.53。COP随冷凝温度升高而降低,当蒸发温度为 $-20\text{ }^{\circ}\text{C}$,冷凝温度由 $20\text{ }^{\circ}\text{C}$ 升至 $40\text{ }^{\circ}\text{C}$ 时,COP从约3.58降至约2.29。系统存在最佳中间压力以获得最大COP,推导出的最佳一级压缩比关联式与模型计算值的偏差主要集中在 $\pm 3\%$ 以内,可为系统运行参数设定和智能控制提供理论指导。

关键词: R1233zd(E); 闪蒸罐; 蒸汽注入制冷系统; 性能系数(COP)

中图分类号: TB61