EXPERIMENTAL STUDY ON ALTERNATIVE PERFORMANCE OF R32/R1270 IN LOW-TEMPERATURE HEATING HEAT PUMP

by

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Original scientific paper
https://doi.org/10.2298/TSCI250818213J

The effect of refrigerant charge on the system performance of quasi-two-stage compression heat pump (QCHP) with replacement refrigerant of R32/R1270 (15/85 wt.%) (B for short) for district heating was studied under nominal conditions based on the self-designed QCHP test system and the feasibility of replacing R22 with B was discussed under variable evaporation temperature conditions. The results indicated that the refrigerant charge showed distinct impact on the COP_h, heating capacity and discharge temperature of the system using B. The system with B at the optimal charge of 1.2 kg (B_{opt}) achieved the maximum heating capacity of 7.455 kW and the maximum COP_h of 3.35 at nominal conditions. The COP_h and heating capacity of both systems with B_{opt} and R22 had similar trends with the change in evaporating temperature, while their discharge temperatures showed different changes from the heating capacity. Compared to those of R22 system, B_{opt} system showed improvement in COP_h and heating capacity of 9.73%~17.54% and 50.84%~53.40% respectively in the evaporating temperature of –15~–5 °C, with the distinctly lower discharge temperature. Thus, B_{opt} has remarkable replacement advantages in the QCHP. The lower discharge temperature highlights the outstanding benefits of B_{opt} system in the reliable heating at lower ambient temperature and longer service life. This study provides reliable reference data for refrigerant replacement in heat pumps and low-carbon heating.

Keywords: *QCHP, R32/R1270, low-carbon heating, refrigerant charge, refrigerant replacement*

Introduction

Given that heating energy consumption (electricity, coal, gas, oil included) makes up 50% of building energy use, China has been actively engaged in the exploration of efficient and cleaner heating technologies, with the objective of responding to the Paris Climate Agreement [1, 2]. Heat pump heating has the advantages of saving energy, and it has become a necessary

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way to achieve carbon neutrality in the building division [3]. High GWP refrigerants are required by regulations such as the Kigali Amendment to be explicitly restricted or even phased out. Thus, the search for efficient alternatives is the current priority of exploration in the field of heat pumps [4]. Zeotropic blends with temperature glide effects that can prominently improve system performance are believed by many scholars to have great potential for replacement.

The feasibility of low GWP refrigerant blends replacing conventional refrigerants in heat pumps has been widely researched by many scholars and excellent outcomes have been achieved. Wang *et al*. [5] demonstrated theoretically that R32/R1234ze(E) (40/60) has potential substitution advantages in QCHP for clean hot water. Ju *et al*. [6] experimentally found that heat pump water heater (HPWH) with R744/R290 (12/88) achieved distinctly better COP_h and heating capacity, as well as markedly lowered discharge temperatures, than those of R22 system. Lee *et al*. [7] experimentally revealed that the COP_h of heating heat pump with R32/R152a (50/50) under heating conditions was increased by 15.8% in comparison with that of R22 system. Tian *et al*. [8] carried out the experimental exploration of the feasibility of R32/R290 (68/32) as an alternative to R410A in heating heat pump and found that the heating capacity of R32/R290 (68/32) system was increased by 8.2%~16.4% compared with that of R410A system at environmental temperature of 7 ℃. Hakkaki-Fard *et al*. [9] theoretically investigated R32/R744 (80/20) heating heat pump in cold climate of –8.3 ℃ with a 30% increase in heating capacity compared to that of R410A system. Chien *et al*. [10] simulated and experimentally found that R32/R1270 (70/30) has a pool boiling performance 8.67% higher than that of R410A and performs better than R32 at low heat fluxes, which showed a good substitution potential.

One of the most remarked elements influencing the performance of small heat pumps is refrigerant charge. Yang *et al*. [11] experimentally studied how refrigerant charge affects the performance of R290 heating heat pump, with results showing that the system had an optimal charge (300 g) to achieve maximum heating capacity and highest COP_h at environmental temperature of 0 °C. Mehrabi *et al*. [12] conducted a theoretical assessment of how refrigerant charge impacted the heat pump's performance. It was shown by the results that undercharging affected the COP_h and heating capacity of heat pump more markedly than overcharging did. Wang *et al*. [13] probed experimentally that the COP_h, economic returns and environmental benefits of R744 instant HPWH were maximized at the optimal charge level. Ghoubali *et al*. [14] undertook an experimental exploration into the impact of the charge on superheat degree and found that the superheat degree of HPWH with R290 decreases and then tended to become stable as the charge increases. Corberan *et al*. [15] theoretically showed that the charge mainly affected the subcooling degree and condensing pressure, which in turn affected the COP_h and optimal charge of HPWH with R290.

In summary, the alternative use of appropriate refrigerant blends can ameliorate the performance of heat pump, and the charge is a crucial variable influencing the efficiency of energy utilization. The R32 has no ozone depletion potential (ODP) and good thermodynamic characteristics, but a slightly higher GWP (675) and high discharge pressure and temperature [16]. On the basis of superior thermodynamic performance, R1270 also has significant advantages including very low GWP, zero ODP, and low acquisition cost [17]. The R32/R1270 can provide complementary benefits and is expected to be a promising alternative option. However, the potential of R32/R1270 in low temperature QCHP for heating hot water has yet to be fully investigated. Based on a self-designed and constructed QCHP test system for heating hot water, the bearing of refrigerant charge on the performance of the QCHP with low GWP blend of R32/R1270 (15/85 wt.%) (B for short) was investigated through experimental analysis under

the nominal heating condition and the potential for replacing R22 with B_{opt} was discussed. The aim was to promote the development of refrigerant substitution for heating heat pumps and cleaner and more efficient heating technologies.

Materials and methods

Experimental methods such as the test set-up (including measuring instruments), data processing and their uncertainty analysis and design of test conditions were introduced in detail, aiming to clarify the experimental details and the feasibility of the results for the research on optimizing the charge quantity of the working fluid and verifying the feasibility of working fluid substitution. The preliminary theoretical evaluation was conducted on the substitution performance of B based on the analysis of its thermodynamic properties and flammability.

Experimental set-up and measurement system

As shown in fig. 1, the QCHP test system is mainly composed of a refrigerant circulation system, a heating brine reservoir, a cooling water reservoir and a data measuring and collecting system. Comprising primarily a rotor vapor-injection compressor, an economizer, a condenser, an evaporator, electronic expansion Valve 1, electronic expansion Valve 2, and other components is the refrigerant circulation system. The electronic expansion Valves 1 and 2 respectively control the refrigerant flow rate of the main circuit and the refrigerant flow rate that reaches the intermediate intake port of the compressor. By adjusting the opening degree of ex-

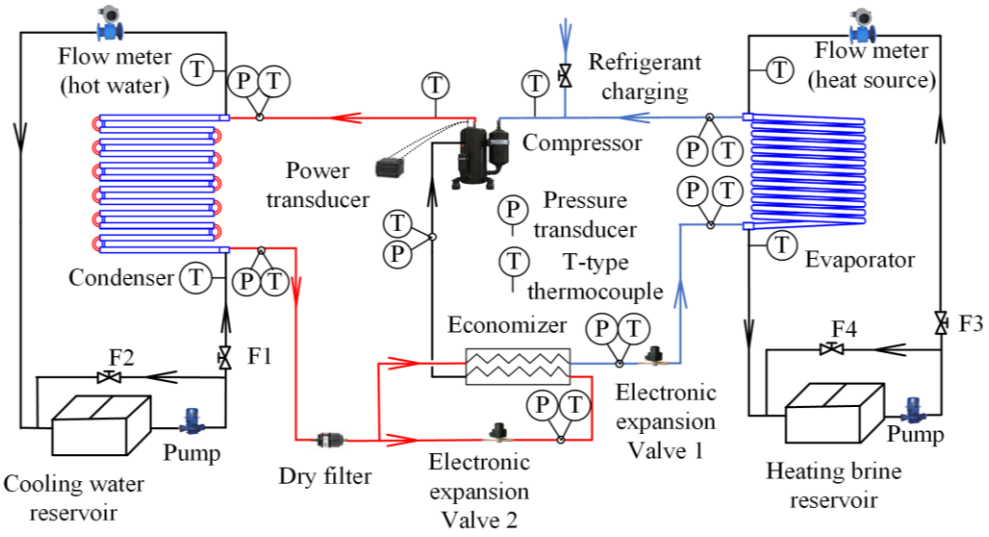

Figure 1. Schematic diagram of the QCHP test system

pansion Valve 2, it is possible to significantly reduce the discharge temperature and significantly increase the COP_h and the heating capacity under low-temperature conditions. Considering the high operation pressure of R32, the R410A rotor vapor-injection compressor specifically designed for heat pump with the displacement of 32.5 ml per rev was opted for to guarantee the safety of test procedure. The condenser is a counter-flow serpentine casing heat exchanger. The evaporator is a counter-flow spiral casing heat exchanger with a spiral coil diameter of 30 cm. Other structural parameters for both of the heat exchangers are contained in tab. 1. The economizer is a parallel flow plate heat exchanger with the heat transfer area of 0.144 m^2. Heating brine

reservoir and cooling water reservoir provide heat source and hot water at the inlet temperature required for the experiment, respectively. The brine was used to simulate the low temperatures of outdoors, and it could be used normally in the temperature range of –50~218 °C.

Table 1. Structural dimension of condenser and evaporator

Parameters		Condenser	Evaporator
Outer tube	External diameter [mm]	20	28
	Internal diameter [mm]	18	25
Inner tube	External diameter [mm]	12.7	15.88
	Internal diameter [mm]	11.1	13.48
Length [m]		17.6	14

The data measuring and collecting system includes *T*-type thermocouples, pressure transmitters, flow meters, power transducer and so on. And their detailed distributions of measurement points were revealed in fig. 1. The *T*-type thermocouples (±0.2 °C) served to measure the temperature of the fluids. The pressure transducers (±0.3%) were designed for measuring refrigerant pressure. Flow meters (±0.2%) were intended for measuring hot water flow and brine flow, and power transducers (±0.2%) were utilized for measuring compressor power consumption. The real-time collection of experimental data was accomplished using the Keithley data acquisition instrument.

Data processing and uncertainty analysis

The heating capacity, coefficient of performance (COP_h), pressure ratio and relative middle pressure, K, can be calculated by eqs. (1)-(4), respectively, [18, 19]. The P_m is middle pressure. The quadratic power method was used to calculate the uncertainties of the indirect parameters. The uncertainties of heating capacity, COP_h and pressure ratio for the QCHP with B at the optimal charge under the normal condition, as in tab. 2, were 1.40%, 1.44%, and 0.56% respectively:

$$Q_h = \Delta M_{hw} c_p (t_{hw,o} - t_{hw,i}) \tag{1}$$

$$COP_h = \frac{Q_h}{W_{com}} \tag{2}$$

$$R = \frac{P_{dis}}{P_{suc}} \tag{3}$$

$$K = \frac{P_m}{\sqrt{P_{suc} P_{dis}}} \tag{4}$$

Experimental condition

To evaluate the substitution potential of B and to find its optimal charge, the system performance test conditions of the QCHP unit were set concerning the [20] and GB/T25127.2-2020 [21], and the parameters were listed in tab. 2. The test study on the optimization of refrigerant charge was carried out under the nominal heating condition. The evaporating temperature was identified as the refrigerant temperature at the evaporator inlet and its was reduced by 12 °C compared to the inlet temperature of the brine. For the convenience of experiment, the brine

flow rate was kept constant during the test. As indicated in eq. (4), the set values of relative middle pressures were 1 in this study.

Table 2. Setting of experimental parameters

Classification	$t_{hw,I}$ [°C]	$t_{hw,o}$ [°C]	t_b [°C]	$\dot{m}_b$ [kgs^{-1}]	t_e [°C]
Normal heating conditions	35	50	7	0.443	–5
Variable evaporating temperature conditions	35	50	–3~7	0.443	–15~–5

Refrigerant characteristics

The thermodynamic properties of refrigerant can directly or indirectly reflect the system performance, so it is an important evaluation index of the potential replacement. The main refrigerant properties of B and R22 were gained from REFPROP 10.0 as shown in tab. 3. The normal boiling point (NBP) of B is the arithmetic mean of the normal dew point and normal bubble point. The B has zero ODP and low GWP. The B has the marked temperature glide, which improves the temperature matching between the fluids in the heat transfer processes and is expected to increase the system energy efficiency. The NBP of B is markedly less than that of R22, which indicates that B system operates at higher pressures under the same condition and is more beneficial in low-temperature settings. The critical temperature of B is 10.62 °C less than that of R22, resulting in a remarkably enhanced heating capacity for B system under identical operation parameters.

Table 3. Main refrigerant characteristics

Parameters	B	R22
ODP	0	0.034
GWP	103	1760
Δt_g [°C], P_{sa} = 101.3 kPa	5.52	–
t_{NBP} [°C], P_{sa} = 101.3 kPa	–52.83	–40.81
P_{cr} [Mpa[	4.8619	4.9900
t_{cr} [°C]	85.53	96.15
P_{sv} [Mpa], t_{sa} = 25 °C	1.2940	1.0439

Table 4 primarily presented multiple combustion characteristic parameters for R32 and R1270, which served as the basis for a theoretical analysis of how their safety performance changed when mixed. From tab. 4, it could be inferred that mixing R32 in R1270 can improve the safety level of R1270. The addition of R32 increases both the lower and upper flammable limits (LFL and UFL) of B compared to those of R1270. In addition, the addition of R32 can improve the minimum ignition energy of R1270, which has a positive impact on improving the safety of R1270.

Results and discussion

Effect of refrigerant charge

The bearings of refrigerant charge on COP_h, power consumption and heating capacity of B system at the nominal condition were revealed in fig. 2. The increase in refrigerant charge resulted in a speedy rise and subsequent slow rise in COP_h of B system, reaching a maximum before declining rapidly. This outcome was because of the combined effect of heating capacity

Table 4. Main combustion characteristics of R32 and R1270

Parameters	R32	R1270
Safety level	A2L	A3
LFL [vol.%]	14.4	2.0
UFL [vol.%]	29.3	11.1
Minimum ignition energy [°C]	648	460

and power consumption, with the former playing a dominant role. The COP_h of system with B at the optimal charge of 1.2 kg (B_{opt} for short) reached a maximum of 3.35. The power consumption showed an overall slow increase with increasing charge of B, chiefly because refrigerant charge in the compressor increased. The heating capacity of B system demonstrated a fast increase followed by a gradual increase with the rise in refrigerant charge. After attaining the maximum of 7.455 kW at the optimal charge, a gradual decline in heating capacity was observed. This phenomenon was primarily attributable to the fact that when the refrigerant charge was at a relatively low level, an increase in refrigerant could precipitate a swift surge in the condenser's heat transfer capacity, which led to a sharp increase in the heating capacity. When the charge gradually approached its optimal level, the condenser's heat transfer capacity would be fully utilized, accompanied by a continuous increase in discharge pressure, which caused a tendency for the growth of heating capacity to level off. Excessive refrigerant charge resulted in the refrigerant accumulation in the area near the condenser outlet, that brought about a sharp reduction in the effective heating transfer area of condenser and a decline in heating capacity [22]. Power consumption rose slowly as the charge of B increased, which resulted in a small change in power consumption as the experimental operating conditions were kept constant.

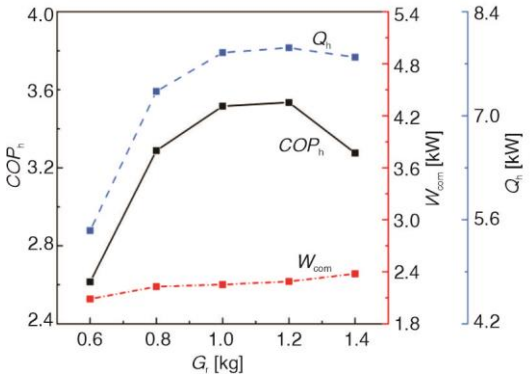

Figure 2. Variation of COP_h, power consumption and heating capacity for different refrigerant

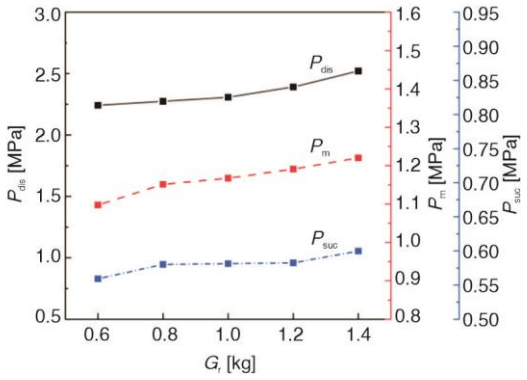

Figure 3. Variation of refrigerant charges of B pressures for different refrigerant charges of B

Figure 3 illustrates the changes in discharge pressure and suction pressure with the charge of B under the nominal heating condition. As refrigerant charge of B increased, the discharge pressure increased gradually, mainly because refrigerant B accumulated in the condenser [23]. The discharge pressure of B_{opt} system stood at 2.397 MPa. The bearing of charge of B on the suction pressure was slight in all cases, mainly because the evaporator pressure decrease changed little across different charge levels and the evaporating temperature remained constant. The middle pressure of B system was also slowly increased by the combined effects of the discharge and the suction pressures.

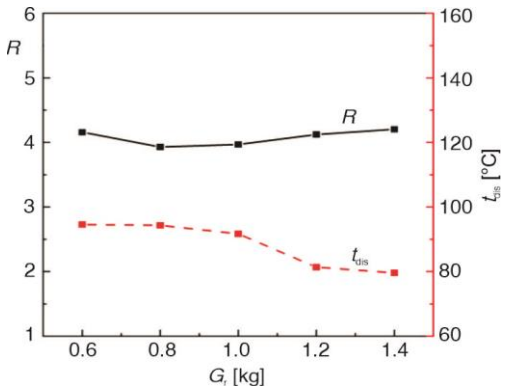

Figure 4. Variation of pressure ratio and discharge temperature for different refrigerant charges of B

The pressure ratio and discharge temperature as affected by refrigerant charge of B under the nominal condition were depicted in fig. 4. Under the combined effects of discharge pressure and suction pressure, the pressure ratio of B system first decreased and then increased with the increase in refrigerant charge of B. The discharge temperature exhibited a gradual decline followed by a steep drop as the refrigerant charge of B increased. Its discharge temperature was 81.37 °C at the optimal charge of 1.2 kg. It was evident that when the refrigerant charge of B was altered, the discharge temperature and the pressure ratio exhibited opposing trends. This suggests that the pressure ratio had a relatively minor impact on the discharge temperature, while it was mainly influenced by parameters such as the suction temperature, isentropic efficiency of compression and the concentration migration of B.

Evaluation of alternative performance

In the following section, the feasibility of replacing R22 with B_{opt} in the QCHP was investigated under different evaporating temperature conditions. Our team's experimental research found that the optimal charge of R22 in the QCHP test system, see fig. 1, was 2.2 kg under the nominal heating condition. The optimal charge of B_{opt} was 45.45% lower than that (2.2 kg) of R22. Consequently, The secondary greenhouse impact connected to refrigerant manufacturing, distribution and reclamation was remarkably diminished through the employment of B_{opt}.

As observed in fig. 5(a), the heating capacity of both systems exhibited a notable decline with the reduction in evaporating temperature. The reason was the drop in evaporating temperature caused a sudden decrease in suction refrigerant density of compressor, which in turn led to the marked lessening in the corresponding flow rate of refrigerant. The B_{opt} system was established to have a heating capacity 50.84%~53.40% higher than that of R22 system in the evaporating temperature range of –15~–5 °C. This was primarily resultant from the higher suction density and refrigerant flow rate of B_{opt} system under identical operating condition, which was caused by the lower critical temperature of B, see tab. 3. As fig. 5(b) presents, the precipitous decline in COP_h with the reduction in evaporating temperature for both systems was primarily attributable to the rise in the enthalpy disparity of B_{opt} and R22 between the compressor inlet and outlet and the diminution in heating capacity resulting from the decrease in evaporating temperature. The COP_h of B_{opt} system exhibited an improvement of 9.73%~17.54% compared with that of R22 system within the studied evaporating temperature scope. This improvement was primarily attributed to the temperature glide of B_{opt}, see tab. 3, which was capable of boosting the temperature matching between the heat exchange fluids inside the heat exchangers. Consequently, this led to a notable reduction in the irreversible losses that occurred in the course of heat exchange. Therefore, using B_{opt} as an alternative can contribute to enhancing the heating capacity of the QCHP as well as the effects of energy saving and carbon reduction.

Figure 6 shows the discharge pressure and suction pressure of B_{opt} system and R22 system at different evaporating temperatures. At an equal evaporation temperature, the discharge and suction pressures of B_{opt} system were markedly higher than those of R22 system,

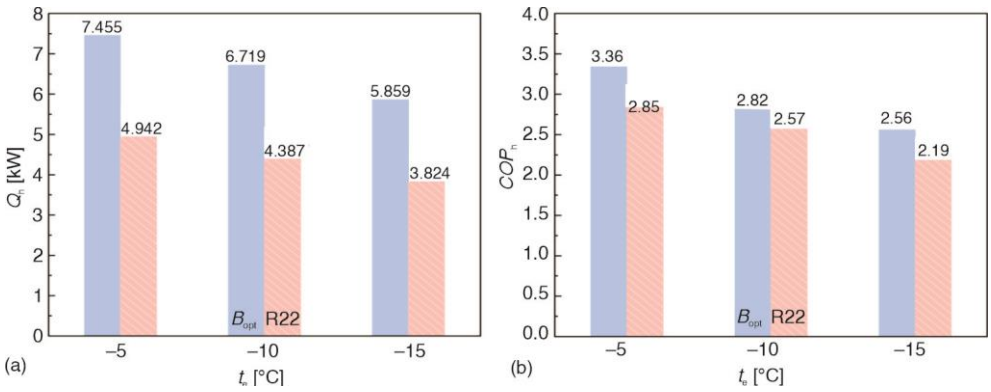

Figure 5. Variation of heating capacity and COP_h for different evaporating temperatures; (a) heating capacity and (b) COP_h

primarily attributed to the lower NBP of B_{opt}, as illustrated in tab. 3. For both systems, the discharge pressures changed slightly as the evaporating temperature declined. This phenomenon was primarily associated with the relatively minor alterations in condenser operational parameters. The suction pressures of both systems declined precipitously as the evaporation temperatures decreased. When the ambient temperature at the time of heating was lower than the ambient temperature under the experimental conditions of this study (7 °C), there was no safety hazard due to high pressure during the operation of the QCHP using B_{opt} instead of R22. Furthermore, the higher suction pressure indicates that negative pressure operation could be more effectively avoided when R22 is replaced by B_{opt} at low temperatures.

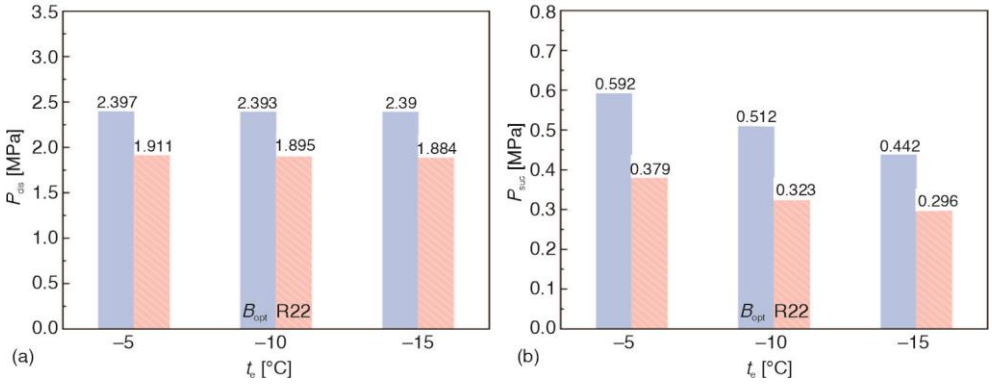

Figure 6. Variation of refrigerant pressures for different evaporating temperatures; (a) discharge pressure and (b) suction pressure

At different evaporating temperatures, the pressure ratio and discharge temperature of both B_{opt} system and R22 system are presented in fig. 7. As fig. 7(a) demonstrates, the pressure ratios of both systems experienced a notable increase with the lessening in evaporating temperature. This phenomenon might be mainly linked to the quick drop in suction pressure, see fig. 6(b). The B_{opt} system had much lower pressure ratios than that of R22 system, mainly as the former's suction pressure was far higher. With the evaporating temperature on the decline, the discharge temperatures of two systems soared fast, primarily owed to the swiftly increasing pressure ratio. As fig. 7(b) shows, B_{opt} system exhibited a discharge temperature 5.94~14.17 °C below the R22 system in the investigated evaporating temperatures scope. This finding conveys that B_{opt}

system has notable advantage of being able to provide reliable district heating at low outdoor temperature, which will give rise to a great extension of the service life of compressor [24].

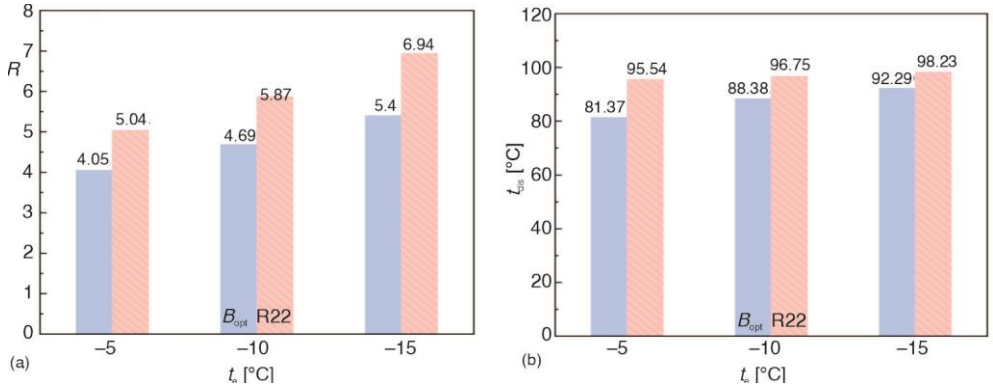

Figure 7. Variation of pressure ratio and discharge temperature for different evaporating temperatures; (a) pressure ratio and (b) discharge temperature

Conclusions

This paper presents the findings of an experimental investigation into the effect of refrigerant charge on the system performance of the QCHP using B at nominal heating condition based on a QCHP test system for district heating. Additionally, the feasibility of replacing R22 with B_{opt} in the QCHP was explored under variable evaporating temperature conditions. The following main research conclusions were obtained:

- With the rise in refrigerant charge, both COP_h and heating capacity of B system at the nominal condition increased rapidly first and then slowly, and then rapidly decreased, while its discharge temperature dropped slowly first and then rapidly.
- The B_{opt} system achieved respectively the highest heating capacity of 7.455 kW and the highest COP_h of 3.35, under the nominal heating condition,. The optimal charge of B was 1.2 kg, which was 45.45% lower than that of R22, hence using B as alternative can realize energy conservation and carbon emission reduction in refrigerant production, transportation and recycling.
- The influence of evaporation temperature on discharge pressures of both systems was negligible, while its impact on their other parameters is significant. The COP_h and heating capacity of both systems rose rapidly with increasing evaporation temperature, while their discharge temperatures exhibited a rapid drop.
- Compared to those of R22 system, the COP_h and heating capacity of B_{opt} system respectively rose by 9.73%~17.54% and 50.84%~53.40% under variable evaporating temperature. Additionally, its discharge temperature dropped by 5.94~14.17 °C. Thus, B_{opt} demonstrates the remarkable potential for replacing R22 in the QCHP, and B_{opt} system can achieve reliable district heating at the lower ambient temperatures and the extension of compressor life.

Acknowledgment

This work was sponsored by the Development Project of Henan Province (No. 231111320900) and Pingdingshan City Industrial Joint Fund Project.

Nomenclature

c_p – specific heat capacity [kJkg^{-1}K^{-1}]
G – charge [kg]

$\dot{m}$ – mass-flow [kgs^{-1}]
P – pressure [MPa]
R – pressure ratio
t – temperature [°C]
Q – heating capacity [kW]
W – power consumption [kW]

Greek symbols

Δ – difference

Subscripts

B – blend of R32/R1270
b – brine
cr – critical
com – compressor
dis – compressor discharge
e – evaporating
g – glide
h – heating
hw,i – heat water inlet
hw,o – heat water outlet
m – middle
opt – optimal
r – refrigerant
sa – saturation
sv – saturation vapor
suc – compressor suction

Acronyms

HPWH – heat pump water heater
LFL – lower flammable limits
ODP – ozone depletion potential
QCHP – quasi-two-stage compression heat pump
NBP – normal boiling point
UFL – upper flammable limits

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Paper submitted: August 18, 2025
Paper revised: October 9, 2025
Paper accepted: October 13, 2025
2026 Published by the Vinča Institute of Nuclear Sciences, Belgrade, Serbia.
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