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A New Mixture Refrigerant for Space Heating Air Source Heat Pump: Theoretical Modelling and Performance Analysis

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Featured Application: This paper proposed a new mixture refrigerant named RHR-1with high energy efficiency, no ODP (ozone depletion potential) and low GWP (global warming potential), which suitable for ASHP (air source heat pump) heating in residential buildings of rural areas in North China.

Abstract: Air source heat pump (ASHP) is becoming a substitute for small coal boilers in rural residential buildings of North China. However, the application of ASHP faces challenges of heating capacity, energy efficiency, ozone depletion potential (ODP) and global warming potential (GWP). Proper refrigerant is a key factor influences the performance of ASHP. In this paper, a new mixture refrigerant named RHR-1 is proposed, which aimed to improve energy efficiency, eliminate ODP and reduce GWP of ASHP refrigerant. The performance of RHR-1 was analyzed and compared with commonly used refrigerants including R134a, R410A, R407C and R22 in terms of heating coefficient of performance (COP_h), compression ratio (CR) and discharging temperature (DT). The results show that, under the design cases, where supply water temperatures vary from 35 °C to 50 °C and outdoor air temperatures vary from −15 °C to 15 °C, the COP_h of RHR-1 are in the range of 2.43–4.93. The COP_h of RHR-1 is higher than other candidates when the supply water temperature is above 40 °C. The CR and DT of RHR-1 are in medium levels of the compared samples. A logarithmic regression equation was deduced to get the relationship of COP_h with temperature difference between supply water and outdoor air which suggested the temperature difference should be controlled within 47.5 °C to get reasonable COP_h. In addition, RHR-1 has no ODP, and its GWP is 279, which is much lower than other candidates. RHR-1 could be a reasonable refrigerant used in ASHP for space heating in North China.

Keywords: air source heat pump; HFC refrigerant; global warming potential; space heating

1. Introduction

Due to air pollutant emissions such as particles, nitrogen oxides and sulfur oxides, coal and gas boiler heating systems were recognized as main causes of atmospheric pollution in North China which led to adverse impacts on human health [1–3]. Air source heat pump (ASHP) is becoming a promising device for space heating in North China due to its energy-savings, environmental friendliness, and application flexibility [4,5]. ASHP has been carried out as a main method in the "coal to electricity" clean heating reform work and appointed to be a promoted substitution for existing small coal boilers in rural residential buildings of North China [5]. However, the utilization of ASHP for space heating still faces challenges such as heating capacity, frosting and low coefficient of performance (COP). The ozone depletion potential (ODP) and global warming potential (GWP) of refrigerants in ASHP constitute other challenges for environment.

To evaluate ASHP for space heating, studies compared its initial investment, running cost and environmental impacts with conventional space heating systems [5–10]. Zhang et al. [5] showed that low temperature ASHP heating mode has lower primary energy consumption and lower dioxide pollution emissions compared with coal boiler, gas boiler and direct electric heating mode. T Chen et al. [6] indicated that heat pump heating system would have lower cost and less environment impact than other systems if its COP is above 3. Greening and Azapagic [7] showed that up to 36% of CO_2 emission can be saved with water source heat pump and 6% with ASHP in comparison with boiler in the UK. Zhang et al. [8] reported the application of ASHP for heating in Harbin which showed the temperature difference between indoor and outdoor air should be controlled within 41 °C to achieve an acceptable COP. Poppi et al. [9] evaluated solar thermal and ASHP combisystems which indicated that variations in electricity price affect the additional investment far more than other economic parameters and the potential for achieving cost benefit vary a lot depending on load and climate boundary conditions. Braun and Rowley [10] indicated that heat pumps can realize emissions reductions when installed at high penetration levels combined with a grid decarbonization strategy.

To promote ASHP for space heating, efforts have been taken on new technologies of ASHP including two-stage compression, quasi-two-stage compression, new throttling devices, thermal storage, and defrosting technologies. Jiang et al. [11] studied two-stage compression ASHP and found the seasonal COP with cylinder volume ratio of 2 is 10.3% and 17.6% higher than that of 3 when outdoor design temperature is −4 °C and −8 °C. Li and Yu [12] analyzed a two-stage compression ASHP and found the COP_h of the heat pump can be maximized by optimally allocating thermal conductance inventory of the heat exchangers. Xu and Ma [13] did exergy analysis for quasi-two-stage compression heat pump coupled with ejector and indicated that the ejector could decrease exergy loss of compressor and improve the system exergy efficiency. Peng et al. [14] compared the performance of ASHP water heater using different expansion devices and showed that short tube orifice is more suitable for heat pump water heater than capillary tube. Zeng et al. [15] found that sensible heat storage system can achieve improved indoor thermal comfort and lower energy consumption compared with normal VRV (Variable Refrigerant Volume) systems. Jiang et al. [16] proposed a novel defrosting method based on superheat degree control which demonstrated reasonable strategy that initiated defrosting before the performances of ASHP deteriorated rapidly. Touchie and Pressnail [17] tested a low-temperature ASHP operating in a thermal buffer zone created by an enclosed balcony space which could improve the COP of ASHP in cold climates. Hu et al. [18] proposed a self-optimizing control scheme using the ESC (extremum seeking control) strategy which could search and even track both fixed and slowly varying optimum COP. Dai et al. [19] numerically studied a hybrid solar assisted loop heat pipe/heat pump water heater system, and the results showed that, on typical sunny days in spring or autumn, the proposed system could save 40.6% power consumption compared to heat pump mode. The influence of wet compression on a heat pump system was experimentally investigated by Seong et al. [20], who found that the heating capacity and power input of wet compression increased more than that of dry compression, with a superheat of 10 °C.

In addition to the heat pump cycle and components of ASHP, studies on refrigerant alternatives have been carried out for years because refrigerant is important for both energy performance and environment protection. The refrigerant R22 which is commonly used in China will be phased out in 2030. Ozone-friendly refrigerants such as R134a, R410A, R417A, R404A and R507 used in air conditioners were analyzed in several previous studies [21–25]. Chen [21] compared the performance of residential air conditioners using R410A and R22 and concluded that the adoption of R410A could be helpful for air conditioner to decrease their heat exchanger size or improve their energy efficiency. Bolaji [22] investigated the performance of R22 and its ozone-friendly alternative refrigerants (R404A and R507) in a window air-conditioner and showed the average COP of R507 increased by

10.6%, while that of R404A reduced by 16.0% compared to that of R22. Wang et al. [23] proposed a numerical model on the performance of a novel frost-free ASHP using three different refrigerants and indicated that R134a has higher COP than R22 and R407C at given ambient temperature of −10 °C and RH of 85%. Fannou et al. [24] presented a comparative analysis of a direct expansion geothermal evaporator using R410A, R407C and R22 as refrigerants. The results showed R410A would be a better choice to minimize pressure drop. Cabello et al. [25] evaluated the performance of a vapor compression plant using R134a, R407C and R22 as working fluids and indicated that the power consumption with R22 tends to decrease more slowly with increasing compression ratios than others.

The studies above proposed and compared ozone friendly refrigerants. However, refrigerants with high GWP are also unfriendly to the environment, which pushed researchers to investigate and propose mixing refrigerant alternatives with low GWP. Han et al. [26] showed that a new ternary non-azeotropic mixture of HFC-161/125/143a's thermodynamic properties are similar to those of R404A, but its global warming potential (GWP) is much lower than those of R507A and R404A. Ln et al. [27] found that the SEER and SCOP of R32 and mixing refrigerant R446A were higher than R410A by 8% and 1% in a residential heat pump system. A comparison of R404A with six low-GWP mixing refrigerants including mid-term alternatives R407A and R407F and long-term alternatives L40, DR-7, N40 and DR-33 was conducted by Mota-Babiloni et al. [28] in terms of cooling capacity, volumetric flow rate and COP, and conclusions were given that the most efficient alternatives were the low-flammable refrigerants L40 and DR-7 considering two-stage and diverse operating conditions. Devecioglu [29] evaluated seasonal efficiency of four new low-GWP mixing refrigerants including R446A, R447A, R452B and R454B whose GWP values were lower than 750 and concluded that R452B has the most suitable SCOP among the alternative refrigerants. Feasibility studies on R32 refrigerant mixtures, such as R32/R290, R32/L41A and R32/CO_2 replacing R410A in heat pumps or household air conditioners were conducted in several studies [30–32], and the results demonstrated that, compared with R410A, higher cooling/heating capacities and similar COP could be achieved by R32 refrigerant mixtures. Devotta et al. [33] presented simulated and experimental performance evaluations of a few selected low GWP refrigerants, i.e., HC-290, HC-1270, HFC-32, and HFC-1234yf as alternatives to HCFC-22 for room air conditioners. It indicated that HC-290 offers the best performance. The discharge temperatures of HFC-32 are relatively high, and HFC-161 is also a potential ultra-low global warming potential alternative to HCFC-22, once its safety classification is established. Ma et al. [34] proposed a precooling cycle that is used to reduce the power consumption and improve the heat transfer efficiency of a mixed refrigerant cycle. The results indicated that the optimum mixed refrigerant component ratios varied based on the pressure and temperature of the feed gas. Wu et al. [35] established mathematical model to calculate the thermodynamic properties of zeotropic mixtures and concluded that this simulation program could analyze the system cycle effectively and provide a direction for improvement of the auto-refrigerating cascade system.

The literature above showed that the ASHP and refrigerants performance vary largely in different operation conditions. In previous studies, most of the refrigerant alternatives analyses and comparisons were focused on ASHP water heater or air conditioning units. Research on proper refrigerant in ASHP for space heating, especially for dynamic space heating under varied outdoor environment, is sparse. In this study, a new mixture refrigerant named RHR-1 was proposed, which aimed to find a refrigerant with high energy efficiency, no ODP and low GWP for dynamic space heating ASHP in North China or similar climate zones. A theoretical model for refrigerants performance simulation and comparison was developed. The performance of the proposed refrigerant RHR-1 was then analyzed and compared with commonly used refrigerants including R134a, R410A, R407C and R22 on the heating coefficient of performance, compression ratio and discharging temperature. Based on the performance analyses, the control strategies for RHR-1 ASHP under dynamic space heating operations were given.

2. Methods

The new refrigerant was proposed based on mixing of pure HFC refrigerants with considerations of cycle COP_h, reducing compression ratio (CR) and realizing low GWP. The analyses and comparisons on the performance of different refrigerant candidates were based on theoretical modeling and simulation.

2.1. Mixture Refrigerant Proposal

The proposed mixture refrigerant RHR-1 was mixed by R152a and R32 with proportions of 75% and 25%, respectively. Inputting the components and their proportions of RHR-1 into the software REFPROP 9.1 (Reference Fluid Thermodynamic and Transport Properties) released from NIST (United States National Institute of Standards and Technology, Gaithersburg, MD, USA, 2013), the thermodynamic parameters including its saturation temperature/pressure, density, enthalpy, and entropy of the mixture could be inquired and obtained. REFPROP covers the thermos-physical properties of refrigerants consisting of pure substances or predefined mixtures. The thermo-physical properties of defined substance mixtures can also be inquired with REFPROP [36]. Figure 1 shows the relationship between temperature and saturated liquid phase/vapor phase enthalpy of RHR-1. A regression analysis was conducted for the enthalpy-temperature relationship based on the physical parameters of RHR-1, and the regression results are given as Equations (1) and (2).

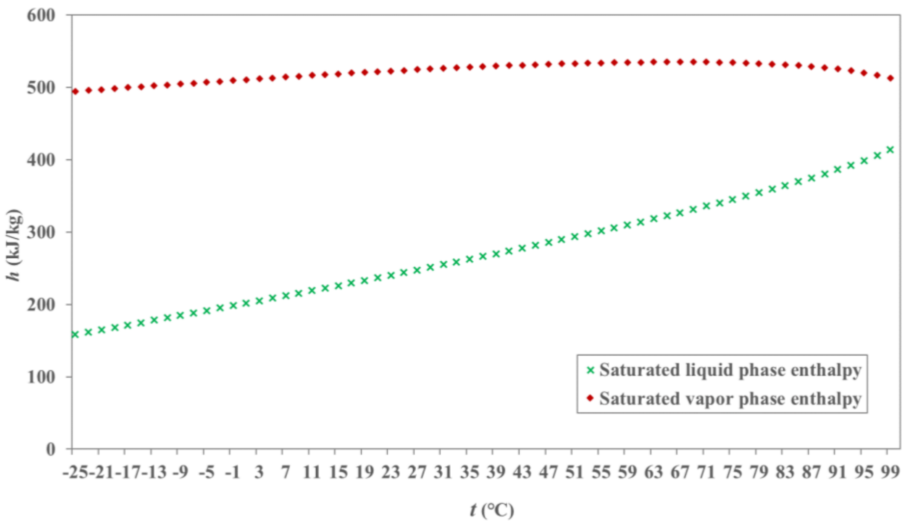


Figure 1. Relationship between temperature and enthalpy of RHR-1.

$$h_{li} = (6E-07)t^4 - 0.0001t^3 + 0.0081t^2 + 1.476t + 157.49 \quad (1)$$

$$h_{va} = (-8E-07)t^4 + 0.0001t^3 - 0.0103t^2 + 0.873t + 492.52 \quad (2)$$

where h_{li} and h_{va} denote the liquid enthalpy and vapor enthalpy of RHR-1 at saturation points, kJ/kg. t is the temperature of the refrigerant, °C.

The HFC refrigerants used to be analyzed and compared with RHR-1 in this study were R134a, R410A and R407C, which are commonly used in heat pump and air conditioners. To better demonstrate the performance of RHR-1, the HCFC refrigerant R2,2 which is still widely used in air conditioner and ASHP in China, was also included in the analyses and comparisons. The physical properties of the refrigerant candidates used as analyzing samples are shown in Table 1 [37]. The flammability and toxicity of refrigerants are expressed alphanumerically which have meanings, as shown in Figure 2.

Table 1. Physical properties of compared refrigerants.

Items	RHR-1	R134a	R410A	R407C	R22
Molecular weight (g mol^{-1})	62.54	102.03	72.58	86.20	86.47
Critical temperature (°C)	101.2	101.1	71.4	86	96.1
Critical pressure (MPa)	5.18	4.06	4.90	4.63	4.99
Critical density (kg m^{-3})	377	512	489	527	526
Boiling point at 101.3 kPa (°C)	−51.7 to −24	−26.1	−51.4	−43.6	−40.8
Toxicity/Flammability	A2	A1	A1	A1	A1
ODP [a]	0	0	0	0	0.04
GWP (100 year) [b]	279	1370	2100	1700	1790

[a] Ozone Depression Potential; [b] Global Warming Potential.

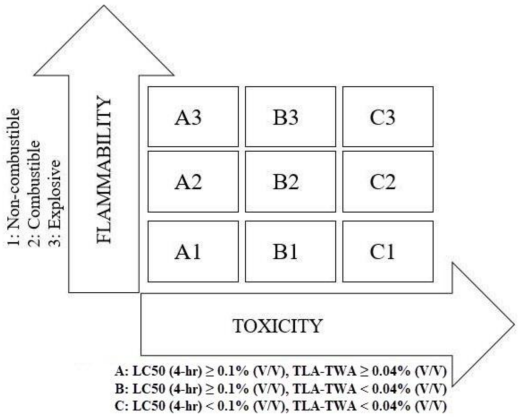


Figure 2. Classifications of refrigerants flammability and toxicity.

2.2. Theoretical Modelling and Simulation Tools

The ASHP for space heating used in North China normally adopt single stage compression or quasi-two-stage compression refrigeration cycles. The thermal processes of refrigerants in the typical single stage vapor-compression and quasi-two-stage compression ASHP cycle are shown in Figures 3 and 4.

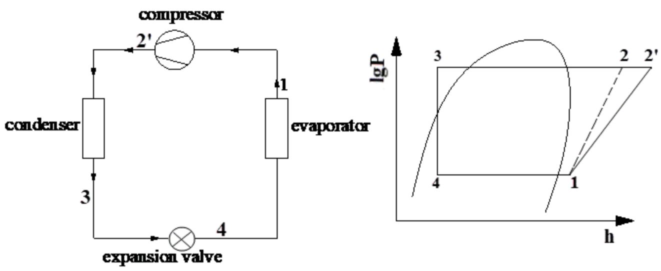


Figure 3. The schematic and pressure-enthalpy diagram of single stage vapor-compression air source heat pump (ASHP).

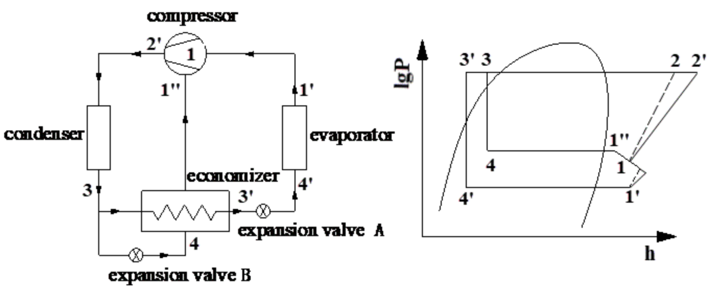


Figure 4. The schematic and pressure-enthalpy diagram of quasi-two-stage vapor-compression air source heat pump (ASHP).

In the single stage cycle, refrigerant condenses from Point 2′ to Point 3 and releases heat to circulation water which is used as heating medium. The high-pressure liquid refrigerants at pPoint 3 then depressurizes to low pressure and low temperature liquid–vapor mixture (Point 4) through throttling valve. The refrigerant evaporates from Point 4 to Point 1 and takes heat from outdoor environment. During the compression process from Point 1 to Point 2′, electrical power is consumed by compressor. The theoretical COP_h of the heat pump can thus be calculated with the enthalpy differences of refrigerant during the condensing and compression process together with the refrigeration efficiency.

The theoretical COP_h of a single stage compression ASHP system for heating can be calculated with Equation (3)

$$COP_{h-th} = \frac{h_2' - h_3}{h_2' - h_1} \tag{3}$$

where h_1 is the refrigerant enthalpy at the inlet point of the compressor, kJ/kg. h_2' is the refrigerant enthalpy at the outlet point of the compressor, kJ/kg. h_3 is the enthalpy of refrigerant leaving the condenser, kJ/kg.

h_2' can be calculated by Equation (4)

$$h_2' = h_1 + \frac{h_2 - h_1}{\eta_{is}} \tag{4}$$

where h_2 is the enthalpy of refrigerant at condensing pressure and has equal entropy with the refrigerant at h_1, kJ/kg. η_{is} is the isentropic efficiency of the compression.

The COP_h calculated with Equation (3) is the theoretical COP of refrigerant cycle regardless of energy loss through power or heat transfer (energy transferred to surrounding environment which is not used for heating circulation water). Considering the energy loss through power-engine transfer in the compressor and heat loss during heat transfer between refrigerant and water in condenser, Equation (5) was used to formulate the actual COP_h of ASHP.

$$COP_h = \eta_e \eta_d \eta_m \eta_t \frac{h_2' - h_3}{h_2' - h_1} \tag{5}$$

where η_m, η_d, and η_e are the motor efficiency, engine transmission efficiency (from motor to compressor) and friction efficiency of the compressor. η_t is the heat transfer efficiency from refrigerant to water in the condenser.

In the quasi-two-stage vapor-compression ASHP, the high temperature and high-pressure refrigerant vapor discharged from the compressor condenses to high temperature and high-pressure refrigerant liquid in the condenser. The heat released from the condensing process of refrigerant is used for space heating in buildings. The refrigerant liquid from the condenser is then divided two ways. In 3-3′-4′ circuit, the refrigerant changes into super-cooled liquid after heat transfer in economizer and then expands through expansion valve A, the super-cooled liquid changes into low temperature and low-pressure gas–liquid mixture and is delivered into evaporator. In the evaporator, the refrigerant evaporates to low temperature and low-pressure gas phase refrigerant after heat exchanging with outdoor air. The refrigerant vapor from the evaporator is sucked into the compressor to start a new cycle. In the auxiliary circuit (3-4-1″), refrigerant changes into low temperature and low-pressure gas–liquid mixture after expansion valve B, and transfers heat with high temperature refrigerant in circuit 3-3′-4′ through the economizer. After evaporating in the economizer, the gas phase refrigerant is directly sucked into the auxiliary refrigerant inlet of the compressor, and the new cycle begins.

The theoretical COP_h of a quasi-two-stage compression ASHP system for heating can be calculated with Equation (6)

$$COP_{h-th} = \frac{h_2' - h_3}{\alpha(h_2' - h_1') + (1-\alpha)(h_2' - h_1'')} \tag{6}$$

where h'_1 is the refrigerant enthalpy at the outlet point of the evaporator, kJ/kg. h'_1 is the refrigerant enthalpy at the outlet point of the economizer in 3-4-1″ circuit, kJ/kg. α is the proportion of refrigerant going through 3-4′-1′ circuit.

The h'_2 in quasi-two-stage compression refrigeration cycle can also be calculated by Equation (4), and the h_1 could be calculated with Equation (7)

$$h_1 = \alpha h'_1 + (1-\alpha)h'_1 \quad (7)$$

Considering the energy loss through power–engine transfer in the compressor and heat loss during heat transfer between refrigerant and water in condenser, the actual COP_h of quasi-two-stage compression ASHP could be calculated with Equation (8).

$$COP_h = \eta_e \eta_d \eta_m \eta_t \frac{h'_2 - h_3}{a(h'_2 - h'_1) + (1-\alpha)(h'_2 - h''_1)} \quad (8)$$

To calculate the COP_h of ASHP with Equations (3)–(8), the physical parameters of different HFC refrigerants could be inquired with the software REFPROP. With the mixing HFC refrigerants and their proportions, the properties of RHR-1 could be calculated and obtained with REFPROP as well.

The COP_h of different refrigerants could thus be obtained using the theoretical Equations (3)–(8) integrating REFPROP. Other parameters that affect the performance and practical utilization of ASHP using different refrigerants such as condensing pressure, evaporating pressure, compression ratio, and discharging temperature could also be calculated based on the theoretical model. During the simulation, an overall refrigeration efficiency η_{oval} was used to represent the effects of η_e, η_d, η_m, η_t and η_{is}. The η_{oval} was pre-set to be 0.65 in the original computation process. The isentropic efficiency was pre-set to be 0.76 [38].

A schematic is given as Figure 5 to show the input, output parameters and the computation process of the proposed model. The simulation equations were coded within MATLAB 9.0-R2016a (MathWorks, Natick, MA, USA, 2016) assisted by MS-Excel invoking REFPROP 9.1.

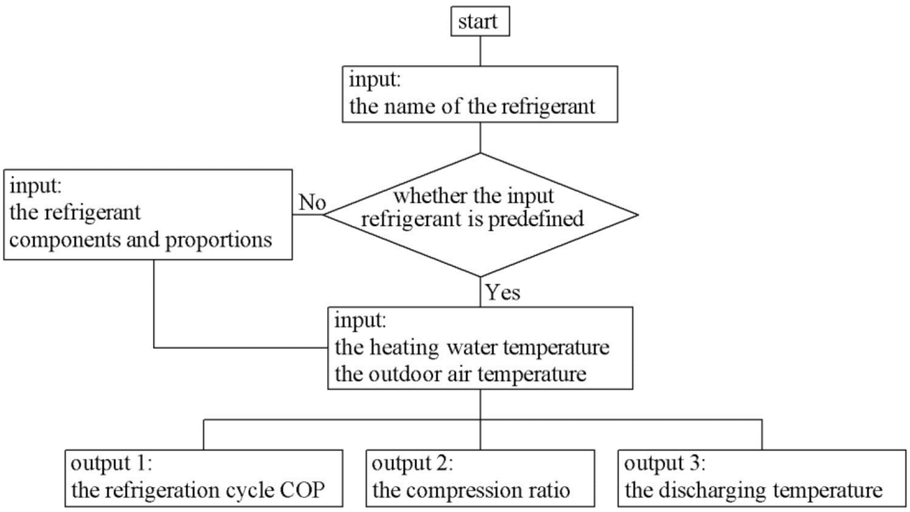


Figure 5. The computation process of the refrigerant performance simulation. COP: coefficient of performance.

2.3. Case Design

Due to the dynamic outdoor thermal environments and varied heating systems of buildings, different combinations of outdoor air temperatures and supply water temperatures were set as operation cases in this study. In the simulation cases, the required supply water temperatures were set to be from 35 °C to 50 °C considering the economical heating operation of ASHP together with heating terminals including heating radiator, warm air heating system and radiant floor heating. The outdoor air temperatures were set to be from −15 °C to 15 °C referring to Tianjin outdoor meteorological parameters, which is a typical city of North China. The outdoor air temperatures were divided into six categories with steps of 5 °C. The number of hours of the six winter categories in Tianjin were calculated and summarized in Table 2.

Table 2. Outdoor air temperature categories and its number of hours in Tianjin.

Class No.	Outdoor Air Temperature Range (°C)	Representative Outdoor Air Temperature (°C)	Number of Hours (h)
Class 1	−15 to −10	−12.5	38
Class 2	−10 to −5	−7.5	504
Class 3	−5–0	−2.5	1020
Class 4	0–5	2.5	871
Class 5	5–10	7.5	356
Class 6	10–15	12.5	98

The supply water temperatures were designed to be in the range of 35 °C to 50 °C with steps of 5 °C. Combinations of outdoor air temperature and supply water temperature were set as the simulation cases shown in Table 3. During the simulation, the condensing temperature was assumed to be 8 °C higher than the supply water temperature, and the sub-cooling temperature of the refrigerant was controlled to be 5 °C. The evaporating temperature was set 8 °C lower than outdoor air temperature, and the superheat temperature during evaporation was controlled to be 5 °C as well.

Table 3. Outdoor air and supply water temperatures in the simulation cases.

Outdoor Air Temperature (°C)	Supply Water Temperature (°C)			
−12.5	35	40	45	50
−7.5	35	40	45	50
−2.5	35	40	45	50
2.5	35	40	45	50
7.5	35	40	45	50
12.5	35	40	45	50

With the varied outdoor air temperatures and supply water temperatures, 24 operation cases were thus designed. The simulation and comparison were conducted within the deigned cases.

3. Results

Based on the theoretical model and designed operation cases for refrigerants performance simulation, the COP_h, CR and DT (discharging temperature) of ASHP using RHR-1, R134a, R410A, R407C and R22 under dynamic heating processes were analyzed and compared.

3.1. Heat Pump COP_h

The COP_h simulation results of ASHP using different refrigerants as working fluids for space heating under the designed cases are shown in Figures 6–9.

Because the critical temperature of all refrigerant candidates are high enough to condense at 58 °C, all the refrigerants are available choices to supply heating water at 50 °C, but the COP_h are quite different from each other. Figure 6 shows that, to supply heating water with 50 °C under varied outdoor air temperature, the COP_h of ASHP using RHR-1, R134a, R410A, R407C and R22 are in the ranges of 2.43–3.59, 2.37–3.56, 2.24–3.28, 2.25–3.33 and 2.43–3.57, respectively.

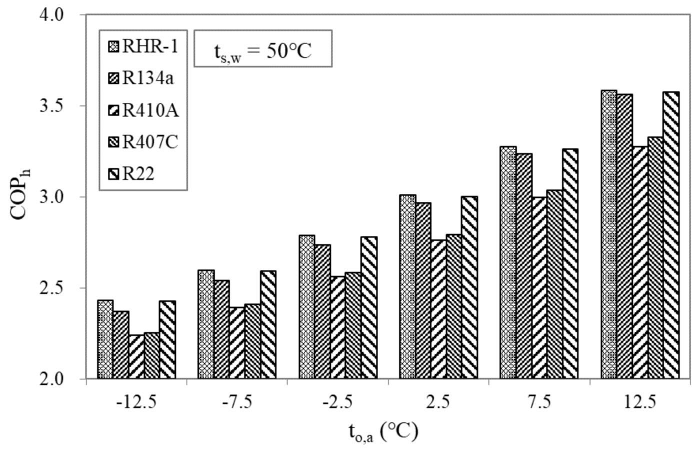


Figure 6. The COP_h of ASHP with different refrigerants to supply water at 50 °C under varied thermal environments.

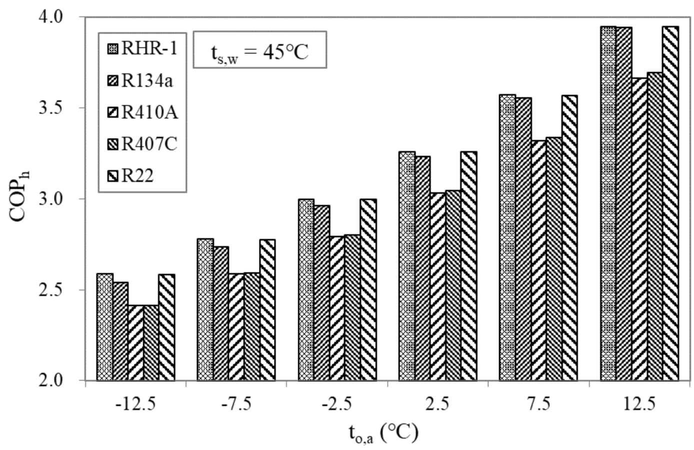


Figure 7. The COP_h of ASHP with different refrigerants to supply water at 45 °C under varied thermal environments.

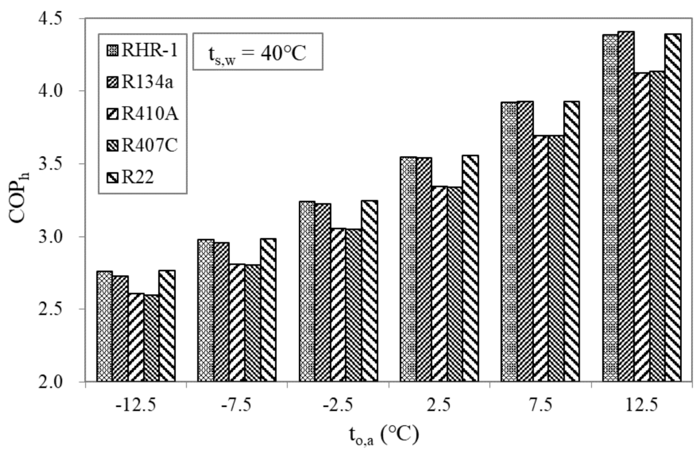


Figure 8. The COP_h of ASHP with different refrigerants to supply water at 40 °C under varied thermal environments.

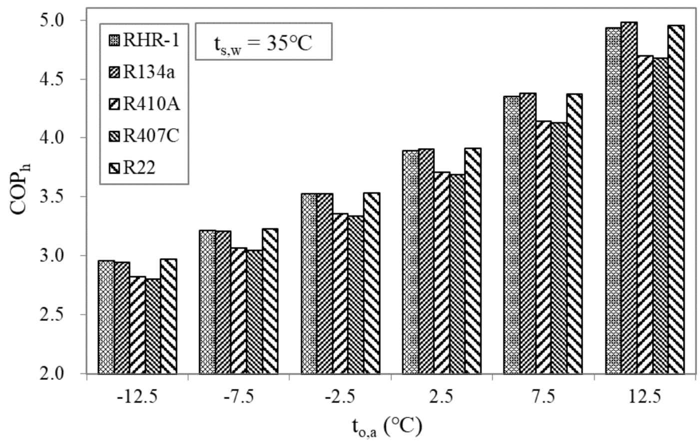


Figure 9. The COP_h of ASHP with different refrigerants to supply water at 35 °C under varied thermal environments.

To supply heating water at 45 °C under varied outdoor thermal environment, the COP_h of ASHP with RHR-1, R134a, R410A, R407C and R22 are in the ranges of 2.59–3.95, 2.54–3.94, 2.41–3.66, 2.42–3.69 and 2.59–3.94, respectively (Figure 7).

To supply heating water at 40 °C under varied outdoor thermal environment, the COP_h of ASHP with RHR-1, R134a, R410A, R407C and R22 are in the ranges of 2.76–4.38, 2.73–4.41, 2.61–4.13, 2.60–4.13 and 2.77–4.39, respectively (Figure 8).

To supply heating water at 35 °C under varied outdoor thermal environment, the COP_h of ASHP with RHR-1, R134a, R410A, R407C and R22 are in the ranges of 2.96–4.93, 2.94–4.98, 2.82–4.70, 2.80–4.68 and 2.97–4.95, respectively (Figure 9).

Theoretically, all the refrigerant candidates in this study could be used for ASHP to supply heating water from 35 °C to 50 °C, but the COP_h vary largely from one to another. Under the designed operation cases, the COP_h of ASHP using RHR-1, R134a and R22 are higher than that with R410A and R407C. Among RHR-1, R134a and R22, the COP_h comparisons show different characteristics in different cases. RHR-1 shows highest COP_h when the supply water temperatures are 50 °C and 45 °C, which are normally adopted in space heating systems. When the supply water temperature decreases to 40 °C, R22 shows highest COP_h when outdoor air temperatures are from −12.5 °C to 7.5 °C, and R134a show highest COP_h when outdoor air temperature is 12.5 °C. If the supply water temperature further decreases to 35 °C, R22 shows highest cycle COP_h when outdoor air is from −12.5 °C to 2.5 °C, R134a shows highest cycle COP_h when outdoor air is 7.5 °C and 12.5 °C. Overall, under the 24 designed cases, RHR-1 showed highest COP_h in 12 cases, while R134a and R22 had the highest COP_h three and nine times, respectively.

3.2. Compression Ratio

The COP_h simulation and comparison were based on the isentropic efficiency of compression being the same for different refrigerants. However, different refrigerants could lead to large differences of compression ratio, which can affect the clearance volume and isentropic efficiency of the compressor. Compression ratio is thus an important factor in refrigeration and heat pump cycle. The high compression ratio would result in low isentropic efficiency for single stage compression or the requirement of two-stage compression. The compression ratio of ASHP using different refrigerants for space heating under the designed cases were thus simulated, as shown in Figure 10.

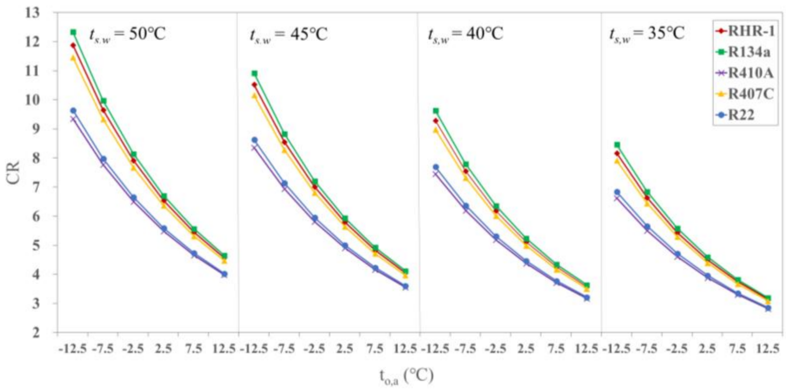
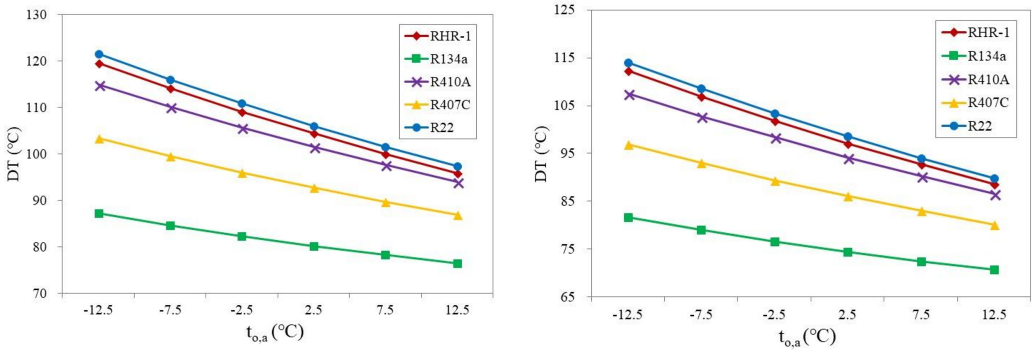


Figure 10. Compression ratio of ASHP with different refrigerants under varied thermal environments. CR: compression ratio.

In Figure 10, the compression ratio of ASHP using R134a is highest, and R410A has the lowest compression ratio among the refrigerant candidates. The compression ratio of proposed refrigerant RHR-1 is lower than R134a, but higher than R22. High compression ratio would lead to low isentropic efficiency and result in low refrigeration efficiency. Under the extreme design case in which supply water temperature is 50 °C and outdoor air temperature is −12.5 °C, the maximum compression ratio requirements are 12.35, 11.89, and 9.66 for R134a, RHR-1 and R22, respectively.

3.3. Refrigerant Discharging Temperature

Discharging temperature of refrigerant influences the performance of compressor and is the key factor for selecting lubricant of compressor. Discharging temperature of ASHP using different refrigerants for space heating under the designed cases are shown in Figure 11a–d.



(**a**) DT of refrigerants to supply water of 50 °C. (**b**) DT of refrigerants to supply water of 45 °C.

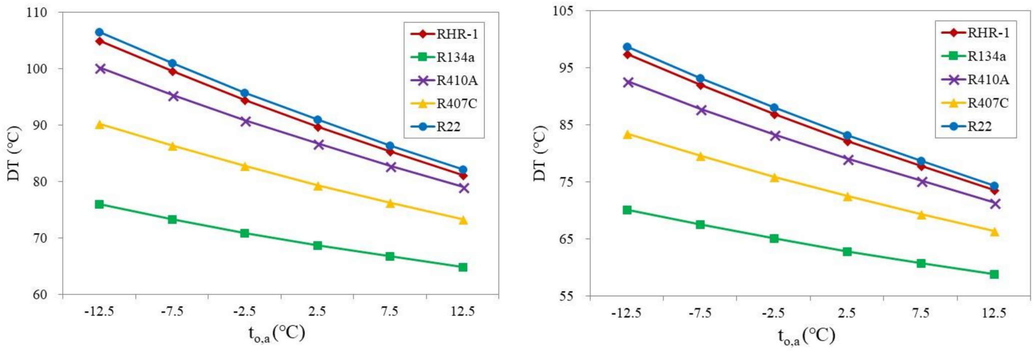


Figure 11. DT of ASHP using different refrigerants under varied thermal environments. DT: discharging temperature.

The results show that the discharging temperatures of ASHP using R22 as refrigerant is highest among the candidates, and R134a has the lowest discharging temperature under the designed cases. The discharging temperature of RHR-1, R410A and R407C are in the middle level. To supply heating water of 50 °C under −12.5 °C outdoor air temperature which is the extreme design case, the DT of R22, RHR-1 and R134a are 122 °C, 120 °C and 87 °C, respectively.

4. Discussions

For space heating air source heat pump, since the COP_h is significantly affected by the supply water temperature, low temperature heating terminals such as radiant floor heating, fan-coil warm air heating or expanded radiators are normally adopted. Thus, in the designed cases of this paper, the highest supply water temperature was set as 50 °C. For space heating, the refrigerants that are commonly used in existing heat pumps are R22, R134a, R410A and R407C. Through the comparisons above, the proposed RHR-1 has higher COP_h than others in most of the design cases, especially when supply heating water are 50 °C and 45 °C. R134a has higher COP_h compared with R407C in this study, which accords with a previous study [23] that also reported R134a has longer frost-free operation time than R407C and R22. The compression ratio of RHR-1 is lower than R134a but higher than R22. The discharging temperature of RHR-1 is higher than R134a, but lower than R22. Considering the performance of refrigerants including COP_h, CR and DT, RHR-1 has shown relatively strong comprehensive properties.

Besides energy performance, the global warming potential of RHR-1 is 279 which is much lower than R134a (GWP 1370), R407C (GWP 2100), R410A (GWP 1700) and R22 (GWP 1790). As an alternative for R22, RHR-1 has not only no ozone depletion potential, but also low global warming potential, which strengthen the environment friendliness of ASHP refrigerants. With the above characteristics, RHR-1 might be the proper choice for space heating in rural residential buildings of North China or similar climate zones.

In addition to the comparison of different refrigerants, the operation cases such as supply water temperature and outdoor air temperature could largely affect the performance of ASHP. The COP_h of ASHP using RHR-1 as refrigerant for space heating under the deigned cases were analyzed, as shown in Figure 12.

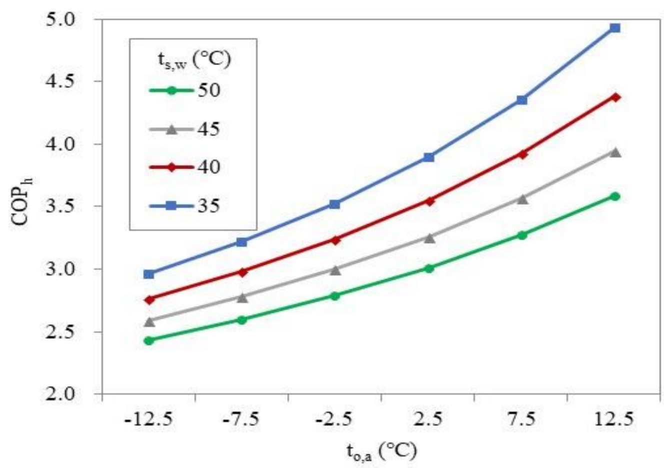


Figure 12. The COP_h of space heating ASHP using RHR-1 under varied thermal cases.

In Figure 12, when the outdoor air temperature increases 5 °C, the COP_h will increase 0.17–0.46. If the supply water temperature decreases 5 °C, the COP_h will increase 0.15–0.55. Regression analysis was then conducted to get the relationship between COP_h of ASHP using RHR-1 and the temperature differences between supply water and outdoor air. The regression results are shown in Figure 13.

Using a logarithmic equation to regress the relationship of COP_h with temperature difference between supply water and outdoor air, it could be written as Equation (9)

$$COP_h = -2.437 \ln(t_{s,w} - t_{o,a}) + 12.418 \quad (9)$$

From the study of Chen et al. [6], the COP of heat pump should be above 3 to achieve lower cost and less environment impact than other heating systems. To get the reasonable COP_h, the temperature difference between outdoor air and supply water of ASHP using RHR-1 is suggested to be controlled within 47.5 °C. Considering the temperature difference between supply water and indoor air, this result accords with the study by Zhang et al. [8] that indicated that the application of ASHP should control indoor and outdoor air temperature be within 41 °C to achieve an acceptable COP. In this study, the outdoor meteorological parameters of Tianjin were used as the typical thermal climate in North China, and the average outdoor air temperature in a heating season (15 November–15 March) of Tianjin is −0.16 °C. Thus, to get the reasonable COP_h that was reported by Chen et al. [6], the average supply water temperature of ASHP using RHR-1 in the heating season is suggested to be controlled below 47 °C.

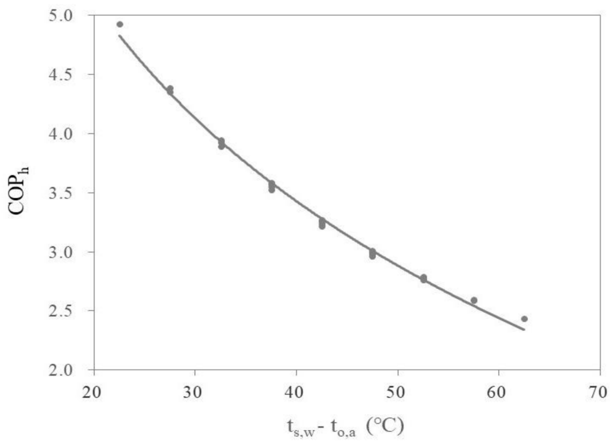


Figure 13. The regressed relation between COP_h of ASHP using RHR-1 with temperature differences of supply water and outdoor air.

The influence of supply water temperature to COP_h determines that the control strategy of ASHP should be different from gas or coal boiler whose efficiency changes less under varied heating load or outdoor environment. Under partial heating load cases, the variable flow control strategy used in coal or gas boiler heating system, which decreases water flow and keeps supply water temperature constant, might not be appropriate for ASHP. Keeping constant heating water flow and decreasing supply water temperature would improve the COP_h and thus optimize the energy efficiency of ASHP units. However, the water circulating pump energy would not be saved under the partial load cases in this way. A new control strategy that keeps the total energy consumption of ASHP system including the compressor and the circulating water pump minimum should be developed for specific ASHP applications.

Because the COP_h varies greatly with supply water temperature, appropriate building renovation measures in buildings such as enhancing building envelope insulation and window air tightness, and renovation from heating radiators to radiant floor heating would decrease the required water temperature and thus improve the energy efficiency of the ASHP.

5. Conclusions

To improve energy efficiency and environment friendliness of space heating ASHP, a new mixture refrigerant RHR-1 was proposed in this study. A theoretical model for refrigerants performance analysis was developed based on refrigeration cycle equations integrating REFPROP. The performance of RHR-1 including heating coefficient of performance, compression ratio and discharging temperature were simulated and compared with R134a, R410A, R407C and R22. The following conclusions were obtained from this study.

1. RHR-1 has no ozone depletion potential and relatively low global warming potential compared to commonly used refrigerants including R134a, R410A, R407C and R22.
2. The COP_h of RHR-1 is in the range of 2.43–4.93, which is higher than other candidates in most design cases. The CR and DT of RHR-1 are in the middle levels among the compared refrigerants.
3. RHR-1 might be a reasonable refrigerant in ASHP for space heating due to its high COP_h, appropriate compression ratio and discharging temperature, no ODP and low GWP.
4. According to the regression analysis, the temperature difference between outdoor air and supply water of ASHP using RHR-1 is suggested to be controlled within 47.5 °C to get a reasonable COP_h.
5. Constant water flow control strategy was suggested to improve the energy efficiency of air source heat pump units under partial heating load cases.

At present, clean heating reforms are underway in North China. This study proposed a new mixture refrigerant RHR-1 for ASHP used in North China which is a typical region facing energy and environment problems. The analysis was conducted using North China as a representative area. The conclusions of this study can also be used in countries and regions having similar meteorological parameters.

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Author Contributions: Jinzhe Nie conceived the refrigerant study and developed the theoretical model, the mixture refrigerant; Yufeng Zhang suggested the mixture refrigerant components; Xiangrui Kong analyzed the simulation results and wrote the paper; Jinzhe Nie improved the paper organization and English of the manuscript.

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

Nomenclature

COP	coefficient of performance
CR	compression ratio
DT	discharging temperature, °C
ODP	ozone depletion potential
GWP	global warming potential
h	enthalpy, kJ/kg
η	efficiency
t	temperature, °C
α	proportion of refrigerant
Subscripts	
o	outdoor
a	air
s	supply
w	water
is	isentropic
li	liquid
va	vapor
m	friction
d	engine transmission
e	motor

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