IMPACT OF THE R407c AND R134yf REFRIGERANT CHARGE ON THE PERFORMANCE OF A NEW PURE ELECTRIC VEHICLE HEAT PUMP AIR CONDITIONING SYSTEM

by

Zhiyong SU[a*], Gang CHEN[a], Haiyan ZENG[b], Haijun LI[a], Yiwei YU[a], Yayu DAI[a], Mengqi XU[a], and Xinyu TANG[a]

[a]School of Energy and Environment, Zhongyuan University of Technology, Zhengzhou, China
[b]Jinjiang Middle School in Zhongyuan District, Zhengzhou, China

Original scientific paper
https://doi.org/10.2298/TSCI2503943S

The objective of this study was to investigate the impact of varying refrigerant charge levels of R407c and R134yf on the refrigeration performance of a vehicle heat pump air conditioning system. The study was conducted under high temperature conditions (45 °C) to assess the influence of different refrigerant systems on refrigeration performance parameters. The results demonstrated that the optimal refrigerant charge level can be determined by analyzing the change in refrigeration performance parameters with varying charge levels. The results indicated that an increase in the refrigerant charge of R407c from 2.3 kg to 2.6 kg resulted in a maximum cooling capacity and COP of the system reaching 4.935 kW and 2.1, respectively. The optimal charging capacity of the system was determined to be 2.5 kg. Similarly, an increase in the refrigerant charging capacity of R134yf from 2.3 kg to 3.5 kg resulted in a maximum cooling capacity and COP of the system reaching 5.08 kW and 2.71, respectively. The optimal charging capacity of the system was determined to be 2.9 kg. The COP of the R134yf system is 0.61 higher than that of the R407c system. Furthermore, the energy saving of the R134yf system is more pronounced under extreme conditions.

Key words: *heat pump air conditioner, low pressure tonic, R134yf, R407c, optimal filling amount*

Introduction

In recent years, the rapid development of new energy vehicles and pure electric vehicles has been accompanied by a number of advantages, including energy saving, low carbon emissions, and environmental protection, all of which align with the principles of sustainable development [1, 2]. The Kigali Amendment to the Montreal Protocol essentially prohibits the utilization of GWP < 150 refrigerants [3]. The recently developed refrigerant R134yf is environmentally benign and represents the most *via*ble alternative refrigerant at present [4]. Automotive heat pump air conditioning systems continue to be the second most energy-consuming equipment in pure electric vehicles [5], particularly in extreme environments, which significantly impacts range [6]. The selection of a refrigerant with optimal physical properties and an appropriate system configuration directly influences the performance of the system [7]. An inadequate or excessive refrigerant charge affects the evaporating and con-

* Corresponding author, e-mail: 13938489682@163.com

densing temperatures of the system in operation, which in turn affects the cooling performance of the new automotive heat pump air conditioning system [8]. Consequently, it is of paramount importance to identify a refrigerant with superior physical properties and an optimal charge to enhance the cooling performance of heat pump air conditioners for pure electric vehicles under high temperature operating conditions (45 °C) [9, 10].

In light of the existing issues, numerous scholars have identified certain inherent limitations in refrigerants, primarily through an examination of the impact of the physical properties of various refrigerants on the overall performance of the system. This research has been extensively documented in literature, including references [11-13]. In a study conducted by Navarro-Esbi *et al.* [14], an experimental analysis was performed on the effect of different evaporation temperatures, condensation temperatures, superheat, and compressor speeds on the cooling capacity and COP of a vapor compression cycle. Wei *et al.* [15] presented an overview of the research and development of alternatives to refrigerant HCFC, as well as an analysis of the physical properties of various types of refrigerant alternatives. In their study, Lee *et al.* [16] analyze the exhaust pressure and unit cooling capacity of refrigerants in terms of thermophysical properties, material compatibility, oil solubility, and thermal properties. Jing *et al.* [17] examined the performance of R153A as a replacement for R407c in locomotive air conditioning systems. Their findings indicated that refrigerants with higher molar mass exhibited higher charge volumes. Lee and Jung [18] conducted an investigation into the physical properties, heat transfer, and lubrication properties of R134yf under different operating conditions with the objective of determining the optimal charge of R134yf refrigerant. In a study conducted by Li *et al.* [19], experiments were performed on a two-loop heat pump air conditioning system. The results indicated that the refrigerant charge has a significant impact on the cooling performance. Vaghela [20] reached the conclusion that R134yf can directly replace R134a by conducting experiments on the same automobile air conditioning system.

Currently, numerous scholars are investigating the impact of high GWP refrigerants, single refrigerant charge, and refrigerant properties on the performance of a single component. However, the study of high temperature conditions, which examines the effects of varying refrigerant properties and refrigerant charge on the system refrigeration performance, is less prevalent. In this paper, a heat pump air conditioning system test bed for automotive low pressure make-up air for R407c and R134fy was constructed using the quasi-secondary compressor cycle principle. Further study of the physical parameters of different refrigerants, different refrigerant charge adjustment, and the impact on the exhaust temperature, cooling capacity, COP, undercooling degree, and exhaust pressure is necessary for the subsequent study of low pressure make-up technology. This experimental basis will be provided by the aforementioned studies.

Experimental system

A vehicle heat pump system with an economizer has been designed. The system circulation principle is depicted in fig. 1. The cooling and heating cycle of the system is completed through the switching of a four-way valve and the change of a check valve, and the working mode of low pressure gas supplement is realized through the stop valve. The present experimental study of refrigerant R407c and R134yf charge on the cooling performance of vehicle heat pump air conditioning system was conducted in a standard enthalpy laboratory. Figure 2 depicts the pressure-enthalpy diagram for the theoretical cycle of low pressure gas supplement.

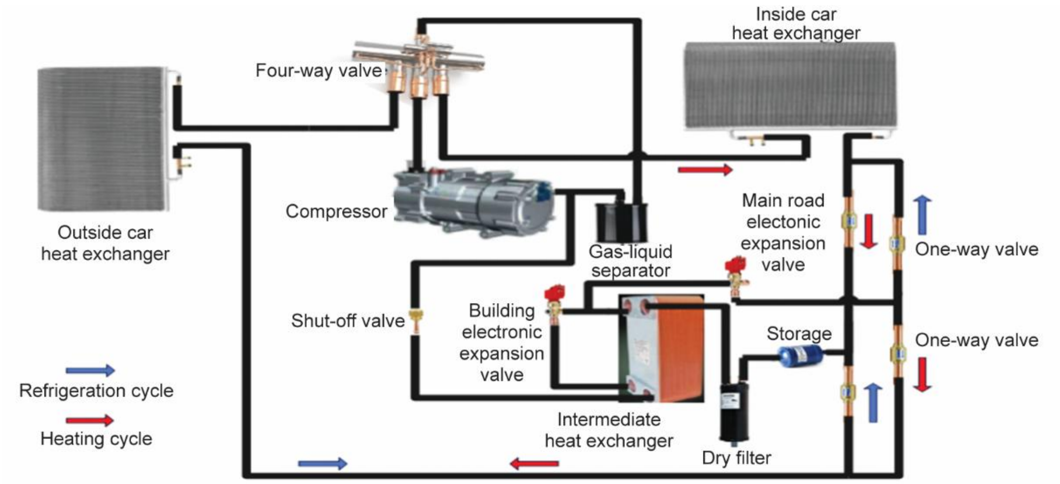

Figure 1. Low pressure gas supplement system

– Condenser heat capacity:

$$Q_k = m_r(h_2 - h_5)$$

where Q_k [kW] is the condenser heat production, m_r [kgs^{-1}] – the compressor discharge refrigerant mass-flow rate, h_2 [kJkg^{-1}] – the enthalpy of refrigerant discharge from compressor, and h_5 [kJkg^{-1}] – the enthalpy of refrigerant at condenser outlet.

– Compressor power:

$$W = m_r(h_9 - h_2)$$

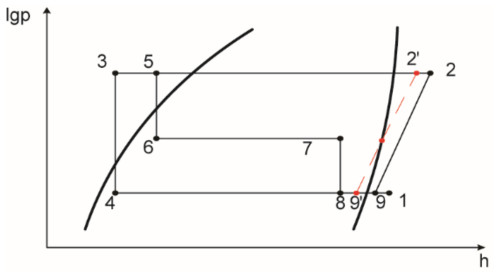

Figure 2. Pressure-enthalpy diagram for low pressure gas supplement theoretical cycle

where W [kW] is the compressor power, m_r [kgs^{-1}] – the compressor discharge refrigerant mass-flow rate, h_9 [kJkg^{-1}] – the enthalpy of refrigerant entering the suction chamber of the compressor, and h_2 [kJkg^{-1}] – the enthalpy of refrigerant discharge from compressor.

– Evaporator cooling capacity:

$$Q_o = m_o(h_1 - h_4)$$

where Q_o [kW] is the evaporator cooling capacity, m_o [kgs^{-1}] – the refrigerant mass-flow rate at evaporator, h_1 [kJkg^{-1}] – the enthalpy of refrigerant at outlet of evaporator, and h_4 [kJkg^{-1}] – the enthalpy of refrigerant inlet to evaporator.

– Heat pump system heating factor:

$$EER = \frac{Q_k}{W} = \frac{W + Q_o}{W} = 1 + \frac{Q_o}{W}$$

– Mass-flow rate of gas-supplemented refrigerants:

$$m' = m_r - m_o$$

– Economizer heat exchange:

$$Q' = m'(h_7 - h_6) = m_o(h_5 - h_3)$$

where Q_o [kW] is the economizer heat exchange, h_7 [kJkg^{-1}] – the enthalpy of refrigerant exported from the economizer on the refrigerant side of the make-up circuit, h_6 [kJkg^{-1}] – the enthalpy of refrigerant inlet to economizer on refrigerant side of make-up circuit, h_5 [kJkg^{-1}] – the enthalpy of refrigerant inlet to economizer on refrigerant side of main circuit, and h_3 [kJkg^{-1}] – the enthalpy of refrigerant outlet from economizer on refrigerant side of main circuit.

Experimental set-up

The experimental bench was set up, and the Hailey inverter scroll EVS34 compressor was selected. Its operating range is 1000~7000 rpm, and its maximum refrigeration capacity is 7 kW. The Zhengzhou Kelin heat exchangers, which are used inside and outside the car, have a heat transfer area of 2.07 m^2 and 1.40 m^2, respectively, tab. 1.

Table 1. The main parameters of the experimental prototype and the test instrument

Equipment name	Specifications
Main circuit expansion valve	CAREL E^{2}V-24, capacity: 16.3 kW, adjustment range: 10%~100%; suitable working medium: R407c or R134yf
Make-up expansion valve	CAREL E^{2}V-14, capacity: 5.7 kW, adjustment range: 10%~100%; suitable working medium: R407c or R134yf
Economizer	Weal Yield (Jiangsu) plate heat exchanger model B3-014-20D-3.0; design capacity: 6.06 kW; design pressure: 3.0 MPa; design temperature: –160~200°C
Four-way valve	Dunan DSF-20, applicable capacity: 7.1~25 kW

In the context of a high temperature 45 °C working condition, the main expansion valve superheat setting is 5 °C, while the make-up expansion valve superheat setting is 10 °C. The air volume on the side of the heat exchanger outside the car is 7830 m³/h, while the air volume on the side of the heat exchanger inside the car is 1080 m³/h. The compressor speed is 4000 rpm. The objective of this study is to examine the impact of varying refrigerant charges on the refrigeration performance of automotive heat pump air conditioning systems. The study will focus on the changes in refrigeration performance when the refrigerant charge is increased from 2.3 kg to 2.6 kg for R407c and from 2.3 kg to 3.5 kg for R134yf, tab. 2.

Table 2. Experimental test conditions

Outside temperature [°C]		Inside temperature [°C]		Replenishment technology	Refrigerant charge [kg]	
Dry-bulb	Wet-bulb	Dry-bulb	Wet-bulb		R407c	R134yf
45	37	27	19.5	Low pressure gas replenishment technology	2.3, 2.35, 2.4, 2.45, 2.5, 2.55, 2.6	2.3, 2.5, 2.7, 2.9, 3.1, 3.3, 3.5

Experimental analysis

Figure 3 illustrates the impact of high temperature operating conditions on the relationship between R407c and R134yf refrigerants on exhaust temperature change with varying charge amounts. As shown in the figure, an inverse relationship between the two refrigerants is observed, with a decreasing trend in exhaust temperature as the charge of R407c refrigerant increases from 2.3 kg to 2.6 kg, reaching a minimum of 80. A decrease in exhaust temperature was observed when the charge of R134yf refrigerant was increased from 2.3 kg to 3.5 kg, with a minimum temperature of 80.2 ℃. The low pressure make-up air system with economizer reduces the temperature of the compressor inlet mass, and the system can operate stably under high temperature conditions of 45 ℃. An increase in the refrigerant charge results in an increase in the system circulating mass-flow of the work mass, an expansion of the effective heat transfer area of the work mass, a reduction in the evaporating temperature of the refrigeration side, a decrease in the evaporator outlet superheat, and a subsequent reduction in the exhaust temperature. The refrigerant system charge for the R134yf refrigerant is higher than that for the R407c refrigerant due to the greater molar mass of the former. The molar mass of a refrigerant is a physical parameter that affects the refrigerant charge. The exhaust temperature of the R134yf refrigerant system is slightly lower than that of the R407c refrigerant system. The physical properties of different refrigerants, including their latent heat of vaporization, differ. The latent heat of vaporization of the R407c refrigerant is relatively high, which directly affects the effective heat transfer area and causes slightly higher exhaust temperatures in R407c refrigerant systems. In contrast, the R134yf refrigerant system has a small latent heat of vaporization and a large effective heat transfer area, which is the reason for the slightly lower exhaust gas temperature.

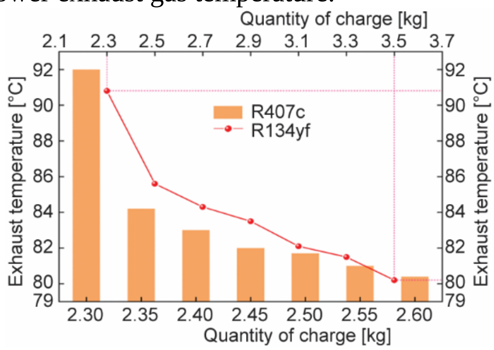

Figure 3. Effect of different charges of R407c and R134yf refrigerants on exhaust temperature

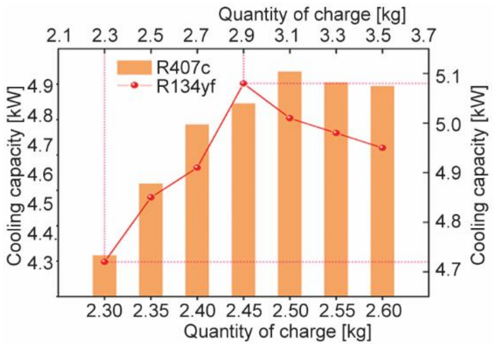

Figure 4. Effect of different charges of R407c and R134yf refrigerants on cooling capacity

Figure 4 illustrates the relationship between the R407c and R134yf refrigerants with varying charge levels and the resulting change in cooling capacity. As shown in the figure, when the charge of the R407c refrigerant is increased from 2.3 kg to 2.6 kg, the refrigerating capacity exhibits a trend of initially increasing and then decreasing. The cooling capacity reaches a maximum of 4. A cooling capacity of 935 kW was observed under the high temperature 45 °C working condition. When the charge of R134yf refrigerant was increased from 2.3 kg to 3.5 kg, the tendency was to increase and then decrease, and the cooling capacity reached a maximum of 5.08 kW at the high temperature 45 °C working condition. As the refrigerant charge continues to increase, the system reaches a point where the refrigerant charge is at saturation. This results in a larger heating side of the condensing temperature, while the cooling side remains unchanged. The evaporator inlet temperature increases, and the heat transfer temperature difference is reduced. This is the dominant factor, and the pressure drop is reduced. This is not conducive to the flow of the circulating workpiece, affecting the amount of the workpiece cycle. This results in another downward trend in cooling capacity. The molar mass of R134yf refrigerant is greater than that of R407c refrigerant, resulting in a higher charge for the R134yf refrigerant system. Despite the high gas density and large mass-flow rate of R407c refrigerant, the charge for the R134yf refrigerant system is still higher, leading to a slightly higher cooling capacity for the R134yf refrigerant system. As the charging amount increases, the maximum value of cooling capacity is observed in both the R134yf refrigerant system and the R407c refrigerant system. The maximum values are 5.08 kW and 4.935 kW, respectively. The optimum charging amount for the R134yf refrigerant system is 2.9 kg, while the optimum charging amount for the R407c refrigerant system is 2.5 kg. In the same system form, the R134yf systems demonstrate superior performance to the R407c systems.

Figure 5 illustrates the impact of high temperature operating conditions on the relationship between R407c and R134yf refrigerants on COP changes with different charging rates. As shown in the figure, when the charge of R407c refrigerant is increased from 2.3 kg to 2.6 kg, the COP exhibits a trend of increasing and then gradually decreasing, reaching a maximum value of 2.1. Conversely, when the charge of R134yf refrigerant is increased from 2.3 kg to 3.5 kg, the trend is one of increasing and then gradually decreasing, reaching a maximum value of 2.71. The substantial molar mass of the R134yf refrigerant results in an optimal charge of 2.5 kg for the R407c system, which is less than the optimal charge of 2. The R134yf system requires 9 kg of refrigerant, with the latent heat of vaporization of the R134yf refrigerant being relatively small. This results in a larger evaporator effective heat transfer area, and the refrigerant is utilized more effectively in the same system form. Consequently, the R134yf system has a higher COP than the R407c system. In a high temperature 45 °C working condition, the cooling capacity of the R134yf system is slightly higher than that of the R407c system. However, the COP of the R134yf system is higher than that of the R407c system, resulting in a higher efficiency and greater energy savings for the R134yf system.

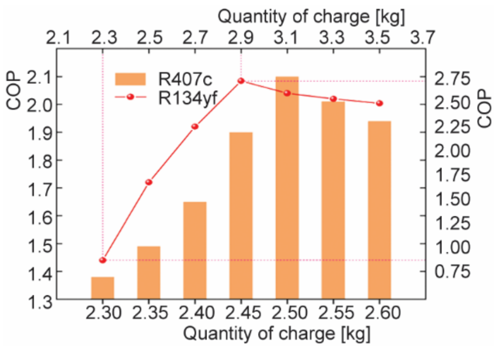

Figure 5. Effect of different charges of R407c and R134yf refrigerants on COP

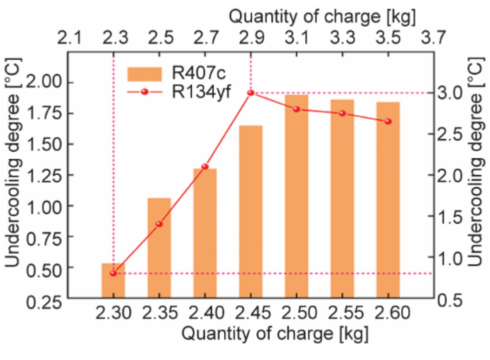

Figure 6. Effect of different charges of R407c and R134yf refrigerants on undercooling degree

Figure 6 illustrates the impact of high temperature operating conditions on the relationship between R407c and R134yf refrigerants on undercooling degree changes with varying charging amounts. As depicted in the figure, an increase in the charge of R407c refrigerant from 2.3 kg to 2.0 kg results in a notable shift in the undercooling degree. At a charge of

6 kg, the undercooling degree exhibits a trend of rapid increase followed by a slow decrease, reaching a maximum of 1.9 °C at a working temperature of 45 °C. When the charge of R134yf refrigerant was increased from 2.3 kg to 3.5 kg, there was also a trend of rapid increase followed by a slow decrease, with the undercooling degree at the 45 °C working temperature reaching a maximum of 3 °C. When the cooling capacity of the R407c system reaches its maximum, the degree of undercooling also reaches its maximum. This indicates that the optimal charging volume of the R407c system is 2.5 kg. Similarly, when the cooling capacity of the R134yf system reaches its maximum, the degree of undercooling also reaches its maximum. This indicates that the optimal charging volume of the R134yf system is 2.9 kg. In the high temperature 45 °C conditions, the temperature difference on the heating side of the heat exchanger is relatively small due to the relatively low latent heat vaporization of the R134yf refrigerant. This results in a more effective utilization of the heat transfer area by the same heat exchanger, which is reflected in the higher undercooling degree of the R134yf system compared to the undercooling degree of the R407c system. In the extreme conditions, the R134yf system exhibits a performance trend that is superior to that of the R407c system.

Conclusions

- A comparison of the saturated vapor pressure of R407c and R134yf refrigerant indicates that the same heat pump air conditioning system can be utilized. The physical molar mass of R134yf is greater than that of R407c refrigerant, and thus, the charging amount of R134yf refrigerant is found to be greater than that of R407c refrigerant. Consequently, the charging amount of R407c is increased from 2.3 kg to 2.6 kg, while the charging amount of R134yf refrigerant is increased from 2.3 kg to 3.5 kg. In the event of elevated temperatures (45 °C), the increase in refrigerant sufficiency results in a reduction in exhaust temperature for the R134yf system, while the exhaust pressure of the R134yf system is elevated in comparison to the R407c system.
- Under the high temperature operating condition (45 °C), when the R407c charge was increased from 2.3 kg to 2.6 kg, the cooling capacity, COP, and the undercooling degree appeared to be the maximum values of 4.935 kW, 2.1, and 1.9 °C, respectively. When the R134yf charge was increased from 2. When the R407c charge was increased from 2.3 kg to 2.6 kg, the maximum values of cooling capacity, COP, and undercooling degree were observed at 4.935 kW, 2.1, and 1.9 °C, respectively. When the R134yf charge was increased from 2.3 kg to 2.6 kg, the maximum values of cooling capacity, COP, and undercooling degree were observed at 5.08 kW, 2.71, and 3 °C, respectively. The R134yf system demonstrated superior cooling performance and more stable operation under extreme working conditions.
- The optimal sufficiency was found to be 2.5 kg at the high temperature operating condition (45 °C), with peaks in cooling capacity, COP, and undercooling degree at 2.5 kg of R407c system charge. The cooling capacity, COP, and undercooling degree also peaked at 2.9 kg for the R134yf system adequacy, and the optimal adequacy was found to be 2. At the peak value, it is found that the cooling capacity of the two systems is similar, but the COP of the R134yf system is higher than that of the R407c system by 0.61. Furthermore, the energy saving of the R134yf system is better under the condition that the form of the system does not change. This is an important guidance for the study of low GWP and low energy consumption systems.

References

[1] Donateo, T., *et al.*, A Method to Estimate the Environmental Impact of an Electric City Car during Six Months of Testing in an Italian City, *Journal of Power Sources, 270* (2014), Dec., pp. 487-498

[2] Zhuang, X., Jiang K., Research on the Development Roadmap of Pure Electric Vehicles in China, *Automotive Engineering, 34* (2012), 2, pp. 91-97

[3] Bing, L., *et al.*, Seasonal Performance Analysis of a Heat Pump with a New R134a Alternative Work Material, *Journal of Engineering Thermophysics, 43* (2022), 12, pp. 3177-3183

[4] Celil, A. M., *et al.*, Performance Evaluation of an Automotive Air Conditioning and Heat Pump System Using R1234yf and R134a, *Science and Technology for the Built Environment, 27* (2021), 1. pp. 44-60

[5] Kwon, C., *et al.,* Performance Evaluation of a Vapor Injection Heat Pump System for Electric Vehicles, *International Journal of Refrigeration, 74* (2016), Feb., pp. 138-150

[6] Zhang, Z., *et al.*, The Solutions to Electric Vehicle Air Conditioning Systems: A Review, *Renewable and Sustainable Energy Reviews, 91* (2018), Aug., pp. 443-463

[7] Ye, W. L., *et al.,* Effect of Mixed Refrigerant Composition on Performance of an Auto-Cascade Refrigeration System Using R600a/R1150/R14, *Thermal Science, 28* (2024), 5B, pp. 4307-4320

[8] Han, S., *et al.*, Influence of Refrigerant Charge on Make-Up Gas Enthalpy Heat Pump System, *Journal of Refrigeration, 43* (2022), 6, pp. 114-120

[9] Hu, X., *et al.*, Experimental Analysis on Refrigerant Charge Optimization for Cold Storage Unit, *Procedia Engineering, 205* (2017), Dec., pp. 1108-1114

[10] Liu, M., *et al.,* Experimental Study on Refrigerant Charge of Heat Pump Air Conditioning System for an Electric Vehicle, *Fluid Machinery, 48* (2020), 5, pp. 82-88

[11] Meng, Y., *et al.,* Comparison and Experiment of Thermodynamic Performance of Refrigerant Blend R134a/R1234yf(R513A) and R134a, *Chemical Progress, 38* (2019), 3, pp. 1182-1189

[12] Wang, Z., *et al.*, An Investigation into the Influences of Refrigerants' Thermal - Physical Properties on Temperature Separation Effect of a Vortex Tube, *Thermal Science, 27* (2023), 3B, pp. 2513-2524

[13] Hongli, W., *et al.*, Energy and Analysis of R1234yf Heat Pump System, *Low Temperature and Superconductivity, 47* (2019), 1, pp. 62-65

[14] Navarro-Esbrí, J., *et al.*, Experimental Analysis of R1234yf as a Drop-In Replacement for R134a in a Vapor Compression System, *International Journal of Refrigeration, 36* (2013), 3, pp. 870-880

[15] Wei, C. *et al.*, Research Progress and Performance Analysis of HCFC Refrigerant Substitutes, *Low Temperature and Superconductivity, 39* (2011), 12, pp. 41-44

[16] Lee, Y. *et al.*, Performance of Virtually Non-Flammable Azeotropic R1234yf/R134a Mixture for R134a Applications, *International Journal of Refrigeration, 36* (2013), 4, pp. 1203-1207

[17] Jing, Z. *et al.*, Performance Study of R513A Replacing R407C in Locomotive Air conditioning System, *Journal of Refrigeration, 44* (2023), 3, pp. 49-57

[18] Lee, Y., Jung, D., A Brief Performance Comparison of R1234yf and R134a in a Bench Tester for Automobile Applications, *Applied Thermal Engineering, 35* (2012), Mar., pp. 240-242

[19] Li, K., *et al.* An Experimental and Theoretical Investigation of Refrigerant Charge on a Secondary Loop Air Conditioning Heat Pump System in Electric Vehicles, *International Journal of Energy Research, 43* (2019), 8, 3381

[20] Vaghela, J. K., Comparative Evaluation of an Automobile Airconditioning System Using R134a and Its Alternative Refrigerants, *Energy Procedia, 109* (2017), Mar., pp. 153-160

Paper submitted: December 12, 2023
Paper revised: May 16, 2024
Paper accepted: May 17, 2024