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Comparative Analysis of Heat Pump System with IHX Using R1234yf and R134a

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Abstract

This research presents an energy performance analysis of the heat pump system with internal heat exchanger (IHX). The mathematical model of the heat pump outlined in this paper has been created by the author, it is steady-state with lumped parameters. The experimental validation of the model has been carried out using R1234yf and R134a as refrigerant. The aim of this work is to compare the energy performance in a wide range of operating conditions of a monitored heat pump system using both refrigerants. Finally, the heating capacity for R1234yf was lower from 0.63 % to 7.54 % compared with R134a, while the compressor power was similar from 0.12 % to 3.51 %. The COP values of R1234yf were lower than those obtained of R134a, ranging from 1.39 % to 4.22 %.

Keywords

heat pump, IHX, R1234yf, R134a, COP

1 Introduction

Safely accessible and affordable energy are vital factors in the future path of Hungary, assuring social security and high performance of modern industries. The most efficient and successful method to solve this issue in the short run is by reducing energy consumption and improving energy efficiency. A key element in increasing energy efficiency is implementing improvements in building energy. Currently an estimated 40 % of overall energy in Hungary is used within buildings, and two-thirds of this amount is used up in the form of heating and cooling [1].

The implementation of heat engines is one of the most efficient heating-cooling technologies as well as a key factor in lowering the global CO_2 emission and air pollution.

In the past few years much, scientific attention has been paid to the use of vapor compression cycles with and without internal heat exchangers [2–5]. Research works have dealt with the comparison of cooling capacity amounts, power consumption of the system and COP parameters when operating at various evaporation and condenser temperatures implementing R1234yf and R1234ze(E) as pos– sible alternatives for R134a [6–9].

Navarro-Esbrí et al. [10] experimentally showed that for a vapor compression plant, the replacement of R-134a for R-1234yf reduces the cooling capacity and the COP by 6 % and 13 %, respectively, but with the inclusion of IHX, these reductions become 2 % and 6 %, respectively. Devecioğlu and Oruç [11] compared the energy performance in a refrigeration system using R-1234yf and R-1234ze(E) with the energy performance of R-134a using an IHX. Their results showed that, despite a reduced refrigeration capacity with R1234ze(E), the energy consumption was also reduced, and comparing it with R-1234yf, the COP of R1234ze(E) was higher. They also concluded that, with IHX, the COP of the system for R-1234ze(E) was 3 % higher than that obtained for R134a without IHX.

Yang et al. [12] compared R513A and R-134a in a domestic refrigerator without IHX and reported that R-513A offered a higher refrigeration capacity in comparison with R134a. They also evaluated the energy consumption and found that using R513A, the energy was reduced by 3.5 % in comparison with R-134a. Mota-Babiloni et al. [13] presented an exergy analysis of an experimental setup which operated with R134a and the alternative HFO/HFC mixture R513A. The evaporating temperature ranged between −15 °C and 5 °C, while the condensing temperature was set at 30 °C and 35 °C. Pérez-García et. al [14] presented a study on the first and second law of thermodynamics using experimental data from a medium capacity

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refrigeration system using R-450A, R-513A and R-134a as working fluids.

Mota-Babiloni et al. [15] performed an exergy study in an experimental installation with R-513A without IHX, demonstrating that the global exergy efficiency of R-513A was slightly higher than the one obtained with R-134a under the same operation conditions, though R-513A presented a higher amount of exergy destruction. Mota-Babiloni et al. [16] performed an energy comparison in an experimental installation where they found that the refrigeration capacity was higher for R-134a in comparison with R-450A with and without IHX. Wantha [17] outlined the experimental and theoretical evaluation of heat transfer characteristics of a tube-in-tube internal heat exchanger for R1234yf and R134a refrigerants. They investigated the relationship between the COP and the overall heat transfer coefficient, the influence on the length and effectiveness of the heat exchanger, annular space, and pressure drops under operating conditions between −6.4 °C and +6.4 °C evaporation and 46 °C condensation temperature. It was revealed that the exergetic COP increased with the effectiveness of the internal heat exchanger, i.e., 4.4 % for R1234yf and 1.35 % for R134a. Belman-Flores et. al [18] presented an energy and exergy analysis of a domestic refrigeration system using R1234yf as a drop-in replacement for R134a. Mendoza-Miranda et. al [19] in their comparative evaluation of the compressor predictions showed a reduction in the cooling capacity obtained with R1234yf, R450A and R1234ze(E), in comparison with R134a. Also, COP values for R1234yf, R450A, and R1234ze(E) were lower than those obtained from R134a. Makhnatch et al. [20] energetically evaluated R-450A and R-513A in comparison with R-134a in an experimental system at higher condensation temperatures. While R-513A surpassed the COP of R-134a in 1.8 %, R-450A presented a decrease of 5.3 %. Daviran et al [21] concluded that the refrigerant-side overall heat transfer coefficient of HFO-1234yf was 18–21 % lower than that of HFC-134a, and the pressure drop was 24 % and 20 % smaller than HFC-134a during condensing and evaporating processes, respectively. Further, in a constant cooling capacity, the COP of HFO-1234yf was lower than HFC-134a by 1.3 % – 5 %, and in the second case, the COP of HFO-1234yf was about 18 % higher than that of HFC-134a. Navarro-Esbrí et al. [22] studied R1234yf performance in a vapor compression system varying a wide range of condition, concluding that the cooling capacity and COP for R1234yf were about 9 % and 19 % lower than those obtained using R134a. Meng et. al [23] deduced that the capacities of R1234yf/R134a and R134a were similar in cooling and heating modes. The COP of R1234yf/R134a was lower than that of R134a by 4–9 % in cooling mode, whereas it was lower by 4–16 % in heating mode. Lee and Jung [24] compared the performance of refrigeration systems using R1234yf and R134a as working fluid in the drop-in tests. The results of the experiment showed that the COP of R1234yf systems was 2.7 % lower than that of R134a system. Zilio et al. [25] concluded that the R1234yf cooling capacity and COP in a MAC were considerably lower than those obtained with R134a, and they suggested some hardware modifications in order to reduce the difference between both refrigerants. Cho et al. [26] concluded that when R134a and R1234yf were compared, R1234yf systems without internal heat exchangers (IHX) experienced a decrease of approximately 7 % and 4.5 % in cooling capacity and COP.

The aim of this paper was to create a mathematical model that would enable the detailed description of thermodynamic processes within heat pump system using internal heat exchangers, as well as determine the parameter changes in any point of the cycle.

The research will compare the COP, the heating capacity and the compressor power consumption of the heat pump system with internal heat exchanger in a wide range of operating conditions of a monitored heat pump system using R134a and R1234yf refrigerants. The mathematical model of the heat pumps was steady-state with lumped parameters. The experimental validation of the model was carried out with 36 tests using R134a and R1234yf as working fluid.

2 Design of the mathematical model for the heat pump system

This concentrated parameter mathematical model takes into account all main components of the heat exchanger heat pump system. These are the evaporator and condenser which are shell and tube heat exchangers, the IHX which is a plate heat exchanger, the reciprocating compressor, and the capillary tubes.

The evaporator is divided into one zone, namely evaporation, the condenser is divided into two zones, namely superheating, two-phase zones. Lumped parameters are considered in each zone in the heat exchangers, and also in the compressor and in the expansion valve. The model disregards the following:

- Steady-state operating conditions.
- The superheated degree at the outlet of the evaporator and condenser are fixed as 0 K.
- Pressure losses in the heat exchangers are neglected.
- Isenthalpic expansion process.

The corresponding p-h diagram of vapor compression cycle considered is presented in Fig. 1 [27].

2.1 Evaporator

Equations (1–3) detail the steady state energy balance in the evaporator:

- For the heat flux from the cooling water:

$$q_{eva} = \dot{m}_w \cdot c_{p_w} \cdot \left(T_{in} - T_{out}\right). \tag{1}$$

- For the heat flux absorbed by the refrigerant:

$$q_{eva} = \dot{m}_r \cdot \left(h_1 - h_{4'}\right). \tag{2}$$

- For the heat transfer between the cooled water and refrigerant is:

$$q_{eva} = U \cdot A \cdot \Delta T_{lm}. \tag{3}$$

2.2 Condenser

Equations (4–6) describe the steady state energy balance in the condenser:

- The heat flux absorbed by heated water is given in:

$$q_{con} = \dot{m}_w \cdot c_{p_w} \cdot \left(T_{in} - T_{out}\right). \tag{4}$$

- The heat flux from the refrigerant is:

$$q_{con} = \dot{m}_r \cdot \left(h_{2'} - h_3\right). \tag{5}$$

- The heat transfer between the refrigerant and heated water is:

$$q_{con} = U \cdot A \cdot \Delta T_{lm}. \tag{6}$$

2.3 Internal heat exchanger

The heat transfer rate can be defined for three equations defined on the refrigerant – vapor, liquid-side and in the wall interface region. These are, respectively:

- Heat flux from the liquid phase of the refrigerant:

$$q_{ihx} = h_3 - h_{3'} = \dot{m}_r \cdot c_{p_{liq}} \cdot \left(T_3 - T_{3'}\right). \tag{7}$$

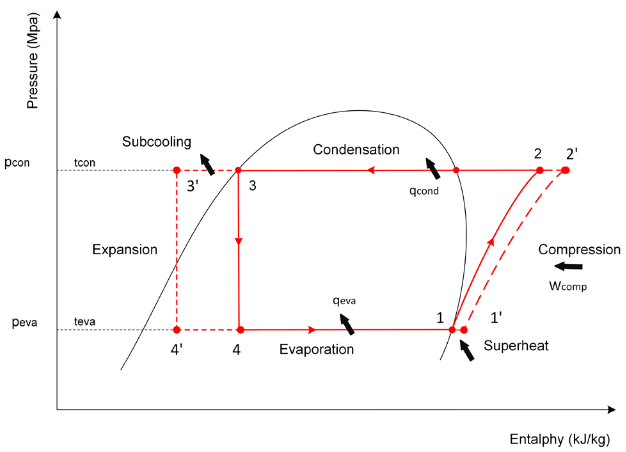

Fig. 1 p-h diagram of the heat pump cycle [27]

- Heat flux absorbed by the vapor phase of the refrigerant:

$$q_{ihx} = h_{1'} - h_1 = \dot{m}_r \cdot c_{pvap} \cdot \left(T_{1'} - T_1\right). \tag{8}$$

- The heat transfer between the vapor and liquid phase of the refrigerant:

$$q_{ihx} = U \cdot A \cdot \Delta T_{lm}. \tag{9}$$

2.3.1 Auxiliary equations

In addition to the afore-mentioned basic energy equations, several auxiliary equations are required in the simulation of the heat pump.

The intensity of the heat transfer between the vapor and liquid phase of the refrigerant or water and refrigerant were determined by the overall heat transfer coefficient:

$$U = \frac{1}{\frac{1}{\alpha_{vap}} + \frac{\delta}{\lambda} + \frac{1}{\alpha_{liq}}}. \tag{10}$$

After neglecting the thermal resistance of the wall:

$$U = \frac{\alpha_{vap} \cdot \alpha_{liq}}{\alpha_{vap} + \alpha_{liq}}. \tag{11}$$

The literature lists numerous condensation heat transfer correlations for the various types of refrigerants and different conditions. In the model, the well-known Shah correlation [28] was applied:

$$\alpha = \alpha_f \cdot \left(1 + \frac{3.8}{Z}\right), \tag{12}$$

where:

$$Z = \left(\frac{1-x}{x}\right)^{0.8} \cdot \left(p\right)^{0.4}. \tag{13}$$

For single-phase heat transfer correlation determined with Eq. (17).

For the definition of the evaporation heat transfer correlation, the well-known Chan correlation [29] was used:

$$\alpha_{tp} = F \cdot \alpha_f + S \cdot \alpha_{nb}. \tag{14}$$

The two-phase correction multiplier for convective boiling:

$$F = b \cdot \frac{Co^c}{X_{tt}{}^d}. \tag{15}$$

The two-phase correction multiplier factor for bubble boiling:

$$S = a \cdot (1 - x). \tag{16}$$

For the single-phase heat transfer, it can be used Dittus-Boelter [30] correlation:

$$\mathrm{Nu} = 0.023 \cdot \mathrm{Re}_e^{0.8} \cdot \mathrm{Pr}^n . \tag{17}$$

The logarithmic mean temperature differences in the heat exchangers can be calculated as seen here:

$$\Delta T_{lm} = \frac{\Delta T_{\max} - \Delta T_{\min}}{\ln \dfrac{\Delta T_{\max}}{\Delta T_{\min}}} . \tag{18}$$

2.4 Expansion valve

The same reference gives the capacity of TEV to be [31]:

$$Q_{tev} = C_o \cdot \sqrt{\rho \cdot (t_{con} - t_{eva})} \cdot [h_{con} - h_{eva}] . \tag{19}$$

2.5 Compressor

The compressor power consumption is the enthalpy difference and is expressed as a function of the compressor efficiency:

$$W = \dot{m}_r \cdot (h_{con} - h_{eva}) \cdot \frac{1}{\eta} . \tag{20}$$

3 Solution of the mathematical model of the heat pump system

The solution of the algorithm makes it possible to understand and trace the logical structure of the solution of the formulated mathematical model since it gives the model its accuracy and applicability. The block diagram of the algorithm is shown in Fig. 2.

4 Experimental procedure

In the course of this research the author performed laboratory and service tests for the fully comprehensive computerized modeling of heat pump systems with IHX, to determine the primary and secondary water parameters, and the refrigeration thermodynamic parameters in the cycle. The specification of the heat pump with internal heat exchanger is presented in Table 1.

Measuring points were formed on the heat pump system, which is shown in Fig. 3. Various measuring instruments have been used to measure temperature, pressure, mass and volume flow as well as compressor power consumption.

The accuracy and uncertainty of the measuring equipment used in the experiments are summarized in Table 2.

Total number of measurements was 36. 18 measurements were performed with R134a refrigerant and 18 measurements with R1234yf refrigerant. The measurement ranges are summarized in Table 3.

5 Results and discussion

The simplest way to verify the goodness of a mathematical model set up to describe the behavior of a heat pump system with an internal heat exchanger was to perform laboratory measurements and compare the obtained results with the results provided by the model, which are presented in Figs. 4–14.

The model error was determined by the difference between the real measured and predicted values of the energy parameters, according to Eq. (21):

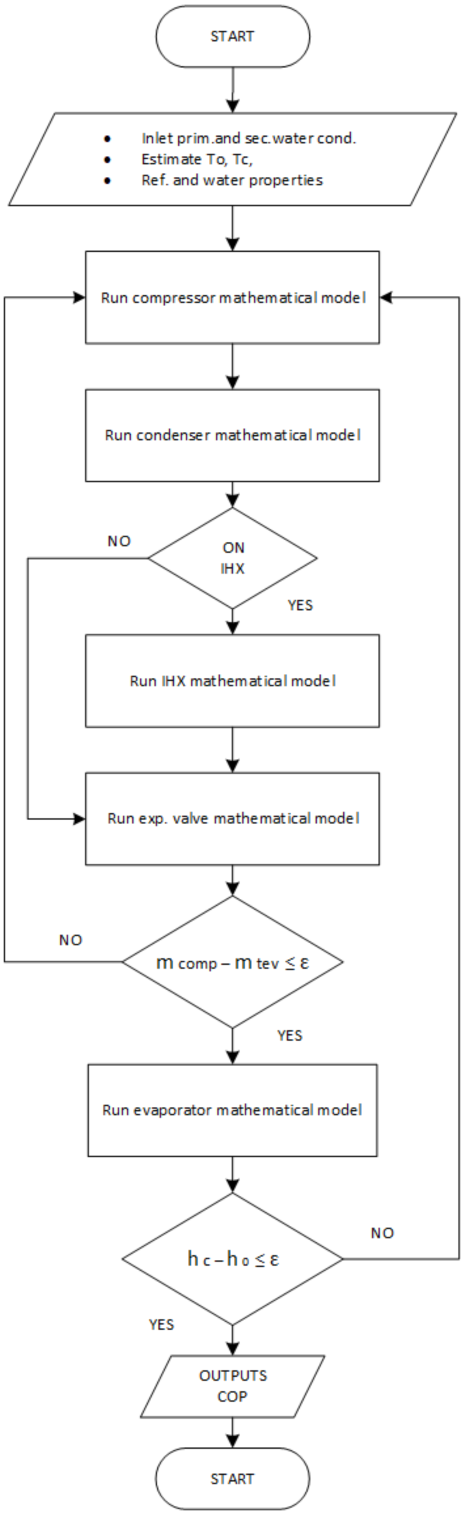

Fig. 2 Flowchart of the simulation program

Table 1 Specifications of the main components of the heat pump system.

Name	Remarks
Compressor	Reciprocating compressor, CAJ4511Y 1hp Valve Tecumseh Compressor R134a
Evaporation	Shell and Tube Heat Exchanger. Shell diameter: 32 mm. Tube diameter: 6 mm. Number of tubes: 5. Evaporator length: z = 3 m
Condensation	Shell and Tube Heat Exchanger. Shell diameter: 32 mm. Tube inner diameter: 6 mm. Number of tubes: 5. Condenser length: 3 m
Expansion valve	Internally-equalized type
Internal heat exchanger	Plate heat exchanger, SWEP no. 14314-010

$$H = \frac{\left|V_{\exp} - V_{pred}\right|}{V_{\exp}}. \quad (21)$$

In the current case, both refrigerants flow through in a tube bundle of the heat exchangers, while the primary and secondary fluid flows in the shell across the bundle. The evaporator was always operated with constant inlet cooling water temperatures T_{cw} = 13 °C, while the outlet heated water temperatures of the condenser were T_{hw} = 35 °C. The mass flow rate in the shell side of the evaporator and condenser were $\dot{m}_w = 0.07 \frac{\text{kg}}{\text{s}}$.

The temperature profiles of the primary and secondary fluid at the outlet in the shell side of the evaporator and the condenser are shown in Figs. 4 and 5.

In Fig. 4, the maximum difference between the measured and predicted values for the temperature profile of the cooling water flow in the shell side of evaporator is 0.46 °C, while Fig. 5 shows the deviation of the heated water flow in the shell side of condenser, which is 0.58 °C.

Figs. 6 and 7 show the difference between the measured and calculated values of the vapor phase refrigerant temperature profiles inlet and outlet of the compressor for refrigerants R134a and R1234yf. At the compressor inlet the discrepancy is a prediction error of 0.44 °C, while at the compressor outlet, it is 0.57 °C.

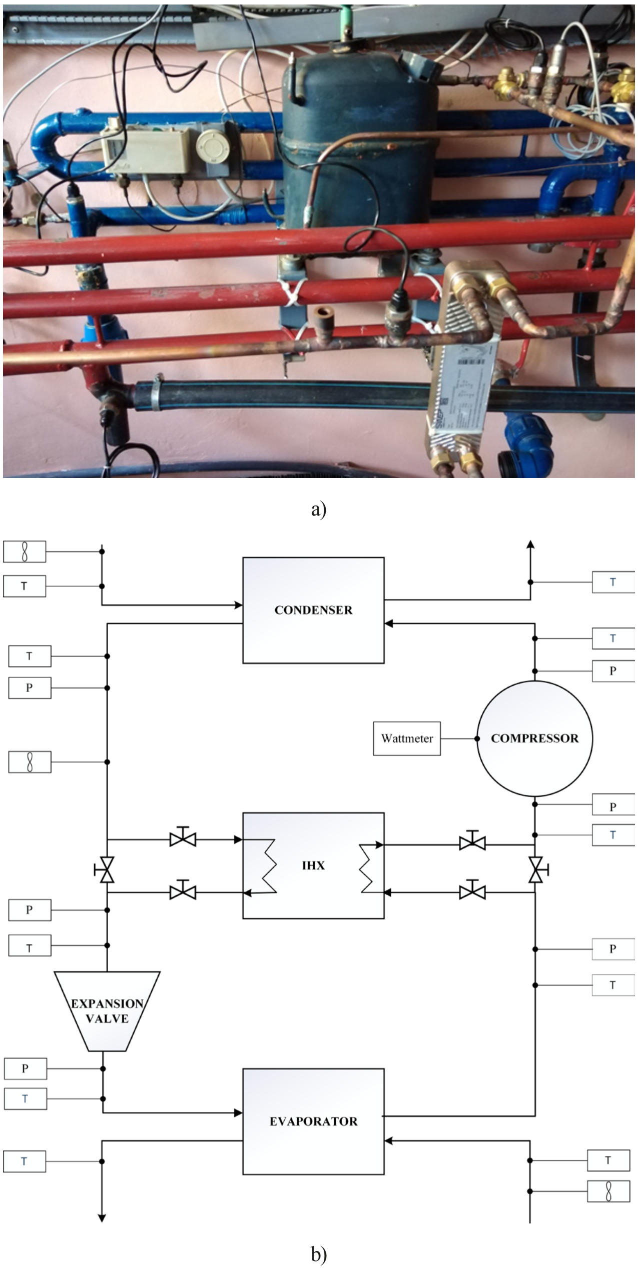

Fig. 3 a) Photo and b) schematic diagram of the experimental setup

Table 2 Measured parameters and equipment uncertainty.

Measured parameters	Sensor	Uncertainty
Temperatures	K-type thermocouples DS18B20 90807A	± 0.2 K
Pressures	Transducers TD220030 ELIWELL EWPA 030	± 1 %
Flow meters	Coriolis mass flow meter, and water flow sensors turbine flowmeters	± 0.2 %
Compressor power consumption	Digital watt meter	± 0.15 %

Table 3 Range of operating condition in the experimental tests.

Controlled parameters	Range values
Condensing temperature (T_{con})	41–62 °C
Evaporating temperature (T_{eva})	3–6 °C
Compressor power consumption (W)	600–1400 W

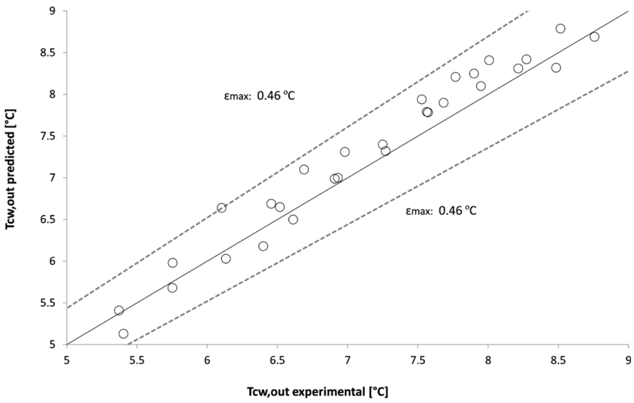

Fig. 4 Prediction of the water temperature in the shell side evaporator

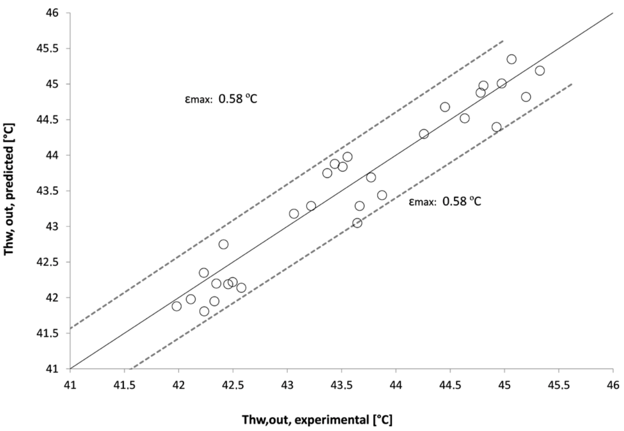

Fig. 5 Prediction of the water temperature in the shell side condenser

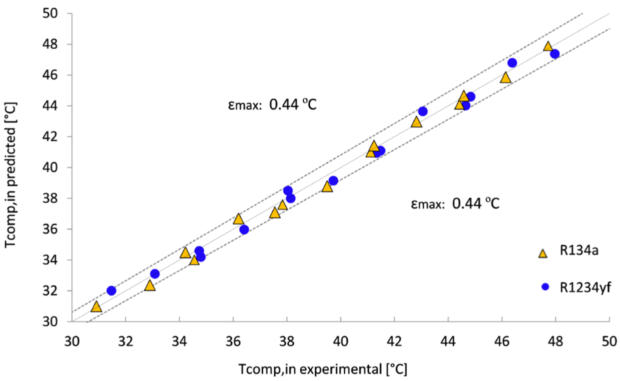

Fig. 6 Prediction of the temperature at the compressor inlet

Fig. 8 shows the difference between the temperature values obtained from the model and the measured values for the two refrigerants. The relative error prediction for the internal heat exchanger outlet temperature is about 0.39 °C, when the refrigerant is in the liquid phase.

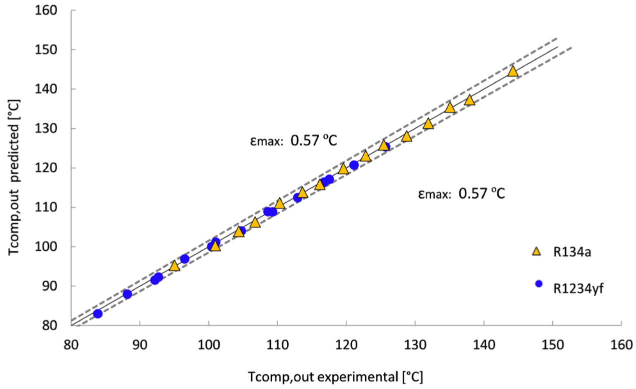

Fig. 7 Prediction of the temperature at the compressor outlet

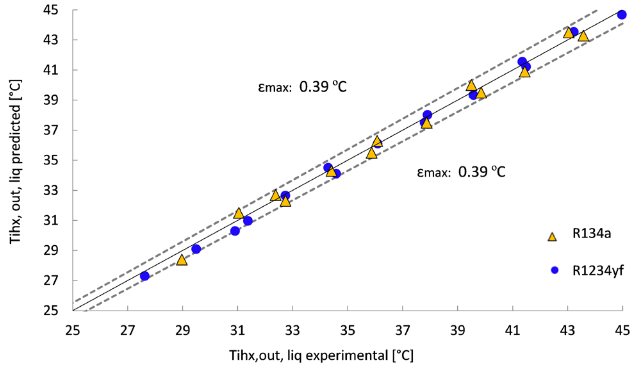

Fig. 8 Prediction of the temperature at the internal heat exchanger outlet

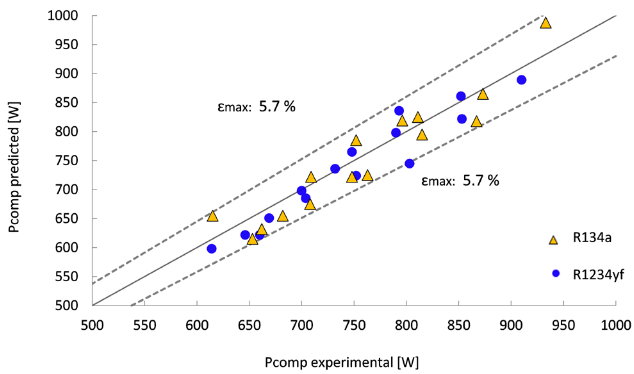

Fig. 9 Prediction of compressor power consumption

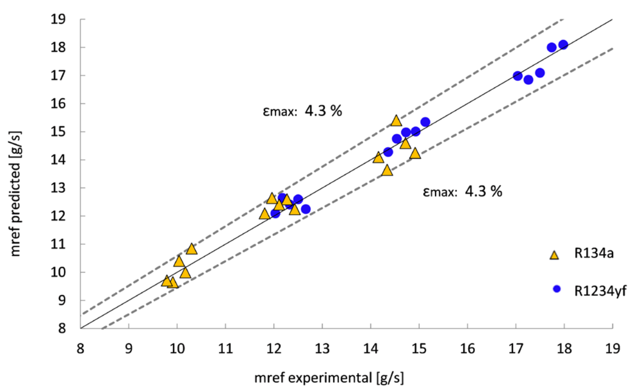

Fig. 10 Prediction of the refrigerant mass flow rate

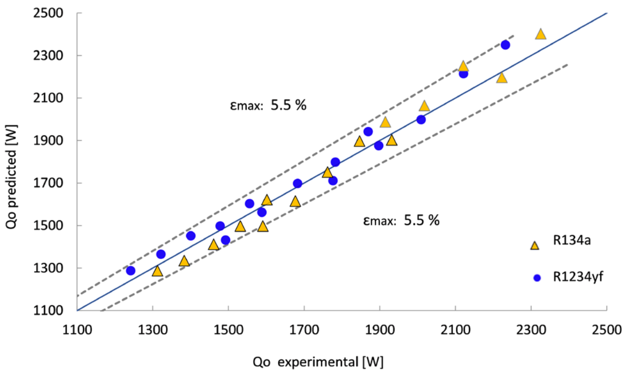

Fig. 11 Prediction of the cooling capacity

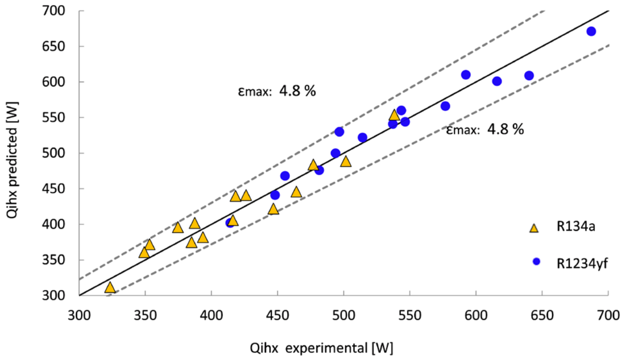

Fig. 12 Prediction of the heat transfer rate in the internal heat exchanger

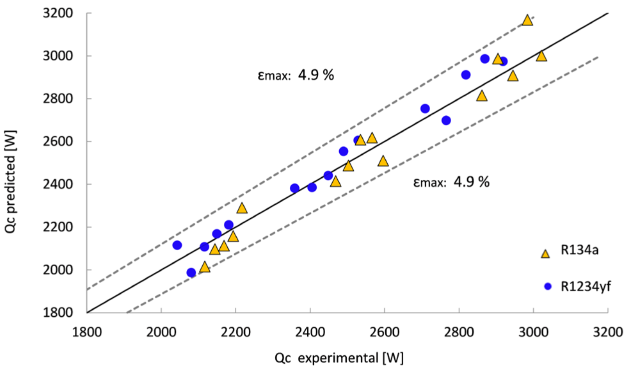

Fig. 13 Prediction of condenser capacity

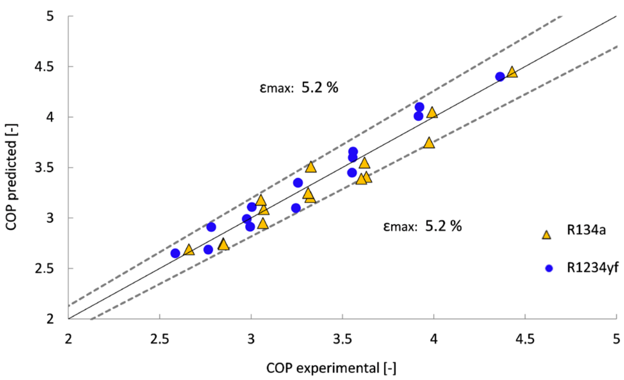

Fig. 14 Prediction of the coefficient of performance COP

As shown in Fig. 9, the maximum deviation of the compressor power consumption values obtained from the model from the measured values is 5.7 % for the R1234yf and R134a refrigerants.

The comparison between the measured mass flow rate values with the model values is given in Fig. 10. The predicted values are within an error of 4.3 %.

Fig. 11 shows that the maximum deviation of the cooling capacity values provided by the model from the measured values is 5.2 % for R134a and R1234yf refrigerants.

Figs. 12 and 13 show the difference between predicted and measured values of the heat transfer rate in the internal heat exchangers and the condenser capacity. The predicted values are within an error of 4.8 % and about 4.9 %, respectively.

Fig. 14 shows the difference between the model and the experimental results for the COP values using both refrigerants. It can be seen that the maximum relative difference was 5.4 %. The mathematical model error is acceptable, similar to other research results mentioned in literature [7, 10, 17]. Figs. 15–17 compare the various energy performance values of the heat pump system with internal heat exchanger in a wide range of operating conditions using the R134a and R1234yf refrigerants.

Fig. 15 shows the variation of compressor power consumption at different condensation temperatures from 50 °C to 80 °C and evaporation temperatures from −10 °C to 10 °C using R134a and R1234yf. For R1234yf the values of compressor power consumption are very similar to those for R134a at low condensation temperature 50 °C. When the condensation temperature is higher, 80 °C and the evaporation temperature rises to 10 °C, the difference is 3.51 %. In fact, the values determined for R134a are higher than those determined for R1234yf.

Fig. 16 shows the variation of heating capacity at different condensation temperatures from 50 °C to 80 °C and evaporation temperatures from −10 °C to 10 °C using R134a and R1234yf. For R1234yf, the heating capacity values are smaller than those obtained for R134a under the above-mentioned conditions. In smaller evaporation temperatures −10 °C and condensation temperatures 50 °C, the difference between heating capacities of R1234yf and R134a is very similar at 0.63 %. When the evaporation temperature rises from −10 °C to 10 °C, the difference is 3.74 % in the case of R134a. When higher condensation temperature 80 °C is considered, the difference between heating capacities of R1234yf and R134a is increased. The difference between both refrigerants increases between 4.18 % and 7.54 % when

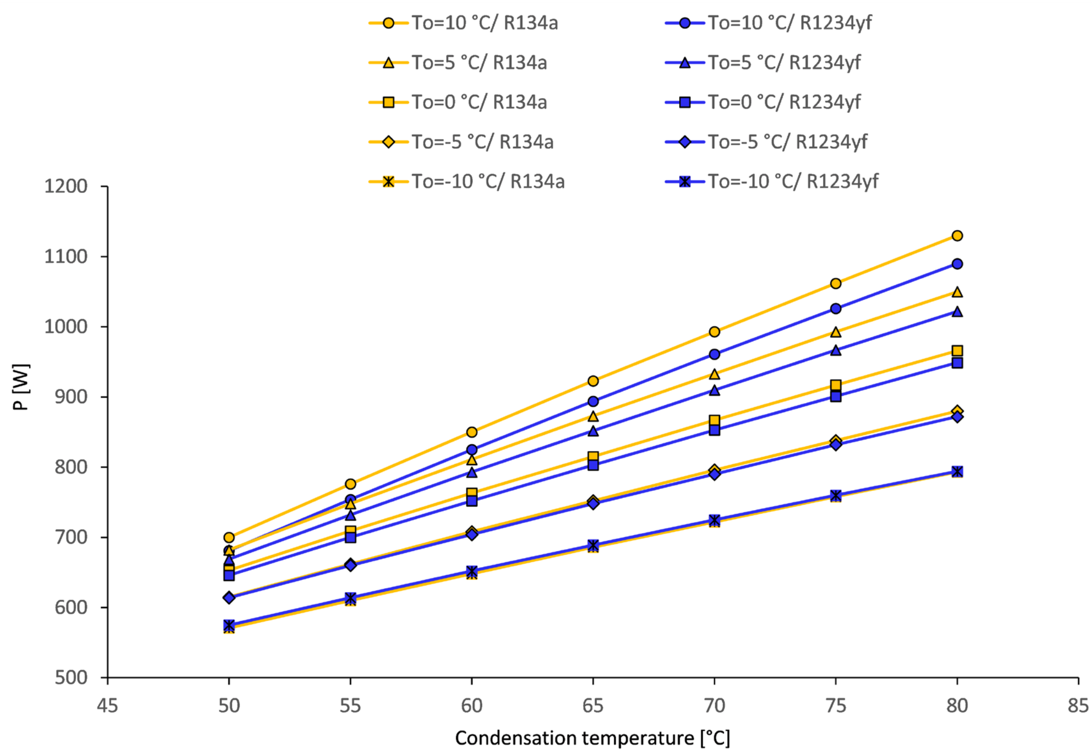

Fig. 15 Compressor power consumption versus condensation temperature

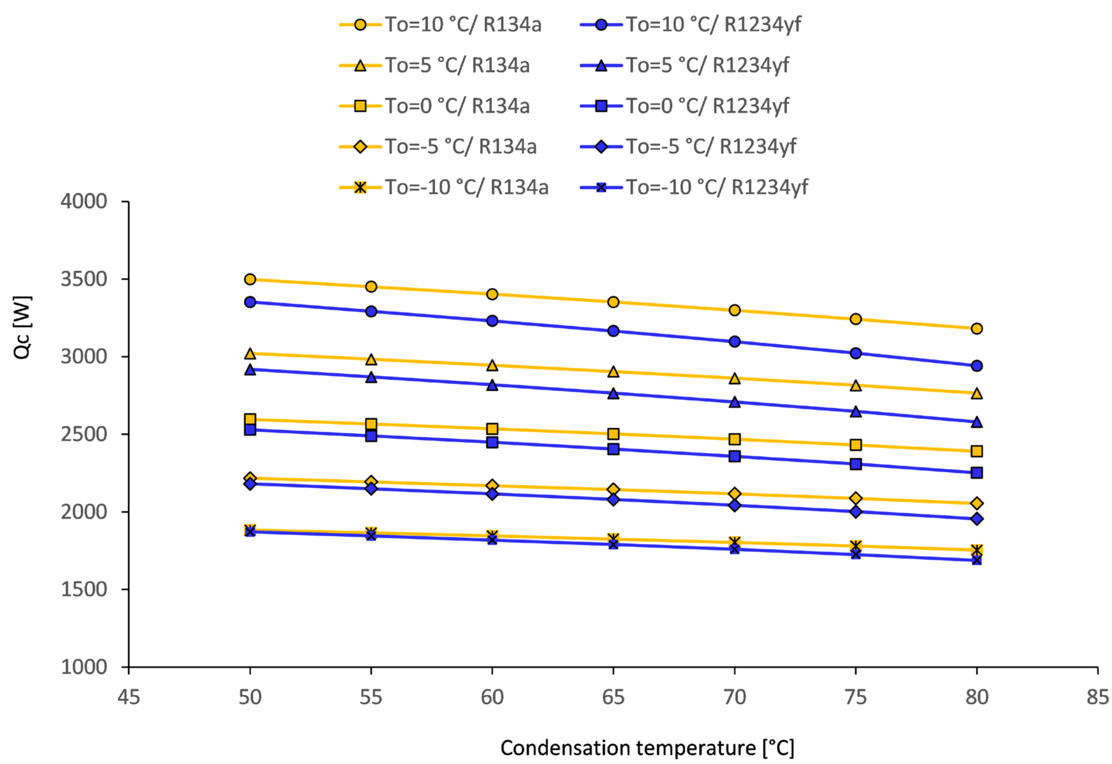

Fig. 16 Heating capacity versus condensation temperature

evaporation temperatures change from −10 °C to 10 °C, i.e., in all cases R134a provides the greater values.

Fig. 17 shows the variation of COP at different condensation temperatures from 50 °C to 80 °C and evaporation

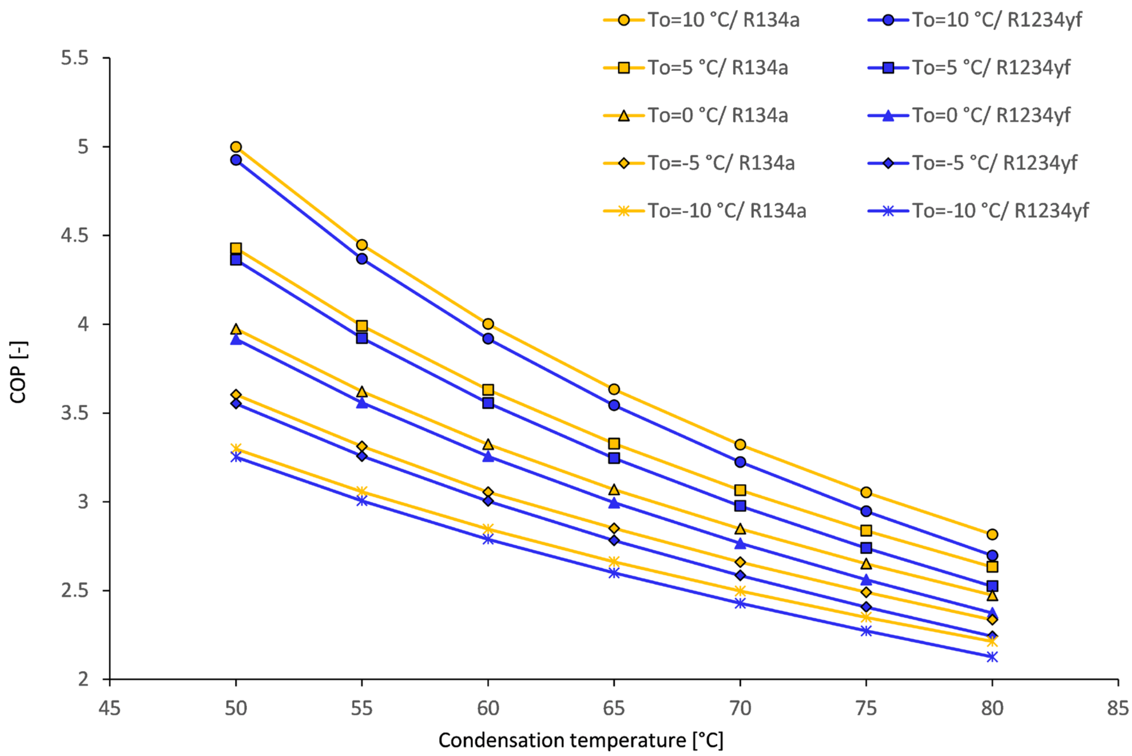

Fig. 17 COP versus condensation temperature

temperatures from −10 °C to 10 °C using R134a and R1234yf. For R1234yf the COP values are smaller than those for R134a as measured in the above-mentioned conditions. In smaller evaporation −10 °C and condensation temperatures 50 °C, the difference between COP values of R1234yf and R134a is 1.38 %, with R134a being greater. At condensation temperature 50 °C, when the evaporation temperature rises from −10 °C to 10 °C, the difference in COP values is higher, at 3.93 % where the higher values stem from R134a. When higher condensation temperatures 80 °C are considered, the difference between COP values of R1234yf and R134a is increased. The difference between both refrigerants increases between 1.48 % and 4.22 % when evaporation temperatures change from −10 °C to 10 °C, again the higher value is that for R134a.

6 Conclusions

This paper presents a steady-state model for a vapor compression heat pump with internal heat exchanger. During the research, the author compared the proposed and experimental results obtained in the test bench using R1234yf and R134a, for the main energy performance parameters: cooling and heating capacity, heat transfer rate in the IHX, compressor power consumption, and COP. Other variables of interest, such as refrigerant mass flow rate, temperature of refrigerant in the main points of the cycle and primary and secondary water temperature were also included. The results are summed up in Figs. 4–14. A total of 36 experimental tests were conducted for the model validation using a wide range of operating conditions, which allowed for thoroughly checking the model for robustness. Furthermore, the energy performance of the heat pumps was compared using both refrigerants in function of the evaporation temperature between −10 °C and 10 °C, and the condensation temperature between 50 °C and 80 °C. The compressor power consumption for R1234yf is similar than that obtained for R134a when the evaporation −10 °C and condensation temperature 50 °C are low, as summarized in Fig 15. The difference between both refrigerants decreases by between 2.7 % and 3.53 % when the condensation temperatures was 80 °C, the greater value is that of R134a.

The heating capacity for R1234yf is between 0.63 % and 7.54 % lower than for R134a in the tested range, which is presented in Fig. 16, whereas Fig. 17 illustrates the COP difference obtained using R1234yf, which is between 1.48 % and 4.22 % lower than that obtained for R134a. The tests highlight that in conditions when the condensation temperature rises, the COP difference will also increase. Further research in the topic could be conducted in the field of investigating energetics parameters of the heat pump systems, both with or without the use of IHX.

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