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# Investigations of the performance of a heat pump with internal heat exchanger

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## Abstract

The aim of this study was to develop a mathematical model with lumped parameters to investigate the energetic parameters of the heat pump with an internal heat exchanger (IHX). The developed mathematical model is validated with 25 tests using R134a as a working fluid. The results show that the maximum prediction error between the modeled and experimental results for the COP is 7.06%. The main objective of this paper was to investigate the COP value of a heat pump as a function of the efficiency of the internal heat exchanger and present a new equation to calculate the COP value. When the efficiency of IHX was increased from 0.65 to 0.95, the COP value increased by 5.41% under the conditions of minimum evaporation $-20\ °C$ and maximum condensation temperature $90\ °C$.

**Keywords** Heat pump · Efficiency · IHX · COP

## List of symbols

| | |
|---|---|
| $A$ | Heat transfer area ($m^2$) |
| $C$ | Characteristic constant of the TEV valve (–) |
| $Cp$ | Specific heat at constant pressure ($J\ kg^{-1}\ K^{-1}$) |
| COP | Coefficient of performance (–) |
| $D$ | Diameter (m) |
| $h$ | Enthalpy ($J\ kg^{-1}$) |
| Nu | Nusselt number (–) |
| $\dot{m}$ | Mass flow rate ($kg\ s^{-1}$) |
| $P$ | Compressor power (kW) |
| Pr | Prandtl number (–) |
| Rc | Capacity ratio (–) |
| Re | Reynolds number (–) |
| $Q$ | Heat transfer rate (kW) |
| $T$ | Temperature (K) |
| $U$ | Overall heat transfer coefficient ($W\ m^{-2}\ K^{-1}$) |

## Greek symbols

| | |
|---|---|
| $\alpha$ | Heat transfer coefficient ($W\ m^{-2}\ K^{-1}$) |
| $\eta$ | Compressor efficiency (–) |
| $\lambda$ | Thermal conductivity ($W\ m^{-1}\ K^{-1}$) |
| $\rho$ | Density ($kg\ m^{-3}$) |
| $\varepsilon$ | Efficiency of the IHX (–) |

## Subscripts

| | |
|---|---|
| $r$ | Refrigerant |
| $c$ | Condenser |
| $o$ | Evaporator |
| liq | Liquid phase |
| vap | Vapor phase |
| cw | Cooling water |
| hw | Hot water |
| in | Inlet |
| out | Outlet |

## Introduction

The direction of energy developments is clearly determined by energy-saving and environmental issues. Resources must be invested into developing material and energy-saving technologies, improving their efficiency, and reducing their environmental impact.

The greatest energy savings can be achieved by rationalizing the production and use of energy, reducing heat loss in buildings, and optimizing selection and operation of the heating equipment. With the modernization of heating technologies, the heat pump can be expected to be an indispensable device in the near future. There are several studies in the literature describing the behavior of the heat pumps under different conditions, with smaller to larger approximations [1–3]. Recent years have seen a considerable number of studies published on heat pump systems with an internal heat

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exchanger, in which the role of the internal heat exchanger was investigated in the cycle [4–8].

In their work describing the heat transfer of the internal heat exchanger, Klein et al. [9] showed the effectiveness of the heat exchanger as the most important parameter further, they revealed the relationship between the relative capacity index (RCI) and the effectiveness of liquid-suction heat exchangers to be almost linear. Their study involved the following refrigerants: R507A, R404A, R600, R290, R134a, R407C, R410A, R12, R22, R32, and R717. The key issue in the study by Kwon et al. [10] was the heat transfer characteristics of internal heat exchangers for $CO_2$ heat pump systems. The outcomes of their experiment demonstrated that the with a greater length of the internal heat exchanger, the capacity, effectiveness, and pressure drop would be greater as well. Mota-Babiloni et al. [11] used 30% of the effectiveness of the tube-in-tube heat exchanger with vapor flows through the inner tube while the liquid flowed in the annulus between the inner and outer tubes of heat exchanger. Their results confirmed that the cooling capacity for R1234yf and R1234ze was lower than that of R134a, specifically for R1234ze. Wantha [12] described the experimental and theoretical evaluation of heat transfer characteristics of a tube-in-tube internal heat exchanger for R1234yf and R134a refrigerants. The research studied how the COP and the overall heat transfer coefficient are related to one another, what impact the length and effectiveness of the heat exchanger, annular space, and pressure drop had, given the operating conditions between −6.4 and 6.4 °C evaporation and 46 °C condensation temperature. The study showed that the exergetic COP increased with the efficiency of the internal heat exchanger, i.e., 4.4% for R1234yf and 1.35% for R134a.

Another research by Torrella et al. [13] sought to analyze the performance of an IHX operating in a $CO_2$ transcritical refrigeration plant at three evaporating levels (−5 °C, −10 °C and −15 °C) and two different gas-cooler outlet temperatures (31 °C and 34 °C). The findings indicated the following: a maximum increment on cooling capacity of 12% and an increment of the cycle efficiency up to 12%. Mota-Babiloni et al. [14] conducted an energy comparison in an experimental installation. The results confirmed that the refrigeration capacity was higher for R134a in comparison with R450A with and without IHX. The study by Direk et al. [15] investigated how internal heat exchanger (IHX) effectiveness affected the performance parameters of the refrigeration cycle with R1234yf. Perez et Al. [16] analyzed the first and second law of thermodynamics with the implementation of experimental data from a medium capacity refrigeration system using R450A, R513A and R134a as working fluids.

The aim of this research is to create a concentrated parameter mathematical model that takes into account the parameters defining of the heat pump cycle, temperature and

enthalpy values, couplings and boundary conditions with sufficient accuracy, but neglects all secondary characteristics that are not considered crucial for the COP analysis. The aspect of the presented model is to express the laws of nature describing the heat pump system in algebraic mathematical formulas as well as to communicate the error and the applicability limit of the model. The main objective of the study is to accurately describe the heat transfer and flow processes in the heat pump cycle with internal heat exchanger, and thus to improve the COP value of the equipment and present a new calculation formula that takes into account the efficiency of the internal heat exchanger. The investigated conditions range between −20 °C and 10 evaporation temperature, 40 °C–90 condensation temperature and the internal heat exchanger between 0.65 and 0.95 interval.

## Modeling

The given concentrated parameter mathematical model, set up by the author, is a mathematical description of the phenomena, the processes, and their interrelationships in a real physical system. This model allows the understanding of the processes in the heat pump and makes it possible to study the energy parameters of the system. The investigation of the heat pump with IHX is performed with the implementation of a steady-state mathematical model.

A heat pump system consists of five main parts, three heat exchangers, namely the evaporator, condenser and internal heat exchanger, as well as a reciprocating compressor and an expansion valve. The working fluid of the circuit is the R134a refrigerant, while the working fluid of the primary and secondary circuit is water. The evaporator and condenser are shell and tube type, the refrigerant flow in the tubes, while heated and cooled water flows in the shell side of the heat exchangers. Refrigerant evaporation takes place over the entire length of the evaporator, while in the condenser the refrigerant is overheated and condensed.

The mathematical model observed each region in the heat exchangers. The internal heat exchanger is a plate heat exchanger and its efficiency was considered with the NTU model $\varepsilon = f(NTU, Rc)$. The Number of Transfer Units (NTU) Method is used to calculate the rate of heat transfer in heat exchangers. Capacity ratio $R_c = \frac{C_{min}}{C_{max}}$.

The following simplifications and omissions have been made in the mathematical model:

- The pressure drop of the refrigerant is neglected in the heat exchangers.
- Negligible heat loss to the surroundings.
- Constant refrigerant properties.

- The superheated temperature degree at the outlet of the evaporator is fixed as 0 K.

Figure 1 shows the heat pump cycle in the logp-h diagram.

## Evaporator

The evaporator is where the heat transfer between the cooled water and the refrigerant takes place. The heat transfer between the cooling water and refrigerant determined by the following Eqs. (1–4) is defined, respectively, as follows:

The heat transfer rate for the cold water is：

$$Q_{cw} = \dot{m}_{cw} \cdot c_{pcw} \cdot \left(T_{in,cw} - T_{out,cw}\right) \tag{1}$$

The overall energy balance for the evaporator is：

$$Q_{ihx} = U \cdot A \cdot \Delta T_{lm} \tag{2}$$

The heat transfer rate for the refrigerant is：

$$Q_o = \dot{m}_r \cdot \left(h_1 - h_{4\}}\right) \tag{3}$$

The heat balance in the evaporator for the lossless case：

$$Q_{cw} = Q_{ihx} = Q_o. \tag{4}$$

## Condenser

The condenser is where the heat transfer between the refrigerant and the heated water takes place. The superheated refrigerant enters the condenser, where it is condensed



**Fig. 1** p–h diagram of the refrigeration cycle

before turning into a saturated liquid. The heat transfer between the refrigerant and heated water determined for Eqs. (5–8) is defined, respectively, as follows:

The heat transfer rate for the refrigerant is：

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

The overall energy balance for the condenser is：

$$Q_{ihx} = U \cdot A \cdot \Delta T_{lm} \tag{6}$$

The heat transfer rate for the heating water is：

$$Q_{hw} = \dot{m}_{hw} \cdot c_{phw} \cdot \left(T_{in,hw} - T_{out,hw}\right) \tag{7}$$

The heat balance in the condenser for the lossless case：

$$Q_c = Q_{ihx} = Q_{hw} \tag{8}$$

## Internal heat exchanger

The internal heat exchanger is a counterflow heat exchanger. The liquid phase refrigerant flows on one side of the heat exchanger, while the vapor phase refrigerant flows on the other side. The heat transfer in the heat exchanger is described by the following Eqs. (9–12):

The heat transfer rate for the liquid phase of the refrigerant:

$$Q_{liq} = \dot{m}_r \cdot \left(h_3 - h_{3\}}\right) = \dot{m}_r \cdot c_{pliq} \cdot \left(T_3 - T_{3'}\right) \tag{9}$$

The heat transfer rate for the vapor phase of the refrigerant:

$$Q_{vap} = h_{1\}} - h_1 = \dot{m}_r \cdot c_{pvap} \cdot \left(T_{1'} - T_1\right) \tag{10}$$

The heat transfer rate for a crossflow internal heat exchanger is:

$$Q_{ihx} = U \cdot A \cdot \Delta T_{lm} \tag{11}$$

The heat balance in the internal heat exchanger for the lossless case:

$$Q_{liq} = Q_{vap} = Q_{ihx} \tag{12}$$

While the mass flow rate in the internal heat exchanger is the same, specific heat is different of the flow refrigerants. The temperature change of the vapor phase refrigerant is higher than that of the liquid phase. The change of enthalpy is the same as of the refrigerant.

The following Eq. (13) is the balance equation of the heat exchanger.

$$q = c_{pliq} \cdot \left(T_c - T_{1\}}\right) = c_{pvap} \cdot \left(T_{1\}} - T_o\right) \tag{13}$$

The maximum possible heat transfer rate is expressed as:

$$q_{\max} = c_{\text{pvap}} \cdot (T_3 - T_1) \tag{14}$$

The effectiveness of the heat exchanger is defined as the ratio of the actual heat transfer rate to the maximum possible heat transfer rate, seen in Eqs. (15) and (16):

$$\varepsilon = \frac{q}{q_{\max}} \tag{15}$$

$$\varepsilon = \frac{c_{\text{p,vap}} \cdot (T_{1\}} - T_{\text{o}})}{c_{\text{p,vap}} \cdot (T_3 - T_1)} = \frac{T_{1\}} - T_{\text{o}}}{T_3 - T_1} \tag{16}$$

From Eqs. (13) and (16), the outlets' temperature of the refrigerant at the IHX can be calculated [13] using the equation below:

$$T_{1\}} = T_o + \varepsilon \cdot (T_{\text{c}} - T_{\text{o}}) \tag{17}$$

$$T_{3\}} = T_{\text{c}} - \varepsilon \cdot \frac{c_{p,\text{vap}}}{c_{p,\text{liq}}} \cdot (T_{\text{c}} - T_{\text{o}}) \tag{18}$$

The intensity of the heat transfer between the vapor and liquid was determined by the overall heat transfer coefficient [17] using the equation:

$$U = \frac{1}{\frac{1}{\alpha_{\text{vap}}} + \frac{\delta}{\lambda} + \frac{1}{\alpha_{\text{liq}}}} \tag{19}$$

after neglecting the thermal resistance of the wall, the obtained equation is:

$$U = \frac{\alpha_{\text{vap}} \cdot \alpha_{\text{liq}}}{\alpha_{\text{vap}} + \alpha_{\text{liq}}} \tag{20}$$

where the fluid flow is turbulent, it can be determined from the Dittus-Boelter correlation [17] for single-phase heat transfer seen in Eq. (21):

$$\text{Nu} = 0.023 \cdot \text{Re}^{0.8} \cdot \text{Pr}^{n} \tag{21}$$

where $n$ is a constant, $n = 0.4$ for heating and $n = 0.3$ for cooling.

The logarithmic temperature difference:

$$\Delta T_{lm} = \frac{\Delta T_{\max} - \Delta T_{\min}}{\ln \frac{\Delta T_{\max}}{\Delta T_{\min}}} = \frac{(T_{\text{c}} - T_{1\}}) - (T_{3\}} - T_{\text{o}})}{\ln \frac{(T_{\text{c}} - T_{1\}})}{(T_{3\}} - T_{\text{o}})}} \tag{22}$$

## Expansion valve

Expansion devices are located between the condenser and the evaporator in the circuit, reducing the pressure from the condensing pressure to the evaporating pressure. The capacity of the TEV is defined as [18]:

$$\dot{Q}_{\text{r}} = C \cdot \sqrt{\Delta p \cdot \rho_{\text{r}}} \cdot [h_{\text{c}} - h_{\text{o}}] \tag{23}$$

where $C$ is a constant dependent of valve geometry.

Additionally, expansion is assumed to be an isenthalpic process seen in Eq. (24).

$$h_{\text{c}} = h_{\text{o}} \tag{24}$$

## Compressor

The compressor continuously suctions vapor phase of refrigerant from the evaporator to maintain a sufficiently low pressure and then compresses the vapor phase of the refrigerant and discharges it to the condenser. The compressor power consumption is determined by the mass flow rare, efficiency of compressor and enthalpy of the function of the partial pressures of the refrigerant.

$$P = \dot{m}_{\text{r}} \cdot (h_{\text{c}} - h_{\text{o}}) \cdot \frac{1}{\eta} \tag{25}$$

## Experimental procedure

The simplest way of verifying the validity of a mathematical model set up to describe the behavior of the system, performing measurements in the system and comparing the results with provided by the model. During the research, the author carried out numerous full-scale laboratory and operational experiments on a heat pump system with an internal heat exchanger.

The design of the measuring points on the heat pump system is shown in Fig. 2, while the type and accuracy of the measuring instruments is summarized in Table 1. The overall number of measurements was 25, the indicated measuring points directly measured the temperature of the R134 refrigerant which flowed in the heat pump system, and the compressor capacity and indirectly the heat capacities of the heat exchangers and COP value of the heat pump system.

The physical characteristics of the heat pump's main components are summarized in Table 2.

In order to determine the error of the mathematical model, the calculated and measured values of the investigated parameters were compared.
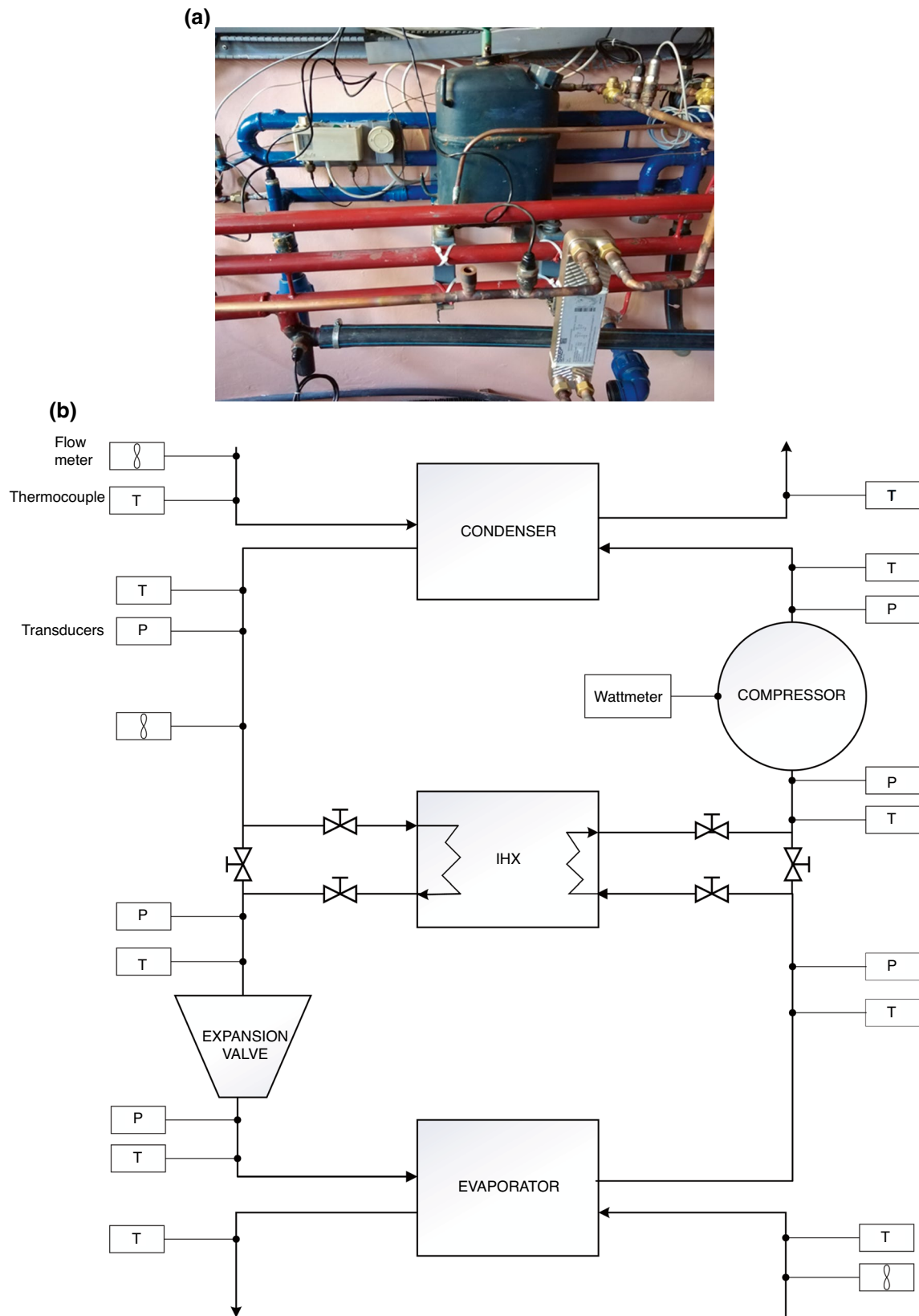
**(a)**



**(b)**



**Fig. 2** **a** Photo and **b** schematic diagram of the experimental setup

**Table 1** Measured parameters and equipment uncertainty [1]

| Measured parameters | Sensor | Uncertainty |
| --- | --- | --- |
| Temperatures | K-type thermocouples DS18B20 90807A | ± 0.2K |
| 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 wattmeter | ± 0.15% |

**Table 2** Specifications of the main components of the heat pump system

| Name | Remarks |
| --- | --- |
| Compressor | Reciprocating compressor, R134a, piston displacement: 32.7 cm$^3$ |
| Evaporation | Shell and tube heat exchanger Shell inner diameter: 32 mm Number of tubes: 5 Tube inner diameter: 6 mm Length of evaporator: 3 m |
| Condensation | Shell and tube heat exchanger Shell inner diameter: 32 mm Tube inner diameter: 6 mm Number of tubes: 5 Length of condenser: 3 m |
| Expansion valve | Internally equalized type |
| Internal heat exchanger | Plate heat exchanger, SWEP no 14314–010 |



**Fig. 4** Temperature prediction of the two-phase refrigerant in the condensator



**Fig. 3** Temperature prediction of the two-phase refrigerant in the evaporator

$$\text{Error} = \frac{\left| V_{\text{pred}} - V_{\text{exp}} \right|}{V_{\text{pred}}} \tag{26}$$

Figures 3–8 present the comparison of the differences between the thermodynamic parameters obtained from the measurement series and the model .
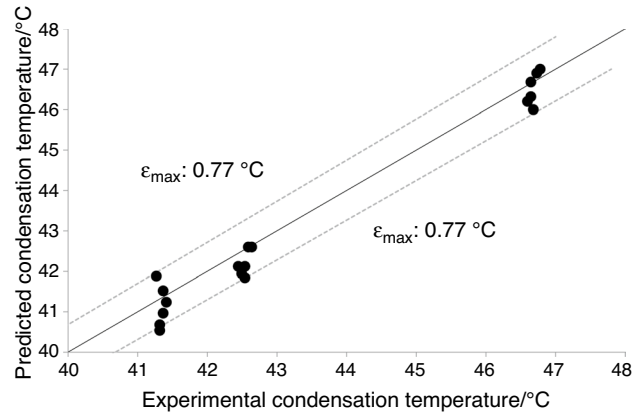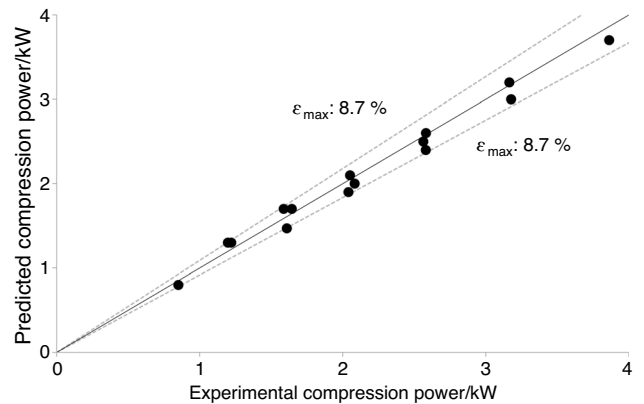


**Fig. 5** Prediction of the compressor power consumption

The figures show the maximum deviation, relative error of the direct (temperature, power), and indirect (COP, heat rate) measured variables.

Figure 3 presents the differences in the temperature values of the two-phase refrigerant flowing in the evaporator between the model and the measured results. The maximum deviation is 0.61 °C. The comparison of the condensation temperature values obtained from the measurement and the model is demonstrated in Fig. 4. It can be seen that
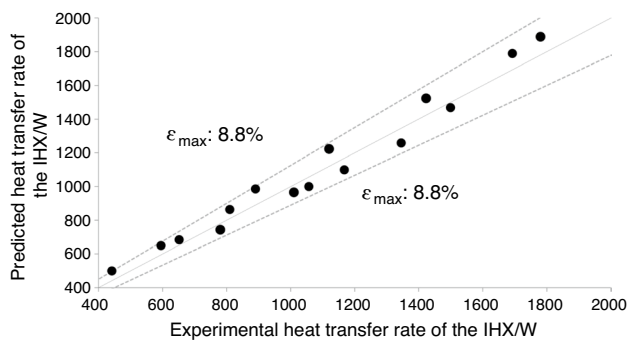
Fig. 6 Prediction of the heat transfer in IHX



Fig. 7 Prediction of the condenser capacity

the largest temperature difference between the model and the measured values is 0.77 °C.

Figure 5 shows the compressor power consumption values. The model values describe the values obtained from the measurements with a relative error of 8.7%. The compressor efficiency is $\eta_{comp} = 0.7$ in the mathematical model.

In Fig. 6, the heat capacity values in the internal heat exchanger between liquid phase and vapor phase refrigerant are summarized. The mathematical model has a relative error of 8.8% compared to the measured values.

Figure 7 outlines the deviation of the heat capacity values of the condensator obtained from the mathematical model from the values obtained from the measurement series. The relative deviation is 6.1%.

Figure 8 presents the difference between the COP values obtained from the heat pump experiment and from the mathematical model. The mathematical model describes the physical system with an accuracy of 7.06%.

## Results and discussion

This section discusses the main objectives of this research, namely the role of IHX efficiency on the COP. The simulation of the mathematical model, the COP value was analyzed as a function of the evaporation and condensation temperatures, and the efficiency of the internal heat exchangers.

The predicted ranges of the investigation are within the following values:

- internal heat exchanger efficiency $\varepsilon = 0.65, 0.75, 0.85, 0.95$,
- evaporation temperature $T_o = -20, -10, 0, 10°C$,
- condensation temperature $T_c = 40, 50, 60, 70, 80, 90°C$,
- cooling capacity $Q_o = 5$ kW,
- compressor efficiency $\eta_{comp} = 0.7$.

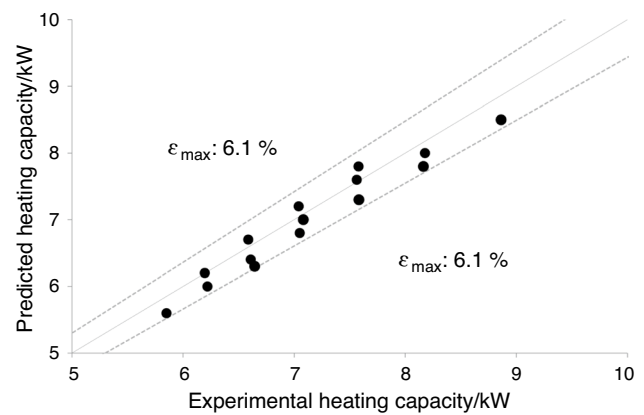In Fig. 9, the change of the COP value is presented as a function of the internal heat exchanger efficiency ranges

from $\varepsilon = 0.65$ to 0.95, and of the condensation temperature ranges from $T_c = 40°C$ to $90°C$, while the evaporation temperature is kept constant at $T_0 = 10$ °C.

As can be seen, at the condensation temperature $T_c = 40°C$, the change of efficiency of the heat exchanger has a very small effect on the COP value, under only 0.43%, from $\varepsilon = 0.65$ to 0.95 examined conditions. However, at higher condensation temperature $T_c = 90°C$ and at higher heat exchanger efficiency $\varepsilon = 0.95$, the COP value will be 4.51% higher than at $\varepsilon = 0.65$.

Figure 10 presents the change of the COP value as a function of the condensation temperatures ranges from $T_c = 40°C$ to $90°C$ and the efficiency of the internal heat exchanger ranges from $\varepsilon = 0.65$ to 0.95, and the evaporation temperature at $T_0 = 0$ °C.

The change of the efficiency value from $\varepsilon = 0.65$ to 0.95 has a very small effect on the COP value at the condensation temperature $T_c = 40$ °C, this value is only 0.66%. However, at higher condensation temperature $T_c = 90$ °C and at higher heat exchange efficiency $\varepsilon = 0.95$, the COP value is 5.1% higher than at $\varepsilon = 0.65$.
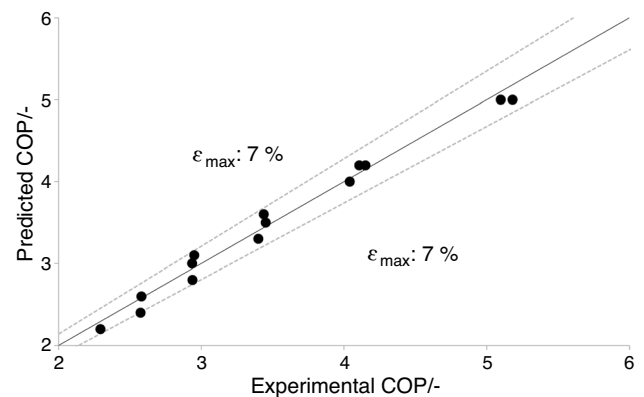


Fig. 8 Prediction of the coefficient of performance COP

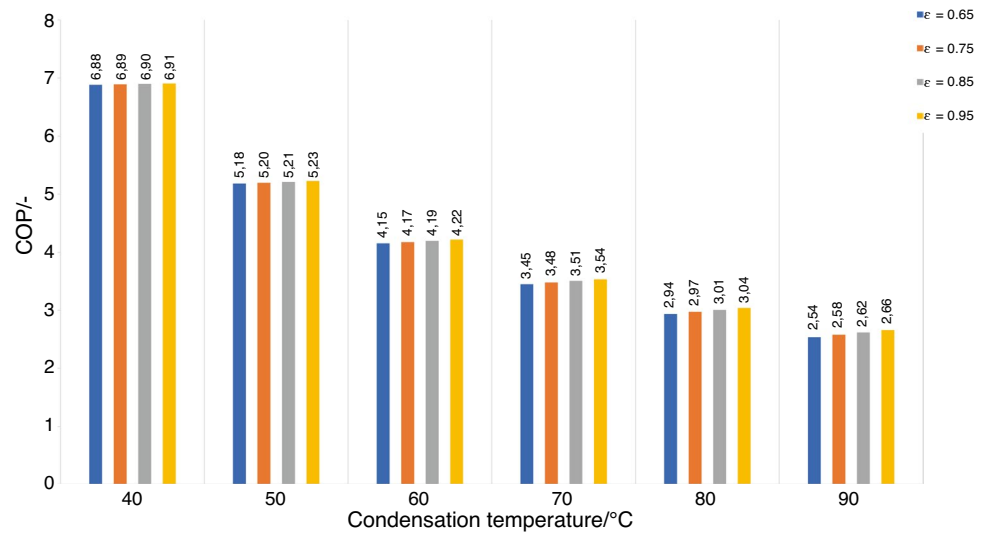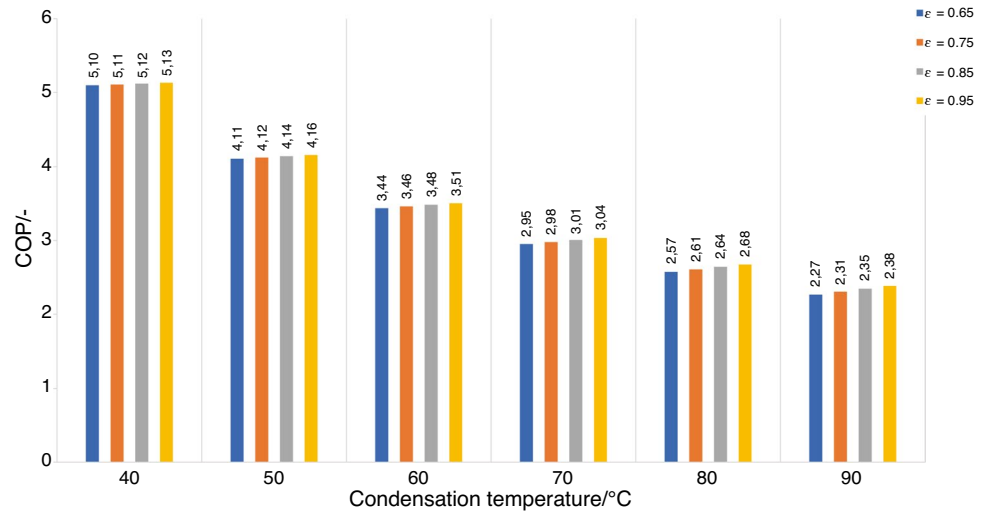**Fig. 9** COP as a function of the condensation temperature and of the efficiency at $T_0=10$ °C



**Fig. 10** COP as a function of the condensation temperature and of the efficiency at $T_0=10$ °C



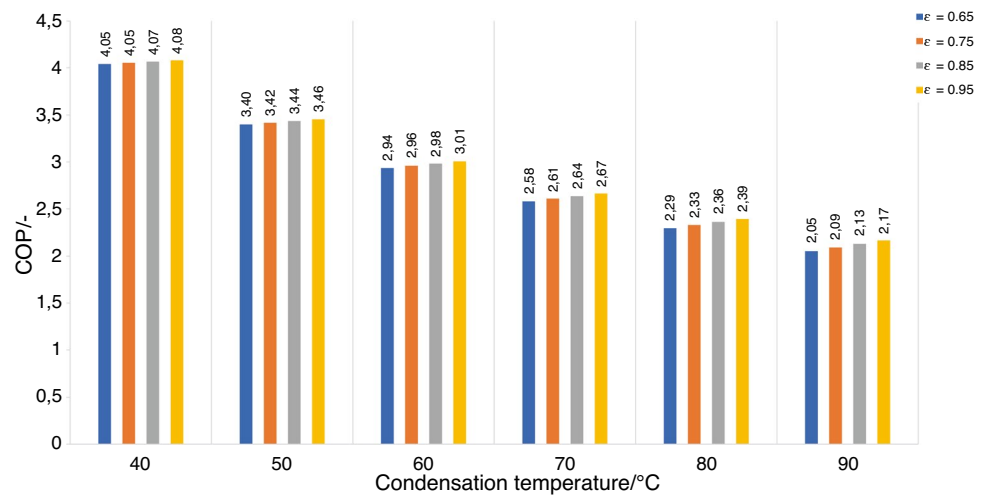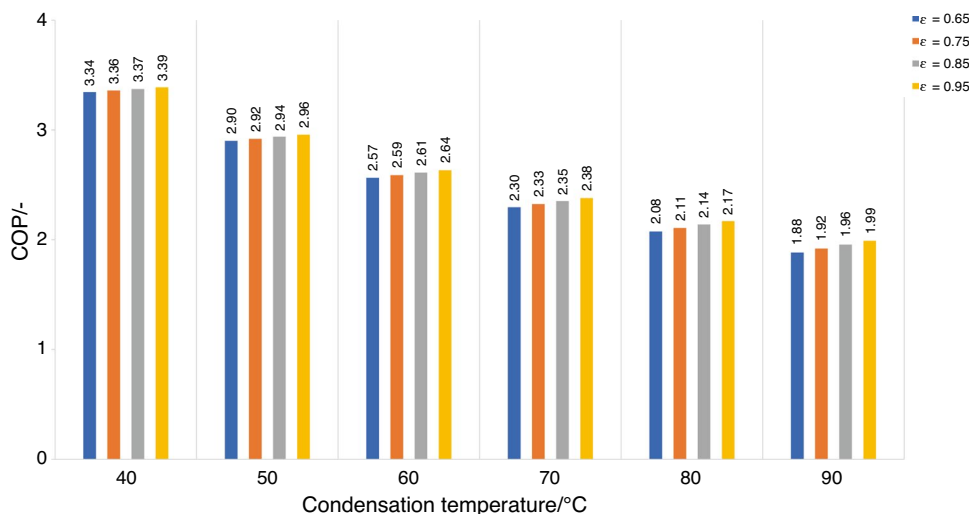**Fig. 11** COP as a function of the condensation temperature and of the efficiency at $T_0=10$ °C

**Fig. 12** COP as a function of the condensation temperature and of the efficiency at $T_0 = -20\,°C$



As outlined in Fig. 11, the efficiency of the heat exchanger from $\varepsilon = 0.65$ to 0.95 has a very small effect on the COP value, only 0.99% at the condensation temperature $T_0 = 40\,°C$, and at the evaporating temperature $T_0 = -10\,°C$. At higher condensation temperature $T_c = 90\,°C$ and at higher efficiency $\varepsilon = 0.95$, the COP value is 5.85% higher than at $\varepsilon = 0.65$.

Figure 12 shows the efficiency of the heat exchanger from $\varepsilon = 0.65$ to 0.95 has a very small effect on the COP value at the condensation temperature $T_c = 40\,°C$, and at the evaporating temperature $T_0 = -20\,°C$, this value is only 1.49%. Yet at higher condensation temperature $T_c = 90\,°C$, and at higher heat exchange efficiency $\varepsilon = 0.95$, the COP value is 5.95% higher than at $\varepsilon = 0.65$.

Another aim of this research is to develop a more accurate and generally applicable calculation formula for calculating the COP than the existing correlations in the literature for R134a refrigerant, which takes into account the efficiency of the internal heat exchanger.

The new formula for calculating the COP value of a heat pump with IHX:

$$COP = a_0 + a_1 \cdot T_0 + a_2 \cdot \varepsilon - a_3 \cdot T_c \qquad (27)$$

The expected values of the constants of the multivariate linear first-order model, given the correlation coefficient, are: $R = 0.8576$.

$$a_0 = 6,6542$$

$$a_1 = 0,0555$$

$$a_2 = 0,25225$$

$$a_3 = -0,0507$$

The presented Eq. (27) was defined in the following conditions and criteria:

- Refrigerant: R134a.
- Compressor power consumption: $P = 0.85\,kW \div 3.8kW$.
- Efficiency of the heat exchanger: $\varepsilon = 0.65, 0.75, 0.85, 0.95$
- Evaporation temperature: $T_o = -20\,°C \div 10\,°C$.
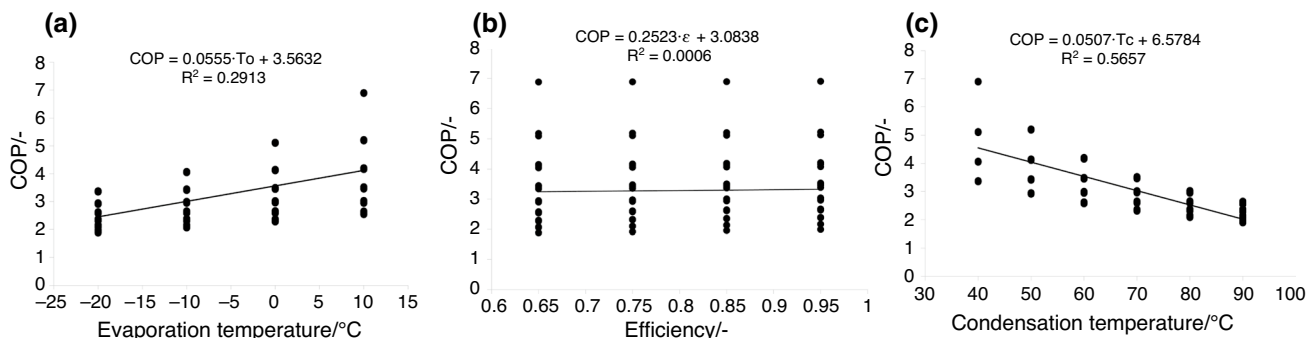- Condensation temperature: $T_c = 40\,°C \div 90\,°C$



**Fig. 13** Linear correlation coefficients, COP as a function of **a** evaporation temperature, **b** efficiency, **c** condensation temperature

## Conclusions

This paper has described a lumped parameter steady-state model for a vapor compression heat pumps with internal heat exchanger using R134a as working fluid. The experimental tests for the model validation were performed in a wide range of operating conditions, which allowed the author to check the robustness of the model. The mathematical model makes it possible to estimate the COP of the heat pump with internal heat exchangers with a maximum relative error of 7.06%. Figures 3–8 summarize the comparison of the results of the mathematical model with the measured values.

The investigation of the COP value was performed as a function of the evaporation temperature $To = -20, -10, 0, 10 \,°C$, of the condensation temperature $Tc = 40, 50, 60, 70, 80, 90 \,°C$ and of the efficiency of the internal heat exchanger $\varepsilon = 0.65, 0.75, 0.85, 0.95$.

Based on the simulation results it can be concluded that the efficiency of the internal heat exchanger is almost negligible for the change of the COP value at low condensing temperatures and high evaporation temperatures. By raising the efficiency of the internal heat exchanger from $\varepsilon = 0.65$ to 0.95, the COP value increased by only 0.66% at the evaporation $T_0 = 10 \,°C$ and the condensation $T_c = 40 \,°C$ temperatures, as shown in Fig. 9.

The actual significance of the efficiency of the internal heat exchanger is manifested at high condensation and low evaporation temperatures. The change of the COP value was most pronouncedly demonstrated in Fig. 12, when the evaporation temperature was low, in the present studies this value was $T_0 = -20°C$, while the condensation temperature was high $T_c = 90 \,°C$. In this case, the change of the COP value was 5.95% higher when the efficiency value was increased from $\varepsilon = 0.65$ to 0.95 (Fig. 13)

The last part of the research introduced a new equation Eq. (27) to calculate COP value of the heat pump with internal heat exchanger. This new equation for calculating COP value takes into account the efficiency of the internal heat exchanger in addition to the evaporation and condensation temperatures known so far.

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