Advances in Science and Technology Research Journal

Advances in Science and Technology Research Journal 2023, 17(5), 12–27
https://doi.org/10.12913/22998624/170971
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Received: 2023.07.05
Accepted: 2023.09.11
Published: 2023.10.20

Compressor Heat Pump Model Based on Refrigerant Enthalpy and Flow Rate

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ABSTRACT

This paper introduces a compressor heat pump model using flow rate and enthalpy values for the calculation of heat pump control parameters. The aim of in this article presented research work was to develop an universal model of a compressor heat pump which will enable simulation tests for compressors of any volumetric efficiency with variable operating parameters (e.g. temperature of the lower source and condensation temperature) worked with any refrigerant. An additional assumption was the possibility of conducting simulation tests continuously. The model was developed on the basis of equations describing the thermodynamic transformations taking place in the refrigeration system. The data needed for the model are the input signals of saturation and condensation temperatures and the refrigerant mass flow rate entered as a time series. The output signals are heating capacity, cooling capacity, power consumed by the compressor, and the temperature at the second point of the thermodynamic cycle of the refrigerant. The paper presents equations for examples of the frequently used refrigerants R410a and R290 and practical verification of the presented algorithm in the laboratory made for refrigerant R410a. A description of the laboratory stand is also included. Comparison of the capacity provided by the condenser and the coefficient of performance with the values simulated using the proposed model confirms the correctness of the applied model and its practicality. The results obtained by simulation and measurements show very good convergence. The developed model makes it possible to calculate the operational parameters of the device, e.g. such as COP and SCOP for given boundary conditions, which then enables to estimate the coverage degree of the heat load of the building by the heat pump due to central heating and to estimate momentary and seasonal operating costs of the heat pump.

Keywords: heat pump, R410A refrigerant, R290 refrigerant, COP (coefficient of performance).

INTRODUCTION

International agreements on the reduction of greenhouse gases ratified by the EU (Montreal Protocol, Kyoto Protocol) introduced a ban on the use of refrigerants belonging to the chloro-fluoro-carbons (CFCs) and hydro-fluoro-carbons (HFCs) groups in refrigeration systems, including heat pumps, and significant restrictions on the use of refrigerants belonging to the hydro-chloro-fluoro-carbons (HCFCs) group. In addition, the EU has introduced a timetable for reducing the use of agents belonging to this group, and so from 1 January 2030 there is to be a complete phase-out of agents belonging to this group. Therefore, there is a need to develop new refrigerants that could act as substitutes for the popular ones currently used, such as R410A and R134a. Preference is given to natural refrigerants belonging to the hydro-carbon (HC) group, which are characterised by low ozone deep potential (ODP) and global warming potential (GWP) values. Such agents include R290 propane, R1270 propylene, and R600 isobutane. The use of environmentally friendly agents belonging to the combustible A3 group requires the development of new design solutions for refrigeration systems. To achieve this, it is necessary to carry out numerous simulation studies not only of the individual components of the refrigeration system, such as the condenser or evaporator, but also of the entire thermodynamic system of the heat pump, taking into account changes in the heat load of both the condenser and the evaporator. Furthermore, during the design phase, it is very important to determine the seasonal performance of the heat pump installed in a specific building. Therefore, it is important to carry out simulation studies of the heat pump in real time with the possibility of analysing the performance of the system depending on the refrigerant used at a given operating point. The developed compressor heat pump model, which is the subject of this article, enables such simulation studies to be carried out.

RESEARCH METHODS

Most commonly used compressor heat pump models

The simulation studies described in the literature mainly concern the modelling of the individual components of the heat pump: compressor [1], expansion valve [2, 3], condenser [4], and the entire unit, to determine the temperatures of the refrigerant at various points of the thermodynamic cycle or the COP (coefficient of performance) [5] of the heat pump in terms of the use of different refrigerants, such as R134a, R404a, R407c or R410a, at the selected operating point. The models are mainly implemented in the Matlab&Simulink, TRNSYS or CYCLE_D packages [6-8] and are often not verified under operating conditions. The heat pump model presented in [9] was based on mathematical equations describing the adiabatic process, resulting in the calculation of the refrigerant temperature at the condenser supply. The model was not verified, and the analyses presented were purely theoretical. A compressor heat pump model allowing more accurate thermodynamic calculations was presented in [10]. It was based on an equation relating the specific enthalpy of points 1–2 of the heat pump thermodynamic cycle (Figure 1) to the volumetric flow rate of the fluid. The model allowed the estimation of electricity consumption to drive the compressor with assumed electrical and mechanical efficiencies. Uhlmann and Bertsch [11] developed a model for a compressor heat pump based on the analysis of heat fluxes on the low- and high-pressure side, taking into account the mechanical power and electrical energy required to drive the compressor. The model developed was valid for a certain range of input parameters—evaporating temperature and pressure—and was based on the model described by Klein, for which the EES (Engineering Equation Solver) software was used to solve. The advantage of the model presented was that simulation studies could be carried out for different refrigerants [11]. A similar approach to modelling the operation of a compressor heat pump was presented in [12]. The model developed allowed basic thermodynamic calculations of the refrigeration system and its performance to be made on the basis of input data, which were taken to be the efficiencies (of compression and expansion) and the volumetric efficiency of the compressor. A different approach to modelling the operation of a compressor heat pump is presented in [13]. The developed model for a compressor heat pump was based on a polynomial function and allowed the calculation of the heating capacity of the heat pump and the COP value for a set value of evaporation and condensation temperatures of the refrigerant. The paper only presented the results of simulation studies, and the developed model was not verified. A much more extensive model is presented in [14]. To simulate the operation of the heat pump, a complex mathematical apparatus, which, in addition to basic thermodynamic calculations, was able to perform hydraulic calculations and calculations of the electricity consumed by the fans, was used. Equally complex, difficult-to-solve compressor heat pump models were presented in [15, 16]. An advantage of the model presented in [16] was that it was possible to analyse the heat pump's interaction with a buffer tank representing the condenser load. A holistic approach to simulating the operation of a compressor heat pump was presented in [17]. The developed model of the heat pump took into account its basic components: compressor, condenser, evaporator, and expansion valve, and made it possible to analyse its operation parameters: heating capacity, cooling capacity, and power consumed by the compressor, depending on the operating point defined by the value of evaporation, condensation temperature of the refrigerant, and volumetric efficiency of the compressor. An advantage of the developed model was the possibility of analysing the cooperation of the heat pump with the central heating system, while a disadvantage was the complicated mathematical apparatus of the model, requiring the use of significant computer computing power to solve it.

The presented analysis shows that the models of the compressor heat pump described in the literature have numerous assumptions, simplifications, or are very complicated, which makes their implementation difficult and significantly prolongs the period of simulation studies. In addition, most of the models used in simulation studies have not been verified.

Heat pump model base on flow rate

The developed compressor heat pump model is based on the thermodynamic cycle of the refrigeration system inscribed in the pressure–enthalpy diagram of any refrigerant, which is its advantage. An advantage of the developed model is its utilitarian nature. The developed model can be used to simulate the operation of a compressor heat pump fed with any refrigerant. In this paper, all analyses are presented for a compressor heat pump supplied with the R410a thermodynamic refrigerant; additionally, the created equations are elaborated for refrigerant R290. The thermodynamic properties of R410a are shown in the pressure–enthalpy diagram (Figure 1). The input parameters of the heat pump model are the refrigerant saturation temperature, the refrigerant condensation temperature, and the refrigerant mass flow rate (Figure 1). The output parameters of the heat pump model are cooling capacity, heating capacity, and power consumed by the compressor, which are determined from the relations (1–3). In addition, the output parameter of the heat pump model is the refrigerant temperature at the second point of the thermodynamic cycle (Figure 1).

In order to determine the heating and cooling capacity and power consumed by the compressor from relations 1–3 for a known mass flow rate, refrigerant saturation, and condensation temperature, it is necessary to know the enthalpies h1-h4 of the individual points in the thermodynamic cycle. The enthalpy values of the individual points of the thermodynamic cycle can be determined under the simplifying assumption that subcooling and superheating of the refrigerant are neglected in the thermodynamic cycle. Therefore, points 1 and 3 of the thermodynamic cycle are assumed to be on the humidity saturation line.

$$Q_c = (h_1 - h_4) \cdot \dot{m} \tag{1}$$

$$Q_g = (h_2 - h_3) \cdot \dot{m} \tag{2}$$

$$Q_s = (h_2 - h_1) \cdot \dot{m} \tag{3}$$

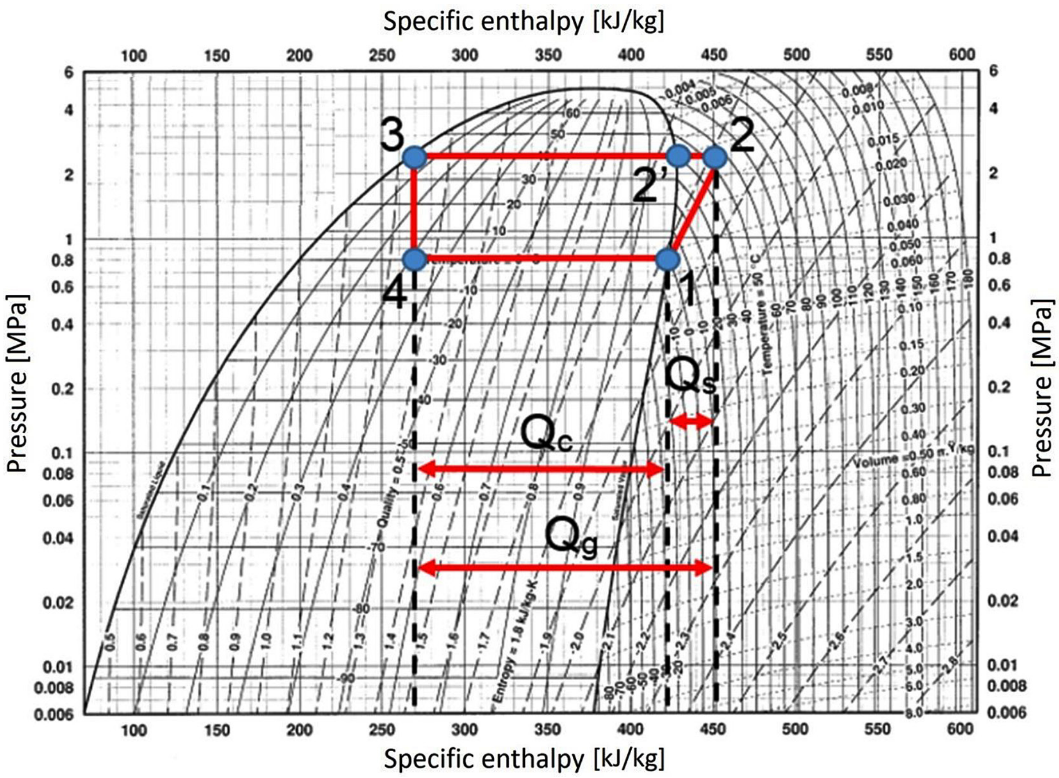

Fig. 1. Thermodynamic cycle of a compressor heat pump shown in the pressure–enthalpy diagram for refrigerant R410a [own elaboration]

$$COP = \frac{h_2 - h_3}{h_2 - h_1} = \frac{Q_g}{Q_s} \tag{4}$$

Under operating conditions, the operating parameters of a heat pump can change dynamically. For this reason, a compressor heat pump model based on a thermodynamic cycle should enable continuous simulations of the unit. Performing continuous simulations in time requires continuous calculation of the values of specific enthalpy, specific entropy, pressure, and volume of the working medium at individual points of the cycle as a function of the temperature of the working medium. For this purpose, the p-h diagram of the working medium should be divided into two parts with respect to the critical point k, and the variations of pressure, enthalpy, and entropy as a function of temperature should be mapped by polynomial functions. The mathematical polynomial description of the dependencies of the individual refrigerant physical quantities readable from the p-h diagram and the variables as a function of temperature in the humidity range x = 0 – point k will conventionally be indexed as "fluid", while the dependencies of the individual refrigerant physical quantities readable from the p-h diagram and the variables as a function of temperature in the range point k - x = 1 will conventionally be indexed as "steam".

The tested model is based on equations (1–4), which use flow rate and enthalpy. Using these equations and thermodynamic cycle of a compressor heat pump (Figure 1), the enthalpy at point h2 remains to be determined. To solve this problem, the algorithm described by equations (19–23) is used. This algorithm requires knowledge of the low and high source temperatures, together with technical parameters of the refrigerant. In this paper, the presented model was practically verified using refrigerant R410a, whose parameters are described by equations (5–11); additionally, the same parameters were calculated using the same procedure for refrigerant R290 by equations (12–18). In both cases, the polynomials, whose coefficients were calculated for the model, were based on tabulated data resulting from extensive measurement studies, while the equations for calculating the specific heat of the refrigerant at constant pressure and volume were based on the Martin–Hou equation of state derived from technical materials [18]. Parameters used in polynomials are dependent on the refrigerant (in the case of this study, refrigerants R410a and R290).

Calculation for refrigerant R410a (values used for practical verification)

The steam pressure of the working medium as a function of temperature (Figure 2) can be described by relation 5 with a coefficient of determination of 99%.

$$p_{steam} = 0.00210 \cdot T^3 + 0.2874 \cdot T^2 + 25.46 \cdot T + 797.5 \tag{5}$$

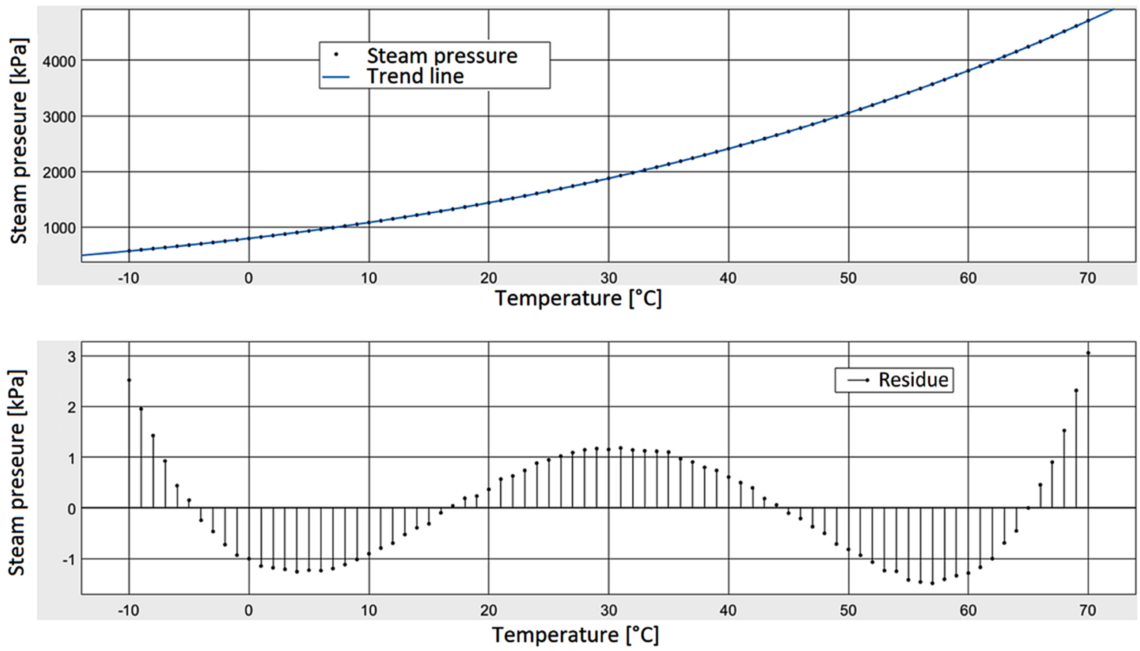

Fig. 2. Steam pressure of the medium as a function of temperature (for refrigerant R410a)

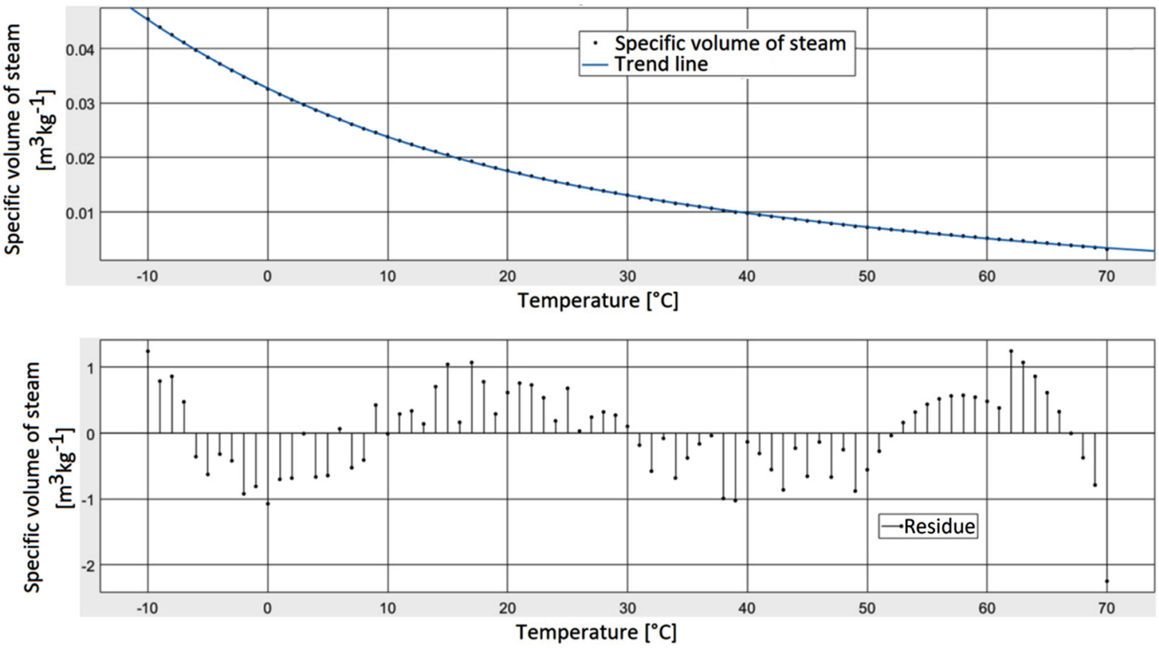

Fig. 3. Steam volume of the medium as a function of temperature (for refrigerant R410a)

The variation of the steam volume of the working medium as a function of temperature can be described by relation 6 with a coefficient of determination of 99% (Figure 3).

$$V_{steam} = 7.845 \cdot 10^{-10} \cdot T^4 - 1.854 \cdot 10^{-7} \cdot T^3 + \\ +1.823 \cdot 10^{-5} \cdot T^2 - 0.001055 \cdot T + 0.03271 \quad (6)$$

The variation of the working fluid pressure as a function of temperature is shown in Figure 4 and can be described by relation 7 with a coefficient of determination of 99%.

$$p_{fluid} = 0.002004 \cdot T^3 + 0.2952 \cdot T^2 + \\ +25.48 \cdot T + 799.6 \quad (7)$$

The changes in the specific enthalpy of the working medium in the fluid (Figure 5) and gaseous (Figure 6) phases as a function of temperature can be described using relations 8 and 9, respectively, while the changes in the specific entropy of the working medium in the fluid (Figure 7) and gaseous phases can be described using relations 10 and 11, respectively.

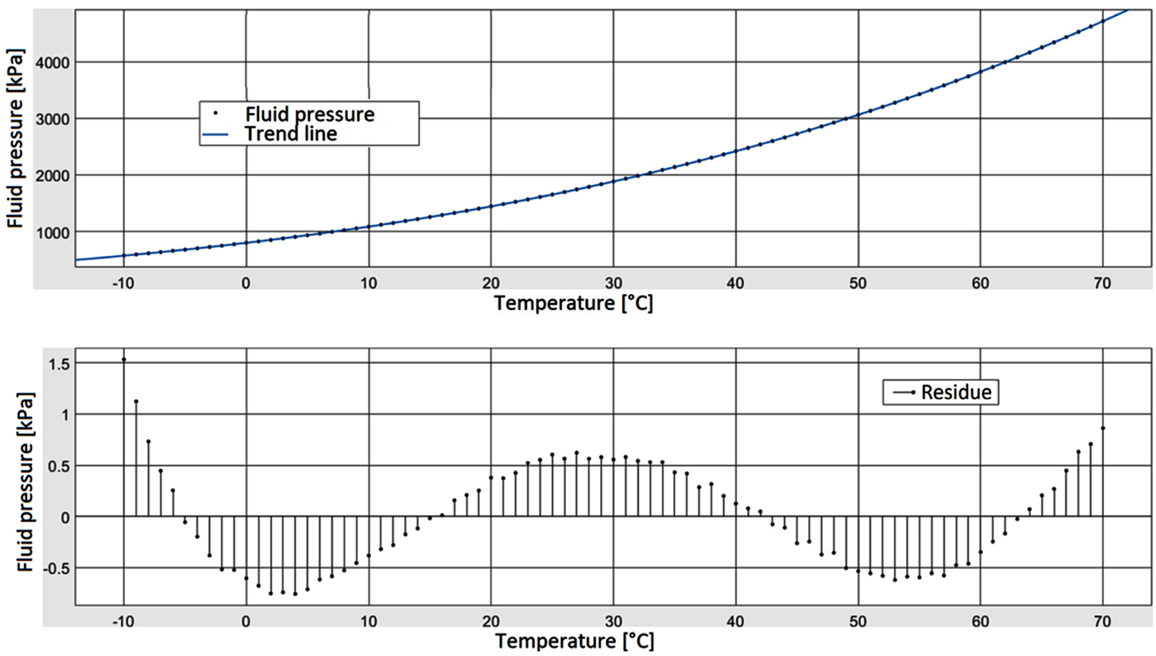

Fig. 4. Fluid pressure of the medium as a function of temperature (for refrigerant R410a)

$$h_{fluid} = 1.149 \cdot 10^{-7} \cdot T^5 - 1.279 \cdot 10^{-5} \cdot T^4 + \\ +0.0004736 \cdot T^3 - 0.001878 \cdot T^2 + \\ +1.517 \cdot T + 200.2 \tag{8}$$

$$h_{steam} = -1.49 \cdot 10^{-11} \cdot T^7 + 2.331 \cdot 10^{-9} \cdot T^6 - \\ -1.307 \cdot 10^{-7} \cdot T^5 + 2.507 \cdot 10^{-6} \cdot T^4 - \\ -2.665 \cdot 10^{-5} \cdot T^3 - 0.003454 \cdot T^2 + \\ +0.2874 \cdot T + 422.6 \tag{9}$$

$$s_{fluid} = 3.268 \cdot 10^{-10} \cdot T^5 - 3.639 \cdot 10^{-8} \cdot T^4 + \\ +1.356 \cdot 10^{-6} \cdot T^3 - 1.533 \cdot 10^{-5} \cdot T^2 + \\ +0.005539 \cdot T + 1.001 \tag{10}$$

$$s_{steam} = 0.9982 \cdot p_{fluid} - 2.142 \tag{11}$$

It should be noted that in order to calculate the power available at the condenser, it is necessary to know the value of the specific enthalpy of

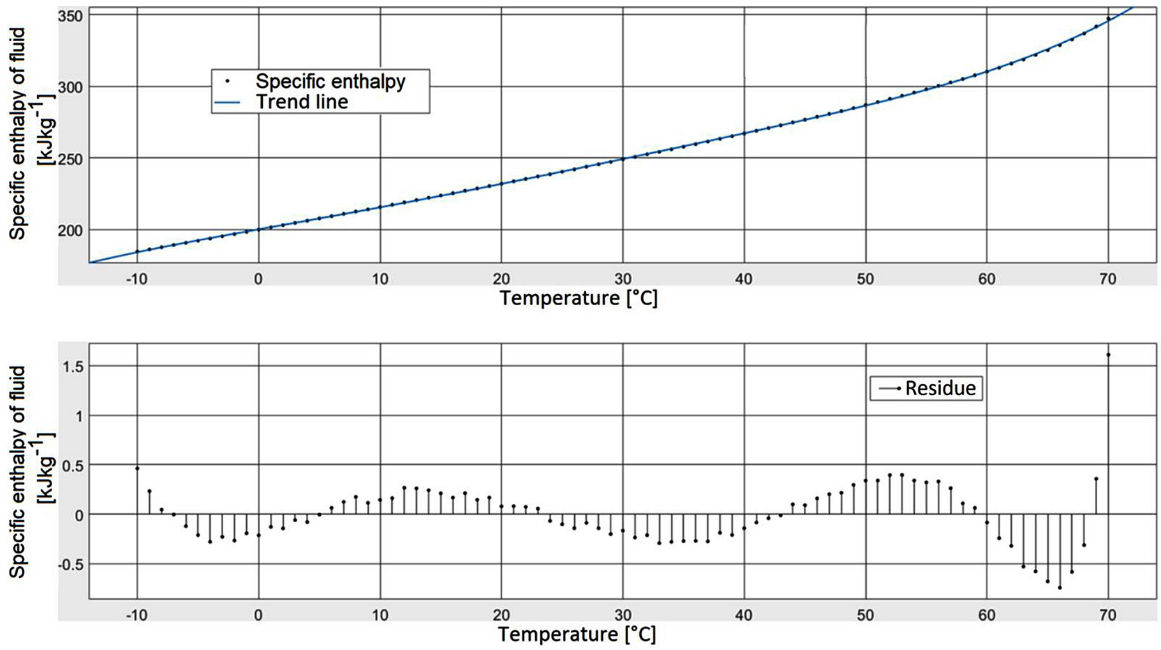

Fig. 5. Variation of the specific enthalpy of a working medium in the fluid phase as a function of temperature (for refrigerant R410a)

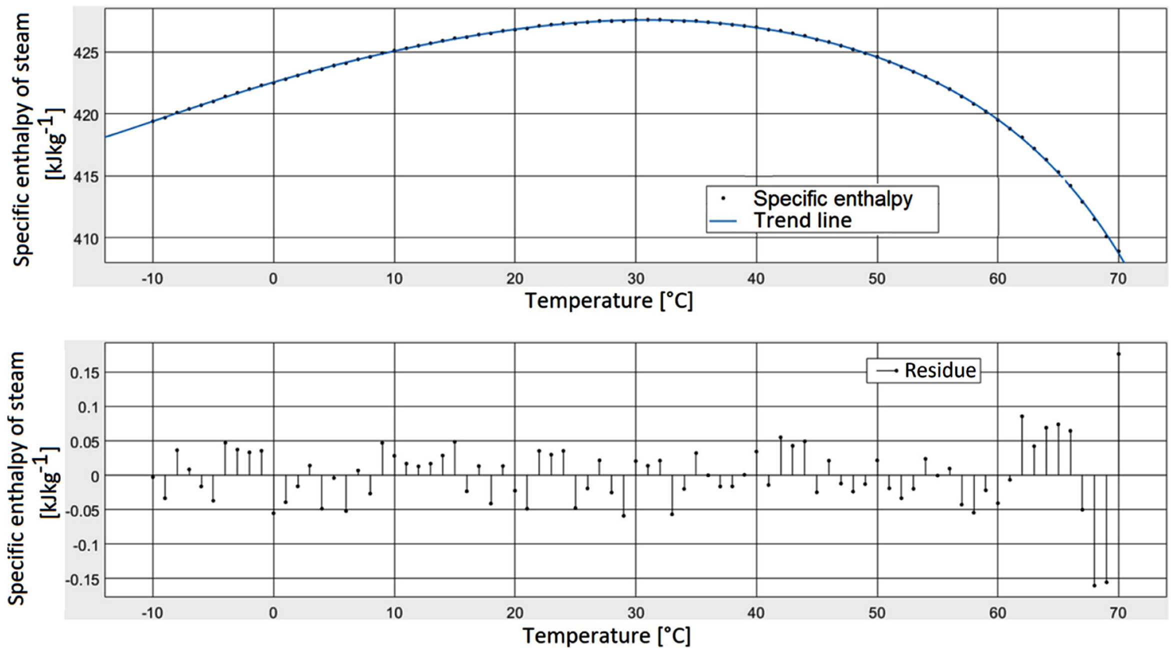

Fig. 6. Variation of the specific enthalpy of a working medium in the gas phase as a function of temperature (for refrigerant R410a)

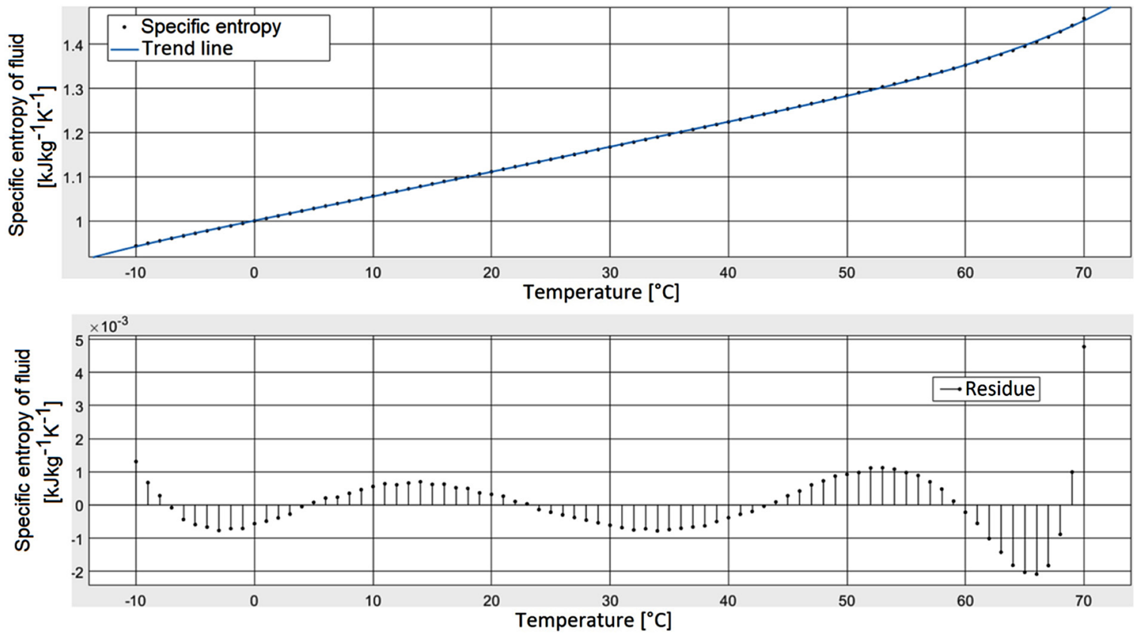

Fig. 7. Variation of the specific entropy of a thermodynamic agent in the fluid phase as a function of temperature (for refrigerant R410a)

point 2 of the thermodynamic cycle of the compressor heat pump, which is in the superheated steam region. Assuming that the compression of the medium is isentropic, then the specific entropy of point 1 and the specific entropy of point 2 are identical, s1 = s2.

Calculation for refrigerant R290 - additional calculation as example of refrigerant different from R410a

The steam pressure of the working medium as a function of temperature (Figure 8) can be described by relation 12 with a coefficient of determination of 99%.

$$p_{steam} = 0.000908 \cdot T^3 + \\ +0.168 \cdot T^2 + 141911 \cdot T + 469 \quad (12)$$

The variation of the steam and fluid volume of the working medium as a function of temperature can be described by relations 13 and 14 with a coefficient of determination of 99% (Figure 9 and 10).

$$V_{steam} = 7.38 \cdot 10^{-11} \cdot T^5 + 1.46 \cdot 10^{-8} \cdot T^4 - \\ -1.035 \cdot 10^{-6} \cdot T^3 + 0.000046 \cdot T^2 + \\ +0{,}00258 \cdot T + 0.00973 \quad (13)$$

$$V_{fluid} = 2.97 \cdot 10^{-13} \cdot T^7 - 2.79 \cdot 10^{-11} \cdot T^6 - \\ -1.29 \cdot 10^{-9} \cdot T^5 + 1.38 \cdot 10^{-7} \cdot T^4 + \\ +1.9 \cdot 10^{-6} \cdot T^3 - 0.0001 \cdot T^2 + 0.0035 \cdot T + 1.94 \quad (14)$$

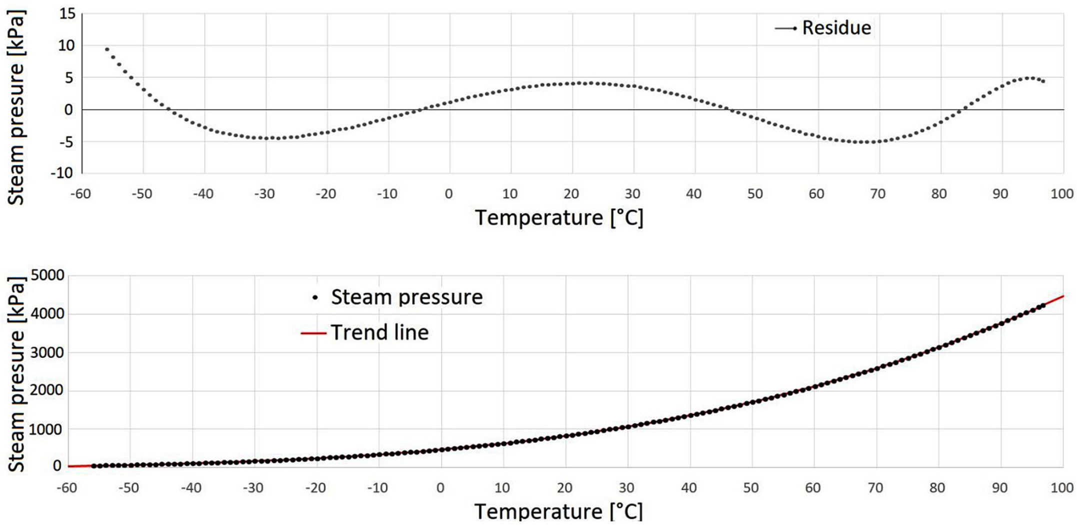

Fig. 8. Steam pressure of the medium as a function of temperature (for refrigerant R290)

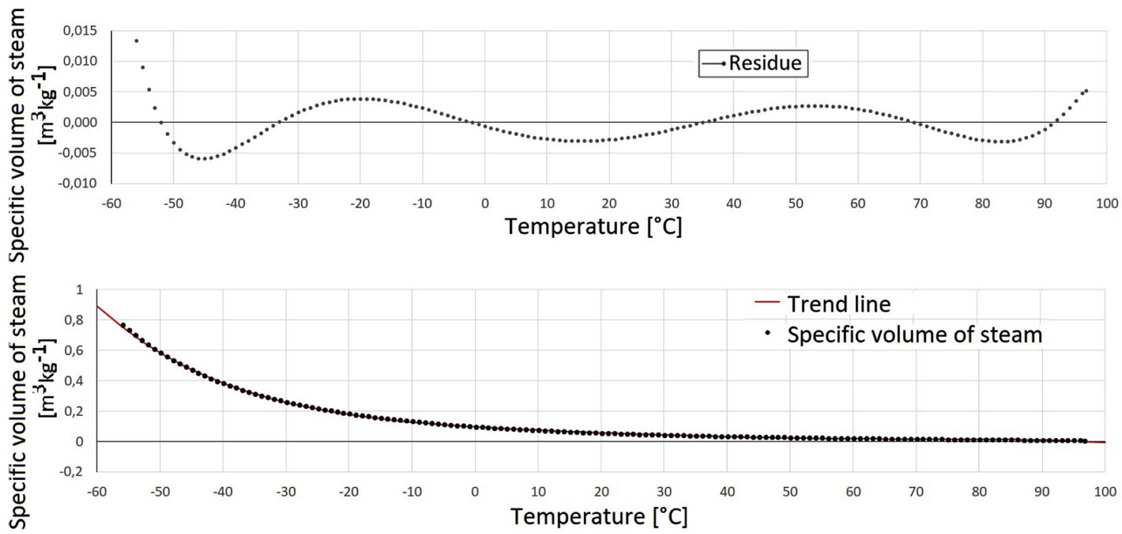

Fig. 9. Steam volume of the medium as a function of temperature (for refrigerant R290)

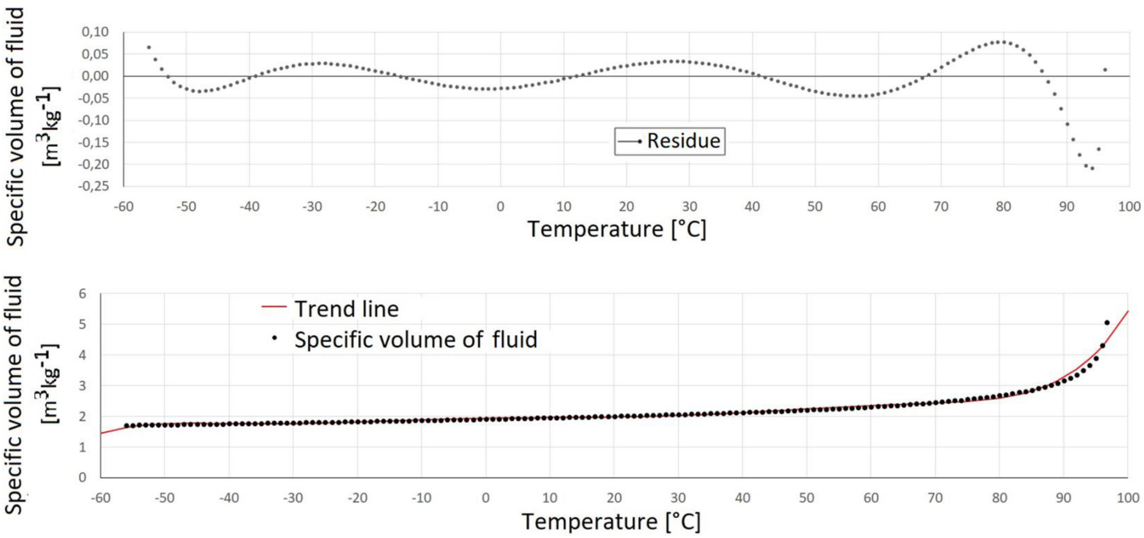

Fig. 10. Fluid volume of the medium as a function of temperature (for refrigerant R290)

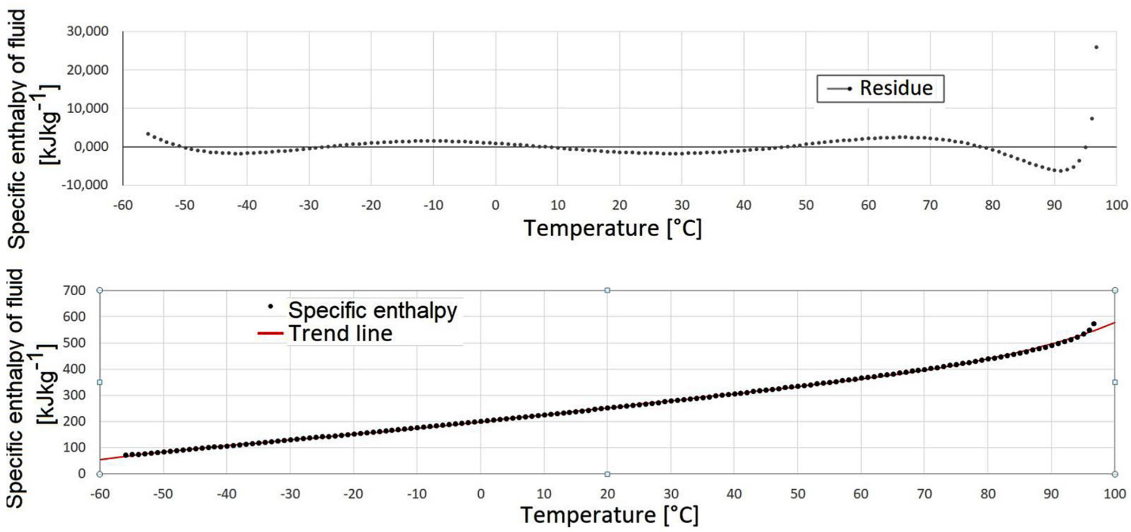

Fig. 11. Variation of the specific enthalpy of a working medium in the fluid phase as a function of temperature (for refrigerant R290)

The changes in the specific enthalpy of the working medium in the fluid (Figure 11) and gaseous (Figure 12) phases as a function of temperature can be described using relations 15 and 16, respectively, while the changes in the specific entropy of the working medium in the fluid (Figure 13) and gaseous phases (Figure 14) can be described using relations 17 and 18, respectively.

$$h_{fluid} = 2.6 \cdot 10^{-8} \cdot T^5 - 1.18 \cdot 10^{-6} \cdot T^4 - \\ -0.0001 \cdot T^3 - 0.0068 \cdot T^2 + 2.55 \cdot T + 199 \quad (15)$$

$$h_{steam} = -6.77 \cdot 10^{-12} \cdot T^7 + 6.2 \cdot 10^{-10} \cdot T^6 + \\ +2.88 \cdot 10^{-8} \cdot T^5 - 3.15 \cdot 10^{-6} \cdot T^4 - \\ -0.0001 \cdot T^3 + 0.0019 \cdot T^2 + 1.09 \cdot T + 572.8 \quad (16)$$

$$s_{fluid} = 2.73 \cdot 10^{-7} \cdot T^3 - \\ -8.59 \cdot 10^{-6} \cdot T^2 + 0.0084 \cdot T + 1 \quad (17)$$

$$s_{steam} = 2.62 \cdot 10^{-7} \cdot T^3 + \\ +1.53 \cdot 10^{-5} \cdot T^2 - 0.0008 \cdot T + 2 \quad (18)$$

Calculation of model value parameters

Knowing the saturation temperature of the medium, its condensation temperature and the value of the pressure of the working medium in the gaseous phase at point 1 (p1), its volume (*v1*) and the pressure of the working medium in the gaseous phase at point 2’ (p2’), and its volume at point 2’ (*v2*’), using the properties of the adiabatic process (in the

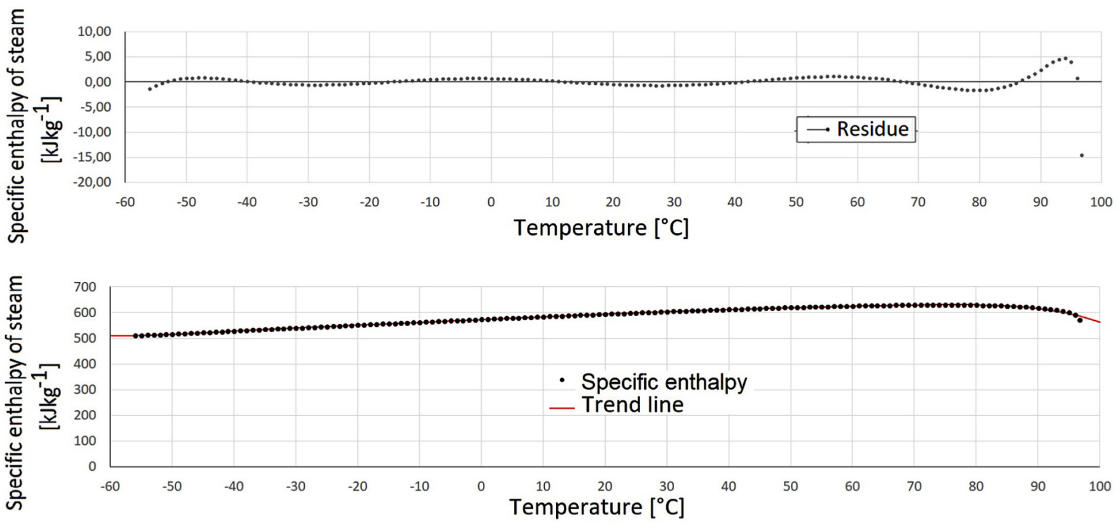

Fig. 12. Variation of the specific enthalpy of a working medium in the gas phase as a function of temperature (for refrigerant R290)

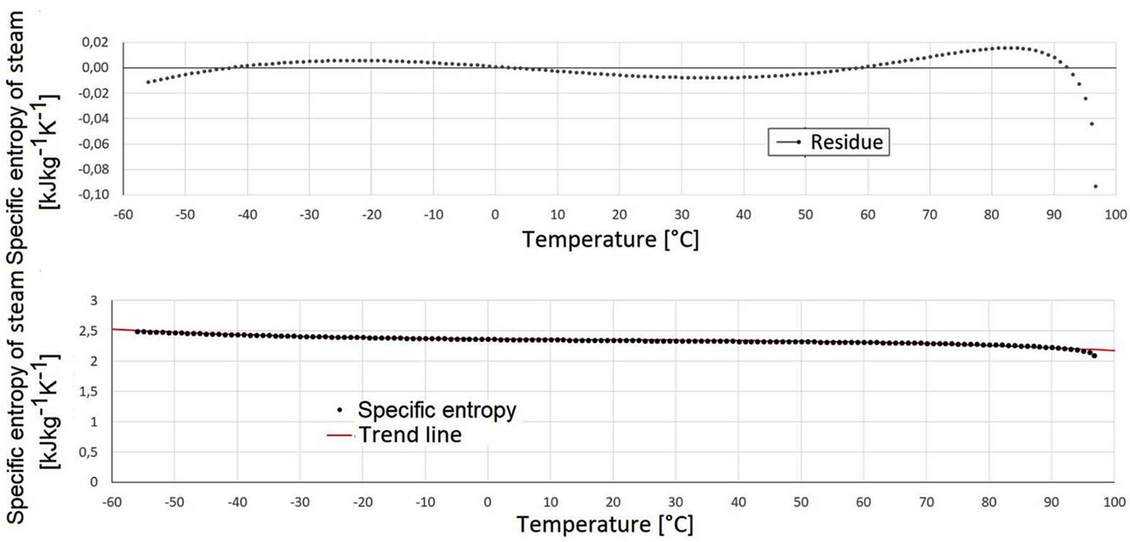

Fig. 13. Variation of the specific entropy of a thermodynamic agent in the fluid phase as a function of temperature (for refrigerant R290)

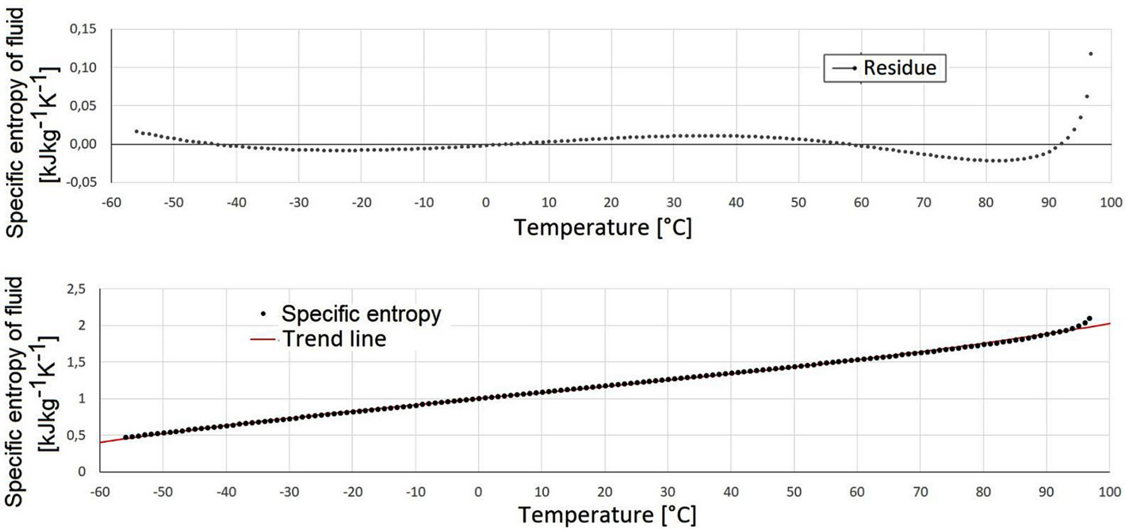

Fig. 14. Variation of the specific entropy of a thermodynamic agent in the gas phase as a function of temperature (for refrigerant R290)

superheated steam region p = const, v = const), it is possible to calculate the value of the temperature of the gas at point 2 ($T2$) using relation 19.

$$T_2 = T_1 \left(\frac{V_2'}{V_1}\right)^{\varkappa - 1} \tag{19}$$

The heat capacity ratio, κ, can be calculated from relation 20.

$$\varkappa = \frac{c_p}{c_v} \tag{20}$$

The specific heat capacity of the working medium at constant pressure for R410a can be calculated using relation 21, while the specific heat capacity of the medium at constant volume can be calculated using relation 22.

$$c_p = a + b \cdot T + c \cdot T^2 + d \cdot T^3 \left[\frac{kJ}{kgK}\right] \tag{21}$$

$$c_v = c_p - R \left[\frac{kJ}{kgK}\right] \tag{22}$$

The values of the coefficients for R410a in Equation 21 are: a = 2.676084e^{-1}, b = 2.115353e^{-3}, c = -9.848184e^{-7}, d = 6.493781e^{-11}.

The value of the R coefficient in Equation 22 for the refrigerant R410a is R = 0.114550 [kJ/(kg·K)]. Knowing the value of the temperature of point 2 ($T2$) of the heat pump thermodynamic cycle, using relation 23, it is possible to calculate the specific enthalpy of point 2 ($h2$) needed to calculate the capacity provided by the condenser Q_g, the power consumed by the compressor Q_s, and the COP value.

$$h_2 - h_3 = T_2 \cdot (s_1 - s_3) \tag{23}$$

Knowing the mass flow rate of the refrigerant and the enthalpy values h_1, h_2, h_3, and h_4 for the saturation temperature and the condensation of the refrigerant using relations 1–3, the heating capacity, the cooling capacity, and the power consumed by the compressor can be calculated. With the capacity provided by the condenser and the power consumed by the compressor, it is possible to calculate the COP value using relation 4, which is a parameter in the quality assessment of the heat pump determined by normative testing in accordance with EN 14511.

The model described by equations (1–4) and (19–23) is universal, while the knowledge of specific heat capacity c_p and c_v is individual and depends on the refrigerant properties, as presented above for two cases: R410a and R290. Experimental validation was carried out in a dedicated heat pump laboratory using a heat pump with the refrigerant R410a. The scheme and components of the installation are described in chapter 4. The correctness of the model parameter selection was confirmed by practical verification in the laboratory, presented in Fig. 22–24.

Compressor heat pump model verification

The developed compressor heat pump model was verified. Verification of the heat pump model was carried out using data recorded during the performance of service tests of an air-to-water heat pump with a compressor, for which the heat load was a hydraulic system.

Description of the test stand

The hydraulic design of the test stand (Figure 15) allows service tests to be carried out at various condenser heat loads and heat carrier water volumetric flow rates within the operating ranges of the circulation pumps.

An air-to-water heat pump consists of two units: an outdoor and an indoor unit. The outdoor unit with a nominal heating capacity of 7 kW, in which the main components are the compressor and evaporator, was placed on a freestanding structure. A water pump with infinitely variable mass flow rate adjustment in the range (8–26 kg/min) was placed in the indoor unit. The load for the condenser was a 200 dm^3 buffer tank connected parallelly to the hydraulic system. Two heating circuits were connected to the buffer tank: a high-temperature and a low-temperature circuit. Each of the heating circuits has a pump group with an infinitely variable adjustment pump capacity and a three-way mixing valve. In order to measure the heat stream received from the condenser, a capillary was installed on the supply and return of the heating circuits to enable the installation of a temperature sensor and a turbine flow meter measuring the mass flow rate of the medium in the range of 2–40 kg/min, which corresponds to the analogue 4–20 mA signal generated. Pt1000 resistance sensors made in class A in a four-wire system were used to measure the temperature in the individual nodes of the system.

The variable heat load of the condenser was applied by attaching a fan cooler to the buffer tank, with infinitely variable capacity adjustment possible by varying the speed of the fans, which was mounted outside the building (Figure 16). Between the fan cooler and the tank, a plate heat exchanger was installed, which acts as a separator between the water circuit on the primary side of the exchanger and the glycol circuit on the secondary side of the plate heat exchanger (Figure 17). The fan cooler circuit uses propylene glycol, which has a specific heat capacity of 3.580 J/(kg-K). Circulation pumps with infinitely variable volumetric flow rate adjustment were installed on both the primary and secondary sides of the exchanger. Infinitely variable adjustment of the volume flow rate on both the primary and secondary sides of the exchanger, as well as infinitely variable adjustment of the capacity of the fan cooler, enabled precise adjustment of the capacity of the heating system and, as a result, the possibility of simulating the heating circuits of the central heating system, which constitute the thermal load of the condenser. In order to regulate the supply temperature of the primary side of the exchanger, a three-way mixing valve with infinitely variable position adjustment was installed, which, as a result, reflects the supply temperature of the heating circuits between the buffer tank and the plate heat exchanger (Figure 17). The position of the mixing valve was controlled by an external voltage signal of 0-10 VDC. A control signal value of 0 VDC corresponds to complete closure of the valve. The control signal value equal to 10 VDC corresponds to complete opening of the mixing valve.

A freely programmable modular PLC was used to implement the adjustment algorithm for the heating system (Figure 18).

The central unit was expanded with a discrete input and output module, an analogue input

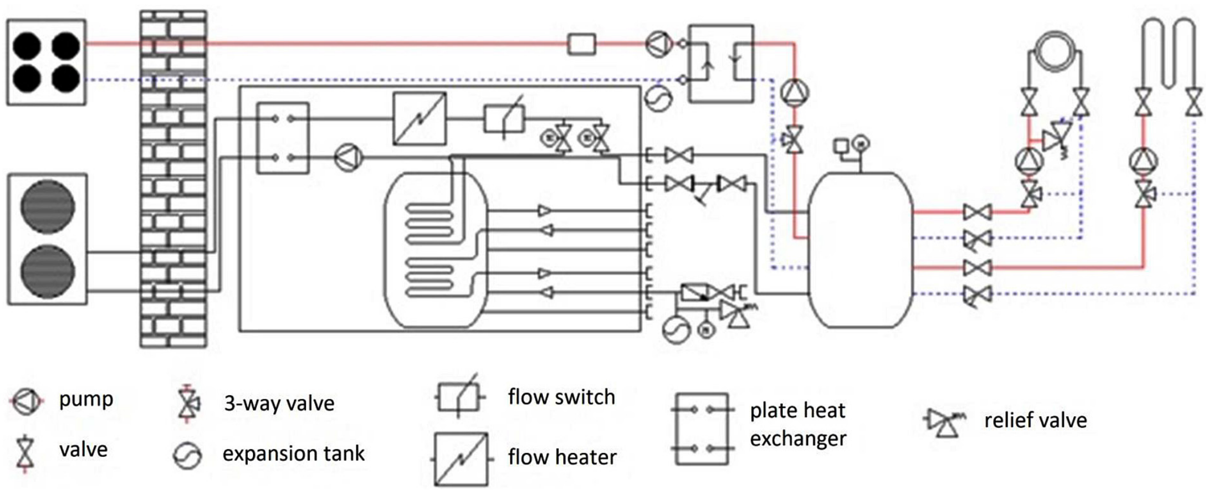

Fig. 15. Hydraulic diagram of the installation [own elaboration]

Fig. 16. Fan cooler installed outside the building [Photo: P. Obstawski]

Fig. 17. Three-way mixing valve with plate heat exchanger [Photo: P. Obstawski]

Fig. 18. Control cabinet [Photo: P. Obstawski]

and output module, four eight-channel modules dedicated to the connection of thermal resistive sensors, and a module dedicated to the connection of thermocouples. The PLC central unit was connected using TCPIP protocol to a computer on which a SCADA programme was installed, enabling visualisation, monitoring, and archiving of the values of the measured physical quantities. The SCADA programme also allows remote access to the application and measurement data via Ethernet.

Comparison of operational data with model-simulated data

To verify the developed heat pump model, a series of service tests were carried out with the compressor heat pump at different lower source temperatures. During the experiments, the heat pump supplied heat to the buffer tank by raising the buffer water temperature from 20 °C to 50 °C. Tests were carried out at four pump capacities, P1, corresponding to the flow rates: 10 kg/min, 12 kg/min, 18 kg/min, and 23 kg/min (constant flow during the experiment) and compressor control of 20%, 30%, 50%, and 70% of the maximum power value, respectively. During the test, the following were measured at a sampling frequency of 1 Hz: heating water volumetric flow rate, working medium temperature at the condenser supply and return, condenser supply and return water temperature, lower source temperature, and power consumed by the compressor (Figure 20–23). With measurements of the condenser supply and return temperatures and flow rate, it was possible to determine the capacity provided by the condenser water $Q_{g\_sk\_w}$ according to relation 24.

$$Q_{g_{sk_w}} = \dot{m} \cdot c_p \cdot (T_{in} - T_{out}) \tag{24}$$

In order to verify the model, many operational tests for the established boundary conditions were carried out on test stand described in the Chapter 4. Results obtained in the laboratory were compared with the results calculated with simulation tests. During the operational tests were recorded: the power of the condenser, the compressor electrical energy consumption, the value of the COP coefficient, the temperature of the working medium and refrigerant on the primary and secondary side of the condenser and the temperature of the atmospheric air (lower source temperature).

The compressor heat pump model was implemented in the Matlab Simulink environment (Figure 19). Simulation studies were carried out by introducing the lower source temperatures and refrigerant condensation temperatures recorded during service tests as input signals in the form of time series.

Figure 20 shows temperatures during the experiment and Figure 21 the power values of the

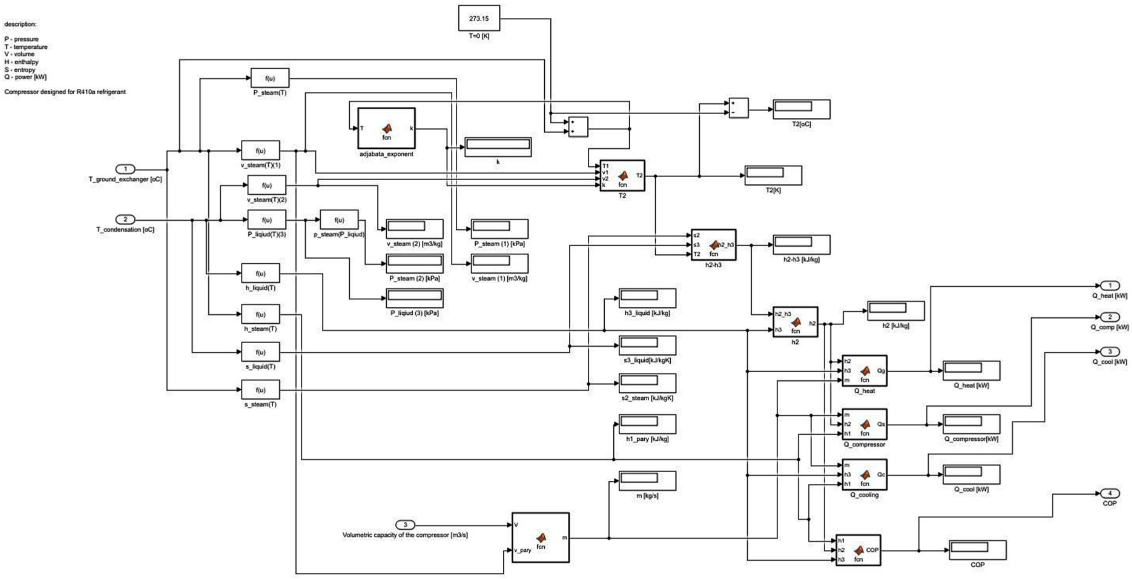

Figure 19. The compressor heat pump model implemented in the Matlab Simulink environment

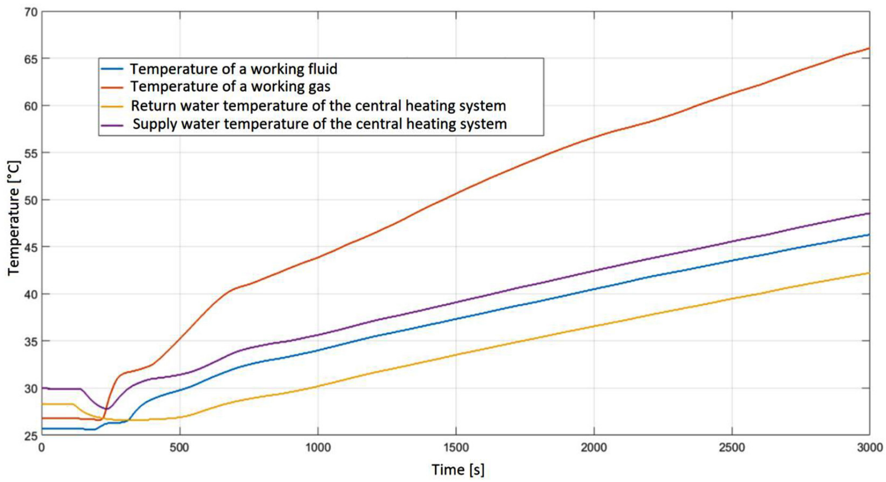

Fig. 20. Curve showing the temperature of the working medium and the heating water in the condenser supply and return for a water flow of 12 kg/min and 50% compressor control (transient state visible in the initial phase up to approx. 300 s)

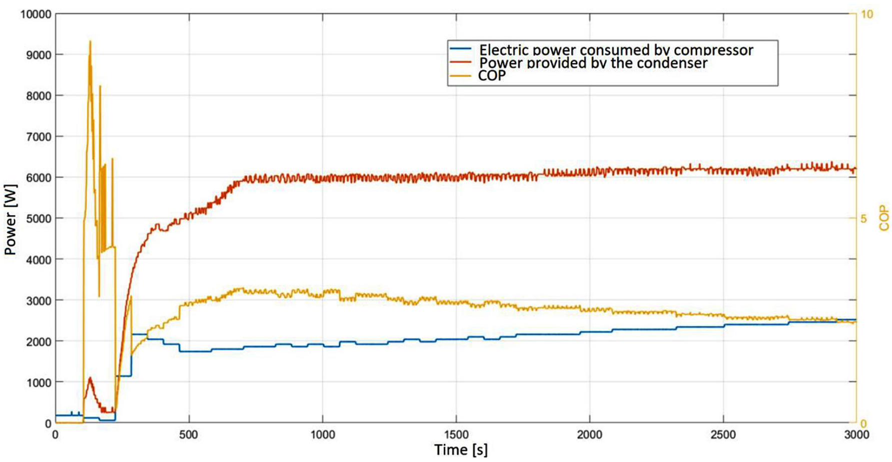

Fig. 21. Curve showing the capacity of the heating water provided from the condenser, the power consumed by the compressor and the COP for a water flow of 12 kg/min and 50% compressor control (transient state visible in the initial phase up to approx. 300 s)

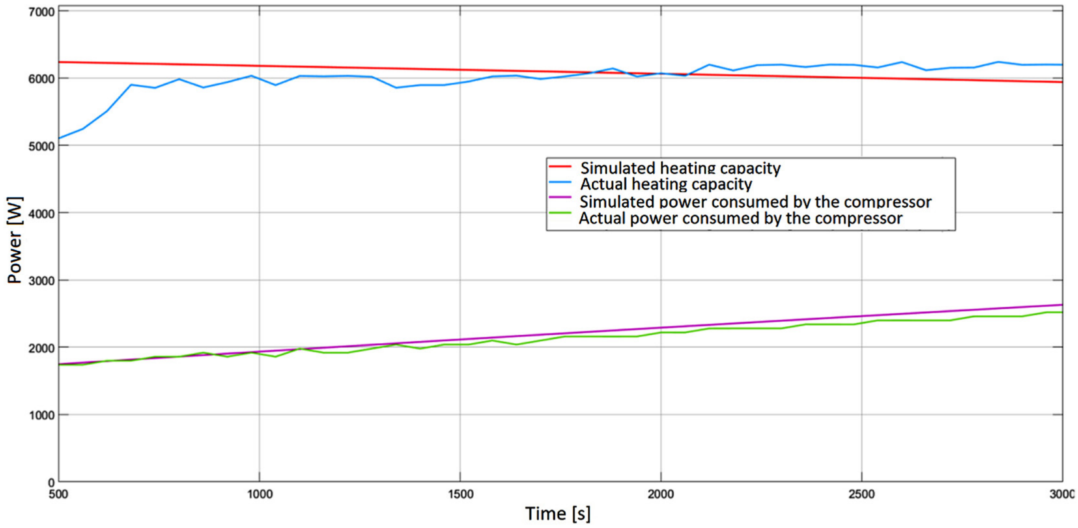

Fig. 22. Comparison of simulated and actual values of heating capacity (R^2 = 95.8%) and electricity consumption (R^2 = 97.4%)

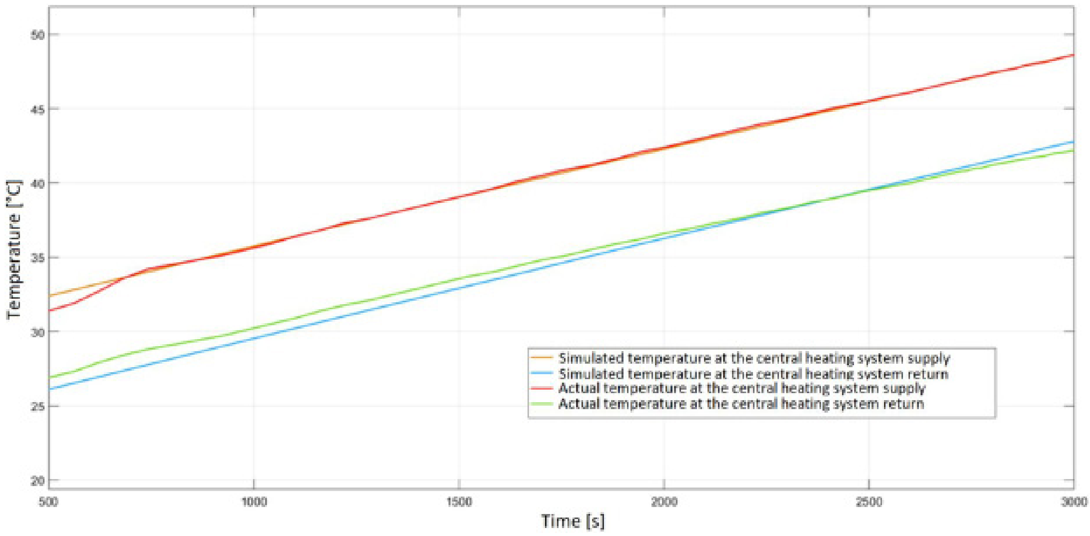

Fig. 23. Comparison of simulated and actual condenser supply (R^2 = 98.8%) and return water temperatures (R^2 = 97.4%)

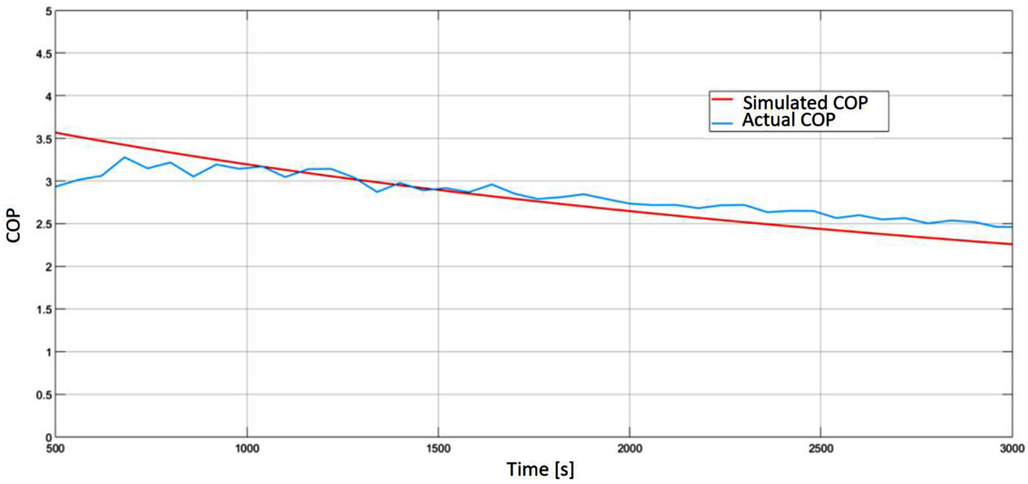

Fig. 24. Comparison of simulated and actual COPs (R^2 = 95.6%)

system. On Fig. 20 and Fig. 21 transient state is visible in the initial phase up to approx. 300 s, the following figures (22–24) show the curves after stabilization of the transient state.

Figures 22–24 show a comparison of the actual and simulated values: heat pump heating power, electricity consumption (Figure 22), condenser supply and return water temperature (Figure 23) and COP value (Figure 24). When analyzing the course of real and simulated operating parameters of the heat pump shown in Figures 22 - 24, it should be noted that the time series overlap, which proves the very high quality of the developed heat pump model, as confirmed by the high value of the coefficient of determination ($R^2 > 95\%$).

CONCLUSIONS

This paper presents a universal compressor heat pump model that allows to conduct simulation studies of a heat pump for any refrigerant based on saturation and condensation temperature values and refrigerant mass flow rate and model parameters calculation for two refrigerants R410a and R290.

The practical laboratory verification of the compressor heat pump model presented in this article, described in chapter 4, confirmed its correctness and purposefulness of application. The model was verified by comparing operational data recorded during operation of an air-to-water heat pump with a compressor, with data simulated using the model. In this way, an effective and easy-to-implement control algorithm of the heat pump was obtained. The model inputs were the actual refrigerant saturation and condensation temperature and refrigerant mass flow. The output signals of the model are: heating power, cooling power, the compressor electrical energy consumption. The verification of the model consisted in comparing the operational parameters simulated with the use of the model with the operational parameters recorded during the operational tests of the compressor heat pump supplied with the R410a factor for the given boundary conditions.

A very good convergence between the simulated and measured values was obtained for the heat pump with the refrigerant R410a (simulated and actual values curves have good convergence, as confirmed by the high value of the coefficient of determination $R^2 > 95\%$). Additionally, the presented model parameters were used for calculations for the heat pump with refrigerant R290.

The developed model enables conducting simulation tests for any refrigerant and any compressor, which makes it universal. The developed model makes it possible to calculate the operational parameters of the device, e.g. such as COP and SCOP for given boundary conditions, which then enables to estimate the coverage degree of the heat load of the building by the heat pump due to central heating and to estimate momentary and seasonal operating costs of the heat pump.

Nomenclatures

Symbol	Meaning	Unit
Q	Capacity	W
R^2	Coefficient of determination	
k	Critical point	–
h	Enthalpy	kJ/kg
S	Entropy	
$\dot{m}$	Flow rate	dm^3/s
R	Gas constant	
Cp, Cv	Specific heat capacity	$J/(kg\cdot K)$
T	Temperature	$°C$
V	Volume	dm^3
v	Volumetric efficiency of compressor	m^3/h
Indexes		
g	Heating	
c	Cooling	
w	Water	
s	Compressor	
$steam$	Steam	
$fluid$	Fluid	
in	Inlet	
out	Outlet	
Acronyms		
COP	Coefficient of performance	
SCOP	Seasonal coefficient of performance	

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