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A novel numerical methodology for modelling simple vapour compression refrigeration system

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Abstract

The extended use of cooling and refrigeration systems in industrial, building and transport sectors may have a negative impact on the climate change and ozone depletion. Thus, important aspects related to these systems, such as refrigerant charge level, malfunction, or refrigerant leakage, must be taken into account. In this sense, the study of refrigeration system performance under different conditions can be very useful. In this paper, a novel methodology for modelling a simple compression refrigeration system is described. Starting from three input parameters, i.e. the ambient air temperature, the cold room air temperature, and the degree of superheating, a calculation algorithm based on iterative loops is used in the model to determine the operating point of the system. An experimental set-up consisting of a walk-in freezer unit was used for the development and validation of the model. The model is system dependent, i.e. empirical correlations must be derived for determining some of the features of system components. The numerical results are in good agreement with the experiment, therefore the model can be a reliable tool for the detection of a system malfunction.

Keywords: Numerical simulation; Compression refrigeration system; Experimental validation.

Nomenclature

A	surface area [m ²]
c _p	specific heat at constant pressure [J kg ⁻¹ K ⁻¹]
$\dot{D}$	actual displacement [m ³ s ⁻¹]
f	fraction of the total heat exchanger occupied by a specific region [-]
frac	fraction of air blown by the fan with respect to the maximum value [-]
h	specific enthalpy of the refrigerant [J kg ⁻¹ K ⁻¹]
h·A	effective thermal conductance of heat exchanger [W K ⁻¹]
$\dot{m}$	mass flow rate [m ³ s ⁻¹]
P	pressure [kPa]
$\dot{Q}$	heat transfer rate [W]
r	compression ratio [-]
R ²	coefficient of determination
T	temperature [°C]
T _c	condensation temperature [°C]
T _e	evaporation temperature [°C]
T _w	wall temperature of the condenser tubes [°C]
U·A	overall thermal conductance [W K ⁻¹]
$\dot{V}$	volumetric flow rate [m ³ s ⁻¹]
$\dot{W}$	electrical power consumption [W]
x	vapour quality of the refrigerant [-]
Greek symbols	
α	convective heat transfer coefficient [W m ⁻² K ⁻¹]
η	isentropic efficiency of the compression process [-]
ρ	density [kg m ⁻³]
ξ	fraction of energy losses from the compressor to the environment [-]
Abbreviations and subscripts	
a	air
ACWC	air cooled water chiller
ANN	artificial neural networks
c	compressor
calc	calculated
cond	condenser
COP	coefficient of performance
EES	Engineering Equation Solver
evap	evaporator
exp	expected, experimental
GRBF	generalized radial basis function
GWP	global warming potential
HFCs	hydrofluorocarbons
HVAC	Heating, Ventilation, and Air Conditioning
int	internal, inner
max	maximum
MRE	mean relative error
ODP	ozone depletion potential
RMSE	root-mean-square error

s	isentropic
SC	subcooling
SH	superheating
theor	theoretical
w	condenser tubes wall

1. Introduction

In the last few decades, a lot of concerns were raised regarding the climate change and ozone depletion produced by a highly increasing demand of energy resources in the industrial, building and transport sectors. As a result, different regulations and protocols were established, such as the Montreal and Kyoto protocols, which aim to reduce the emissions of gases with high global warming potential (GWP) or ozone depletion potential (ODP). A study on estimations of hydrofluorocarbons (HFCs) emissions in 2050 shows that HFCs will account for 9-19% of the projected global CO₂ emissions [1]. Therefore, issues such as proper refrigerant charge, malfunction or refrigerant leakage are important aspects to be taken into account because they can have dramatic economic and environmental impact.

In order to prevent possible system malfunctions or performance degradation, and limit the maintenance costs of the installation, it is important to develop numerical tools which can help in determining the correct operating regime of a system, as well as the optimum charge level, for design purposes.

There are many studies in the literature focusing on the optimal refrigerant charge and the possibility of charge reduction in refrigeration equipment. Some of these studies only include experimental research; however, others also deal with the modelling of vapour-compression liquid refrigeration systems for predicting the system operating performance. Poggi et al. [2] presented a review of available knowledge in terms of refrigerant charge studies, and describes and evaluates the possibilities of charge reduction in refrigeration systems and heat pumps. The review also includes some methodologies for charge measurement, and evaluates the influence of the refrigerant charge on the performance of the system. Palm [3] discussed the reasons and possibilities to decrease the refrigerant charge in both refrigeration and heat pump systems, both at the system and the component levels. A rationally based algorithm was proposed by Vjacheslav [4] to evaluate the optimal refrigerant charge in refrigerating machines. Their results indicated a strong relation between the system performance and the refrigerant charge level, and that there was an optimal charge for which the coefficient of performance (COP) reached a maximum value.

Many experimental researches were carried out regarding the influence of charge level on the performance of different types of vapour compression systems. Palmiter et al. [5] studied the effect of the refrigerant charge and blower airflow variations on the seasonal performance of residential heat pump, for both heating and cooling modes in the six climatic zones in the US. Their results showed that an increase in the charge may be beneficial in heating mode, but it is problematic for cooling mode, while a decreased airflow produces significant penalties in both heating and cooling modes. Choi et al. [6] developed a refrigerant detection algorithm for predicting the refrigerant charge in a ground-source heat pump system. They conducted experiments for developing a refrigerant amount detection algorithm, and investigated the effects of varying refrigerant charge amount on the performance of a ground source heat pump unit. The influence of refrigerant charge and expansion restrictions in household refrigerators were investigated experimentally [7,8]. Goswami et al. [9] investigated the effect of reducing refrigerant charge in air-conditioning systems, and their results indicated that a charge level below 80% considerably reduced the system performance.

Modelling of refrigeration systems for predicting the influence of the total refrigerant charge on the performance is a valuable tool, since it can help in estimating the optimal charge, in detecting systems under- or overcharge, or improper functioning conditions. There are many approaches for system modelling, but they can be classified into two main groups: physical and empirical (grey- or black-box type) models.

The physical models require detailed information of the different system components, and their performance is based on equations derived from physics laws. Corberán et al. [10] carried out an extensive experimental research to study the influence of the refrigerant charge on the performance of a reversible water-to-water propane heat pump. Their research also included a theoretical study using IMST-ART software [11] for estimating the refrigerant distribution among the different system components. The experimental study performed by Corberán et al. [10] was followed by another research [12], in which a simulation study was performed to elucidate the influence of both the source and sink temperatures on the optimal charge of the system. Their model was able to accurately predict the performance of the unit under different working conditions, for both chiller and heat pump modes. An interesting study was performed by Kim et al.

[13] on air-to-water single and cascade heat pumps. They studied both experimentally and numerically the effect of the charge amount on the performance of heat pumps. For the numerical study they proposed a flow chart for simulating the cycle assuming a constant degree of superheating at compressor inlet, based on an iterative process that stopped once the charge amount restriction was met. The numerical simulation was conducted using a MATLAB code, and it was able to effectively predict the system variables.

In empirically based models, which include regression analysis, polynomial curve fits and artificial neural networks (ANN), there is no need of detailed information of system components, and they are able to estimate the performance of the system given a set of input variables. Among them, the radial basis function neural network models may be preferred for predicting system performance [14]. However, one has to be aware of the fact that empirically based models may not be able to accurately extrapolate beyond the parameter space values used for estimating model parameters. Swider et al. [15] presented a model that used a generalized radial basis function (GRBF) neural network to predict the performance of two different chillers in steady-state conditions. The GRBF model only required input parameters that were readily known to the operating engineer, and was able to make predictions in agreement with experimental data within $\pm 5\%$. Bechtler et al. [16] used an approach based on ANN that used a GRBF, to model dynamic processes in refrigeration systems, which required as inputs only those parameters that were easily measurable. Jiang et al. [17] applied the grey forecasting for predicting the operating energy performance of an air cooled water chiller (ACWC) unit. Their results indicated that the grey forecasting method was able to make accurate predictions of system performance, so that it could be integrated into building Heating, Ventilation, and Air Conditioning (HVAC) installations to prevent low operating efficiency, for energy saving purposes and sustainable development.

This paper presents a novel methodology for predicting the steady-state performance of a simple vapour compression refrigeration system based on a physical model approach. However, the modelling of the different components of the system is not determined exclusively on the basis of physical laws, and some empirical correlations are used. In order to obtain the required empirical correlations under different operating conditions,

a set of experimental tests were conducted using an experimental rig. This simplifies the overall theoretical description of the model, and reduces the possible uncertainties inherent to complex physical laws or an incomplete knowledge of some of the components, but still allows a physical interpretation of the parameters. Finally, the model predictions are compared and validated against experimental data.

2. Methodology

2.1 Overview of the model

The main objective of the research described in this paper is to present a novel methodology for determining the performance of a simple vapour compression system in quasi-stable operating conditions. A numerical model was built, based on a walk-in freezer installation consisting of a condensing unit, a throttle device, and an evaporator. The schematic of the refrigeration cycle is shown in Figure 1.

Figure 1: Scheme of the refrigeration cycle

The points represented in Figure 1 correspond to different states of the refrigerant along the cycle, and also to different states of the air in contact with the heat exchangers. Since the different components of the system are connected through relatively short and well insulated pipes, the model assumes no heat transfer from the connecting pipes. A short description of these states is given in Table 1.

Table 1: Description of the different points shown in Figure 1

Point	Description
1	Evaporator outlet and compressor inlet
2	Compressor outlet and condenser inlet
3	Saturated vapour inside the condenser
4	Saturated liquid inside the condenser
5	Condenser outlet and expansion valve inlet
6	Expansion valve outlet and evaporator inlet
7	Saturated vapour inside the evaporator
8	Outdoor air (ambient) at condenser inlet
9	Outdoor air (ambient) at condenser outlet
10	Indoor air (freezer) at evaporator inlet
11	Indoor air (freezer) at evaporator outlet

In order to determine the steady-state operating point of the system, the model requires three input variables:

- The outdoor air (ambient) temperature (T_8);
- The indoor air (freezer) temperature (T_{10});
- The degree of superheating at the compressor inlet ($T_1 - T_7$).

For a given a set of values of the three input variables, the operating point of the system is completely determined. The methodology presented in this paper is based on different iterative loops for the determination of the unique solution for the evaporation and condensation pressures, given a set of input parameters values. Engineering Equation Solver (EES) software was used to perform the simulations, because it provides a highly accurate thermodynamic and transport property database for hundreds of substances,

and is able to numerically solve thousands of coupled non-linear algebraic and differential equations.

Once the two system pressures are determined, all output parameters of the system, such as refrigerant and air temperatures at different points, compressor work, cooling capacity, and COP can be obtained from the governing equations of each system component. The unique solution for the two pressures is determined based on the following requirements:

- 1) It has to predict the correct degree of subcooling at condenser outlet,
- 2) Starting from one point of the cycle, if one follows the different processes performed in each of the system components, one has to end up on the starting point (“closing” the cycle).

Taking into account these two requirements, the solution is obtained using a calculation algorithm based on iterative processes, grouped in three blocks. The details of the calculation algorithm are described below, in Section 2.3.

2.2 Energy balances inside components

As previously mentioned, the model described in this paper uses a physical approach, based on energy balances of each of the system components. The governing equations of each of the main system components are shown in this section.

2.2.1 Compressor

There are three equations that characterize compressor behaviour in a refrigeration system: one equation for the refrigerant flow rate, one equation for the enthalpy at compressor discharge, and one for the electrical power consumption. The actual volumetric displacement of the compressor was supposed to be a linear function of the compression ratio ($r=P_2/P_1$), as shown in Eq. 1:

$$\dot{D}_c = a_1 + a_2 \cdot r$$

Eq. 1

where coefficients a_1 and a_2 can be determined from the compressor performance data provided by the manufacturer.

The refrigerant mass flow rate ($\dot{m}$) is then determined from Eq. 2:

$$\dot{m} = \rho_1 \dot{D}_c \quad \text{Eq. 2}$$

where ρ_1 is the density of the superheated refrigerant at compressor inlet, and it can be inferred using EES database from the refrigerant pressure and temperature.

The specific enthalpy at compressor discharge (h_2) is obtained from Eq. 3:

$$h_2 = h_1 + \frac{h_{2s} - h_1}{\eta_c} \quad \text{Eq. 3}$$

where h_1 is the specific enthalpy of the refrigerant at compressor inlet, h_{2s} is the specific enthalpy of the refrigerant at the isentropic compressor discharge, and η_c is the isentropic efficiency of the compression process. The isentropic efficiency is one of the several parameters of the model that has to be determined experimentally.

The electrical power consumption of the compressor ($\dot{W}_c$) takes into account the possible heat losses to the environment, and is given by Eq. 4:

$$\dot{W}_c = \dot{m} \cdot \frac{h_2 - h_1}{1 - \xi} \quad \text{Eq. 4}$$

where ξ accounts for the fraction of the electrical power consumed by the compressor that is lost to the environment, which is another model parameter that has to be determined experimentally.

2.2.2 Condenser

The following assumptions have been considered in the analysis of the condenser:

- Negligible pressure loss between the refrigerant inlet and outlet of the condenser;
- The heat transfer properties are uniform across the condenser surface, except for the heat transfer coefficient on the inner tube surface, which may be different in each of the three regions;
- The specific heat of the refrigerant is constant in both the superheated vapour and the subcooled liquid regions, with the corresponding values evaluated at the mean temperature of each region;
- Steady state conditions, meaning that the air- and refrigerant mass flow rates do not vary between condenser inlet and outlet;
- The airflow is uniformly distributed across the outer surface of the condenser, so that the volumetric flow rate of air crossing each of the three regions is proportional to the surface area occupied by each region;
- All the heat released by the refrigerant is accepted by the cooling air.

According to the manufacturer, the condenser fan speed varies with the head pressure P_2 , thus it is essential to determine the volumetric flow rate of the air flowing across the condenser ($\dot{V}_{a,cond}$) in order to perform a theoretical analysis of the condenser. It can be obtained from an energy balance applied to the condenser, as shown in Eq. 5, by first determining the mass flow rate of condenser air ($\dot{m}_{a,cond}$), and then dividing it by the air density:

$$\dot{m} \cdot (h_2 - h_5) = \dot{m}_{a,cond} \cdot c_{p,a,cond} (T_9 - T_8)$$

Eq. 5

where $c_{p,a,cond}$ is the specific heat of the air flowing across the condenser.

The volumetric flow rate can be expressed as a fraction ($frac_{cond}$) of the maximum value provided by the manufacturer ($\dot{V}_{a,cond,max} = 1650 \text{ m}^3 \text{ h}^{-1}$), as shown in Eq. 6:

$$frac_{cond} = \frac{\dot{V}_{a,cond}}{\dot{V}_{a,cond,max}} \quad \text{Eq. 6}$$

An empirical correlation for $frac_{cond}$ has been obtained from experimental data, which can be found in Section 4.1.

In the heat transfer analysis of the condenser, one must distinguish between the three different regions defined by the physical state of the refrigerant inside the tubes: a) superheated vapour refrigerant, b) saturated liquid-vapour mixture of refrigerant corresponding to the condensation process, and c) subcooled liquid refrigerant. Thus, the condenser can be treated as a set of three different heat exchangers connected in series. The heat transfer process in each of these three heat exchangers is calculated using the ϵ -NTU method [18].

a) Superheated vapour refrigerant (region 2→3)

In this region both the refrigerant and the cooling air have finite calorific capacities. At the inlet of the region 2→3, the refrigerant temperature is the same as the compressor discharge temperature, while at the outlet of this region the refrigerant is at the saturated temperature corresponding to the head pressure P_2 . The cooling air enters this region at environmental temperature, and exits at a temperature that can be inferred from an energy balance. However, the air flow rate flowing through this region is unknown, but it is proportional to the fraction f_{23} occupied by the region 2→3. Thus, by applying the ϵ -NTU method one obtains an equation for f_{23} , which is shown in Eq. 7:

$$f_{23} = (U \cdot A)_{23} \cdot \left(\frac{1}{\alpha_{int,23} \cdot A_{int,cond}} + \frac{1}{(h \cdot A)_{cond}} \right) \quad \text{Eq. 7}$$

where $(U \cdot A)_{23}$ is the overall thermal conductance of the region 2→3, $\alpha_{int,23}$ is the convective heat transfer coefficient inside the tube surface in the region 2→3, $A_{int,cond}$ is the total area of the inner surface of the condenser tubes, and $(h \cdot A)_{cond}$ is an effective thermal conductance of the condenser, which includes all possible thermal resistances except the internal convection: fouling resistances, contact resistances, tubes and fins conduction resistances, external convection, etc.

In order to determine the value of $(U \cdot A)_{23}$, one has to know the value of the fraction f_{23} , so that Eq. 7 has to be solved iteratively.

The internal convection coefficient $\alpha_{int,23}$ is determined by the use of “PipeFlow” procedure of EES, which assumes simultaneous hydrodynamic and thermally developing flow as reported in Section 5.2.3 by Nellis and Klein [19], and requires as inputs the fluid type, its bulk temperature and pressure, the mass flow rate, and the length, diameter and relative roughness of the inner tube surface.

Regarding $(h \cdot A)_{cond}$ parameter, it can be estimated from the geometrical and physical characteristics of the condenser. However, since it is a difficult task to obtain reliable results based on this approach, an empirical correlation for $(h \cdot A)_{cond}$ was determined from experimental data.

b) Saturated liquid-vapour refrigerant mixture (region 3→4)

This region corresponds to the condensing process, to which the refrigerant enters as saturated vapour and exits as saturated liquid corresponding to the head pressure P_2 . Thus, refrigerant is changing phase so that it has an infinite heat capacity. A relation similar to Eq. 7 can be obtained using the ε -NTU method applied to region 3→4, which is shown in Eq. 8:

$$f_{34} = (U \cdot A)_{34} \cdot \left(\frac{1}{\alpha_{int,34} \cdot A_{int,cond}} + \frac{1}{(h \cdot A)_{cond}} \right)$$

Eq. 8

where again $(U \cdot A)_{34}$ depends on the fraction f_{34} occupied by the region 3→4, so that Eq. 8 has to be solved by iteration. Notice that $(h \cdot A)_{cond}$ parameter is the same as in Eq. 7, which justifies the assumption that the heat transfer properties are uniform across the condenser surface, except for the heat transfer coefficient on the inner tube surface.

The internal convection coefficient $\alpha_{int,34}$ corresponds to the condensation process. It is determined by means of the “Cond_HorizontalTube_avg” procedure of EES software, which calculates the average heat transfer coefficient for condensation of vapour entering at quality x_1 as it condenses to a state with quality x_2 inside a circular tube. In region 3→4 $x_1 = 1$ (saturated vapour) while $x_2 = 0$ (saturated liquid). Apart from the entering and exiting qualities, the procedure also requires as inputs the fluid type, the mass flow rate and saturation temperature of the incoming vapour, the temperature of the inside surface of the tube, and the inner diameter of the tube. The wall temperature of the condenser tubes ($T_{w,cond}$) in the condensation region strongly depends on the condensation temperature, and it is estimated using an iterative approach.

c) Subcooled liquid refrigerant (region 4→5)

The refrigerant enters this region as saturated liquid at a temperature corresponding to saturation at pressure P_2 . The refrigerant temperature at condenser outlet is unknown, so that it depends on the heat transfer between refrigerant and the cooling air flowing through the region 4→5. Previous equations Eq. 7 and Eq. 8 allow to determine the fractions f_{23} and f_{34} occupied by the regions 2→3 and 3→4, respectively, and since the sum of the fractions occupied by the three regions must be equal to one, this sets the value of f_{45} . This allows to calculate the value of $(U \cdot A)_{45}$ product by means of Eq. 9:

$$(U \cdot A)_{45} = \frac{f_{45}}{\frac{1}{\alpha_{int,45} \cdot A_{int,cond}} + \frac{1}{(h \cdot A)_{cond}}}$$

Eq. 9

By applying the ϵ -NTU method to the region 4→5, the heat transfer rate, $\dot{Q}_{45}$, and the refrigerant temperature at condenser outlet, T_5 , can be inferred. Consequently, the calculated degree of subcooling ($T_{SC,calc}$) is obtained from Eq. 10:

$$T_{SC,calc} = T_4 - T_5$$

Eq. 10

where T_4 is the temperature of saturated liquid refrigerant.

2.2.3 Evaporator

An approach similar to the one used for analysing the condenser, and similar assumptions can also be applied in the case of the evaporator, though there are some differences to be taken into account. The main difference with respect to the condenser is that in the evaporator there are only two distinct regions: a) superheated vapour refrigerant, and b) saturated liquid-vapour mixture of refrigerant corresponding to the evaporation process. Another difference with respect to the condenser consists in the constant speed feature of evaporator fan.

a) Superheated vapour refrigerant (region 7→1)

The refrigerant enters this region as saturated vapour at the low branch pressure P_1 , and exits it as superheated vapour. Since the superheated at evaporator exit is one of the model inputs, this means that the exit state (1) is completely determined for a given evaporation pressure P_1 . Thus, by applying the ε -NTU method one can obtain an equation for the fraction f_{71} occupied by the region 7→1 shown in Eq. 11, which can be solved by iteration:

$$f_{71} = (U \cdot A)_{71} \cdot \left(\frac{1}{\alpha_{int,71} \cdot A_{int,evap}} + \frac{1}{(h \cdot A)_{evap}} \right)$$

Eq. 11

where $(U \cdot A)_{71}$ is the product of the overall thermal conductance of the region 7→1, $\alpha_{int,71}$ is the convective heat transfer coefficient inside the tube surface in the region 7→1, $A_{int,evap}$ is the total area of the inner surface of the evaporator tubes, and $(h \cdot A)_{evap}$ is an effective thermal conductance of the evaporator, which includes all possible thermal resistances except the internal convection.

The internal convection coefficient $\alpha_{int,71}$ is determined by the use of “PipeFlow” procedure of EES, in a similar way as the $\alpha_{int,23}$ coefficient in the superheated vapour region of the condenser.

As in the condenser case, $(h \cdot A)_{evap}$ was also determined from experimental data, and the correlation obtained is shown in Section 4.1.

b) Saturated liquid-vapour refrigerant mixture (region 6→7)

In this region, the refrigerant that exits the expansion valve as a saturated mixture of liquid and vapour at the low branch pressure P_1 , evaporates due to heat absorption from the air flowing across the evaporator. Since the value of the fraction f_{71} is already known from Eq. 11, this fixes the value of $f_{67} = 1 - f_{71}$. Consequently, one is able to calculate the value of $(U \cdot A)_{67}$ product from Eq. 12:

$$(U \cdot A)_{67} = \frac{f_{67}}{\frac{1}{\alpha_{int,67} \cdot A_{int,evap}} + \frac{1}{(h \cdot A)_{evap}}}$$

Eq. 12

where $\alpha_{int,67}$ is the heat transfer coefficient inside the evaporator tubes in the evaporation process. This coefficient was determined experimentally by measuring the evaporator tube surface temperature and applying a heat transfer analysis. The corresponding correlation can be found in Section 4.1.

By applying the ϵ -NTU method one can calculate the heat transfer rate, $\dot{Q}_{67}$, and also the refrigerant specific enthalpy at the outlet of the region 6→7, from equation Eq. 13:

$$h_{7,calc} = h_6 + \frac{\dot{Q}_{67}}{\dot{m}}$$

Eq. 13

By comparison to the expected saturation enthalpy, h_7 , the value of the $h_{7,calc}$ parameter is used by the algorithm to restrict the allowed range of values for the low pressure P_1 , as explained in Section 2.3.

2.2.4 Throttling device

The methodology described in this paper is based on a refrigeration system that contains a stepper-motor electronic expansion valve. The operation of this kind of expansion valve is such that it keeps a fixed degree of refrigerant superheating (T_{SH}) at evaporator outlet, which can be determined experimentally. Another feature of the expansion valve consists in the isenthalpic nature of the expansion process, from the condensation pressure to the evaporation pressure. Therefore, the model assumes that the refrigerant enthalpy at evaporator inlet is equal to the refrigerant enthalpy at condenser outlet ($h_5 = h_6$).

2.2.5 Other components

Other components that are frequently found in refrigeration systems could also be taken into account. For instance, the connecting pipes between the main components may produce both pressure and temperature drops along the flow direction. However, in condensing refrigeration units, pressure and temperature variations between the inlet and the outlet of the different connecting pipes are usually small. Therefore, the model assumes ideal conditions in which both pressure and temperature variations are disregarded.

2.3 Calculation algorithm

Each of the processes that take place in the main system components involves the use of a number of equations and unknowns. However, when all equations are put together one can see that, in order to be able to solve the equations system, one additional equation is needed. As discussed in [20], there are two alternatives for providing the required equation: the first one consists of a mass inventory for the refrigerant, and the second one consists of a linear equation for the subcooling fitted to reproduce the measured values at different amounts of refrigerant charge. The mass inventory of the system is a very difficult task because it requires a very precise description of each element of the system, including piping and auxiliary elements, and also a reliable and accurate method for estimating the refrigerant mass in each element. For simplicity, the methodology

presented in this paper relies on the use of a non-linear equation for subcooling inferred from experimental data ($T_{SC,exp}$), shown in Section 4.1.

The algorithm used in the model is described below, and is divided into three different blocks, in such a way as to avoid exploring the entire space of parameter values for determining the solution, which reduces the computational time and resources.

2.3.1 Block 1

The first block, shown in Figure 2, is aimed at determining a rough upper bound for the pressure P_2 . The first step consists in reading the values of the three inputs of the model: T_8 , T_{10} and T_{SH} . Next, the maximum value of the evaporation pressure ($P_{1,max}$) is determined as the saturation pressure corresponding to temperature maximum value of $T_{7,max} = T_{10} - T_{SH}$, which corresponds to the limiting case in which refrigerant temperature at evaporator outlet equals the freezer temperature ($T_{1,max} = T_{10}$). On the other hand, the minimum value of the condensation pressure ($P_{2,min}$) is given by the saturation pressure at ambient temperature, since the actual condensation temperature (T_c) must always be higher than the ambient temperature. Finally, the expected degree of subcooling ($T_{SC,exp}$) is determined from the empirical correlation given in Eq. 22.

Figure 2: First block of the algorithm

The second step consists of setting the initial values of the low pressure to its maximum, and the head pressure to its minimum. Once the values of the two pressures are fixed, the main thermo-physical properties of refrigerant at states 7 (saturated vapour inside evaporator) and 1 (evaporator outlet and compressor inlet) can be determined using EES database. At this point, a loop for P_2 is initiated and its value is increased by an amount $\Delta P_2 = 10$ kPa, which is relatively large in order to obtain a relatively short computation time. The compression process is determined by solving the equations system formed by Eq. 1 to Eq. 4, which allow determining the refrigerant mass flow rate ($\dot{m}$), refrigerant specific enthalpy (h_2) and temperature (T_2) at compressor discharge, and also

the electrical power consumption of the compressor ($\dot{W}_c$). The condensation process is solved using the ε -NTU method described in Section 2.2.2, which allows determining the degree of subcooling ($T_{SC,calc}$) defined in Eq. 10. As long as the calculated subcooling remains lower than the expected subcooling ($T_{SC,calc} < T_{SC,exp}$), the head pressure P_2 has still not reached its maximum value ($P_{2,max}$). The loop stops when the calculated subcooling gets higher than the expected subcooling ($T_{SC,calc} > T_{SC,exp}$). When this happens, it means that the head pressure has reached a maximum value $P_{2,max}$. This is so because P_1 is kept constant to its maximum value along loop calculations, for which the refrigerant mass flow rate is higher than the actual flow rate (because both refrigerant density and volumetric rate are higher than their actual values). Thus, in order to get the expected subcooling, a higher condensation pressure is needed for condensing the surplus of refrigerant flowing inside the condenser.

At this stage the cycle cannot be “closed”, and it looks like in Figure 3 (a). It can be seen that the specific enthalpy at the outlet of the region 6→7 ($h_{7,calc}$), calculated by means of Eq. 13, is less than the value corresponding to saturated vapour. This is so because the evaporation temperature is too high, and the heat transfer rate at the evaporator is not able to evaporate the entire refrigerant flow to reach saturation.

At the end of this first block, the maximum value $P_{2,max}$ is saved for its later use in the second block of the algorithm.

(a)

(b)

Figure 3. Schematic representation of the thermodynamic cycle in the different blocks: (a) at the end of block 1; (b) for an intermediate iteration of block 2; (c) at the end of block 3 (final solution)

2.3.2 Block 2

The maximum values of both head- and low system pressures, as determined in the first block, are the starting points of the iterative process performed in the second block, shown in Figure 4. Both pressures are initially above the actual solution, so that they are lowered until they get close to the actual solution, but still above it. The steps of pressure variations are less than in block 1, but still larger than the desired resolution in order to reduce the computation time. The second block consists of two loops: an external one for P_2 , and an internal one for P_1 . After having set both pressures to their respective maximum values determined in block 1, P_2 is lowered by an amount $\Delta P_2 = 2$ kPa, which is five times lower than in block 1, after which a second loop is initiated for P_1 . Firstly, the main thermo-physical properties of refrigerant at states 7 and 1 are determined using the EES database. Next, compression and condensation processes are calculated according to the methodology explained in Sections 2.2.1 and 2.2.2,

respectively. As long as the calculated subcooling remains lower than the expected subcooling ($T_{SC,calc} < T_{SC,exp}$), P_1 is lowered by an amount $\Delta P_1 = 2$ kPa, and the internal loop for P_1 is restarted. Once $T_{SC,calc} > T_{SC,exp}$, the internal loop stops, and the throttling and evaporation processes are calculated following the methodology explained in Sections 2.2.4 and 2.2.3, respectively. If the specific enthalpy at the outlet of the region 6→7, calculated by means of Eq. 13, is lower than the theoretical value ($h_{7,theor}$) corresponding to saturated vapour at pressure P_1 , the thermodynamic cycle is not “closed”, and looks like in Figure 3 (b). At this stage, the two pressures are still above their actual values. Thus, the external loop for P_2 is restarted, and the last value of P_1 becomes the new maximum for the next external loop ($P_{1,max} = P_1$). This is done for two purposes: on one hand it reduces the computation time by avoiding P_1 values already discarded; on the other hand it allows storing the value of $P_{1,max}$ used in the previous external loop, which is needed at the end of the block as explained below. As a consequence of this approach, the calculated refrigerant state $7_{,calc}$ moves to the right towards the saturation curve (to which it should belong by definition). The external loop stops when the calculated enthalpy at state 7 becomes higher than the theoretical value ($h_{7,calc} > h_{7,theor}$). This is an indication of the fact that the pressures have entered a close vicinity of the actual solution or even below. Therefore, the initial values for P_1 ($= P_{1,max}$) and P_2 ($= P_2 + \Delta P_2$) used in the previous external loop iteration are the latest values that are still above the actual values, and they are saved and used in the final block as initial values for the two pressures.

Figure 4: Second block of the algorithm

2.3.3 Block 3

Conceptually, block 3 is very similar to block 2 with regard to the way in which both pressures are decreased from their initial maximum values. The only difference is that in block 3 the actual values of P_1 and P_2 are searched, within the desired resolution, and not only upper bounds as in block 2.

As shown in Figure 5, pressure P_1 is decreased by $\Delta P_1 = 0.2$ kPa in the internal loop until the calculated subcooling reaches the expected value, within a desired accuracy range $\pm \delta T = 0.02^\circ\text{C}$. Once T_{SC} reaches the expected value, the external loop checks whether the thermodynamic cycle gets “closed”, by comparing the calculated enthalpy $h_{7,calc}$ with the theoretical value $h_{7,theor}$ (saturated vapour at pressure P_1). If $h_{7,calc}$ differs from $h_{7,theor}$ by more than the desired accuracy, pressure P_2 is decreased by $\Delta P_2 = 0.2$ kPa, and the internal loop is started again. The solution for P_1 and P_2 is found when $h_{7,calc} \approx h_{7,theor}$ within the desired accuracy range $\pm \delta h = 0.5$ kJ kg⁻¹, meaning that the calculated refrigerant state 7_{calc} lies on the saturation curve (see Figure 3 (c)).

Once the solution for P_1 and P_2 is found, all other output variables (COP, $\dot{W}_{abs}$, $\dot{Q}_{evap}$, T_9 , T_{11} , etc.) can also be calculated.

Figure 5: Third block of the algorithm

3. Experimental procedure

3.1 Experimental set-up

The experimental set-up used for the development and validation of the model consists of a walk-in freezer unit located in the laboratory room of AKO ELECTROMECAÁNICA, S.A.L. (see Figure 6). The freezing unit employs R404A as refrigerant. The system is equipped with a Silensys condensing unit (SIL2464Z) from Tecumesh consisting of a three-phase hermetic reciprocating compressor, fin-and-tube air-cooled condenser with a variable speed fan. The system also contains a fin-and-tube evaporator equipped with a constant speed fan that blows air over the evaporator. As expansion device the system uses an E2V electronic expansion valve with AKO controller that sets the superheating to a constant value.

The test facility was comprehensively instrumented to measure:

- temperature of refrigerant at the main points in the refrigeration cycle,
- temperature of air at both inlet and outlet of heat exchangers,
- head- and low pressures, using pressure sensors ranging from -0.5 to 8 bar for the low pressure, and 0 to 30 bar for the head pressure, and
- the active electrical power consumption of the compressor using a CVM Mini power analyser.

The temperature measurements were performed using PT-100 temperature sensors with an accuracy of $\pm(0.3+0.005 \cdot |T|)^{\circ}\text{C}$ (DIN EN 60751 F 0.3 class B), and all the data was registered every 5 seconds through IPC-CON modules (ET-7015 and ET-7019Z).

The accuracy of the pressure measurements is, according to the manufacturer's specifications, less than 2%, while the accuracy of the electrical power measurements is 1%.

Figure 6: Walk-in freezer in the laboratory of AKO Electromecánica S.A.L.

3.2 Set of experiments

To obtain a robust simulation model capable of making accurate predictions in a wide range of working conditions, a series of experiments was performed at different levels of external temperature and internal thermal loads, under conditions of optimal refrigerant charge. The external temperature was controlled by varying the set-point of the room's air conditioning system. The cold room temperature was controlled by an AKO ProPlus Basic controller and the superheating degree was set at 9 K, which was regulated by an EEV controller from AKO.

The experimental procedure consists of different tests performed at three different external temperatures, and three levels of internal cooling loads, as summarized in Table 2. The internal loads were simulated using a variable heat resistance located inside the freezer. Heat infiltrations through the freezer walls and door must be added to the different loads supplied by the heat resistance.

Table 2: Set of experiments performed at different external and internal conditions

External temperature (°C)	Internal heat load (W)		
18	0	650	1300
24	0	650	1300
30	0	650	1300

The tests were carried out following the same pattern. A defrost was done before each test, in order to avoid the influence of the frost in the results. Then, the compressor was switched on, after which the internal heat loads were also activated. After an initial transition period, the system entered a quasi-stable regime before reaching the set point. On or more points in this quasi-stable regime were used for determining the different experimental correlations required in the model.

4. Results and discussion

4.1 Determination of empirical correlations

The energy balances of the different system components described in Section 2.2 require some parameters that are difficult to be determined only from theory-based considerations. For that reason, empirical correlations have been developed based on experimental measurements. The correlations were determined in such a way as to maximize the value of the coefficient of determination R^2 .

First of all, an equation was determined for the actual displacement of the compressor, from the data provided by the manufacturer data, which is given in Eq. 14:

$$\dot{D}_c = (1.55 \cdot 10^{-3} - 5.28 \cdot 10^{-6} \cdot T_c) - (6.24 \cdot 10^{-5} - 8.38 \cdot 10^{-7} \cdot T_c + 6.43 \cdot 10^{-9} \cdot T_c^2) r \quad \text{Eq. 14}$$

where T_c is the condensation temperature, and r is the compression ratio.

The correlation obtained for the fraction $frac_{cond}$ of air blown by the condenser fan with respect to the maximum value only depends on the condensation temperature T_c , and is given in Eq. 15:

$$frac_{cond} = \begin{cases} 0.3 & \text{if } T_c \leq 23.45^\circ C \\ 0.934 & \text{if } T_c \geq 42.52^\circ C \\ -2.218 + 0.1483 \cdot T_c - 0.00174 \cdot T_c^2 & \text{if } 23.45 < T_c < 42.52^\circ C \end{cases} \quad \text{Eq. 15}$$

According to the manufacturer, the fan speed varies in a certain range of pressures (condensation temperatures) from a minimum of 30% of the maximum speed at low pressures, to a maximum of 100% at high pressures. In all the experimental tests the

condensation pressure never reached the minimum bound, so that a value of 0.3 was assumed for $frac_{cond}$ at low pressures. On the other hand, at high pressures the maximum value obtained experimentally was less than 1, but this may be attributed to experimental errors or to an actual performance of the fan below the value supplied by the manufactures. Between these extreme bounds, a parabolic variation of $frac_{cond}(T_c)$ was found to fit quite well experimental data, with a coefficient of determination $R^2 = 0.891$.

Another empirical correlation required by the model was obtained for the effective thermal conductance of the condenser $(h \cdot A)_{cond}$, defined in Section 2.2.2, which takes into account all possible thermal resistances except the internal convection. The correlation is given in Eq. 16:

$$(h \cdot A)_{cond} = 2714.71 \cdot \dot{m}^{0.449} - 0.15 \cdot (42.52 - T_c)^2$$

Eq. 16

Interestingly, $(h \cdot A)_{cond}$ depends on the mass flow rate of refrigerant, $\dot{m}$, in addition to the condensation temperature T_c . This may be interpreted as a consequence of the inaccuracy of the correlations used for determining the convective heat transfer coefficients inside condenser tubes. The correlation in Eq. 16 is in good agreement with the experimental data, with a coefficient of determination $R^2 = 0.977$.

The wall temperature of condenser tubes ($T_{w,cond}$) varies linearly with the condensation temperature (T_c), according to the correlation of Eq. 17, which has a coefficient of determination $R^2 = 0.999$:

$$T_{w,cond} = -0.275 + 0.98436 \cdot T_c$$

Eq. 17

The empirical correlation for the effective thermal conductance of the evaporator, $(h \cdot A)_{evap}$, defined in Section 2.2.3, is shown in Eq. 18:

$$(h \cdot A)_{evap} = 256.89 + 1.0919 \cdot T_e$$

Eq. 18

It can be noticed from Eq. 18 that $(h \cdot A)_{evap}$ slightly depends on the evaporation temperature, T_e , which can be explained by a slight variation of air properties with the temperature. In this case, the relatively low value of the coefficient of determination $R^2 = 0.616$ is attributed to evaporator icing, which cannot be completely avoided and it may also vary depending on the operating conditions.

An empirical correlation was found for the heat transfer coefficient inside the evaporator tubes in the evaporation process, $\alpha_{int,67}$, as shown in Eq. 19:

$$\alpha_{int,67} = -711.4 + 1.77 \cdot 10^5 \cdot \dot{m} \quad \text{Eq. 19}$$

One can see from Eq. 19 that $\alpha_{int,67}$ increases linearly with the mass flow rate of refrigerant, with a coefficient of determination $R^2 = 0.993$.

The isentropic efficiency of the compression process (η_c) defined in Eq. 20, was also determined from experimental data. Its value remains almost constant irrespective of the operation conditions, and only a slight dependence of the condensation temperature (T_c) was obtained, as shown in Eq. 20:

$$\eta_c = 0.599 - 0.00088 \cdot T_c \quad \text{Eq. 20}$$

Therefore, a low value of the coefficient of determination $R^2 = 0.379$ is obtained, which confirms the slight dependence of η_c on T_c . However, all experimentally determined values of η_c differ from the values given by the correlation of Eq. 20 by less than 2%.

Another parameter required for compressor modelling is the fraction of energy that is lost by the compressor to the environment, ξ , defined in Eq. 4. The empirical correlation obtained for this parameter, shown in Eq. 21, only depends on the evaporation temperature, T_e .

$$\xi = 0.0854 \cdot \exp(-0.0265 \cdot T_e) \quad \text{Eq. 21}$$

The coefficient of determination of the correlation of Eq. 21 is $R^2 = 0.850$, and the deviations of the experimental values with respect to the empirical correlation are less than 10%.

Finally, an important empirical correlation required by the model is the one obtained for the expected degree of subcooling, shown in Eq. 22:

$$T_{SC} = 26.65 \cdot T_8^{-0.405} + 0.057 \cdot (T_{10} + 16.8)$$

Eq. 22

This correlation was obtained in terms of two of the input variables (T_8 and T_{10}), in the conditions of a constant degree of superheating (set at 9 K in all experiments). The coefficient of determination of the correlation shown in Eq. 22 is $R^2 = 0.978$, and the experimental values differ from the values given by the correlation by less than 0.2°C.

4.2 Numerical results

After performing the nine experiments as described in Section 3.2, the first task was to find the time intervals in which the system had been operating in steady or near steady states. Once steady-states were identified, the recorded values of the different experimental parameters were then averaged over some time period in order to reduce the fluctuations and the noise in the experimental data. For some of the experiments performed, more than one experimental point was selected at different time periods in order to increase the reliability of the data sets and to reduce possible errors associated to a particular measurement. A total of 15 experimental points have been selected for determining the empirical correlations used in the model. Besides these 15 experimental points, an additional subset of 9 experimental points was used for checking the accuracy of the model. The model was then run for the complete set of 24 experimental points, and the results were compared with the experimental data.

In Figure 7, the predictions of the model for some of the main parameters of the system are compared to the experimental values: low pressure (P_1), head pressure (P_2), gas temperature at evaporator exit (T_1), gas temperature at compressor discharge (T_2), liquid temperature at condenser exit (T_5), electrical power consumption of the compressor

($\dot{W}_c$), and degree of subcooling (T_{SC}). In each of these figures, the root-mean-square error (RMSE), and the mean relative error (MRE) are indicated as a way to quantify the robustness of model predictions, which were calculated taking into account only the additional subset of 9 experimental points.

(a)

(b)

(c)

(d)

(e)

(f)

(g)

Figure 7. Comparison between model predictions and experimental data for: (a) the low pressure, P_1 ; (b) the head pressure, P_2 ; (c) the refrigerant temperature at evaporator exit, T_1 ; (d) the refrigerant temperature at compressor discharge, T_2 ; (e) the refrigerant temperature at condenser exit, T_5 ; (f) the electrical power consumption of the compressor, $\dot{W}_c$; (g) the degree of subcooling, T_{SC}

One can see from Figure 7 (a) and (b) that the two pressures of the cycle are predicted by the model with high accuracy, with an RMSE of around 6.2 kPa and 18.3 kPa, respectively, and an MRE of 2.7% and 0.8%, respectively. Regarding the temperature predictions at the different points of the cycle, the most accurate values are obtained for the temperature at condenser outlet (T_5), with an RMSE value of 0.3°C, and an MRE of 0.9% (see Figure 7 (e)). By comparing Figure 7 (a) and (c), one can notice that temperature predictions at evaporator exit (T_1) are strongly related to the accuracy of predictions for the low pressure (P_1), as expected. The RMSE value of the predictions for T_1 temperature is 0.6°C, with an MRE value of 2.3%. The most difficult to predict is the temperature at compressor discharge (T_2), with an RMSE value of 1.9°C, and an MRE of 2.0%. From Figure 7 (d), it can be observed that in some cases the deviations with respect to the experiment can be as large as 3°C. This can be a consequence of having different levels of compressor overheating, depending on the time the compressor had been operating continuously before the measurement was made. The model is also able to predict the electrical power consumption of the compressor, as shown in Figure 7 (f). The predictions are in good agreement with the experiment, with an RMSE value of 14.1 W, and a MRE of 1.3%. Regarding the degree of subcooling (T_{SC}), it is very important to be able to predict it accurately, since it is one of the parameters used in the calculation algorithm on which the model is based. In Figure 7 (g), the predictions of the model are compared with the experiment, and one can see a good agreement, with a RMSE value of 0.2°C, and an MRE of 2.5%.

4.3 Error analysis

An important issue for model validation is to perform an analysis of the influence of the different errors that may affect the parameters of the model and its outputs. Therefore,

both an error sensitivity analysis of the variables of interest, and an error propagation analysis of the experimental data were performed.

First, a sensitivity analysis has been performed to determine how sensitive the main output variables are with respect to uncertainties in the empirical correlations. To perform such an analysis, the maximum deviation of each parameter with respect to the fitting curve was first determined, for a given experimental point ($T_8 = 23.7^\circ\text{C}$, $T_{10} = -20.4^\circ\text{C}$, $T_{SH} = 9.5^\circ\text{C}$). After that, each of the nine experimental correlations used in the model were modified by adding or subtracting the maximum deviation determined in the first step, while keeping all the other parameters as in the reference case. The model was then run for each of the combinations required for sensitivity analysis, and the relative deviations of the different output variables were calculated with respect to the reference case. The results of the sensitivity analysis are summarized in Table 3.

Table 3: Results of the error sensitivity analysis

Parameter and value		P_1 (kPa)	ΔP_1 (%)	P_2 (kPa)	ΔP_2 (%)	T_1 (°C)	ΔT_1 (%)	T_2 (°C)	ΔT_2 (%)	T_5 (°C)	ΔT_5 (%)	$\dot{W}_c$ (W)	$\Delta \dot{W}_c$ (%)	T_{sc} (°C)	ΔT_{sc} (%)
$\dot{D}_c$	0.001045	188.8		1512.0		-22.12		79.98		25.05		948.7		7.20	
	ref+0.38%	188.6	0.11	1512.4	0.03	-22.15	0.14	80.01	0.04	25.08	0.12	951.8	0.33	7.17	0.42
	ref-0.38%	189.0	0.11	1511.4	0.04	-22.10	0.09	79.93	0.06	25.06	0.04	945.5	0.34	7.17	0.42
$frac_{cond}$	0.7631	188.8		1512.0		-22.12		79.98		25.05		948.7		7.20	
	ref+8.26%	188.8	0.00	1507.2	0.32	-22.12	0.00	79.80	0.23	24.95	0.40	948.2	0.05	7.17	0.42
	ref-8.26%	189.0	0.11	1517.6	0.37	-22.10	0.09	80.16	0.23	25.21	0.64	949.8	0.12	7.18	0.28
$(hA)_{cond}$	326.4	188.8		1512.0		-22.12		79.98		25.05		948.7		7.20	
	ref+4.56%	188.8	0.00	1506.2	0.38	-22.12	0.00	79.76	0.28	24.91	0.56	948.1	0.06	7.18	0.28
	ref-4.56%	189.0	0.11	1518.4	0.42	-22.10	0.09	80.19	0.26	25.23	0.72	949.9	0.13	7.18	0.28
$T_{w,cond}$	31.83	188.8		1512.0		-22.12		79.98		25.05		948.7		7.20	
	ref+1.00%	188.8	0.00	1510.6	0.09	-22.12	0.00	79.93	0.06	25.03	0.08	948.5	0.02	7.18	0.28
	ref-1.00%	188.8	0.00	1513.0	0.07	-22.12	0.00	80.01	0.04	25.08	0.12	948.8	0.01	7.19	0.14
$(hA)_{evap}$	222.3	188.8		1512.0		-22.12		79.98		25.05		948.7		7.20	
	ref+17.81%	192.4	1.91	1515.4	0.22	-21.67	2.03	79.71	0.34	25.15	0.40	960.2	1.21	7.18	0.28
	ref-17.81%	183.4	2.86	1507.0	0.33	-22.81	3.12	80.40	0.53	24.93	0.48	930.9	1.88	7.19	0.14
$\alpha_{int,67}$	1028	188.8		1512.0		-22.12		79.98		25.05		948.7		7.20	
	ref+6.42%	189.2	0.21	1512.4	0.03	-22.07	0.23	79.95	0.04	25.07	0.08	950.0	0.14	7.19	0.14
	ref-6.42%	188.4	0.21	1511.4	0.04	-22.17	0.23	80.00	0.03	25.06	0.04	947.3	0.15	7.17	0.42
η_c	0.5706	188.8		1512.0		-22.12		79.98		25.05		948.7		7.20	
	ref+1.89%	188.6	0.11	1511.4	0.04	-22.15	0.14	78.67	1.64	25.04	0.04	930.4	1.93	7.19	0.14
	ref-1.89%	188.6	0.11	1512.2	0.01	-22.15	0.14	81.37	1.74	25.06	0.04	966.3	1.86	7.19	0.14
ξ	0.1975	188.8		1512.0		-22.12		79.98		25.05		948.7		7.20	
	ref+9.82%	188.8	0.00	1512.0	0.00	-22.12	0.00	79.98	0.00	25.05	0.00	972.2	2.48	7.20	0.00
	ref-9.82%	188.8	0.00	1512.0	0.00	-22.12	0.00	79.98	0.00	25.05	0.00	926.3	2.36	7.20	0.00
T_{sc}	7.2	188.8		1512.0		-22.12		79.98		25.05		948.7		7.20	
	ref+2.50%	188.8	0.00	1514.4	0.16	-22.12	0.00	80.07	0.11	24.96	0.36	948.9	0.02	7.35	2.08
	ref-2.50%	188.6	0.11	1509.2	0.19	-22.15	0.14	79.90	0.10	25.17	0.48	947.8	0.09	7.00	2.78

From Table 3, one can see that, even though the relative error of some of the empirical correlations can be quite large, the effect on the output variables is substantially reduced. For instance, the parameter with the highest relative error, $(hA)_{evap} = \pm 17.81\%$, produces a maximum deviation of the low pressure that is less than 3%, while the effect on the head pressure is even smaller (less than 0.4%).

Next, since all the experimental correlations are expressed in terms of parameters that are affected by experimental uncertainties, a study on the propagation of experimental uncertainties was performed [21]. In Table 4, the absolute and the relative uncertainties of the different parameters are shown, which were calculated for a given experimental point ($T_8 = 23.7^\circ\text{C}$, $T_{10} = -20.4^\circ\text{C}$, $T_{SH} = 9.5^\circ\text{C}$).

Table 4: Results of the error propagation analysis

Parameter	Absolute uncertainty	Relative uncertainty (%)
$\dot{D}_c$	$9.6 \cdot 10^{-6} \text{ (m}^3 \text{ s}^{-1}\text{)}$	0.92
$frac_{cond}$	0.0259 (-)	3.39
$(hA)_{cond}$	3.1 (W K ⁻¹)	0.95
$T_{w,cond}$	0.74 (°C)	2.32
$(hA)_{evap}$	0.25 (W K ⁻¹)	0.11
$\alpha_{int,67}$	24.6 (W m ⁻² K ⁻¹)	2.39
η_c	0.0007 (-)	0.12
ξ	0.0012 (-)	0.61
T_{SC}	0.058 (°C)	0.80

5. Conclusions

In this paper, a novel methodology is described for modelling a simple vapour compression refrigeration system, implemented in EES software. The model only requires three known inputs: the ambient air temperature (T_8), the indoor (cold room) air temperature (T_{10}), and the degree of superheating (T_{SH}), which is regulated by the expansion valve and is kept at a constant value in all experiments. Depending on the values of the input parameters, a calculation algorithm is used in the model to determine the operating point of the system, i.e. the evaporation and condensation pressures, the temperatures at different points of the thermodynamic cycle, and compressor power consumption. The calculation algorithm is based on iterative loops, and is structured in three main blocks, in such a way to reduce the computational time and resources. In the first block a rough value for the upper limit of the condensation pressure is inferred, which is then used in the second block as initial value for the iteration loops. The second block determines upper values for the evaporation and condensation pressures with a higher accuracy than block 1, which are then used in block 3 as initial values for determining the solution within the desired accuracy. The solution is found when the calculated degree of subcooling equals the expected value, and the cycle gets “closed”, meaning that the starting point and the ending point coincide, within a certain error interval.

The model requires some of the features of the system components, in order to be able to describe their operation. Both heat exchangers are modelled using the ε -NTU method, while in the case of the compressor, the volumetric displacement curve determined from the data provided by the manufacturer is used. However, some of the features of these components are not known, or cannot be determined with acceptable accuracy. For this reason, some empirical correlations were derived and implemented in the model. Thus, the model is system dependent, i.e. for a different system, new empirical correlations must be derived. Also, all the correlations were obtained for a fixed degree of superheating in all experiments (around 9 K). If the superheating were set to a different value, the correlations would probably change. The methodology described here can be applied to any similar system, by adapting the model to the specific features of the main system components.

The results show that the model can predict with good accuracy some of the main parameters of the system, such as evaporation and condensation pressures, temperature at different points of the thermodynamic cycle, and compressor consumption. Moreover, an error analysis has been performed, including error propagation and sensitivity analysis, showing that even though the relative error of some of the empirical correlations can be significant, its effect on the output variables is reduced. This means that simpler equations (e.g. linear equations) could be used in most of the empirical correlations instead of accurate but more complex ones in future researches intended to apply the methodology described here to any other similar refrigeration systems. The model allows predicting the performance of the system in different operating conditions, and can be useful for detecting any deviation of the experimental facility from the expected functioning, which can alert on some malfunction of the system or inadequate refrigerant charge level.

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