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# A dynamic model of a shell-and-tube condenser operating in a vapour compression refrigeration plant

R. Llopis [a,*], R. Cabello [a], E. Torrella [b]

[a] Department of Mechanical Engineering and Construction, Campus de Riu Sec, Jaume I University, E-12071 Castellón, Spain
[b] Department of Applied Thermodynamics, Camino de Vera 14, Polytechnic University of Valencia, E-46022 Valencia, Spain

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**Abstract**

This work presents a mathematical model of a shell-and-tube condenser based on mass continuity, energy conservation and heat transfer physical fundamentals, whose methodology can be easily adapted for modelling any type of condenser. The model is formulated as a combination of control volumes that represents all the refrigerant states in the condenser and the liquid receiver function, which is carried out by the condenser of the experimental plant. Model validation is performed by using steady-state data and transient tests from an experimental vapour compression plant; the prediction error of the model is lower than 5% and a good representation of the dynamic performance of the condenser is achieved. A theoretical comparison involving the importance of the dynamic responses of the evaporator and the condenser at the plant is also presented.


*Keywords:* Condenser; Dynamic model; Shell-and-tube; Refrigeration

## 1. Introduction

Due to the amount of energy required by the refrigeration sector, which is estimated to be around 15% of the total energy consumption in the world [1], the scientific community is working hard to optimise the operation of refrigeration plants. To do so, mathematical models constitute one of the best tools both for analysing and for controlling the systems. Researchers are mainly using two types of strategies: statistical models [2,3], based on mathematical routines that obtain the formulation of the system from experimental data, or physical-based models, which approximate the behaviour of the system by using physical relations. This latter group is the one that provides more information about the system behaviour, since all the operating variables are considered, being it represented by the finite-elements models [4–6]. But finite-element models are limited by the time required to solve the mathematical routines, and they are not efficient when a dynamic representation of the system is required, for example, when they are used for fault detection and diagnosis or dynamic control of the cycle. In this case, research is focused on obtaining simplified models that work with low calculation times, being they represented by models that consider average properties in the control volumes used to model the system [7–10].

In this work a dynamic mathematical model of a shell-and-tube condenser operating in a vapour compression refrigeration plant is presented and validated. The model is formulated from mass continuity, energy conservation and heat transfer physical fundamentals by using a lumped-parameter formulation for the condenser that is similar to the ones presented by Deng S. [9] and Finn P. et al. [10], but with some differences in the selection of control volumes and including the refrigerant dynamics in a simplified way. These differences, in addition to the small integration time-step selected, make the model capable of representing the quick dynamics of the condenser without using complicated formulation. It is therefore a good tool to understand how the condenser operates and its influence on the complete vapour compression system. This effect is analysed by comparing the theoretical response of a previously presented evaporator model [11] with the one obtained using the condenser model.

* Corresponding author. Tel.: +34 964728136; fax: +34 964728106.
*E-mail address:* rllopis@emc.uji.es (R. Llopis).

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**Nomenclature**

| | | |
|---|---|---|
| $c_p$ | specific heat capacity | $\mathrm{J\,kg^{-1}\,K^{-1}}$ |
| $h$ | specific enthalpy | $\mathrm{J\,kg^{-1}}$ |
| $M$ | mass | kg |
| $\dot{m}$ | mass flow rate | $\mathrm{kg\,s^{-1}}$ |
| $P$ | pressure | Pa |
| $\dot{Q}$ | heat flux | $\mathrm{W\,m^{-2}}$ |
| $T$ | temperature | K |
| $t$ | time | s |
| $u$ | specific internal energy | $\mathrm{J\,kg^{-1}}$ |
| $V_C$ | condensing zone total volume | $\mathrm{m^3}$ |
| $V_S$ | de-superheating zone total volume | $\mathrm{m^3}$ |
| $V_{SB}$ | subcooling zone total volume | $\mathrm{m^3}$ |

*Greek symbols*

| | | |
|---|---|---|
| $\rho$ | density | $\mathrm{kg\,m^{-3}}$ |
| $\alpha$ | convection heat transfer coefficient | $\mathrm{W\,m^{-2}\,K^{-1}}$ |

*Subscripts*

| | |
|---|---|
| env | environment |
| $i$ | equations referred to control volumes 1 and 2 |
| in | inlet |
| $l$ | saturated liquid |
| $m$ | metal |
| out | outlet |
| $r$ | refrigerant |
| shell | condenser shell |
| $S$ | superheated refrigerant |
| $SB$ | subcooled refrigerant |
| $v$ | saturated vapour |
| $w$ | secondary fluid |
| 1 | de-superheating refrigerant zone |
| 2 | condensing refrigerant zone |
| 3 | subcooling refrigerant zone |

## 2. Mathematical model of the condenser

The condenser considered in this work is of the shell-and-tube type, in which the refrigerant flows through the shell and the secondary fluid inside the tubes; it also has the function to regulate the refrigerant mass variations in the facility, therefore the model was developed to consider both the whole heat exchange process and the liquid receiver function. Other authors [9,10] have included the liquid receiver in the model formulation but not in the condenser model because it was a different element in the modelled facility. In this model, however, it is included because this function is performed by the condenser and it is needed to complete the differential equation system. Classical equations for mass continuity, energy conservation and heat transfer, in a one-dimensional formulation are used to obtain a simplified dynamic model of the element, which represents the refrigerant mass and energy storages, as well as the thermal capacities of the shell and tubes.

Each state of the refrigerant in the condenser is modelled by using a control volume (see Fig. 1) whose properties are considered to be constant at each time-step. The selected control volumes in the refrigerant region are as follows: one for the de-superheating zone, another for the condensing one and a last one for the subcooling region. For the physical characteristics of the condenser, which is described in the following section, only the de-superheating and the condensing zones are cooled with the internal tubes of the condenser. Subcooling of the refrigerant is mainly caused by heat transfer through the shell, so a secondary fluid control volume is only considered in the de-superheating and condensing zones. Furthermore, both the de-superheating and the subcooling refrigerant zones transmit heat to the shell, which is treated as a single control volume.

The inlet mass flow rate, pressure and temperature in the refrigerant and the inlet mass flow rate and temperature in the secondary fluid are taken as model inputs; the environmental temperature is also taken into account. The outlet refrigerant enthalpy, the outlet secondary fluid temperature and the total heat transfer rates are considered as model outputs in order to validate the model.

The assumptions made in order to formulate the model are: pressure drop in refrigerant through the condenser is neglected; kinetic and potential energy variations are not taken into account; in the condensing zone all the refrigerant is considered as saturated vapour and the heat transferred is only used to

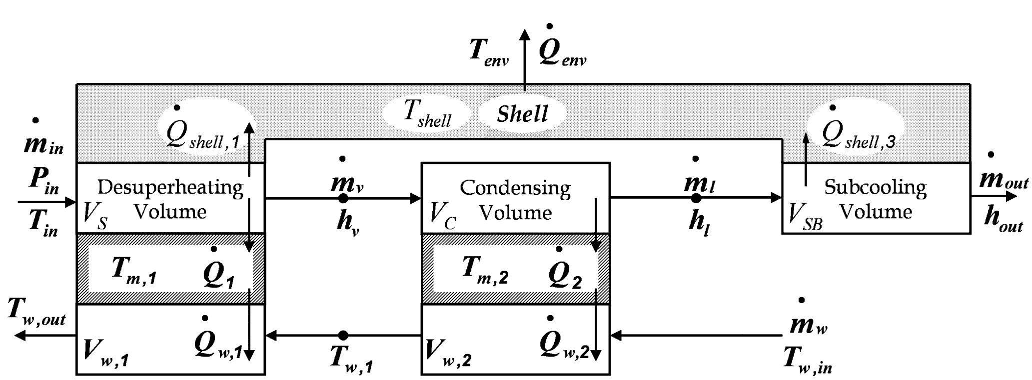


Fig. 1. Conceptual diagram of the condenser.

condense it, and it is assumed that refrigerant subcooling only occurs through the shell; the dynamics of the secondary fluid are neglected.

The equations presented below are those used to model each control volume in Fig. 1.

### 2.1. De-superheating zone

The inlet refrigerant enthalpy, $h_{\text{in}}$, is evaluated from the model inputs and it is used to apply the mass continuity (1) and energy conservation (2) equations that are expressed as a function of model inputs and the average properties of the refrigerant. They are expanded using the total volume of refrigerant in this zone as state variable, following the same procedure as in the evaporator model presented by Llopis R. et al. [11].

$$\rho_S \cdot \frac{\mathrm{d}V_S}{\mathrm{d}t} + \frac{\mathrm{d}\rho_S}{\mathrm{d}t} \cdot V_S = \dot{m}_{\text{in}} - \dot{m}_v \tag{1}$$

$$(\rho_S \cdot u_S) \cdot \frac{\mathrm{d}V_S}{\mathrm{d}t} + \left(u_S \cdot \frac{\mathrm{d}\rho_S}{\mathrm{d}t} + \rho_S \cdot \frac{\mathrm{d}u_S}{\mathrm{d}t}\right) \cdot V_S = \dot{m}_{\text{in}} \cdot h_{\text{in}} - \dot{m}_v \cdot h_v - \dot{Q}_1 - \dot{Q}_{\text{shell},1} \tag{2}$$

Zhukauskas' correlation [12] for an ideal bundle of tubes is used to evaluate the convection heat transfer to the tubes, taking into account the deviations from the ideal case due to the low number of tube rows in the condenser. In this case, the average superheated vapour temperature and the adjacent metal temperature $T_{m,1}$ are used to obtain the convection heat transfer coefficient. The heat transferred from the refrigerant to the shell is computed by evaluating the convection coefficient using both the same correlation and the same velocity pattern, but now with the shell temperature.

### 2.2. Condensing zone

Treatment of the condensing zone is based on the assumption that the refrigerant contained in this volume is saturated vapour and the heat transferred is only used to condense it. So the mass (3) and energy (4) equations are applied using saturated vapour properties and by considering the outlet refrigerant as saturated liquid.

$$\rho_v \cdot \frac{\mathrm{d}V_C}{\mathrm{d}t} + \frac{\mathrm{d}\rho_v}{\mathrm{d}t} \cdot V_C = \dot{m}_v - \dot{m}_l \tag{3}$$

$$(\rho_v \cdot u_v) \cdot \frac{\mathrm{d}V_C}{\mathrm{d}t} + \left(u_v \cdot \frac{\mathrm{d}\rho_v}{\mathrm{d}t} + \rho_v \cdot \frac{\mathrm{d}u_v}{\mathrm{d}t}\right) \cdot V_C = \dot{m}_v \cdot h_v - \dot{m}_l \cdot h_l - \dot{Q}_2 \tag{4}$$

In order to simplify the model, Dhir and Lienhard's correlation [13] for laminar condensation over circular tubes is used. The decrease in the convection heat transfer coefficient due to the rows of tubes is neglected, in accordance with Karlsson T. et al. [14] who state that this decrease is not important and the heat transfer coefficient is almost constant for the different rows of tubes. To evaluate this coefficient the saturated vapour refrigerant temperature and the metal temperature $T_{m,2}$ are used.

### 2.3. Subcooling zone

Due to the physical construction of the shell-and-tube condenser (refrigerant flowing through the shell and secondary fluid inside the tubes, in which there is no physical contact between the subcooled liquid and the internal tubes for a correct refrigerant mass charge), subcooling is modelled by considering only the heat exchange through the shell and any possible subcooling with the tubes is neglected. Furthermore, the condenser also has the function to regulate the refrigerant mass variations in the cycle, so this control volume contains the greater part of the refrigerant mass in the condenser. Again, the mass (5) and energy (6) equations are applied considering the average properties of the subcooled refrigerant.

$$\rho_{SB} \cdot \frac{\mathrm{d}V_{SB}}{\mathrm{d}t} + \frac{\mathrm{d}\rho_{SB}}{\mathrm{d}t} \cdot V_{SB} = \dot{m}_l - \dot{m}_{\text{out}} \tag{5}$$

$$(\rho_{SB} \cdot u_{SB}) \cdot \frac{\mathrm{d}V_{SB}}{\mathrm{d}t} + \left(u_{SB} \cdot \frac{\mathrm{d}\rho_{SB}}{\mathrm{d}t} + \rho_{SB} \cdot \frac{\mathrm{d}u_{SB}}{\mathrm{d}t}\right) \cdot V_{SB} = \dot{m}_l \cdot h_l - \dot{m}_{\text{out}} \cdot h_{\text{out}} - \dot{Q}_{\text{shell},3} \tag{6}$$

To simulate the subcooling of the refrigerant, the heat transferred to the shell is evaluated by applying McAdams' correlation [15] for natural convection modified by Goldstein R.J. et al. [16] for non-flat elements, using the average refrigerant temperature and the shell temperature $T_{\text{shell}}$.

In Deng's work [9] the total volumes are obtained by considering the difference between the total volume of the condenser but in this case, since vapour zones dynamics are considered, the differential equation system obtained with Eqs. (1) to (6) is represented by seven state variables ($\frac{\mathrm{d}V_S}{\mathrm{d}t}$, $\frac{\mathrm{d}V_C}{\mathrm{d}t}$, $\frac{\mathrm{d}V_{SB}}{\mathrm{d}t}$, $\dot{m}_v$, $\dot{m}_l$, $\dot{m}_{\text{out}}$, $h_{\text{out}}$) and only six equations to solve. Therefore, to complete the system, the same reasoning is applied but in this case using the differential equation that represents the invariability of the total volume of the condenser (7).

$$\frac{\mathrm{d}V_S}{\mathrm{d}t} + \frac{\mathrm{d}V_C}{\mathrm{d}t} + \frac{\mathrm{d}V_{SB}}{\mathrm{d}t} = 0 \tag{7}$$

### 2.4. Secondary fluid side

The volumes of the secondary fluid side ($V_{w,1}$, $V_{w,2}$) are modelled by considering that the net convection heat exchange causes a change in the stored energy of the fluid (8) induced by the heat transferred from the metal control volumes. All the dynamics in the secondary fluid side are neglected, since the total density variation of the secondary fluid is small over a modest temperature change.

$$\dot{Q}_{w\_i} = \dot{m}_w \cdot c_{p,w\_i} \cdot (T_{w,\text{out}\_i} - T_{w,\text{in}\_i}) \tag{8}$$

The heat exchange from the tubes to the secondary fluid is computed by applying Gnielinski's correlation [17] for turbulent flow inside straight smooth tubes using the average secondary fluid temperature and the adjacent metal temperature.

### 2.5. Metal control volumes

These control volumes are one of the main parts that influence the dynamic behaviour of the condenser due to energy storages, their dynamics being represented by (9) for the metal tube control volumes and by (10) for the shell.

$$M_{m\_i} \cdot c_{p,m\_i} \cdot \frac{dT_{m\_i}}{dt} = \dot{Q}_i - \dot{Q}_{w\_i} \quad (9)$$

$$M_{\text{shell}} \cdot c_{p,\text{shell}} \cdot \frac{dT_{\text{shell}}}{dt} = \dot{Q}_{\text{shell},1} + \dot{Q}_{\text{shell},3} - \dot{Q}_{\text{env}} \quad (10)$$

*2.6. Heat transfer to the environment*

It is more important to consider heat transfer to the environment in the case of the condenser than in the evaporator since in most refrigerating applications the condenser is not isolated and this heat transfer cannot be neglected.

In this case, the heat transferred to the environment is evaluated by using a correlation of our own that was obtained from experimental data of the plant (11) and works well inside the range in which the condenser was validated.

$$\alpha = 4.723 \cdot (T_{\text{shell}} - T_{\text{env}})^{0.25} \quad (\mathrm{W\,m^{-2}\,K^{-1}}) \quad (11)$$

*2.7. Refrigerant and secondary coolant modelling*

The model is able to work with both pure and blend refrigerants, and their properties as well as those of the secondary fluid (in our case water) are evaluated by using the Refprop dynamic libraries [18].

## 3. Model simulation program

The whole model, programmed in Fortran 90, consists of a set of ten first-order differential equations that are solved by explicit fourth-order Runge–Kutta methods. Fig. 2 presents the working structure of the program. The dynamic response calculation starts with the model initialization using the first values of the input data. With these values and considering steady-state equations for the heat transfer rates in the control volumes, the initial state of all the variables of the model (metal temperatures, heat transfer coefficients, refrigerant outlet enthalpy, secondary fluid outlet enthalpy and total volumes of each zone) are obtained using a steady-state convergence criterion. The convergence condition is based on the model outlet variables (namely, the refrigerant outlet enthalpy and the secondary fluid outlet temperature). In the initialization the heat transfer rates from the refrigerant to the metal control volumes are equalled to the heat transfer rates from the metal control volume to the secondary fluid in order to initialize the model. Once the model has been initialized, the dynamic model evaluates the temporal evolution of the state variables using a time-integration-step of 0.2 seconds, being the differential equations solved in each step using the model outputs of the step before. This small integration step allows the model to represent in more detail the quick dynamics of the condenser. The average program time used to solve the problem is approximately 65 seconds per hour of test in a Pentium IV at 2.4 GHz.

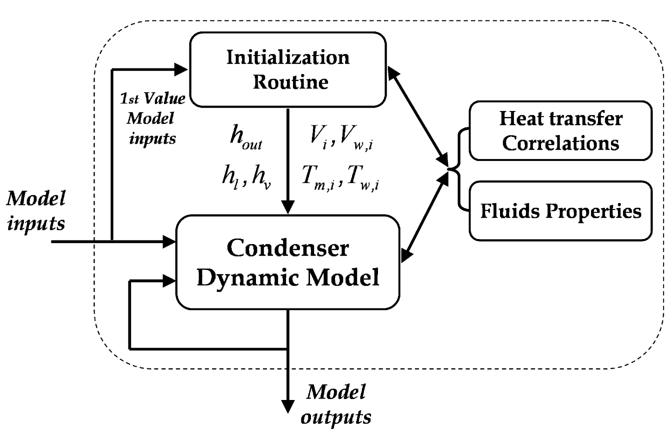


Fig. 2. Schematic program diagram.

## 4. Validation

Validation of the dynamic model is achieved using data from a test campaign performed with an experimental vapour compression plant previously presented by the authors [11,19]. This consists of a single-stage vapour compression plant with an open-type compressor driven by a 4 kW electric motor, a shell-and-tube evaporator (1–2), a shell-and-tube condenser (1–2) and a thermostatic expansion valve.

The condenser used to apply and validate the dynamic model is a shell-and-tube condenser (1–2) in which refrigerant is flowing through the shell and water inside the internal tubes. The characteristics of the main elements are as follows: 20 × 0.8 m long horizontal cooper tubes with internal and external diameters of $13 \times 10^{-3}$ m and $16 \times 10^{-3}$ m respectively, with 26 circular fins per inch with that have a diameter of $17.62 \times 10^{-3}$ m. The condensing heat transfer coefficient is evaluated by considering the external diameter of the tubes. The shell, made of cast iron, has an internal diameter of 0.183 m, an external diameter of 0.196 m and is 1 m long. The condenser has no internal baffles and is no isolated.

The refrigerant HCFC-22 was selected for the tests. The model was validated both by using steady-state data (see validation range in Table 1) and by transient tests to ensure the stability of the model within the whole operating range in the first case and to validate the dynamic response of the model in the second.

The calculated versus the measured condensing net heat exchanges are presented in Fig. 3, both for the refrigerant and for the secondary fluid. The absolute maximum prediction er-

Table 1
Steady-state test range. Measured values

| | *Min* | *Max* |
|---|---|---|
| Condensing pressure (bar) | 13.1 | 19.5 |
| Refrigerant mass flow rate ($\mathrm{kg\,s^{-1}}$) | 0.051 | 0.078 |
| Reheating degree at condenser inlet (K) | 32.6 | 41.0 |
| Subcooling degree at condenser outlet (K) | 2.2 | 4.2 |
| Sec. fluid mass flow rate ($\mathrm{kg\,s^{-1}}$) | 0.24 | 0.48 |
| Sec. fluid inlet temperature (K) | 286.8 | 302.1 |
| Sec. fluid temp. increment in condenser (K) | 5.4 | 11.2 |
| Condenser refrigerant power (kW) | 10.4 | 16.3 |
| Compressor speed (rpm) | 405 | 560 |

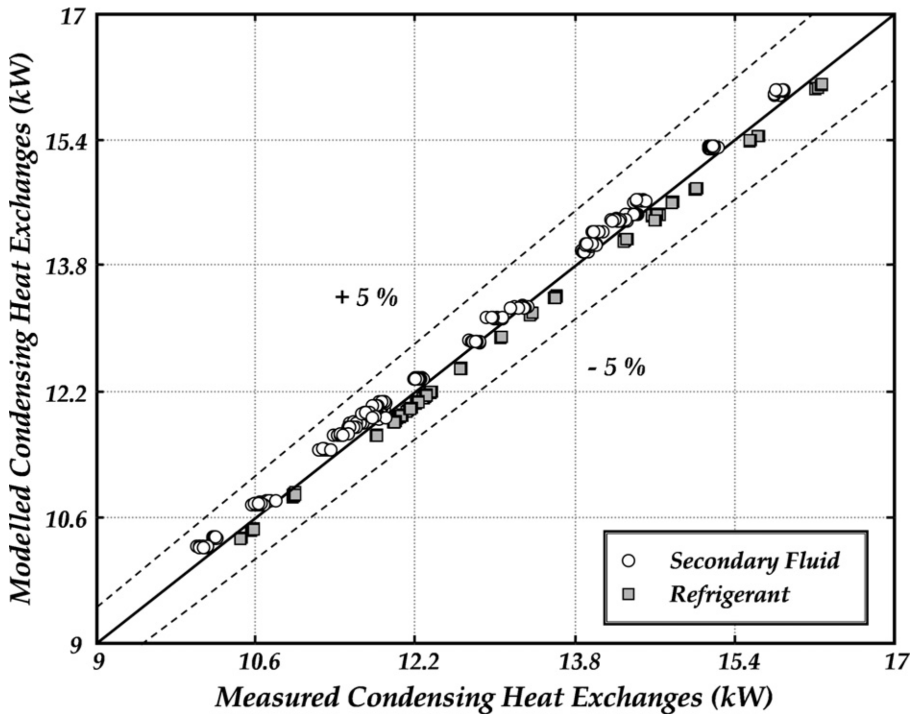


Fig. 3. Predicted vs. measured powers in condenser.

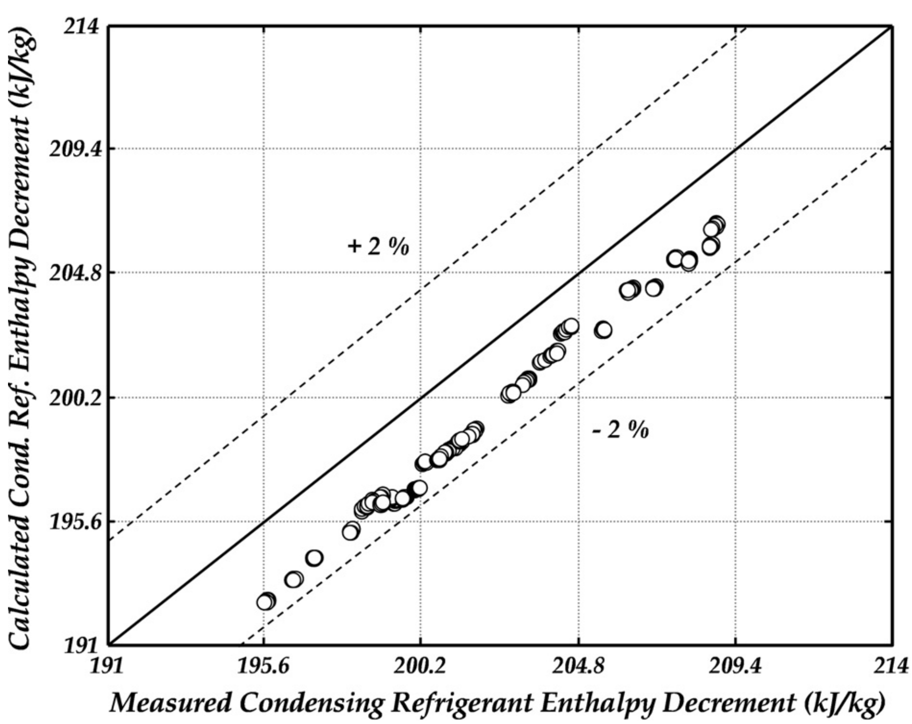


Fig. 4. Predicted vs. measured refrigerant enthalpy decrement.

ror over the entire test range for the secondary fluid is 1.30% and for the refrigerant 1.66 %.

The output properties of the fluids are also validated, being they expressed in this case as an enthalpy decrement in the refrigerant (Fig. 4) and as a temperature increment in the secondary fluid (Fig. 5). The predicted change in refrigerant enthalpy is always less than the measured one, but the prediction error is less than 2% of the total enthalpy change. In the case of the secondary fluid, the prediction error is below 5% of the total temperature change.

In order to ensure an adequate prediction of the dynamic performance by the model, it was validated under the main transient processes that the condenser could experience, namely, compressor rotation speed variations (Fig. 6); secondary fluid

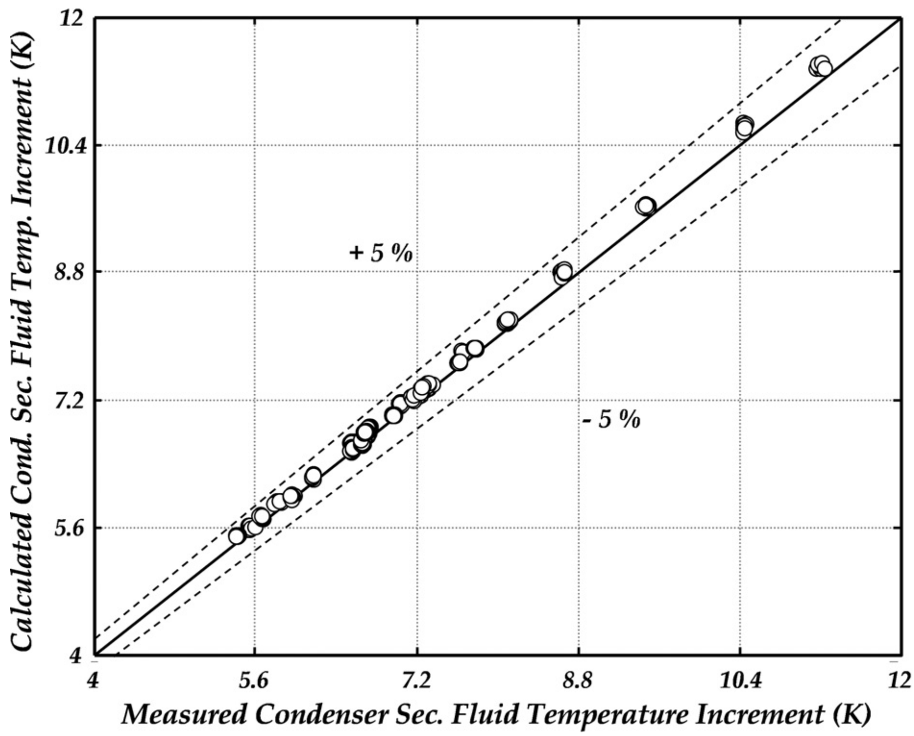


Fig. 5. Predicted vs. measured sec. fluid temperature increment.

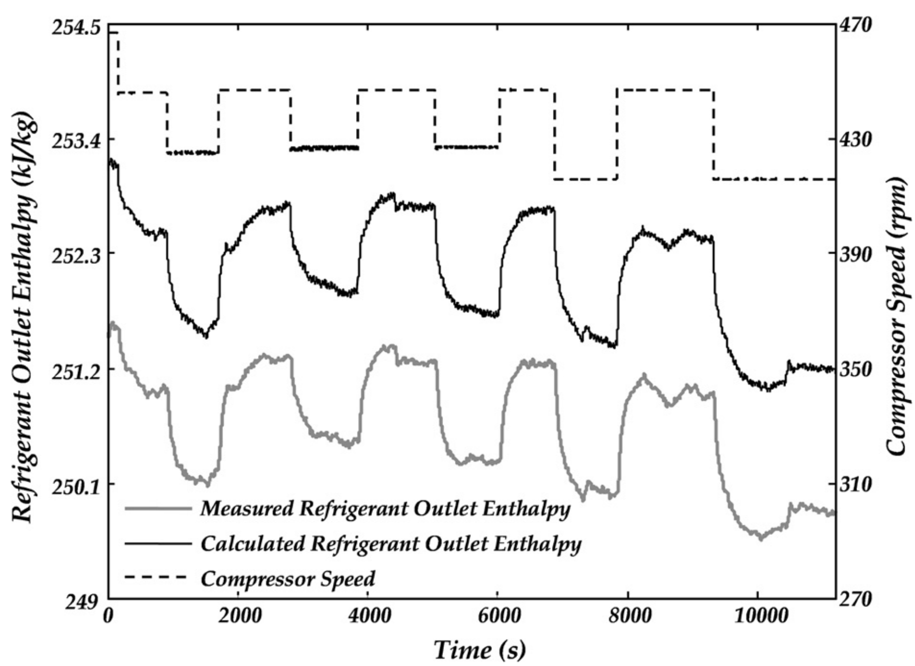


Fig. 6. Predicted vs. measured refrigerant outlet enthalpy. Compressor speed variation test.

inlet temperature changes (Fig. 7); and mass flow rate variations in the secondary fluid (Fig. 8).

The dynamic response of the refrigerant outlet enthalpy caused by compressor speed changes (Fig. 6) represents the behaviour of the refrigerant leaving the condenser with a high degree of accuracy, although there is a constant deviation of the enthalpy value. This position error is not important if the objective of the model is to analyse the global dynamic behaviour of the condenser. The dynamic response of the secondary fluid outlet temperature (Fig. 7) is also good although the model presents an initial oscillation that is not present in the measured response.

Fig. 8 shows the evolution of the condenser powers in a secondary fluid temperature variation test. The agreement between the calculated and the measured heat transfer exchanges is good but, according to Figs. 7 and 8, the modelled inertia of the sec-

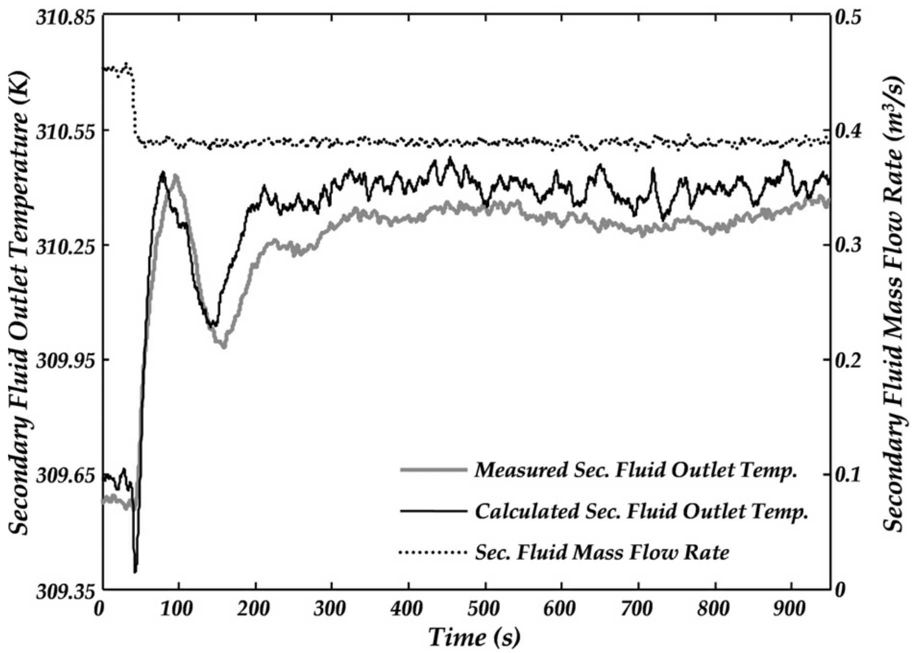


Fig. 7. Predicted vs. measured secondary fluid outlet temperature. Secondary fluid mass flow variation test.

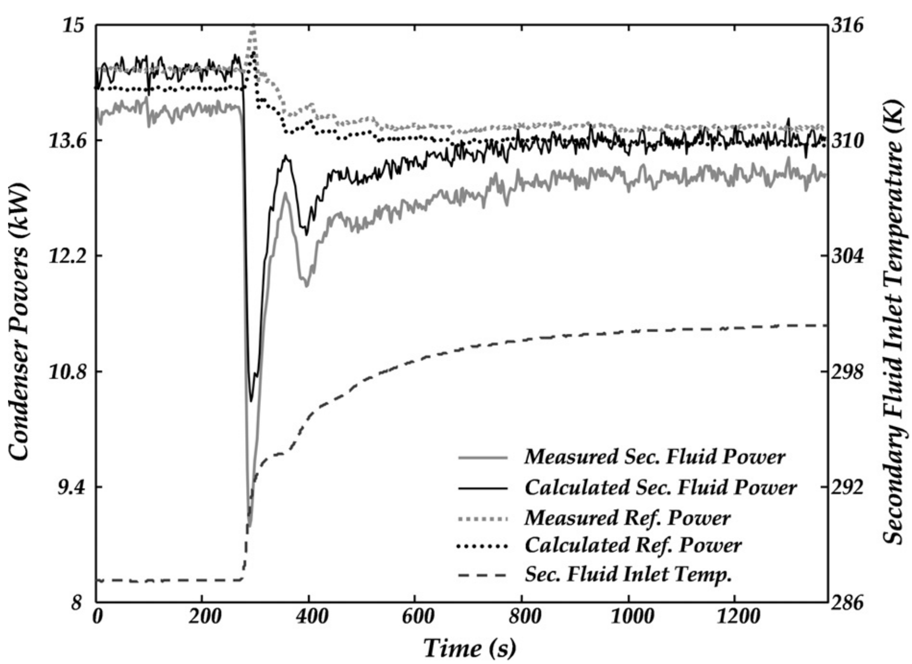


Fig. 8. Predicted vs. measured condenser heat exchanges. Secondary fluid inlet temperature variation test.

ondary fluid is less than the actual one because the dynamic response of the model is quicker. We assume that this deviation is caused by omitting in the model the inertia of some thermal storage elements that are not in contact with the refrigerant, such as the condenser covers. Fig. 8 shows the difference between the dynamic response of the refrigerant with respect to that of the secondary fluid, being the response of the secondary fluid slower than of the refrigerant one, which is the opposite to what could have been expected due to the presence of subcooled liquid in the condenser.

## 5. Dynamic responses of the condenser and evaporator

A dynamic model should only represent the main or slow responses of the element, since they determine its global dynamics [20]. Therefore, in order to obtain a simplified and useful model of a complete vapour compression system only the slowest processes should be taken into account, that in refrigerating cycles are usually represented by elements that contain refrigerant in liquid form. In this section the dynamic responses of the evaporator, whose model has been previously validated [11], and the condenser of the plant, whose validation is pre-

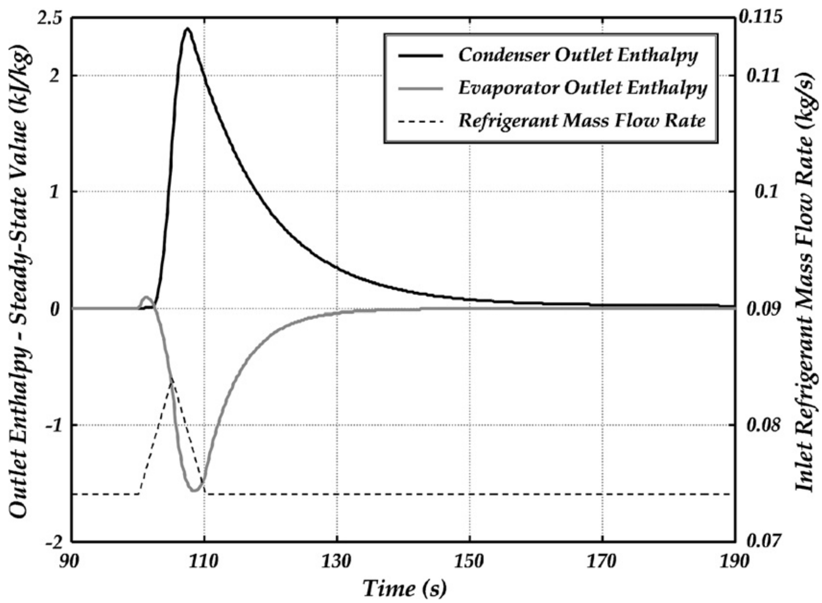


Fig. 9. Simulated heat exchanger dynamic responses when faced with a change in the inlet mass refrigerant flow rate.

sented in this work, are analysed. To analyse response of the heat exchangers in a vapour compression plant is not easy since the output conditions of one element influence the input conditions of the other. Thus, one way to compare the dynamics of the heat exchangers is by using mathematical models that represent the element.

The dynamic responses both of evaporator and condenser models when faced with a simulated sudden change in the inlet refrigerant mass flow are presented in Fig. 9. To obtain the response of Fig. 9, a steady-state of the vapour compression plant was selected and used to initialize the models. Once the models had been initialized, the model input parameters were kept constant and at time 100 s the inlet refrigerant mass flow rate was incremented in the models (represented with dashed line).

In Fig. 9, it can be observed that the dynamics of the condenser are slower than those of the evaporator; the condenser response takes 28.3 s to return to a value of a 17% of the peak value, whereas that of the evaporator only takes 19.3 s. The physical meaning of the responses shown in Fig. 9 is that the element that influences more the global dynamic response of the vapour compression plant, that is the slowest dynamic element, corresponds to the condenser, mainly because it contains the higher proportion of liquid refrigerant in the plant since it performs the liquid receiver function in the plant. But the evaporator dynamics cannot be neglected in order to obtain an accurate dynamic model of the refrigeration plant, since there is no large time variation with respect to those of the condenser, in other words, the amount of liquid refrigerant in the evaporator is similar to those contained in the condenser.

## 6. Conclusions

A dynamic model of a condenser constructed from physical equations has been developed and validated in this work. The model takes into account the main dynamics in the element, including mass storages and thermal capacities in the refrigerant side, and considers all the properties for fluids and materials constant in each control volume at each time step, while they change step by step.

The model was validated over a wide operating range in a vapour compression refrigeration plant (working with the refrigerant HCFC-22) by using steady-state data and several transient tests that include compressor speed, secondary fluid inlet temperature and refrigerant mass flow variations. A discrepancy of less than 5% was observed with respect to the steady-state measurements and good agreement with the dynamic responses of the element was also noted. The disadvantage of the model is that only average properties can be analysed but it can be used to model this type of heat exchangers if all that is needed is a good representation of the fluids outlet conditions or the heat transfer rates, as occurs in fault detection and diagnosis or dynamic control.

A theoretical comparison between the dynamic response of the condenser and the evaporator of the plant is also presented. It is shown that the evaporator response is quicker than that of the condenser, but this cannot be neglected to obtain a simplified model of a complete refrigeration vapour compression system.

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