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Dynamic model of a shell-and-tube condenser. Analysis of the mean void fraction correlation influence on the model performance

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ABSTRACT

A moving-boundary dynamic model of a shell-and-tube condenser is presented. Within this approach, the mean void fraction is a relevant parameter which is obtained, in this work, using different correlations proposed in the literature for the flow pattern analyzed. In order to evaluate the performance of the model with each different mean void fraction correlation, a set of experimental tests using R134a as working fluid, varying the main operating variables (refrigerant mass flow rate, secondary fluid mass flow rate and inlet temperature), are performed. The model performance is analyzed from the system model outputs, namely, condensing pressure and refrigerant and secondary fluid outlet temperatures. The results, comparing model predictions and experimental data, show the great influence of the mean void fraction correlation on the model predictions with noticeable discrepancies depending on the correlation used. It is also observed that the model using the homogeneous correlation frequently provides acceptable results in all the tests analyzed, although the most appropriate correlation depends on the transient characteristics.

1. Introduction

Dynamic models of refrigeration facilities based on vapor compression cycles can be useful tools to predict system behavior during transients and, in this way, can be used to design efficient controllers (optimizing energy consumption and performance) and minimize time and expense of experimentation [1]. So, although stationary models can be used for systems and components design and to simulate specific operating situations, they are not suitable enough to correctly describe the system response due to changes in operating conditions. This makes dynamic models appropriate for intelligent control design and transients simulation [2].

A review of papers with different approaches to transient modeling of both individual components and complete systems is given by Bendapudi and Braun [3], who summarize the adopted methodologies and their applicability to chiller systems. In these models, the dominant dynamics are typically those of the evaporator and condenser, which are the most complex parts, and the description of their dynamic behavior is generally performed by means of two approaches, namely, finite-volume distributed-parameter (FV) and moving-boundary lumped parameter (MB) models, although techniques which are a hybrid of the two aforementioned approaches are also used.

The MB approach was pioneered by Wedekind et al. [4]. They proposed to use a mean void fraction (MVF) to predict transient phenomena associated with two-phase evaporating and condensing flows. The main advantage of the moving-boundary models is that they require significantly less computation time than the finite-volume approach. So, although the moving-boundary approach utilizes lumped characteristics for each of the control volumes, which could lead to lower accuracy [1], it is faster than FV models while maintaining the accuracy within a reasonable limit [2,3,5]. This execution speed, along with the relatively simplicity associated with the lumped form, makes this approach a useful tool for control and diagnosis purposes.

As mentioned before, within the MB approach, the MVF is a key physical value for determining important flow parameters in two-phase flow such as average fluid density, viscosity and relative average velocity of the two phases. However, some authors take a constant value for this parameter [6]. The MVF can be obtained by means of different correlations, which can be either for internal or external flow. In the literature one can find research works focused on obtaining and evaluating empirical and semi-empirical correlations to predict void fraction and frictional pressure drops [7–11].

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These works have also been conducted on in-line and staggered tube arrangements, for upward, downward and side-to-side flows. In some works, it is stated that simulation results show that time constants may change significantly with changes in the average void fraction [12].

Therefore, in this work, we propose an MB dynamic model of a shell-and-tube condenser where the MVF correlation used can be changed in order to analyze the influence of the MVF correlation on the model performance, comparing the predictions obtained with experimental data using MVF correlations frequently mentioned in the literature. The experimental data used consists of transients obtained varying the working conditions in a wide range, and being the MVF correlations used derived from different local void fraction expressions which allow a relatively easy analytical or numerical integration. Furthermore, whereas many researchers analyze either the upward flow or the downward flow inside tubes, there are much less works about two-phase flow in external flow in baffled shell-and-tube condensers, being this our case of study.

The rest of the paper is organized as follows. In Section 2, the proposed MB dynamic model and the different mean void fraction correlations used are presented. In Section 3, the experimental test bench and the tests performed are briefly described. Section 4 shows the comparison between the experimental data and the model predictions, using the different MVF correlations, and analyzes the influence of the MVF correlations on the model performance. Finally, Section 5 summarizes the main conclusions of the work.

2. Model description

The proposed condenser model takes four input variables, namely, refrigerant mass flow rate ($\dot{m}_{\mathrm{r}}$), condensing water flow rate ($\dot{V}_{\mathrm{W}}$), condensing water inlet temperature ($T_{\mathrm{w,ic}}$), and refrigerant inlet enthalpy (h_{ic}). The model direct outputs are: length of the superheating and condensing zones, ($L_{1\mathrm{c}}$ and $L_{2\mathrm{c}}$), condensing pressure (P_{c}), refrigerant outlet enthalpy (h_{oc}), and tube wall temperatures ($T_{\mathrm{t,1c}}$,$T_{\mathrm{t,2c}}$,$T_{t,3c}$) in the superheating zone, 1c, condensing zone, 2c, and subcooling zone, 3c, respectively. From these direct outputs one can straightforward derive the other measurable outputs: refrigerant outlet temperature ($T_{\mathrm{r,oc}}$) and condensing water outlet temperature ($T_{\mathrm{w,oc}}$). The general structure of the proposed model is shown in Fig. 1.

In the following, we present the condenser model and the mean void fraction correlations used in this work.

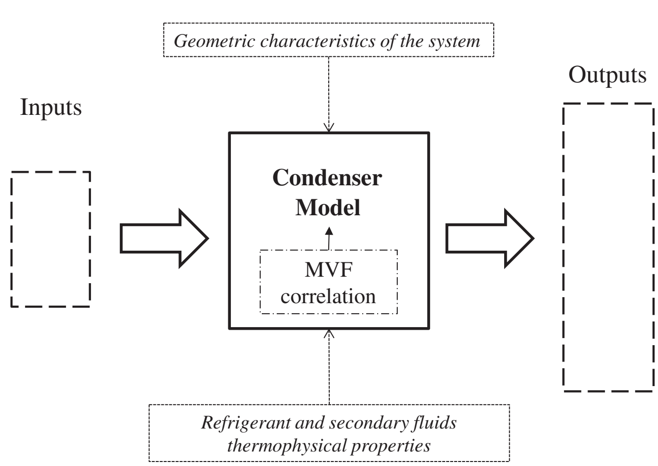

Fig. 1. General model structure.

2.1. Condenser model

In the analyzed condenser the refrigerant is flowing on the shell side, entering at one end of the shell and crossing the tube bundles directed by baffles until leaving at the other end, while the water is flowing in the tubes. For modeling simplicity, a typical design of heat exchanger is developed to form an equivalent axial tube heat exchanger with counter-flow assumption [1,13], considering the thermal mass, heat transfer area and mass flux equivalence with the actual component [14]. Thus, the true flow can be approximated by an outer tube which is formed by the shell and has the refrigerant flowing through it and an inner tube that carries the condensing water [1]. Accordingly, the condenser is modeled with three zones, that is, a superheated vapor zone, a two-phase zone and a subcooled liquid zone, as shown in Fig. 2, where the lengths of these zones, namely, $L_{1\mathrm{c}}$ (superheat), $L_{2\mathrm{c}}$ (two-phase) and $L_{3\mathrm{c}}$ (subcooled), are model outputs.

In what follows, the governing partial differential equations (PDEs) are described, as well as the way to obtain the governing ordinary differential equations (ODEs) of the lumped parameter model.

In order to reduce the complexity of the conservation equations, the main simplifying assumptions of the model are:

- The mass flow rate of the refrigerant is assumed to be the same throughout the condenser.
- The fluid flow in the condenser is one-dimensional.
- Pressure drop of the fluid flow is negligible.
- There is no axial thermal heat conduction in the fluid flow.
- There is no axial thermal heat conduction in the tube wall, and there is no wall temperature variation along its cross section.
- Heat conduction through the shell is much lower that the heat transfer from refrigerant to condensing water.

The condenser can be modeled from the Navier–Stokes generalized equations [15], and from the energy conservation in the condenser's tube wall. Due to the aforementioned simplifying assumptions, these equations can be written as:

$$\frac{\partial \rho A_{\mathrm{cs}}}{\partial t} + \frac{\partial \dot{m}_{\mathrm{r}}}{\partial z} = 0 \quad \text{Refrigerant mass balance} \tag{1}$$

$$\frac{\partial(\rho A_{\mathrm{cs}} h - A_{\mathrm{cs}} P_{\mathrm{c}})}{\partial t} + \frac{\partial(\dot{m}_{\mathrm{r}} h)}{\partial z} = \alpha_{\mathrm{ex}} \pi D_{\mathrm{ex}} (T_{\mathrm{t}} - T_{\mathrm{r}}) \quad \text{Refrigerant energy balance} \tag{2}$$

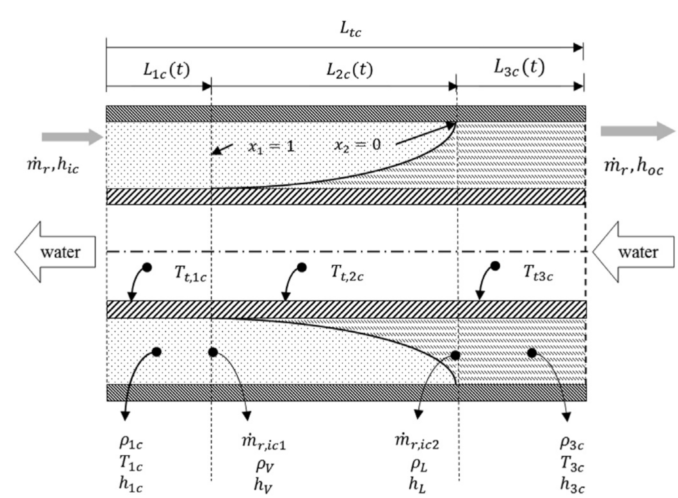

Fig. 2. Condenser zones.