Energetic and Exergetic Analysis of Double Pipe Condensers and Evaporators Interacting with Phase Change Material System

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Abstract

This paper presents a numerical simulation of the thermal and fluid dynamic behaviour of double-pipe: thermal energy storage system consisting of a phase changing material (PCM) interacting with a two phase flow. The governing equations of the fluid flow (continuity, momentum and energy) in the tube (evaporating or condensing) and both the energy equation inside the annulus (PCM) and in the tube wall, are solved iteratively in a segregated manner using a one-dimensional transient formulation based on an implicit step by step numerical scheme in the zone with fluid flow, an enthalpy method to solve the phase change process and an implicit central difference numerical scheme in the tube wall. This formulation requires the use of empirical information for the evaluation of convective heat transfer, shear stress and void fraction. The mathematical model of both evaporator and condenser has been validated by comparison with experimental obtained from literature. Irreversibility due to heat transfer and that to flow-related pressure losses is evaluated using the second law of thermodynamics.

Résumé

Dans ce travail nous présentons un modèle basé sur les équations de conservations de la masse, de l'énergie et du bilan de la quantité de mouvement pour étudier le comportement dynamique d'un échangeur coaxial : évaporateur ou condenseur. Sa particularité est le couplage de ces deux échangeurs avec une unité de stockage d'énergie utilisant un matériau à changement de phase (eau/glace). Pour suivre le front de la fusion/cristallisation nous avons utilisé la méthode enthalpique. Pour évaluer les irréversibilités locales au sein de l'échangeur nous avons utilisé le deuxième principe de la thermodynamique.

Key words:

Heat exchanger, condenser, evaporator, numerical simulation, PCM, irreversibility.

NOMENCLATURE

c_p	specific heat, J/K.kg	$\dot{m}$	mass flow rate, kg/s
e	specific total energy, J/kg	T	temperature, °C
f	friction factor	x	vapour quality
g	acceleration of gravity, m/s^2	u	velocity, m.s^{-1}

Greek symbols

λ	thermal conductivity, W/m.°C		g	gas or vapour
ρ	masse volumique, kg.m^{-3}		l	liquid
			pcm	phase change material

Subscripts

1. INTRODUCTION

The process of two phase flow of refrigerants in an evaporator/condenser has been playing an important role in thermal control of different systems and especially in the thermal latent energy storage (TES) systems [1]. The efficiency of the conventional refrigeration cycle can be improved by incorporating TES unit using phase change material (PCM) [2]. In such system we can have the coupling between the melting/freezing of the PCM and the two phase flow. The objective of this work is to develop numerical model which allow the simulation in both steady and transient state, of the thermal and fluid dynamic behaviour of double pipe evaporators and condensers and the kinetic of the melting/freezing process inside the annulus. The total irreversibility due to the heat transfer and the pressure losses is evaluated by using the second law of thermodynamics.

2. Physical model

2.1 Two phase flow inside tube

Taking into account the characteristic geometry of ducts, the governing equations have been integrated assuming the following assumptions:
- One dimensional flow.
- Non participant radiation medium and negligible radiant heat exchange between surfaces.
- Axial heat conduction inside the fluid is neglected.

The semi-integrated governing equations over the control volume have the following form [1]:

- **Continuity**

$$\left[\dot{m}\right]_i^{i+1} + \frac{\partial m}{\partial t} = 0 \tag{1}$$

- **Momentum**

$$\left[\dot{m}_g v_g\right]_i^{i+1} + \left[\dot{m}_l v_l\right]_i^{i+1} + \Delta x \frac{\partial \dot{m}}{\partial t} = -\left[p\right]_i^{i+1} S - \tau P \Delta z \tag{2}$$

- **Energy**

$$\dot{m}\left[e_l\right]_i^{i+1} + \left[\dot{m}_g \left(e_g - e_l\right)\right]_i^{i+1} + \left(e_g - e_l\right)\frac{\partial m_g}{\partial t} + m_g \frac{\partial e_g}{\partial t} +$$

$$m_l \frac{\partial e_l}{\partial t} - S \Delta x \frac{\partial p}{\partial t} = \dot{q} P \Delta z \tag{3}$$

where the specific energy is defined as $e = h + \dfrac{u^2}{2}$. In order to relate the convective heat transfer and the wall temperature, the convective two phase heat transfer coefficient α is introduced, which is defined from the equation: $\dot{q} = \alpha\left(T_{wall} - T\right)$. The evaluation of the

shear stress is performed by means of a friction factor f. This factor is defined from the expression: $\tau = \phi(f/4)(G^2/2\rho)$ where ϕ is the two-phase factor multiplier. The void fraction ε_g is used to determine the volume fraction of the vapour phase in the mixture, which can be expressed in terms of the gas to liquid velocity ratio in the form: $\varepsilon_g = (1 + (1-x)\rho_g u_g / x\rho_l u_l)^{-1}$, where x is the vapour quality.
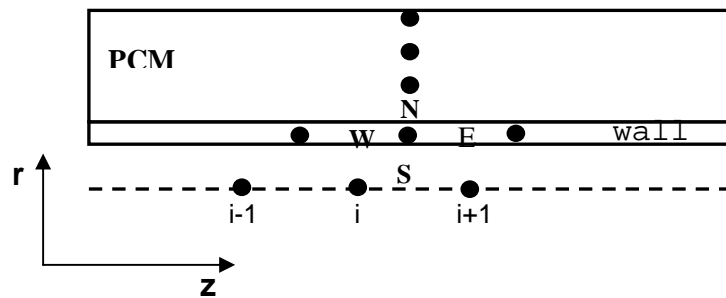

Figure1 : Double pipe heat exchanger scheme.

The different flow regimes produced in both condensation and evaporation phenomenon, have been taken into account. The refrigerants thermophysical properties are evaluated locally using the REFPROP v.6.01 program [3]. In the condensing flow, α is estimated using, in the case of stratified flow, the Nusselt equation, and in the case of annular flow, the expression proposed by [1]; the equation of Wallis [1] is taken as a differentiation criterion between annular and stratified condensation. The void fraction is calculated from the semi-empirical equation by Zivi. In the beginning of the subcooled boiling, α is estimated according to Frost et al. [1]. The correlation proposed by Bergles et al. [1] is used to consider the transition between pure liquid convection heat transfer and boiling heat transfer, which is evaluated from the correlation proposed by Forster et al. [1]. Before the point of dryout, α is evaluated using Kandlikar's correlation [1], and in the post-dryout region (considered to begin at $x = 0.9$) α is calculated from the Groeneveld's correlation [1].The friction factor f in two phase regions is calculated from the literature correlations [1].

2.2 Heat conduction in the internal tube wall

The conduction equation has been written assuming the following hypotheses:
-   One-dimensional temperature distribution.
-   Negligible heat exchanged by radiation.

$$m\frac{\partial h}{\partial t} = (\dot{q}_s P_s - \dot{q}_n P_n)\Delta x + (\dot{q}_w - \dot{q}_e)S \qquad (4)$$

where $\dot{q}_{s/n}$ is evaluated using the respective convective heat transfer coefficients and $\dot{q}_{w/e}$ is determined from the Fourier law, that is: $\dot{q}_{w/e} = -\lambda\left(\dfrac{\partial T}{\partial z}\right)$

1.3   Phase change in the external tube

From **Fig.1** the energy is discharged (or charged) from the lower face while the upper face keeps insulated. For a mathematical description of the phase change process, the following assumptions are made:

- heat transfer in the PCM is conduction dominated and is two –dimensional conduction along the radial r and axial x directions.
- initially, all the PCM is at a uniform temperature.
- PCM is homogeneous and isotropic

$$\left(\rho\, c_p\right)_{pcm} \frac{\partial T_{pcm}}{\partial r} = \lambda_{pcm} \frac{\partial^2 T_{pcm}}{\partial r^2} + \lambda_{pcm} \frac{\partial^2 T_{pcm}}{\partial z^2} \pm \rho_{pcm} L_F \frac{\partial \beta}{\partial t} \qquad (5)$$

where β is the liquid fraction of the PCM. The numerical resolution of the above equation is presented in Ref **[3]** and it is deemed to repeat it in the present work.

The total irreversibility of the two phase flow due to the heat transfer and the pressure losses is evaluated using the following **[4]**:

$$Irr_{tot} = \sum_{i=1}^{n} Irr_i \qquad (6)$$

where Irr_i is the total irreversibility rate inside each volume control:

$$Irr_i = T_o \left[\dot{m}_i \left(s_{i+\frac{1}{2}} - s_{i-\frac{1}{2}} \right) - \Gamma_i \right] \qquad (7)$$

The quantity in square bracket on the right hand side represents the net entropy generation rate at local state position i. Assuming refrigerant as an ideal gas, specific entropy change of the fluid can be calculated:

$$s_{i+\frac{1}{2}} - s_{i-\frac{1}{2}} = c_p \ln\left(\frac{T_{i+\frac{1}{2}}}{T_{i-\frac{1}{2}}} \right) + R \ln\left(\frac{p_{i+\frac{1}{2}}}{p_{i-\frac{1}{2}}} \right) \quad \text{and} \quad \Gamma_i = \frac{\dot{q}_i}{T_i}$$

2. RESULTS AND DISCUSSIONS

Based on the above mentioned physical model, a code has been developed to describe the two phase flow and the melting process of PCM inside the double heat exchangers. The code has been carefully verified using whenever possible analytical solution.

Figs **2** and **3** depict the fluid, wall and PCM temperatures during the condensation and evaporation processes inside a tube with 13 m length, 0.08 m and 0.095 are inner and extern diameters. The refrigerant used is R22. In these cases, both evaporating and condensing flows inside tubes are characterized by two different phenomena: two phase region (saturated liquid + vapour) and single phase flow. In the condenser the single phase corresponds to subcooled liquid but in the evaporator this region corresponds to the superheated vapour.

Figs **4** and **5** present the effect of Reynolds number on both condensation and evaporation heat transfer. During evaporation, the increase in the Reynolds number leads to an increase in the rate of heat transferred from the PCM.
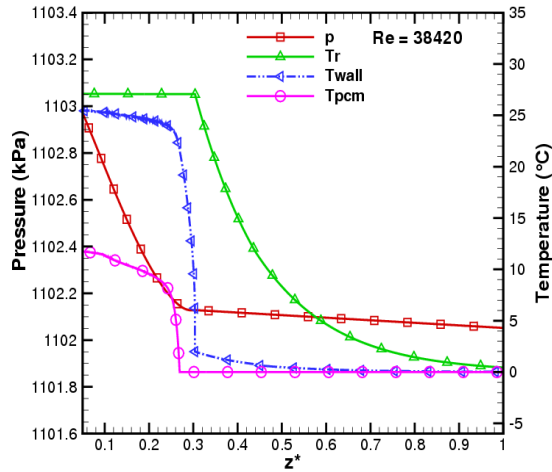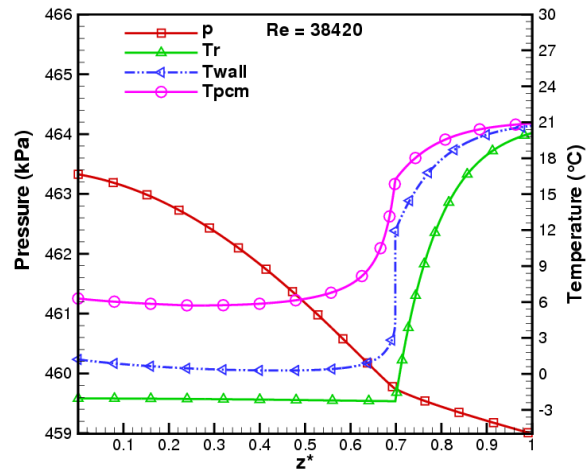
Figure 2 : Condensing flow.

Figure 3 : Evaporating flow.

Nusselt number not only increases with the magnitude of Reynolds number but its peak shifts from being in lower vapour quality mixture at low Reynolds to higher vapour quality mixture at high Re. Condensation heat transfer does not show the same behaviour as that of evaporation. Peak heat transfer occurs around 95% vapour for all annular flow condensation and at 100% vapour for stratified flow.
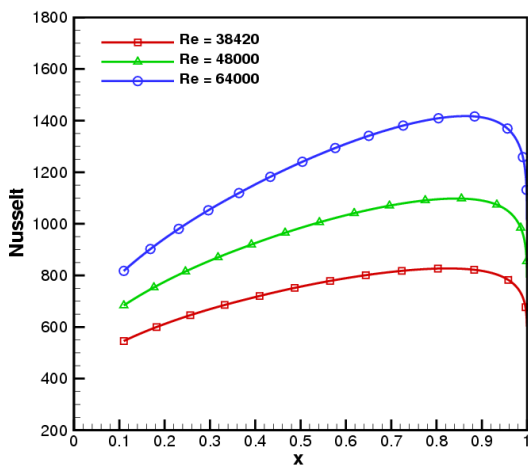

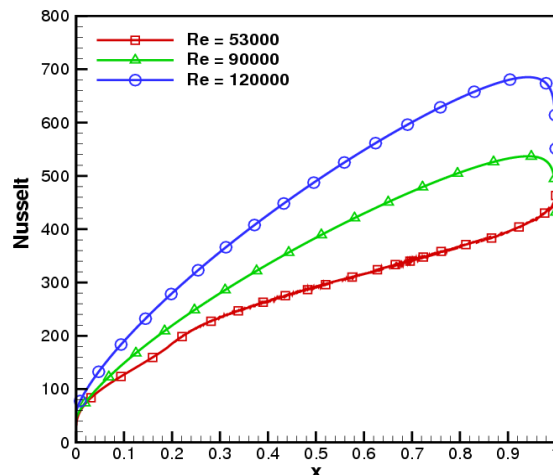*Figure 4 : Effet of Reynolds number on the heat transfer : evaporating flow*

Figure5 : Effet of Reynolds number on the heat transfer : condensing flow

We can also note that the heat transfer coefficient has smaller magnitude for a given Reynolds number than does evaporation. That is due to the resistance of the liquid film that is formed at the refrigerant inner wall as well as to the pool of the liquid that accumulates at the bottom of the tube. The latter occurs primarily during stratified flow.

Fig.6 shows the temperature distribution inside the tube (R22) and the annulus (PCM). From the figure, a typical logarithmic temperature distribution for the single phase flow can be observed. The two phase flow is characterized by a slight temperature drop due to the pressure drop. The PCM temperature decreases by increasing time. Phase change transition of PCM produce at the melting temperature.

Fig.7 displays the variation of local irreversibility rate for two Reynolds number during the two phase region and for the evaporating flow. We note that the local irreversibility rate decreases by increasing the vapour quality until it reaches the critical point where it starts to increase. This is because in this region (correspond to the nucleate boiling) the heat transfer dominates the local irreversibility inside the volume control. After the critical quality, the nucleate boiling is suppressed and the local heat transfer inside the volume increase due to the heat exchange between the refrigerant and the wall. We can also note that the local irreversibility increases by increasing the Reynolds number. This is due to the effect of the pressure losses inside the volume control.
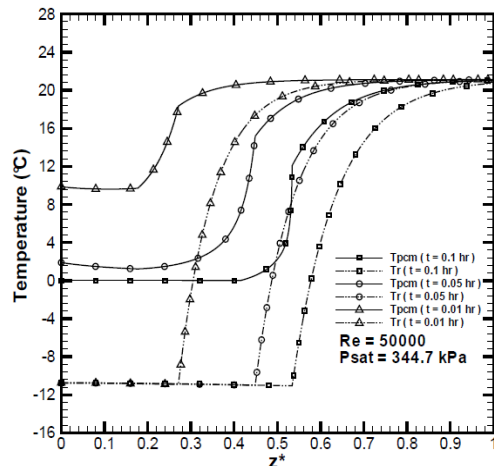

Figure 6: Refrigerant and PCM temperature distribution in a double pipe condenser at different instants
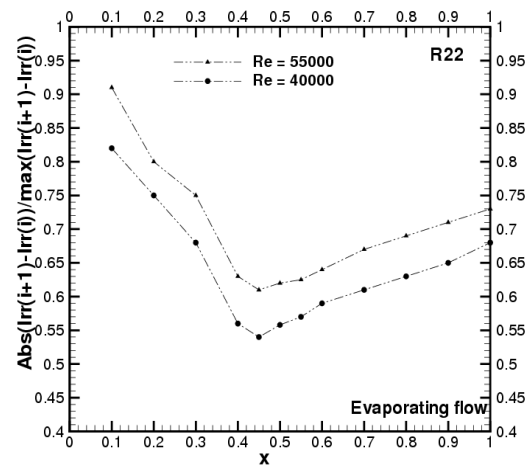
Figure 7: Effects of Reynolds number on the local irreversibility: evaporating flow

CONCLUSION

A numerical model of the thermal and fluid dynamic behaviour of double pipe condensers and evaporators interacting with a phase change material system has been developed by means of a transient one dimensional analysis of the two phase flow and two dimensional model for the PCM system with detail analysis of the heat conduction in the tube wall. The empirical coefficients proposed in the literature are used in the model to evaluate the shear stress, heat flux and two phase flow structure. The dependence of the local irreversibility on flow regime was also discussed.

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