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Void fraction correlations analysis and their influence on heat transfer of helical double-pipe vertical evaporator

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HIGHLIGHTS

- 50 void fraction correlations were evaluated on heat transfer in vertical evaporators.
- Two-phase flow model based on control volume formulation was used.
- The drift flux parameter is common in all correlations with satisfactory results.

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ABSTRACT

An analysis of 50 void fraction correlations available in the literature was performed to describe two-phase flow mechanism inside two helical double-pipe vertical evaporators. The evaporators considered water as working fluid connected in countercurrent so the change of phase was carried out into the internal tube. The discretized equations of continuity, momentum and energy in each flow were coupled using an implicit step by step method. The selection of the void fraction correlations for the mathematical model was based on inclusion of some theoretical limits. The results of this analysis were compared with the experimental data in steady state for two different evaporators, obtaining good agreement in the evaporation process for only 7 void fraction correlations. The Armand and Massena correlation had a mean percentage error (*MPE*) of 3.08%, followed by Rouhanni and Axelsson I adquired $MPE = 3.16\%$, Chisholm and Armand obtained $MPE = 3.18\%$, Steiner as well as Rouhanni and Axelsson II with $MPE = 3.19\%$, Bestion reached $MPE = 3.20\%$ and Flanigan presented $MPE = 3.21\%$. Furthermore, the experimental and simulated heat flux were acceptable ($R^2 = 0.939$). Finally, the results showed that the drift flux parameter was important to evaluate the void fraction.

1. Introduction

Efficient use of applied energy is an essential element to mitigate global warming as well as economic development and social progress in all countries. In recent years, the demand for energy resources has been increasing by the result of a worldwide fast population growth [1]. Based on this information, some countries are investing considerable amounts of money in the development of equipment as well as innovative process control strategy that facilitates the recovery and efficient use of energy [2–8]. One of the most interesting of these devices is the absorption heat transformer (AHT), which has the capacity to upgrade waste heat temperature to a higher level to be reused in the industrial process, consuming just a negligible amount of primary energy [9].

Consequently, the AHT has been set in water purification (WP) systems because it has several advantages over conventional purification systems [10]. The integration of both systems (AHT–WP) enables to increase the temperature of the impure water system, and thus obtain pure water and useful heat [9,10]. Fig. 1 showed the schematic diagram of the AHT–WP with recycling energy. The five main components of the AHT are: generator (GE), condenser (CO), evaporator (EV), absorber (AB), and economizer (EC). These components were operated with different temperatures

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Nomenclature

A	Cross area section (m^2)
AB	Absorber
AHT	Absorption heat transformer
AHT–WP	Absorption heat transformer integrated to the water purification system
b	Coil pitch
CO	Condenser
Cp	Specific heat at constant pressure $\left(\frac{J}{kg\,^\circ C}\right)$
CV	Control volume
D	Helical diameter (m)
d	Internal diameter (m)
Dn	Dean number $\left[Dn = Re\left(\frac{d}{D}\right)^{\frac{1}{2}}\right]$
EV	Evaporator
f	Friction factor (dimensionless)
F_t	Froude rate $\left[F_t = \left(\frac{x^2G^2}{\rho_g^2 gd(1-x)}\right)^{0.5}\right]$
Fr	Froude number $\left[Fr = \left(\frac{U_{sl}^2}{gD}\right)\right]$
g	Acceleration due to gravity $\left(\frac{m}{s^2}\right)$
GE	Generator
He	Helical number $\left[He = \frac{Dn}{\left[1+(b/2\pi(d/2))^2\right]^{\frac{1}{2}}}\right]$
G	Mass velocity $\left(\frac{kg}{m^2\,s}\right)$
h	Enthalpy $\left(\frac{J}{kg}\right)$
$\dot{m}$	Mass flow rate $\left(\frac{kg}{s}\right)$
m	Mass (kg)
MPE	Mean percentage error
n_z	Number of control volumes
Nu	Nusselt number $\left[Nu = \frac{\alpha \cdot d}{\lambda}\right]$
p	Pressure (bar)
P	Perimeter (m)
Pr	Prandtl number $\left[Pr = \frac{Cp\mu}{\lambda}\right]$
$\dot{q}$	Heat flux per unit of area $\left(\frac{W}{m^2}\right)$
Q	Heat flux (W)
Re	Reynolds number $\left[Re = \frac{Gd}{\mu}\right]$
Re_l	Liquid Reynolds number $\left[Re_l = \frac{Gd}{\mu_l}\right]$
RMSE	Root mean square error
t	Time (s)
T	Temperature (°C)
U_{sl}	Liquid superficial velocity $\left[U_{sl} = \frac{G(1-x)}{\rho_l}\right]$ $\left(\frac{m}{s}\right)$
U_{sg}	Steam superficial velocity $\left[U_{sg} = \frac{Gx}{\rho_g}\right]$ $\left(\frac{m}{s}\right)$
v	Velocity $\left(\frac{m}{s}\right)$
We_l	Weber number $We_l = \frac{G^2 d}{\sigma \rho_l}$ (dimensionless)
X_{tt}	Martinelli parameter $\left[X_{tt} = \left(\frac{1-x_g}{x_g}\right)^{0.9}\left(\frac{\rho_g}{\rho_l}\right)^{0.5}\left(\frac{\mu_l}{\mu_g}\right)^{0.1}\right]$ (dimensionless)
x_g	Vapor mass fraction (dimensionless)

Greek letters

ρ	Density $\left(\frac{kg}{m^3}\right)$
δ	Convergence criterion (dimensionless)
σ	Superficial tension $\left(\frac{N}{m}\right)$
Φ	Two-phase frictional multiplier (dimensionless)
α	Heat transfer coefficient $\left(\frac{W}{m^2\,^\circ C}\right)$
τ	Shear stress (Pa)
λ	Thermal conductivity $\left(\frac{W}{m\,^\circ C}\right)$
μ	Dynamic viscosity (Pa s)
ϵ_g	Void fraction (dimensionless)

Subscript

exp	Experimental
g	vapor
l	liquid
sim	Simulation

against pressure levels (see Fig. 1). In the AHT–WP, the water purification system removes the useful heat obtained in the AB of the AHT. The impure water reached the boiling point and changed in two-phases (liquid and steam). In liquid phase, the impure water was pumped the AB, meanwhile the steam was transferred to an auxiliary condenser where the heat was conducted to the heat source [10].

In the literature, two theoretical mathematical models have been trying to explain the thermal and fluid dynamic behavior of the fluid flow into the heat exchanger (EV and CO) [11,12]. These dynamic models consider equations of continuity, momentum and energy in each flow. The models were evaluated to determine the principal operation variables that affect the evaporation and condensation. Nevertheless, one unknown critical parameter involved in the physical mathematical model was the void fraction [13]. Moreover, Dalkilic et al. [14] mentioned also that the void fraction is very important parameter during condensation in vertical downward flow in a smooth tube. Void fraction was defined as the ratio of the volume occupied by the gas and liquid [15]. Due to difficulty of calculations and lack of appropriate instrumentation, most analyzes were inclined towards empirical correlations [16,17].

The main objective of this work is to discover correlations of void fraction, among those reported in the literature, and test on the mathematical model in order to describe and analyze heat transfer and fluid dynamic behavior of a helical double-pipe vertical evaporator. It is important to remark that this analysis of void fraction is carried out by comparing the theoretical model and experimental data in two different size evaporators used in each AHT–WP. Both evaporators used water as working fluid connected in countercurrent.

2. Mathematical formulation of two phase flow

The mathematical formulation applied in this research was described in detail by Colorado-Garrido et al. [11] to transient two-phase flow in helical coil evaporator.

★ The outlet mass flow rate ($\dot{m}_{i+1}$) was obtained from the continuity equation,

$$\dot{m}_{i+1} = \dot{m}_i - \frac{A\Delta z}{\Delta t}\left(\bar{\rho}_{tp} - \bar{\rho}^o{}_{tp}\right) \tag{1}$$

where the two-phase flow density was evaluated from: $\rho_{tp} = \epsilon_g \rho_g + (1-\epsilon_g)\rho_l$.

In terms of the mass flow rate, gas (v_g) and liquid (v_l) velocities were obtained as,

$$v_g = \frac{\dot{m}x_g}{\rho_g \epsilon_g A}$$
$$v_l = \frac{\dot{m}(1-x_g)}{\rho_l(1-\epsilon_g)A} \tag{2}$$