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Pressure drop data and prediction method for enhanced external boiling tube bundles with R-134a and R-236fa

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ABSTRACT

The pressure drops of external flow over enhanced tube bundles were experimentally obtained at both adiabatic and diabatic conditions using R-134a and R-236fa as test fluids. The tests were carried out at saturation temperatures of 5 and 15 °C, mass fluxes from 4 to 40 kg m^{-2} s^{-1}, heat fluxes from 15 to 70 kW m^{-2} and inlet vapour qualities ranging from 10% to 90%. The frictional pressure drop was found to be primarily a function of mass flux and vapour quality. After comparisons were made with prediction methods in literature a new pressure drop prediction method was proposed for adiabatic and diabatic conditions. The proposed method is based on local measurements (4 and 8 tube rows) and flow conditions (evaluated per tube pitch) and the prediction method is well adapted to local incremental implementation for flooded evaporator design.

Données sur la chute pression et méthode prévisionnelle pour l'ébullition externe améliorée du R-134a et du R-236fa sur des faisceaux de tubes

Mots clés : ébullition en écoulement externe ; chute de pression ; transfert de chaleur amélioré ; méthode prévisionnelle ; écoulement diphasique

1. Introduction

For design purposes, the two-phase pressure drop is an important consideration in tube bundle evaporation due to flow related losses. Furthermore, in flooded type evaporators with close temperature approaches of only ≈1–2 K, such as in refrigeration and heat pump applications, the effect of the two-phase pressure drop on the local saturation temperature

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Nomenclature

Roman symbols

A	Area, m^2
a	Inter-tube gap
a–g	Constants, –
C	Constant, –
Cap	Capillary number, –
D	Diameter, *m*
Δp	Pressure drop, Pa
Eu	Euler number, –
Fr	Froude number, –
f	Friction factor, –
G	Mass flux, kg m^{-2} s^{-1}
g	gravitational acceleration, m s^{-2}
j_v	Superficial gas velocity, m s^{-1}
j_v^*	Dimensionless superficial velocity, –
m	Mass flow rate, kg s^{-1}
N_R	Number of tube rows, –
P	Pitch, *m*
q	Heat flux, W m^{-2}
Re	Reynolds number, $4\Gamma/\mu_l$, $\rho v D_h/\mu$
Ri	Richardson number, –
S	Slip ratio, –
T	Temperature, *K*
U_{gs}	Velocity, –
u	Velocity, m s^{-1}
We	Weber number, –
X_{tt}	Martinelli number, –
x	Length, *m* or Vapour quality, –
z	Height, m

Greek symbols

Δ	Difference
ρ	Density, kg m^{-3}
δ	Film thickness, *m*
ε	Void fraction, –
λ	Two-phase multiplier, –
Λ	Dimensionless mass velocity, –
Φ	Two-phase multiplier, –
μ	Dynamic viscosity, Pas
σ	Surface tension, N m^{-1}

Subscripts

H	Homogeneous
hex	Hexagonal
2Φ	Two-phase
f	Frictional
gap	Inter-tube gap
l	Liquid
m	Momentum
sat	Saturation temperature
s	Static or gravitational
t	Total
v	Vapour or gas phase
wat	Water

may be crucial in evaluating the temperature difference for incremental thermal design methods (i.e. local calculation of thermal performance).

The two-phase pressure drop components in a vertical flow require a void fraction model for their calculation and are therefore sensitive to modelling assumptions related to the prediction of void fraction. The void fraction is thus an important parameter for evaluating the two-phase pressure drop since it is directly related to the local two-phase density of the shell-side flow and the relative mean velocities of each phase. In particular, for thermosyphon evaporators, the circulation rate depends directly on the two-phase pressure drop across the tube bundle and hence the difference in void fraction is of primary importance. Furthermore, at low mass fluxes, the static pressure drop becomes dominant and its calculation is directly dependent on the accuracy of the void fraction profile in the bundle.

1.1. *Void fraction*

As pointed out by Ribatski and Thome (2007), several authors recorded void fraction values significantly different from those predicted by the homogeneous flow model. For tube bundle flow, and especially at low liquid velocities, the slip ratio can be much higher because the vapour phase buoyancy dominates.

The method of Ishihara et al. (1980) is based on the two-phase frictional multiplier of the liquid, which was defined as a function of the Martinelli parameter. Cornwell et al. (1980) also proposed a void fraction method based on the Martinelli parameter, whereas Fair and Klip (1982) proposed a method based on a different two-phase friction multiplier and the Martinelli parameter.

The prediction method of Schrage et al. (1988) used a function of liquid Froude number as a multiplier in the homogeneous model. This model was one of the earliest models that included the important effect of mass flux directly:

$$\frac{\varepsilon}{\varepsilon_H} = 1 + 0.123\left(\frac{\ln x}{Fr_l^{0.191}}\right) \quad \text{with} \quad Fr_l = \frac{G}{\rho_l (gD)^{0.5}} \tag{1}$$

Dowlati et al. (1996) proposed a void fraction model based on data from R-113 flow at mass fluxes higher than 50 kg m^{-2} s^{-1}, which was based on the dimensionless gas phase superficial velocity and included two empirical constants (C_1 = 30 and C_2 = 50):

$$\varepsilon = 1 - \frac{1}{\left(1 + C_1 j_v^* + C_2 j_v^{*2}\right)} \tag{2}$$

$$\text{with} \quad j_v^* = \frac{\rho_v^{0.5} j_v}{\sqrt{gD(\rho_l - \rho_v)}} \tag{3}$$

The Feenstra et al. (2000) method is based on dimensionless parameters that were used to fit to their database and that of other researchers. Combining the continuity equations for the