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A semi-empirical model for free-convection condensation on horizontal pin–fin tubes

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ABSTRACT

A simple semi-empirical correlation accounting for the combined effect of gravity and surface tension has been developed for condensation on horizontal pin–fin tubes. The model divides the heat transfer surface into five regions, i.e. two types of pin flank, two types of pin root and the pin tip. Data for three fluids (i.e. steam, ethylene glycol and R113) condensing on eleven tubes with different geometries were used in a minimization process to find three empirical constants in the final expression. The model gives good overall agreement (within ±20%) with the experimental data, as well as correctly predicting the dependence of heat-transfer enhancement on the various geometric parameters and fluid types.

1. Introduction

A significant number of experimental investigations have been reported on free-convection condensation heat-transfer on horizontal integral-fin tubes; see for example [1–11]. During the condensation process, liquid retained on the lower part of tube insulates the fin flanks and root from heat transfer, This condensate retention on integral-fin tubes was first observed by Katz et al. [12] and afterwards experimentally investigated by many other investigators for a wide range of fluid and tube combinations [1,3,13–15]. The development of an analytical correlation to predict this condensate retention angle (measured from the top of the tube up to the point where whole fin flanks become flooded with condensate) was a pivotal step for the development of a theoretical heat-transfer model for condensation on integral-fin tubes. Such an analytical correlation to predict condensate retention angle on integral-fin tube was first reported by Honda et al. [1] (later developed by Owen et al. [16] and Rudy and Webb [13]) to accomplish the requirement, the following expression was produced for retention angle, ϕ_f, measured from the top of the tube,

$$\phi_f = \cos^{-1}\left[\left(\frac{2\sigma\cos\theta}{\rho g s R_o}\right) - 1\right] \text{ for } s < 2h \quad (1)$$

Reliable and simple heat-transfer models for integral-fin tubes (i.e. Honda and Nozu [17], Rose [18] and Briggs and Rose [19]) accounting for the combined effects of surface tension and gravity on heat-transfer were later developed which are now readily available for design engineers. With the help of above experimental and theoretical work, optimal tube geometries are now identified for a wide range of working fluids condensing on integral-fin tubes.

In the recent past, attention has been focused on more complex pin–fin tubes (a schematic of three dimensional pin–fin tube with condensate retention angle is shown in Fig. 1). Many experimental investigations on pin–fin tubes [20–25] have shown their superior heat transfer performance (up to 25%) over the equivalent integral-fin tubes (i.e. with the same fin height, root diameter and longitudinal pin thickness and spacing). When Briggs [22] tested steam, four out of six pin–fin tubes were fully flooded with condensate i.e. the only available area for heat transfer was the pin tips. When compared with equivalent integral-fin tubes these fully flooded tubes gave about 20% more heat transfer, despite the fact that available area was only about half of the equivalent integral-fin tube. Qin et al. [26] tested R134a condensing on two pin–fin tubes of different geometries, one made of copper and another made of stainless steel. Heat transfer enhancements were found to be 7.9 and 3.3 for copper and stainless steel pin–fin tubes respectively. The superior performance of copper was due to its longer pin height and high thermal conductivity.

In order to exploit the superior experimental performance of pin–fin tubes, it is necessary to develop a heat-transfer model to optimize these tubes to discover their full potential. For the development of an accurate heat-transfer model for pin–fin tubes, the development of a predictive correlation of condensate retention

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Nomenclature

A	area
A_D	constant in Eq. (4)
$A_{flank\ 1}$	area of pin flank 1
$A_{flank\ 2}$	area of pin flank 2
A_L	constant in Eq. (3)
$A_{root\ 1}$	area of root 1
$A_{root\ 2}$	area of root 2
A_{tip}	area of pin tip
B	constant in Eq. (3)
B_{flank}	empirical constant for pin flank
$B_{flank\ 1}$	empirical constant for pin flank 1
$B_{flank\ 2}$	empirical constant for pin flank 2
B_{root}	empirical constant for root
$B_{root\ 1}$	empirical constant for root 1
$B_{root\ 2}$	empirical constant for root 2
B_{tip}	empirical constant for pin tip
C	constant in Eq. (2)
d	fin or pin tip diameter of fin or pin tube
g	specific force of gravity
j	number of pins in unflooded region
h	fin or pin height
h_{fg}	specific enthalpy of vaporization
h_v	mean vertical fin or pin height
k	thermal conductivity of condensate
L	length of flat plate
n	total number of pins per circumference
P	perimeter
$P_{flank\ 1}$	perimeter of pin flank 1
$P_{flank\ 2}$	perimeter of pin flank 2
$P_{root\ 1}$	perimeter of root 1
$P_{root\ 2}$	perimeter of root 2
P_{tip}	perimeter of pin tip
$Q_{flank\ 1}$	heat-transfer rate through all pin flanks 1
$Q_{flank\ 2}$	heat-transfer rate through all pin flanks 2
$Q_{root\ 1}$	heat-transfer rate through root 1
$Q_{root\ 2}$	heat-transfer rate through root 1
Q_{tip}	heat-transfer rate through all pin tips
q_d	heat flux on outside of a horizontal tube defined by Eq. (4)
q_{flank}	heat flux to fin flank in unflooded part of tube
$q_{flank\ 1,i}$	heat flux to flank 1 for pin *i* defined by Eq. (10)
$q_{flank\ 2,i}$	heat flux to flank 2 for pin i defined by Eq. (14)
q_L	heat flux on a plate defined by Eq. (3)
$q_{root\ 1}$	heat flux through root 1 defined by Eq. (19)
$q_{root\ 2,i}$	heat flux to pin root 2 for a pin *i* defined by Eq. (22)
q_{tip}	heat flux to fin tip
$q_{tip,i}$	heat flux to pin tip *i* defined by Eq. (6)
$q_{tip,flood}$	heat flux to fin tip in flooded part of tube
R_o	pin tip radius
s	fin spacing at fin root or longitudinal pin spacing at pin root
s_c	circumferential pin spacing
t	fin tip thickness or longitudinal pin tip thickness
t_c	circumferential pin thickness
x_D	linear dimension of tube diameter
x_L	linear dimension of plate length
x_σ	characteristic length for surface tension driven flow in model

Greek letters

β	angle defined by Eq. (17)
ΔT	temperature difference across the condensate film
$\varepsilon_{\Delta T}$	vapour-side, heat-transfer enhancement ratio, heat flux for finned or pinned tube based on fin or pin root diameter divided by heat flux for smooth tube with same fin/pin root diameter, at same vapour-side, temperature difference
μ	dynamic viscosity of condensate
$\xi(\phi)$	function given by Eq. (5)
ρ	density of condensate
ρ_v	density of vapour
$\tilde{\rho}$	$\rho - \rho_v$
σ	surface tension
θ	half angle at fin tip
ϕ	angle measured from the top of a fin or pin tube
ϕ_f	condensate flooding or retention angle measured from the top of a fin or pin tube

*Subscripts*calc

	calculated
obs	experimental
rel	pertaining to relative residuals
Std	pertaining to standard deviation

angles on pin–fin tubes was the start point which was recently proposed by Ali and Briggs [27] as following equation;

$$\phi_f = \cos^{-1}\left[\left(1 - C \times \frac{s_c}{t_c}\right)\left(\frac{2\sigma}{\rho g s R_o}\right) - 1\right] \textit{for } s < 2h \tag{2}$$

Eq. (2) was found to give agreement within ±15% with experimental retention angle data on pin–fin tubes reported by the authors and also by other investigators [13,14,20] for a wide range of fluid and tube combinations.

Kumar et al. [28] proposed a generalized empirical model to predict the vapour-side, heat-transfer coefficient on integral-fin and pin–fin tubes (the only theoretical model so far proposed for condensation on pin–fin tubes). They proposed that the heat-transfer coefficient was a function of fluid properties, tube geometry and condensate mass flow rate. They claimed agreement to within ±15% with their own experimental data for one tube for steam and one for R-134a, respectively. Cavallini et al. [29] and Namasivayam [30] reported the poor performance of this model for copper integral-fin tubes. Later, Ali and Briggs [23] when compared this model with experimental data of pin–fin tubes, it showed poor agreement with most of the data. One possible reason for the inadequate performance of the model might be neglect of condensate retention on the lower part of the tubes. In addition, the model is based on the assumption of a linear pressure gradient along the pin or fin flank which has been shown to give poor results for integral-fin tubes (see Briggs and Rose [31]).

More recently, Kundu and Lee [32] reported optimized profiles for vertical fins of variable cross section subjected to condensation of saturated vapour under free convection, while Kundu [33] and Kundu and Ghosh [34] extended the analysis to horizontal circular pins under free and forced convection condensation respectively. These included the conjugate effects of conduction in the pins. The choice of fin profiles, however, meant that surface tension effects could be neglected and the condensation process was modeled assuming gravity drainage alone (in 32 and 33) and gravity plus vapour shear (in 34).

Finally, Nagarani et al. [35] presented a detailed review covering a wide range of extended surfaces applications in heat transfer problems, including condensation on pin–fin tubes.