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THE COMBINED EFFECT OF VAPOUR VELOCITY AND CONDENSATE INUNDATION DURING CONDENSATION ON INTEGRAL-FIN TUBES: A THEORETICAL ANALYSIS

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ABSTRACT

This work presents procedures to compute the heat transfer coefficient when downwards flowing pure vapours (or azeotropic mixtures) condense on horizontal integral-fin tubes.

At first, single tubes are considered, and the suggested procedure takes shear, gravity and surface tension force into account. The method is compared with a vast data bank demostrating a sound concordance.

The second part of the paper considers condensation on tube bundles, also taking the effect of condensation inundation on heat transfer into account. The correlations proposed are compared with available experimental points demostrating satisfactory concordance.

NOMENCLATURE

C	= Constant in eq.(2)
d	= Tube diameter
E	$= 100 \sum_{1}^{n} \lvert (Nu_{cal} - Nu_{exp})/Nu_{exp} \rvert / n$
fpm	= Fins per meter
h	= Fin heigth
n	= Number of experimental runs
N	= Tube row number
Nu	= Nusselt number = $\alpha\, d_o/\lambda_L$
p	= Fin pitch
Re_{eq}	= Equivalent Reynolds number (eq. 3)
Re_G	= Vapour Reynolds number = $\rho_G u_{max} d_o/\mu_G$
t	= Fin thickness
u	= Vapour velocity
α	= Film heat transfer coefficient
ρ	= Density
λ	= Thermal conductivity
μ	= Dynamic viscosity

Subscripts:

cal	= calculated
exp	= experimental
fc	= forced convection
G	= vapour phase
L	= condensate
max	= maximum value
o	= outside of finned tube at the fin tip
r	= outside of finned tube at the fin root
st	= stationary vapour

INTRODUCTION

Several theoretical models for the prediction of the heat transfer coefficients during condensation on single horizontal integral-fin tubes are available in the relevant literature. Honda et al. (1987a,b) analysis, Adamek and Webb (1990) procedure, Murata and Hashizume (1992) model, the correlation by Briggs and Rose (1994) refer to stationary pure vapour condensation on a single horizontal tube; they display the effect of surface area increase, surface tension and gravity forces, though none of them take vapour shear effect into account. This effect may be particularly relevant in the inlet region of condensers.

Cavallini et al. (1995) presented a model for condensation on single horizontal integral-fin tubes under high vapour velocity. This model was based on the experimental data and the flow pattern observation was carried out during condensation of R11 and R113 on three integral-fin tubes in a wide range of vapour velocities and driving temperature differences. The model is able to predict the data by Honda et al. (1989b, 1991) for halogenated refrigerant 113 and the data for steam by Michael et al. (1989) fairly well.

When condensing pure vapours on a bundle of finned tubes, experimental data shows that the heat transfer coefficient is affected by the combined effect of tube geometry and tube layout, condensate inundation and vapour shear, surface tension and gravity. Much experimental work has beeen carried out during condensation of stagnant vapors: Katz and Geist (1948), Webb and Murawski (1990), Murata et al. (1990, 1992), Blanc (1994) and during condensation of vapours flowing downwards (Honda et al. (1989b, 1991, 1995a,b, 1996), Chu and McNaught (1992, 1994) and McNaught and Chu (1993)). Several theoretical models have been proposed to predict the local heat transfer coefficient in the tube bundle during condensation of a pure stagnant vapour: Katz and Geist (1948), Honda et al. (1989a) and Murata and Hashizume (1992). No equation or analysis is available for condensation of pure vapours flowing at high velocity on a bundle of horizontal integral fin tubes.

This paper presents a method for calculating the heat transfer coefficient during condensation on single tubes and bundles of finned tubes when both inundation and vapour shear stress are present. The method is tested against available experimental data.

CONDENSATION WITHOUT INUNDATION

Bella et al. (1993) and Cavallini et al. (1994a,b,c) tested three different integral-fin tubes: the first had 1333 fins per meter (fpm), the other two had 2000 fpm and different fin heights. Table 1 lists their main geometrical characteristics.Two different test sections were used: the first was designed to measure heat transfer coefficients and presented a test tube instrumented with thermocouples to measure the tube wall temperature; the second had two transparent side-walls so as to observe the condensate flow patterns directly. Pure R-11 and R-113 were tested, with vapor inlet pressure ranging between 110 and 200 kPa, temperature difference between vapor and tube wall varying between 4 and 18.5 K and vapor Reynolds number Re_G between 11,000 and 350,000.

Analysis of the above data showed that the effects of vapour shear are linked to the geometry of the extended surface. They are appreciable only at vapour Reynolds numbers Re_G greater than 80,000-100,000 with a 50-60% maximum enhancement for the 1333 fpm tube and of 70-80% for the 2000 fpm tubes at the highest vapour velocity tested. At very low vapour velocities, the condensate leaves the tube forming equally spaced columns and flooding the lower portion of the tube. When vapour velocity increases, the columns break into droplets and a dispersed flow takes place. Vapour shear reduces the thickness and promotes turbulence in the liquid film, while the extension of the flooded portion of the tube does not seem to be influenced by vapour shear, as described by Cavallini et al. (1994b,c).

In the region where vapour shear stress effects are negligible and the condensation on integral-fin tubes is controlled only by surface tension and gravity the experimental heat transfer coefficients can be well predicted by the available models for stationary vapour condensation on integral-fin tubes as shown by Cavallini et al (1995). A model which reproduces data within ± 20% and is quite simple is the Briggs and Rose (1994) equation.

In the region where vapour shear stress effects are important, the heat transfer coefficient (referred to the envelope surface area of the tube at the fin tip and to the temperature difference between the vapour and the tube wall) can be calculated by an asymptotic model (Cavallini et al. (1995)), according to:

$$\alpha = \left(\alpha_{st}^2 + \alpha_{fc}^2\right)^{1/2} \qquad (1)$$

where the first asymptote α_{st} is the heat transfer coefficient under stationary vapour conditions when shear stress effects are insignificant, whereas the second asymptote α_{fc} is the heat transfer coefficient when forced convection mechanisms are predominant, and gravity and surface tension effects become negligible. The first term α_{st} can be calculated by suitable models for stationary conditions. The second term α_{fc} can be expressed as a function of the geometry of the extended surface, of the equivalent Reynolds number Re_{eq} and of the liquid Prandtl number as:

$$Nu_{fc} = \left(\alpha_{fc} \cdot d_o\right) / \lambda_L = C \cdot Re_{eq}^{0.8} \cdot Pr_L^{1/3} \qquad (2)$$

where: $$Re_{eq} = Re_f + Re_G\,(\mu_G/\mu_L)\,(\rho_L/\rho_G)^{1/2} = [\rho_G u_{max} d_o)/\mu_L)\,(\rho_L/\rho_G)^{1/2} \qquad (3)$$

$$C = 0.03 + 0.166 \cdot (t_o/p) + 0.07 \cdot (h/p) \qquad (4)$$

Equations (1-4) were determined by fitting a set of more than 150 data points, as described in Cavallini et al. (1995), with $3 < Pr_L < 8$ and $22{,}000 < Re_{eq} < 110{,}000$.

Authors	Cavallini et al.			Honda et al.			Michael et al.		
Test tube	1333 fpm	2000a fpm	2000b fpm	A	B1	B2	200 fpm	400 fpm	800 fpm
Diameter d_o [mm]	16.4	15.8	15.8	15.6	16.1	15.8	21.05	21.05	21.05
Fin thickness t_o [mm]	0.10	0.20	0.20	0.21	0.05	0.22	1.00	1.00	1.00
Fin thickness t_r [mm]	0.35	0.20	0.20	0.45	0.17	0.36	1.00	1.00	1.00
Fin pitch [mm]	0.75	0.50	0.50	0.96	0.50	0.52	5.00	2.50	1.25
Fin height [mm]	0.70	1.50	0.60	1.43	1.30	1.09	1.00	1.00	1.00

Table 1. Geometrical characteristics of the test tubes.

Other available experimental data sets for condensation on a horizontal integral-fin tube without condensate inundation, such as first row of a tube bundle or core tube in an otherwise uncooled bundle, are those by Honda et al. (1989b, 1991, 1995a, 1995b, 1996) with refrigerants R-11, R-113, R-123, R-134a, and Michael et al. (1989) with steam. Honda et al. data covers temperature differences between 1 and 13 K with vapor Reynolds number ranging from 37,000 to 240,000 ($10{,}000 < Re_{eq} < 69{,}000$). Michael et al. data is relative to condensation of steam at 11.4 kPa, with vapor Reynolds number between 2,000 and 15,000 ($4{,}000 < Re_{eq} < 30{,}000$) and temperature difference between 4 and 12 K. Table 1 reports the geometry of the test tubes.

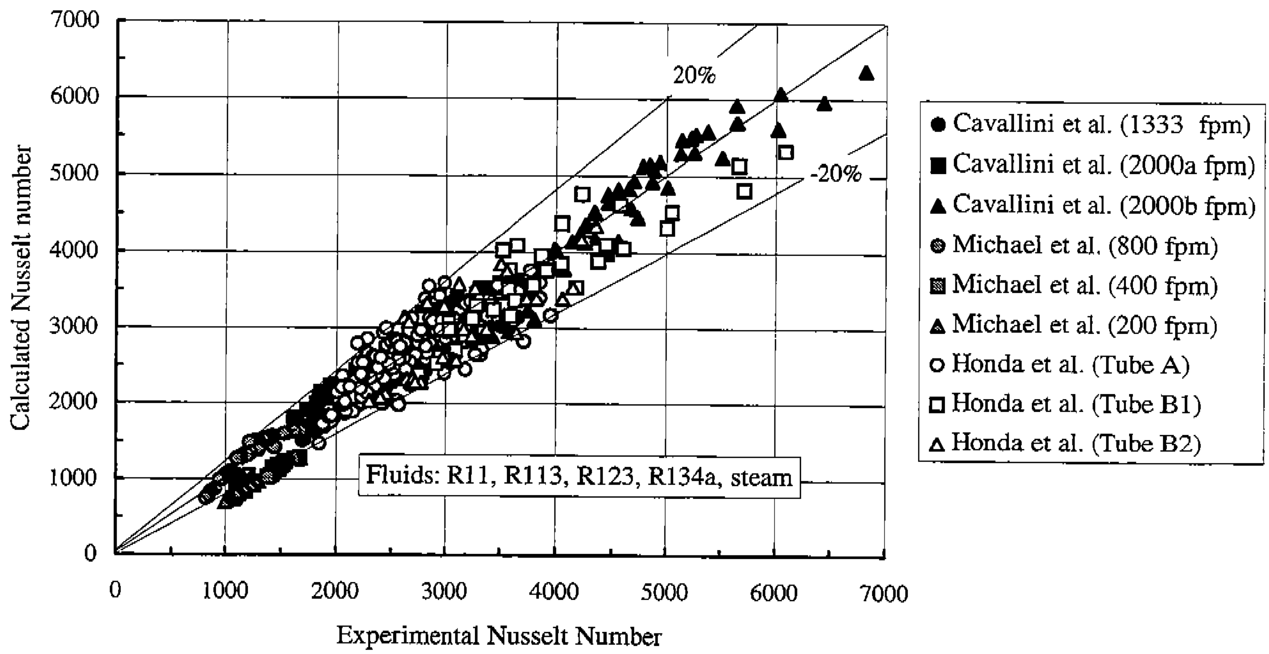

Figure 1. Comparison between the experimental and calculated Nusselt numbers (eq. 1-4).

The 226 points with $Re_{eq} < 22{,}000$ (< 10,000 for steam) were predicted by Briggs & Rose (1994) model with an absolute mean deviation E of 10.4%. The 404 points with $Re_{eq} > 22{,}000$ (> 10,000 for steam) were predicted by equation (1-4) used together with the model of Briggs & Rose (1994) for the stationary vapour component with an absolute mean deviation E of 9.9%. Figure 1 shows the relative comparison between the calculated and experimental Nusselt numbers for all the available experimental data from different sources.

CONDENSATION IN BUNDLES WITH INUNDATION

Heat transfer during condensation on bundles of integral-fin tubes at low vapor velocity is influenced by tube geometry and layout, surface tension and gravity forces and condensate inundation. For stationary vapour conditions a regular column flow occurs and the falling condensate interests only limited portions of the tubes under the columns, while the portions between the columns operate as single tubes. Only a moderate inundation effect has been observed for quiescient vapour conditions (Honda et al. 1989b, 1991).

Marto (1987) reports the following inundation coefficient based on Katz and Geist (1948) data:

$$\alpha_N/\alpha_1 = N^{0.96} - (N-1)^{0.96} \tag{5}$$

where α_N is the heat transfer coefficient on the N-th row, α_1 is the heat transfer coefficient for condensation without condensate inundation, at the same temperature difference, N is the row number in the flow direction. Blanc (1994) and Signe et al. (1995) suggested 0.93 instead of 0.96, as the exponent.

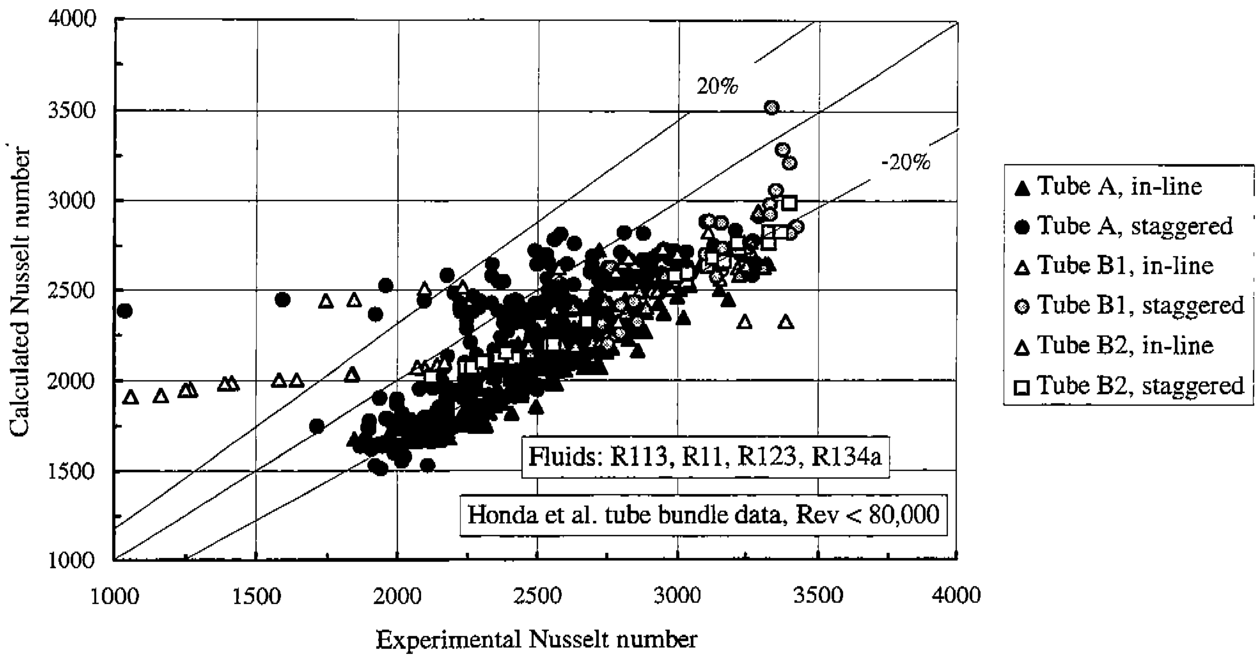

Figure 2. Comparison between the experimental and calculated Nusselt numbers by modified Briggs & Rose model.

Fig. 2 compares available experimental data by Honda et al. (1989b, 1991, 1995a, 1995b, 1996) on in-line and staggered finned bundles with inlet vapour Reynolds number Re_G less than 80,000 against predictions of Briggs and Rose (1994) equation multiplied by the inundation factor (equation (5)). For staggered bundles the tube row number was obtained by numbering only those tubes directly beneath one another. Table 1 lists test tubes main geometrical characteristics. Horizontal and vertical tube pitches of the bundles were 22 mm. The 591 points were predicted with an absolute mean deviation E of 14.2%. Only 59 data measured on tube B2 (in-line bundles) cannot be predicted with E better than 20%. This tube, which has narrow interfin spacings, shows a sharp decrease of in-line data with inundation. This is probably due to the fact that the vertical distance between tubes is smaller for in-line than for staggered bundles, resulting in a lower velocity of falling condensate on the lower tube, as suggested by Honda et al. (1995b).

At high vapor velocity, a relevant inundation effect was observed by Honda et al. (1989b, 1991). This behaviour can be explained by the different flow patterns observed: when vapour velocity increases, there is a transition to a dispersed flow and the falling condensate interests the whole surface of the inundated tubes. In the region where vapour shear stress effects are important, the heat transfer coefficient can be calculated by the asymptotic model eq. (1) where the first asymptote α_{st} is the heat transfer coefficient under stationary vapour conditions computed with the Briggs and Rose (1994) equation multiplied by the inundation factor (equation (5)), whereas the second asymptote α_{fc} is the heat transfer coefficient when forced convection mechanisms are predominant. α_{fc} is the product of the value for single tube (equations (2-4)) multiplied by the same inundation factor, in the form:

$$\alpha_N = \left(\alpha_{st}^2 + \alpha_{fc}^2\right)^{1/2} \cdot \left(N^{0.96} - (N-1)^{0.96}\right) \qquad (6)$$

Fig. 3 compares data by Honda et al. (1989b,1991) at high vapor velocity (inlet vapour Reynolds number greater than 80,000) against predictions of the above model. In the computation the experimental values of vapour velocity have been used. The 832 points were predicted with an absolute mean deviation E of 9.7%.

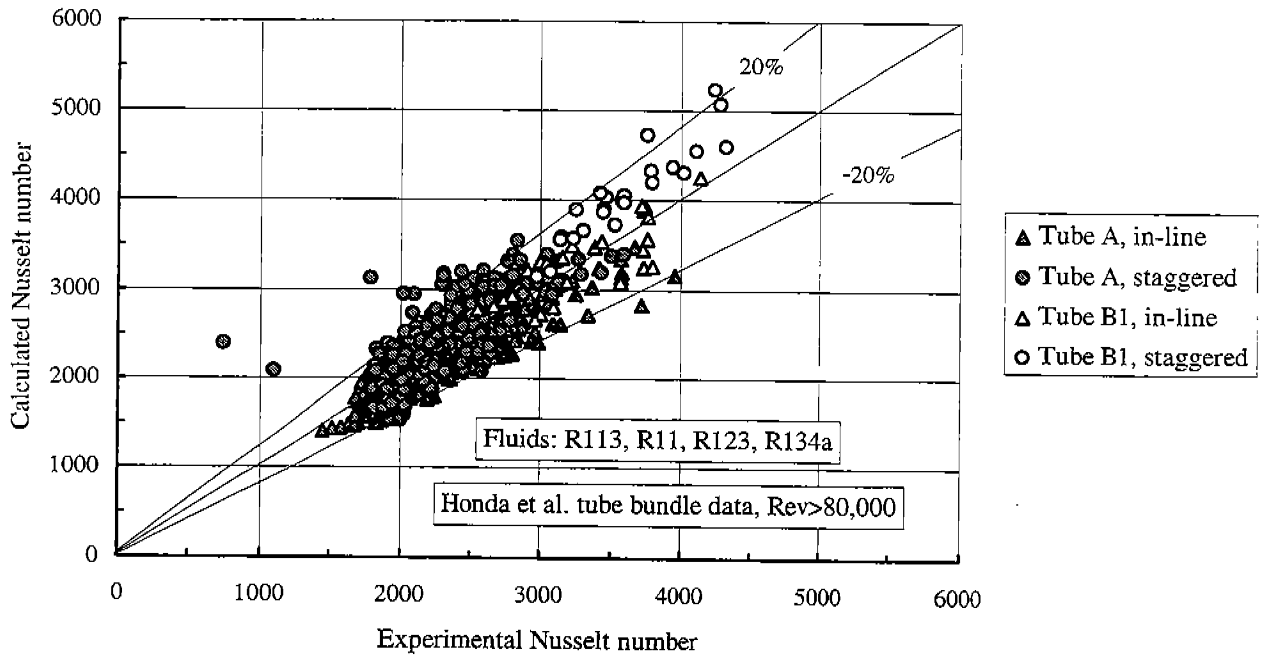

Figure 3. Comparison between the experimental and calculated Nusselt numbers by suggested model (eq. 6).

CONCLUSIONS

The first part of this work has compared a computational procedure of the heat transfer coefficient during condensation on a single horizontal integral-fin tube with available experimental data. The method accounts for the shear stress effect, gravity and surface tension forces. The 226 points with stagnant vapour condensation were predicted by Briggs & Rose (1994) model with an absolute mean deviation of 10.4%. The 404 points with forced convection condensation were predicted by equation (1-4) used together with the model of Briggs & Rose (1994) for the stationary vapour component with an absolute mean deviation of 10.1%.

This work also presents a computational method of the heat transfer coefficient realized during condensation on a bundle of horizontal integral-fin tubes. The proposed equations quantify the combined effects of shear, gravity and surface tension forces and of condensate inundation. The 591 points with stagnant vapour condensation were predicted by the Briggs and Rose (1994) equation multiplied by the inundation factor (equation (5)) with an absolute mean deviation of 14.2%. The 832 points with forced convection condensation were predicted by an asymptotic model presented above with an absolute mean deviation of 9.7%.

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