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Technical Note

Experimental analysis for the determination of the convective heat transfer coefficient by measuring pressure drop directly during annular condensation flow of R134a in a vertical smooth tube

A.S. Dalkilic [a,*], I. Teke [a], S. Wongwises [b,*]

[a] Heat and Thermodynamics Division, Department of Mechanical Engineering, Yildiz Technical University, Yildiz, Besiktas, Istanbul 34349, Turkey
[b] Fluid Mechanics, Thermal Engineering and Multiphase Flow Research Lab. (FUTURE), Department of Mechanical Engineering, King Mongkut's University of Technology Thonburi, Bangmod, Bangkok 10140, Thailand

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ABSTRACT

This study investigated the direct relationship between the measured condensation pressure drop and convective heat transfer coefficient of R134a flowing downward inside a vertical smooth copper tube having an inner diameter of 8.1 mm and a length of 500 mm during annular flow. R134a and water were used as working fluids on the tube side and annular side of a double tube heat exchanger, respectively. Condensation experiments were performed at mass fluxes of 260, 300, 340, 400, 456 and 515 kg m^{-2} s^{-1} in the high mass flux region of R134a. The condensing temperatures were around 40 and 50 °C; the heat fluxes were between 10.16 and 66.61 kW m^{-2}. Paliwoda's analysis, which focused mainly on the determination of the two-phase flow factor and two-phase length of evaporators and condensers, was adapted to the in-tube condensation phenomena in the test section to determine the condensation heat transfer coefficient, heat flux, two-phase length and pressure drop experimentally by means of a large number of data points obtained under various experimental conditions.

1. Introduction

Over the years, there are a large number of papers on the in-tube condensation heat transfer coefficient and pressure drop. The total pressure drop of two-phase flow in tubes is combination of the frictional, accelerational, and gravitational components. Determination of accelerational and gravitational components depends on the void fraction, and in a similar way, determination of either the two-phase friction factor or the two-phase frictional multiplier is required to compute the frictional component of pressure drop. Commonly, the computation of the condensation heat transfer coefficients in smooth tubes has been performed by empirical methods which are mostly the modifications of the Dittus–Boelter single-phase forced convection correlation [1], as in Akers et al. [2], Cavallini and Zecchin [3], and Shah [4].

The investigation of the characteristics of downward condensation heat transfer and pressure drop of refrigerants in small diameter vertical tubes has received relatively little interest in comparison to those in horizontal tubes. One of the most important flow pattern is annular two-phase flow which is characterized by a phase interface separating a thin liquid film from the gas flow in the core region. Two-phase annular flow occurs usually in the processes of film heating and cooling, mainly in power generation and especially in nuclear reactors. This flow pattern has taken the most notice, both analytically and experimentally regarding with its practical significance.

The aim of this work was to show the independence of an annular flow model from tube orientation and the general applicability for a relatively short vertical tube. Although convective heat transfer coefficient and pressure drop models and correlations have been extensively developed in the past, no attention has been paid to the determination of the convective heat transfer coefficient by direct measurement of the pressure drop. Experiments included a wide range of high mass fluxes with different saturation temperatures. Apart from the authors' previous publications [5–15], few studies exist in the literature on the investigation of heat transfer characteristics in a small diameter vertical tube during co-current downflow condensation. Furthermore, there is no study with the experimental parameters and content used here in the open literature. In this study, determination of the shear-dominated forced convective condensation heat transfer coefficients for refrigerant 134a is presented by means of Paliwoda's model [16], by relating the heat transfer to the pressure gradient depending mainly on the two-phase flow parameter during annular flow. In addition, the measured heat flux and two-phase length were validated by

* Corresponding authors. Tel.: +90 212 3832819; fax: +90 212 2616659 (A.S. Dalkilic), tel.: +66 2 470 9115; fax. +66 2 470 9111 (S. Wongwises).

E-mail addresses: dalkilic@yildiz.edu.tr (A.S. Dalkilic), somchai.won@kmutt.ac.th (S. Wongwises).

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Nomenclature

A	inside surface area (m^{-2})
d	internal tube diameter (m)
f	friction coefficient
G	mass flux ($kg\ m^{-2}\ s^{-1}$)
h	heat transfer coefficient ($W\ m^{-2}\ K^{-1}$)
i_{fg}	latent heat of condensation ($J\ kg^{-1}$)
L	length of test tube (m)
m	mass flow rate ($kg\ s^{-1}$)
Re	Reynolds number
T	temperature (°C)
Q	heat transfer rate (W)
x	mean vapour quality
z	axial coordinate
ΔP	pressure drop (Pa)

Greek symbols

μ	dynamic viscosity ($kg\ m^{-1}\ s^{-1}$)
υ	specific volume ($m^3\ kg^{-1}$)
β	two-phase flow factor
θ	dimensionless ratio of liquid–vapour pressure gradient
ε	dimensionless two-phase parameter

Subscripts

exp	measured
F	frictional
g	gas/vapour
i	inlet
l	liquid
o	outlet
ph	preheater
ref	refrigerant
sat	saturation
TS	test section
w	water
wi	inner wall

means of a large number of data points obtained under various experimental conditions.

2. Experimental apparatus and method

The detailed information on the experimental apparatus and test section can be seen from authors' previous publications [5–13].

3. Data reduction and experimental uncertainty

The equations of data reduction section regarding the inlet vapour quality of the test section, the outlet vapour quality of the test section, the average heat transfer coefficient and uncertainty analysis can be obtained from authors' previous publications [5–13].

3.1. The average heat transfer coefficient

$$h_{exp} = \frac{Q_{TS}}{A_{wi}(T_{ref,sat} - T_{wi})} \tag{1}$$

where h_{exp} is the experimental average heat transfer coefficient, Q_{TS} is the heat transfer rate in the test section, T_{wi} is the average temperature of the inner wall, $T_{ref,sat}$ is the average saturation temperature of the refrigerant at the test section inlet and outlet, and A_{wi} is the inside surface area of the test section:

$$A_{wi} = \pi\, dL \tag{2}$$

4. Calculation procedure for the generalized two-phase model

According to Paliwoda [16], the pressure gradient of any two-phase flow depends on liquid and vapour phase gradients and the mixture quality of the liquid–vapour phases, and he used Müller-Steinhagen and Heck's correlation [17] in his analysis due to its simple mathematical form. This can be seen in Eq. (3), which has the minimum number of correlating parameters, for the prediction of a two-phase flow gradient as follows:

$$\frac{dP_F}{dL} = \left[\left(\frac{dP_F}{dL}\right)_l + 2\left\{\left(\frac{dP_F}{dL}\right)_g - \left(\frac{dP_F}{dL}\right)_l\right\}x\right](1-x)^{1/3} + \left(\frac{dP_F}{dL}\right)_g x^3 \tag{3}$$

where the Darcy–Weisbach equation is used to calculate the liquid phase gradient $\left(\frac{dP_F}{dL}\right)_l$ and the vapour phase gradient $\left(\frac{dP_F}{dL}\right)_g$ as follows:

$$\frac{dP_F}{dL} = f\frac{G^2\upsilon}{2d} \tag{4}$$

where the friction coefficient can be calculated by means of the Hagen-Poiseulle and Blasius equations according to the Reynolds numbers of the liquid or vapour as follows:

$$f = \frac{64}{Re} \quad \text{for } Re = \frac{Gd}{\mu} \leqslant 1187 \tag{5}$$

$$f = \frac{0.3164}{Re^{0.25}} \quad \text{for } Re = \frac{Gd}{\mu} > 1187 \tag{6}$$

Paliwoda [16] defined the two-phase flow factor (β) regarding the transformation of two-phase flow pressure gradients from any vapour phase pressure gradient at any temperature using mixture quality with the liquid–vapour pressure ratio parameter of θ. He paid attention to the validation range of his model developed for boiling and condensing refrigerants for a wide range of saturation temperatures. For that reason, he developed Müller-Steinhagen and Heck's correlation [17] due to its limited operating range of pressures. His two-phase flow factor can be seen in Eq. (7) as:

$$\beta = \frac{\left(\frac{dP_F}{dL}\right)}{\left(\frac{dP_F}{dL}\right)_g} \tag{7}$$

Eq. (7) is obtained by dividing both sides by the liquid phase gradient $\left(\frac{dP_F}{dL}\right)_l$ in Eq. (3), and is presented in Eq. (8) after some rearrangements as follows:

$$\beta = [\theta + 2(1-\theta)x](1-x)^{1/3} + x^3 \tag{8}$$

where the dimensionless ratio of the liquid–vapour pressure gradient (θ) can be calculated by means of the Hagen-Poiseulle and Blasius equations according to the Reynolds numbers of liquid or vapour as follows:

$$\theta = \frac{64}{0.3164}\frac{\mu_l}{\mu_g^{0.25}}\frac{\upsilon_l}{\upsilon_g}(Gd)^{-0.75} \quad \text{for } Re_l \leqslant 1187 \text{ and } Re_g > 1187 \tag{9}$$

$$\theta = \frac{\upsilon_l}{\upsilon_g}\left(\frac{\mu_l}{\mu_g}\right)^{0.25} \quad \text{for } Re_l > 1187 \text{ and } Re_g > 1187 \tag{10}$$