International Journal of Heat and Mass Transfer 91 (2015) 494–501

Contents lists available at ScienceDirect

International Journal of Heat and Mass Transfer

journal homepage: www.elsevier.com/locate/ijhmt

Reflux condensation of steam inside a short vertical large diameter tube

P.V. Muratov, R.I. Pashkevich *

Research Geotechnological Center, Far Eastern Branch, Russian Academy of Sciences, Severo-vostochnoe shosse 30, 683002 Petropavlovsk-Kamchatsky, Russian Federation

ARTICLE INFO

Article history:
Received 12 June 2015
Received in revised form 15 July 2015
Accepted 17 July 2015
Available online 13 August 2015

Keywords:
Condensation
Upward steam flow
Vertical tube
Large diameter
Nusselt number
Correlation

ABSTRACT

An experimental investigation of the reflux steam condensation inside a vertical tube at the atmospheric pressure is presented. The data were obtained for duralumin tube with the internal diameter of 200 mm, length of 220 mm and roughness height less than 0.05 mm. The film and vapor Reynolds numbers were varied within the range of 1–25 and 790–3680, respectively. The data were lower than Nusselt theory prediction. In contrast to Nusselt theory according to which the modified Nusselt number decreases with the film Reynolds number, the obtained data show inverse dependence when the film Reynolds number is less than ten. For fixed vapor Reynolds numbers there are the maximum values of the Nusselt number for definite film Reynolds numbers. The correlations are suggested for the Nusselt number in the case of a constant and variable wall temperature along the tube. The general correlation based on the presented and earlier published experimental data for the tubes with different height and diameter ratio is suggested in addition.

1. Introduction

The heat transfer during laminar film condensation inside vertical tubes with upward vapor flow and inside closed thermosyphons was theoretically and experimentally investigated in [1–25]. The part of the experimental works was devoted to the research of a mean heat transfer coefficient (Table 1).

The reviews of investigations of countercurrent condensation in closed thermosyphons and tubes are presented in the papers of Gross [14] and Lips and Meyer [15]. The paper [25] presents an overview of [1–15] and shows that there are only few experimental investigations of laminar film condensation inside tubes with upward steam flow. Majority investigations were performed for the tubes with large aspect ratios ($AR > 10$). Small amount of the experimental works is devoted to the studies of heat transfer in a reflux film condensation inside the tubes with a large diameter. Experimental values of the Nusselt number fell below Nusselt prediction at a low film Reynolds number ($Re_f < 10$) [9,14,20,24,25].

Pohner and Desai [17] have developed the mathematical model of a steady steam condensation in a vertical circular tube for the cases of laminar and turbulent vapor cores and laminar liquid film. Computed values of the heat transfer coefficient well agreed with the experimental results of other authors for downward vapor flow. Recently Shabgard et al. [16] have presented the mathematical model of transient condensation in a closed thermosyphon. The results obtained by the authors [16] were in a good agreement with the experimental data obtained by Baojin et al. [23] and Jouhara and Robinson [24].

Shiraishi et al. [21] have experimentally studied the heat transfer mechanism of a two-phase closed thermosyphon and presented a mathematical model. A good agreement between the thermal resistance of the thermosyphon estimated from the experimental data and the prediction from the mathematical model was obtained [21]. Bezrodnyi and Moklyak [22] have conducted the experimental investigation of the hydrodynamic characteristics of a two-phase layer and heat transfer behavior under the characteristic operating conditions of the condensing section of a vertical closed thermosyphon. Abou-Ziyan et al. [19] have reported the experimental results of thermosyphon working under stationary and vibrated conditions for a case of water. Hashimoto and Kaminaga [20] have experimentally showed that at a low heat flux the condensation heat transfer in a thermosyphon is lower than that one predicted by Nusselt theory. Baojin et al. [23] have carried out the experimental investigations of heat transfer characteristics of titanium and copper two-phase thermosyphons filling by water. Authors [23] have found out that the Nusselt theoretical correlation is not suitable for simulating of the heat transfer coefficient in a condenser. Jouhara and Robinson [24] have obtained the experimental data for a small diameter two-phase closed thermosyphon charged with water and discovered that Nusselt equation notably over predicts the measurements at a low film Reynolds number. The authors [24] have also showed that the experimental measurements follow very closely the Nusselt

* Corresponding author.
E-mail addresses: pashkevich@kscnet.ru, teplosnab@rambler.ru (R.I. Pashkevich).

http://dx.doi.org/10.1016/j.ijheatmasstransfer.2015.07.075
0017-9310/

Nomenclature

AR	aspect ratio of the tube, $AR = H/D$
D	inner diameter of measuring section
g	gravitation acceleration
r	latent heat
G	mass flow rate
H	height of the measuring section
$\bar{L}$	characteristic length, $\bar{L} = (\bar{v}_f^2/g)^{1/3}$
max	maximum value
ΔT	steam and inner wall temperatures difference in the measuring section
Subscripts	
f	film
i	inlet of the measuring section
l	laminar
lw	laminar-wavy
v	vapor
Superscript	
—	mean value
Greek letters	
α	heat transfer coefficient
Δ	roughness height
ε	condensate and vapor rate ratio, i.e. ratio of condensate mass flow rate to vapor mass flow rate at the inlet of the measuring section
λ	thermal conductivity
μ	dynamic viscosity
μ^*	viscosity ratio, $\mu^* = \mu_{v,i}/\mu_{f,i}$
v	kinematic viscosity
Dimensionless numbers	
$\overline{Nu}^*$	modified Nusselt number, $\overline{Nu}^* = \bar{\alpha}\bar{L}/\bar{\lambda}_f$
Re	Reynolds number

number versus the film Reynolds number dependency of the Hashimoto and Kaminaga [20] correlation.

Lips and Meyer [15] marked that additional experimental investigations are needed for better understanding of condensation in vertical tubes at low mass fluxes, particularly for upward vapor flow. Recently the experimental data for heat transfer in a film condensation inside a large diameter rough pipe with roughness height $\Delta = 0.14$ mm and relative roughness height $\Delta/D = 7 \times 10^{-4}$ with upward steam flow were presented in [25]. It was found that the modified Nusselt number values increased with the vapor Reynolds number and depended on the film Reynolds number, viscosity ratio and condensate and vapor rate ratio [25]. The presented above literature review shows that heat transfer in reflux condensation has unsufficiently been studied, especially at a low condensate flow rate in a large diameter tube with a low aspect ratio.

It is known that heat transfer coefficients in a film condensation at a rough surface are lower than at a smooth surface [26]. There is an interest to investigate heat transfer in a reflux condensation inside a smooth pipe at a low film Reynolds number. The purpose of the present paper is to study the laminar condensation inside a smooth vertical tube of a large diameter (0.2 m), small aspect ratio (1.1) and relative roughness height of the inner tube wall less than 2.5×10^{-4} with upward steam flow of small flow rate (1.5×10^{-3}–7.1×10^{-3} kg/s) and develop the correlations for the Nusselt number. The accompanying aim of current research is to investigate the influence of roughness on the heat transfer coefficient during reflux condensation.

2. Experimental setup

The test facility is schematically drawn in Fig. 1. The measuring module consists of three sections (S1, S2 and S3) with the cooling jackets. To maintain the saturation conditions under the atmospheric pressure at the inlet of the measuring section S2 the superheater (H) is installed after the cyclone separator (C). The measuring section (S2) and condensation section (S3) are cooled by water. The water flow rate is adjusted for maintenance of a constant temperature of the section S2 inner wall in every experimental series. The condensate volumes running in the sections S2 and S3 are measured by the vessels V1 and V2. The condensate gathering rings are installed in *i*–*i* and *o*–*o* cross sections. The jacket of the stabilizing section (S1) is filled by air. Sections S1 and S2 are thermally insulated. Characteristics of the measuring module are given in Table 2.

Uncertainties of measured variables are presented in Table 3.

Detailed description of the test facility is given in [25]. All experiments were carried out under a partial steam condensation in the measuring section and subsequent full condensation in the condensation section. The internal surface of the measuring section was treated by polishing paste and it had a roughness height Δ less than 0.05 mm and relative roughness height less than $\Delta/D = 2.5 \times 10^{-4}$. The values of characteristic length are less than a wall roughness height. The internal surface of the measuring section was cleaned by a water solution of 95 percentages of alcohol. Before the experiment the steam blowing of the measuring module with steam exhausting through the tube section on the upper flange was proceeded. Then during at least a quarter of an hour the test facility worked in a full condensation regime only in a condensation section and the condensate was removed from the measuring module. Water degassing was produced during the boiling in a steam generator.

3. Data reduction

Accordingly the classical Nusselt model for laminar film condensation of a quiescent vapor along an isothermal vertical plate, the mean modified Nusselt number is defined as [14,27,28]:

$$\overline{Nu}_l^* = 0.925 Re_f^{-\frac{1}{3}} \tag{1}$$

where the Re_f is the film Reynolds number at the lower end of cooling zone.

In the laminar-wavy range ($Re_f > 2$) from Uehara et al. [14]:

$$\overline{Nu}_{lw}^* = 0.884 Re_f^{-\frac{1}{4}}. \tag{2}$$

The vapor Reynolds number at the inlet of the measuring section (S2) is evaluated as:

$$Re_{v,i} = \frac{4G_{v,i}}{\pi D \mu_{v,i}}. \tag{3}$$

The film Reynolds number at the cross section *i*–*i* is evaluated as (Fig. 1):