International Journal of Heat and Mass Transfer 94 (2016) 403–410

Contents lists available at ScienceDirect

International Journal of Heat and Mass Transfer

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

Experimental analysis of steam condensation in vertical tube with small diameter

Milan Kubín, Jiří Hirš, Josef Plášek *

Brno University of Technology, Faculty of Civil Engineering, Veveří 331/95, Brno 602 00, Czech Republic

ARTICLE INFO

Article history:
Received 14 September 2015
Received in revised form 9 November 2015
Accepted 9 November 2015
Available online 11 December 2015

Keywords:
Water steam
Condensation
Vertical tube
Heat transfer coefficient

ABSTRACT

Thermal design of tubular heat exchanger based on condensation heat of water steam requires knowledge of condensation heat transfer in the each tube. This paper is just aimed on experimental analysis of steam condensation in vertical copper tube in length of 1285 mm with 2.0 mm inner diameter and 0.5 mm wall thickness. Experimental measurement is performed in 12 steps with variable inlet temperature and mass flow rate of water steam. The heat transfer coefficient on the inner surface of tube in condensation zone is calculated by Thermal resistance method and Wilson plot method. The correlation quality of results obtained from both methods is 98.8%. The results are compared with other experimental studies and also correlated with five chosen equations for prediction of condensation heat transfer coefficient. The correlation quality of results obtained from this experimental analysis and four tested equations is over 96.6%, only theoretically determined Nusselt equation undervalues condensation heat transfer coefficient as is known. The Nusselt equation does not take to account waves on condensate surface. These waves on condensate surface are caused by flow of water steam in tube and the wave's effect is approximately 20.5% in this presented case.

1. Introduction

Heat exchangers are commonly part of technology systems and tubular heat exchanger is one of often applied types. The most efficient tubular heat exchangers use latent heat of fluids as is phase change from liquid to gas (evaporation) or reverse phase change from gas to liquid (condensation). Thermal design of tubular heat exchanger based on condensation heat requires knowledge of phase change process in the tubes. This paper is focused on experimental analysis of steam condensation in vertical tube with small diameter.

The first article about laminar film condensation is published by Nusselt [1] in 1916, where Nusselt analytically expressed condensation heat transfer coefficient dependent on amount of steam condensate. The Nusselt equation (see Eq. (11)) assumes smooth and uniform liquid film on wall surface and condensation heat transfer coefficient is equal to ratio of thermal conductivity and thickness of film condensate. The effect of sub-cooling condensate on surface wall is published later by Bromley [2] and non-linear temperature distribution in film condensate is studied by Rohsenow [3]. The classical Nusselt theory is also extended regard to momentum changes of film condensate by Sparrow [4–6] and stability of laminar flow down of film condensate is published by Bankoff [7] or Marschall and Lee [8] and recently others [9–11].

The Nusselt equation with assumption of smooth liquid film on wall surface is valid for stationary steam, because the flowing steam in tube causes waves on condensate surface and these waves improve condensation heat transfer. The wave's effect is studied by Kapitsa [12] in 1948 and later McAdams [13] recommended multiplied Nusselt equation by the wave's effect factor 1.2 as a discrepancy correction between experimental results and theoretical Nusselt solution. The Nusselt equation is increased about 20.6% by Whitham [14] as the wave's effect on condensation heat transfer coefficient, see Eq. (12). The next theoretically determined equation (see Eq. (13)) which includes the wave's effect is published by Hobler [15] and the wave's effect is continuously studied for example in [16–19]. The equation for prediction of condensation heat transfer coefficient can be also determined by experimental way and usually is formulated in exponential function. The bases of exponential function are often fluid properties (*Nu*, *Pr*, *Re* etc.) for example Hausen [20] in Eq. (15) or boundary conditions (q, p, ΔT etc.) where the exponents of bases are determined by experimental measurement, for example Kutateladze [21] in Eq. (14).

* Corresponding author.
E-mail address: plasek.j@fce.vutbr.cz (J. Plášek).

http://dx.doi.org/10.1016/j.ijheatmasstransfer.2015.11.022
0017-9310/

Nomenclature

Latin symbols

A	slope parameter of linear regression function [m^{-1}]
B	constant term of linear regression function [$m\,K\,W^{-1}$]
c	specific thermal capacity [$J\,kg^{-1}\,K^{-1}$]
C	multiple constant in exponential function [–]
d	characteristic length in Nusselt number [m]
D	diameter of tube [m]
g	gravity acceleration [$m\,s^{-2}$]
h	specific enthalpy [$J\,kg^{-1}$]
H	level of steam condensate [m]
k	overall heat transfer coefficient [$W\,m^{-1}\,K^{-1}$]
l_{23}	latent heat of phase change [$J\,kg^{-1}$]
L	total length of tube [m]
m	mass flow rate [$kg\,s^{-1}$]
n	total count of tubes [pcs]
p	static pressure [Pa]
q	specific heat flux [$W\,m^{-2}$]
Q	total heat flux [W]
R	Thermal resistance [$m\,K\,W^{-1}$]
t	temperature in Celsius scale [°C]
T	temperature in Kelvin scale [K]
ΔT	logarithmic mean temperature difference [K]
V	volume flow rate [$m^3\,s^{-1}$]
x	variable on x-axis [–]
y	variable on y-axis [–]

Greek symbols

α	heat transfer coefficient [$W\,m^{-2}\,K^{-1}$]
ε	percentage differences [%]
λ	thermal conductivity [$W\,m^{-1}\,K^{-1}$]
μ	dynamic viscosity [Pa s]
ν	kinematic viscosity [$m^2\,s^{-1}$]
π	mathematical constant [–]
ρ	bulk density [$kg\,m^{-3}$]
σ	surface tension [$N\,m^{-1}$]

Subscripts

c	condensate
e	external
i	internal
min	minimal value
T	wall of tube
v	vapour
w	water
in	inlet
out	outlet

Dimensionless numbers

Nu	Nusselt number
Pr	Prandtl number
Re	Reynolds number

The theme of water steam condensation is continuously studied in many articles by theoretical or experimental way, for example [22–25] and recently others [26–29]. Water steam condensation in vertical tube is also recently studied in article [30], but the study is for wide tube with ratio of length to inner diameter $L/D = 1.10$. The purpose of this paper is parametric experimental study of vertical copper tube with ratio of length to inner diameter $L/D = 642.50$ [–]. Impact of variable inlet temperature and mass flow rate of water steam on condensation heat transfer coefficient is also studied. The obtained results are correlated with five chosen equations for prediction of condensation heat transfer coefficient and compared with other experimental studies [25,31–34]. The wave's effect on condensation heat transfer coefficient is evaluated, too.

2. Experimental setup

The experimental analysis is realized in 12 steps on vertically oriented copper tube in length of 1285 mm with 2.0 mm inner diameter and 0.5 mm wall thickness. Tubes are intentionally measured in bundle of 37 tubes, because some small geometric imperfections of each tube are eliminated. Concurrently measured values in the bundle more correspond with statistical average and edge effect is eliminated, too. The bundle of 37 tubes is surrounded by outer copper tube of diameter 400 mm. The experimental setup can be for description divided to loop of water steam and loop of cooling water, see Fig. 1.

The loop of water steam is composed from steam generator (A), where water steam is produced with known temperature $t_{v,in}$ [°C] and pressure $p_{v,in}$ [Pa]. After that water steam enters into measured bundle of 37 copper tubes (B), where the condensation process is realized. Volume flow rate of condensate V_c [m^3/s] and temperature of condensate $t_{c,out}$ [°C] is measured on outflow from the bundle before collection tank (C). The interspace of bundle is counter-flow cooled by loop from source of cold water (D). Inlet temperature $t_{w,in}$ [°C] and volume flow rate V_w [m^3/s] of cooling water is monitored on enter to the bundle of tubes. Outlet temperature $t_{w,out}$ [°C] and pressure $p_{w,out}$ [Pa] of cooling water is measured on outflow from the bundle of tubes. The volume changes of cooling water are compensated by expansion vessel (E). The steam condensate level H [m] in the measured tubes is displayed on external gauge level (G).

The inlet temperature of water steam is changed in 12 steps during the experimental measurement in range from $t_{v,in} = 100.2$ °C to $t_{v,in} = 117.9$ °C. Concurrently is changed mass flow rate of water steam in range from $m_v = 0.00898$ kg/s to $m_v = 0.01154$ kg/s. The changed input parameters of water steam are kept for a sufficiently long period to obtain of thermal steady state, according to monitored values. Mass flow rate $m_w = 0.274 \pm 0.001$ kg/s and inlet temperature $t_{w,in} = 11.0 \pm 0.2$ °C of cooling water is almost on constant value for the whole time, more Table 1. The uncertainty of certificated gauges is on temperature sensor ±0.3%, pressure gauge ±0.6%, external gauge level ±0.2% and uncertainty of volume flow rate is ±0.5%. The internal surface of measured tubes is cleaned by high percentage alcohol cleaner and the purity of water steam from steam generator is over 99.9%

3. Solution methods

The transferred condensation heat Q_v [W] between water steam and cooling water is calculated from Eq. (1), where specific enthalpy of water steam condensate is $h_{c,out} = 419.10$ kJ/kg and condensation temperature is $t_{v,out} = 100$ °C. The logarithmic mean temperature difference ΔT [K] for counter-flow involvement is determined from Eq. (2). The one-dimensional state steady overall heat transfer coefficient k [W/(m K)] for cylindrical wall in