Applied Thermal Engineering 71 (2014) 378–388

Contents lists available at ScienceDirect

Applied Thermal Engineering

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

Effect of sensible heat, condensation in superheated and subcooled region incorporated in unified model for heat rejection in condensers in horizontal round smooth tubes

Rahul Agarwal [a], Pega Hrnjak [a, b, c, *]

[a] Air Conditioning and Refrigeration Center, Department of Mechanical Science and Engineering, University of Illinois at Urbana-Champaign, 1206 West Green Street, Urbana, IL 61801, USA
[b] Creative Thermal Solutions, 2209 North Willow Road, Urbana, IL 61802, USA
[c] Institute of Refrigeration and Biotechnologies, ITMO University, Saint-Petersburg, Russia

HIGHLIGHTS

- Review of current models for condensation and proposal of 5 zones for heat rejection.
- Discussion of proposed model which includes the effect of sensible heat rejection in condensation.
- Validation of model with experimental data for three fluids R1234ze(E), R134a and R32 at various operating conditions.
- Qualitative discussion of condensation in superheated, two-phase and subcooled zone separately by comparing experimental data with model results.

ARTICLE INFO

Article history:
Received 7 August 2013
Accepted 22 May 2014
Available online 10 June 2014

Keywords:
Heat transfer
Condensation
Model
Horizontal
Subcooled
Superheated

ABSTRACT

Heat transfer in condensers is typically divided into 3 zones: superheated, two-phase and subcooled .The heat transfer coefficients (HTC) for these zones show discontinuity at qualities of 0 and 1, where two-phase correlations meet single phase flow correlations. The paper presents a model that bridges that discontinuity. It takes into account the condensation in de-superheating zone and subcooled liquid in two-phase zone which results in condensation in subcooled region. Cavallini et al. (2006) and Gnielinski correlations have been used as a baseline correlations in the model to calculate HTC in two-phase and single phase zone respectively. The model has been compared to experimental data in condensation for R1234ze(E), R134a and R32 conducted from superheated to subcooled region and predicts the HTC satisfactorily within an accuracy of 16% for these fluids.

Published by Elsevier Ltd.

1. Introduction

Heat transfer in condensers is usually modeled in the two-phase zone with various correlations predicting HTC as a function of quality, mass and heat flux. These models implicitly assume thermodynamic equilibrium during condensation and in single phase regions. The modeling of the process in de-superheating zone assumes that the first drop of condensate forms when bulk enthalpy of vapor reaches saturation enthalpy. This assumption is not entirely correct as the phenomena of condensation in superheated zone have been acknowledged by many. Kondou and Hrnjak [1] identified the presence of condensation in superheated region for CO_2. They identified the criteria for beginning of condensation to be the point where the wall temperature drops below saturation temperature of the refrigerant at corresponding operating condition. Kondou and Hrnjak [2] also conducted experiments with R410A and termed the region exhibiting condensation in superheated zone as Condensing Superheated (CSH) Zone. The HTC in CSH zone was shown to be much higher compared to the prediction by Gnielinski correlation. The importance of CSH zone is shown through experiments conducted for CO_2 near critical pressure where the latent heat of the refrigerant is small [3]. The Kondo–Hrnjak correlation was then proposed based on the argument that the heat rejection in CSH zone is a combination of sensible and latent heat which satisfied their data satisfactorily. Lee et al. [4] experimentally investigated condensation in superheated R22 vapor proposed a model taking sensible heat rejection into account.

---

* Corresponding author. Air Conditioning and Refrigeration Center, Department of Mechanical Science and Engineering, University of Illinois at Urbana-Champaign, 1206 West Green Street, Urbana, IL 61801, USA. Tel.: +1 217 244 6377; fax: +1 217 244 6534.

E-mail address: pega@illinois.edu (P. Hrnjak).

http://dx.doi.org/10.1016/j.applthermaleng.2014.05.071
1359-4311/Published by Elsevier Ltd.

Nomenclature

P	pressure [Pa]
T	temperature [°C]
H	specific enthalpy [J kg^{-1}]
Cp	Specific heat capacity [J kg^{-1} K]
x	thermodynamic vapor quality [−]
f_b	friction factor [−]
$\dot{Q}$	heat transfer rate [W]
ΔZ	segment of test tube [m]
$\dot{m}$	mass flow rate [kg s^{-1}]
G	mass flux [kg m^{-2} s^{-1}]
d	diameter of test tube [m]
Δh_{LV}	latent heat [J kg^{-1}]
g	gravitational acceleration [m s^{-2}]
Nu	Nusselt number [−]
Pr	Prandtl number [−]
HTC	see α
A	surface area [m^{-2}]
J_G	dimensionless gas velocity [−]
J_G^T	transition gas velocity [−]
X	Lockhart–Martinelli parameter [−]
Re	Reynolds number [−]

Greek symbols

α	(same as HTC) [W m^{-2} K^{-1}]
ρ	density [kg m^{-3}]
μ	viscosity [Pa s]
λ	thermal conductivity [W m^{-1} K]
δ	film thickness [m]
θ	angle [rad]
ε	void fraction [−]

Subscripts

TP	two-phase
i	inner
in	inlet
o	outer
out	outlet
LO	liquid only
Sensible	sensible heat
strat	fully stratified flow regime
V	vapor
L	liquid
r	refrigerant
w	tube wall
b	evaluated at bulk temperature
f	evaluated at film temperature
sat	evaluated at saturation temperature
SH	superheat
SC	subcool
latent	latent heat
tt	turbulent–turbulent

Webb [5] simplified this model by expressing the HTC as a function of two-phase and single phase HTC with the inclusion of the F-factor which asymptotically approaches 0 to satisfy the boundary condition at $x = 1$. Balekjian and Katz [6] investigated film condensation of superheated vapor on horizontal tubes for water and Freon-114. They looked into the temperature profile of refrigerant within the tube with both superheated vapor and liquid film existing simultaneously and defined interfacial film coefficient as a function of vapor superheat. These models have been proposed separately for CSH zone and not many attempts have been made to combine the effect of sensible heat rejection in two-phase models.

Condensation heat transfer in two-phase zones has been modeled using various approaches. Correlations proposed by Dobson and Chato [7], Jung et al. [8], Haraguchi et al. [9] predict HTC in two-phase zone with reasonable accuracy. These correlations consider the effect of fluid properties, parameters like Lockhart–Martinelli, HMFR (Heat Mass Flux Ratio) and single phase heat transfer etc. in their models. Shah formulated a simple dimensionless correlation analytically for predicting heat transfer coefficients during film condensation [10] with wide variety of experimental data for water, R11, R12, R22, R113, methanol, ethanol, benzene, toluene and trichloroethylene for condensation in horizontal, vertical and inclined pipes of diameter from 7 to 40 mm. The correlation worked very well universally for refrigerants with reasonable accuracy except for highly turbulent flows. The correlation was therefore modified again to fit into a wider range of parameters [11]. More physical models have been also made by taking into account the flow regimes at various qualities. Thome et al. [12] proposed a flow pattern map for condensation analogous to Kattan et al. [13] and expressed HTC as a function of convective and nusselt film condensation. The film thickness and wetted perimeter of the tube were calculated as a function of flow regime identified through the flow pattern map. The model predicted wide range of experimental data available in literature for various fluids with reasonable accuracy. Cavallini et al. [14] proposed a simplified correlation for heat exchanger design where HTC is predicted through two basic equations which also take the flow regime into account. The model proves to be quite successful with variety of fluid under wide range of operating conditions. Cavallini et al. [15] reported experimental data for R134a, R32 along with 3 other refrigerants for a mass flux of 100–750 kg m^{-2} s^{-1}, quality of 0.15–0.85 and saturation temperature of 30–50 °C. The study provided a good comparison for the experimental data shown in this work as the refrigerants and operating conditions are fairly similar. Hossain et al. [16] conducted experiments with R1234ze(E), R32 and R410A in horizontal tubes for mass flux of 100–450 kg m^{-2} s^{-1}, saturation temperature of 35–45 °C and compared their data with Haraguchi correlation with satisfactory agreement. These models, although captures the trend and physics behind the phenomena do not consider the effect of subcooling of liquid film. As a result they do not asymptotically satisfy single phase correlations at the exit of condensers. Hashizume et al. [17] analyzed the temperature profile of refrigerants near the exit of condensers at very low quality. They measured the temperature in adiabatic section at the exit of condensers and found the sub-cooling of approximately 10 °C and proposed a numerical model for condensation at vapor condensate surface. The paper proposes a unified model in condensers which takes the effect of condensation in de-superheating zone and sub-cooling in two-phase zone to show the smooth transition of HTC from single to phase zones.

2. Conventional approach in modeling of heat rejection in condensers

Heat transfer models in condensers usually categorize the process in de-superheating (single phase, typically turbulent), two-phase condensation and subcooled (single phase, laminar or turbulent) zones. The HTC in these zones are calculated independently in the design of heat exchangers as shown in Fig. 1(a). Condensation that