International Journal of Heat and Mass Transfer 78 (2014) 170–180

Contents lists available at ScienceDirect

International Journal of Heat and Mass Transfer

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

Numerical analysis of mixed-convection laminar film condensation from high air mass fraction steam–air mixtures in vertical tubes

F. Hassaninejadfarahani, M.K. Guyot, S.J. Ormiston *

Dept. of Mechanical Engineering, University of Manitoba, 75A Chancellors Circle, Winnipeg, MB R3T 5V6, Canada

ARTICLE INFO

Article history:
Received 19 July 2013
Received in revised form 13 June 2014
Accepted 13 June 2014
Available online 17 July 2014

Keywords:
Laminar film condensation
Vertical tube
Steam–air
Detailed model
High air mass fraction

ABSTRACT

A numerical analysis is performed for condensing downward flow from steam–air mixtures in vertical tubes with constant wall temperature. The focus of this study is on the case of high inlet non-condensing gas (air) mass fractions (greater than 0.80). The parabolic governing equations are solved for steady, axisymmetric, laminar flow in the liquid film and in the vapour–gas mixture. A complete two-phase model, based on the conservation of mass, momentum, and energy in each phase, is presented. Detailed results are presented in both the film and mixture regions. Those results include radial-direction profiles of axial velocity, temperature, and air mass fraction, as well as axial variation of film thickness, Nusselt number, interface and bulk temperatures, interface and bulk air mass fraction, and proportion of latent heat transfer. In all the cases studied the interface temperature is very close to the wall temperature and the condensation rates are small due to high interface air mass fractions.

The inlet relative humidity is varied from 30% to 100% and the inlet temperature is varied from 25 °C to 90 °C. These conditions cover the range of inlet air mass fraction approximately in the range from 0.865 to 0.995 for an inlet pressure of 1 bar. In addition, the effects of varying the inlet Reynolds number (500 to 2000), the wall temperature (5 °C and 15 °C), and the tube radius (2.5 mm to 10.0 mm) are examined. It was found that decreasing the inlet relative humidity reduced the heat transfer rate and that the condensate film thickness increased with an increase in the inlet Reynolds number, inlet-to-wall temperature difference, and inlet relative humidity. Due to high air mass fraction, in some cases only 40% or less of the total heat transfer to the tube wall was due to condensation. For a fixed Reynolds number, decreasing the tube radius increased the condensation rate.

1. Introduction

Condensation heat transfer is relevant to many industrial applications [1]. Applications involving condensing two-phase flow phenomena occur in the refrigeration, chemical processing, and thermal power generation industries. Condensation in the presence of high mass fraction of non-condensing gas is an important phenomenon in the design of humidifier-dehumidifier desalination systems [2]. Numerous studies have been made on modelling the fluid flow and heat and mass transfer in condensing two-phase flows in order to provide predictions that could improve the design of equipment in those application areas. Models of filmwise condensation in downward flow in a vertical tube have been developed for flows of pure vapour and of vapour–gas mixtures in both laminar and turbulent flow regimes.

For laminar pure vapour flows in a vertical tube, Dobran and Thorsen [3] modelled laminar flow in both the film and the vapour. Starting from the full governing equations, they obtained a set of ordinary differential equations by using an integral analysis and profile assumptions for the velocity in the liquid and in the vapour. They also used a fully-developed inlet velocity profile and studied the effect of selected dimensionless groups.

For laminar film condensation from a gas-vapour mixture in a vertical tube, Groff et al. [4], Dharma Rao et al. [5] and El-Hammami et al. [6] modelled steady axisymmetric film condensation from an air–water–vapour mixture. Groff et al. [4] solved implicitly the complete parabolic set of governing equations for laminar film and vapour flow. A marching procedure was used to obtain the solution field results along the tube. They presented results on the effects of changing the inlet Reynolds number, the inlet-to-wall temperature difference, the inlet pressure, and the inlet air mass fraction. Inlet air mass fraction values were limited to 0.80 or less.

Dharma Rao et al. [5] used an implicit finite difference method to solve the parabolic momentum, energy, and diffusion equations

* Corresponding author. Tel.: +1 204 474 8639; fax: +1 204 275 7507.
E-mail address: Scott.Ormiston@UManitoba.ca (S.J. Ormiston).

http://dx.doi.org/10.1016/j.ijheatmasstransfer.2014.06.047
0017-9310/

Nomenclature

C_p	specific heat (J kg^{-1} K^{-1})
D	diffusion coefficient (m^2 s^{-1})
f_z	grid expansion variable
g	gravitational acceleration (m s^{-2})
h_{fg}	latent heat of vapourisation (J kg^{-1})
J	mass flow at η-direction control volume faces (kg s^{-1})
J_i''	η-direction mass flux at the interface (kg m^{-2} s^{-1})
k	thermal conductivity (W m^{-1} K^{-1})
L	tube length (m)
m	mass (kg)
$\dot{m}$	total mass flow rate (kg s^{-1})
M	molar mass (kg kmol^{-1})
N	number of moles
N_L	number of grid spacings in the η-direction in the liquid
N_M	number of grid spacings in the η-direction in the mixture
Nu_z	local Nusselt number
$\mathrm{Nu}_{s,z}$	local sensible Nusselt number
$\overline{\mathrm{Nu}}_s$	average sensible Nusselt number
q''	heat flux (W m^{-2})
N_z	number of grid spacings in the z-direction
P	pressure (N m^{-2})
r	radial coordinate (m)
r_o	radius of tube (m)
$\mathcal{R}$	universal gas constant, 8314 (J^{-1} kmol^{-1} K^{-1})
$\mathrm{Re}_{\mathrm{in}}$	inlet Reynolds number ($\rho_{\mathrm{in}} u_{\mathrm{in}} 2r_o/\mu_{\mathrm{in}}$)
T	temperature (K)
$T^\star$	dimensionless temperature ($(T - T_{\mathrm{wall}})/(T_{\mathrm{in}} - T_{\mathrm{wall}})$)
u	velocity in the z direction (m s^{-1})
$u^\star$	dimensionless velocity in the z direction (u/u_{in})
v	velocity in the r direction (m s^{-1})
V	volume (m^3)
W	gas mass fraction
$W^\star$	gas mass fraction relative to the inlet (W/W_{in})
y	mole fraction
z	axial coordinate (m)
$z^\star$	dimensionless axial coordinate ($z/(2r_o)$)

Greek symbols

γ	atomic diffusion volume
δ	condensate film thickness (m)
$\delta^\star$	dimensionless condensate film thickness (δ/r_o)
η	transformed coordinate defined by Eqs. (20) and (21)
μ	dynamic viscosity (kg m^{-1} s^{-1})
ρ	density (kg m^{-3})
ϕ	relative humidity
χ	transformed coordinate (m), Eq. (19)
ω	specific humidity

Subscripts

a	referring to the air
b	bulk
g	referring to the gas
i	interface
in	at the tube inlet
lat	latent
L	referring to the liquid
M	referring to the vapour–gas mixture
sat	saturation
v	referring to the vapour
v, sat	saturation condition for the vapour
wall	at the wall
s	sensible
tot	total
z	axial

in the mixture region. They also used a marching scheme along the tube. They based the mixture solution on the vapour density variation rather than the air mass fraction, and applied an impermeability condition as an equation for the interface mass flow rate. They coupled the mixture region solution to a calculation method for the film region that is based on Nusselt's theory for film condensation from a pure quiescent vapour next to a plane wall. They also used an iterative scheme for the interface temperature based on the energy balance at the interface. Dharma Rao et al. changed the relative humidity of the inlet air–water–vapour mixture from 60% to 100%. Although they did not explicitly state the inlet vapour density or equivalent air mass fraction, this range of relative humidity corresponds to very high inlet air mass fractions (*i.e.*, well above 0.8).

El-Hammami et al. [6] modelled film condensation from a steam–air mixture with small concentrations of vapour (*i.e.*, high air mass fraction) in a vertical tube. They solved parabolic governing equations in both phases and neglected advection terms in the liquid axial momentum and energy equations. They used a finite difference discretisation approach and a marching procedure to determine the velocities, temperature, vapour mass fraction, and film thickness along the tube. In their solution scheme, the values of the pressure gradient and the film thickness are first guessed and then corrections are applied to them and the axial velocity, based on imbalances in mass conservation.

Other models have been developed for the case of turbulent flow of a gas-vapour mixture. Some of those are simplified theoretical models (*e.g.*, [7–13]) and others are detailed models based on the governing differential equations (*e.g.*, [14,15]). Among the examples listed, some considered a laminar film and a turbulent mixture flow. All of these and other models for turbulent mixture flow did not study the case of very high gas mass fraction. Therefore, to the authors' best knowledge, besides [5,6] there have been no other published models for laminar film condensation from a gas-vapour mixture with high gas mass fraction (above 0.8) for downward flow in a vertical tube.

This article presents the results from a complete two-phase model for laminar film and mixture downward flow in a vertical tube. The model is based on work of Groff et al. [4]. It solves the complete parabolic set of governing equations for conservation of mass, momentum, and energy in the film and in the mixture. Inertia terms and energy convection terms are not neglected and conservation of energy and shear at the liquid–mixture interface are accounted for fundamentally. Furthermore, the variation of thermophysical and transport properties is also included. An efficient, fully coupled numerical solution approach is employed that enables conditions with very large inlet gas mass fraction to be solved easily.

In this article, detailed comparisons with previous work are presented. Also included is a thorough presentation of many new results including axial variation of bulk air mass fraction, bulk temperature, interface air mass fraction, interface temperature, local Nusselt number, film thickness, and proportion of latent heat transfer. Radial profiles of axial velocity, temperature, and air mass fraction at various axial stations are also presented to illustrate the evolution of the two-phase flow. Many of these new, detailed results have not been presented in previous similar studies. These profiles will be useful as laminar flow tests for developers of