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Theoretical modeling of condensation of steam outside different vertical geometries (tube, flat plates) in the presence of noncondensable gases like air and helium

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ABSTRACT

Condensation of steam coming out from the coolant pipe during a loss of coolant accident (LOCA) plays a key role in removing heat from the primary containment of the advanced nuclear reactor (ANR). The presence of large mass fractions of air (W_{air} = 0.25–0.9) and a small mass fraction of helium (W_{He} = 0.017–0.083) reduces the overall heat transfer coefficient (HTC) substantially. The present work emphasizes on the issue that modeling the diffusion of water-vapor through the gas-liquid interface is the key to give good predictions in HTC. In this, condensation conductivity and effective diffusivity plays a key role. Therefore, modifications have been made in the derivation for calculation of condensation conductivity in the case of steam-air mixture and effective diffusivity (in the case of multicomponent mixture). The model validation has been done with the experimental data of Dehbi et al. [Dehbi, A.A., Golay, M.W., Kazimi, M.S., 1991. National Conference of Heat Transfer AIChE Symposium Series, pp. 19-28] and Anderson et al. [Anderson, M.H., Herranz, L.E., Corradini, M.L., 1998. Experimental analysis of heat transfer within the AP600 containment under postulated accident conditions. Nucl. Eng. Des. 185, 153-172] and other analytical models available in the literature [Herranz, L.E., Anderson, M.H., Corradini, M.H., 1998a. The effect of light gases in noncondensable mixtures on condensation heat transfer. Nucl. Eng. Des. 183, 133-150; Herranz, L.E., Anderson, M.H., Corradini, M.L., 1998b. A diffusion layer model for steam condensation within the AP600 containment. Nucl. Eng. Des. 185, 153-172; Peterson, P.F., Schrock, Y.E., Kageyama, T., 1993. Diffusion layer theory for turbulent vapor condensation with noncondensable gases. J. Heat Transf., 115; Dehbi, A.A., Golay, M.W., Kazimi, M.S., 1991. National Conference of Heat Transfer AIChE Symposium Series, pp. 19-28]. Since the validations of the results were found satisfactory, the datasets [Dehbi, A.A., Golay, M.W., Kazimi, M.S., 1991. National Conference of Heat Transfer AIChE Symposium Series, pp. 19-28; Anderson, M.H., Herranz, L.E., Corradini, M.L., 1998. Experimental analysis of heat transfer within the AP600 containment under postulated accident conditions. Nucl. Eng. Des. 185, 153-172] have been compared with a wide range of subcooling and the operating pressures. An extensive comparison has been reported and the results predicted by the present model were found to be satisfactory.

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1. Introduction

One of the most important areas in research related to the safety of nuclear reactors, is the condensation heat transfer occurring in containment atmospheres during the loss of coolant accident (LOCA). In the advanced nuclear reactors (ANR), the nuclear reactor decay heat removal is accomplished by passive features called passive containment cooling systems (PCCS). In these systems, the study of condensation of steam in the presence of noncondensables is of prime interest since during LOCA steam is released into the reactor, which contains air and other noncondensables (helium, argon, etc.).

If any of the pipes of the primary loop or steam generators break, large amount of steam is released into the containment due to the fast evaporation of water. This process increases the pressure within the containment. To prevent the containment structure from collapsing, and the efficient cooling of steam and air mixture is required. If the temperature of any of the containment's part is below the saturation temperature the steam may condense on these surfaces. Large amount of air, which in normal operation conditions is present inside the reactor containment, creates additional resistance and influence significantly the steam condensation process. In such a case diffusion model of the surface condensation is adequate to analyze such a phenomenon. Hence effort has been





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Nomenclature

i

liquid/vapor interface

Nomen	
Α	function used to calculate thermal conductivity for
	multicomponent mixture
с	concentration (kmol m^{-3})
Cn	specific heat $(I kg^{-1} K^{-1})$
d	diameter of the vertical tube on which the gas mix-
u	ture condenses
Л	diffusion coefficient $(m^2 s^{-1})$
D'	diffusion coefficient from Sutherland's kinetic-
D	theory model
σ	acceleration due to gravity $(m s^{-2})$
8 Cr	Crashof number
Gi h	best transfer coefficient $(Wm^{-2} K^{-1})$
n h	latent heat of vanorization (Ukr=1)
n _{fg}	mana transfer as officient
$n_{\rm m}$	latent hast of connection of defined in Eq. (2)
n _{fg}	latent neat of vaporization as defined in Eq. (3)
h_1^i	liquid enthalpy at T _i (J kg ⁻¹)
$h_{\rm s}^{\rm b}$	Steam enthalpy at $T_{\rm b}$ (J kg ⁻¹)
н	enthalpy (J)
k	thermal conductivity (W m ^{-2} K ^{-1})
k	Boltzmann's constant as in Eq. (20)
L	characteristic length (m)
<i>m</i> ″	mass flux (kg m ^{-2} s ^{-1})
M	steam molecular weight in kmol kg $^{-1}$
P	pressure in noncondensable equation
Pr	Prandtl number $C_{\mu}\mu k$
л л	heat flux (W/m ⁻²)
Ч R	universal gas constant
Ra	$\mathbf{R}_{\mathbf{r}} = \mathbf{R}_{\mathbf{r}} \mathbf{R}_{\mathbf{r}} \mathbf{R}_{\mathbf{r}}$
Ru Ro	film Poynolds number (Urra)
Kej Se	Schmidt number, $\mu/\rho D$
SC Sh	Schilling humber, $\mu/\rho D$
511 T	Sile wood number, $n_{\rm m} a/\rho D$
	temperature (K) (rr^3/lrr)
V	specific volume (m ² /kg)
VV	mass fraction of noncondensables
x	molar fraction
Ζ	axial co-ordinate
Crook sy	mbols
S GIEER SY	diffusion laver thickness
0	delta (sumbel used for temperature difference)
Δ	is the second of malaxylar attraction
$\mathcal{E}_{a,b}$	is the energy of molecular attraction
θ	
μ	
$\frac{\rho}{2}$	density
ν	molar average velocity (m/s)
ψ	waviness factor Re ^{0.04}
Subscrip	ts
2 Subscrip	notation for helium/air as in Eq. (20)
a	air
α 2Vσ	3V/2T302
avg h	hulk
b	potation for steam as in Eq. (20)
u	condensation
conu	convection
off	convection
ell	enecuve value
пım	nim
gas	gas
He	helium

i,j	components in a multicomponent mixture for the estimation of viscosity and thermal conductivity
1	liquid phase
m	gas mixture
п	number of components in the mixture
nc	noncondensable
t	total
v	vapor phase or vapor species
w	wall
x	denotes whether bulk or interface property, temper-
	ature or pressure

made to understand the phenomena governing the condensation of steam in the presence of noncondensables over the last three decades with different types of transport models. The next subsection gives an overview of the work that has been reported in the literature.

1.1. Previous work

Nuclear power plant containments and atmospheric conditions that can prevail in them during the course of an accident are quite unique in terms of scale, complexity, and parameter range. The studies in containment analysis have been divided into two main categories: experimental and theoretical investigations. The real challenge of containment analysis model is to be able to reproduce the conditions measured in such facilities. Theoretical models have been mainly classified into boundary layer similarity solution (Sparrow et al., 1967) and heat and mass transfer analogy (Herranz et al., 1998a,b; Peterson et al., 1993). The analogy-based models are simple and easily implementable. Hence, for the containment accident analysis the latter option is preferable. Despite extensive validation of analogy-based models (Corradini, 1984; Bunker and Carrey, 1986) in the past, the new hypothetical accident scenarios associated with advanced reactors require an upgrade in their formulation and a validation under the anticipated accidental conditions. Dehbi et al. (1991) and Uchida et al. (1965), proposed semi-empirical correlations based on the experimental data and depended solely on the noncondensable gas mass fraction in the condensing steam. The importance of variables can be classified in following three categories: (a) primary variables like noncondensable gas mass fraction, subcooling temperature difference and operating pressures, (b) secondary variables like suction effect, mist formation and film waviness, and (c) tertiary variables like effect of the type of noncondensable light gases like argon, helium and the condensing surface orientation. The modeling and experimental effort was made to understand mostly the primary variables. The analogy-based models underestimate experimental results without empirical correction factors to account for the effects of mixed convective bulk motions, mist formation and rippling on the liquid surface of the condensing film. Hence incorporating these factors theoretically was an important aspect of the research carried out in the late 1990s. Modeling condensation conductivity in analogybased models has been carried out using the Clapeyron equation (Peterson et al., 1993; Herranz et al., 1998a,b). The summary of the previous work with their limitations for each has been given in Table 1. The heat transfer coefficients (HTC) predicted by different models has been specified in Tables 2A and 2B.

For steam-air mixture the effective mass diffusion coefficient provides accurate predictions for condensation heat transfer. When noncondensable gases consist of mixtures of heavy, low mass diffusivity species and light, high mass diffusivity species, then during condensation the heavy species accumulates preferentially at the

Table 1

Summary	/ of	previous	work: ex	perimental	details.	parameters studied	. conclusions	and limitations
					,		,	

Author	Approach	Technique	Type of nuclear reactor	Parameters studied	Conclusions	Limitations
Uchida et al. (1965)	Correlating experimental data	Experimental	SBWR	Noncondensable mass fraction, pressure	Empirical correlation was developed with varying noncondensable mass fraction the data for which was used for validating codes	1,2,3,4,5,6
Sparrow et al. (1967)	Boundary layer solutions	Theoretical modeling	SBWR	Noncondensable mass fraction, subcooling		2,3,4,5
Corradini (1984)	Boundary layer solutions	Theoretical modeling	SBWR	Noncondensable mass fraction		2,3,4,5
Tagami (1965)	Heat and mass transfer analogy	Iterative Procedure	SBWR	Noncondensable mass fraction	Empirical correlation was developed with varying noncondensable mass fraction the data for which was used for validating codes	1,2,3,4,5
Dehbi et al. (1991)	Correlating experimental data and boundary layer solutions	Experimental and theoretical modeling	Westing house AP600 EBWR	Noncondensable mass fraction, subcooling, pressure	The conditions of AP600 were considered while developing the empirical correlation	4,5,6
Peterson et al. (1993)	Heat and Mass Transfer Analogy	Theoretical Modeling	Westing house AP600 EBWR	Noncondensable-mass fraction, subcooling, pressure	Model deals with large gas wall temperature differences, high mass fluxes and wavy structure of condensate	
Herranz et al. (1998a,b)	Heat and mass transfer analogy	Theoretical modeling	Westing house AP600 EBWR		Importance of variation of pressure and the presence of light gases in noncondensable gases have been considered while developing the model	6,7
Anderson et al. (1998)	Heat flux measurement and coolant energy balance	Experimental	Westing house AP600 EBWR		Valuable database to validate models. In Advanced Nuclear Reactors in which primary heat removal is from condensation of steam in the containment it is necessary to use a more complex model. When pressure changes there is a change in the concentration	5,6,7

Limitations: (1) Effect of pressure on the condensation was not considered, (2) geometry and orientation not considered according to the actual containment scenario, (3) complex methodology of estimating heat transfer coefficient, (4) type of reactor under consideration is simpler, (5) theoretical model does not account effective diffusivity properly which properly cannot predict the effect of helium, hydrogen, etc., (6) error in performing experiments, and (7) condensation conductivity was defined with less insight into the physics which affected the condensation heat transfer.

condensing surface. The neutral behavior (no significant effect on the HTC) of helium concentration in a gas mixture has been verified experimentally and theoretically (Pernsteiner et al., 1992; Anderson et al., 1998; Herranz et al., 1998a,b, respectively). The fact that the presence of helium alters substantially the effective mass diffusion, making even large mass fractions of light gases remarkably ineffective in augmenting condensation heat transfer in the presence of heavy gases (Peterson, 2000).

A brief description of the experimental apparatus of Dehbi et al. (1991) and Anderson et al. (1998) is described to compare the

Table 2A

Summary of previous work: derivables used for the steam-air mixture

Author	Heat transfer coefficient	Film heat transfer coefficient	Convective heat transfer coefficient	Condensation heat transfer coefficient
Uchida et al. (1965)	$h_{\text{Uchida}} = 380 \left(\frac{W}{1-W}\right)^{-0.7}$			
Dehbi et al. (1991)	$h_{\text{Debhi}} = \frac{L^{0.05}[(3.7+28.7P)-(2438+458.3P)\log W}{t_{\text{b}}-t_{\text{w}}}$	1		
Peterson et al. (1993)	$\frac{h_{\text{film}}(h_{\text{conv}} + h_{\text{cond}})}{h_{\text{film}} + h_{\text{conv}} + h_{\text{cond}}}$	$h_{\text{film}} = \left[\frac{\rho_{\text{l}}(\rho_{\text{l}} - \rho_{\text{gb}})gh_{\text{fg}}^{\prime}k_{1}^{3}}{\mu I(t_{1} - t_{\text{W}})}\right]^{1/4}$	$h_{\rm conv} = 0.13 \frac{k_{\rm g}}{L} Gr^{1/3} Pr^{1/3}$	$h_{\text{cond}} = \frac{Sh_0}{L} k_{\text{cond}} k_{\text{cond}} = \frac{1}{\frac{1}{\phi_p t_{\text{avg}}} \frac{P_0 M_v^2 h_{\text{fg}}^2 D_0}{R_v^2 t_0^2}}$
Herranz et al. (1998a,b	$) \frac{h_{\rm film}(h_{\rm conv} + h_{\rm cond})}{h_{\rm film} + h_{\rm conv} + h_{\rm cond}}$	$h_{\text{film}} = \left[\frac{\rho_{\text{l}}(\rho_{\text{l}} - \rho_{\text{gb}})gh'_{\text{fg}}k_{\text{l}}^{3}}{\mu L(t_{\text{i}} - t_{\text{w}})}\right]^{1/4}$	$\psi_{\rm conv} = 0.13 \frac{k_{\rm g}}{L} Gr^{1/3} Pr^{1/3}$	$h_{\rm cond} = \frac{Sh_0}{L} k_{\rm cond} k_{\rm cond} = \frac{PM_v \bar{h}_{\rm fg}^2 D}{R_v^2 t_1^2 t_{\rm avg}} \phi$
Present model	$\frac{h_{\text{film}}(h_{\text{conv}}+h_{\text{cond}})}{h_{\text{film}}+h_{\text{conv}}+h_{\text{cond}}}$	$h_{\text{film}} = \left[\frac{\rho_{\text{l}}(\rho_{\text{l}} - \rho_{\text{gb}})gh_{\text{fg}}k_{1}^{3}}{\mu I(t_{\text{i}} - t_{\text{w}})}\right]^{1/4}$	$\psi_{\text{conv}} = \left(0.825 + \frac{0.387(GrPr)^{1/6}}{(1 + (0.492/Pr^{0.5625}))^{0.2963}}\right)$	${}^{2}h_{\text{cond}} = \frac{Sh_{0}}{L}k_{\text{cond}} \frac{k_{\text{cond}}}{\frac{\rho Dh_{\text{fg}}}{(t_{\text{b}} - t_{\text{i}})}} \frac{(W_{\text{nc,i}} - W_{\text{nc,b}})}{W_{\text{nc,i}}}$

Table 2B

Summary of previous work: dea	rivables used for the	steam-air-helium mixture
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Author	Heat transfer coefficient	Diffusivity coefficient
Uchida et al. (1965)	$h_{\text{Dehbi}}^{\text{He}} = L^{0.05} \left[\frac{(3.7 + 28.7P) - (2438 + 3483P)}{(T_{\text{b}} - T_{\text{w}})^{0.25}} \right] \left[\frac{1.051 - 1.149X_{\text{He}} - 0.0553X_{\text{air}}}{+1.371X_{\text{He}}^2 + 1.371X_{\text{He}}X_{\text{air}}} \right]$	$D_{\text{eff}} = \frac{x_{\text{He}}}{x_{\text{He}} + x_{\text{air}}} D_{\text{helium-steam}} + \frac{x_{\text{air}}}{x_{\text{He}} + x_{\text{air}}} D_{\text{air-steam}}$
Dehbi et al. (1991)		
Peterson et al. (1993)		
Herranz et al. (1998a,b)		$D_{\text{eff}} = \frac{x_{\text{He}}}{x_{\text{He}} + x_{\text{air}}} D_{\text{helium-steam}} + \frac{x_{\text{air}}}{x_{\text{He}} + x_{\text{air}}} D_{\text{air-steam}}$
		$D_{\text{eff}} = \frac{x_{\text{He}}}{x_{\text{He}} + x_{\text{air}}} D_{\text{helium-steam}} + \frac{x_{\text{air}}}{x_{\text{He}} + x_{\text{air}}} D_{\text{air-steam}}$
Present model		$D_{\text{eff}} = \left(\frac{x_{\text{g,ave}}}{\sum_{j=1}^{n} (x_{j,\text{ave}}/D_{jv})}\right)$

model assumptions. The experimental apparatus of Dehbi et al. (1991) consisted of a 3.5 m long, 3.8 cm in diameter, copper cylinder located inside a 5 m long, 0.45 m in diameter stainless vessel able to withstand pressures of up to 10 atm. Steam was generated at the bottom of the vessel by a set of immersion heaters with a total capacity of 36 kW. Air was injected in the vessel from an air supply unit from a single point source. The vessel was fully insulated so that condensation takes place only on the cold copper wall. Anderson et al. (1998) on the other hand used two different test vessels namely (a) 'atmospheric test section' to study the effects of temperature difference between the bulk atmosphere and the wall side steam-air and steam-air-helium wall side pressures and with transparent walls for visualization and (b) 'the pressurized test section', to withstand more prototypical pressures and temperatures anticipated in the accident scenarios. The combined usage of both vessels along with a proper test matrix allowed a thorough understanding of the role of the major variables involved in the condensation process under realistic conditions and representative test protocols. The dimensions of the vessel were 2.8 m tall, 1.7 m wide and 0.32 m in depth. The pressurized section was designed for a design pressure of 4 bar absolute, which could be achieved in operation. The steam-air experiments involved three isobaric sets of runs, at absolute pressures of 1.5, 3.0 and 4.5 atm, respectively. The air mass fraction varied from 25% to 90%. At the start of each run, the pressure vessel was filled with room temperature air until the pressure reading indicated about 90% of the desired equilibrium pressure. The step ensured that the mixture would be in the air rich region at the start of the run.

The present work focuses on the modification of modeling the condensation conductivity (steam-air mixture) and the effective mass diffusivity (steam-air-helium mixture) needed to model the phenomena correctly. An improvement in the prediction of total HTC has been achieved and is compared with the experimental dataset (Dehbi et al., 1991; Anderson et al., 1998) and other models from the literature (Dehbi et al., 1991; Peterson et al., 1993; Herranz et al., 1998a,b). Further, effect of subcooling and pressure has also been widely studied.

2. Basis of present model

Fig. 1 shows the schematic of the theory of diffusion layer model. Initially, the vapor in the vicinity of a cooler surface condenses and forms a liquid film. Subsequently the interface is no longer the structure surface but it is the outermost edge of this liquid film. Condensation depletes the immediately adjacent vapor concentration, and as the noncondensables are not removed, they build up to higher concentrations. Thus two boundary layers develop namely the boundary layer due to condensate liquid film and gas-vapor boundary layer adjacent to the condensing surface. The liquid film boundary layer poses a minor fraction of the overall resistance to heat transfer. When noncondensable gas accumulates at the liquid-vapor interface, it reduces the interface saturation temperature T_i^s below the bulk saturation temperature T_b^s . The heat flux q'' through the condensate film, wall and external thermal resistances driven by the temperature difference $T_i - T_{\infty}$ must equal the flux of latent and sensible heat to the interface from the vapor/gas mixture. When the concentration of noncondensable gases is high the condensate film thermal resistance becomes the rate-controlling step. Similarly, for intermediate noncondensable gas concentrations the mass transfer resistance to the phenomena of phase change condensation plays a dominant role. To address condensation problems in the case of LOCA in nuclear reactors, the gas concentration of noncondensable gases often covers a wide range (mass fraction W_{air} = 0.25–0.9, Dehbi et al., 1991), requiring adequate models for the estimation of condensate film, sensible heat and mass transfer resistances.

In the present model both, the film and gas resistances to heat transfer have been accounted for by considering them to be acting in series to calculate the total heat flux from the atmosphere to the heat transfer surface. The product of the total HTC and the dif-



Fig. 1. Schematic of diffusion layer model.

ference of equivalent temperature driving force between the bulk stream and the interface give the total heat flux. The mathematical representation of the heat flux is given as follows:

$$q'' = h_{\rm t}(T_{\rm b} - T_{\rm w}) = \left(\frac{h_{\rm film}(h_{\rm conv} + h_{\rm cond})}{h_{\rm film} + h_{\rm conv} + h_{\rm cond}}\right)(T_{\rm b} - T_{\rm w}) \tag{1}$$

where h_t is the total HTC, T_b and T_w are the bulk and wall temperatures, respectively and h_{film} , h_{conv} and h_{cond} are the film HTC, convection HTC and condensation HTC, respectively.

The interface temperature is given by (Herranz et al., 1998a,b):

$$T_{\rm i} = \frac{h_{\rm film} T_{\rm w} + (h_{\rm conv} + h_{\rm cond}) T_{\rm b}}{h_{\rm film} + h_{\rm conv} + h_{\rm cond}}$$
(2)

The influence of variation of physical properties across the condensate film can be introduced by using Nusselt equation at some effective condensate film temperature T_{film} which is calculated using:

$$T_{\rm f} = T_{\rm W} + F(T_{\rm i} - T_{\rm W}) \tag{3}$$

The factor F has been considered as 0.25 (Drew, 1954).

Assumptions.

- (1) Interface temperature is equal to the surface temperature and thus the model considers only the resistance provided by the boundary layer.
- (2) Energy transfer caused by radiation and aerosols can under particular conditions, be important to containment heat transfer but these are neglected due to low temperature differences.
- (3) Flow is distributed uniformly across entire cross-section of volume.
- (4) Temperature difference ($\Delta T < 60 \circ C$) between bulk and wall is maintained in such a way that fog formation does not occur (Brouwers, 1991).

2.1. Primary variables

2.1.1. Film heat transfer coefficient

The film HTC is calculated by a modified Nusselt equation as in Collier and Thome (1994):

$$h_{\rm film} = \left[\frac{\rho_{\rm l}(\rho_{\rm l} - \rho_{\rm v})gh'_{\rm fg}k_{\rm l}^3}{\mu L(T_{\rm i} - T_{\rm w})}\right]^{1/4}\psi$$
(4)

where $\rho_{\rm l}$ and $\rho_{\rm v}$ represent the density of the liquid (water) and density of vapor, respectively. $k_{\rm l}$ and μ represent the thermal conductivity and viscosity of liquid, respectively. *L* represents the length of the tube. $h'_{\rm fg}$ accounts for the condensate subcooling and the temperature jump across the film due to phase change and difference in the specific heat values,

$$h'_{\rm fg} = h_{\rm fg} \left[\frac{1 + 0.68C_{pl}(T_{\rm i} - T_{\rm w})}{h_{\rm fg}} \right]$$
(5)

where $h_{\rm fg}$ is latent heat of vaporization in J/kg.

It is known from the literature (Kutateladze and Gongonin, 1979) that any mechanism that causes turbulence in the liquid film would enhance energy transfer rates. The rippling effect on the liquid film, which alters the boundary layer characteristics, has been considered by incorporating the correlation proposed by Kutateladze and Gongonin (1979) given by

 $\psi = Re_1^{0.04} \tag{6}$

where Re_1 is the film Reynolds number ($5 < Re_1 < 100$). $Re_1 = Lu\rho_{\text{film}}/\mu_{\text{film}}$, where *L* is the characteristic length of tube, *u* is the velocity of the liquid film and ρ_{film} and μ_{film} are density and viscosity (properties of the film).

2.1.2. Convective heat transfer coefficient

During a hypothetical accident condition, the vaporous flow pattern in the containment after the initial rapid blow down event, in which forced convection is dominant, are expected to be governed by turbulent natural convection (Green and Almenas, 1996). The correlation for convective HTC proposed by Churchill and Chu (1975) is considered here and is as follows:

$$h_{\rm conv} = \left(0.825 + \left(\frac{0.387 \times (Gr Pr)^{1/6}}{\left(1 + (0.492/\Pr^{0.5625})\right)^{0.2963}}\right)\right)^2 \tag{7}$$

Gr represents the dimentionless Grashof number $(g\beta \Delta T L^3 \rho^2 / \mu^2)$ and *Pr* is the Prandtl number $(C_p \mu / k)$.

2.1.3. Condensation heat transfer coefficient

The condensation heat transfer formulation relies on the application of the heat and mass transfer analogy (Herranz and Campo, 2002). As a result, the condensation HTC (h_{cond}) can be written as

$$h_{\rm cond} = \left(\frac{Sh_0}{L}\right) k_{\rm cond} \tag{8}$$

where $k_{\rm cond}$ is referred to as condensation conductivity (W/mK), and Sh_0 ($h_{\rm m}d/\rho D$) is the Sherwood number without the consideration of suction effect.

The mass flux m''_{cond} is found by applying Fick's law (Treybal, 1981) and multiplying it by ρ_{avg} the liquid density, *D* is the diffusivity of water-vapor in water (m²/s), W_v is the mass fraction of water-vapor and $W_{v,i}$ is the mass fraction of water-vapor at the interface. The equation at the interface is then given by

$$m_{\rm cond}^{\prime\prime} = \left[\frac{-\rho_{\rm avg} D \,\partial W_{\rm v}}{\partial y}\right]_{\rm l} + W_{\rm v,i} (m_{\rm cond}^{\prime\prime})_{\rm i} \tag{9}$$

As the condensate surface (being a solid boundary) is impermeable to the noncondensables. Eq. (12) can be simplified as

$$m_{\rm cond}'' = \left[\frac{-\rho_{\rm avg} D \partial W_{\rm v} / \partial y}{1 - W_{\rm v,i}}\right] = h_{\rm m} \frac{W_{\rm v,b} - W_{\rm v,i}}{1 - W_{\rm v,i}} \tag{10}$$

where $h_{\rm m}$ is the mass transfer coefficient.

The condensation conductivity k_{cond} has been derived from the basic Fick's law in mass transfer the resulting expression being given by

$$k_{\rm cond} = \frac{\rho_{\rm avg} D h_{\rm fg}}{(T_{\rm b} - T_{\rm i})(W_{\rm nc,i} - W_{\rm nc,b})/W_{\rm nc,i}}$$
(11)

This k_{cond} has been compared to that obtained from other methods of Herranz et al. (1998a,b), Peterson et al. (1993), and from the experimental observations (Dehbi et al., 1991; Anderson et al., 1998). The equations are listed as follows:

Condensation conductivity as proposed by Peterson et al. (1993), is given by

$$k_{\rm cond} = \frac{1/\phi P_{\rm t} M^2 D h_{\rm fg}^2}{R^2 T_{\rm avg}^3}$$
(12)

where P_t is the total pressure in atm, R is the Rydberg constant ($R = 0.08206 \text{ m}^3/\text{kmol kg}$), and T_{avg} is the average temperature in K. Condensation conductivity as proposed by Herranz et al. (1998a,b), is given by

$$k_{\rm cond} = \frac{1/\phi P_{\rm t} M^2 D h_{\rm fg}^2}{R^2 T_{\rm i} T_{\rm b}^2}$$
(13)

Table 3

Formulae for pro	operties and m	le fractions of l	binary and	multicom	ponent mixtures
------------------	----------------	-------------------	------------	----------	-----------------

	Steam-air	Steam-air-helium
Diffusion coefficient	$D = 3.4439 \times 10^{-05} \frac{T_{\rm avg}^{2.072}}{P_{\rm tot}}$ (Cenzel, 2002)	$D_{\rm eff} = \left(\frac{x_{\rm g,ave}}{\sum_{i=1}^{n}(x_{i,ave}/D_{\rm jv})}\right)$
Grashof number	$Gr = \frac{g\rho_{\rm gb}(\rho_{\rm gi} - \rho_{\rm gb})L^3}{\mu^2} $ (Herranz et al., 1998a,b)	
Viscosity	$\mu_x = W_{\rm nc} \mu_{\rm nc} + W_{\rm v} \mu_{\rm v}(T_{\rm x})$	$\mu_{\rm m} = \sum_{i=1}^{n} \frac{\mu_i(T_{\rm avg})}{\left[1 + \sum_{j=1}^{n} D'_{ij}(x_j/x_i)\right]} \text{ (Reid et al., 1987)}$
Specific heat of the mixture	$C_{P_X} = W_{\rm nc}C_{P_{\rm nc}} + W_{\rm v}C_{P_{\rm v}}(T_{\rm v})$	L j≠1
Thermal conductivity	$k_{\rm X} = W_{\rm nc} \kappa_{\rm nc} + W_{\rm V} \kappa_{\rm V} (I_{\rm V})$	$k_{\rm m} = \sum_{i=1}^{n} \frac{1}{\left[1 + \sum_{\substack{j=1\\j \neq 1}}^{n} A_{ij}(x_j/x_i)\right]} \text{ (Reid et al., 1987)}$
Mass fraction of noncondensables	wnc(x) = $\frac{(P_{\rm T} - P_{\rm v}(T_x)/P_{\rm v}(T_x))(M_{\rm nc}/M_{\rm v})}{1 + (P_{\rm T} - P_{\rm v}(T_x)/P_{\rm v}(T_x))(M_{\rm nc}/M_{\rm v})}$	
Mole fraction of noncondensables	$x_{\rm nc} = \frac{P_{\rm T} - P_{\rm Snc}}{P_{\rm T}}$	$x_{j,\text{ave}} = \left(\frac{x_{jb} - x_{ji}}{\ln x_{jb} / x_{ji}}\right) \text{(Peterson, 2000)}$

The validity of the analogy of heat mass transfer is restricted to dilute concentrations of the diffusive component. In systems where high mass fluxes are expected, the predictive capability of the analogy-based models must be improved by introducing a correction factor, θ , in the definition of the Sherwood number (Herranz and Campo, 2002):

$$Sh = Sh_0 \theta = 0.13 \, Gr^{1/3} \, Pr^{1/3} \theta \tag{14}$$

Dehbi et al. (1991) assumed that suction in gas-vapor boundary layer creates a single turbulent core and eliminates the need for the consideration of thin laminar sublayer in the development of the analogy. Thus the suction factor accounts for the effect of volume reduction as a result of condensation. This accounts for the variations in thermal and mass diffusivities. Based on the derivation of Bird et al. (1960) a simple equation for suction factor is as follows (Herranz et al., 1998a,b):

$$\theta = \frac{X_{\rm nc}^{\rm i} T_{\rm avg}}{X_{\rm nc}^{\rm avg} T} \tag{15}$$

where x_{nc}^{i} is the mass fraction of noncondensable gas (air/air+helium) at the interface and x_{nc}^{avg} is the average mass fraction of noncondensable gas and T_{avg} and T_{i} are the average and interface temperature K.

It can be noted that θ is a non-dimensional factor that measures the decrease in the steam content at the interface relative to the gaseous bulks due to the formers selective condensation.

The overall heat transfer rate can be thus, is expressed in the final form (Appendix A) as

$$q'' \propto \frac{P_{\text{tot}}^{0.66} T_{\text{avg}}^{0.39}}{T_{\text{b}}^{0.33}} \left[(M_{\text{mix}})_{\text{avg}}^{0.67} (M_{\text{mix}})_{\text{b}}^{0.33} [(M_{\text{mix}})_{i} - (M_{\text{mix}})_{\text{b}}]^{0.33} \right] \\ \times \left(\frac{1 - W_{\text{nc},\text{b}}}{W_{\text{nc},\text{i}}} \right)$$
(16)

 $M_{\rm mix}$ is the mixture molecular weight, $W_{\rm nc,b}$ and $W_{\rm nc,i}$ are the mass fraction of noncondensable gas at the bulk and interface, respectively. And the total HTC can be written as

$$h_{\text{tot}} = f(P_{\text{tot}}, T_{\text{avg}}, W_{\text{nc,b}}, (T_{\text{b}} - T_{\text{w}}))$$
(17)

2.2. Secondary variables

2.2.1. Mist formation

Condensation of vapor is expected to start before the vapor reaches the interface and presence of dust particles, present potential nucleation sites upon which such condensation droplets could form. The sensible heat carried by fog mist particles to condensing surface can therefore contribute to the overall energy removal from the atmosphere. The fog formation phenomenon is important if the large values of local subcooling are observed (>60 °C). In such situation nucleation of the small droplets and further steam condensation on droplets surfaces influences overall mass and heat transfer process described by the diffusion layer model. Since the present analysis considers the case of subcooling (<50 °C) fog formation is not expected to occur and is not taken into account. Also, in case of LOCA diffusion layer model is adequate in order to analyze such condensation phenomena (Karkoszka, 2004). Hence, the effect of fog formation has been neglected in the present analysis.

2.3. Calculation of properties of mixture

The properties for both binary (steam–air) and multicomponent (steam–air–helium) mixtures have been described in detail in Table 3.

The effective diffusion coefficient proposed by Dehbi et al. (1991) is given by

$$D_{\rm eff} = \left(\frac{x_{\rm He}}{x_{\rm He} + x_{\rm air}}\right) D_{\rm helium-steam} + \left(\frac{x_{\rm air}}{x_{\rm He} + x_{\rm air}}\right) D_{\rm air-steam}$$
(18)

Herranz et al. (1998a,b) explained the effect of helium on the HTC as

$$h_{\rm cond} = B[(\rho_{\rm mix,i} - \rho_{\rm mix,b})D_{\rm eff}^2]$$
⁽¹⁹⁾

where *B* is a variable, practically independent of noncondensable gas composition, D_{eff} is given by Eq. (20) and the binary components, i.e. $D_{\text{helium-steam}}$ and $D_{\text{air-steam}}$ are given by Wilke and Lee correlation (Wilke and Lee, 1955):

$$D_{a,b} = 10^{-4} \left[1.084 - 0.249 \sqrt{\frac{1}{M_a} + \frac{1}{M_b}} \right] \frac{T^{3/2} \sqrt{(1/M_a) + (1/M_b)}}{P(r_{a,b})^2 f(kT/\varepsilon_{a,b})}$$
(20)

where 'a' and 'b' denote helium/air and steam, respectively, *T* is the absolute temperature in K, M_a and M_b are molecular weight of 'a'



Fig. 2. Flowchart for the calculation of heat transfer coefficient.

and 'b', respectively, kg/kmol, *P* is the absolute pressure in atm, $r_{a,b}$ is the molecular separation at collision in nm, $\varepsilon_{a,b}$ is the energy of molecular attraction, *k* is the Boltzmann's constant and $f(kT/\varepsilon_{a,b})$ is the collision function.

3. Results and discussion

Fig. 2 shows the flowchart of the code developed for the calculation of HTC of condensation of steam outside a pipe in the presence of air. The iterative procedure of Peterson (2000) has been considered for finding the effective diffusion coefficient for diffusion in presence of multiple gas species.

Theoretical model for the prediction of overall HTC proposed in this work has been validated using experimental data of separate effect test as of Dehbi et al. (1991), and integral tests of Green and Almenas (1996) and along with the comparison with other model predictions (Dehbi et al., 1991; Peterson et al., 1993; Herranz et al., 1998a,b). The variation in HTC with system parameters such as noncondensable gas mass fraction, degree of subcooling and pressure has been studied. The specific observations are discussed below.

3.1. Results

3.1.1. Steam-air mixture

3.1.1.1. Variation of heat transfer coefficient with mass fraction. Fig. 3A–C depicts comparisons of the various models with experimental data of Dehbi et al. (1991), with varying mass fraction, the degree of subcooling and the operating pressure remaining constant. Model of Peterson et al. (1993) and Herranz et al. (1998a,b), both overestimate for lower mass fractions while they underestimate for higher mass fractions as compared with the model and data of Dehbi et al. (1991).

3.1.1.2. Variation in heat transfer coefficient with subcooling. The analysis has been carried out at different operating pressures each with the variation in subcooling $(10-50 \,^{\circ}\text{C})$, mass fraction being kept constant. The predicted results for variation of HTC with the degree of subcooling have been compared with the experimental results of Dehbi et al. (1991) and Anderson et al. (1998). The predictions of the present model have also been compared with various models (Dehbi et al., 1991; Peterson et al., 1993; Herranz et al., 1998a,b) as well. This is discussed in the following sections:

(A) P = 1.5 atm

Fig. 4A–C gives a clear picture of the variation in HTC predicted by various models with the variation in the degree of subcooling. The HTC predicted by the model of Dehbi et al. (1991), is in good agreement with the experimental data of Dehbi et al. (1991), for the different mass fractions and pressures considered. An exponential decrease in the HTC is predicted by the model of Dehbi et al. (1991), for all the mass fractions and pressures when the degree of subcooling is varied.

An underestimation (30-50%) in the HTC by the model proposed by Peterson et al. (1993), is mainly due to the difference in the estimation of the value of the condensation conductivity as has been identified in the present analysis.

The model of Herranz et al. (1998a,b), slightly overestimates (\sim 6%) the HTC for lower mass fractions of noncondensable gases but underestimates (25–40%) for higher gas mass fractions as compared with the experimental data base of both Dehbi et al. (1991) and Anderson et al. (1998).

Present model (Eq. (17)) shows an underestimation (\sim 3–8%) in HTC for lower mass fractions (W_{air} = 0.25), an underestimation of about (\sim 16–25%) for intermediate gas mass fractions (W_{air} = 0.5), an underestimation of about (\sim 7–35%) for higher gas mass fractions (W_{air} = 0.8) when compared with the experimental data of Dehbi et al. (1991), and that of Anderson et al. (1998), respectively.

As the mass fraction of noncondensable gas increases, the saturation temperature of steam decreases at any specific operating pressure and so does the HTC. All the models under consideration show a decrease in the HTC with the variation in subcooling at this mass fraction ($W_{air} = 0.5$). Model of Peterson et al. (1993), significantly underestimates the HTC when compared with the experimental data of both Dehbi et al. (1991), and Anderson et al. (1998). There is a steep decrease in the HTC beyond a subcooling of 20 °C, below which the degree of subcooling does not significantly affect the HTC. Model developed by Herranz et al. (1998a,b), also underestimates HTC when compared with the experimental data of both Dehbi et al. (1991) and Anderson et al. (1998). The trend in the variation in HTC is similar to that observed by Peterson et al. (1993). The present model shows a peculiar variation in HTC. The model underestimates by 16% when compared to the experimental data of Dehbi et al. (1991), while underestimation is of the order of 26% when compared with the experiments of Anderson et al. (1998). At some midway mass fractions ($W_{air} = 0.5$) and intermediate subcooling ($\Delta T = 25 - 40 \circ C$) the HTC remains nearly constant as observed from the predictions of the present model. The reason for the constant HTC predicted by the present model at the above-mentioned conditions is as follows. For at intermediate mass fraction ($W_{air} = 0.5$), the gradient ($(W_{nc,i} - W_{nc,b})/W_{nc,i}$) increases in such a way that it counterbalances the increase

in the value of $(T_b - T_i)$. Thus, the ratio between the gradient $((W_{nc,i} - W_{nc,b})/W_{nc,i})$ and subcooling $T_b - T_i$ causes no significant increase in the predicted value of the HTC. This observation is consistent with the experimental results of Dehbi et al. (1991), for which only a marginal deviation (±5%) have been reported under these operating conditions.

For a higher mass fraction ($W_{air} = 0.8$) the HTC decreases with an increase in the degree of subcooling for all the models under consideration. Large underestimation is shown by all models (Peterson et al., 1993; model by Herranz et al., 1998a,b; and the present model) when compared with the experimental results of Anderson et al. (1998).

(B) P=3 atm

Fig. 5A–C gives a clear picture of the variation in HTC predicted by various models with a variation in subcooling. The predictions of HTC by all the models at higher pressure (P=3 atm) showed fairly satisfactory. For intermediate mass fractions ($W_{air} = 0.56$) there is a significant decrease in

the HTC. The three models (Peterson et al., 1993; Herranz et al., 1998a,b; present model) underestimate the experimental results of Dehbi et al. (1991). It is evident from Fig. 5B that at low mass fractions both the models of Peterson et al. (1993), and Herranz et al. (1998a,b), show a steep decrease (high slope) in the HTC with an increase in the degree of subcooling, whereas the present model shows gradual decrease in the HTC with an increase in the degree of subcooling.

(C) P=4.5 atm

Fig. 6A–C gives a clear picture of the variation in HTC predicted by various models with a variation in subcooling. For an even higher pressure (P=4.5 atm) the decrease in HTC is similar to the model prediction of Dehbi et al. (1991) and Peterson et al. (1993). The models of Dehbi et al. (1991) and Peterson et al. (1993) are in good agreement with the experimental results of Dehbi et al. (1991). The reasons for the different trends have been discussed in Section 3.



Fig. 3. Variation of heat transfer coefficient with mass fraction of air for various authors. Subcooling = $24 \degree C$. (A) P = 1.5 atm, (B) P = 3 atm, and (C) P = 4.5 atm. (1) Dehbi et al. (1991) model, (2) Peterson et al. (1993) model, (3) Herranz et al. (1998a,b) model, and (4) present model. (\blacksquare) Experimental data of Dehbi et al. (1991).



Fig. 4. Variation of heat transfer coefficient with subcooling for various authors. P = 1.5 atm. (A) mass fraction = 0.33 $T_b = 100$ °C, (B) mass fraction = 0.56 $T_b = 90$ °C, and (C) mass fraction = 0.8 $T_b = 79$ °C. (1) Dehbi et al. (1991) model, (2) Peterson et al. (1993) model, (3) Herranz et al. (1998a,b) model, and (4) present model. (\blacksquare) Experimental data of Dehbi et al. (1991).



Fig. 5. Variation of heat transfer coefficient with subcooling for various authors. P = 3 atm. (A) mass fraction = $0.34 T_b = 125 \circ C$, (B) mass fraction = $0.56 T_b = 113 \circ C$, and (C) mass fraction = $0.85 T_b = 85 \circ C$. (1) Dehbi et al. (1991) model, (2) Peterson et al. (1993) model, (3) Herranz et al. (1998a,b) model, and (4) present model. (\blacksquare) Experimental data of Dehbi et al. (1991). (\blacklozenge) Experimental data of Anderson et al. (1998).



Fig. 6. Variation of heat transfer coefficient with subcooling for various authors. P = 4.5 atm. (A) mass fraction = 0.35 $T_b = 137 \degree$ C, (B) mass fraction = 0.58 $T_b = 127 \degree$ C, and (C) mass fraction = 0.88 $T_b = 95 \degree$ C. (1) Dehbi et al. (1991) model, (2) Peterson et al. (1993) model, (3) Herranz et al. (1998a,b) model, and (4) present model. (\blacksquare) Experimental data of Dehbi et al. (1991). (\blacklozenge) Experimental data of Anderson et al. (1998).



Fig. 7. Variation of heat transfer coefficient with pressure for various authors. Subcooling = 30 °C. (A) Mass fraction = 0.25, (B) mass fraction = 0.5, and (C) mass fraction = 0.8. (1) Dehbi et al. (1991) model, (2) Peterson et al. (1993) model, (3) Herranz et al. (1998a,b) model, and (4) present model. (■) Experimental data of Dehbi et al. (1991). (♦) Experimental data of Anderson et al. (1998).

3.1.1.3. Variation in heat transfer coefficient with system pressure. Fig. 7 shows the variation in HTC with varying system pressure. The model of Dehbi et al. (1991) correlates exactly his experimental data for all the mass fractions under consideration at the pressures used by them. Model of Peterson et al. (1993), underestimates the experimental data of both Dehbi et al. (1991) and Anderson et al. (1998), for all the noncondensable gas mass fractions considered while the model developed in the present study and that of Herranz et al. (1998a,b), show much lower deviations from all the experimental data. The model developed in the present study shows a good agreement with both the experimental data of Dehbi et al. (1991) and Anderson et al. (1991)

3.1.2. Steam-air-helium mixture

Light gases like hydrogen or helium also inhibits the process of heat transfer. When light gases like helium are present along with air (heavy species) then during condensation of steam–air–helium mixture the effective mass diffusion is affected substantially (Peterson, 2000).

Fig. 8A–C depicts comparisons of the various models with experimental data of Dehbi et al. (1991), with varying mass fraction of helium ($W_{\text{He}} = 0.017, 0.047, \text{and } 0.083, \text{respectively with}$ air mass fraction varying from $W_{\text{air}} = 0.2-0.9$) the subcooling and pressure remaining constant. The predictions given by the present model are better than the predictions of correlation proposed by



Fig. 8. Variation of heat transfer coefficient with change in mass fraction of helium and air for various authors. Subcooling = 32 °C. (A) W_{He} = 0.017, (B) W_{He} = 0.047, (C) W_{He} = 0.083. (1) Dehbi et al. (1991) model, (2) Peterson et al. (1993) model, (3) Herranz et al. (1998a,b) model, and (4) present model. (\blacksquare) Experimental data of Dehbi et al. (1991). (\blacklozenge) Experimental data of Anderson et al. (1998).



Fig. 9. Variation of heat transfer coefficient with subcooling for various authors. P = 3 atm. $W_{He} = 0.017$. (A) Mass fraction of air = 0.459 $T_b = 102.5 \degree C$ and (B) mass fraction of air = 0.56 $T_b = 90\degree C$. (1) Dehbi et al. (1991) model, (2) Herranz et al. (1998a,b) model, and (3) present model. (\blacksquare) Experimental data of Dehbi et al. (1991). (\blacklozenge) Experimental data of Anderson et al. (1998).

Dehbi et al. (1991), to explain his own experimental results. The present model shows a maximum error of the HTC only $\pm 6\%$ for ($W_{\text{He}} = 0.017, 0.047, 0.083$).

Fig. 9A and B depicts comparisons of the various models with experimental data of Dehbi et al. (1991), with varying degrees of subcooling, mass fraction of helium ($W_{\text{He}} = 0.017$) the pressure remaining constant (P=3 atm) and mass fraction of air remaining constant in each figure ($W_{air} = 0.459$ and 0.698, respectively). The present model overestimates the HTC, while the model of Herranz et al. (1998a,b), underestimates the HTC. Similarly Fig. 10A-C and Fig. 11A and B show the effect of the variation in subcooling with mass fraction of helium ($W_{\text{He}} = 0.047$ and 0.083, respectively). For a lower mass fraction of air, helium mass fraction remaining constant ($W_{\text{He}} = 0.047$) the predictions for the present model overestimates (20%) as compared with the experimental data of Dehbi et al. (1991). For a high mass fraction of air the present model underestimates the HTC by $\pm 20\%$. For higher mass fraction of helium ($W_{\text{He}} = 0.083$) the predictions of the present model are in good agreement with the experimental data of Dehbi et al. (1991) (underestimation of 10% maximum). All Figs. 9-11 show that at a lower degree of subcooling there is only a marginal decrease in HTC due to subcooling as long as subcooling was below 40 °C and a steep decrease when the subcooling increases to 40 °C and beyond. The reasons for the observed variation in the trend and the predictions are discussed in the next section.

3.2. Discussion

The present section discusses the probable reasons for the observed deviation in the magnitude of HTC predicted by the present model in the context of the available experimental data and also the other models.

3.2.1. Steam-air mixture

3.2.1.1. Mass fraction. The magnitude and trends of the present model are well agreement with the experimental results of Dehbi et al. (1991).

3.2.1.2. Subcooling. Underestimation of the experimental results by the model of Peterson et al. (1993), of the experimental data of Peterson et al. (1993), is possibly due to the following reasons: condensation conductivity predicted by the model of Peterson et al. (1993), is dependent on T_{avg} (Eq. (12)). In other words, it has been assumed that both, latent heat (h_{fg}) and specific volume (v_{fg}) could be represented by constant average values over the entire temperature range. Latent heat shows a variation of approximately 10% over the specified temperature range (25–130 °C) while the specific volume change shows a substantial variation (about a factor of four) in the same temperature range as correctly pointed out by Herranz et al. (1998a,b). Thus the condensation conductivity estimated by the model of Peterson et al. (1993), is expected to be underestimated.

On the other hand Herranz et al. (1998a,b) model depends on T_b and T_i (Eq. (13)). The dependence of noncondensable gases at the interface has been considered only as a ratio ϕ . ϕ is a function of mole fractions of air at the interface and bulk. For a given mass fraction of air, the amount of noncondensable gas (air) accumulating at the interface decreases the HTC. The constant ϕ fails to give an accurate measure of this increasing amount of noncondensable gas accumulated at the interface and hence its



Fig. 10. Variation of heat transfer coefficient with subcooling for various authors. P = 3 atm, $W_{He} = 0.047$. (A) Mass fraction of air = 0.34, $T_b = 117 \circ C$, (B) mass fraction of air = 0.565, $T_b = 117 \circ C$, (a) C, mass fraction of helium = 0.761, $T_b = 117 \circ C$. (1) Dehbi et al. (1991) model, (2) Herranz et al. (1998a,b) model, and (3) present model. (\blacksquare) Experimental data of Dehbi et al. (1991).

predictions deviate from the experimental results of Dehbi et al. (1991).

Present model on the other hand depends on T_{avg} , $T_{b} - T_{i}$ and gradient ($(W_{nc,i} - W_{nc,b})/W_{nc,i}$) (Eq. (11)). The consideration of the temperature and the concentration gradient gives a better prediction of the resistance offered by the gas film. Condensation conductivity is a function of local concentration and temperature. Hence the correct estimation depends on correct local concentration and hence considering the variation in local concentration through gradient of noncondensable gases $((W_{nc,i} - W_{nc,i})/W_{nc,i})$ is a better option. It is a weak function of $T_{\rm b}$ and $T_{\rm i}$. As more noncondensable gas accumulates near the interface, HTC decreases. When the degree of subcooling increases, T_b and T_i both increase. Since condensation conductivity is an inverse function of bulk and interface temperature $T_{\rm b}$ and $T_{\rm i}$, there is a decrease in condensation conductivity. Moreover, as the amount of noncondensable gas (air) accumulating near the interface increases, it causes a decrease in the gradient $((W_{nc,i} - W_{nc,b})/W_{nc,i})$. Both these factors cause an effective decrease in the estimation of HTC. Although all the models selected for comparison show the same overall trend of decrease in the HTC with a variation in subcooling it is interesting to check these effect (trends) at different mass fractions and pressures.

For low pressures as described in Section 3.1.1.2 there is a steep decrease in the HTC. This is due to the fact that the boundary layer resistance (concentration gradient) remains more or less constant and since HTC is inversely proportional to subcooling the HTC decreases. For $W_{air} = 0.56$ the HTC remains nearly constant for $\Delta T = 25-40$ °C and then decreases. Here the resistance due to concentration gradient term decreases. The decrease in the boundary layer resistance is counterbalanced by an increase in ΔT and hence there is no effective increase in HTC. But for $\Delta T > 40$ the boundary layer resistance remains constant and hence the HTC decreases. For a higher pressure of P=4.5 atm and $W_{air}=0.35$ a similar trend of decrease in the HTC for lower degree of subcooling $\Delta T = 10-20$ °C after which the HTC remaining constant at ΔT = 25–40 °C and then again a decrease in HTC for ΔT > 40 °C is observed. However for W_{air} = 0.56 and ΔT = 10–40 °C HTC, remains constant upto $\Delta T < 40 \,^{\circ}$ C and then decreases. This suggests that there is a critical subcooling temperature above, which the HTC starts decreasing. The trend suggests that there is transition of regime for different sets of subcooling due to which such abrupt change in the HTC are seen.

3.2.1.3. Pressure. Increase in the HTC with a pressure is mainly due to the variation in gas properties with pressure. Due to an increase in the operating pressure, the bulk and mixture average

temperatures increase. Hence, the term containing mixture molecular weights in Eq. (16) increases, as the average and interface air mass fractions increases while air mass fraction in the bulk remains the same. In other words, due to an increase in the pressure, density of the mixture increases resulting in an increase in the HTC. Higher air mass fraction at the gas-vapor boundary layer resists the hear transfer due to which the increase in pressure also does not enhance the HTC to a greater extent. This is the reason why there is a marginal increase in the HTC.

3.2.2. Steam-air-helium mixture

As per the discussion in Section 3.1.1.2 the HTC decreases due to an effective decrease in $((W_{nc,i} - W_{nc,b})/W_{nc,i})$ and $T_{avg} (T_b - T_i)$. The trends of steam–air–helium are similar to those of steam–air. The only difference being, the rate of decrease in the HTC is low due to the introduction of helium. This proves that the introduction of helium in a condensing system enhances the mass transfer by diffusion. But, as the mole fraction of noncondensable increases, the effective diffusion coefficient decreases and even at higher mole fraction of helium there is a steep decrease in the HTC. This is because as the subcooling increases the resistance of the thermal boundary layer increases along with the gas–vapor film due to which there is a steep decrease beyond a degree of subcooling of 40 °C.

Also when a multicomponent mixture of steam-air-helium is considered at lower pressures (P=1.5, 3 atm) and with varying subcooling (ΔT =10-50 °C) the trends similar to steam-air mixtures are seen but as the operating pressure increases the HTC decreases with increase in subcooling. Thus the inhibiting effect of helium can be clearly seen when the concentration gradient is kept constant.

Fig. 12 shows that the average diffusivity calculated by Dehbi et al. (1991), is nearly three times that calculated by the iterative procedure suggested by Peterson (2000). Fig. 12 has been specifically shown to emphasize the effect of higher air mole fraction on the effective diffusion coefficient. Due to the iterative procedure, the mole fractions are predicted at the interface based on the interface temperature. Effective diffusivity is a function of interface mol fractions of air and helium, which are functions of individual diffusivities of steam, air and helium. Hence an optimum value of effective diffusivity is obtained using the procedure proposed by Peterson (2000), which is followed in the present model. Thus the effect on helium on the HTC is seen at lower mass fractions of air while at higher mass fractions of air the effective diffusivity depends on the mole fraction of air and the presence of helium does not show any effect on HTC.



Fig. 11. Variation of heat transfer coefficient with subcooling for various authors. *P* = 3 atm, W_{He} = 0.083. (A) Mass fraction of air = 0.508, *T*_b = 102.5 °C and (B) mass fraction of air = 0.805 *T*_b = 90 °C. (1) Dehbi et al. (1991) model, (2) Herranz et al. (1998a,b) model, and (3) present model. (■) Experimental data of Dehbi et al. (1991). (♦) Experimental data of Anderson et al. (1998).

 $n_{\rm s}$



Fig. 12. Comparison of effective diffusivity predicted by present model with correlation of Dehbi et al. (1991) for steam–air–helium mixture. Helium mole fraction = 0.083, mole fraction of air = 0.805, and pressure = 3 atm. Subcooling = $30 \degree C.(1)$ Present model and (2) model of Dehbi et al. (1991).

4. Conclusions

A theoretical model for predicting the HTC for free convective film condensation on the vertical surface in the presence of air has been developed. The effects of influencing variables such as noncondensable gas mass fraction, operating pressure and the degree of subcooling are numerically studied. The HTC decreases due to an increase in the air mass fraction for constant wall subcooling and pressure due to the higher resistance for the diffusion of uncondensed air from the boundary layer on the condensing wall to the bulk. The predictions of present model are satisfactory when compared with the experimental results of Dehbi et al. (1991) and Anderson et al. (1998). The following conclusions can be made:

- (1) The HTC decreases with an increase in the mass fraction of air and also with an increase in mass fraction of air and both air-helium. The deviations of the present model are well found to be satisfactory.
- (2) The effect of subcooling has been extensively studied and it has been observed that at intermediate air mass fractions and high pressures (P=4.5 atm) the decrease in HTC is not exponential. At lower degree of subcooling there is a steep decrease in HTC but for temperature difference in the range 25–40 °C, the HTC remains constant and steeply decreases after 40 °C there is a steep decrease in HTC. When helium is added as a third component the HTC remains constant for subcooling less than 40 °C but steep for subcooling (>40 °C).
- (3) HTC increases with pressure for lower mass fractions of air but for higher mass fractions there is a marginal increase in HTC.

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Appendix A

A.1. Calculation of mass and mole fraction

The mass fraction and mole fraction at the interfaces are calculated as follows.

The partial pressure of steam at any given temperature is given as

$$P_{\rm nc} = P_{\rm T} - P_{\rm v}(T_{\rm X}) \tag{A1}$$

The system under condition is a constant volume and isothermal system PV = nRT for both air and steam.

$$\frac{p_{\rm nc}M_{\rm nc}}{p_{\rm v}M_{\rm v}} = \frac{n_{\rm nc}M_{\rm nc}}{n_{\rm v}M_{\rm v}} \qquad (V, T \text{ and } R \text{ are constant})$$
(A2)

$$n_{\rm nc} m_{\rm nc} = {\rm mass} \, {\rm or} \, {\rm an}$$
 (A3)

$$M_{\rm nc} = {\rm mass \ of \ steam}.$$
 (A4)

Using ideal gas laws for steam and air, the ratio of mass of air to that of steam is

$$\frac{\text{mass of steam}}{\text{mass of air}} = \left(\frac{P_{\text{nc}}}{P_{\text{v}}(T_{x})}\right) \left(\frac{M_{\text{nc}}}{M_{\text{v}}}\right) = \left(\frac{P_{\text{T}} - P_{\text{v}}(T_{x})}{P_{\text{v}}(T_{x})}\right) \left(\frac{M_{\text{nc}}}{M_{\text{v}}}\right) \quad (A5)$$

Hence mass fraction is calculated as

mass of air

wnc(x) =
$$\frac{(P_{\rm T} - P_{\rm v}(T_{\rm x})/P_{\rm v}(T_{\rm x}))(M_{\rm nc}/M_{\rm v})}{1 + (P_{\rm T} - P_{\rm v}(T_{\rm x})/P_{\rm v}(T_{\rm x}))(M_{\rm nc}/M_{\rm v})}$$
(A6)

Mole ratio can also calculated from the ideal gas law:

$$\frac{p_{\rm nc}}{p_{\rm v}} = \frac{n_{\rm nc}}{n_{\rm v}} \tag{A7}$$

The mole fraction can thus be given as

$$x_{\rm nc} = \frac{P_{\rm T} - P_{\rm S_{\rm nc}}}{P_{\rm T}} \tag{A8}$$

A.2. Physical significance

The heat flux on the condensing surface is proportional to the condensing mass flux:

$$q'' \propto m''_{\rm cond}$$
 (A9)

The proportionality of the heat flux can be written as

$$q'' \propto Sh_0(\rho_{\rm avg}D) \left(1 - \frac{W_{\rm nc,b}}{W_{\rm nc,i}}\right) \tag{A10}$$

where

$$\rho_{\rm avg} \propto \frac{P_{\rm tot}(M_{\rm mix})_{\rm avg}}{RT_{\rm avg}} \tag{A11}$$

and

$$D \propto \frac{T_{\text{avg}}^{2\text{J}}}{P_{\text{tot}}} \tag{A12}$$

where

$$M_{\rm mix} = \frac{M_{\rm nc}M_{\rm v}}{M_{\rm nc} - W_{\rm nc}M_{\rm v}} \tag{A13}$$

So.

$$\rho_{\rm avg} D \propto T_{\rm avg}^{1.072} (M_{\rm mix})_{\rm avg} \tag{A14}$$

The Sherwood number is a function of Grashof number and Schmidt number.

The Grashof number is function of mixture density $[Gr \propto \rho_{\rm gb}(\rho_{\rm gi} - \rho_{\rm gb})]$, which can be again written from Eq. (A14), as

$$Gr \propto P_{\rm T}^2 \frac{(M_{\rm mix})_{\rm b}}{T_{\rm b} T_{\rm avg}} \left[\frac{(M_{\rm mix})_{\rm i}}{T_{\rm i}} - \frac{(M_{\rm mix})_{\rm b}}{T_{\rm b}} \right]$$
(A15)

and introducing Eq. (A14) into the definition of Schmidt number it can be found that

$$Sc = \frac{\mu}{\rho_{\text{avg}}D} \propto T_{\text{avg}}^{-1.072} (M_{\text{mix}})_{\text{avg}}^{-1}$$
(A16)

(12)

So from Eqs. (A15) and (A16) Sherwood number dependency can be written as

$$Sh_0 \propto (Gr Sc)^{0.33} \\ \propto \left[\rho_{\rm gb}(\rho_{\rm gi} - \rho_{\rm gb})T_{\rm avg}^{-1.072}(M_{\rm mix})_{\rm avg}^{-1}\right]^{0.33}$$
(A17)

$$Sh_0 \propto \frac{P_{\text{tot}}^{0.66}}{T_b^{0.33} T_{\text{avg}}^{0.684}} \left[\frac{(M_{\text{mix}})_b}{(M_{\text{mix}})_{\text{avg}}} \right]^{0.33} [(M_{\text{mix}})_i - (M_{\text{mix}})_b]^{0.33}$$
(A18)

$$Sh_0 \propto \frac{P_{\rm tot}^{0.66}}{T_b^{0.33} T_{\rm avg}^{0.684}} \left[\frac{(M_{\rm mix})_b (M_{\rm mix})_i - (M_{\rm mix})_b}{(M_{\rm mix})_{\rm avg}} \right]^{0.33} \tag{A19}$$

From Eqs. (A10) and (A19) heat transfer rate can be written as

$$q'' \propto \frac{P_{\text{tot}}^{0.66} T_{\text{avg}}^{0.39}}{T_{\text{b}}^{0.33}} [(M_{\text{mix}})_{\text{avg}}^{0.67} (M_{\text{mix}})_{\text{b}}^{0.33} [(M_{\text{mix}})_{\text{i}} - (M_{\text{mix}})_{\text{b}}]^{0.33}] \times \left(1 - \frac{W_{\text{nc,b}}}{W_{\text{nc,i}}}\right)$$
(A20)

Thus the total HTC can be written as

$$h_{\text{tot}} = f(P_{\text{tot}}, T_{\text{avg}}, W_{\text{nc,b}}, (T_{\text{b}} - T_{\text{w}}))$$
(A21)

A.3. Resistances to heat transfer rate

The heat transfer rate from bulk to wall is given by

$$q_{\rm bw} = h_{\rm t}(t_{\rm b} - t_{\rm w}) = \frac{1}{(1/h_{\rm film}) + (1/(h_{\rm conv} + h_{\rm cond}))}(t_{\rm b} - t_{\rm w}) \quad (A22)$$

The resistance offered by the condensate film $= \frac{1}{h_{\text{film}}}$ (A23)

The resistance offered by the gas/vapor boundary layer

$$=\frac{1}{h_{\rm conv}+h_{\rm cond}}\tag{A24}$$

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