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Laminar film condensation of saturated vapor on an isothermal vertical cylinder

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ABSTRACT

An analytical model for laminar film condensation of pure vapor on the outer surface of an isothermal vertical tube is presented. Based on the assumptions employed in the Nusselt's analysis on a flat plate, the explicit expressions for the condensate film thickness as well as local and average Nusselt numbers are derived. The heat transfer from the liquid–vapor interface is dealt with for two contrary situations: with and without the condensate subcooling. The deviation of the Nusselt number of a cylinder from that of a flat plate is presented in terms of the ratio of the liquid film thickness to the radius of a tube and the Jakob number. The variations of the Nusselt number with the tube radii are calculated from the derived models to investigate the curvature effects of a condensing surface. It is demonstrated that the effect of the curvature on the condensation heat transfer becomes important when the radius of a tube is of the similar order of magnitude as the liquid film thickness.

1. Introduction

Condensation is a heat transfer process of great importance in a variety of industrial applications, including refrigeration facilities, thermal power plants, heat exchangers, and distillation systems and so on. In particular, in the aftermath of Fukushima accident in 2011, this heat removal by phase changes has drawn extensive attention in nuclear industries since advanced nuclear power plants (NPPs) planned to introduce the Passive Containment Cooling System (PCCS) which protects the integrity of the containment building by condensing the vapor [1]. S. Korea, in which all the NPPs employ the concrete containment, may adopt a PCCS using internal condensers and its configurations are shown in Fig. 1. When a nuclear reactor encounters an accident, the vapor released from nuclear systems is condensed on the outer surface of vertical condenser tubes where cooling water flows inside.

The theory of laminar film condensation on isothermal vertical flat surface was first established by pioneering analysis by Nusselt [2]. Afterwards a number of attempts have been made to improve the simplifying assumptions embodied in the Nusselt's analysis. For instance, the effects of the subcooling of the condensate film or the shear stress at the film–vapor interface were taken into account in deriving the rate of vapor condensing and heat transfer [3]. Some investigations accounted for the effect of ripples and waves at the interface as the Reynolds number (Re) of the condensate film increases [4,5]. For more elaborated analysis of the vapor condensation on an isothermal plate, a couple of analytical studies were conducted on the basis of the boundary layer approximation to present the numerical results [6–9].

It should be noted that most of previous analytical and experimental studies are concentrated on the film condensation on a flat plate or a horizontal tube, whereas many industrial condensing systems employ vertical tube condensers recently. Revankar and Pollock [10] and Kim et al. [11] presented the analytical models for laminar film condensation in a cylindrical coordinate, but they dealt with the vapor condensation inside the vertical tube. To the best knowledge of the authors, the analytical model of laminar film condensation on the outer surface of isothermal vertical cylinder is not found. Especially, the curvature effect of the tubes on heat transfer is a crucial factor in design optimization of a condenser system, and thus it is essentially required to investigate the variation of the Nusselt number (Nu) with the tube radius.

In this paper, the analytical model of laminar film condensation of saturated vapor on an isothermal vertical cylinder is presented on the basis of fundamental assumptions used in the Nusselt's analysis. From the momentum and energy conservation equations, the expressions for the thickness of the condensate film as well as local and average Nusselt numbers are derived. This is performed for two cases in the absence and presence of the condensate

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Nomenclature

c_p	specific heat at constant pressure (J kg^{-1} K^{-1})
g	gravitational acceleration (m s^{-2})
h	heat transfer coefficient (W m^{-2} K^{-1})
$\bar{h}$	average heat transfer coefficient over length (W m^{-2} K^{-1})
h_{fg}	latent heat (J kg^{-1})
Ja	Jakob number
k	thermal conductivity (W m^{-1} K^{-1})
$\dot{m}$	mass flow rate (kg s^{-1})
Nu	Nusselt number
$\overline{Nu}$	average Nusselt number over length
q''	heat flux (W m^{-2})
r	radial coordinate (m)
R	radius of cylinder (m)
T	temperature (K)
u	axial velocity (m s^{-1})
x	axial coordinate (m)

Greek letters

δ	liquid film thickness (m)
Θ	sensible heat transfer rate (W)
μ	viscosity (kg m^{-1} s^{-1})
ρ	density (kg m^{-3})

Subscripts

L	entire length
P	flat plate
sat	saturation state
V	vapor
W	wall

Superscripts

sub	in the presence of the condensate subcooling

subcooling. The deviation of the Nusselt number of a cylinder from that of a flat plate is expressed in terms of the thickness of the condensate film, the radius of a vertical tube and the Jakob number (in the presence of the condensate subcooling). The variation of the heat transfer coefficient on a vertical tube according to the curvature of the tube is studied with the derived theoretical model.

2. Analytical model

A physical model of the laminar film condensation on the outer surface of a vertical cylinder is described in Fig. 2. All relevant variables are defined in cylindrical coordinates. A saturated vapor is condensed on the outer wall whose temperature is lower than the vapor saturation temperature and constant along the axial direction. The annular film first formed at the top of the cylinder grows thicker as it flows downwards along the wall due to continuous condensation at the liquid–vapor interface. Then the velocity and thermal gradients develop across the condensate film.

In the proposed analytical model, most assumptions embodied in the Nusselt's analysis are also accepted for the derivation procedure herein [2]. For a pure, saturated vapor condensing on an isothermal vertical cylinder,

(1) Steady-state laminar flow and constant properties are assumed for the condensate film.
(2) The vapor is stationary and has no temperature gradient in the radial direction.
(3) The shear stress at the liquid–vapor interface is negligible.
(4) Heat transfer across the film occurs only by conduction.

On the basis of the above assumptions, the governing equations for the momentum and the energy of the liquid film are expressed as:

$$\mu\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial u}{\partial r}\right)+(\rho-\rho_v)g=0, \quad (1)$$

$$\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial T}{\partial r}\right)=0. \quad (2)$$

Note that the free stream pressure gradient in the quiescent region is replaced by the gravitational body force of the vapor. For the velocity and the temperature of the condensate film, the following boundary conditions are implemented:

$$u|_{r=R}=0, \quad (3)$$

$$\mu\left.\frac{\partial u}{\partial r}\right|_{r=R+\delta}=0, \quad (4)$$

$$T|_{r=R}=T_w, \quad (5)$$

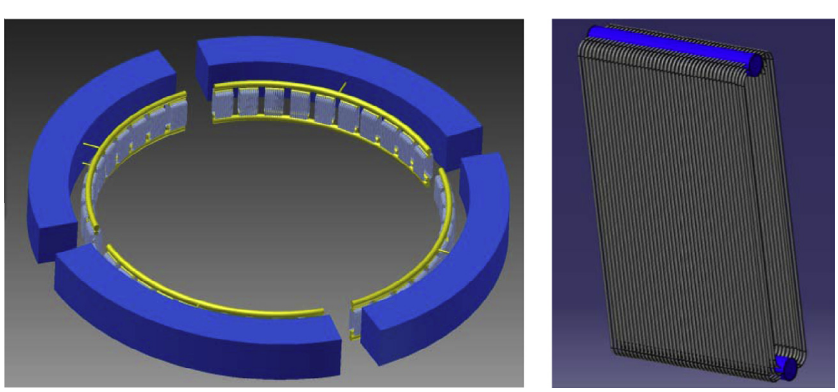

Fig. 1. Passive Containment Cooling System with vertical internal condensers.