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# Mini-channel flow condensation enhancement through hydrophobicity in the presence of noncondensable gas

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## ABSTRACT

Steam condensation is important for a broad range of industrial applications, including power generation and nuclear containment systems. The presence of noncondensable gases in these systems significantly reduces heat transfer, prompting the need for condensation heat transfer enhancement. Steam was condensed in the presence of nitrogen in hydrophilic and hydrophobic 1.82-mm rectangular mini-channels for a range of experimental conditions: steam mass flux (i.e., 35–75 $\mathrm{kg/m^2\,s}$), steam quality (i.e., $0.3 < x < 0.9$), and nitrogen mass fraction (i.e., 0–30%). In the hydrophilic channel, nitrogen mass fractions of 10–30% reduced condensation heat transfer coefficients by 24–55%. Experimental results were well predicted by the Caruso et al. (2013) correlation. Dropwise condensation was observed in the hydrophobic channel, although the addition of nitrogen suppressed nucleation. In the hydrophobic channel, heat transfer was enhanced by 34–205% over the hydrophilic channel in presence of 10–30% nitrogen, particularly at low vapor mass fractions. Heat transfer coefficients in the hydrophobic channel with 30% nitrogen were identical or higher than those of pure steam in the hydrophilic channel at the same mass flux and quality. Heat transfer coefficients strongly depended on vapor mass fraction, defined as the vapor mass flow rate divided by the three-phase (vapor, liquid and nitrogen) mass flow rate.

## 1. Introduction

Condensers play an important role in many industrial applications, yet condensation heat transfer performance can be greatly degraded by the presence of noncondensable gases (NCG). Noncondensables, such as air, gather at the liquid-vapor interface and impede condensation heat transfer; in many cases, the thermal resistance across the air becomes the dominant thermal resistance [1–4]. Condensation in the presence of noncondensables is found in many industrial systems, such as thermal desalination units [5], thermosiphons [6], containment systems in nuclear power plants [7], and air-cooled condensers [8]. A further understanding and mitigation of the effects of noncondensable gases on condensation is required.

Based on Nusselt's falling film theory [9], several analytical approaches (e.g. boundary layer method and diffusion layer method) have been employed to understand the impacts of noncondensable gases on condensation heat transfer. In the boundary layer method, convection in the liquid condensate film and vapor-NCG mixture are solved separately by setting temperature boundary conditions at the liquid-gas interface [1,10–15]. The diffusion layer method solves heat transfer through three-layer (i.e. vapor, gas, and liquid) mixtures [3,4,16–24] in which heat transfer from the vapor to the liquid through NCG layer is driven by both diffusion and sensible heat transfer caused by vapor partial pressure difference between vapor-NCG and NCG-liquid interfaces. Both methods demonstrate the reduction of condensation heat transfer coefficient with noncondensables.

Minkowycz and Sparrow [1] modeled steam-air mixture condensation on an isothermal vertical plate with laminar free film convection using the boundary layer method. A 0.5% bulk mass fraction of air reduced condensation heat transfer coefficients by up to 50%. Sparrow et al. [10] analytically studied horizontal and vertical flows; at low NCG mass fractions (i.e. 0.5–1%), the effects of NCG on heat flux in horizontal forced flow were negligible while in gravity-driven laminar vertical flows, heat fluxes were reduced by 80–90%. To investigate the effects of noncondensable gases for a wider range of flow conditions and predict heat transfer of practical systems, experiments [25–31] were conducted and empirical correlations were developed using the degradation factor method [16,25–27] and heat and mass transfer analogy [3,20,28–34]. Vierow [25] developed the degradation method, defined as the ratio of experimentally measured condensation heat transfer coefficent with NCG to that of pure steam, typically based on the

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## Nomenclature

| | |
|---|---|
| A | area ($m^2$) |
| C | flow-based constant |
| Cp | specific heat (kJ/kg K) |
| D | diameter (m) |
| F | heat transfer coefficient degradation factor |
| h | heat transfer coefficient ($W/m^2$ K) |
| i | evaporative enthalpy (kJ/kg) |
| k | thermal conductivity (W/m K) |
| $\dot{m}$ | mass flow rate (kg/s) |
| $Nu$ | Nusselt number |
| P | pressure (kPa) |
| $P_{red}$ | reduced pressure |
| $\dot{Q}$ | heat transfer rate (W) |
| $q''$ | heat flux ($W/m^2$) |
| $Re_l$ | liquid Reynolds number $Re_l = \frac{G(1-x)D}{\mu_l}$ |
| $Re_v$ | vapor Reynolds number $Re_v = \frac{GxD}{\mu_v}$ |
| $Re_g$ | gas Reynolds number $Re_g = \frac{(\dot{m}_v+\dot{m}_N)D}{\mu_g A}$ |
| $Sc$ | Schmidt number |
| T | temperature (K) |
| TCP | thermocouple |
| $\dot{V}$ | volumetric flow rate ($m^3/s$) |
| w | uncertainty |
| x | steam quality |
| y | vertical distance from the condensation surface (m) |

*Greek letters*

| | |
|---|---|
| $\rho$ | density ($kg/m^3$) |
| $\lambda$ | relative mass fraction |
| $\omega$ | mass fraction in vapor-nitrogen-liquid mixture |

*Subscript*

| | |
|---|---|
| cond | condensation |
| exp | experiment |
| f | flow |
| fg | liquid-vapor phase change |
| g | gas (including vapor and nitrogen) |
| l | liquid |
| N | nitrogen |
| pre | prediction |
| st | steam |
| ts | test section |
| v | vapor |
| w | wall |

Nusselt analysis for laminar filmwise condensation heat transfer [35–38]. Lee and Kim [27] developed a degradation factor correlation based on dimensionless shear and NCG mass fraction. Results showed that the effects of small amounts of noncondensables on heat transfer coefficients could be eliminated by reducing channel size (e.g. 3% nitrogen by mass did not significantly affect heat transfer coefficients in a 13-mm inner diameter tube).

Through analysis and experiments, it has been shown that noncondensables reduced steam condensation heat transfer. Therefore, it is important to consider approaches which could mitigate the effects of noncondensables in condensing systems. Reduced channel size (i.e., mini- and microchannels [39–43]) and hydrophobicity [44–48] are two heat transfer enhancement approaches studied in this paper. In mini- and micro-channels, surface tension becomes significant and changes the flow regime to improve heat transfer, and increased surface hydrophobicity promotes dropwise condensation and facilitates liquid removal [48–50]. To understand the influence of steam flow conditions, noncondensable gas fractions, and wettability on flow condensation heat transfer, this work studies steam condensation heat transfer in 1.82-mm, hydrophilic and hydrophobic rectangular mini-channels at steam mass fluxes of 35–75 $kg/m^2$ s, steam qualities of 0.3–0.9, and NCG mass fractions from 0 to 30%.

## 2. Experimental method and apparatus

### 2.1. Experimental apparatus

In order to study the impacts of noncondensables on steam flow condensation, nitrogen was introduced to condensing steam in an open-loop experimental apparatus [48], shown in Fig. 1. Steam was sourced from the campus system at 550 kPa, regulated to 250 kPa, and filtered to remove debris. Liquid was removed in the separation tank and the steam was superheated 20–30 °C in order to determine its enthalpy. Steam was partially condensed in a pre-condenser; in order to conduct an energy balance on the coolant, cooling water mass flow rates were rate measured by a Coriolis flow meter (Micro Motion$^{TM}$ F-series sensor and 2700 transmitter) and inlet and exit temperatures were measured with T-type thermocouples. Ultra-pure nitrogen (mass purity >99.9%, Matheson) was introduced into the system at a pressure 103 kPa higher than the steam pressure to prevent backflow. Nitrogen volumetric flow rates (Omega$^{TM}$ 7211 rotameter), temperature, and pressure were measured prior to introducing nitrogen between the pre-condenser and test section. Flow stability was monitor visually through the acrylic rotameter, which has a manufacturer-reported full scale uncertainty of 2%. Room temperature nitrogen was injected into the steam flow; the steam quality change from heating the nitrogen was less than 0.02. Test section fluid inlet and exit temperatures and pressures were measured; heat transfer measurements and flow visualization could be conducted simultaneously. Following the test section, the steam was fully condensed and its flow rate was determined.

### 2.2. Test section

The test section simultaneously measured condensation heat transfer coefficients of the steam/liquid/nitrogen mixture and visualized the flow (Fig. 2). The mixture entered through an inlet in the PEEK block, entered the oxygen-free copper coupon, and exited through the PEEK block. The 40-mm long, 10-mm wide, and 1-mm deep mini-gap was milled into an oxygen-free copper coupon, resulting in a hydraulic diameter of 1.82 mm. The mini-gap incorporated three-side cooling (i.e. bottom and two side walls) and an adiabatic top glass window for visualization. Heat transfer coefficients were determined using the cooled areas. Surface temperature was determined from a near-wall thermocouple 0.5 mm below the channel and heat flux was determined through through Fourier's law. Five type T thermocouples were installed in 3.5-mm thermocouple holes spaced 8 mm apart in the oxygen-free copper block. The heat transfer rate measurements were previously validated using single-phase cooling [48]. An indium thermal interface material provided good thermal contact between the coupon and copper block. Due to the three-phase flow (i.e., liquid, vapor, and $N_2$), fluid temperature was directly measured with a 0.5 mm-diameter thermocouple (TC Direct$^{TM}$ 206-494) inserted