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# Rate correlation for condensation of pure vapor on turbulent, subcooled liquid

J. STEVEN BROWN, BOO CHEONG KHOO† and AIN A. SONIN‡
Department of Mechanical Engineering, Massachusetts Institute of Technology, Cambridge, MA 02139, U.S.A.

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**Abstract**—An empirical correlation is presented for the condensation of pure vapor on a subcooled, turbulent liquid with a shear-free interface. The correlation expresses the dependence of the condensation rate on fluid properties, on the liquid-side turbulence (which is imposed from below), and on the effects of buoyancy in the interfacial thermal layer. The correlation is derived from experiments with steam and water, but under conditions which simulate typical cryogenic fluids.

## 1. INTRODUCTION

CONDENSATION of pure vapor at a turbulent liquid interface is a liquid-side heat transfer process, the rate being limited by the turbulent transport of the latent heat from the interface to the bulk of the liquid. Theoretically, this is still an unsolved problem, largely because the structure of the turbulence very near the free surface is still open to speculation. At lower turbulence intensities the condensation problem is further complicated by stable thermal stratification at the interface, with attendant turbulence damping. Simplistic models have been proposed for the analogous gas absorption problem, where thermal stratification is absent [1–8]. However, each of these models is tailored largely to specific experimental conditions. The models disagree with each other, and there is no consensus on a unified model which expresses the condensation rate in terms of the local turbulence parameters and fluid properties (e.g. see ref. [9]). Progress toward such a model has been hindered not only by the lack of understanding of the interfacial turbulence structure, but also by the fact that accurate comparison with experiment has been difficult: the turbulence parameters which appear in a general model (e.g. turbulence intensity and turbulence macroscale) have not been directly measured in most investigations of condensation.

Simultaneous data on vapor condensation rate and liquid-side turbulence are relatively scarce. Thomas [10] made measurements with steam and water in several different systems in which turbulence was imposed on the liquid from below, without shear on the interface. Jensen and Yuen [11] report measurements in a channel flow in which the liquid-side turbulence was induced largely by interfacial shear from the steam side. Ueda *et al.* [12], Mizushina *et al.* [13], Komori *et al.* [14, 15] and Ogino [16] have published significant basic data on the turbulence structure in a channel flow with interfacial heat transfer. They did not, however, report simultaneous measurements of the heat transfer rate at the interface, and their measurements of turbulent diffusivity do not cover the very thin region near the free surface where most of the temperature drop occurs when buoyancy effects are not dominant.

More recently, Sonin *et al.* [9] investigated the condensation of pure steam on a shear-free water interface, on which a calibrated turbulence was imposed from below. Using relatively high turbulence intensities where buoyancy effects were negligible, they concluded that the condensation rate could be correlated in terms of a constant Stanton number based on the liquid-side turbulence intensity.

In this paper we present a more general empirical correlation for the rate of pure vapor condensation on a turbulent subcooled liquid. The correlation accounts not only for the dependence on the interfacial turbulence conditions, but also establishes the dependence on liquid-side Prandtl number and buoyancy. One of the major objectives of this work has been to obtain a rate correlation that can be applied to predict the condensation rate of cryogenic fluids under a broad range of turbulence conditions.

The present work is based on experiments with steam and water, and generalized to other fluids by means of scaling laws (Section 5). Our apparatus is similar to the one used in ref. [9], but experimental accuracy has been improved, the system has been modified to operate over a range of saturation conditions, and our data correlation is based on more precise information on the turbulence structure in the system (Section 3). Our correlation covers the scaling

† Presently at the Department of Mechanical and Production Engineering, National University of Singapore.
‡ Author to whom correspondence should be addressed.

## NOMENCLATURE

| | |
|---|---|
| $c_p$ | specific heat at constant pressure [J kg$^{-1}$ K$^{-1}$] |
| $d$ | nozzle exit diameter, Fig. 1 [m] |
| $D$ | test cell inside diameter, Fig. 1 [m] |
| $f$ | frequency [Hz] |
| $g$ | gravitational (or reference frame) acceleration [m s$^{-2}$] |
| $Gr$ | Grashof number, equation (39) |
| $h_{fg}$ | latent heat of condensation [J kg$^{-1}$] |
| $Ja$ | Jakob number, equation (17) |
| $k$ | turbulence intensity in $k$–$\varepsilon$ model [m$^2$ s$^{-2}$] |
| $l$ | turbulence macroscale [m] |
| $L$ | length scale in $k$–$\varepsilon$ turbulence model, $k^{3/2}/\varepsilon$ [m] |
| $\dot{m}$ | condensation mass flux across interface [kg s$^{-1}$ m$^{-2}$] |
| $Q$ | volume flow rate circulating through system, Fig. 1 [m$^3$ s$^{-1}$] |
| $r$ | radial coordinate [m] |
| $R(t)$ | Eulerian autocorrelation function measured at a fixed point, equation (8) |
| $Re$ | Reynolds number, equation (17) |
| $Re_*$ | system Reynolds number, $Q$ $dv$ |
| $Ri$ | Richardson number, equation (17) |
| $St$ | condensation Stanton number, equation (17) |
| $St_0$ | reduced Stanton number, equation (37) |
| $t$ | time [s] |
| $\bar{t}$ | characteristic time, equations (7) and (9) [s] |
| $T$ | absolute temperature [K] |
| $T_b$ | value of $T_B$ extrapolated to the interface, Fig. 7(b) [K] |
| $T_B$ | temperature in bulk of liquid, outside interfacial layer, Fig. 7(b) [K] |
| $T_s$ | liquid saturation temperature [K] |
| $\Delta T$ | liquid subcooling, $T_s - T_b$ [K] |
| $u_c$ | condensation induced bulk flow, $\dot{m}/\rho_b$, equation (20) [m s$^{-1}$] |
| $v$ | r.m.s. value of either the horizontal or vertical component of turbulent velocity [m s$^{-1}$] |
| $v_b$ | value of $v$ extrapolated from the bulk liquid to the surface disregarding the interfacial layer, Fig. 7(a) [m s$^{-1}$] |
| $y$ | coordinate measured vertically downward from the surface into the bulk liquid, Fig. 1 [m] |
| $z$ | coordinate measured vertically upward from the nozzle exit into the bulk liquid, Fig. 1 [m] |
| $z_s$ | elevation of surface from nozzle exit, Fig. 1 [m]. |

Greek symbols

| | |
|---|---|
| $\alpha$ | thermal diffusivity [m$^2$ s$^{-1}$] |
| $\alpha_T$ | turbulent thermal diffusivity, equation (22) [m$^2$ s$^{-1}$] |
| $\alpha_{TB}$ | turbulent thermal diffusivity outside the interfacial layer, Fig. 7(c) [m$^2$ s$^{-1}$] |
| $\beta$ | coefficient of thermal expansion [K$^{-1}$] |
| $\delta$ | thermal layer thickness, equation (18) [m] |
| $\delta_v$ | viscous layer thickness [m] |
| $\varepsilon$ | viscous dissipation rate in $k$–$\varepsilon$ model [m$^2$ s$^{-3}$] |
| $\lambda$ | thermal conductivity [kg m s$^{-3}$ K$^{-1}$] |
| $\Lambda$ | integral turbulence length scale, equation (9) [m] |
| $\mu$ | viscosity [kg m$^{-1}$ s$^{-1}$] |
| $\nu$ | kinematic viscosity [m$^2$ s$^{-1}$] |
| $\rho$ | density [kg m$^{-3}$] |
| $\tau$ | statistical property, equations (31) and (32) [s] |
| $\phi(Re_*)$ | function defined in equation (3) |
| $\Phi(f)$ | Eulerian time spectrum, equation (11) [m$^2$ s$^{-1}$]. |

Subscripts

| | |
|---|---|
| b | liquid at bulk temperature (extrapolated to interface) |
| g | vapor |
| s | liquid at saturation temperature. |

parameters characteristic of most cryogenic fluids, and establishes the dependence of the condensation rate on the liquid and vapor properties, the liquid-side turbulence intensity and turbulence macroscale, and the effects of buoyancy.

## 2. TEST CELL

Experiments were performed with steam and water in a test cell (Fig. 1) which is geometrically similar to those used in ref. [9]. The cell consists of a pyrex tube of inside diameter $D$, partially filled with water. A statistically steady turbulence is generated in the water by a single submerged nozzle located far below the interface ($z_s \gg d$, where $z_s$ is the interface elevation relative to the nozzle exit and $d$ the nozzle diameter), so that the region near the interface is in the far field of the jet. The water is circulated in a closed loop, with the temperature controlled by a heat exchanger. Sufficiently far from the nozzle (about $z > 3D$), this system produces an essentially bulk-flow-free turbulence which is approximately isotropic in the hori-