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Technical Note

Condensation of FC-72

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ABSTRACT

Heat transfer coefficients for condensing FC-72® vapor on vertical copper and Teflon plates are reported as a function of sub-cooling at one atmosphere. Results include the evolution of the average heat transfer coefficient with time and a visual record of droplet formation and coalescence owing to non-condensable gas. Experiments are run for a Reynolds number of 15.1, wall heat flux of 0.92–2.88 W/cm^2, and 6–44 K sub-cooling. Film condensation heat transfer coefficients compare reasonably well with those of prior studies run under different convective conditions. High resolution video captures evolution of droplet size (average diameter) and number density. A correlation is shown to exist between overall heat transfer coefficients and droplet size and number density. When droplet number density exceeds 10 cm^{-2} and droplet area exceeds ~1.5 mm^2, average heat transfer coefficients approach a limiting value.

1. Introduction

Phase change in a fluorocarbon, e.g., Fluorinert FC-72®, has become a viable option for cooling of electronics because of the high heat transfer rates that can be achieved and the inert nature of fluorocarbons. The overall thermal management strategy with immersion cooling involves boiling and condensation, but for fluorocarbons relatively little research on condensation heat transfer has appeared. We report measurements that characterize condensation of this class of heat transfer fluids, and we focus on condensation of FC-72 at one atmosphere for which few experimental studies have appeared in the literature.

A wide range of experimental studies of film and drop wise condensation heat transfer have appeared over the past 90 years, and excellent summaries are available [1–4]. Most of the early work has focused on the condensation of water vapor (steam) owing to its central role in power generation and chemical engineering processes. Several condensation models have been developed and generally corroborated experimentally.

Drop wise condensation is characterized by overall heat transfer coefficients that are an order of magnitude larger than those of film condensation, and two models of droplet formation have been proposed. One approach postulates that condensation occurs on a thin unstable film covering either all or part of the surface [6]. At a critical thickness the film ruptures, and the liquid coalesces into droplets under surface tension forces. Several experiments [7–9] have supported this proposed mechanism. The classical model due to Eucken [10] assumes that droplet formation is a heterogeneous nucleation process, and subsequent studies support this mechanism [11,12]. McCormick and Baer [13] suggest that innumerable sub-microscopic droplets are randomly nucleated at active sites, such as, wetted pits and grooves and grow by direct condensation and coalescence. The rate of direct condensation on the larger droplets is less than for the smaller droplets because of their resistance to heat conduction. Larger drops grow mainly by coalescence, whereas smaller drops grow mainly by condensation and are responsible for a major fraction of heat flux. Gose et al. [14] propose a model for drop wise condensation which accounts for drop nucleation and growth, removal and re-nucleation on sites exposed by removal, and coalescence of drops. Collier and Thome [1] describe heterogeneous droplet formation, nucleation and growth based on mechanical equilibrium of a spherical drop. The equilibrium temperature of the liquid is determined by the Clausius–Clapeyron equation. Once nucleation has occurred, droplet growth requires that the latent heat of condensation be removed by conduction and convection to the surrounding fluid. In the case of conduction only, a model proposed by Mason [5] predicts growth proportional to the excess vapor pressure at the liquid–vapor interface. Ohtani et al. [15] have studied fluctuations of surface temperature that occur as a result of unsteady nature of nucleation, growth and coalescence process because of the large heat flows through much reduced surface area occupied by smaller droplets. The thermal diffusivity of the condensing surface thus has an important influence via transient heat conduction near the surface [16].

Mahajan et al. [17] studied laminar film condensation of a fluorocarbon vapor via temperature measurement of suddenly immersed cold copper spheres in a saturated fluorocarbon vapor. Their heat transfer rates agree well with the Nusselt–Rohsenow

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Nomenclature

A_p	area of condensing surface [m^2]
c_p	specific heat of water [J/kgK]
G	mass flux [kg/m^2s]
h	heat transfer coefficient, $q/(T_{sat}-T_w)$ [W/m^2K]
h_{fg}	latent Heat of vaporization, [J/kg]
Ja	Jakob number, $c_p(T_{sat}-T_w)/h_{fg}$
k	thermal conductivity [W/mK]
q	heat flux [W/m^2]‘
Re	Reynolds number, $\rho VH/\mu$
H	channel height above condensing surface [m]
N	Droplet number
Nu	Nusselt number, hL/k
T	temperature [K]
ΔT	subcooling, T_w-T_{sat} [K]
t	time [s]
V	mean velocity of flow [m/s]
Greek Symbols	
μ	dynamic viscosity [Pa-s]
ρ	density, kg/m^3
Subscripts	
w	wall or surface
sat	saturation

[3] model, but experimental heat transfer coefficients (corrected for viscosity variations) are ~10% lower than theoretical values for $Ja \approx 3.5$. The discrepancy between experiment and theory increases with decreasing Jakob number owing to non-condensable gases in the vapor. Condensation of FC-72 in a horizontal rectangular duct in forced convection is reported by Lu and Suryanarayana [18] for inlet vapor Reynolds numbers $\sim 10^6$, and heat transfer rates increase significantly after the appearance of interfacial waves. Boyack [19] reports data on wavy film condensation of FC-72 vapor with film Reynolds numbers ranging from 17 to 238 in a stationary vapor, and heat transfer rates are found to be in a good agreement with condensation data for other fluids.

2. Apparatus and procedure

The condensation plate and flow geometry are shown in Fig. 1. Liquid FC-72 is brought to a boil in an external evaporator, and its vapor flows across the condensing surface. Temperatures of the cooled surface and its cooling loop, as well as the condensate and cooling loop flow rates, are used with an energy balance to determine heat transfer coefficients. The uncondensed vapor flows to an auxiliary condenser from which it returns to the primary FC-72 reservoir. Full details of the design are given elsewhere [20].

The condensation chamber is constructed of polycarbonate, and the condensation plate and heat exchanger are copper. An observation window mounted over the chamber opening provides a view of the condensate as it forms. Condensate is collected in the trough at the bottom of the condensation plate and directed into a glass gauge for volumetric flow rate measurement. The condensing surface is finished with 400 grit sand paper (rms roughness = 23 μm). For experiments with a non-wetting surface, a 3.2 mm thick Teflon® film is bonded to the copper surface.

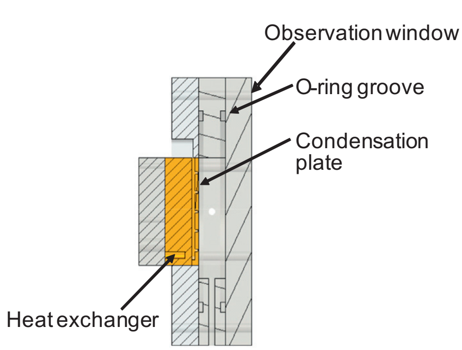

Fig. 1. Condensation assembly. Vapor flow is vertically downward over the condensation plate.

The average temperature of the heat transfer surface is obtained with 25 imbedded 30 Ga thermocouples with junctions located 0.8 mm beneath the condensing surface. One thermocouple is placed in the inlet and outlet of the heat exchanger to measure the rise in water temperature during condensation. Six thermocouples monitor the overall assembly temperature and one thermocouple measures the temperature of vapor in the evaporator. Temperature scans of the entire system are made every 3 s. Average surface temperature, mass flow rate of the cooling water, and condensate mass flow rate are used to calculate the heat transfer rate. Two methods are used to calculate heat transfer during condensation. The first uses the temperature difference between the inlet and outlet of the cooling loop, while the condensate mass flow rate at steady state with latent heat of vaporization is used to calculate the heat transfer rate in the second method. Heat flux estimates via these methods are in fair agreement (within ~25%).

Before each data run the condenser assembly is heated to 333 K, and the evaporator is set to 333 K as well (slightly above the nominal boiling point of FC-72). The heat exchanger is set to a temperature to yield the desired degree of sub-cooling at the condensing surface. Once the condenser assembly and the water bath have reached thermal equilibrium, liquid FC-72 is injected into the evaporator, and the condensing chamber is purged of non-condensable gases. Experiments are run for inlet Reynolds number based on the gap height above the condensing surface of ~15.1, corresponding to a mass flux of 0.264 kg/m^2 s across the condensing surface. The mean fluid speed used to compute the Reynolds number is based on the mass balance for FC-72. Heat flux is $0.92 \leqslant q \leqslant 2.88$ W/cm^2, and sub-cooling is $6.9 \leqslant \Delta T \leqslant 44$ K. For each run, data is collected over 10 min intervals at steady state, typically $t > 300$ s when surface temperature has reached a stable value.

The effect of non-condensable gas (air) on heat transfer coefficients is obtained by partially purging the test chamber with FC-72 vapor. Estimates of the volume fraction of the non-condensable gas are made on the basis of the fraction of time elapsed before a full purge of the test chamber is achieved. We present data for ~13.2% mass fraction of air where significant drop wise condensation is observed. For these runs $Re \approx 3.4$. A camera records the condensation process to determine the rate of droplet formation, number density, and coalescence. Images are extracted from and modified in Adobe Photoshop CS®, and droplet number and size are quantified using ImageHub SpotoGraphics Edition® (v. 1.02).

3. Results

Data runs at each level of surface sub-cooling are repeated four times to capture natural data variance. Heat flux data and the heat