# EXPERIMENTAL STUDIES ON STEAM CONDENSATION IN HORIZONTAL AND VERTICAL MICROTUBES

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## ABSTRACT

Microchannels have increasingly been used in the industry to miniaturize heat transfer equipment, improve energy efficiency, and minimize heat transfer fluid inventory. A fundamental understanding about condensation in microscale will yield far reaching benefits for the automotive and HVAC&R industries. In this study, the effect of microchannel diameter and orientation on condensation heat transfer is investigated. Steam is used as the working fluid, microtubes with inner diameters of 500 and 900 μm inner diameters, in horizontal and vertical orientations were used. The working fluid was pumped into the vapor generator from the reservoir after passing through a micro-filter. Saturated vapor was generated via electrical heating and was then led into the condensing section. Flow condensation occurred in the microtubes. The condensate leaving from the outlet of the condensing section was cooled in the cooler before flowing through the micro flow-meter. It was found that the condensation heat transfer coefficient increased with mass flux, heat flux, and vapor quality, while pressure drop increased with the mass flux and vapor quality. At low mass fluxes, it was found that the channel orientation had a considerable effect on heat transfer coefficient, while this difference diminished as mass flux increased. As mass velocity increased, differences in heat transfer coefficient are reduced.

## INTRODUCTION

Due to the high compactness of latest heat transfer systems, microtubes and microchannels are widely used in different area of industry. Different kinds of condensers have been used in equipments such as heat pumps, air conditioners and chillers. Though many studies have been investigated about two phase heat transfer such as boiling and evaporation, there are less studies on condensation compared to those of them. Coleman and Garimella presented first studies on condensation through microtubes in 1998 [1-3]. Wongwises and Polsongkram [4] studied condensation heat transfer and pressure drop of HFC-134a in tube-in-tube heat exchangers. They investigated the effect of mass flux and condensation temperature on heat transfer and pressure drop and new correlations were developed for heat transfer coefficient and pressure drop in condensation. The effect of tube geometry in condensation has been studied by many researchers with different working fluids such as water, R-11, R-12, R-22, R-113, methanol, ethanol [5] and R-134a [6] . Wilson et al. [7] conducted an experimental study on the effect of different tube geometry in condensation heat transfer.

## NOMENCLATURE

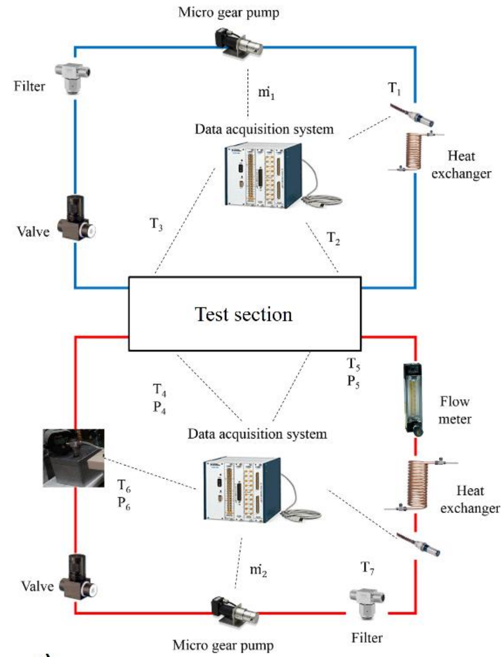
| | | |
|---|---|---|
| $A$ | [-] | Annular flow |
| $h_{tp}$ | [W/m$^2$.K] | Two-phase heat transfer coefficient |
| $C_p$ | [J/kg.K] | Specific heat |
| $d$ | [m] | Diameter |
| $G$ | [kg/m$^2$.s] | Mass flux |
| $g$ | [m/s$^2$] | Gravity acceleration |
| $i$ | [J/kg] | Enthalpy |
| $I$ | [A] | Electrical current |
| $k$ | [W/m.K] | Thermal conductivity |
| $L$ | [m] | Length |
| $m$ | [kg/s] | Mass flow rate |
| $P$ | [pa] | Pressure |
| $q$ | [W/m$^2$] | Heat flux |
| $Q$ | [W] | Heat transfer rate |
| $t$ | [s] | Time |
| $T$ | [K] | Temperature |
| $U$ | [V] | Voltage |
| $x$ | [-] | Vapor quality |
| Special characters | | |
| $\eta$ | [-] | Efficiency |
| $\rho$ | [kg/m$^3$] | Density |
| $\mu$ | [Pa.s] | Viscosity |
| $\sigma$ | [N/m] | Surface tension |
| $\Phi^2$ | [-] | Two-phase flow multiplier |
| $X$ | [-] | Martinelli parameter |
| Subscripts | | |
| *1* | | Evaporator inlet |
| *2* | | Evaporator outlet |
| *a* | | Acceleration pressure |
| *ave* | | Average |
| *c* | | Cooling water |
| con | | Condenser |
| e | | Effective heat transfer |
| eva | | Evaporator |
| f | | Frictional pressure drop |
| g | | Vapor – gravity pressure |
| i | | Inside surface |
| in | | Condenser inlet |
| l | | liquid |
| o | | Outer surface |
| out | | Condenser outlet |
| s | | Shell side |
| t | | Tube side |
| to | | Total |
| TP | | Two-phase flow |
| w | | Wall - water |

They used flattened tubes with different heights in their experiments and heat transfer enhancement were reported as tube were flattened. Corrugated surfaces [8], twisted-tape inserts with different twisted ratios of 6, 9, 12 and 15 [9] and micro-fin tubes [10] have been used due to their high condensation heat transfer performance. The effect of microfin tubes with different aspect ratios are investigated by Lee et al. [11]. They reported that for high mass fluxes, heat transfer enhancement increases as tube aspect ratio increases. Garimella et al. [12] studied condensation heat transfer in rectangular microscale geometries.

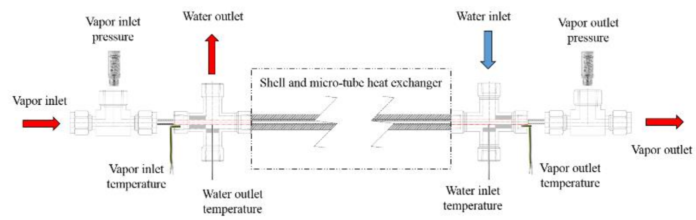
Heat transfer coefficient and pressure drop of refrigerant R134a condensation were determined in rectangular channels with hydraulic diameter ranging $100 < D_h < 160$ μm. They showed that as $T_{sat}$ decreases, due to the decrease in the vapor to liquid density ratio, void fraction increases which leads to an increase in condensation heat transfer coefficient of refrigerant.
Fieg and Roetzel [13] conducted an experimental work to calculate the laminar film condensation in/on inclined elliptical tube. They showed that the elliptical form increases the heat transfer coefficient. Laminar film condensation heat transfer inside inclined heat pipe in solar collector was studied theoretically by Hussein et al. [14]. They reported that pipe inclination angle has a great effect on the heat transfer coefficient of condensation inside the inclined heat pipes. Fiedler et al. [15] investigated the effect of tube inclination on heat transfer in reflux condensation through a tube with diameter of 7mm. They found optimum inclination angle for the heat transfer about 40°. Fiedler and Auracher [16] presented an experimental and theoretical study on reflux condensation inside a 7mm diameter inclined tube. They developed correlations for the film thickness and the mean heat transfer coefficient of condensation. The combined effect of microfin tube and tube inclination was investigated by Akhavan-Behabadi et al. [17]. They showed that the effect of inclination angle is more eminent at low mass fluxes and vapor quality. A correlation also developed to predict condensation heat transfer coefficient for different vapor qualities and mass velocities. Lips and Meyer [18] investigated the effect of inclination on flow pattern and heat transfer coefficient in convective condensation of R134a. 20% heat transfer coefficient enhancement was reported for -15° downward flow and on the other hand for upward flow, condensation heat transfer coefficient can be decreased. Khoeini et al. [19] presented an experimental work on condensation heat transfer of R134a flow in corrugated tube with different inclination ranging from -90° to +90°. Highest condensation heat transfer coefficient was obtained for 30° at low mass velocities and vapor qualities. They also developed a correlation to predict condensation heat transfer of R134a in inclined corrugated tube. The effect of saturation temperature ranging from 30°C to 50°C on condensation heat transfer in inclined tubes for R134a was investigated by Meyer et al. [20]. As they reported, an increase in saturation temperature increases the effect of inclination on two phase heat transfer.

## EXPERIMENTAL SETUP AND DATA REDUCTION

Figure 1a shows a schematic of the experimental setup. The setup consists of two separate loops, including filters, gear pumps, valves, spiral heat exchangers, flow meter, temperature and pressure sensors located in the specified locations, and test section. The test section is designed as a counter-flow micro heat exchanger (shown in Figure 1b).



a)



b)

**Figure 1** Schematic of the a) experimental setup and b) test section

### Data reduction

Before performing the condensation experiments, the efficiencies of the evaporator and condenser were obtained by calibrating their thermal performance. The calibration experiment was performed under single-phase convective heat transfer conditions [21]. Since single phase studies were performed in different system pressures, in higher pressure conditions, the working temperature was as high as 160 °C, which is in the range of two phase condensation and boiling thermal conditions. The thermal efficiency of the evaporator is calculated using the following equation

$$\eta_{eva} = \frac{m_w c_{p,w}\left(T_{w,in} - T_{w,out}\right)}{UI} \tag{1}$$

where $m_w$, $c_{p,w}$, $T_{w,in}$, $T_{w,out}$, $U$ and $I$ are the mass flow rate and specific heat of water, the water temperatures at the evaporator outlet and, water temperatures at the evaporator inlet (correspond to the $T_7$ and and $T_4$ shown in Figure 1, respectively), the voltage and current, respectively. According to the obtained results, the average thermal efficiency of the evaporator was obtained as 0.89.

A similar procedure was performed to obtain the condenser efficiency as follows:

$$\eta_{con} = \frac{m_{w,s} c_{p(w,s)} \left(T_{(w,s),in} - T_{(w,s),out}\right)}{m_{w,t} c_{p(w,t)} \left(T_{(w,t),in} - T_{(w,t),out}\right)} \quad (2)$$

Here the subscripts (w,t) and (w,s) stand for the water in tube side and the water in shell side of shell-tube heat exchanger (condenser). The measured average condenser efficiency is about 0.98.

The condenser inlet enthalpy ($i_{r,in}$) and quality ($x_{in}$), stem from the evaporator efficiency:

$$i_{r,in} = i_{r,1} + \frac{UI\eta_{con}}{m_r} \quad (3)$$

$$x_{in} = \frac{i_{r,in} - i_{l,in}}{i_{\lg,in}} \quad (4)$$

where $i_{r,1}$ is the water's enthalpy at the evaporator inlet, $i_{l,in}$ and $i_{lg,in}$ are the saturated liquid enthalpy and latent heat of evaporation based on the inlet pressure, respectively.

The condenser outlet parameters were obtained in terms of the condenser efficiency. Heat received by the cooling water in the shell tube is given as:

$$Q = m_m C_{p,c} \left(T_{c,out} - T_{c,in}\right) \quad (5)$$

Heat flux based on the inner copper tube wall surface is expressed as:

$$q = \frac{Q}{\pi\, d_i\, L_e} \quad (6)$$

where $L_e$ is the effective heat transfer length. The outlet mixture enthalpy and quality are

$$i_{r,out} = i_{r,in} - \frac{Q}{m_r\, \eta_{con}} \quad (7)$$

$$x_{out} = \frac{i_{r,out} - i_{l,out}}{i_{\lg,out}} \quad (8)$$

where $i_{l,out}$ and $i_{\lg,out}$ are the saturated liquid enthalpy and latent heat of evaporation based on the outlet pressure, respectively. It should be noted that the inlet and outlet vapor qualities are calculated based on the corresponding pressures.

The condensation heat transfer coefficient h is based on the average vapor mass quality, stated as $x_{ave} = \left(x_{in} + x_{out}\right)/2$. Thus, h is computed as [8]:

$$h_{tp} = \frac{1}{\dfrac{1}{h_{to}} - \dfrac{d_i}{2k_w}\ln\left(\dfrac{d_{outer}}{d_{inner}}\right) - \dfrac{d_{inner}}{d_{outer}}\dfrac{1}{h_c}} \quad (9)$$

where $k_w$ is the thermal conductivity of the microtube, $h_{to}$ is the total heat transfer coefficient, and $h_c$ is the heat transfer coefficient of cooling water. The calculation of $h_{to}$ and $h_c$ can be found in Ref [22].

Pressure drops were correlated based on Chisholm method, where two-phase multiplier ratio is expressed as:

$$\phi_l^2 = \frac{\Delta P_{Tf,f}}{\Delta P_{sp.f}} \quad (10)$$

where $\Delta P_{Tf,f}$ and $\Delta P_{sf,f}$ are the two-phase friction pressure drop and friction pressure drop if liquid flows alone. The Lockhart-Martinelli parameter X is calculated as:

$$X^2 = \frac{\Delta P_{liquid,f}}{\Delta P_{gas.f}} \quad (11)$$

where $D_{gas,f}$ is the friction pressure drop if vapor phase flows alone. Finally, the following expression is included to correlate the two-phase friction pressure drop.

$$\phi_l^2 = 1 + \frac{C}{X} + \frac{1}{X^2} \quad (12)$$

## RESULTS AND DISCUSSION

Figure 2 shows the obtained heat transfer coefficients for microtubes with inner diameters of 899 and 500 μm at mass fluxes of 50, 100, 150, and 200 kg/m$^2$.s for different average vapor qualities in horizontal and vertical configurations. Accordingly, it can be observed that the microchannel orientation has no considerable effect on the microchannel with a larger diameter, while the obtained results for the microtube of the inner diameter of 500 μm reveal that the vertical configuration has lower heat transfer coefficients in comparison to the horizontal microtubes. The difference is more pronounced at lower mass flux and vapor qualities. The gravitational force plays a critical role in heat transfer at low flow rate, but its effect is weakened at larger flow rate. This shows competitive balance between gravitational and inertial forces.

Five flow patterns were observed as stratified-wavy, stratified-smooth, intermittent flow, churn flow and annular flow. This is in parallel with the reported results [23, 24]. The stratified-smooth flow happens when the liquid layer height signal is stable. If the liquid layer thickness signal becomes oscillating, the stratified-wavy flow is exist. When the wave crest of the stratified-wavy flow reaches the top of tube wall, the intermittent flow occurs. The stratified-wavy and intermittent flow flows involve apparent interface waves.

The larger tube diameter destroys the occurrence criterion of bubble flow and slug flow. Thus, the two flow patterns are not observed for the microtube with the inner diameter of 889 μm.

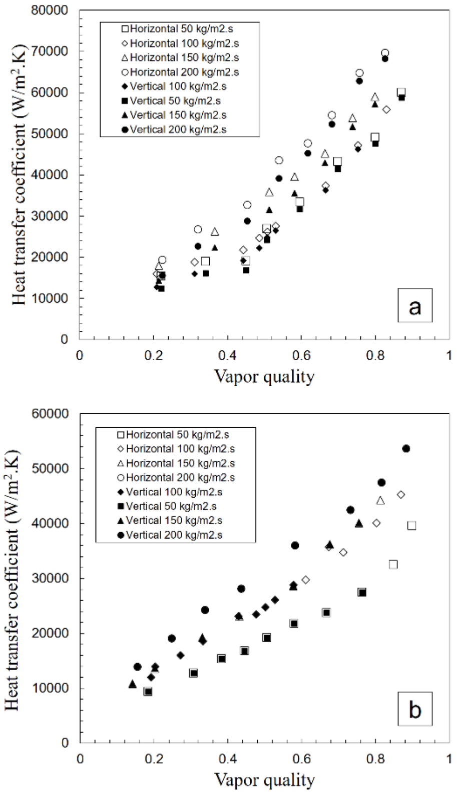



**Figure 2** Variation in heat transfer coefficients with average vapor quality at different mass fluxes in the microtube with a) 500 μm and b) 889 μm inner diameters

The heat transfer coefficient variation for microtubes with the inner diameter of 500 and 889 μm at mass flux of 100 kg/m$^2$.s and vapor quality of 0.4 within 2.5 seconds time is shown in Figure 3. As seen, the heat transfer coefficient in the microtube with inner diameter of 500 μm is more stable than the other microtube. One reason might be related to the transition from annular to intermittent regimes in this microtube. The rather unstable behaviour in the larger microtube might be related to the vapor-liquid interface wave.

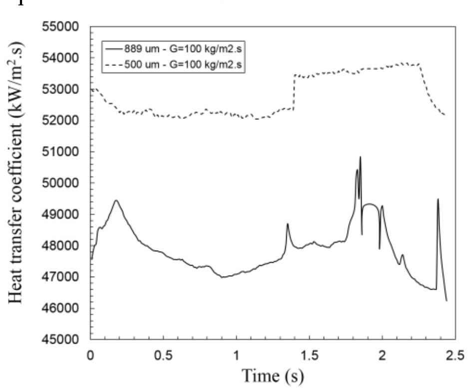



**Figure 3** Heat transfer coefficient variation with time for 2.5 second time period

Figure 4 shows the comparison between obtained experimental results and those predicted by equation 12. As seen for the horizontal configuration, $\phi_l^2$ based on the two-phase frictional pressure drop has a good agreement with the Chisholm method for C=12.

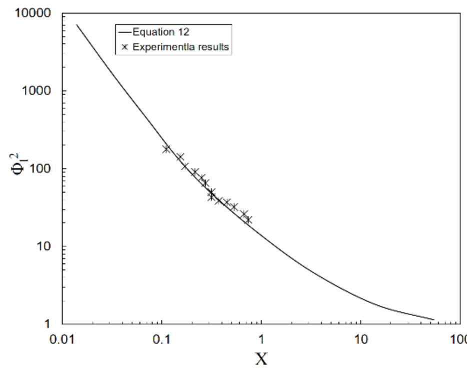



**Figure 4** Comparison between obtained experimental results and those available in the literature

## CONCLUSION

Condensation heat transfer in microtubes with inner diameters of 500 and 889 μm was investigated in for vertical and horizontal configurations. Four mass fluxes of 50, 100, 150, and 200 kg/m$^2$.s were considered to characterize heat transfer coefficients and pressure drops in microtubes. The effect of orientation on heat transfer coefficients were observed. The experimental pressure drops were compared with a correlation available in the literature and good agreement was achieved. The channel orientation has no considerable effect in microtubes with a larger diameter (889 μm inner diameter), while some variations are observed for the smaller microtube at low mass fluxes.

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