SATURATED FLOW BOILING HEAT TRANSFER CHARACTERISTICS
OF PROPANE IN A SMOOTH VERTICAL MINI CHANNEL UP TO
DRY-OUT INCIPIENCE

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ABSTRACT

In this article the two phase heat transfer results of propane are presented. Experiments are performed up to
dryout incipience in a circular mini channel made of stainless steel (AISI 316) having an internal diameter of
1.70 mm and a heated length of 245 mm. Experimental tests are done at a saturation temperature of 23 °C
and for a heat flux range 5 - 240 kW/m². Mass flux ranges from 100 to 400 kg/m²s. The effects of mass flux,
heat flux and vapour quality on heat transfer coefficient are explored in detail. The results shows that the
local heat transfer coefficients of propane are more or less independent of mass flux and vapour quality and
is a strong function of heat flux until the occurrence of partial dryout. Finally the experimental results are
compared with well known micro and macro scale correlations from the literature.

Keyword: Mini channel, Two-phase, Heat transfer, Propane.

1. INTRODUCTION

The need to dissipate high heat fluxes is an important issue in a number of applications, including electronic
cooling and MEMS (Micro-Electro-Mechanical-Systems) devices (Steinke and Kandlikar, 2004). Extensive
investigation of flow boiling in mini and micro channels has been done in recent years because enhanced
heat transfer could be achieved by reducing hydraulic diameter of the channels in heat exchangers (Palm,
2001). Apart from the enhanced heat transfer, mini and micro channels also offer other advantages like
reduced cost (less material requirement), low fluid inventory and low weight. The implementation of micro
channel heat sinks operating in the two-phase regime in practical applications has been slow due to the
complexity of boiling phenomena at the micro scale and the resulting difficulty in predicting heat transfer
performance (Garimella and Harirchian, 2009).

The research of flow boiling in small diameter channels has been focussed on HFC refrigerants. Recent
awareness of environmental concerns has led to a demand to get knowledge of flow boiling phenomena of
natural refrigerants in small diameter channels. Propane is a natural refrigerant with zero Ozone Depletion
Potential and a very low Global Warming Potential. Moreover it is chemically stable, compatible with most
of the materials and miscible with commonly used compressor lubricants. Propane has very good thermo-
physical properties that resemble those of HFC refrigerants (Fernando, 2007). The low fluid inventory in
small diameter channels mitigates concerns of high flammability of propane.

Choi et al. (2009) performed experiments to investigate the flow boiling of propane in horizontal mini
channels of 1.5 and 3.0 mm internal diameters. Flow boiling results were obtained for heat flux ranging
from 5 to 20 kW/m², mass flux ranging from 50 to 400 kg/m²s, saturation temperatures of 0, 5 and 10 °C and
quality up to 1. Mass flux was observed to have insignificant effect on heat transfer coefficient at low quality
region and the heat transfer coefficient is higher for higher mass flux at medium and high quality region.
Contrary to mass flux, the heat transfer coefficient was observed to be higher for higher heat flux at low
vapour quality while heat flux had insignificant effect at higher quality region. The heat transfer coefficient
was increased with decrease in internal diameter and increase in saturation temperature. A new heat transfer
correlation based on the superposition model was presented with 10 % mean deviation.

Fernando (2007) conducted experiments with propane in a multiport mini channel heat exchanger mounted
as an evaporator in a small heat pump test rig. Tests were done for evaporating temperature range from -15
to 10 °C, heat flux from 2 to 9 kW/m² and mass flux ranging from 13 to 66 kg/m²s. The heat transfer coefficient was observed to be higher for higher heat flux and for increasing evaporation temperatures. For a fixed evaporating temperature and for heat flux higher than 4.5 kW/m², the heat transfer coefficient was observed to be independent of heat flux. It was observed also that the correlations emphasizing nucleate boiling were in good agreement with experimental data.

Very few flow boiling studies of propane are found in the literature. Experimental data for heat transfer of propane in small hydraulic diameter tubes is therefore needed, both because this may be an interesting fluid in the near future and because data for a wider range of fluids may help in the understanding of the mechanisms of heat transfer. In these experiments, the heat transfer coefficient has been determined on the basis of measured parameters in a vertical 1.70 mm diameter tube. The effects of mass flux, heat flux, and vapour quality on the heat transfer coefficient has been explored in detail. The experimental data was compared with well known correlations.

2. EXPERIMENTAL SET-UP AND METHODOLOGY

The experimental vertical mini channels test rig is shown in Figure 1. The experimental set-up consists of a close loop: the subcooler, the magnetic gear pump, the coriolis mass flow meter, the pre heater, the test section and the condenser. The subcooler fully subcools the refrigerant to reduce the cavitation risk of the magnetic gear pump. The subcooled refrigerant flows to the inlet of the magnetic gear pump (ISIMATEC, type MCP-Z standard) which circulates it to the test rig. The mass flow rate is controlled by adjusting the speed of the pump and is measured by a (MicroMotion DS006) coriolis mass flow meter. The refrigerant then enters the pre heater where it is heated up to obtain the desired inlet temperature at the inlet of the test section. After passing through the test section, the refrigerant enters the condenser, after which the condensate enters the subcooler to complete the loop. The system pressure is maintained by a tank connected to main loop placed in a water bath whose temperature is maintained by a thermostat. The system pressure is measured by an absolute pressure transducer (Druck PDCR 4060, 20 bar). A differential pressure transducer (Druck PDCR 2160, 350 mbar) is used to measure pressure drop across the test section. The refrigerant temperature at the inlet and the outlet of the test section is measured by T-type thermocouples. The refrigerant absorbs heat in the test section and boils when power is applied to the wall of the test section by a DC power supplier. The test rig is insulated by thermal insulation to prevent heat loss to the surroundings. The outer wall temperatures of the test section are measured by T-type thermocouples. The thermocouples are attached to the outer wall of the test section after covering them with a special epoxy which is thermally conductive and electrically insulating. The inner wall temperatures are calculated by the heat conduction equation (Owhaib, 2007) from the measured outer wall temperatures.

The test section consists of a stainless steel (AISI 316) tube with inner diameter of 1.70 mm and 245 mm in length. A glass tube of the same inner diameter as of test section is inserted after and before the test section to visualize the flow regime and to insulate the test section electrically and thermally from the test rig. The roughness of the test section was determined by scanning the inner surface using a method called conical stylus profilometry. Five profiles of the inner surface of test section were obtained. The details of the roughness test result are shown in table 1 where \( R_a \) represents the arithmetic mean roughness, \( R_v \) the maximum valley depth and \( R_p \) is the maximum peak height.

The desired test conditions are maintained at the start of the experiment. All tests are performed in steady state conditions. After achieving steady state conditions, the data is recorded by a data logger connected to a computer for almost five minutes and around 100 data points are recorded. The average values of these data points are used in calculations. Mainly four types of measuring parameters are determined:

- The outer wall temperatures of the test section at 9 positions and the refrigerant bulk fluid inlet and outlet temperatures.
- The mass flow rate of the refrigerant to calculate mass flux through the test section.
- The power applied to the test section to calculate the heat flux applied to the test section.
- The absolute pressure during tests to know saturation conditions across the test section.

All the thermal and transport properties including enthalpy, density, viscosity and thermal conductivity of propane are calculated using REFPROP 7 developed by NIST (National Institute of Standards and
The uncertainty propagation in derived parameters from the uncertainties in the measurement of diameter, tube length, power input, temperature pressure and mass flow rate is calculated by EES (Engineering Equation Solver) software which uses method for uncertainty propagation mentioned in (Taylor B.N. and Kuyatt, C.E., Guidelines for Evaluating and Expressing the Uncertainty of NIST Measurement Results, National Institute of Standards and Technology Technical Note 1297, 1994). The propagation in experimental uncertainties is tabulated in table 2.

![Schematic diagram of experimental test rig](image)

Figure 1: Schematic diagram of experimental test rig

<table>
<thead>
<tr>
<th>Tube Inner Dimension</th>
<th>R_a (μm) Arithmetic Mean Roughness</th>
<th>R_v (μm) Max. Valley Depth</th>
<th>R_p (μm) Max. Peak Height</th>
</tr>
</thead>
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<tr>
<td>1.70 mm</td>
<td>0.21</td>
<td>-0.73</td>
<td>0.80</td>
</tr>
</tbody>
</table>

Table 2: Operational Conditions and Uncertainties

<table>
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<tr>
<th>Parameter</th>
<th>Operating Range</th>
<th>Uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>D (mm)</td>
<td>1.70</td>
<td>± 0.007</td>
</tr>
<tr>
<td>Z_{bus} (mm)</td>
<td>245</td>
<td>± 0.2</td>
</tr>
<tr>
<td>G (kg/m³s)</td>
<td>100- 400</td>
<td>± 3.5%</td>
</tr>
<tr>
<td>q'' (kW/m²)</td>
<td>5-240</td>
<td>± 2.5%</td>
</tr>
<tr>
<td>h (kW/m²K)</td>
<td>4-35</td>
<td>±11%</td>
</tr>
<tr>
<td>X</td>
<td>0-0.98</td>
<td>± 6%</td>
</tr>
</tbody>
</table>
3. DATA REDUCTION

The local boiling heat transfer coefficient is calculated according to Newton’s law of cooling given below;

\[
h = \frac{q''}{T_{w,z} - T_{sat,z}}
\]  

(1)

\(q''\) is the heat flux which is the ratio of the power applied to the test section and the internal surface area based on the heated length. The power can be determined by multiplying the voltage and the current values from the power source. \(T_{w,z}\) is the inner wall temperature calculated from the measured outer wall temperature by using heat conduction equation (Owhaib, 2007). The assumptions of this calculation are: one dimensional conduction in radial direction, steady state, uniform heat distribution in the tube wall and no heat loss to surroundings due to insulation around test section. \(T_{sat,z}\) is obtained from the corresponding pressure, calculated from the measured inlet pressure and pressure drop along the tube, assuming a linear profile along the whole test section.

The average heat transfer coefficient is determined by averaging local heat transfer coefficients arithmetically.

The vapour quality at any vertical location (\(z\)) is determined from the heat transferred to the fluid as;

\[
x_z = \frac{q'' \pi D (Z-Z_s)}{A_c G \cdot l_{lv}}
\]  

(2)

\(Z\) is the desired location where vapour quality is estimated, \(Z_s\) is the location where saturation conditions along the test section are reached and can be determined as;

\[
Z_s = \frac{m c_p (T_{sat} - T_{in})}{q'' \pi D}
\]  

(3)

\(q''\) is the heat flux, \(D\) is the internal diameter of the test section, \(A_c\) is the cross sectional area, \(G\) is the mass flux, \(l_{lv}\) is the latent heat of vaporization, \(m\) is the mass flow rate, \(c_p\) is the specific heat of the fluid, \(T_{sat}\) is the saturation temperature and \(T_{in}\) is the inlet temperature at the test section.

4. RESULTS AND DISCUSSION

The local heat transfer coefficients are plotted against local vapour qualities for different mass fluxes in Figure 2a. It is clearly seen that for the lower heat flux of 40 kW/m² and also for higher heat flux of 100 kW/m², the local heat transfer coefficients are independent of mass flux. The similar trends were also observed by Callizlo (2010), Ali (2010) and Arima et al. (2010). To see the effect of mass flux on the heat transfer coefficient for all range of heat fluxes, the average heat transfer coefficients are plotted against heat flux.
flux in Figure 2b. It can be seen that the data points of all mass fluxes overlap each other which is also an indication of the independence of the heat transfer coefficient on mass flux. The similar trends are also observed by Garimella and Harirchian (2009). This type of results indicates that the flow boiling is more influenced by phenomena similar to nucleate boiling and the convective effects are of minor importance.

The local heat transfer coefficients are plotted in Figures 3a and 3b against local vapour qualities for mass fluxes of 200 kg/m²s and 400 kg/m²s respectively. Both the diagrams are plotted for a range of heat fluxes up to the occurrence of dryout. From the figures the main observations are;

- The local heat transfer coefficient increases with heat flux except at the highest vapour fractions where partial dryout is assumed to take place.
- The local heat transfer coefficient is more or less independent of vapour quality until the partial dryout occurs.
- A local maximum is observed at third thermocouple located 90 mm from the entrance of the test section. This corresponds to local vapor qualities of about 0.35 for 200 kg/m²s in Figure 3a and about 0.3 for 400 kg/m²s in Figure 3b for the highest heat fluxes just before the start of the deterioration of heat transfer, probably caused by beginning dryout of the tube surfaces. This local maximum in the heat transfer coefficient may be explained by frequent rewetting of the inner wall of the test section by incoming liquid waves and evaporation of thin liquid films on the surface.
- The deterioration of the heat transfer coefficients at the highest heat fluxes in both figures is thought to be due to the occurrence of partial dryout.

The strong dependence of the heat transfer coefficient on heat flux and independence of vapour quality again show that flow boiling in the minichannel has parameter dependence similar to that expected for nucleate boiling. Similar trends are also observed by Vlasie et al. (2004), Callizo (2010) and Ali (2010).

Based on previous studies of evaporation in electrically heated glass tubes, it is believed that the dryout incipience occurs when part of the thin liquid film on the inner wall of the test section evaporates in between waves coming from the upstream direction. The liquid waves again rewet the inner wall unless permanent dryout occurs, (which will happen at high enough vapour fractions).
The dryout incipience heat flux at the last thermocouple for a given mass flux can be determined by the change of slope in the boiling curve, at that position as can be seen in Figure 4b. The boiling curve is plotted for all the tested mass fluxes. The heat flux at dryout incipience is indicated by $q''_{id}$ for each mass flux. It can also be seen that the heat flux at dryout incipience increases with increase in mass flux, indicating the importance of the vapour fraction.

![Figure 4: (a) Average heat transfer coefficient versus mass flux. (b) Dryout incipience heat fluxes at last thermocouple position for each mass flux.](image)

5. COMPARISON WITH CORRELATIONS

The experimental heat transfer coefficients are compared with existing generalized macro and micro scale correlations to test their predicting capability.

Figure 5a shows the comparison of the local heat transfer coefficient with Cooper (1984) nucleate pool boiling correlation. The experimental data of propane is nicely captured by this correlation. For flow boiling in narrow tubes, surface tension forces become highly important. The evaporation of thin liquid films during flow boiling in the tube has similarities with pool boiling which can be a reason that this correlation predicts the data with Mean Absolute Deviation (MAD) of only 14%.

Tran et al. (1996) suggested a correlation for mini and micro channels. The comparison with this correlation was expected to work well as this correlation gave good predictions in previous studies (Ali, 2010) with R134a as working fluid. But for propane, this correlation under predicted the local heat transfer coefficients with a MAD of 33%. The predictions of Tran et al. (1996) are plotted in Figure 5b.

The experimental local heat transfer coefficients are compared with the power type asymptotic correlations proposed by Gungor and Winterton (1986) and Liu and Winterton (1991). Both these correlations include Cooper (1984) correlation as the nucleate boiling contribution. These correlations both use an enhancement factor and a suppression factor but with different definitions to take into account of convective and nucleate boiling heat transfer mechanisms. Gungor and Winterton (1986) correlation predicts the data with a MAD of 19% as shown in Figure 6a. Liu and Winterton (1991)correlation captures the boiling phenomena and predicts the data with a MAD of only 8%.The predictions of Liu and Winterton (1991) correlation are shown in Figure 6b.

Lazarek and Black (1982) developed a correlation for small diameter channels. The dependence of heat transfer coefficient in this correlation on heat flux is comparable to that in Cooper (1984) pool boiling correlation. This correlation predicts the data better at low mass and heat fluxes than at high mass and heat fluxes. Overall this correlation predicts the data with MAD of 13% as shown in Figure 7a. Kew and Cornwell (1997) suggested a correlation which is a modified form of Lazarek and Black (1982) correlation. The local heat transfer coefficients calculated on the basis of Kew and Cornwell (1997) correlation is compared with our local heat transfer data and the result is shown in Figure 7b. As in comparison with Lazarek and Black (1982), this correlation also over predicted the data at high mass and heat fluxes but overall this correlation predicted the data with a MAD of 18%.
Figure 5: (a). Comparison of local heat transfer coefficients with Cooper (1984) correlation (b). Comparison of local heat transfer coefficients with Tran et al. (1996) correlation

All MAD have been calculated using all data points, including those after inception of dryout. As shown in the figures, the fit in all cases would be better if these data points had been omitted. By excluding data point after dryout inception, Cooper (1984) correlation, Tran et al. (1996) correlation, Gungor and Winterton (1986) correlation, Liu and Winterton (1991) correlation, Lazarek and Black (1982) correlation and Kew and Cornwell (1997) correlation predicted the experimental data with MAD of 11%, 33%, 17%, 6%, 10% and 11% respectively.

Figure 6: (a). Comparison of local heat transfer coefficients with Gungor and Winterton (1986) correlation (b). Comparison of local heat transfer coefficients with Liu and Winterton (1991) correlation

6. CONCLUSIONS

Experiments are performed to investigate heat transfer in a circular vertical mini channel made of stainless steel (AISI 316) with an internal diameter of 1.70 mm and a uniformly heated length of 245 mm using propane as working fluid. The heat transfer coefficient in general is independent of mass flux and vapour qualities and is a strong function of heat flux up to dryout incipience which shows that the flow boiling is controlled by a mechanism similar to nucleate boiling. The deterioration in the heat transfer coefficient is observed at partial dryout condition and the heat transfer coefficient sharply decreased at the occurrence of full dryout. The experimental data of propane is compared with generalised correlations of macro and micro scale and apart from Tran et al. (1996) correlation; all correlations included in this study predicted our experimental data reasonably well.
Figure 7: (a). Comparison of local heat transfer coefficient with Lazarek and Black (1982) correlation (b). Comparison of local heat transfer coefficient with Kew and Cornwell (1997) correlation

7. REFERENCES