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TWO PHASE HEAT TRANSFER OF AMMONIA IN A MINI/MICRO CHANNEL

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ABSTRACT

Experiments have been performed to investigate heat transfer in a circular vertical mini channel made of stainless steel (AISI 316) with internal diameter of 1.70 mm and a uniformly heated length of 245 mm using ammonia as working fluid. The experiments are conducted for a heat flux range of 15 to 350 kW/m^2 and mass flux range of 100 to 500 kg/m^{2}s. The effects of heat flux, mass flux and vapour quality on the heat transfer coefficient are explored in detail. The experimental results show that the heat transfer coefficient increases with imposed wall heat flux while mass flux and vapour quality have no considerable effect. Experimental results are compared to predictive methods available in the literature for boiling heat transfer. The correlations of Cooper et al. [1] and Shah [3] are in good agreement with our experimental data.

Keywords: Heat Transfer, Two phase, Mini channel

1. INTRODUCTION

Two phase flow heat transfers in mini channels has been attracting a lot of attention during the last decade due to its possible applications in fuel cells, cooling systems for high performance microelectronics etc. Due to global warming and ozone depletion concerns, it is necessary to study the behavior of natural refrigerants in mini channels to meet the future requirements. The interest in mini channel heat exchangers during the last decade has resulted in a large number of experimental studies of heat transfer of two-phase flow. The majority of these studies have been performed using HFC refrigerants with fairly similar properties. There are still discrepancies which need to be clarified despite great efforts made by researchers to understand the phenomena of flow boiling in mini and micro channels. Therefore to understand the flow boiling phenomena, it is important to do tests with different fluids so that on the basis of experimental observations, a better understanding can be reached about the behavior of two phase flow in mini and micro channels.

Due to the need of experimental observations of different fluids, ammonia heat transfer results are reported in this article. Ammonia has been used in large refrigeration systems for more than a hundred years. Since ammonia is toxic at low concentration levels so it has not been considered for domestic appliances. But low inventory of ammonia in mini and micro channels make it possible to take advantage of its excellent thermodynamic and heat transfer properties and also to address the global warming and ozone depletion concerns. The operating pressures of ammonia are comparable with other refrigerants.

2. LITERATURE SURVEY

Ammonia heat transfer studies for mini and micro channels have not been reported before in the open literature. The available studies are only for in macro scale channels. Some of the related studies are mentioned below.

Zamfirescu et al. [4], Kabelac et al. [5], Zurcher et al. [6] and Boyman et al. [7] conducted experiments for the heat transfer of ammonia inside macro channels of inner diameter range of 14 to 32 mm and wide range of mass fluxes and heat fluxes. Generally it was observed that for same heat flux heat transfer is higher for higher mass flux.

Tahar et al. [8] conducted experiments for forced convective boiling heat transfer for ammonia water mixture in 6 mm vertical smooth tube. The two phase heat transfer coefficient increased up to vapor quality 0.3 then it stayed constant for higher vapor quality. When compared with models, Mishra et al. [9] model predicted the convective boiling quite well.

Thome et al. [10] reviewed the available data of ammonia. They compared the Zurcher et al. [6] and Kabelac et al. [5] experimental data and concluded that heat transfer coefficient increases with increase of mass flux. For mass flux greater than 80 kg/m^{2}s, heat transfer coefficient increase with increase of vapor quality which shows dominance of convective boiling. For mass flux less than 50 kg/m^{2}s, heat transfer coefficient remains almost constant with increase of vapor quality which the authors suggests shows dominance of nucleate boiling. They compared the results with available predictive methods but no method gave good results.

In the present investigation, measurements have been done in a test rig previously used with other fluids. Owhaib et al. [11] did experiments in 1.70, 1.224 and 0.826 mm tube using R134a as test fluid. It was observed that the heat transfer coefficient was dependent on heat flux but independent of vapor quality and mass flux. It was also observed that the heat transfer coefficient was higher for small diameter channel. Ali et al. [12] did heat transfer tests up to dry out conditions using R134a in the same test section as used for the present study and it was observed that the heat transfer coefficient increased with increase of heat flux and system pressure but mass flux and vapor quality had negligible effect. Pool boiling correlations predicted the experimental data well.

3. EXPERIMENTAL FACILITY

The experimental apparatus is schematically illustrated in Figure 1. The refrigerant coming from the sub cooler was pumped by a magnetic gear pump, type MCP-Z standard, to the test section. This pump allows a wide range of flow rates. The circuit included a Coriolis mass flow meter to measure the flow rate. To adjust the inlet temperature of test section, a pre heater was used. A filter of 7 micro meters was used to restrict any particles to enter the test section.

An absolute pressure transducer (Druck, 25bar) was used to measure the system pressure and the pressure drop across the test section was measured by a differential pressure transducer (Druck, 350mbar). The test section consists of metal (AISI 316 stainless steel) tube with inner diameter of 1.70 mm.

Ten T-type thermocouples were mounted on the surface of the test section to measure the wall temperature. The tip of each thermocouple was electrically insulated and then attached at the outer wall with special epoxy which is thermally conductive and electrically insulating. Temperatures were measured at outer wall then calculation was done to get inner wall temperature. To measure temperature at the inlet of test section, at the outlet of test section and at different system points, T type thermocouples of 0.1mm diameter were installed.

The test section was heated using an electric DC power supply by applying a potential difference over the test tube itself. This direct heating ensured homogeneous heat flux over the test section. After the test section, the fluid was condensed in the condenser and further sub cooled in sub cooler. As a first step, mass flow, pressure and inlet temperature was set then electric power was applied step by step. Data was recorded for each step when steady state conditions were achieved. The temperatures, the mass flow and the system pressure were recorded using a data logger connected to a computer. Thermal and transport properties of ammonia were taken from REFPROP 7.

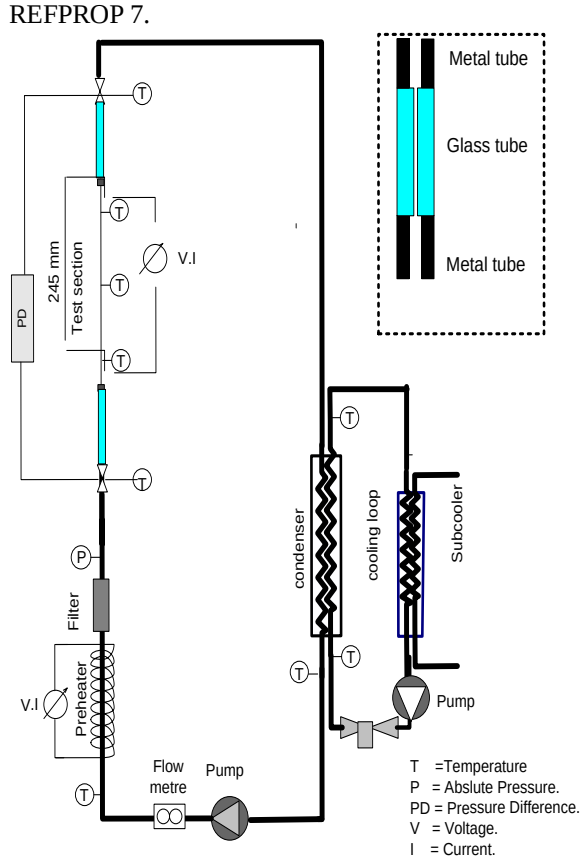

Figure 1. Schematic diagram of experimental test rig

4. DATA REDUCTION

For a given test, the heat flux added to the test section is calculated as;

$$q'' = \frac{Q}{A} \quad (1)$$

Where

$Q = IV$

And

$A = \pi D L_h$

where I and V are the current and voltage, A is the heat transfer area, D is the inner diameter of test section and L_h is the heated length. At the inlet of test section, there is sub cooling of 1K.

The vapor quality at any vertical location (z) is calculated as;

$$x(z) = \frac{q'' P(z - z_\circ)}{A_c G i_{fg}} \quad (2)$$

Where

$$z_\circ = \frac{\dot{m}_{NH_3} C_p (T_{sat} - T_{in})}{q'' P} \quad (3)$$

C_p is the specific heat of the fluid, $\dot{m}$ is the mass flow rate of ammonia, T_{sat} is the saturation temperature, T_{in} is the inlet temperature of test section, q'' is the heat flux and P is the perimeter. z_o is the location on the heated section at which saturated conditions would be reached.

Under subcooled conditions, the bulk temperature at any axial position is calculated from the inlet temperature and the heat added to the test section:

$$T_{f,z} = T_{f,in} + z \frac{q'' \pi D_i}{m c_p} \quad (4)$$

The local heat transfer coefficient under sub cooled conditions can be calculated as;

$$h = \frac{q''}{T_{wall,in} - T_{f,z}} \quad (5)$$

The inside wall temperature, $T_{wall,in}$ is calculated from the measured outside surface temperature using the solution of the steady-state one-dimensional heat conduction equation [15] shown below;

$$T_{wall,in} = T_{wall,out} + \frac{Q}{4\pi k L_h} \left[\frac{\varphi(1 - ln\varphi) - 1}{\varphi - 1} \right] \quad (6)$$

$$\varphi = \left(\frac{D_{out}}{D_{in}} \right)^2$$

k is the thermal conductivity of the test section and Q is the heat applied to heating length.

The local boiling heat transfer coefficient under saturated conditions can be calculated as;

$$h = \frac{q''}{T_{wall,in} - T_{sat,z}} \quad (7)$$

The local saturation temperature, $T_{sat,z}$, is obtained from the corresponding pressure, calculated from the measured inlet pressure and pressure drop, assuming the latter to present a linear profile along the whole test section. The average heat transfer coefficient is determined by averaging local heat transfer coefficients arithmetically.

5. RESULTS AND DISCUSSION

5.1. Boiling Curve

Figure 2 shows the typical boiling curve for T_{sat}=23° C and G=100-500 kg/m^{2}s. Boiling curve is achieved by maintaining the mass flux and increasing the heat flux on the tube wall. Note that the temperature difference in this plot is the average difference between the tube wall and the saturation temperature.

The last point of 100 kg/m^{2}s deviates from boiling curve. This deviation is caused by dry out in the upper part of the test section, causing the wall superheat to increase sharply with small increase in heat flux. It can be observed that for different mass fluxes, wall superheat is almost the same for a given heat flux. From this it can be concluded that the boiling curve is more or less independent of mass flux up to dry out point.

Boiling curve for mass flux of 100 kg/m^{2}s is presented in Figure 3. In this boiling curve, heat flux is plotted versus the local temperature difference between the tube wall and the saturation temperature at the different locations along the tube. At heat flux of 204kW/m^2, dry out occurs and the wall superheat of the last two thermocouples starts to increase sharply which is shown by arrows. At low heat flux, the temperature difference is about the same along the tube, while at higher heat fluxes, the influence of position, and thereby of vapor quality, is higher, indicating a larger influence of convective effects.

Figure 4 shows the boiling curve for mass flux of 300 kg/m^{2}s. The encircled points deviate from the boiling curve and indicate temperature overshoot due to suppression of nucleation. The heat transfer coefficient is independent of heat flux for encircled points which shows the dominance of convective heat transfer for these points, as can be expected in single phase flow. At incipience of nucleate boiling, the wall temperature decreases suddenly which can be seen from the

Figure 4. As for the lower mass flux, the temperature differences along the tube are larger at higher heat fluxes, where the difference in vapor fraction between inlet and outlet is larger.

5.2. Effect of Mass Flux

The effect of mass flux on the average heat transfer coefficient for a wide range of heat fluxes is plotted in Figure 5. It is seen that in general, the average heat transfer coefficient is weakly dependent on mass flux. It is also observed that the heat transfer coefficient is a function of heat flux. This type of dependence is often taken as an indication that nucleate boiling is dominating, and that convective evaporation is of minor importance. However, from previous visualization studies, we expect nucleation of bubbles to take place only close to the inlet of the tube. The dependence of the heat transfer coefficient on the heat flux seem to be important also during plug/slug flow, and perhaps even in annular flow, in mini channels. Similar results were also observed by Owhaib et al. [11].

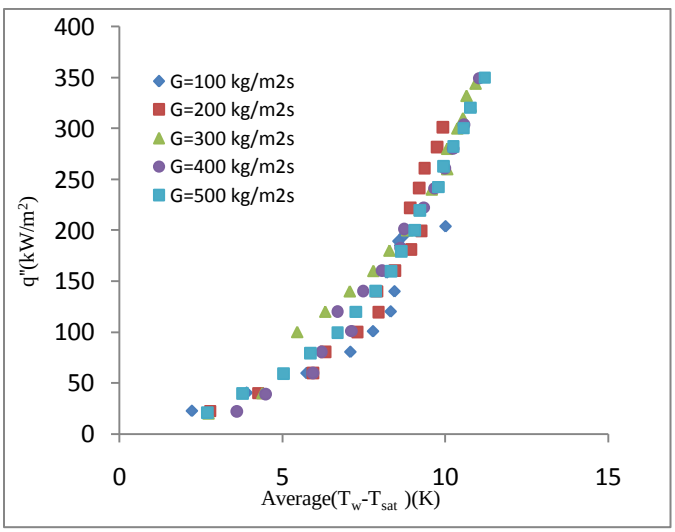

Figure 2. Boiling Curve at T_{sat}=23ºC

In Figure 6, average heat transfer coefficients are plotted as a function of heat flux for different mass fluxes. It can be seen that heat transfer coefficient is independent of mass flux as curves of different mass fluxes cluster together. Obviously, the convective contribution to the average heat transfer coefficients is low. It can be seen that heat transfer coefficients of different mass fluxes diverge from each other at lower heat fluxes which can be due to suppression of nucleation at these points i.e. may be due to transition from single phase superheated region to nucleate boiling region. Similar results were also observed by Garimella et al. [13].

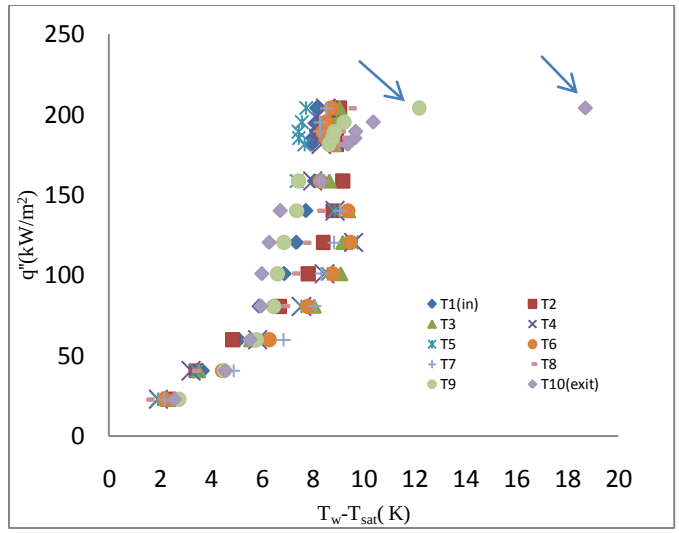

Figure 3. Boiling Curve for 100 kg/m^{2}s

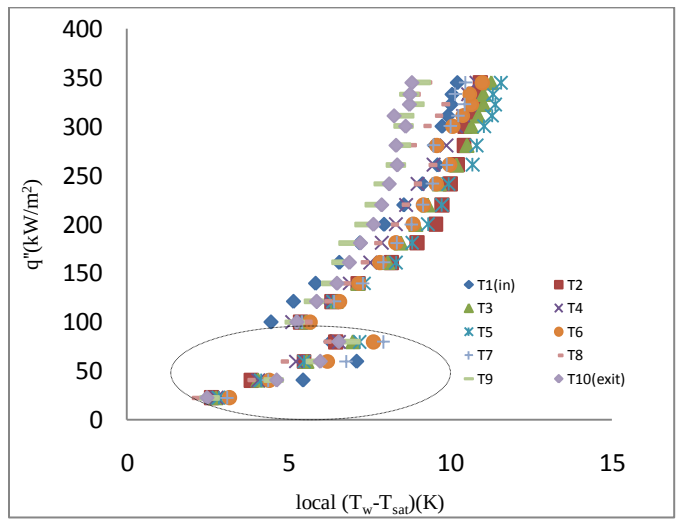

Figure 4. Boiling Curve for 300 kg/m^{2}s

5.3. Effect of Heat Flux and Vapor Fraction

Local heat transfer coefficients are plotted against vapor quality for mass flux of 100 kg/m^{2}s in Figure 7. It can be seen that heat transfer coefficient is higher for higher heat flux. It can also be seen that the heat transfer coefficients are mainly independent of vapor fraction. There are a few exceptions worth mentioning: The first data point at 195 kW/m2 shows very low heat transfer coefficient, probably due to suppression of boiling leading to temperature overshoot at this point. Secondly, there is an indication, for all heat fluxes,

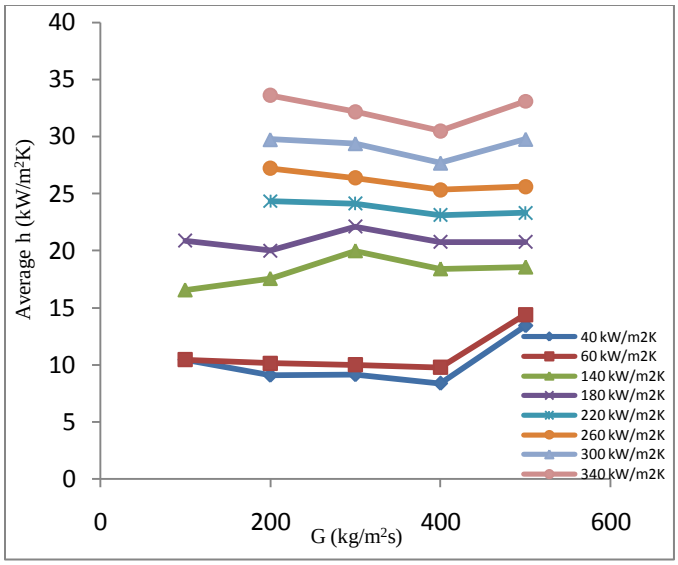

Figure 5. Average Heat Transfer Coefficient at T_{sat}=23ºC

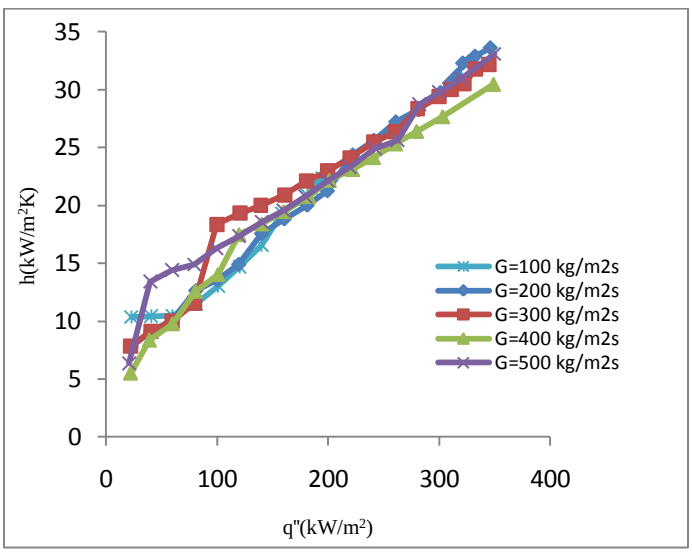

Figure 6. Average heat transfer Coefficient versus heat flux at T_{sat}=23ºC

that there is a local minimum in the heat transfer coefficients at vapor fractions of about 0.2. Thirdly, for the heat flux 204 kW/m^2 K, the heat transfer coefficients decreases sharply at the highest vapor fractions, probably due to partial dry out.

In Figure 8 and Figure 9, local heat transfer coefficients are plotted against local vapor quality for a mass flux of 300 kg/m^{2}s and 500 kg/m^{2}s respectively. It can be seen that at 300 kg/m^{2}s, the heat transfer coefficients are almost constant, independent of vapor fraction, for all heat fluxes. Only a slight increase may be seen at the highest vapor fractions.

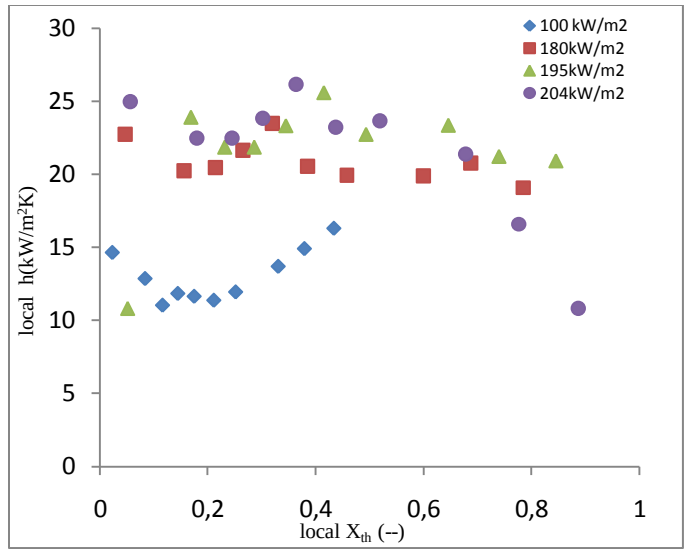

Figure 7. Local heat transfer coefficient versus local vapor quality at 100 kg/m^{2}s.

At 500 kg/m^2 s, there is a clear indication that the heat transfer coefficients increase with vapor fraction above 0.1. More tests are needed to confirm this trend. Note that the range of vapor fractions in these tests are limited by the length of the test section.

It can also be seen that for higher mass flux, dependence of heat transfer coefficient on vapor quality starts at lower vapor quality. This means that at higher mass flux, contribution of convective boiling phenomena increases while this effect is lower at lower mass flux.

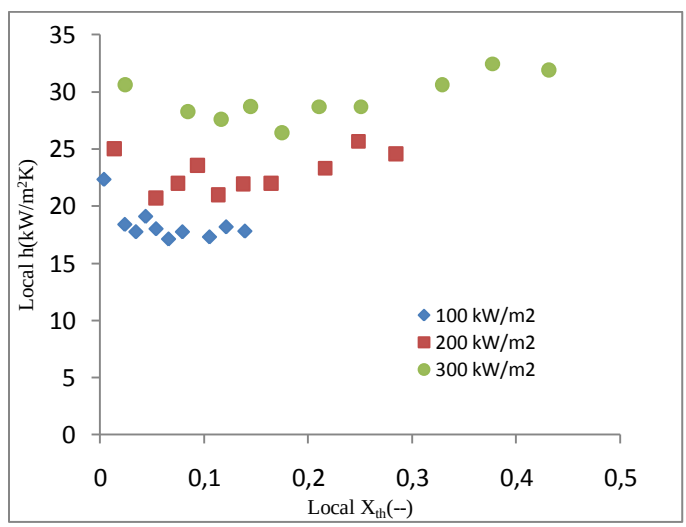

Figure 8. Local heat transfer coefficient versus local vapor quality at 300 kg/m^{2}s.

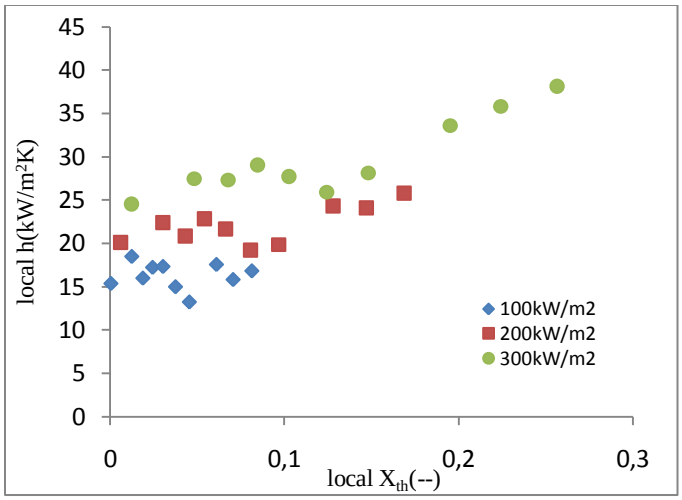

Figure 9. Local heat transfer coefficient versus local vapor quality at 500 kg/m^{2}s.

6. COMPARISON WITH CORRELATIONS

The experimental data is compared with many correlations available in literature. The correlations which are presented here are combined in tabular form in table 1.

Figure 10 shows the average experimental heat transfer coefficient plotted versus heat transfer coefficients predicted by Cooper's [1] pool boiling correlation. At lower heat fluxes, predictions are not so good, probably due to partial suppression of nucleation and resulting superheating of the fluid, but overall it predicts the data very well with MAD of 8.5%. For flow boiling in narrow tubes, surface tension forces are highly important, leading to evaporation of thin liquid films in the tube, at least at lower vapor fractions. May be due to this reason, Cooper [1] pool boiling correlation predicts the data well. The local experimental heat transfer coefficient versus the local predicted heat transfer coefficient are also shown in Figure 11.

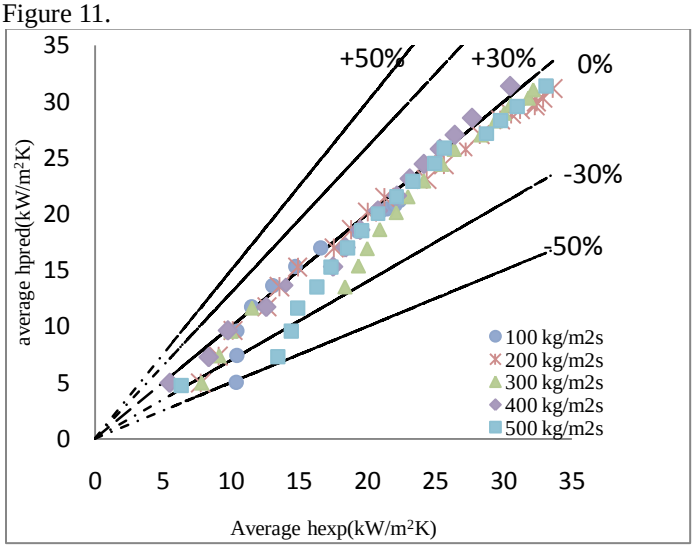

Figure 10. Comparison of average heat transfer coefficient with Cooper [1] correlation.

Shah correlation [3] is added in this study because it was developed on the basis of ammonia data, but in larger diameter tubes.

Figure 12 shows the comparison of experimental data with Shah correlation [3]. Local experimental heat transfer coefficients are plotted against local predicted heat transfer coefficients in Figure 12. Shah correlation [3] accounts for both nucleate and convective boiling mechanism but for calculation of two phase heat transfer coefficient, it takes maximum of the two. Shah correlation does not predict data well at high heat flux and local data points are quite scattered.This may be explained if both mechanisms are important at high heat fluxes. Shah correlation [3] predict the average values quite well with a MAD of 22.5% which can be seen in Figure 13.

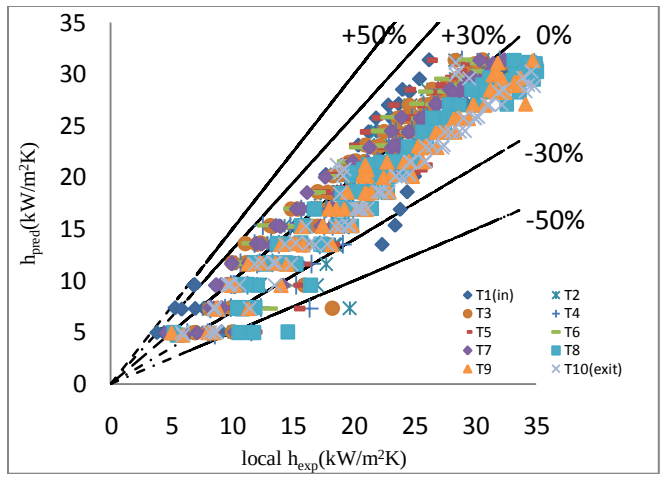

Figure 11. Comparison of local heat transfer coefficient with Cooper [1] correlation.

Table1: Correlations presented in article

Reference	Correlation
Cooper [1]	$h_{tp} = 35P_r^{0.12}(-log_{10}P_r)^{-0.55}M^{-0.5}q''^{0.67}$
Shah [3]	$h_{tp} = Maximum(\varphi(Bo,Co))h_{D-B,l}$

$$\varphi(Bo,Co) = \left\{ \varphi_{cb} = \frac{1.8}{N^{0.8}}, \left\{ \begin{matrix} N > 1.0 \\ \varphi_{cb} = 230Bo^{0.5}, Bo > 3*10^{-5} \\ \varphi_{cb} = 1 + 46Bo^{0.5}, Bo < 3*10^{-5} \end{matrix} \right\} \right\}$$

$$\varphi(Bo,Co) = \left\{ \begin{matrix} 0{,}1 < N \leq 1.0 \\ \varphi_{cb} = \frac{1.8}{N^{0.8}}, \{\varphi_{cb} = FBo^{0.5}exp^{2{,}74N^{-0.1}}\} \end{matrix} \right\}$$

$$\varphi(Bo,Co) = \left\{ \begin{matrix} N \leq 1.0 \\ \varphi_{cb} = \frac{1.8}{N^{0.8}}, \{\varphi_{cb} = FBo^{0.5}exp^{2{,}47N^{-0.15}}\} \end{matrix} \right\}$$

$$F = \left\{ \begin{matrix} 14.7, Bo \geq 11*10^{-4} \\ 15.43, Bo < 11*10^{-4} \end{matrix} \right\},$$

$$N = \left\{ \begin{matrix} Co, Fr_l \geq 0.04 \\ 0.38Fr_l^{-0.3}Co, Fr_l < 0.04 \end{matrix} \right\}$$

Choi et al. [2]

$$h_{tp} = Sh_{nbc} + Fh_{lo}$$

$$S = 469.1689\left(\Phi_f^2\right)^{-0.2093} Bo^{0.7402}$$

$$h_{nbc} = 55P_r^{0.12}(-0.4343lnP_r)^{-0.55}M^{-0.5}q''^{0.67}$$

$$F = 0.042\Phi_f^2 + 0.958$$

$$h_{lo} = 0.023\frac{k_f}{D}\left[\frac{G(1-x)D}{\mu_f}\right]^{0.8}\left(\frac{c_{pf}\mu_f}{k_f}\right)^{0.4}$$

$$\Phi_f^2 = 1 + \frac{C}{X} + \frac{1}{X^2}$$

$$X = \left(\frac{\mu_f}{\mu_g}\right)^{1/8}\left(\frac{1-x}{x}\right)^{7/8}\left(\frac{\rho_g}{\rho_f}\right)^{1/2}$$

$$C_{tt} = 20, C_{vt} = 12, C_{tv} = 10, C_{vv} = 5$$

Tran et al. [14]

$$h_{tp} = 840000(Bo^2We_l)^{0.3}\left(\frac{\rho_f}{\rho_g}\right)^{-0.4}$$

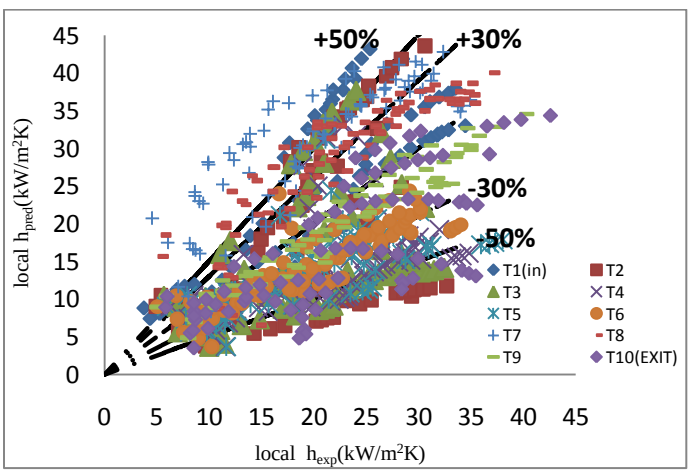

Figure 12. Comparison of local heat transfer coefficient with Shah [3] correlation

Tran et al. [14] correlation's predicted heat transfer coefficients are plotted versus local experimental heat transfer coefficient in Figure 14. This correlation fails to predict the data and under predicts the data with MAD of 86%.

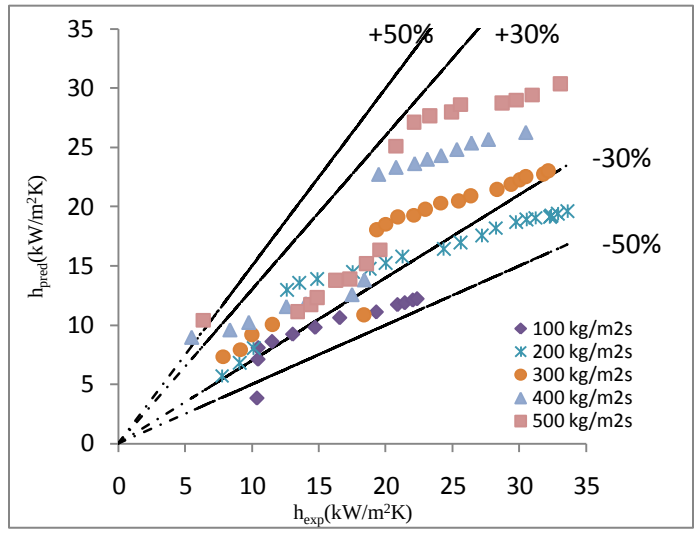

Figure 13. Comparison of average heat transfer coefficient with Shah [3] correlation.

An interesting thing to note is that Tran et al. [14] correlation predicts the data in smooth pattern . This shows that this correlation is capturing the changing phenomena in two phase flow with the increase of heat flux and mass flux. If this correlation is adjusted by some factor then this correlation can be a good predicting method for mini and micro channels.

Figure 15 shows the comparison of Choi et al. [2] superposition model correlation with average heat transfer coefficient. This correlation is able to predict our experimental data in smooth pattern and under predict the data with MAD of 47%. In this case also the local values are quite scattered which are presented in Figure 16.

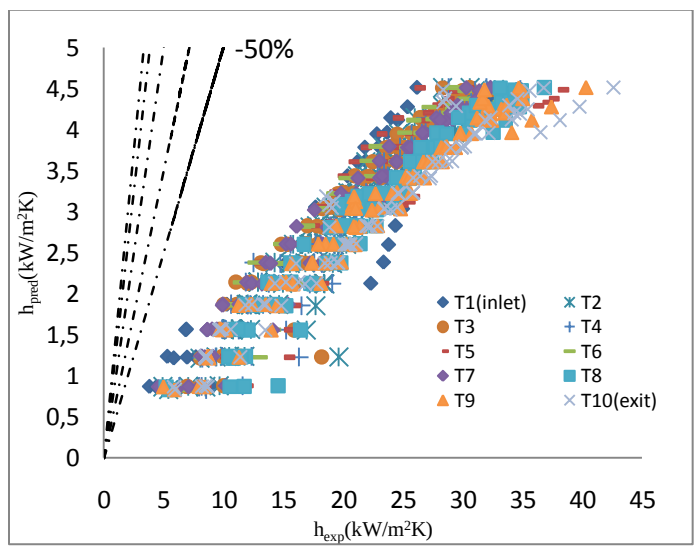

Figure 14. Comparison of local heat transfer coefficient with Tran et al. [14] correlation

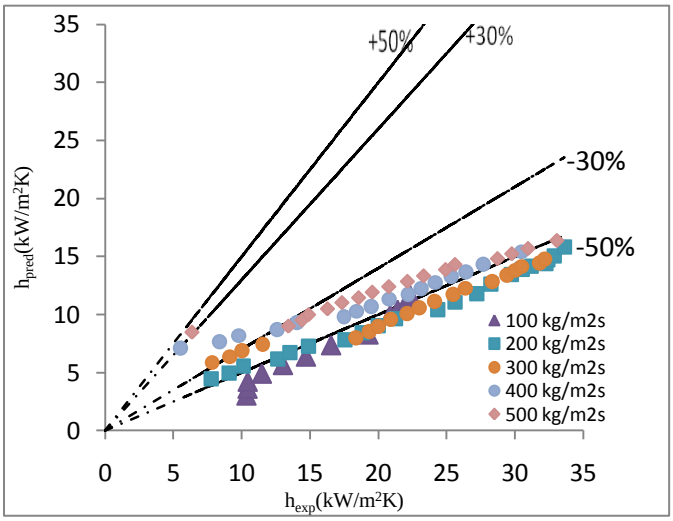

Figure 15. Comparison of average heat transfer coefficient with Choi et al. [2] correlation

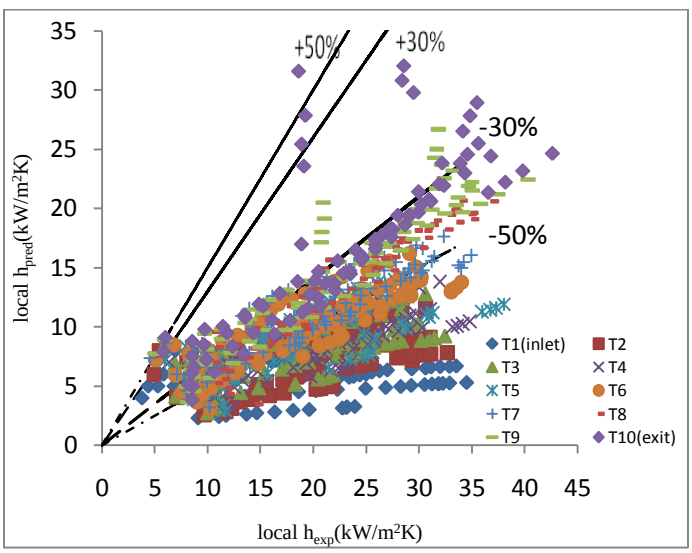

Figure 16. Comparison of local heat transfer coefficient with Choi et al. [2] correlation

CONCLUSION

Experiments have been performed to investigate heat transfer in a circular vertical mini channel made of stainless steel (AISI 316) with internal diameter of 1.70 mm and a uniformly heated length of 245 mm using ammonia as working fluid. To our best knowledge, this is the first time heat transfer tests with ammonia in minichannels have been reported.

The experimental results show that in general heat transfer coefficient depends on heat flux and is less dependent of mass flux and vapor quality. At higher heat fluxes, the influence of convective boiling is observed for higher mass fluxes. There is an indication that higher vapor fractions can be reached without dry out with ammonia compared to HFC refrigerants.

The pool boiling correlation of Cooper [1] predict the average heat transfer coefficients along the test section well, with MAD of 8.5 %.

The correlation by Choi et al. [2] under predicts almost all average heat transfer coefficients by about 47%. The local values, however, are much more scattered.

Shah correlation [3] predict the average values quite well with a MAD of 22.5%. Also in this case, the local data points are much more scattered.

Tran et al. [14] is found to under predict our experimental data by a factor of 0.15. However, even the local data points are predicted within a narrow band by about the same factor. The large under prediction by this correlation is unexpected as previous results with HFC refrigerants show good predictions.

NOMENCLATURE

A	heat transfer area (m^2)
C_p	specific heat (J/kg K)
D	diameter (m)
G	mass flux (kg/m2s)
h	heat transfer coefficient (W/m^2K)
I	current (A)
i_{fg}	latent heat of vaporization (J/kg)
k	thermal conductivity (W/mK)
L	length(m)
MAD	mean absolute deviation, $=1/N\ \Sigma\lvert Xpred - Xexp\rvert/Xexp$ (%)
$\dot{m}$	mass flow of refrigerant (kg/s)
P	pressure (bar)
Q	power (W)
q''	heat flux (W/m^2)
T	temperature (ºC)
V	voltage (V)
x_{th}	thermodynamic vapor quality (-)
z	axial position (m)

Subscripts

exp	experimental
h	heated
i	inside
in	inlet
f	liquid
g	gas
o	outside
pred	predicted
r	reduced
sat	saturation

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