

**FLOW BOILING HEAT TRANSFER CHARACTERISTICS OF R600A AND R290 IN
VERTICAL MINI-CHANNELS**

Z. Anwar¹, I. Ahmed¹, F. Noor¹, Z. Ali*², S. Imran¹, S. H. Siyal³, T. Sultan⁴

¹Mechanical Engineering Department, University of Engineering and
Technology Lahore, Pakistan

²Mining Engineering Department, University of Engineering and Technology
Lahore, Pakistan

³Energy and Environment Department, Quaid e Awam University of Engineering
Sciences and Technology, Nawabshah.

⁴Chemical Engineering Department, COMSATS Institute of Engineering and
Technology Lahore, Pakistan

Abstract

High heat dissipation from compact modern electronic devices demands smart cooling solutions to ensure proper functionality. Two phase heat transfer has the capability to cope with high heating/cooling demands with better temperature uniformity along the chip. Refrigerant related environmental concerns (ozone depletion, global warming etc.) instigated recent surge for new alternatives. Natural refrigerants are regaining their fame, however reliable experimental studies under wide operating conditions are required for designing novel practical devices. Keeping in view the current demands, this study is focused on the thermal performance of two natural hydrocarbon refrigerants (Isobutane R600a, and Propane R290) in uniformly heated small vertical stainless steel tubes. The heat transfer performance of natural refrigerants is also compared with R134a under similar experimental conditions. The experiments were carried out under wide operating conditions and continued till completion of dryout. The effects of operating parameters (heat flux, mass flux, vapor quality, saturation temperature etc.) along with assessment of macro and micro-scale correlations are the very subjects of this study.

Keywords: Mini-channel, Heat Flux, Isobutane, Propane, Correlation

*Corresponding author E-mail: zulfi@vt.edu

1. Introduction

While there is no general consensus on the definition of the micro channel, researchers do agree that there is a threshold limit below which the knowledge gained from conventional scale may not be well extrapolated. Kandlikar and Grande [1] named "mini-channel" for channels with $d < 3$ mm whereas Kew and Cornwell's [2] criteria is based on the confinement of a single vapor bubble and defines the "micro-channels" where $Co > 0.5$. In this study, "mini-channel" is used as the channel diameter is < 3 mm whereas Co in most of the cases were either closer to or higher than the threshold limit ($Co \sim 0.5$) of Kew and Cornwell [2].

Mini-channels offers enhanced heat transfer due to increased surface area per unit volume of the fluid in contact. Furthermore, ambitious goals of reduced charge and material inventories can also be materialized [4-6]. Coupled with two-phase heat transfer applications they can cope with high heat fluxes over fairly small temperature lifts. Generally the heat transfer coefficient increases with the increase of applied heat flux, which makes this arrangement a good solution for controlling the local hot spots.

Flow boiling heat transfer and the pressure drop characteristics of R600a within a horizontal mini-channel (2.60 mm in diameter and 185 mm heated length) has been reported by Copetti et al. [7]. In the low vapor quality region ($x < 0.4$), the heat transfer coefficients (HTCs) increased with the increase of heat flux and were not affected by the vapor quality. This study also reports increase in HTCs with the increase of mass flux. Their experimental data set is satisfactorily predicted by correlation from Kandlikar and Balasubramanian [3] with a modified value for fluid surface parameter.

Anwar et al. [6] reported experimental results on flow boiling heat transfer and dryout characteristics of R152a with a vertical mini-channel (1.60 mm in diameter and 245 mm heated length). Experimental results were reported for 100-500 kg/m²s, for two saturation pressures

corresponding to 27 and 32 °C saturation temperature. They reported a strong dependence of the heat transfer performance on the heat flux with insignificant contribution of the mass flux and the quality of the vapor.

Evaporative heat transfer using ammonia in small vertical tubes (1.224 and 1.70 mm in diameter and 245 mm heated length) has been reported by Maqbool et al. [8]. Their results were collected at 23, 33 and 43 °C saturation temperature with mass fluxes in the range of 100 to 500 kg/m²s. Nucleate boiling appeared as a dominant heat transfer mechanism in large tube (1.70 mm) where HTC's were mainly controlled by the heat flux. Convective boiling dominance (strong effect of mass flux and vapor quality) was reported with the small tube (d=1.20mm).

Forced convective heat transfer of pure refrigerants and their mixtures has been reported by Shin et al. [9]. The experiments were carried out using a horizontal 7.7 mm stainless steel tube with five pure refrigerants (R22, R32, R134a, R600a and R290) and their mixtures in various proportions. For Isobutane, the mass flux was varied from 265 to 583 kg/m²s at the saturation temperature of 12 °C. At low vapor quality a strong hold of heat flux on heat transfer was reported, thereafter at higher quality regions strong convective contributions were reported in the same study.

Ali et al. [5] reported experimental findings of flow boiling heat transfer of R134a in a uniformly heated vertical tube (1.70 mm in diameter and 220 mm heated length). The experiments were carried out with 50-600 kg/m²s mass flux at 27 and 32 °C saturation temperatures. Their heat transfer results were strongly controlled by the applied heat flux.

Enormous experimental and numerical studies has been reported in the literature from last two decades, however, the intrinsic mechanisms are not fully understood yet. This is clearly reflected

from different trends reported from various laboratories and from the absence of any mechanistic prediction model. Natural refrigerants are free from chlorine, fluorine and have good environmental footprints (Zero ODP and negligible GWP). Hydrocarbons have good transport properties and good compatibility with currently used components and materials. However, these materials are flammable in nature and require additional practical measures (hermetic components, reduced charge etc). Generally, the hydrocarbons have high heat of vaporization, this favors in reducing the amount of charge for a given heating/cooling duty. The flow boiling heat transfer characteristics of hydrocarbon based refrigerants have been reported by limited investigators and only a few studies are focused on the mini/micro-scale level.

In this study, the results obtained from the previous experimentations which are independently focused on flow boiling heat transfer of R290, R600a and R134a in vertical mini-channels are combined [10-12]. It should be noted that all the experimentations were carried out in the same laboratory using the same instrumentation set with two test sections (see **Table 1** for details).

Main objectives of this study are to explore the parametric effects on the heat transfer and to compare the heat transfer performance of hydrocarbons with widely used HFC refrigerant (R134a). The predictions of macro and micro scale correlations are also assessed.

2. Experimental Setup

The experimental setup is schematically shown in **Figure 1**. This is a closed refrigerant loop with heating and cooling provisions. All the experiments were carried under vertical upward flow direction whereas the test section was heated by the Joules effect.

A variable speed gear pump (Ismatec MCP-Z) was used for fluid circulation through the loop. A Coriolis type mass flow meter was used to record the mass flow rate. A preheater coil (coil wrapped on the outer periphery of the tube and heated by dc electricity) was used to adjust the

inlet conditions. A $2\mu\text{m}$ filter was used before the inlet of test-section to prevent its clogging by any small particles. Uniformly spaced t-type thermocouples were fixed on the outer periphery of the tube for recording the wall temperature whereas insertion type counterparts were used for recording the bulk flow temperature at inlet and outlet of the test section. **Figure 2** shows test section before and after fixation of the thermocouples. Thermally conductive epoxy from Omega Inc. was used for attachment of thermocouples. The system pressure was measured with an absolute pressure sensor (Druck PDCR 4060, 0-20 bar) before the inlet of test section whereas differential type counterpart (Druck PTX 5072) was used for measuring the pressure drop across the test section. A water cooled plate type heat exchanger was used as a condenser. Similar sized (test section) glass tubes were used at the inlet and outlet of the test section to electrically isolate it from other components in the test rig.

A data-logger connected with a computer and HP Agilent VEE was used for the data acquisition. For all experiments, the data was recorded for 3 to 5 minutes after achieving the steady state conditions. The average values were used for later calculations. The refrigerant property data was calculated by REFROP 9.

Table 1. Summary of test matrix

Refrigerant	Test section l/d [mm]	t_{sat} [°C]	Mass flux [kg/m²s]	Quality change [-]
R134a	245/1.60	27, 32	100-500	Till dryout
R600a	245/1.60	27, 32	50-350	Till dryout
R290	245/1.70	23, 33, 43	100-500	Till dryout

The literature on pool boiling shows that the surface roughness has an effect on the bubble nucleation and this, in turn, can control the heat transfer performance. The inner surface of both test tubes was scanned with Talysurf from Taylor Hobson. Main parameters for surface

characterization are summarized in Table 2. The inner surface of the tube having $d=1.60$ mm was more rough than the inner surface of 1.70 mm tube, this is clearly reflected from higher average roughness, peak height and valley depth values.

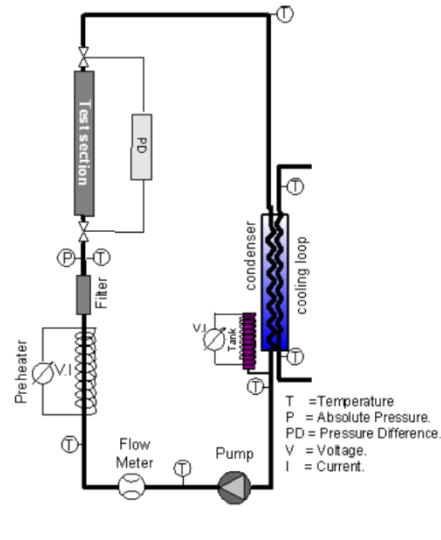


Figure 1. Schematic diagram of the experimental setup

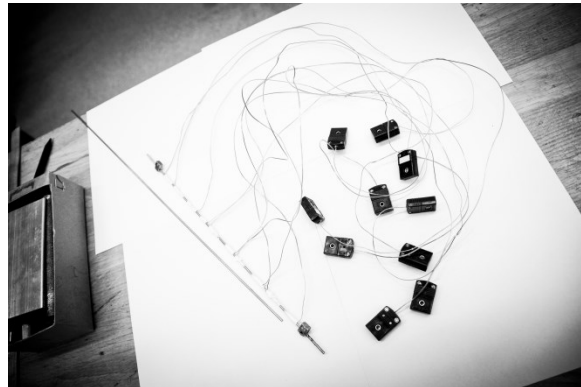


Figure 2. Test section before and after attaching thermocouples

Table 2. Roughness values

Test section	Ra ¹ [μm]	Rp ² [μm]	Rv ³ [μm]
245/1.60	0.95	2.69	6.44
245/1.70	0.21	0.80	0.73

3. Data reduction

Following procedure was used for calculation of various parameters of the interest.

The heat flux applied to the test section was calculated by,

$$q'' = \frac{V \cdot I}{A_h} \quad (1)$$

Where I and V are the applied current and the voltage respectively. A_h is the heated area,

$$A_h = \pi d_{in} l$$

The inner wall temperature was calculated from the outer wall temperature by using solution of the steady state one dimensional heat conduction equation (with heat generation) for cylinders,

$$t_{wall\ in} = t_{wall\ out} + \frac{Q}{4\pi\pi k} \left[\frac{\xi(1 - \ln\xi) - 1}{\xi - 1} \right] \quad (2)$$

Where $\xi = \frac{d_{out}^2}{d_{in}^2}$ and Q is the applied heat power.

Under sub-cooled conditions, the bulk temperature at any axial location was calculated with the information of inlet temperature and the added heat to the test section,

$$t_{fluidz} = t_{fluidin} + \frac{q'' \pi d_{in}}{mC_p} z \quad (3)$$

The local heat transfer coefficient at any location z was calculated by,

¹ Arithmetic mean value for the roughness

² Maximum peak height

³ Maximum valley depth

$$h_z = \frac{q''}{t_{\text{wall in}} - t_{\text{sat}}} \quad (4)$$

The quality/vapor fraction at any axial location is calculated by,

$$x_{\text{th}} = \frac{q'' \pi d_{\text{in}} (z - z_0)}{A_c G h_{\text{fg}}} \quad (5)$$

Where $z - z_0$ is the boiling length and h_{fg} is the latent heat of vaporization.

$$z_0 = \frac{m' C_p (t_{\text{sat}} - t_{\text{in}})}{q'' \pi d_{\text{in}}} \quad (6)$$

4. Results and Discussion

All the experiments were carried out with 1 to 1.5 °C degrees of sub-cooling. The inlet conditions were controlled with adjustment of power input to the pre-heater coil. In all cases, the heat flux was gradually increased in small steps and experiments were carried out till completion of the dryout.

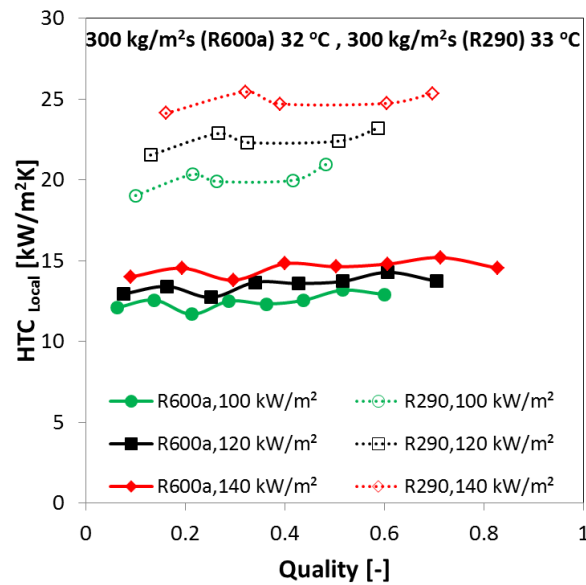


Figure 3. Local HTCs versus vapor quality for R600a and R290

The effect of heat flux on heat transfer is shown in Figure 3. The local HTCs for different heat fluxes with Isobutane and Propane (32 and 33 °C) are shown in this Figure. For both refrigerants,

the increase of heat transfer with the increase of heat flux can clearly be noticed for all vapor qualities. Similar trend was observed at other operating conditions (mass flux and vapor quality). The effect of mass flux on heat transfer is clarified in Figure 4. The local HTC's for various mass velocities under constant applied heat flux for each refrigerant are shown there. For both refrigerants duplication of identical results shows insignificant effect of mass flux on the heat transfer. Furthermore HTC's remained almost unaffected with variation of vapor quality.

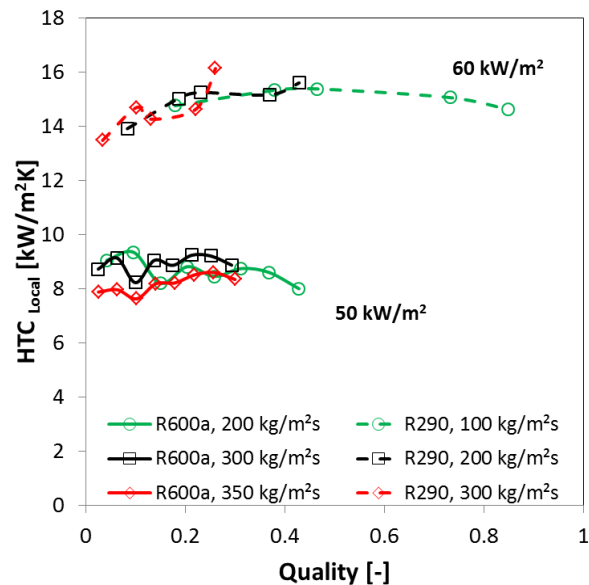


Figure 4. Variation of heat transfer coefficient with mass flux and vapor quality

The effect of system pressure on heat transfer performance is shown in **Figure 5**. The HTC's increased with increase in system pressure. The increased saturation temperature/pressure results in reduced density, surface tension and viscosity ratios (liquid/vapor). The reduced density ratio results in reduced vapor velocity which in turn results in lower suppression of nucleate boiling contribution and this probably enhances the heat transfer performance[13].

Thermal performance of natural flammable refrigerants was compared with conventionally used R134a (Figure 6). Under similar operating conditions (mass flux, saturation temperature and applied heat flux) identical heat transfer results were observed with propane whereas in case of isobutane much lower results were found. This can be due to different thermo-physical properties

(operating pressure, viscosity, surface tension as listed in **Table 3**) which affects the bubble nucleation activity and in turn controls the heat transfer performance. Isobutane and Propane are characterized with high latent heat of vaporization. This favors in reducing the required charge amount for a given application and hence reduces the explosion related hazards.

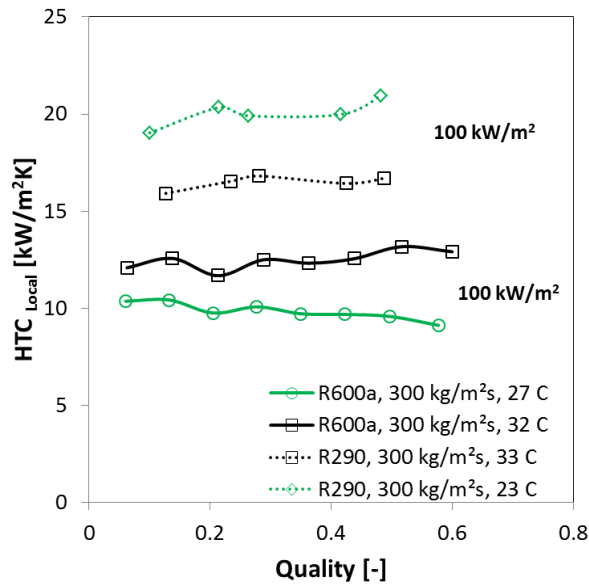


Figure 5. Local HTCs versus vapor quality for different saturation temperatures

Table 3. Thermo-physical properties of refrigerants at 27 °C saturation temperature

	P [bar]	h_{lg} [kJ/kg]	ρ_l/ ρ_g	k_l/k_g	μ_l/μ_g	σ [*10⁻³ N/m]
R600a	3.72	326.71	56.8	5.19	19.6	9.875
R134a	7.06	175.94	34.9	5.72	16.1	7.814
R290	10.01	332.18	22.5	4.81	11.3	6.744

Trends observed in this study and the dependency of the heat transfer on the heat flux and the system pressure with insignificant effect of the vapor quality and the mass velocity has been explained in the literature with dominance of nucleate boiling mechanism[4, 7, 14, 15]. However, based on the previous visualization studies from our group[16, 17], nucleation only occurs in the

close vicinity of inlet of the test section and in the lateral segment bubble's radial growth is restricted by the channel confinement. As bubble elongates in lateral direction, there appears a thin liquid film on the wall of channels. It is the evaporation of the thin liquid film in mini/micro-channels that has resemblance with the nucleate boiling.

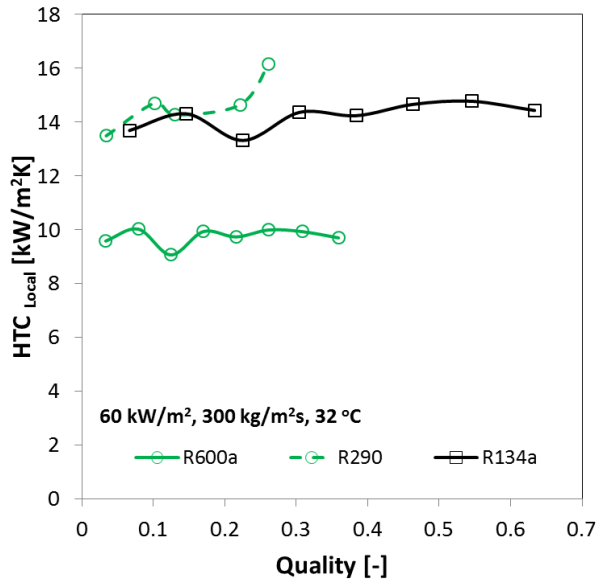


Figure 6. Local HTCs versus vapor quality for three refrigerants

5. Comparison with correlations

The experimental results with all three refrigerants are compared with the predictions of the renowned macro and micro-scale correlations from the literature. This comparison is based on the Mean Bias Error (MBE) and the percentage of data within $\pm 30\%$ of experimental results. MBE gives information about variance of predicted values from experimental ones and was calculated

$$\text{by, MBE} = \frac{1}{N} \left(\sum \frac{h_{\text{Predicted}} - h_{\text{Experimental}}}{h_{\text{Experimental}}} \right) * 100 \quad (7)$$

Mathematical formulation for the correlations and summary of comparison can be found in Table

4.

Some researchers [5, 6, 15, 16] have reported a satisfactory prediction for average flow boiling heat transfer coefficient in the mini/micro-channels with correlations originally developed for the pool boiling.

The average heat transfer coefficients were compared with the Cooper's pool boiling correlation [18]. This correlation showed (Figure 7a) reasonable prediction for R290 whereas coherent predictions were noticed with R134a where the data was highly under-predicted. For R600a scattered predictions with highly over-predicted values especially at higher vapor qualities are visible.

The average HTC's were compared with the correlation reported by Tran et al. [4]. This micro-scale correlation was originally developed from R12 based database where the nucleate boiling was reported as a dominant heat transfer mechanism. The Comparison of current database shows (Figure 7b) coherent prediction whereasthe data was highly under-predicted for R134 and R290 and scattered predictions were noticed for R600a.

Owhaib[17] proposed a correlation for prediction of saturated flow boiling heat transfer in the mini/micro-channels. It should be mentioned that Owhaib's data was collected from the same experimental setup with different test section and different test fluids. This correlation showed (Figure 7c) reasonable predictions within $\pm 30\%$ for R134a and R290 whereas the data was over predicted at low vapor quality and under predicted at higher vapor fractions. For isobutene, scattered predictions were noticed for the whole set of experimentation.

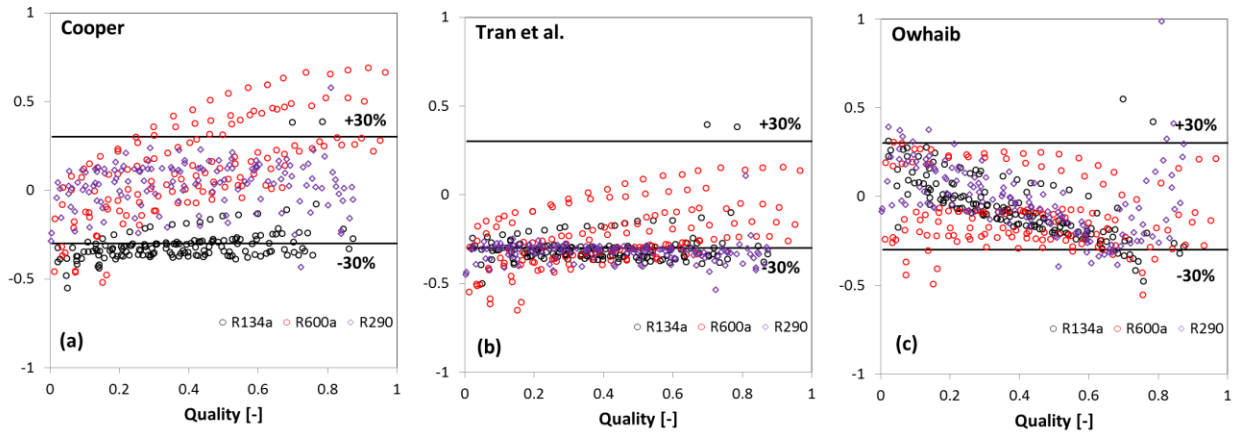


Figure 7. Comparison with correlations for average heat transfer coefficient

The local heat transfer coefficients were also compared with many macro-scale models. Better predictions were noticed using Gungor&Winterton [19]and Liu &Winterton [20] correlations as shown in Figure 8a and Figure 8b respectively. Both models considered contribution from nucleate boiling and convective boiling mechanisms with suppression and enhancement factors as shown in Table 4. The nucleate boiling contribution was calculated by the Cooper’s pool boiling correlation [18] whereas the Dittus-Boelter model was used for accounting the convective contribution. Gungor and Winterton[19]model coherently predicted the results for R134a and R290 whereas Liu and Winterton[20] model worked well for R290 and R600a.

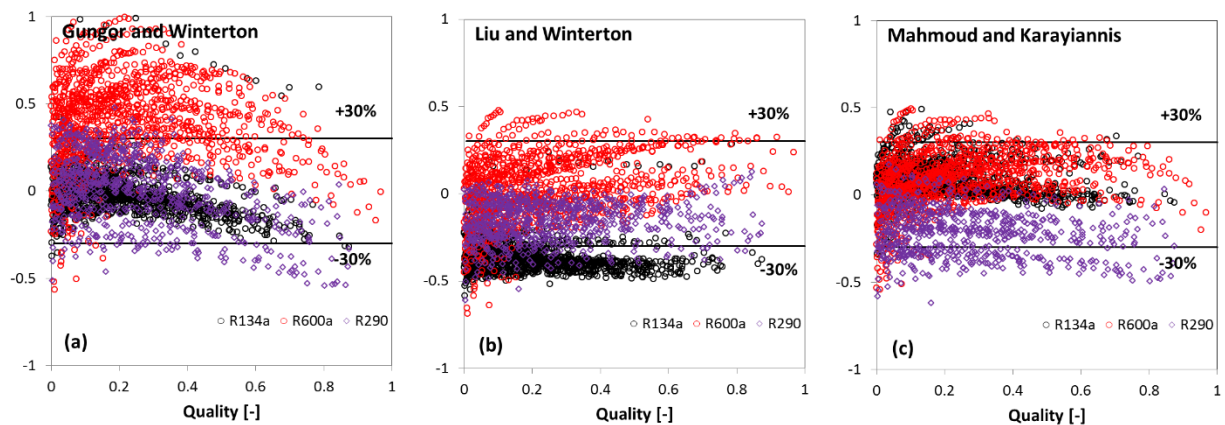


Figure 8. Comparison of local heat transfer coefficients with correlation from the literature

The local HTC's were compared with Mahmoud and Karayiannis model [21]. This micro-scale model was developed from their R134a based database. Comparison of current database showed (Figure 8c) a good prediction with correlation for all three test fluids. The test conditions of this study matches well with applicability range for this correlation furthermore similar trends (dominance of heat flux and system pressure on heat transfer) were reported by Mahmoud and Karayiannis[21]. These could be the possible reasons for good prediction noticed with correlation.

6. Conclusions

Experimental findings on saturated flow boiling heat transfer of two flammable refrigerants under wide operating conditions were reported in this article. Main findings are summarized below,

- Experimental results showed a strong hold of heat flux and system pressure on heat transfer whereas almost insignificant effect of vapor quality and mass flux was observed with all three refrigerants.
- Thermo-physical properties and surface characteristics affected the bubble nucleation and its growth. This in turn controlled the overall thermal performance.
- Compared with R134a, identical heat transfer results were observed with R290 whereas R600 showed significantly lower values.
- The pool boiling correlation predicted trends for experimental data. It was observed that satisfactory predictions were noticed with correlations developed for pool boiling or with those where observed trends (Bo for heat flux, Co for compactness) were properly addressed.
- The trends reported in this study have similarity with expectations from conventional pool boiling experiments. The nucleation is active close to the inlet of test section only, thereafter evaporation of thin liquid film was the dominant mechanism.

Nomenclature

A_c	Cross sectional area [m^2]
Bo	Boiling No [-]
Co	Confinement No [-]
C_p	Specific heat capacity [$J/kg.K$]
d	Inner diameter [m]
E	Enhancement factor [-]
G	Mass Flux [kg/m^2s]
I	Current [A]
k	Thermal conductivity [w/m^2K]
l_h	heated length [m]
M	Molecular weight [$kg/kmol$]
m'	Mass flow rate [kg/sec]
MBE	Mean Bias Error [%]
P_R	Reduced pressure [-]
Q	Applied electric power [W]
q''	Heat Flux [W/m^2]
Re	Reynolds No [-]
S	Suppression factor [-]
t_{sat}	Saturation temperature [$^{\circ}C$]
V	Voltage [V]
We	Weber No [-]
x	Vapor quality [-]
z	Axial position [m]

Greek letters

μ	viscosity [Pa-s]
-------	------------------

- ρ Density [kg/m³]
 σ Surface Tension [N/m]

Subscript

- g gas phase
 D-BI Dittus-Boelter
 Cooper Cooper's correlation
 l liquid phase

Table 4. Summary of comparison with correlation

Correlation	Expression	MBE	Percentage of data within $\pm 30\%$
Tran et al. [4]	$h = 8.4 * 10^{-5} (Bo \cdot We_l)^{0.3} \left(\frac{\rho_l}{\rho_g} \right)^{0.4}$	-30.22 -23.10 -33.05	32.67 (R134a) 63.76 (R600a) 19.20 (R290)
Cooper[7]	$h = 55P_R^{0.12} (-\log_{10} P_R)^{-0.55} M^{-0.5} q^{0.67}$	-31.14 13.21 19.20	29.35 (R134a) 70.28 (R600a) 98.67 (R290)
Owhaib[17]	$h = 400(Re_{lo} \cdot Bo)^{0.5} (1 - x_{exit})^{0.1} Co^{0.55} P_R^{1.34} \left(\frac{\rho_l}{\rho_g} \right)^{0.37} \frac{k_l}{d}$	-6.88 -10.24 -1.26	92.67 (R134a) 89.85 (R600a) 88.74 (R290)
Gungor and Winterton et al.[19]	$h = Eh_{D-BI} + Sh_{Cooper}$ $E = 1 + 24000Bo^{1.16} + 1.37 \left(\frac{1}{X_{tt}} \right)^{0.86}$	-1.52 39.73 -1.20	97.18 (R134a) 30.80 (R600a) 85.10 (R290)

	$S = (1 + 1.15 \times 10^{-6} E^2 Re_1^{1.17})^{-1}$		
Liu and Winterton[20]	$h_{tp}^2 = (Eh_{D-B})^2 + (Sh_{Cooper})^2$ $E = \left[1 + xPr \left\{ \frac{\rho_l}{\rho_g} - 1 \right\} \right]^{0.35}$ $S = \left[1 + 0.055E^{0.1} Re_{lo}^{0.16} \right]^{-1}$	-36.34 2.50 -15.16	15.14 (R134a) 88.40 (R600a) 84.63 (R290)
Mahmoud and Karayiannis[21]	$h = 3320 * \frac{Bo^{0.63} We_1^{0.2} Re_1^{0.11} k_l}{Co^{0.6} d}$	10.35 8.00 -21.78	97.68 (R134a) 96.92 (R600a) 69.50 (R290)

References

- [1] S. G. Kandlikar and W. J. Grande, "Evolution of Microchannel Flow Passages—Thermohydraulic Performance and Fabrication Technology," *Heat Transfer Engineering*, vol. 24, no. 1, pp. 3-17, 2003.
- [2] S. G. Kandlikar and P. Balasubramanian, "An extension of the flow boiling correlation to transition, laminar and deep laminar flows in minichannels and microchannels," *Heat Transfer Engineering*, vol. 25, pp. 86-93, 2004.
- [3] P. A. Kew and K. Cornwell, "Correlations for the prediction of boiling heat transfer in small diameter channels," *Applied Thermal Engineering*, pp. 705-715, 1997.
- [4] T. N. Tran, M. W. Wambsganss and D. M. France, "Small circular and rectangular channel boiling with two refrigerants," *International Journal of Multiphase flow*, pp. 485-498, 1996.
- [5] R. Ali, B. Palm and M. H. Maqbool, "Flow Boiling Heat Transfer Characteristics of a Minichannel up to Dryout Condition," *Journal of Heat Transfer*, vol. 133, pp. 081501-10, 2011.
- [6] Z. Anwar, B. Palm and R. Khodabandeh, "Flow boiling heat transfer and dryout characteristics of R152a in a vertical minichannel," *Experimental Thermal and Fluid Science*, vol. 53, pp. 207-217, 2014.
- [7] J. B. Copetti, M. H. Macagnan and F. Zinani, "Experimental study on R600a flow boiling in 2,6 mm tube," *International Journal of Refrigeration*, vol. 36, pp. 325-334, 2013.
- [8] M. H. Maqbool, B. Palm and R. Khodabandeh, "Boiling heat transfer of ammonia in vertical smooth mini channels: Experimental results and predictions," *International Journal of thermal sciences*, pp. 1-9, 2011.
- [9] J. Y. Shin, M. S. Kim and S. T. Ro, "Experimental study on forced convective boiling heat transfer of pure refrigerants and refrigerant mixtures in a horizontal tube," *International Journal of Refrigeration*, vol. 20, no. 4, pp. 267-275, 1997.
- [10] Z. Anwar, B. Palm and R. Khodabandeh, "Flow boiling of R600a in a uniformly heated vertical minichannel," in *13TH UK Heat Transfer Conference*, London, 2013.
- [11] M. H. Maqbool, B. Palm and K. Rahmatollah, "Investigation of two phase heat transfer and pressure drop of propane in a vertical circular minichannel," *Experimental Thermal and Fluid Science*, vol. 46, pp. 120-130, 2103.

- [12] Z. Anwar, "Evaporative heat transfer with R134a in a vertical minichannel," *Pakistan Journal of Engineering and Applied Sciences*, vol. 13, pp. 101-109, 2013.
- [13] C. B. Tibirica and G. Ribatski, "Flow boiling heat transfer of R134a and R245fa in a 2.3 mm tube," *International Journal of Heat and Mass transfer*, pp. 2459-2468, 2010.
- [14] G. Lazarek and S. H. Black, "Evaproative heat transfer, pressure drop and critical heat flux in a small veratical tube with R-113," *International Journal of Heat and Mass transfer*, pp. 945-960, 1982.
- [15] S. S. Bertsch, E. A. Groll and S. V. Garimella, "A composite heat transfer correlation for saturated flow boiling in small channels," *International Journal of Heat and Mass transfer* , pp. 2110-2118, 2009.
- [16] C. M. Callizo, "Flow boiling heat transfer in single vertical channels of small diameters," KTH, Stockholm, 2010.
- [17] W. Owhaib, "Experimental heat transfer, pressure drop and flow visualization of R-134a in vertical mini/micro tubes," *Doctoral Theis in Energy Technology KTH*, Stockholm, 2007.
- [18] M. Cooper, "Saturated nucleate pool boiling, a simple correlation," in *Proceedings of the 1st UK National Heat Transfer conference*.
- [19] K. E. Gungor and R. Winterton, "A general correlation for flow boiling in tubes and annuli," *International Journal of Heat and Mass transfer*, pp. 351-358, 1986.
- [20] Z. Liu and R. H. S. Winterton, "A general correlation for saturated and subcooled flow boiling in tubes and annuli, based on a nucleate pool boiling equation," *International Journal of Heat and Mass transfer*, pp. 2759-2766, 1991.
- [21] M. M. Mahmoud and T. G. Karayiannis, "A statistical correlation for flow boiling heat in micro tubes," in *3RD European Conference on Microfluidics*, Heidelberg, 2012.