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FLOW BOILING HEAT TRANSFER OF PROPANE IN MPE TUBE

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Abstract. *The local heat transfer coefficient and flow patterns were studied experimentally during flow boiling of R-290 (propane) inside multiport extruded (MPE) mini channel tube made up of 7 channels with hydraulic diameter of 1.47 mm. The tests analyzed the effects of heat flux, mass velocity and vapor quality. The study were performed with heat fluxes from 5.3 to 20 kW/m^2, mass velocities varying from 35 to 170 kg/m^2 and vapor quality between 0.07 to 0.98. As result, heat transfer coefficients between 1-18 kW/m^{2}K were obtained. Five types of flow patterns were observed, with predominance of plug and slug and churn. The results unveil the significant effect of flow patterns on heat transfer characteristics.*

Keywords: *Flow boiling, Heat transfer, Propane, Multiport extruded tube, Flow patterns.*

1. INTRODUCTION

Due to the advancement of "miniaturization" in the technology sector coupled with the fact that companies have tried to increase the efficiency of their systems without major investments, compact heat exchangers have emerged as a solution to these problems. The advantage of working with mini and microchannel is the possibility of having higher heat transfer coefficients, which represents a great capacity to remove heat by refrigerant fluid for specific mass velocities and heat fluxes as well as have lower cost in manufacturing and operation for those types of heat exchangers (Kandlikar *et al.*, 2006).

Research related to flow boiling in reduced diameter channels is of interest in the design and development of automotive air conditioners, compact evaporators and condensers, extruded tubes, etc.

Compact heat exchangers are devices dimensioned for carrying out the heat exchange in small equipment such as cooling microprocessors, high-power laser, MEMS (Micro Mechanical Systems) applied to engineering, biomedical and genetic research, as well as other various applications. In addition to their small sizes and low cost, compact heat exchangers are highlighted for allowing the use of different refrigerants.

This study presents experimental results during flow boiling of R-290 in multiport extruded aluminum tubes (MPE) made up of 7 channels with hydraulic diameter of 1.47 mm. Both heat transfer and flow patterns will be analyzed for different heat flux, mass velocity and vapor quality conditions.

From the Montreal Protocol to reduce the use of refrigerants like HFCs and HCFCs it has been established for several countries, including Brazil, seeking constantly for new refrigerants with best results and with low rate of ozone depletion potential (ODP). In this context, hydrocarbons are highlighted because they are considered natural refrigerants, which do not affect the ozone layer, in addition to having a global warming potential (GWP) almost nil compared to synthetic refrigerants. An example of this category is the refrigerant R290 (propane) which also has advantages with respect to their thermophysical properties, low toxicity, high solubility with conventional lubricants and ester oils. So it is gaining importance in research in the field of refrigeration, especially in automotive air conditioners (Del Col *et al.*, 2014).

The characteristics of heat transfer and two-phase flow in multi minichannels have been studied with different refrigerants.

Kaew-On *et al.* (2011) performed tests with R-134a and observed a heat transfer coefficient of 11 kW/m^{2}K for heat fluxes between 34 and 36 kW/m^2 and 18 kW/m^{2}K for heat fluxes ranging from 63 to 65 kW/m^2 for MPE tubes with channel 1.1 mm hydraulic internal diameter.

Vakili-Farahani *et al.* (2013), studied flow boiling heat transfer of R245fa and R1234ze(E), in a MPE composed of 7 parallel rectangular channels (1.1 mm x 2.1 mm) with hydraulic diameter of 1.4 mm, for heat fluxes between 3 and 107 kW/m² and mass velocities ranging from 50 to 400 kg/m²s. As result, they observed that heat transfer coefficient were between 150 – 6000 W/m^{2}K.

Copetti *et al.* (2016) analyzed flow boiling hat transfer of R600a (isobutane) for heat fluxes between 5 and 30 kW/m² and mass velocities ranging from 50 to 200 kg/m²s. As result, the local heat transfer coefficients were found to increase with heat flux and mass velocity.

Park *et al.* (2009) performed tests in multi minichannels with hydraulic diameter of 0.89 mm, using CO_2 as refrigerant and heat conditions established with power between 10 and 90 W. In their tests it was also observed the increase of heat transfer coefficient with increasing the heat flux, mass velocity and vapor quality.

Costa-Patry *et al.* (2011) conducted tests with f R236fa and R245fa and analyzed heat transfer coefficients in 12.7mm silicon evaporator composed of 135 microchannels with 85 µm wide and 560 µm high channels separated by 46 µm wide fins. They used heat fluxes from 45 kW/m^2 to 190 kW/m^2, with mass velocities ranging from 500 kg/m^{2}s to 900 kg/m^{2}s. They observed that heat transfer coefficients for R236fa and R245fa were between 5–20 kW/m^{2}K.

Huai *et al.* (2004) conducted experiments of pressure drop and heat transfer flow boiling of CO2 in a multi-port extruded aluminum test section with 10 circular channels, and inner diameter of 1.31 mm. The measurements were performed for pressures ranging from 3.99 to 5.38 MPa, inlet temperatures from -3.08 to 16.96 °C, heat fluxes from 10.1 to 20.1 kW/m^2, mass velocities from 131.4 to 399.0 kg/m^{2}s, and vapor quality from 0.0 to 1.0. They observed that the mass velocity and the applied heat flux have significant effects on flow boiling heat transfer characteristics.

Besides studying the heat transfer coefficient, many studies relating to the flow patterns. The flow patterns study shows the great importance to understand how the heat transfer process works between fluid and tube.

In some research on flow boiling, the flow patterns maps are also focus of study to check the flow relation to the parameters studied. The analysis of flow patterns is an important tool, which allows us to view the regime transitions, fluid behavior under certain conditions, in addition to allowing the relationship to the mass flow, heat flow, vapor quality and other parameters (Bratland, 2010).

Charnay *et al*. (2015) conducted an experimental investigation for study the heat transfer characteristics of R-245fa during flow boiling in a 3.00 mm inner diameter tube at a saturation temperature ranging from 100 to 120 ºC. They built a flow patterns map, and observed that heat transfer coefficient increases with increasing vapor quality and/or mass velocity. As result, they found four types of flow patterns, as intermittent flow, annular flow, dry-out flow and mist flow.

Keepaiboon *et al.* (2015) analyzed flow patterns and heat transfer characteristics of R134a refrigerant during flow boiling in a single rectangular micro-channel with 0.68mm hydraulic diameter. In their experiment, heat transfer coefficients were studied at a heat flux range of 7.63–49.46 kW/m^2, mass flux range of 600 – 1,400 kg/m^{2}s, and saturation temperature range of 23–31 °C. They observed six different flow patterns occurring during boiling, as bubbly flow, bubbly-slug flow, slug flow, throat-annular flow, churn flow, and annular flow. As result, they found that heat flux, mass flux, and saturation temperature have significant influence on both change of flow pattern and heat transfer coefficient.

Thiangtham *et al*. (2016) conducted an experimental study of flow patterns and heat transfer characteristics during flow boiling of R-134a in a multi-microchannel heat sink made of a section with 27 parallel rectangular channels with a depth of 470 µm, a width of 382 µm, a length of 40 mm and a fin thickness of 416 µm. Mass velocities of 150, 400, and 600 kg/m^{2}s, heat fluxes range from 3 to 127 kW/m^2, and vapor quality from 0.05 to 0.92, were considered in their experiments. With the heat flux range of 40–120 kW/m^2, for low heat flux conditions, the heat transfer coefficient increases with increasing heat flux, and flow patterns were bubbly and slug. For higher heat flux conditions, heat transfer coefficient increases with increasing mass velocity, and consequently wavy annular and annular were the main patterns observed. At low vapor quality, the heat transfer coefficient slightly increased when heat flux increases, and for higher vapor quality the heat transfer coefficient increases with increasing mass velocity.

2. EXPERIMENTAL SET UP AND PROCEDURE

2.1 Experimental apparatus

An experimental facility was developed to investigate the flow boiling and pressure drop in a multiport mini channel and is schematically shown in Figure 1. The system consists of a loop that provides controlled mass velocity, and designed to test different fluids under a wide range of flow conditions. The main part of the circuit has a pre-heater (*PH*), a test section (*TS*) and a visualization section (*VS*). The secondary part consists of a condenser, a refrigerant reservoir, a liquid refrigerant vessel, a magnetic driven micro gear pump and a subcooler. The condenser and the subcooler have independent circuits and each uses an ethylene-glycol/water solution as the secondary refrigerant, and the temperatures are controlled by a thermostatic bath. This set up controls the refrigerant saturation pressure. The refrigerant reservoir, connected to the main circuit of the bench, operates as a pressure regulator, maintaining stable conditions during the experiments, and a liquid refrigerant vessel maintains constant static pressure at the pump suction. This procedure assures that the pump works uniformly and under immersion, avoiding cavitation.

The pre-heater establishes the experimental conditions entering the *TS* just downstream. It consists of a horizontal stainless steel tube with 4.4 mm ID, 465 mm length and absolute internal roughness of 1.03 µm, uniformly heated by direct application of an electrical current in the tube wall (Joule effect), the intensity of which is controlled by the DC power supply. The refrigerant enters the *PH* as subcooled liquid and reaches saturation condition to the exit. This condition sets vapor quality in the inlet of *TS*, and varies according to the heat flux imposed on the *PH*.

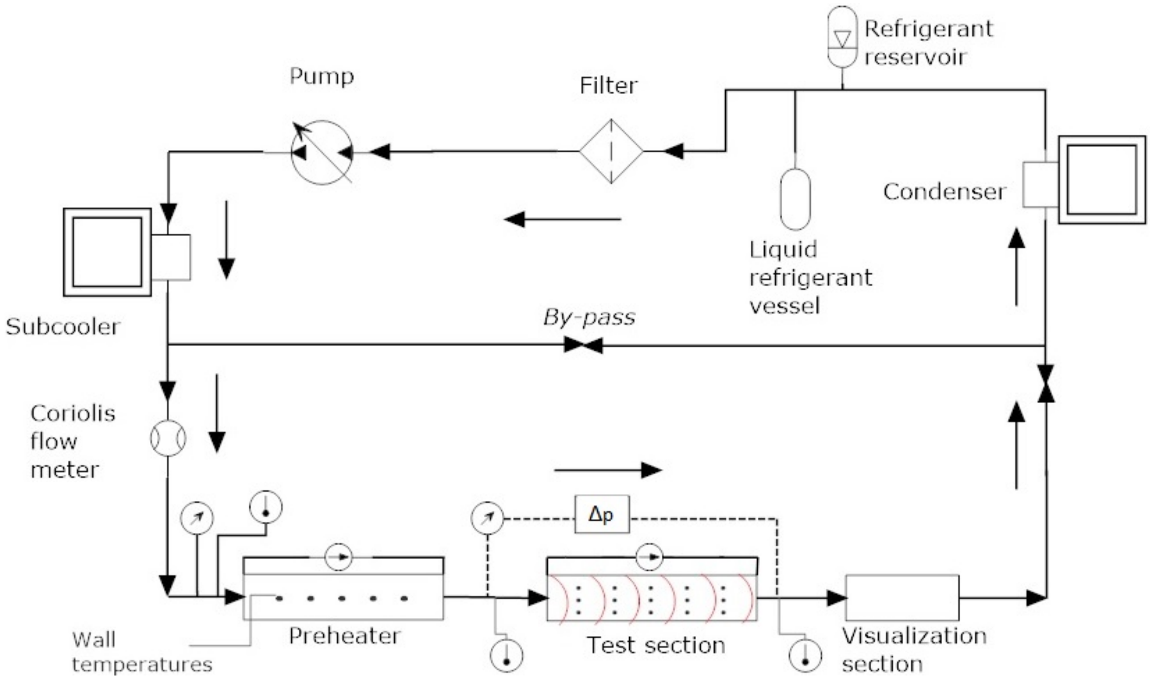


Figure 1. Experimental apparatus

The test section consists of an extruded aluminium flat tube with multi mini channels with 300 mm length. The external dimensions of tube are 1.97 mm height and 16.48 mm width with 7 parallel rectangular channels, 5 rectangular channels with 1.83 mm x 1.23 mm and 2 external channels with semi-circular geometry with radius equal to 0.63 mm. The average hydraulic diameter of channels is 1.47 mm. The cross-sectional area of all channels is 15.44 mm² and the internal perimeter is 44.27 mm. The *TS* is heated by an electrical tape resistance (11.7 Ω/m) wrapped around its surface to guarantee a uniform heat flux to the refrigerant, which intensity is controlled by a voltage regulator (Variac). The electrical resistance is insulated from tube with a kapton conductive tape and the good thermal contact between the wire and the surface is guarantee by a bakelite plate. The absolute internal roughness (Ra) of the MPE measured is 0.295 µm. Both the *PH* and *TS* are thermally insulated. More details about the experimental apparatus can be seen in Copetti *et al.* (2016).

2.2 Test conditions with R-290

The tests were performed for different conditions of heat flux, mass velocity and saturation temperature of 17 °C, according to Tab. 1.

Table 1. Experimental conditions.

Heat flux in TS, *q''* (kW/m²)	5.3,10, 16, 20
Mass velocity, *G* (kg/m²s)	35, 55, 70, 100, 170
Average saturation pressure, *p* (kPa)	772
Average saturation temperature , T_{sat} (ºC)	17

2.3 Data reduction

The heat flux, *q''*, is calculated considering the electrical power supplied, the internal surface area, A_{is}, and the number of channels for heated length, *N*, according to Eq. (1).

$$q''=\frac{q}{NA_{is}} \quad (1)$$

The internal wall temperature was calculated assuming radial conduction through the wall as given by Eq. (2)

$$T_{iw} = T_{ew} - \frac{qe}{kA_{es}} \tag{2}$$

where T_{ew} is the external wall temperature, e is the wall thickness of the channel, k is the thermal conductivity of the aluminum and A_{es} represents the external surface area.

For each axial location z along the test tube, the external wall temperature was assumed to be the average of three measured top wall temperatures, given by Eq. (3).

$$T_{ew} = \frac{T_{ew,1} + T_{ew,2} + T_{ew,3}}{3} \tag{3}$$

Consequently, the local heat transfer coefficient, h_z, is determined using Eq. (4), as follows:

$$h_z = \frac{q''}{(T_{wi,z} - T_{sat,z})} \tag{4}$$

In this case, the local saturation temperature, $T_{sat,z}$ is obtained as a function of local saturation pressure, considering the pressure drop as being linear along the tube.

The vapor quality in test section inlet was calculated from energy balance in *PH*, using Eqs. (5) and (6), as follows:

$$i_{i,ST} = i_{o,PH} = \frac{q_{PH}}{\dot{m}} + i_{i,PH} \tag{5}$$

$$x_{i,TS} = \frac{i_{i,TS} - i_l}{i_{lv}} \tag{6}$$

3. RESULTS

Table 2 presents the maximum relative uncertainties of measured and calculated parameters for some experimental conditions. The maximum uncertainty for the heat transfer coefficient was 9.1% in the operating conditions of highest heat flux.

Table 2. Uncertainty of parameters.

Temperature	0.3 ºC
Heat Flux	0.1%
Heat transfer coefficient	9.1%
Local pressure	1.3%FS
Mass velocity	0.17%FS

3.1 Heat transfer

3.1.1 Effect of heat flux

In all cases observed, it appears that the heat transfer coefficient increases with increasing heat flux and vapor quality. Similar results for different refrigerants were observed by Kaew-On *et al.* (2011), Vakili-Farahani *et al*. (2013), Thiangtham *et al*. (2016), Charnay *et al*. (2015), Szczukiewicz et al. (2013), Costa-Patry *et al.* (2011) and Copetti *et al.* (2016).

For lower mass velocity, G=35 kg/m²s, it was observed that the strong dependence of heat flux causing a variation of heat transfer coefficient from 2 to 8 kW/m²K with vapor quality varying from 0.07 to 0.84.

For mass velocity of G=55 kg/m²s, the variation of the heat transfer coefficient was from 1 to 13 kW/m²K, and vapor quality from 0.14 to 0.98, according Figure 2. In this case, it perceives an influence of all heat flux on the heat transfer coefficient until vapor quality of 0.8, approximately. For the vapor quality between 0.8 and 1 is observed the effect of the heat flux of 16 kW/m^2.

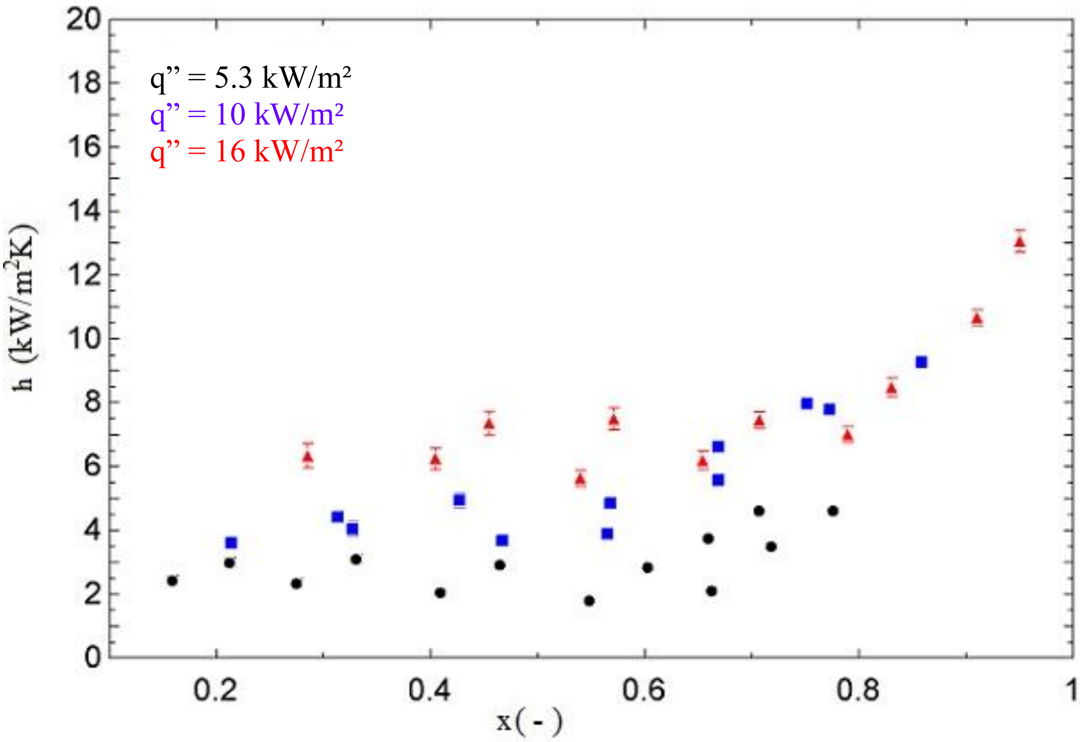


Figure 2. Effect of heat flux on the local heat transfer coefficient for *G*=55 kg/m²s.

Figure 3 shows de influence of heat flux for mass velocity G=70 kg/m²s, it is possible to verify that for higher G, the effect of heat flux on heat transfer coefficient vanish, for higher vapor qualities. The variation of heat transfer coefficient was in the range from 1 to 15 kW/m^{2}K with vapor quality varying from 0.14 to 0.91.

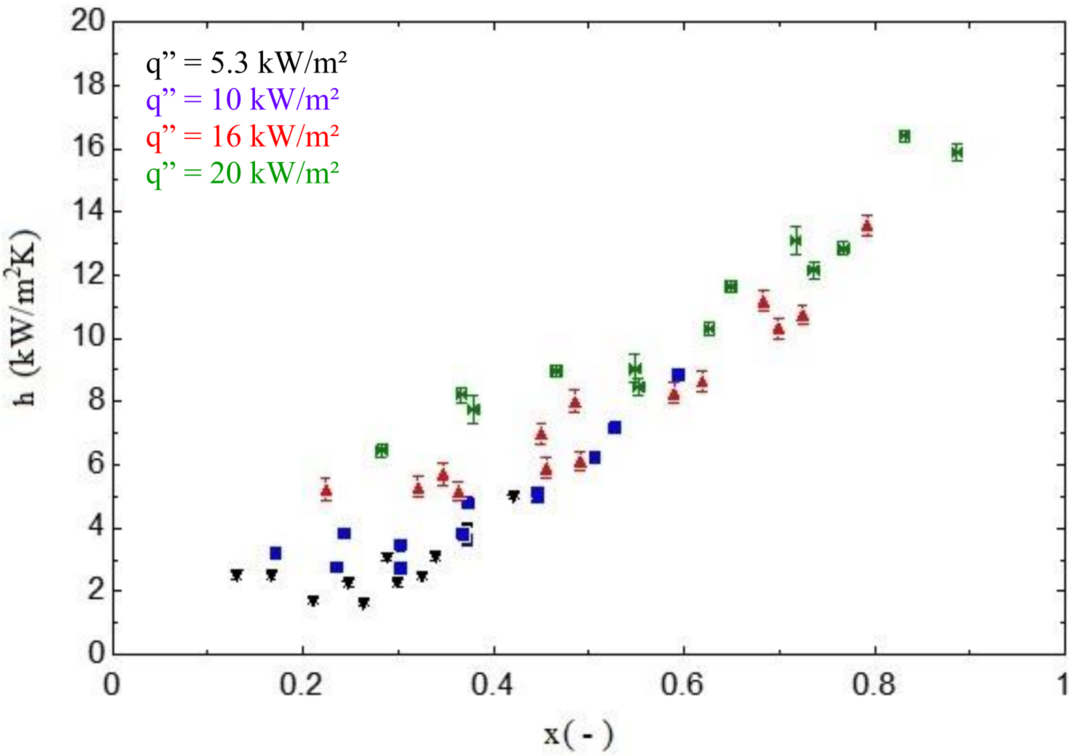


Figure 3. Effect of heat flux on the local heat transfer coefficient for *G*=70 kg/m²s.

Figure 3 presents the influence of heat flux on the local heat transfer coefficient for G = 70 kg/m²s. The results shows the influence of all heat flux on the heat transfer coefficient until vapor quality of 0.5, approximately. For the vapor quality between 0.5 and 1 is observed the effect of the heat flux of 16 and 20 kW/m^2.

For the mass velocities G=100 kg/m²s, a variation occurred of heat transfer coefficient from 1.5 to 18 kW/m^{2}K due to the heat flow, with vapor quality varying from 0.08 to 0.86.

And lastly, for the mass velocities G=170 kg/m²s, a variation of heat transfer coefficient was in the range from 1 to 16 kW/m^{2}K with vapor quality varying from 0.14 to 0.49.

3.1.2 Effect of mass velocity

Figures 4 and 5 show the effect of the mass velocity on the local heat transfer coefficient for q"= 10 kW/m^2 and q"= 16 kW/m^2.

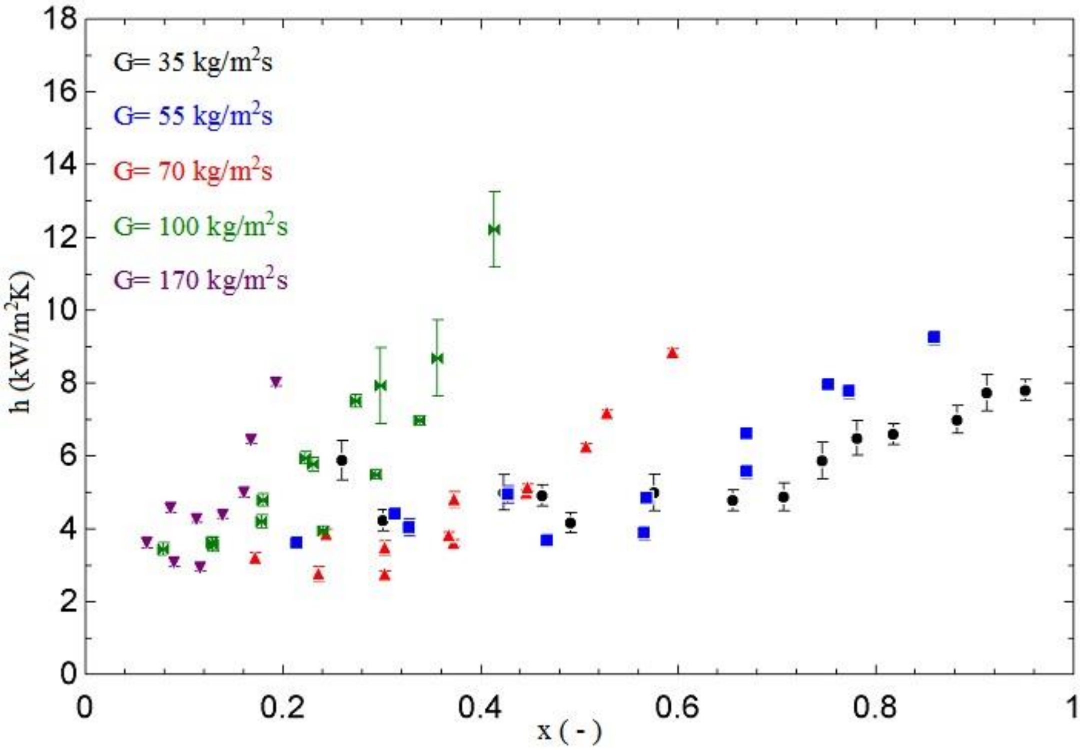


Figure 4. Effect of mass velocities on the local heat transfer coefficient for q''=10 kW/m².

First, the local heat transfer coefficient increases with G, but this effect is more evident as vapor quality increases and for higher heat flux. The same behavior was observed by Copetti *et al.* (2016) for the hydrocarbon isobutane, and by Vakili-Farahani *et al.* (2013) for R245fa and R1234ze(E).

Secondly, the enhancement of the heat transfer coefficient becomes minimal at the low qualities, where Dispersed bubbly regimes are identified, and the values converge approximately on the same magnitude attributed to the onset of boiling.

Thirdly, the effect of mass velocity is more seen in the heat flow of 16 kW/m² than 10 kW/m² where a greater variation is observed in the heat transfer coefficient.

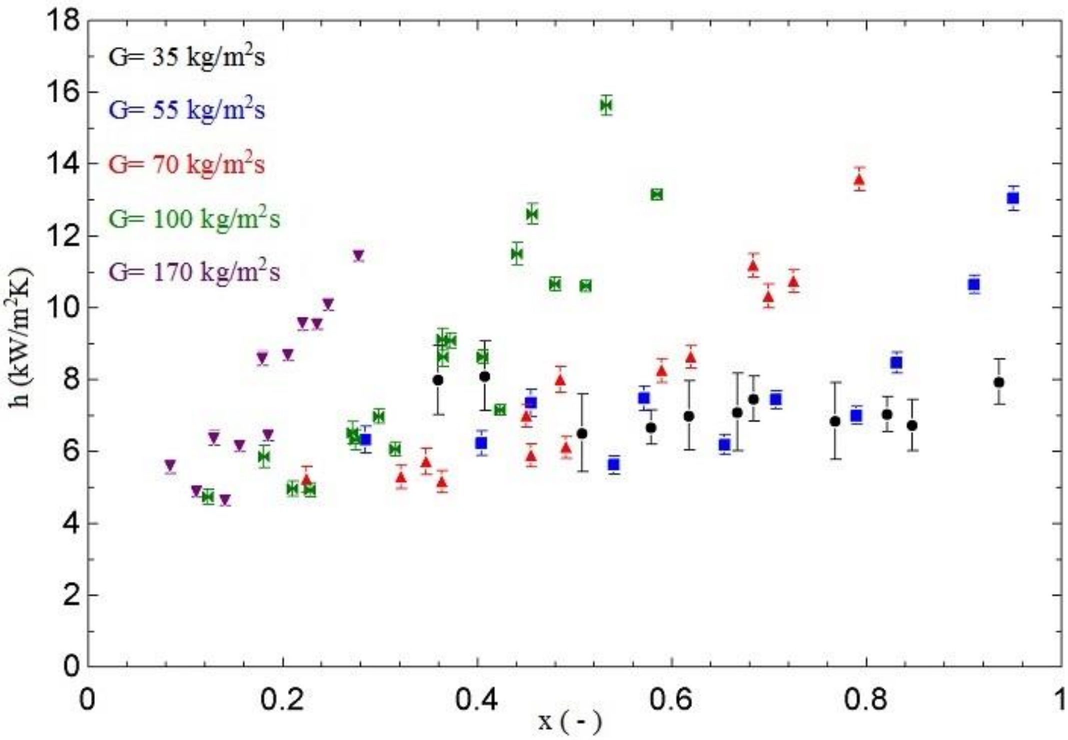
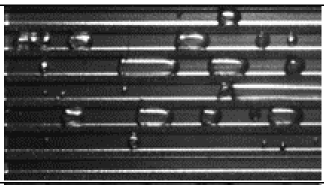
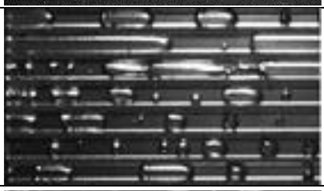
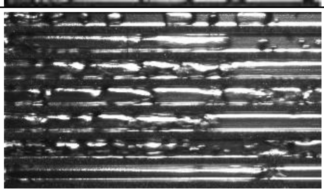


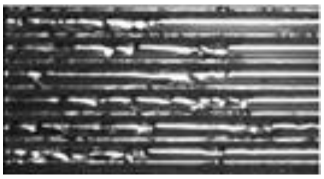
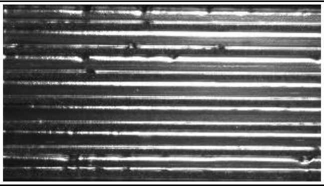
Figure 5. Effect of mass velocities on the local heat transfer coefficient for q''=16 kW/m².

3.2 Flow Patterns

For the analysis of flow patterns, images and videos were captured during the tests on the top face of VS. The flow patterns observed in this study are show in Tab. 3.

Table 3. Flow patterns observed in this study.

Image	q'' (kW/m^2)	$x_{o,TS}$ (-)	G (kg/m²s)	Flow pattern
	5.3	0.09	170	Dispersed bubbles
	10	0.43	70	Plug and slug
	5.3	0.41	100	Churn

	20	0.98	70	Wavy annular
	16	0.98	55	Annular

Five different types of flow patterns were observed in this study, the dispersed bubbles, plug e slug, churn, wavy annular and annular, in flow multiport extruded channels. The results show that the flow pattern changes with increasing the heat flux. This is possibly due to the increase of heat flux, which leads to the rise in incidence rate of bubbles in MPE tube.

The maldistribution of flow across all channels was observed in different conditions of tests. Thus, the patterns have a heterogeneous characteristic when the flow patterns of the central channels are compared with those channels of the edges. The differences among the flow patterns in the channels are probably due to non-uniform fluid distribution, caused by the bubble containment, and also the heat distribution in the TS.

The flow patterns observed in this study have similar characteristics to results presented by Thiangtham et al. (2016). Figure 6 shows the flow patterns of R-290 for the mass velocities of 35, 55, 70, 100 e 170 kg/m²s.

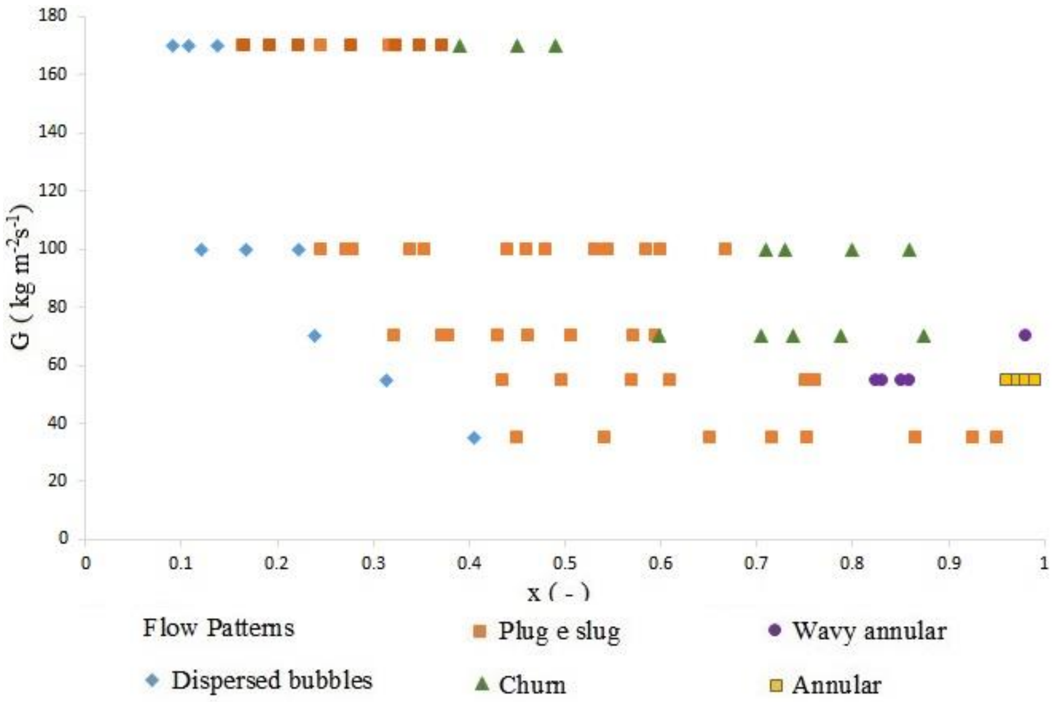


Figure 6. Flow patterns of propane R-290.

According the results of Fig. 6, for the operational conditions of the tests the patterns more frequently observed were the plug and slug and churn. The dispersed bubbles pattern was verified only at lower vapor quality, for all mass velocities. The wavy annular pattern was observed only for the mass velocities of G=55 and 70 kg/m²s, and annular pattern was observed for the G=55 kg/m²s, vapour quality equal 0.98 and heat flux of 16 kW/m^2.

4. CONCLUSIONS

Experimental results during flow boiling of R-290 in a horizontal multiport mini-channels tube, under variations in the mass velocity, heat flux and vapor quality, were presented. The behavior of the heat transfer coefficient was investigated for different conditions and the flow patterns were identified. Some observations from this study can be made:

- The local heat transfer coefficient increases with heat flux and vapor quality.
- Different flow patterns in parallel channels were observed, at the same conditions, probably associated to the maldistribution of the mass flow rate into the channels and transient thin-film evaporation mechanism with the entrainment of the liquid in the core flow (even at moderate vapor fractions);
- It is noted, that as the mass velocity increases, the vapor qualities tends to have lower values, for a given value of *h*.
- The heat flux have significant effect on the variation of flow patterns;
- In general, five flow patterns were more frequent: dispersed bubbles, plug and slug, churn, wavy annular and annular. Isolate bubble and plug patterns were observed only in specific conditions, as high mass velocity and low heat flux, for low vapor quality.

5. ACKNOWLEDGEMENTS

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