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Condensation Heat Transfer and Pressure Drop with Propane in a Minichannel

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ABSTRACT

The use of propane as a refrigerant is a good opportunity to develop environmentally friendly heat pump and air-conditioning equipment, since the direct effect on the anthropogenic global warming due to atmospheric emissions is almost completely avoided, while the indirect effect can be reduced by exploiting the favourable thermodynamic properties of these fluids.

Because of flammability, charge minimization is a major design objective for such equipment using hydrocarbons. From previous experience, it appears that the estimated charge of unitary air conditioners is expected to be mainly trapped in the heat exchangers. In this regard, microchannel technology appears to be a very good opportunity to minimize the charge without energy performance loss. This was demonstrated at the University of Padova in a heat pump with 100 kW heating capacity, using minichannel shell-and-tube heat exchangers within the European project Sherpa. Nevertheless, a very limited number of data of condensation of propane in minichannels is available to validate predicting correlations.

In this paper, the local heat transfer coefficient measured during condensation of propane in a 0.96 mm diameter circular minichannel is reported and compared versus available correlations. Tests are carried out on the experimental apparatus available at the Heat Transfer Lab of the University of Padova. During condensation tests, the heat is subtracted from the fluid by using cold water. The heat transfer coefficient is obtained through the measurement of the local heat flux and the saturation-to-wall temperature difference. Tests are carried out at mass fluxes ranging between 100 and 800 kg m⁻² s⁻¹.

Since the saturation temperature drop directly affects the heat transfer rate, also pressure drop during adiabatic two phase flow of propane is measured and compared to predicting models.

1. INTRODUCTION

Over the past decades, increased attention to environmental problems has brought substantial changes and improvements in the technique of refrigeration. In particular, with regards to refrigerants, the transition from CFCs to HCFCs and to HFCs has decreased the environmental impact of common refrigeration. Recently, global warming has been one of the most important issues facing mankind; indeed, while HFC refrigerants have no ozone depletion potential (ODP), many of them have relatively large values of global warming potentials (GWP) comprising one of the six so-called “baskets of gases” contained in the Kyoto Protocol (1997). As a result, much effort is being invested in finding alternative refrigerants to conventional HFCs, with natural refrigerants such as hydrocarbons, ammonia, and carbon dioxide receiving considerable research and development focus. The new refrigerant must have adequate stability to be used in the typical conditions of refrigeration equipment but also have a short atmospheric lifetime so that the contribution to global warming is small.

The use of propane as a refrigerant is a good opportunity to develop environmentally friendly heat pump and air-conditioning equipment, since the direct effect on the anthropogenic global warming due to atmospheric emissions is almost completely avoided, while the indirect effect can be reduced by exploiting the favorable thermodynamic properties of these fluids.
Because of flammability, charge minimization is a major design objective for such equipment using hydrocarbons. From previous experience, it appears that the estimated charge of unitary air conditioners is expected to be mainly trapped in the heat exchangers. In particular Harms et al. (2003) estimated the charge in three unitary air conditioners from 9 kW up to 26 kW and using R22 and R407C; they found that the computed charge in the condenser may vary from 30% up to 70% of the total amount, while the charge in the evaporator is lower (around 20%). Similar results have been obtained by Corberán and Martínez (2008) for a water-to-water propane heat pump using plate heat exchangers: 50% of the total charge is expected to be found in the condenser, while about 20% should be trapped in the evaporator.

In this regard, microchannel technology appears to be a very good opportunity to minimize the charge without energy performance loss. Cavallini et al. (2010) presented the experimental performance of a 100 kW heat pump using propane. A shell-and-tube heat exchangers using minichannels and providing low charge have been installed in the unit. It is shown that a 100 kW heat pump without a liquid receiver could be run with around 3 kg of propane using a plate condenser; when using the minichannel condenser, around 0.8 kg reduction can be obtained with a negligible performance loss.

Nevertheless, a very limited number of heat transfer data of propane in minichannels is available and in most cases experimental data is taken during vaporization.

Fernando et al. (2008) measured heat transfer during propane condensation inside a minichannel aluminium heat exchanger vertically mounted. The condenser was constructed of 36 multiport tubes; each multiport tube contained six rectangular parallel channel having a 1.42 mm hydraulic diameter. Experiments were performed at constant condensation temperatures of 30°C, 40°C and 50°C and mass flux ranging between 20 and 50 kg m⁻² s⁻¹. The authors showed that the experimental heat transfer coefficients were higher than those predicted by available correlations.

Shao et al. (2009) developed a distributed-parameter model of serpentine microchannel condensers; the Cavallini et al. (2002) correlation has been selected to calculate the refrigerant condensation heat transfer coefficient. The model was experimentally validated with a serpentine microchannel condenser using working fluid R-290; the predictions on the heat capacity and the pressure drop fall into ±10% error band.

Choi et al. (2009) examined the two-phase flow boiling pressure drop and heat transfer for propane in horizontal minichannels with inner diameters of 1.5 mm and 3.0 mm. The pressure drop and local heat transfer coefficients were obtained for heat fluxes ranging between 5 and 20 kW m⁻², mass fluxes ranging between 50 and 400 kg m⁻² s⁻¹ and saturation temperatures of 10, 5 and 0°C. They also developed new correlations for pressure drop and boiling heat transfer coefficients.

In this paper, the local heat transfer coefficient measured during condensation of propane in a 0.96 mm diameter circular minichannel is reported and compared versus available correlations. Tests are carried out in the experimental apparatus available at the Heat Transfer Lab of the University of Padova at mass fluxes ranging between 100 and 800 kg m⁻² s⁻¹. A new test section has been built for pressure drop measurements inside minichannels; the experimental setup has been validated during R290 liquid-phase laminar flow. Two-phase frictional pressure drop tests have been performed at mass velocity ranging from 200 kg m⁻² s⁻¹ to 800 kg m⁻² s⁻¹ and 40°C saturation temperature.

2. EXPERIMENTAL APPARATUS

2.1 General description of the test facility

Test runs have been performed in the test rig depicted in Fig. 1. The test facility includes a primary refrigerant loop which was cleaned and underwent several washing cycles to remove all the contaminants before filling it with propane. A washing cycle consists of creating a vacuum followed by pressurization with nitrogen and new vacuum. The composition of the refrigerant was certified by the provider (R290 purity 99.95%). Two thermal baths are employed: one provides the distilled water used as secondary fluid for the heat transfer in the test section, the other serves brine at 5°C to the auxiliary loop of the post-condenser, in which the working refrigerant is subcooled after exiting the measuring sector. The subcooled refrigerant passes through a filter drier before entering an independently controlled gear pump, which allows setting the mass flow measured by a Coriolis-effect mass flow meter. The evaporator is a tube in tube heat exchanger in which the primary fluid is heated using hot water flowing in a closed auxiliary loop provided with PID-controlled electrical heaters. The refrigerant is finally sent through the test section for pressure drop measurement or through the test section for condensation heat transfer investigation. Each test section is described below. In every test run, when the apparatus is working in steady state conditions,
measurements of thermo-fluidynamic parameters are recorded for 50 s. Each recording is averaged and then reduced by calculating the fluid properties with NIST Refprop Version 9.0 (2010).

2.2 Test section for heat transfer coefficients measurements

The test section is made of a straight single minichannel with two diabatic sectors separated by an adiabatic segment. The diabatic sectors work as heat exchangers by using a secondary fluid, which is water. The two sectors are made from an 8 mm external diameter copper rod with a 0.96 mm internal bore which is the test minichannel itself. The thick-walled tube was machined externally so as to obtain a cooling water channel within the wall thickness. The tortuous path for the secondary fluid, closed with plastic sheath, enables a good water mixing and thus allows precise local coolant temperature measurements. In this test section, fifteen thermocouples have been inserted into the water channel along the measuring sector in order to measure the coolant temperature profile. In order to measure precise local heat transfer coefficient values, fifteen thermocouples have been inserted into the wall thickness, near the minichannel along the measuring sector, without having the thermocouple wires cross the coolant path. Furthermore, one single thermocouple and one thermopile are used to know the refrigerant temperature by recording the external wall surface temperature of the diabatic sectors – stainless steel capillary tubes at the two extremes of the measuring sector; the saturation temperature is checked against pressure in the two adiabatic sectors upstream and downstream of the measuring sector. When operating in condensation mode, the first diabatic sector works as a desuperheater. To avoid large temperature gradients at the inlet of the measuring sector the desuperheater is used to cool down the superheated refrigerant to the saturation state at the inlet of the measuring sector.

2.3 Test section for pressure drop measurements

The temperature of the water in the evaporator is set so that the refrigerant enters the pre-sector as subcooled liquid or superheated vapour. The state of the refrigerant at the inlet of the pre-sector is determined from temperature and pressure measurements.

![Diagram](image_url)

**Figure 1:** Experimental test rig: PS (pre-sector); MF (mechanical filter); FD (filter dryer); PV (pressure vessel); CFM (Coriolis-effect mass flow meter); TV (valve); P (pressure transducer); DP (differential pressure transducer); T (thermocouple).
Figure 2: Experimental apparatus for heat transfer and pressure drop measurements inside a single minichannel.

The pre-sector is a counter-flow heat exchanger in which distilled water is used to have the refrigerant at the desired vapour quality condition at the inlet of the measuring sector. The mass flow rate of the distilled water is measured by a Coriolis-effect mass flow meter, the water outlet temperature in the pre-sector is measured by a T-type thermocouple and the water temperature difference between inlet and outlet is measured by a thermopile. Static mixers have been positioned upstream of the water temperature sensors and therefore the measured temperatures can be considered as the mean effective water temperature. The refrigerant vapour quality at the inlet of the measuring sector is obtained from a heat balance in the pre-sector.

The measuring sector consists of a copper rod with a 0.96 mm internal bore having an inner surface roughness $Ra = 1.3 \, \mu m$. The minichannel was placed in horizontal and was connected to the pre-sector and the rest of the facility through stainless steel tubes. Two pressure ports were made directly on the copper rod and they are connected through warmed pressure lines to digital strain gauge transducers for the measurement of the absolute pressure at the inlet and the pressure drop through the measuring sector. The length of the measuring sector between the pressure ports is 0.22 m. T-type thermocouples were placed upstream and downstream of the minichannel on the stainless steel tubes that act as adiabatic sectors and allow checking the agreement between temperature and pressure under saturated conditions. The minichannel was insulated to the external environment in order to perform frictional pressure drop tests in adiabatic conditions. The temperature transducers were calibrated according to the procedure described in Del Col et al. (2011) and the pressure strain gauge transducers were checked against a pressure calibrator. Furthermore, the energy balance in pre-sector was checked by comparing the water side heat transfer rate to the one determined on the refrigerant side both during complete condensation from superheated vapour to subcooled liquid and during complete vaporization from subcooled liquid to superheated vapour. The heat dissipation towards the external environment in the measuring sector was found to be negligible in the temperature range of the present test runs.

3. CONDENSATION TESTS

3.1 Data reduction

The following three parameters are used for the determination of the local heat transfer coefficient: the local heat flux, the saturation temperature and the wall temperature. The heat flux is determined from the temperature profile of the coolant in the measuring sector. The wall temperature is directly measured along the test section and the saturation temperature is measured in the adiabatic segments at the inlet and outlet of the test minichannel and checked through pressure transducers.
The coolant temperature profile is obtained from the thermocouples set in the water channel along the measuring sector. The derivative of the temperature profile is proportional to the local heat flux:

\[ q'(z) = -\dot{m}_{\text{water}} \cdot c_{p,\text{water}} \cdot \frac{1}{\pi \cdot d} \cdot \frac{dT_{\text{water}}(z)}{dz} \]  

(1)

and it is associated to the local heat transfer coefficient:

\[ \text{HTC}(z) = \frac{q'(z)}{T_{\text{sat}}(z) - T_{\text{wall}}(z)} \]  

(2)

The local saturation temperature \( T_{\text{sat}}(z) \) of the fluid along the measuring sector is calculated from the two measured values in the adiabatic sectors. The calculation, which considers frictional pressure drop and pressure recovery due to condensation, is iterative. The local pressure is calculated from the inlet pressure by taking into account the local pressure gradient in order to match the saturation temperature curve to the saturation temperature measurement at the outlet of the measuring sector. On the other hand, the wall temperature \( T_{\text{wall}}(z) \) is measured locally.

By considering the conservation of energy in the sector, the coolant temperature change is directly associated to the corresponding enthalpy variation of the refrigerant. Therefore, the local vapour quality is calculated as follows:

\[ x(z) = x_{in} - \frac{\pi \cdot d \cdot \int_0^z q'(z)dz}{\dot{m}_{R290} \cdot h_{LV}} \]  

(3)

In the present technique, the dominant thermal resistance during the condensing process is on the refrigerant side; this is favourable to the reduction of the experimental uncertainty associated to the determination of the heat transfer coefficient. In fact, one contribution of the experimental heat transfer coefficient uncertainty is related to the saturation to wall temperature difference. The other main uncertainty terms are associated to the heat flux, the mass flow rate and the hydraulic diameter. Since the heat flux is obtained from the water temperature gradient, this uncertainty contribution depends on the operating conditions, mainly mass flux and vapour quality, yielding higher uncertainty at lower mass flux. In the end, at high mass velocity, the total heat transfer coefficient uncertainty is below 4%, while at the lowest mass velocity, 100 kg m\(^{-2}\) s\(^{-1}\), it ranges between 9% and 15%. More details on the error analysis are reported in Del Col et al. (2011).

Prior to the tests, a proper calibration procedure has been performed. Besides, several tests have been run to verify that the heat transfer coefficient does not depend on the conditions of the secondary fluid.

3.2 Heat transfer during condensation of propane

The local heat transfer coefficient has been measured during condensation of R-290. The experiments have been performed over the entire range of vapour quality at 40°C saturation temperature and mass velocity ranging from 100 kg m\(^{-2}\) s\(^{-1}\) up to 800 kg m\(^{-2}\) s\(^{-1}\). The complete set of the experimental heat transfer coefficients measured during condensation is plotted in Fig. 3 versus vapour quality. As expected for forced convective condensation inside conventional pipes, the heat transfer coefficient increases with mass velocity and vapour quality.

3.3 Comparison against models

Experimental results have been compared against two models available in the open literature and developed for HTC predictions inside macro-scale tubes, which have been shown to work pretty well inside minichannels by Matkovic et al. (2009). The first correlation is the one by Cavallini et al. (2006), which was also developed for condensation inside channels with hydraulic diameters higher or equal to 3 mm. It also accounts for the transition from \( \Delta T \)-independent to \( \Delta T \)-dependent region, where \( \Delta T \) is the saturation minus wall temperature difference. All the data points at mass velocity higher or equal to 100 kg m\(^{-2}\) s\(^{-1}\) lay in the \( \Delta T \) independent region and may be predicted
**Figure 3:** Local experimental condensation heat transfer coefficient versus vapour quality for R290 at mass velocities ranging from 100 to 800 kg m$^{-2}$ s$^{-1}$.

**Figure 4:** Calculated versus experimental R290 heat transfer coefficient. Left: the model by Cavallini et al. (2006) is applied. Right: the model by Moser et al. (1998), with the pressure drop correlation by Zhang and Webb (2001).

by using a model for annular flow condensation. Fig. 4 reports the comparison between experimental heat transfer coefficients vs. predicted values by using the correlation by Cavallini et al. (2006). R290 data are well predicted by this correlation, which is able to catch the experimental trend.

The second model used in the present comparison is the one by Moser et al. (1998), which was initially developed for conventional pipes and later modified by using the Zhang and Webb (2001) method for pressure drop calculation inside small-diameter tubes. The comparison between experimental values and predictions is depicted in Fig. 4. As one can see, the model by Moser et al. (1998) modified with the Zhang and Webb (2001) pressure drop correlation is able to predict almost all experimental data within ±20% band.
4. PRESSURE DROP

4.1 Single-phase pressure drop

Single-phase pressure drop has been measured to validate the data acquisition system and to gain critical insight into the test section hydraulic performance. The friction factor has been determined during adiabatic flow of R290 in subcooled liquid state and in superheated vapour state from pressure drop and mass flow rate measurements, according to Eq. 4. In the laminar region, it has been used to check the geometry of the channel, while, in the turbulent region, it has been employed to control the measured value of the surface roughness.

\[ f = \frac{\rho \cdot d \cdot \Delta p}{2 \cdot G^2 \cdot L} \]  

(4)

In Fig. 5, the measured friction factor is plotted as a function of the Reynolds number and two different symbols are utilized in order to distinguish the experimental points taken with liquid phase and vapour phase flow. Experimental data is compared against Hagen-Poiseuille (1839, 1840-1841) equation for laminar flow and Blasius (1913) equation for turbulent flow inside smooth tubes; the Churchill (1977) equation, that covers both laminar and turbulent flow and accounts for channel roughness, is also reported. The agreement between experimental and predicted values is very good.

4.2 Pressure drop during adiabatic vapour-liquid flow

Two-phase frictional pressure drop tests have been performed during adiabatic flow of R290 at 40-42°C inside the circular cross section minichannel at mass velocities ranging from 200 kg m\(^{-2}\) s\(^{-1}\) to 800 kg m\(^{-2}\) s\(^{-1}\). The vapour quality for each experimental point is calculated as given in Eq. 5:

\[ x = \frac{h_{m,MS} - h_L}{h_v - h_L} \]  

(5)

where \( h_L \) and \( h_v \) are the specific enthalpy of saturated liquid and saturated vapour at the mean pressure in the measuring sector and \( h_{m,MS} \) is the specific enthalpy of the refrigerant at the inlet of the measuring sector and it results from the energy balance in the pre-sector heat exchanger, according to Eq. 6:
\[ h_{in,MS} = h_{in,PS} = \frac{\dot{m}_{water} \cdot c_{p,water} \cdot \Delta T_{water,PS}}{\dot{m}_{R290}} \]  

(6)

The specific enthalpy of the refrigerant at the inlet of the pre-sector \( h_{in,PS} \) is calculated from temperature and pressure at the inlet, the specific heat \( c_{p,water} \) is referred to the mean water temperature in pre-sector. The maximum experimental uncertainty of the measured pressure drop is ±0.13 kPa, while the maximum uncertainty of the vapour quality is equal to 0.01 at \( G = 200 \) kg m\(^{-2}\) s\(^{-1}\). The experimental pressure drop gradient is plotted against vapour quality in Fig. 6.

When performing test runs with vapour quality lower than 0.5, the refrigerant enters the pre-sector as subcooled liquid at 20-23°C, therefore in the pre-sector heat exchanger a partial vaporization occurs. On the contrary, to get experimental points with vapour quality higher than 0.5, the fluid enters the pre-sector as superheated vapour at 50-55°C and a partial condensation occurs there. Thanks to the present experimental technique, the difference between the mean water temperature in pre-sector and the saturation temperature at the inlet of the measuring sector ranges from -15°C and +15°C. The experimental points taken by partial vaporization of R290 and those obtained by partial condensation of the refrigerant are in good agreement at vapour quality equal to 0.5 and the global trend of the points does not show discontinuity. At high mass velocities of \( G = 600 \) kg m\(^{-2}\) s\(^{-1}\) and \( G = 800 \) kg m\(^{-2}\) s\(^{-1}\), the pressure drop gradient shows an inflection point and the slope becomes steep at high vapour qualities. This may be due to the liquid entrainment that appears during shear dominated flow regime, where the shear stress strips liquid drops away from the liquid-vapour interface. As a consequence, the slope of the pressure drop gradient profile changes when the liquid entrainment occurs.

4.3 Comparison against models

In Fig. 7, the frictional pressure drop gradient calculated at 40°C using the Cavallini et al. (2009) model is plotted along with the experimental data measured during adiabatic flow of R290. The effect of surface roughness is taken into account in the model. Deviations between experiments and predictions are very low, with average deviation \( e_R = -5.4\% \) and standard deviation \( \sigma_N = 9.8\% \).

In Fig. 8 the same experimental data are compared against the Friedel (1979, 1980) model and the correlation by Zhang and Webb (2001) for two-phase pressure drop in small diameter channels. The agreement is satisfactory, above all when using the Friedel method (\( e_R = -5.1\% \) and \( \sigma_N = 12.3\% \)), while the Zhang and Webb correlation tends to underestimate more the present data (\( e_R = -17.8\% \) and \( \sigma_N = 9.7\% \)).

![Figure 7: Experimental pressure drop gradient and calculated trends by Cavallini et al. (2009) model.](image)
Figure 8: Experimental pressure drop gradient and calculated trends. On the left: model by Friedel (1979, 1980); on the right: model by Zhang and Webb (2001).

At high mass flux, $G = 800$ kg m$^{-2}$ s$^{-1}$, the predicted trends of the models by Friedel (1979, 1980) and Zhang and Webb (2001) do not exhibit any change of slope, while the calculated trend by Cavallini et al. (2009) model, because of the effect of liquid entrainment, displays an inflection point as the one shown in the experimental curve. Therefore, the important role of liquid entrainment at high mass velocities during two-phase flow in minichannels seems to be confirmed.

5. CONCLUSIONS

In this paper, condensation heat transfer coefficients and two-phase frictional pressure drop measured with R290 in a 0.96 mm diameter single round channel have been reported. The experimental values of the heat transfer coefficient show that the condensation is shear stress dominated for most of the data points and they can be well predicted by using the Cavallini et al. (2006) model and the Moser et al. (1998) model modified by using the Zhang and Webb (2001) method for pressure drop calculation. Adiabatic pressure gradient has been measured inside the same channel; the experimental setup has been validated with measurements during R290 liquid-phase laminar flow. Two-phase frictional pressure drop tests have been performed at mass velocity ranging from 200 kg m$^{-2}$ s$^{-1}$ to 800 kg m$^{-2}$ s$^{-1}$ and 40°C saturation temperature. The experimental pressure gradient data have been compared against the models by Cavallini et al. (2009), Friedel (1979, 1980) and Zhang and Webb (2001); the models show a satisfactory agreement with the measured values.

NOMENCLATURE

- $c_p$: specific heat (J kg$^{-1}$ K$^{-1}$)
- $d$: tube inside diameter (m)
- $f$: friction factor
- $e_p$: percentage deviation = 100($y_{CALC} - y_{EXP}$)/$y_{EXP}$
- $e_R$: average deviation = $(1/N_p)\Sigma e_p$
- $G$: mass velocity (kg m$^{-2}$ s$^{-1}$)
- $h$: specific enthalpy (J kg$^{-1}$)
- $h_{LV}$: heat of vaporization (J kg$^{-1}$)
- HTC: heat transfer coefficient (W m$^{-2}$ K$^{-1}$)
- $L$: distance between pressure ports (m)
- $m$: mass flow rate (kg s$^{-1}$)
- $N_p$: number of data points

Subscripts
- CALC: calculated
- EXP: experimental
- $f$: frictional
- in: inlet
- L: saturated liquid
- MS: measuring sector
- PS: pre-sector
- sat: saturation
- V: saturated vapour
- wall: wall
- water: water
$q'$: specific heat flux (W m$^{-2}$)
$\bar{R}a$: arithmetical mean deviation (µm)
of the profile (ISO 4297:1997)
$Re$: Reynolds number ($-$)
$T$: temperature (K)
$x$: thermodynamic vapour quality ($-$)
$z$: axial position (m)
$Δp$: pressure drop (Pa)
$Δp/Δz$: pressure drop gradient (kPa m$^{-1}$)
$ΔT$: temperature difference (K)
$\rho$: density (kg m$^{-3}$)
$σ_N$: standard deviation = $[\Sigma (e_{ir} - e_{ir})^2/(N-1)]^{1/2}$ ($-$)

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