



# Heat transfer characteristics of hydrocarbon mixtures refrigerant during condensation in a helical tube

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## ABSTRACT

In this paper, the condensation heat transfer of hydrocarbon mixtures refrigerant (methane/propane) flowing in a helical tube with 10 mm hydraulic diameter was investigated by experiments and numerical simulations. The experiments were conducted at mass flux of 200–400 kg/(m<sup>2</sup>·s), saturation pressure of 0–4 Mpa and heat flux of 4.8–5 kW/m<sup>2</sup>. Also, the effects of mass flux, vapor quality, and saturation pressure on heat transfer coefficient and vapor quality, while the heat transfer coefficients generally decreased as the saturation pressure increased. Meanwhile, annular flow (half-annular flow, annular flow) and Non-annular flow patterns (slug flow, stratified flow, wavy flow, transition flow) were observed during the condensation of methane/propane mixtures refrigerant. And a new flow pattern transition line for hydrocarbon refrigerants condensation process was proposed. The condensation heat transfer coefficient data were compared with several existing correlations, and a modified heat transfer correlation for different flow patterns was proposed, which predicted the results well with a mean absolute error of 10.03%.

## 1. Introduction

Nowadays, liquefied natural gas (LNG) has been widely used in the world, and helically coiled tubes have been introduced as one of the passive heat transfer enhancement techniques are widely used in LNG liquefying applications due to their high heat transfer coefficient and compact structure. The two-phase heat transfer data of LNG are crucial parameters for design and optimization the liquefaction and gasification equipment, and LNG predominantly consists of hydrocarbons refrigeration. Therefore, to know the accurate flow and heat transfer characteristics of hydrocarbons refrigeration is crucial for understanding the multi-components mixture.

For hydrocarbons refrigeration, they have greater heat transfer coefficients than that of the synthetic refrigerant [1–3]. Park et al. [4] found that the condensation heat transfer coefficients of hydrocarbons increased as the quality and mass flux increased, and the correlation proposed by Jung et al. [5] could well predict the experimental results. Del Col [6] presented the results from an experimental investigation of propane two-phase heat transfer in a minichannel. The results showed that the heat transfer coefficients increased with mass velocity and with

vapor quality. The heat transfer coefficients were well predicted by Cavallini et al. [7]. Macdonald [8] experimentally conducted propane condensation inside horizontal tubes. They conducted that the heat transfer coefficients increased slightly with the tube diameter. The same conclusion was obtained by Del Col [6] on the effect of vapor quality and mass flux on the heat transfer coefficients, but it decreased with the increase of saturation temperature. All correlations performed large deviation at higher reduced pressures. Nevertheless, Hossain et al. [9] presented that the saturation temperature had no effect on heat transfer coefficients under the lower mass flow rate in case of the heat transfer mechanism was forced convection. Ağra and Teke [10] pointed out that the saturation temperature of the working fluid had no effect on heat transfer coefficient at low mass fluxes for condensation of iso-butane in a 4-mm tube. And the average condensation heat transfer coefficients decreased with the increase of temperature difference. Lots of experimental results showed that the increase of mass flow rate and vapor quality could increase the condensation heat transfer coefficients. However, Dobson [11] and Cavallini et al. [12] conducted that the heat transfer coefficients increased with vapor quality at high mass flow rate, and it had no relationship with vapor quality during low mass flow

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**Nomenclature**

$A$	heat transfer surface from inlet to middle of test section, $\text{m}^2$
$A_o$	total heat transfer surface in test section, $\text{m}^2$
$C_p$	specific heat at constant pressure, $\text{J kg}^{-1} \text{K}^{-1}$
$d_i$	inside diameter of inner tube, $\text{m}$
$d_o$	outside diameter of inner tube, $\text{m}$
$D$	the hydraulic diameter, $\text{m}$
$E_q$	the correction factor for the heat flux
$Fr_l$	liquid Froude number, $Fr_l = [G(1-x)]^2/(\rho_l^2 g D)$
$g$	gravity, $\text{m s}^{-2}$
$G$	mass velocity, $\text{kg}/(\text{m}^2 \text{s})$
$Ga$	Galileo number, $Ga = g \rho_l (\rho_l - \rho_g) D^3 / \mu_l^2$
$h$	heat transfer coefficient, $\text{W m}^{-2} \text{K}^{-1}$
$i$	enthalpy, $\text{J kg}^{-1}$
$i_{lg}$	latent heat, $\text{J kg}^{-1}$
$Ja_l$	liquid Jakob number, $Ja_l = C_{p,l}(T_{sat} - T_o)/i_{lg}$
$J_G$	dimensionless gas velocity, $J_G = xG/[gD\rho_g(\rho_l - \rho_g)]^{0.5}$
$l$	length, $\text{m}$
$m$	the mass flow rate, $\text{kg s}^{-1}$
$MAE$	mean absolute error
$Pr$	Prandtl number
$P$	Pressure, $\text{Pa}$
$P_{red}$	reduced pressure
$Q$	the heat transfer rate
$Q_{pre}$	the output power of pre-heater
$q$	heat flux, $\text{W m}^{-2}$
$R$	the diameter of curvature
$Re$	Reynolds number $Re = GD/\mu$
$Re_{lo}$	liquid-only Reynolds number $Re_{lo} = GD/\mu_l$
$T_o$	average temperature of thermocouples along the test section surface
$\Delta T_{in}$	the temperature difference between test fluid and cooling fluid at the inlet of test fluid side
$\Delta T_{out}$	the temperature difference between test fluid and cooling

	fluid at the outlet of test fluid side
$T$	the temperature, $^\circ\text{C}$
$Su_g$	Suratman number, $Su_g = \rho_g \sigma D / \mu_g^2$
$v$	Velocity, $\text{m s}^{-1}$
$We$	Weber number, $We = G^2 D / (\rho_g \sigma)$
$We^*$	modified Weber number
$X_{tt}$	Lockhart-Martinelli parameter
$x$	vapor quality

**Greek symbols**

$\alpha$	void fraction
$\lambda$	thermal conductivity, $\text{W m}^{-1} \text{K}^{-1}$
$\Delta T$	temperature difference, $\text{K}$
$\mu$	viscosity, $\text{Pa s}$
$\rho$	density, $\text{kg m}^{-3}$
$\theta$	Angle subtended from the top of tube to the liquid level
$\eta$	percentage of points predicted within $\pm 30\%$ range
$\sigma$	surface tension

**Subscripts**

$c$	cooling fluid
$cal$	calculated
$g, l$	vapor phase and liquid phase, respectively
$go$	Vapor only
$i$	inner wall
$in$	inlet
$lo$	liquid only
$out$	outlet
$pred$	predicted
$sat$	the saturation condition
$STRAT$	fully stratified flow regime
$t$	test fluid
$tot$	total
$tp$	two-phase

rate. Meanwhile, Yan et al. [13] also found that the mass flux showed slight influence in the low vapor quality region. Marák [14] investigated the condensation heat transfer of methane and binary methane mixtures in small vertical channels. For pure fluids, the heat transfer coefficients increased with the increasing mass flux, and it generally decreased with the increasing pressure. The heat transfer coefficients were dependent on the heat flux for binary fluid condensation. Chang et al. [15] considered that pure hydrocarbons performed better than binary mixtures for condensation heat transfer.

For refrigerant condensation in a helical tube, the condensation heat transfer coefficients were higher than those in a straight tube [16–18]. The existing experimental investigations on the condensation heat transfer characteristics of two-phase flow in a helical tube focused on the non-hydrocarbon flow. Kang et al. [19] investigated the condensation heat transfer characteristics of R134a inside a helical tube with inside diameter of 12.7 mm. The results showed that the refrigerant-side heat transfer coefficients decreased with the increase of mass flux or Reynolds number. These results were contrary to those gained by Al-Hajeri et al. [20], who found that the heat transfer coefficient increased with the mass flux increased, however, the refrigerant-side heat transfer coefficients decreased with the increase of saturation temperature. According to the above studies, it is clearly seen that very limited results are available for condensation heat transfer of hydrocarbon mixtures refrigerant in a helical tube.

In this paper, a new test facility was designed to realize the convective condensation heat transfer in a helical tube. The numerical simulation method was used to expand the data application range based

on the experiment data. Meanwhile, the visualization of flow patterns of methane and propane condensation was obtained. In addition, the influence of mass flux, heat flux and vapor quality on the heat transfer characteristics of hydrocarbon mixtures refrigerant had been discussed. According to the results of the visual flow patterns, the transition line between different flow patterns was derived. Finally, the data was also compared with various existing condensation heat transfer correlations. An better prediction heat transfer correction for different flow patterns was proposed.

**2. Experimental and numerical simulation methods****2.1. Experimental apparatus**

The experimental setup was designed to measure both single-phase and two-phase heat transfer for hydrocarbon refrigerants inside a helical tube. Based on the previous experimental apparatus [21], the flow pattern observation device was added. Its temperature range was approximate  $-123^\circ\text{C}$  to  $20^\circ\text{C}$  and the maximum pressure was 4 MPa, and refrigerant flow rates could be varied up to  $400 \text{ kg}/(\text{m}^2 \text{s})$ .

**2.1.1. Test loop**

Fig. 1 shows a schematic diagram of the experimental facility. It consisted of a test, condensation and a cooling flow loop (liquid nitrogen as the cold resource). In the test loop, the sub-cooled test fluid was circulated through the Coriolis mass flow meter with an accuracy of  $\pm 0.1\%$  by the Cryogenic piston pump. The mass flux was controlled

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