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Saturated vapour condensation of R410A inside a 4 mm ID horizontal smooth tube: Comparison with the low GWP substitute R32

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ABSTRACT

This study performs the comparative analysis of R32 and R410A condensation inside a 4 mm ID smooth tube. The experimental tests were carried out at three different saturation temperatures, 30 °C, 35 °C, and 40 °C, at different vapour quality and mass velocities to evaluate the specific contribution of refrigerant mass flux, mean vapour quality, and condensation temperature (pressure). The frictional pressure drops exhibit great sensitivity to all the operating variables considered, while the condensation heat transfer coefficients show great sensitivity only to refrigerant mass flux and mean vapour quality. The transition between gravity controlled and forced convection condensation occurred in the range of the equivalent Reynolds number 10,000–20,000. The Friedel (1979) correlation was able to predict properly the entire set of frictional pressure drops data, while the Akers et al. (1959) model gave a fair estimation of the forced convection condensation heat transfer coefficients. R32 exhibits condensation heat transfer coefficients and frictional pressure drops higher than those of R410A and it seems a valuable low GWP substitute for R410A.

1. Introduction

The substitution of traditional Hydrofluorocarbon (HFC) refrigerants affected by large Global Warming Potential (GWP) involves the use of fluids with a reduced chemical stability, such as flammable refrigerants (ASHRAE classification A3) or mildly flammable refrigerants (ASHRAE classification A2L). These types of refrigerants require the re-design of the refrigerating machines, adopting for example a secondary fluid configuration, or the use of heat transfer equipment with low refrigerant charge, such as plate heat exchangers, or tubular heat exchangers with small-diameter round tubes or multiport tubes.

The authors of the present paper have already investigated this item considering in [1] the condensation of R290 (propane) and R1270 (propylene) inside a small-diameter smooth tube as long-term low GWP substitutes for the traditional refrigerants R404A and R507A, that have dominated commercial refrigeration in the last two decades. Condensation of R404A, R290 and R1270 inside a small-diameter smooth tube evidences a transition point from gravity controlled to forced convection condensation for an equivalent Reynolds number around 10,000. Moreover, the traditional heat transfer models for refrigerant condensation inside smooth tubes of conventional diameter, such as for example Cavallini and Zecchin [2] and Dobson and Chato [3], greatly over-predicted this data for small-diameter tubes, whereas the classical Akers et al. (1959) equation [4] showed a good ability in predicting the experimental data in forced convection condensation.

The substitution of R410A, the world-wide leading refrigerant in small and medium size air-conditioners, with low GWP refrigerants involves similar critical design issues, as it may require the use of highly flammable refrigerants or the adoption of non-azeotropic refrigerant mixtures with large temperature glides. The unique high pressure pure HFC refrigerant with a relatively low GWP, 675, is R32 that exhibits a mild flammability (ASHRAE classification A2L). Currently R32 has already found large use in Japan, China, and India and has been recently commercialized in Europe as a substitute for R410A in split air-conditioners. In order to reduce the residual risk associated with its mild flammability, R32 should be applied in heat transfer equipment with low refrigerant charge such as small-diameter tubes.

In the open literature it is possible to find some experimental works on R32 in-tube condensation. Cavallini et al. [5] in 2001 presented R32 condensation heat transfer coefficients and pressure drops in an 8 mm horizontal smooth tube with a refrigerant mass flux in the range 100–750 kg m^{-2} s^{-1}. Hossain et al. [6] in 2012 measured the condensation heat transfer coefficient of R32 inside a 4.35 mm horizontal smooth tube with a refrigerant mass flux

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Nomenclature

c_p	specific heat capacity, J kg^{-1}K^{-1}
d	tube diameter, m
f.s.	full scale
G	refrigerant mass flux, kg m^{-2}s^{-1}
h	heat transfer coefficient, W m^{-2} K^{-1}
ID	inside diameter
J_H	heat transfer factor
J_G^*	dimensionless gas velocity
k	coverage factor
L	length of the measurement section, m
MAPE	mean absolute percentage deviation
Nu	Nusselt number
p	pressure, Pa
Pr	Prandtl number
R_a	arithmetic mean roughness (ISO4271/1), μm
Re	Reynolds number
R_p	roughness (DIN 4762/1), μm
T	temperature, K (°C)
X	vapour quality
X_{tt}	Martinelli parameter

Greek symbol

ρ	density, kg m^{-3}
μ	dynamic viscosity, kg m^{-1} s^{-1}

Subscript

eq	equivalent
f	frictional
G	gas/vapour
h	hydraulic
L	liquid
LG	liquid vapour phase change
m	mean value
r	refrigerant
sat	saturation

varying from 150 to 445 kg m^{-2} s^{-1} over the vapour quality range 0.00–1.00. Agarwal and Hrnjak [7] in 2015 measured R32 condensation heat transfer coefficients inside a 6.1 mm horizontal smooth tube considering also the effects of vapour de-super-heating. Guo et al. [8] in 2015 compared R22, R32, and R410A during condensation inside a 11.43 mm smooth tube: R32 exhibited the best heat transfer performance. Therefore, there is a specific need for a sound amount of new experimental data concerning R32 condensation inside small-diameter tubes in order to determine the different heat transfer regimes, to select the more adequate computational procedures, and to compare its performance to that of R410A.

The present paper carried out the comparative analysis of R32 and R410A condensation inside a 4 mm horizontal smooth tube: the effects of refrigerant mass flux, vapour quality and saturation temperature were investigated separately to identify the dominant heat transfer regimes and the relative transition conditions. The experimental data points were also compared with different two-phase heat transfer and pressure drop models.

2. Experimental measurements and results

The experimental facility and procedures were already described in details in a previous work [1] and here they are only briefly summarised.The experimental rig is based on a closed loops scheme consisting of a refrigerant circuit (primary loop), a water-glycol loop and a refrigerated/cooling water loop (supply loops). The test-section is a tube-in-tube heat exchanger with the refrigerant condensing in the inner tube and the cooling water flowing in the annulus. The inner tube is instrumented with four T-type (copper-constantan) thermocouples embedded in its wall to measure surface temperature. The geometrical characteristics of the test-section are reported in Table 1, while Table 2 shows the characteristics of the instrumentation used in the experimental circuit. The experimental results are reported in terms of condensation heat transfer coefficients h_r and frictional pressure drop Δp_f. The condensation heat transfer coefficient is computed by measuring the surface temperature of the tube wall, while the condensation frictional pressure drop is obtained from the total pressure drop by subtracting the inlet/outlet local pressure drop and adding the momentum pressure rise. The refrigerant vapour quality at the measurement section inlet and outlet are computed starting from the refrigerant temperature and pressure at the inlet of the pre-evaporator (sub-cooled liquid condition) considering the heat flow rate exchanged in the pre-evaporator, in the pre-section and in the measurement section and the pressure at the inlet and outlet of the test section. It should be noted that in the present experimental tests the vapour quality changes through the measurement section and pre-section were limited. The refrigerant properties are evaluated accordingly with [9].The experimental runs include two sets of 126 saturated vapour condensation data points carried out with refrigerant R410A and R32, respectively, at three different condensation temperatures, 30 °C, 35 °C, and 40 °C, and different vapour quality. Table 3 shows the operating conditions in the experimental test: condensation temperature T_{sat} and pressure p_{sat}, mean vapour quality X_m, and refrigerant mass velocity G. An error analysis carried out following the approach [10] evidences an overall uncertainty within ±19.8% and ±19.6% for the condensation heat transfer coefficient and within ±12.5% and ±13.0% for the total pressure drop for R410A and R32, respectively. At each condensation temperature (pressure), seven different refrigerant mass velocities (G = 100 kg m^{-2} s^{-1}, 150 kg m^{-2} s^{-1}, 200 kg m^{-2} s^{-1},

Table 1
Geometrical characteristics of the test-section.

Parameter	
Tube inside diameter d (mm)	4.0
Measurement section length L (mm)	800
Pre-section length (mm)	200
Total section length (mm)	1300
Inside tube surface roughness R_a (m) (ISO 4287/1)	0.7
Inside tube surface roughness R_p (m) (DIN 4762/1)	1.8

Table 2
Specification of the different measuring devices.

Devices	Uncertainty (k = 2)	Range
T-type thermocouples	0.1 K	−20/80 °C
T-type thermopiles	0.05 K	−20/80 °C
Abs. pressure transducers	0.075% f.s.	0/2.0 MPa
Diff. pressure transducers	0.075% f.s.	0/0.05 MPa
Coriolis effect flow meters	0.1%	0/300 kg h^{-1}
Magnetic flow meters	0.15% f.s.	100/1200 l h^{-1}
Data logger	± 2.7 μV	0/100 mV