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ABSTRACT

This paper presents the comparative analysis of HFC134a and its low GWP substitutes HFO1234yf, and HFO1234ze(E) in saturated vapour condensation inside a 4 mm ID horizontal smooth tube. The experimental tests were carried out at 30, 35, and 40°C of saturation temperatures, with refrigerant mass flux in the range 100 - 600 kg $m^{-2}s^{-1}$ at decreasing vapour quality. A transition point from gravity-dominated and forced convection condensation was found in the range of the equivalent Reynolds number 10,000 – 20,000. The experimental heat transfer coefficients in the forced convection condensation regime were very well predicted by the Akers et al. (1959) model, whereas the Friedel (1979) correlation was able to reproduce the frictional pressure drop data in the whole experimental range. HFO1234yf and HFO1234ze(E) exhibit heat transfer coefficients and frictional pressure drops similar to those of HFC134a and both the HFO refrigerants seem to be very promising as long-term low GWP substitutes for HFC134a.

1. INTRODUCTION

The substitution of HFC134a with low GWP refrigerants is one of the most important challenge for refrigeration and air conditioning. The possible substitutes include natural refrigerants, such as HC600 (Butane) and HC600a (Isobutane), and also synthetic refrigerants, such as HFO1234yf and HFO1234ze(E). The HC refrigerants exhibit very low GWP, 3 and 4 HC600a and HC600 respectively, good thermodynamic and transport properties, and pressure and volumetric performance very similar to HFC134a. The major drawback of HC refrigerants is their high flammability, being classified in class A3 according to ASHRAE classification. Also the HFO refrigerants present a mild flammability, being classified in class A2L. In fact it is very difficult to found low GWP substitutes for traditional HFC refrigerants with no flammability, as a weak chemical stability and / or a big chemical reactivity are presuppositions for low GWP. Both HFO1234yf and HFO1234ze(E) seem to be very promising as substitute for HFC134a, showing a GWP lower than 1 together with pressure and volumetric properties closely near to those of HFC134a.
In the open literature it is possible to find some experimental work on HFO1234yf and HFO1234ze(E) intube condensation, however the major part of the data refers to traditional 3/8" and 1/2" OD smooth or microfin tubes and only a few data considers small-diameter tubes. This paper presents the comparative analysis of HFC134a HFO1234yf and HFO1234ze(E) during saturated vapour condensation inside a 4 mm ID horizontal smooth tube: the effects of refrigerant mass flux, mean vapour quality and saturation temperature (pressure) are investigated.

2. EXPERIMENTAL MEASUREMENTS AND DATA REDUCTION

The experimental facility, shown in Figure 1, consists of a refrigerant circuit, a water-glycol loop and a refrigerated / cooling water loop. The test-section is a double tube condenser in which the refrigerant vapour condenses in the inner tube while the cooling water flows in the annulus.

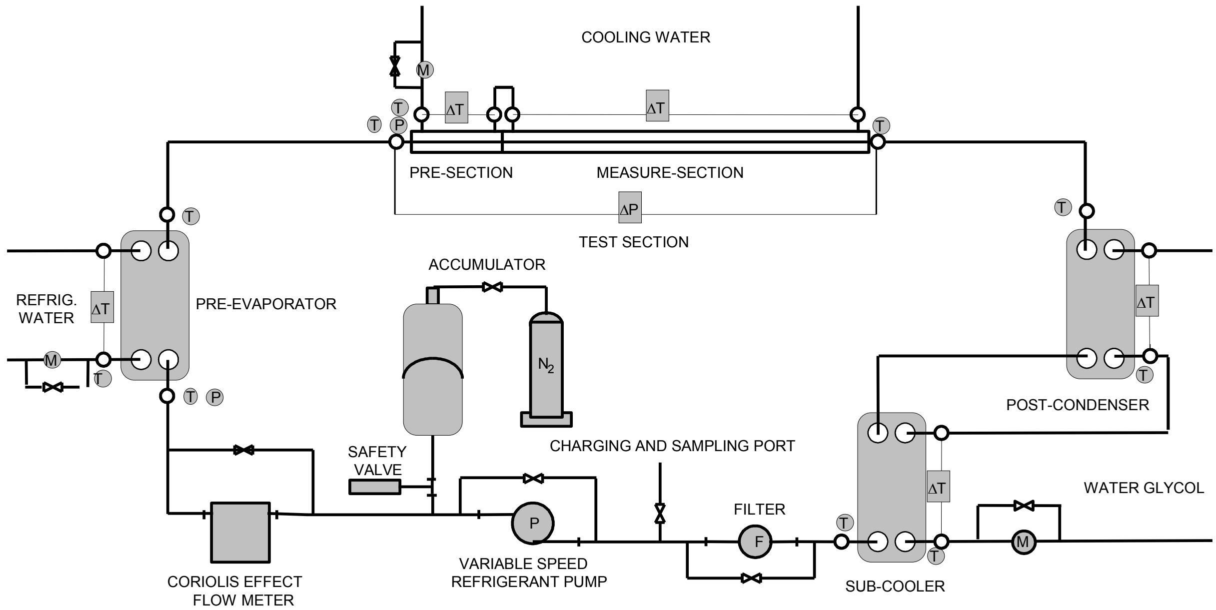

Figure 1: Schematic view of the experimental test rig.

The test-section is subdivided into two different parts: a pre-section, 200 mm long, in which the refrigerant flow achieves a fully developed flow regime and the measurement section, 800 mm long, in which the heat transfer coefficient is measured. This arrangement is obtained using a single inner smooth tube, 4 mm ID, 1300 mm long and two separated cooling water jackets fed in series. The inner tube is instrumented with four copper-constantan thermocouples (uncertainty (k= 2) within ±0.1 K) embedded in its wall to measure surface temperature. The thermocouples are inserted into two equidistant axial grooves, at the top and the bottom of the cross section, 100 mm from the inlet and outlet of the cooling water. Each groove is sealed with a copper wire fixed by epoxy.
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 h_r is equal to the ratio between the heat flow rate Q, the heat transfer area A and the mean temperature difference ΔT:

$$h_r = Q / (A \, \Delta T) \qquad (1)$$

The heat flow rate Q is derived from a thermal balance on the water-side, the heat transfer area A is equal to the area of the inner surface of the test tube, and the mean temperature difference ΔT is equal to the difference between the arithmetical mean value of the reading of the four thermocouples embedded in the tube wall and the average saturation temperature, derived from the average pressure on refrigerant side.
The frictional pressure drop on the refrigerant side Δp_f is computed by subtracting the inlet / outlet local pressure drop Δp_c, and adding the momentum pressure recovery Δp_a from the total pressure drop measured Δp_t:

$$\Delta p_f = \Delta p_t - \Delta p_c + \Delta p_a \qquad (2)$$

Being the test section horizontal, no gravity pressure drops Δp_g occur.

3. ANALYSYS OF THE EXPERIMENTAL RESULTS

Three sets of condensation tests with refrigerant and water counter-flow were carried out at three different saturation temperatures (30, 35, and 40 °C) at decreasing vapour quality up to subcooled liquid condition with HFC134a, HFO1234yf, and HFO1234ze(E) refrigerant, respectively. A detailed error analysis performed in accordance with Kline and McClintock (1953) indicates an overall uncertainty within ±21.6%, ±22.3% and ±21.6% for the refrigerant heat transfer coefficient measurement and within ±24.1%, ±21.6% and ±19.7% for the total pressure drop measurement of HFC134a, HFO1234yf, and HFO1234ze(E), respectively.

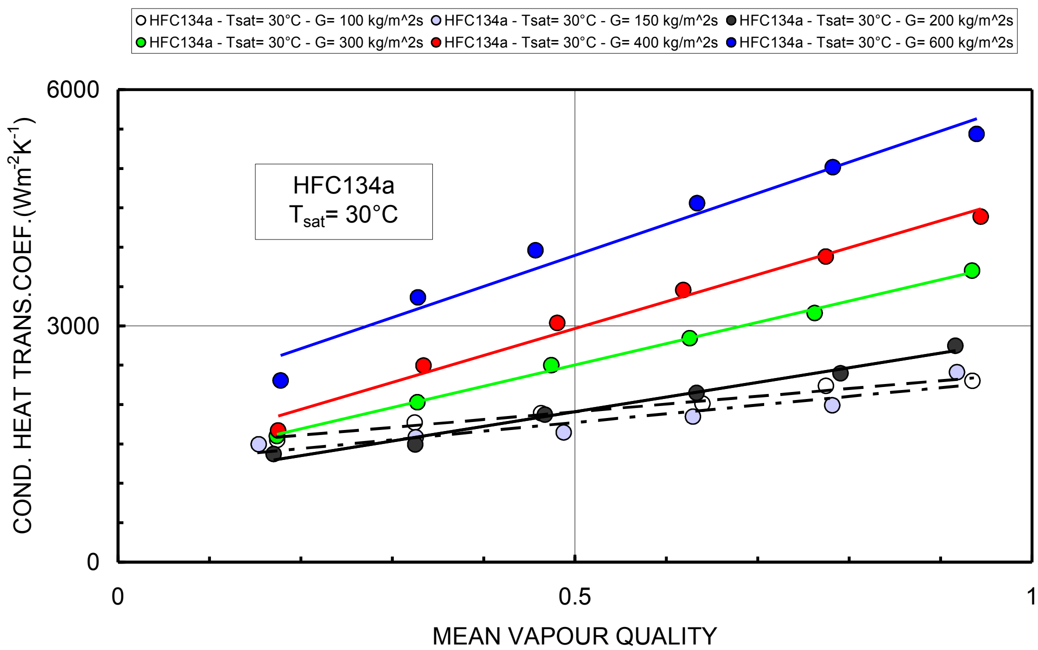

Figure 2a: Condensation heat transfer coefficient vs. mean vapour quality and mass flux at 30°C for HFC134a.

Figures 2a, 2b, and 2c show the condensation heat transfer coefficient h_r against mean vapour quality at 30 °C saturation temperature and six different refrigerant mass fluxes for HFC134a, HFO1234yf, and HFO1234ze(E), respectively. HFO1234yf and HFO1234ze(E) show condensation heat transfer coefficients very similar to those of HFC134a. The heat transfer coefficients show a positive slope versus vapour quality and the slope increases with refrigerant mass flux indicating a dominant effect of forced convection condensation mechanisms, except at the lowest refrigerant mass fluxes tested (100-200 kg m^{-2} s^{-1}), where condensation process is gravity controlled.

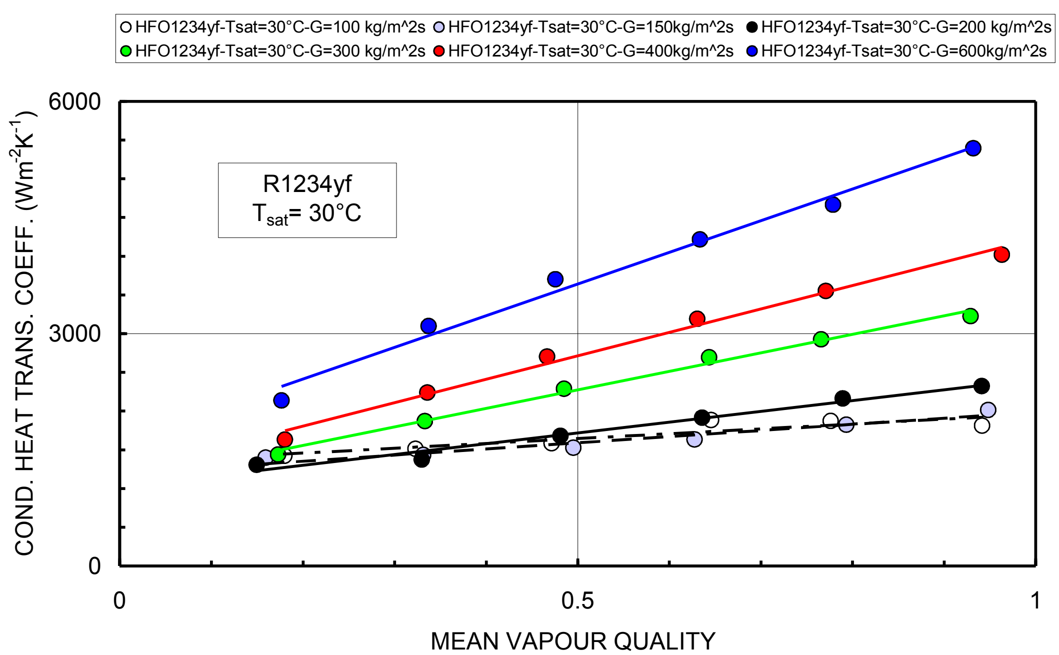

Figure 2b: Condensation heat transfer coefficient vs. mean vapour quality and mass flux at 30°C for HFO1234yf.

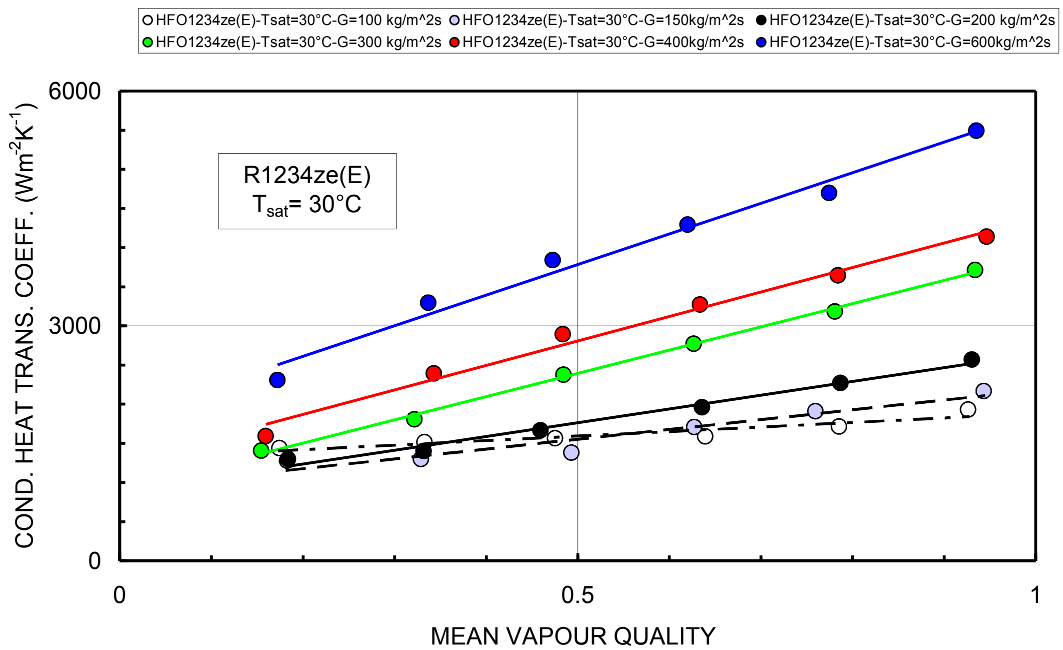

Figure 2c: Condensation heat transfer coefficient vs. mean vapour quality and mass flux at 30°C for HC1270.

For determining the dominant heat transfer regime, the experimental data points were plotted in figure 3 in non-dimensional co-ordinates giving the heat transfer factor $J_H = Nu_r/Pr_L^{1/3}$ versus the equivalent Reynolds number Re_{eq}. The transition from gravity-controlled and forced convection condensation occurs for an equivalent Reynolds number in the range 10,000 – 20,000 that is consistent with a refrigerant mass velocity of 150 – 300 kg m^{-2}s^{-1}.

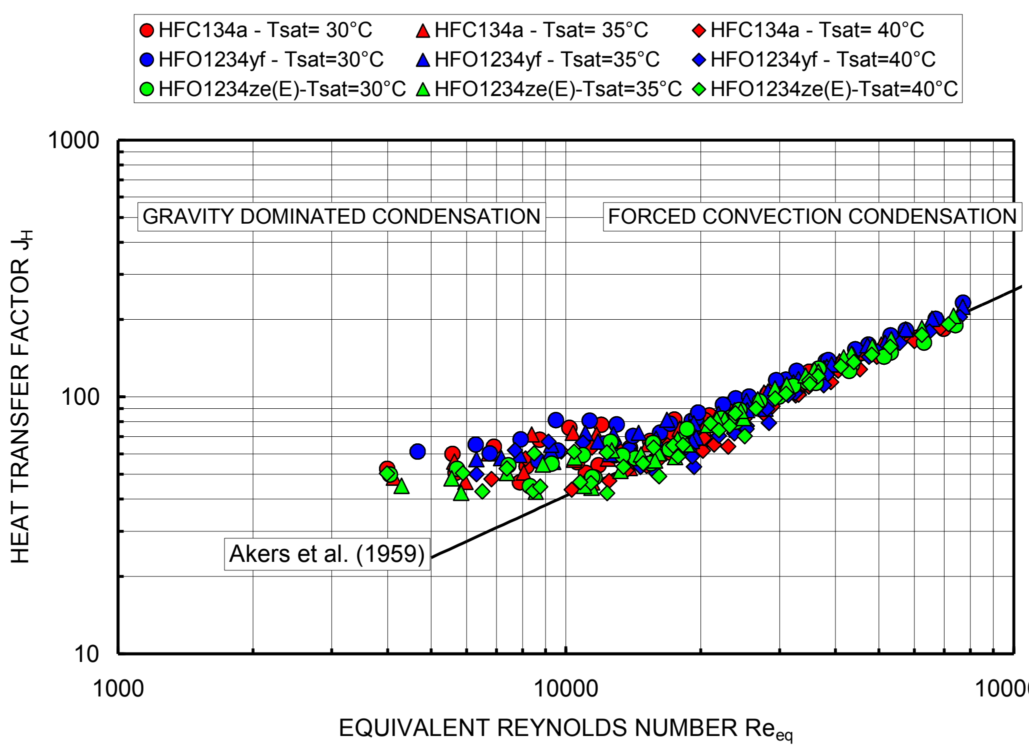

Figure 3: Experimental data plotted on the non-dimensional co-ordinates J_H vs. Re_{eq}.

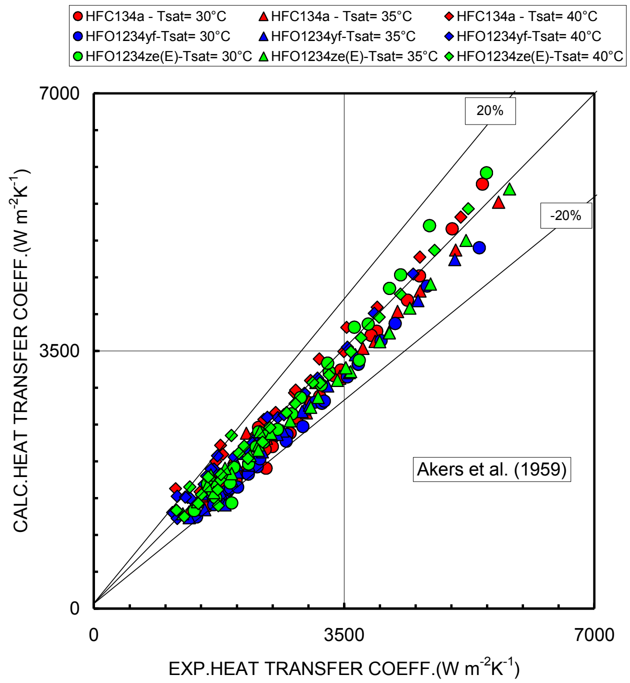

Figure 5: Comparison between experimental heat transfer coefficients and calculated by Akers et al. (1959) model.

The present experimental heat transfer coefficients in forced convection condensation were compared against different heat transfer correlations for condensation inside smooth tube: the classical Akers et al. (1959) equation shows the best performance in reproducing the experimental data with a mean absolute percentage deviation of 7.6%, 9.7%, and 5.8% for HFC134a, HFO1234yf, and HFO1234ze(E) data, respectively. Figure 4 shows the comparison between experimental and calculated data by Akers at al. (1959) model.

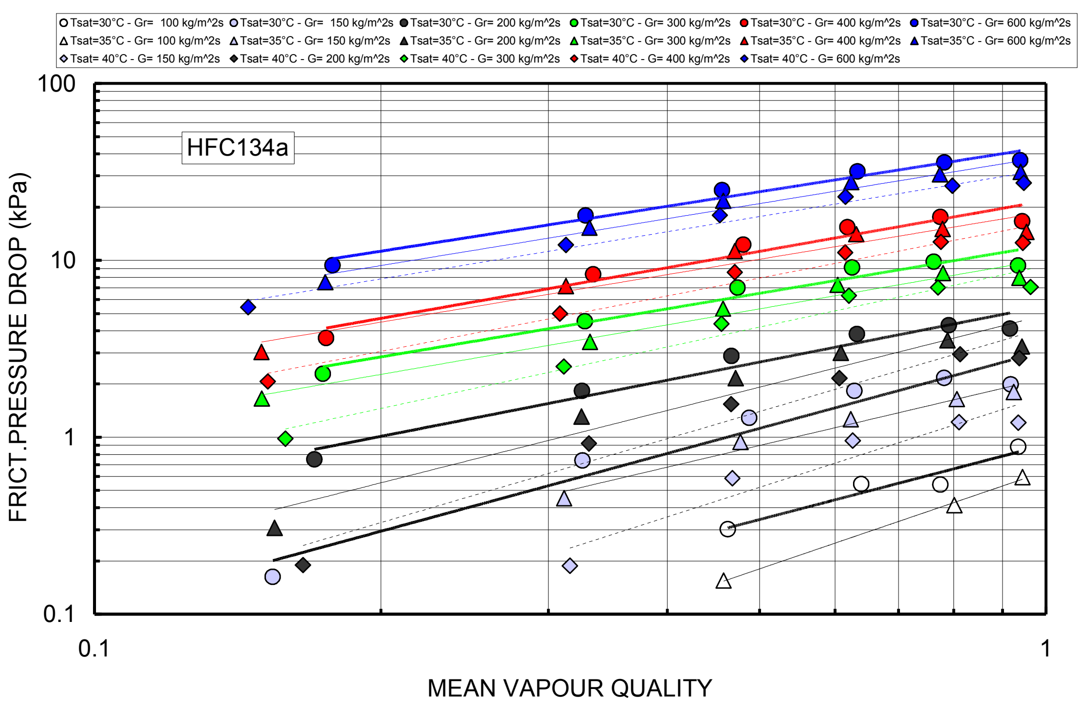

Figure 6a: Condensation frictional pressure drop vs. refrigerant mass flux for HFC134a.

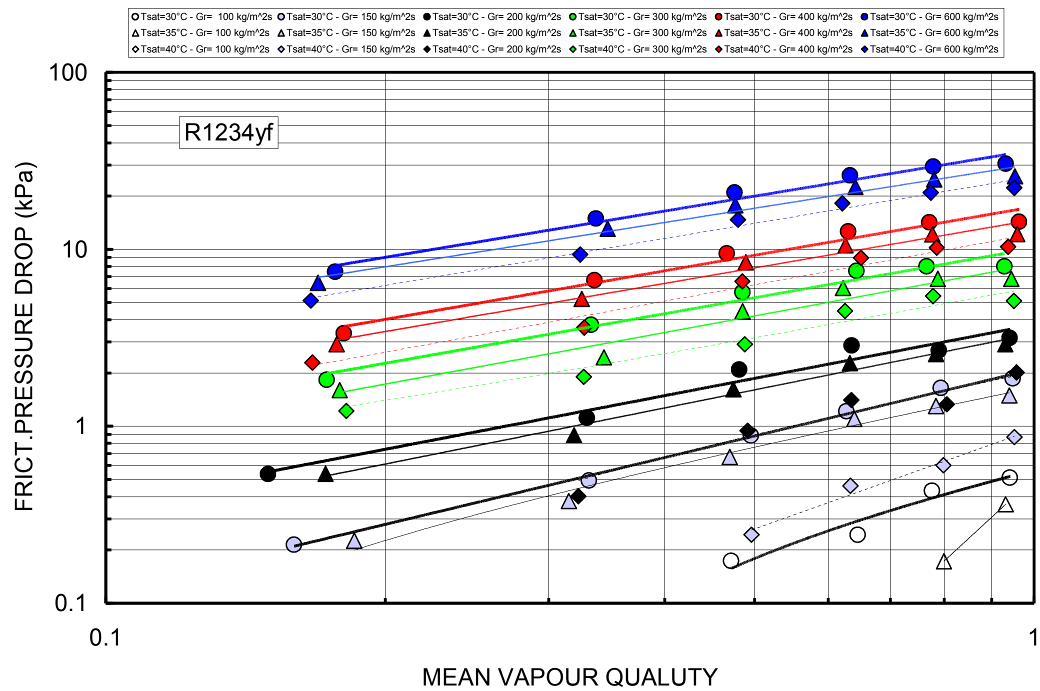

Figure 6b: Condensation frictional pressure drop vs. refrigerant mass flux for HFO1234yf**.**

Figures 6a, 6b, and 6c show the condensation frictional pressure drop against refrigerant mass flux at three different saturation temperatures (30°C, 35°C, and 40°C) for HFC134a, HFO1234yf, and HFO1234ze(E), respectively. HFO1234yf exhibits frictional pressure drops 5-15% lower than HFC134a and 15-30% lower than HFO1234ze(E) under the same operating conditions.

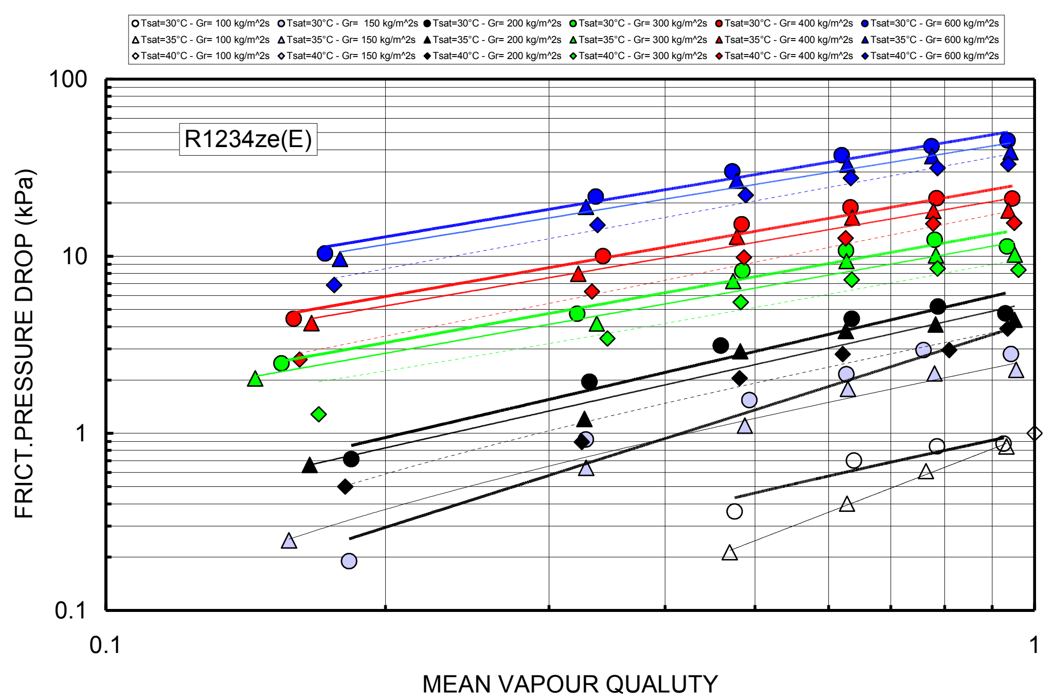

Figure 6c: Condensation frictional pressure drop vs. refrigerant mass flux for HFO1234ze(E)**.**

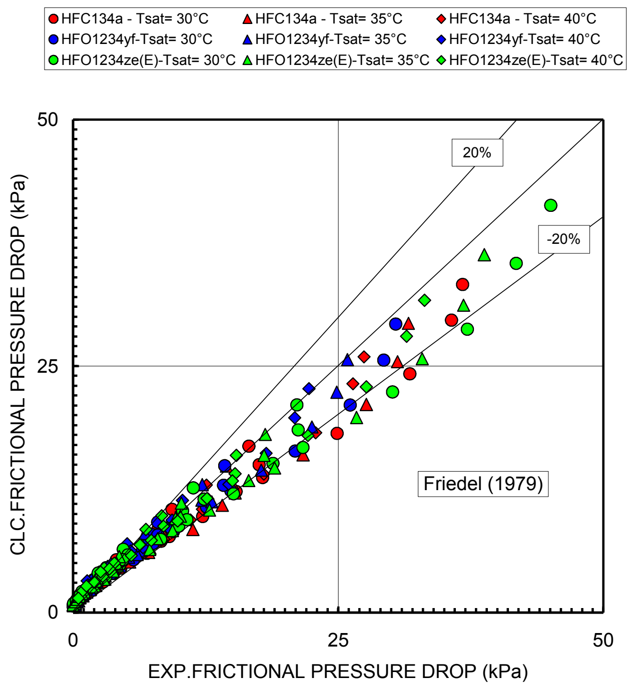

Figure 7: Comparison between experimental frictional pressure drop and calculated by Friedel (1979) equation.

Present experimental data points were compared against different correlations for two-phase pressure drop inside tube: Friedel (1979) correlation shows the best performance with a mean absolute percentage deviation of 14.2%, 10.1%, and 13.0% for HFC134a, HFO1234yf, and HFO1234ze(E), respectively. Figure 7 shows the comparison between experimental and calculated data by Friedel. (1979) equation.

4. CONCLUSIONS

This paper presents the comparative analysis of HFC134a and its low GWP substitutes HFO1234yf, and HFO1234ze(E) during saturated vapour condensation inside a 4 mm ID horizontal smooth tube: the effect of refrigerant mass flux, mean vapour quality, and saturation temperature (pressure) are evaluated. A transition point from gravity-dominated to forced convection condensation was found in the range of the equivalent Reynolds number 10,000 – 20,000. which corresponds, for the tested tube, the present operating conditions and the present refrigerants, to a refrigerant mass flux around 150-300 kg $m^{-2}s^{-1}$. The experimental heat transfer coefficients in the forced convection condensation regime were very well predicted by the Akers et al. (1959) model, whereas the Friedel (1979) correlation was able to reproduce the frictional pressure drop data in the whole experimental range. HFO1234yf and HFO1234ze(E) exhibit heat transfer coefficients and frictional pressure drops similar to those of HFC134a, therefore the present experimental measurements confirm that both HFO1234yf and HFO1234ze(E) are very promising as long-term low GWP substitutes for HFC134a.

NOMENCLATURE

A	heat transfer area	(m^2)
G	refrigerant mass flux	(kg $m^{-2}s^{-1}$)
h	heat transfer coefficient	(W m^{-2} K^{-1})
J_H	heat transfer factor	(–)
k	coverage factor	(–)
Nu	Nusselt number	(–)
p	pressure	(Pa)
Pr	Prandtl number	(–)

q	heat flux	(Wm^{-2})
Q	heat flow rate	(W)
Re	Reynolds number	(–)
T	Temperature	(K, °C)
X	Vapour quality	(–)
Δ	Difference	

Subscript

a	momentum
c	local
eq	equivalent
f	frictional
g	gravity
L	liquid phase
t	total

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