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Experimental results on boiling heat transfer coefficient, frictional pressure drop and flow patterns for R134a at a saturation temperature of 34 °C

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A B S T R A C T

The current study presents heat transfer and pressure drop data for R134a at a saturation temperature of approx. 34 °C. Study cases have been set for two different mass fluxes and heat fluxes. The flow patterns have also been characterized by means of a fast speed camera and a visualization section between the heat transfer and pressure drop measurements. Comparing with the available results in the open literature for lower saturation temperatures it can be seen that the heat transfer coefficient has a more planar profile as the mass flux decreases and the saturation temperature and the heat flux increase. For the frictional pressure drop the studied cases show a maximum around a quality of 0.85, which seems not to be influenced by the mass flux. Finally, the flow patterns observed show a good agreement with the K–T–F flow pattern map (N. Kattan, D. Favrat, J. R. Thome, Flow boiling in horizontal tubes. Part 1: Development of a diabatic two-phase flow pattern map, Journal of Heat Transfer 120(1):140–147, 1998a).

Résultats expérimentaux sur le coefficient de transfert de chaleur en ébullition, la chute de pression due au frottement et les schémas d'écoulement pour le R134a à une température de saturation de 34 °C

Mots clés : R134a ; Ebullition ; Coefficient de transfert de chaleur ; Chute de pression due au frottement ; Schémas d'écoulement

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1. Introduction

Many different working fluids for refrigeration applications have been investigated in the last years due to new regulations regarding environmental issues. Among these new widely used working fluids, R134a represents the largest demand in applications of domestic, commercial and industrial refrigeration (Billiard, 2003, UNEP, 2006).

The accuracy in the prediction of the boiling heat transfer coefficient for R134a is important for the modeling of the phenomenon, especially in transition regions such as subcooled boiling and dryout due to the possible development of high temperature gradients in the heated pipe walls (Manavela Chiapero et al., 2012). During the last years, several experimental studies have been carried out for the characterization of the two-phase heat transfer coefficient and the frictional pressure drop in pipes (Akhavan-Behabadi et al., 2009, Greco and Vanoli, 2005, Ribatski and Thome, 2006, de Rossi and Mauro, 2009, da Silva Lima et al., 2009, Wongwises et al., 2000, Zhang and Webb, 2001, Zhang et al., 1997). Most of the experimental studies dealt with saturation temperatures below 25 °C, except Greco and Vanoli (2005) with one test at 28 °C. At higher saturation temperatures (36, 42 and 48 °C) Al-Hajeri et al., (2007) measured average heat transfer coefficients for condensing R134a through annular helical tubes. Huo et al., (2004) and Shiferaw et al., (2007) made a broad experimental analysis comparing existing correlations with their experiments on R134a boiling heat transfer in the limit between conventional channels and minichannels. Two test sections inner diameters were used, 2.01 mm and 4.26 mm, with pressures varying from 8 to 12 bar, mass fluxes from 100 to 500 kg m^{-2} s^{-1} and heat fluxes from 13 to 150 kW m^{-2}. Nevertheless, for the lowest heat flux values, below 30 kW m^{-2}, no heat transfer coefficient values were reported for vapor qualities higher than 0.2. Regarding two-phase frictional pressure drop, Ould Didi et al. (2002) presented an extensive data base for five refrigerants including R134a. These experimental points were compared against seven of the most-quoted methods available in the literature, finding the method of Grönnerud (1979) as the most accurate one. The tests for R134a were performed at saturation temperatures between 0 and 10 °C. Raush et al., (2009) analyzed the heat transfer coefficient and the diabatic pressure drop for R134a in an electrically-heated, 8.15 mm inner-diameter pipe, for saturation temperatures ranging from −24 to 8 °C and compared the experimental data with numerical simulations of two-phase flow evaporation models. For those test cases, the Kattan et al., (1998b) correlation showed the best performance describing the complete evaporation process at mass fluxes higher than 100 kg m^{-2} s^{-1}. Even though not many applications deal with boiling R134a at a working pressure higher than 8 bar, a validation of the most widely used correlations for pressure drop and heat transfer is needed. For example, it is a common practice during design in research and development activities to scale down equipments or processes by means of similarity and dimensional analysis theory. These types of situations sometimes force researchers to use working fluids outside its usual application range, and thus, to use correlations that have not been tested at such conditions. Therefore, it is important to expand the experimental data bases available in the literature.

In this study, the heat transfer coefficient and frictional pressure drop in a 5 mm smooth horizontal stainless steel pipe are analyzed. The tests are performed for a saturation temperature of approx. 34 °C (pressure of 8.6 bar), range where there is not much available data. The mass flux and the heat flux are varied from 300 to 500 kg m^{-2} s^{-1} and from 10 to 20 kW m^{-2} respectively, covering the whole span on quality from subcooled boiling to dryout. Pictures of the different flow patterns are recorded with a high speed camera, which helps understanding the acquired data and contributes to the flow pattern maps development.

2. Description of the facility

The goal of the present study is to run in-tube boiling cases under different experimental conditions in order to characterize the heat transfer coefficient and the frictional pressure drop for refrigerant R134a at high saturation temperatures (above 25 °C) where the data available in the literature is very limited. The experimental facility (shown in Fig. 1) is a R134a loop consisting of a main tank, a pump, a pre-heater or conditioner, a heated test section, a sight glass (Fig. 2), an adiabatic test section and a condenser. The loop is schematically represented in Fig. 3. The fluid pressure is set by controlling the temperature in the main tank where the refrigerant is at saturation conditions. To assure that the fluid in the tank is at saturated conditions the system is left for 1 h at the desired working pressure before starting the experiments with gas and liquid present in the tank. However, equilibrium conditions in the main tank are not a need as long as the pressure in the tank is kept constant, since the flow conditions at the inlet of the heated section are always subcooled liquid (fluid temperature and pressure are registered at the inlet of the heated section). The fluid is driven by a magnetically coupled gear pump. The conditioner is a shell and tube heat exchanger with glycol in the shell side which is used for adjusting the R134a inlet temperature. Before entering the heated section the refrigerant flows through a Coriolis mass flow meter. The heated stainless steel test section (Fig. 4) is electrically heated by Joule effect with a rectified sine wave. The heating is splitted into 5 different sections, where the power applied can be specified independently. Each

Fig. 1 – Experimental facility.