International Journal of Multiphase Flow 80 (2016) 1–14

Contents lists available at ScienceDirect

International Journal of Multiphase Flow

journal homepage: www.elsevier.com/locate/ijmultiphaseflow

The influence of inclination angle on void fraction and heat transfer during condensation inside a smooth tube

Stefan P. Olivier[a], Josua P. Meyer[a,*], Michel De Paepe[b], Kathleen De Kerpel[b]

[a] Department of Mechanical and Aeronautical Engineering, University of Pretoria, Private Bag X20, Hatfield 0028, Pretoria, South Africa
[b] Department of Flow, Heat and Combustion Mechanics, Ghent University – UGent, Sint-Pietersnieuwstraat 41, 9000 Gent. Oost-Vlaanderen, Belgium

ARTICLE INFO

Article history:
Received 29 May 2015
Revised 29 September 2015
Accepted 13 October 2015
Available online 1 December 2015

Keywords:
Void fraction
Two-phase flow
Capacitance
Flow pattern
Condensation
Inclination angle
Heat transfer coefficient

ABSTRACT

Most work in literature on condensation in tubes has been done for smooth tubes in the horizontal and vertical configurations. Recent experimental works with condensation at different inclination angles showed that the heat transfer characteristics were a function of inclination angle. These works were limited to heat transfer and pressure drop measurements with visual observations. However, no work has been done on measuring the void fractions during condensation at different inclination angles. The purpose of this study was to measure void fractions and heat transfer coefficients during condensation for tube inclinations ranging from vertical downwards (−90°) to vertical upwards (90°) at a saturation temperature of 40 °C. Measurements were taken in an experimental set-up in which condensation occurred on the inside of a test section. The test section was a circular tube with an inner diameter of 8.38 mm and a heat transfer length of 1.488. The refrigerant used was R134a, and the void fractions were measured using two capacitive void fraction sensors. Mass fluxes ranging from 100 to 400 kg/m^2 s and vapour qualities ranging from 10–90% were considered. Heat transfer coefficients were also compared with void fractions. It was found that at combinations of low mass fluxes and vapour qualities, void fraction and heat transfer were significantly affected by changes in inclination angle. Generally, void fractions and heat transfer coefficients increased with downward inclination angles with an optimum angle between −10° and −30° (downward flow). At some intermediate mass flux and vapour quality conditions, the void fraction and heat transfer coefficients were observed to be independent of the inclination angle despite changes in the prevailing flow patterns.

1. Introduction

Many systems in the air-conditioning and refrigeration industries where cooling and/or heating is required make use of vapour-compression systems. The condenser in the vapour compression system needs to reject heat by condensing the refrigerant from a gas to a liquid.

Other industries where condensation occurs are in the power generation industries where steam is condensed in air and water cooled cooling towers. In the majority of cases condensation occurs in tubes which are configured horizontally. An exception is steam condensation in large air cooled condensers used in the power generation industry where water is not readily available as a cooling medium. These condensers are constructed in an A-frame structure configured to provide condensing flow at an approximate 60° downward inclination. However, no work has been published in the open literature that justifies this angle. It is therefore not clear if this angle is based on heat transfer and/or structural considerations.

Other applications where condensation occurs in inclined tubes are steam condensers used for air-cooling, and also in some rooftop industrial air-cooled refrigeration systems. Again, no work has been published in the open literature which justifies the angles being used.

A recent review of the state of the art in condensation flow was conducted and the conclusion was drawn that the majority of work was done for horizontal tubes and a limited amount in vertical tubes (Lips and Meyer, 2011). However, very little work was done for condensation in inclined tubes. Past work have highlighted the importance of the identification of the prevailing flow pattern since it is used in the development of equations that estimate the local and average heat transfer and pressure drop coefficients during condensation (Da Riva et al., 2012; Doretti et al., 2013; Nebuloni and Thome, 2013; Thome et al., 2013).

Past work on the subject of condensation in inclined tubes found that downward inclination angles could have significant advantages since gravity assisted in thinning the liquid layer inside the tube and reduce the thermal resistance to heat transfer (Lips and Meyer, 2012b).

An increase in heat transfer of up to 20% at an inclination of −15° (downward flow) was observed for combinations of low mass flux

* Corresponding author. Tel.: +27124203104; fax: +274206632.
E-mail address: josua.meyer@up.ac.za (J.P. Meyer).

http://dx.doi.org/10.1016/j.ijmultiphaseflow.2015.10.015
0301-9322/

and vapour quality. The heat transfer decreased with upward flow inclinations. At higher mass flux and vapour quality conditions, shear forces were dominant, which meant that the flow patterns tended to remain annular with the heat transfer coefficients exhibiting independence of the tube inclination angle. Also, the observed flow patterns did not correlate well with the considered models for inclined flow.

From the work of Lips and Meyer (2012c) for horizontal pressure drops, the model of Moreno Quibén and Thome (2007) along with the El Hajal et al. (2003) flow pattern map best represented the experimental results. Void fraction models were chosen by Lips and Meyer (2012c) in order to use pressure drop correlations for comparison with experimental results. The void fraction correlations considered in their study led to a ±25% agreement between predicted pressure drops and measured pressure drops for upward flows. The correlations failed to predict pressure drops for downward flows. The apparent gravitational pressure drop (which was defined as the difference between the pressure drops for inclined tubes versus horizontal tubes) was used to estimate the apparent void fraction. The apparent void fraction was defined as the void fraction value which would have led to the aforementioned apparent gravitational pressure drop. From the results, it appeared that the void fraction remained constant for upward flows. For downward flows, the apparent void fraction could not be considered to represent the actual void fraction since the frictional pressure drop was dependent on the inclination angle.

At lower mass fluxes (i.e. 100–200 kg/m^2 s), the heat transfer coefficients were observed to exhibit a maximum in the region of −30° to −15° (downward flow) due to gravity thinning of the stratified liquid layer (Meyer et al., 2014), which led to less thermal resistance. The heat transfer coefficients were observed to reduce with increased saturation temperature for all inclination angles and mass fluxes. The reason provided was that the thermal conductivity of the particular refrigerant (R134a) decreased with increasing temperature meaning a reduction in heat transfer coefficient.

However, what has not yet been looked into is the determination of the void fractions and their influence on heat transfer coefficients in tubes at different inclination angles. The knowledge of the void fractions is important in the developing of accurate heat transfer coefficient equations, which has also not yet been done in inclined tubes.

The challenge is that the heat transfer and pressure drop during condensation in a tube are directly related to the temporal and spatial distribution of the liquid and vapour phases. These phases are directly related to the void fraction and a considerable number of measurements have been taken to relate the void fraction to heat transfer and pressure drop equations during condensation (Adelaja et al., 2014a, 2014b; Meyer et al., 2014). However, no work has been done to determine the void fraction and its influence on heat transfer coefficients during condensation in inclined tubes. In addition, no work has been done on the development of new and/or revision of existing equations, which takes into consideration the changes that occur in void fraction as a function of the inclination angle.

Accurate void fraction measurements are challenging, and in most cases where void fractions were measured it was done using only one sensor for the entire test section (Ahmed, 2011; Da Silva et al., 2010; De Kerpel et al., 2013; Demori et al., 2010). This is perhaps not an accurate representation of the average void fraction of a test section that changes between the inlet and outlet. As the void fractions and heat transfer correlations are so strongly related there is a need for accurate void fraction measurements, however, instrumentation that can measure void fractions without influencing the flow patterns are not readily available commercially. Such a void fraction sensor has been developed at the Ghent University (Canière et al., 2010, 2009, 2008, 2007), but is not yet commercially available on the open market.

The purpose of this study was to measure void fractions with two void fraction sensors developed at Ghent University, capture prevailing flow patterns visually and measure the heat transfer coefficients over a wide range of inclination angles during condensation in a smooth tube. The effect of the void fractions on the flow patterns and heat transfer coefficients was also investigated qualitatively.

2. Experimental apparatus and test conditions

2.1. Experimental set-up

The experimental system (Fig. 1) was inherited from past research work in which detailed descriptions and explanations have been provided (Adelaja et al., 2014a, 2014b; Canière et al., 2007; Lips and Meyer, 2012a, 2012b, 2012c; Meyer et al., 2014; Suliman et al., 2009; Van Rooyen et al., 2010). Therefore, only a brief overview will be provided in this section.

The experimental set-up consisted of a vapour compression refrigeration cycle operated with refrigerant R134a, as well as a water cycle. The refrigeration cycle consisted of two high-pressure lines, i.e. the test line and bypass line, through which the condensing fluid was pumped. Fluid flow was obtained using a hermetically sealed scroll compressor with a nominal cooling capacity of 10 kW. The flow of refrigerant through each respective line was controlled using a set of electronic expansion valves (EEVs).

The test line was served by three respective condensers. A pre-condenser was responsible for controlling the vapour quality at the inlet of the test condenser where temperature, pressure, flow pattern and void fraction measurements were taken. The void fraction measurements were made using sensors which measured the capacitance of the flow with an accuracy of $\pm 4 \times 10^{-15}$ F. The absolute uncertainty in the void fraction measurements was estimated by the sensors' developers to be 0.02. Two sensors were used, one at the inlet of the test section and the other at the outlet of the test section. A cross-sectional illustration of the void fraction sensor construction is provided in Fig. 2. The internal diameter of the void fraction sensor was 8 mm, the inscribed electrode angle (β) was 160°, the tube wall thickness was 50 μm, and the axial length of each electrode was 8 mm, i.e. one tube diameter. The average of the two measurements was taken as the average void faction in the test section.

A post-condenser ensured that the refrigerant reached the EEVs in a liquid state to ensure accurate mass flow measurements. The bypass condenser controlled the mass flow rate, saturation temperature as well as saturation pressure of the refrigerant in the test section. The bypass condenser was also adjusted so only liquid refrigerant reached the EEV. Downstream of the respective EEVs, the test and bypass lines merged into a single lower-pressure line, which led to the suction of the scroll compressor through an evaporator. A suction accumulator ensured that only vapour was available at the compressor suction.

Hot and cold water was supplied by a 50 kW heating and 70 kW cooling dual-function heat pump. Thermostat control was used to maintain the hot water at 25 °C–30 °C and the cold water at 15 °C–25 °C respectively in two tanks of 5000 litre capacity each. The hot water was used in the evaporator while the cold water was used in the various condensers mentioned previously.

The test condenser was constructed as a tube-in-tube counter-flow heat exchanger with refrigerant flowing in the inner tube and water flowing in the annulus. The inner tube was constructed from an 8.38 mm inner diameter (0.6 ±0.002 mm wall thickness) smooth, hard-drawn copper tube inside an outer tube from 15.9 mm inner diameter smooth, hard-drawn copper tube. This tube diameter was selected for this study as it was used in several previous studies (Adelaja et al., 2014a, 2014b; Lips and Meyer, 2012b, 2012c; Meyer et al., 2014) and therefore better comparisons are possible.

The refrigerant stream flowed through the inner tube and the water stream through the annulus in a counter flow direction. A wire with an outside diameter of 2 mm was wrapped around the outside of the inner tube at a constant pitch to improve water mixing within