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Evaluation of the heat transfer enhancement and pressure drop penalty during flow boiling inside tubes containing twisted tape insert

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HIGHLIGHTS

- Pressure drop and heat transfer coefficient for R134a two-phase flow in tubes with twisted-tape inserts.
- Comparison of experimental results against predictive methods available in the literature.
- An analysis of the pressure drop penalty by using twisted tape is performed.
- Overall performance enhancement by using twisted tape were analyzed.

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ABSTRACT

This paper presents experimental data for pressure drop and heat transfer coefficient for two-phase flows in tubes containing twisted-tape inserts. Based on these data, an analysis of the heat transfer enhancement and pressure drop penalty due to the use of twisted-tapes is performed. Experimental results were obtained for horizontal 12.7 and 15.9 mm ID tubes without and with twisted-tape for twist-ratios of 3, 4, 9 and 14. The experiments were performed for R134a as working fluid covering mass velocities from 75 to 200 kg/m^2 s, heat fluxes of 5 and 10 kW/m^2 and saturation temperatures of 5 and 15 °C. The predictive methods available in the literature were compared against the experimental data obtained in the present study. In general, it was concluded that the use of reduced twist-ratios values provide higher overall heat transfer enhancement factors for intermediary vapor quality. Additionally, it was found that the pressure drop correlation proposed by Kanizawa and Ribatski (2012) is suitable to be used as a design tool because it predicted 99% of the present database within an error band of ±30%.

1. Introduction

According to Manglik and Bergles [1], twisted-tape insert has been used since 1896 as a heat transfer enhancement technique. Bergles [2] defined twisted-tape insert as a passive heat transfer enhancement technique because it does not require any external energy source to promote the enhancement of the heat transfer coefficient. The twisted-tape insert, schematically shown in Fig. 1, is geometrically characterized by the twist ratio, defined as the ratio between 180° turn length H and the internal diameter d as follows:

$$y = \frac{H}{d} \tag{1}$$

Unfortunately, the heat transfer augmentation provided by the twisted-tape is accompanied by pressure drop penalty. The pressure drop increases due to the cross section restriction by the tape and the promotion of secondary flows. In general, the twisted-tape presents the following advantages: (i) low manufacturing costs; (ii) easy installation and maintenance; (iii) the possibility of being used to retrofit heat exchangers already in use as mentioned by Thome and Ribatski [3]. The overall system efficiency is affected by the increment of the pressure drop and the heat transfer coefficient, consequently, both parameters must be evaluated accordingly in

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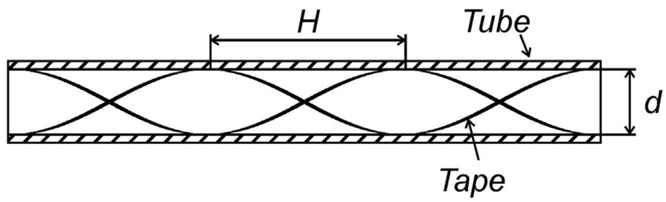

Fig. 1. Schematic view of twisted tape insert in a tube.

order to verify if the use of this technique is advantageous to the overall system performance.

The increase of pressure drop can be overcome by heat transfer augmentation, since through the use of twisted-tape the heat exchanger size can be reduced keeping the same heat transfer capacity. According to Agarwal and Raja Rao [4], heat transfer augmentation factors higher than 20% can be obtained for the same pumping power through the use of twisted-tape inserts during single-phase flow. For flow boiling of R12, Agrawal and Varma [5] have pointed out heat transfer coefficient augmentations up to 235% for the same pumping power through the use of twist tape inserts. The use of swirl devices during convective boiling seems more advantageous under conditions that the twist-tape may induce an earlier transition from stratified to annular flow. This last flow pattern provides higher heat transfer coefficient than stratified flows.

Recently, Kanizawa and Ribatski [6] presented an experimental study focusing on pressure drop and two-phase flow patterns in horizontal tubes containing twisted-tape inserts. The authors also proposed a correlation to predict frictional pressure drop based on their data for R134a flow in horizontal 15.9 mm ID tube, for twist ratios y between 3 and 14, mass velocities G from 75 to 250 kg/m^2 s, and saturation temperatures T_{sat} of 5 and 15 °C. Kanizawa and Ribatski [6] have pointed out the following flow patterns: *stagnant flow*, verified for reduced vapor qualities, mass velocities and twist ratios and that is characterized by stagnant portions of liquid in the bottom region of the cross section, periodically propelled by liquid slugs; *intermittent flow*, verified for intermediary vapor qualities and twist ratios and that is characterized by periodical passage of liquid slugs without stagnant portion of liquid in the lower region of the cross section; *stratified flow*, verified for higher twist ratios and reduced mass velocities. This pattern is similar to the stratified flow in plain tubes, with the liquid phase in the lower region of the section and vapor phase flowing above it; *annular-stratified flow*, observed for intermediary mass velocities and vapor qualities and that is characterized by a continuum liquid film in the tube wall thicker close to the tape edges; and *annular flow*, verified for higher vapor qualities and mass velocities and that is characterized by uniform liquid film along the tube perimeter.

In the present study, new experimental results for pressure drop and heat transfer coefficient during flow boiling of R134a are presented. These data were carefully analyzed and compared against predictive methods from literature. An analysis of the pressure drop penalty and heat transfer enhancement is also performed.

2. Experimental apparatus

This section aims to present the experimental facility, the data regression, procedure and the validation of the experimental apparatus based on single-phase flow experiments. Kanizawa and Ribatski [6] and Mogaji et al. [7] have also presented descriptions of the experimental facility, and these studies are indicated as supplementary literature.

2.1. Experimental loop

The experimental setup is comprised of refrigerant and ethylene-glycol/water circuits, named also as primary and secondary circuits, respectively. Fig. 2 presents schematically the refrigerant loop that corresponds to a closed loop. The secondary circuit, partially shown in Fig. 2, is responsible for the cooling effects in the condenser and in the sub-cooler. This circuit contains a vapor compression refrigeration cycle responsible for cooling the mixture of 60% of ethylene glycol in water. This anti-freezing solution is driven by a centrifugal pump through the condenser and the sub-cooler. The refrigeration cycle rejects heat to the external environment through an evaporative cooling tower.

The condenser is a shell-and-tube heat exchanger with refrigerant flowing in the shell side. The flow rate of anti-freezing solution through the condenser is adjusted with the help of a needle valve, as shown in Fig. 2. The sub-cooler is a tube-in-tube heat exchanger type with the test fluid flowing in the internal tube. The sub-cooler is used to ensure that the fluid is sub-cooled at the pre-heater inlet. Thus, the thermodynamic state of the fluid at the pre-heater inlet is estimated based on local temperature and pressure measurements. The temperature at the sub-cooler outlet and pre-heater inlet is measured with a thermocouple (0.120 mm wire diameter).

The pre-heater is used to control and adjust the thermodynamic state, and consequently the vapor quality, at the test section inlet. It consists of a 12.7 mm ID Copper tube, wrapped with 20 electrical resistances that totalize 12 kW. The electrical powers supplied to the pre-heater and test sections were measured using YOKOGAWA 2285A power transducers. The system is thermally insulated with ceramic fiber and elastomeric foam in order to minimize heat exchange with the environment. At the pre-heater outlet and stabilization section inlet, a thermocouple is used to determine the local temperature.

The stabilization section consists of a horizontal 1.4 m long tube of the same inner and outer diameters and material of the test section, and is used to guarantee that the flow at the test section inlet is fully developed. For the setup with 15.9 mm ID tube, this length corresponds to approximately 90 diameters, and for 12.7 mm ID tube it corresponds to 110 diameters. Both values are considered sufficient for the two-phase flow development. The stabilization section and the pre-heater are contained in the same thermal-insulation device separated by a wide ceramic fiber layer.

The test sections consist of horizontal 2.0 m long tubes, wrapped with electrical resistances that totalize 3.0 kW, insulated with ceramic fiber and elastomeric foam. The test sections characteristics are given in Table 1. The first test section has an internal diameter of 15.9 mm (5/8 in.) and is made of Copper. The second one has an internal diameter of 12.7 mm (1/2 in.) and is made of Brass. Both test sections present thick walls with thickness of 3.2 mm (1/8 in.). Table 1 presents the geometric characteristics of each tested tube and the corresponding visualization sections. The test section wall temperature is measured at four equally distant cross sections as depicted in Fig. 3. At each cross section, four thermocouples are spread apart by 90° along the tube perimeter. These thermocouples are nested in longitudinal grooves 0.5 mm distant from the internal tube surface. These grooves are filled with highly-conductive epoxy.

The visualization sections are installed in order to visually determine the flow patterns at the test section inlet and outlet. They are borosilicate tubes with internal diameter of 14.1 and of 16.4 mm used with the test sections of internal diameters of 12.7 and 15.9 mm, respectively.

The twisted tape inserts are made from an aluminum foil 1.0 mm thick and 2.0 m long, and are manufactured as suggested by