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Experimental Study of Evaporation Heat Transfer of R-134a Inside a Corrugated Tube with Different Tube Inclinations

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Abstract

Experimental heat transfer studies during evaporation of R-134a inside a corrugated tube have been carried out. The corrugated tube has been provided with different tube inclination angles of the direction of fluid flow from horizontal, α. The experiments were performed for seven different tube inclinations, α, in a range of - 90° to + 90° and four mass velocities of 46, 81, 110 and 136 kg/m2s for each tube inclination angle during evaporation of R-134a. Data analysis demonstrate that the tube inclination angle, α, affects the boiling heat transfer coefficient in a significant manner. The effect of tube inclination angleα , on heat transfer coefficient, h, is more prominent at low vapor quality and mass velocity. In the low vapor quality region the heat transfer coefficient, h, for +90° inclined tube is about 62% more than that of the - 90° inclined tube. The results also showed that at all mass velocities, the highest average heat transfer coefficient were achieved for α =+90°. An empirical correlation has also been developed to predict the heat transfer coefficient during flow boiling inside a corrugated tube with different tube inclinations.

Keywords

Corrugated Tube, Evaporation, Heat Transfer, Inclination, Two Phase Flow

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1. Introduction

As the energy sources are limited and for their conservation, appropriate design and optimization of evaporators is very important. Therefore, the design of efficient evaporators has always been significant to equipment designers. In this regard, the use of augmentative techniques, either active or passive, to increase the convective heat transfer rates on tube side has been studied for quite some time (Agrawal &Varma 1991, Akhavan-Behabadi & Kumar 2009, Wongwises & Polsongkram 2005). Evaporator is an important and widely used heat exchanger in air conditioning and refrigeration industries.

One of the passive techniques to enhance heat transfer coefficient is the application of corrugated tubes. Corrugated tubes have corrugation on their surface that can increase heat transfer through creating rotational flow and limiting the growth of thermal boundary layer. This enhancement in heat transfer is also accompanied by slight increase in flow friction factor (Lohalertdecha & Dalkilic & Wongwises 2011, Lohalertdecha & Wongwises 2011). These kinds of tubes are better heat transfer devices compared to finned tubes, sand-grain textures, wire-coil inserted tubes and transverse rib roughened tubes due to the following reasons: a) ease of construction, b) limited fouling and c) considerable increase in heat transfer, while having a slight impact on pressure drop (Zimparov & Vulchanov 1991).

There have been many studies concerning heat transfer and pressure drop of single-phase flow in corrugated tube. As for two-phase flows, Targanski and Cieslinski (2007),

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Laohalertdecha and Wongwises (2010, 2011), Aroonrat and Wongwises (2011), Withers and Young (1971), Akhavan-behabadi and khoeini (2011), Mohseni Garakani (2014), F. T. Kanizawa *et al.* (2014) and SH. Zhang (2014) investigated heat transfer, pressure drop, flow pattern effect of corrugation pitch, effect of corrugation depths and other geometrical parameters of corrugated tubes during evaporation of refrigerants inside corrugated tubes and reported the increase in heat transfer and pressure drop as compared with smooth tube.

The heat transfer coefficient generally keeps changing as the flow pattern changes along an evaporator tube. On the other hand, the flow regime is influenced by interfacial shear stress, buoyancy, surface tension and gravitational force. A review of the existing literature reveals that, although vast studies have been done on heat transfer enhancement in these tubes, yet the focus of almost all of the studies is on two phase refrigerant flow only in horizontal and vertical (downward flow) tubes. Thus there is a great necessity to consider and study the effect of magnitude and direction of gravity field on evaporation flow inside corrugated tube.

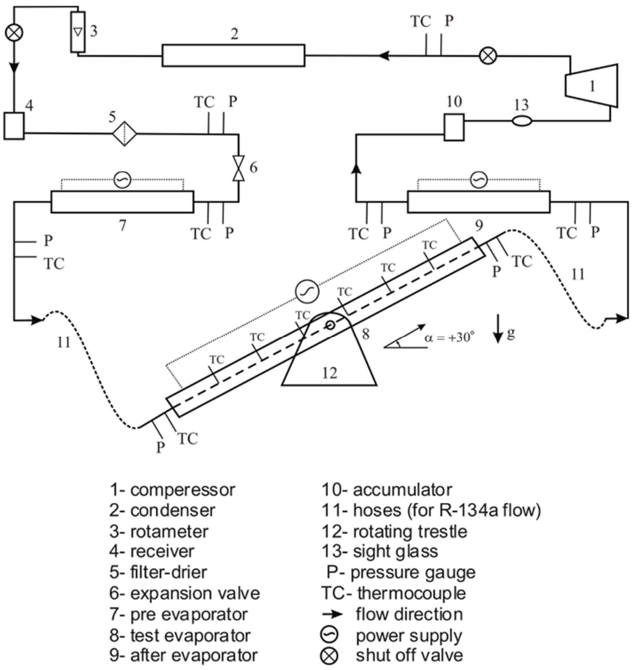
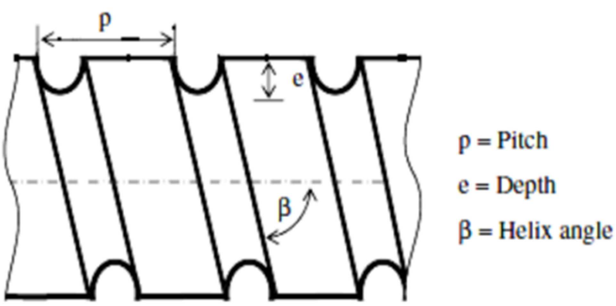


Figure 1. Schematic diagram of test apparatus with vapor quality at refrigerant mass velocity of 136 kg m^{-2} s^{-1}.



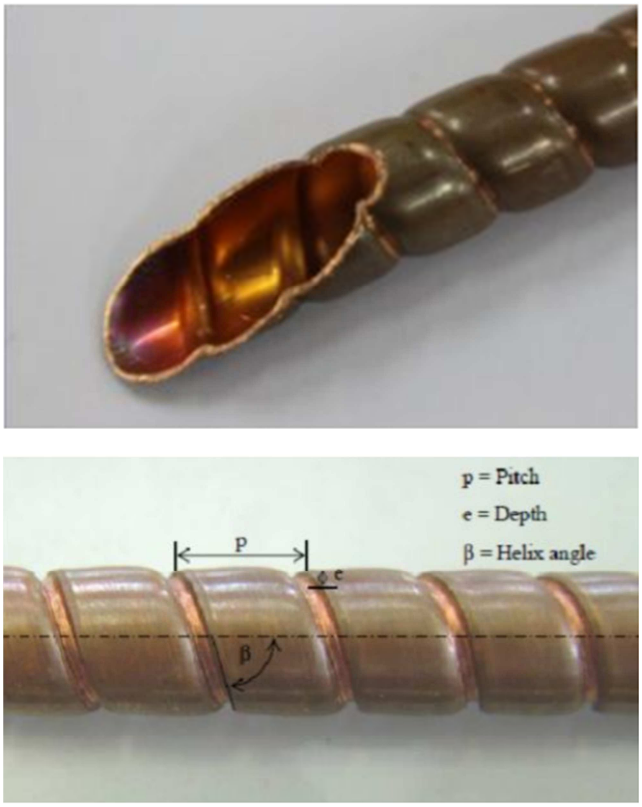


Figure 2. Geometry of corrugated tube.

di=8.3mm do=9.5mm
p=8mm W_t=0.6mm
β=75 e=1.5mm

2. Experimental Facility Procedure

The schematic diagram of the test apparatus has been shown in Figure 1. In fact, the experimental set-up was a well instrumented vapor compression refrigeration system. The test apparatus consisted of a test-evaporator (8) having a test-section length of 1100 mm. The test-section was a copper corrugated tube. The geometrical parameters of this tube are shown in Figure 2. This tube was heated uniformly by a flexible electrical heating tape of 3.5 kW capacity (with a calibrated accuracy of 2W). Also a cylindrical brass block was located between heater and corrugated tube to ensure that the heat flux is uniform along the channel. In order to change the inclination angle of the test-evaporator the connections to this evaporator were made by flexible pressure hoses (11).

During the experiments, a short range of vapor quality could be achieved in a single test run. Therefore, to cover the entire ranges of vapor quality a pre-evaporator (7) was installed before the test-evaporator. By regulating the voltage to the electrical coil around the pre-evaporator, the vapor qualities at the inlet of test-evaporator was controlled. It was ensured that the refrigerant coming from the test-evaporator is superheated vapor before it entered the compressor. This was attained by installing an after-evaporator (9) downstream of the test-evaporator. All the three evaporators were thermally insulated. An accumulator (10) was also provided before the suction of compressor. The refrigerant mass flow rate was measured by a rotameter (3) installed downstream of condenser. It was ensured that the complete liquid refrigerant enters the rotameter by providing a suitable condenser (2). The rotameter was calibrated prior to installation and the accuracy of it is within ±1% of the measured value. Refrigerant enters rotameter (3) and enters pre-evaporator (7) after passing through receiver (4), filter drier (5) and expansion valve (6).

The outside wall temperature of the test-evaporator tube was measured at six axial locations. At each location the temperature of tube was measured at top and bottom positions (when corrugated tube is in horizontal position).

$$t_{ws}=\frac{t_1+t_b}{2} \quad (1)$$

Thus, the average outside tube wall temperature of the test evaporator t_{wo}, was calculated as the arithmetic mean of outside tube wall temperatures at six axial stations:

$$t_{wo}=\frac{\Sigma t_{ws}}{6} \quad (2)$$

A total of 224 test runs with four different refrigerant mass velocities of 46, 81, 110 and 136 kg/m2s were performed for seven different tube inclinations from α = -90˚ to α = +90˚ (with intervals of 30˚). The range of operating parameters is given in table 1.

Table 1. List of operating parameters.

Working fluid	R-134a
Refrigerant mass velocity	46-136 kg/m2s
Average evaporating temperature	-26 to -2°C
heat flux	4.56 to 9.13 kw/m2
Inlet vapor quality	0.2-0.9
Exit vapor quality	0.3-1.0

3. Results and Discussion

First of all, the experimental heat transfer coefficients have been compared with those of Lohalertdecha & Wongwises (2011) for the evaporation inside a horizontal corrugated tube. In Figure 3 such a comparison has been made taking experimental heat transfer coefficient as abscissa and predicted heat transfer coefficient as ordinate.

From Figure 3, it is observed that the correlation of Laohalertdecha and Wongwises (2011) predicts the experimental heat-transfer coefficient, from the present investigation within an error range of-13.76% to +14.11%. The agreement of the experimental results with those predicted by the correlation suggested by Laohalertdecha and Wongwises (2011) establishes the integrity of the experimental apparatus.

The variation of evaporation heat transfer coefficient, h, with vapor quality for seven different tube inclination angles (α = -90° to α = +90°) at the mass velocity of 136 kg/m^{2}s, has been shown in Figure 4. From this figure it is found that for all the tube inclination angles the heat transfer coefficient increases with the increase of vapor quality until a vapor quality near 70–85% when it begins to decrease because of dryout. This is because the thickness of liquid film on inner wall of the tube decreases during evaporation in the flow direction and the void fraction increases, and the density of the liquid-vapor mixture decreases which would result in lower thermal resistance. As a result, higher evaporation heat transfer coefficients are observed as vapor quality increases until dryout occurs. Change in orientation of heat transfer coefficient with vapor quality might be because of change in flow pattern and different boiling mechanism (nucleate and convective).

The maximum heat transfer coefficient occurs at a vapor quality which depends upon mass velocity and the tube inclination. With increase in mass velocity and increase in tube inclination angle (from -90 to +90), the dryout occurs in lower vapor quality thus the maximum heat transfer coefficient occurs earlier. for example, for mass velocity of 136kg/m^{2}s, maximum heat transfer coefficient occurs at a lower vapor quality comparably with mass velocity of 46kg/m$_2$s. For the mass velocity of 136 kg/m^{2}s, dryout occurs at x=78-80% in the tube with inclination angle of -90° and at x=70–71% for the tube with inclination angle of +90°. While for the mass velocity of 46kg/m^{2}s, dryout occurs at x=79% and x=85% for the tubes with inclination angle of +90° and -90°, respectively.

The tube inclination angle influences the heat transfer coefficient, *h*, in a significant manner. The tube with +90˚ inclination angle (vertical upward flow) turns out to be the best performing tube at low vapor quality region. The tube having inclination angle of -90° (vertical downward flow) has the lowest heat transfer coefficient, *h*. Akhavan-Behabadi et al (2010) also reported similar results on heat transfer of evaporating flow in inclined microfin tubes. The performance of the tube with +90˚ inclination angle is much superior to that of tube with -90˚ inclination angle in the low vapor quality region where the heat transfer coefficient, h, for +90° inclined tube is about 39.9% more than that of the -90° inclined tube. However, the average heat transfer coefficient, *h*, for the tube with +90° inclination is 30% more than that for the tube with -90° inclination angle for the entire ranges of vapor qualities. In the low vapor quality region and at the lowest mass velocity the heat transfer coefficient, h, for +90° inclined tube is about 62% more than that of the -90° inclined tube.

The variation of heat transfer coefficient with vapor quality for a corrugated tube with different inclinations is shown in Figs. 5, 6 and 7 for the mass velocity of 46 kg/m^2-s, 81 kg/m^{2}s and 110 kg/m^{2}s respectively.

In these figures it is observed that the change in tube inclination has considerable effect on evaporation heat transfer coefficient. In fact, at low vapor quality the vapor phase velocity is reduced resulting in decrease of interfacial shear stresses and inertia force as well. Therefore, at low vapor quality the gravitational force has considerable effect on two-phase flow and this is the major reason for different rates of heat transfer coefficient for different inclinations of corrugated tube.

In other words, interfacial shear stress is dominant at high vapor velocity and high vapor quality, while at low vapor qualities, the vapor phase motion is slowed down and this would result in decrease of interfacial shear stress and consequently lower inertia forces near the wall. Therefore, the effect of gravity force on the evaporation of R134a flow at low vapor qualities is more considerable as it can be seen in Figs.4, 5, 6 and 7 for different inclinations of test evaporator.

The random variation in heat transfer coefficient, h, is also visible in these figures. This is due to the instability of heat transfer, in which, both the phases are flowing together during phase change. In addition, the presence of the different flow regimes inside the tube with different tube inclinations causes these random variations.

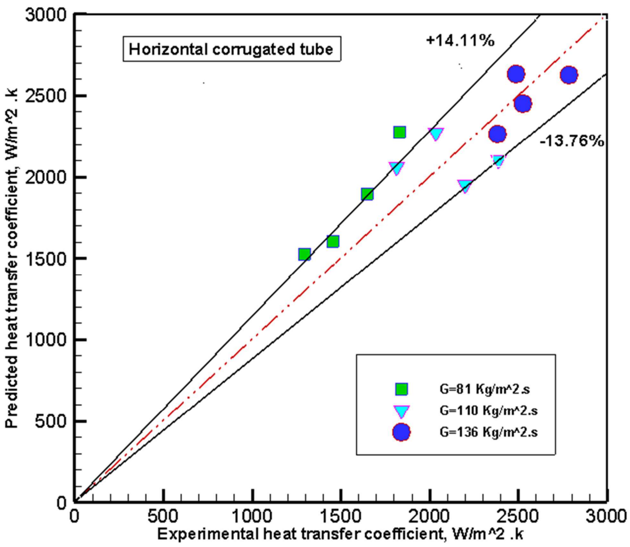


Figure 3. Comparison of experimental heat transfer coefficients with those predicted by Laohalertdecha and Wongwises (2001).

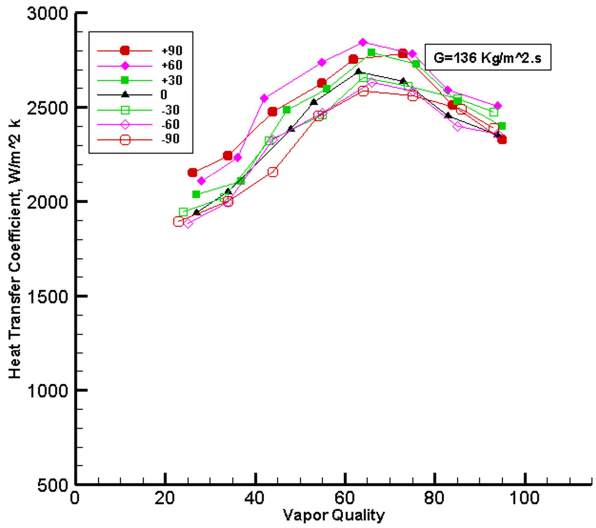


Figure 4. Variation of heat transfer coefficient.

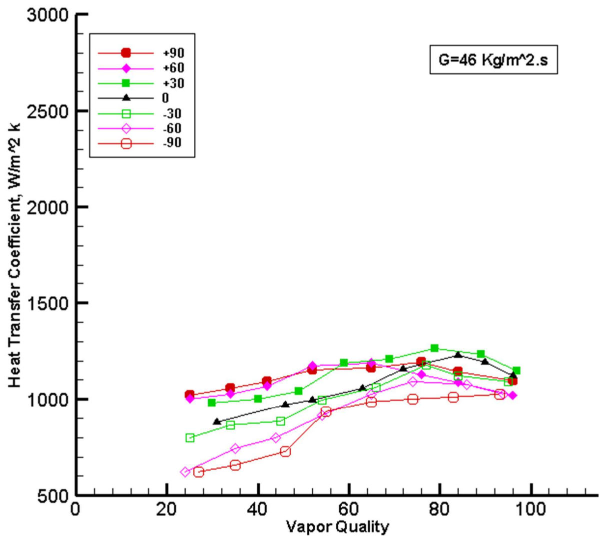


Figure 5. Variation of heat transfer coefficient with vapor quality at refrigerant mass velocity of 46 kg m^{-2} s^{-1}.

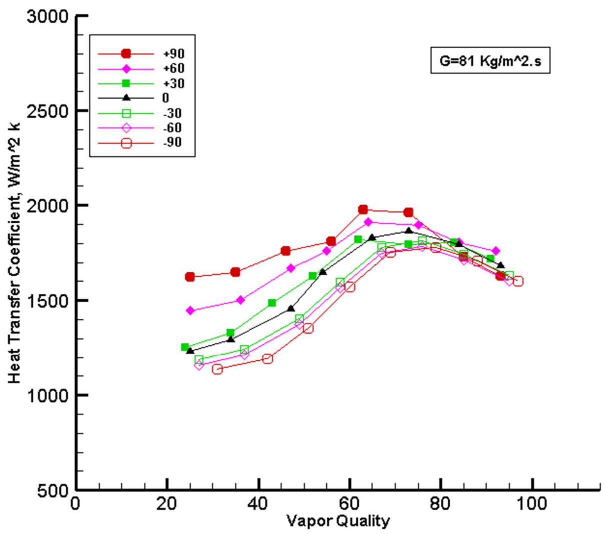


Figure 6. Variation of heat transfer coefficient with vapor quality at refrigerant mass velocity of 81 kg m^{-2} s^{-1}.

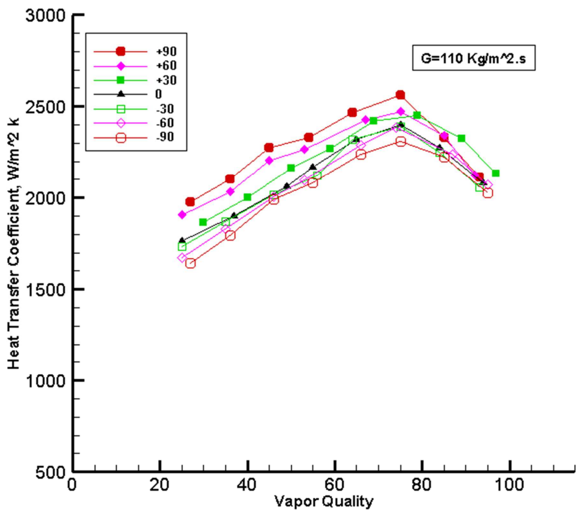


Figure 7. Variation of heat transfer coefficient with vapor quality at refrigerant mass velocity of 110 kg m^{-2} s^{-1}.

In the dry out region, at high vapor quality, upward vertical boiling flow has the lower heat transfer coefficient than that for the horizontal flow. In other word, when dry out happens, the wetted perimeter of the tube is the most effective region for heat transfer. At high vapor quality in horizontal tube (also tubes with inclinations close to horizontal), the liquid tends to flow in lower side of the tube and as a result the flow boiling heat transfer in the flooded part of the tube takes place while mist flow in the form of entrained droplets occurs during upward vertical flow.

From Figs 4 to7, it is also noted that the vertical tube with inclination angle of -90° has the lowest heat transfer coefficient. Wongwises et al. (2011) also reported the same observations in their study of orientation effects for R-134a flow boiling in vertical downward flow inside a corrugated tube. In the downward vertical flow, since the interfacial shear stress and the gravity are acting in the same direction, the flow pattern will remain in annular status (Khoeini & Akhavan-Behabadi 2011). The phenomenon of annular flow causes to form a thick layer of liquid film around the periphery of tube as a result the heat transfer coefficient, h, is reduced. Since, the gravitational force and vapor shear stress are in same direction, the interfacial turbulence is the lowest (Akhavan-Behabadi & Kumar 2007). In Fig. 7, the variation of heat-transfer coefficient with vapor quality at different mass velocities of R-134a is shown for the tube having inclination angle of +90˚. The experimental data clearly show that the heat transfer coefficients increase with increasing the refrigerant mass velocity. Indeed, increasing the refrigerant mass velocity increase the fluid velocity, thus enhancing convective boiling. In convective boiling, when fluid velocity increases, the heat transfer coefficient increase. In higher mass velocity, effects of convection are much higher and more turbulence occurs, as a result heat transfer coefficient increase.

With increase of mass velocity, flow pattern changes from stratified flow to annular flow in a much lower vapor quality, and the increase in vapor quality in annular flow causes more increase in heat transfer relative to stratified flow. Also with the increase of mass velocity, maximum heat transfer coefficient will occur in much lower vapor quality because with the increase of mass velocity, dryout will occur sooner. The average heat transfer coefficient, $\overline{h}$, at the mass velocity of 136 kg/m2s is nearly 128% more than that for the mass velocity of 46 kg/m2s.

4. Conclusions

The following conclusions have been drawn from the present investigation:

- For all tube inclination angles, the heat transfer coefficient increases with the increase of vapor quality until a vapor quality near 70–85% when it begins to decrease because of dryout.
- The tube inclination angle, a, affects the boiling heat transfer coefficient in a significant manner. The effect of inclination angle is more prominent at low vapor qualities and mass velocities.
- At low vapor qualities, the highest heat transfer coefficient is attained at inclination angle of +90° (vertical upward flow) and at high vapor qualities the highest heat transfer coefficient occurs when the corrugated tube is horizontal. The vertical tube with inclination angle of -90° has the lowest heat transfer coefficient for the entire range of vapor quality. The corrugated tube having inclination angle of +90° outperforms the tube with -90° inclination in a range of 7-62%.

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Biography

Hesamoddin Salarian is now Assistant Professor in the Department of Mechanical Engineering, Islamic Azad University, Nour Branch, He has been involved in teaching and research in the field of Mechanical Engineering, Heating &Cooling System and Energy.

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1 Denoting a heating system in which heat is transported using circulating water.